

Helmut Bannwarth

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Liquid Ring Vacuum Pumps, Compressors and Systems

Conventional and Hermetical Design



H. Bannwarth

**Liquid Ring Vacuum Pumps,
Compressors and Systems**

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Helmut Bannwarth

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*Translated by
Christine Ahner*



**WILEY-
VCH**

WILEY-VCH Verlag GmbH & Co. KGaA

Author**Helmut Bannwarth**

Emil-Gött-Strasse 7
D-79194 Gundelfingen
Germany

Translation**Christine Ahner**

translate economy
Freiherr von Eichendorff-Str. 8/1
D-88239 Wangen/Allgäu
Germany

Cover Picture

Two-stage vacuum system with hermetic liquid ring vacuum pumps for recovery of aromatic compounds (Hermetic-Pumpen GmbH, Gundelfingen, Germany)

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Library of Congress Card No.:

applied for

British Library Cataloguing-in-Publication Data

A catalogue record for this book is available from the British Library.

Bibliographic information published by**Die Deutsche Bibliothek**

Die Deutsche Bibliothek lists this publication in the Deutsche Nationalbibliografie; detailed bibliographic data is available in the Internet at
<<http://dnb.ddb.de>>.

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Printed in the Federal Republic of Germany.

Printed on acid-free paper.

Typesetting Kühn & Weyh, Satz und Medien, Freiburg

Printing Betzdruck GmbH, Darmstadt

Bookbinding J. Schäffer GmbH i. G., Grünstadt

ISBN-13 978-3-527-31249-8

ISBN-10 3-527-31249-8

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Dedicated to my wife Karin

Foreword

Modern technology is based on both craftsmanship and scientific knowledge. The further development in technology depends decisively on how far scientific results are brought in purposefully in view of economical aspects. Here, physics is of great importance.

When trying to determine today's relation between physics and technology it can be assumed that physics is pure science while technology means designing on a scientific basis. Physics is one of the bases technology needs. Those who are capable of utilizing physical understanding for their developing and designing skills can avoid lengthy and costly experimenting.

In 1991, on the occasion of the 125th birthday of LEDERLE GmbH, the technical manual "Liquid ring vacuum pumps, compressors and plants, conventional and hermetic" was issued for the first time in German language.

The author succeeded in communicating physical and technical basics in a remarkable way. The book met with great interest both among planners and operators.

Vacuum technology has become indispensable for many branches of industry.

The demand for more protection of health, workplace and environment in modern process engineering didn't stop at the vacuum pump either. The product range of LEDERLE GmbH in the vacuum sector has been further developed according to these requirements and has been based on the experiences of the plant operators. As a result, nowadays liquid ring vacuum pumps and compressors in hermetic design are on the market.

The great success and the active interest the first issue of this reference book met with induced us to issue an edition in English. With this, an international clientele and interested circles will have a specialist book in their hands that deals with the design and application of pumps and plants in the vacuum range. The author, Helmut Bannwarth, once more substantiates his expert competence in an impressive way.

We cherish the hope that this book will find a wide and attentive readership, and that owing to the continuous cooperation between manufacturers, planners and operators ideas and suggestions for further progress will arise.

Dr. Roland Krämer
Managing Director – Engineering
Lederle-Hermetic GmbH

Wolfgang Krämer
Managing Director – Sales
Lederle-Hermetic GmbH

December 2004

Preface

In 1991, the first edition of the technical manual “Liquid ring vacuum pumps, compressors and plants” was published in German language by the publishing house VCH Verlag in D-69496 Weinheim, Germany. Three years later, in 1994, the second, revised edition came out and I took advantage of the opportunity to update and complete it according to the progressing technical developments.

With this first edition in English language, again updated, I could fulfil the request for a translated version of the book expressed by many interested students and practitioners of industry and engineering offices at home and abroad.

I express my thanks to the publishing house Wiley-VCH Verlag particularly for the again pleasant cooperation and the continuous support I enjoyed.

I would also like to express my gratitude to the managing directors of the company group Lederle GmbH and Hermetic-Pumpen GmbH, Mr. Wolfgang Krämer and Dr. Roland Krämer for their generous support. Many thanks to all companies and publishing houses not mentioned here for kindly providing me with the respective documents.

December 2004

Helmut Bannwarth

Preface of the first edition in German language in 1991

The Greek philosopher Democritus and other well-known scholars as well as the metaphysicians of the Middle Ages have already dealt with the subject vacuum.

In 1640, Otto von Guericke, the Mayor of Magdeburg, conducted the first experiments regarding the generation of vacuum. On the occasion of the Reichstag in Regensburg in 1654, he demonstrated the effect of air pressure on two evacuated hemispheres (known as Magdeburger Halbkugeln). Owing to his thorough knowledge in this field and the machines and plants he had designed, he is regarded as the founder of the entire vacuum technology.

In 1643, the Italian mathematician and physicist Evangelista Torricelli succeeded in inventing the barometer, the first device for the measurement of vacuum.

Today, neither modern physical-technical basic research nor industrial process engineering is conceivable without methods and appliances based on vacuum technology. There is hardly any field of technology offering so many possibilities of application as the field of vacuum technology does. Meanwhile, in this sector a lot of industrial companies developed components and vacuum systems that are partly available on the market as standard units. Due to the progressing development in the field of vacuum technology, project and design engineers will not find all the required equipment on the market, but will have to convert already existing plants for new experiments or will have to design new pilot plants or production plants by themselves.

The intention of this book is to design and manufacture a vacuum plant suitable for rough vacuum making use of the conventional components, the practical experience and standards valid in the vacuum sector.

At the beginning, we cast light on the field of gas physics in vacuum technology and provide an overview of the whole vacuum field. Thereby, all machines used in practice for the generation of vacuum will be taken into consideration. In particular, the liquid ring vacuum pumps and compressors are being elucidated, as well as components usually applied in industry and their combination to vacuum systems. Here, great importance is attached to the hermetic liquid ring machines and components nowadays used for closed and environment-friendly cycles. Furthermore, we will also report comprehensively on the practical layout of vacuum pumps, pipework and vacuum containers, on the assembly and control of machines and plants, the

surface quality in vacuum technology, vacuum hygiene, safety-at-work, explosion protection and explosion-proof electrical resources. Some chapters are completed with practical calculation examples.

As far as standards, recommendations and guidelines in vacuum technology and the adjacent fields exist they have been quoted to a large extent.

Physical values shall be SI-units, however, even tables and charts with old units that are still valid and in use, such as the former internationally introduced pressure unit "Torr", are taken into consideration.

The appendix contains an extensive compilation of the international SI-unit system, conversion tables and common constants, national and international standards as well as recommendations, pictograms, and material data of fluids often found.

This book is written from the engineer's practical point of view and is mainly addressed to students, technicians and engineers involved in designing and operating of machines and plants in the field of vacuum or to those keen on familiarizing themselves with this subject.

With this work, a supplementary reference book, practically oriented and reflecting the latest state of knowledge, will be available on the specialist book market.

I want to express my thanks to the management of the company Lederle GmbH, Pumpen- und Maschinenfabrik, D-79194 Gundelfingen for providing me with a large part of photos and drawings that actually made the publication of this book possible. My particular thanks go to Mr. Hermann Krämer, graduate engineer and General Manager of the company, for his generous support. I also thank the companies and publishing houses for providing me with pictures and charts and their permission to reproduce them.

January 1991

Helmut Bannwarth

1

Gas Physics and Vacuum Technology

1.1

The term “vacuum”

In standard specification list DIN 28400, Part 1, the term “vacuum” is defined as follows:

Vacuum is the state of a gas, the particle density of which is lower than the one of the atmosphere on the earth’s surface. As within certain limits the particle density depends on place and time, a general upper limit of vacuum cannot be determined.

In practice, the state of a gas can mostly be defined as vacuum in cases in which the pressure of the gas is lower than atmosphere pressure, i.e. lower than the air pressure in the respective place.

The correlation between pressure (p) and particle density (n) is

$$p = n \cdot K \cdot T \quad (1-1)$$

k Boltzmann constant

T thermodynamic temperature

Strictly speaking, this formula is valid only for ideal gases.

The legal pressure unit is Pascal (Pa) as SI unit. The usual pressure unit in vacuum technology is millibar (mbar). This pressure unit is valid for the whole vacuum range from coarse vacuum to ultrahigh vacuum.

1.2

Application of vacuum technology

Vacuum is often used in chemical reactions. It serves to influence the affinity and therefore the reaction rate of the phase equilibrium gaseous – solid, gaseous – liquid and liquid – solid. The lowering of the pressure causes a decrease in the reaction density of a gas. This effect is used e. g. in the metallurgy for the bright-annealing of metals. There are several kilograms of metal for 1 liter annealing space, whereas less than 1/3 of the total volume is filled with gas; as a result, the oxygen content of

the residual air is less than 1 mg in the pressure range of 10 mbar. Compared to the metal mass, the oxygen content decreases to 10^{-5} . This leads to a retardation of the oxidation process, thus allowing higher process temperatures. It also causes an increase in the ductility of the products. When teeming melted materials, such as metals, apart from a retarded oxidation also degassing (desorption) takes place at the same time. The result is metal of particular purity. In the metal and sinter ceramics industry, sintering processes are based on the same principle. The impediment of fermentation caused by aerobic micro-organisms with the help of vacuum can also be called reactive retardation, an example of which is vacuum packaging. On the other hand, a reactive acceleration is reached, e. g. when after the evacuation of the materials to be treated, gases or liquids are discharged in order to increase the reaction density. The reaction density can also be controlled as required by means of a decrease in pressure, e.g. when chlorinating. In this case, diluting gases are not required.

The selection of the adequate technology for a chemical-physical process depends on pressure-related parameters and the specific characteristics of the material to be treated. This requires e.g.

- the determination of the optimal ranges of vacuum and temperature,
- the determination of the required equipment,
- the determination of all necessary auxiliary means, which vacuum pumps or vacuum devices belong to.

The dimensions of a vacuum plant are not only determined by the performance data of the process devices, but also by the operating range of the vacuum. In the range of high vacuum the sizes of the individual devices are not as important for the dimensions of the total plant as the required suction capacity and the sizes and dimensions of the vacuum pumps, i.e. the vacuum pump stations.

Generally, in vacuum process engineering of the chemical industry or related branches vacuum plants usually consist of the following main components:

- Vacuum devices for the execution of the process
- Condensation devices for the compression of the arising vapor
- vacuum pump or combination of pumps
- accessories, such as separators, heat exchangers, vacuum vessels, metering and control devices.

1.2.1

Basic operations in process engineering

In the industrial process engineering, basic operations are usually carried out in coarse vacuum, seldom in fine vacuum. The application of high vacuum is considered only in particular cases.

The machines used here are vacuum pumps and compressors. With lower vacuums and higher flow rates mostly extractor fans are used.

Regarding the use of waste heat and the careful heating of thermally sensitive material it is advantageous for the performing of the vacuum process to work at low temperatures. The most different processes are carried out through vaporizing, drying, condensing, degassing, filtering etc. under vacuum. Generally, it can be said that the total operating costs of vacuum plants increase with higher vacuum.

Mechanical vacuum pumps can be designed as dry or wet running pumps with pistons or rotating elements.

Dry running vacuum pumps are used for pumping dry and non-condensable gases. In case of existing condensable vapors, condensers have to be installed on the suction side in which the condensable particles are condensed through cooling. In the field of coarse vacuum, usually surface condensers or mixing condensers are used, while low-temperature condensers or absorption condensers are used in the fine vacuum range.

Wet running vacuum pumps are particularly suitable for the suction of condensable vapors or gases, as well as for mixtures of gases and liquids. In wet running vacuum pumps driven by an operating liquid (e.g. water or another liquid chosen according to the process), the process gas can be condensed. Owing to this fact, condensers installed on the suction side of the pumps are not required. The diagram of the basic layout of a vacuum device is shown in fig. 1-1.

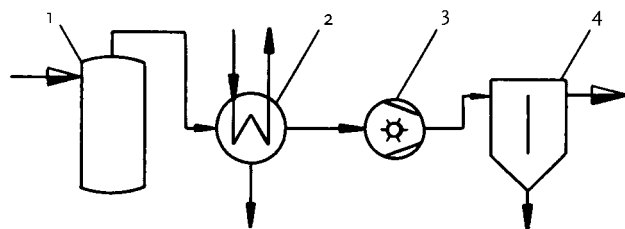


Figure 1-1. Basic scheme of a vacuum plant

- 1 vacuum vessel
- 2 condenser
- 3 liquid ring vacuum pump
- 4 liquid separator

1.2.2

Basic fields and worked-out examples for the application of vacuum technology

Vacuum technology is dominant in many fields of research and industry (Table 1-1) and is applied by using the most different process technologies (Table 1-2).

Table 1-1. Fields of application of vacuum technology [1.1]

Field of knowledge	Branch of industry/technology
Physics (mechanics, continuum mechanics, thermodynamics, electrodynamics, optics, nuclear physics, surface physics- and chemistry)	Scientific instrument production (precision mechanics) Mechanical engineering and heavy engineering industry Electronics (for measuring and control problems) Automation and controlling Cryogenic engineering
Biophysics	Chemical process engineering (oils, greases, waxes, resins etc.)
Physical chemistry	Metallurgy
Chemistry	
Material engineering	State-of-the-art technologies (glass, ceramics and metallic compounds)
Pharmacy	
Medicine	
Field of application	Examples
Nuclear technology	Crystal growing (scintillation detectors) Evaporation (solid-state detectors) Working with closed systems (hot laboratories, plutonium, etc.) Filtration Sintering under vacuum (nuclear metals, ceramics, carbide)
Optical industry	Vaporization technologies (interference layers, laser, maser, glass fiber optics, optoelectronics)
Electrical engineering/electronics	Drying (insulation oils, coolants) Impregnation (insulation material) Hermetic sealing (boosters) Evacuation and degassing (e.g. tubes, lamps) Evaporation and sputtering (e.g. condenser production, thin-film technology) Encapsulation (tubes, semiconductor elements) Welding and surface treatment (micro-circuits) Crystal growing (epitaxial growth) Surface reactions (transistors, circuits)
Scientific instrument production	Physical and chemical analyses Analyzing appliances (surface analysis, UV examination, electron and ion microscopes, X-ray analyzers, microwave devices) Lowest temperature analyses Particle accelerators, storage rings Fusion plants
Chemical industry	Distillation (fatty acids, oils, alcohol, etc.) Filtration Drying, dehydration vaporization, sublimation

Field of application	Examples
Food industry	Freeze-drying (fresh and cooked food) Preservation and conservation Dehydration and concentration (milk, coffee ...) Crystallization (e.g. sugar)
Pharmaceutical industry	Distillation (vitamin A, E, ...) Freeze-drying (blood, ...) Drying (antibiotics, hormones, ...) Sterilization (dressing materials, ...)
Metallurgy and semiconductor manufacturing	Distillation (Mg, Ca, Li, Se, Na, K, ...) Reduction (Ti, Mg, Zr, Fe, Cr, ...) Sintering (high-melting and reactive metals, carbides, ...) Melting and casting (Pb, Sn, Mn, Ge, alloys, high-melting and reactive metals) Drying (powder) Heat treatment
Production engineering	Impregnation (molds for casting) Injection molding (Mg alloy components without voids) Fastening (chucks) Welding and soldering (precision devices) Surface finish (hard material or anticorrosion coating)
Space engineering	Biological processes and developments Material development and control Development and control of devices (motors, gauges, ...)
Office machines industry	Welding and treatment Registration Heat insulation
Transportation in various industry branches	Lifting and transporting (paper, metal sheets, pavement plates, cathode ray tubes, ...)
Miscellaneous applications	Evaporation (paper, plastics, fabrics, ...) Thermal insulation (Dewar flasks, ...) Forming (plastics, vacuum casting) Concrete hardening

1.2.3

Overview of the most important vacuum processes

Tab. 1-2 contains processes preferably carried out under vacuum.

Table 1-2. Vacuum processes in process engineering [1.2]







































Process	Important advantages through vacuum
<i>Endothermic processes</i>	
Vacuum vaporization	Low temperature of material and heating agent Increased heat efficiency
Vacuum distillation	Better separation effect, molecular distillation; Oxide-free and gas-free metal distillation
Vacuum sublimation	Under triple point (freeze-drying)
Vacuum drying	Quick and careful drying without shrinking; increased dissolving speed
Vacuum calcination	Shifting of phase equilibrium, decomposing temperature drops
Vacuum annealing and sintering	Bright annealed products are free from oxides, gases and scale
Vacuum melting	Gas-free melted product, high-purity metals, chemicals, plastics, sealing compounds
Vacuum casting	Non-porous cast products with high density
Vacuum soldering	Furnace soldering without flux, oxide-free hard soldering
Vacuum evaporation	Surface finish through vapor deposition of thin films of metals and non-metals
Vacuum reaction	Thermal conversion at low temperatures and decreased reaction density
Vacuum steam generation	Water vapor heating below 100 °C, rapid control
<i>Processes without catalytic oxidation</i>	
Vacuum degassing	Gas-free liquid, viscose, plastic masses
Vacuum gas injection	Fumigation, disinfection, sterilisation, sorption
Vacuum mixing	Modified sorption, improved wettability
Vacuum extraction	Higher dissolver speed, dissolver recovery
Vacuum filtration	Continuous residue decreasing
Vacuum impregnation	Complete impregnation of porous bodies, agglutination
Vacuum transport	Fluidized bed transport of bulk materials by means of induced draught
Vacuum insulation	Thermo-barochamber
Vacuum packaging	Improved shelf life, no aroma losses
<i>Exothermic processes</i>	
Vacuum condensation	Distillate recovery, higher energy yield
Vacuum cooling	Ice generation without coolants
Vacuum crystallization	Higher crystal yield through flash distillation of solvents
Vacuum reaction	Higher distribution rate, low reaction density
Vacuum presses	Non-porous agglomeration or agglutination of powders and laminates

1.2.4

Basic designs of apparatus for mass transfer and mass combination

The most important vacuum processes applied in process engineering are given in Table 1-3. They are subdivided according to thermal processes and grouped together according to the apparatus equipment.

Table 1-3. Basic symbols, apparatus and process technique in vacuum engineering [1.2]

Component	Basic symbol	Symbol range					
Vacuum apparatus		Mass transfer				Mass combination	
	Heat tone	endothermic	endothermic	exothermic	without	without	endothermic
							
	Process	evaporating distilling deodorising	drying subliming calcining	degassing cooling crystallising	filtering extracting	agglomerating shaping impregnating	sintering, sold- ering, melting, casting, coating
Condensers	Condenser	liquid	liquid and solid		solid	liquid	dissolved
	Cooling fluid	through pipes	around pipes	Condensation trap	rotat. cooler	Injection	Absorption
							
	Process	Surface condensation with condensate separation				Co-condensation	
Pumps		Piston	Rotor	Rotor	Rotor	Motive fluid	Motive fluid
							
	Process	Positive displacement vacuum pump oscillating	rotating	rotating	Kinetic vacuum pump rotating		
Separators	Fluid	liquid	solid	solid	solid	gaseous	
							
	Process	Droplet separation	Dust separation by Gravity		Filter	Submerged washing	Circulatory washing
Vacuum vessels	Fluid	liquid	liquid	liquid or solid	liquid or solid	solid	solid
							
		Heating, cooling coil	Heating, cooling jacket	with cover opening	with stirrer	Bottom opening	Bottom opening with stirrer
Heat exchangers	Fluid	liquid	liquid	liquid or solid			
							
		Tube system	Tube coil	Double shell with stirrer			

1.2.5

Limits to the application of vacuum in process engineering

In the field of vacuum, the kind of gas flow depends on the respective prevailing vacuum.

According to the Hagen-Poiseuille law, laminar gas flow exists in coarse vacuum. In the range of high vacuum, the internal friction is no longer decisive, as the collision of the molecules and the tube wall occurs more often than the collision among the molecules themselves. This kind of flow is called Knudsen molecular flow, i.e. the average molecular speed and the mean particle path of the gas molecules determine the flow process. The range between coarse and high vacuum is called fine vacuum. The fine vacuum range is the transition zone between the Hagen-Poiseuille flow and the Knudsen flow. The range of vacuums higher than in high vacuum is called ultrahigh vacuum.

According to the Knudsen equation

$$K = \frac{\bar{l}}{d} \quad (1-2)$$

the different types of flow are subdivided as shown in table 1-4.

Table 1-4. Flow types in vacuum [1.3]

K	< 0.5	0.5 – 3.0	> 3.0
Type of flow	Hagen-Poiseuille flow	Transition zone	Knudsen molecular flow

- K Knudsen number
 \bar{l} mean free path [m]
d diameter of the flow channel [m]

Therefore, for the type of flow arising in tubes, the ratio of the mean free path (a gas molecule does on average until its collision with another molecule), which increases with decreasing pressure and the diameter of the flow channel is decisive.

Material transport. With the increasing vacuum, the transport of gases and vapors gets more and more difficult. This is a result of the fact that with decreasing pressures the available forces diminish and the volumes increase. With pressures lower than 0.1 kPa (= 1.0 mbar), in practice only insignificant quantities of gas and vapor are transported in pipes.

Heat transport. Only in the range of atmospheric pressure heat transfer through convection is technically applicable, whereas high vacuum is a good heat insulator. In vacuum processes, the heating-up of the material occurs practically only in direct contact with heating elements through radiation, rarely through dielectric heating or inductive heating.

1.3

Operating ranges and measuring ranges of vacuum

Vacuum ranges are ranges of pressures or particle densities according to which it is agreed to classify vacuum.

The rounded down limits of these ranges are listed as pressure values or equivalent particle density values in Tab. 1-5.

The particle density values given in the table apply to a temperature of $\delta = 20^\circ\text{C}$.

1.3.1

Vacuum pressure ranges

Table 1-5. Vacuum ranges (acc. to DIN 28400, Part 1, July 1979)

Formula character	Unit	Coarse vacuum, CV	Fine vacuum, FV	High vacuum, HV	Ultrahigh vacuum, UHV
P	[Pa]	$1 \cdot 10^{15}$ to $1 \cdot 10^2$	$1 \cdot 10^2$ to $1 \cdot 10^{-1}$	$1 \cdot 10^{-1}$ to $1 \cdot 10^{-5}$	$< 1 \cdot 10^{-5}$
P	[mbar]	$1 \cdot 10^3$ to 1	1 to $1 \cdot 10^{-3}$	$1 \cdot 10^{-3}$ to $1 \cdot 10^{-7}$	$< 1 \cdot 10^{-7}$
n	$[\text{m}^{-3}]$	$2.5 \cdot 10^{25}$ to $2.5 \cdot 10^{22}$	$2.5 \cdot 10^{22}$ to $2.5 \cdot 10^{19}$	$2.5 \cdot 10^{19}$ to $2.5 \cdot 10^{15}$	$< 2.5 \cdot 10^{15}$

It seems to suggest itself to divide measuring ranges in decimal powers, as follows:

<i>Millibar range</i>	from 1000 to 1 mbar, essentially the normal and coarse vacuum range
<i>Microbar range</i>	from 1 to 10^{-3} mbar, the fine vacuum range
<i>Nanobar range</i>	the high vacuum range
<i>Picobar range</i>	and below, the ultrahigh vacuum range

1.3.2

Vapor pressure curve of water in vacuum

For vacuum process engineering with a prevailing thermal mass transfer, it is practical and clearer to divide into vacuum operating ranges following the thermometric fixed points of water as so-called fundamental material which the chemists, process engineers and technicians have to deal with every day (Fig 1-2). According to this, in the boiling range of pure water between 0 and 100°C corresponding to 6.11 mbar to 1013mbar, the normal or basic vacuum range results, in which the boiling process always occurs as pure vaporization.

Processes with lower pressures at which vaporization takes place through sublimation from the solid phase (ice) below 0°C are to be allocated to the fine or high vacuum range.

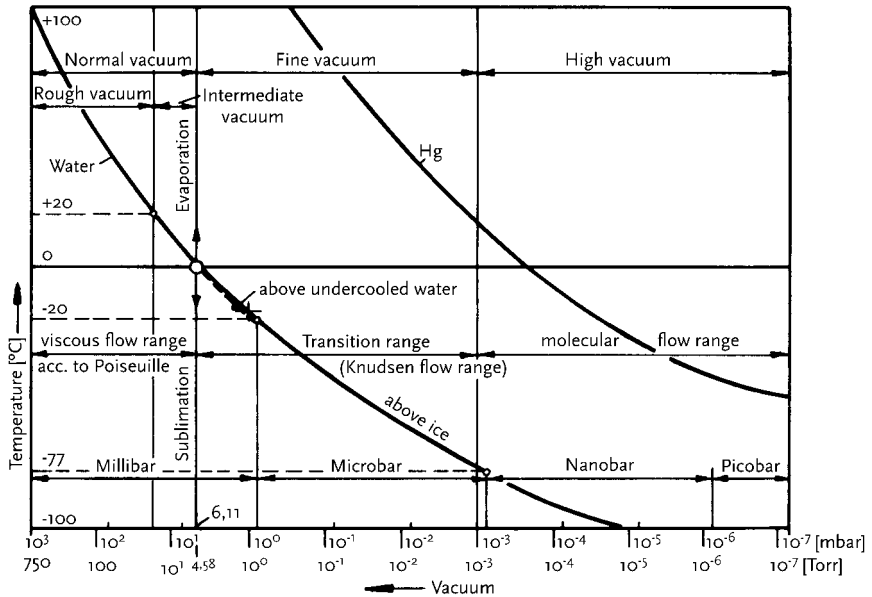


Figure 1-2. Vapor pressure curve of water in different vacuum ranges [1.2]

1.3.3

Vacuum operation ranges, temperature pressure table

In Tab. 1-6, vacuum operating and measuring ranges in millibar and Torr are compared to the specific boiling points of pure water (H_2O), mercury (Hg), methanol (CH_3OH) and ethyl alcohol ($\text{C}_2\text{H}_5\text{OH}$). From this, the difference between the vapor pressures of the individual fluids and the dependence on the pressure temperature are clearly deriving. In fine and high vacuum, the operating ranges coincide with the measuring ranges.

Table 1-6. Operating ranges and measuring ranges of vacuum [1.4]

		Measuring ranges		Boiling ranges in °C			
		mbar ¹⁾	Torr ²⁾	H ₂ O	Hg	C ₂ H ₃ · OH	CH ₃ OH
Normal vacuum	Rough vacuum	1013	760	100	357	78,3	64,7
		1000	750	99,6	356,2	78	64,2
		800	600	93,5	344	71	56,5
		600	450	85,5	329	61,5	48
		400	300	75	309	54	40,5
		200	150	60,1	278	45	32
		100	75	45,7	251	29,5	16,5
		80	60	41	242	24	11
		60	45	36,4	232	17	5,5
		50	37,5	32,5	225	14	3
		40	30	29	218	11,5	0
		30	22,5	24	208	8	-3,5
		23,4	17,53	20	201	5	-6
		20	15	17,2	196	3,5	8
Intermediate vacuum	Intermediate vacuum	15	12	14	190	-1	-11,5
		12	9	9,7	181	-4	-15,5
		10	7,5	7	176	-9	-19,6
		8	6	3,8	170	-12,5	-24
		6,11	4,58	0	162,5	-15,6	-25
		6	4,5	-0,2 ³⁾	162	-17	-20,5
		4	3	-5	152	-21,5	-32
		2	1,5	-13	135,5	-28	-38
		1	0,75	-20,3	119	-38,5	-47,5
		1000	750	-22,7	115,5	-42	-50,5
		800	600	-25,4	110	-45,5	-53
		600	450	-29,3	102	-48,5	-56,5
		400	300	-36	89	-53	-61
		200	150	-42	77,5	-60	-67,5
Fine vacuum range	Fine vacuum range	80	60	-43,7	73,5	-63	-70
		40	30	-50	63	-68	-74,5
		10	7,5	-60,5	42,5	-75	-83
		8	6	-62	39,8	-77	-85
		4	3	-67	31	-81	-88,5
		1	0,75	-76,3	14,4	-87	-94
		1000	750	-77,8	12	-88,5	-95
		800	600	-82	5	-91,5	-98
		400	300	-90	-9	-94	-100
		100	75	-91,5	-11		
		80	60	-96	-17		
		40	30	-101,5	-29		
		10	7,5	-103	-31		
		8	6	-106	-36		
High vacuum range	High vacuum range	4	3	-112	-42		
		1	0,75	-113			
		1000	750	-116			
		800	600	-121,5			
		400	300	-122			
		100	75	-124,5			
		80	60	-129,5			
		40	30	-130			
		10	7,5	-132,5			
		8	6	-137	-100		
		4	3				
		1	0,75				
		1000	750				
		800	600				
Ultra vacuum range	Ultra vacuum range	400	300				
		100	75				
		80	60				
		40	30				
		10	7,5				
		8	6				
		4	3				
		1	0,75				
		1000	750				
		800	600				
		400	300				
		100	75				
		80	60				
		40	30				
		10	7,5				
		8	6				
		4	3				
		1	0,75				
		1000	750				
		800	600				
		400	300				
		100	75				

¹⁾ 10³ N/m²²⁾ mm Hg³⁾ above ice

1.3.4

Total pressure measuring

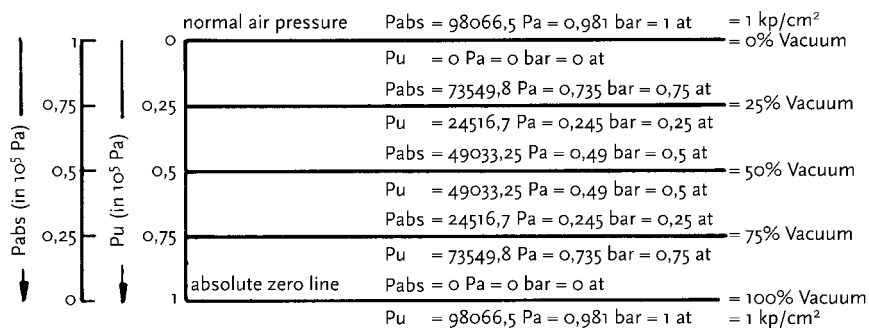
Pressure units. In vacuum technology, the Torr unit is used as the practical unit for pressure, which is tantamount to millimeter mercury column (mm Hg), the same applies to the unit millibar (mbar). In the international unit system (SI), Pascal (Pa) or Newton/square meter (N/m^2) are used as pressure units. There are still other customary pressure units which should no longer be used, however. When pressure values are quoted, usually negative decimal powers (e.g. 2×10^{-1} Torr) are employed. Table 1-7 may help in converting old units, no longer permitted in business and official communication since 31/12/1977, into new units.

Table 1-7. Conversion table for pressure units (acc. to DIN 28400, part 1 July 1979)

	Pa [$\text{N} \cdot \text{m}^{-2}$]	1 bar = 1000 mbar	Atm	Torr
1 Pa = $1 \text{ N} \cdot \text{m}^{-2}$ =	1	10^5	$0.986923 \cdot 10^{-5}$	$0.750062 \cdot 10^{-2}$
1 bar = 1000 mbar =	10^5	1	0.986923	$0.750062 \cdot 10^3$
1 atm =	$1.01325 \cdot 10^5$	1.01325	1	$0.760000 \cdot 10^3$
1 Torr =	$1.33322 \cdot 10^2$	$1.33322 \cdot 10^3$	$1.315789 \cdot 10^3$	1

Depression and vacuum. In technology, the terms overpressure (p_o), depression (p_u), and vacuum are used. The lowest pressure theoretically possible is $0 \text{ Pa} = 0 \text{ N} \cdot \text{m}^{-2} = 0 \text{ mbar}$.

This would correspond to a 100 per cent vacuum (Fig. 1-3). Vacuum is a rarefied air or gas space. With 100 percent vacuum there is a total absence of air and gas.

**Figure 1-3.** Underpressure and vacuum [1.5]

According to the following formula, vacuum can be calculated in percent:

$$\text{Vacuum} = \frac{P_u}{P_{\text{air}}} \cdot 100 [\%] \quad (1-3)$$

As this is a pressure ratio, any pressure unit can be used for calculation. However, for p_u and p_{air} the same pressure units have to be used in the formula.

Vacuum in percent can also be determined from barometer readings B in Pa and the readings of a vacuum gauge H in Pa using the following formula:

$$\text{Vacuum} = \frac{1.01325 \cdot 10^5 \text{ Pa} - (B - H)}{1.01325 \cdot 10^5 \text{ Pa}} \cdot 100 [\%] \quad (1-4)$$

Example 1.1:

How much underpressure is in a vessel with a vacuum of 80% and an air pressure of $0.98 \cdot 10^{-5} \text{ Pa}$?

Solution:

According to equation (1-3), the result is:

$$p_u = \text{Vacuum in } \% \cdot p_{air}$$

$$p_u = 0.80 \cdot 98000 \text{ Pa} = 78400 \text{ Pa} = 784.0 [\text{mbar}]$$

Absolute pressure, overpressure and underpressure. A pressure higher than the outside air pressure is called overpressure p_e . A pressure below the outside air pressure is called underpressure p_u . (Fig. 1-4)

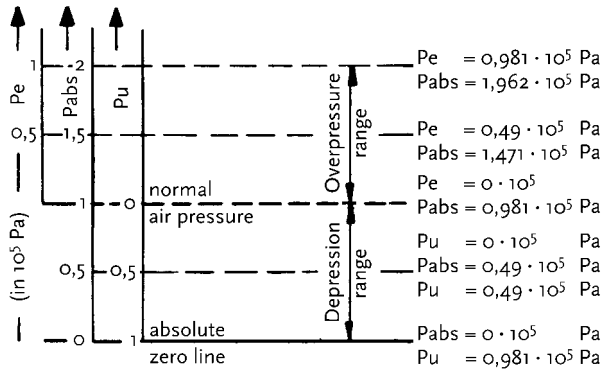


Figure 1-4. Absolute pressure, overpressure and underpressure [1.5]

In technology, the absolute pressure p_{abs} is used for calculations taking the currently prevailing air pressure into consideration.

The absolute pressure p_{abs} is the pressure calculated from the absolute zero line.

$$p_{abs} = p_e + p_{air} \quad \left[\text{Pa}; \frac{\text{N}}{\text{m}^2}; \text{ mbar} \right] \quad (1-5)$$

$$p_{abs} = p_{air} - p_u \quad \left[\text{Pa}; \frac{\text{N}}{\text{m}^2}; \text{ mbar} \right] \quad (1-6)$$

Example 1.2

In a condenser, an underpressure of 650.00 mbar prevails, with the barometer showing 980.00 mbar.

How much is the absolute pressure in mbar, in Pa and the vacuum in percent?

Solution:

Acc. to equation (1-6), the result is:

$$p_{\text{abs}} = p_{\text{air}} - p_{\text{u}}$$

$$p_{\text{abs}} = 980 \text{ mbar} - 650 \text{ mbar} = 330 \text{ [mbar]} = 3,30 \cdot 10^4 \text{ [Pa]}$$

and acc. to equation (1-3) the result is:

$$\text{Vacuum} = \frac{p_{\text{u}}}{p_{\text{air}}} \cdot 100 \text{ [%]}$$

$$\text{Vacuum} = \frac{650 \text{ mbar}}{980 \text{ mbar}} \cdot 100 \text{ [%]}$$

$$\text{Vacuum} = 66,32 \text{ [%]}$$

1.3.5

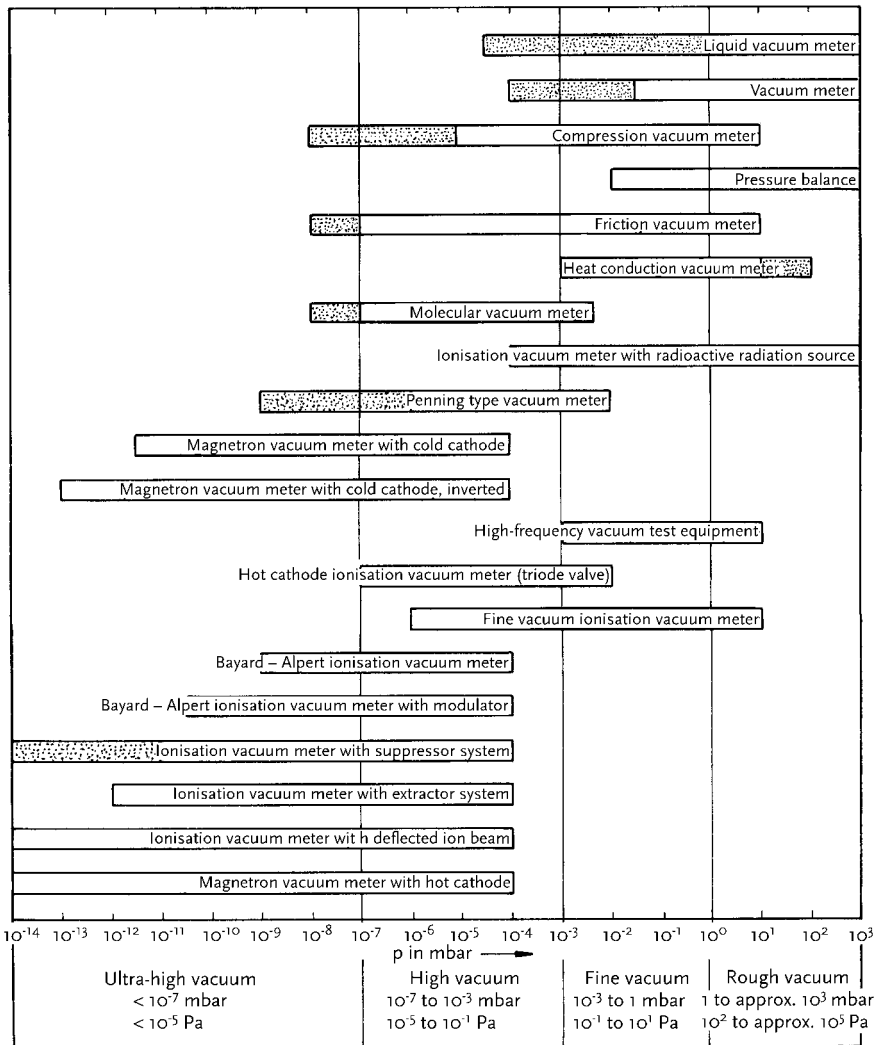
Pressure meters

In order to be able to record the pressure range from 10^5 to 10^{-11} Pa metrologically, various physical basic principles are employed according to which the pressure meters are subdivided.

There are

- mechanical gauges
- heat conduction vacuum manometers
- friction manometers
- radiometer manometers
- ionization manometers.

Up to pressures of 1 Torr, the direct measuring and display of the mechanical force acting upon a surface is carried out by means of mechanical manometers. Pressures below 1 Torr are determined by other measuring methods. Here, physical quantities such as friction, thermal conduction and ionization are measured. The measured quantities can be converted into electric currents or voltages, thus enabling the application of most modern electric and electronic methods of analysis.



The limits of the measuring ranges are without engagement. In the diagram are shown the usual limits.

Measuring range for special execution or for special operating data.

Figure 1-5. Measuring ranges of common vacuum gauges
 (acc. to DIN 28400, Part 3, October 1980)

Mechanical gauges

In the following, only the group of mechanic gauges will be described [1.1]. These devices utilize the action of forces of the pressure for pressure display. To this group includes:

- Bourdon pressure gauge
- Diaphragm gauge
- Modulation meter
- Liquid manometer

Bourdon pressure gauge. This gauge (Fig. 1-6a) is a bent tube made of elastic material with an oval cross section. One tube end is closed while the other one is connected to a vacuum vessel. Owing to different actions of forces onto the upper and lower broadside of the tube (Fig. 1-6b) the tube bends during evacuation. By means of a lever mechanism, the tube deformation is transferred to a pointer.

Features

- Pressure readings independent from kind of gas
- Measuring range: atmospheric pressure up to about 1000 Pa
- special versions for laboratories up to about 1 Pa, e.g. versions made of corrosion-resistant materials (glass or quartz) are known
- Accuracy of readings depends on design, usually relatively low

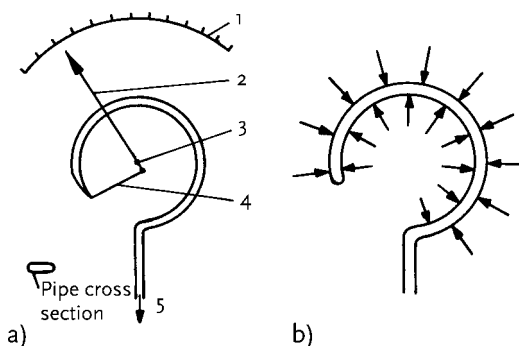


Figure 1-6. Bourdon tube

a) basic design b) force balance

1 scale, 2 pointer, 3 pointer's centre of motion, 4 lever, 5 vacuum vessel

Diaphragm gauge. This type of gauge uses the deformation of a diaphragm (or the changing of the length of a bellows) dividing two ranges of different pressures for pressure display. The deformation is displayed mechanically, optically or electrically (fig. 1-7). As the diaphragm deformation does not occur proportionally to the pressure, gauging devices with linearization appliances are partly used. The reference pressure is mostly selected in the lower pressure range, i.e. in fine and high vacuum, to be independent from altitude, fluctuating air pressure and weather. The readings are independent from temperature and outside air pressure, as the sensing element and the display are not in contact with the measured gas. In these devices, the clearance between the solid wall and the nesting ripple diaphragm, or the so-called pressure capsule in versions with two nesting ripple diaphragms, (acc. to [1.6]) is evacuated (fig. 1-8).

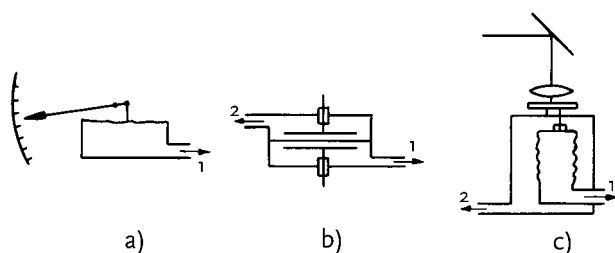


Figure 1-7. Diaphragm vacuum gauge with different kinds of display
 a) mechanical display (pointer) b) electrical display c) optical display
 1 to vacuum vessel, 2 to reference vacuum

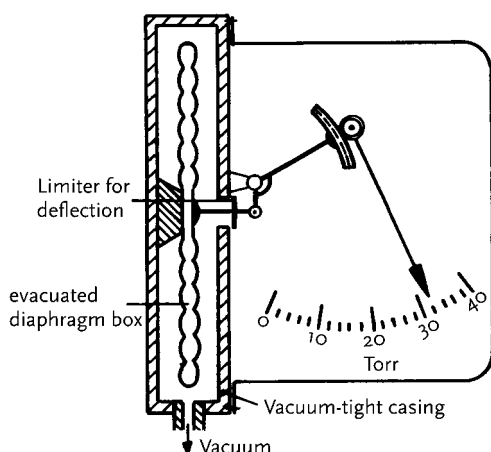


Figure 1-8. Diaphragm vacuum gauge with diaphragm box

Features

- Pressure readings independent from kind of gas
- Measuring range: depending on the versions, from 10^5 Pa to 10^{-2} Pa or for pressure differences of some hundreds of Pa up to 10^{-2} Pa, in laboratory versions up to some 10^{-3} Pa
- Accuracy: depending on the version, in some ranges up to several percent. Owing to the combination with switches, suitable for controlling; with electric display, remote readings and registration are possible.

Modulation manometer. The change of pressure according to the state equation for ideal gases arising from the periodical change of volume is used for the pressure measurement.

$$\Delta p = -p_o \frac{\Delta V}{V} \quad (1-7)$$

p_o average pressure

V_o average volume

$$\Delta V = \Delta V_o \cdot \cos \omega \cdot t$$

(constant gas mass being presupposed)

The periodic volume change can be produced e.g. with a piezo-ceramic measuring transmitter, the periodic pressure change can be detected by means of a sensitive microphone (fig. 1-9).

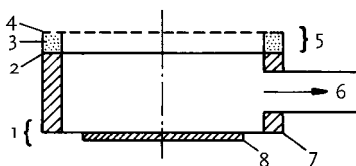


Figure 1-9. Modulation gauge according to Hartung and Jurgeit
1 – modulator, 2 – diaphragm, 3 – insulator, 4 – electrode, 5 – receiver,
6 – vacuum vessel, 7 – metal diaphragm, 8 – piezo-oscillator

Features

- In case of suitable design, pressure display independent from the kind of gas – at least in certain pressure ranges
- Measuring ranges: from atmospheric pressure up to about 10^{-4} Pa
- Accuracy: several percent
- Due to electronics, process control in vacuum, remote indication and registration of pressure are possible

U-tube gauge. The hydrostatic pressure of a liquid column compensating the pressure of the gas serves the pressure measurement.

$$p = \rho \cdot g \cdot h \quad (1-8)$$

ρ liquid density

h liquid column height

g acceleration of free fall

p hydrostatic pressure

The open, simplest type of U-tube gauge (fig. 1-10) is not really common in vacuum engineering, as for the measuring required for the determination of the gas pressure the air pressure has to be read as well. For technical purposes, a shortened closed U-tube gauge has become established (Fig. 1-10b). In order to record the measuring range up to air pressure, a U-tube with a branch length of more than 760 mm

would be required in case of mercury filling. Mercury is mostly used as filling medium. Over the mercury in the closed branch, the Torricelli vacuum exists.

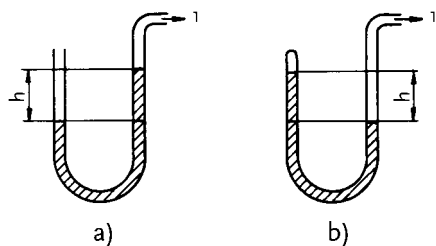


Figure 1-10. U-tube gauge
a) open type b) closed type (1 to vacuum vessel)

Features:

- Pressure readings independent from gas type
- Liquid mostly mercury ($\rho = 13.55 \text{ g/cm}^3$ at $T = 281 \text{ K}$), in some cases dibutylphtalate ($\rho = 1.05 \text{ g/cm}^3$) is used
- Measuring range depending on design, from 10^5 Pa to about 150 Pa with mercury as liquid, and from 10^3 Pa to about 15 Pa with dibutylphtalate
- In case of mercury filling in laboratory versions, switching operations, remote display and registration of pressure can be realized (fig. 1-11)

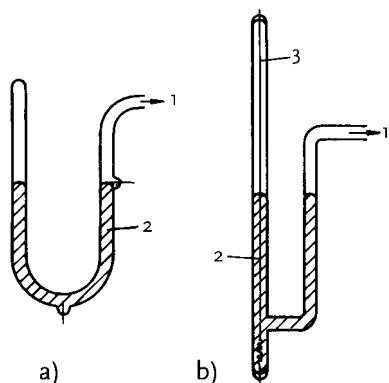


Figure 1-11. U-tube gauge in laboratory version
a) with a switch
b) for pressure registration
1) vacuum vessel
2) mercury
3) resistance wire

Compression gauge acc. to McLeod. With the compression gauge invented by McLeod in 1879, even pressure values below 150 Pa can be measured. Through isothermal compression of a measuring volume V_M onto the compression volume V'_M , the gas pressure is increased to such an extent that it can still be compensated by a mercury column of the readable height h . The McLeod compression gauge shown in Fig 1-12 disposes of a pressure-proportional display.

In the state (a), the measuring volume is connected with the measuring point. If h is the height difference between the mercury menisci in the closed and in the reference capillary after compression, the following formula is applicable:

$$p \approx \rho \cdot g \cdot h \cdot \frac{V'_M}{V_M} \quad (1-9)$$

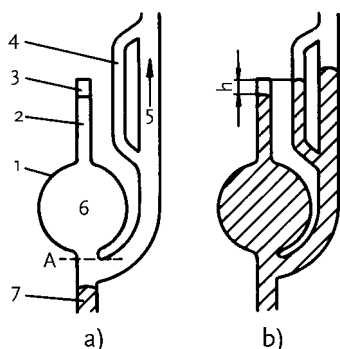


Figure 1-12. McLeod compression gauge with a linear scale

- a) before compression, b) after compression
 1) ball, 2) measuring capillary
 3) compression volume V'_M
 4) corresponding capillary
 5) to vacuum vessel, 6) measuring volume V_M
 7) mercury column

Fig. 1-13 shows a measuring layout in which the mercury in the measuring capillary does not rise up to a fixed mark (so that the known and constant compression volume V'_M is generated), but to the same (fixed) height in the reference capillary as the end of the measuring capillary. Then the mercury column in the measuring capillary shows different heights. For the pressure goes (r =capillary radius):

$$p \approx \rho \cdot g \cdot \pi \cdot r^2 \cdot \frac{h^2}{V_M} \quad (1-10)$$

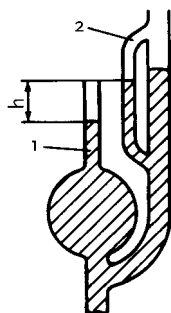


Figure 1-13. McLeod compression gauge with square display
 1 measuring capillary, 2 corresponding capillary tube

Features:

- Pressure readings independent from kind of gas
- Measuring range depends on design (V_M and V'_M sizes)
- Upper measuring limit is up to some hundreds of Pascal, in exceptional cases up to 10^4 Pa; the lowest limit is at about 10^{-2} Pa to 10^{-3} Pa, in special cases up to 10^{-4} Pa

- The measuring is discontinuous
- Filling liquid: mercury of purest quality
- Measuring accuracy decreases with decreasing pressure and achieves values of 50–100% at the lowest measurable pressures (height difference of mercury columns about 1 mm). The compression gauge is not suitable for the measuring of condensable vapors, as its function is based on the application of the equation for ideal gases in thermodynamic balance ($p \cdot V = m \cdot R \cdot T$). If such vapors are contained in the gas to be measured, extremely high measuring errors occur.

1.3.6

Definition of terms for vacuum measuring devices

(extract from DIN 28400, Part 3, October 1980)

Absolute vacuum gauge. An absolute vacuum gauge is an absolute pressure gauge for vacuum.

Absolute pressure gauge, absolute gauge. An absolute pressure gauge is a gauge which is used for the determination of pressure as quotient from the pressure force applied onto a surface and the surface area. An absolute pressure gauge is independent from the kind of gas.

Operation and display device of a vacuum gauge. An operation and display device of the vacuum gauge is the part of a vacuum gauge which contains power supply and appliances for the functioning of the vacuum gauge as well as for pressure readings.

Differential pressure gauge, differential pressure measuring device. A differential pressure gauge is a device measuring the difference between pressures acting on one of both sides of a pressure-sensitive partition surface at the same time, e. g. on an elastic diaphragm or a movable separation liquid. Differential pressure gauges are independent from the kind of gas.

Differential pressure vacuum gauge. A differential pressure vacuum gauge is a differential pressure gauge for vacuum.

Pressure gauge. Manometer. According to the corresponding norm, a pressure gauge is a device for measuring gas and vapor pressures, independent from the pressure range.

Piston manometer. A piston manometer is an absolute vacuum gauge in which the pressure to be measured acts onto a piston-cylinder-combination with a very small clearance in between and a known cross section. The force acting onto the piston is determined by weighing.

Vacuum gauge sensitiveness. The sensitiveness of a vacuum gauge is a quotient of the change of value displayed on a vacuum gauge and the corresponding change of pressure within a small pressure interval. With certain kinds of vacuum gauges, the sensitiveness depends on the kind of gas. In such cases, the kind of gas has to be specified. In case of a lacking specification, the sensitiveness refers to nitrogen. Together with the specification of sensitiveness the operating conditions and pressure ranges must be indicated.

Integrated measuring systems. An integrated measuring system is a vacuum measuring system without a special casing. The pressure-sensitive elements are situated directly in the vacuum vessel.

Elastic spring vacuum gauge. This vacuum gauge is a differential pressure vacuum gauge in which the pressure-sensitive partition is an elastic element. The pressure difference can be determined either by changing of the position of the elastic element (direct method) or through the force needed for maintaining the original position (zero method). (Examples: diaphragm vacuum gauge, Bourdon tube).

Note: Such devices are also used in pressure ranges above vacuum. Then they are called elastic spring manometers or elastic spring barometers.

Liquid vacuum gauge. A liquid vacuum gauge is a differential vacuum gauge in which the pressure-sensitive element is a movable separation liquid (e.g. mercury in a U-tube vacuum gauge). The pressure difference is determined by measuring the differences of the liquid level.

Note: Such devices are even used in pressure ranges above vacuum. Then they are called liquid manometer. A form frequently used is the U-tube manometer.

Compression vacuum gauge. A compression vacuum gauge is a vacuum gauge in which a known volume of a gas is compressed by a known compression ratio at the pressure to be measured (e.g. by moving of a liquid column, usually mercury) and with which the higher pressure resulting from this is measured.

Measuring with a compression vacuum gauge is independent from the kind of gas. In case of existing vapors, condensation processes must be taken into consideration. A well-known example is the McLeod vacuum gauge.

Measuring range of a vacuum gauge. The measuring range of a vacuum gauge is the pressure range in which the inaccuracy of a single pressure reading under defined conditions does not exceed the maximum permissible inaccuracy. For certain kinds of vacuum gauges, this range depends on the type of gas. In such cases, the type of gas has to be specified. In case of lacking specification, the measuring range refers to nitrogen.

Partial pressure vacuum gauge. A partial pressure vacuum gauge is a vacuum gauge for the measuring of partial pressures of the individual components of a gas mixture.

Relative sensitiveness of vacuum gauge. The relative sensitiveness of a vacuum gauge for a certain gas is the quotient resulting from the sensitiveness of this gas and the nitrogen sensitiveness at the same pressure and under the same operating conditions.

Nitrogen equivalent pressure. Nitrogen equivalent pressure is the pressure of pure nitrogen which would cause the same vacuum gauge reading as the pressure of the gas to be measured.

Total pressure vacuum gauge. A total pressure vacuum gauge is a vacuum gauge for measuring the total pressure of a gas or gas mixture.

Vacuum measuring system. Vacuum measuring system is the part of a vacuum gauge consisting of pressure-sensitive elements and connected to the area in which the gas pressure is to be measured.

Some kinds of vacuum measuring systems are also called vacuum measuring tubes, vacuum measuring cells or vacuum measuring heads.

Vacuum gauge; vacuum measuring device. A vacuum gauge is a pressure measuring instrument for vacuum. It can consist of a vacuum measuring system, as well as of an operation and display device.

Note: Some common kinds of vacuum gauges do not measure pressure directly (i.e. acc. to quotients from force and surface), but another physical quantity depending on pressure or particle number.

1.4

Gas flow and vacuum ranges

1.4.1

Vacuum ranges and types of flow

For the designing of pipeworks and vacuum plants, exact knowledge about flow processes is significant, i.e. in vacuum technology gas flows play an important role. In the following, steady flows will be discussed. Non-steady flow processes, as for example in ducts of rotary piston pumps and rotary plunger pumps or gas mass vibration in pipework cannot be described within the framework of this treatise.

Gas flows are subdivided into very low gas velocities, and flow speeds in the range of speed of sound and supersonic speed. In case of low gas-flow velocities, the gas temperature can be regarded as constant, if the temperature is not affected by external influences. For flows in the sound and supersonic range, the laws of gas dynamics are applicable.

In any flow, whether liquid or gas – jointly called fluids – forces that produce, accelerate or delay flows are active. There is a distinction between pressure forces and friction forces; gravity forces may generally be ignored in gas flow.

Flow processes, in which friction forces are small or do not exist at all, are called inviscid flows. In this case, the acceleration or delay of the mass elements of the gas is determined by the pressure forces; here the Bernoulli-equation is applied. Otherwise, the flow is influenced by friction forces. Usually, pressure forces and friction forces are equally high and counteract each other. Friction forces are determined by the inside friction or the viscosity. In these cases, this type of flow is called viscous flow.

1.4.2

Mean free path

For the definition of flow processes in different vacuum ranges, the ratio between the mean free path \bar{l} (Fig. 1-14) of the gas molecules and the flow channel width d (e.g. pipe diameter) is particularly important.

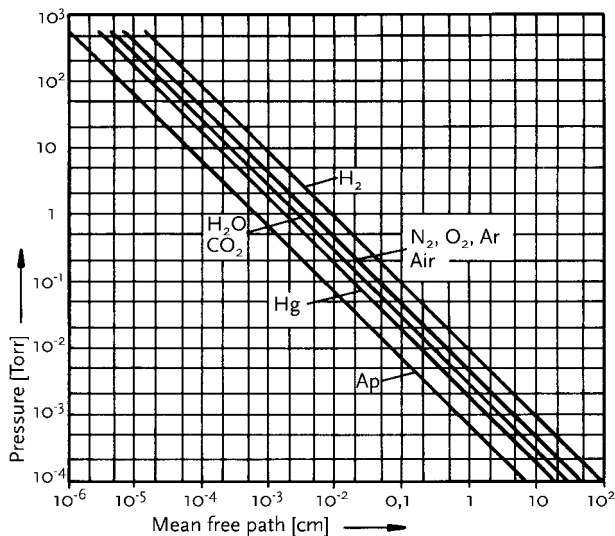


Figure 1-14. Mean free path of some substances at 20 °C [1.6]

Ar	argon	CO ₂	carbon dioxide
H ₂	hydrogen	H ₂ O	water vapor
N ₂	nitrogen	O ₂	oxygen
Hg	mercury	Ap	apiezon oil vapor

This ratio

$$K = \frac{\bar{l}}{d} \tag{1-2}$$

\bar{l} mean free path (m)
d flow channel width

is called Knudsen number.

As shown in table 1-8, the kind of gas flow is determined by the Knudsen number (K) acc. to the pressure ranges.

Table 1-8. Vacuum ranges and kinds of flow [1.7]

Coarse vacuum	Fine vacuum	High-/ultrahigh vacuum
Viscous flow	Knudsen flow	molecular flow

p = gas pressure, d = pipe diameter, \bar{l} = mean free path,
K = \bar{l}/d = Knudsen number, Re = Reynolds number

In coarse vacuum, gas molecules frequently bang together with neighboring molecules, i.e. the molecular structure can be ignored. As the dynamic viscosity is independent from pressure, the flow is viscous. Viscous flow distinguishes between

laminar flow and turbulent flow. The turbulent flow occurs when in the range of higher speeds random flow instead of laminar flow exists.

1.4.3

Reynolds number

Reynolds number Re determines the boundary between laminar and turbulent flow.

$$Re = \frac{\rho \cdot \bar{v} \cdot d}{\eta} \quad (1-11)$$

Re Reynolds number

ρ fluid density [kg/m^3]

\bar{v} average flow speed [m/s]

d characteristic length, e.g. pipe diameter [m]

η dynamic viscosity [$\text{kg}/(\text{m} \cdot \text{s})$]

Viscous flow exists whenever $K < 10^{-2}$.

The critical Reynolds number for round pipes has the value of $Re_{\text{crit}} = 2300$. With this limit value for viscous flow, the laminar flow changes to turbulent flow.

Laminar flow exists with $Re < 2300$

Turbulent flow exists with $Re > 4000$

While in viscous flow, i. e. in the coarse vacuum range, the molecules close to each other continuously touch each other, and the gas flows like a viscous liquid with adhesion to the pipe wall (fig. 1-15) in high and in ultra-high vacuum, i.e. with thinner gas, the gases behave differently. In this flow range, the mean free path \bar{l} of the gas molecules is almost equal to the width of the flow channel, or larger. The number of molecule collisions decreases to such an extent that at a certain lowest pressure collisions of the molecules do not take place any more. Thus the flow is influenced only by the collisions of the gas particles with the pipe wall (fig. 1-16). This type of flow in high and ultra-high vacuum is called molecular flow.

Molecular flow takes place with $K > 0.5$.

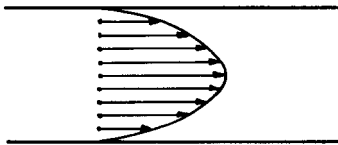


Figure 1-15. Scheme of laminar flow [1.6]

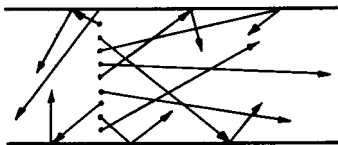


Figure 1-16. Scheme of molecular flow [1.6]

Turbulent flow is characterized by a strongly flattened speed profile and strongly decreasing boundary layers near the pipe wall.

Laminar flow shows a parabolic speed course formed across the cross section with a maximum in the middle and speed decreasing to zero near the pipe wall.

In the *molecular flow*, an abrupt velocity jump occurs near the pipe wall as soon as the mean free path increases compared to pipe diameter at very small pressures.

In practice, turbulent-laminar and laminar-molecular types of flow occur between the three typical types of flow (Fig. 1-17).

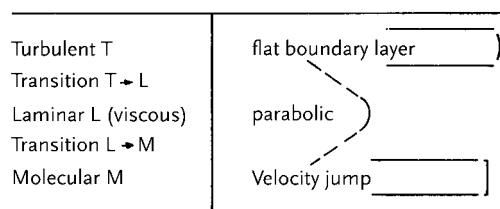


Figure 1-17. The three typical types of flow and their limits [1.2]

The range between viscous flow and molecular flow is called Knudsen flow, as defined acc. to DIN 28400. This type of flow occurring in fine vacuum is also called Knudsen number.

Knudsen flow occurs when $0.5 > K > 0.01$.

1.4.4

Gas flow, suction power, suction capacity [1.7]

The gas quantity, gas mass, the quantity of volume, particle numbers or substances ΔM flowing through a definite cross section within a definite period of time Δt is called flow rate q .

$$q = \frac{\Delta M}{\Delta t} \quad (1-12)$$

Therefore, we get

$$\text{volume flow rate} \quad q_v = \frac{\Delta V}{\Delta t} = \dot{V} \left[\frac{\text{m}^3}{\text{s}} \right] \quad (1-13)$$

$$\text{mass flow rate} \quad q_m = \frac{\Delta m}{\Delta t} = \dot{m} \left[\frac{\text{kg}}{\text{s}} \right] \quad (1-14)$$

$$\text{amount of substance flow rate} \quad q_v = \frac{\Delta V}{\Delta t} = \dot{v} \left[\frac{\text{mol}}{\text{s}} \right] \quad (1-15)$$

$$\text{particle number flow rate} \quad q_N = \frac{\Delta N}{\Delta t} = \dot{N} \left[\frac{1}{\text{s}} \right] \quad (1-16)$$

Instead of flow rate, the terms volume flow, mass flow, molecular flow and particle number flow can be used.

For ideal gases in a steady condition where pressure and temperature remain constant acc. to the general state equation of ideal gas, the following is applicable:

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

for time period Δt

$$p \cdot \frac{\Delta V}{\Delta t} = p \cdot \dot{V} = \frac{\Delta m}{\Delta t} \cdot R \cdot T$$

or

$$p \cdot \frac{\Delta V}{\Delta t} = \dot{m} \cdot R \cdot T = q_m \cdot R \cdot T \quad (1-18)$$

Substance mass flow rate \dot{m} is proportional to the product $p \cdot \dot{V}$.

The $p \cdot \dot{V}$ -flow rate or the $p \cdot \dot{V}$ -flow

$$q_{pV} = p \cdot \dot{V} \left[\frac{\text{Pa} \cdot \text{m}^3}{\text{s}}; \frac{\text{mbar} \cdot \text{l}}{\text{s}} \right] \quad (1-19)$$

is proportional to the substance mass flow rate \dot{m} .

The gas volume (= volume flow rate) flowing into a suction nozzle of a vacuum pump and related to the time and surface of the suction nozzle is called suction volume flow rate \dot{S} , which is the suction capacity of a vacuum pump.

$$\dot{S} = \dot{V}_{\text{pump}} = q_{V\text{pump}} \left[\frac{\text{m}^3}{\text{h}}; \frac{\text{m}^3}{\text{s}}; \frac{\text{l}}{\text{s}} \right] \quad (1-20)$$

On the other hand, the $p \cdot \dot{V}$ flow rate at the pump inlet is called suction power \dot{Q} of the vacuum pump.

$$\dot{Q} = p \cdot \dot{V}_{\text{pump}} = q_{pV\text{pump}} \left[\frac{\text{Pa} \cdot \text{m}^3}{\text{s}}; \frac{\text{mbar} \cdot \text{l}}{\text{s}} \right] \quad (1-21)$$

From equations (1-20) and (1-21), follows:

$$\dot{Q} = p \cdot \dot{S} \left[\frac{\text{mbar} \cdot \text{l}}{\text{s}} \right] \quad (1-22)$$

In vacuum pumps with a suction capacity independent from the intake pressure – this is the case in many vacuum pumps within a certain pressure range – the suction power \dot{Q} decreases proportionally to the intake pressure. The suction power \dot{Q} decreasing with lower pressure and having the same pump suction capacity can be explained with the fact that with smaller intake pressures fewer molecules per volume unit exist.

The characteristic curve of a vacuum pump with a constant suction capacity within a determined pressure range is shown in the fig. 1-18. Fig. 19 shows the suction power curve of a vacuum pump with constant suction capacity.

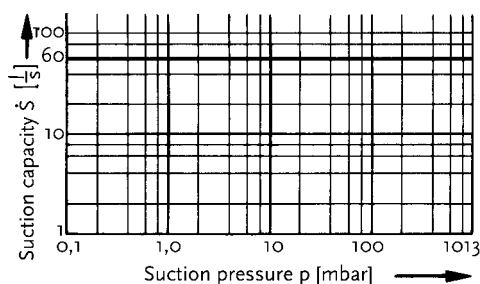


Figure 1-18. Characteristic curve, vacuum pump with constant suction capacity of 60 l/s within the pressure range of 1013 mbar to 0.10 mbar

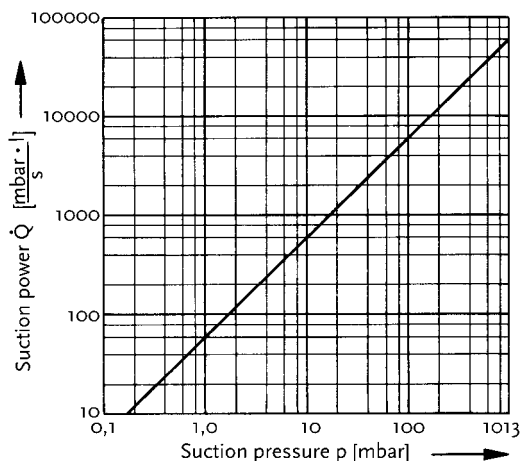


Figure 1-19. Characteristic curve with suction power of a vacuum pump with a constant suction capacity of 60 l/s within the pressure range from 1013 mbar to 0.10 mbar

1.4.5

Flow losses in pipework [1.7]

Vacuum pumps are usually connected with the vacuum vessels via pipes (fig.1-20).

With gas flow q flowing through a pipe, a flow resistance occurs in the pipe resulting in a pressure difference

$$\Delta p = p_K - p_A > 0 \quad (1-23)$$

between the pipe ends.

As in electrical engineering, even in vacuum technology the ratio $\Delta p/q$ is called resistance or more precisely, flow resistance W .

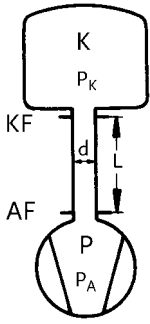


Figure 1-20. Vacuum pump with suction pipe and vacuum vessel Vessel K (pressure p_K) and pump P (pressure p_A). Pressure on suction flange AF equal to p_A . Vacuum vessel flange KF.

$$W = \frac{\text{difference in pressure}}{\text{flow rate}} = \frac{\Delta p}{q} \quad (1-24)$$

The flow resistance in vacuum pipes depends on the pressure difference or gas flow rate. The reciprocal value of the resistance is called conductivity L , just like in electrical engineering. In laminar ranges for different gases, conductivities behave like their viscosities. In the molecular range, conductivities depend only on the temperature and the molecular mass.

$$L = \frac{1}{W} = \frac{q}{\Delta p} \quad (1-25)$$

In vacuum technology, resistance or conductivity can be different quantities depending on the flow rate applied.

If the $p \cdot \dot{V}$ flow rate is used, the following results:

$$W = \frac{\Delta p}{q_{pV}} = \frac{\text{Pa}}{\frac{\text{Pa} \cdot \text{m}^3}{\text{s}}} \left[\frac{\text{s}}{\text{m}^3} \right] \quad (1-26)$$

and

$$L = \frac{q_{pV}}{\Delta p} = \frac{\frac{\text{Pa} \cdot \text{m}^3}{\text{s}}}{\text{Pa}} \left[\frac{\text{m}^3}{\text{s}} \right] \quad (1-27)$$

With volume flow rate \dot{S} , the equation is:

$$W = \frac{\Delta p}{q_V} = \frac{\text{Pa}}{\frac{\text{m}^3}{\text{s}}} \left[\frac{\text{Pa} \cdot \text{s}}{\text{m}^3} \right] \quad (1-28)$$

and

$$L = \frac{q_V}{\Delta p} = \frac{\frac{\text{m}^3}{\text{s}}}{\text{Pa}} \left[\frac{\text{m}^3}{\text{Pa} \cdot \text{s}} \right] \quad (1-29)$$

With a series connection of electrical resistances, the single resistances are added up, whereas with a parallel connection, the reciprocals of the electrical resistances, i.e. of the conductivities are added up.

These rules were applied analogously to the gas flow.

Series connection

$$W = W_1 + W_2 + W_3 + \dots \quad (1-30)$$

$$\frac{1}{L} = \frac{1}{L_1} + \frac{1}{L_2} + \frac{1}{L_3} + \dots \quad (1-31)$$

Parallel connection

$$L = L_1 + L_2 + L_3 + \dots \quad (1-32)$$

$$\frac{1}{W} = \frac{1}{W_1} + \frac{1}{W_2} + \frac{1}{W_3} + \dots \quad (1-33)$$

In contrast to the electrical calculation of resistances, the calculation of the gas flow resistance shows slight deviations. This difference results from the fact that the gas flow in a pipe is not constant throughout the whole length. Therefore, in case of very short pipes the total resistance is not proportional to the pipe length due to the inlet flow. Strictly speaking, for the calculation e.g. in case of series connections, respective transition resistances have to be considered.

For the calculation of several pipework systems connected to each other, for example, a subdivision in individual sections is recommendable.

The calculation method described above is applicable for steady flow processes. In calculations of non-steady flows, complex resistances are used. The pipe volumes can be compared to the capacity of condensers and the inertia of the flowing gas masses to the inductivity of coils, according to the electric calculations.

1.4.6

Effective suction capacity of vacuum pumps

In a connecting pipe between vacuum chamber and vacuum pump, the gas flow, or at a constant temperature, the $p \cdot \dot{V}$ flow through each cross-section of the pipe is the same, as according to the continuity equation gas particles cannot get lost or be created anywhere. Considering the gas flow from a chamber with a higher gas density to another chamber with lower gas density, flow practically occurs only in direction of the pressure drop, as due to the higher density at the beginning of the pipe there are more moving molecules than at the end of the pipe on the gas outlet side. This is the explanation for an increasing flow rate owing to an increased gas temperature; even here the molecular velocity is higher than at low temperatures.

If, for example, in front of a vacuum pump with flow rate \dot{S} and intake pressure p , a valve is installed that increases the line resistance (fig. 1-21), then, according to the rules described above, a so-called effective suction rate on the suction side of the installed valve is an inevitable result. [1.6]

Referring to the vessel flange KF and pump flange AF in fig. 1-20, the following correlation results acc. to [1.7]:

$$q_{pV} = p_K \cdot \dot{V}_K = p_A \cdot \dot{V}_A \quad (1-34)$$

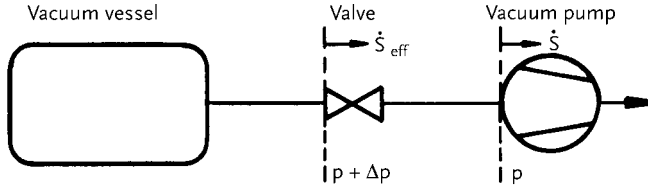


Figure 1-21. Scheme vacuum pump with valve installed on the suction side

$$\dot{V}_A = \dot{S} = \text{suction capacity of the pump} \quad (1-35)$$

$$p \cdot \dot{V} = \dot{Q} = \text{suction power of the pump} \quad (1-36)$$

If

$$\dot{V}_K = \dot{S}_K \quad (1-37)$$

\dot{S}_K = effective suction capacity on vessel flange then the following will result from equation (1-34):

$$\dot{S}_K = \frac{p_A}{p_K} \cdot \dot{S} \quad (1-38)$$

Therefore \dot{S}_K is smaller than \dot{S} , as already stated above, because in order to maintain the flow through the pipe, $p_K > p_A$.

With regard to the continuity equation, however, the suction power $\dot{Q} = \dot{Q}_K$ remains constant.

According to the equations

$$L = \frac{1}{W} = \frac{q}{\Delta p} \quad (1-25)$$

$$q_{pV} = p_K \cdot \dot{V}_K = p_A \cdot \dot{V}_A \quad (1-34)$$

$$\dot{V}_K = \dot{S}_K \quad (1-37)$$

the following correlation between pressure ratio p_K/p_A , suction capacity \dot{S} and conductivity L results:

$$\frac{p_K}{p_A} = 1 + \frac{\dot{S}}{L} \quad (1-39)$$

and from the equation (1-38), the effective suction capacity

$$\frac{1}{\dot{S}_K} = \frac{1}{\dot{S}} + \frac{1}{L} \quad (1-40)$$

or

$$\dot{S}_K = \frac{\dot{S}}{1 + \frac{\dot{S}}{L}} \quad (1-41)$$

In fig. 1-22, the relation between the effective suction capacity \dot{S}_K , the actual suction capacity \dot{S} of the vacuum pump, and conductivity L is shown acc. to equation (1-40).

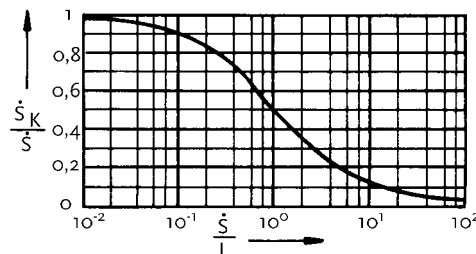


Figure 1-22. Pump efficiency \dot{S}_K / \dot{S} in relation to the ratio \dot{S}/L [1.7]

\dot{S}_K effective suction capacity on the vacuum vessel flange

\dot{S} suction capacity of the pump

L flow conductivity

It is apparent that a pump efficiency $\dot{S}_K / \dot{S} \approx 0.9 \approx 90\%$ can only be achieved if the conductivity is 10 times as high as the suction capacity of the vacuum pump. If conductivity $L \ll \dot{S}$, then acc. to the equation

$$\dot{S}_K = \frac{\dot{S}}{1 + \frac{L}{\dot{S}}} \quad (1-41)$$

it results $\dot{S}_K \approx L$, i.e. the effective suction capacity is influenced only by conductivity and no longer by the suction capacity of the pump. From this derives that with flow losses in the vacuum pipe being too high even vacuum pumps with any form of suction capacity are not able to increase the effective suction capacity. This principle should always be taken into consideration when planning vacuum pipes and systems

The rule says that a vacuum pump can only be operated at a capacity of up to 90% if the conductivity of the respective pipe is 10 times as high as the suction capacity of the pump.

A pump utilization of 50% can be achieved acc. to [1.7], if the conductivity is at least as high as the suction capacity of the pump.

1.4.7

Gas-inflow and outflow on a vacuum chamber

According to fig. 1-23, gas inflow and gas outflow at a vacuum vessel is distinguished [1.1]

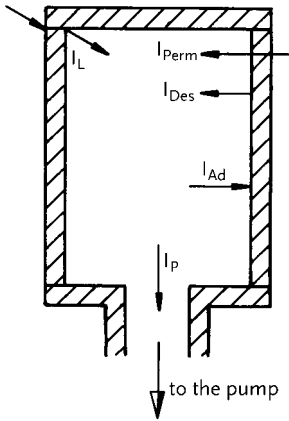


Figure 1-23. Gas inlet and outlet flow of the vacuum vessel

Inflow I_E

- Leakage gas flow I_L
- Permeation gas flow I_{Perm}
- Desorption gas flow I_{Des}

Outflow I_A

- Adsorption flow I_{Ad}
- Gas flow to the pump I_P

Adsorption and desorption are of particular importance in vacuum processes (comp. fig. 1-24).

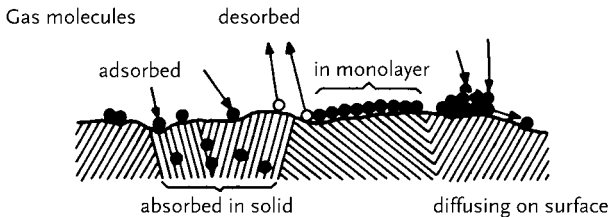


Figure 1-24. Sorption on interfaces of vacuum vessels [1.6]

The higher the vacuum in a chamber is, the greater is the influence of foreign substances on inner surfaces on evacuation. Coarse contaminations can be removed by mechanical cleaning and rinsing. The deeper penetration of adsorbed molecules in a material is called absorption. Adsorption is the intake of molecules, desorption is the discharge.

From a molecular point of view, sorption is an exchange process of different molecules on solid and liquid boundary surfaces of the vacuum chamber. The exchange on the surfaces is accelerated by increasing the temperature. By means of cooling,

for example in working processes in vacuum, gas discharge from surfaces can be diminished or even totally prevented.

Examination of the pumping process in time units

For a constant volume of the vacuum chamber, a differential equation for pumping process can be elaborated according to [1.1]

$$\tau = \frac{V_0}{\dot{S}_{\text{peff}}} \quad (1-42)$$

$$p_e = \frac{I_E}{\dot{S}_{\text{peff}}} \quad (1-43)$$

The differential equation is:

$$V_0 \cdot \left(\frac{dp}{dt} \right) = -p \cdot \dot{S}_{\text{peff}} + I_E \quad (1-44)$$

For constant \dot{S}_{peff} , the following solution results:

$$p - p_e = (p_0 - p_e) \cdot e^{-\frac{t}{\tau}} \quad (1-45)$$

or

$$p = (p_0 - p_e) \cdot e^{-\frac{t}{\tau}} + p_e \quad (1-46)$$

V_0	Volume of vacuum vessel
\dot{S}_{peff}	suction capacity of the pump acting effectively at the vacuum vessel
I_E	Inflow into the vacuum vessel
p	pressure in the vacuum vessel
p_0	Starting pressure at the time $t=0$ (initial state)
p_e	Stationary end pressure of the system for $t \rightarrow \infty$
t	pumping-out time
τ	time constant of the system

With a suction capacity \dot{S}_{peff} supposed to be constant at the vacuum vessel, the pressure p prevailing in the vacuum vessel can be determined for each period of evacuation.

If the suction capacity \dot{S}_{peff} is not constant throughout the whole pressure range, this calculation has to be carried out gradually by means of iteration with the suction capacity corresponding to the pressure ranges.

1.4.8

Practice oriented application of the gas flow calculation [1.6]

For the calculation of flow resistances, it is necessary to determine the existing type of flow. In order to ascertain whether laminar or turbulent flow exists in the coarse

and fine vacuum range, the so-called Reynolds number can be calculated. The Reynolds number Re is a dimensionless number calculated with the help of the known formula:

$$Re = \frac{\rho \cdot \bar{v} \cdot d}{\eta} \quad (1-11)$$

For dry air at 20 °C temperature, the following formula is applicable in practice:

$$Re = 11.2 \cdot \frac{\dot{Q}}{d} \quad (1-47)$$

\dot{Q} flow power [(Torr · l)/s]
 d pipe diameter [cm]

Laminar flow takes place if

$$\frac{\dot{Q}}{d} > 200$$

Turbulent flow is to be expected if

$$\frac{\dot{Q}}{d} > 360$$

For values between 200 and 360, i.e. in the transition range between laminar and turbulent flow, the flow type is significantly influenced by the existing marginal conditions.

The calculation of the resistance of vacuum pipes is based on the following known consideration.

The flow power in a pipe is directly proportional to the pressure difference $\Delta p = p_1 - p_2$ between the beginning and the end of the pipe and inversely proportional to the pipe resistance W .

For dry air at 20 °C, the resistances of round pipes can be determined by the following formulas:

$$\bar{p} = \frac{p_1 + p_2}{2} \text{ [Torr]} \quad (1-48)$$

\bar{p} mean pressure [Torr]
 p_1 pressure at the beginning of the pipe [Torr]
 p_2 pressure at the pipe end [Torr]

$$W_{\text{laminar}} = \frac{a}{182 \cdot d^4 \cdot \bar{p}} \quad \left[\frac{\text{s}}{\text{l}} \right] \quad (1-49)$$

$$W_{\text{molecular}} = \frac{a}{12.1 \cdot d^3} \quad \left[\frac{\text{s}}{\text{l}} \right] \quad (1-50)$$

W resistance [s/l]
 a pipe length [cm]
 d pipe diameter [cm]

For components without normal pipe cross-sections, the calculation in case of molecular flow can be carried out by replacing these components (e. g. valves or control devices) by pipe sections of the respective rated width and of a length approximately corresponding to the gas flow path inside the component.

The calculation of the resistance can be also carried out according to the diagram in the fig. 1-25.

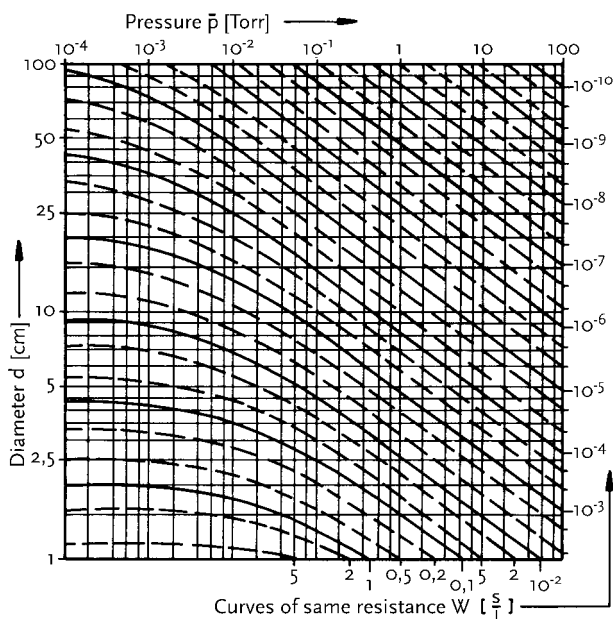


Figure 1-25. Flow resistance of a pipe of 100 cm length, for dry air at room temperature. [1.6]

The diagram in fig. 1-25 shows resistance curves of a pipe with a length of 100 cm and a clear width diameter of 1 to 100 cm, as a function of the mean pressure in the middle of the pipe section.

$$\bar{p} = \frac{p_1 + p_2}{2} \text{ [Torr]} \quad (1-48)$$

From fig. 1-25 follows that in the range of laminar flow the line resistance continuously increases with falling pressure. During the transition to the molecular flow, a constant maximum value is achieved at smaller pressures in the left half of the diagram. Furthermore, it is apparent that the maximum line resistances at small pipe diameters already occur at pressures that are not so low.

Example 1.3

For dry air at room temperature, the flow resistance in a pipe with a clear width of 10 cm and a length of 100 cm can be determined for pressures of 10^{-1} Torr and $3 \cdot 10^{-3}$ Torr with the help of the diagram shown in fig. 1-25.

Pressure \bar{p} in the pipe (mean pressure in the pipe)	Flow resistance W in the pipe
$\bar{p} = 10^{-1}$ [Torr]	$W = 6.8 \cdot 10^{-4}$ [s/l]
$\bar{p} = 3 \cdot 10^{-3}$ [Torr]	$W = 7.00 \cdot 10^{-3}$ [s/l]

Calculation of the resistance of short pipes

For short pipes with a length of less than 1m, the resistance cannot be calculated proportionally, because even with very small pipe lengths or orifices in a thin wall (where the maximum diameter of the orifice may be only 1/10 of the space before it), finite resistance remains.

With the diameter d at the orifice, the resistance for the medium dry air at 20 °C can be calculated according to the following formulas:

$$W_{\text{laminar}} = \frac{1}{15.7 \cdot d^2} \quad \left[\frac{\text{s}}{\text{l}} \right] \quad (1-51)$$

$$W_{\text{molecular}} = \frac{1}{9 \cdot d^2} \quad \left[\frac{\text{s}}{\text{l}} \right] \quad (1-52)$$

d diameter (cm)

The resistances ascertained by the formulas and the diagram above are valid for dry air of 293 K. These values determined for air can be converted for other gases and temperatures with the help of the formulas given below, with the air-specific values being multiplied by the factor C .

For molecular flow, the following formula is applicable:

$$C_{\text{mol}} = 3.18 \cdot \sqrt{\frac{M}{T}} \quad (1-53)$$

M molecular mass [kg/kmol]

T thermodynamic temperature [K]

For laminar flow, the following formula is applicable:

$$C_{\text{lam}} = \frac{\eta_{\text{gas}}}{\eta_{\text{air}}} \quad (1-54)$$

η dynamic viscosity [micropoise]

Table 1-9. Dynamic viscosity of different gases at 293 K

Gas	H ₂	He	H ₂ O	Air	Ar	CO ₂	Hg
η_{293} [micropoise]	88	196	97	181	221	146	280

At temperatures T deviating from 293 K, the viscosity values (comp. Table 1-9) referring to 293 K have to be multiplied by the root of the temperature ratio.

$$\eta_T = \eta_{293} \cdot \sqrt{\frac{293}{T}} \quad (1-55)$$

With increasing temperature the line resistance usually decreases.

Example 1.4:

The following is to be determined:

Resistances of a pipe with a length of 2 m and a diameter of 5 cm for

- 1) air and
- 2) water vapor

at pressures of 10^{-4} Torr and 1 Torr at 20 °C and 150 °C.

1) Solution for air

- a) $p = 10^{-4}$ Torr
 $t = 20$ °C

$$\bar{l}_{\text{air}} = 66 \text{ cm} \quad (\text{acc. to figure 1-14})$$

$$K = \frac{\bar{l}}{d} \quad (\text{equation 1-2})$$

$$K = \frac{66 \text{ cm}}{5 \text{ cm}} = 13.2$$

$K > 0.5$, i.e. molecular flow

$$W_{\text{mol}} = \frac{a}{12.1 \cdot d^3} \quad (\text{equation 1-50})$$

$$W_{\text{mol}} = \frac{200}{12.1 \cdot 125} = 1.32 \cdot 10^{-1} \left[\frac{\text{s}}{\text{l}} \right]$$

- b) $p = 10^{-4}$ Torr
 $t = 150$ °C

$$C_{\text{mol}} = 3.18 \cdot \sqrt{\frac{M}{T}} \quad (\text{equation 1-53})$$

$$M_{\text{air}} = 29 \cdot \frac{\text{kg}}{\text{kmol}}$$

$$T = 423 \text{ K}$$

$$C_{\text{mol}} = 3.18 \cdot \sqrt{\frac{29}{423}} = 0.832$$

$$W = 1.32 \cdot 10^{-1} \left[\frac{\text{S}}{\text{I}} \right] \cdot 0.832$$

$$W_{\text{mol}} = 1.10 \cdot 10^{-1} \left[\frac{\text{S}}{\text{I}} \right]$$

c) $p = 1 \text{ Torr}$
 $t = 20^\circ\text{C}$

$$\bar{I}_{\text{air}} = 6.4 \cdot 10^{-3} \quad (\text{acc. to figure 1-14})$$

$$K = \frac{\bar{I}}{d} \quad (\text{equation 1-2})$$

$$K = \frac{0.0064 \text{ cm}}{5 \text{ cm}} = 0.00128$$

$K < 10^{-2}$, i.e. viscous respectively laminar flow

$$W_{\text{lam}} = \frac{a}{182 \cdot d^4 \cdot \bar{p}} \quad (\text{equation 1-49})$$

$$W_{\text{lam}} = \frac{200}{182 \cdot 625 \cdot 1} = 1.76 \cdot 10^{-3} \left[\frac{\text{S}}{\text{I}} \right]$$

d) $p = 1 \text{ Torr}$
 $t = 150^\circ\text{C}$

$$\eta_T = \eta_{293} \cdot \sqrt{\frac{293}{T}} \quad (\text{equation 1-55})$$

$$T = 423 \text{ K}$$

$$\eta_{\text{air}} = 181 \text{ micropoise}$$

$$\eta_T = 181 \cdot \sqrt{\frac{293}{423}} = 150.56$$

$$C_{\text{lam}} = \frac{\eta_{\text{gas}}}{\eta_{\text{air}}} \quad (\text{equation 1-54})$$

$$C_{\text{lam}} = \frac{150.56}{181} = 0.832$$

$$W = 1.76 \cdot 10^{-3} \left[\frac{\text{S}}{\text{I}} \right] \cdot 0.832$$

$$W_{\text{lam}} = 1.47 \cdot 10^{-3} \left[\frac{\text{S}}{\text{I}} \right]$$

2) Solution for water vapor

a) $p = 10^{-4} \text{ Torr}$
 $t = 20^\circ \text{C}$

$$\bar{l}_{\text{H}_2\text{O}} = 50 \text{ cm} \quad (\text{acc. to figure 1-14})$$

$$K = \frac{\bar{l}}{d} \quad (\text{equation 1-2})$$

$$K = \frac{50 \text{ cm}}{5 \text{ cm}} = 10.0$$

$K > 0.5$, i.e. molecular flow

$$C_{\text{mol}} = 3.18 \cdot \sqrt{\frac{M}{T}} \quad (\text{equation 1-53})$$

$$M_{\text{H}_2\text{O}} = 18 \frac{\text{kg}}{\text{kmol}}$$

$$T = 293 \text{ K}$$

$$C_{\text{mol}} = 3.18 \cdot \sqrt{\frac{18}{293}} = 0.7876$$

$$W = 1.32 \cdot 10^{-1} \frac{\text{S}}{\text{I}} \cdot 0.7876$$

$$W_{\text{mol}} = 1.04 \cdot 10^{-1} \left[\frac{\text{S}}{\text{I}} \right]$$

b) $p = 10^{-4} \text{ Torr}$
 $t = 150^\circ \text{C}$

$$C_{\text{mol}} = 3.18 \cdot \sqrt{\frac{M}{T}} \quad (\text{equation 1-53})$$

$$M_{\text{H}_2\text{O}} = 18 \frac{\text{kg}}{\text{kmol}}$$

$$T = 423 \text{ K}$$

$$C_{\text{mol}} = 3.18 \cdot \sqrt{\frac{18}{423}} = 0.656$$

$$W = 1.32 \cdot 10^{-1} \frac{\text{S}}{\text{I}} \cdot 0.656$$

$$W_{\text{mol}} = 8.7 \cdot 10^{-2} \left[\frac{\text{S}}{\text{I}} \right]$$

c) $p = 1 \text{ Torr}$
 $t = 20^\circ \text{C}$

$$\bar{l}_{\text{H}_2\text{O}} = 4.5 \cdot 10^{-3} \text{ cm} \quad (\text{acc. to figure 1-14})$$

$$K = \frac{\bar{l}}{d} \quad (\text{equation 1-2})$$

$$K = \frac{0.0045 \text{ cm}}{5 \text{ cm}} = 0.0009$$

$$K < 10^{-2}, \text{ i.e. viscous respectively laminar flow}$$

$$C_{\text{lam}} = \frac{\eta_{\text{gas}}}{\eta_{\text{air}}} \quad (\text{equation 1-54})$$

$$\eta_{\text{H}_2\text{O}} = 97 \text{ micropoise}$$

$$\eta_{\text{air}} = 181 \text{ micropoise}$$

$$C_{\text{lam}} = \frac{97}{181} = 0.535$$

$$W = 1.76 \cdot 10^{-3} \frac{\text{S}}{\text{I}} \cdot 0.535$$

$$W_{\text{lam}} = 9.4 \cdot 10^{-4} \left[\frac{\text{S}}{\text{I}} \right]$$

d) $p = 1 \text{ Torr}$
 $t = 150^\circ \text{C}$

$$\eta_T = \eta_{293} \cdot \sqrt{\frac{293}{T}} \quad (\text{equation 1-55})$$

$$T = 423 \text{ K}$$

$$\eta_{\text{H}_2\text{O}} = 97 \text{ micropoise}$$

$$\eta_T = 97 \cdot \sqrt{\frac{293}{423}} = 80.69$$

$$C_{\text{lam}} = \frac{\eta_{\text{gas}}}{\eta_{\text{air}}} \quad (\text{equation 1-54})$$

$$C_{\text{lam}} = \frac{80.69}{181} = 0.445$$

$$W = 1.76 \cdot 10^{-3} \frac{\text{s}}{\text{l}} \cdot 0.445$$

$$W_{\text{lam}} = 7.836 \cdot 10^{-4} \frac{\text{s}}{\text{l}}$$

Summary of results

20 °C (293 K)			150 °C (423 K)		
1) air [Torr]	resistance [s/l]	C_{mol}	η T	C_{lam}	resistance [s/l]
10^{-4}	a) $1.32 \cdot 10^{-1}$	0.832	–	–	b) $1.1 \cdot 10^{-1}$
1	c) $1.76 \cdot 10^{-3}$	–	150.56	0.832	d) $1.47 \cdot 10^{-3}$
2) H ₂ O					
10^{-4}	a) $1.04 \cdot 10^{-2}$	0.656	–	–	b) $8.7 \cdot 10^{-2}$
1	c) $9.4 \cdot 10^{-4}$	–	80.69	0.445	d) $7.83 \cdot 10^{-4}$

Example 1.5

A vacuum pump shall suck off from a vacuum chamber in the high vacuum range. The suction line is to be regarded as an orifice in a thin wall.

- 1) Which is the maximum possible suction capacity of the pump to enable the suction of all molecules flowing through the orifice?
- 2) Which is the specific suction capacity of the pump with reference to the suction cross-section?

Solution:

As the pump is operating in the high vacuum range, the calculation can be based on the molecular type of flow.

- 1) Flow resistance can be determined by following formula

$$W = \frac{1}{9 \cdot d^2} \left[\frac{\text{s}}{\text{l}} \right] \quad (1-52)$$

For the flow capacity, the following relation exists:

$$W = \frac{\Delta p}{q_{\text{pv}}} \left[\frac{\text{s}}{\text{l}} \right] \quad (1-26)$$

Consequently,

$$q_{\text{pv}} = \Delta p \cdot 9 \cdot d^2 \left[\frac{\text{Torr} \cdot \text{l}}{\text{s}} \right] \quad (1-56)$$

From the equation

$$\dot{Q} = p \cdot \dot{S} = q_{\text{pv}} \left[\frac{\text{Torr} \cdot \text{l}}{\text{s}} \right] \quad (1-22)$$

the result is

$$\dot{S} = \frac{q_{pV}}{p} \left[\frac{1}{s} \right] \quad (1-57)$$

and as $p = \Delta p$:

$$\dot{S} = \frac{\Delta p}{\Delta p} \cdot 9 \cdot d^2 \left[\frac{1}{s} \right]$$

and therefore

$$\dot{S} = 9 \cdot d^2 \left[\frac{1}{s} \right] \quad (1-58)$$

2) The suction capacity referred to the circular cross-section is calculated according to the equation

$$\dot{S} = 9 \cdot d^2 \left[\frac{1}{s} \right] \quad (1-58)$$

Through multiplication by the factor $\pi/4$ the result is

$$\dot{S} \cdot \frac{\pi}{4} = 9 \cdot d^2 \frac{\pi}{4}$$

with the relation

$$F = \frac{d^2 \cdot \pi}{4} \left[\text{cm}^2 \right] \quad (1-59)$$

d diameter [cm]

The calculation is:

$$\dot{S} \cdot \pi / 4 = 9 \cdot F$$

and the specific suction capacity is:

$$\frac{\dot{S}}{F} = 9 \cdot \frac{4}{\pi} \left[\frac{1}{s \cdot \text{cm}^2} \right] \quad (1-60)$$

$$\frac{\dot{S}}{F} = 11.46 \left[\frac{1}{s \cdot \text{cm}^2} \right]$$

$\frac{\dot{S}}{F} = S_{\text{spec}}$ = specific suction capacity referring to the inlet cross-section of the vacuum pump.

1.5

Physical states of matter

1.5.1

The terms gases, vapors, vacuum

The three physical states of matter of a substance - solid, liquid and gaseous – differ as follows:

In the solid state of matter, strong forces of attraction exist between the particles, i.e. the distance between these particles is relatively small.

In the liquid state of matter, there are larger distances between these particles which are able to move relatively to each other. The forces of attraction are smaller compared to the solid state of matter.

In the gaseous state of matter, there aren't any forces of attraction between the molecules or atoms in the ideal state. In this case, the gas is called "ideal gas".

In practice, in vacuum technology there are pure gases or pure vapors, or mixtures of gases and vapors at pressures which are lower than the atmospheric pressure.

Gas is the state of an element which cannot be transferred into another state of matter in the existing pressure and temperature range.

Vapor is a substance in a gaseous state of matter which however can be transferred into a liquid or solid state within the temperature or pressure range relevant for the process flow.

Vacuum is defined as the thermodynamic state in a chamber filled with gas or vapor at pressures below the atmospheric pressure.

1.5.2

Physical basic principles of ideal gases

Boyle's/Mariotte's law. For calculations in which the gas temperature can be considered as constant, the Boyle Mariott law is applicable (equation 1-61).

$$p \cdot V = \text{constant} \quad (1-61)$$

$$T = \text{constant}$$

$$p \quad \text{pressure [N/m}^2\text{; Pa; bar]}$$

$$V \quad \text{volume [m}^3\text{]}$$

Gay-Lussac's law. With changing volume at varying temperatures and constant pressure the Gay-Lussac's law according equation 1-62 is applicable.

$$\frac{V_{T1}}{V_{T2}} = 1 + \alpha(T_1 - T_2) \quad (1-62)$$

$$p \quad \text{constant}$$

$$T \quad \text{thermodynamic temperature [K]}$$

$$0^{\circ}\text{C} = 273.15 [\text{K}]$$

$$\alpha = \frac{1}{273} \quad (1-63)$$

General state equation. With the general state equation (1-17), the pressure or volume of the gas mass can be determined for any temperature. R is the specific gas constant dependent on the substance. With equation (1-64), even the apparent molecular mass of gas mixtures can be determined.

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

$$\frac{\text{N}}{\text{m}^2} \cdot \text{m}^3 = \text{kg} \cdot \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot \text{K} [\text{Nm}]$$

$$p \cdot V_m = M \cdot R \cdot T \quad (1-64)$$

$$\frac{\text{N}}{\text{m}^2} \cdot \frac{\text{m}^3}{\text{kmol}} = \frac{\text{kg}}{\text{kmol}} \cdot \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot \text{K} \left[\frac{\text{Nm}}{\text{kmol}} \right]$$

$$p \cdot V = m \cdot \frac{R_0}{T} \cdot T \quad (1-65)$$

$$\frac{\text{N}}{\text{m}^2} \cdot \text{m}^3 = \text{kg} \cdot \frac{\frac{\text{Nm}}{\text{kmol} \cdot \text{K}}}{\frac{\text{kg}}{\text{kmol}}} \cdot \text{K} [\text{Nm}]$$

m gas mass [kg]

R specific gas constant [Nm/(kg · K)]

R_0 general molar gas constant [Nm/(kmol · K)]

M relative molecular mass [kg/kmol]

V_m molar volume = 22.41 [m³/kmol] at 0 °C and 1013 mbar

Volume. The volume is determined by the product of gas mass and the specific volume according to equation (1-66).

$$V = m \cdot v \quad (1-66)$$

v specific volume [m³/kg]

General gas constant. According to equation (1-67), the general gas constant is connected to the specific gas constant by the relative molecular mass ratio.

$$R_0 = R \cdot M \quad (1-67)$$

$$R_o = \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot \frac{\text{kg}}{\text{kmol}} = 8314 \left[\frac{\text{Nm}}{\text{kmol} \cdot \text{K}} \right]$$

or

$$R_o = \frac{\text{Pa} \cdot \text{m}^3}{\text{kg} \cdot \text{K}} \cdot \frac{\text{kg}}{\text{kmol}} = 8314 \left[\frac{\text{Pa} \cdot \text{m}^3}{\text{kmol} \cdot \text{K}} \right]$$

or

$$R_o = \frac{\text{mbar} \cdot 1}{\text{kg} \cdot \text{K}} \cdot \frac{\text{kg}}{\text{kmol}} = 83140 \left[\frac{\text{mbar} \cdot 1}{\text{kmol} \cdot \text{K}} \right]$$

or

$$R_o = \frac{\text{mbar} \cdot 1}{\text{kg} \cdot \text{K}} \cdot \frac{\text{kg}}{\text{kmol}} = 83,14 \left[\frac{\text{mbar} \cdot 1}{\text{mol} \cdot \text{K}} \right]$$

or

$$R_o = \frac{\text{J}}{\text{kg} \cdot \text{K}} \cdot \frac{\text{kg}}{\text{kmol}} = 8314 \left[\frac{\text{J}}{\text{kmol} \cdot \text{K}} \right]$$

or

$$R_o = \frac{\text{kpm}}{\text{kg} \cdot \text{K}} \cdot \frac{\text{kg}}{\text{kmol}} = 848 \left[\frac{\text{kpm}}{\text{kmol} \cdot \text{K}} \right]$$

$$1 \text{ Nm} = 1 \text{ J}$$

$$1 \text{ Pa} = 1 \text{ N/m}^2$$

Molar volume. The molar volume is calculated according to equation (1-68) with a value of

22.41 m³/kmol at 0 °C and 1013 mbar. It can also be calculated from the quotient of the Avogadro Number and the Loschmidt's Number acc. to equation (1-69).

$$V_m = \frac{R_o \cdot T}{p} \quad (1-68)$$

$$V_m = \frac{8314 \frac{\text{Nm}}{\text{kmol} \cdot \text{K}} \cdot 273.15 \text{ K}}{1.01325 \cdot 10^5 \frac{\text{N}}{\text{m}^2}} = 22.41 \left[\frac{\text{m}^3}{\text{kmol}} \right]$$

or

$$V_m = \frac{N_A}{N_L} \quad (1-69)$$

$$V_m = \frac{6.022 \cdot 10^{26} \frac{1}{\text{kmol}}}{2.686 \cdot 10^{25} \frac{1}{\text{m}^3}} = 22.41 \left[\frac{\text{m}^3}{\text{kmol}} \right]$$

N_A Avogadro Constant = $6.022 \cdot 10^{26}$ [molecules/kmol]

N_L Loschmidt's Number = $2.686 \cdot 10^{25}$ [molecules/m³]

Density. Acc. to equation (1-70), the quotient of relative molecular mass and molar volume results in the gas density.

$$\rho = \frac{M}{V_m} \quad (1-70)$$

$$\rho = \frac{M \frac{\text{kg}}{\text{kmol}}}{22.41 \frac{\text{m}^3}{\text{kmol}}} = 4.462 \cdot 10^{-2} \cdot M \left[\frac{\text{kg}}{\text{m}^3} \right]$$

ρ density

Absolute molar mass. Absolute molar mass can be calculated from the relative molar mass and the Avogadro Constant acc. to equation (1-71).

$$m_u = \frac{M}{N_A} \quad (1-71)$$

$$m_u = \frac{1 \frac{\text{kg}}{\text{kmol}}}{6.022 \cdot 10^{26} \frac{1}{\text{kmol}}} = 1.6605 \cdot 10^{-27} [\text{kg}]$$

m_u absolute molar mass

Boltzmann Constant. The Boltzmann Constant is calculated from the general gas constant and the Avogadro Number acc. to equation (1-72).

$$k = \frac{R_o}{N_A} \quad (1-72)$$

$$k = \frac{8314 \frac{\text{Nm}}{\text{kmol} \cdot \text{K}}}{6.022 \cdot 10^{26} \frac{1}{\text{kmol}}} = 1.380 \cdot 10^{-23} \left[\frac{\text{Nm}}{\text{K}} \right]$$

or

$$k = \frac{8314 \frac{\text{J}}{\text{kmol} \cdot \text{K}}}{6.022 \cdot 10^{26} \frac{1}{\text{kmol}}} = 1.380 \cdot 10^{-23} \left[\frac{\text{J}}{\text{K}} \right]$$

k Boltzmann constant

Loschmidt's and Avogadro constants. In case of equal volume, equal pressure and equal temperature, each ideal gas has the same number of molecules or atoms. The Loschmidt's Number N_L gives the number of molecules or atoms contained in 1 m³ of gases or gas mixtures.

The Avogadro Constant N_A gives the number of molecules or atoms contained in 1 mol or kilomol.

One mol in grams weighs as much as the relative molecular mass of the gas states. The ratio of the mass of the respective molecules to one twelfth of the mass of a carbon isotope (^{12}C) is called relative molecular mass.

Loschmidt's number. From the division of the Avogadro Constant by the molar volume, the Loschmidt's Number results acc. to equation (1-73).

$$N_L = \frac{N_A}{V_m} \quad (1-73)$$

$$N_L = \frac{6.022 \cdot 10^{26} \frac{1}{\text{kmol}}}{22.41 \frac{\text{m}^3}{\text{kmol}}} = 2.686 \cdot 10^{25} \left[\frac{1}{\text{m}^3} \right]$$

N_L Loschmidt's number

Avogadro Constant. By the division of the relative molecular weight by the absolute molar mass, the Avogadro Constant is determined according to equation (1-74).

$$N_A = \frac{M}{m_u} \quad (1-74)$$

$$N_A = \frac{1 \frac{\text{kg}}{\text{kmol}}}{1.6605 \cdot 10^{-27} \text{kg}} = 6.022 \cdot 10^{26} \left[\frac{1}{\text{kmol}} \right]$$

N_A Avogadro's constant

Particle density. For ideal gases, the number of particles (molecules, atoms) in a system is connected to pressure and temperature. This derives from the general state equation for ideal gases.

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

$$p \cdot V = \frac{m}{M} \cdot R_o \cdot T \quad (1-65)$$

$$m = N \cdot m_u \text{ [kg]} \quad (1-75)$$

m mass [kg]

N particle number

m_u particle mass, absolute molar mass [kg]

$$p \cdot V = \frac{N \cdot m_u}{M} \cdot R_o \cdot T \quad (1-76)$$

$$N = n \cdot V \quad (1-77)$$

N particle number density [$1/\text{m}^3$]
 V volume [m^3]

$$\frac{m_u}{M} = \frac{1}{N_A} \quad (1-78)$$

$$p \cdot V = \frac{n \cdot V}{N_A} \cdot R_o \cdot T \quad (1-79)$$

$$\frac{R_o}{N_A} = k \quad (1-72)$$

and consequently

$$p = n \cdot k \cdot T \quad (1-1)$$

$$n = \frac{p}{k \cdot T} = \frac{\frac{N}{\text{m}^2}}{1.38 \cdot 10^{-23} \frac{\text{Nm}}{\text{K}} \cdot \text{K}} \left[\frac{1}{\text{m}^3} \right] \quad (1-80)$$

With the equation (1-80) the particle density for any temperature and pressure can be calculated.

Example 1.6

Determination of the standard particle density at 1013 mbar and 0 °C.

Solution:

$$n = \frac{p}{k \cdot T} \quad (1-80)$$

$$p = 101325 \text{ [N/m}^2\text{]}$$

$$T = 273.15 \text{ [K]}$$

$$K = 1.380 \cdot 10^{-23} \text{ [Nm/K]}$$

$$n = \frac{101325 \frac{\text{N}}{\text{m}^2}}{1.38 \cdot 10^{-23} \frac{\text{Nm}}{\text{K}} \cdot 273.15 \text{ K}} = 2.686 \cdot 10^{25} \left[\frac{1}{\text{m}^3} \right]$$

Consequently, the particle density n determined for the standard temperature and pressure corresponds to the Loschmidt's Number N_L .

Dalton's law, gas mixtures. According to the Dalton's law, the total pressure of a gas mixture is the sum of the partial pressures of the gas particles, as the individual gas particles fill the whole chamber as if other particles did not exist.

The calculation of a gas mixture can occur according to equation (1-81).

According to the general state equation

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

the following is applicable for the gas mixture:

$$p \cdot V = (m_1 \cdot R_1 + m_2 \cdot R_2 + \dots + m_k \cdot R_k) \cdot T \quad (1-81)$$

V total volume [m^3]

For the calculation of the total pressure, equation (1-82) is used.

$$p = \Sigma p_1 + p_2 + \dots + p_k \quad (1-82)$$

p total pressure [N/m^2 ; Pa; bar]

p_1, p_2, p_k partial pressures

Example 1.7: [1.6]

How many liters does 1 gram water comprise at 1 Torr and $+15^\circ\text{C}$?

Solution:

As water vapor of 1 Torr begins to condense only at -17°C , it can be considered as an ideal gas at 15°C .

The calculation occurs acc. to equation (1-17).

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

$$m = 1 \text{ gram} = 0.001 [\text{kg}]$$

$$T = 288 [\text{K}]$$

$$P = 1 \text{ Torr} = 1.36 \text{ mbar} = 1.36 \cdot 10^2 \left[\frac{\text{N}}{\text{m}^2} \right]$$

$$R_o = 8314 \left[\frac{\text{J}}{\text{kmol} \cdot \text{K}} \right]$$

$$M_{\text{H}_2\text{O}} = 18.02 \left[\frac{\text{kg}}{\text{kmol}} \right]$$

From

$$R_o = R \cdot M \quad (1-67)$$

results

$$R_{\text{H}_2\text{O}} = \frac{R_o}{M_{\text{H}_2\text{O}}} = \frac{8314 \frac{\text{J}}{\text{kmol} \cdot \text{K}}}{18,02 \frac{\text{kg}}{\text{kmol}}} = 461,4 \frac{\text{J}}{\text{kg} \cdot \text{K}} = 461,4 \left[\frac{\text{Nm}}{\text{kg} \cdot \text{K}} \right]$$

$$V = \frac{m \cdot R \cdot T}{p} = \frac{0.001 \text{ kg} \cdot 461.4 \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot 288 \text{ K}}{1.36 \cdot 10^{-6} \frac{\text{N}}{\text{m}^2}} = 0.977 \text{ m}^3 = 977 \text{ [l]}$$

Example 1.8: [1.6]

Which gas volume results from an oil spot with a mass of 1 mg, if it is exposed to an absolute pressure of 10^{-8} Torr and a temperature of 127°C ?

Solution:

Surmised molecular weight of oil is 200 kg/kmol.

The calculation of the gas volume occurs acc. to equation (1-17).

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

$$V = \frac{m \cdot R \cdot T}{p} = \frac{\text{kg} \cdot \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot \text{K}}{\frac{\text{N}}{\text{m}^2}} \left[\text{m}^3 \right]$$

$$m = 0.000001 \text{ [kg]}$$

$$T = 400 \text{ [K]}$$

$$p = 10^{-8} \text{ Torr} = 1.36 \cdot 10^{-8} \text{ mbar} = 1.36 \cdot 10^{-6} \frac{\text{N}}{\text{m}^2}$$

$$\text{From } R_o = R_M, \quad (1-67)$$

results

$$R_{\text{oil}} = \frac{R_o}{M_{\text{oil}}} = \frac{8314 \frac{\text{J}}{\text{kmol} \cdot \text{K}}}{200 \frac{\text{kg}}{\text{kmol}}} = 41.57 \frac{\text{J}}{\text{kg} \cdot \text{K}} = 41.57 \left[\frac{\text{Nm}}{\text{kg} \cdot \text{K}} \right]$$

$$V = \frac{m \cdot R \cdot T}{p} = \frac{0.000001 \text{ kg} \cdot 41.57 \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot 400 \text{ K}}{1.36 \cdot 10^{-6} \frac{\text{N}}{\text{m}^2}}$$

$$V = 12227 \text{ m}^3 = 12227000 \text{ [l]}$$

Under the conditions mentioned above the oil spot will expand to a volume of 12,227,000 liters.

Example 1.9: [1.6]

In a vessel, there is a pressure of 10^{-10} Torr at a temperature of 0°C .

How many molecules are there in one cm^3 ?

Solution:

The number of particles can be determined according to equation (1-80).

$$n = \frac{p}{k \cdot T} = \frac{\frac{N}{m^2}}{\frac{Nm}{K} \cdot K} \left[\frac{1}{m^3} \right] \quad (1-80)$$

$$\begin{aligned} p &= 10^{-10} \text{ Torr} = 1.36 \cdot 10^{-8} \frac{N}{m^2} \\ k &= 1.38 \cdot 10^{-23} \frac{Nm}{K} \\ T &= 273.15 \text{ K} \end{aligned}$$

$$n = \frac{1.36 \cdot 10^{-8} \frac{N}{m^2}}{1.38 \cdot 10^{-23} \frac{Nm}{K} \cdot 273.15 \text{ K}} = 3.607 \cdot 10^{12} \left[\frac{1}{m^3} \right]$$

$$n = 3.607 \cdot 10^{12} \frac{1}{m^3} \cdot \frac{1}{1000000} \cdot \frac{m^3}{cm^3} = 3.607 \cdot 10^6 \left[\frac{1}{cm^3} \right]$$

There are $3.607 \cdot 10^6$ molecules in 1 cm^3 .

1.5.3

Standard temperature and pressure

Standard temperature and pressure is the state of a liquid, solid or gaseous substance as defined in DIN 1343 Standard issued in November 1975.

The standard temperature $T_n = 273.15 \text{ K}$, $\delta = 0^\circ \text{ [C]}$.

The standard pressure $p_s = 101325 \text{ Pa} = 1013.25 \text{ [mbar]}$.

In order to enable the comparison of gases, a standard temperature and a standard pressure are taken as a basis. The standard volume is the volume a gas takes up at a certain pressure ($p = 760 \text{ Torr} = 1.01325 \cdot 10^5 \text{ Pa}$) and a certain temperature ($t = 0^\circ \text{C} = 273.15 \text{ K}$). Therefore, the standard volume (also defined as Nm^3) is an exactly defined quantity of gas.

Example 1.10:

The mass of the standard volume of 10 Nm^3 of air is to be determined.

Solution:

The mass is calculated according to equation (1-17).

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

$$m = \frac{V \cdot p}{R \cdot T}$$

$$\begin{aligned} V &= 10 \text{ [Nm}^3\text{]} \\ P_s &= 1.013 \cdot 10^5 \text{ [N/m}^2\text{]} \\ T_n &= 273.15 \text{ [K]} \end{aligned}$$

$$R_{\text{air}} = 0.2871 \frac{\text{kJ}}{\text{kg} \cdot \text{K}} = 287.1 \left[\frac{\text{Nm}}{\text{kg} \cdot \text{K}} \right] \quad (\text{from table 1-15})$$

$$m = \frac{10 \text{ m}^3 \cdot 1.013 \cdot 10^5 \frac{\text{N}}{\text{m}^2}}{287.1 \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot 273.15 \text{ K}} = 12.92 \text{ [kg]}$$

According to the Avogadro law, a mole of any gas takes up the same volume at the same pressure and the same temperature. This is the molar volume.

For ideal gases, the molar volume at standard temperature and pressure is

$$V_m = 22.41 \text{ [m}^3/\text{kmol]}$$

Example 1.11

The density of a substance at standard temperature and pressure is to be determined from molar weight and molar volume.

Solution:

$$\rho_n = \frac{M}{V_m} \quad (1-70)$$

ρ_n density at standard temperature and pressure [kg/m³].

According to this, the density of nitrogen is calculated as follows:

$$M_{\text{N}_2} = 28.01 \left[\frac{\text{kg}}{\text{kmol}} \right] \quad (\text{from the table 1-15})$$

$$\rho_n = \frac{28.01 \frac{\text{kg}}{\text{kmol}}}{22.41 \frac{\text{m}^3}{\text{kmol}}} = 1.249 \left[\frac{\text{kg}}{\text{m}^3} \right]$$

1.5.4

Real gases and vapors

In real gases and vapors, the forces of attraction between the particles cannot be neglected as it is the case with ideal gases that do not dispose of specific volumes and the particles of which do not exercise interaction forces on each other. Compared to the volume of the vacuum vessel provided for the gases and vapors, the specific volume of real gases and vapors is not very low and cannot be neglected. The state equation according to Van der Waal [comp. equation (1-83 and (1-84)] records this state. [1.1]

$$\left[p + \left(\frac{a \cdot m^2}{V^2} \right) \right] (V - m \cdot b) = \frac{N \cdot \mu u}{M} \cdot R_o \cdot T \quad (1-83)$$

or for practical application:

$$\left[p + \left(\frac{a \cdot m^2}{V^2} \right) \right] (V - m \cdot b) = m \cdot R \cdot T \quad (1-84)$$

The specific volume and the forces of attraction of the particles are considered by the constants a and b .

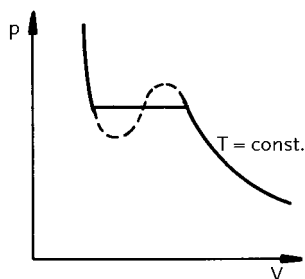


Figure 1-26. Isotherm curve of a real gas [1.1]

The curve trace (fig. 1-26) corresponds to the equation 1-83. However, in practice this kind of curve trace is hardly observed, more likely is a trace along the horizontal straight line caused by the condensation of gas. The trace of the broken line occurs only partly and in exceptional cases. The curve trace drawn in with the aid of a broken line corresponds to the instable behavior of supercooled vapor or overheated liquid.

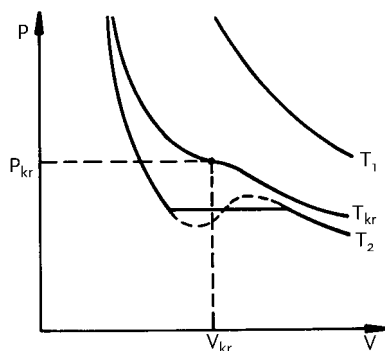


Figure 1-27. Complete p,V -diagram of a uniform chemical substance [1.1]

The curve trace in fig. 1-27 corresponds to a chemically homogeneous substance that does not decompose within the relevant temperature range.

With increasing temperature the distances between the minimum and the maximum diminish continuously and the inflectional tangent is getting flatter (fig. 1-27). At the critical temperature T_{kr} , the inflectional tangent is horizontal, and the inflection point still exists. This is the critical isotherm with the critical point p_{kr} appearing as inflection point. The inflection point has the coordinates $V=V_{kr}$ and

$p = p_{kr}$. Above the critical point, the gas cannot be condensed anymore, and the isotherms above the critical temperature show hyperbola characteristics. With increasing temperatures, the isotherms adopt more and more the trace corresponding to the state equation for ideal gases. The constants a and b can be calculated from the critical point data.

1.5.5

Phase transitions and their descriptions

For the transition from one phase to another, transition heat (latent heat quantity) is required or released. Their descriptions are given in the table 1-10.

Table 1-10. Phase transitions and their respective transition heat [1.1]

Phase transition from/to	Description of phase transition	Description of transition heat
Liquid/solid	congealing (for water: freezing)	solidification heat
Solid/liquid	melting	heat of fusion
Solid/gaseous	subliming	sublimation heat
Liquid/gaseous	vaporizing	vaporization heat
Gaseous/liquid	condensing	condensation heat
Gaseous/solid	condensing	condensation heat

Transition heats usually depend on temperature. For instance, vaporization heat decreases with increasing temperature until it reaches the value zero when reaching the critical point. Over the solid or liquid phase of a substance, there is always its gaseous phase. In the state of equilibrium, the gaseous phase adopts the saturation vapor pressure (fig. 1-28) which depends on the prevailing temperature acc. to equation (1-85) of Clausius and Clapeyron [1.1]. The quantity of heat required for phase transition or released through it at constant pressure corresponds to the enthalpy change and is specific for each material.

$$\frac{dp_s}{dT} = \frac{\Delta H}{T \cdot \Delta V} \quad (1-85)$$

p_s saturation vapor pressure

T thermodynamic temperature

ΔH transition heat = enthalpy change

ΔV volume change during the phase transition

The application of vacuum technology in process engineering requires the knowledge of vapor pressure of e.g. substances that are to be thermally treated. Comprehensive tables of temperature pressure and diagrams of pressure temperatures are available in specialist literature. In fig. 1-29, a graph shows pressure/temperature curves.

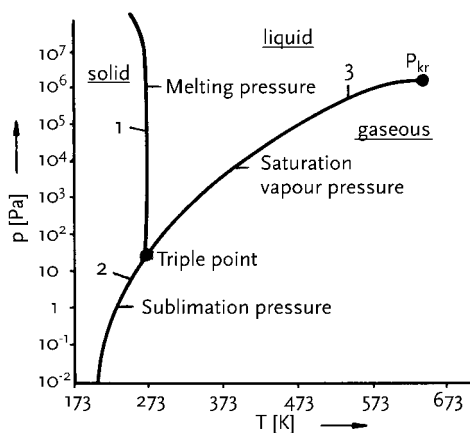


Figure 1-28. Dependence of melting temperature, sublimation and saturation temperature on pressure (water curves)
1) melting curve 2) sublimation curve 3) saturation curve

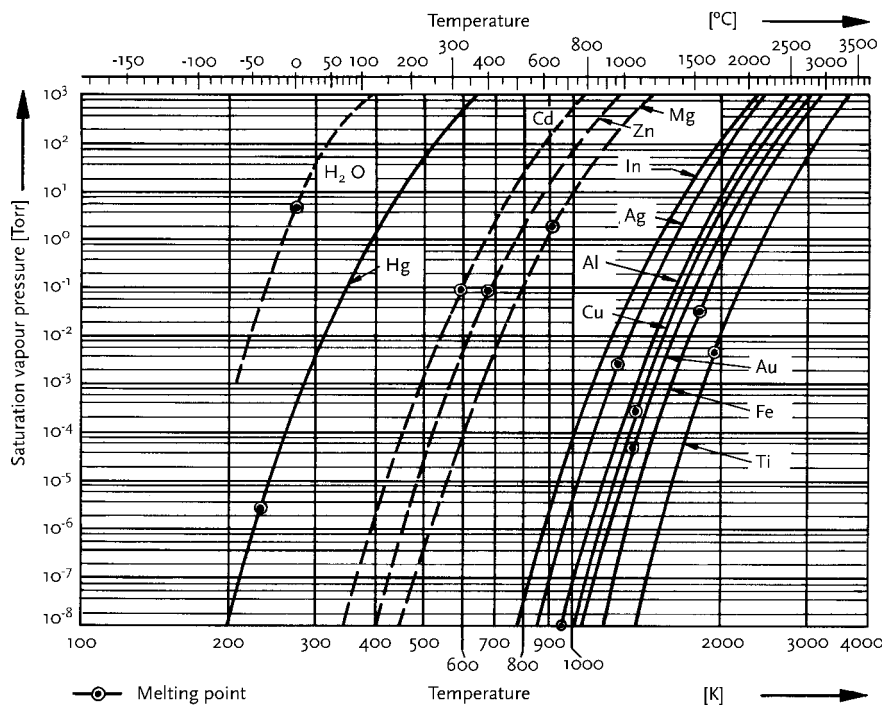


Figure 1-29. Pressure curves for saturated vapor of different substances [1.8].

Figure 1-29 contains the metals important in vacuum technology as well as those that should be avoided due to their high inherent vapor pressure (cadmium, zinc and magnesium). These metals are often contained in alloys and because of their

high vapor pressure especially in high vacuum they can evaporate, contaminate the plant and deteriorate the vacuum. That is why brass in vacuum should never be exposed to temperatures over 200 °C, as otherwise zinc will evaporate.

The drawing up of vapor pressure curves as shown in fig.1-29 is often time-consuming and difficult. The depiction in the diagram, however, according to the relation

$$\lg p = f\left(\frac{1}{T}\right) \quad (1-86)$$

which is based on equation (1-85) by Clausius & Clapayron [1.2] is a simplification. Mostly, it is possible to depict the vapor pressure curves with sufficient accuracy for practical purposes as straight lines according to equation (1-87), (comp. fig. 1-30).

$$\lg p = -\frac{B}{T} + A \quad (1-87)$$

p [mbar]

A and B are material-specific values and are mostly (comp. tab. 1-11) found in literature.

$$A = \lg p_o \quad (1-88)$$

p_o vapor pressure at standard temperature and pressure

$$B = \frac{\Delta H \cdot \lg e}{m \cdot R} \quad (1-89)$$

ΔH transition heat

m gas mass

R specific gas constant

Table 1-11. Values A and B for $\lg p = f(1/T)$ -diagram [1.2].

Kind of gas/steam	Formula	t (°C)	A (A_s)	B (B_s)
bromobenzene	C_6H_5Br	155.6	8.20 (8.90)	-2230 (-2700)
heptane	C_7H_{16}	98	8.11	-1890
propyl alcohol	C_3H_7OH	97.20	9.01	-1800
water	H_2O	100	9.01 (10.52)	-2244 (-2660)

t evaporation temperature at 1013 bar

A, B for evaporation range

A_s , B_s for sublimation range

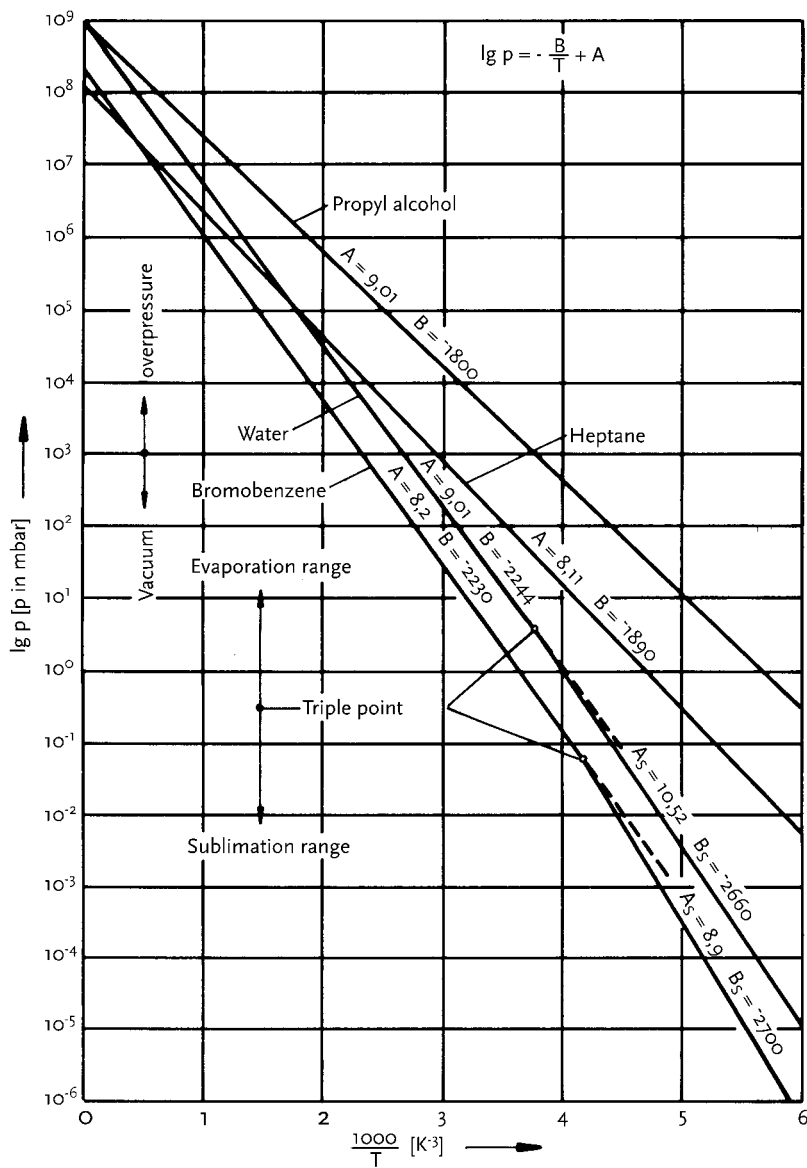


Figure 1-30. $\lg p = f(1/T)$ -diagram acc. to Clausius-Clapeyron [1.2]

1.6

Mixtures of ideal gases [1.9]

In the following, mixtures of ideal individual gases are dealt with that

- a) do not chemically react with each other
- b) exist in mechanical and thermal equilibrium.

Generally it can be said that a mixture of ideal individual gases behaves like an ideal gas. For instance, air is a gas mixture consisting mainly of the individual gases nitrogen and oxygen.

For the calculation of gas mixtures, volumes and mass portions of individual components as well as their partial pressures, molar masses and gas constants must be known.

In practice, the determination of the individual gas portions in a gas mixture is particularly interesting. The interrelations and different solutions are described below.

1.6.1

Mass composition

If a mixture consists of

$$\begin{aligned} m_1 &= 3 \text{ kg H}_2 \\ m_2 &= 1 \text{ kg N}_2 \\ m_3 &= 4 \text{ kg CH}_4 \end{aligned}$$

then the total mass is

$$m = m_1 + m_2 + m_3 \quad (1-90)$$

If a gas mixture consists of k components, the following can be generally formulated:

$$m = m_1 + m_2 + \dots + m_k \quad (1-91)$$

$$m = \sum_{i=1}^k m_i \quad (1-92)$$

The composition of gas mixtures can be stated e. g. in mass portions ξ .

Referring to the total mass of a gas mixture, the mass portion of a gas has the value ξ , and can be determined as follows:

$$\xi_i = \frac{m_i}{m} \quad (1-93)$$

ξ_i mass portion

m_i mass of components

m total mass of gas mixture

With three different gases, for the mixture results:

$$\frac{m_1}{m_1 + m_2 + m_3} + \frac{m_2}{m} + \frac{m_3}{m} = \frac{m}{m} \quad (1-94)$$

$$m = m_1 + m_2 + m_3$$

$$\xi_1 + \xi_2 + \xi_3 = 1 \quad (1-95)$$

Or in general terms

$$\sum_{i=1}^k \xi_i = 1 \quad (1-96)$$

This confirms that the addition of all mass portions existing in a gas mixture has the value 1.

1.6.2

Molar composition

The composition of a gas mixture can also be determined by the quantity of the substance n , e.g. in kmol resulting in the following relation:

$$n = \sum_{i=1}^k n_i \quad (1-97)$$

ψ_1 is taken as a base for molar portion:

$$\psi_1 = \frac{n_i}{n} \quad (1-98)$$

and

$$\sum_{i=1}^k \psi_i = 1 \quad (1-99)$$

corresponding to the examination of the mass portions.

1.6.3

Volumetric composition

Ideal gas mixtures are often calculated according to the volume percentage r . The following consideration refers to a mixture of three different gas quantities at the same pressure and temperature. The gases are ideal gases (fig. 1-31).

Before mixing, the three gas quantities are divided by three thin walls, however all three quantities are under exactly the same pressure of e.g. $p = 0.3 \text{ mPa}$, and all three percentages have the same temperature of $T = 300 \text{ K}$. The three individual percentages can be regarded as partial volume V_1 , V_2 , and V_3 . As temperatures and

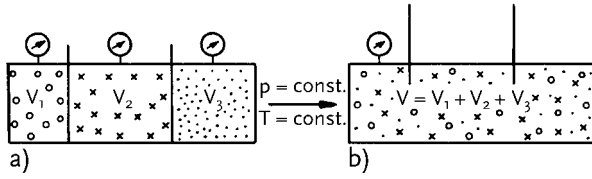


Figure 1-31. Composition of the ideal gas mixture [1.9]

a) before mixing, b) after mixing

pressures of the three components are the same, the volume of a quantity of substances existing in the mixture will have the partial volume described above after the dividing walls have been removed.

$$V = V_1 + V_2 + V_3 = \text{total volume} \quad (1-100)$$

Therefore, the volume percentages r is calculated according to the ratio of the volume proportions:

$$r_1 = \frac{V_1}{V} \quad (1-101)$$

$$r_2 = \frac{V_2}{V} \quad (1-102)$$

$$r_3 = \frac{V_3}{V} \quad (1-103)$$

Generally:

$$r_i = \frac{V_i}{V} \quad (1-104)$$

and

$$\sum_{i=1}^k r_i = 1 \quad (1-105)$$

After mixing, the total pressure of the mixture is $P = 0.3 \text{ MPa}$ and the mixture temperature is $T = 300 \text{ K}$. Therefore, the total pressure is equal to the pressure of each component in the mixture.

1.6.4

Ideal gas mixtures and general equation of gas state

Examining the individual gas proportions during the mixing of different gases it can be seen that the proportions expand from their partial volume V_i to the volume V of the gas mixture, which causes the pressure of each proportion to drop from its original value p before mixing to the partial pressure p_i after mixing (fig 1-32).

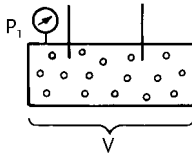


Figure 1-32. State of gas component 1 in the mixture [1.9]

Before mixing the three gases mentioned above, component 1 is under pressure p and has the partial volume V_1 . According to the state equation for ideal gases, for the partial volume V_1 or partial volumes V_2 or V_3 the following is applicable:

$$p \cdot V_1 = m_1 \cdot R_1 \cdot T \quad (1-106)$$

and the partial volume is

$$V_1 = \frac{m_1 \cdot R_1 \cdot T}{p} \quad (1-107)$$

During the mixing process, each single gas expands to the total available volume V . This process corresponds to the *Dalton Law*.

The pressure of component 1 decreases to partial pressure p_1 . For component 1, the following state equation results:

$$p_1 \cdot V = m_1 \cdot R_1 \cdot T \quad (1-108)$$

and the partial pressure is

$$p_1 = \frac{m_1 \cdot R_1 \cdot T}{V} \quad (1-109)$$

This finding can be formulated as follows:

The partial pressure is the pressure the ideal individual gas would exert onto the walls, if this gas alone filled the whole volume V at the mixing temperature T .

Partial pressures cannot be directly and individually measured. The measurable total pressure p is equal to the sum of the individual partial pressure p_i .

$$p = \sum p_1 + p_2 + p_3 \quad (1-82)$$

Generally:

$$p = \sum_{i=1}^k p_i \quad (1-110)$$

as well as

$$p \cdot V_1 = p_1 \cdot V \quad (1-111)$$

and

$$\frac{V_1}{V} = \frac{p_1}{p} \quad (1-112)$$

and also

$$\frac{V_1}{V} = r_1 \quad (1-101)$$

The following ensues

$$r_1 = \frac{V_1}{V} = \frac{p_1}{p} \quad (1-113)$$

Generally

$$r_i = \frac{p_i}{p} = \frac{V_i}{V} \quad (1-114)$$

or

$$p_i = r_i \cdot p \quad (1-115)$$

The partial pressure = volume percentage · total pressure

$$p \cdot V_2 = p_2 \cdot V \quad (1-116)$$

goes also for component 2 and

$$\frac{V_1}{V_2} = \frac{p_1}{p_2} = \frac{r_1}{r_2} \quad (1-117)$$

The result is: in a mixture of ideal gases, the partial pressures behave as the volume percentage.

1.7

Gas mixtures and their calculation [1.9] [1.10]

In thermodynamic calculations, gas mixtures can be treated like simple gases, as from gas mixtures the average values of densities, specific volumes, molar masses, gas constants, specific heat capacities and other quantities of the percentage components can be calculated (see also chapter 6.6.).

1.7.1

Density of an ideal gas mixture

The total mass

$$m = m_1 + m_2 + \cdots + m_k \quad (1-91)$$

or

$$V \cdot \rho_m = V_1 \cdot \rho_1 + V_2 \cdot \rho_2 + \cdots + V_k \cdot \rho_k \quad (1-118)$$

$$\rho_m = \left(\frac{V_1}{V}\right) \cdot \rho_1 + \left(\frac{V_2}{V}\right) \cdot \rho_2 + \cdots + \left(\frac{V_k}{V}\right) \cdot \rho_k \quad (1-119)$$

$$\rho_m = r_1 \cdot \rho_1 + r_2 \cdot \rho_2 + \cdots + r_k \cdot \rho_k \quad (1-120)$$

ρ density

ρ_m gas mixture density

and generally,

$$\rho_m = \sum_{i=1}^k r_i \cdot \rho_i \quad (1-121)$$

The densities ρ_i of the individual gases refer to the total pressure p , e.g. acc. to equation (1-106) it results:

$$p = \rho_1 \cdot R_1 \cdot T \quad (1-122)$$

Generally

$$\rho_i = \frac{p}{R_i \cdot T} \quad (1-123)$$

The density ρ_m of an ideal gas mixture is equal to the sum of the products of the volume percentage and the density of the individual gases.

1.7.2

Molar mass of gas mixture

The mean molar mass M_m is a pure operand value without physical meaning, as molecules of gas mixtures do not exist. The mean or apparently molar mass of an ideal gas mixture is calculated in compliance with the following equation:

$$\rho = \frac{M}{V_m} \quad (1-70)$$

V_m = molar volume at 0 °C and 1013 mbar

For the ideal gas mixture, the following is applicable:

$$\frac{M_m}{\bar{v}_m} = \frac{r_1 \cdot M_1}{\bar{v}_1} + \frac{r_2 \cdot M_2}{\bar{v}_2} + \dots + \frac{r_k \cdot M_k}{\bar{v}_k} \quad (1-124)$$

The specific quantities are indicated with the symbol of the extensive value, however in small letters. Molar quantities are indicated in small letters with a line above.

\bar{v}_m specific molar volume of a gas mixture
 $\bar{v}_1, \bar{v}_2, \bar{v}_k$ specific molar volumes of individual gases

Acc. to Avogadro, molar volumes of all ideal gases are equally high at the same pressure and the same temperature.

The result is:

$$\bar{v}_m = \bar{v}_1 = \bar{v}_2 = \dots = \bar{v}_k \quad (1-125)$$

and

$$M_m = r_1 \cdot M_1 + r_2 \cdot M_2 + \dots + r_k \cdot M_k \quad (1-126)$$

or, generally

$$M_m = \sum_{i=1}^k r_i \cdot M_i \quad (1-127)$$

The mean molar mass of an ideal gas mixture is equal to the sum of the products of volume percentage and molar mass of each individual gas.

With standard temperature and pressure of a gas mixture

$$\rho_{mn} = \frac{M_m}{V_m} \quad (1-128)$$

ρ_{mn} mean standard density at 0 °C and 1013 mbar

1.7.3

Gas constant of an ideal gas mixture

The universal gas constant R_o has the same value for all gases.

The product $M \cdot R$ represents the gas constant R_o referred to a kmol.

The following applies

$$R_o = R \cdot M \quad (1-67)$$

and for the gas mixture, the relation $R_o = R_m \cdot M_m$ or rather

$$R_m = \frac{R_o}{M_m} \quad (1-129)$$

is applicable.

R_m = gas constant of ideal gas mixture

After mixing, the following relationship results if the mass proportions ξ_i , e.g. of three gases are known:

$$p_1 \cdot V = m_1 \cdot R_1 \cdot T \quad (1-108)$$

$$p_2 \cdot V = m_2 \cdot R_2 \cdot T \quad (1-130)$$

$$p_3 \cdot V = m_3 \cdot R_3 \cdot T \quad (1-131)$$

through addition, it results

$$(p_1 + p_2 + p_3) \cdot V = (m_1 \cdot R_1 + m_2 \cdot R_2 + m_3 \cdot R_3) \cdot T \quad (1-132)$$

$$p = p_1 + p_2 + p_3$$

$$p \cdot V = m \left(\frac{m_1 \cdot R_1}{m} + \frac{m_2 \cdot R_2}{m} + \frac{m_3 \cdot R_3}{m} \right) \cdot T \quad (1-133)$$

For the gas mixture the following applies:

$$p \cdot V = m \cdot R_m \cdot T \quad (1-134)$$

and

$$R_m = \xi_1 \cdot R_1 + \xi_2 \cdot R_2 + \xi_3 \cdot R_3 \quad (1-135)$$

or generally

$$R_m = \sum_{i=1}^k \xi_i \cdot R_i \quad (1-136)$$

In an ideal gas mixture, the molar proportion ψ_i is equal to the volume percentage r_i , i.e. in the equations (1-115), (1-121) and (1-127) r_i can be replaced by ψ_i .

The mean gas constant of an ideal gas mixture is equal to the sum of the products of the mass proportions and the gas constant of each individual gas.

1.7.4

Relation between mass proportions and volume percentage

$$m_1 = \rho_1 \cdot V_1 \quad (1-137)$$

$$m = \rho_m \cdot V \quad (1-138)$$

consequently:

$$\frac{m_1}{m} = \frac{\rho_1}{\rho_m} \cdot \frac{V_1}{V} \quad (1-139)$$

or generally

$$\frac{m_i}{m} = \frac{V_i}{V} \cdot \frac{\rho_i}{\rho_m} \quad (1-140)$$

and

$$\xi_i = \frac{r_i \cdot \rho_i}{\rho_m} \quad (1-141)$$

For the density ratio in equation (1-141), the following equations can be elaborated:

$$\rho_m = \frac{M_m}{\bar{v}} \quad (1-142)$$

$$\rho_i = \frac{M_i}{\bar{v}} \quad (1-143)$$

and

$$\frac{\rho_i}{\rho_m} = \frac{M_i}{M_m} \quad (1-144)$$

thus

$$\xi_i = \frac{r_i \cdot M_i}{M_m} \quad (1-145)$$

For gas 1 the following is applicable:

$$p_1 \cdot V = m_1 \cdot R_1 \cdot T \quad (1-108)$$

and for the gas mixture

$$p \cdot V = m \cdot R_m \cdot T \quad (1-134)$$

Thus

$$\frac{p_1}{p} = \frac{m_1}{m} \cdot \frac{R_1}{R_m} \quad (1-146)$$

and

$$r_1 = \frac{\xi_1 \cdot R_1}{R_m} \quad (1-147)$$

generally

$$\xi_i = \frac{r_i \cdot R_m}{R_i} \quad (1-148)$$

1.7.5

Gas laws and their special application in vacuum technology**Clapeyron's state equation for ideal gases**

$$V_{\text{total}} = \frac{R_o \cdot \left(\frac{m_1}{M_1} + \frac{m_2}{M_2} + \dots + \frac{m_k}{M_k} \right) \cdot T}{p_{\text{total}}} \quad (1-149)$$

For calculation purposes, gases and vapors are supposed to be ideal.

V_{total} Total volume of gas and/or vapor mixture [m^3]

R_o molar (general) gas constant = $83.14 [(\text{mbar} \cdot \text{m}^3)/(\text{kmol} \cdot \text{K})]$

m_1, m_2 mass of the different gases or vapors [kg]

M_1, M_2 molar mass (relative molecule masses, molecular weight) [kg/kmol]

T temperature of the gas-vapor mixture [K]

p_{total} total pressure of the mixture [mbar]

$p_{\text{total}} = p_1 + p_2 + \dots + p_k$

Partial pressure in a gas and/or vapor mixture

The formula is to be used for vapors only in case of the liquids not being soluble in each other.

$$p_1 = p_{\text{total}} \cdot \frac{\frac{m_1}{M_1}}{\frac{m_1}{M_1} + \frac{m_2}{M_2} + \dots + \frac{m_k}{M_k}} \quad (1-150)$$

p pressures [mbar]

m masses [kg]

M molar mass [kg/kmol]

Mass of an individual gas or vapor in a gas-vapor mixture

$$m_1 = m_2 \cdot \frac{M_1}{M_2} \cdot \frac{p_1}{p_2} \quad (1-151)$$

p_1, p_2 partial pressures [mbar]

m_1, m_2 masses [kg]

M_1, M_2 molar masses [kg/kmol]

Dalton's Law

This law is also applicable for gas/vapor mixtures.

$$\frac{V_1}{V_2} = \frac{p_1}{p_2} = \frac{p_{\text{total}} - p_2}{p_2} \quad (1-152)$$

$$P_{\text{total}} = P_1 + P_2 \quad (1-153)$$

V_1, V_2 volumes [m^3]
 p_1, p_2 partial pressures [mbar]
 p_{total} total pressure [mbar]

Example 1.12

For a fuel gas with the following composition in vol. % or rather volume percentages it has to be determined:

- 1) density at standard temperature and pressure ρ_{mn} ,
- 2) fuel gas composition in mass proportions ξ_i ,
- 3) molar mass M_m
- 4) gas constant R_m
- 5) standard volume flow rate in m^3/h with the fuel gas flowing through a pipe having a diameter of $d_i = 200 \text{ mm}$ at $t = 20^\circ\text{C}$, $p = 2.5 \text{ bar}$, $c = 18 \text{ m/s}$

The standard density ρ_{in} of the individual gases can be taken from table 1-12.

Fuel gas composition

No.	Gas	Vol. %	r_i
1	H_2	42.0	0.420
2	CO	14.0	0.140
3	CH_4	33.0	0.330
4	C_2H_4	3.5	0.035
5	CO_2	3.0	0.030
6	N_2	4.5	0.045
		100.0 vol. %	1.000

Table 1-12. Standard densities of some gases

Gas	H_2	CO	CH_4	C_2H_4	CO_2	N_2
Standard density ρ_n [kg/m^3]	0.09	1.25	0.717	1.26	1.977	1.25

Solution:

$$1) \rho_{\text{mn}} = \sum_{i=1}^k r_i \cdot \rho_{\text{in}} \quad (1-121)$$

$$2) \xi_i = \frac{r_i \cdot \rho_{\text{in}}}{\rho_{\text{mn}}} \quad (1-141)$$

Results for 1) and 2)

No.	Gas	r_i	$\rho_{in} [\text{kg/m}^3]$	$r_i \cdot \rho_{in} [\text{kg/m}^3]$	$\xi_i = \frac{r_i \cdot \rho_{in}}{\rho_{mn}}$
1	H ₂	0.420	0.090	0.0378	$\xi_1 = \frac{0.0378}{0.6091} = 0.0621$
2	CO	0.140	1.250	0.1750	$\xi_2 = \frac{0.1750}{0.6091} = 0.2873$
3	CH ₄	0.330	0.717	0.2366	$\xi_3 = \frac{0.2366}{0.6091} = 0.3884$
4	C ₂ H ₄	0.035	1.260	0.0441	$\xi_4 = \frac{0.0441}{0.6091} = 0.0724$
5	CO ₂	0.030	1.977	0.0593	$\xi_5 = \frac{0.0593}{0.6091} = 0.0974$
6	N ₂	0.045	1.250	0.0563	$\xi_6 = \frac{0.0563}{0.6091} = 0.0924$
		$\Sigma r_i = 1.000$	$\rho_{mn} = 0.6091 [\text{kg/m}^3]$	$\Sigma \xi_i = 1.0000$	

3) The molar mass can be determined acc. to equation (1-127).

$$M_m = \sum_{i=1}^k r_i \cdot M_i \quad (1-127)$$

For the gas mixture at standard temperature and pressure, the molar mass is easier calculated acc. to equation (1-128):

$$\rho_{mn} = \frac{M_m}{V_m} \quad (1-128)$$

and

$$M_m = \rho_{mn} \cdot V_m$$

$$\rho_{mn} = 0.6091 \left[\frac{\text{kg}}{\text{m}^3} \right]$$

$$V_m = 22.41 \left[\frac{\text{m}^3}{\text{kmol}} \right]$$

$$M_m = 0.6091 \frac{\text{kg}}{\text{m}^3} \cdot 22.41 \frac{\text{m}^3}{\text{kmol}} = 13.649 \frac{\text{kg}}{\text{kmol}}$$

4) Acc. to equation (1-129), the gas constant can be determined:

$$R_m = \frac{R_o}{M_m} \quad (1-129)$$

$$R_m = \frac{8.314 \frac{\text{kJ}}{\text{kmol} \cdot \text{K}}}{13.649 \frac{\text{kg}}{\text{kmol}}} = 0.6091 \left[\frac{\text{kJ}}{\text{kg} \cdot \text{K}} \right]$$

5) The volume flow rate at current conditions is

$$\dot{V} = F (\text{m}^2) \cdot v \left(\frac{\text{m}}{\text{s}} \right) \cdot \frac{3600 \cdot \text{s}}{1 \text{h}} \left[\frac{\text{m}^3}{\text{h}} \right] \quad (1-154)$$

$$\dot{V} = \frac{0.20^2 \cdot \pi}{4} \text{m}^2 \cdot 18 \left(\frac{\text{m}}{\text{s}} \right) \cdot \frac{3600 \cdot \text{s}}{1 \text{h}} = 2034.72 \left[\frac{\text{m}^3}{\text{h}} \right]$$

at $p = 2.5 \text{ bar}$ and $t = 20^\circ \text{C}$.

The conversion into standard temperature and pressure values is carried out according to the thermal state equation (1-166) for ideal gases.

$$\frac{p_1 \cdot V_1}{T_1} = \frac{p_2 \cdot V_2}{T_2} \quad (1-166)$$

$$p_1 = 1013 [\text{mbar}]$$

$$T_1 = 273.15 [\text{K}]$$

$$p_2 = 2500 [\text{mbar}]$$

$$T_2 = 293 [\text{K}]$$

$$\dot{V}_2 = 2034.72 \left[\frac{\text{m}^3}{\text{h}} \right]$$

$$\dot{V}_1 = \frac{p_2 \cdot \dot{V}_2 \cdot T_1}{T_2 \cdot p_1}$$

$$\dot{V}_1 = \frac{2500 \text{ mbar} \cdot 2034.72 \frac{\text{m}^3}{\text{h}} \cdot 273.15 \text{ K}}{293 \text{ K} \cdot 1013 \text{ mbar}}$$

$$\dot{V}_1 = 4681.32 [\text{m}^3/\text{h}] \text{ at } 1013 \text{ mbar and } 0^\circ \text{C}.$$

$$\dot{V}_1 = 4681.32 [\text{Nm}^3/\text{h}]$$

Example 1.13 [1.6]

In table 1-13, the composition of dry atmospheric air in volume percents is shown.

- 1) Partial pressures are to be calculated for the given individual gases with 760 Torr and 1 Torr total pressure.
- 2) In the air of 760 Torr, a partial pressure of water vapor of 12.3 Torr is assumed. Which are the partial pressures of the other individual gases?
- 3) The mass of dry atmospheric air consists of vol. % values according to table 1-13. How much is the molar mass of the air?

Table 1-13. The composition of atmospheric air

	air	H ₂ O	nitrogen	oxygen	argon	carbon dioxide
Vol. %	100.00	0	78.09	20.95	0.93	0.03

Solution:

$$1) \text{ Total pressure } p = 760 \text{ Torr} = p_1 + p_2 + p_3 + p_4$$

$$\frac{V_1}{V} = \frac{p_1}{p} \quad (1-112)$$

or rather

$$\frac{p_2}{p} = \frac{V_2}{V}$$

Partial pressure can be calculated, for instance, for nitrogen as follows

$$p_1 = \frac{V_1}{V} \cdot p$$

$$p_1 = \frac{78.09}{100} \cdot 760 \text{ Torr} = 593.484 \text{ Torr} = \text{partial pressure of N}_2$$

Dry air	V ₁ nitrogen N ₂	V ₂ oxygen O ₂	V ₃ argon Ar	V ₄ carbon dioxide CO ₂
Vol. %	78.09	20.95	0.93	0.03
Total pressure	p ₁	p ₂	p ₃	p ₄
p = 760 [Torr]	593.484 [Torr]	159.22 [Torr]	7.068 [Torr]	0.228 [Torr]
Total pressure	p ₁	p ₂	p ₃	p ₄
p = 1 [Torr]	0.7809 [Torr]	0.2095 [Torr]	0.0093 [Torr]	0.0003 [Torr]

$$2) \text{ Total pressure } p = 760 \text{ Torr} = p_1 + p_2 + p_3 + p_4 + p_5$$

Water vapor partial pressure: p₁ = 12.30 Torr

$$p - p_1 = 760 - 12.30 = 747.70 \text{ Torr}$$

$$\frac{p_2}{p} = \frac{V_2}{V}$$

$$p_2 = \frac{V_2}{V} \cdot p = \frac{78.09}{100} \cdot 747.70 \text{ Torr} = 583.878 \text{ Torr} = \text{partial pressure of N}_2$$

Air containing water vapor	V ₁ water vapor H ₂ O	V ₂ nitrogen N ₂	V ₃ oxygen O ₂	V ₄ argon Ar	V ₅ carbon dioxide CO ₂
Dry air, vol. %	–	78.09	20.95	0.93	0.03
Total pressure	p ₁	p ₂	p ₃	p ₄	p ₅
p = 760 [Torr]	12.3 [Torr]	583.878 [Torr]	156.643 [Torr]	6.953 [Torr]	0.224 [Torr]

- 3) The mean molar mass of an ideal gas mixture is calculated according to the relation

$$M_m = \sum_{i=1}^k r_i \cdot M_i \quad (1-127)$$

or rather acc. to formula

$$M_m = r_1 \cdot M_1 + r_2 \cdot M_2 + \dots + r_k \cdot M_k \quad (1-126)$$

Thus, for the product $r_i \cdot M_i$, for nitrogen the following value results:

$$r_1 \cdot M_1 = \frac{78.09}{100} \cdot 28 \frac{\text{kg}}{\text{kmol}} = 21.865 \left[\frac{\text{kg}}{\text{kmol}} \right]$$

Dry air	V ₁ nitrogen N ₂	V ₂ oxygen O ₂	V ₃ argon Ar	V ₄ carbon dioxide CO ₂
Rel. molecular weight, [kg/kmol]	28	32	40	44
Vol. %	78.09	20.95	0.93	0.03
$r_i \cdot M_i$, [kg/kmol]	21.865	6.704	0.372	0.0132

Therefore, the molar mass of air is

$$M_m = \sum_{i=1}^4 r_i \cdot M_i = 21.865 + 6.704 + 0.372 + 0.0132 \left[\frac{\text{kg}}{\text{kmol}} \right]$$

$$M_m = 28.954 \left[\frac{\text{kg}}{\text{kmol}} \right]$$

1.8

Discharge of gases and vapors

During the discharge of liquids, their volumes do not change with occurring discharge pressures, i.e. liquids are incompressible. Gases, however, change their volumes when being discharged according to the arising pressures. An ideal gas with mass m and molar mass M fulfils the state equation of ideal gases at pressure p , volume V and absolute temperature T .

1.8.1

General state equation of gas

$$p \cdot V = m \cdot \frac{R_o}{M} \cdot T \quad (1-65)$$

General gas constant $R_o = 8314 \text{ [J/(kmol} \cdot \text{K)]}$

For mixtures of ideal gases, the mean molar mass M_m of the mixture is inserted into the equation.

The ideal state equation is valid only for lower pressures where the intermolecular forces and the specific volume need not be considered. The lower the pressure of the gas is, the better these prerequisites are fulfilled. At a pressure of 3 MPa (30 bar), the state equation for ideal gases deviates from the real value by about $\pm 1\%$ to 2% .

1.8.2

Real gas factor Z

At high pressures, gas type, temperature and pressure in the state equation must be included by the so-called real gas factor Z. Then the state equation goes as follows [1.3]:

$$p \cdot V = Z \cdot m \cdot \frac{R_0}{M} \cdot T \quad (1-155)$$

In ideal gas,

$$\frac{p \cdot v}{R \cdot T} = 1 \quad (1-156)$$

and in real gas

$$\frac{p \cdot v}{R \cdot T} = Z \quad (1-157)$$

$Z = 1$ ideal behaviour

$Z \geq 1$ real behaviour

The deviation from the ideal state increases the more the gas state approaches the range of gas condensation. The real gas factor of nitrogen is $Z = 0.99955$ at a pressure $p = 1013$ mbar and a temperature $T = 273$ K. Even at this low pressure, this gas does not behave according to the formula for ideal gas. However, the deviation from factor $Z = 1.000$ is insignificant.

In table 1-14, some real gas factors $Z = f(p, T)$ for hydrogen and air are given.

Table 1-14. Real gas factors for hydrogen and air [1.9]

p [bar]	Z values for H ₂				Z values for dry air		
	t [°C]				t [°C]		
	-100	0	100	200	0	100	200
0	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
19.62	1.0130	1.0122	1.0098	1.0078	0.9895	1.0027	1.0064
49.05	1.0345	1.0307	1.0246	1.0196	0.9779	1.0087	1.0168
98.10	1.0756	1.0625	1.0492	1.0392	0.9699	1.0235	1.0364

The table above shows that for H_2 and air, $Z \approx 1$ at medium and low pressures. Thus in practice, the state equation for ideal gases is applicable for air and hydrogen with sufficient accuracy at pressures of up to approx. 20 bar. The ideal state equation can be applied to practically all gases without correction if it is about gases with very low condensation temperatures and with a gas state temperature being significantly higher than the condensation temperature of the gas. This applies also to gases such as nitrogen, oxygen, natural gas, town gas, helium, carbon dioxide or methane.

Consequently, the state equation for ideal gases can also be applied to all really existing gases, if the stated prerequisites are fulfilled.

The ideal gas state equation:

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

or

$$p \cdot V = m \cdot \frac{R_o}{M} \cdot T \quad (1-65)$$

Into the state equation, absolute pressures in Pa or N/m^2 and absolute temperatures in K have to be inserted.

1.8.3

General gas constant

From the state equation for an ideal gas according to equation (1-156), results

$$\frac{p \cdot v}{T} = R \quad (1-158)$$

with

$$V = m \cdot v \quad (1-66)$$

and

$$v = \frac{V}{m} \quad (1-159)$$

it results

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

and when introducing molar quantities, the gas mass (m) will be replaced by

$$m = N \cdot M \quad (1-160)$$

resulting in

$$p \cdot V = N \cdot M \cdot R \cdot T \quad (1-161)$$

The molar volume:

$$V_m = \frac{V}{N} \quad (1-162)$$

Dividing equation (1-161) by N , it results

$$p \cdot V_m = M \cdot R \cdot T \quad (1-163)$$

With

$$R_o = R \cdot M \quad (1-67)$$

results

$$p \cdot V_m = R_o \cdot T \quad (1-164)$$

For standard temperature and pressure, the following is applicable:

$$p_s \cdot V_m = R_o \cdot T_s \quad (1-165)$$

The product $R \cdot M$ is the gas constant R_o referring to a kmol and is called universal gas constant depending on the gas type. It has the same value for all ideal gases.

p	pressure [N/m ²]
V	volume [m ³]
m	mass [kg]
N	particle quantity,
T	thermodynamic temperature [K]
M	molar mass [kg/kmol]
R_o	general molar gas constant [Nm/(kmol · K)]
R	specific gas constant depending on molar mass [Nm/(kg · K)]
v	specific volume [m ³ /kg]
V_m	molar volume 22.41[m ³ /kmol]
p_s	standard pressure= 101325 [N/m ²]
T_s	standard temperature= 273.15 [K]

Example 1.14

The general (molar) gas constant is to be determined.

Solution:

The general state equation for 1 kmol at standard temperature and pressure:

$$p_s \cdot V_m = R_o \cdot T_s \quad (1-165)$$

$$R_o = \frac{p_s \cdot V_m}{T_s} = \frac{1.01325 \cdot 10^5 \frac{N}{m^2} \cdot 22.42 \frac{m^3}{kmol}}{273.15 K}$$

$$R_o = 8314 \left[\frac{Nm}{kmol \cdot K} \right]$$

1.8.4

The special gas constant depending on the type of gas**Example 1.15**

Determination of the special gas constant for air.

The molar mass for air is given in table 1-15.

Solution:

From the general gas constant, the special gas constant for air is calculated as follows:

$$R_o = R \cdot M \quad (1-67)$$

$$R = \frac{R_o}{M}$$

$$M_{\text{air}} = 28,96 \left[\frac{\text{kg}}{\text{kmol}} \right]$$

$$R_o = 8314 \frac{\text{Nm}}{\text{kmol} \cdot \text{K}} = 8314 \left[\frac{\text{J}}{\text{kmol} \cdot \text{K}} \right]$$

$$R_{\text{air}} = \frac{8314 \frac{\text{J}}{\text{kmol} \cdot \text{K}}}{28,96 \frac{\text{kg}}{\text{kmol}}} = 287,10 \left[\frac{\text{J}}{\text{kg} \cdot \text{K}} \right] = 0,2871 \left[\frac{\text{kJ}}{\text{kg} \cdot \text{K}} \right]$$

Table 1-15. Gas table (extract) [1.9]

Gas type	M [kg/kmol]	V _m [m ³ /kmol]	ρ _s [kg/m ³]	R [kJ/kgK]
H ₂	2.016	22.43	0.0899	4.1243
He	4.003	22.43	0.1785	2.0042
N ₂	28.01	22.41	1.250	0.2968
O ₂	32.00	22.39	1.429	0.2598
Air	28.96	22.40	1.293	0.2871
CO	28.01	22.41	1.250	0.2968
CO ₂	44.01	22.26	1.977	0.1889
H ₂ O vap.	18.02	22.40	0.804	0.4615
SO ₂	64.07	21.89	2.926	0.1298
NH ₃	17.03	22.08	0.771	0.4883
CH ₄	16.04	22.38	0.717	0.5184
C ₂ H ₂	26.04	22.17	1.175	0.3193
C ₂ H ₄	28.05	22.26	1.260	0.2964
C ₂ H ₆	30.07	22.17	1.357	0.2765
C ₃ H ₈	44.10	21.94	2.010	0.1886
n-C ₄ H ₁₀	58.12	21.28	2.732	0.1430

M – molar mass, V_m – molar volume, ρ_s – standard density at t_s = 0 °C
and p_s = 1.013 · 10⁵ Pa, R – special gas constant

1.8.5

Thermal state equation for ideal gases

state 1 state 2

$$\frac{p_1 \cdot V_1}{T_1} = \frac{p_2 \cdot V_2}{T_2} \quad (1-166)$$

The formula is applicable for the same gas type, i.e. the special gas constant R is the same for both states 1 and 2, as well as the masses $m_1 = m_2$.

$$\text{In state 1: } \frac{p_1 \cdot V_1}{T_1} = R$$

$$\text{In state 2: } \frac{p_2 \cdot V_2}{T_2} = R$$

The Boyle & Marriot gas law $p_1 \cdot V_1 = p_2 \cdot V_2$ is applicable here at constant temperature, i.e. if $T_1 = T_2$.

$$\frac{p_2}{p_1} = \frac{V_1}{V_2} \quad (1-167)$$

or rather

$$p \cdot V = \text{constant} \quad (1-61)$$

The 1st law of Gay-Lussac is valid if p remains constant.

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \quad (1-168)$$

The 2nd law of Gay-Lussac is applicable in a special case when $V_1 = V_2$ remains unchanged.

$$\frac{p_1}{T_1} = \frac{p_2}{T_2} \quad (1-169)$$

Example 1.16[1.9]

An oxygen cylinder with a capacity of 15 l has a temperature of 20 °C and an overpressure of 15 bar = 16 [bar (abs)], air pressure \approx 1 bar.

- 1) How many kilograms of oxygen does the cylinder contain?
- 2) Which is the specific volume of the enclosed oxygen?
- 3) How high does the pressure rise, if, owing to insulation, the temperature of the gas rises to 67 °C?

Given: $V = 0.015 \text{ [m}^3\text{]}, T_1 = 293 \text{ [K]}, p_e = 15 \text{ [bar]}, p_1 = 16 \text{ [bar]}, t_2 = 67 \text{ [}^\circ\text{C]}, T_2 = 340 \text{ [K]}$

Solution:

1) The weight is determined through equation 1-17:

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

$$R_o = 0.2598 \frac{\text{kJ}}{\text{kg} \cdot \text{K}} = 259.8 \left[\frac{\text{Nm}}{\text{kg} \cdot \text{K}} \right] \quad (\text{from table 1-15})$$

$$m = \frac{p \cdot V}{R \cdot T} = \frac{16 \cdot 10^5 \frac{\text{N}}{\text{m}^2} \cdot 0.015 \text{m}^3}{259.8 \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot 293.15 \text{K}} = 0.315 [\text{kg}]$$

2) The specific volume is determined acc. to equation 1-159:

$$v = \frac{V}{m} \quad (1-159)$$

$$v = \frac{0.015 \text{ m}^3}{0.315 \text{ kg}} = 0.0476 \left[\frac{\text{m}^3}{\text{kg}} \right]$$

3) At constant volume, the pressure depending on temperature can be calculated acc. to equation (1-169):

$$\frac{p_1}{T_1} = \frac{p_2}{T_2} \quad (1-169)$$

$$V_1 = V_2$$

$$T_1 = 293.15 [\text{K}]$$

$$T_2 = 340 [\text{K}]$$

$$p_2 = \frac{p_1 \cdot T_2}{T_1} = \frac{16 \cdot 10^5 \frac{\text{N}}{\text{m}^2} \cdot 340 \text{K}}{293.15 \text{K}} = 18,56 \cdot 10^5 \left[\frac{\text{N}}{\text{m}^2} \right]$$

$$p_2 = 18.56 [\text{bar (abs), resp. 17.56 bar overpressure}].$$

1.8.6**Suction of dry gases and saturated air-water vapor mixture by liquid ring vacuum pumps**

In this section, the suction behavior of liquid ring pumps under different influences is to be examined. With the aid of calculation examples, the operation behavior of those machines while pumping of dry and humid gas-vapor mixtures at different operating and coolant temperatures is explained. Possibilities to correct suction characteristics possibly appearing during operation are shown on the basis of generally valid or empirically determined factors. An example describes the application of a condenser installed on the suction side of the pump.

An overview of some physical laws applicable to the calculations is given below.

The general state equation

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

R specific gas constant [J/(kg · K)]

or

$$p \cdot V = m \cdot \frac{R_o}{M} \cdot T \quad (1-65)$$

R_o general universal gas constant = 8314 [J/(kmol · K)]

Clapeyron's state equation for ideal gases

$$V_{\text{total}} = \frac{R_o \cdot \left(\frac{m_1}{M_1} + \frac{m_2}{M_2} + \dots + \frac{m_k}{M_k} \right) \cdot T}{p_{\text{total}}} \quad (1-149)$$

V_{total} total volume of a gas and/or vapor mixture [m³]

R_o the molar (general) gas constant = 83.14 [(mbar · m³)/(kmol · K)]

m_1, m_2 masses of different gases or vapors [kg]

M_1, M_2 molar masses (relative molecule masses, molecular weight) [kg/kmol]

T temperature of gas-vapor mixture [K]

p_{total} total pressure of the mixture [mbar]

$p_{\text{total}} = p_1 + p_2 + \dots + p_k$

Partial pressure in a gas and/or vapor mixture

$$p_1 = p_{\text{total}} \cdot \frac{\frac{m_1}{M_1}}{\frac{m_1}{M_1} + \frac{m_2}{M_2} + \dots + \frac{m_k}{M_k}} \quad (1-150)$$

For vapors only valid for liquids non soluble in one another.

p pressure [mbar]

m mass [kg]

M molar mass [kg/kmol]

Mass of an individual gas or an individual vapor in a gas–vapor mixture

$$m_1 = m_2 \cdot \frac{M_1}{M_2} \cdot \frac{p_1}{p_2} \quad (1-151)$$

p_1, p_2 partial pressure [mbar]

m_1, m_2 mass [kg]

M_1, M_2 molar mass [kg/kmol]

Dalton's Law

$$\frac{V_1}{V_2} = \frac{p_1}{p_2} = \frac{p_{\text{total}} - p_2}{p_2} \quad (1-152)$$

$$p_{\text{total}} = p_1 + p_2 \quad (1-153)$$

V_1, V_2 volumes [m³]
 p_1, p_2 partial pressures [mbar]
 p_{total} total pressure [mbar]

This law is also applicable for gas-vapor mixtures as follows:

$$\frac{V_G}{V_{\text{vap}}} = \frac{p_G}{p_{\text{vap}}} = \frac{p_{\text{total}} - p_{\text{vap}}}{p_{\text{vap}}} \quad (1-170)$$

$$\frac{V_{\text{vap}}}{V_G} = \frac{p_{\text{vap}}}{p_{\text{total}} - p_{\text{vap}}} \quad (1-171)$$

g dry gas
 vap vapor
 $p_{\text{total}} = p_G + p_{\text{vap}}$
 p_G, p_{vap} partial pressures

The total volume of a gas-vapor mixture can be calculated acc. to equation (1-172).

$$V_{\text{total}} = \frac{83.14 \cdot \left(\frac{m_G}{M_G} + \frac{m_{\text{vap}}}{M_{\text{vap}}} \right) \cdot T}{p_{\text{total}}} \quad (1-172)$$

Individual volumes are also calculated using the same formula:

$$V_G = \frac{83.14 \cdot \frac{m_G}{M_G} \cdot T}{p_{\text{total}}} \quad (1-173)$$

$$V_{\text{vap}} = \frac{83.14 \cdot \frac{m_{\text{vap}}}{M_{\text{vap}}} \cdot T}{p_{\text{total}}} \quad (1-174)$$

Basically, with $V_{\text{vap}}/V_G \geq 1$, the probability of vapor condensation in front of the pump should be controlled.

Example 1.17

Given: Dry-saturated mixture of gas and water vapor to be sucked.

$$\dot{m}_{\text{total}} = 92.5 \text{ [kg/h]}$$

Individually consisting of:

$$\dot{m}_1 = 40 \text{ [kg/h], methanol CH}_4\text{O}$$

$$\dot{m}_2 = 41.3 \text{ [kg/h], air}$$

$$\dot{m}_3 = 11.2 \text{ [kg/h], water vapor, H}_2\text{O}$$

$$m = m_1 + m_2 + m_3 \quad (1-90)$$

$$\text{Suction pressure } p_2 = 65.7 \text{ [mbar]}$$

$$\text{Suction temperature: } t_2 = 12 \text{ [}^\circ\text{C]}$$

$$\text{Compression pressure } p_1 = 1013 \text{ [mbar]}$$

The adequate liquid ring vacuum pump is searched. A selection can only be made on the basis of the suction capacity $[\text{m}^3/\text{h}]$. For the determination of the suction capacity the following two possibilities are shown:

a) *solution by determination of total molar volume*

Molecular weights acc. to tables

Kind of gas/steam	Molecular weight M
Methanol	32.04 [kg/kmol] = M_1
Air	28.96 [kg/kmol] = M_2
Water vapor	18.02 [kg/kmol] = M_3

At first, the molar quantity of the individual gas-vapor proportions has to be determined. This is possible through rearranging equation (1-160).

$$m = N \cdot M \quad (1-160)$$

m gas mass [kg]

N particle quantity [kmol]

M molecular weight [kg/kmol]

$$N = \frac{m}{M}$$

respectively

$$\dot{N}_{M_1} = \frac{\dot{m}_1}{M_1}$$

or

$$\dot{N}_{M_k} = \frac{\dot{m}_k}{M_k}$$

consequently:

$$\dot{N}_{M_1} = \frac{40 \frac{\text{kg}}{\text{h}}}{32.04 \frac{\text{kg}}{\text{kmol}}} = 1.248 \left[\frac{\text{kmol}}{\text{h}} \right]$$

$$\dot{N}_{M_2} = \frac{41,3 \frac{\text{kg}}{\text{h}}}{28.96 \frac{\text{kg}}{\text{kmol}}} = 1.426 \left[\frac{\text{kmol}}{\text{h}} \right]$$

$$\dot{N}_{M_3} = \frac{11.2 \frac{\text{kg}}{\text{h}}}{18.02 \frac{\text{kg}}{\text{kmol}}} = 0.621 \left[\frac{\text{kmol}}{\text{h}} \right]$$

$$\text{total particle quantity } \dot{N}_M = 3.295 \left[\frac{\text{kmol}}{\text{h}} \right]$$

$$\text{molar volume } V_m = 22.41 \left[\frac{\text{m}^3}{\text{kmol}} \right] \text{ acc. to equation (1-68)}$$

At standard temperature and pressure, the total volume of the existing mixture to be sucked can be calculated by rearranging equation (1-162):

$$V_m = \frac{V}{N} \quad (1-162)$$

$$V = N \cdot V_m$$

$$V \quad \text{volume [m}^3\text{]}$$

$$N \quad \text{particle quantity [kmol]}$$

$$V_m \quad \text{specific volume of 1 kmol [m}^3\text{/kmol]}$$

$$\dot{V}_{\text{tot}} = \dot{N}_M \cdot V_m$$

$$\dot{V}_{\text{tot}} = 3.295 \frac{\text{kmol}}{\text{h}} \cdot 22.41 \frac{\text{m}^3}{\text{kmol}}$$

$$\dot{V}_{\text{tot}} = 73,84 \left[\frac{\text{m}^3}{\text{h}} \right] \text{ at } 0^\circ\text{C and } 1013 \text{ mbar}$$

$$\dot{V}_{\text{tot}} = 73.84 \left[\frac{\text{Nm}^3}{\text{h}} \right]$$

Converted to the current suction state ($p_2 = 65.70 \text{ mbar}$, $t_2 = 12^\circ\text{C}$), the following suction volume \dot{V}_2 to be sucked by the pump per hour can be determined.

Calculation applying the thermal state equation for ideal gases:

$$\frac{p_1 \cdot V_1}{T_1} = \frac{p_2 \cdot V_2}{T_2} \quad (1-166)$$

$$T = 273.15 \text{ K} + t^\circ\text{C [K]}$$

Original mixture at standard temperature and pressure	Mixture in suction state
$p_1 = 1013 \text{ [mbar]}$	$p_2 = 65.7 \text{ [mbar]}$
$\dot{V}_1 = 73.84 \text{ [Nm}^3/\text{h]}$	$\dot{V}_2 = ?$
$T_1 = 273.15 \text{ [K]}$	$T_2 = 273.15 \text{ K} + 12^\circ\text{C} = 285.15 \text{ [K]}$

$$\dot{V}_2 = \frac{1013 \text{ mbar} \cdot 73.84 \frac{\text{m}^3}{\text{h}} \cdot 185.15 \text{ K}}{273.15 \text{ K} \cdot 65.7 \text{ mbar}} = 1188.50 \left[\frac{\text{m}^3}{\text{h}} \right]$$

b) *solution applying the general state equation for ideal gases*

An alternative calculation using the general state equation for the ideal gas mixture leads to the same result.

Acc. to equation (1-134), the mean gas constant R_m of the gas mixture has to be determined at first.

$$p \cdot V = m \cdot R_m \cdot T \quad (1-134)$$

$$R_m = \sum_{i=1}^k \xi_i \cdot R_i \quad (1-136)$$

$$\xi_i = \frac{m_i}{m} \quad (1-93)$$

Gas constant of each gas in the mixture can be taken from the tables.

Kind of gas/steam	Specific gas constant R_i	
	$\left[\frac{\text{kJ}}{\text{kg} \cdot \text{K}} \right]$	$\left[\frac{\text{Nm}}{\text{kg} \cdot \text{K}} \right]$
Methanol	0.2594	259.48
Air	0.2871	287.10
Water vapor	0.4615	461.50

With the equations (1-93) and (1-136), the gas constant R_m is determined.

Kind of gas/steam	\dot{m}_i [kg/h]	$\xi_i = \dot{m}_i / \dot{m}$	R_i [Nm/kg·K]	$\xi_i \cdot R_i$ [Nm/kg·K]
Methanol	$\dot{m}_1 = 40$	$40 : 92.5 = 0.4322$	259.48	112.14
Air	$\dot{m}_2 = 41.3$	$41.3 : 92.5 = 0.4460$	287.10	128.05
Water vapor	$\dot{m}_3 = 11.2$	$11.2 : 92.5 = 0.1210$	461.50	55.85
$R_m = \sum_{i=1}^k \xi_i \cdot R_i$				$= 296.04 \left[\frac{\text{Nm}}{\text{kg} \cdot \text{K}} \right]$

$$\dot{m} = \dot{m}_1 + \dot{m}_2 + \dot{m}_3 = 92.5 \text{ [kg/h]}$$

From equation (1-134), we get:

$$\dot{V} = \frac{\dot{m} \cdot R_m \cdot T}{p}$$

\dot{V} volume per time unit in suction state [m^3/h]

\dot{m} gas mass per time unit [kg/h]

R_m mean gas constant [Nm/(kg · K)]

T thermodynamic temperature in suction state [K]

p pressure in suction state [N/m^2]

$$\dot{V} = \frac{92.5 \frac{\text{kg}}{\text{h}} \cdot 296.04 \frac{\text{Nm}}{\text{kg} \cdot \text{K}} \cdot 285.15 \text{ K}}{6.57 \cdot 10^3 \frac{\text{N}}{\text{m}^2}}$$

$$\dot{V} = 1188.50 \text{ [m}^3/\text{h]} \quad \text{in suction state}$$

Based on the suction capacity determined under a) respectively b), the suitable vacuum pump can be chosen.

The characteristic curves of the respective pump are shown in fig. 1-33. The operation data of the pump are stated as follows:

- Suction pressure = 65.70 [mbar]
- Suction capacity $\dot{S} = 1188.50 \text{ [m}^3/\text{h]}$
- Rotational speed $n = 1130 \text{ [min}^{-1}\text{]}$
- Max. pump power input $P = 36.50 \text{ [kW]}$

In this case, vapor condensates and water can be used as operating liquid, for instance.

Data of the operating liquid on the pump inlet:

- Temperature $t = 15 \text{ [}^\circ\text{C]}$
- Density $\rho = 1000 \text{ [kg/m}^3\text{]}$
- Viscosity $\eta = 1 \text{ [mPas]}$

Example 1.18:

An air mass $\dot{m}_{\text{air}} = 12 \text{ [kg/h]}$ is to be sucked.

Gas mixture to be sucked: air, saturated with water vapor.

- Suction pressure: 100 [mbar]
- Suction temperature: 35 [°C]

Molar masses:

- air: $M_{\text{air}} = 29 \text{ [kg/kmol]}$
- water vapor, $M_{\text{WV}} = 18 \text{ [kg/kmol]}$

To be determined:

- partial pressure of the air
- proportion of vapor volume
- vapor mass to be sucked
- total volume to be sucked
- influence of the temperature of the operating liquid on the suction capacity of the pump
- suction capacity of the pump owing to the condensation effect
- condensation in the vacuum pump and air volume at the pump outlet
- condensation in the condenser before the vacuum pump

Solution:

- partial pressure of the air is to be determined acc. to equation (1-153):

$$p_{\text{total}} = p_1 + p_2 \quad (1-153)$$

$$p_{\text{air}} = p_{\text{total}} - p_{\text{WV}}$$

p_{air} partial pressure of the air

p_{WV} partial pressure of the water vapor

p_{total} total pressure = suction pressure = 100 mbar

$p_{\text{WV}} = 56.23 \text{ mbar}$ at 35 °C (from the tables)

$p_{\text{air}} = 100 \text{ mbar} - 56.23 \text{ mbar} = 43.77 \text{ mbar}$

- The proportion of the vapor volume is determined acc. to equation (1-171):

$$\frac{V_{\text{WV}}}{V_{\text{air}}} = \frac{p_{\text{WV}}}{p_{\text{total}} - p_{\text{WV}}} \quad (1-171)$$

$$\frac{V_{\text{WV}}}{V_{\text{air}}} = \frac{56.23 \text{ mbar}}{100 \text{ mbar} - 56.23 \text{ mbar}} = 1.28$$

V_{WV} water vapor volume

V_{air} air volume

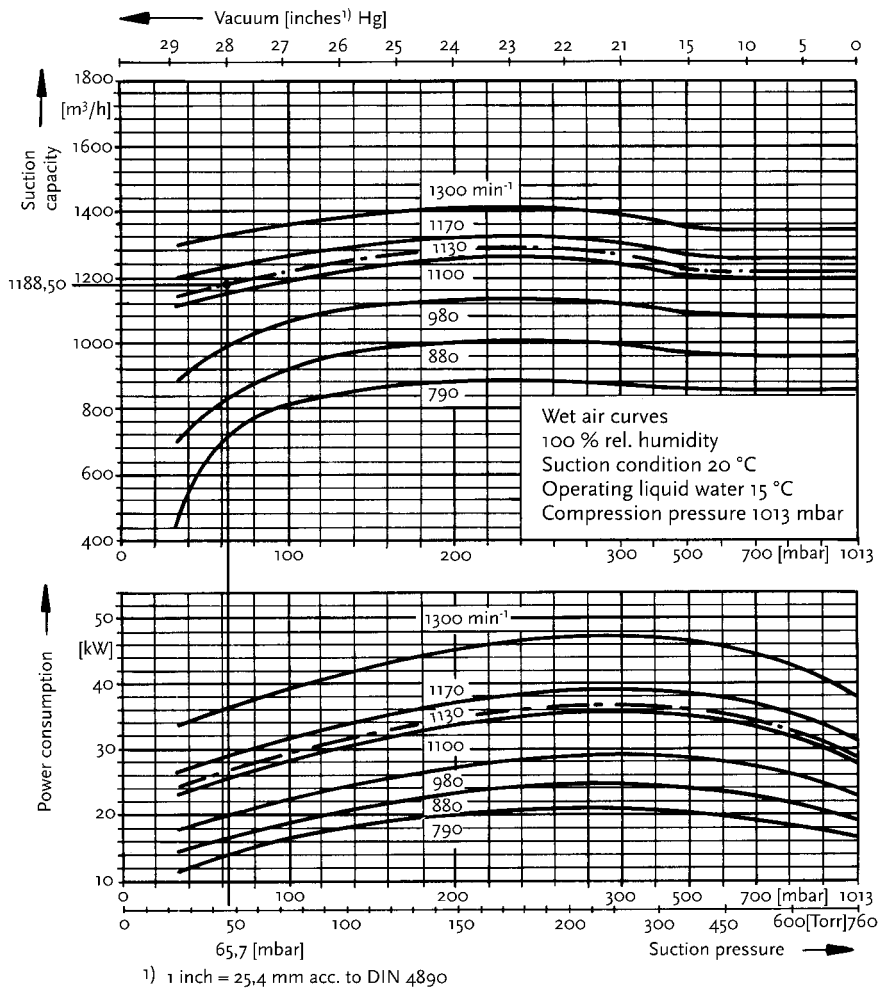


Figure 1-33. Characteristic curve field with wet-air curves of a single-stage liquid ring vacuum pump

Therefore, the volume of the water vapor has a value 1.28 times as high as the volume of the dry air.

- c) The vapor mass results from equation (1-151):

$$\dot{m}_{\text{WV}} = \dot{m}_{\text{air}} \cdot \frac{M_{\text{WV}}}{M_{\text{air}}} \cdot \frac{p_{\text{WV}}}{p_{\text{air}}} \quad (1-151)$$

$$\dot{m}_{wv} = 12 \frac{\text{kg}}{\text{h}} \cdot \frac{18 \frac{\text{kg}}{\text{kmol}}}{29 \frac{\text{kg}}{\text{kmol}}} \cdot \frac{56.23 \text{ mbar}}{43.77 \text{ mbar}} = 9.56 \left[\frac{\text{kg}}{\text{h}} \right]$$

\dot{m}_{wv} water vapour mass

\dot{m}_{air} air mass

Therefore, 9.56 kg water vapor is to be sucked in one hour.

- d) The total occurring mixture volume before the pump is determined acc. to equation (1-172).

$$\dot{V}_{\text{total}} = \frac{83.14 \cdot \left(\frac{\dot{m}_{\text{air}}}{M_{\text{air}}} + \frac{\dot{m}_{wv}}{M_{wv}} \right) \cdot T}{P_{\text{total}}} \quad (1-172)$$

$$T = 273.15 \text{ K} + 35^\circ \text{C} = 308.15 \text{ K}$$

$$\dot{V}_{\text{total}} = \frac{83.14 \frac{\text{mbar} \cdot \text{m}^3}{\text{kmol} \cdot \text{K}} \cdot \left(\frac{12 \frac{\text{kg}}{\text{h}}}{29 \frac{\text{kg}}{\text{kmol}}} + \frac{9.56 \frac{\text{kg}}{\text{h}}}{18 \frac{\text{kg}}{\text{kmol}}} \right) \cdot 308.15 \text{ K}}{100 \text{ mbar}}$$

$$\dot{V}_{\text{total}} = 241.84 [\text{m}^3/\text{h}]$$

The liquid ring vacuum pump must be designed for a suction quantity of 241.84 m³/h at 100 mbar suction pressure.

- e) Initially, the influence of the temperature of the operating liquid on the suction capacity of the vacuum pump is described for the suction of dry gas. The suction capacity of a liquid ring vacuum pump changes according to the temperature of the operating liquid, as a part of the impeller chamber is filled with vaporizing operating liquid until it is saturated. For the discharge of gas this part of the volume is lost.

For water as operating liquid, this change of the suction capacity can be determined according to the diagram shown in fig. 1-34. Usually the characteristic catalogue curves of liquid ring vacuum pumps are based on inflow temperatures of the pressure water of 15 °C.

With a suction pressure of 100 mbar (74.1 Torr) and an assumed water inflow temperature of 25 °C, a correction factor \dot{S}_{x1}/\dot{S}_k of 1.20 is determined. In this case, the suction capacity of the pump to be used should be 1.2 times higher than at the water inflow temperature of 15 °C in order to achieve the suction quantity of 241.84 m³/h.

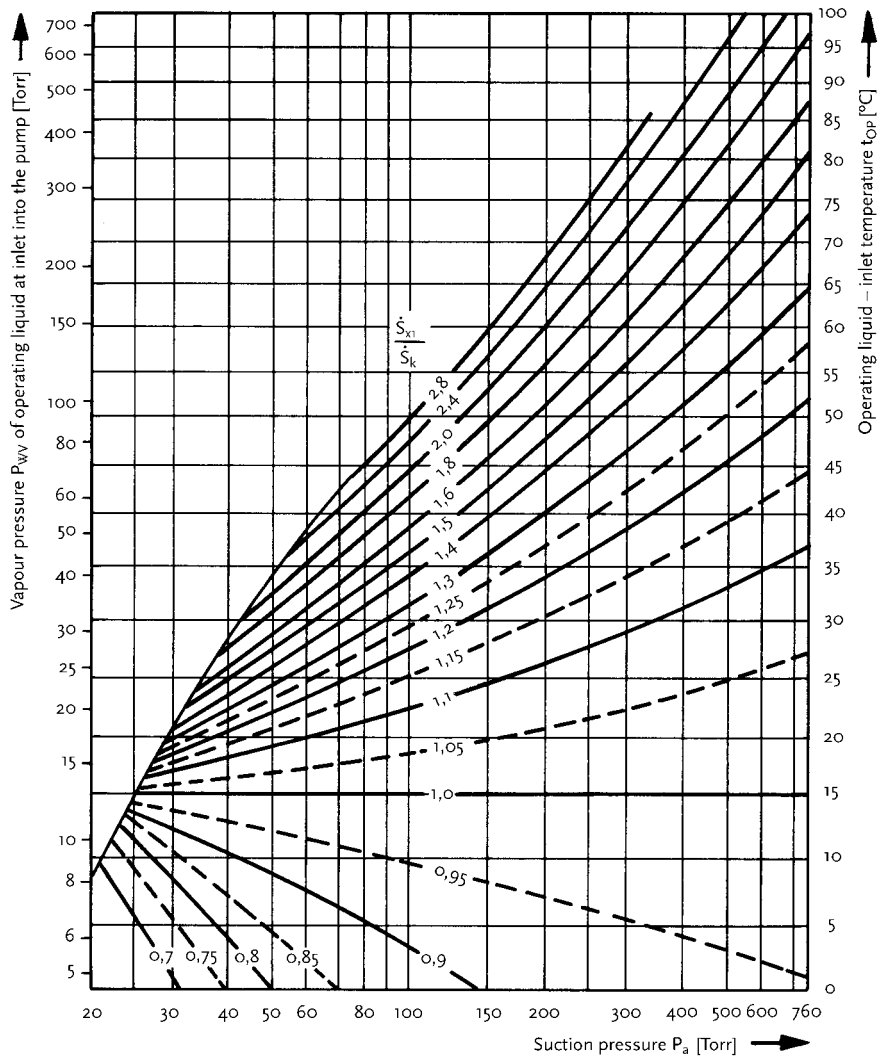


Figure 1-34. Changing of the suction capacity of liquid ring vacuum pumps depending on suction pressure and inflow temperature of the operating liquid. [1.11]

Operating liquid: water

\dot{S}_k Suction capacity of a vacuum pump acc. to dry-air curve at suction temperature of 20 °C and inflow temperature of 15 °C of the operating liquid (water)

\dot{S}_{x1} Required suction capacity of a vacuum pump at an inflow temperature of the operating liquid (water) of t_{op} °C

With the sucked mixture in the present example d) being dry, the required pump suction capacity would be

$$\dot{S}_{x1} / \dot{S}_k = f_1$$

$$\dot{S}_{x1} = 1.2 \cdot \dot{S}_k$$

$$\dot{S}_{x1} = 1.2 \cdot 241.84 \text{ m}^3/\text{h}$$

$$\dot{S}_{x1} = 290.21 \text{ m}^3/\text{h}$$

- f) The condensing effect can occur during the pumping of a gas-vapor mixture. If a liquid ring vacuum pump is used for pumping vapors, ring liquid will vaporize in the impeller cells only until the state of saturation is reached, i. e. equilibrium is established. If the pump sucks gas already saturated with operating liquid, the vaporization of the operating liquid is theoretically impossible. Here, the suction capacity of the vacuum pump is higher by the proportion of the operating liquid otherwise vaporizing. If the temperature of the gas-vapor mixture to be sucked is higher than the temperature of the ring liquid, then cooling takes place in the vacuum pump when both fluids meet and the vapor will condense after having reached the saturation vapor pressure. Thus the vacuum pump has the function of a mixing condenser. Owing to condensation, an additional suction effect arises which is also called condensation effect. With liquid ring machines disposing of characteristic curves based on the suction of dry gases at 20 °C, a correction of the suction capacity similar to the correction curves given in fig. 1-35 is possible for the pumping of dry saturated air-water vapor mixtures within a temperature range of 10 °C and 50 °C and with water as operating liquid.

The exact determination of the corrective factors is to be made acc. to the geometric ratios of the pump. Therefore, the correction diagram should be used only for information purposes.

For a suction temperature of 35 °C and an inlet temperature of the ring liquid of 21 °C the actually arising suction quantity of the vacuum pump is to be determined for a dry saturated air-water vapor mixture acc. to example d). With a presumable inlet temperature of the operational water of 21 °C and on the assumption that dry gas is used, according to the curves in fig. 1-34 the corrective factor $\dot{S}_{x1} / \dot{S}_k = 1.1 = f_1$ results at a suction pressure of 100 mbar (= 74.1 Torr).

This means that the suction capacity of the pump to be used shall be larger by this factor compared to a pump with an inlet temperature of the operating water of 15 °C.

In the case of the given saturated air-water vapor mixture and the mentioned operating conditions the occurrence of the condensation effect at the intake of the vacuum pump has to be considered.

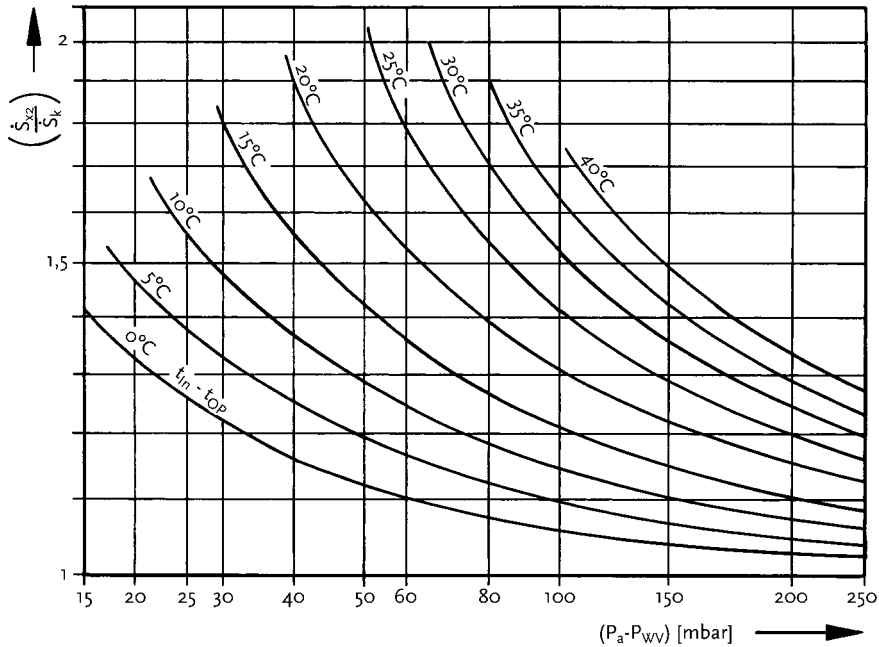


Figure 1-35. Improvement of the suction capacity of liquid ring vacuum pumps due to the condensing effect during the pumping of saturated air-water vapor mixtures [1.10]

Diagram valid for lower limit values in the range of $10^\circ\text{C} \leq t \leq 50^\circ\text{C}$

Operating liquid: water

p_a suction pressure

p_{WV} vapor pressure of the operating liquid

t_{in} temperature of air-water-vapor mixture at pump's suction flange

t_{op} inflow temperature of the operating liquid (water)

\dot{S}_{x2} suction capacity of the vacuum pump resulting from condensation effect

\dot{S}_k suction capacity of the vacuum pump acc. to the dry-air curve at a suction temperature of 20°C and an inflow temperature of the operating liquid (water) of 15°C

Acc. to the curves in fig. 1-35, another corrective factor results which is determined as follows:

$$t_{in} = 35^\circ\text{C}$$

$$t_{op} = 21^\circ\text{C}$$

$$t_{in} - t_{op} = 35^\circ\text{C} - 21^\circ\text{C} = 14^\circ\text{C}$$

$$p_{WV} = 24.86 \text{ mbar at } 21^\circ\text{C (from tables)}$$

$$p_a - p_{WV} = 100 \text{ mbar} - 24.86 \text{ mbar} = 75.14 \text{ mbar}$$

From the diagram the following factor results:

$$\dot{S}_{x2}/\dot{S}_k = f_2$$

$$\dot{S}_{x2}/\dot{S}_k = 1.27$$

which confirms the improvement of the suction capacity of the vacuum pump.

As described above, the suction capacity of liquid ring vacuum pumps depends on the type of suction flow, the features of the operating liquid, the suction pressure, as well as on the temperature of the pumped gas or gas mixture and the inlet temperature of the operating liquid. Based on the correction curves acc. to fig. 1-34 and fig. 1-35, the actual suction capacity of those machines can be approximately determined according to equation (1-175) considering the existing operating conditions.

$$\dot{S}_{\text{act}} = \dot{S}_k \cdot \frac{f_2}{f_1} \quad (1-175)$$

\dot{S}_{act}	actual suction capacity
\dot{S}_{x1}	required suction capacity due to the inlet temperature of the operating water
\dot{S}_{x2}	arising suction capacity resulting from condensation effect
\dot{S}_k	suction capacity acc. to the dry air curve and an inlet temperature of the operating liquid of 15 °C.
$f_1 = (\dot{S}_{x1} / \dot{S}_k)$	Factor influence of the inlet temperature of the operating water
$f_2 = (\dot{S}_{x2} / \dot{S}_k)$	Factor influence of the condensation effect

In the present example, the increase in the suction quantity required under the influence of the increased temperature of the operating liquid could be achieved by increasing the rotational speed or selecting a larger vacuum pump. If, however, the pump ($\dot{S}_k = 241.84 \text{ m}^3/\text{h}$) selected according to the dry air curve is kept at the same rotational speed (see fig. 1-36), the suction capacity deteriorates by the factor f_1 . However, considering the pump condensation effect arising in the pump the suction capacity of this machine will increase, as the following calculation shows:

$$\dot{S}_{\text{act}} = \dot{S}_k \cdot \frac{f_2}{f_1} \quad (1-175)$$

$$\dot{S}_k = 241.84 [\text{m}^3/\text{h}] \text{ (determined in example d)}$$

$$\dot{S}_{\text{act}} = 241.84 \frac{\text{m}^3}{\text{h}} \cdot \frac{1.27}{1.10} = 279.21 \left[\frac{\text{m}^3}{\text{h}} \right]$$

Therefore, the actual suction capacity of the vacuum pump is slightly higher than the required pumping quantity. In practice, this is usually tolerated as a safety margin. The calculated operating points are entered in the characteristic curve field (fig. 1-36).

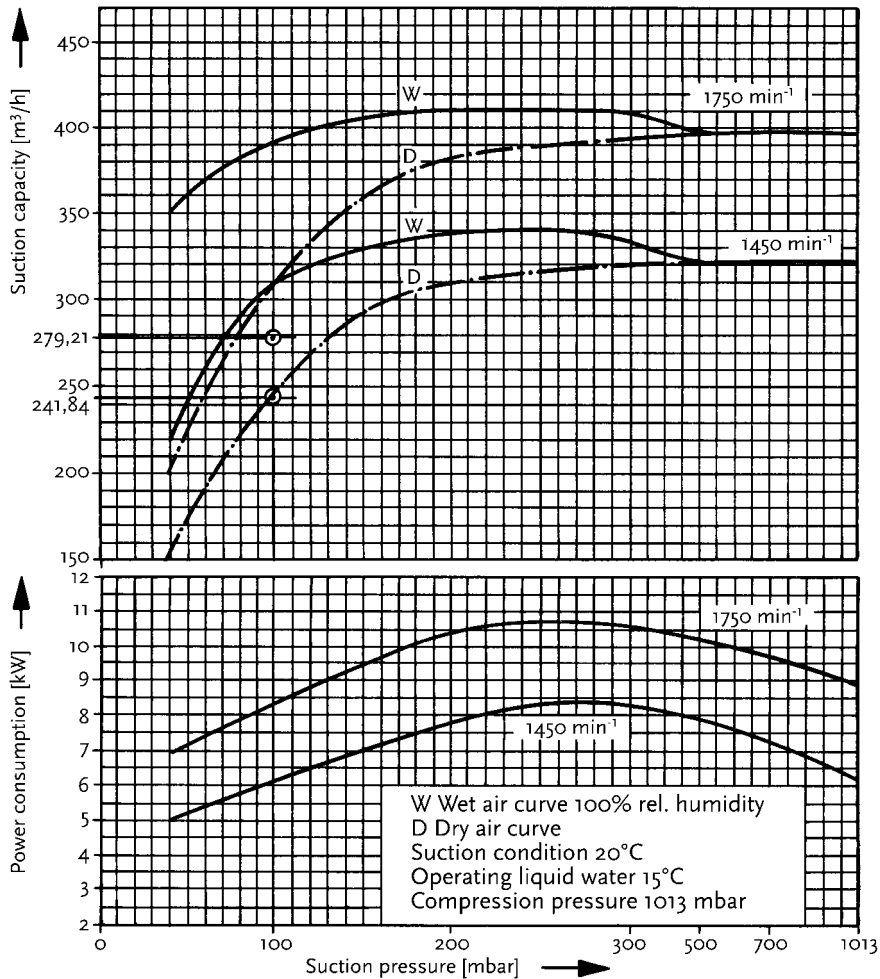


Figure 1-36. Characteristic curve field with dry and wet-air curves of a single-stage liquid ring vacuum pump

- | | |
|--------------------------|---|
| 279.21 m ³ /h | Operating point at the inflow temperature of the operating liquid (water) of 21 °C and a saturated air/-water-vapor mixture of 35 °C at pump's suction flange |
| 241.84 m ³ /h | Operating point at the inflow temperature of the operating liquid (water) of 15 °C and dry gas mixture at pump's suction flange |

- g) During compression to 1013 mbar, a certain quantity of water vapor condenses in the pump. As the temperature of the gas mixture and the temperature of the ring liquid in the pump equalize, we assume that based on a suction temperature of 35 °C and a chosen inlet temperature of the pressure water of 15 °C the common temperature at the pump outlet will be 20 °C.

The proportion of water vapor still existing in the gas mixture at the pump outlet port can be determined according to equation (1-151):

$$\dot{m}_{wv} = \dot{m}_{air} \cdot \frac{M_{wv}}{M_{air}} \cdot \frac{p_{wv}}{p_{air}} \quad (1-151)$$

p_{wv} partial pressure of the water vapor

p_{air} partial pressure of the air

\dot{m}_{wv} water vapor existing at pump outlet

$p_{wv} = 23.37 \text{ mbar at } 20^\circ\text{C (from tables)}$

$p_{air} = 1013 \text{ mbar} - 23.37 \text{ mbar} = 989.63 \text{ mbar}$

$$\dot{m}_{wv} = 12 \frac{\text{kg}}{\text{h}} \cdot \frac{18 \frac{\text{kg}}{\text{kmol}}}{29 \frac{\text{kg}}{\text{kmol}}} \cdot \frac{23.37 \text{ mbar}}{989.63 \text{ mbar}} = 0.175 \left[\frac{\text{kg}}{\text{h}} \right]$$

The total quantity of vapor at the pump inlet has been calculated in example c) with 9.56 kg/h.

Thus the vapor quantity condensed during the compression process in the pump is

$$9.65 \text{ kg/h} - 0.175 \text{ kg/h} = 9.385 \text{ [kg/h]}$$

The air volume of 12 kg/h at the pump outlet at 1013 mbar/20°C is calculated acc. to equation (1-173):

$$\dot{V}_{air} = \frac{83.14 \cdot \frac{\dot{m}_{air}}{M_{air}} \cdot T}{p_{total}} \quad (1-173)$$

$$T = 273.15 \text{ K} + 20^\circ\text{C} = 293.15 \text{ K}$$

$$\dot{V}_{air} = \frac{83.14 \frac{\text{mbar} \cdot \text{m}^3}{\text{kmol} \cdot \text{K}} \cdot \frac{12 \frac{\text{kg}}{\text{h}}}{29 \frac{\text{kg}}{\text{kmol}}} \cdot 293.15 \text{ K}}{1013 \text{ mbar}} = 9.95 \left[\frac{\text{m}^3}{\text{h}} \right]$$

h) In the condenser (see fig. 1-37), condensation will already begin before the vacuum pump.

- Suction temperature $t_1 = 35^\circ\text{C}$
- Suction pressure $p_1 = 100 \text{ mbar}$
- Cooling water temperature for condenser $t_{in} = 25^\circ\text{C}$
- Suction mixture air, saturated with water-vapor
- Air mass to be sucked $\dot{m}_{air} = 12 \text{ kg/h}$

The temperature t_2 after the condenser is assumed to be 30°C . The vapor mass flow before the condenser is 9.56 kg/h acc. to the calculation in example c).

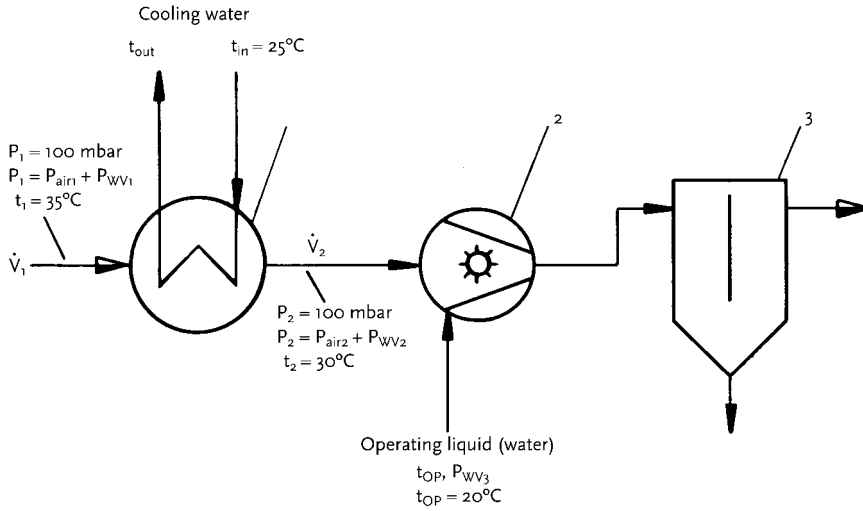


Figure 1-37. Vacuum pump with liquid separator and a condenser on the suction side
1 condenser, 2 liquid ring vacuum pump, 3 liquid separator

The condenser is assumed to show no pressure losses while pressure p_2 after the condenser corresponds to suction pressure p_1 .

The vapor mass after the condenser is calculated according to equation (1-151).

$$\dot{m}_{WV2} = \dot{m}_{air} \cdot \frac{M_{WV}}{M_{air}} \cdot \frac{p_{WV2}}{p_{air2}} \quad (1-151)$$

$$p_2 = p_1 = 100 \text{ mbar}$$

$$p_{total} = p_1 = p_2$$

$$p_{air2} = p_2 - p_{WV2}$$

$$p_{WV2} = 42.43 \text{ mbar}/30^\circ\text{C} \text{ (from tables)}$$

$$p_{air2} = 100 \text{ mbar} - 42.43 \text{ mbar} = 57.57 \text{ mbar}$$

\dot{m}_{WV2} existing vapor mass after the condenser

$$\dot{m}_{WV2} = 12 \frac{\text{kg}}{\text{h}} \cdot \frac{18 \frac{\text{kg}}{\text{kmol}}}{29 \frac{\text{kg}}{\text{kmol}}} \cdot \frac{42.43 \text{ mbar}}{57.57 \text{ mbar}} = 5.48 \left[\frac{\text{kg}}{\text{h}} \right]$$

In the condenser $9.56 \text{ kg/h} - 5.48 \text{ kg/h} = 4.08 \text{ kg/h}$ are condensing.

The accruing volume \dot{V}_2 before the pump is calculated acc. to equation (1-172).

$$\dot{V}_{total} = \frac{83.14 \cdot \left(\frac{\dot{m}_{air}}{M_{air}} + \frac{\dot{m}_{WV2}}{M_{WV}} \right) \cdot T_2}{p_{total}} \quad (1-172)$$

$$T = 273.15 \text{ K} + 30^\circ\text{C} = 303.15 \text{ K}$$

$$\dot{V}_{\text{total}} = \frac{83.14 \frac{\text{mbar} \cdot \text{m}^3}{\text{kmol} \cdot \text{K}} \cdot \left(\frac{12 \frac{\text{kg}}{\text{h}}}{29 \frac{\text{kg}}{\text{kmol}}} + \frac{5.48 \frac{\text{kg}}{\text{h}}}{18 \frac{\text{kg}}{\text{kmol}}} \right) \cdot 303.15 \text{ K}}{100 \text{ mbar}}$$

$$\dot{V}_{\text{total}} = \dot{V}_2 = 180.71 \left[\frac{\text{m}^3}{\text{h}} \right]$$

In example d), a total mixture volume of 241.84 m³/h was determined without a condenser at the suction side. With a condenser installed on the suction side of the vacuum pump, the mixture volume flow in the pump is lower. This may allow the selection of a smaller pump and therefore savings in driving power.

In this case, the vacuum pump installed after the condenser can be designed for a suction capacity of 180.71 m³/h, according to the dry-air curve at a suction pressure of 100 mbar. Due to the existing operating conditions, the actual suction capacity of this vacuum pump must be determined according to the temperature of the operating water, the suction temperature and the suction pressure, as in example f).

According to the diagram in fig. 1-34, it results:

$$\frac{\dot{S}_{x1}}{\dot{S}_k} = 1.08 = f_1$$

$$p_2 = 100 \text{ mbar} (= 74.1 \text{ Torr})$$

$$t_{\text{op}} = 20^\circ\text{C}$$

Through correction due to the condensation effect acc. to diagram 1-35 it results:

$$\frac{\dot{S}_{x2}}{\dot{S}_k} = 1.20 = f_2$$

$$p_2 = 100 \text{ mbar}$$

$$p_{\text{wv}_3} = 23.37 \text{ mbar}/20^\circ\text{C}$$

$$t_2 = 30^\circ\text{C}$$

$$t_{\text{op}} = 20^\circ\text{C}$$

$$p_2 - p_{\text{wv}_3} = 100 \text{ mbar} - 23.37 \text{ mbar} = 76.63 \text{ mbar}$$

$$t_2 - t_{\text{op}} = 30^\circ\text{C} - 20^\circ\text{C} = 10^\circ\text{C}$$

Thus, at 100 mbar suction pressure, the actual suction capacity of the vacuum pump (S_{act}) is

$$\dot{S}_{\text{real}} = \dot{S}_k \cdot \frac{f_2}{f_1} \quad (1-175)$$

$$\dot{S}_k = 180.71 \left[\frac{\text{m}^3}{\text{h}} \right]$$

$$\dot{S}_{\text{real}} = 180.71 \frac{\text{m}^3}{\text{h}} \cdot \frac{1.20}{1.08} = 200.79 \left[\frac{\text{m}^3}{\text{h}} \right]$$

The actual suction capacity of the vacuum-pump of $200.79 \text{ m}^3/\text{h}$ is slightly higher than the accruing volume flow rate of $180.71 \text{ m}^3/\text{h}$ measured at the outlet of the condenser at a pressure of 100 mbar.

The difference between the discharge rate of the pump and process gas quantity to be sucked can be fed to the vacuum pump through ventilation valves or through bypass via orifices, as well as via manually or automatically controlled valves.

1.8.7

Gases in mixtures with overheated vapors [1.11]

Engineering usually deals with ideal gases or gas-vapor-mixtures. With gas-vapor-mixtures it often happens that the vapors existing in the mixture are dry saturated vapors. For the calculations of these mixtures, the laws for ideal gas are applied, i. e. these mixtures are assumed to behave almost like ideal gases. In practice, even mixtures of gases and overheated vapors occur. These mixtures are also calculated roughly according to the laws applied for ideal gases.

Any mixing ratio of the respective gas proportions is possible for mixtures of ideal gases. However, the miscibility of ideal gases with vapors is subject to certain laws. The temperatures of vapors can be randomly high, i. e. higher than the boiling temperatures corresponding to their partial pressures. The occurring vapor temperatures, however, are by no means lower than the boiling temperatures belonging to their respective partial pressures. This results in a partial vapor pressure of a vapor arising in a mixture that takes on its specific vapor pressure corresponding to the temperature of the mixture. As in an ideal gas mixture the volume proportions of the existing substances behave like their partial pressures, the maximal vapor proportion in the mixture is determined by the boiling pressure of the vapor assigned to the mixing temperature. If partial pressure p_p of a vapor in the mixture already corresponds to its boiling pressure p_s at the existing temperature of the mixture, this vapor exists with its highest volume proportion. So the gas mixture contains vapor in dry saturated state. Such a mixture is also called mixture saturated with vapor.

If vapor in the mixture exists with a smaller proportion, this means that its partial pressure p_p is smaller than its saturated pressure p_s .

- p_p partial pressure
- p_s saturated pressure
- φ factor for relative humidity, without dimensions

Using the factor φ , the value of which can lie between zero and one, and which is a measure for the relative humidity the following equation results:

$$p_p = p_s \cdot \varphi \quad (1-176)$$

If p_d is smaller than p_s , the vapor is overheated, i.e. the gas mixture could take up further vapor.

If the proportion of vapor of a mixture is given in kilogram it is called absolute humidity. The quantity of vapor in kilogram contained in one kilogram of the vapor-free dry mixture is usually given with the letter

$$\kappa \left[\frac{\text{kg}}{\text{kg}} \right].$$

A mixture consisting of an ideal gas and an overheated vapor can be calculated according to the general state equation for ideal gases. For the proportion of vapor, however, this calculation is only approximately valid.

General state equation:

$$p \cdot V = m \cdot R \cdot T \quad (1-17)$$

Definitions for the mixture

units	ideal gas	overheated vapor
mass	m_g	m_v
partial pressure	p_g	p_v
specific gas constant	R_g	R_v
	mixture of gas and overheated vapor	
total volume		V_{total}
total pressure		p_{total}
temperature		T_{mix}

The state equation for ideal gases:

$$V_{\text{total}} \cdot p_g = m_g \cdot R_g \cdot T_{\text{mix}} \quad (1-177)$$

The state equation for overheated vapors:

$$V_{\text{total}} \cdot p_v = m_v \cdot R_v \cdot T_{\text{mix}} \quad (1-178)$$

Dividing both equations, it results

$$\frac{p_v}{p_g} = \frac{m_v}{m_g} \cdot \frac{R_v}{R_g} \quad (1-179)$$

and

$$\kappa = \frac{m_v}{m_g} = \frac{R_g}{R_v} \cdot \frac{p_v}{p_g} \quad (1-180)$$

Using factor φ the result is,

$$p_v = p_s \cdot \varphi \quad (1-176)$$

$$p_g = p_{\text{total}} - p_v$$

resp.

$$p_g = p_{\text{total}} - p_s \cdot \varphi \quad (1-181)$$

and therefore

$$\kappa = \frac{R_g}{R_v} \cdot \frac{p_s \cdot \varphi}{p_{\text{total}} - p_s \cdot \varphi} \left[\frac{\text{kg}}{\text{kg}} \right] \quad (1-182)$$

$$\kappa \cdot (p_{\text{total}} - p_s \cdot \varphi) = \frac{R_g}{R_v} \cdot \varphi \cdot p_s$$

$$\varphi = \frac{\kappa}{\left(\frac{R_g}{R_v} + \kappa \right)} \cdot \frac{p_{\text{total}}}{p_s} \quad (1-183)$$

The largest proportion of vapor results from the maximum possible relative humidity, i. e. $\varphi = 1.0$:

$$\kappa_{\text{max}} = \frac{R_g}{R_v} \cdot \frac{p_s}{p_{\text{total}} - p_s} \left[\frac{\text{kg}}{\text{kg}} \right] \quad (1-184)$$

Example 1.19.

For the gas-vapor-mixture, i.e. humid air often occurring in practice the general equations for atmospheric pressure of

- a) absolute humidity κ
- and
- b) relative humidity φ

are to be drawn up.

Solution:

$$\text{gas constant air } R_g = 0.2871 \left[\frac{\text{kJ}}{\text{kg} \cdot \text{K}} \right]$$

$$\text{gas constant water vapor } R_v = 0.4615 \left[\frac{\text{kJ}}{\text{kg} \cdot \text{K}} \right] \quad (\text{from table 1.-15.})$$

Therefore:

$$\frac{R_g}{R_v} = \frac{0.2871 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}}{0.4615 \frac{\text{kJ}}{\text{kg} \cdot \text{K}}} = 0.622$$

$$p_{\text{total}} = 1013 \text{ [mbar (abs)]}$$

$$p_s \text{ [mbar (abs)]}$$

a) from equation

$$\kappa = \frac{R_g}{R_v} \cdot \frac{p_s \cdot \varphi}{p_{\text{total}} - p_s \cdot \varphi} \quad (1-182)$$

it results:

$$\kappa = 0.622 \cdot \frac{p_s \cdot \varphi}{1013 - p_s \cdot \varphi} \left[\frac{\text{kg}}{\text{kg}} \right]$$

b) from the equation

$$\varphi = \frac{k}{\left(\frac{R_g}{R_v} + \kappa \right)} \cdot \frac{p_{\text{total}}}{p_s} \quad (1-183)$$

it results:

$$\varphi = \frac{\kappa}{0.622 + \kappa} \cdot \frac{1013}{p_s}$$

1.8.8

Condensation and cavitation

In cases with high proportions of vapor on the suction side condensing in the pump during compression it may happen that the pump lacks in sufficient inert gases, and towards the end of compression there is not enough compressible media. Then, cavitation on the pressure side occurs. As is known, cavitation on the suction side occurs when the liquid ring vacuum pump is operated at a suction pressure close to the boiling temperature of the operating liquid belonging to this pressure.

To avoid cavitation on the pressure side, the pump must always deliver as much gas – mainly inert gases – as corresponds to the suction capacity of the pump at the lowest suction pressure acc. to its characteristic curve.

1.9

Change of gas state during the compression process

In thermodynamics, two ideal borderline cases of the change of the gas state are distinguished, i.e. isothermal and adiabatic (isentropic) processes which can only be approximately realized in practice.

Distinctive quantities of these processes are pressure ratio and compression work.

1.9.1

The isothermal compression

In the isothermal compression process, the temperature of the compressed gas remains constant, i. e. the total heat accruing during compression is evacuated by respective measures. Therefore, between its initial and final state, the compressed gas behaves acc. to the Boyle-Mariotte Law (equation 1-167).

This process presupposes ideal heat conductivity.

$$\frac{p_2}{p_1} = \frac{V_1}{V_2} \quad (1-167)$$

The volume V_1 equals, for example, the piston displacement of a piston compressor. According to the equation

$$p \cdot V = m \cdot \frac{R_o}{M} \cdot T \quad (1-65)$$

the required isothermal compression work W_{is} depends on this pressure ratio and on the temperature, it is, however, independent from the type of gas. The compression work is calculated as follows [1.3]:

$$W_{is} = p_1 \cdot V_1 \cdot \ln \frac{p_2}{p_1} \quad (1-185)$$

W_{is} corresponds to the heat to be evacuated in the isothermal compression process.

1.9.2

The adiabatic compression

In the adiabatic compression process, the heat accruing during compression process is not evacuated, but remains preserved by means of ideal heat insulation. The whole energy is used for increasing the gas temperature. According to S.D. Poisson, the following applies to the pressure ratio [1.3]:

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1} \right)^{\frac{\kappa}{\kappa-1}} = \left(\frac{V_1}{V_2} \right)^{\kappa} \quad (1-186)$$

The adiabatic compression work W_{ad} to be used is calculated acc. to equation (1-187).

$$W_{ad} = \frac{\kappa}{\kappa-1} \cdot p_1 \cdot V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \quad (1-187)$$

The compression work is determined by the type of gas, the temperature and the pressure ratio.

1.9.3

Adiabatic exponent κ

The adiabatic exponent κ corresponds to the ratio of the specific heat capacity of the gas at constant pressure c_p and at constant volume c_v (comp. table 1-116).

$$\kappa = \frac{c_p}{c_v} \quad (1-188)$$

c_p specific heat capacity at constant pressure

c_v specific heat capacity at constant volume

κ adiabatic exponent

As the heating-up process of a gas quantity enclosed in a chamber may proceed differently with the same change of temperature, different specific heat capacities of the same gas occur. Two borderline cases are distinguished. In one case, the gas volume is kept constant while the temperature of the gas increases. The gas pressure rises with the increasing temperature. In the other case, the pressure remains constant, temperature and volume, however increase. Between these two borderline cases, there is an unlimited number of processes with simultaneous changes of pressure and volume.

Fig. 1-38 shows the possible changes of state. [1.5].

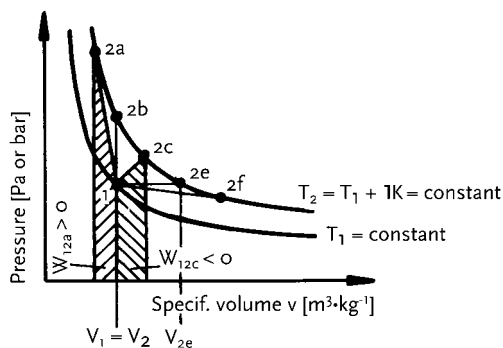


Figure 1-38. Change of the state of a gas through heating up by one degree Kelvin.

In the figure, the change of state from state 1 to state 2b is a change of state at constant volume with no external work being performed. The change of state from point 1 to point 2e takes place at constant pressure, with the output work quantity $W_{12} = p(v_{2e} - v_1)$.

In the final state 2a, the gas is supplied with a specific external work W_{12a} , in the final state 2b specific external work is neither supplied nor put out. In the final state 2c the gas puts out specific external work.

Owing to the fact that there are a lot of possibilities of change-of-state work, there are just as many specific heat capacities for one and the same gas. In practice, only two changes of state are considered:

- The change of state at constant volume and the specific heat capacity c_v and
- the change of state at constant pressure and the specific heat capacity c_p .

Although the specific heat capacities c_v and c_p are different for each gas, the ratio of c_v to c_p is the same for all biatomic gases. Those ratio values (κ) are known for most of the gases. For gaseous water in the ideal state $\kappa = 1.33$.

Table 1-16. Specific heat capacity c_p , c_v and κ -values of different technical gases [1.9]

Type of gas	$c_p \left[\frac{\text{kJ}}{\text{kg} \cdot \text{K}} \right]$	$c_v \left[\frac{\text{kJ}}{\text{kg} \cdot \text{K}} \right]$	κ
H ₂	14.21	10.09	1.41
He	5.194	3.117	1.66
N ₂	1.040	0.743	1.40
O ₂	0.915	0.655	1.40
Air	1.004	0.718	1.40
CO	1.040	0.743	1.40
CO ₂	0.818	0.628	1.30
H ₂ O	1.858	1.396	1.33
SO ₂	0.607	0.477	1.27
NH ₃	2.055	1.567	1.32
CH ₄	2.158	1.640	1.32
C ₂ H ₂	1.629	1.310	1.23
C ₂ H ₄	1.496	1.200	1.24
C ₂ H ₆	1.654	1.377	1.20
C ₃ H ₈	1.512	1.325	1.14
n-C ₄ H ₁₀	1.623	1.480	1.10

specific heat capacity c_p and c_v at $p=0$ bar and $t=0^\circ\text{C}$, $\kappa = c_p / c_v$ at $t=0^\circ\text{C}$

The compression work to be performed in the isothermal compression process is always lower than the adiabatic compression work. With the increasing pressure ratio, this difference in compression work increases. For air, the ratio values given in table 1-17 result when compressing from p_1 to p_2 [1.3].

Table 1-17. Isothermal compression work in relation to adiabatic compression work of air
Compression from $p_1 = 101$ kPa to values of p_2 acc. to the table.

W_{is} = isothermal compression work

W_{ad} = adiabatic compression work

p_2 [kPa]	200	500	1000
$\frac{W_{is}}{W_{ad}}$	0.896	0.786	0.704

Real compression processes are called polytropic processes. They lie between the two borderline cases of ideal heat dissipation and ideal heat insulation. By means of cooling, a state is reached in which the gas is not so intensely heated, i.e. the convergence to an isothermal process. Processes running in cooled compressors are subject to the laws according to the equations (1-186) and (1-187), replacing the adiabatic exponent κ by the polytropic exponent n [1.3].

This change of state is subject to the law

$$p \cdot V^n = \text{const.} \quad (1-189)$$

For technically normal changes of state, n lies mostly within the range of $1 < n < \kappa$.

For isothermal, polytropic and adiabatic compression work at the same pressure ratios, the following applies:

$$W_{\text{is}} < W_{\text{pol}} < W_{\text{ad}}$$

W_{is} isothermal compression work
 W_{pol} polytropic compression work
 W_{ad} adiabatic compression work

The polytropic exponent of biatomic gases (e. g. air) ranges from about 1.25 to 1.35. In centrifugal compressors, the polytropic exponent n can even be larger than the adiabatic exponent κ , due to higher flow speeds and higher internal friction.

1.9.4

Especially distinguished changes of state

The four changes of states which are different according to their thermal state quantities may be regarded as special cases of polytropic curves. An overview is given in table 1-18.

Table 1-18. Changes of state in the p, V diagram [1.9]

change of state	exponent	Explanation acc. to $p \cdot V^n = \text{const.}$
isobars	$n=0$	$p \cdot V^0 = \text{const.}: p = \text{const.}$
isotherms	$n=1$	$p \cdot V^1 = \text{const.}: p \cdot V = \text{const.}$
isentropes	$n=\kappa$	$p \cdot V^\kappa = \text{const.}$
isochores	$n=\pm \infty$	$p^{\frac{1}{n}} \cdot V = \text{const.}: V = \text{const.}$

n = polytropic exponent

1.10

Names and definitions in vacuum technology

(extract from DIN 28400 Part 1 – July 1979)

Absorption. Absorption is a sorption with the gas (absorbate) penetrating the solid body or liquid (absorbent).

Adsorption. Adsorption is a sorption with the gas (adsorbate) being bound on the surface of a solid body or a liquid (adsorbent).

Average particle velocity. The average particle velocity \bar{v} is the average amount of the particle velocities.

Characteristic flow conductivity, Intrinsic conductivity. The characteristic flow conductivity is the flow conductivity in the case of an orifice or a duct connecting two vessels with Maxwell's velocity distribution existing in each vessel. In the case of molecular flow, the characteristic flow conductivity is the product of flow conductivity at the inlet orifice of the duct and the throughput probability of the particles. In this flow range, the characteristic flow conductivity is independent from pressure. The flow conductivity usually given in catalogues corresponds to this definition.

Characteristic flow resistance. Characteristic flow resistance is the inverse value of the characteristic flow conductivity.

Chemisorption. Chemisorption is an adsorption developing chemical compounds.

Collision rate. Collision rate is the quotient of the average number of impacts of a particle with other particles during its moving through a gas within a period of time and that period of time.

Gassing off, gas release. The gassing-off is a spontaneous desorption.

Degassing. Degassing is desorption accelerated by physical processes.

Desorption. Desorption is the release of sorbed gases from a sorbent. The release may be spontaneous or accelerated by physical processes.

Diffusion coefficient. The diffusion coefficient is the absolute amount of the quotient of particle flow density and the concentration gradient. The particle flow refers to a surface element perpendicular to the direction of the gradient.

Dwell time. Dwell time is the average period within which a particle in adsorbed state remains on the surface.

Flow conductivity. The flow conductivity of an orifice in a thin wall, or of a duct or of a duct piece between two defined cross-sections is the quotient of the $p\dot{V}$ -flow and the difference of the pressures existing on both sides of the orifice, or of the duct or on the two defined cross-sections of the duct piece, with the temperature being constant in the system. The obtained measured value depends on the method of pressure measurement.

Flow resistance. Flow resistance is the inverse value of the flow conductivity.

Gas (in general). Gas is a matter in a state of aggregation in which the mean distances between the particles are large in comparison to their sizes, and in which the mutual order of the individual particles changes continuously.

Gas (in the stricter sense). Gas in the stricter sense is a matter in the gaseous state of aggregation which cannot be transformed into the liquid or solid state of aggregation by compression at prevailing temperatures.

Gas consumption. Gas consumption is a sorption resulting from an electric discharge.

Gas diffusion. The gas diffusion is the movement of the gas resulting from its concentration gradient in another medium. The medium can be gaseous, liquid or solid.

Gas dynamic flow. The gas dynamic flow is the flow of a gas with comparably large changes of density characterized by high Mach-Numbers (e. g. supersonic flow in nozzle- or steam jet pumps).

Gas permeability. The gas permeability is a parameter (a quantity related to the type of matter) for permeation of a gas through a certain solid or liquid at a certain temperature. It is calculated from the quotient of the $p\dot{V}$ flow through a solid or liquid of finite thickness and a quantity being a function of the pressures existing on both sides of a solid or liquid. The kind of function depends on the physical processes taking part in the current permeability.

Gas permeation. Gas permeation is the penetration of gas through a solid or liquid of finite thickness. Permeation includes diffusion and surface effects.

Gettering. Gettering is a link of gas, mostly through chemical reactions. Frequently, getters (gettering matters) have large real surfaces.

Ideal vacuum. In vacuum technology ideal vacuum is the state in a space with the particle density being zero (up to now called “absolute vacuum”).

Knudsen flow. Knudsen flow is a gas flow in the area between viscous flow and molecular flow.

Laminar flow. The laminar flow (layer- or thread flow) is viscous flow without mixing movements at small Reynold’s numbers.

Mean free path. Mean free path is the average value of distances a particle flies through between two successive impacts with other particles.

Mass flow. Mass flow is the quotient of the mass of a gas flowing through a duct cross-section during a period of time and that period of time.

Maxwell’s velocity distribution. Maxwell’s velocity distribution is the velocity distribution according to the Maxwell-Boltzmann distribution function. This velocity distribution exists for particles of a gas in equilibrium at a certain temperature and for sufficiently large distances from the limiting wall.

Molecular effusion. Molecular effusion is a gas flow through an orifice in a wall with insignificant thickness with the largest width of the orifice being smaller than the mean free path.

Molecular flow. The molecular flow is a gas flow through a duct with the mean free path being very large compared with the largest bore size of the duct cross-section. In molecular flow, the flow resistance is independent from pressure.

Most probable particle velocity. The most probable particle velocity v_w is the particle velocity which corresponds to the maximum value of the Maxwell & Boltzmann distribution function.

Occlusion. Occlusion is the encapsulation of gas quantities in solids or liquids.

Partial pressure. Partial pressure is the pressure of a certain gas or vapor in a mixture of gases and/or vapors.

Particle flow conductivity. Particle flow conductivity of an orifice in a thin wall (wall with insignificant thickness) or of a duct or of a duct piece between two defined cross-sections is the quotient of the particle flow rate and the difference of the average particle number densities on the both sides of the orifice, or of a duct or on the both cross-sections of the duct piece.

Particle flow density. The particle flow density is the quotient of the particle flow rate through a surface element and the volume of the surface element.

Particle flow rate. The particle flow rate through a given surface element is the quotient of the difference of the average number of particles flowing through the surface element in one direction within a period of time and the average number of particles flowing through the element in the opposite direction divided by that period of time.

Particle number density (volume-related particle number). The particle number density is the quotient of the number of particles contained in a suitably selected volume element and the volume of that element.

Particle number density of a component in gas mixture. The particle number density of a component in a gas mixture is the quotient of the number of particles of this gas component contained in a suitably selected volume element and the volume of that element.

Permeation coefficient. The permeation coefficient is the product of gas permeability and the quotient of the thickness of the solid or liquid and the content of the surface on the side of the higher pressure.

Physical adsorption. Physical adsorption is an adsorption caused by physical forces which does not result in certain chemical compounds.

Poiseuille's flow. Poiseuille's flow is the special case of the laminar flow in a long pipe with a circular-cylindrical cross-section. In that case, a parabolic velocity distribution occurs.

Pore penetration; transporisation. Transporisation is a gas flow through a porous body due to a pressure difference.

Pressure units. The legal pressure units are Pascal as SI-unit (Pa),

1 Pascal = 1 Newton divided by 1 square meter,

1 Pa = $1 \text{ N} \cdot \text{m}^{-2}$,

and bar as a special unit name for 10^5 Pa,

1 bar = 1000 mbar = $10^5 \text{ Pa} = 10^5 \text{ N} \cdot \text{m}^{-2}$

The common unit in vacuum technology is millibar.

pV-flow. The pV-flow is the quotient of the pV-value of a gas flowing through a pipe cross-section at the prevailing temperature within a period of time and that period of time.

pV-value. The pV-value is the product of pressure and volume of a certain quantity of a gas at the currently prevailing temperature. If the pV-value is to be a measure for the quantity of matter or the mass of a gas, then it needs to be an ideal gas with the temperature being specified.

Root of the average velocity square. The root of the average velocity square $\sqrt{\bar{v}}$ is the root of the average amount of squares of the particle velocities.

Saturation vapor pressure. Saturation vapor pressure is the pressure exerted by a vapor in thermodynamic equilibrium with one of its condensed phases at a certain temperature.

Sorption. Sorption is the binding of gas (sorbate) by a solid or liquid (sorbent). (Sorbents are also called sorption means).

Flow of the amount of substances, molar flow. The molar flow is the quotient of the amount of substances of a gas flowing through a duct cross-section within a period of time and that period of time.

Surface diffusion. Surface diffusion is a surface migration owing to a gradient of the chemical potential, e.g. due to a concentration gradient.

Surface migration. Surface migration is the movement of particles on a surface. The surface migration of a liquid film is called creeping.

Thermal effusion. Thermal effusion is a gas flow in the range of molecular flow through a duct system between two vessels with different temperatures. It leads to a state of equilibrium which maintains a pressure difference between the two vessels.

Total pressure. The total pressure is the sum of the existing partial pressures. This word is used when the shorter term “pressure” does not allow the clear distinction between the individual partial pressures and their sum in the given context.

Flow-through probability (Clausing factor). The flow-through probability is the probability that a gas particle penetrating a duct element through an orifice in any direction will leave it through the duct outlet without having passed the inlet again in the opposite direction. In this case, the reflexion according to the Cosine law is assumed for the interaction with the walls.

Turbulent flow. The turbulent flow is a viscous flow with mixing movement above a critical Reynold’s number (for circular-cylindrical pipes, $Re \approx 2300$).

Vapor. Vapor is a substance in gaseous state of aggregation existing either in thermodynamic equilibrium with its liquid or solid phase (*saturated vapor*), or which can be brought to thermal equilibrium (condensing) through compression at the prevailing temperature (*unsaturated vapor*).

Note: In vacuum engineering, the word “gas” in the broader sense is used both for a non-condensable gas and for vapor, if a distinction is not required.

Viscous flow. The viscous flow is a gas flow through a duct with the mean free path being very small compared with the smallest bore size of the pipe cross-section. The flow depends on the viscosity of the gas and can be laminar or turbulent. In case of viscous flow, flow resistance depends on the pressure.

Volume flow rate. Volume flow rate is the quotient of the volume of a gas flowing through a pipe cross-section at currently prevailing temperature and pressure during a period of time and that period of time.

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2

Machines for Vacuum Generation

2.1

Overview of vacuum pumps

In German industry standards, vacuum pumps known in vacuum technology are classified by their operation principle and design. A large number of subgroups are allocated to two main groups of pumps (gas transfer vacuum pumps and gas-binding vacuum pumps). Figure 2-1 shows an overview of the whole spectrum of vacuum pumps.

2.2

Description of vacuum pumps and their functioning

(Extract from DIN 28400, Part 2 – October 1980)

Machines for vacuum generation usually used in industry are mentioned below and described acc. to their functioning.

2.2.1

Gas transfer vacuum pumps

A gas transfer vacuum pump is a pump which transfers gases from an inlet port to an outlet port.

Displacement vacuum pump

A displacement vacuum pump is a gas transfer vacuum pump in which a gas-filled suction chamber is periodically separated from the inlet port with the gas being afterwards transferred to the outlet port. In most displacement vacuum pumps, the gas is compressed before its discharge through the outlet port. Displacement vacuum pumps are preferably applied in the range of viscous flow.

The different types of displacement vacuum pumps have special names.

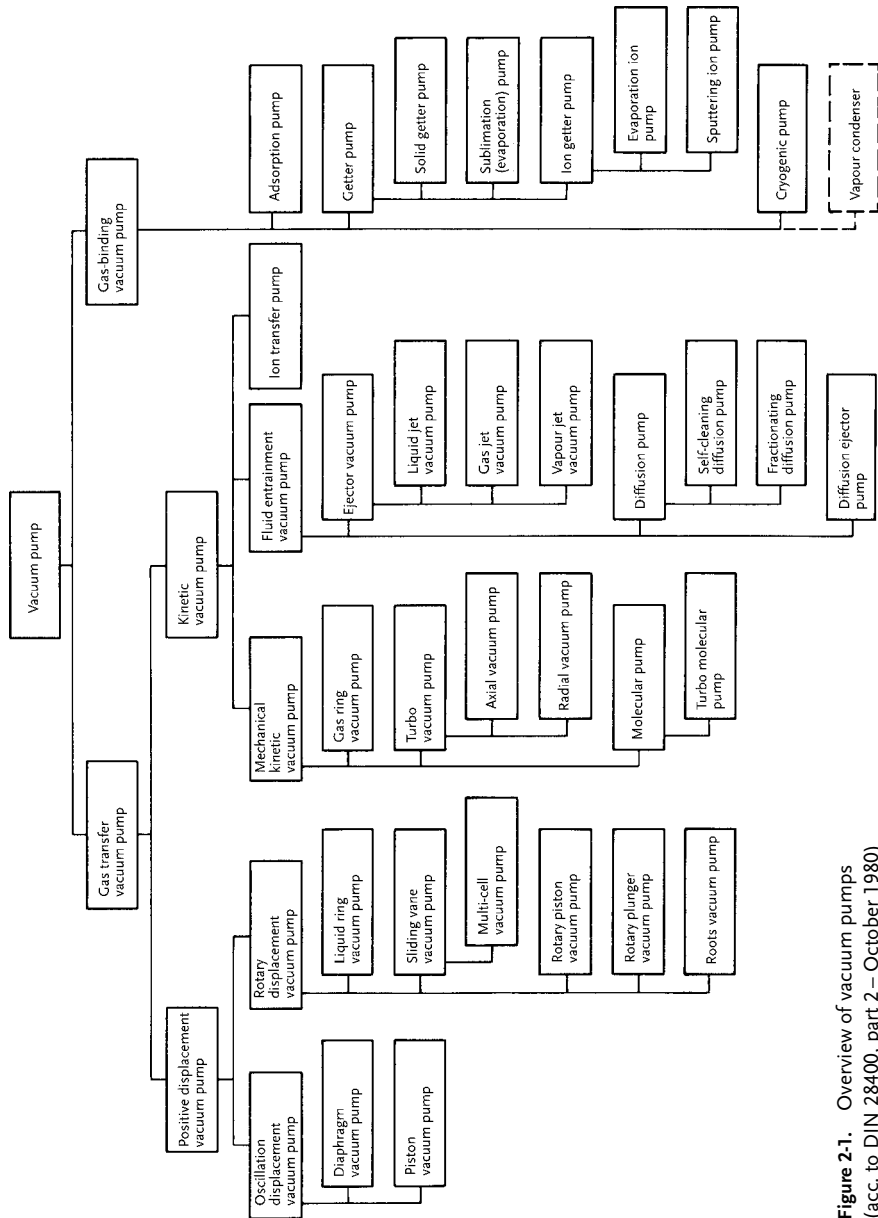


Figure 2-1. Overview of vacuum pumps
(acc. to DIN 28400, part 2 – October 1980)

Oil-sealed (liquid-sealed) vacuum pump

An oil-sealed (liquid-sealed) vacuum pump is a positive displacement vacuum pump in which oil (liquid) is used to seal the clearance between the parts moving against each other and to reduce the dead zone in the pump body at the end of the compression process.

Dry-running vacuum pump

A dry running vacuum pump is a displacement vacuum pump which works without oil sealing (liquid sealing) according to the description above (e.g. diaphragm pump).

Note: It shall be observed that in this connection, dry running does not necessarily mean “with dry bearing”.

Gas-ballast vacuum pump

A gas-ballast vacuum pump is a displacement vacuum pump operating with compression in which a controlled quantity of a gas (gas ballast) is additionally let in (usually into the compression chamber) in order to avoid or to diminish condensation inside the pump.

Reciprocating displacement vacuum pump

A reciprocating displacement vacuum pump is a displacement vacuum pump, in which the suction chamber volume is periodically changed by moving a wall (e.g. piston or diaphragm) to and fro.

Reciprocating piston vacuum pump

A reciprocating piston vacuum pump (fig.2-2) is a reciprocating displacement vacuum pump in which the wall moved to and fro is a movable piston inside a cylinder.

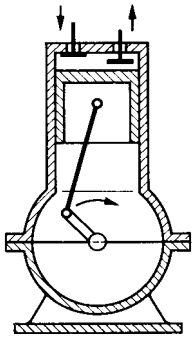


Figure 2-2. Reciprocating piston vacuum pump

Diaphragm vacuum pump

A diaphragm vacuum pump is a reciprocating displacement vacuum pump in which the moved wall is a diaphragm.

Rotary displacement vacuum pump

A rotary displacement vacuum pump is a displacement vacuum pump in which the suction chamber volume is periodically changed by revolving components (e.g. pistons).

Liquid ring vacuum pump

A liquid ring vacuum pump (fig.2-3) is a rotary displacement vacuum pump in which a rotor on eccentric bearing with fixed blades forces a liquid against the wall of the stator (casing). The liquid takes the shape of a ring concentrically to the stator and forms several chambers with changing volume together with the rotor blades.

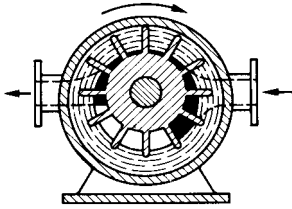


Figure 2-3. Liquid ring vacuum pump

Sliding vane vacuum pump

Sliding vane vacuum pump (fig. 2-4) is a rotary displacement vacuum pump in which a rotor with eccentric bearing tangentially slides past the inside wall of the stator. Two or more gate valves placed in rotor slots (radially, as a rule) and moving there slide along the inside wall of the stator and divide the pump chamber into sections of changing volumes.

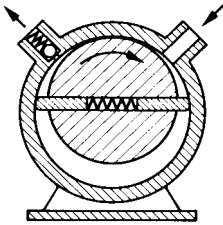


Figure 2-4. Sliding vane vacuum pump

Multi cell vacuum pump

A multi cell vacuum pump (fig. 2-5) is a sliding vane vacuum pump with more than 3 sliding vanes.

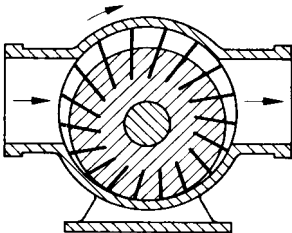


Figure 2-5. Multi cell vacuum pump

Rotary piston vacuum pump

A rotary piston vacuum pump is a rotary displacement vacuum pump with the piston cross-section having a trochoidal shape (e.g. ellipse) the centre of gravity of which describes a circular path.

Rotary plunger vacuum pump

A rotary plunger vacuum pump (fig. 2-6) is a rotary displacement vacuum pump in which the piston moves eccentrically to the inside wall of the casing. A gate valve rigidly connected with the rotor divides the pump chamber in two parts with changing volume. The flat slide gate valves slides within a bushing journaled in the casing.

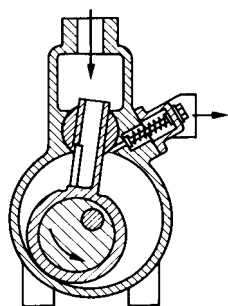
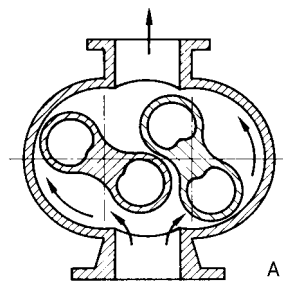


Figure 2-6. Rotary plunger vacuum pump

Roots vacuum pump

A Roots vacuum pump (fig. 2-7) is a rotary displacement vacuum pump in which two (mostly eight-shaped) impellers with identical cross-sections counter-rotate along the casing wall without touching each other. The two rolling pistons are synchronised so that they pass by each other with a slight clearance. In the suction chamber of the pump, compression doesn't occur.



A Figure 2-7. Roots vacuum pump

Kinetic vacuum pump

A kinetic vacuum pump is a gas transfer vacuum pump in which a pulse is given to a certain gas quantity or to individual particles in order to make the gas move from the inlet port to the outlet port.

Mechanical kinetic vacuum pump

A mechanical kinetic vacuum pump is a kinetic vacuum pump in which the pulse is given by a rapidly moving rigid wall.

Turbo vacuum pump

A turbo vacuum pump is a mechanical kinetic vacuum pump working in the range of viscous flow. By means of rapidly rotating components, the pulse is transferred directly to the gas to be pumped by the pump.

Radial vacuum pump

A radial vacuum pump (fig. 2-8) is a turbo vacuum pump in which the gas to be pumped flows through the casing preferably in a radial direction.

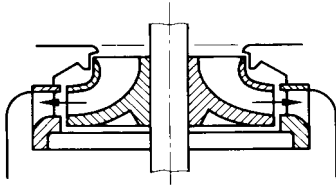


Figure 2-8. Radial vacuum pump

Axial vacuum pump

An axial vacuum pump (fig. 2-9) is a turbo vacuum pump in which the gas to be pumped flows through the casing in an axial direction.

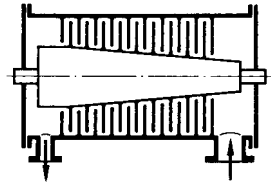


Figure 2-9. Axial vacuum pump

Gas ring vacuum pump

A gas ring vacuum pump (fig. 2-10) is a mechanical kinetic vacuum pump working in the range of viscous flow. In this kind of pump, the rapidly moving rigid wall is a rotating impeller with or without blades in a ring-shaped casing moving along the circumferential boundary of the ring, with a small clearance. The gas flows through an axial inlet port into the chamber on the side of the impeller (lateral channel) and

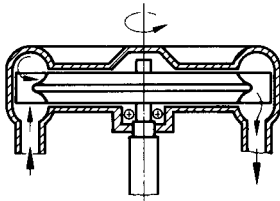


Figure 2-10. Gas ring vacuum pump

receives a pulse in the direction of rotation aimed onto the likewise axial outlet port. A gas ring vacuum pump can be provided with one side channel or two parallel lateral channels, one on each side of the impeller.

Molecular pump

A molecular pump (fig. 2-11) is a mechanical kinetic vacuum pump working in the range of molecular flow in which gas particles receive a pulse through their collision with the rotor surfaces with high rotation speed. The pulse directs the movement of the particles to the outlet port. The circumferential speed of the rotor is of the order of the average particle speed.

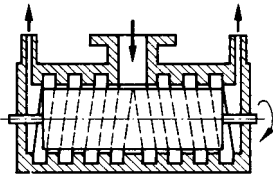


Figure 2-11. Molecular pump

Turbo-molecular pump

A turbo-molecular pump is a molecular pump with a rotor consisting of discs with conveying channels. These discs rotate between corresponding discs of the stator. There are single-flow turbo-molecular pumps (fig. 2-12) and double-flow turbo-molecular pumps (fig. 2-13).

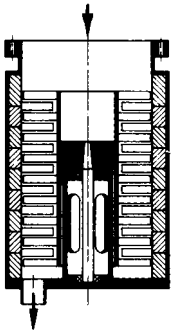


Figure 2-12. Single-flow turbo-molecular pump

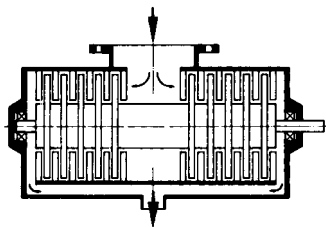


Figure 2-13. Double-flow turbo-molecular pump

Motive fluid vacuum pump, kinetic pump with fluid

A motive fluid vacuum pump is a kinetic vacuum pump in which the pulse is given by a rapidly moving liquid, gaseous or vaporous fluid entrainment.

Motive fluid ejector vacuum pump

A motive fluid ejector vacuum pump is a jet vacuum pump in which the pressure drop of the Venturi effect is utilised and the gas is carried away to the outlet port by a rapidly moved jet. A motive fluid ejector vacuum pump works in the range of viscous flow and the Knudsen flow of the gas to be pumped.

Liquid jet vacuum pump

A liquid jet vacuum pump (fig. 2-14) is a jet ejector vacuum pump with a motive fluid (usually water). Liquid jet vacuum pumps work in the range of viscous flow of the gas to be pumped.

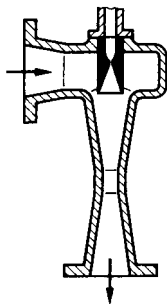


Figure 2-14. Liquid jet vacuum pump

Gas jet vacuum pump

A gas jet vacuum pump (fig. 2-15) is a jet ejector vacuum pump with gas as a motive fluid. Gas jet vacuum pumps work in the range of viscous flow of the gas to be conveyed.

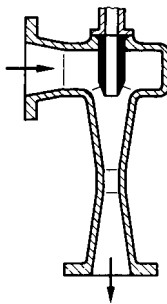


Figure 2-15. Gas jet vacuum pump

Vapour jet vacuum pump, propelling vapour vacuum pump, ejector vacuum pump

A vapour jet vacuum pump (fig. 2-16) is a jet ejector vacuum pump with vapour as motive fluid jet (usually water vapour). Vapour jet vacuum pumps work in the range of viscous flow and the Knudsen flow of the gas to be conveyed.

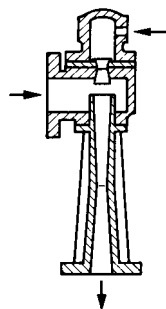


Figure 2-16. Vapor jet vacuum pump

Diffusion pump

A diffusion pump (fig. 2-17) is a motive fluid vacuum pump in which the particle density of the gas to be conveyed in the motive fluid jet is so low that, owing to diffusion the gas particles penetrate into the jet freely expanding to the pre-vacuum and get a pulse in the direction of conveyance. Diffusion pumps work in the range of molecular flow of the gas to be pumped.

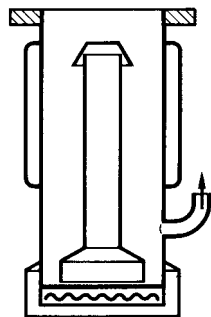


Figure 2-17. Diffusion pump

Self-cleaning diffusion pump

A self-cleaning diffusion pump is a diffusion pump in which lightly volatile contamination of the fluid entrainment does not return to the boiler, but is channelled to the outlet port by means of special measures.

Fractionating diffusion pump

A fractionating diffusion pump is a multiple-stage diffusion pump (fig. 2-18) in which the stage on the lowest pressure side is provided with motive fluid components of higher density and lower vapour pressure, and the highest pressure side is provided with motive fluid components of lower density and higher vapour pressure.

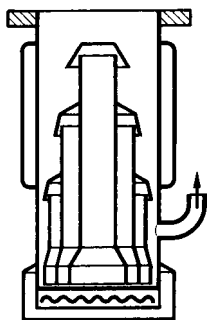


Figure 2-18. Fractionating diffusion pump

Diffusion ejector pump

A diffusion ejector pump is a jet vacuum pump in which one or more stages with diffusion pump characteristics follow one or more stages with jet ejector pump characteristics. Diffusion ejector pumps work in the range of viscous flow, Knudsen flow and molecular flow of the gas to be pumped.

Ion-transfer pump

An ion-transfer pump is a kinetic vacuum pump in which gas particles are being ionised and then moved to the outlet port by means of an electric field, individually or combined with another magnetic field.

Ion-transfer pumps work in the range of molecular flow of the gas to be pumped.

2.2.2

Gas binding vacuum pumps

A gas binding vacuum pump is a vacuum pump in which the gas particles are withheld on inner surfaces through sorption or condensation.

Depending on the predominating gas binding process (sorption or condensation), they are called sorption pumps or condensing pumps.

Adsorption pump

An adsorption pump is a sorption pump in which the gas is withheld mostly by physical sorption forces on material of a large actual surface (e. g. porous substance). Adsorption pumps work in the range of viscous flow and Knudsen flow.

Getter pump

A getter pump is a sorption pump in which the gas is held on a getter. As getter materials, mostly metals or metal alloys are used that occur either as solids or in form of vapour-deposited thin layers.

Sublimation pump (evaporator pump)

A sublimation pump is a getter pump in which the getter material is sublimed (evaporated), mostly intermittently. Sublimation pumps work in the range of molecular flow.

Note: In this context, evaporation and sublimation are regarded as equally matched processes.

Solid getter pump

A solid getter pump is a getter pump in which the gas is sorbed on the surfaces and inside the solid getter material. Solid getter pumps work in the range of viscous flow and Knudsen flow.

Ion getter pump

An ion getter pump is a getter pump in which the gettering process is supported by ionisation of gas particles. Ions are driven onto the getter on the inside surfaces of the pump by means of an electric field (individually or in combination with a magnetic field).

Evaporation ion pump

An evaporation ion pump is an ion pump in which the getter is produced through sublimation or evaporation, continuously or intermittently.

Evaporation ion pumps work in the range of molecular flow.

Sputtering ion pump

A sputtering ion pump is an ion pump in which the getter is continuously produced through cathode sputtering.

Sputtering ion pumps work in the range of molecular flow.

Cryogenic pump

A cryogenic pump is a gas binding vacuum pump in which gases condense on deep-frozen surfaces and/or adsorb on deep-frozen sorption matters (solids or condensate). The condensate and/or adsorbed substance are kept at the temperature at which the pressure of the gaseous phase is less than the desired pressure in the vacuum chamber.

Cryogenic pumps work in the range of molecular flow.

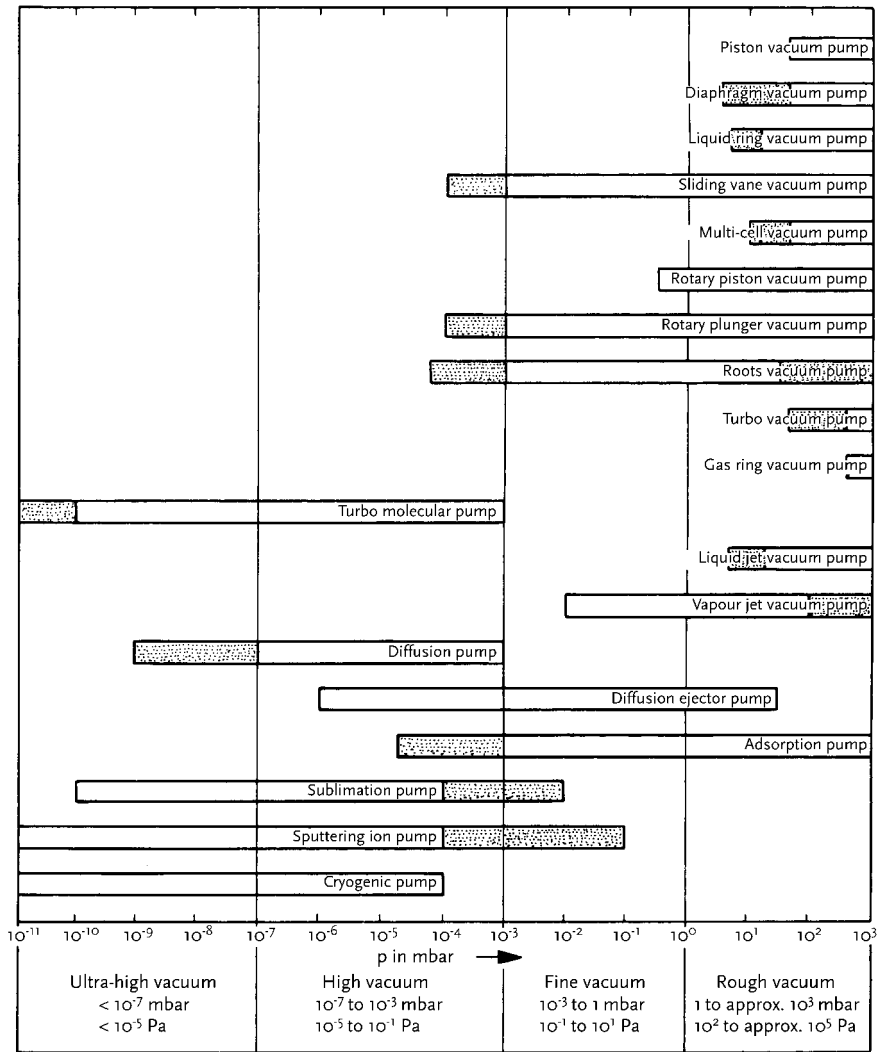
Note: Only vacuum pumps working at temperatures below 120 K are regarded as cryogenic pumps. The selected temperature depends on the type of gas to be pumped.

Condensing pumps working at higher temperatures are called vapour condensers.

2.3

Operating fields of pumps acc. to suction pressure

The classification of vacuum pumps by their functioning principle in fig. 2-1 is completed (acc. to the standard) by a diagram showing the operating fields of vacuum pumps. In this diagram (Fig. 2-19), the operating fields are differentiated by suction pressure. The economic application and the combination of different vacuum pumps should be individually adjusted to the respective operating conditions.




 Working range for special execution or for special operating data

Figure 2-19. Standard operating fields of vacuum pumps (acc. to DIN 28400, part 2 – October 1980)

2.4

Suction pressure and suction capacity of different pump designs

Figure 2-20 and 2-21 give an overview of some operating fields of different vacuum pump designs which are relevant to process technology. Operating fields of pumps of the same design are compared in terms of suction pressure and suction capacity.

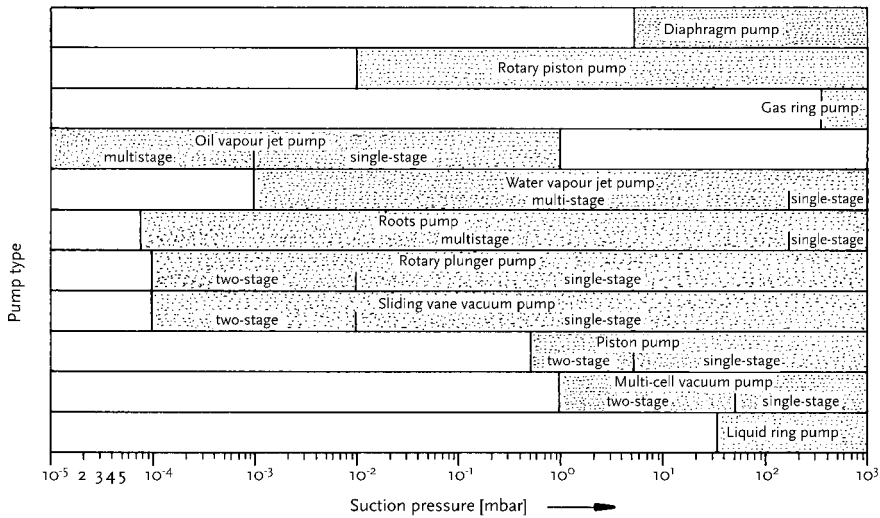


Figure 2-20. Suction pressure ranges of vacuum pumps

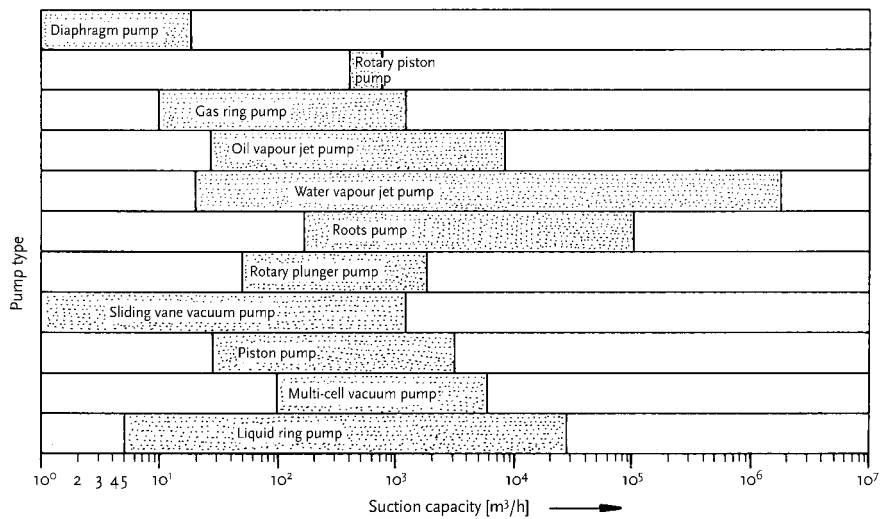


Figure 2-21. Suction capacity ranges of vacuum pumps

2.5

Usual designs and combinations of vacuum pumps

The volume flows of gas-vapour-mixtures arising in technological processes usually range between $1.0 \text{ m}^3/\text{h}$ and $100\,000 \text{ m}^3/\text{h}$ with pressures between 10^3 mbar and about 10^{-2} mbar . In individual cases, pumping can be performed with only one pump or a pump system. However, quite often several pumps of the same or varying working principles are to be series-connected in order to achieve the required compression ratio. Owing to different gas densities and flow types (viscous flow, Knudsen flow or molecular flow) occurring with large pressure ranges, these various pressure ranges require different working principles of pumps. For instance, in the rough vacuum range, single-stage discharge pumps can be applied efficiently, while in the fine- and high vacuum range a combination of discharge pumps and jet pumps is used. The choice of vacuum pumps is to be adjusted mainly to the suction pressure range in which a pump is intended to be operated. In process technology, liquid ring pumps, Roots pumps, rotary vane pumps and jet pumps belong to the most important machines for vacuum processes. The condenser often used in pump combinations is an especially economic type of vacuum pump.

In this chapter, only those pumps and pump combinations are considered which are applied in the range of rough and fine vacuum, i.e. industrial vacuum.

2.5.1

Sliding vane vacuum pump

In pumps of this kind, a rotor eccentrically supported in a cylindrical casing rotates with one or several slides. Owing to the expansion and reduction of the chambers between the gate valves which is caused by the rotary motion, the gas volume is sucked, compressed inside the pump and discharged on the delivery side. Oil lubricated pumps are used in the range of rough and fine vacuum and as pre-vacuum pumps in the high and ultra-high vacuum range. Sliding vane vacuum pumps are available for suction volumes of $1.0 \text{ m}^3/\text{h}$ to approx. $1200 \text{ m}^3/\text{h}$ [2.1] [2.2]. Figure 2-22 depicts a single-stage pump. The two-stage design disposes of two series connected pumps in one casing (fig. 2-23). With the single-stage pump, final pressures of about 10^{-2} mbar are achieved, and with a two-stage pump some 10^{-4} mbar can be achieved. For pressures of about 10^{-5} mbar , it is more convenient to use pump units with diffusion pumps and a sliding vane pump as forepump, or combinations of turbo-molecular pumps and a sliding vane pump as forepump. Sliding vane pumps are also used as pre-vacuum pumps to Roots pumps. A series connection of two sliding vane pumps with an interposed condenser is another customary combination. With the combination of two sliding vane vacuum pumps, the pump which works against atmosphere pressure has a gas ballast valve and an outlet valve. The end vacuum pump has no valve. Ordinarily, a sliding vane pump is provided with a safety valve closing the suction side of the pump in case of power failure or standstill of the pump. In such cases, the pump is vented at the same time, so that it will not have to run against the vacuum after re-starting.

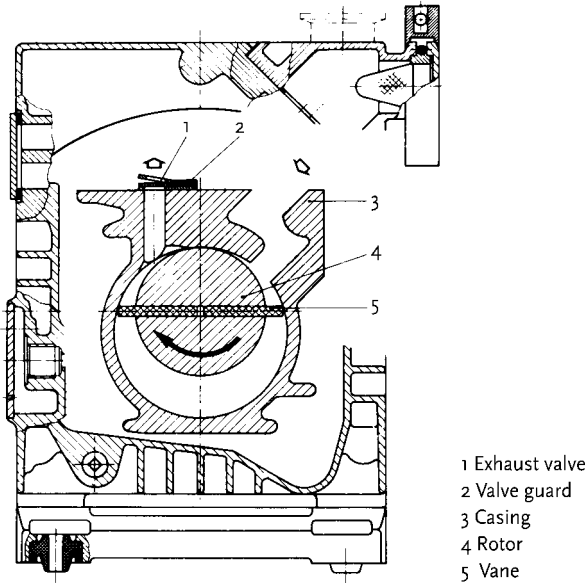


Figure 2-22. Sliding vane vacuum pump, single-stage design with circulation lubrication (acc. to Leybold-Heraeus)

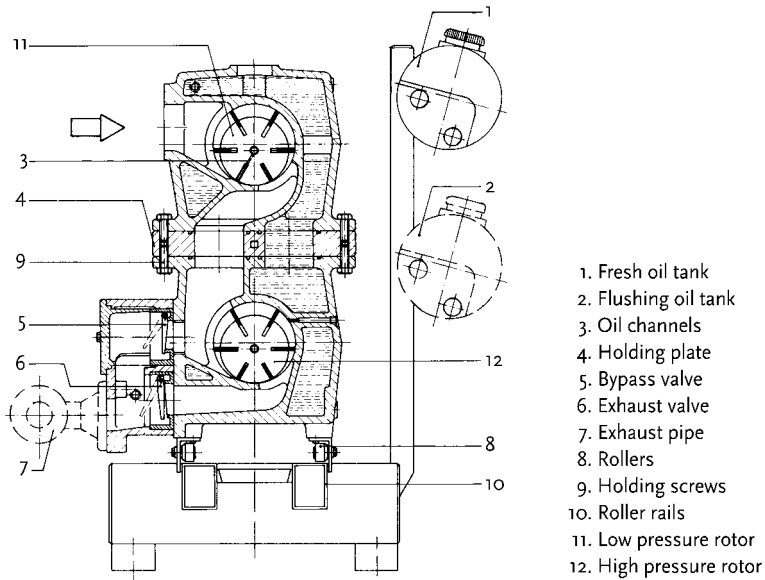


Figure 2-23. Sliding vane vacuum pump, two-stage design (Huckepack), fresh-oil lubrication (acc. to Dr. Ing. K. Busch GmbH)

The effect of oil lubrication may be considered to be the largest disadvantage of sliding vane pumps. Thanks to the installation of filters and automatic safety valves, the return flow of oil vapours from the pump into the vacuum vessel can be avoided to a great extent. Due to their design, sliding vane pumps work through internal compression, i.e. the gas to be pumped is brought to a higher pressure inside the pump. If any vapours are admixed to the gas, they can condense in the pump during the compression process. Thus the resulting condensate can mix with the pump oil and impair its lubricity or evaporate at the suction side and diminish efficiency.

Owing to this, it is necessary to let a certain quantity of gas (so-called Gaede gas ballast) enter the suction chamber through a gas ballast valve with each compression. This gas can be ambient air or, if required by the process, any other gas. Shortly before compression begins, the ballast quantity is exactly measured and let into the chamber. Thus, the partial pressure of non-condensing gases increases compared to the partial pressure of vapours. With this it is achieved that the saturated vapour pressure of the existing vapours is not exceeded and condensation inside the pump is avoided. The gas ballast quantity may amount to a maximum of about 10% of the pump suction capacity. Of course, the functioning of the pump is negatively influenced by the ballast, which means the final vacuum of the pump will decrease. On the other hand, the quantity of added ballast gas is limited and depends on the pressure of the respective kind of pump. This pressure is specified for the pump as water vapour compatibility. The water vapour compatibility is approx. 60 mbar, i. e. the highest pressure at which a discharge pump can suck and discharge pure water vapour under normal ambient conditions. If this limiting value is exceeded, condensers shall be installed in the suction pipe. To achieve higher vacuum, working without gas ballast is attempted. In high and ultrahigh vacuum this is quite possible, as vapours hardly accrue there.

Another possibility to avoid condensation of vapours during the compression process in the pump is to operate the pump at higher temperatures. With this it is achieved that e. g. in the case of compression of water vapour at a temperature of 100 °C the respective saturated vapour pressure is 1013 mbar and therefore, condensation takes place only at the end of compression process at the gas outlet. Then, before condensing, sucked vapours can be emitted into the air together with gas ballast. The use of synthetic oils allows operation of the pumps at temperatures up to approx. 100 °C. In the general sense, the same principle is applicable also to sliding-vane-, rotary-plunger- and trochoidal vacuum pumps.

During operation, these pumps should have a temperature of 60–80 °C, in order to limit water vapour condensation and the creation of an oil emulsion.

Oil lubrication common in sliding vane vacuum pumps serves the heat removal, the lubrication, sealing and filling of detrimental spaces on the pressure side. In this type of machine, the sealing of the clearance between the slide valve and casing is force closed. When pumping aggressive corrosive mixtures of gases and vapours, the lubricating oil may be decomposed or destroyed in these pumps. Then, fresh oil lubrication instead of circulatory lubrication can be applied. In this system, a pump injects the fresh oil into the compression chamber. If condensates arise in the pump, the fresh oil lubrication may be a remedy. The lubrication of bearings occurs

in an oil bath separated from the compression chamber. Recently, also dry running sliding vane pumps are partly used. They allow oil-free gas delivery.

In certain applications, sliding vane pumps are in competition with liquid ring pumps. They are usually used for suction pressures below 60 mbar. However, those pumps are quite sensitive to contamination. The pumping of condensable vapours often requires special precautions to prevent condensation in the pump. An advantage of sliding vane pumps compared to liquid ring vacuum pumps and vapour jet vacuum pumps is their low energy consumption despite their high volumetric efficiency. Sliding vane pumps have an almost constant suction capacity at suction pressures up to approx. 1 mbar and can be operated at high compression ratios. However, machines of this design require certain maintenance costs. Compact versions are designed for air cooling, while larger machines are usually cooled by coolant circuits.

The installation of special filters on the pressure side ensures oil-free exhaust air to a great extent. The application of the sliding vane pumps is advantageous in batch processing or in processes with frequently changing flow media.

In such oil-lubricated pumps, grey-cast iron material proved to be ideal for the pumping of basic and acidic vapours. Oil as a service fluid creates an effective protective coating against corrosion. Sliding vane pumps are also manufactured in corrosion-resistant versions, with stainless steel components or made of special materials.

2.5.2

Multi cell vacuum pump

Like piston-, rotary-vane-, rotary-plunger- and trochoidal pumps, this pump type belongs to the oil-lubricated pumps.

The layout of the multi cell vacuum pump (fig. 2-24a) is similar to the one of a liquid ring pump. In multi cell vacuum pumps, the slide valves are arranged flexibly while the liquid ring vacuum pumps are equipped with fixed vanes. The vacuum chambers that periodically become larger and smaller are separated from each other by movable slides. If suitable self-lubricating materials are used for slide valves, e.g. high-quality carbons or other special materials, the pump can also be operated as a dry runner, i.e. oil-free. Multi cell vacuum pumps work at relatively small compression ratios. In single-stage pumps, the ultimate pressure is approx. 50 mbar, while in two-stage machines it is approx. 1 mbar. Multi cell vacuum pumps are manufactured in sizes with suction capacities of 100 m³/h to 6000 m³/h. [2.3]

Cooling of these machines usually occurs by means of water. As lubrication system, fresh-oil lubrication of the pump is used. Designs with oil-injection are applied as well. In this version, compression heat is not carried off through cooling water but through oil injected into the compression chamber. Continuous oil injection into the compression chamber results in an almost isothermal compression process. In the discharge nozzle of the pump, more than 90% of the cooling oil is already separated from the air by a separator and fed back to the multi cell vacuum pump

via a heat exchanger through a pump with downstream oil filter. In a fine separator on the pressure side, residual oil is sucked out and led back to the pump (fig. 2-24b).

Due to low end temperatures, a good separation level and, consequently, an extremely small quantity of residual oil in the exhaust gas flow are achieved. The pumps are provided with inlet and outlet slots, manufactured and allocated according to the pressure ratio in the pump. Besides the standard version with pressure-independent metering notches, there are also versions with cell valves. Multi cell pumps are used in distillation and evaporation processes, in drying technologies and for the suction of filters. To avoid condensation of sucked vapours in the compression chamber, the machines can be operated at temperatures up to approx. 100 °C.

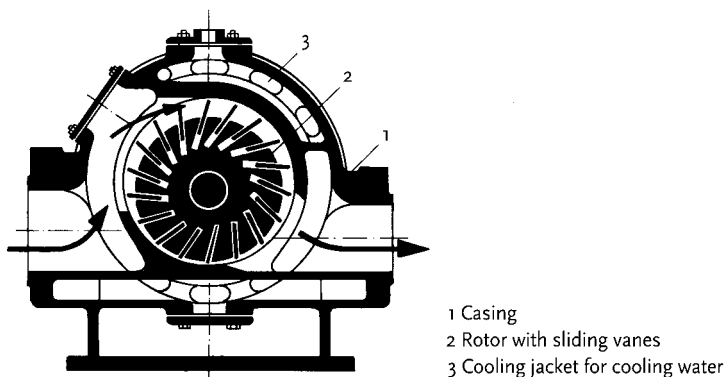


Figure 2-24a. Multi cell vacuum pump

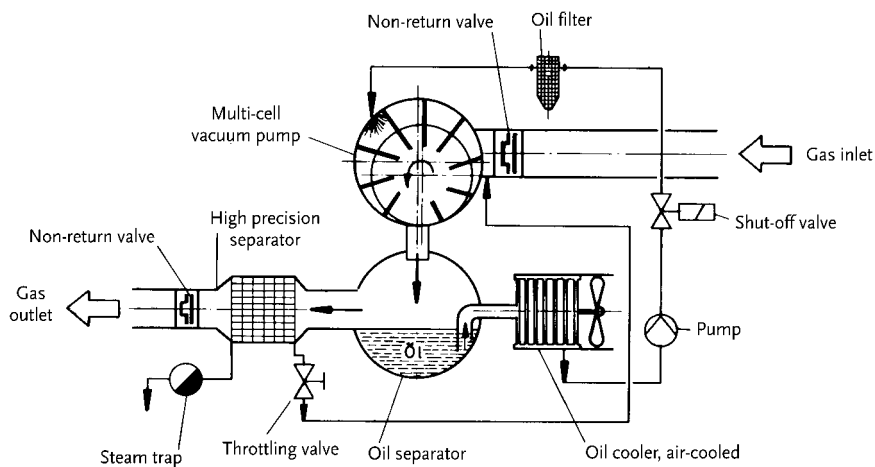


Figure 2-24b. Function scheme of a multi cell vacuum pump with oil injection cooling

2.5.3

Liquid ring vacuum pump

The main feature of a liquid ring vacuum pump is the energy transfer from a rotating impeller via liquid ring to the flow medium. A rotating impeller is arranged eccentrically to the casing. It transfers energy to the liquid ring running concentrically to the casing.

The ring liquid in a liquid ring vacuum pump is usually water or another liquid adapted to the current technological process flow. Vacuum- or compression chambers between impeller vanes are isolated from the casing by the ring liquid (fig. 2-25). The inlet and outlet slots for the gas flow are arranged axially in the end faces of the casing. These pumps are particularly suitable for pumping humid gases or vapours, to which they are sensitive. Therefore, liquid ring vacuum pumps are especially suitable for the application in wet processes e. g. in plants for impregnation, distillation, as well as in drier plants used in the chemical industry, the food industry, the pharmaceutical and the paper industry. Due to intensive heat exchange between the gas pumped and the operating liquid and due to permanent heat evacuation, an almost isothermal compression process develops in these machines. In this respect, the liquid ring vacuum pump differs from almost any other rotating vacuum pump. Unlike oil-lubricated discharge pumps, where condensation in the compression chamber is not desired, vapour condensation in liquid ring vacuum pumps has a positive effect on the suction capacity. The economic advantages of this pump become apparent when instead of dry gases, humid gas and vapour mixtures are pumped. The optimal operating range of such machines is about 20 mbar over vapour pressure of the applied service liquid at ring liquid temperature. If a vapour ejector is installed on the suction side, the optimal intermediate pressure (suction pressure of a liquid ring vacuum pump) ranges from 60 mbar to 80 mbar. Due to isothermal compression, liquid ring vacuum pumps are particularly suitable for the pumping of e.g. explosive and reactive mixtures, as owing to the constant gas temperature, the reactions of such mixtures are reduced. Another advantage is that in process technology these machines, belonging to the group of gas transfer pumps, work as absorption devices in case of the respective selection of gas and ring liquid. Besides delivery and compression of gases, other technological processes, such as cooling, vapour condensation, gas absorption and chemical reactions can run in these pumps. The suction capacity of these machines ranges from 5 m³/h to 25000 m³/h with the main operating range lying between 100 m³/h to 3000 m³/h. With water as operating liquid, the operating range of atmospheric pressure can be down to 33 mbar. If oil is used, final pressures between 10 mbar and 30 mbar are achieved.

As the suction pressure of these pumps directly depends on the vapour pressure of the ring liquid at operation temperature, liquid ring vacuum pumps are often combined with a dry-running air-jet pump or a Roots pump installed upstream on the suction side, in order to achieve smaller suction pressure values. Placing a gas ejector on the suction side before the pump is economical if at the end of an evacuation process for a short time a vacuum is to be achieved that is higher than it would

be physically possible with a liquid ring vacuum pump. The motive fluid for the gas ejector can be e.g. atmospheric air or the process gas from the liquid separator installed on the pressure side. Installing a gas ejector between the liquid ring pump and a Roots pump has the additional advantage that e.g. a Roots pump works against a counter-pressure of only 5 mbar and can develop very high ultimate pressures.

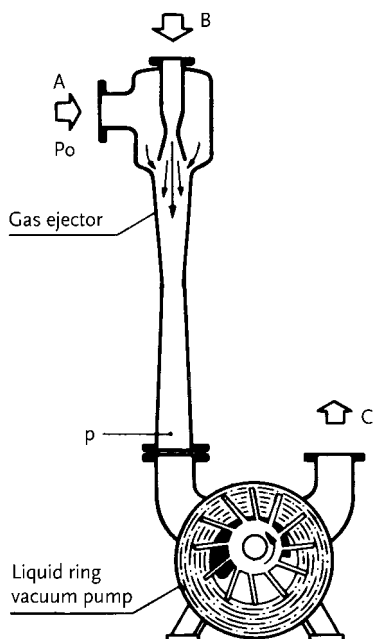


Figure 2-25. Liquid ring vacuum pump with gas ejector

- A suction medium
- B atmospheric air as motive fluid
- p_o suction pressure
- p counter-pressure of air-jet vacuum pump = suction pressure of liquid ring vacuum pump
- C outlet for sucked medium and motive fluid

Due to their high compression ratio, liquid ring vacuum pumps can be used in rough and fine vacuum, favourably in combination with other pump types. So these pumps are profitably operated as complete vacuum pump systems, e.g. together with vapour injectors or gas jet pumps and Roots pumps. In such systems, the liquid ring vacuum pump is installed as forepump, i.e. as pump stage on the atmospheric side. This type of pump is very robust, resistant to contamination, solvents, oil-soluble gases, humid vapours, and conveyed liquids. The maximum gas suction temperature lies at approx. 100 °C. If the suitable ring liquids and sealing materials are used, also higher temperatures are permissible. In these pumps, usually gland seals as well as single and double mechanical seals are employed as shaft sealings. The liquid ring vacuum pump is available in a hermetic version, too (see chapters 3.16 and 3.17). Here, pumps without seals are meant which are equipped with permanent magnet couplings and with canned motors. Machines of this design are used in industry wherever absolutely hermetic circulation is needed regarding both gas and operating liquid.

2.5.4

Rotary plunger vacuum pump

Rotary plunger vacuum pumps (fig. 2-26) achieve a final pressure of 10^{-2} mbar in a single-stage version and approx. 10^{-4} mbar in the two-stage version. Depending on the pump size, the suction capacity ranges from 50 m³/h to approx. 1700 m³/h [2.4]. Cooling occurs by cooling fins of the casing in certain cases even water-cooled series are available. Like sliding vane vacuum pumps, rotary plunger vacuum pumps are equipped with gas ballast valves. The flat slide gate valve of such a pump is designed as suction channel. Like in sliding vane vacuum pumps, the gases to be pumped are emitted through an oil-lubricated exhaust valve. Rotary plunger vacuum pumps are very robust and reliable, as the rotary piston is positively driven, it is not in direct contact with the stator casing and it is sealed up against the casing only by the lubricant film.

Therefore, these pumps are relatively resistant to contamination and stand out due to their long life-cycle. A disadvantage of the rotary plunger pumps is the existence of large masses circulating and moving to and fro. The rotating masses can be equalised, however oscillating masses cannot be fully balanced. Due to limited maximum revolutions, these machines turn out to be much larger compared to sliding vane pumps.

Fields of application of rotary plunger vacuum pumps are vacuum processes in chemistry, metallurgy and electronics, in experimental and production plants.

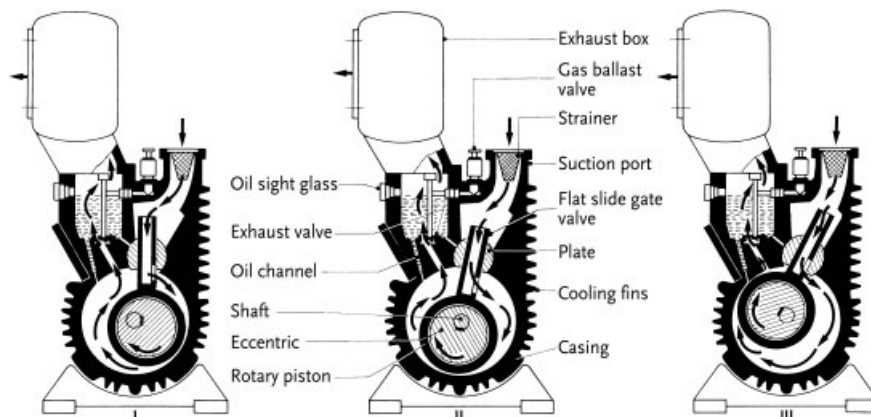
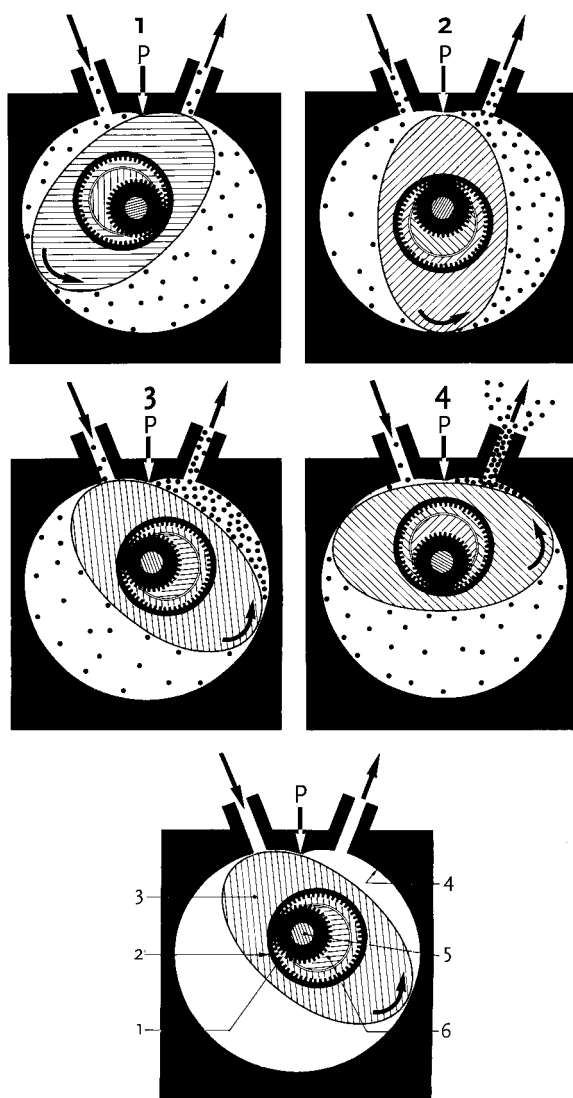


Figure 2-26. Functioning scheme of a rotary plunger vacuum pump (acc. to Leybold-Heraeus)

2.5.5

Trochoidal vacuum pump

Among oil-lubricated and oil-sealed positive displacement vacuum pumps, trochoidal pumps are probably the latest development. This pump has a piston with the shape of an elliptical hypertrochoid. Among vacuum pumps it is also called rotary



Cross-section of a trochoidal pump
 1 gear-wheel connected with drive shaft; 2 gear-wheel fixed on piston;
 3 elliptical piston; 4 inside wall of casing; 5 drive shaft; 6 eccentric

Figure 2-27. Cross-section and functioning scheme of a trochoidal vacuum pump (acc. to Leybold-Heraeus)

piston vacuum pump (fig.2-27). As shown in the functioning scheme of this pump, there are two work cycles per piston revolution. There is no contact between the casing wall and the rotary piston. A sealing between inlet and exhaust port impedes the return flow of the gas to be pumped from the pressure side to the vacuum vessel.

Up to now, these pumps are manufactured only in a single-stage version with a suction capacity between approx. 400 m³/h and 750 m³/h [2.4]. Without gas ballast, the ultimate pressure lies at about 10⁻² mbar, and with gas ballast at approx. 10⁻¹ mbar. Trochoidal pumps work with internal compression, like the otherwise known oil-sealed displacement pumps. In order to keep the temperature constant, these pumps are equipped with cooling jackets (e. g. for water cooling).

The advantage this design shares with the rotary compressors is the possibility to carry out the balancing of rotating masses without problems. Trochoidal pumps have small dimensions and offer continuous gas suction.

2.5.6

Roots pump

The action of the Roots pump is based on the principle of the known Roots blower. In the pump casing, two eight-shaped pistons are contra-rotating without touching each other (fig.2-28). The synchronous running of both pistons occurs through an external timing gear box. The pistons rotate even without touching the casing. The dimensions of the initial diametric clearances arise from manufacturing processes and of course under consideration of an optimal suction performance. The suction capacity of these machines depends directly on the clearances between the suction pistons and on those between the pistons and inside walls of the casing, with the suction performance changing together with the suction pressure.

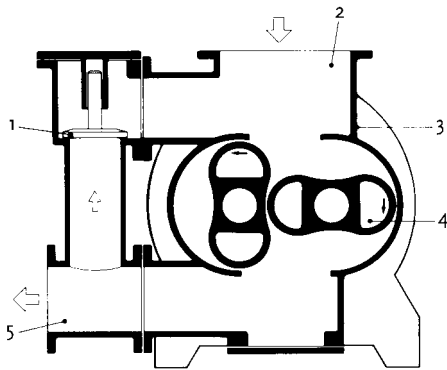


Figure 2-28. Principle presentation of a Roots vacuum pump with an overflow valve (acc. to Balzers)

- 1 overflow valve
- 2 suction side
- 3 casing
- 4 piston
- 5 discharge side

In high vacuum, the suction capacity of these pumps is much higher than in the range of atmospheric pressure and above. The reason is that according to gas laws, line resistance in pipes and clearances increases with reducing pressures. Consequently, also the return flow from the pressure side to the suction side in the pump reduces with falling pressure, and suction capacity increases. It increases until the gas quantity at a higher vacuum becomes smaller compared to the clearance loss, and then the suction volume of the pump drops again. Like the suction volume, also the maximum compression ratio (K_m) of these pumps reacts in relation to the suction pressure. This interdependence is shown in fig. 2-29.

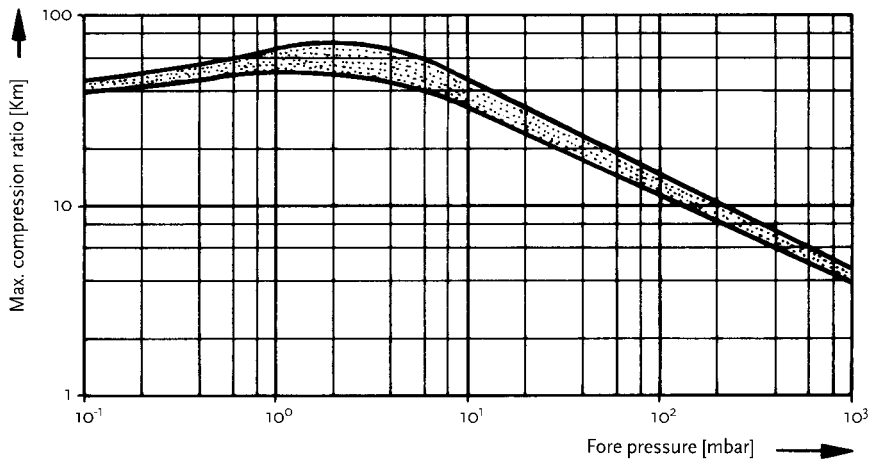


Figure 2-29. Maximum compression ratio K_m of Roots vacuum pumps interdependent on backing pressure (acc. to Balzers)

Therefore, the suction capacity of a Roots pump is always depending on the respective forepump and the suction pressure, i.e. it cannot be determined by a single number. For this purpose, so-called suction capacity curves are used which are applicable for the existing pump combination (fig. 2-30).

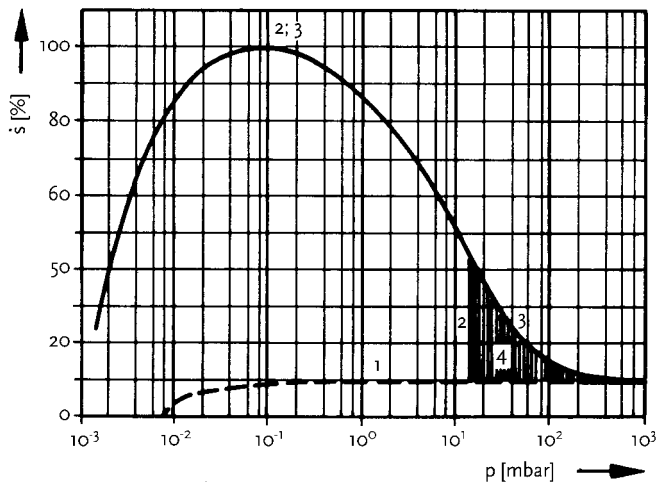


Figure 2-30. Suction capacity curve of a Roots vacuum pump with and without overflow valve in combination with single-stage sliding vane vacuum pump as forepump (acc. to Balzers)

Roots pumps work without internal compression, because during the pumping the volume in the pump does not change. Gas compression takes place only against the pressure on outlet side of the pump, when the gas leaves the pump. In order to

avoid overheating caused by heat generation during compression, especially at high counter-pressures, these pumps are mostly used at lower pressures. Therefore, on the pressure side, Roots pumps are often combined with suitable forepumps compressing against higher pressures, e. g. against the atmosphere. As a rule, pressures as from approx. 50 mbar up to atmosphere pressure are taken over by other displacement pumps, because otherwise the Roots pump, especially the rotors, gets too hot.

When a Roots pump is started, it has a particularly large suction capacity compared to the front-end pre-vacuum pump. Without appropriate measures, this would cause an excess pressure between the two pumps as well as overheating. The pressure difference permissible for continuous operation decreases with the increasing size and revolutions of those pumps. This pressure difference varies between approx. 80 mbar and 20 mbar. A possibility to avoid over-compression and overheating would be a pressure switch control to start the Roots pump only after the fore-pump has evacuated the medium down to the pressure which would be acceptable for the Roots pump as counter-pressure. In practice, there is another elegant solution: between suction and pressure port, a weight-loaded excess pressure valve is installed, as shown in fig. 2-28. The valve opens at a certain pressure difference thus causing the return flow of a part of the delivered gas to the suction side. So the Roots pump is relieved. If a certain lower pressure difference is achieved, the valve closes, and the Roots pump works with full capacity at the desired compression ratio. An alternative to the overload protection is driving the pump by means of a hydrodynamic torque converter, the design of which should be adjusted to the drive data of the Roots pump. However, due to price and economical reasons, this kind of drive is used only for large Roots pumps.

Suction capacity curve (fig. 2-30) shows the typical behaviour of a Roots pump combined with a sliding vane pump as forepump.

Curve 1 is the suction capacity curve of the sliding vane pump (forepump). Curve (2;3) is the common curve of the Roots pump and sliding vane pump. If there is no overflow valve, the suction capacity initially corresponds to curve 1. If the Roots pump is then actuated by a pressure switch, the suction capacity of both pumps changes according to the vertical line (2). If an overflow valve is installed and both pumps are switched against atmospheric pressure from the start, the Roots pump is active right from the beginning, and suction capacity curve 3 is applicable. The hatched surface (4) can be seen as additional suction capacity when starting the pump.

Figure 2-29 shows the dependence of the maximum achievable compression ratio (K_m) on the counter-pressure, against which the Roots pump works. On the abscissa, pre-vacuum of 10^{-1} mbar up to atmospheric pressure is recorded. On the ordinate, the compression ratio (K_m) is registered within the range of 1:1 to 100:1. If compression against atmospheric pressure is intended, a compression ratio (K_m) of 4:1 to 6:1 is achieved. With compression against 2 mbar counter-pressure, values of 50:1 up to 70:1 are reached. From this, it can be clearly concluded that practically Roots pumps can be used only together with suitable forepumps. In combination with other pumps, these machines can be efficiently used at final pressures between approx. 50 mbar and 10^{-3} mbar, i. e. in the transition range between rough and fine vacuum.

Figure 2-31 [2.5] shows the suction capacity curves of a Roots pump with a rated suction capacity of $2000 \text{ m}^3/\text{h}$ and single-stage sliding vane pumps as forepumps of different sizes during operation with or without gas ballast of the forepumps. The maximum suction capacity of the Roots pump with forepump (1) is approx. $1560 \text{ m}^3/\text{h}$ in the best operating range. In the case of pump (2) the suction capacity of which is $90 \text{ m}^3/\text{h}$ larger compared to pump 1, a suction volume of $1800 \text{ m}^3/\text{h}$ is achieved, i.e. an increase of $240 \text{ m}^3/\text{h}$. With forepump 3, approx. $1900 \text{ m}^3/\text{h}$ are being sucked, thus an increase of only $100 \text{ m}^3/\text{h}$ is reached, although the suction capacity of this forepump is $180 \text{ m}^3/\text{h}$ larger than that of forepump (2). It is apparent that the suction quantities resulting from the application of forepumps with different dimensions in the range of small pre-vacuum pressure do not show an equal suction capacity growth as e. g. at atmospheric pressure. On the other hand, in operation modes where faster and more frequent evacuation in high pressure ranges is important, the application of larger forepumps is recommended. The selection of the most suitable pump combination is to be adjusted to the existing operation conditions.

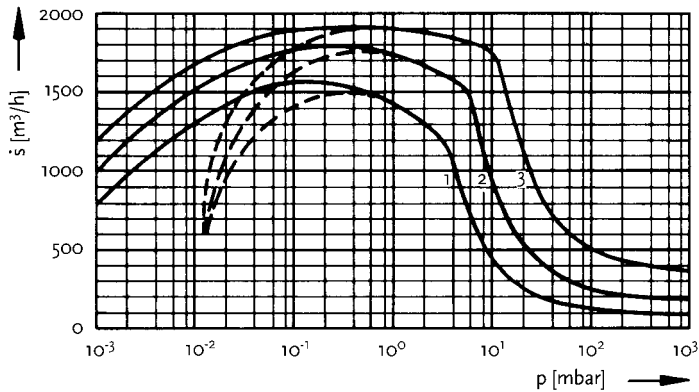


Figure 2-31. Suction capacity curves of a Roots vacuum pump combined with single-stage sliding vane forepumps of different sizes (acc. to Balzers)

- 1 sliding vane vacuum pump, $90 \text{ m}^3/\text{h}$, forepump
- 2 sliding vane vacuum pump, $180 \text{ m}^3/\text{h}$, forepump
- 3 sliding vane vacuum pump, $360 \text{ m}^3/\text{h}$, forepump
- sliding vane pump operation without gas ballast
- sliding vane pump operation with gas ballast

Furthermore, for this pump combination the suction capacity curves (fig. 2-31) show that the operation with gas ballast causes a reduction in final pressure and suction capacity compared to the operation mode without gas ballast. Moreover, it is clear that these three pump combinations dispose of the same final pressure at which the suction capacity has almost reached its minimum. This is also applicable to the 3 combinations in the operating mode without gas ballast which is not apparent on the chart.

In addition, it can be concluded that the suction capacity curve of a pump combination is greatly influenced by the choice of the respective pre-vacuum pump and highest vacuum achievable by it. As seen, the final vacuum of sliding vane pumps operated with gas ballast is poorer than without gas ballast.

In the chemical industry, Roots pumps have proved their worth also in combination with liquid ring vacuum pumps as forepumps. In high-vacuum technologies, liquid ring vacuum pumps should not be used as forepumps due to their rather poor final vacuum.

An economical forepump to a Roots pump is the so-called gas ballast pump. In cases in which the lubricating quality of the oil in the forepumps is impaired or the oil is decomposed due to process gases, liquid ring vacuum pumps are used as well. If a Roots pump is employed to discharge vapours, attention must be paid to the vapours not condensing in the pump. Then condensers are to be installed before or between the Roots pumps. In the range of rough and fine vacuum, usually, pump systems consisting of Roots pumps and oil-sealed vacuum pumps or liquid ring vacuum pumps are used.

Up to certain dimensions, also two-stage Roots pumps are manufactured. The pump design corresponds to the one of single-stage pumps, however two pump stages are housed in one casing.

With pump systems consisting of a two-stage oil-sealed vacuum pump and a two-stage Roots pump, pressures below 10^{-3} mbar, i.e. in the beginning of high vacuum range, can be achieved.

If a fine vacuum adsorption trap restraining possible vapours from the forepump is installed between the forepump and Roots pump, a pressure of approx. 10^{-5} mbar can be kept up. The adsorbent being saturated in time in the adsorption trap can be regenerated again by heating.

In case of oil-free compression of process gases, multiple-stage Roots pumps with intermediate condensers are often used, with the forepump being equipped with gas circulation cooler.

In Roots pumps with gas circulation cooling (fig.2-32), compression heat generated in the range of high pressure is evacuated by the cooler. This innovation, tested during the last years, enables the use of Roots pumps also in the pressure range up to atmospheric pressure, i.e. the operation of several Roots pumps installed in series. With this design, Roots pumps, up to now mostly used within the range of fine vacuum at suction capacities up to 10^{-4} mbar can be also used in rough vacuum.

The pump is designed in a way that the pumped gas is cooled on the pump outlet in a heat exchanger located there and is partially led back into to the suction chamber which is closed due to a certain position of pistons, between pistons and casing wall. The gas entering at the suction port and already compressed is then mixed with the cooled gas flow and transferred to the outlet port. Single-stage pumps of this design compress up to atmospheric pressure at suction pressures between approx. 150 mbar and 200 mbar without thermal overload. In case of series connection of two Roots pumps with gas circulation cooling, a suction pressure of 20 mbar is achieved. For higher vacuums, Roots pumps of normal design are installed on the suction side. In these cases, gas circulation Roots pumps are the forepumps to normal Roots pumps.

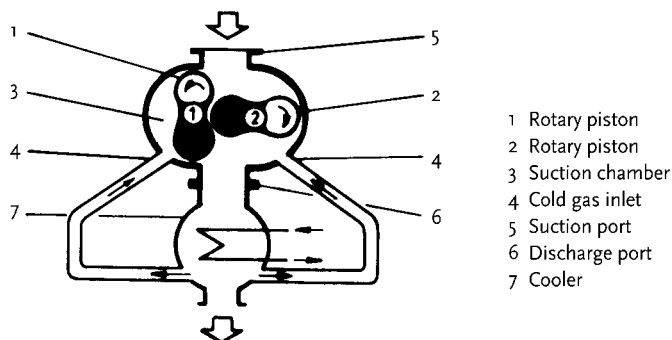


Figure 2-32. Functioning scheme of a single-stage Roots pump with gas circulation cooling (acc. to Leybold Heraeus)

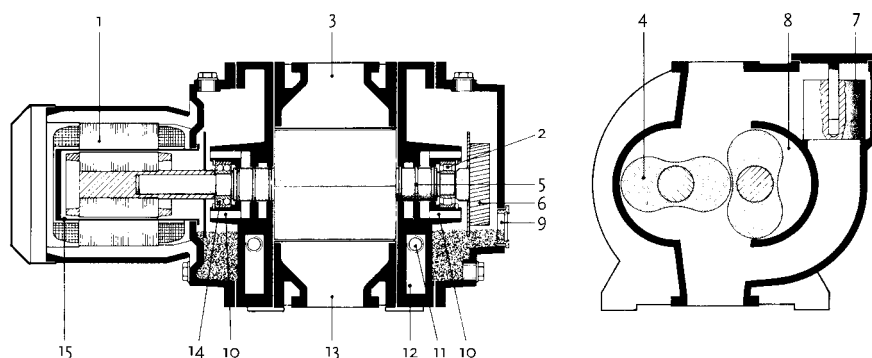


Figure 2-33. Roots vacuum pump with canned motor (acc. to Balzers)
1 canned motor, 2 movable bearing, 3 suction port, 4 rotary piston
5 labyrinth packing, 6 gear, 7 overflow valve, 8 suction chamber
9 oil-level indicator, 10 oil return flow, 11 barrier gas port
12 heating chamber, 13 pressure port, 14 fixed bearing, 15 can

These pumps are of particular importance in cases where e. g. dry and expensive gases are to be sucked from high vacuum and then have to be recovered. An advantage of Roots vacuum pumps is a clean and dry suction chamber, which is free from oil as lubrication is not required.

Roots pumps are usually designed for suction capacities between $150 \text{ m}^3/\text{h}$ and approx. $50000 \text{ m}^3/\text{h}$. The largest pumps of this type manufactured up to now have a suction capacity of $100000 \text{ m}^3/\text{h}$. However, in the range of high and ultra-high vacuum, such capacities are not needed. Roots pumps get by on small dimensions owing to their high specific suction capacity, as due to their functioning principle each piston emits the suction volume twice per revolution. As the pistons work without contact and the mass distribution is symmetrical, these machines can be operated at relatively high speed. Usually, they are driven by three-phase A.C. motors, V-belt drives or gears. The speed range lies between 1500 rpm and 3000 rpm. The drive shaft is mostly sealed with radial sealing rings against atmosphere.

In case of discharging radioactive, dangerous or toxic gases where very low leakage is required, drives in the form of canned motors are possible (fig. 2-33) and dynamic sealing is no longer necessary. These machines are suitable for maintaining high vacuums in apparatus without maintenance for a long time. In these seal-less vacuum pumps, the leakage rate lies at about 10^{-3} [(mbar · l)/s] [2.12]. Pumps of this design also meet the requirements of environmental protection.

A ceramic Roots pump for pumping high-corrosive gases and vapours is shown in the figures 2-34a and 2-34b. In this machine, all components in contact with the

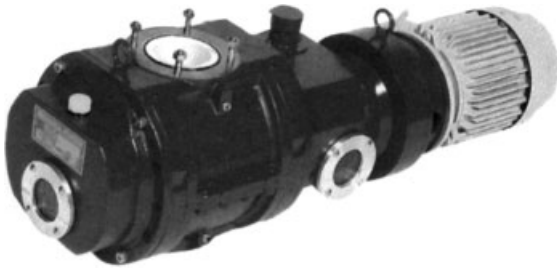


Figure 2-34a. Ceramic Roots vacuum pump with stable steel reinforced components and flanged driving motor (acc. to NGK)

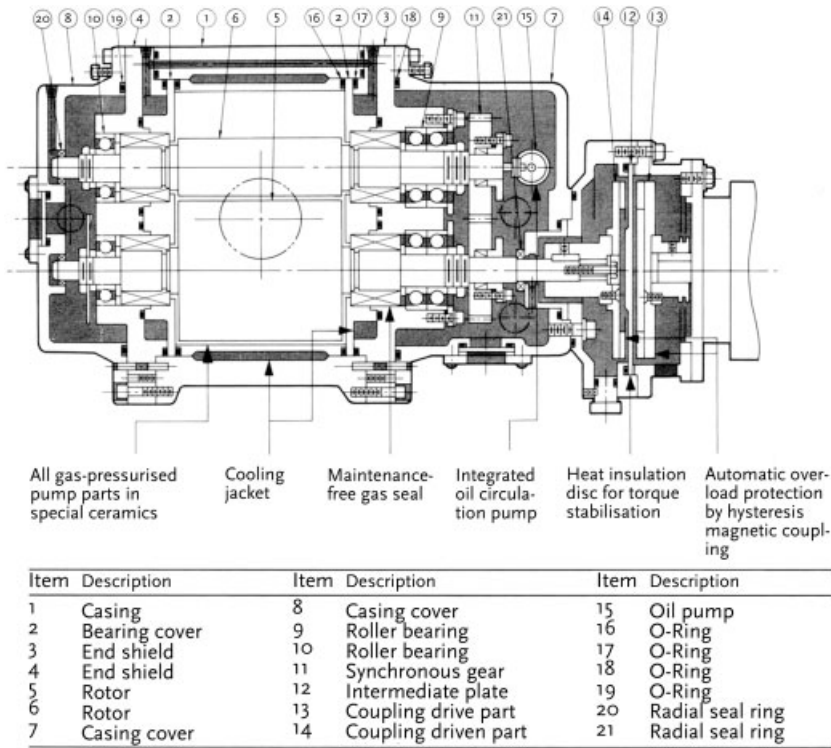


Figure 2-34b. Cross section of the ceramic Roots vacuum pump with gas-lubricated mechanical seals and hysteresis magnetic coupling (acc. to NGK)

process gas are made of stable ceramic (partially steel-reinforced). Equipping such apparatus with contact-free gas-lubricated mechanical seals, barrier gas, cooling and lubricating systems, as well as a hysteresis magnetic coupling ensure high availability and low maintenance costs.

2.5.7

Jet pump

Compression in fluid entrainment pumps and vacuum pumps such as gas jet and vapour jet pumps results from the kinetic energy of the entrained fluid used. Here, motive fluid and gas to be pumped are mixed, leading to a modified total quantity. Depending on the motive fluid, pumps are classified as liquid jet pumps, vapour jet pumps and gas jet pumps.

Such displacement vacuum pumps, as e. g. sliding vane pumps, Roots pumps and piston pumps get their energy from moving mechanical components. Here, motive fluids are not used, and the conveyed mixture changes neither in its composition nor in its quantity during the compression process. The liquid ring vacuum pump belonging to the displacement pumps, however, behaves in another way. Owing to continuous heat evacuation through the operating liquid, and the thus arising isothermal compression, condensation of existing vapours can occur in the pump during the discharge process.

Liquid jet vacuum pumps

The design of a jet pump is simple as they do not dispose of moving components. In a flow channel, the pressure of the in-flowing motive fluid is at first reduced by modifying the respective cross-sections and is then raised again to a higher value. With an increasing contraction (comp. fig. 2-35), jet speed also increases and the static pressure is reduced from p_1 to p_2 . Then in the divergent area, speed decreases again and pressure increases to p_3 . The suction inlet of a jet pump is situated on the spot of the lowest pressure (fig. 2-36), i.e. where jet speed equals the suction pressure of the ejector. In practice, the Venturi effect is used for non-compressible media in jet pumps. After the inflow of the medium to be pumped with pressure p_s , it gets into contact and mixes with the jet flow coming out of a driving nozzle. The common speed c_2 is reduced in a down-stream diffuser and gas flow escapes with speed c_3 and pressure p_3 . Then pressure of the sucked medium increases from p_s to counter pressure p_3 . In industry liquid jet pumps are used according to this principle.

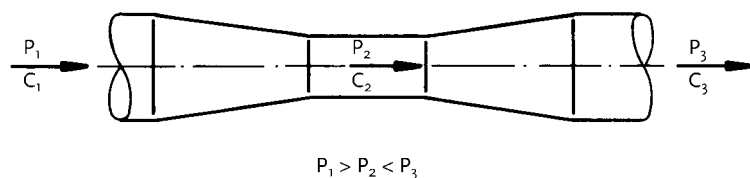


Figure 2-35. Venturi pipe [2.6]

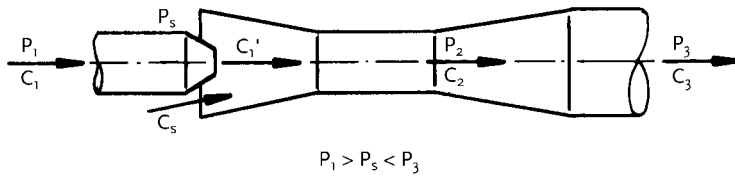


Figure 2-36. Jet pump for fluids with small changes of volume [2.6]

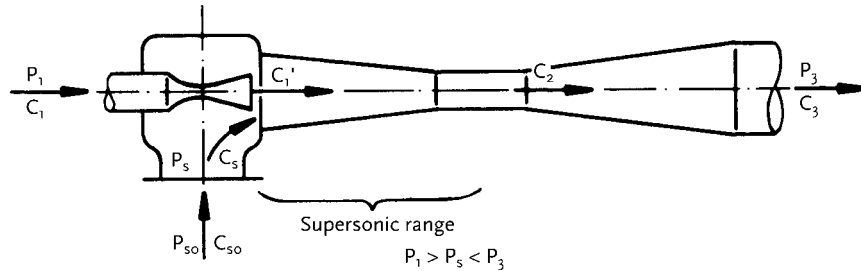


Figure 2-37. Jet pump for compressible media, e.g. gas jet – and vapour jet vacuum pumps [2.6]

Here, the sucked medium is compressed in an isothermal process. Water jet pumps are mostly used when low purchase costs are more important than low operating costs. With water being used as operating liquid, suction pressures of approx. 35 mbar can be achieved, and in the case of ethylene glycol as pump fluid, suction pressures lie at approx. 2 mbar. The final suction pressure of a liquid jet vacuum pump corresponds to the saturation pressure of the motive fluid at the prevailing liquid temperature. Concerning the execution of technological processes, liquid jet pumps offer possibilities similar to those of liquid ring vacuum pumps. As here again, an operating liquid is needed apart from the mere compression process even condensation and absorption processes are basically realisable. Here, the operating liquid – the motive fluid – has to be fed with relatively higher pressure. Liquid jet vacuum pumps are convenient in cases with the medium to be sucked containing a relatively high proportion of condensable particles.

Vapour jet vacuum pumps

In vapour jet pumps as well as in gas jet vacuum pumps, the flowing media, the motive vapour or motive gas and the sucked vapour or gas are compressible. Different pressure zones existing in the flow channel from the inlet to the outlet of the jet cause these media to be subject to large volume changes. In the driving nozzle, an overcritical expansion ratio occurs. With a motive vapour pressure of 10 bar and a suction pressure of 10^{-2} mbar, the pressure ratio is $p_1/p_s = 1 \times 10^6$. According to the supersonic flow, the nozzle is designed as Laval nozzle with expanding cross-section, (fig. 2-37). The downstream converging element of the diffuser has a decelerating effect on the supersonic flow. In this flow segment, a considerable pressure increase takes place. In a so-called diffuser neck, the supersonic flow is converted

into infrasonic flow by a compression shock with rapidly rising pressure (fig. 2-38). In the following expanding part of the diffuser infrasonic flow prevails. Here, the flow speed is decelerated and the pressure increases up to p_3 . Similar to liquid jet vacuum pumps, also in vapour jet pumps or gas jet vacuum pumps, suction flow is let in at the point of the lowest static pressure and the highest flow speed.

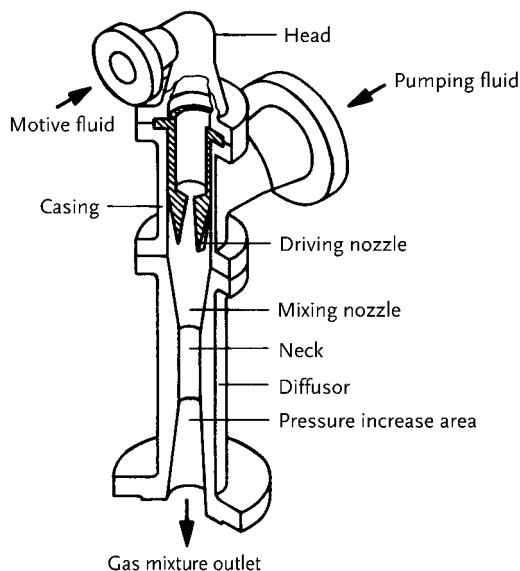


Figure 2-38. Third section of a vapour jet pump

Water vapour is the cheapest motive fluid available in almost all industrial enterprises. High vapour pressure allows the application of high motive vapour pressures and the compression of gas mixtures from smallest suction pressures to the atmospheric pressure with water vapour jet pumps, without changing the pumping principle. Water vapour jet vacuum pumps are designed for up to 6 stages, managing pressure differences between 10^{-3} mbar and 1013 mbar. [2.6]. Dry saturated, slightly overheated water vapour is also very often used as motive fluid. In the case of extremely overheated vapour, a front-end attemperator is usually installed, and in the case of wet vapour, a front-end water separator is installed before the vapour jet pump. The higher the motive vapour pressure, the lower the vapour consumption. The installation of intermediate condensers enables the condensation of vapours arising from the suction flow and the following pump stages are relieved. If only expensive motive vapour is available or only small suction flows exist, it is economically reasonable to install a water ring vacuum pump as atmospheric stage, thus saving one or more vapour jet stages. Here, the liquid ring vacuum pump can take over water evacuation of the intermediate condenser. Owing to the combination of vapour jet vacuum pumps and water ring vacuum pumps lower total operating costs are achieved. Single-stage vapour jet vacuum pumps are used for the pressure range between approx. 150 mbar and 1013 mbar. For suction pressures below 100 mbar

up to 150 mbar, multiple-stage versions are used [2.7]. As liquid ring vacuum pumps with water as operating liquid are efficient in the range between 60 mbar and 1013 mbar, in this pressure range they are preferably combined with vapour jet pumps.

In the case of a multiple-stage arrangement of vapour jet vacuum pumps, vapour consumption per stage is reduced while efficiency increases. In fig. 2-39, a four-stage vapour jet vacuum pump with three mixing condensers is shown. Surface condensers are used, if the condensate must not get into the cooling water. For the last stage, a liquid jet pump can be used as well.

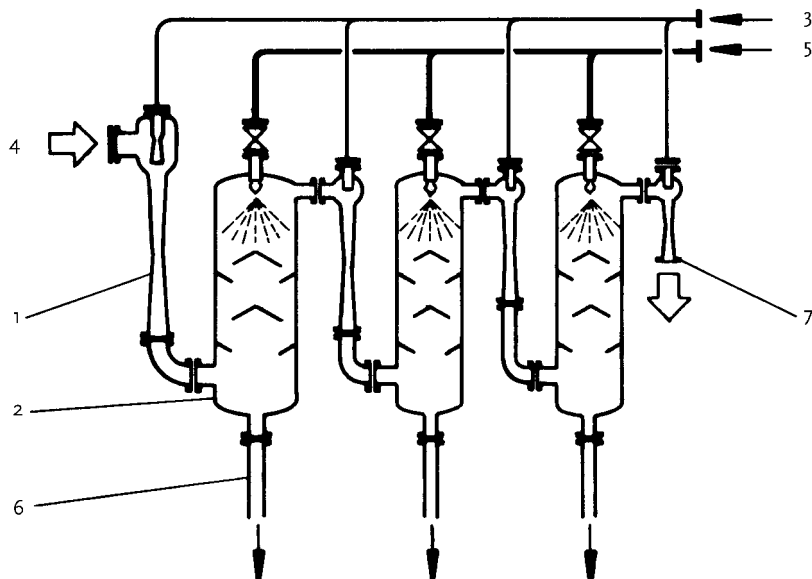


Figure 2-39. Four-stage vapour jet vacuum pump with mixing condensers
 1 vapour jet vacuum pump, 2 mixing condenser, 3 jet vapour
 4 suction port, 5 cooling water for condensers, 6 barometric downpipe
 7 pressure port

Vapour jet vacuum pumps are available for the largest suction flows up to $1,000,000 \text{ m}^3/\text{h}$. These are jet pumps with a rated width of the suction port of 3400 mm and a length of 24.25 m [2.8]. For the degassing of steel, such pumps have been manufactured for suction flows of $1,600,000 \text{ m}^3/\text{h}$ at a suction pressure of 0.5 mbar [2.9]. The dimensions of these pumps have no upper limit. Jet pumps do not have any moving parts, and can therefore be made of all materials required in process technology, e.g. cast iron, steel, stainless steel, porcelain, graphite, rubber or plastics. Figures 2-40a and 2-40b show a gas jet vacuum pump with all components contacting the gas-vapour mixture being made of ceramics. Due to the simple design, they are safe in operation also under most difficult conditions. Compared to the operating costs, investment and maintenance costs can be considered low.

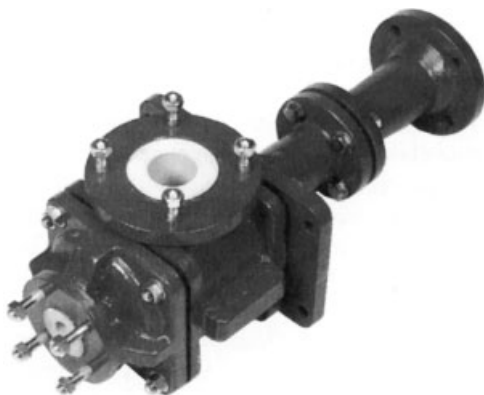
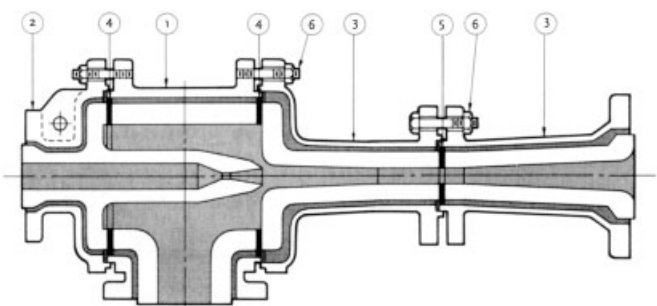


Figure 2-40a. Ceramic gas jet pump with steel reinforced components (acc. to NGK)



Item	Description	Item	Description
1	Suction casing	4	Sealing
2	Motive fluid connection	5	Sealing
3	Diffusor	6	Hexagon nut and hexagon screw

Figure 2-40b. Cross section of a ceramic gas jet pump (acc. to NGK)

Vapour jet vacuum pumps with motive fluids different from water vapour

Besides water vapour, recently vapours of other fluids are used for driving vapour jet vacuum pumps. There are jet pumps running with high-boiling oil or mercury. Mercury-vapour jet vacuum pumps are usually employed at pressures between 10^{-2} and 10^{-1} mbar. Their main fields of application are laboratories. Oil vapour jet pumps are operated with vapours of thermally stable and chemically resistant oils. The vapour generator is mostly part of the oil vapour jet vacuum pump. They are used in degassing and drying processes, as well as in distillation and impregnation plants. The suction pressure range is between approx. 1 mbar and 10^{-3} mbar. To achieve smaller suction pressures, several oil-vapour jet pumps can be installed in series. In combinations with oil diffusion pumps, the suction pressure can reach the high vac-

uum range. Oil-vapour jet pumps are made for suction rates between 25 m³/h and approx. 8000 m³/h [2.6]. These pumps are often used together with oil-sealed sliding vane pumps or rotary plunger pumps as forepumps. According to the process, even liquids as toluen, benzene, chlorobenzene, ethylene-glycol, or acetone are used for the generation of motive vapours in process engineering. As a rule, special vapour generators are needed for the respective motive medium. If the motive fluid and the existing vaporious material to be sucked are identical, the conditions are the same as for the suction of water vapour by means of a vapour jet vacuum pump, i.e. the sucked vapour can be condensed in intermediate condensers with the similar motive vapour. If required a liquid ring vacuum pump or liquid jet vacuum pump can also be used here as forepump for compression to atmospheric pressure.

Gas jet vacuum pumps

The jet pump (fig. 2-38) is an addition to a liquid ring vacuum pump, and therefore often combined with the latter (fig. 2-25). Naturally, the application of gas jet pumps in combination with liquid ring vacuum pumps requires the suction capacity of the liquid ring vacuum pump to be sufficient to suck even the motive fluid of the gas ejector. Gas ejectors work without external energy by using the adiabatic temperature gradient existing between the atmosphere and the outlet nozzle of the diffuser. In the nozzle, the motive medium is accelerated to the speed corresponding to the suction pressure of the ejector with the pumping gas being sucked into the mixing nozzle. In the diffusor, the speed of the mixture is converted into suction pressure corresponding to the pressure of the liquid ring vacuum pump. Referring to water as ring liquid at about 15 °C, gas ejectors are installed front-end to the liquid ring vacuum pumps, if suction pressures between 33 mbar and 5 mbar are to be achieved. If ring liquids other than water are used, the achievable suction pressures will have different values, according to the boiling characteristics of the ring liquids. The installation of gas ejectors is simple, installation costs are low, and the costs for the gas ejector itself are significantly lower than for other vacuum pumps. The power consumption of gas or air ejectors is higher compared with displacement vacuum pumps and mechanical kinetic vacuum pumps. The application of these fluid entrainment vacuum pumps is justifiable even from an economical point of view and especially for small suction capacities as in each individual case aspects such as operating time, energy costs and frequency of starting have to be taken into consideration.

If low energy costs are very important, it is better to install a vapour jet pump with intermediate condenser instead of an air jet pump on the suction side of a liquid ring vacuum pump. In the intermediate condenser, the main part of the motive medium (e. g. water vapour) and possibly even parts of the process gas are already condensed, with the liquid ring vacuum pump being relieved.

Any gas or even atmospheric air may be used as motive gas for the gas pump. Possibly, an adequate inert gas can be used as well, in order to avoid the generation of reactive or explosive gas mixtures. Gas compressed to atmospheric pressure that is taken from the liquid separator of the liquid ring vacuum pump and fed to the gas jet pump can also be used.

2.6

Vacuum pump units and their control

The selection of an optimal combination of several vacuum pumps to form a pump unit presupposes exact knowledge of different pressures and suction volumes in individual operation modes. Further process-related requirements shall also be taken into consideration. For instance, the suitability of possibly needed lubricants, coolants, operating liquids and motive fluids should be examined. For planning, precise data about the type of gas or gas mixtures to be sucked, their molar values, temperatures, and volumetric or mass composition are required. Specific product features (solubility of gases and liquids, the material's tendency to polymerisation, sublimation and explosion) and their behaviour during the compression process are to be considered. When handling or conveying toxic or otherwise dangerous media, special regulations for environmental protection and safety shall be observed.

Transition points between operating ranges of individual pumps are determined according to the intended design of the pumps. According to the layout of the different pumps in a pump unit, the pump installed on the side of the lowest pressure is called final vacuum pump while the pump on the side of the highest pressure is called forepump. All pumps in between are called intermediate pumps.

2.6.1

The three phases of evacuation

Evacuation distinguishes three phases that will partially repeat, i.e. the volume or coarse evacuation, surface degassing and the maintenance of the operating vacuum. For determining the dimensions of the final vacuum pump, parameters as leakages, gas sources and gas sinks are decisive [2.10]. For the layout of final vacuum pumps, gas sources especially at low pressures have to be considered for the worst cases. Therefore, the vacuum process is defined by the final vacuum pump, i.e. with the final vacuum pump being too small the whole process is endangered, while a final vacuum pump being too large means time savings for the process flow. The gas molecules dissolved or absorbed in the solid material of the walls are transported to the surface through diffusion within the period of time in which desorption takes place. Evacuation at increased temperature is shown in the pressure-time diagram (fig. 2-41). The surface under the curve depicts the molecules emitted from the surface to the vacuum chamber [2.10]. The intense gas discharge at the beginning diminishes in time.

Therefore, for the desorption of molecules enclosed in the surfaces, vacuum pumps and pump units intended for operating at higher vacuums are heated in order to reduce evacuation time.

The determination of the time for pre-evacuation has to be adjusted to the pump design. For instance, for pre-evacuation with oil-sealed pumps or with liquid jet pumps, this time should not be extended too much in order to limit the entrance of oil or liquids to the vacuum chamber. Two or more pumps can be operated simultaneously and parallel as well, to reduce evacuation times. Therefore, the cooperation

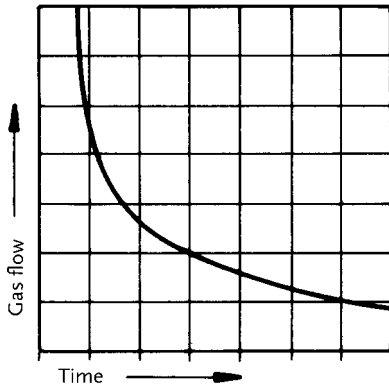


Figure 2-41. Gas desorption of a surface as a function of time during evacuation

of the pumps in one pump unit has to be determined from the beginning of the evacuation and adjusted to the individual pumps.

2.6.2

Vacuum pumps in series

Due to the fact that a single pump is not able to generate a very small suction pressure beginning from the atmospheric pressure, and, depending on the gas quantity, to maintain it for a longer time, in practice certain combinations of pumps have established for certain cases of application. The suction capacity of the forepump shall at least be equal to or slightly higher than the suction capacity of the following pump working at higher vacuum. Owing to the declining suction capacity curve at small suction pressures, the operating pressure of a vacuum pump must not be too close to the final pressure, the lowest achievable pressure of the pump. For safety reasons, an additional suction capacity should always be provided. With two vacuum pumps being series connected it is absolutely possible that these two pumps do not dispose of the same suction capacity. The reasons for this are the possibly different compression ratios. The suction capacity of the last pump which is the nearest to the vacuum vessel is mainly determined by the final pressure required, the inflows and the evacuation time. The suction capacity of the previous pump depends on the continuity of the defined gas flow and the minimal value of the pre-vacuum required for the last pump, i. e. the maximum value of the pre-vacuum pressure at which the pump still possesses the required suction capacity.

Acc. to [2.11], the following equation results:

$$\dot{S}_v \geq \frac{(\dot{S}_h \cdot p_h)_{\max}}{p_{pl}} \quad (2-1)$$

\dot{S}_v suction capacity of the forepump

p_{pl} upper limit value of pre-vacuum of the high vacuum pump

\dot{S}_h suction capacity of the high vacuum pump

p_h pressure on the suction side of the high vacuum pump

Consequently, an excess pressure build-up between the individual pumps is to be avoided. Where this is not the case, e.g. when Roots pumps are used, such operating conditions are avoided by means of special precautionary measures such as the installation of excess pressure- or bypass valves. A further possibility is to start the pump on the high-vacuum side only after the vacuum generated by the pre-vacuum pump has reached a value at which the suction capacity equals or exceeds the suction capacity of the pump on the high-vacuum side.

In a multiple-stage pump unit, coarse evacuation is usually taken on by the pre-vacuum pump. Intermediate and final vacuum pumps can already be installed upstream in series connection. Then, without supporting the suction process, the sucked gas can flow through. In this case, the intermediate pump and the final vacuum pump are energized as soon as their respective compression pressures are reached. Thus, this design does not require valves. Another possibility would be energizing a series connection of the vacuum pumps with valves only when the starting pressure of the subsequent pumps has been reached. In fig. 2-42, the structure of a vacuum pump unit with a diffusion pump according to [2.10] is shown.

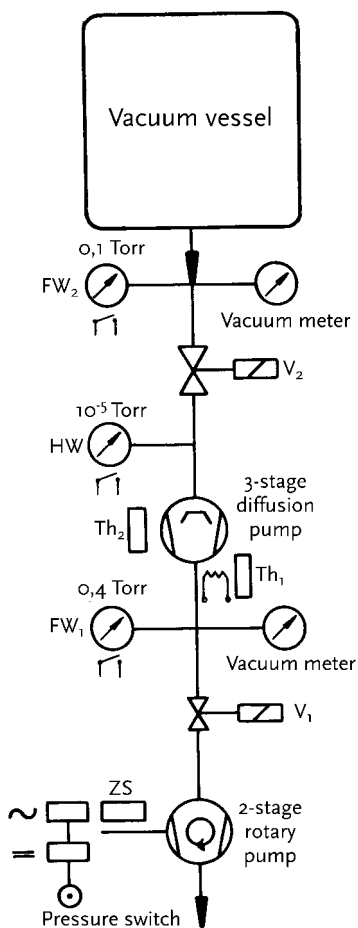


Figure 2-42. Scheme of a pump unit with control unit

The final vacuum pump (fig. 2-42) is directly series connected. Coarse evacuation occurs through the cold diffusion pump. The space up to the high-vacuum valve V_2 will be pre-evacuated. The high vacuum valve V_2 opens when a pressure of 0.4 Torr is reached at FW_1 , and the pressure at FW_2 is not below 0.1 Torr. As it is common practice in vacuum technology, the flow direction is from top to bottom. Following the law of gravity, possibly sucked particles can leave the pump system again at the discharge nozzle.

The moment to switch over to vacuum pumps with higher suction capacities at small suction pressures depends on the permissible compression pressure of the pump to be started. Moreover, at the time of switching over, the forepump already in operation should not have lost too much of its suction capacity [2.10]. If necessary, an intermediate pump shall be provided which is able to manage accruing gas quantities better than the final vacuum pump. In fig. 2-43, some pump combinations

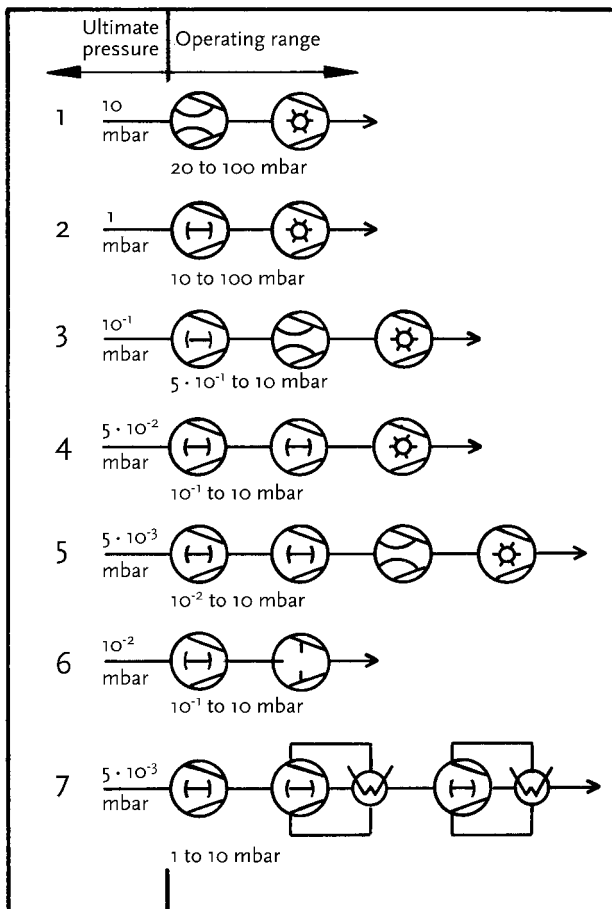


Figure 2-43. Pump combinations with operating pressure ranges and achievable final pressures

used in practice are shown, as well as their operating ranges and the practically achievable final pressure acc. to [2.12].

2.7

Names and definitions of vacuum pumps and their accessories

(extract from DIN 28400 Part 2 – October 1980)

Cooling time of a fluid entrainment vacuum pump. Cooling time of a fluid entrainment vacuum pump is the time needed for cooling the motive fluid from the operating temperature down to the temperature at which the motive fluid of the vacuum pump can be exposed to the atmospheric air without any danger after having switched off the heating.

Liquid separators on vacuum pumps. Vacuum pump liquid separators are appliances on the inlet and outlet side of vacuum pumps for the collection of condensates that may develop during the pumping of vapours in the vacuum pipes or in parts of the vacuum pump, or of other, even solid substances in vacuum pipes and pumps.

Heating-up period of a fluid entrainment vacuum pump. The heating-up period of a fluid entrainment vacuum pump is the period of time required for heating-up the motive fluid in a boiling vessel to its operating temperature. The initial temperature can equal the ambient temperature or the temperature at which the motive fluid of the vacuum pump can be exposed to the atmospheric air without any danger.

Outlet pressure. Outlet pressure is the pressure at the outlet of a gas transfer vacuum pump.

Vacuum pump exhaust filter. An exhaust filter on vacuum pumps is a separator on the outlet side of such pumps that transports the pumped gases to the atmosphere. It is used for the collection of gases or vapours.

Exhaust valve, discharge valve. The exhaust valve is an excess pressure valve in some displacement vacuum pumps for the discharge of the gas from the compression volume.

Booster vacuum pump, intermediate vacuum pump. A booster vacuum pump is one of the pumps installed in series which serves to increase the suction capacity in the vacuum range in which the upstream vacuum pumps are not efficient any more and the down-stream pumps are not yet efficient to the full extent.

Note: Usually, a Booster vacuum pump is a specially designed displacement or fluid entrainment vacuum pump connected between a fluid entrainment vacuum pump and the respective pre-vacuum pump.

Baffle. A baffle is a system of possibly cooled shielding elements which is situated near the inlet of a fluid entrainment vacuum pump or a diffusion pump to reduce the return flow of the motive fluid.

Diffusor. A diffusor of a fluid entrainment pump is the element in which the gas to be pumped is sucked by the motive fluid jet and mixed with the motive fluid (diffusion). The diffusor consists of an inlet cone converging in flow direction followed by a diverging outlet cone mostly in the form of a cylindrical neck.

Admission pressure; suction pressure. The admission pressure is the pressure in the inlet of a vacuum pump.

Note: This term is applicable only for vacuum pumps with own pump casings.

Vacuum pump final pressure. The final pressure is the value the pressure in a standardised test dome asymptotically approaches during normal pump operation and without gas inflow. It can be differentiated between the final pressure caused by gases and the one caused by gases and vapours.

Final vacuum pump. The final vacuum pump is the vacuum pump generating the final pressure in a vacuum pump combination.

Trap. A trap is an appliance in which the partial pressure of undesired components of a mixture of gases and/or vapours is reduced by means of physical or chemical processes.

Gas back diffusion. The gas back diffusion is a diffusion of the gas to be pumped in opposite direction to the pump flow from the outlet of the vacuum pump or from a connected baffle or a connected trap to the inflow.

Appendix pump. An appendix pump is an auxiliary vacuum pump maintaining a low pressure in an already evacuated vessel with small gas quantities.

Appendix pre-vacuum pump. An appendix pre-vacuum pump is an auxiliary pre-vacuum pump which maintains the pre-vacuum pressure for certain types of vacuum pumps, if the continuous operation of a larger pre-vacuum is not worthwhile owing to the low gas flow rate.

Ionisation trap. An ionisation trap is a trap the effect of which is based on gas clean-up. More specifically, an ionisation trap is a trap in which hydrocarbon molecules from oils ionise through gas discharge in a superposed magnetic field. They are then guided to the cathode and stimulated by charge carriers to create fixed polymerisation layers.

Compression ratio. The compression ratio is the ratio of the outlet pressure to the inlet pressure of a specific gas.

Condensate trap on a vacuum pump. A condensate trap on a vacuum pump is a separator for collecting condensates in vacuum pumps.

Cold condensation trap. A cold condensation trap is a trap the effect of which is based on condensation on cooled surfaces.

Maximum permissible mass flow rate of water vapour. The maximum permissible mass flow rate of water vapour is the mass flow rate of water vapour a vacuum pump is able to suck and transport continuously under normal ambient conditions (20 °C, 1000 mbar).

Maximum permissible pre-vacuum pressure; Pre-vacuum constancy. The maximum permissible pre-vacuum pressure is the highest pre-vacuum pressure against which a vacuum pump can work trouble-free. This value depends on different parameters.

Note: In some types of vacuum pumps, the slowdown of the pumping efficiency does not occur suddenly. In such cases, the maximum permissible pre-vacuum pressure can be determined with the aid of a measuring curve.

Multiple-stage vacuum pump. A multiple-stage vacuum pump is a series connection of vacuum pump devices often combined to a structural unit in a common casing.

Molecular filter in a vacuum pump. A molecular filter on vacuum pumps is a separator filled with sorption substances for vapour separation.

Oil trap on a vacuum pump. An oil trap on a vacuum pump is a separator on the outlet side of displacement vacuum pumps for the collection and if needed for the feedback of vacuum pump oil, often occurring in the form of drops, into the pump. In the case of drops the device is even called oil-mist trap or oil-mist filter.

Oil cleaner on a vacuum pump. An oil cleaner on a vacuum pump is an appliance to remove contamination from the vacuum pump oil.

Back creeping of motive fluids, back creeping of vacuum pump oil.

- a) In fluid-entrainment vacuum pumps: penetration of motive fluids into the vessel to be evacuated, attributable to the creeping of motive fluid molecules on the walls.
- b) In oil-sealed vacuum pumps: penetration of vacuum pump oil into the vessel to be evacuated, attributable to the creeping of molecules of the vacuum pump oil on the walls.

Suction performance of a vacuum pump. The suction performance of a vacuum pump is the $p\dot{V}$ -flow rate of the pumped gas.

Suction capacity of a vacuum pump. The suction capacity of a vacuum pump is the volume flow rate of gases removed from the gas phase from a vacuum vessel by the vacuum pump. For practical reasons, it has been established that for the suction capacity of a specific vacuum pump for a certain gas, the quotient of a $p\dot{V}$ -flow rate and an equilibrium pressure shall be defined. To calculate the quotient, it is necessary to use the $p\dot{V}$ -flow rate from a standardised test dome connected with the pump and the equilibrium pressure measured at a certain point of the test dome under specified conditions. This is valid only in cases where the connection of a test dome is possible.

Suction chamber. The suction chamber is the chamber inside the casing of a displacement pump in which the gas to be pumped enters, in which the gas is transported and from which it is discharged. In many displacement vacuum pumps, the volume of the pumping chamber changes during the pumping process.

Sorption trap. A sorption trap is a trap the effect of which is based on sorption.

Starting pressure: switching-on pressure. The starting pressure is the pressure at which a vacuum pump can be switched on and a pumping effect is achieved without causing any damage.

Dust separator on a vacuum pump. A dust separator on a vacuum pump is a trap on the inlet side of a vacuum pump to collect dust. More specifically, dust separators of vacuum pumps are appliances with large cross-sections using mechanical or electric aid for dust separation.

Dust filter of a vacuum pump. The dust filter of a vacuum pump is a dust separator for collecting fine dust by means of porous bodies or strainers.

Motive nozzle. The motive nozzle is the part of a motive fluid jet-vacuum pump or a diffusion pump which is used for generating a motive fluid jet.

Motive fluid. A motive fluid is a liquid or gaseous operating medium in motive fluid jet vacuum pumps and diffusion pumps.

Return flow of the motive fluid. The return flow of the motive fluid is the flight of motive fluid particles through the inlet of a fluid entrainment vacuum pump, or from a connected baffle or connected trap into the vacuum vessel.

Vacuum pump oil. Vacuum pump oil is the liquid used for sealing, cooling and lubrication of an oil-sealed vacuum pump.

Note: Do not mistake this term for oil used as motive fluid in fluid entrainment vacuum pumps.

Compression chamber. The compression chamber is the chamber inside the stator of some displacement pumps, the volume of which decreases and in which the gas is compressed before being discharged.

Pre-vacuum pressure. Pre-vacuum pressure is the pressure on the outlet side of the vacuum pump which is not able to work against atmospheric pressure.

Pre-vacuum pump. A pre-vacuum pump is a vacuum pump enabling the working of another vacuum pump in a vacuum pump combination, by creating the required low pressure on the outlet side of the other pump.

Water vapour compatibility. Water vapour compatibility is the highest pressure at which a vacuum pump can suck and deliver pure water vapour under normal ambient conditions (20 °C, 1000 mbar).

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3

Liquid Ring Vacuum Pumps and Liquid Ring Compressors

Liquid ring machines belong to the group of positive displacement pumps in which the delivery of the gas to be sucked is achieved by a circulating liquid. Positive displacement pumps belong to the most important machines in the field of vacuum engineering. According to DIN 28400, these machines are defined as pumps in which the medium to be pumped is delivered by means of blades, pistons, rotors etc. which are insulated against each other with or without liquid. The operating principle and the simple design of liquid ring vacuum pumps and compressors enable the versatile application of these machines. With the pumps being adequately installed a long-term maintenance-free operation is possible. Most of the designs of the liquid ring vacuum pumps are suitable for the discharge of vapor, liquid and dirt particles. Among mechanical pumps, only liquid ring vacuum pumps dispose, apart from compression, of a condensation effect. Therefore, the main field of application is the chemical industry and process engineering for the suction and compression of dry and especially humid process vapors.

According to the arrangement of the impellers in the pump casing, concentrically and eccentrically installed impellers are distinguished. Machines with cylindrical casings are called single-acting or single-chamber liquid ring pumps. Designs with oval or elliptical casings are called double-acting or two-chamber liquid ring pumps. The further classification is based on the direction of the gas inflow and outflow with regard to the impeller. Units with axial or radial flow are usual. As a rule, liquid ring vacuum pumps are designed as single or two-stage versions. Liquid ring compressors are partly designed as multiple-stage compressors.

3.1

Liquid ring vacuum pumps and compressors with radial flow

In their standard design, these machines dispose of circular casings partially filled with liquid. The impeller is arranged eccentrically in the casing. The characteristic energy transfer from the impeller via liquid ring to the gas to be sucked occurs even in this type of liquid ring vacuum pumps.

With pumps of this design (figure 3-1), the fluid to be pumped is led radially to the impellers via control cones placed in the hub area. With these machines, the dis-

charge ports are usually designed only for a defined pressure ratio so that a change of the suction pressure causes the installation of other control cones corresponding to the pressure ratio. Operation beyond the intended suction pressure and with unchanged control cones would lead to a reduced suction capacity.

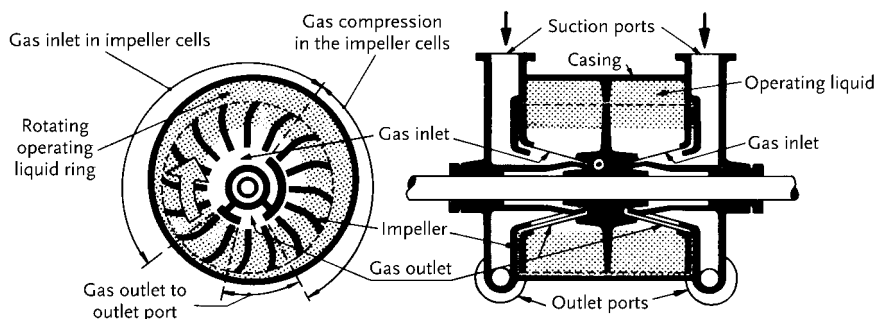


Figure 3-1. Radial flow liquid ring pump with conical control ports (acc. to Nash)

Depending on the machine size, impeller shapes for one-sided and double-sided flow are used. These pumps are also manufactured in single-acting and double-acting versions. Due to the internal separate conduction of gas, in the case of double-sided flow (figure 3-2) and concentrically divided impellers, two different suction pressures can simultaneously be run with one single pump. For normal operation with the same vacuum on both sides, the two impeller sides are parallel-connected via bifurcations.

Machines of this design are used in the vacuum range and also as compressors in the overpressure range.



Figure 3-2. Quarter cross-section of a liquid ring vacuum machine with radial flow admission (acc. to Nash)

3.2

Liquid ring machines with axial flow

In figure 3-3, an overview of liquid ring vacuum pumps and compressors with axial flow is given.

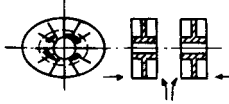
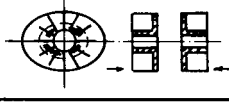
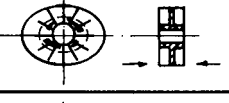
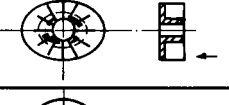
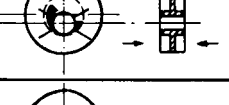
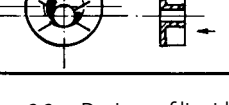
Principle sketch	Number of impellers	Effect	Admission	Impeller arrangement
	2	double	double sided	concentric
	2	double	one-sided	concentric
	1	double	double sided	concentric
	1	double	one-sided	concentric
	1	single	double sided	eccentric
	1	single	one-sided	eccentric

Figure 3-3. Designs of liquid ring vacuum machines (acc. to Sulzer-Burckhardt)

3.2.1.

Liquid ring pump with lateral channel

The characteristic element of a self-priming lateral channel pump is an open, parallel-walled impeller with radially oriented blades and a smaller assemblage clearance opposite to the lateral casing walls. Schmiedchen [3.1] and Engels [3.2] determined the suitable impeller blade profiles for the optimal increase in pressure. These machines belong to the designs with concentrically installed impellers and axial flow. Just like all directly self-priming centrifugal pumps, these pumps also need a service liquid for the evacuation of the suction pipe which must remain in the pump casing when the pump is stopped. For this reason, the suction and discharge ports are arranged vertically upwards. A prerequisite of self-priming is that at the

prevailing suction pressure the temperature of the applied service liquid is far below its boiling temperature. The gas discharge of lateral channel pumps occurs according to the displacement principle as it is the case with liquid ring vacuum pumps.

During gas conveyance the impeller cells of a lateral channel pump are closed by a water ring along the impeller circumference. With the liquid ring vacuum pump, the suction and discharge of the gas occurs through the immersion of the impeller blades into the liquid ring at different depths and through the extension and reduction of the impeller cells connected with it. In a lateral channel pump, this function is taken on by one or two lateral channels running concentrically or eccentrically to the impeller shaft (figure 3-4). The lateral channels extend over almost the whole circumference, however, being interrupted in one place. At the beginning and the end, the lateral channels shallow out to zero, while the rest of the channels show a constant depth. Opposite the beginning of the lateral channel there is a suction port (4) (fig. 3-4), and opposite the end of the lateral channel there is the discharge port (5). When the impeller rotates, the water ring departs from the impeller hub and sucks the gas through the suction port (4), until the suction process is finished at the largest depth of the channel at the end of the suction port. As soon as the impeller cells reach the end and the lowest depth of the channel, the compressed gas is emitted through the discharge port by the liquid inevitably flowing back from the lateral channel into the impeller cells. Machines of this design are mostly used for smaller suction capacities. Experience has shown that these pumps are less suitable for the suction of gases, for the suction of liquids, however, they can achieve pump heads five to fifteen times as high as in the case of centrifugal pump impellers of the same circumferential velocity, due to pulse exchange between the impeller cells and the lateral channel [3.3]. For this reason, lateral channel pumps are mostly used as self-priming machines for the pumping of liquids.

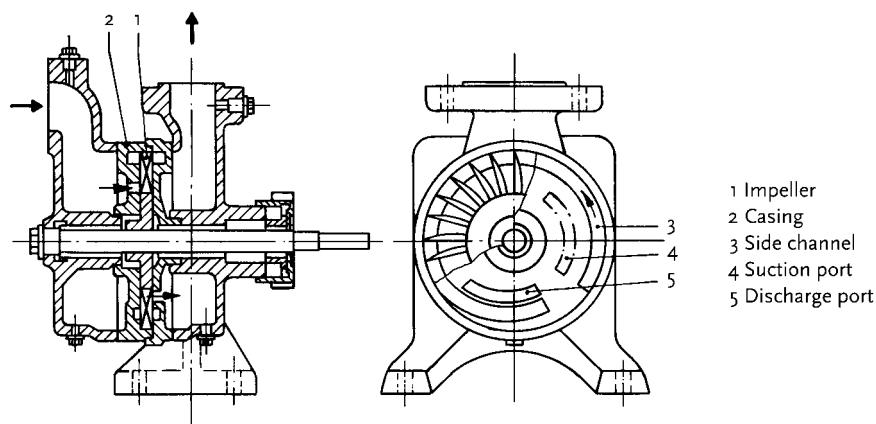


Figure 3-4. Single-stage lateral channel liquid ring pump

3.2.2

Liquid ring pump with eccentric screw wheel

This machine, also called screw pump (fig. 3-5), is a liquid ring vacuum pump with the rotor being designed as screw wheel arranged eccentrically in a circular casing. The service liquid is a suitable liquid of low viscosity such as water, selected acc. to the technical process flow. Here, too, the operating liquid helps in sealing between the suction and discharge side of the pump and helps to convey the accruing compression heat.

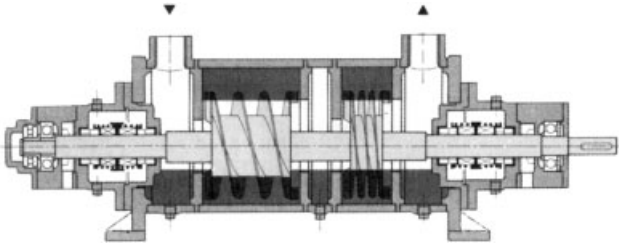


Figure 3-5. Two-stage screw wheel type liquid ring vacuum pump (acc. to AVN-Neltig)

The rotor comparable to a conveyor screw is axially pressurized by the gas to be pumped. With the rotor being in motion, a liquid ring is formed in the casing due to the centrifugal force. The quantity of liquid in the suction chamber and, therefore, the thickness of the liquid ring are determined by a circular orifice placed in the port plate on the suction side. The liquid ring touches the rotating hub only at one point, as to be seen in fig. 3-6. The rotor blades are immersed into the rotating liquid over the whole circumference, creating crescent-shaped gas-filled chambers between the liquid ring and the hub which are moved from the suction side to the discharge side in axial direction as soon as the rotor rotates. Through modification of the gradient of the rotor blades, a pressure increase of the gas to be pumped in the cell-chamber is achieved, if the gradient decreases towards the discharge side. The conveyed volume depends on the rotational speed and turn gradient of the rotor. Pumps of this design are operated at revolutions between 1000 min^{-1} and 3000 min^{-1} and can be used as vacuum pumps and as compressors. Designed with one rotor, vacuums of up to 75 % are achieved while the two-stage version reaches vacuums of up to 95 %. Operated as compressor, the discharge pressures are at about 1.70 bar. These machines are suitable for the pumping of gases even with liquid portions as well as for the discharge of liquids.

The small axial installation clearances of the impellers, ranging at some tenth millimeters and quite common with liquid ring machines, are larger in screw pumps. The axial installation clearance of the rotors is up to 2 mm, the radial installation clearance lies at about 5 mm. Therefore, these machines are suitable for the transport of dirt particles. They are insensitive to contamination, crystallization or sediments such as lime. Owing to the fact that they are able to transport gases and

liquids, these pumps are even suitable for a combined operation, i.e. in cases in which a discharge of both gas and liquids occurs. A combination of the screw rotor pumps with another pump system is possible, e. g. through installation of a centrifugal pump on the suction side.

Single- or double-acting mechanical seals are mainly used as shaft seals.

This pump design belongs undoubtedly to the robust and resistant machines, with a wide application range and versatile possibilities.

Boye [3.4] carried out fundamental tests regarding the operational behavior of these machines.

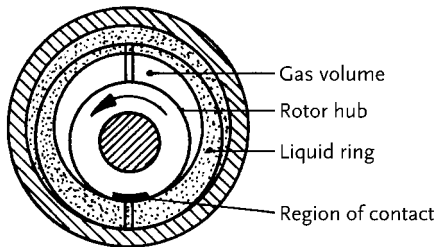


Figure 3-6. Cross-section of a screw wheel-type liquid ring vacuum pump (acc. to AVN-Neltig)

3.2.3

Liquid ring machines with elliptic casing

With these machines, also called two-chamber liquid ring pumps (fig. 3-7b), the impeller in the casing is installed symmetrically in contrast to the single-acting designs (fig. 3-7a). The pump casing has an oval shape. Due to the rotation of the impeller, a rotating liquid ring develops towards the casing contour, touching the hub on two opposite sides in the form of lines. Thus, two crescent-shaped chambers are created, which causes two operating cycles with each revolution of the impeller. Therefore, the theoretical volume flow of a two-chamber liquid ring pump is twice

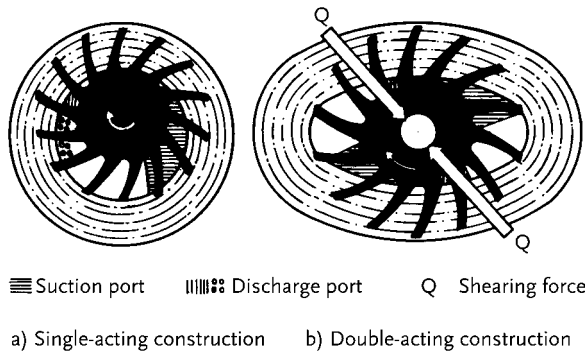


Figure 3-7. Principle of the single and double-action liquid ring machines (acc. to Siemens)

as high as the flow of the single-chamber pump with the same impeller dimensions and with identical impeller eccentricity. The suction and discharge ports of these double-acting machines are positioned diametrically opposite each other. Due to this construction feature, the impeller shafts are completely relieved of radial forces, therefore shaft deflection does not occur and the bearings are only slightly loaded. Thus large torques can be transmitted. Therefore, pumps of this design are mainly used as compressors, as here higher pressure ratios than in vacuum operation must be overcome, or higher suction pressures exist.

3.2.4

Liquid ring compressors

Liquid ring compressors are used both as single and double-acting design. Furthermore, the single- and multi-stage design is distinguished (figs. 3-8 and 3-9). In order to increase the suction capacity, machines are equipped with two-side flow impellers or double-impeller devices are used. According to their field of application, single- and double-impeller compressors are known as single-acting machines and double-acting compressors. The method of calculating the basic design values of a liquid ring machine is the same for both the vacuum pump and the compressor. Depending on the machine design and size, the optimal operation range of a vacuum pump lies at a circumferential impeller speed of between approx. 17 m/s and 20 m/s, compressors reach their optimum between 25 m/s and 30 m/s. Depending on the existing pressure ratio, there is a difference in the geometry of the port plate. Single-acting liquid ring compressors are manufactured in single-stage design for suction capacities of max. 10000 m³/h at a compression pressure of about 2.5 bar. Two-stage machines are available as single-acting versions for volume flow rates of max. ca. 1200 m³/h at discharge pressures of max. approx. 8 bar. Double-acting machines in single-impeller versions allow suction flows of max. 2500 m³/h at a compression pressure of approx. 3.3 bar. Higher compression pressures can be obtained by the

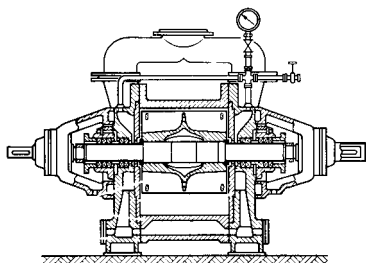


Figure 3-8. Liquid ring compressor, single-stage, with double impeller (acc. to Siemens)

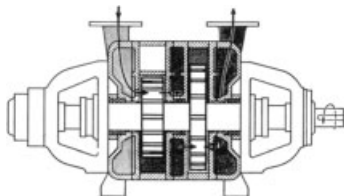


Figure 3-9. Liquid ring compressor two-stage (acc. to Siemens)

series connection of multiple-stage compressors. For the compression based on higher suction pressures double-acting compressors are preferably used.

3.2.5

Liquid ring machines with eccentrically installed impeller

Liquid ring vacuum pumps with impellers installed eccentrically in a circular casing belong to the group of single-chamber or single-acting liquid ring pumps. Here, the casing has not to be exactly cylindrical – it may also have a slightly elliptic profile. Through twisting the casing, an optimum adjustment of the machine to the point of operation is achieved. With these pumps, the drive energy is transmitted from the impeller to the gas via the liquid ring. The liquid ring is accelerated on the suction side thus disposing of higher energy. On the pressure side, the ring liquid penetrates the impeller cells, which causes some of the kinetic energy to be converted into static energy for compression. With the energy available, flow losses of the liquid and the gas are overcome, the gas is compressed and emitted.

Functioning

The functioning of a liquid ring pump is explained with the help of fig 3-10.

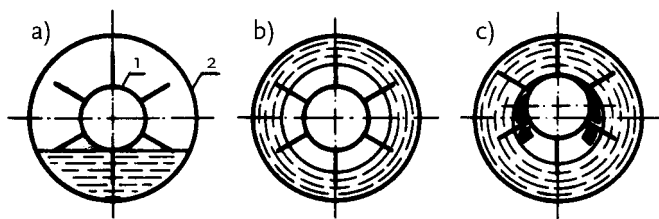


Figure 3-10. Functioning of the liquid ring pump
1 impeller, 2 cylindrical casing

In a circular casing partly filled with liquid, an impeller is installed concentrically (fig. 3-10a). The blades of the impellers can be directed radially. With blades bent forward in rotational direction, higher liquid ring speeds and with this higher compression is possible. With the revolution of the impeller, the liquid is ejected outwards due to centrifugal forces (figure 3-10b). If the impeller in the casing is eccentrically installed, a rotating liquid ring adapting to the shape of the casing develops with the impeller being in motion. Owing to the eccentricity of the impeller to the casing a crescent-shaped space develops between the impeller hub and the liquid ring when the impeller rotates (fig. 3-10c). The impeller blades divide this space into several cells of different volumes. During the rotation of the impeller, the cells in the range of the suction port become larger (fig. 3-11) and suck the gas or gas-vapor mixtures through the suction port. In the part opposite to the suction port, the liquid to be pumped is compressed in the reducing cells and emitted through the discharge port. The suction and discharge ports of axial flow machines are situated on

the lateral port plates. Consequently, liquid ring pumps are rotating machines with piston pump characteristics. Here, the ring liquid takes on the function of a piston.

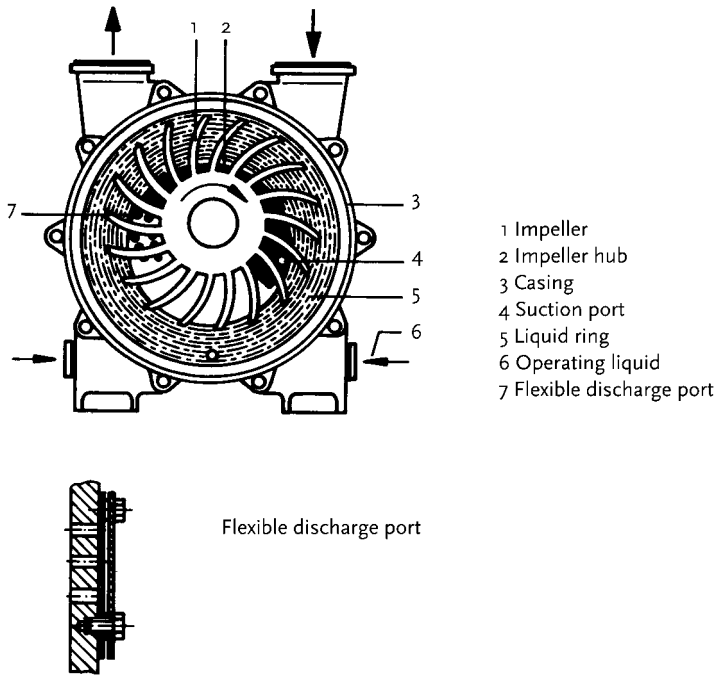
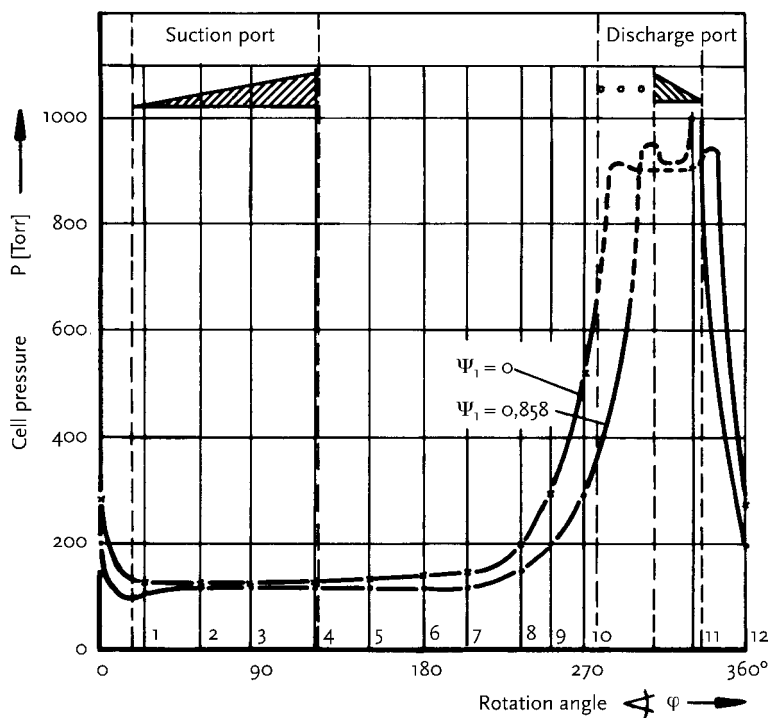


Figure 3-11. Cross-section of a liquid ring pump

Compression ratio

Figure 3-12 shows the cell pressure pattern over the rotational angle [3.5]. The compressed gases or gas/vapor mixtures are emitted in the range of the discharge port or, in multiple-stage devices, transported to the next stage.

Liquid ring vacuum pumps with water as operating liquid can reach compression ratios of up to 1:31 between the suction and the pressure sides. However, with liquid ring pumps disposing of suction and discharge ports of rigid geometric arrangement, it is not possible to run through the entire pressure range. The shape of the suction and pressure ports should be adjusted to the existing pressure ratio. The suction capacity of a liquid ring vacuum pump is determined by the position, size and shape of the suction port. The inlet port must be adjusted to the course of the operating liquid ring. The gas discharge from the impeller cells must begin only after the determined pressure has been reached, i.e. at the position of the impellers corresponding to the prevailing pressure ratio. Actually, the beginning of the discharge port should be individually determined for each pressure ratio. However, this is not feasible in practice as mostly it is not clear at which vacuum or pressure ratio the pump will actually work then.



Test series 21:

$P_1 = 122 \text{ Torr}$; $t_1 = 56^\circ\text{C}$

$G_{k1} = 43 \text{ l/min}$; $t_{k1} = 10,4^\circ\text{C}$

$\Psi = P_{wv} / p$

P_{wv} = Vapour partial pressure of operating liquid (water)

Figure 3-12. The cell pressure measured at different turning angles for two different vapour concentrations ψ_1 (Dissertation of Dr. Ing. Beckmann, 1965)

Single-stage liquid ring vacuum pumps

With single-stage liquid ring vacuum pumps (figs 3-13 and 3-14), different pressure ratios can be achieved, depending on the design and arrangement of ports. Figure 3-15 shows three different versions of port plates [3.6].

The design with a rigid discharge port shown in figure 3-15a can be optimally used only for one specific pressure ratio. If this pressure ratio determined by the design is exceeded, flow back through the discharge port into the impeller cells occurs. Exceeding the intended pressure ratio happens, e.g. if the pump is operated at unplanned suction pressures that are too low. The falling below the determined pressure ratio, for example, at the beginning of the evacuation and thus at suction pressures that are too high, causes supercompression in the impeller cells connected with an increased power consumption and a decrease in suction capacity. In this case, the residual energy of the rotating liquid ring cannot develop the full dis-

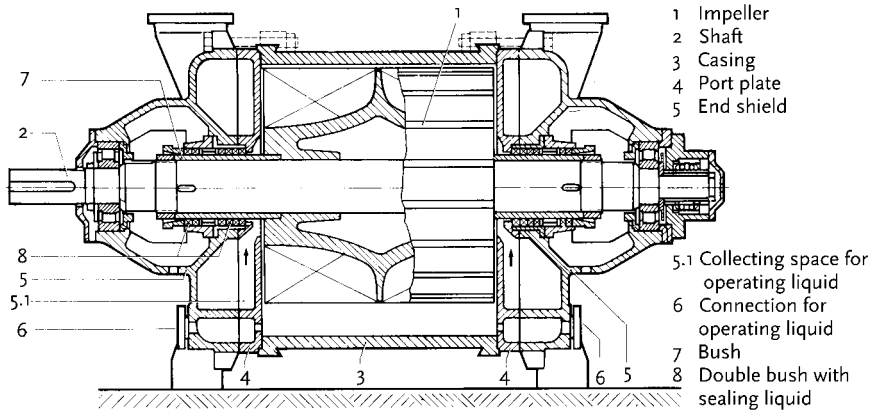


Figure 3-13. Structure of a single-stage double-admission liquid ring vacuum pump (acc. to Siemens)

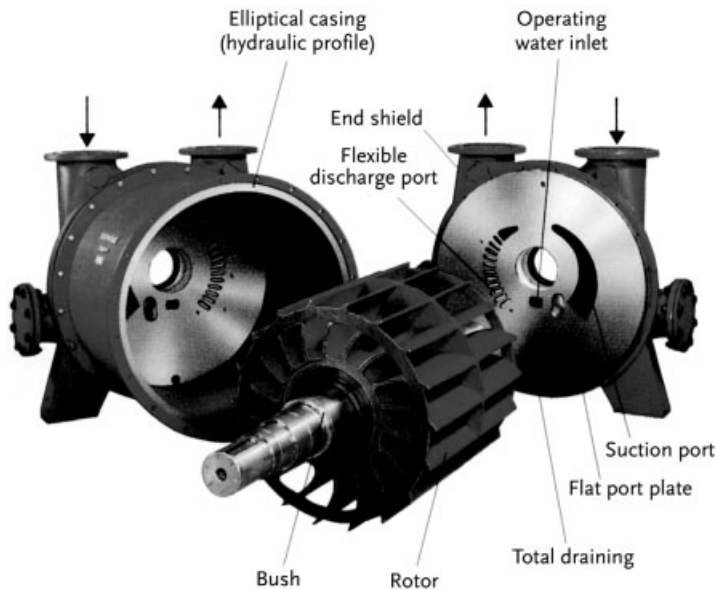


Figure 3-14. Compression chamber with impeller and port plates of a single-stage double-flow liquid ring vacuum pump (factory photo by Siemens)

charge power and part of the gas to be pumped passes from the cells from the discharge side over the peak to the suction side.

In pump designs with radial flow, it is common to design the fixed pressure ratio of the control cones in the hub area for suction pressures of e.g. 100 mbar, 200 mbar and 400 mbar. With the suitable cones with the respective cross-sections being installed, these pumps can be adapted to the operating conditions existing at the time.

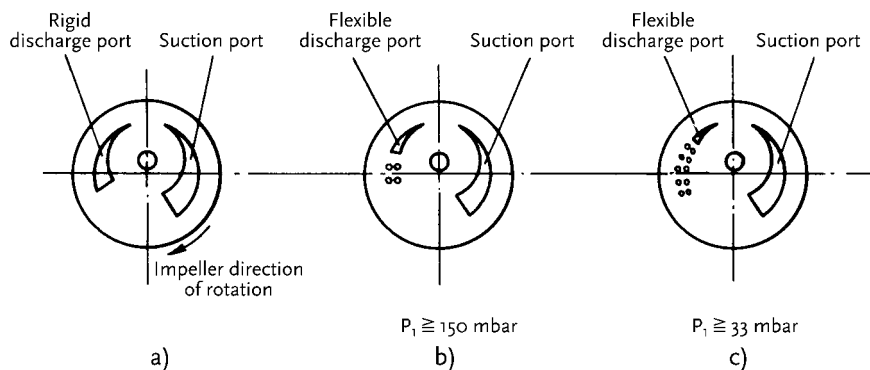


Figure 3-15. Port plate versions

In single-stage machines with axial flow, port plates with variable discharge ports are used to enable operating at higher suction pressure range. The version acc. to figure 3-15b is equipped with pressure relief holes. As soon as the holes are reached, the liquid to be pumped can already be emitted here towards the discharge side of the pump, while supercompression does not occur. Reflow through the wholes is avoided, as compared to gases the flow resistance of liquids in the wholes is higher. The liquid under outlet pressure behind the port plate partially closes the holes. Liquid ring vacuum pumps with port plates of this design can be economically used for pressure ratios of up to about 1:7, beginning with suction pressures of 1013 mbar down to approx. 150 mbar, with the ring liquid water at 15 °C.

For the operation at pressure ratios higher than 1:7, at which the pump capacity of the sucked air decreases considerably the single-stage liquid ring vacuum pumps are equipped with shortened discharge ports and further holes in front of them, as well as with an automatic valve (fig. 3-15c). Nowadays, plate valves are more common. Ball valves used previously are not used any longer. The outlet orifice of the port plate is adjusted to the respective pressure ratio by this valve. The elastic plate applied closes or opens the holes automatically over the entire pressure range thus preventing both supercompression and reflow. With water at 15 °C being used as ring liquid, suction pressures between about 1013 mbar and 33 mbar are possible, which corresponds to a ratio of suction pressure to compression pressure of 1:30.7. If other ring liquids, e. g. oils, are used, suction pressures between about 10 mbar and 30 mbar can be reached. Single-stage pumps with plate valves can be optimally operated at respectively adjusted pressure ratios over a wide range of suction pressures.

Mode of action of the plate valve

Due to the application of an elastic plate, the plate valve is also called flexible discharge port [3.7]. In the following, the function of this valve will be described for 3 different pressure ratios (see fig. 3-16).

At a suction pressure of 500 mbar, the compression pressure of 1013 mbar is already reached prior to the actual discharge port in the impeller cells. The plate valve is fully opened and the compressed gas emits thorough the holes with super-compression being avoided.

At a suction pressure of 150 mbar, the compression pressure of 1013 mbar is reached later than shown in figure 3-16a. The atmospheric air pressure, in this case being also the compression pressure, presses the plate valve against those holes behind which the compression pressure in the impeller cells has not been reached yet. Therefore, in this pressure range both supercompression and reflow are avoided.

With a suction pressure of 33 mbar, a compression pressure of 1013 mbar in the impeller cells will only be achieved at the moment the discharge port is reached. The gas emits via the discharge port. All holes positioned in front of the discharge port are closed by the plate valve through atmospheric pressure. Therefore, the occurrence of reflow is impossible.

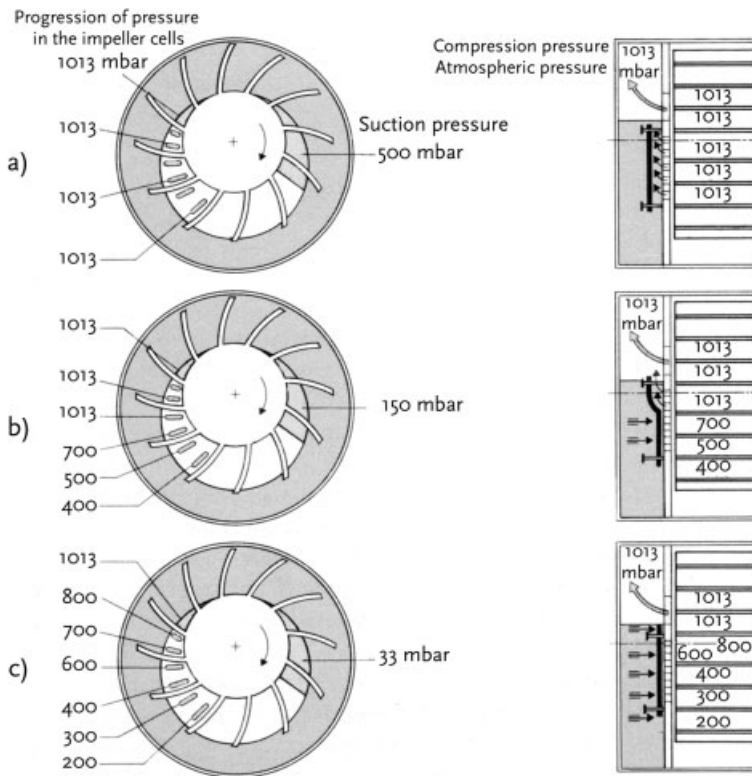


Figure 3-16. Function of the flexible discharge port (acc. to Siemens)

Two-stage liquid ring-vacuum pumps

For suction pressures ranging below 150 mbar, apart from single-stage machines with flexible discharge ports (fig. 3-17) even two-stage liquid ring vacuum pumps are used.

This design disposes of gas outlets without valves. The compression pressure of the first stage corresponds to the suction pressure of the second stage. Depending on the total compression ratio, the two stages have differently designed liquid ring configurations.

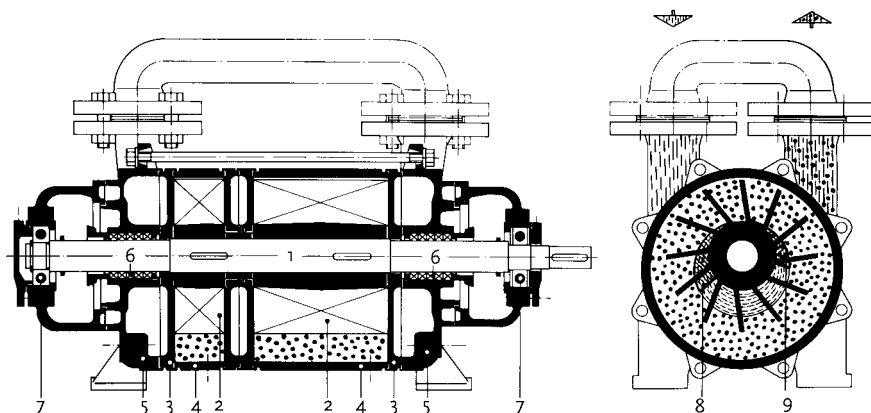


Figure 3-17. Cross section of a two-stage liquid ring-vacuum pump (acc. to Sihi)

1 shaft, 2 impellers, 3 port plate, 4 middle section, 5 casing, 6 shaft seal, 7 shaft bearings, 8 suction port, 9 discharge port

Thereby, a variable partial compression ratio conforming to the suction pressure occurs for each stage. Owing to the rigid discharge ports, a compression ratio smaller than 1:7 per stage results. The total compression ratio is the product of both partial compression ratios. The impeller dimensions are determined in accordance with hydraulics criteria and volume ratios.

In two-stage machines, the suction capacity does not depend so much on the vapor pressure of the operating liquid as it is the case in the single-stage design, since the second compression stage is not as strongly influenced by the vapor pressure as the first stage, due to a higher suction pressure. With water as ring liquid at a temperature of 15 °C, also the two-stage design of the liquid ring-vacuum pump is suitable for pressure ranges from 1013 mbar to 33 mbar.

The liquid ring-vacuum pump can also be applied for suction pressures below 33 mbar. In these cases, vacuum pumps of another pump system are installed on the suction side in front of the liquid ring pump. For this purpose, gas jet pumps, steam jet pumps, Roots pumps, sliding vane pumps and condensers can be used.

Theoretical and actual suction capacity

Disregarding its condensation behavior, the liquid ring vacuum pump belongs to the displacement pumps. The theoretical suction flow can be determined by the

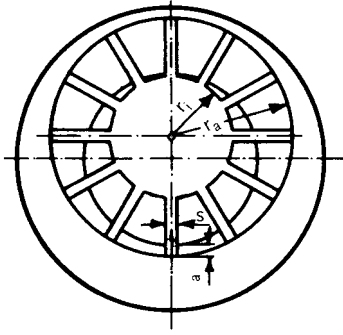


Figure 3-18. Impeller arrangement of a liquid ring vacuum pump

impeller cell volume and the impeller revolutions. The volume flow rate, as shown in figure 3-18, can be calculated acc. to equation (3-1) [3.8].

$$\dot{V}_{\text{theor}} = 60 \cdot n \cdot b \left\{ \pi \left[(r_a - a)^2 - r_i^2 \right] - z \cdot s (r_a - r_i - a) \right\} \quad (3-1)$$

\dot{V}_{theor}	volume flow [m ³ /h]
n	impeller rotation speed [min ⁻¹]
b	impeller width [m]
r_a	impeller radius [m]
r_i	impeller hub radius [m]
a	immersion depth of the blades [m]
s	blade thickness [m]
z	number of blades

From equation (3-1), a horizontal straight line results (fig. 3-19). As the cell volume does not only fill with the sucked gas-vapor mixture, but partly also with the vapor of the operating liquid, the cell volume available for the sucked gas-vapor mixture reduces by the correction factor f acc. to Dalton's Law.

$$\varphi = \frac{P_A - P_s}{P_A} \quad (3-2)$$

φ	portion of pumped gas
P_A	suction pressure of the pump
P_s	saturated vapor pressure of the operating liquid

The real characteristic curve is obtained by including the volumetric coefficient of efficiency η_{vol} . Owing to the rotation of the impeller being eccentrically installed in a casing and partly filled with liquid, a liquid ring develops due to the effective centrifugal forces in which the impeller blades are immersed. Blade chambers occur which are limited by two neighboring blades, the hub and the surface of the liquid. Owing to the respectively adjusted eccentricity, the blade length and the liquid fill-

ing, the liquid ring touches the hub slightly on one side while the tips of the blades immerse into the liquid ring by the measure a on the opposite side (fig. 3-18).

Thus, the theoretical air volume flow rate of liquid ring machines is further reduced by losses due to overflow and leakage. The overflow losses occur because the liquid ring does not fit completely on the impeller hub, and the remaining residual volume flows back in the impeller cells to the suction side. Between the port plate and the impeller blades, a part of the compressed gas also escapes to the suction side. These losses are taken into consideration by the volumetric efficiency η_{vol} . At suction pressures between 260 mbar and 660 mbar, the volumetric efficiency η_{vol} lies at about 0.70–0.90, depending on the machine size. Below 250 mbar, the volumetric efficiency decreases progressively

The actual pump capacity referring to the suction state \dot{V}_{theor} can be calculated acc. to equation (3-3).

$$\dot{V} = \eta_{vol} \cdot \varphi \cdot \dot{V}_{theor} \quad (3-3)$$

Figure 3-19 shows the actual suction capacity \dot{S} of a liquid ring vacuum pump referring to water as ring liquid at 15 °C. The theoretical suction capacity \dot{S}_{theor} results in a horizontal straight line.

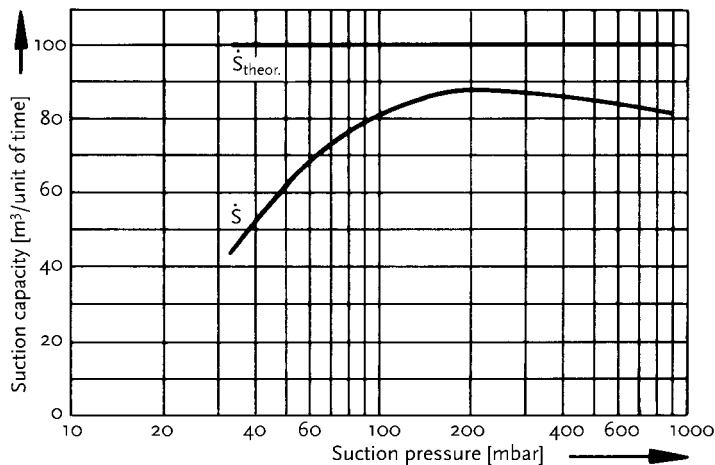


Figure 3-19. Theoretical and real suction capacity of a two-stage liquid ring-vacuum pump (acc. to Sihi)

In comparison, figure 3-20 shows the theoretical volume flow \dot{V}_{theor} and the actual volume flow \dot{V} of a liquid ring compressor depending on the compression pressure.

According to Prager and Bramer [3.10] [3.11], the real suction capacity can also be determined using the pump capacity factor.

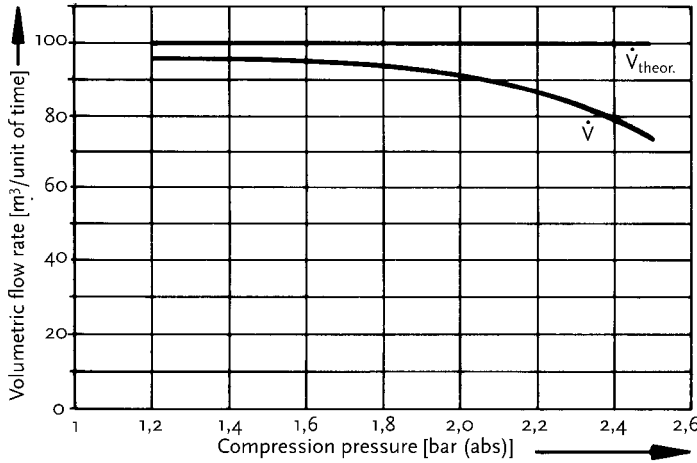


Figure 3-20. Theoretical and real volume flow rate of a single-stage liquid ring compressor (acc. to Sihi)

$$\dot{V} = \dot{V}_{\text{theor}} \cdot \lambda_{\text{nu}} \quad (3-4)$$

λ_{nu} pump capacity factor

λ_f filling ratio

λ_l supply factor

The pump capacity factor λ_{nu} is the product of the filling ratio λ_f and the supply factor λ_l . The filling ratio λ_f takes the volumetric losses into consideration occurring due to incomplete filling of the cells at the end of the suction process caused by the inflow for a limited time. The supply factor λ_l , also known as sealing factor, contains all leakages occurring in the machine at the clearances between the impeller and the casing, as well as the circulation flow at the blade tips and the influence of the existing liquid vapor [3.12]. At the end of the suction period, the gas is completely saturated independent from the saturation state in the inlet, since the pumping of the gas occurs in close contact to the operating liquid.

Acc. to the surveys of Prager's [3.9] [3.10], dependences of the volume flow from the pressure ratio result from similarity analyses.

$$\dot{V} = \frac{\lambda_{\text{nu}}}{\lambda_{\text{nu,max}}} \cdot \lambda_{\text{nu,max}} \cdot \dot{V}_{\text{theor}} \quad (3-5)$$

For compressors

$$\frac{\lambda_{\text{nu}}}{\lambda_{\text{nu,max}}} = 0.707 \cdot X^3 - 1.527 \cdot X^2 - 0.1782 \cdot X + 1 \quad (3-6)$$

For vacuum pumps

$$\frac{\lambda_{\text{nu}}}{\lambda_{\text{nu,max}}} = 0.147 \cdot X^3 - 1.297 \cdot X^2 + 0.150 \cdot X + 1 \quad (3-7)$$

The pressure ratio X is obtained acc. to equation (3-8).

$$X = \frac{p_D/p_A - 1}{(p_D/p_A)_{\max} - 1} \quad (3-8)$$

p_D pressure in the discharge port [Pa]

p_A pressure in the suction port [Pa]

Figure 3-21 shows how the dependence of pump capacity factor λ_{nu} on the circumferential velocity of the impeller at the outer diameter.

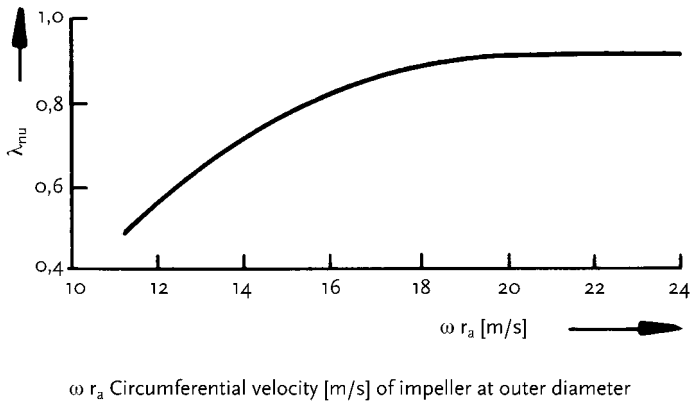


Figure 3-21. Pump capacity factor (volumetric efficiency) and circumferential velocity

The alteration of $\lambda_{nu}/\lambda_{nu, \max}$ depending on the pressure ratio is shown in figure 3-22.

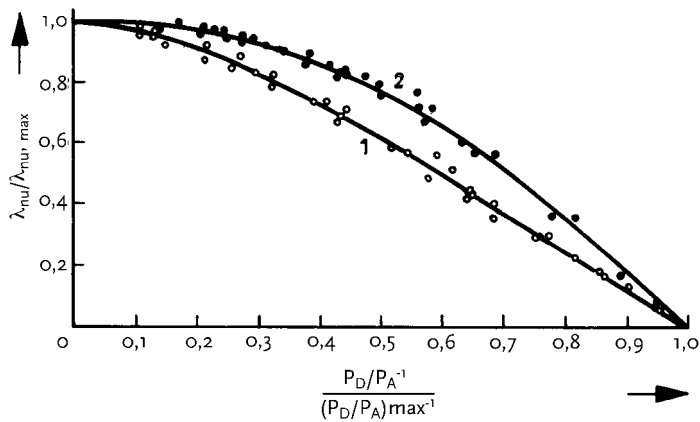


Figure 3-22. Pump capacity factor and pressure ratio
1 compressor, 2 vacuum pump

The pressure gradient ψ . The compression ratio acc. to equation (3-8) can be also expressed by a dimensionless pressure gradient ψ .

$$\psi = \frac{2p_A(p_D/p_A - 1)}{\rho \cdot \omega_{ra}^2} \quad (3-9)$$

- ψ pressure gradient
 p_D pressure at the discharge port [Pa]
 p_A pressure at the suction port [Pa]
 ρ density of the operating liquid [kg/m³]
 r_a outer radius of the impeller [m]
 ω_{ra} circumferential velocity [m/s]
 $1 \text{ Pa} = 1 \text{ [kg/(m} \cdot \text{s}^2)]$

The impeller. According to the pump size, the impellers dispose of 12 to 24 blades, and just like the number of blades even the rotational speed depends on the size of the pump. Speeds of 3000 min⁻¹ are possible for smaller pumps, while larger units are operated at 1450 min⁻¹ and below. The circumferential velocities of the impellers at the blade tips usually range from 17 m/s to 20 m/s. In cases in which the strength conditions are fulfilled, circumferential velocities up to 25 m/s are permissible. Here an increase in pressure of up to 50% is possible, while an improvement of the isothermal coupling efficiency is achieved.

Considering the pump capacity factor λ_{nu} , the dimensions of the impellers for compressors with one-sided axial flow are approx.

$$\frac{l}{d_a} = 0.6$$

and for compressors with double axial flow

$$\frac{l}{d_a} = 1.2 \quad [3.13].$$

- l impeller length
 d_a outer diameter of the impeller

Depending on machine size and pressure ratio, the pump capacity factor λ_{nu} lies between 0.7 and 0.9. In order to achieve these values, the required liquid level in the pump and the smallest possible value for the axial installation clearance between the impeller front and the casing cover have to be maintained with values between 0.10 mm and 0.30 mm depending on the pump size.

The higher the pumped volume flow, the more the rotational speed has to be reduced in order to obtain suitable impeller dimensions. For larger pumps, the two-side flow is preferred, to reach a good pump capacity factor λ_{nu} . The gas is fed to and removed from both sides of the impeller. The pulled-up partition of the hub and the reduction of the hub diameter from the center to the front sides improve the flow conditions in the blade cells. Designs are known in which the casing chamber

is separated by a partition wall and the pulled-up impeller hub. To these machines, two separate suction ports with different suction pressures can be connected.

In figures 3-23 and 3-24, optimal pressure ratios and optimal isothermal coupling efficiencies are given acc. to [3.9].

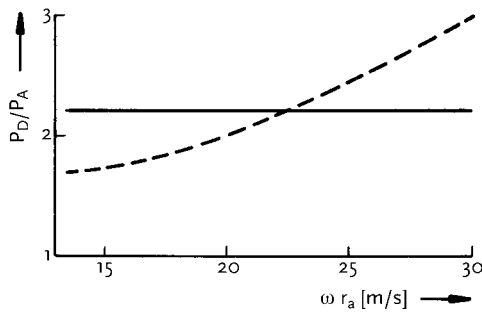


Figure 3-23. Optimal pressure ratios

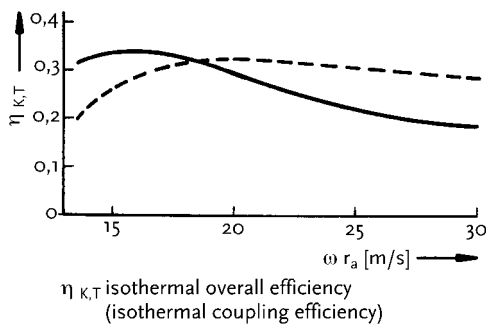


Figure 3-24. Optimal isothermal coupling efficiencies

Usually, the maximal achievable pressure gradient for blades bent forward at an angle of $\beta_2 \approx 135^\circ$ (fig. 3-25) has to be calculated at 1.30 [3.12]. From equation (3-9), the minimally achievable suction pressure of a vacuum pump can be calculated, as in this case, the pressure at the discharge port can be equated with the atmospheric air pressure. The smallest possible suction pressure depends on the vapor pressure of the operating liquid. Inserting the vapor pressure p_s into equation (3-9), the smallest permissible circumferential velocity of the liquid ring vacuum pump can be calculated acc. to equation (3-10).

$$\omega r_a = \sqrt{\frac{2(p_D - p_s)}{\rho \cdot \psi_{\max}}} \quad (3-10)$$

p_s saturated vapor pressure of the operating liquid [Pa]

In the lists of the manufacturers, vacuum pumps are based on a compression pressure of 1013 mbar (abs), whereas the compressors are based on a suction pressure of 1013 mbar (abs).

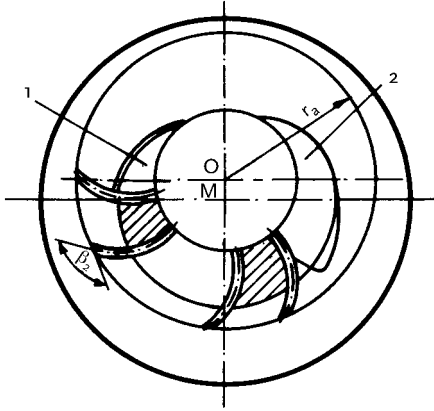


Figure 3-25. Geometric ratios on a liquid ring vacuum machine
1 discharge port, 2 suction port

3.3

The operating liquid

Apart from energy transfer and sealing of the impeller cells and clearances between the impeller, the port plate and the casing, the operating liquid is also necessary for the absorption and the removal of the heat accrued in the pump. Besides the compression heat, further heat flows may accrue in the liquid ring pump, as a result of vapor condensation, absorption of gases, or chemical reactions between the process gas and the ring liquid as well as owing to the cooling of sucked gases with a higher temperature.

The sum of the expected heat quantity to be removed can be arithmetically calculated acc. to equation (3-11), [3.14].

$$\dot{Q}_{\text{tot}} = \dot{Q}_{\text{comp}} + \dot{Q}_{\text{cond}} + \dot{Q}_{\text{cool}} \quad (3-11)$$

$$\dot{Q}_{\text{comp}} = 0.9 \cdot P \cdot 3600 \quad (3-12)$$

$$\dot{Q}_{\text{cond}} = \dot{m}_v \cdot r \quad (3-13)$$

$$\dot{Q}_{\text{cool}} = \dot{m}_G \cdot c_p \cdot (T_{\text{in}} - T_o) \quad (3-14)$$

\dot{Q}_{tot}	heat flow to be removed from the pump [kJ/h]
\dot{Q}_{comp}	isothermal compression flow and heat loss flow [kJ/h]
\dot{Q}_{cond}	condensation heat flow [kJ/h]
\dot{Q}_{cool}	heat exchange gas/operating liquid [kJ/h]
P	power consumed by the pump [kW]
\dot{m}_v	mass flow of the condensing vapor [kg/h]
\dot{m}_G	mass flow of the sucked gas [kg/h]
c_p	spec. heat of the sucked gas [kJ]/(kg · K)
r	spec. evaporation and condensation heat [kJ/kg]
T_{in}	inlet temperature of the sucked gas [K]
T_o	outlet temperature of the operating liquid [K]

During the operation of the pump, a part of the operating liquid is discharged through the discharge port together with the gas to be transported. This heated quantity of liquid conveyed together with the gas must be replaced in the pump. Depending on the mode of operation, this quantity is taken from an available supply system or re-fed to the pump from the liquid separator on the pressure side after being cooled by a heat exchanger. Owing to this continuous circulation, there is a permanent heat transport from the pump which keeps the temperature of the ring liquid constant. About 90% of the arising heat quantities are removed through the ring liquid. Due to the higher heat capacity of the operating liquid compared to the gas to be pumped, most of the energy passes to the liquid ring during the heat exchange between these two matters, causing the temperature of the compressed gas to be only slightly higher than the temperature of the new operating liquid entering the pump. Consequently, the compression typical for this design of liquid ring vacuum pumps is almost isothermal. During the compression of dry gas with water as ring liquid, depending on the operating pressure an increase in temperature of about 3 °C to 10 °C with liquid ring vacuum pumps and of about 5 °C to 20 °C with liquid ring compressors is to be expected compared to the inlet temperature of the operating liquid. [3.8].

The type of ring liquid for these machines is chosen with regard to the chemical qualities of the gas and vapors to be pumped. The respective combination of gas and operating liquid enables engineering processes in the liquid ring machines. Besides water, even various alcohols, oils, gasoline, chloroform, petroleum, phenol, lye or acids are used as ring liquids. Mainly liquids with a high specific heat capacity are preferred in order to ensure good cooling. Concentrated sulphuric acid is usually used as ring liquid for the compression of aggressive humid chlorine gas. At the same time, the concentrated acid cleans the gas. Owing to the fact that the conventional ring liquid is water, these pumps are also called water ring pumps.

3.3.1

Influence of the operating temperature of the ring liquid on suction capacity and suction pressure of the pump

Values listed by the pump manufacturers usually refer to the compression of dry air at a temperature of 20 °C and to an inlet temperature of the operating water of 15 °C. Using a different fluid as ring liquid with different physical qualities or water at a different temperature, the suction capacity and the power consumption will change. In the range of lower suction pressures, the field of application of these machines is mainly limited by the vapor pressure of the operating liquid. To avoid cavitation in the pump, the operation of the pump near the curve of the saturated vapor pressure should be avoided.

With the aid of figure 3-26, the influence of the partial vapor pressure on the suction volume of a liquid ring pump at an inlet temperature of the operating water of 15 °C will be explained. Owing to the direct contact of the gas and ring liquid, the impeller cells are partly filled with water vapor during the suction process. At the

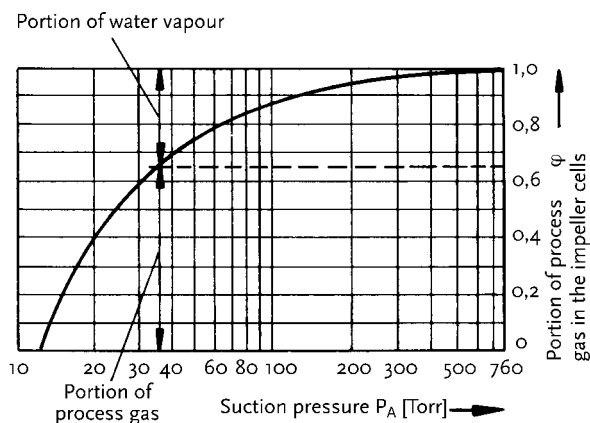


Figure 3-26. Suction capacity of a liquid ring vacuum pump influenced by the partial vapour pressure at an inlet water temperature of 15 °C

same time, the water vapor is under saturated vapor pressure corresponding to the inlet temperature of the water.

The volume portion of the impeller cells available to the gas to be sucked is calculated according to the proportional partial pressure of the water vapor with the aid of equation (3-2).

$$\varphi = \frac{p_A - p_s}{p_A} \quad (3-2)$$

φ portion of pumped gas

p_A suction pressure of the pump

p_s saturated vapor pressure of the operating liquid

Therefore, the lower the suction pressure and the higher the partial vapor pressure of the operating liquid, the lower is the suction volume of the pump. With the suction pressure corresponding to the partial vapor pressure the impeller cells are completely filled with water vapor, i.e. the portion of the gas to be pumped $\varphi = 0$, and the pump cannot suck gas any longer. If the pump is operated at operating water temperatures higher than 15 °C on which the characteristic curves are based, the higher saturated vapor pressure of the liquid affects the suction capacity acc. to Dalton's Law [3.15]. With dry air being sucked, the effective suction capacity of the pump at a constant volume of the impeller cells can be approximately calculated acc. to equation 3-15.

$$\dot{S}_{\text{eff}} = \dot{S}_k \cdot \frac{p_A - p_s}{p_A - p_k} \quad (3-15)$$

\dot{S}_{eff} effective suction capacity

p_A suction pressure of the vacuum pump

- p_k saturated vapor pressure of the operating liquid under conditions given in the catalogue (at 15 °C)
 \dot{S}_k suction capacity for dry air under conditions given in the catalogue (at 20 °C)
 p_s saturated vapor pressure of the operating liquid at the operating temperature

For an exact determination of the performance data of a liquid ring vacuum pump at different suction pressures and operating liquid temperatures, the pump manufacturer make conversion curves available which consider also correction factors resulting from the design. Figure 3-27 shows the correction factors for a single-stage liquid ring vacuum pump as well as the limit curve of critical state for cavitation-free operation.

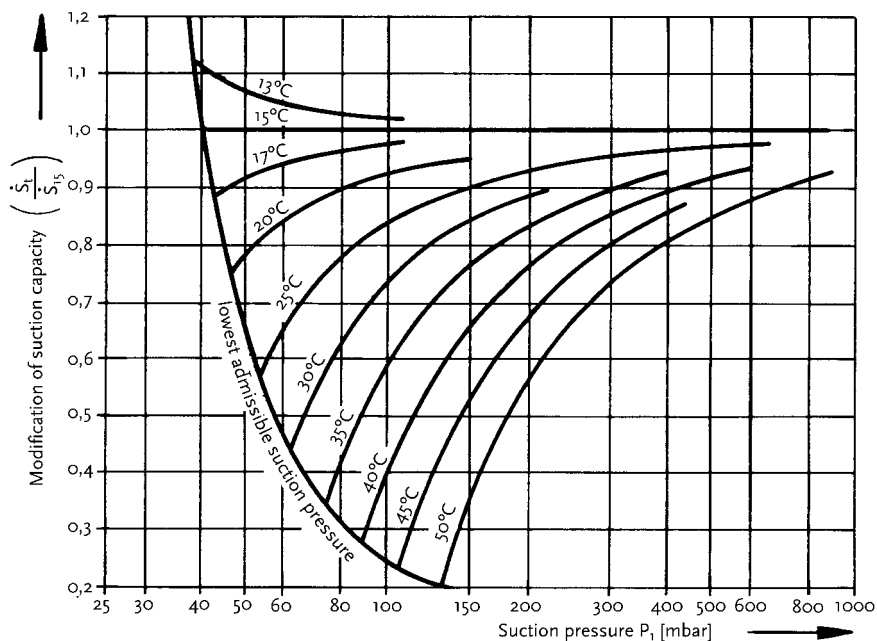


Figure 3-27. Change of suction capacity of single-stage liquid ring vacuum pumps, depending on the operating temperature of the entering water, applicable for dry air at 20 °C (acc. to Sihi)

In contrast to vacuum pumps, the influence of the operating liquid temperature of a compressor is insignificantly small.

3.3.2

Operating behavior at different densities of the operating liquid

The kinetic energy contained in the rotating liquid ring is proportional to the quantity of liquid displaced, to the density of the operating liquid and to the square of the circumferential velocity of the liquid ring.

$$E = V \cdot \rho \cdot v^2 \quad (3-16)$$

- E kinetic energy [$\text{kg} \cdot (\text{m}^2/\text{s}^2)$]
 V volume of circulating liquid [m^3]
 ρ density of ring liquid [kg/m^3]
 v circumferential velocity [m/s]

The power consumption of the pump changes with the specific weight of the ring liquid, and thereby with the energy in the liquid ring. Through this change the inner shape of the ring and with this perhaps the suction capacity may be affected, depending on the constructive design of the pump. Within the range of $\rho = 800 \text{ kg}/\text{m}^3$ to $1200 \text{ kg}/\text{m}^3$, the power consumption changes approximately in proportion to the value of the square root of the operating liquid density [3.16].

$$N_X = N \cdot \sqrt{\rho_X} \quad (3-17)$$

Applicable for values within the range $\rho = 0.8 \text{ [kg}/\text{dm}^3]$ to $1.2 \text{ [kg}/\text{dm}^3]$

- N_X power consumption for ring liquid with a density deviating from water
 N power consumption for ring liquid water, $\rho = 1.0 \text{ [kg}/\text{dm}^3]$
 ρ_X ring liquid density [kg/dm^3]

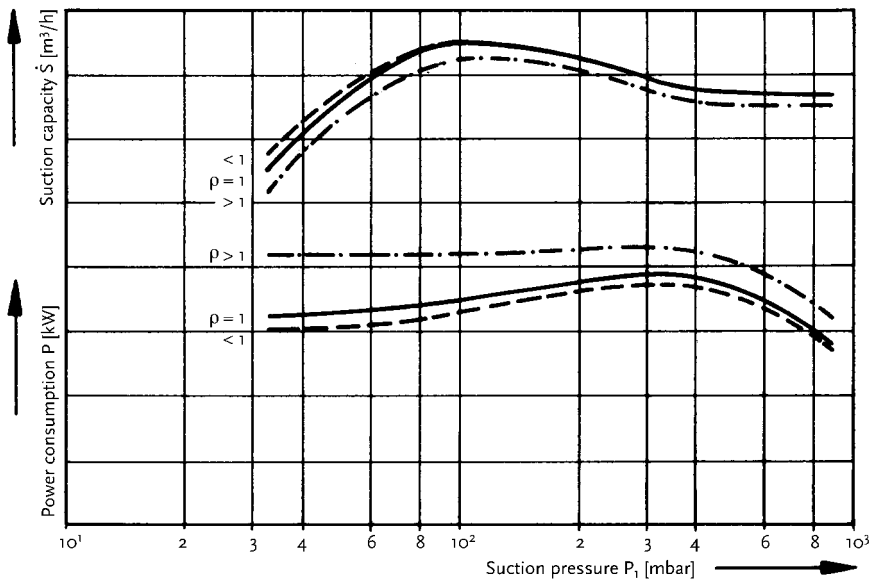


Figure 3-28. Suction capacity and power consumption of a two-stage liquid ring vacuum pump influenced by the operating liquid density (acc. to Sihi)

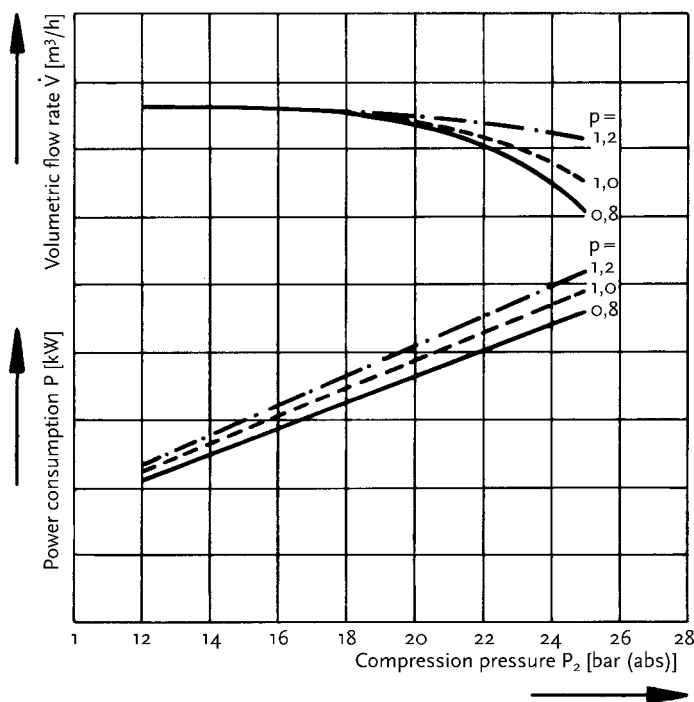


Figure 3-29. Volume flow rate and power consumption of a single-stage liquid ring compressor at different densities of the operating liquid (acc. to Sihi)

The dependence of the suction capacity and power consumption on the density of the ring liquid for a two-stage liquid ring vacuum pump is shown in figure 3-28 and for a single-stage liquid ring compressor in figure 3-29.

3.3.3

Influence of the viscosity of the operating liquid on the discharge behavior of the pump

If liquids other than water are used, the respective dynamic viscosity of the liquid in question has to be taken into consideration as well. With an increasing viscosity a higher driving force is required, causing only a slight change of the suction capacity. Higher viscosities achieve a better sealing of the clearance between the impeller and the casing elements. However, if viscosity is too high too little ring liquid is sucked and the suction capacity of the pump decreases. Therefore, the suction capacity tends to decrease with increasing viscosity.

The power consumption for viscosities of up to 20 mPas can be approximately determined according to equation 3-18 [3.6].

$$\frac{P_X}{P} = 1 + 0.01 \cdot \frac{\eta_X}{\eta} \quad (3-18)$$

P power consumption with water as ring liquid

P_X power consumption for ring liquid with viscosity deviating from water

η dynamic viscosity of water ($\eta = 1.15$ [mPas])

η_X dynamic viscosity of another operating liquid [mPas]

In cases when liquids with other physical properties than water are used, the suction capacity and power consumption curves change in comparison with the original characteristic curves.

3.3.4

Solubility of gases in the operating liquid

The contact between the gas to be pumped and the operating liquid in the pump can lead to chemical reactions between gas and liquid. The liquid ring vacuum pump compresses the sucked gas to the reaction pressure thus causing absorption. The solubility of gases in the liquid is different depending on the pressure. In table 3-1, the solubility of different gases in water is compared at different temperatures and under normal conditions.

Table 3-1. Solubility of gases in water [3.17]

Gas	Solubility m ³ gas under normal conditions in 1 m ³ at t °C				
	0	10	20	30	40
Acetylene	1.73	1.31	1.03	0.84	
Argon	0.0519	0.0395	0.0329	0.0281	0.0247
Chlorine	0.0987	0.0656	0.0472	0.0362	0.0292
Dinitrogen monoxide	0.226	0.162	0.122	0.098	
Ethane	4.61	3.148	2.299	1.799	1.438
Ethylene		0.878	0.630	0.280	0.181
Carbon dioxide	1.713	1.194	0.878	0.665	0.530
Carbon monoxide	0.0354	0.0282	0.0232	0.0200	0.0178
Air	0.0289	0.0227	0.0187	0.0161	0.0142
Methane	0.0556	0.0418	0.0331	0.0276	0.0237
Oxygen	0.0489	0.0380	0.0310	0.0261	0.0231
Sulphur dioxide	79.79	56.65	39.37	27.16	18.77
Hydrogen sulphide	4.67	3.40	2.58	2.04	1.66
Nitrogen	0.0235	0.0186	0.0155	0.0134	0.0118
Hydrogen	0.0215	0.0196	0.0182	0.0170	0.0164

For gases having a good solubility in the existing operating liquid, degassing at the suction side of the pump, a reduction of the active cell volume as well as a reduction of the suction volume flow are to be expected. Experience shows however, that

the reduction of the suction capacity is small, since the interval of the cells passing the suction port is very short. Therefore the time available for a partial degassing is limited.

3.4

The quantity of operating liquid

Fig. 3-30 shows the influence of the quantity of operating liquid on the suction capacity and fig. 3-31 shows the influence of this dependence on the power consumption of a single-stage liquid ring vacuum pump over the suction pressure. Fig. 3-32 shows the characteristic curve of the volume flow of a liquid ring compressor as a function of the operating liquid flow Q_{F1} and the compression pressure.

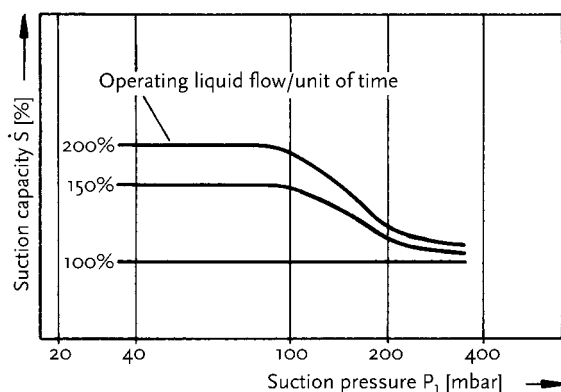


Figure 3-30. Suction capacity of a liquid ring vacuum pump depending on the operating liquid flow rate (acc. to Siemens)

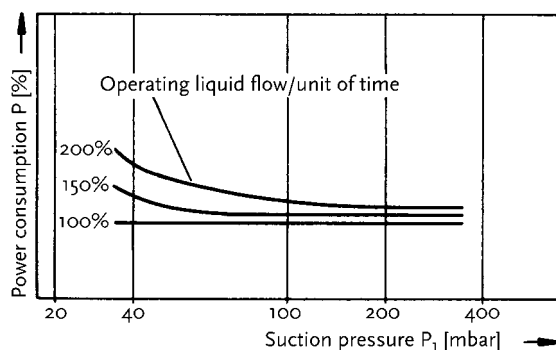


Figure 3-31. Power consumption of a liquid ring vacuum pump depending on the flow rate of the operating liquid (acc. to Siemens)

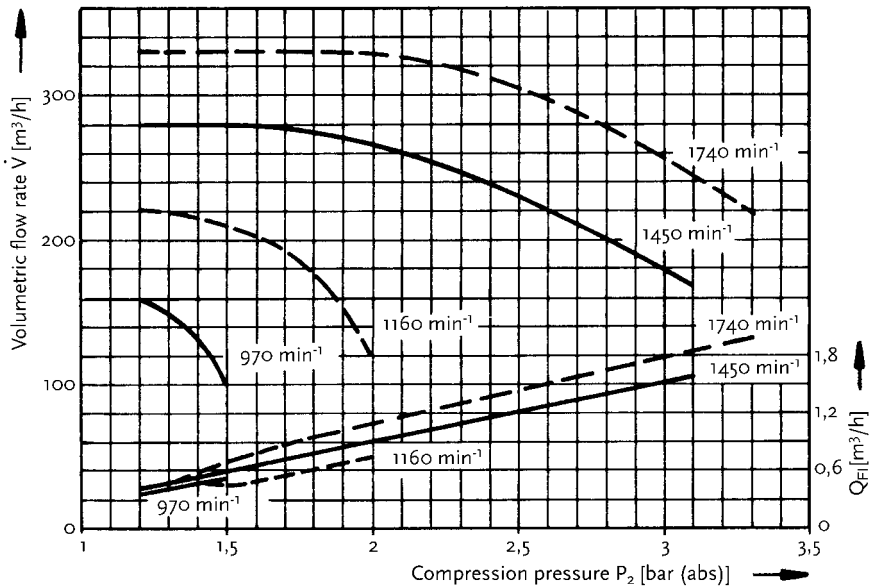


Figure 3-32. Change of the volume flow rate of a liquid ring compressor depending on the discharge pressure, on the pumps speed and on the flow rate of the operating liquid (acc. to Siemens)

With an increasing quantity of fresh liquid an increase in suction capacity is achieved for both compressor and vacuum operation, as the supply of cooled liquid in sufficient or larger quantities provokes a more intensive cooling in the pump. This increase is especially distinctive in vacuum operations at low suction pressures. The reason for this behavior is also the better sealing of clearances between the impeller and the casing parts. According to equation 3-19, the gas portion which can be transported in the impeller cells depends on the suction pressure and on the temperature in the inlet area of the pump.

$$\frac{V_{\text{gas}}}{V_{\text{total}}} = \frac{p_{\text{total}} - p_s}{p_{\text{total}}} \quad (3-19)$$

V_{gas} gas portion

V_{total} total volume

p_{total} suction pressure = total pressure of the suction mixture

p_s saturated vapor pressure of the operating liquid

From this relation follows that the gas portion transportable in the impeller cells or the maximum possible suction volume flow of the liquid ring vacuum pump grows with an increasing total pressure and a decreasing saturated vapor pressure of the ring liquid. Depending on the product, the saturated vapor pressure takes lower values at lower temperatures, too.

Owing to the pressure in the suction area of compressors remaining almost constant, the mean suction capacity of these machines increases regularly over the total pressure range. In the case of vacuum pumps, a larger quantity of fresh liquid enters the pump at lower suction pressures resulting in a larger increase in the suction volume flow due to better cooling. The characteristic pump curves and catalogue values of the manufacturers partly state the quantity of the operating liquid in dependence on the pressure ratio and the rotational speed of the pump.

3.5

The behavior of liquid ring vacuum pumps in case of liquid being carried simultaneously

Besides gases and gas-vapor mixtures, liquid ring vacuum pumps can also carry additional quantities of liquid. Depending on design and dimensions of the pump, the carried quantity is limited, as apart from the reduction of the suction capacity, an increase in the operating energy input occurs as well. Owing to this it is possible to feed liquids arising in the process to the pump without safety appliances on the suction side. In cases in which a continuous carrying of liquids is to be expected, the quantity of operating liquid fed to the pump via separate ports may be reduced. The carrying of liquids can occur by means of a liquid ring vacuum pump equipped with rigid or flexible discharge ports. Figure 3-33 shows the carried quantity of liquid in the ring chamber. In pump designs with rigid discharge ports without pressure relief holes, supercompression occurs as soon as the impeller cells have reached point B, since the gas and the liquid can leave the compression chamber only at point A, where the rigid discharge port begins. However, if the port plate is equipped with pressure relief holes in section B (C), gas and liquid can already leave the pump in this zone. Thus supercompression does not occur.

In machines with rigid discharge ports, the quantity of the carried liquid can be twice the amount of the nominal quantity of the operating liquid [3.6]. Designs with discharge ports allow the carrying of up to five times the value of the nominal quan-

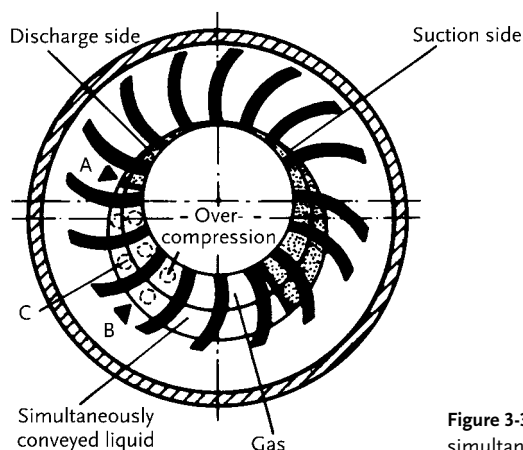


Figure 3-33. Liquid ring vacuum pump simultaneously carrying liquid

tity of operating liquid. Referring to the suction capacity of the vacuum pumps and depending on the pump design, the carried liquid quantity can be up to 5% of the sucked gas flow, without impairing the operational safety [3.18].

3.6

The carrying of contaminants

The robust liquid ring machines with their simple design can carry considerable quantities of contaminants, as the impeller glides without contact to the casing, being sealed by the ring liquid only. Specifically lighter solid particles are continuously expelled together with the operating liquid. With large dirt quantities arising, the contaminated operating liquid can be continuously carried out of the liquid ring. Through a steady or intermittent supply of or rinsing with clean operating liquid, over-contamination is effectively avoided. For heavily contaminated gases, it is common to install a separator or a filter on the suction side of the pump (see section 4.4.3). In such cases, the degree of contamination or flow resistance of these devices must be monitored by means of suitable measuring equipment, in order to avoid the operation at suction pressures that are too low.

3.7

The condensation effect

The suction capacity of liquid ring vacuum pumps also depends on the saturation state of the gas-vapor mixtures to be sucked and on the temperature difference between the process gas mixture and the operating liquid. If the saturation state of the gas-vapor mixture in the pump is reached or exceeded during the pumping process, a part of the vapor will condense. In this state, the cell chambers of the impeller blades work as an injection condenser. Schematically, for this state the liquid ring vacuum pump can be seen as a unit consisting of a displacement vacuum pump and an injection condenser (acc. to Beckmann [3.5]) (fig. 3-34).

Condensation occurs only when the vapor temperature of the sucked gas-vapor mixture is higher than the temperature of the operating liquid and when owing to cooling down and reaching of the saturation vapor pressure, condensation of the vapor portion occurs. However, if the suction temperature and the temperature of the liquid ring are identical, condensation does not occur.

The characteristic pump curves of the manufacturers are usually based on the suction of dry air at 20 °C and a water ring temperature of 15 °C. As soon as other conditions than those the values in the catalogue are based on exist, the actual pump suction capacity of these machines is to be determined according to the correction curves usually drawn up by the pump manufacturers as well.

Figure 3-35 shows the trend towards the change of the suction capacity of a liquid ring vacuum pump during the suction of vapors or gas-vapor mixtures compared to the suction of dry air over the suction pressure. Due to the condensation effect, an

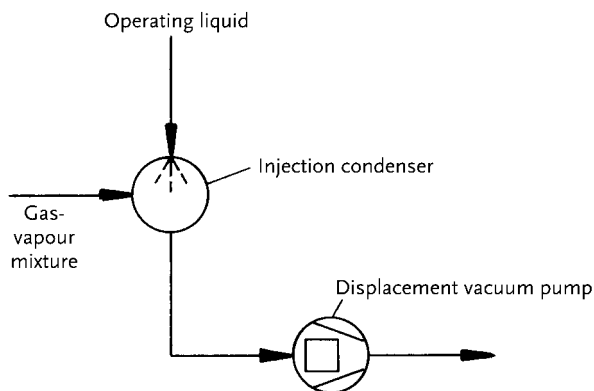


Figure 3-34. Alternative piping diagram of the liquid ring vacuum pump

increase in the suction capacity is achieved. Under favorable conditions the suction capacity for gas-air mixtures can adopt many times the value of the suction volume of dry gas.

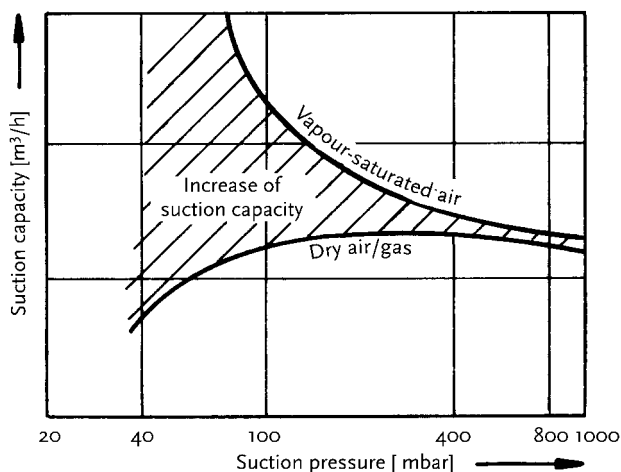


Figure 3-35. Effect on the suction capacity of a liquid ring vacuum machine by the suction mixture

With an already fully saturated mixture being sucked, evaporation of the operating liquid cannot occur and the suction capacity of the pump becomes larger by the proportion otherwise occupied by the evaporating operating liquid in the impeller cells. During the suction of vapors, a vapor balance will develop in the impeller cells, i. e. operating liquid in the impeller chamber will vaporize only to the extent possible at the existing ring liquid temperature and the prevailing suction pressure acc.

to the vapor pressure curve. The developing condensation effect depends on the following influential factors:

- on the suction pressure of the vacuum pump
- on the temperature and the saturation state of the sucked gas-vapor mixture
- on type and temperature of the operating liquid
- on the geometric ratios of impeller width and impeller diameter.

In Section 1, figure 1-35, a diagram for the determination of the actual suction capacity resulting from the condensation effect is given.

This behavior typical for the pump justifies the application of these devices especially for the discharge of humid gas mixtures and vapors.

An increase in the suction volume also occurs when the sucked gases dissolve in the operating liquid. For the determination of the size of a liquid ring vacuum pump, the condensation effect or the increase in suction capacity should always be taken into consideration in order to be able to select smaller pump sizes, if necessary.

3.8

Characteristic curves of liquid ring machines at different compression pressures and suction pressures

The characteristic curves of liquid ring vacuum pumps usually refer to compression pressures of 1013 mbar (abs) while the liquid ring compressors usually refer to suction pressures of 1013 mbar (abs). Both vacuum pumps and compressors can be economically operated at compression and suction pressures other than 1013 mbar (abs). Then the achievable suction data depend on the design and the size of the pump. Fig. 3-36 shows the typical behavior of a two-stage liquid ring vacuum pump, and in figure 3-37 the behavior of the volume flow and the power consumption of a single-stage compressor is depicted depending on compression and suction pressure.

3.9

The similarity law for liquid ring gas pumps

The performance of liquid ring machines is strongly influenced by the geometrical shape of the liquid ring. The outer chamber boundary of the liquid ring is deformed because of the Coriolis force acting in the relativity system and the pressure difference between the neighboring impeller cells. Therefore, the development of the ring surface depends on the pressure ratio existing at that time. An arithmetical recording of the liquid ring contour is possible acc. to Prager [3.9]. For the first approximation, a circle with its centre on the vertical axis of the pump casing can be selected. Decisive for the developing shape of the liquid ring, and with this for the suction volume and the efficiency of a liquid ring machine, are the rotational speed and the

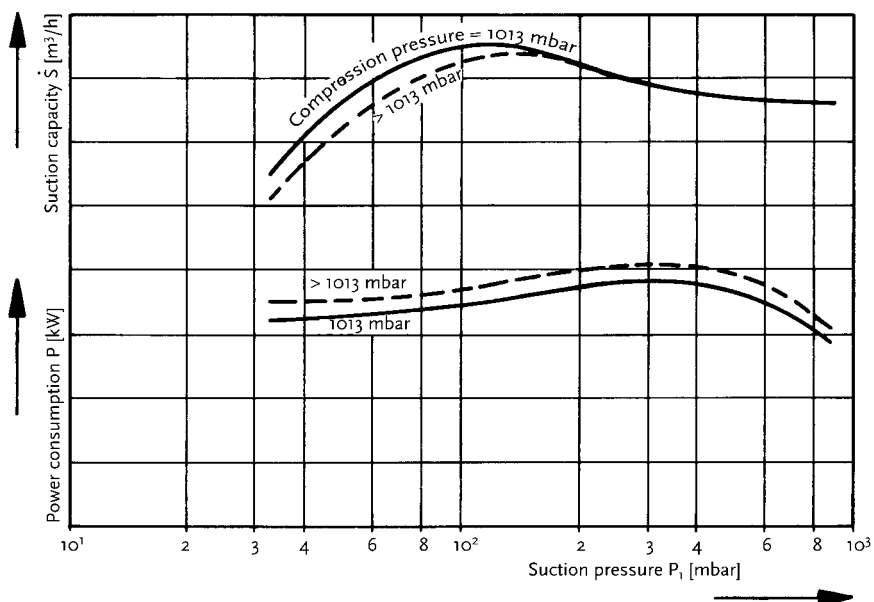


Figure 3-36. Operation characteristics of a two-stage liquid ring vacuum pump at different suction and discharge pressures (acc. to Sihi)

density of the operating liquid. The energy of the liquid ring and thereby the compression performance of the pump depend directly on these two factors.

In accordance with the similarity law [3.16], the compression processes run similarly, if acc. to the equation 3-20

$$k = \frac{p \cdot 2 \cdot g_N}{\rho_N \cdot u^2} = \text{constant.} \quad (3-20)$$

- p pressure, e. g. suction pressure [kg/m^2]
- ρ_N standard density of the operating liquid [kg/m^3]
- g_N standard gravity acceleration [m/s^2]
- u circumferential velocity of the impeller [m/s]

Moreover, the geometric similarity shall be maintained, too, and the ratio between the final gas compression pressure and the gas suction pressure shall remain constant.

Under this condition and due to already existing test characteristic curves and tables, data for operating conditions not yet known can be determined.

The suction capacity of vacuum pumps behaves almost directly proportional to the rotational speed. The power consumption changes on the average squared with the rotational speed ratio. At lower rotational speeds, it can be calculated with a power exponent of 1.5, at higher rotational speeds with the exponent 2.5.

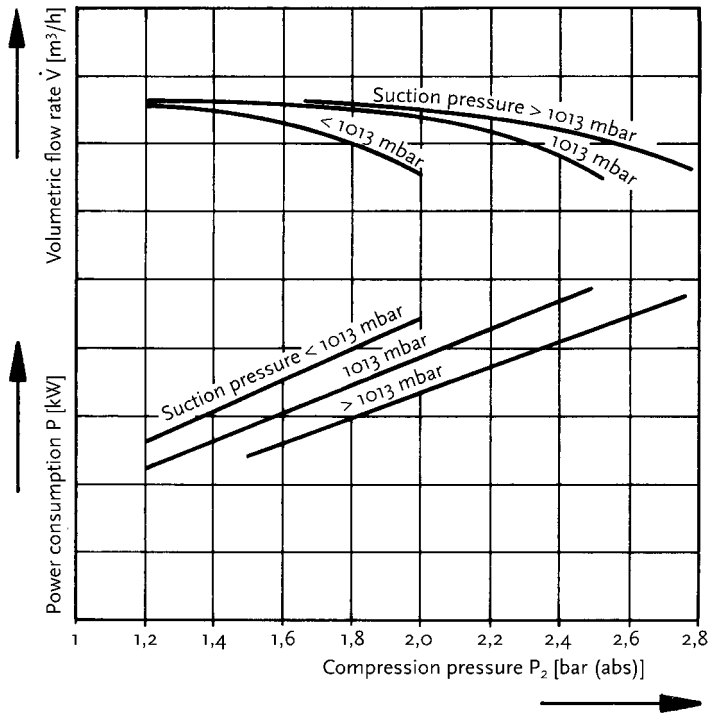


Figure 3-37. Volume flow rate and power consumption of a single-stage liquid ring compressor at different suction and discharge pressures (acc. to Sihi)

3.10

Pump performance and power consumption of liquid ring machines

The gas enclosed in the impeller cells of the liquid ring machine is practically compressed isothermally, since during the compression, the gas temperature changes only slightly. Due to the direct contact between liquid and gas, the compression heat is almost completely absorbed by the liquid. The isothermal compression performance can therefore be equated with the pump performance. The isothermal power consumption is the theoretical power consumption required for the compression of a gas from a certain suction pressure to a certain output pressure at constant temperature in a liquid ring vacuum pump. Theoretically, the liquid vapor possibly carried does not need to be compressed, since it condenses during the compression at constant partial pressure proportional to the decrease in volume so that only the developing condensate has to be evacuated.

The isothermal compression performance P_{is} of a liquid ring machine required for the compression of a gas can be determined acc. to equation (3-21) and (3-22) [3.16].

For vacuum pumps

$$P_{is} = \frac{10^{-3}}{27} \cdot p_a \cdot \dot{S} \cdot \ln \frac{p_e}{p_a} [\text{kW}] \quad (3-21)$$

P_{is} isothermal compression power [kW]
 \dot{S} suction capacity at suction pressure p_a [m³/h]
 p_a suction pressure Torr [mmHg (abs)]
 p_e compression pressure Torr [mmHg (abs)]
 $(10^{-3}/27)$ conversion constant

For compressors

$$P_{is} = 0.0278 \cdot p_a \cdot \dot{V} \cdot \ln \frac{p_e}{p_a} [\text{kW}] \quad (3-22)$$

P_{is} isothermal compression power [kW]
 \dot{V} suction volume at suction pressure p_a [m³/h]
 p_a suction pressure [bar (abs)]
 p_e compression pressure [bar (abs)]
 0.0278 conversion constant

The power input is the power consumption at the driving shaft of the liquid ring vacuum pump. The energy to be supplied to the coupling is larger than the effective performance by the losses arising in the pump, which are about 50% to 60% of the driving power [3.12]. They are hydraulic losses composed of:

- wall friction losses on the cylindrical casing shell
- wall friction losses on the lateral face surfaces
- turbulent exchange losses between the liquid flows inside and outside of the impeller.

According to [3.12.], the volumetric and thermodynamic losses amount to about 10 per cent of the driving power.

The total losses are taken into consideration by the isothermal coupling efficiency η_{is} . The isothermal efficiency is the ratio of isothermal power to the actual power consumption. The optimum isothermal efficiency $\eta_{is}=0.25$ to 0.40 is achievable at about 60 per cent of the absolute vacuum [3.13]. The upper limit value can be reached only at large volume flow rates and low end pressure.

Therefore, the power to be supplied to the pump driving shaft is

$$P = \frac{P_{is}}{\eta_{is}} \quad (3-23)$$

P driving power
 η_{is} isothermal coupling efficiency

The specific power is the ratio of power consumption to the suction volume flow.

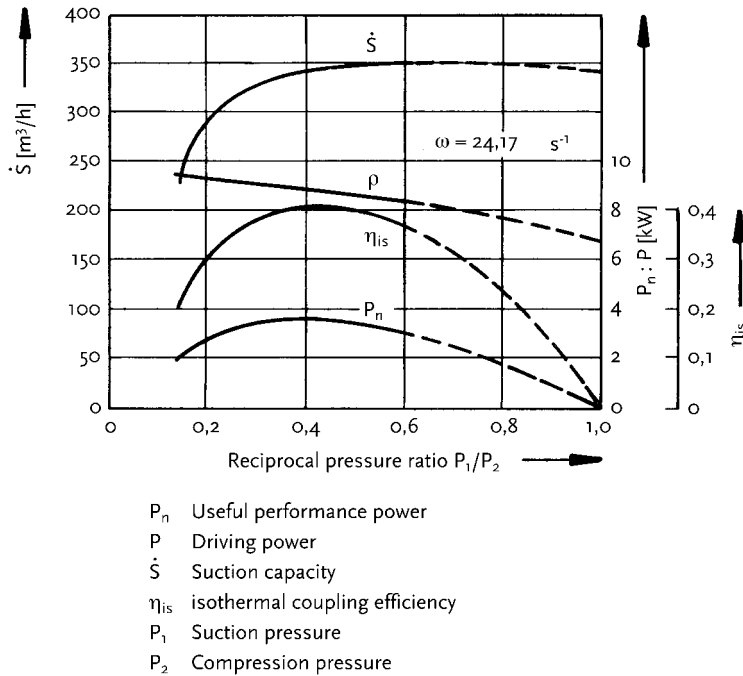


Figure 3-38. Characteristic curves of a single-stage water ring vacuum pump, applicable for a operating water inlet temperature of 15 °C, dry air suction at 20 °C and discharge pressure of 1013 mbar

3.10.1

Characteristic curves of liquid ring vacuum pumps and compressors

Figure 3-40 shows the characteristic curves of a two-chamber water ring vacuum pump operated as vacuum pump and as compressor. Operated as compressor, the pump develops higher isothermal efficiency and higher consumption of the operating liquid compared to the operation as vacuum pump.

The gas quantity pumped by a liquid ring gas pump mainly depends on the suction pressure and the final pressure.

In the vacuum range, lower suction pressures cause smaller suction volumes, because the partial pressure of the vapor of the operating liquid prevents the complete filling of the impeller cells with the gas to be sucked.

If the liquid ring gas pump is operated as compressor, the suction volume decreases with increasing final pressures, as the isothermal compression work becomes too large compared to the ring liquid quantity.

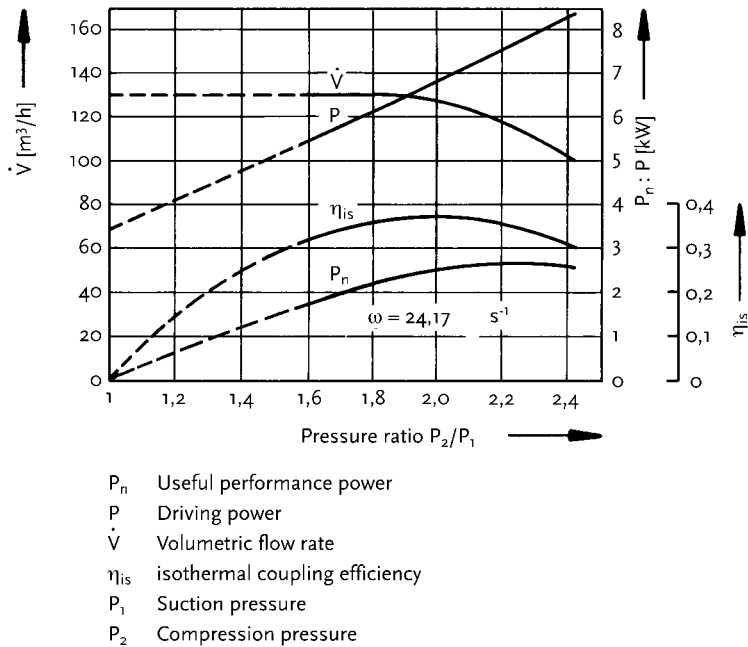


Figure 3-39. Characteristic curves of a single-stage water ring air-compressor, applicable for an operating water inlet temperature of 15 °C, dry air compression at 20 °C and suction pressure of 1013 mbar

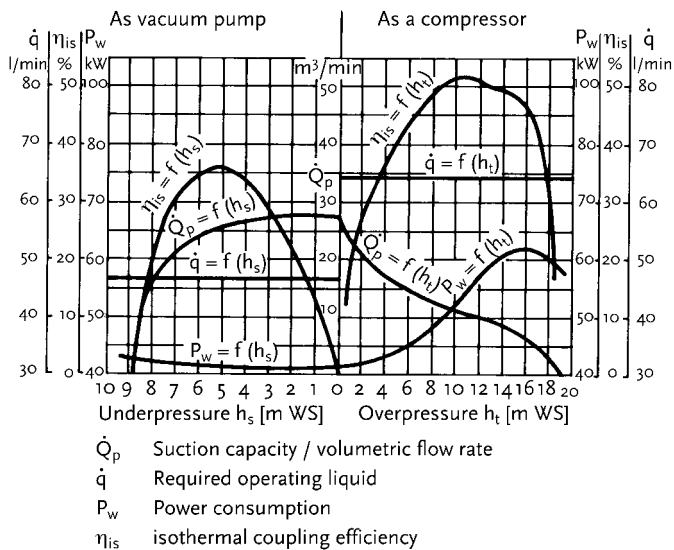


Figure 3-40. Characteristic curves of a two-chamber water ring vacuum pump operated as a vacuum pump and as a compressor (acc. to P. Zworski)

3.11

Cavitation

At low suction pressures the application range of the liquid ring vacuum pumps is restricted by the vapor pressure of the operating liquid used. If it falls below this physical limit, the ring liquid begins to boil, and the impeller cells will be filled with the respective steam. For the most part, the design final pressure of a liquid ring machine is not achieved, as the vapor pressure of the operating liquid is higher. If the suction pressure of the machine approaches the vapor pressure of the applied ring liquid, the suction capacity of the pump reduces to zero, because the impeller cell chambers on the suction side are filled with vapor. With the developing vapor bubbles being compressed during the continuing rotation of the impeller cells towards the pressure side of the pump, the bubbles will implode there. This implosion takes place at very high frequencies and pressures on the smallest surface. It becomes noticeable through a percussive sound and jolts on the pump and is called cavitation. The machines are exposed to very high and varying stresses. The cavitation attack leads to material fatigue and to damages on the impeller blades, impeller hubs and port plates. Therefore, operation under cavitation must be avoided in any case.

At a suction pressure of 30 mbar to 40 mbar, a vapor portion of about 50 per cent of the liquid ring can be permitted as a reference value for a liquid ring vacuum pump. In this case:

$$\frac{V_G}{V_{\text{vap}}} = 1$$

and acc. to Dalton's Law, the limit value can be calculated for the suction pressure [3.19].

$$\frac{V_1}{V_2} = \frac{p_1}{p_2} = \frac{p_{\text{total}} - p_2}{p_2} \quad (1-152)$$

$$\frac{V_G}{V_{\text{vap}}} = \frac{p_G}{p_{\text{vap}}} = \frac{p_{\text{total}} - p_s}{p_s} = 1 \quad (1-170)$$

From

$$\frac{p_{\text{total}} - p_s}{p_s} = 1$$

the following results

$$p_{\text{total}} = 2 \cdot p_s \quad (3-24)$$

V_G	gas volume
V_{vap}	vapor volume
p_{total}	suction pressure = total pressure
p_s	saturated vapor pressure

According to this, the suction pressures of liquid ring vacuum pumps can be reduced to pressures which do not fall below double the value of the saturated vapor pressure of the applied operating liquid, without endangering operation safety.

For a cavitation-free operation, suction pressures are permissible as absolute limiting value, which are at least 16 mbar higher than the vapor pressure of the operating liquid used at ring liquid temperature..

If the temperature of the operating liquid, and with this its vapor pressure rises, the cavitation safety diminishes. Cavitation also occurs in hydraulic flow machines, e. g. in pumps, turbines or ship's propellers.

3.12

Cavitation protection

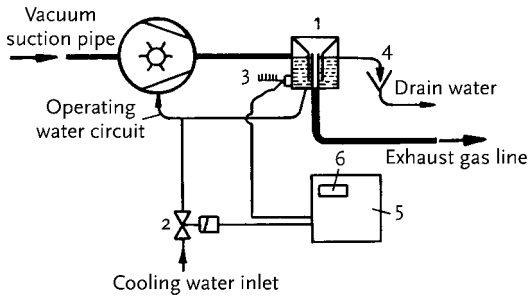
Cavitation can be avoided by suitable measures. The application of especially cavitation-resistant materials may reduce the effects of cavitation on the pump components, cavitation, however, cannot be prevented. In order to avert the occurrence of cavitation, the volume flow of non-condensable gases according to the characteristic curve at the lowest suction pressure of the pump should always be available to the liquid ring vacuum pump. The pump can also be supplied with compressed process gas of the liquid separator on the pressure side over a control valve to the suction side. Moreover, there is a possibility to discharge atmospheric air or gases into the suction side of the pump by means of manual valves, spring-loaded vacuum safety valves, so-called vacuum breakers, or by pressure-regulated fittings. If the gas injectors are installed in the pump combination, then, owing to the design of the ejectors, a gas flow can always pour into the pump over the open motive gas pipe, thus preventing cavitation.

To avoid cavitation damage, the liquid ring vacuum pump can be equipped with a cavitation protection. For this purpose, the working chamber is supplied with gas ballast on the pressure side. The suction capacity is practically not influenced.

A further possibility to prevent cavitation is shown in figure 3-41. The liquid ring vacuum pump is provided with a liquid temperature regulation device and potential vortex separator with integrated cavitation protection valve [3.20]. On the one hand, in this system the water consumption referred to the pre-set suction pressure is reduced to the minimum by means of the temperature regulator in order to avoid cavitation. At the same time, however, cavitation is reliably prevented by the cavitation protection valve in all operating states, i. e. also at the smallest suction pressures and operating liquid quantities.

As is known, the suction capacity of liquid ring vacuum pumps depends very much on the kind and temperature of the operating liquid.

With this arrangement, the ballast gas is not supplied on the suction side, but added to the by-pass of the operating liquid through the cavitation protection valve installed in the separator. Through throttling in the pipe of the operating liquid a variable gas quantity from the gas chamber of the separator is supplied to the pump in the bypass. The system can be controlled manually, semi-automatically or fully-



- 1 Potential vortex separator with cavitation protection valve
- 2 Fresh water valve
- 3 Temperature measurement transmitter for the operating liquid temperature
- 4 Overflow
- 5 Switchboard
- 6 Temperature indicator

Figure 3-41. Diagram of a water quantity regulation device with cavitation protection (acc. to Loewe)

automatically. In the two last-mentioned modes the temperature of the operating water and thus even the vacuum can be kept constant. Owing to the possibility to select the optimum temperature of the operating liquid, this operating mode has the advantage of minimal coolant consumption.

3.13

Gas ejector in combination with the liquid ring vacuum pump

The lowest suction pressures achievable with a liquid ring vacuum pump largely depend on the boiling behavior of the operating liquid used. Besides water, other ring liquids can be used depending on the engineering process. Hydrocarbons, solvents and non-viscous oils, for example, can be considered. If water is used as a ring liquid at a temperature of 15 °C, the corresponding saturated vapor pressure is 17.04 mbar; this would be the theoretically possible operating pressure. However, at this pressure the discharge of gas is not possible, because the liquid is in a boiling state and the impeller cells are filled with saturated water vapor. The lowest possible suction pressure, at which the discharge of the gas with water of 15 °C as ring liquid is practically possible, ranges at about 33 mbar. Lower pressures can be achieved by using higher boiling liquids.

The expansion of the suction pressure range to smaller pressures can be obtained by installing a vacuum pump of another discharge principle on the suction side. Therefore, liquid ring vacuum pumps are often combined with gas ejectors, so that

the pump can be operated at suction pressures of e.g. from 33 mbar to about 5 mbar, at a water ring temperature of 15 °C.

The gas ejector is installed directly on the suction side of the vacuum pump. The ejector can be arranged either vertically, diagonally or horizontally, this has no effect on the suction capacity. The installation of a gas ejector on the suction side does not change the operating behavior of the liquid ring vacuum pump either. The installation of a gas ejector on the suction side of a liquid ring vacuum pump is shown in figure 3-42.

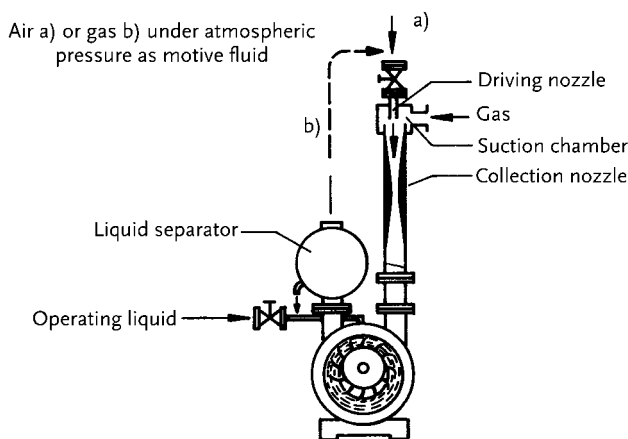


Figure 3-42. Liquid ring vacuum pump with a gas ejector

3.13.1

Operating range of a vacuum pump with gas ejector

The operating range of a vacuum pump with a gas ejector at a water ring temperature of 15 °C is shown in figure 3-43.

The motive fluid for the gas ejector is atmospheric air, the gas to be pumped itself or any other non-condensable gas under atmospheric pressure. With a gas ejector being operated at altitudes above 1000 m, the atmospheric pressure available there is not sufficient to be used as motive fluid pressure. In such cases, the gas needs to be compressed first to a pressure of about 1013 mbar, e.g. in the liquid separator, to be then used as motive fluid. When entering the suction chamber of the gas ejector, the motive fluid is accelerated by the driving nozzle, and the gas to be sucked is thereby discharged into the collection nozzle (s. section 2.5.7). In the diffuser of the collection nozzle, a pressure increase to about 50 mbar to 90 mbar occurs. Thus, the vacuum pump has to convey both the motive fluid and the gas to be pumped. Nevertheless, in the suction pressure range between 50 mbar and 5 mbar with an injector being installed on the suction side a significantly larger suction volume is achieved with the vacuum pump (see fig. 3- 43).

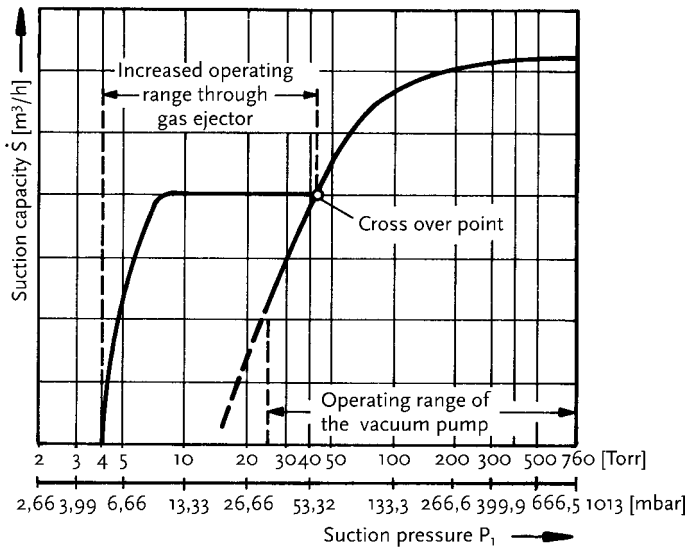


Figure 3-43. Operating range of a liquid ring vacuum pump combined with a gas ejector

The characteristic curves in figure 3-44 show the influence of the inlet temperature of the operating liquid of the vacuum pump on the suction capacity of the gas ejector. At inlet temperatures of the water above the reference temperature of 20 °C, a reduction of the suction capacity of the ejector and of the final pressure is to be expected. In order to exploit the full suction capacity of the ejector, the vacuum pump of the next larger size should be installed in these cases.

As shown in figure 3-44, the cross-over point from the operation of the vacuum pump without ejector to the operating mode with injector is influenced by the inlet temperature of the operating liquid. The dimensions of the gas ejectors are designed for certain volume flows with their sizes being adapted to the suction capacity of the pumps by the manufacturers of vacuum pumps. The suction and counter-pressure curves of jet pumps are usually drawn up as mass flow curves. Compared to the characteristic curves of the liquid ring vacuum pumps, in the case of jet pumps even suction volume flow curves are used.

The curves on the left of the cross-over points apply to the vacuum pump with gas ejector; the curves on the right do only apply to the vacuum pump.

3.13.2

Operation mode of gas ejectors

During volume or coarse evacuation of containers and columns, the vacuum pump combined with a gas ejector is operated within a suction pressure range in which the suction capacity of the liquid ring vacuum pump alone is higher than that of the combination. The necking in the collection nozzle acts as throttle for the volume

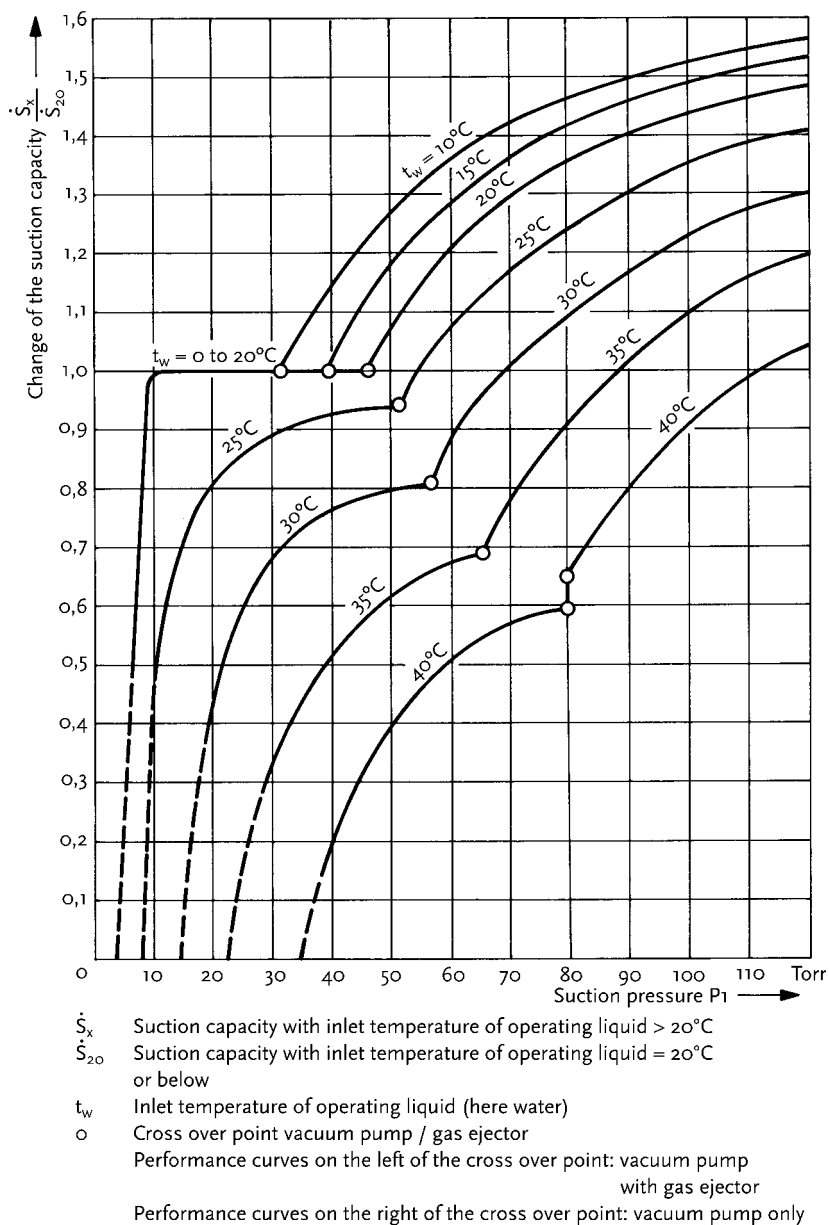


Figure 3-44. Influence of the operating liquid inlet temperature of the liquid ring vacuum pump on the conveying characteristics of the gas ejector (acc. to Siemens)

flow pumped by the liquid ring vacuum pump in this pressure range and the additionally accruing motive fluid flow. This can be resolved by cutting off the motive fluid supply by means of a pressure-regulated valve. A further improvement and, above all, a reduction of the evacuation time in the upper coarse vacuum range can be achieved through the installation of a suction-pressure-regulated bypass. In this case, the gas ejector is bypassed at the beginning of the evacuation, i.e. the total volume flow is supplied to the liquid ring vacuum pump with smaller losses on the suction side via a bypass with a large cross-section (figure 3-45). In this operating state, the motive fluid supply to the gas ejector is cut off e.g. by a magnetic valve. Only after a suction pressure of about 50-90 mbar is achieved – depending on the operation conditions – the supply of the motive air is released by a vacuum switch via magnetic valve, the bypass is closed by the diaphragm valve and thus the operation mode is changed over to the gas ejector installed on the suction side of the pump.

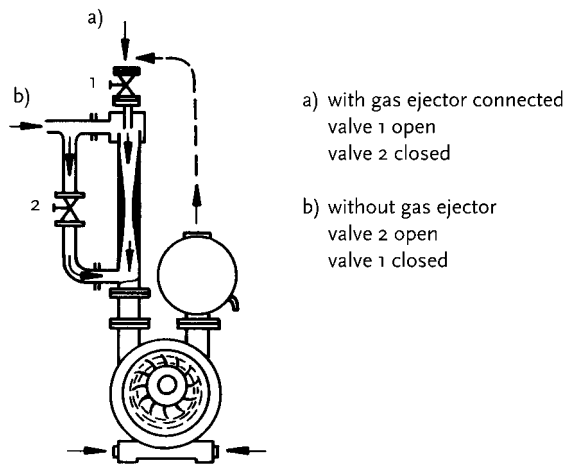


Figure 3-45. Circuit diagram of a liquid ring vacuum pump with a gas ejector provided with a bypass pipe

Figure 3-46 shows the schematic layout of a pressure-dependent automatic control for the operation with and without a gas ejector.

With the gas ejector being installed on the suction side, it is also possible to close the suction side of the gas ejector while the liquid ring vacuum pump is in operation, since the motive fluid pouring into the gas ejector prevents the cavitation otherwise occurring. Thus, the suitable combination provided, the pump equipped with an ejector can be run up to a suction volume of zero. .

Taking the motive fluid for the gas ejector from the liquid separator installed on the pressure side of the liquid ring vacuum pump, additional environmental damages are prevented, especially in the case of toxic or otherwise dangerous media. In process engineering, gas jet vacuum pumps are also used for the mixing of two gas flows. A typical application for this is, for example, the use of inert motive jet gas in

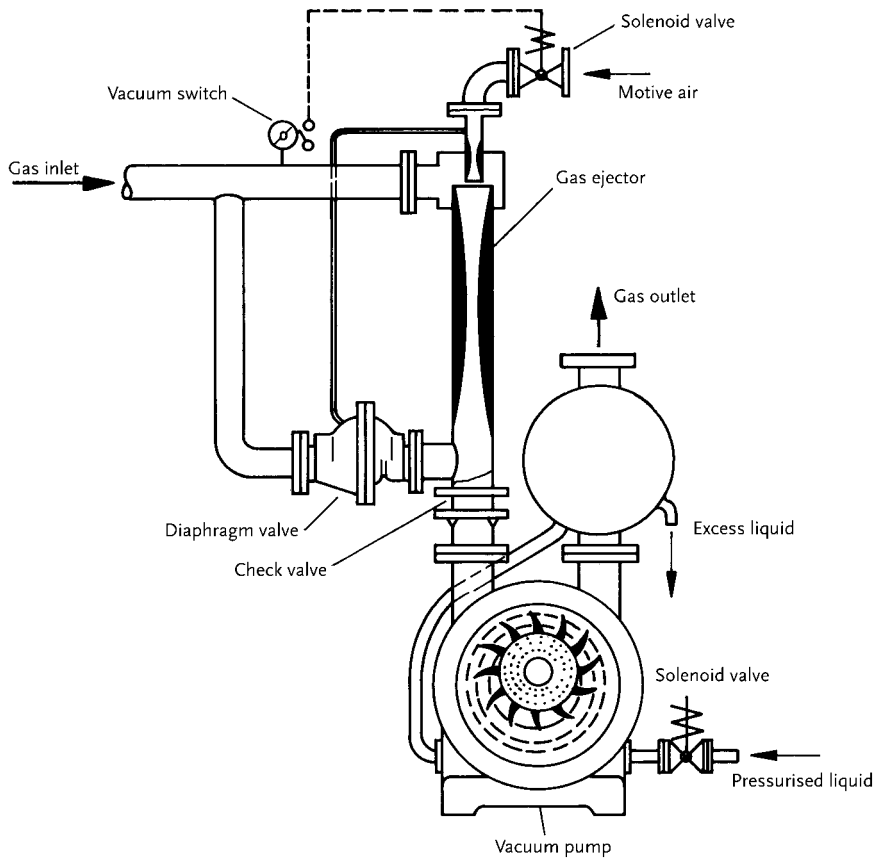


Figure 3-46. Liquid ring vacuum pump with a gas ejector and automatic switch-over equipment

order to prevent the concentration and ignition of an explosive gas to be pumped during the compression process in the liquid ring pump.

3.14

Operating modes, supply of operating liquid

The operating liquid required for liquid ring machines does not only take on the energy transfer and the sealing of the impeller cells, but also the transport of most of the heat quantity generated in the pump. This requires the pump casing to be filled with operating liquid up to a certain level during a standstill to enable the development of a concentric liquid ring during the rotation that takes on this function. Owing to the fact that liquid is always discharged together with the gas, this liquid needs to be replaced continuously, so that the blades are always sufficiently immersed in the liquid ring and isothermal compression is guaranteed. In axial

flow machines, the gas to be pumped coming from the suction port directed upwards enters the impeller cells through the laterally installed channels via the suction port of the port plate. The compressed gas is supplied in the same way to the pressure nozzle via the discharge port.

The operating liquid is also supplied to the machine through channels in the lateral parts arranged in a way that due to the underpressure in the pump the liquid ring vacuum pump can suck its operating liquid independently. As the liquid ring pump allows of operation not only with water but with regard to the engineering processes to be carried out even with other inorganic or organic fluids, different kinds of operating liquid supply and circulation exist in practice. Generally, three different methods are distinguished, i.e. operation without recirculation of the liquid (fresh liquid operation, cooling mode), operation with liquid recirculation (combined operation, saving mode) and operation with closed liquid circulation (circulating liquid operation, circulation mode). The kind of mode to be applied has to be chosen according to the type of the available operating liquid, to the engineering process, to the required suction pressure, to the coolant and its temperature, as well as to the possible requirements regarding the disposal of accruing matters both on the process gas side and on the operating liquid side. The operation of these machines and correspondingly optimized vacuum systems enables the realization of measures regarding the fulfillment of conditions for environmental protection even from an economic point of view. The different kinds of operating liquid circulation and some designs depending on the pump size and customary in practice are described in the following.

3.14.1

Operation without liquid recirculation (fresh liquid operation)

With this mode of operation, only fresh liquid from the supply network or an existing supply system (e.g. re-cooling water) is supplied to the vacuum pump. This brings about a particularly intensive cooling of the pump and the process gas, as it is required e.g. for the generation of low suction pressures. A liquid separator is not absolutely necessary; however, in large pumps, the separator for the discharge of gas and liquid should be kept up. It is practical to supply the pump with fresh liquid at a constant pressure that lies between 0.20 bar and 0.30 bar above the pressure at the discharge port, via a pressure regulating valve. For vacuum pumps with a high consumption of fresh liquid it is favorable to supply the liquid by means of a fresh liquid tank fed by a feed control. In this case, the vacuum pump sucks the required quantity of operating liquid independently from the fresh liquid tank being under atmospheric pressure. The sucked liquid quantity increases with the decreasing suction pressure of the pump, i.e. with the increasing pressure difference. The fresh liquid fed to the pump will not be reused in this mode of operation which is usually applied when economical water is available as operating liquid and can be used for the gases and vapors to be discharged. The supplied liquid flow absorbs the total heat accruing in the pump. The rising of the ring liquid temperature in the pump can be determined acc. to equation 3-28 [3.14].

$$\dot{Q}_{\text{tot}} = \dot{F}B \cdot \rho \cdot c_p \cdot \Delta T \quad (3-25)$$

$$\Delta T = T_A - T_B \quad (3-26)$$

$$T_B = T_F \quad (3-27)$$

$$\Delta T = T_A - T_F = \frac{\dot{Q}_{\text{tot}}}{\dot{F}B \cdot \rho \cdot c_p} \quad (3-28)$$

\dot{Q}_{tot}	heat flow to be evacuated from the pump [kJ/h]
T_A	outlet temperature of gas and liquid at the pump discharge port [K]
T_B	inlet temperature of operating liquid [K]
T_F	temperature of the supplied liquid [K]
$\dot{F}B$	required flow rate of operating liquid [m^3/h]
ρ	density of operating liquid [kg/m^3]
c_p	specific heat capacity of operating liquid [$\text{kJ}/(\text{kg} \cdot \text{K})$]
ΔT	temperature increase of operating liquid in the pump [K]

The mode of operation without the recirculation of the liquid is schematically shown in figure 3-47 [3.14]. Figures 3-48, 3-49 and 3-50 show examples for different designs.

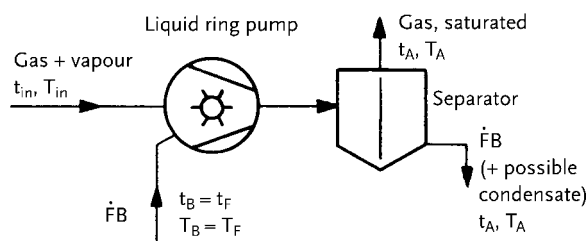
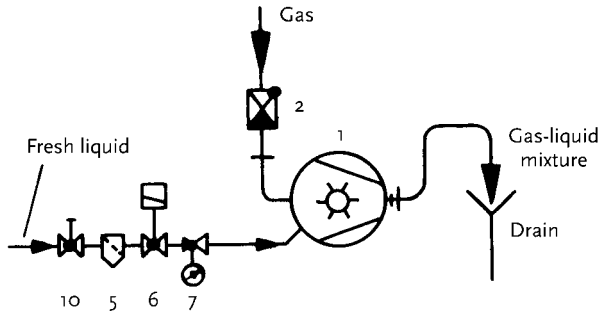


Figure 3-47. Principle scheme of the operation without liquid recirculation



Legend to Fig. 3-48, 3-49, 3-50

- 1 Liquid ring vacuum pump
- 2 Ball check valve
- 3 Liquid build-up separator
- 4 Liquid separator
- 5 Strainer
- 6 Solenoid valve
- 7 Pressure reducer with pressure gauge
- 8 Fresh liquid tank
- 9 Float valve
- 10 Shut-off valve

Figure 3-48. Fresh liquid operation (small pumps)

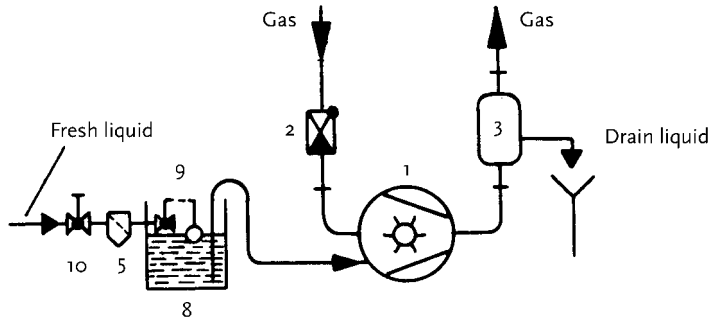


Figure 3-49. Fresh liquid operation (medium-sized pumps)

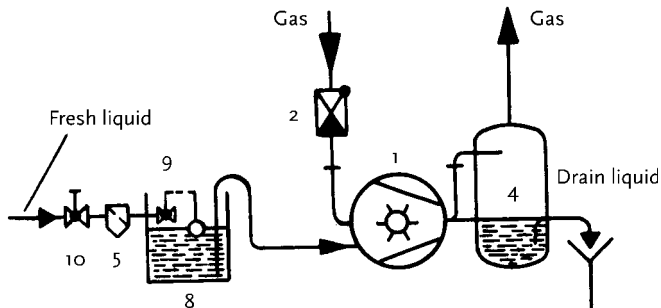


Figure 3-50. Fresh liquid operation (large pumps)

3.14.2

Operation with liquid recirculation (combined operation)

This mode can be described as the standard mode for water as operating liquid. The liquid ring pump sucks the non-cooled circulating liquid independently from the liquid separator. For the limitation of the volume flow rate, an orifice can be integrated in the by-pass between the separator and the pump. The liquid sucked from the separator will only be fed with the quantity of cooled liquid or fresh water from the supply system that is required for recooling. The quantity of liquid continuously discharged by the separator corresponds to the quantity of fresh liquid supplied to the system with developing condensates being discharged, too. Any possible loss of liquid due to evaporation has to be compensated in the system.

The quantity of fresh liquid required in this system can be considerably reduced, since a large part of non-cooled liquid from the liquid separator is led back to the pump. Here, the main characteristic is the fact that the fresh liquid consumption can be limited the best possible way to the quantity of heat to be discharged by the pump. The liquid and gas flows and their temperatures are shown in figure 3-51. The temperature of the operating water T_B is the mixed temperature of the two temperatures of the supplied fresh water flow \dot{K}_B and the liquid flow recirculated from the separator with the temperature T_A . Depending on the temperatures chosen, the ratio of fresh water to the required operating liquid flow can be calculated according to equation 3-29 [3.14]. The basic mode of operation with liquid recirculation is shown in figure 3-51 [3.14]. The figures 3-52, 3-53 and 3-54 show examples for different designs.

$$\frac{\dot{K}_B}{\dot{F}_B} = \frac{T_A - T_B}{T_A - T_F} \quad (3-29)$$

\dot{K}_B supplied fresh water [m^3/h]

\dot{F}_B required operating liquid flow [m^3/h]

T_A temperature of the recirculated liquid flow = outlet temperature at the discharge port of the pump [K]

T_B inlet temperature of the operating water [K]

T_F temperature of the supplied fresh water [K]

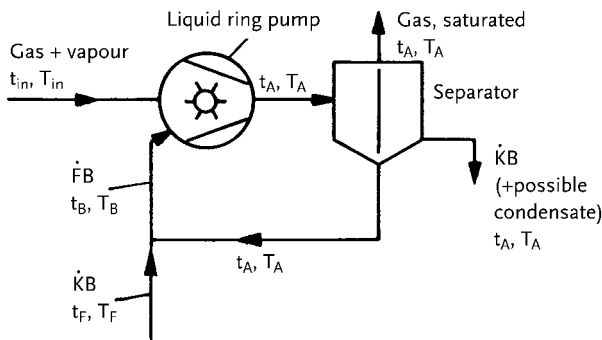
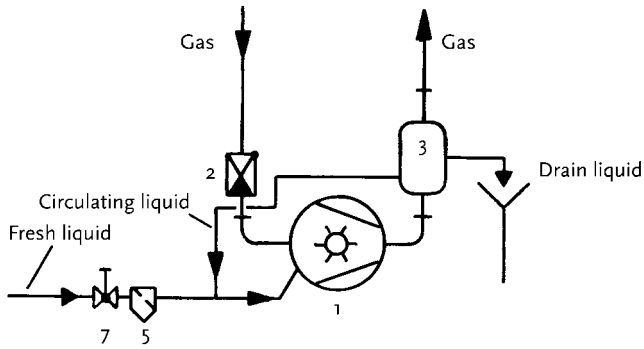


Figure 3-51. Principle scheme of the operation with liquid recirculation



Legend to Fig. 3-52, 3-53, 3-54

- 1 Liquid ring vacuum pump
- 2 Ball check valve
- 3 Liquid build-up separator
- 4 Liquid separator
- 5 Strainer
- 6 Solenoid valve
- 7 Shut-off valve
- 8 Liquid separator in vacuum package units
- 9 Thermostatic control valve

Figure 3-52. Combined operation (small and midsize pumps)

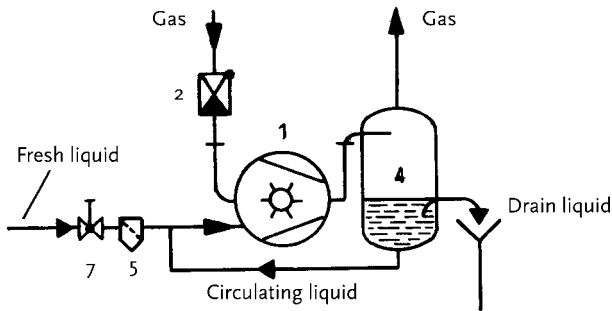


Figure 3-53. Combined operation (large pumps)

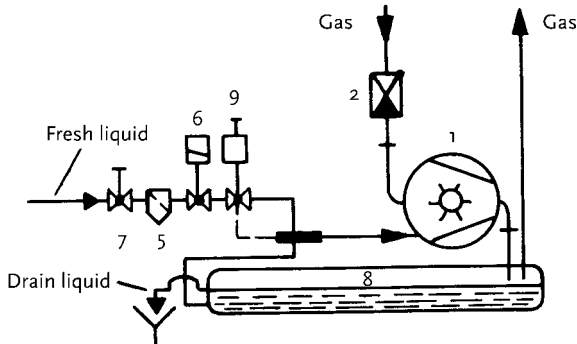


Figure 3-54. Combined operation, package unit with temperature control (small and medium-sized pumps)

3.14.3

Operation with closed circulation (circulating liquid operation)

In this mode of operation, the total operating liquid separated from gas or vapor by the liquid separator is led back into the pump while a heat exchanger provides for the indirect and constant cooling of the operating liquid. Via heat exchanger all accruing heat is removed from the system. In the heat exchanger, the operating liquid is usually led around the pipes. The flow resistance on the operating liquid side should be as low as possible, usually however max. 20 % of the available pressure difference between suction port and discharge port of the vacuum pump. In the case of higher flow losses, a circulation pump can be installed in order to overcome the pressure in the operating liquid circulation.

When dry gases are pumped it is possible that the dynamic liquid loss caused by the gas flow must be fed to the system again. If owing to the compression of condensable vapors condensate occurs it can be discharged at the overflow of the separator. The separator is also designed as separating vessel in case of operating liquid and condensate being different substances. Then the condensate will be discharged separately from the operating liquid. In order to avoid concentration or contamination in the circulation of the operating liquid caused by possible chemical reactions, pure operating liquid can be continuously fed to the pump or to the separator. The supply of fresh liquid can also occur via nozzles on the suction side (see section 5.3.3). A liquid quantity corresponding to the quantity of the pure operating liquid supplied leaves the system by the overflow of the separator. The special feature of this method is that the sucked gases and vapors as well as the operating liquid do not come into contact with the cooling medium of the heat exchanger. Therefore, the question of how to dispose the cooling medium does not occur. Due to the isothermal compression at low gas temperatures, the gases or vapors discharged into the atmosphere are loaded with only little masses of noxious substances. The installation of an immission cooler on the pressure side of the pump which is operated at low temperatures enables the adherence to the legally admissible emission values even at the discharge side of the pump. The closed circulation mode is applied in the chemical industry and in cases where higher demands on leak tightness and environmental protection must be fulfilled.

Fig. 3-55 shows the basic mode for the operation with closed circulation [3.14]. Further examples of plants with circulating liquid operation are given in fig. 3-56 and 3-57.

Fig. 3-58 depicts the design of a system with Booster pump.

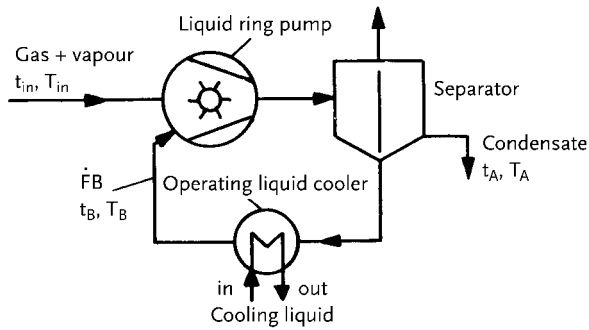


Figure 3-55 Principle scheme for operation with closed liquid circulation

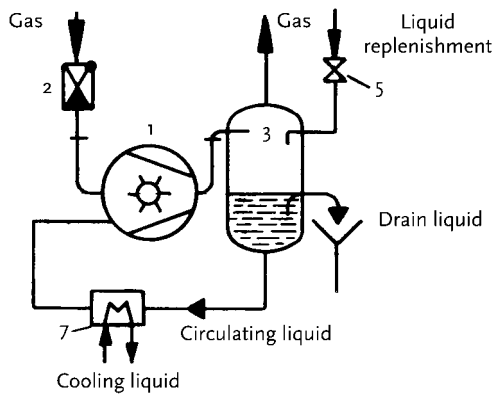


Figure 3-56. Operation method with return circulation (all pump sizes)

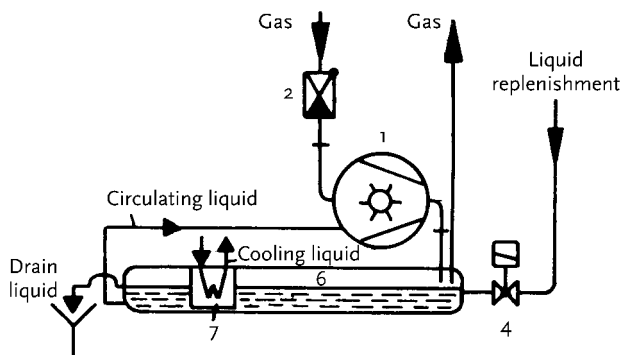


Figure 3-57. Operation method with return circulation (package unit)

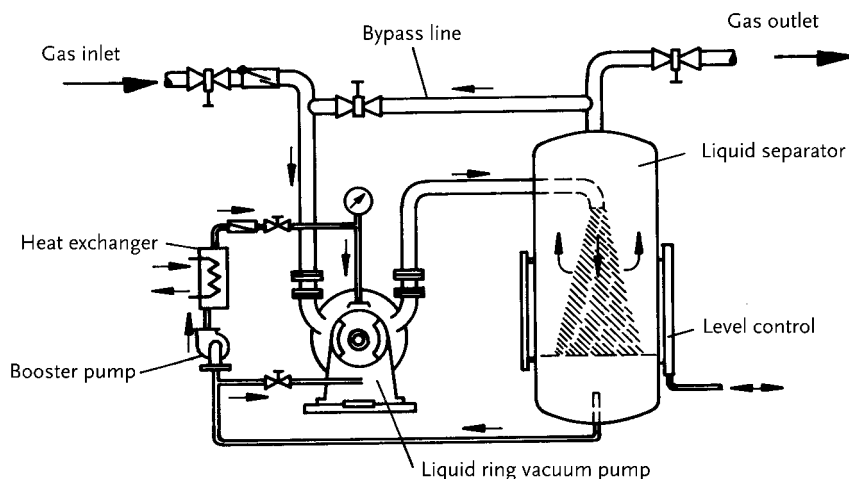


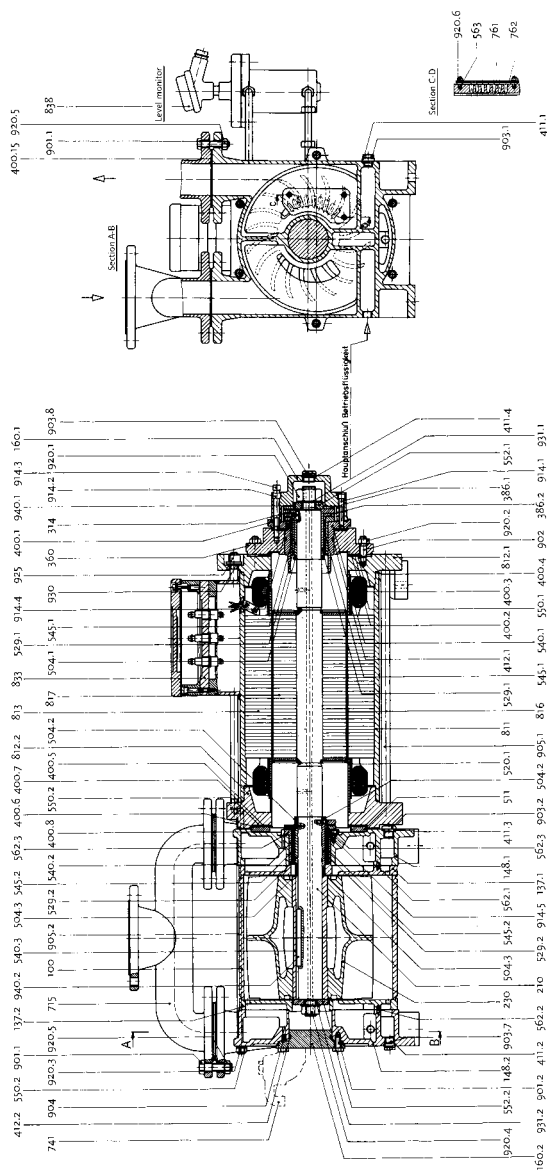
Figure 3-58. Scheme of operation with return circulation and with circulation pump

3.15

Materials for liquid ring machines

The liquid ring pumps belong to the more robust machines with hardly any parts inside the pump that glide on each other. This allows practically any material to be used for manufacturing that is customary in mechanical and process engineering. For liquid ring machines, the kind of the operating liquid must be chosen according to the kind of gas to be pumped. Besides water, oils, acids, lye, hydrocarbons, etc. are used as ring liquid. The materials to be used have to be determined according to the gases to be pumped and the operating liquid applied. Usually, corrosion resistance is of major importance when choosing the appropriate material. Economic aspects such as operation costs, purchasing costs, possibilities and costs of repair work, replacement possibilities, interchangeability, thermal resistance and mechanical characteristics must be taken into consideration, too. Liquid ring machines can be manufactured of grey cast, ductile cast iron, cast steel, chemically resistant chromium-nickel steels, high nickel alloys, titanium, bronze, polypropylene, PVDF, Noryl (PPO), Ryton (PPS), ceramics and stoneware. Combinations of the different materials are possible as well. Components of these pumps are also available with coatings of hard rubber, plastic, enamel or special lacquers for the protection against corrosion and erosion. For machines usually used in vacuum technology, this variety of designs with different materials only exists with liquid ring pumps and gas-/vapor jet pumps. For this reason, they belong to the most frequently used pumps of the chemical, pharmaceutical and food industries.

Table 3-2 gives an overview of some components and their standard material design of a single-stage liquid ring vacuum pump with a canned motor according to figure 3-59.



Item	Description	Item	Description	Item	Description	Item	Description
148	End shield	381	Slide ring	562.1	Cylinder pin	905	Tie bolt
148.2	End shield	381.2	Slide ring	811	Motor casing	905.2	Tie bolt
137.1	Port plate	400.1	Flat gasket	812.1	Motor casing cover	914.1	Hexagon socket head screw
137.2	Port plate	400.2	Flat gasket	812.2	Motor casing cover	914.2	Hexagon socket head screw
130	Casing	400.3	Flat gasket	813	Stator laminations	914.3	Hexagon socket head screw
220	Impeller	400.4	Flat gasket	814	Stator tube	914.4	Hexagon socket head screw
741	Automatic drain valve	400.5	Flat gasket	815	Regulating bush	914.5	Hexagon socket head screw
741.2	Automatic drain valve	400.6	Flat gasket	816	Regulating bush	914.6	Hexagon socket head screw
750	Catch plate	400.7	Flat gasket	817	Rotor	914.7	Hexagon socket head screw
750.2	Catch plate	400.8	Flat gasket	818	Terminal box	914.8	Hexagon socket head screw
565	Threaded bolt	400.9	Flat gasket	819	Level monitor	914.9	Hexagon socket head screw
775	Manifold	401.1	Flat gasket	901.1	Hexagon screw	915.1	Hexagon nut
137.2	Port plate	411.1	Seal ring	901.2	Hexagon screw	915.2	Hexagon nut
137.2	Port plate	411.2	Seal ring	902	Stud and	915.3	Hexagon nut
160.1	Cover plate	411.3	Seal ring	903.1	Locking screw	915.4	Hexagon nut
160.2	Cover plate	411.4	Seal ring	903.2	Locking screw	915.5	Hexagon nut
270	Shaft	412	O-ring	903.3	Locking screw	915.6	Hexagon nut
314	Slide ring	412.2	O-ring	903.4	Locking screw	915.7	Hexagon nut
350	Bearing cover	412.3	O-ring	903.5	Locking screw	915.8	Hexagon nut
		412.4	O-ring	903.6	Locking screw	915.9	Hexagon nut
		412.5	O-ring	903.7	Locking screw	916.1	Hexagon nut
		412.6	O-ring	903.8	Locking screw	916.2	Hexagon nut
		412.7	O-ring	903.9	Locking screw	916.3	Hexagon nut
		412.8	O-ring	904	Grab screw	916.4	Hexagon nut
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Table 3-2. Overview of materials for a liquid ring vacuum pump with a canned motor (acc. to Lederle-Hermetic)

Part definition	Combination of materials: grey-cast iron	Combination of materials: chromium-nickel steel
casing	sheet steel (UST137-2/1.0036)	stainless steel (XlOCrNiMoTi 1810/1.4571)
impeller	ductile cast iron (GGG-40/0.7040)	stainless cast steel (G-X6CrNiMol810/1.4408)
shaft	steel (St50-2/1.0050)	stainless steel (XlOCrNiMoTi 1810/1.4571)
port plate	grey cast iron (GG-20/0.6020)	stainless cast steel (G-X6CrNiMol 810/1.4408)
end shields	grey cast iron (GG-20/0.6020)	stainless cast steel (G-X6CrNiMol810/1.4408)
flexible discharge port	polytetrafluoroethylene (PTFE)	polytetrafluoroethylene (PTFE)
bearing	1.4571/carbon	1.4571/carbon
stator tube	Hastelloy C	Hastelloy C

Figures 3-60, 3-61, 3-62a and 3-62b show vacuum pumps manufactured of special materials.

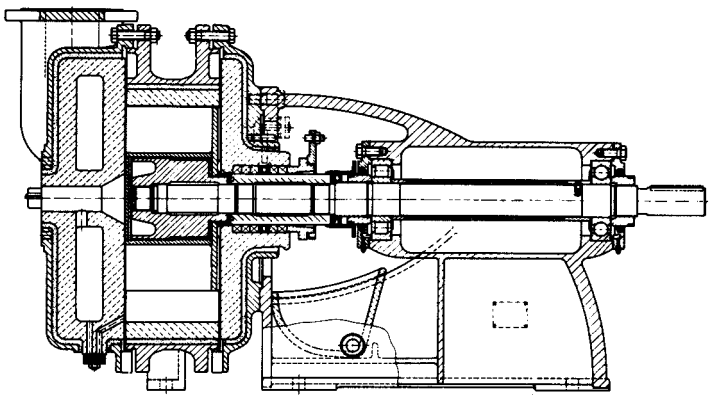


Figure 3-60. Liquid ring vacuum pump made of stoneware (acc. to Friedrichsfeld)

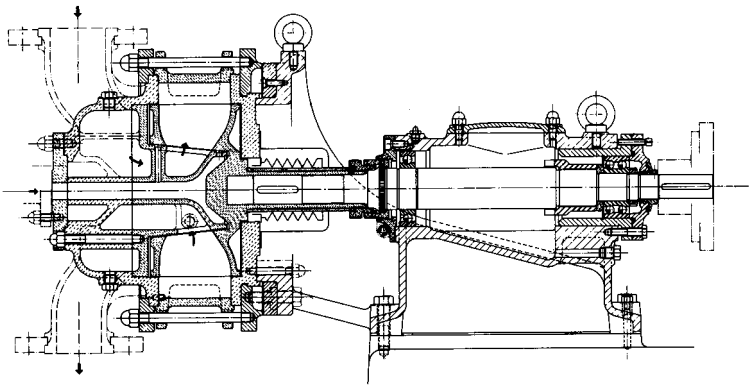


Figure 3-61. Liquid ring vacuum pump made of titanium (acc. to Garo)

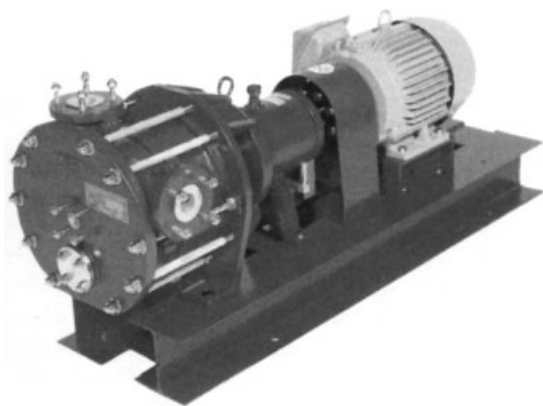


Figure 3-62a. Single-stage ceramic liquid ring vacuum pump with steel reinforced components with driving motor (acc. to NGK)

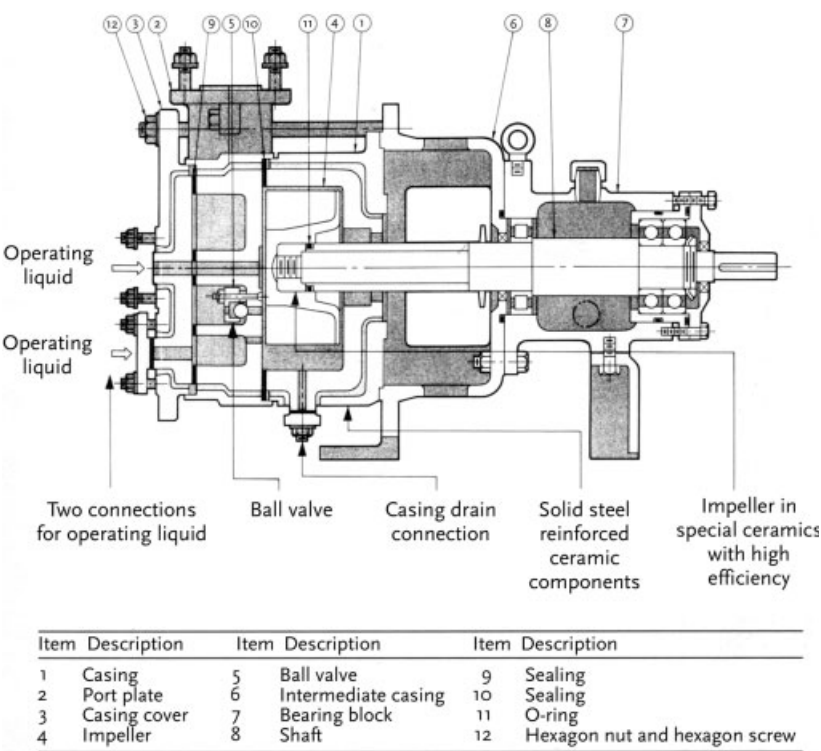


Figure 3-62b. Cross-section of a single-stage ceramic liquid ring vacuum pump with mechanical shaft seal (acc. to NGK)

3.16

Sealing of liquid ring vacuum pumps and compressors

Liquid ring vacuum pumps distinguish between static and dynamic seals. For the sealing of static sealing points on the casing parts liquid seals based on polyvinyl acetate, polyisobutene, and epoxy resins are used. O-rings and flat gaskets on the basis of nitrile rubber, fluorine elastomers, polytetrafluoroethylene, ethylene-propylene rubber and chloroprene rubber are used, too. Where dynamic sealing points, i.e. shaft ducts need to be sealed conventional sealing elements such as stuffing box packing or single or double-acting mechanical seals are used. Stuffing box shaft seals are partly designed with double packing and sealing liquid connection.

The sealing materials have to be chosen with regard to the actual pressures and temperatures of the existing process gases and ring liquids. In particular, the chemical resistance of the seals against the substances occurring on the gas side and operating liquid side of the pump has to be verified. This is particularly important for machines used in the chemical, pharmaceutical and nuclear engineering.

Figures 3-63, 3-64, 3-65 and 3-66 provide an overview of conventional shaft seals.

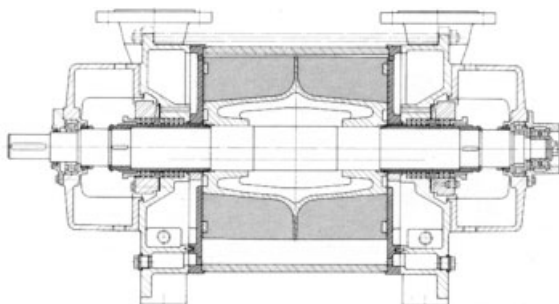


Figure 3-63. Liquid ring machine with stuffing box shaft seal (acc. to Siemens)

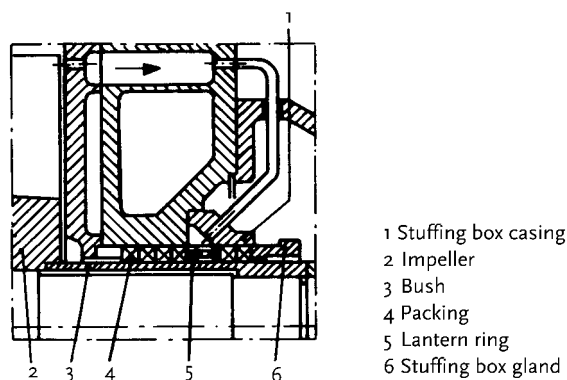


Figure 3-64. Stuffing box shaft seal with internal seal liquid circulation (acc. to Siemens)

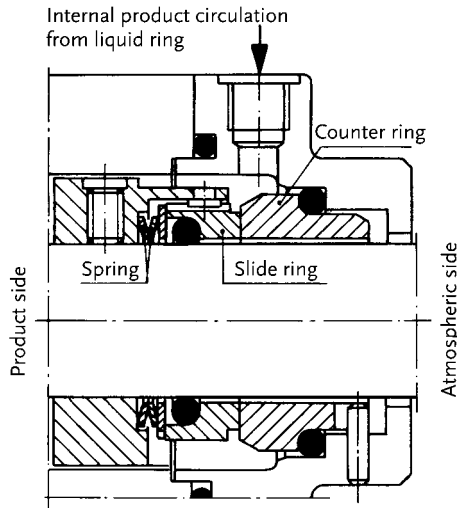


Figure 3-65. Single-acting mechanical seal (acc. to Burgmann)

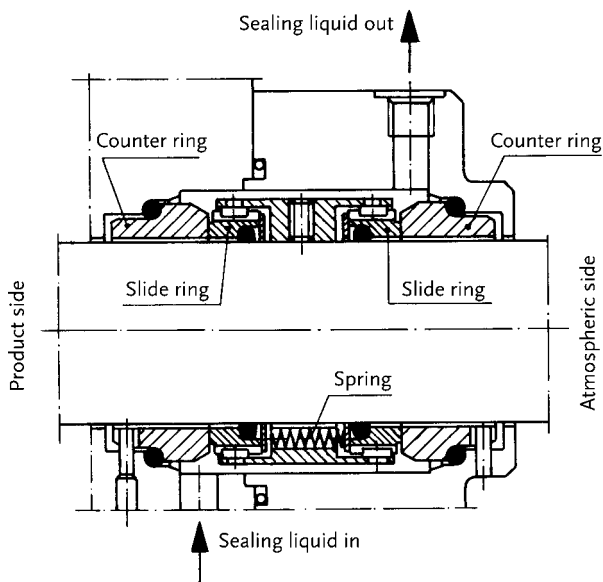


Figure 3-66. Double-acting mechanical seal (acc. to Burgmann)

Vacuum technology demands the leakage of all seals to be kept small both in operation and at a standstill of the machine. An alternative to vacuum pumps designed with conventional dynamic sealings is the liquid ring machine in hermetic design. There are two possibilities: Equipment of the pump with permanent magnetic coupling or the application of a canned motor. The transmission of the torque

as well as the sealing of the shafts towards the atmosphere is performed simultaneously by these driving systems. The seal liquid flow and the necessary supply systems partly required for stuffing box shaft seals or mechanical shaft seals are not required in these designs. Hermetically enclosed designs are recommended in cases where absolutely leak-free and maintenance-free operation is demanded. With machines of this design, the lowest possible leakage rates or even zero-rates are achieved which enables an environmentally friendly operation when discharging toxic, carcinogenic and explosive gases or using hazardous operating liquids. Section 3.17.2 gives a more detailed description of the hermetic sealing systems.

3.17

Drives for liquid ring machines

3.17.1

Electric motor drive

The drive of liquid ring machines mainly occurs by means of two, four or six pole low voltage asynchronous motors with squirrel-cages. For larger machines with driving powers of more than about 150 kW, high voltage motors with squirrel-cages are used. For large pumps with low operating speeds, gears or V-belt drives are interposed which allows the adjustment of the suction capacity to the existing operational data by choosing the respective transmission. The flywheel torques of these machines and the load torques during the start-up are so small that an overload of the motors does not occur. Usually the start-up is carried out by directly starting the three-phase a.c. motor. If the ideal conditions for this are not given at the moment of the start-up, for example, due to the pump casing being overflowed with operating liquid, especially for large machines it is appropriate to arrange a hydrodynamic coupling between the motor and the pump or to provide for starting resistances or frequency control for the start-up of induction motors. In contrast to piston compressors, oscillating inertia forces do not occur in these machines working with symmetrical torque. Therefore, special measures for the anchoring of these devices are not required. Liquid ring vacuum pumps can be equipped with an automatic drain valve for the limitation of the fill level (fig. 3-67). This leads to a start-up of the pump with reduced ring liquid volume at smaller load torques and a smaller engine power can be planned, i.e. according to the characteristic curve of the pump at the

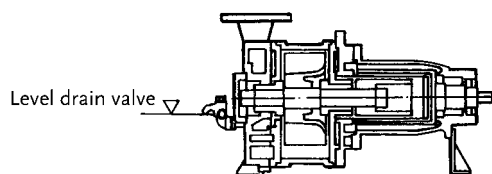


Figure 3-67. Liquid ring vacuum pump with an automatic level drain valve (acc. to Siemens)

design speed. Usually the design speed of the motor is chosen with an allowance of about 10% to 15%, referring to the maximal input capacity of the pump.

Under these conditions, the machine reaches only approx. 30% of the maximum load torque after the start-up time t_a at design speed (fig. 3-68). In this operating mode, the vacuum pump automatically sucks the liquid volume that is still required. The period of time until the liquid ring is completely developed and the maximal load torque is reached, corresponds to three times the start-up time. The total suction capacity is then available. After the closing-down of the pump the operating liquid is automatically discharged to the overflow level of the drain valve.

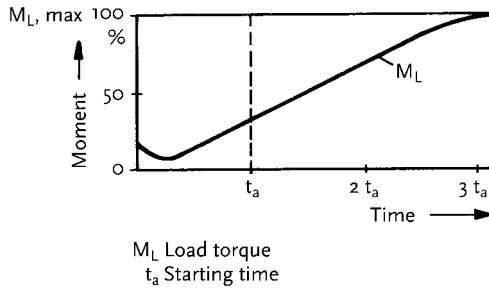


Figure 3-68. Starting-up torque time curve of a liquid ring vacuum pump at a reduced liquid fill level (acc. to Siemens)

For the optimal design of the drive unit, the torque curve of both the machine and the driving machine should be taken into consideration for the calculation. The driving power must be determined in a way that a sufficient acceleration torque is available during the start-up over the whole speed range and the three-phase A.C. motor will not be overloaded in the operating point (see figure 3-69). When driving motors are used which are not oversized they can be operated in their best range of efficiency, thus contributing to a better economicalness of the vacuum pump system.

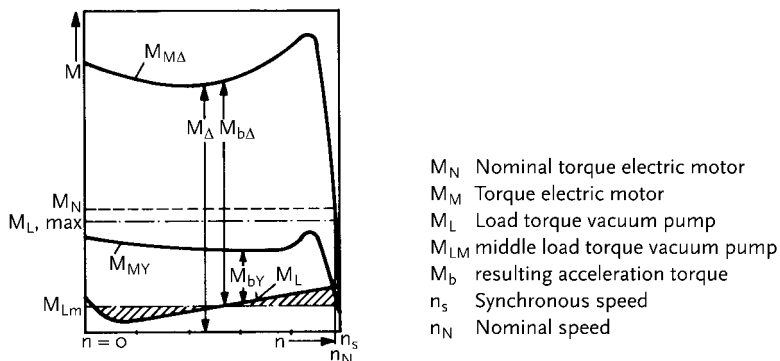


Figure 3-69. Starting-up torque-speed curve of a liquid ring vacuum pump (acc. to Siemens)

3.17.2

Hermetic drive systems

The demand of our modern industrial society regarding more protection of health, workplace and environment require particular efforts in manufacturing, processing and disposal of dangerous products. Environmentally friendly process flows are to ensure that exhaust air, water and soil are kept free of hazardous substances. For the assessment of the existing safety risks, the maximum permissible limiting values for processing and handling of dangerous substances from the MAK values lists issued annually (max. concentration at workplace) have to be consulted. Further standard values can be taken from the "TA Luft" according to the Federal Immission Act Law. The same goes for the directives concerning noise emission given in "TA Lärm". With certain fluids and the suction of air, a dangerous explosive mixture can develop in vacuum pumps. In "open constructions", i.e. those equipped with stuffing box or mechanical shaft seals, emission of pollutants may occur which is not in accordance with the legal regulations. The solution of this problem can only be the hermetic sealing of such units by renouncing a drive shaft leading out of the pressure casing. Then the torque must be transferred to the rotor either by permanent magnetic or electromagnetic means. Machines of this type contribute considerably to an increase in safety in the factories and a reduction of environmental damages. The complete leak tightness of these pumps becomes clear when it is necessary to pump highly toxic, carcinogenic or radioactive gases and/or vapors where leakage of the operating liquid can by no means be tolerated.

Here an extract from substances which are hazardous, detrimental to health and environmentally harmful and their vaporous and gaseous compounds is given:

Benzol, coal oils, vinylchloride, hydrazine, ethylene oxide, ethylamine, dimethylsulfate, phosgene, bromine, chlorine, fluorine, hydrogen sulphide, acrylic acid, aniline, formaldehyde, acrylic acid, hydrogen arsenide, nitrobenzene, phenol, cresols etc.

Further advantages of hermetic drive systems are the good vacuums that can be achieved and maintained even in a stationary machine without sealing liquid equipment and with a compression guaranteed to be free of oil and grease

However, the permanent magnet coupling and the electromagnetic canned motor drive are not equally matched regarding their safety standards, so that it is necessary to choose between the two systems considering the gases to be pumped and the operating liquids applied. In the case of dangerous operating liquids or the discharge of highly toxic and explosive gases, the canned motor drive (fig. 3-77) should be preferred, since compared to the permanent magnet coupling (fig. 3-71), the canned motor disposes of a double sealing jacket – can and stator casing – separating the fluid from the atmosphere.

Machines with canned motors cause less noise emissions than conventionally driven vacuum pumps or pumps equipped with magnet couplings, because noise resulting from ball bearings, motor fans and the coupling do not occur. With the application of these aggregates the demand for less noise pollution can be fulfilled. In table 3-3, the advantages and disadvantages of the permanent magnetic coupling and the canned motor are compared [3.21].

Tab. 3-3. System comparison of permanent magnet couplings and canned motors

Argument	Permanent magnet coupling (with single wall system)	Canned motor
Safety in case of average	lower, as only one sealing jacket (can tube) between the fluid and the atmosphere exists. If the can tube is damaged, the liquid escapes directly into the atmosphere.	Higher, two sealing jackets, the can and the motor casing exist. If the can breaks, the motor casing continues to seal hermetically.
Application range (kW)	smaller: magnet drive is limited by the existing gyrating masses and machine sizes.	wider the canned motor can be considered a normal three-phase motor and can be dimensioned accordingly.
Viscosity range	lower:	higher:
Total efficiency	same	same
Temperature limits for application	approx. -10 to $+400$ °C (with temperature bridge).	-120 to $+500$ °C
Heatability	only pump and the front can tube can be heated.	a full heating of the pump and the canned motor is possible.
System pressure	lower: with increasing wall thickness of the can tube (due to higher system pressures), the transmissible capacity decreases.	higher: the achievable system pressures are independent from the thickness of the can (0.5-1 mm). The current upper limit of the system pressure is at 1200 bar
Variation possibilities regarding fluids	Performance capacities, limits regarding viscosity, temperature and pressure restrict the range of application.	Performance capacities, viscosity, temperature and system pressures allow a wide application range
Protection against explosion	Directives 94/9/EG ATEX 100a legal conditions do not exist. In the interest of the operator, however, level control should be carried out, as dry running damages the pump	is required: legal regulations require pumps in explosion-prone areas to be equipped with a level and temperature monitoring device. Directives 94/9/EG ATEX 100a
Repairs of driving motor	better: the motor (normal three-phase motor) can always be replaced with the pump remaining installed in the pipework.	worse: this driving motor is a special motor requiring the disassembly of the pumping device.
Discharge of solid matters	worse	better: special versions enable suspension discharge with solid matter portion of up to 10%
Starting-up problems	exist	none
Installation possibilities	Dimensions and a required base-frame restrict the possibilities of installation.	The device can be installed horizontally and vertically in niches and equipment.

Drive by permanent magnet coupling. The hydraulics of liquid ring vacuum pumps with magnetic drive corresponds to the design and output of conventional designs. These machines do not have any kind of shaft seals. The drive connection is a magnetic coupling with a can tube sealing the pump part hermetically from the drive. This prevents the escape of the gases and vapors to be pumped and of the operating liquid (figures 3-70, 3-71, and 3-72). With machines of this design, leakage rates smaller than $1 \cdot 10^{-5}$ [(mbar · l)/s] can be maintained.

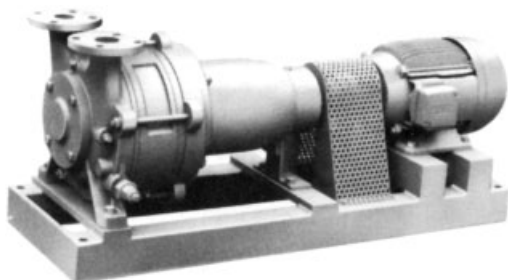


Figure 3-70. Hermetical liquid ring vacuum pump with permanent magnetic coupling (factory photo of Lederle-Hermetic)

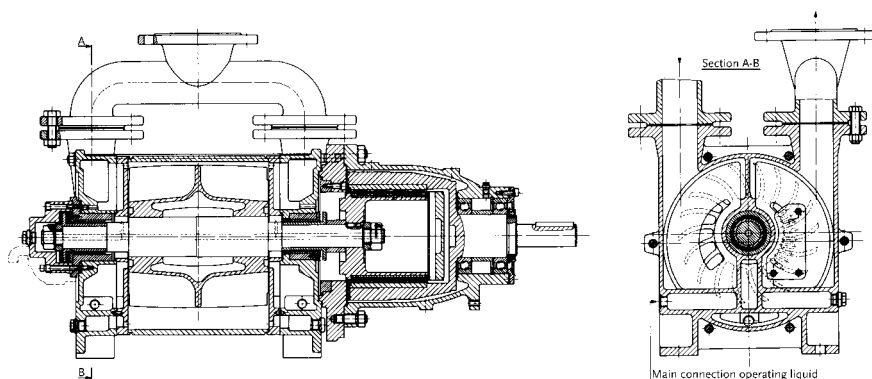


Figure 3-71. Hermetical liquid ring vacuum pump, single-stage, double-flow design with permanent magnetic coupling (factory drawing of Lederle-Hermetic)

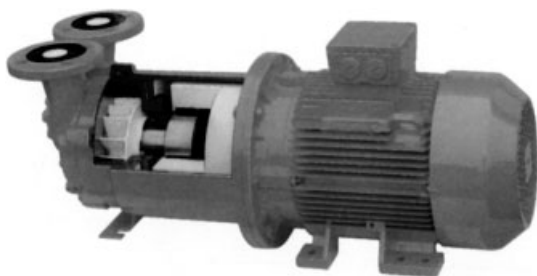


Figure 3-72. Quarter section of a hermetical liquid ring vacuum pump, single-stage, single-flow design with permanent magnetic coupling (factory photo of Lederle-Hermetic)

The permanent magnets applied are based on samarium-cobalt compounds (SmCO_5). They are powerful and are magnetized for a working life. They feature a high stability against demagnetizing effects which may occur e.g. during assembly and removing of the rotor or when exceeding the maximum transmissible torques (fig. 3-73). Both outer and inner runners of the magnetic coupling are rigid, i.e. slip-free connected with each other through the magnetic field, so that the rotor of the pump runs synchronously to the outer drive and thereby also to the driving motor. As hermetic sealing element, the permanent magnet coupling has also the advantage of an overload coupling. Owing to the hermetic system, these pumps/compressors are equipped with internal bearings for the uptake of the runner weight and the radial forces. Axial forces as occurring on the single-flow impellers are balanced hydraulically. For this purpose, the operating liquid is taken from the pressure range of the impeller and led to the backside of the impeller over the corresponding bore-holes. In the case of double-flow impeller blades hydraulic thrust equalization occurs. To achieve high vacuums and good efficiencies, the minimization of the lateral bearing clearance between the port plate and the impeller is necessary. This problem is solved by an appropriate adjustment device (figure 3-74) which balances arising residual axial thrusts at the same time [3.23].

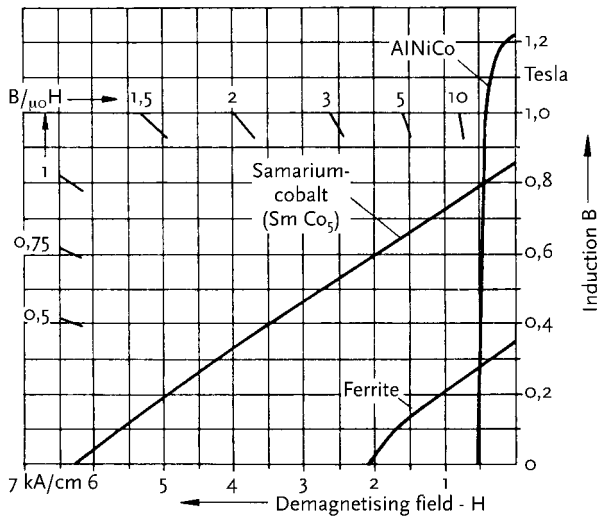


Figure 3-73. Demagnetization curve of the samarium-cobalt magnets compared with AlNiCo and ferrite magnets [3.22]

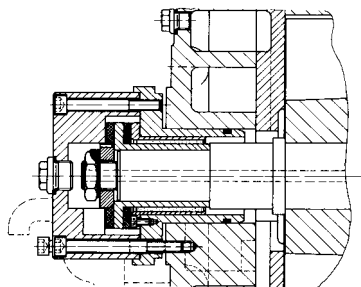


Figure 3-74. Casing-impeller clearance adjusting device (factory drawing of Lederle-Hermetic)

Drive by canned motor. Analogous to the rotor drive by means of the magnetic coupling, an electromagnetic direct drive with the help of a three-phase canned motor is possible (figures 3-75, 3-76, 3-77, 3-78).

The cooling flow necessary for the removal of the dissipated heat of the motor is taken from holes in the casing port on the motor side at the level of the impeller outlet, and led back to the suction area of the impeller, over the rotor through the plain bearings distant from the pump and through the hollow bore of the motor-pump shaft. With such pumps/compressors being used in explosion-prone areas, the motors must be in accordance with the respective explosion safety regulations. The relevant legal regulations require the rotor chamber of the canned motor to be always filled with the operating liquid and the temperature to be monitored. For this purpose, a hermetic level indicator and a temperature monitoring device is installed on the pump/compressor, which releases the start only if the requirements concerning the complete filling of the rotor chamber and the permissible temperature are fulfilled (figures 3-75, 3-79, 3-80).

Canned motors are manufactured and used in the explosion protection classes E Ex de II C T₁–T₆. The advantages of the canned motor drives are their good starting torques and overload capacities – an important demand especially during the start-up. Pump and compressor units of such a design are very compact as the motor is flanged directly on the hydraulics. The adjustment possibilities described



Figure 3-75. Hermetical liquid ring vacuum pump with canned motor (factory photo of Lederle-Hermetic)



Figure 3-76. Quarter section of a hermetical liquid ring vacuum pump, single-stage, single-flow design with canned motor (factory photo of Lederle-Hermetic)

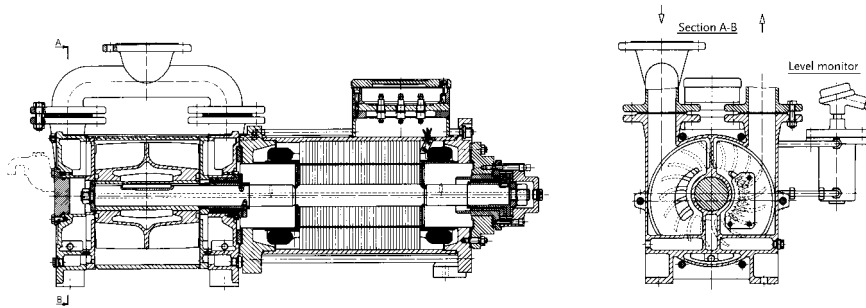


Figure 3-77. Hermetical liquid ring vacuum pump, single-stage, double-flow design, with canned motor (factory drawing of Lederle-Hermetic)



Figure 3-78. Hermetic DUO-Vac liquid ring machine with canned motor (factory photo of Lederle-Hermetic)

before (figure 3-74) can be laid out on the motor or the pump side. The design with canned motor drive meets the highest safety requirements demanded, for example, from nuclear power plants. Such vacuum pumps/compressors remain sealed even in case of average. The total rate of leakage of machines with double canned motor is less than $1 \cdot 10^{-8} \text{ [(mbar} \cdot \text{l)/s]}$.

A pump unit consisting of two liquid ring vacuum pumps with centrally arranged three-phase canned motors is shown in figure 3-78. With pumps of this weight and space saving design, twice the suction capacity of the normal designs is achieved at high rotational speed.

3.17.3

Explosion protection on canned motor machines according to the European Standard “EN”

As far as canned motors have to be operated in explosion-protected areas, legal regulations laid down in a national and European set of rules and regulations are applicable. In the Federal Republic of Germany, their compliance is controlled by the Physikalisch-Technische Bundesanstalt (PTB) in Braunschweig (Germany).

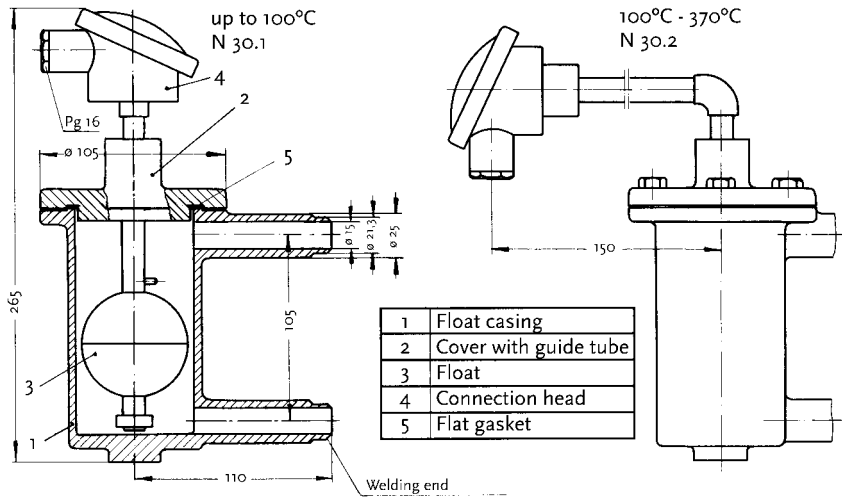
In the test certificates of the PTB, the category “Special Conditions” contains conditions that have to be observed for the operation of such pumps in explosions-prone business premises.

These conditions are:

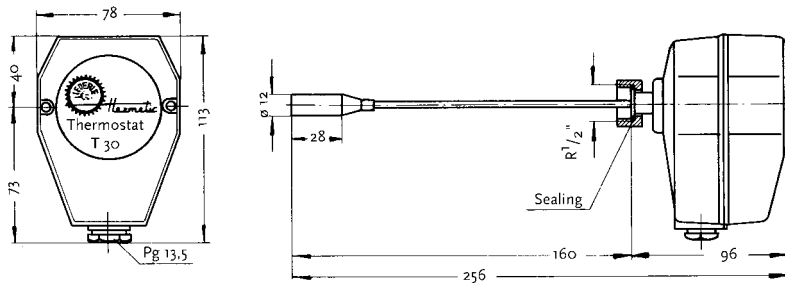
- For safety reasons, the rotor chamber must always be filled with pumping fluid. Therefore, a level indicator has to be installed in the plant or another equivalent protection measure must be taken to ensure that the operation of the rotor is only possible with a sufficient fluid level.
- For the prevention of inadmissible temperatures, the cooling-lubricating flow must be monitored by temperature sensors. This is to ensure that the temperature of the coolant/lubricant does not exceed a certain temperature.

A level-temperature monitoring device fulfilling the conditions of the PTB together with a wiring diagram is shown in figures 3-79 and 3-80 [3.24]. This equipment consists of a thermostat, a level indicator and a switching amplifier. A fluid expansion or resistance thermometer may serve as thermostat. The limiting temperatures stated in the technical specifications of the respective canned motors can be set in the connection head of the device with the aid of a temperature scale.

Level monitor N 30



Thermostat T 30



Switching amplifier S 30

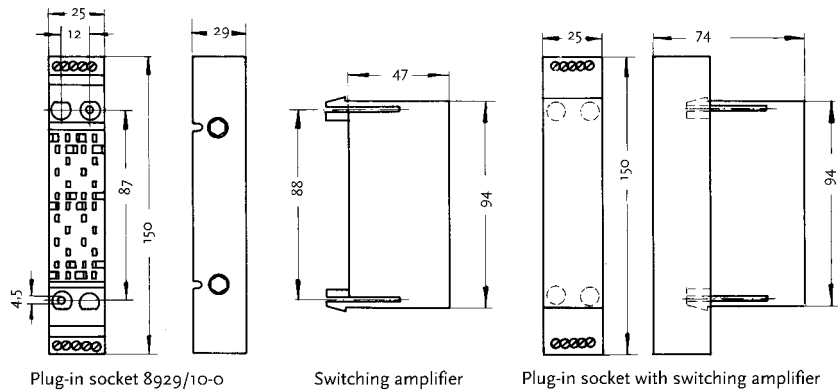


Figure 3-79. Electronic monitoring device for level and temperature indication (factory drawing of Lederle-Hermetic)

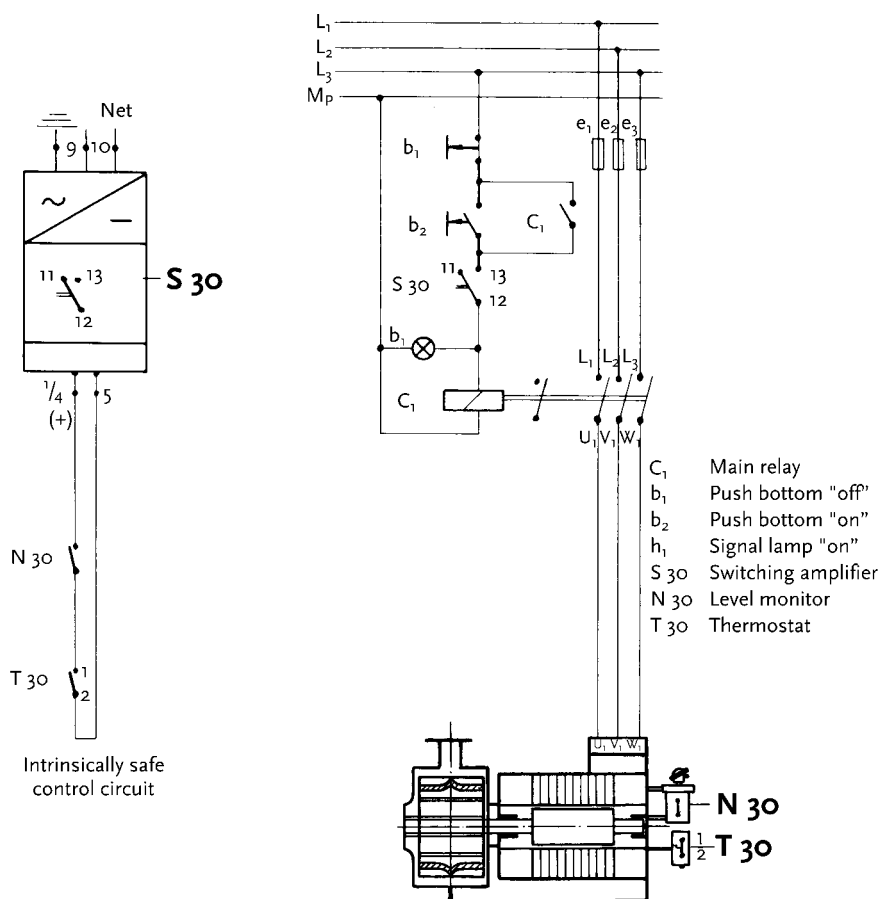


Figure 3-80. Wiring diagram and installation of the electronic monitoring device (factory drawing of Lederle-Hermetic)

3.17.4

Double walled security in hermetic drives (DWS)

A newly developed safety and monitoring system provides additional safety for canned motor and permanent magnetic coupling. Owing to the installation of two co-axially arranged cans with a small clearance in between, it is possible to connect this clearance with sensors which will respond by releasing an optic or acoustic signal or by switching-off the pump/compressor when the inner sealing jacket becomes leaky (fig. 3-81).

If a canned motor is planned as a drive, it disposes of a threefold sealing jacket (2 cans and the motor casing) between the operating circulation liquid and the atmosphere [3.25]. This kind of design has the advantage that in case of a damage of the

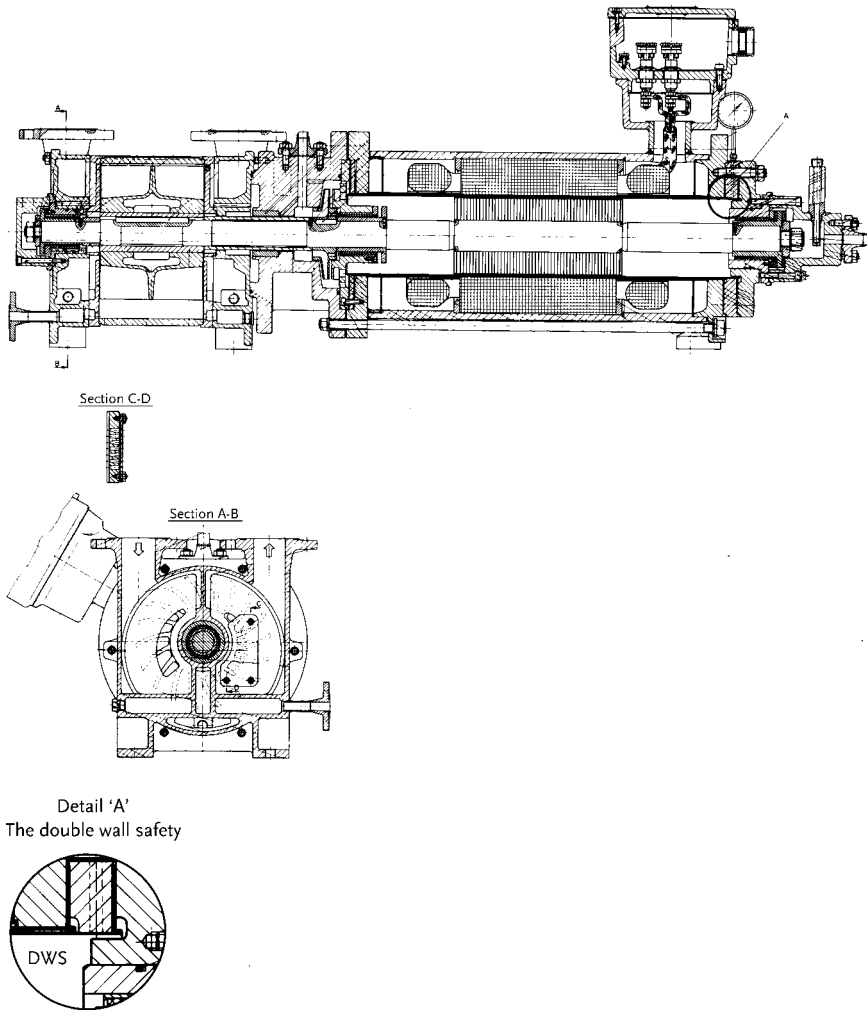


Figure 3-81. Hermetical liquid ring vacuum pump with canned motor and “DWS double wall system” (factory drawing of Lederle-Hermetic)

inner tube, the winding is protected against the entrance of fluid, thus contributing to the reduction of repair time and costs.

3.17.5

Control and monitoring devices for machines with double tube/double can

Fiber-optic level sensor. The leak testing of the double can can be carried out by means of several response modules known from measurement and control engineering. With the aid of a typical example this will be explained more precisely with the appli-

cation of a fiber-optic level sensor. Here, an optical waveguide is immersed into the operating liquid, which, in case of leakage, penetrates the space between inner and outer can, taking advantage of the difference of the refractive index between liquid and gas (fig. 3-82).

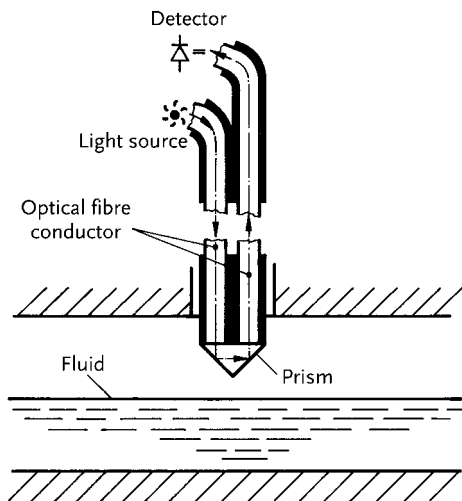


Figure 3-82. Scheme of a fibre-optic measuring method (factory drawing of Lederle-Hermetic)

The special advantage of these sensors is their small size available in thread sizes starting from M5. They are resistant even against high pressures and stand out due to short response times, they can be applied in explosion-hazardous areas and, consisting of fused silica, they are resistant against aggressive fluids. The sensor is supplied with light, coming from a specially developed LED light source over flexible wave guides (\varnothing in the order of 0.5 mm - 1.0 mm). As soon as the sensor contacts the liquid, the light is coupled out into the liquid, so that the arriving light intensity at the detector is much lower than in the case of contact with gas or light. The detector disposes of a downstream switching amplifier used for sounding the respective alarm or triggering the switch-off of the driving engine. These sensors, available in small sizes, in combination with the flexible wave guides of small diameters enable the measuring on spots that are hardly accessible or that dispose of little space. Therefore, they are particularly suitable for the monitoring of the double can system [3.26].

In DWS-systems, the sensor is fed to the measuring spot in the double can via a small borehole in the motor flange (fig. 3-83).

Hermetically sealed pressure sensor. A reliable and safe monitoring device of the double can is also achieved by the application of a sensor for pressure control. The pressure sensor (figure 3-84) is arranged in a suitable position in a way that a pressure change arising in case of a fault can be recorded in the can chamber. The DWS system can be pressurized with nitrogen, for example. The pressure sensor is hermetically sealed and does not contain any movable sealing elements that are subject to wear and tear. In explosion-protected areas, the pressure sensor can be used with-

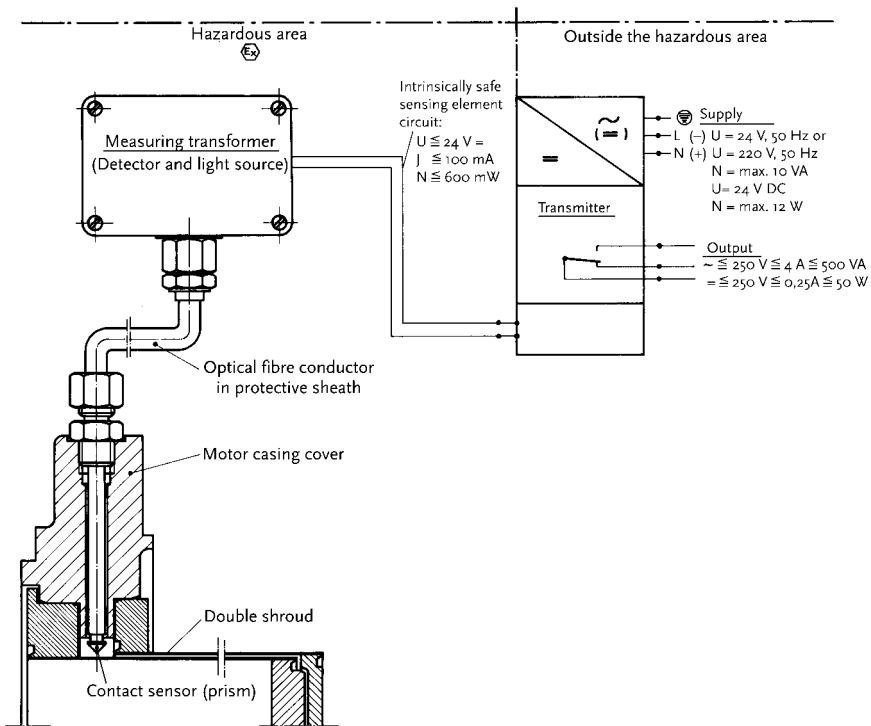


Figure 3-83. Example of a fiber-optic measuring point on a double-can tube design motor (factory drawing of Lederle-Hermetic)

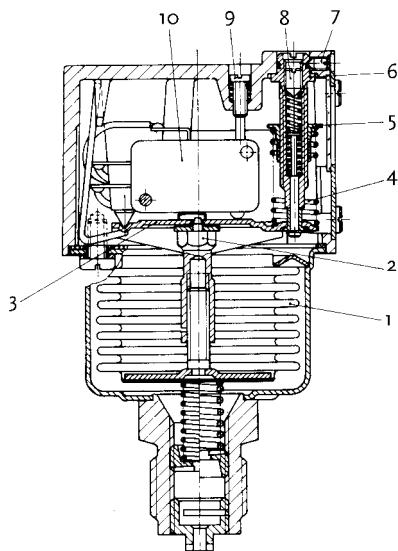
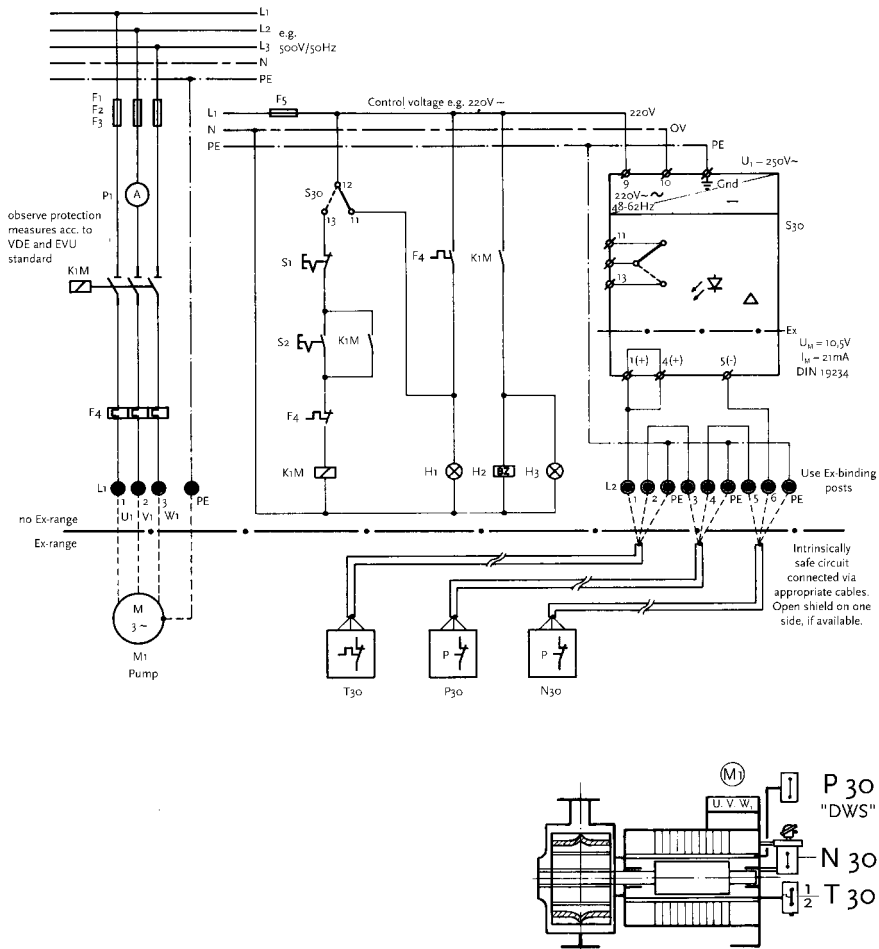


Figure 3-84. Pressure sensor
 1 sensing system 2 tappet 3 switching bridge
 4 pressure spring 5 adjusting nut
 6 set value spindle 7 blocking screw
 8 screw thread pin 9 set screw
 10 micro switch

out problems. By means of a relay amplifier of the protection class “EEx ia IIC”, control commands can be transmitted from an intrinsically safe circuit into an intrinsically non-safe active current circuit. From there, optical and acoustic alarm signals can be given and all further control measures can be taken.



F1	Access security motor	H2	Hours run meter	S2	Push bottom "on"
F2	Access security motor	H3	Signal lamp		
F3	Access security motor			L1	Terminal strip to motor
F4	Motor protection relai			L2	Terminal strip to voltage circuit
F5	Control security	K1M	Safety switch motor	M1	Drive motor pump
		S30	Switching amplifier EE1a II C	P30	Pressostat for DWS
				N30	Level monitor
H1	Signal lamp	S1	Push bottom "off"	T30	Thermostat

Figure 3-85. Wiring diagram for integrated, electronic pump monitoring system (factory drawing of Lederle-Hermetic)

With canned motors operated in the explosion-protected areas and equipped with a temperature level monitoring device, the same amplifier can be used for the signals of the pressure sensor, too. For an optimal monitoring of the canned motor, all three pulse sensors are installed in series according to fig. 3-85, which ensures the optimal electronic monitoring of the electromagnetically driven vacuum pump [3.21].

3.18

Compression of explosible gas-vapor mixtures with liquid ring compressors

The suitability of liquid ring machines for the discharge of reactive mixtures has been verified by the Bundesanstalt für Materialprüfung – BAM (Federal Office for Materials Testing) by means of special explosion tests at a series-manufactured single-stage conventional liquid ring pump of medium size with two flows (Siemens product) [3.27]. The tests were carried out with the flammable gases methane and hydrogen mixed with air. Methane was chosen, because regarding ignition range and minimum ignition energy it matches more or less the gases and vapor occurring in practice, while hydrogen was selected for its excellent ignition qualities. The experimental plant is shown in figure 3-86.

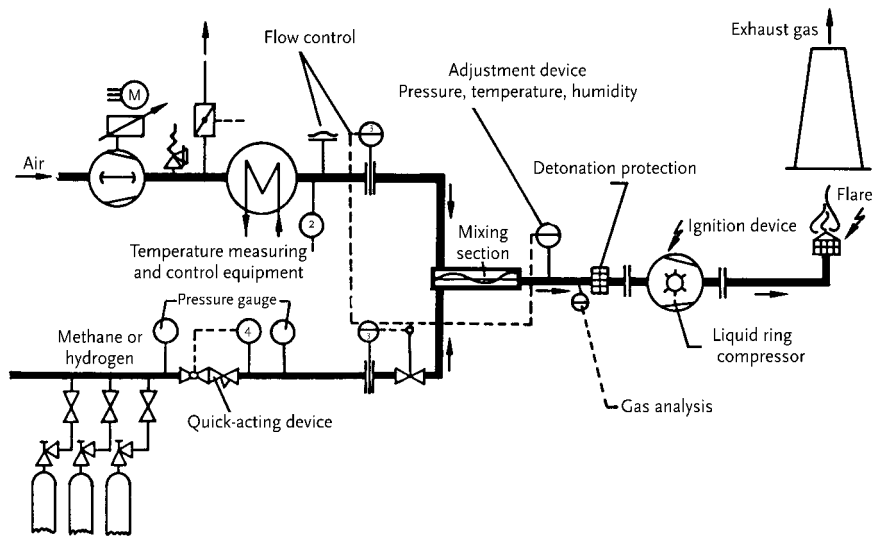


Figure 3-86. Testing device scheme (acc. to BAM)

The explosive mixture was conveyed under following operating conditions:

- volume flow rate: $600 \text{ m}^3/\text{h}$,
- suction pressure: 0.8 bar,
- compression pressure: 1.07 bar,

- rotational speed: 1430 min^{-1} ,
- motor power output 18.5 kW.

In liquid ring machines, the only surfaces gliding with contact to each other (not on each other) are the fronts of the blade impeller with the blades and the port plates. Before switching on as well as during the normal operation of the machine, they are moistened with liquid so that dry friction can hardly occur with perfect bearings of the pump shaft and without axial thrust reacting from outside. Furthermore, the heat arising during compression is removed by the operating liquid, thus achieving an almost isothermal compression. Tests revealed that explosion pressures resulting from an explosion inside the compressor were below the theoretically possible pressures. The pressure rises achieved cannot be called explosions with dangerous effects. Explosions or damages caused by explosions on water ring compressors, for years occasionally used for the discharge of reactive mixtures, are not known. Although the tests have been carried out with only one single medium-sized water ring compressor, in a certain way they can be regarded as typical and applicable to water ring compressors with different capacities, taking into consideration the experiences in practical operation gained with these resources for decades. For the tests, a high-energy ignition source especially arranged for this experiment was used which, in the opinion of the BAM together with the members of the ad-hoc working group “Water Ring Compressors” can hardly occur to such an extent during a normal operation of a liquid ring compressor. Based on results obtained through the tests and their evaluation, it can be assumed that in the case of discharging mixtures that react comparably to methane/air, the commercial water ring compressors made of tough materials fulfill the currently applicable requirements of the UVV (accident prevention regulations) – “compressors”. According to the state-of-the-art technology, series-manufactured water ring pumps are built for operational overpressures of 2.5 bar and above. This would result in a pressure surge resistance for tough materials of at least 4 bar overpressure.

All these considerations lead to the conclusion that liquid ring compressors can well be used for the discharge of explosible mixtures and it can be ruled out almost certainly that with the machine being operated adequately (sufficient quantity of fluid existing at any time) effective ignition sources can hardly arise.

Furthermore, due to theoretical considerations, the working group dealing with the examination came to the conclusion that the requirements of the revised § 10, art. 2 regarding the prevention of ignition sources inside compressors are not only to be regarded as fulfilled in the case of water ring compressors, but also if these machines are operated with different fluids, i.e. with solvents.

3.19

Safety standards for rotating machines

According to the legal regulations [3.28], for the discharge of liquid, organic substances belonging to the danger class A 1 according to § 3, art. 1 of the directive on

flammable liquids – VbF, issued February 27, 1980 (BGBL.I, page 229) revised on May 3, 1982 (BGBL.I, page 569) and having their final boiling point at up to 200 °C, pumps with small leakage losses are to be used. For the discharge of these substances particularly effective measures for the reduction of emission are required depending on the danger and the percentage of contaminants, for example, the application of pumps with double mechanical shaft seals, of canned motors, or of pumps with magnetic coupling as well as the closed diversion of liquid leakage losses, the suction of vaporous or gaseous leakage losses and the cleaning of the sucked exhaust gas in a cleaning facility for exhaust gases. With gases being compressed, the confining liquid of the compressors is not allowed to be degassed outside, if the gases used contain respectively dangerous substances or an inadmissibly high percentage of contaminants. Owing to remarkable further developments it was possible to raise the technological level of rotating machines to a level corresponding to the high requirements of the general process engineering, the chemical industry as well as the nuclear power plants. The solution of the problem concerning rotating machines led above all to highly developed sealing and driving systems, from the stuffing box packing to hermetic pump units with high safety standards with fully electronic monitoring (fig. 3-87).

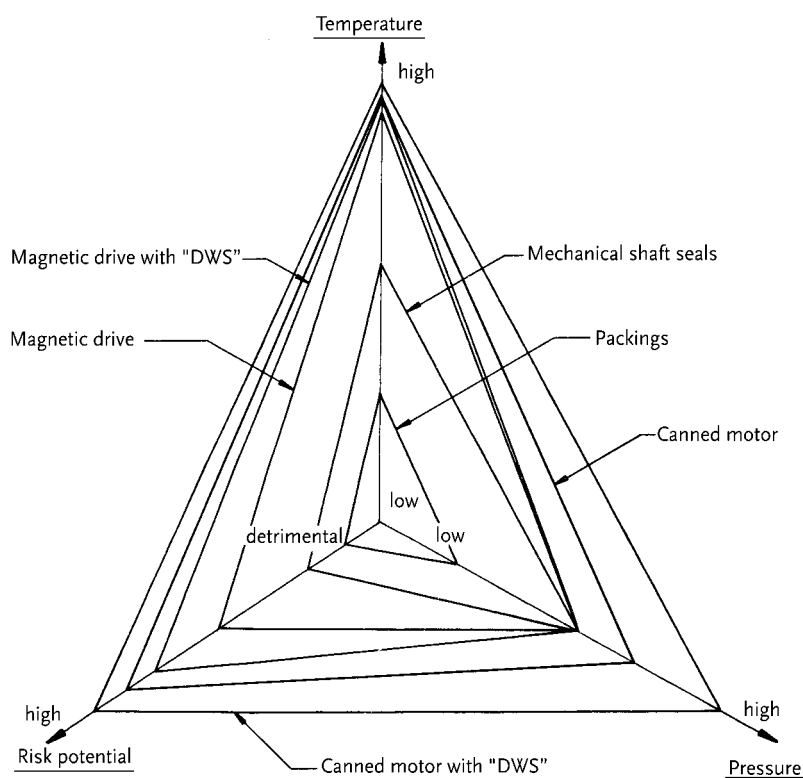
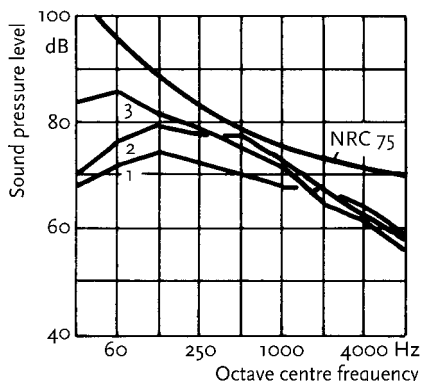


Figure 3-87. Sealing and driving systems in the "hazard diamond" [3.29]

3.20

Characteristics and fields of applications of liquid ring vacuum pumps and compressors

Owing to their plain design and their low susceptibility to damage, there is a wide range of application as vacuum pump and even as compressor, mainly in the chemical industry, in power plants, in the general process engineering of the food and luxury food industry and the pharmaceutical factories. Its features open up a wide range of application to the liquid ring gas pump. Pumps of this design work quietly and almost pulsation-free and vibration-free, dispose of an almost isothermal compression and can be applied for the discharge of dry gases or vapor-gas mixtures without problems. As lubrication in the compression chamber is not required, absolutely oil-free compression is achieved. The carrying of small liquid quantities is possible, and these pumps are even insensitive to the conveying of contaminated or polymerization-prone gases. Due to the isothermal compression, explosible or polymerization-prone gases can be transported safely. The suction of damp, vaporous gases causes condensation in the pump and thus an increase in the suction capacity. Liquid ring machines are not only suitable for the discharge of neutral gases under ideal discharge conditions, they can even take on the functions of condensers, gas coolers, dirt and liquid separators as well as of absorption plants. In process engineering, the gases to be compressed are often aggressive and alkaline or acid operation liquids are required by the respective process. In such cases, liquid ring pumps can be manufactured of a material suitable for the intended application purpose.



- 1 Small ELMO-F vacuum pump,
1450 min⁻¹, $\bar{L}_{PA} = 71$ dB (A)
- 2 Medium size ELMO-F vacuum pump,
420 min⁻¹, $\bar{L}_{PA} = 76$ dB (A)
- 3 Large ELMO-F vacuum pump.
255 min⁻¹, $\bar{L}_{PA} = 75$ dB (A)

Figure 3-88. Average acoustic pressure spectra at normal rotational speed and average operation pressure measured in distances of 1 m (acc. to Siemens)

Owing to different constructive designs with conventional stuffing box, with mechanical shaft seals or in hermetic design with permanent magnetic coupling or canned motor, these pumps can be adapted to all requirements of the chemical and process engineering companies. The hermetic designs are particularly important in

cases where absolutely tight circuits both on the ring liquid side and the gas side are required due to conditions imposed by the environmental protection.

In order to comply with the maximal permissible limit values according to the "TA Lärm", the machines have to be chosen with regard to the max. admissible noise emission. Fig. 3-88 shows the average sound-pressure spectra of different dimensions of vacuum pumps. These are the measured sound-pressure levels to be interpreted as reference values. Liquid ring machines equipped with a canned motor will fall below these noise emission values by 8–10 dB(A) on the average, with the requirements regarding an environmentally friendly noise level being met. Figure 3-89 compares the reduced noise levels of centrifugal pumps for the total pumping unit with canned motor to the conventional centrifugal pump without motor [3.30].

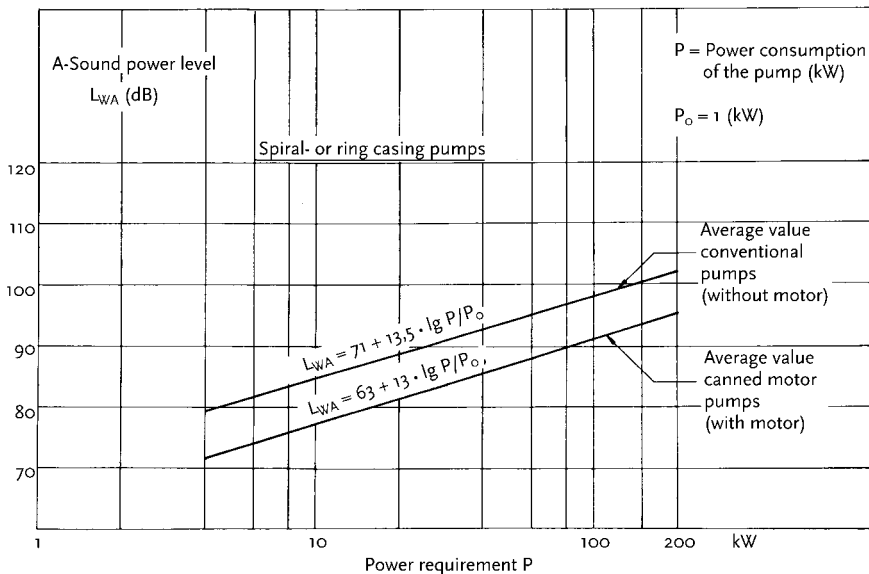


Figure 3-89. Sound power level A as a function of the power consumption P of volute- or ring-housing centrifugal pumps (acc. to Lederle-Hermetic)

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4

Vacuum and Compressor Plants with Liquid Ring Machines

4.1

Demands on pump systems in process engineering

Both design and mode of operation of vacuum plants have to be adapted to the respective vacuum process in consideration of economical aspects. Therefore, when designing vacuum stations, the suction capacities and pressure ratios must be adapted to the relevant operating conditions.

The chart in fig. 4-1 gives an overview of the gas flows in closed systems occurring in vacuum technology.

Vacuum pumps, sinks and cooling traps have to be sufficiently dimensioned already in the planning stages of the plant taking into consideration the influences of residual gases, leakage air and gas development during the process flow.

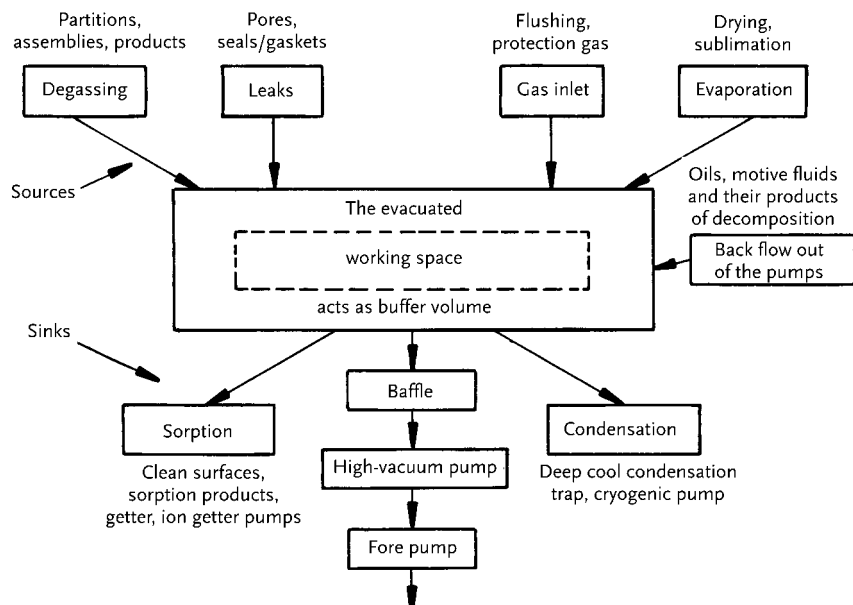


Figure 4-1. Gas flow types in a vacuum system [4.1]

Fields of application for vacuum technology are e.g. drying of food, amongst other things freeze-drying of coffee, vacuum packaging of perishable and other goods, transportation of paper in printing houses, pneumatic dispatch systems, transportation of bulk materials or liquids from ships into dock silos and vacuum distillation. In the vacuum metallurgy, metals are melted and cast under vacuum to avoid chemical reactions and oxidation with oxygen. The degassing of steel and the annealing of metals are accelerated by the application of vacuum i.e. gases and vapors are thereby released and sucked. For electron-beam diffusion welding, vacuum soldering and sintering of hard metal, vacuum is used for the combination of substances as protection against oxidation with oxygen from the atmosphere. In the modern chemical industry and in general process engineering, the degassing, extraction and sublimation processes take place under vacuum. Vacuum is needed in electrical engineering for drying and degassing of insulation materials and insulating liquids.

Vacuum technology is also applied in cleaning processes, for evaporation of anti-reflecting or reflex-increasing layers in glass sheets, as well as for the application of protective coatings to tools and jewellery. For the manufacturing of television tubes, discharge tubes for electrons and gases and for the production process of compact discs, pressures up to the range of high vacuum are partly required.

For reasons of environmental protection, in many branches of industry process steps are carried out under vacuum in order to be able to dispose of hazardous substances or to lead them back into the process instead of emitting them into the atmosphere.

From this wide range of application for vacuum technology, in the following vacuum and compressor plants will be described only for the section from coarse vacuum to fine vacuum.

While in coarse vacuum engineering, seen from a vacuum-technological point of view, the demands on tightness and gas discharge are low, in the range of fine and high vacuum higher demands have to be fulfilled. For instance, in the coarse vacuum engineering, standard screw joints with Teflon tape seals can be used as connecting elements. Shut-off devices are often installed in standard gas pipelines.

Of course, even in coarse vacuum engineering it is common to apply flange connections with elastomere seals or the usual flange and quick-joints standardized in the DIN or PNEUROP, as well as components with radial gasket or cap sleeve seals and versions with bellows seals. Owing to the installation of high-quality parts, the detection of leakages is particularly time-saving.

If dirt and dust accrue in large quantities during evacuation, as it may be the case in coarse vacuum, oil-sealed rotary vacuum pumps must be protected by suitable filters or dust catchers. If there are any oil-soluble or oil disintegrating substances in the suction flow, it is recommended to use e.g. a pump unit consisting of Roots vacuum pump-gas ejector-liquid ring vacuum pump instead of an oil-sealed rotary vacuum pump. These pumps are also more suitable for the pumping of corroding gases and vapours as well as for the suction of contaminated gas mixtures. In cases with low-cost vapor being available, corrosive gases can also be discharged with steam jet pumps of suitable material.

The operating costs of the combination of Roots vacuum pump-gas ejector-liquid ring vacuum pump or of individual or series-installed steam jet pumps can be higher than those of the combination of a Roots vacuum pump with an oil-sealed rotary pump.

Chemical filters installed on the suction side (active carbon, Na_2CO_3 etc.) or special lubricating or sealing liquids instead of common pump oils can be a good remedy when pumping corrosive gases or vapors of chemical solvents such as acetone, trichlorethane, ethyl acetate, tetrachlorocarbon and acids. For the operation of oil-sealed rotary pumps, dry nitrogen can be used as gas ballast instead of air. In order to prevent the pump from corroding, downtime rinsing with dry nitrogen is recommended, too.

Usually, with liquid ring vacuum pumps a trouble-free discharge of corrosive process gases is possible. Due to their design and function and owing to the possibility of manufacturing them of different materials, they can cover a wide spectrum of the vacuum sector. With the suitable ring liquid being selected according to the engineering process, liquid ring machines can be efficiently used e.g. for the recovery of solvents. If the ring liquid is the accruing solvent condensate instead of water, the contamination of the waste water can be avoided and disposal costs can be reduced. The exhaust gas, possibly still containing solvent vapors, can be dried in an immission cooler installed on the discharge side and can then leave the pump system free from pollutants.

4.2

Basic combinations of liquid ring vacuum pumps and equipment in compact plants [4.2]

In the chemical and general process engineering, the liquid ring vacuum pumps are operated not only with water, but frequently also with other ring liquids adapted to the process. This contributes to the fact that these machines are operated with a closed ring liquid circuit (fig. 4-2).

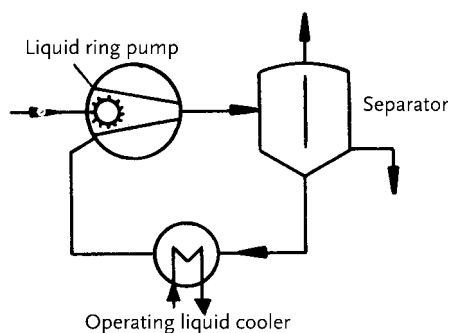


Figure 4-2. Liquid ring machine with a heat exchanger and liquid separator in a closed circuit

Owing to the completion of the design shown in fig. 4-2 with a condenser on the suction side and an exhaust-cooler on the discharge side, process gases such as solvents can be recovered and returned to the process. Fig. 4-3 shows a chart with condenser and exhaust cooler.

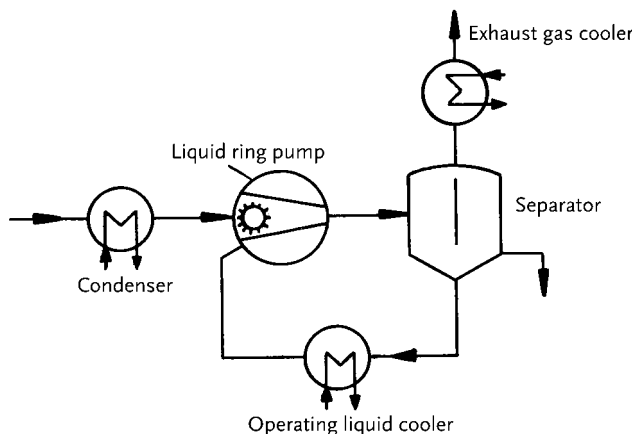


Figure 4-3. Liquid ring pump in a closed circuit with condenser and exhaust cooler

As apparent in fig. 4-4, the liquid ring vacuum pump can be combined with a gas ejector to achieve smaller suction pressures. Instead of gas ejectors, Roots pumps or rotary vane pumps can be used (fig. 4-5). There are also versions with a gas ejector being installed between the liquid ring pump and a Roots vacuum pump to achieve higher vacuums.

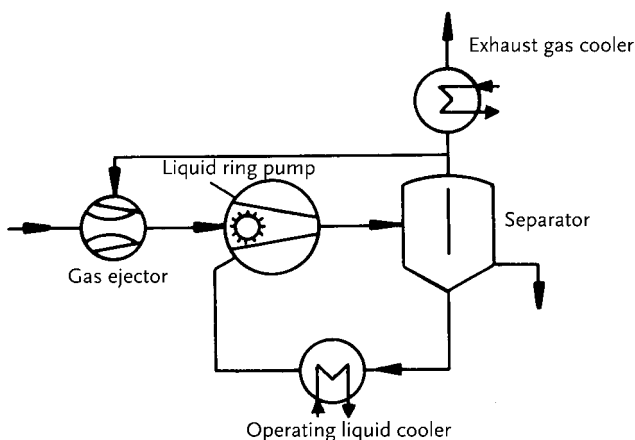


Figure 4-4. Liquid ring vacuum pump in a closed circuit with gas ejector

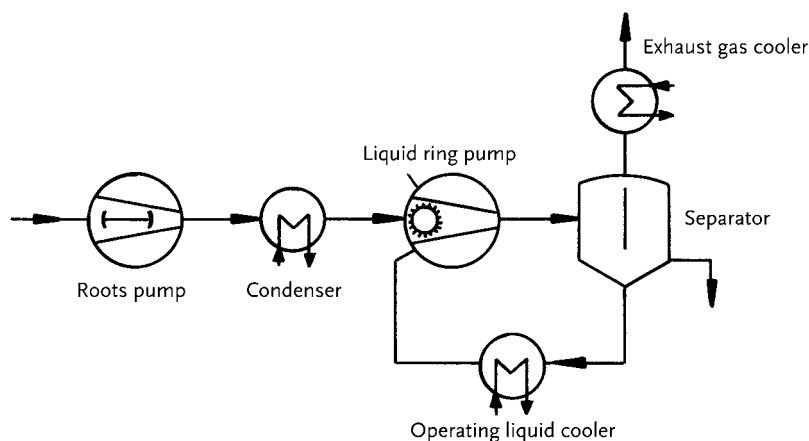


Figure 4-5. Liquid ring vacuum pump in a closed circuit with condenser and Roots pump installed on the suction side

A further possibility to achieve larger pressure ratios is the series-installation of two liquid ring vacuum pumps for the operation at higher vacuums, or to combine a liquid ring vacuum pump with a liquid ring compressor at higher compression pressures (fig. 4-6).

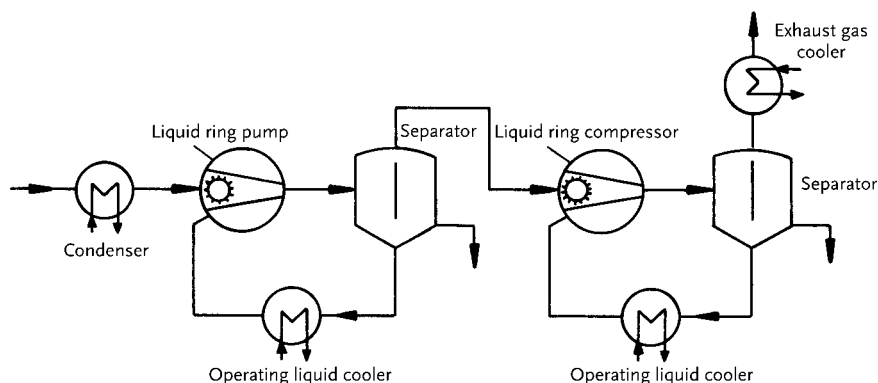


Figure 4-6. Liquid ring vacuum pump and liquid ring compressor installed in series

The selection of the combination of liquid ring machines, condensers and exhaust-coolers should take place with regard to the kind of process gas and possible maximum permissible emission values. With liquid ring vacuum pumps and compressors, pump systems with minimal portions of pollutants in the discharged gas can be designed, provided the pressure and temperature parameters are determined correctly.

4.3

Control of liquid ring pumps and pump systems

Vacuum pumps are usually designed for a pre-determined suction pressure and a defined suction capacity which corresponds to e. g. the leakage rate. In case the calculated leakage does not occur, the selected pump would be too large and would operate at lower suction pressures than planned. Depending on how big the difference is between the suction pressure and the saturation vapor pressure of the operating liquid, a suction pressure control must be provided, in order to avoid cavitation. A very simple solution is e.g. the feeding of atmospheric air or the recirculation of process gas from the pressure side to the suction side of the pump. A control is also necessary if discontinuous leakage is to be expected or if the suction pressure has to be kept constant for process engineering reasons.

As, apart from the condensing effect, a liquid ring vacuum pump is a displacement pump, a control by throttling the vacuum pump on the suction side would not be practical. This would bear the risk of the suction pressure dropping to such a low value that cavitation in the pump is provoked.

For this reason, bypass control is mainly used in process engineering (fig. 4-7). This means that either entrained air from the atmosphere or process gas from the pressure side of the vacuum pump is recirculated to the suction side or to the suction port of the gas ejector. In this way, the suction pressure can be kept constant. Control is carried out manually by operating a valve, via automatic spring-loaded ventilation valves (vacuum breaker) or, as it is mostly the case in automatically controlled plants, by means of fully automatic, electric, pneumatic or electro-pneumatic control devices.

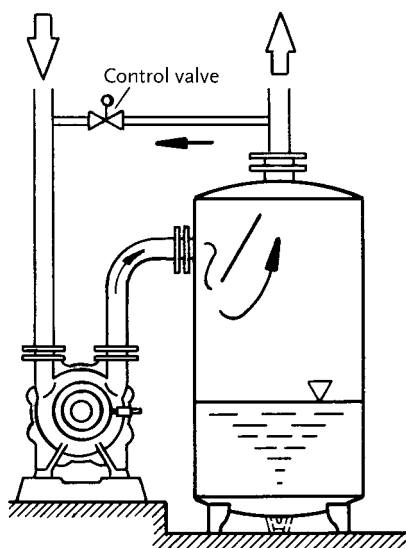


Figure 4-7. Bypass control of a liquid ring vacuum pump

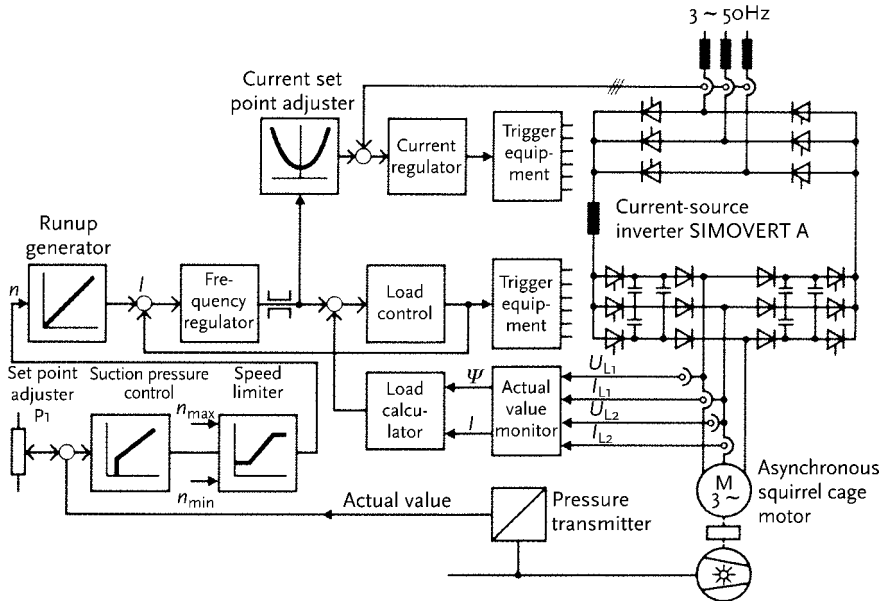


Figure 4-8. Pump speed adjustment for liquid ring vacuum pumps and compressors with frequency-controlled low-voltage three-phase motor (acc. to Siemens)

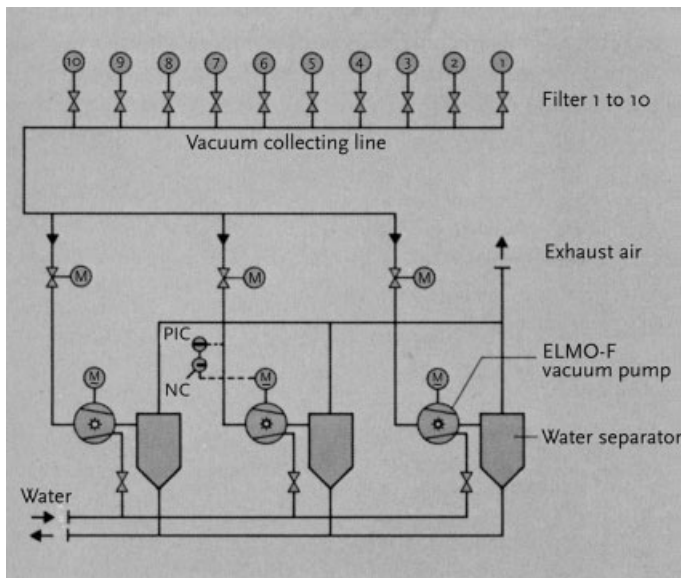


Figure 4-9. Adjustment of the required pump suction capacity to a filtering device by operating 1 to 3 vacuum pumps and a speed-controlled drive of the middle pump unit (acc. to Siemens)

Another possibility of suction volume control offers the rotational speed control of the vacuum pump (fig. 4-8). In general, the liquid ring vacuum pump can be operated at between 75% and 150% of the nominal speed corresponding to a circumferential speed at the vane tips of approx. 14 m/s to 28 m/s. This is an energy-saving alternative mainly used for vacuum pumps with very large suction capacities.

The so-called intermittent control with buffer vessel and stepwise intermittent operation mode of several parallel-connected pumps which are partially speed-controlled (fig. 4-9) are further alternatives for suction volume control.

4.3.1

Electronic vacuum control for distillation in laboratories

In vacuum distillation, it is important to avoid boiling delays and overheating. The application of electronically operating regulators allows the control of the suction pressure in accordance with the solvent to be distilled. In the regulator, the so-called vacuum controller [4.3], a controlled distillation is possible through the input of a code for the solvent, its desired boiling temperature and evacuation period. Regulators of this kind control the absolute pressure and are equipped with micro-computers with the solvent data tables being stored. The advantages of these devices are the automatically controlled boiling process, the easy operation, a decrease in operating costs, as well as a maximum solvent recovery and the environmentally friendly features. For vacuum generation in laboratories, as required and applied in distillation, vacuum filtration and vacuum drying, classical water jet pumps are still used (fig. 4-10).

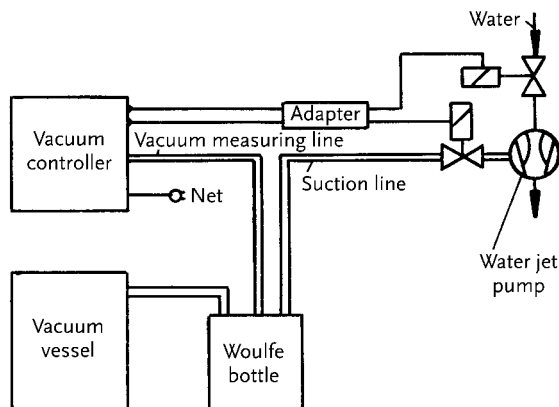


Figure 4-10. Operation with a valve-controlled water jet pump (acc. to Chemie und Filter GmbH, Prominent Pumpenwerk)

Using the electronic controllers described above, the jet pump is operated only when the suction capacity is really needed, which leads to savings of water of up to 95% in these pumps. When stopping the plant, the undesired ventilation of the ves-

sel is contemporarily prevented by the selected electromagnetic valve. Fig. 4-11 shows an arrangement of devices with a sliding vane vacuum pump.

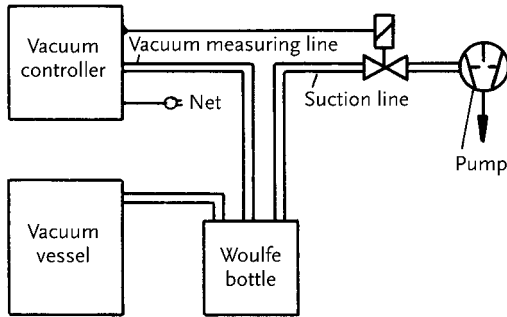


Figure 4-11. Operation with a motor pump and suction pipe valve (acc. to Chemie- und Filter GmbH, Prominent Pumpenwerk)

4.3.2

Liquid ring vacuum pump system with automatic suction pressure control

In the pump system shown in fig. 4-12a and 4-12b, the suction pressure control is taken on by a pneumatic transducer, a PI controller and a pneumatic control device.

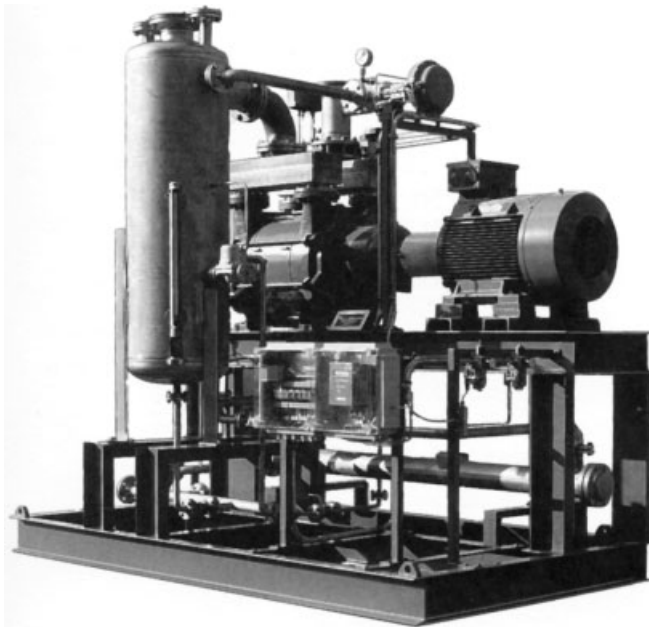


Figure 4-12 a. Hermetical liquid ring vacuum pump with permanent magnet coupling in a pump system with automatic suction pressure control (acc. to Lederle-Hermetic)

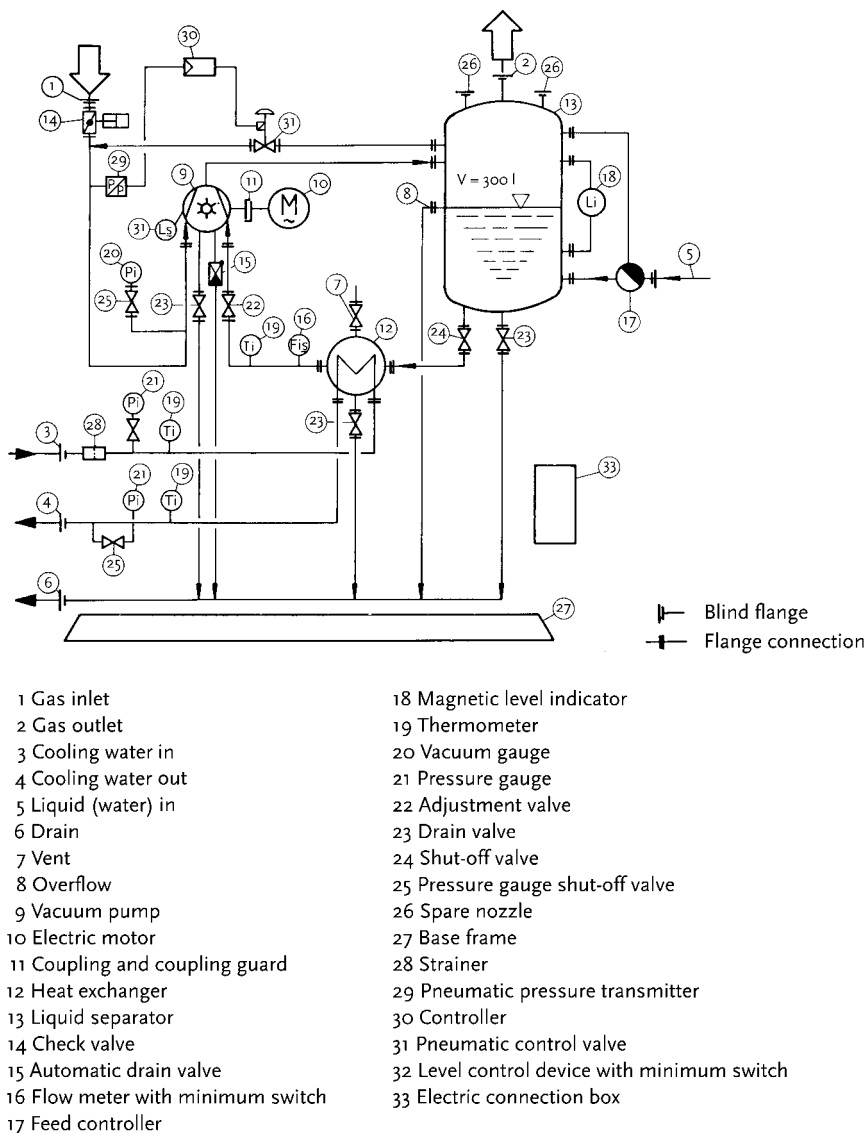


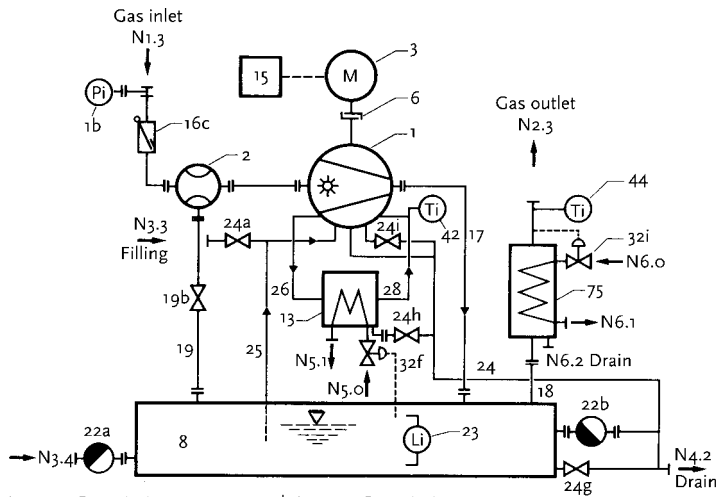
Figure 4-12 b. Functioning and installation diagram

4.3.3

Control of coolant consumption for heat exchanger and immission cooler

When operating liquid ring vacuum pumps in combination with heat exchangers and immission coolers, mechanically working control valves (acc. to [4.4]) can be installed in the feeding pipes of the cooling liquid. As soon as the adjusted set value

temperature of the control valve is exceeded, the control valve is opened via an installed thermostat and the linked users are supplied with coolants. Owing to the optimal adaptation of the coolant consumption to the given operating conditions, the economic efficiency of the plant increases while the coolant consumption is minimized. Fig. 4-13 shows the structure of a vacuum plant with heat exchanger, immission cooler and the respective control valves.



Item	Description	Item	Description
1	Vacuum pump	2	Gas ejector
3	3ph motor	15	Electrical switch equipment
8	Separator/base frame	22 a	Inlet control
13	Heat exchanger	22 b	Outlet control
16 c	Non-return valve	32 f	Control valve for Item 13
17	Pressure pipe	32 i	Control valve for Item 75
18	Pressure pipe	75	Exhaust gas condenser
19	Bypass	42	Thermometer
19 b	Shut-off valve	44	Thermometer
23	Liquid level indicator	1 b	Vacuum gauge
24 a	Shut-off valve		
24 g	Shut-off valve		
24 h	Shut-off valve		
24 i	Shut-off valve		
25	Liquid pipe		
26	Liquid pipe		
28	Liquid pipe		

N1.3	Suction flange
N2.3	Discharge flange
N3.3	Filling
N3.4	Inlet (level control)
N4.2	(Total) drain
N5.0	Cooling water inlet (circulation cooler)
N5.1	Cooling water outlet (circulation cooler)
N6.0	Cooling water inlet (exhaust gas condenser)
N6.1	Cooling water outlet (exhaust gas condenser)
N6.2	Drain (exhaust gas condenser)

Figure 4-13. Liquid ring pump package unit with coolant consumption control (acc. to Siemens)

Auxiliary energy for the operation of the thermostat is not required, since these devices work according to the liquid expansion principle.

4.3.4

Optimal evacuation with liquid ring vacuum pumps

On the one hand, the evacuation of process columns has to occur in a way that a determined suction pressure is reached within a short time and this underpressure is held for a certain period, and on the other hand, a desired temporal pressure march should be maintained. The suction capacity of the liquid ring vacuum pump must be practically adapted to the gas quantities accruing during the process.

The change of the suction capacity of a liquid ring vacuum pump or of the pump plant can occur by means of changing the rotational speed, connecting and disconnecting of individual vacuum pumps, by means of supplying gas externally or internally through a bypass and by changing the operating liquid temperature. Which option is to be used in the individual case depends on the existing operating conditions, on the available control possibilities, on process characteristics or conditions and not least on the number of installed vacuum pumps.

The five control options are listed in table 4-1 acc. to [4.5]. The decision, which type of adjustment is to be used, should be made on consideration of the planned investment costs and their amortization.

Table 4-1. Suction capacity – Adjustment possibilities for liquid ring vacuum pumps [4.5]

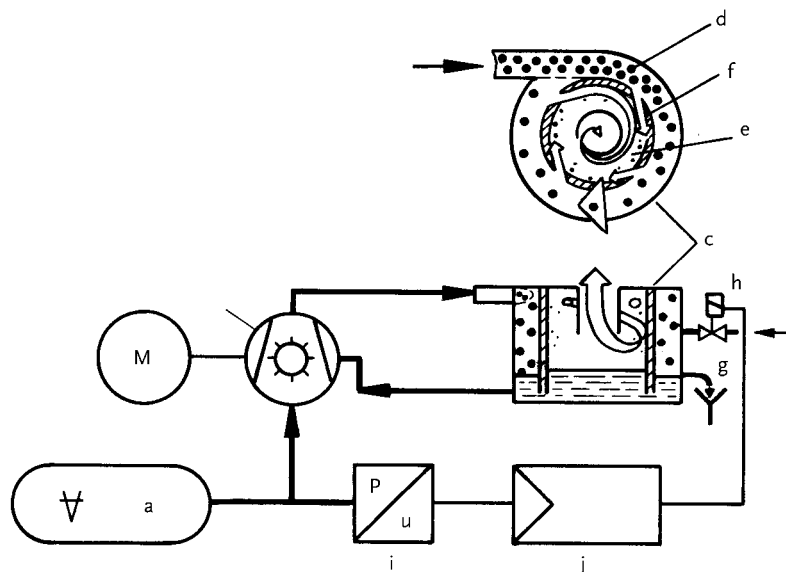
Suction capacity adjustment mode	Suction capacity adjustment range	Advantages	Disadvantages
Pump speed	40–100%	Adjustment times up to 1 Hz. Low electric energy consumption.	Relatively high investment costs with amortization period of up to 2 years.
Parallel operation of pump units	10–100%	Adjustment times up to 1 Hz. Low electric energy consumption.	Moderate investment costs with amortization period of up to 1 year.
External gas supply	0–100%	Adjustment times up to 1 Hz with low investment costs.	High electric energy consumption. No amortization.
Liquid ring temperature	80–100% (dry gas) 40–100% (humid gas)	Low investment costs. Low cooling water consumption.	Adjustment times achievable only up to 10^{-2} Hz.
Combination of different adjustment modes	0–100%	Adjustment times up to 1 Hz with low electric energy and cooling water consumption.	High investment costs with amortization period of up to 1 year.

When changing the rotational speed to its maximum, the pump shaft strength and the dynamic forces, as well as the maximum permissible circumferential speed of the pump impeller vane must be observed. The minimal number of revolutions results from the demand for a stable rotating liquid ring. The maximum possible rotational speed can be chosen at about 20-30 % over the nominal speed at 50 Hz.

Due to a parallel operation of two or more vacuum pumps, the total suction capacity results from adding the suction capacities of the simultaneously working pumps. If one pump is operated with variable speed, the adjustment range can be extended to smaller or larger suction capacities.

Mostly, the control by modifying the external gas supply is used. The gas quantity to be sucked from a vacuum vessel can be adjusted within the range of zero to the maximal suction capacity of the pump unit. A disadvantage is the high consumption of electric power, as the vacuum pump is continuously operated at nominal output.

With the liquid temperature control, the adjustment range does not only change owing to different temperatures of the ring liquid, but also owing to the portion of possibly condensing portions of the suction mixture. The speed needed for the change of the suction capacity depends on the possible velocity of the temperature change. The smaller the liquid volume is, the faster the temperature can change. Therefore, fresh water is added via a magnetic valve (fig. 4-14). Usually, water is used both as coolant and as operating liquid. However, other fluids, e.g. cooling brines, organic or inorganic ring liquids can be used.



Vacuum vessel a, pump unit b, potential vortex separator c with coarse separation stage d and fine separation stage e, diffusor f, drain pipe g, solenoid valve for fresh water supply h, pressure transmitter i, controller j

Figure 4-14. Control system of an automatically programmed march of pressure in a vacuum vessel by changing the operating liquid temperature. [4.5]

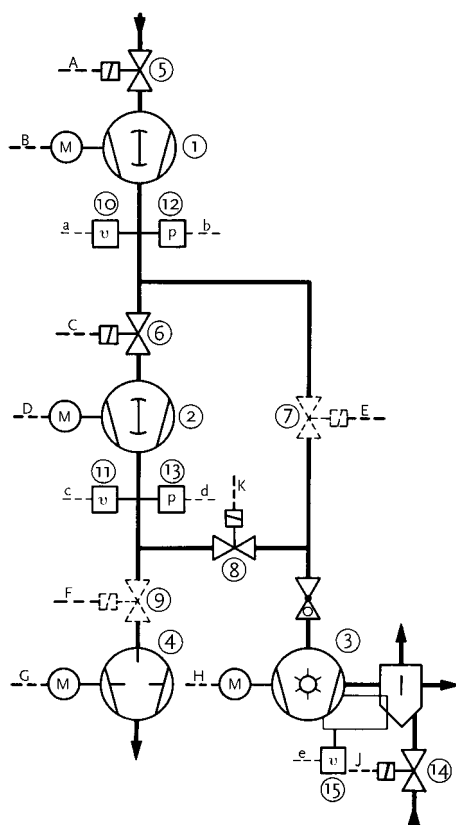


Figure 4-15. Scheme of a Roots pump unit with a liquid ring vacuum pump (acc. to Loewe)

Owing to the combination of different adjustment modes, an optimum between the electric power saving and the water consumption is reached. The first control step is the adjustment of the rotational speed. If the adjusted speed is not sufficient, gas can be supplied externally. When the nominal suction pressure is reached with the two control possibilities described above, the optimal operation point is approached by the slow change of the liquid ring temperature (fig. 4-14). Depending on the measured deviation from the desired pressure value, more or less cold fresh water is fed to the operating liquid circuit. This control mode requires only little cooling energy for the operating liquid. The link of different control loops is carried out by a microcomputer equipped with the appropriate software. The Roots vacuum pump unit shown in fig. 4-15 is controlled fully automatically by a microcomputer [4.6].

Description of functioning for fig. 4-15:

The gas to be sucked enters the Roots vacuum pump 1 by the opened valve 5, flows through the opened valve 7 and is conveyed into the atmosphere by the water ring pump 3. Sensor 15 controls the magnetic valve 14 and minimizes the fresh water consumption of the water ring pump. The sensors 10 and 12 determine the

motor speed of the Roots vacuum pump 1 to protect it against thermal overload. Valve 6 is closed, the Roots vacuum pump 2 is at a standstill, valves 8 and 9 are closed, the sliding vane pump 4 is at standstill.

With the continuously dropping suction pressure, the speed of the Roots vacuum pump 1 increases. At a predetermined pressure on sensor 12, valve 7 closes and the valves 6 and 8 open. The Roots vacuum pump 2 starts up synchronously to the speed of the Roots vacuum pump 1. Then, as desired, the sensors 10, 11, 12, or 13 take over the monitoring and control of the Roots vacuum pumps 1 and 2 to protect them from overheating. Again at a predetermined, however, lower pressure on sensor 12, valve 9 opens, and the sliding vane pump 4 starts at a synchronous speed. After a short time, the valves 8 and 14 close and the water ring pump 3 is switched off. When the pump unit switches off, the valves 5 and 9 close, and shortly afterwards the Roots vacuum pumps 1 and 2 as well as the sliding vane pump 4 stop.

The control, regulation and monitoring functions described above are carried out by a microcomputer installed in an equipment cabinet. Thus, the microcomputer allows an optimal vacuum pump gradation depending on the suction pressure and suction temperature.

4.4

Pump unit designs and possibilities for the application of liquid ring machines with design examples

4.4.1

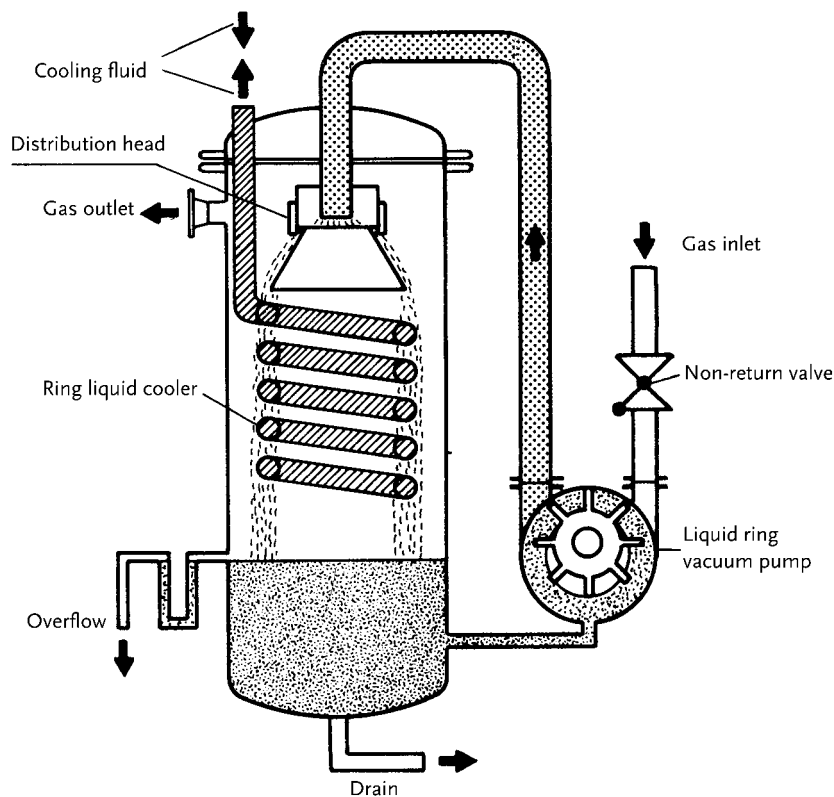
Vacuum systems for condensate recovery

Design of pump units

Vacuum systems with liquid ring vacuum pumps can be operated in suitable combinations with liquid ring coolers, liquid separators and condensers for the suction of gases and vapors in demanding technological processes. Usually, these pump plants are designed with integrated closed circuits (fig. 4-16), in order to avoid a direct contact between the cooling medium and the ring liquid. Thereby, the cooling liquid is not contaminated and the disposal of contaminants is not necessary. Any liquid can be used as ring liquid; the selection usually depends on the technological process. With the operating liquid and the pressure and temperature conditions being adequately selected on the liquid and the gas side, in these plants the sucked gases and vapors can be neutralized and the solvent vapors can be condensed. In systems with closed circuits, the accruing condensate is better used as ring liquid. For the neutralization of acid vapors, for instance, caustic soda or, at lower ambient temperatures, polyethylene-glycol can be chosen as operating liquid. If condensation of process vapors is undesirable, also high-temperature oils can be used as ring liquids. Table 4-2 gives an overview of standard ring liquids. Owing to the installation of an immersion cooler on the discharge side and the application of coolants with lower temperatures (brines), vapor portions not yet condensed can be recovered and returned to the liquid separator.

Table 4-2. Fluids as ring liquids for vacuum systems with closed circuits [4.7]

Alcohols		Ethers		Organic sulphurous compounds	
methanol	CH_4O	dimethyl ether	$\text{C}_2\text{H}_6\text{O}$	methyl-mercaptan	CH_4S
ethanol	$\text{C}_2\text{H}_6\text{O}$	diethyl ether	$\text{C}_4\text{H}_{10}\text{O}$	ethyl-mercaptan	$\text{C}_2\text{H}_6\text{S}$
propanol	$\text{C}_3\text{H}_8\text{O}$	furane	$\text{C}_4\text{H}_4\text{O}$	Aromatic substances	
butanol	$\text{C}_4\text{H}_{10}\text{O}$	Inorganic compounds		benzene	C_6H_6
octanol	$\text{C}_8\text{H}_{18}\text{O}$	water	H_2O	toluene	C_7H_8
i-butanol	$\text{C}_4\text{H}_{10}\text{O}$	caustic soda	NaOH	ethyl benzene	C_8H_{10}
ethylene glycol	$\text{C}_2\text{H}_6\text{O}_2$	Halogenated hydrocarbons		styrolene (vinyl-benzene)	C_8H_8
glycerin	$\text{C}_3\text{H}_8\text{O}_3$	methylene chloride	CH_2Cl_2	Others	
cyclohexanol	$\text{C}_6\text{H}_{12}\text{O}$	chloroform	CHCl_3	high-temperature oils	
benzyl alcohol	$\text{C}_7\text{H}_8\text{O}$	1,1-dichloroethan	$\text{C}_2\text{H}_4\text{Cl}_2$	polyethylene glycol	
Phenols		trichloroethylene	C_2HCl_3		
phenol	$\text{C}_6\text{H}_6\text{O}$	chlorobenzene	$\text{C}_6\text{H}_5\text{Cl}$		
Ketones					
acetone	$\text{C}_3\text{H}_6\text{O}$				
methylethylketone	$\text{C}_4\text{H}_8\text{O}$				

**Figure 4-16.** Scheme of vacuum system with closed ring liquid circuit and an integrated surface irrigation cooler (acc. to Sulzer-Burckhardt)

In closed systems, devices of tube bundle design are mainly used as heat exchangers for the ring liquid and as condensers. Depending on their function principle, their sizes and the available space, they can be installed vertically (figs. 4-16, 4-17, 4-18, 4-19, 4-20a) or horizontally (figs. 4-21a, 4-22a, 4-25a, 4-26a).

For the cooling of ring liquid and gas, even liquid separators with integrated surface irrigation can be used (fig. 4-16). If both ring liquid coolers and exhaust gas condensers are installed in a pump system, they are usually operated with separate coolant circuits at different temperatures.

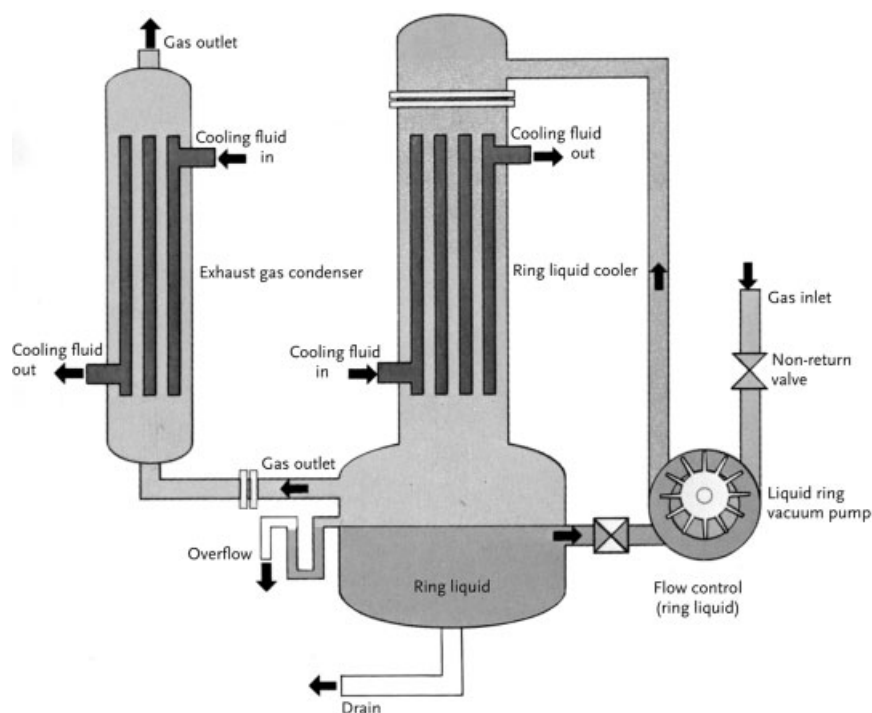


Figure 4-17. Scheme of a vacuum system with closed ring liquid circuit, ring liquid cooler and exhaust gas condenser (acc. to Sulzer-Burckhardt)

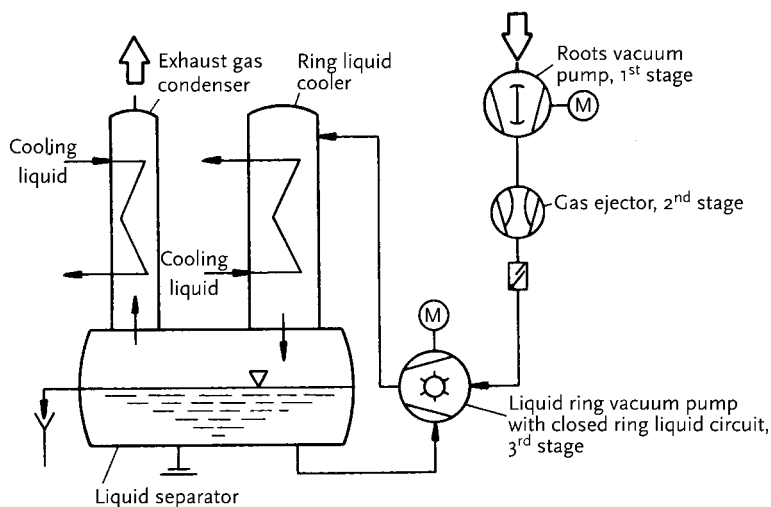


Figure 4-18. Scheme of a three-stage vacuum pump unit

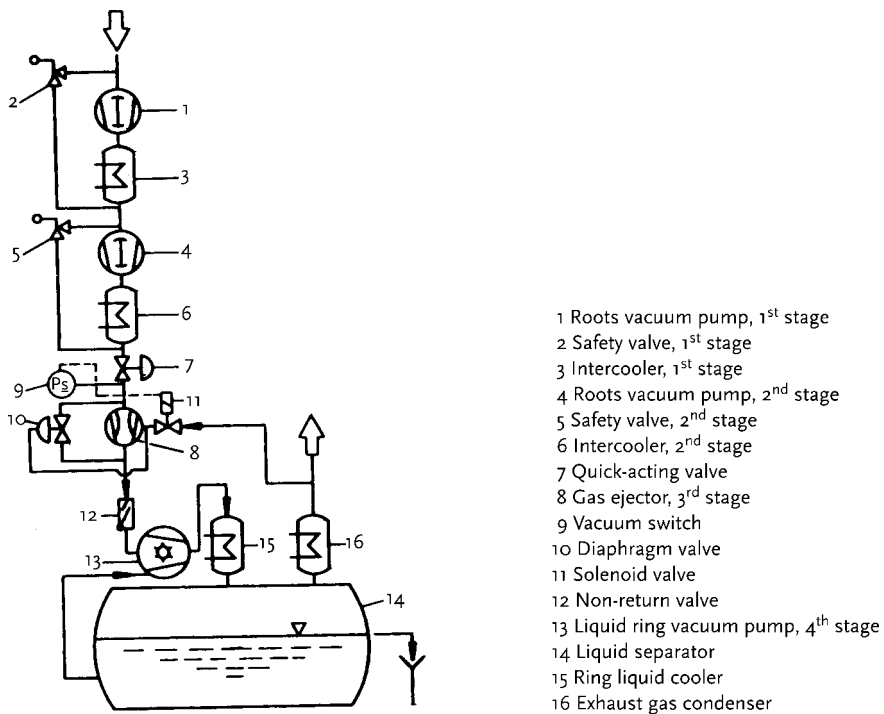


Figure 4-19. Scheme of a four-stage vacuum pump unit

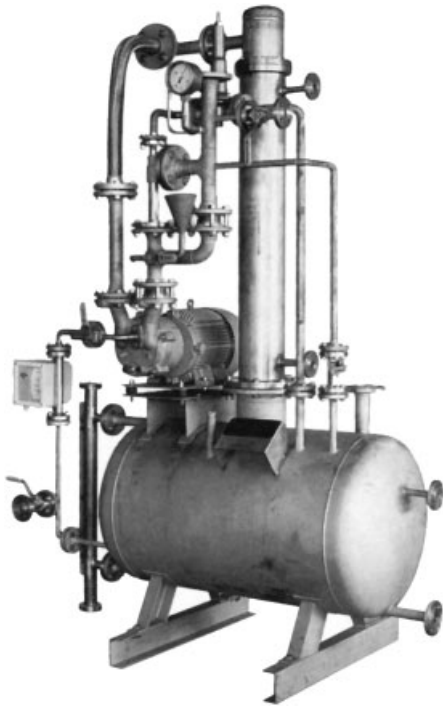


Figure 4-20 a. Two-stage vacuum pump package unit for gas and vapour extraction and for recovery of solvents (acc. to Lederle-Hermetic)

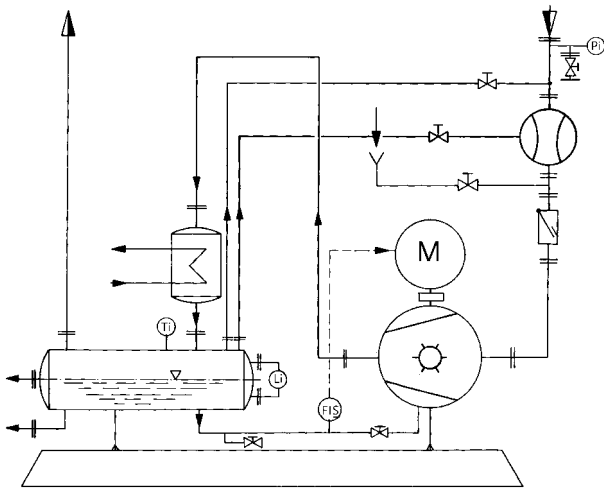


Figure 4-20 b. Functioning and installation diagram

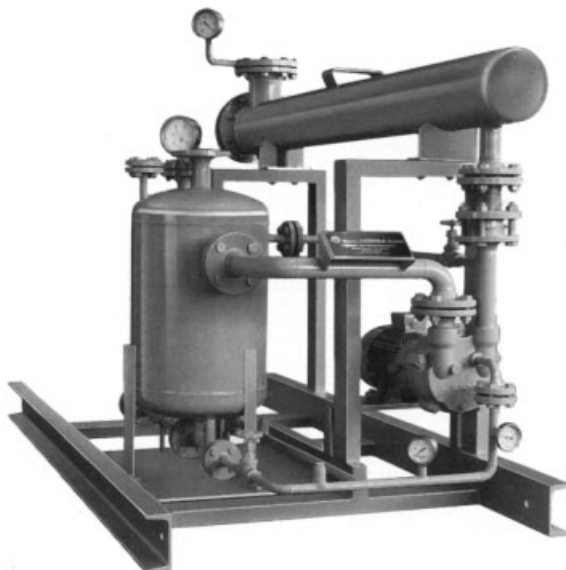


Figure 4-21 a. Single-stage vacuum pump unit for extraction and condensing of water vapour (acc. to Lederle-Hermetic)

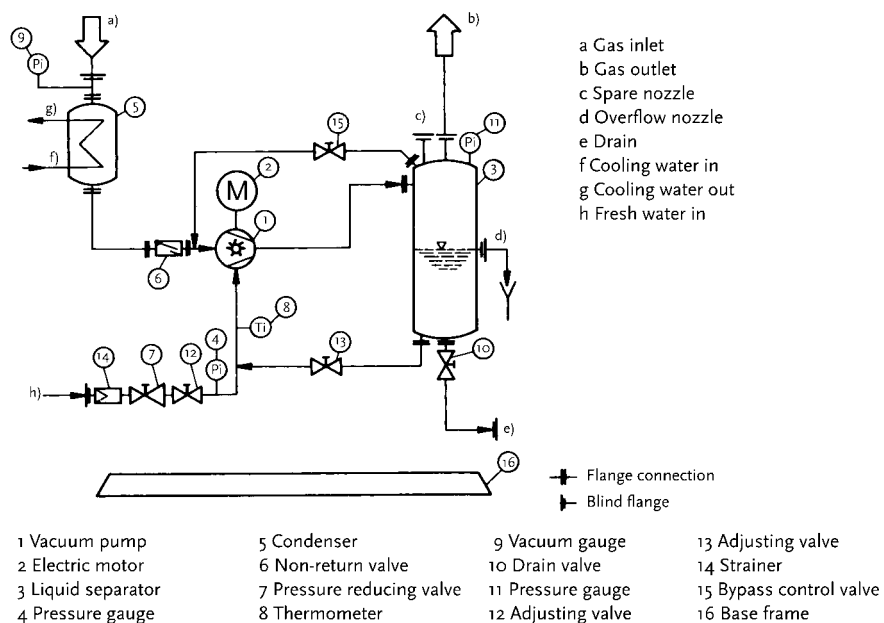


Figure 4-21 b. Functioning and installation diagram

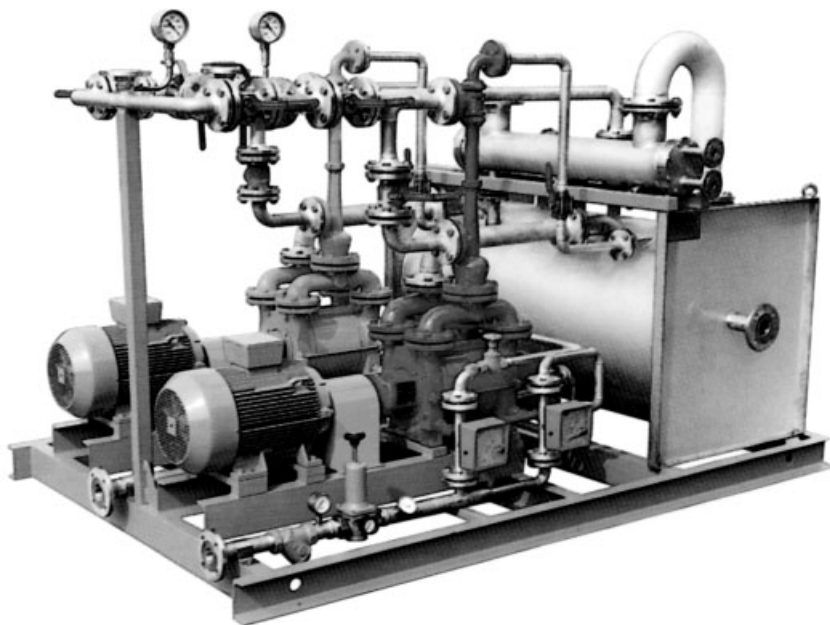


Figure 4-22 a. Two-stage vacuum pump system, for extraction and condensing of process gases from different vacuum vessels (acc. to Lederle-Hermetic)

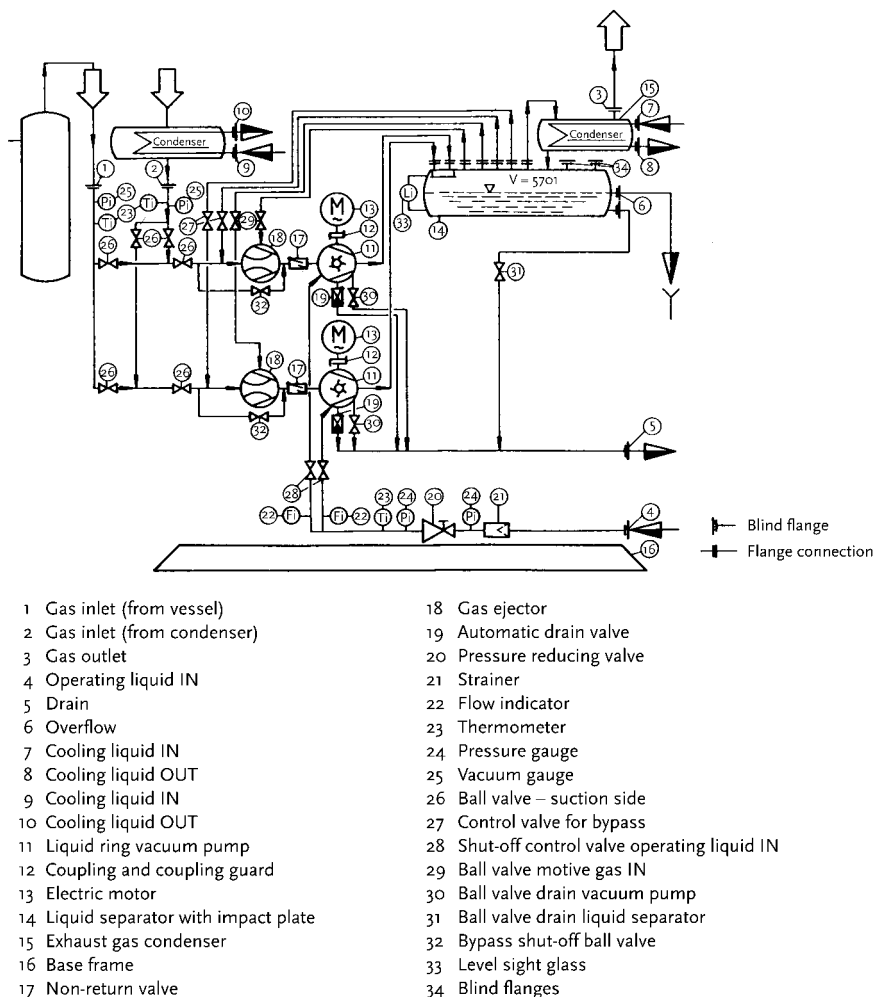


Figure 4-22 b. Functioning and installation diagram

Recovery of polluted solvents

The recovery plant shown in fig. 4-23 is operated on the principle of direct heat transfer with the contaminating liquid. This is a vacuum distillation plant with continuous supply. A solvent vapor acts as heat transferring gas. The solvent vapor is sucked by a ventilator under vacuum via a smoothing zone and a droplet separator (demister) and then led over the contaminated liquid via a heat exchanger where it is heated by means of hot vapor or heat transfer oil. In the following heat exchanger which has the function of a condenser, the distilled purified solvent is generated and then reaches the outlet buffer vessel.

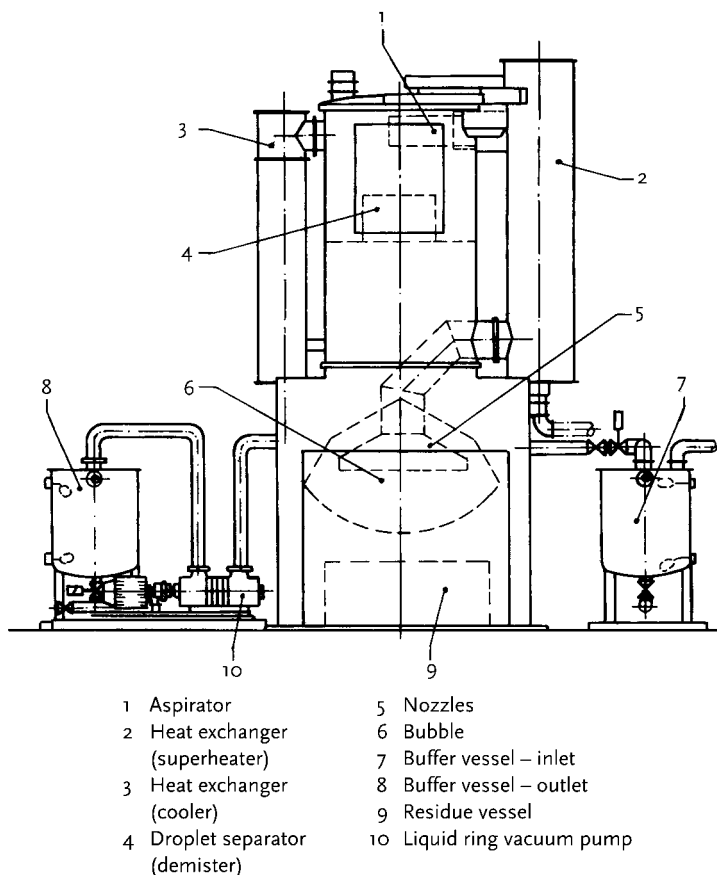


Figure 4-23. Solvent recovery system with liquid ring vacuum pump [4.8]

The residuals are collected in the lower part of the plant. They can be removed manually or, with the appropriate equipment being available, even automatically.

Plants of this design are usually manufactured in sizes with regeneration rates of 40 l/h to about 1000 l/h.

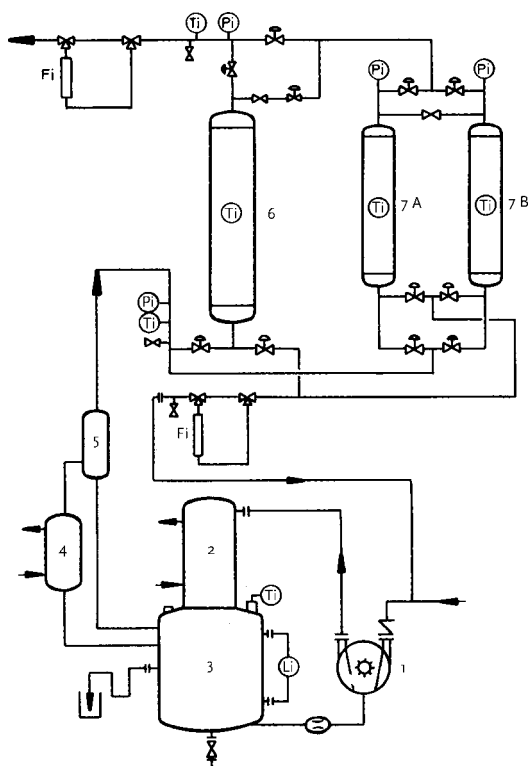
Adsorption of solvents by molecular sieves

With its “General regulation for keeping the air clean”, the “TA Luft” contains basic regulations for the interpretation and application of the Federal Immission Act. The adherence to the emission limits determined in these regulations, i.e. the permissible contaminations of the air arising from a plant, requires further developments in the chemical process engineering, as with conventional cooling liquids only limited condensation effects in the liquid ring vacuum pump system are achieved. For the cleaning of solvent-containing process gases and for the solvent recovery, a method for the adsorption of these substances by means of molecular sieves on the basis of

zeolites has been developed. Zeolites are crystalline aluminum silicates the crystal humidity of which is removed by thermal treatment during their manufacturing process without damaging their crystal structure. In adsorption plants, mostly granulates of different grain size are used. Zeolites can be naturally and synthetically produced.

In adsorption engineering, mainly synthetically generated zeolites are used featuring a large adsorption surface, high electrostatic adsorption forces and defined supply pores to the adsorption cavities. Due to the defined diameter of the supply pores the selective separation of substances according to their molecule diameter is possible.

Owing to the supply of intensely cooled dry air, the molecular sieves can be kept almost free from water and the adsorption effect is improved by the low temperature. For continuous operation, two or more molecular sieve columns must be applied, as the sieve elements will possibly have to be desorbed and cooled after



- | | |
|---------------------------|--|
| 1 Liquid ring vacuum pump | 5 Droplet separator |
| 2 Ring liquid cooler | 6 Starting adsorber |
| 3 Ring liquid tank | 7 Adsorber A and B for continuous pressure alternation |
| 4 Exhaust gas condenser | |

Figure 4-24. Solvent recovery system with three-bed pressure change adsorber (acc. to Sulzer-Burckhardt)

adsorption. The regeneration, also called desorption, usually takes place inertly or under vacuum in a closed circuit until a state of equilibrium is reached. Afterwards, the cooling circuit is activated, in order to cool the sieve element for adsorption.

Fig. 4-24 shows the instrumental structure of a solvent recovery system with a downstream exhaust gas cleaning device (acc. to [4.9]) consisting of a three-bed pressure change adsorber based on molecular sieves.

The process gas is sucked and compressed by the liquid ring vacuum pump (pos. 1). The compression pressure is slightly higher than the atmospheric pressure in order to overcome the pressure loss of the devices on the pressure side. In the ring liquid cooler (pos. 2), the gas-vapor mixture conveyed by the pump is cooled and possibly condensable substances are collected in a liquefied state in the ring liquid tank (pos. 3). The exhaust gas condenser (pos. 4) condenses possibly remaining vapor portions. Liquid carried by the gas, which may be the case especially at the beginning of the evacuation, is separated by the droplet separator (pos. 5). A further separation of vapors and gases takes place in the exhaust gas cleaning system installed after the droplet separator which works in three stages.

With the application of liquid ring vacuum pump systems combined with application-related adsorption technology, additional solvent portions can be recovered and very low emission is achieved.

4.4.2

Pump systems with hermetic liquid ring vacuum pumps and compressors

With the application of liquid ring machines and components without seals, the hermetic design is possible even for complete vacuum and compressor plants.

When installing safety heat exchangers with double-pipe bundles, possibly occurring leakages must be detected in time in order to avoid the contamination of the coolant. Shut-off devices can also be equipped with bellows or with drives with permanent magnet couplings, thus fulfilling the legal regulations of "TA Luft" (Technical Directions for Air Pollution Control).

The application of hermetic vacuum and compressing plants is preferred for environmentally-friendly production processes in chemistry, petrochemistry, pharmacy, nuclear technology, for the suction and neutralization of aggressive, toxic and malodorous gases and vapors, as well as for the economical recovery of solvents.

Advantages of the hermetic design are the maintenance-free operation, no sealing problems, no contaminated coolants and waste water, clean air.

Figs. 4-25a, 4-26a, 4-27a, 4-28a, 4-29a, 4-30 and 431a show pump systems of hermetic design as they are applied in different branches of industry for the extraction of environmentally harmful substances or when handling hazardous ring liquids.

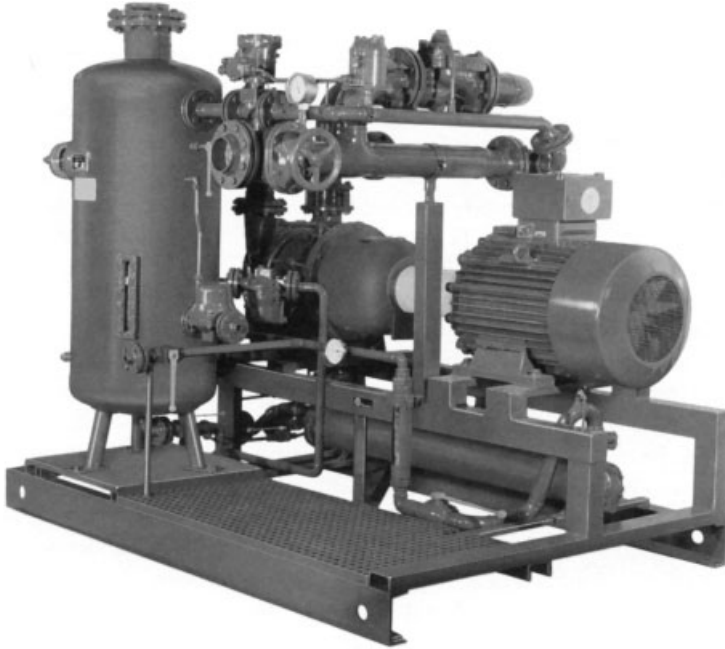


Figure 4-25 a. Hermetical two-stage vacuum degassing system (acc. to Lederle-Hermetic)

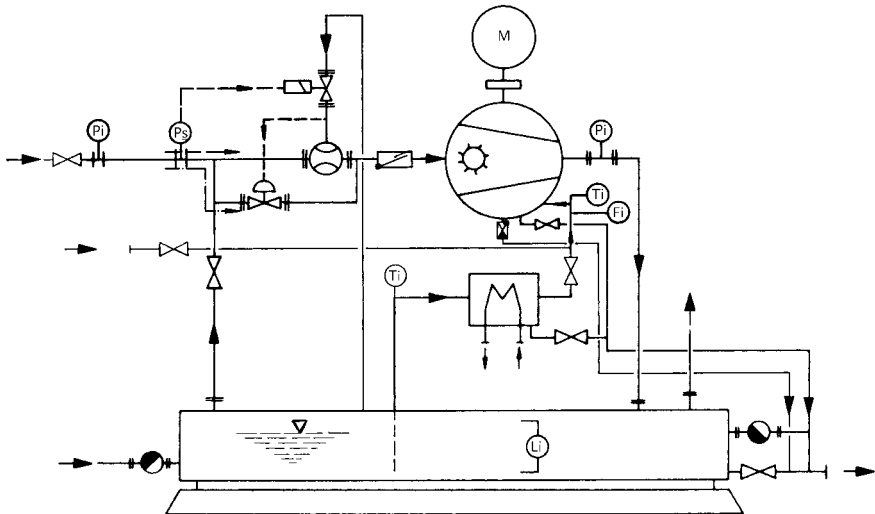


Figure 4-25 b. Function and installation diagram



Figure 4-26 a. Hermetical single-stage vacuum pump unit of special design (acc. to Lederle-Hermetic)

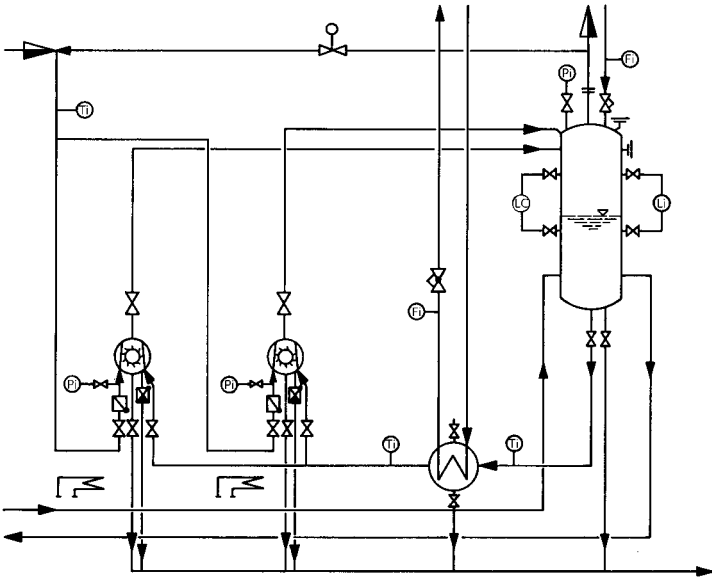


Figure 4-26 b. Functioning and installation diagram

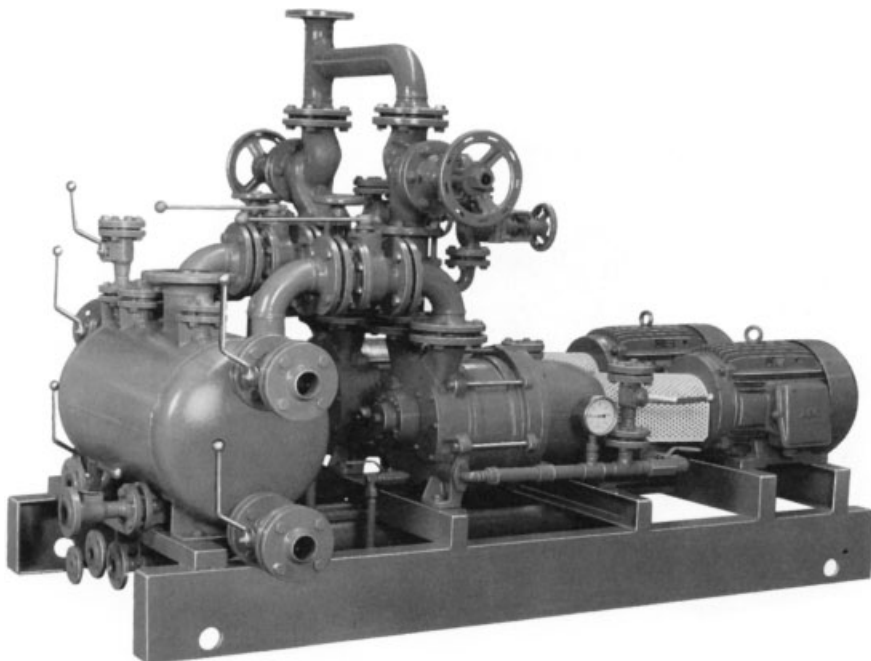


Figure 4-27 a. Single-stage hermetical liquid ring compressor with horizontally arranged liquid separator (acc. to Lederle-Hermetic)

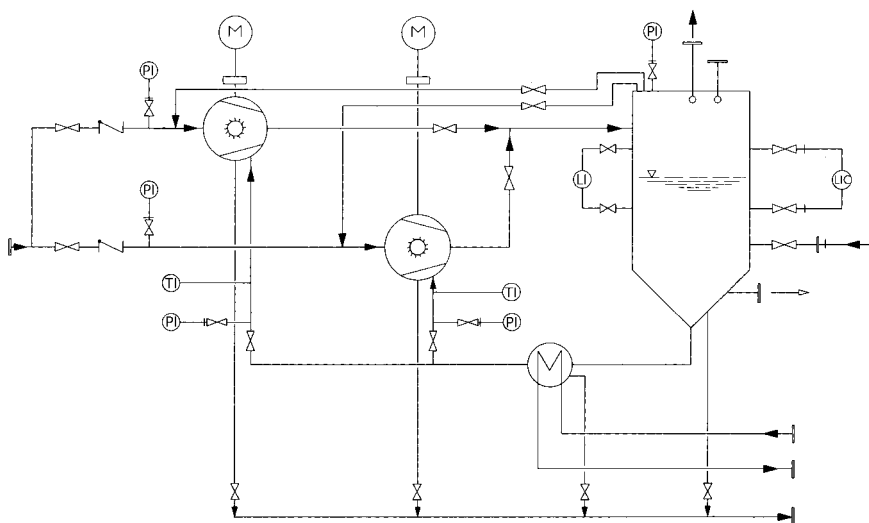


Figure 4-27 b. Functioning and installation diagram

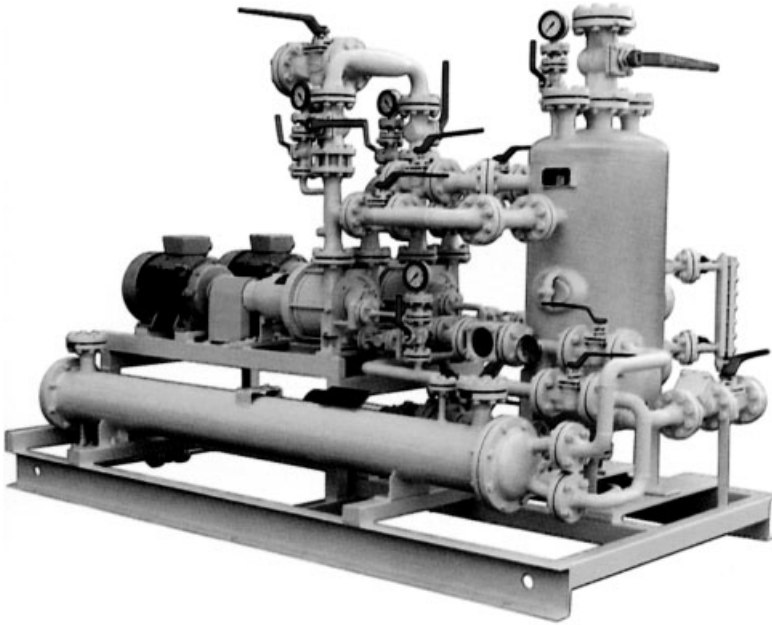


Figure 4-28 a. Compressor package unit in single stage hermetical design (acc. to Lederle-Hermetic)

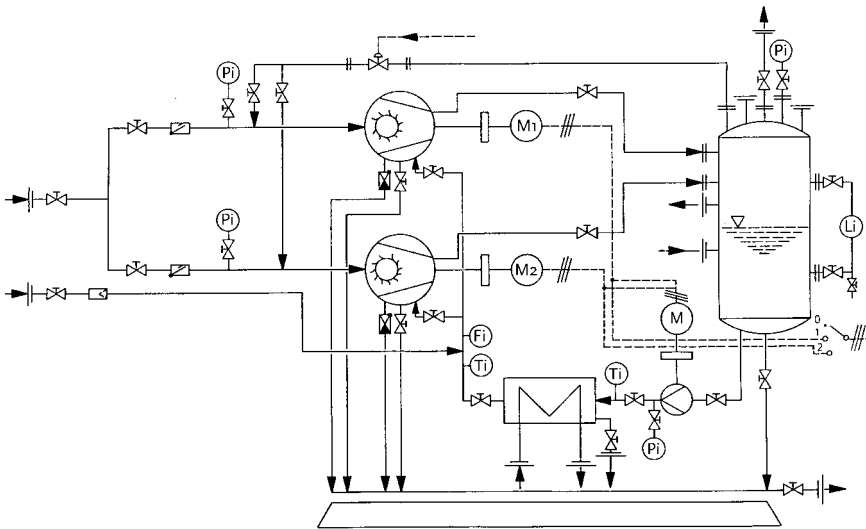


Figure 4-28 b. Functioning and installation diagram

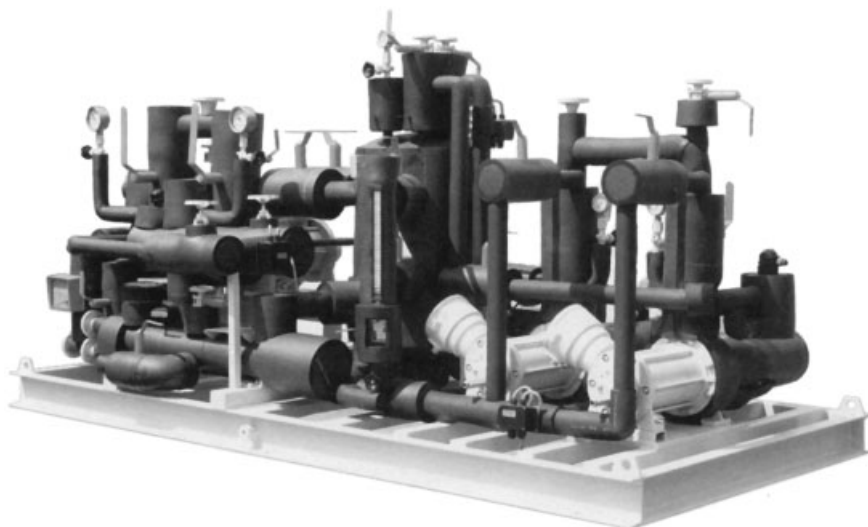


Figure 4-29 a. Electrically heated and isolated vacuum system equipped with hermetical liquid ring pump with canned motor (acc. to Lederle-Hermetic)

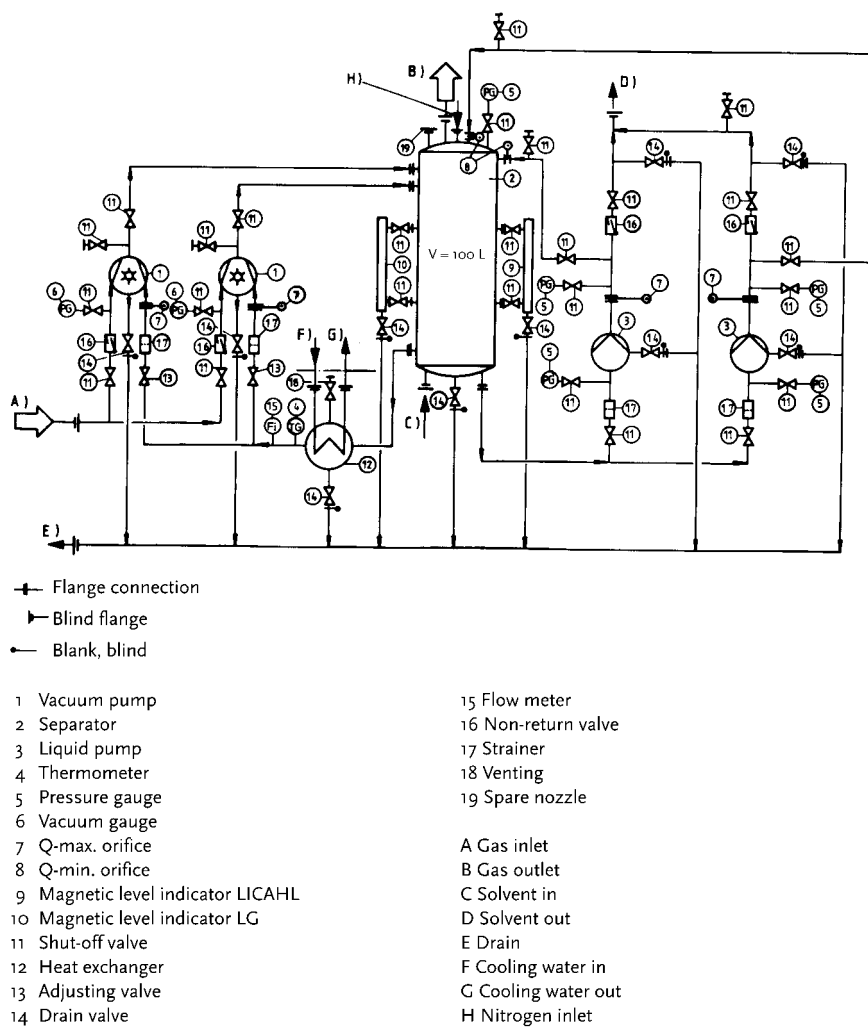


Figure 4-29 b. Functioning and installation diagram

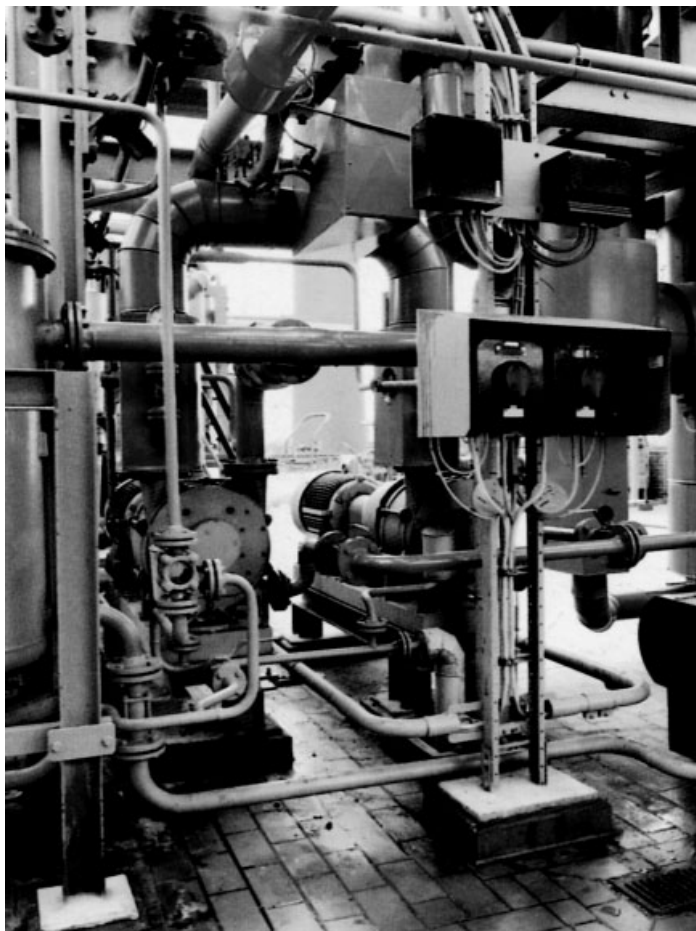


Figure 4-30. Hermetic liquid ring machines in the vacuum system of a chemical plant (acc. to Lederle-Hermetic)

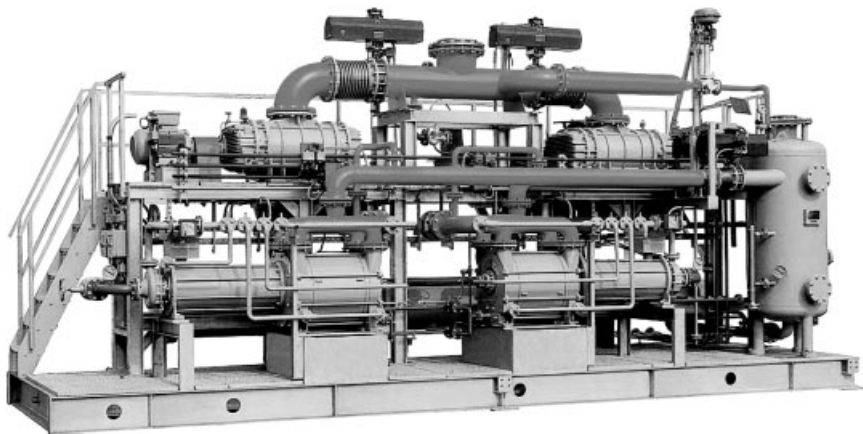


Figure 4-31 a. Two-stage vacuum system with hermetical liquid ring vacuum pumps for recovery of aromatic compounds (acc. to Lederle-Hermetic)

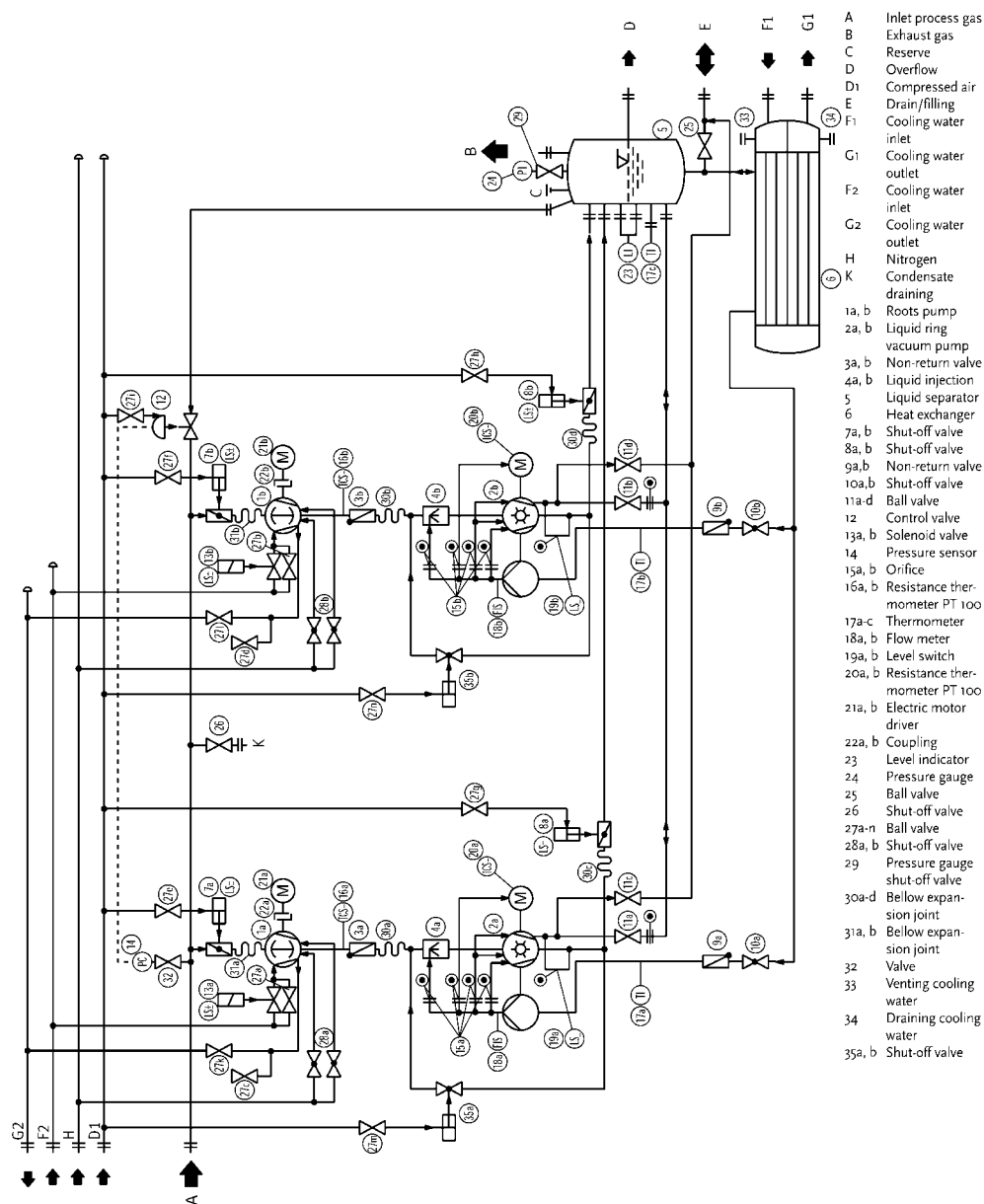


Figure 4-31b. Functioning and installation scheme

4.4.3

Vacuum pump unit of special design for the suction of polluted process gases

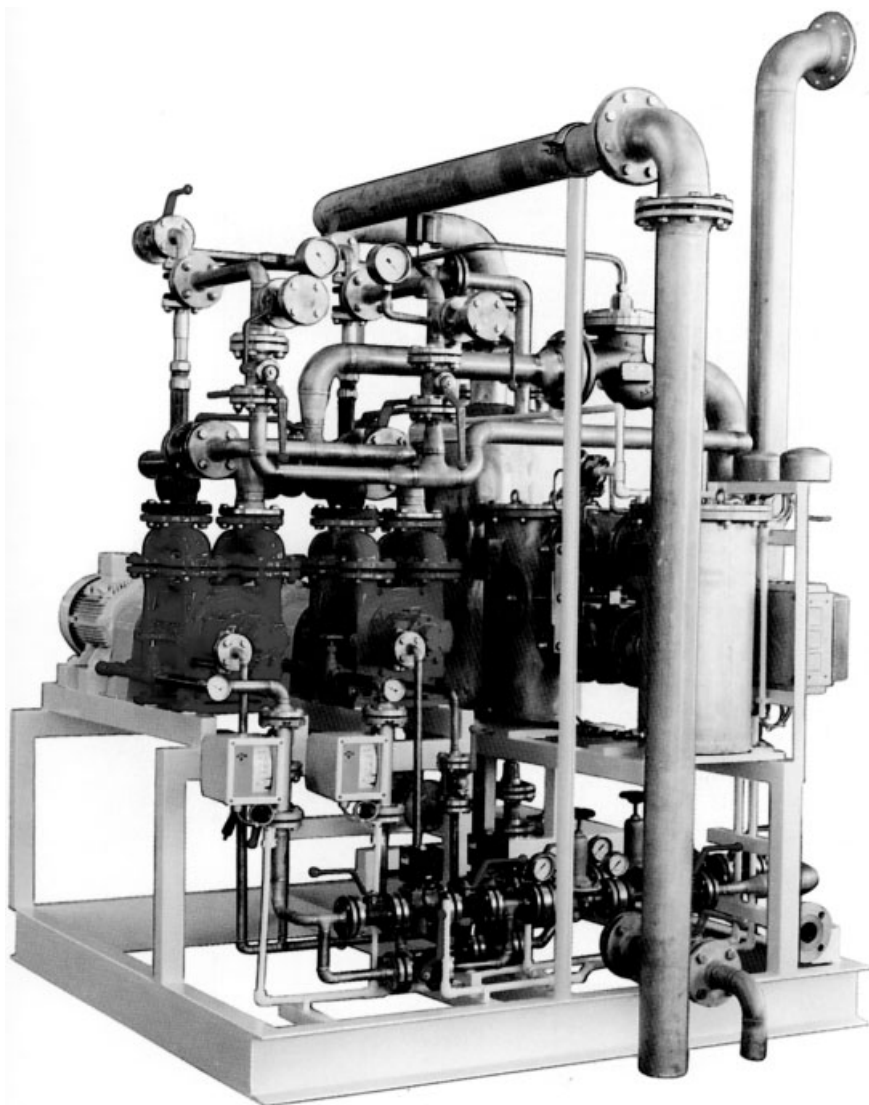


Figure 4-32 a. Two-stage vacuum pump unit with special equipment (acc. to Lederle-Hermetic)

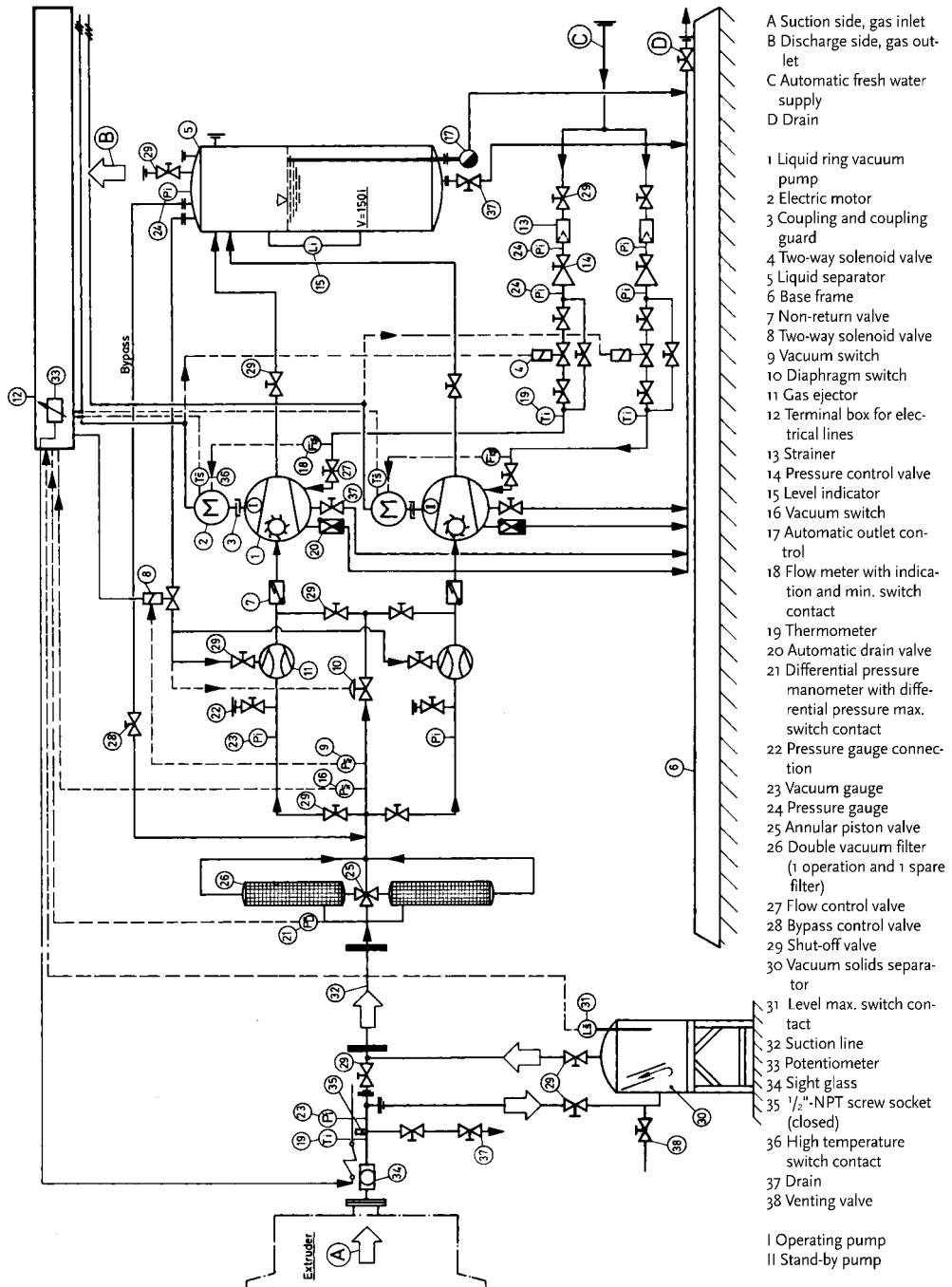
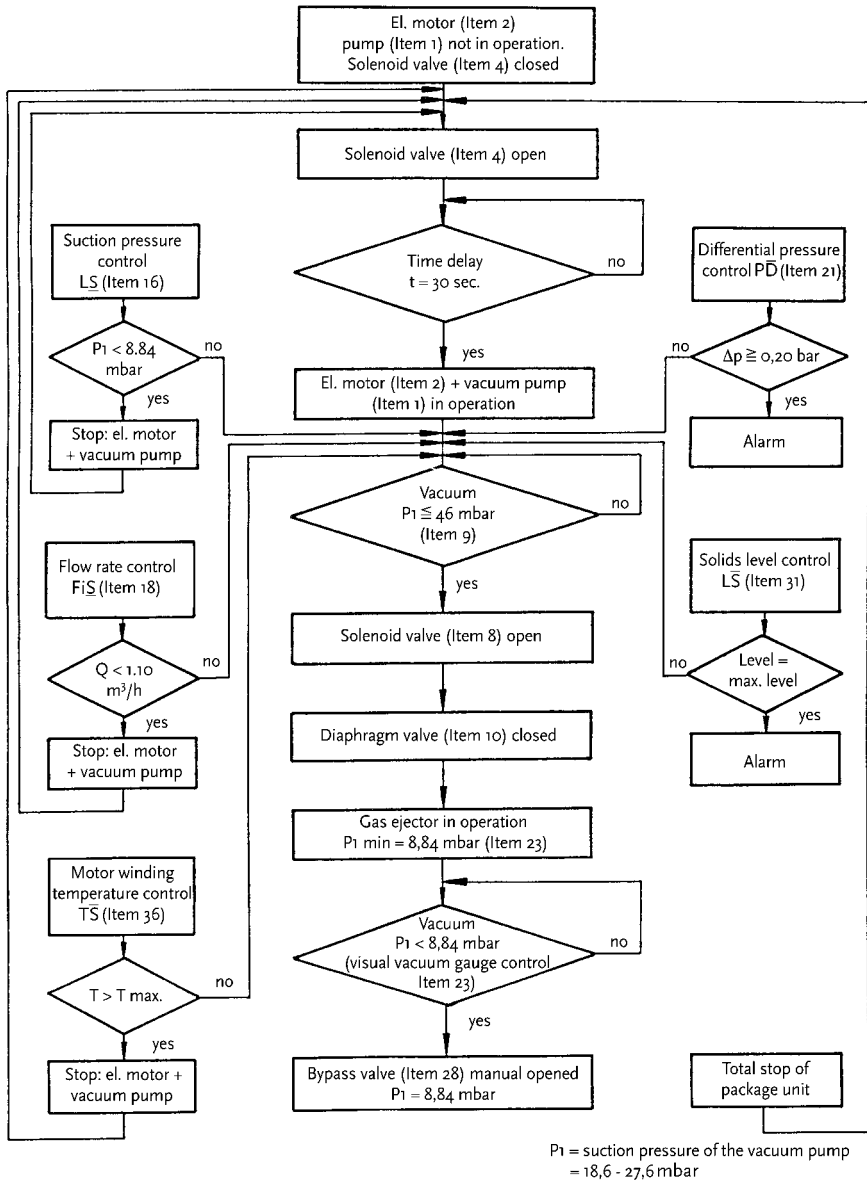


Figure 4-32 b. Functioning and installation diagram



Start conditions:

1. Operation pump: all manual operated valves are open in vacuum pumps suction and discharge pipes and in operation fluid supply pipe. Vacuum pump filled with operation fluid. Operation liquids pressure is adjusted. Drain valve D opened. The shut-off valve bypassing solenoid valve (Item 4) is closed. Bypass valve (Item 28) is closed. Heating (Item 33) adjusted on $29,5^\circ\text{C}$.
2. Spare pump: all manual operated valves are closed.

Figure 4-32 c. Timing diagram of vacuum pump unit acc. to fig. 4-32 a, 4-32 b

4.4.4

Steam jet liquid ring vacuum system of corrosion-resistant design

For the generation of vacuum, in process engineering, devices such as evaporators or dryers and e.g. in distillation and rectification plants, steam-jet water-ring vacuum pumps are used. This combination with a water ring vacuum pump is especially practical when barometric heights of fall do not exist. Further advantages of steam jet compressors with water ring vacuum pumps on the pressure side are low costs for vapor, water and energy consumption.

For cases with particularly high demands, such as the suction of aggressive substances in the chemical and pharmaceutical industry, materials such as porcelain, graphite and plastics are used for steam jet water ring pumps acc. to [4.10].

Porcelain as a material for steam jet pumps is resistant against almost all acids, lye and solvents, and against temperature changes. With wall thicknesses of 10 mm

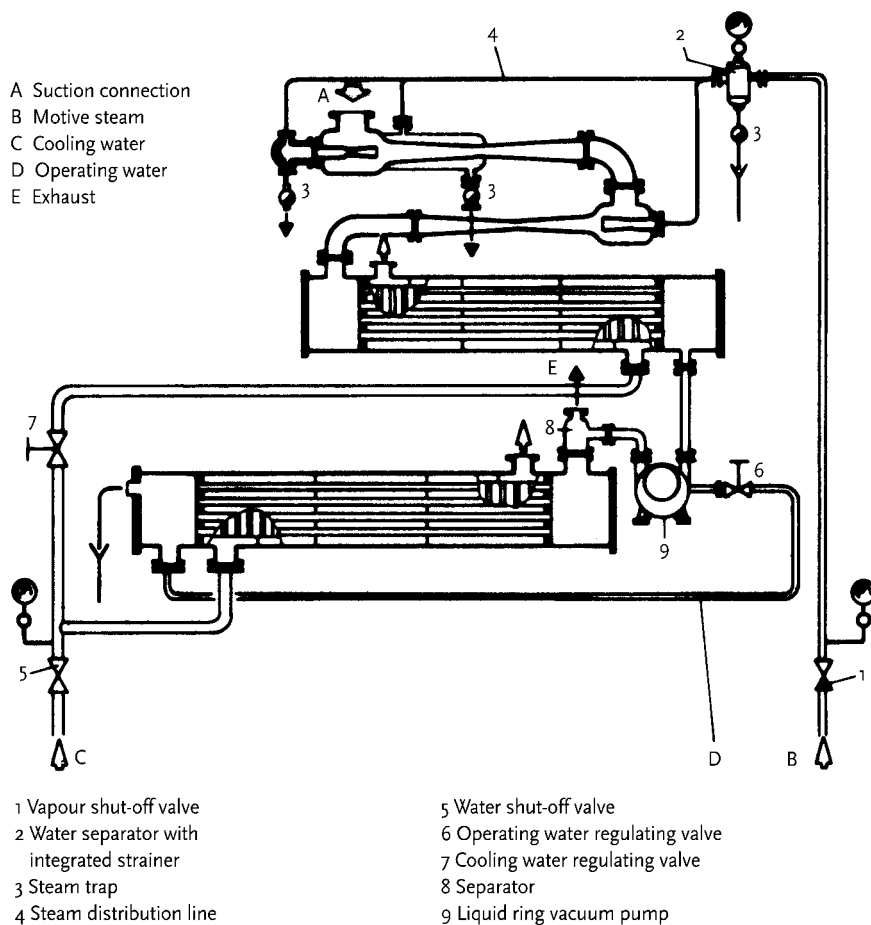


Figure 4-33. Two-stage steam-jet liquid ring vacuum pump (acc. to Wiegand)

to 30 mm, there is no danger of breakage, proper use provided. Graphite proved its worth as material for steam jet pumps and surface condensers, above all in cases in which rustproof and acid-proof steels do not show sufficient resistance.

Fig. 4-33 shows a two-stage steam jet water ring vacuum pump. The steam ejectors are made of porcelain, while the surface condenser and the block heat exchanger – a cooler for the operating liquid of the water ring vacuum pump – are of graphite. The hermetic liquid ring vacuum pump with magnetic drive can be manufactured of Ryton, Noryl, PTFE or similar material. Materials have to be selected in accordance with physical and chemical properties of the conveyed process gases and the ring liquid used. Fig. 4-33 shows a pump combination the air suction mass flow of which amounts to 2.5 kg/h at a suction pressure of 3.0 [mbar (abs)].

4.4.5

Selection of application examples for liquid ring machines

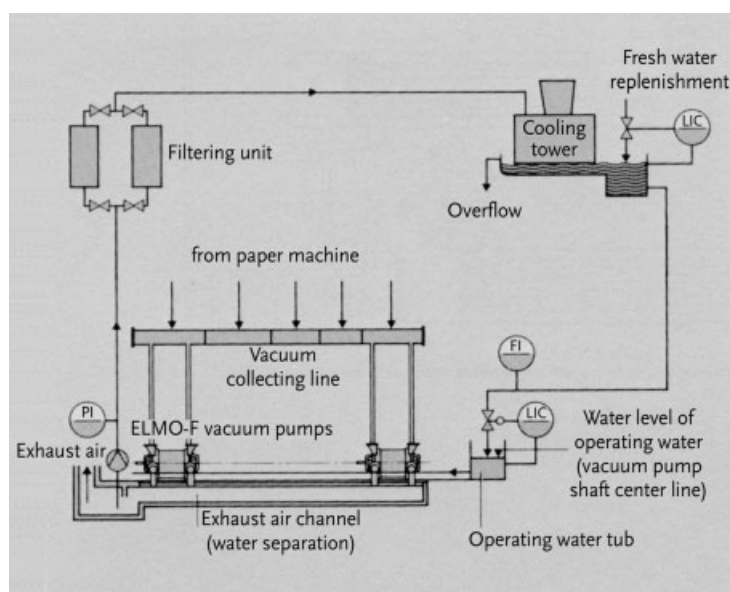


Figure 4-34. Liquid ring vacuum pump with closed operating liquid circuit installed in a paper production plant (acc. to Siemens)

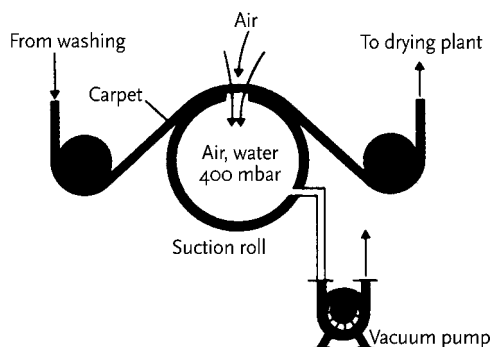


Figure 4-35. Extraction of residual water with liquid ring vacuum pumps in carpet manufacturing (acc. to Siemens)

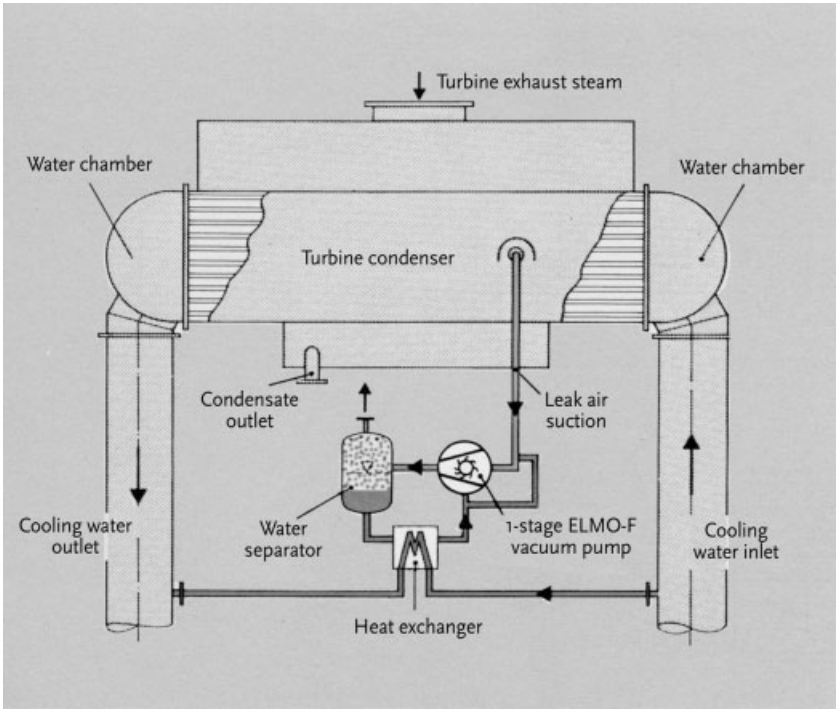


Figure 4-36. Liquid ring vacuum pump with suction capacity $>3000 \text{ m}^3/\text{h}$ for start-up and condensation operation of a steam turbine condenser (acc. to Siemens)

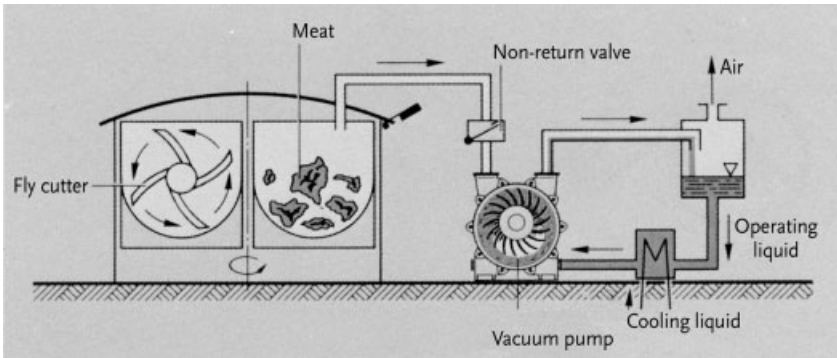


Figure 4-37. Liquid ring vacuum pump with closed operating liquid circuit used in a meat processing plant (acc. to Siemens)

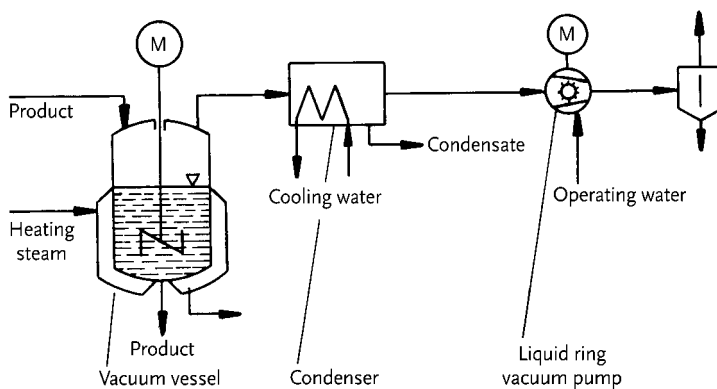


Figure 4-38. Liquid ring vacuum pump used in a vacuum evaporation plant (acc. to Siemens)

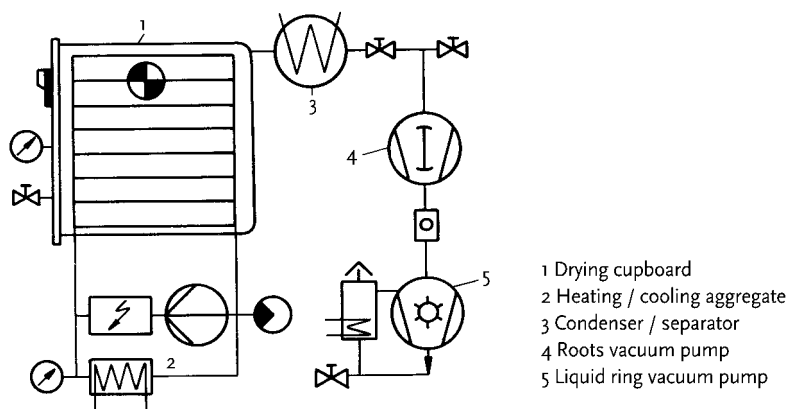


Figure 4-39. Vacuum drying plant with a liquid ring vacuum pump and closed operating liquid circuit (acc. to Hedrich)

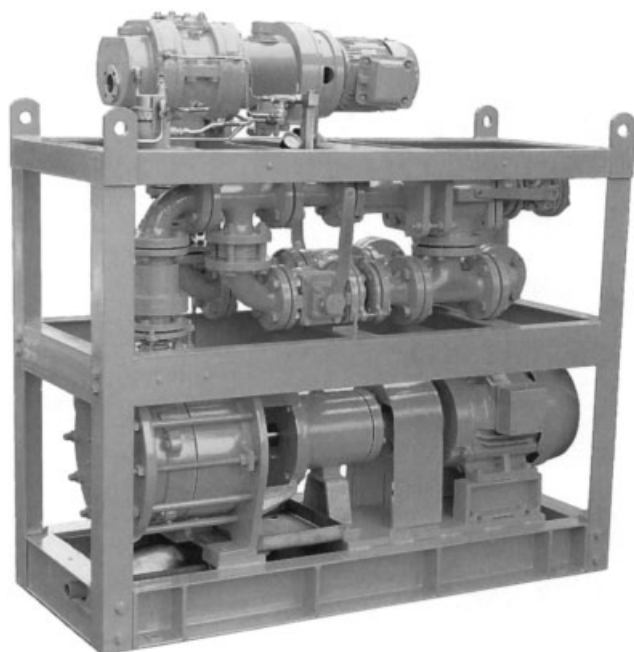


Figure 4-40. Three-stage ceramic vacuum pump unit with Roots vacuum pump, gas ejector and liquid ring vacuum pump, manufactured of ceramics, for extraction of highly corrosive gases and vapours (acc. to NGK)

4.5

Electric heating and insulation on pumps and plants

In production plants of the chemical and petrochemical industry, as well as in heavy industries, it is common to equip the pipework, measuring pipes, vessels and other plant components with electric heating and insulation casing. Depending on the fluid to be conveyed, unprotected pipes and elements are targets for the effects of frost. In order to keep the conveyed medium flowable, to prevent the solidification and to avoid the destruction of plant parts by freezing, an appropriate heating or insulation of the endangered parts is recommended. There is a risk of the gases and vapors condensing during their discharge if the respective measures are not taken. In plant parts in which fluids with temperatures below the ambient temperature are conveyed, sweat condensation or icing is threatening. These conditions may occur both during operation and at a standstill of the plant. Arising heat losses are then compensated by conventional insulation or additional heating on pipework, vessels, heat exchangers and also on the pumps. For the electrical heating, e.g. mineral-insulated heating cables, plastic-insulated heating pipes, parallel resistance heating bands or self-limiting temperature-regulated heating bands are available. The heating bands, heating cables and heating pipes can be used in explosion-prone areas, if

suitable connections are provided. For the layout of heating systems, design data, temperature conditions and possible explosion protection regulations have to be considered, as for example:

- pipe diameter
- insulation thickness
- desired temperature
- minimum ambient temperature
- temperature maintenance or heating-up
- heating-up time when heating
- temperature class (ignition group)
- medium properties and
- maximum permissible temperature.

In the vacuum plant shown in fig. 4-41, gas conveying components are equipped with an electrical heating. The gas- and liquid conveying components are insulated.



Figure 4-41. Liquid ring vacuum pump system with electrical heat tracing and insulation (acc. to Lederle-Hermetic)

4.6

Names and definitions – vacuum systems, components and equipment

(extract from DIN 28400, Part 8 – October 1980)

Initial pressure. The initial pressure is the pressure in a vacuum vessel, at which a certain process can begin.

Note: In some vacuum processes, e. g. for surface coating analyses, the initial pressure is also called “basic pressure”.

Operating pressure. The operating pressure is the pressure (or even pressure range) in the vacuum vessel required for a satisfactory execution of the process.

Pumping-out time of a vacuum system. The pumping-out time of a vacuum system is the period of time which is needed to lower the pressure in a vacuum system from the atmospheric pressure to a certain pressure, e.g. initial pressure.

Bypass valve. A bypass valve is a vacuum shut-off valve in a bypass tube.

Pressure rise rate of a vacuum vessel. The pressure rise rate of a vacuum vessel separated from a pump system is the pressure rise at a constant temperature during a period of time, divided by the period of time. Possibly, this quotient is not constant.

Final pressure. The final pressure is the limit value which the pressure in the unloaded (dry and empty) vacuum vessel approaches asymptotically while the pump is in operation.

Final vacuum valve. A final vacuum valve is the vacuum shut-off valve which is used for shutting-off the vacuum vessel at the final vacuum pump.

Liquid-sealed vacuum joint. A liquid-sealed vacuum joint is a separable vacuum joint which is sealed by means of a liquid with low vapor pressure.

Flooding valve. A flooding valve is a vacuum shut-off valve which is used for the direct inlet of gas into the vacuum system. If it is flooded with air, the valve is called “air valve” or “air inlet valve”.

Flooding time of a vacuum system. The flooding time of a vacuum system is the period of time necessary to increase the pressure in the vacuum system (or in the vacuum vessel) from the operating pressure to a higher pressure, mostly atmospheric pressure, by means of gas inlet over the provided facility. In case of an air inlet this period is called “ventilation time”.

Gas discharge rate of a vacuum system. The gas discharge rate of a vacuum system is the gas quantity (stated as $p\dot{V}$ flow), which is to be attributed to the desorption of gases from all surfaces of the vacuum system.

Note: In practice, a gas discharge inside a vacuum system feigns a leak. In this case, it is called “virtual leak”.

Gas inlet valve. A gas inlet valve is a vacuum control valve for the inlet of gases into a vacuum system.

Coarse vacuum line. A coarse vacuum line is a vacuum line system connecting a vessel to be evacuated with a vacuum pump intended for the coarse evacuation of the vessel.

Coarse vacuum valve. A coarse vacuum valve is a vacuum shut-off valve in the coarse vacuum line needed for shutting-off the vacuum vessel from its coarse vacuum pump.

High vacuum valve. A high vacuum valve is a vacuum valve the characteristic properties of which meet the requirements of high vacuum technology. It is mostly used in this vacuum range.

Leakage rate of a vacuum system. The leakage rate of a vacuum system is the quantity of gas penetrating the vacuum system through the leaks (stated as $p\dot{V}$ -flow rate) affecting the pressure in the vacuum vessel.

Separable vacuum joint. A separable vacuum joint is a vacuum joint which can be easily disconnected and connected again, mostly mechanically.

Permanent vacuum joint. A permanent vacuum joint is a vacuum joint which cannot be easily connected and disconnected again. Examples: soldered vacuum joint, welded vacuum joint, glass-glass- or glass-metal fusion.

Residual pressure; residual gas pressure. The residual pressure is the total pressure of the still existing gas or gas mixture (residual gas) in a vacuum vessel after a certain evacuation time or after finishing a vacuum process. In some cases, the residual pressure equals the final pressure.

Note: In vacuum engineering, the word “gas” is used, in its broader sense, both for a non-condensable gas and for vapor, if a distinction is not required.

Residual gas spectrum. The residual gas spectrum is the mass spectrum of the residual gas in a vacuum chamber.

Suction power of a vacuum pump unit. The suction power of a vacuum pump unit is the suction power ($p\dot{V}$ flow) in the inlet port of the vacuum pump unit.

Suction capacity of a vacuum pump unit. The suction capacity of a vacuum pump unit is the suction capacity (volume flow) in the inlet port of the vacuum pump unit.

Sight glass. A sight glass is a vacuum window serving as insight device.

Note: In some cases of application, certain requirements concerning optic properties of sight glasses must be fulfilled.

Ground-glass-joint. A ground-glass joint is a separable vacuum joint consisting of two ground surfaces. The ground surfaces can be of flat, globular or conical shape; they are usually greased.

Melted metal vacuum joint. A melted metal vacuum joint is a separable vacuum joint sealed with metal of a low melting point. For connecting or disconnecting the seal, the metal is heated.

Semi-permanent vacuum joint. A semi-permanent vacuum joint is a vacuum joint made of wax, glue, lacquer or similar substances.

Ultrahigh vacuum valve, UHV valve. An ultra-high vacuum valve is a vacuum valve the characteristic properties of which fulfill the requirements of the ultra-high vacuum engineering and which is mostly used in this range of vacuum. Valve seats and seals of ultra-high vacuum valves are usually made of metal. Ultra-high vacuum valves are heatable.

Bypass line. A bypass line is a vacuum line system installed parallel to the parts of the vacuum system. It can be operated parallel to these parts or even individually.

Vacuum shut-off valve. A vacuum shut-off valve is a vacuum valve which separates two parts of a vacuum system from each other. Usually, it is not used as control valve.

Pre-vacuum vessel. A pre-vacuum vessel is a vessel installed between a vacuum pump requiring a pre-vacuum and its pre-vacuum valve. It is used to collect the conveyed gas when the pre-vacuum pump is switched off, and/or to balance pressure fluctuations.

Vacuum vessel, vacuum chamber, vessel. A vacuum vessel is a vacuum-tight chamber the mechanical design of which allows the generation of pressures inside the vessel which are lower than the pressure of its environment.

Vacuum sealing ring. A vacuum sealing ring is a ring shaped vacuum seal.

Note: There are vacuum sealing rings of different cross-section shapes, e.g. O-ring seal, V-ring seal, L-ring seal and other profile seals (metal profile seals).

Vacuum seal. A vacuum seal is a deformable component to be inserted between the two parts of a separable vacuum joint, in order to seal it. In some cases, it is completed by supporting parts (e.g. supporting rings). The selection of the material depends on the desired vacuum range; usually it is elastomere or metal.

Vacuum throttle. A vacuum throttle is a special piece of a vacuum line, which is used for limiting the gas flow through the line. As a rule, it is an orifice or a capillary.

Vacuum transmission line. A vacuum transmission line is a device for the transmission or carrying out of

- movements,
- gases or liquids,
- electric currents or
- electric voltages

through the wall of a vacuum vessel. Such a device is usually carried by a flange closing the vacuum vessel against the atmosphere. A vacuum transmission line for the transmission of movements which enables many movements in vacuum, mostly translations and rotations, is called “manipulator”.

Vacuum window. A vacuum window is an appliance in the wall of a vacuum vessel, which allows the penetration of electromagnetic or corpuscular radiation (e.g. Lenard’s window).

Vacuum flat gasket. A vacuum flat gasket is a vacuum seal manufactured of a flat material.

Vacuum flange joint. A vacuum flange joint is a separable vacuum joint creating a vacuum-tight sealing by means of a suitable deformable seal between two flanges.

Note: The sealing surface of the flanges can be smooth or grooved, or it can have a profile which, together with a specific seal, is suitable to develop a sufficient sealing effect. The flanges can be pressed together by means of screws, clamps or quick-clamping elements.

Vacuum bell. A vacuum bell is a bell-shaped component mounted onto another component, usually a base plate, by means of a separable joint creating a vacuum chamber together with this other component.

Vacuum condenser; vapor condenser. A vacuum condenser is a vacuum vessel with cooled inner surfaces which is installed between vacuum chamber and the pump system for the pumping of large vapor quantities. Usually it has a vessel for conden-

sate collection which can be shut off and allows draining of liquid condensate without interruption of the vacuum process.

Vacuum manifold. A vacuum manifold is a vacuum tight distributing element, with the possibility of connecting two or more vacuum chambers for simultaneous evacuation.

Vacuum control valve. A vacuum control valve is a vacuum valve enabling the adjustment of the $p\dot{V}$ flow between the parts of a vacuum system which are separated by the vacuum valve. (Frequent application: needle valve)

Vacuum lock. A vacuum lock is a vacuum chamber applied between two chambers of different pressures. It is equipped with appliances through which its pressure can be adapted to the pressure of one or the other neighboring chamber, and with orifices (completely or partly lockable) through which objects can be transported from one chamber to the other without creating disturbing pressure changes in these chambers. They serve mainly for input and output of objects from the atmosphere into the vacuum vessel or vice versa.

Vacuum jacket. A vacuum jacket is a vacuum-tight vessel fully or partly surrounding another vacuum vessel. It serves the reduction of the $p\dot{V}$ -flow through the leaks and/or of the mechanic forces acting onto the walls. The vacuum existing inside is called "protective vacuum".

Vacuum valve. A vacuum valve is a valve the design of which fulfils certain vacuum-specific requirements. Valve designs particularly often used in vacuum engineering are e.g. stuffing box packing, spring-bellows valve, diaphragm valve and all-metal valves.

Vacuum shaft seal. A vacuum shaft seal is a vacuum seal for the sealing of a shaft, which enables the transmission of rotations and/or translations relatively to the walls of the vacuum vessel. For shaft seals, lip packing is preferably used.

Sealed vacuum vessel. A sealed vacuum vessel is a vacuum vessel which is sealed-off after evacuation or closed by means of another method with the help of a permanent joint. Examples: electron tubes, X-ray tubes.

Pre-pumping time. The pre-pumping time is the period of time a pre-vacuum pump or a pre-vacuum pump system needs to evacuate a vacuum system from the atmospheric pressure to the initial pressure or starting pressure of a vacuum pump operating at low pressure.

Pre-vacuum line. A pre-vacuum line is a vacuum line system connecting a pre-vacuum pump with its corresponding vacuum pump.

Pre-vacuum valve. A pre-vacuum valve is a vacuum shut-off valve in the pre-vacuum line, for shutting-off the pre-vacuum pump from the corresponding vacuum pump.

Time constant of a vacuum system. The time constant of a vacuum system is the period of time required for decreasing the pressure in the vacuum chamber to an e-value. At a constant suction capacity, the time constant equals the quotient of the vacuum vessel volume and the suction capacity of the pump system.

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5

Components for Pump Units with Liquid Ring Vacuum Pumps and Compressors

5.1

General criteria

When designing vacuum plants, for economical reasons industrially manufactured components at the present state of the art should be taken into consideration. Apart from pumps, this comprises heat exchangers, separators, measuring gauges, simple vessels, bushings of all kinds, pipework components, shut-off and throttling valves as well as seals.

Components for rough, fine, high and ultra-high vacuum do not only differ from each other regarding their flanges and seals, but also with regard to the materials used and the surface finish that has to be adapted. In the rough and fine vacuum range, mainly elastomere seals are used. In technological processes, organic substances cannot be used as sealing material in the high and ultra-high vacuum range. In this case, the materials of these components have to meet particular requirements. Since valves and plant components will probably have to be heated and the emission of organic vapor should be avoided, metal seals are used. Valve components such as valve heads and casings are mostly made of high-alloy nickel chromium steel. The valve seat seals have to be designed in a way to be able to withstand a lot of closing cycles. Only for lower demands Viton or PTFE seals are used, essentially in consideration of heating temperatures that may be up to a maximum of 180 °C. Shaft ducts are usually designed with metal bellows.

Nominal widths of components such as pipeworks, valves etc. are chosen from a vacuum-technological point of view with regard to suction capacity and flow conductivity. In the interest of small losses, high flow conductivity is required. Therefore, even in the planning stage attention has to be paid to cable runs as short and straight as possible and sufficiently large flow cross sections have to be considered.

The optimal layout of a vacuum system presupposes the application of a minimum number of flange joints, shut-off and throttling valves and ducts of any kind – as far as this is permissible for the respective kind of problem. Each valve – even in an opened state – means additional flow resistance.

Angle valves dispose of specifically larger resistances than straight-way valves. Each sealing point on components such as valves, flanges or ducts shows leakage rates. Elastomeric seals, organic lubricating and sealing materials emit gases and

vapors into the vacuum chamber that affect the final pressure and the residual gas composition. Owing to the application of vacuum components with metal-bellows ducts or magnetic couplings as well as liquid and vacuum pumps in hermetic design, these problems can almost be avoided. Furthermore, the use of these “Hermetic Components” reduces the fair wear and tear, and with this costs for maintenance and repair. Even in cases that require absolutely tight circuits and systems due to reasons concerning the environmental protection, it is advisable to install hermetic machines and components without any leakage.

5.2

Liquid separators

With liquid ring vacuum pumps and compressors, together with the compressed gas, a part of the operating liquid is carried. Therefore, usually a liquid separator is installed in the pressure pipe. In the liquid separator the heavy liquid particles are separated from the light vapor or gas. The dimensions of the liquid separator depend on the accruing gas flow and the operating liquid quantity in the liquid ring machine.

Liquid separators are manufactured as round vessels with torospherical heads, in vertical or horizontal design. Round vessels with flat bases are used as well as square or rectangular vessels. Economical solutions are the horizontally installed gravity liquid separators in double-flow design acc. to [5.14]. There is a variety of most different designs, only some of which will be described in the following (see also sections 3.14, 4.3, and 4.4).

Liquid separator with impact plate

The impact plate separator as shown in figure 5-1 is one of the most frequently used designs. Liquid particles entering together with the gas flow are separated by change of direction using the mass forces. The impact plate forces such a sharp deviation of the gas flow that the liquid is not able to follow. The liquid particles touching the impact plate reflect and fall into the separator chamber.

Potential vortex separator

The two-stage potential vortex separator shown in fig. 5-2 separates about 90% of the arriving liquid particles in the rough separation stage. By means of a diffuser, a potential-similar distribution of velocity and pressure is caused. The remaining liquid drops move to the chamber wall of the fine separation stage from where they get down as liquid film. Separators working according to this principle enable an almost complete separation of the liquid from the conveyed gas in the most confined of spaces.

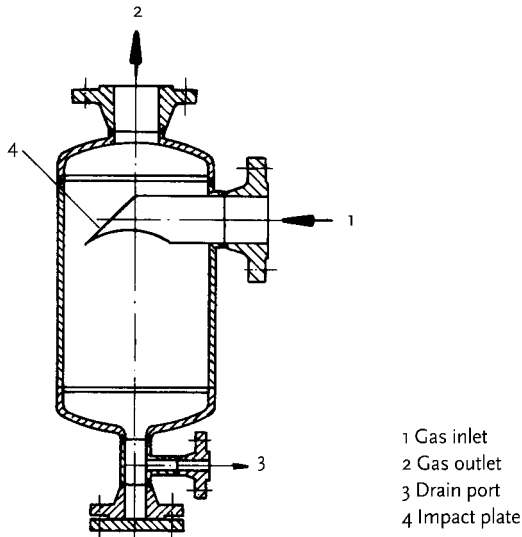


Figure 5-1. Liquid separator with impact plate

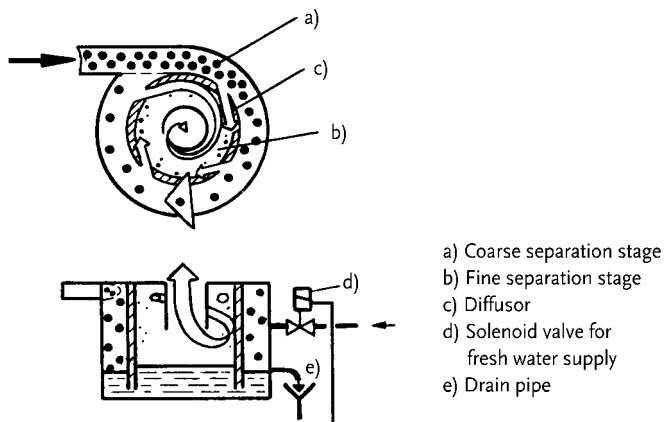


Figure 5-2. Potential vortex separator (acc. to Loewe)

High power cyclone separator

In this separator, a gas loaded with liquid receives a twist by a diffuser (fig. 5-3). Owing to the radial acceleration and the centrifugal forces acting on the liquid particles, these particles are spun onto the wall of the separator where they slide down into the collecting chamber. A screen above the collecting chamber prevents the separating liquid from being swirled up and carried away. The gas separated from the liquid leaves the separator through a central dip pipe. A trailing edge on the blade rim surrounding the dip pipe prevents the liquid from passing into the dip pipe. With devices of this design, an almost perfect separation is achieved.

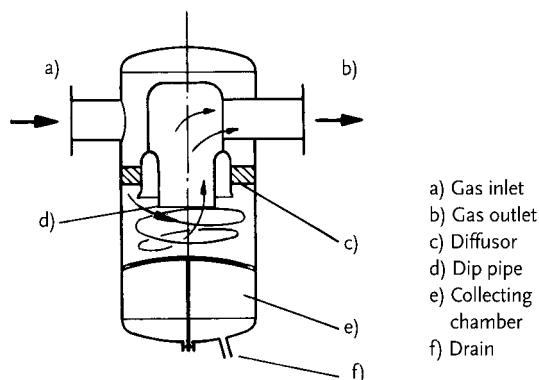
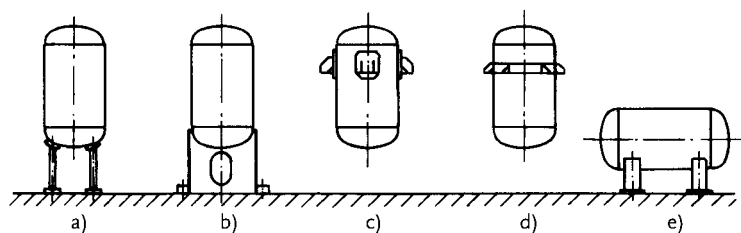


Figure 5-3. High power cyclone separator (acc. to Funke)

5.2.1

Vessel arrangements

Vessels and apparatus are usually mounted horizontally or vertically with regard to the given installation areas, space available, regulations and process technological conditions. For the installation of the vessels support elements are used, the selection of which depends on the vessel arrangement, the vessel design and on the foundation. The support elements have to hold the total weight of the device, as well as possible external forces and moments, and transmit them onto the foundations or steel structures. Figure 5-4 gives an overview of standard versions and arrangements of the support elements also defined in the applicable standards regarding the vessel arrangement.



Item	Vessel assembly	Support element
a)	vertical	Feet
b)	vertical	Shell support
c)	hanging	Lug support
d)	hanging	Supporting rings
e)	horizontal	Saddle bearing

Figure 5-4. Vessel arrangement and allocation of supporting elements [5.2]

5.3

Auxiliary appliances for vessels and pipework

5.3.1

Inflow control unit

The inflow control unit (fig. 5-5) is usually mounted on a liquid separator or on a storage vessel for the operating liquid of the liquid ring machine. In conjunction with the outflow control unit (see section 5.3.2), it takes on the level control of the liquid level in the vessel. The liquid to be supplied is taken, for example, from the water supply net or from an internal supply system and fed to the inflow control unit under a pressure that corresponds to the existing operational conditions. Especially when conveying dry gases where condensation does not occur, the small portion of operating liquid which is continuously carried with the discharge flow must be replaced.

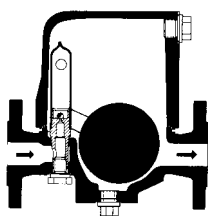


Figure 5-5. Inflow control unit (acc. to Rifox)

5.3.2

Outflow control unit

For keeping a constant liquid level in the separator, as described in section 5.3.1, an outflow control unit acc. to fig. 5-6 can be used. The outflow control unit installed on the liquid separator takes on the removal of the liquid possibly carried from the process or of the condensate accruing during compression.

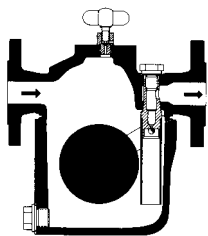


Figure 5-6. Outflow control unit (acc. to Rifox)

5.3.3

Injection segments

In order to reduce the gas suction temperature, to dilute aggressive suction mixtures, as well as to increase the suction capacity by condensation, or to feed larger ring liquid quantities into the liquid ring vacuum pumps, injection segments can be installed in the pump suction pipe (fig. 5-7). Via nozzles mounted in the nozzle holders, the ring liquid or the selected fluid is injected into the suction pipe. If required, the arrangement of several nozzle holders connected in series is possible, too. Due to the prevailing suction velocities, the nozzle holder must dispose of an aero-dynamic shape.

Usually, injection is carried out in flow direction. The pressure in the nozzle must be higher than the pressure in the suction pipe in order to ensure a continuous injection. In order to counter thermal stress arising because of large differences in temperature, the injection adapting piece is equipped with an internal protection pipe – thermal shock pipe – which absorbs thermal stress and reduces its influence on the main pipe of the adapting piece.

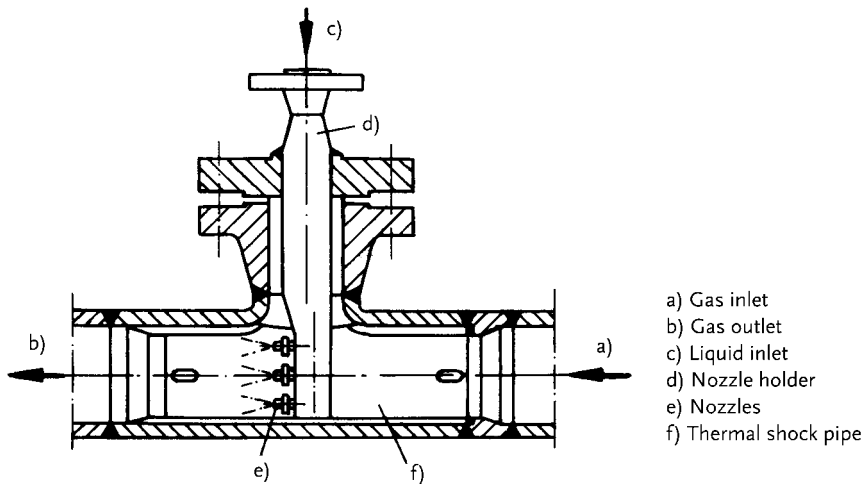


Figure 5-7. Injection segment [5.1]

5.3.4

Purging equipment

Purging equipment as shown in fig. 5-8 is often installed in a way that cleaning before putting into service and evacuating and rinsing after stopping the plant are possible. This is mainly the case with shut-off slides with long pipework running in a complicated manner, on pipe branching or shut-off devices in front of and after rotary machines. In these cases, machines such as pumps and compressors can be effectively protected against the penetration of dirt by cleaning the pipework and plant components.

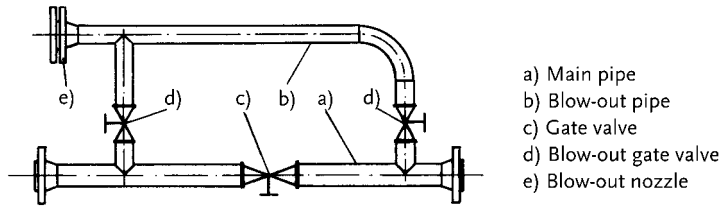


Figure 5-8. Purging equipment

5.3.5

Aeration and ventilating facilities [5.1]

In plant manufacturing, aeration and ventilating facilities are used in liquid and gas-containing systems. The penetration of gases, vapors or liquids into machines is safely prevented by installing an aeration facility in one of the pipework shut-off sections in front of the unit to be protected. In this case, the shut-off section, e.g. in a suction pipe leading to a pump, is installed between two closed shut-off valves.

In pipework for liquids, these facilities are used in order to remove the air accruing in higher places and to aerate the pipes during draining.

In figure 5-9, a simple aeration element is shown. The arrangement shown in figure 5-10 is to be considered in case the aeration element is inaccessible and the fittings must be installed at operation level. Pressureless pipes do not require fittings for aeration. Figure 5-11 shows the possibility of aerating such a pipe.

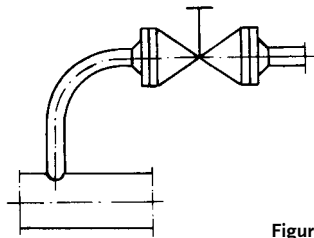


Figure 5-9. Simple aeration and ventilation element

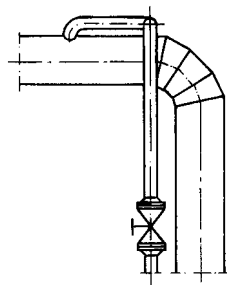


Figure 5-10. Aeration with lowered valve

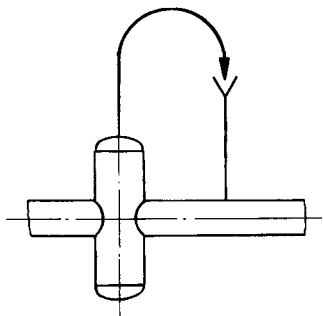


Figure 5-11. Aeration of an unpressurised liquid drain pipe

5.3.6

Sieves for liquids

For the protection of pumps and highly sensitive control, measuring and regulating devices, especially in liquid supply pipes, permanently installed protective fittings are mainly used in pipes where contamination is to be expected. For instance, sieves as shown in figure 5-12 and vessel sieves as shown in figure 5-13 with single or double sieves, as well as magnetic filters in designs acc. to figures 5-14 and 5-15 are customary.

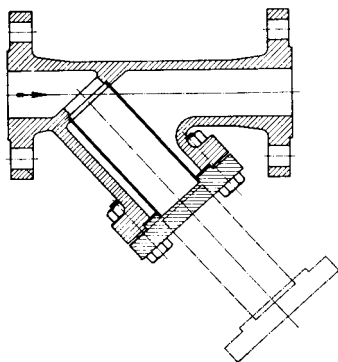


Figure 5-12. Angular sieve (acc. to Rifox)

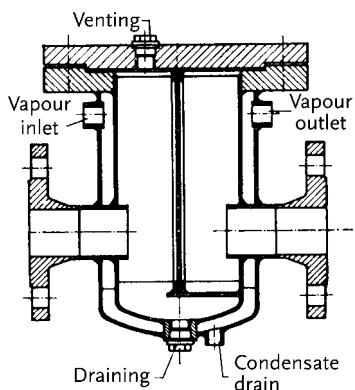


Figure 5-13. Sieve vessel with heating jacket (acc. to Gestra)

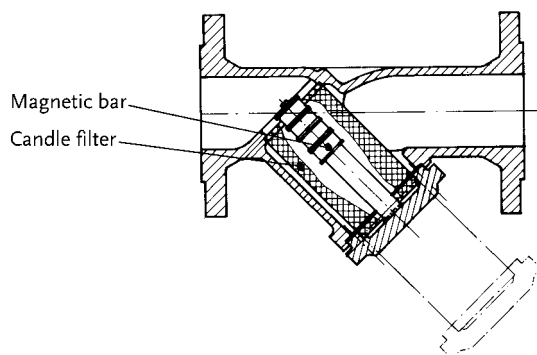


Figure 5-14. Angular sieve with magnetic filter (acc. to Koepp)

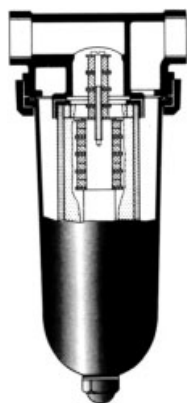


Figure 5-15. Pot-type magnetic filter (acc. to Koepp)

For the collection of rough dirt, supplementary start-up sieves are partly installed for the initial start-up, which are removed after the start-up operation.

5.4

Gas cleaning devices

Wherever there is a risk of contaminating particles arriving with the gas flow when starting-up or during regular operation it is advisable to clean the gas before it enters the vacuum pump. The devices available for this are working according to different principles and differ from each other in their degree of separation depending on the gas to be cleaned. When selecting and designing these devices, possible pressure losses in the entire system should be taken into consideration.

The devices usually applied to the cleaning of gases in the chemical industry separate the solid or liquid disperse phase by mass forces – such as gravity, centrifugal force, impact and sound – or by filters. Wet separation, acoustic separation and other

processes belonging to electromagnetic basic operations such as electrostatic separation or magnetic separation are also applied.

5.4.1

Chamber separator

In this separator (fig. 5-16), suspended particles are separated under the influence of gravity. The arriving larger solid particles precipitate after the inflow of the gas in wide collecting vessels, since the flow speed falls below the suspension speed. The particles fall down along parabolic paths. The horizontal gas flow carries smaller or lighter particles over longer paths than larger and heavier particles.

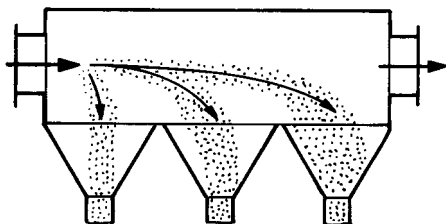


Figure 5-16. Chamber separator [5.3]

5.4.2

Impact plate separator

In the impact plate separator, the sharp change of the gas flow direction which the particles are not able to follow is used for the separation of larger dust particles or liquid drops. The particles crashing onto the plates reflect and fall into the separation chamber (figures 5-17 and 5-18). Even these separators belong to the gas cleaning devices utilizing mass forces.

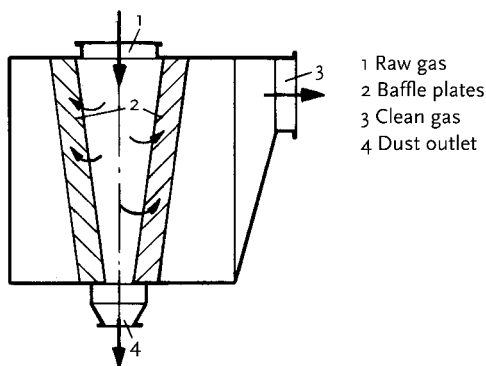


Figure 5-17. Impact plate separator with top side gas inflow [5.4]

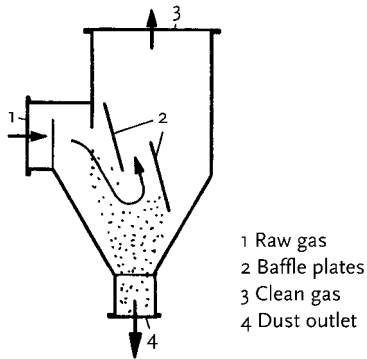


Figure 5-18. Impact plate separator with horizontal gas inflow [5.4]

5.4.3

Centrifugal separator

These separators separate particles suspended in the gas phase and not prone to stick, by means of centrifugal force and repeated deviation of the gas flow. The separated solid particles reach down into the separation chamber of the vessel (fig. 5-19).

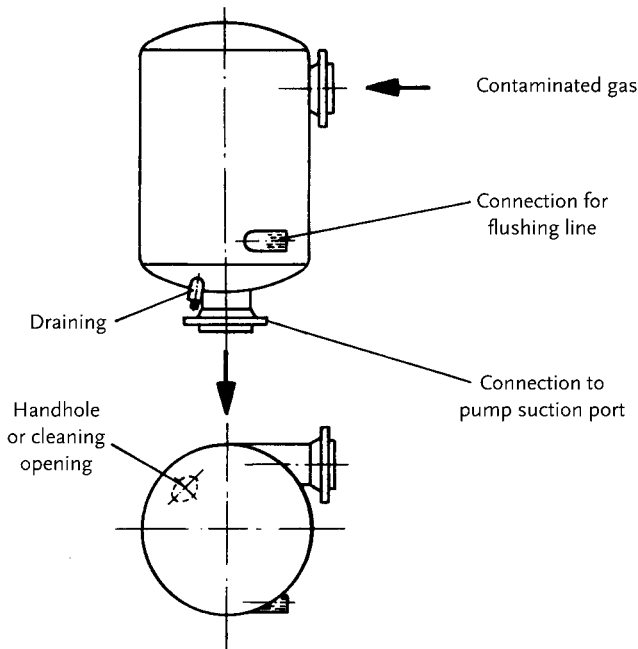


Figure 5-19. Centrifugal separator

5.4.4

Aero-cyclones

With aero-cyclones, the interaction of centrifugal force and gravity is utilized for sedimentation of even the finest suspended particles which are not prone to stick. The gas flow entering tangentially into the cylindrical cyclone casing moves down spirally in a three-dimensional flow field (fig. 5-20). Separated dust or precipitating liquid glides in the cone to the collecting chamber. In the upper area of the cone, the direction of the gas whirl is changed upwards and escapes through the axial dip pipe. This spiral whirl rotating with these separators led to the name “cyclone”.

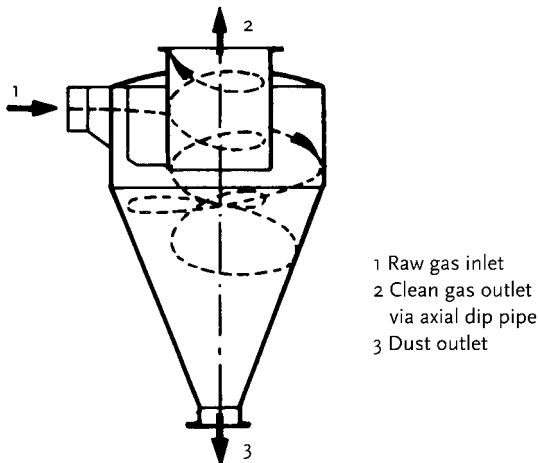


Figure 5-20. Standard air-cyclone [5.3]

5.4.5

Filters

Filtering is the separation of the components of a gas-solid-system by means of a filter medium permeable for gas in which the solid particles are retained. Filtration is a process in which solid particles are separated from liquids. The filtrate flow through the filter layer can occur owing to overpressure, underpressure, hydrostatic pressure or capillary pressure. Filter layers can be filter fabrics made of cloth, paper or synthetic substances, porous filter masses and metallic material. Sedimentation is often the preliminary stage to the filtration, i.e. if solid particles are not deposited owing to their density, filtration is the alternative principle of separation.

Filtering devices are designed in various versions for continuous and discontinuous operating modes.

Fig. 5-21 shows a candle filter in which hose-like sacks are put up over a support rack. The interior of this so-called filter candle is connected to a vacuum pump. Accumulating dust can be tapped off by a device equipped with cams. Filters of this kind are used e.g. for the dedusting of air and gas.

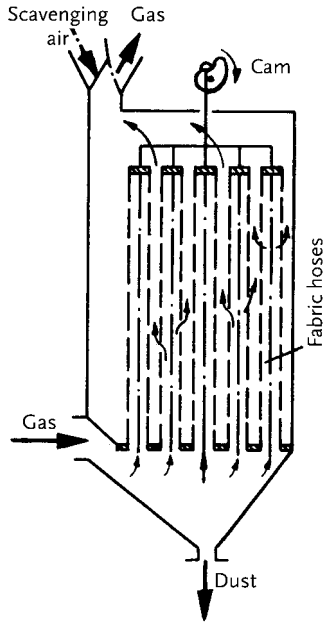


Figure 5-21. Candle filter [5.5]

5.5

Heat transfer devices

Heat transfer devices – also called heat exchangers or temperature changers – are basic equipment that transfers thermal energy from one substance to another.

These devices are also used for the removal of compression and condensation heat when liquid ring vacuum pumps are operated as well as for the removal of heat flows resulting from chemical reactions.

There is a distinction made between three basic designs: recuperators, regenerators and contact heat transfer devices.

- *Recuperators*
The substance flows are separated by a thin wall. Examples are steam boilers, oil coolers, surface condensers and flow heaters.
- *Regenerators*
The heat-storing mass – firebricks, sheet steel – is heated by a heat releasing substance and cooled by a heat absorbing cooler substance after switching over. This discontinuous mode of operation is applied e.g. in stoves or blast furnaces.
- *Contact heat transfer device*
These devices have no heating surface. The heat releasing and the heat absorbing substances contact each other directly. Examples for this are mixing condensers, evaporating cooling towers and air washers.

According to their purpose, heat exchangers are subdivided into coolers, heaters, condensers and evaporators.

In mechanical engineering, mainly continuously operating recuperative heat exchangers are applied. Therefore, they will be described here in detail.

5.5.1

Heat transition

In the heat transfer device, the heat transport occurs through the existing material heat carrier. The heat transition is responsible for the functioning of a recuperator. First, the heat is transferred by a fluid heat carrier onto a solid, single or multi-layered wall and led through this wall. The other side of the wall gives off the heat to the fluid to be heated. This process involves heat transmission, thermal conductivity and heat radiation together (see fig. 5-22). The individual shares of these processes are summarized in the so-called thermal transition coefficient. The measuring unit of the thermal transition coefficient is identical to the measuring unit of the thermal transmission coefficient. The thermal transition coefficient k is calculated from the thermal transmission coefficients of the fluids and the thermal conductivity coefficients and thicknesses of the wall layers.

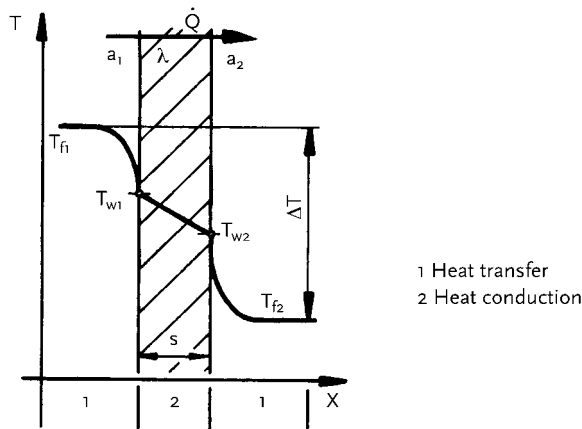


Figure 5-22. Heat transition through a plain wall [5.6]

For a plain, single-layer wall the thermal transition coefficient is calculated acc. to equation 5-1.

$$k = \frac{1}{\frac{1}{\alpha_1} + \frac{s}{\lambda} + \frac{1}{\alpha_2}} \quad (5-1)$$

- k thermal transition coefficient [$\text{W}/(\text{m}^2 \cdot \text{K})$]
- α_1 thermal transmission coefficient fluid 1 [$\text{W}/(\text{m}^2 \cdot \text{K})$]
- α_2 thermal transmission coefficient fluid 2 [$\text{W}/(\text{m}^2 \cdot \text{K})$]

- s wall thickness [m]
 λ thermal conductivity coefficient of the wall layer [W/(m · K)]

The thermal transition coefficient is always smaller than the smallest thermal transmission coefficient. The amount of heat transferred from a warmer fluid to a colder fluid through a partition wall of the surface A within the period of time t at the temperature difference ΔT is:

$$Q = k \cdot A \cdot \Delta T \cdot t \quad (5-2)$$

and the heat flow is

$$\dot{Q} = k \cdot A \cdot \Delta T \quad (5-3)$$

- Q amount of heat [W · s]
 \dot{Q} heat flow [W]
 A transfer surface [m²]
 ΔT temperature difference [K]
 t time [s]

5.5.2

Contamination of transfer surfaces

Experience shows that transfer surfaces of heat transfer devices cover themselves with deposits or have to be covered with a protective coating because of corroding substances.

Deposits and protective coatings hinder the heat flow through the wall. The thickness of the layer, its thermal conductivity and the thermal conductivity coefficient of the wall itself are decisive for the heat transition.

The reduction of the thermal transition can be expressed by the correction factor φ (see equation 5-4), [5.7].

$$k = \varphi \cdot k_o \quad (5-4)$$

The factor k_o is the thermal transition coefficient without consideration of the layers, while k is the real thermal transition coefficient resulting from the correction. For plain heating surfaces, the correction factor acc. to equation (5-5) is

$$\varphi = \frac{1}{1 + k_o \sum_{i=1}^n \left(\frac{s_i}{\lambda_i} \right)} \quad (5-5)$$

- φ correction factor
 s_i layer thicknesses [m]
 λ_i thermal conductivity coefficients [W/(m · K)]
 k_o thermal transition coefficient [W/(m² · K)]

With good approximation, equation (5-5) is also applicable for curved heating surfaces. In table 5-1, thermal conductivity coefficients of some protective coatings and contaminating substances are listed.

Specific values for substances such as thermal transition coefficients and thermal transmission coefficients are stated in the relevant literature (see [5.7]).

Table 5-1. Heat conductivity λ [W/(m · K)] at room temperature [5.7]

Protective coatings				Contaminating substances	
Tin	65	Oppanol	0.2 to 0.35	Scale, gypseous	0.6 to 2.3
Lead	35	acid-proof stones	≈ 1.2	Scale, silicate-rich	0.08 to 0.18
Glass	0.76 to 0.84	Carbon bricks	1.6 to 4.7	Soot, dry	0.035 to 0.07
Quartz glass	1.34	Porcelain	1.7 to 3.5	Coal dust, dry	0.11
Enamel	0.9 to 1.2			Ice	1.75 to 2.3
Rubber	0.15 to 0.17			Cooling water jelly-layer	≈ 0.35
Asphalt	0.76			Brine-jelly layer	≈ 0.46
Igelit (PVC film)	0.16			Salt	≈ 0.6

Devices with over-dimensioned designs or the selection of factors that are too large will support contamination.

5.5.3

Designs of heat exchangers

Owing to the different fields of application of heat exchangers, a variety of different designs are on the market. The flow directions of the media in heat exchangers have a significant influence on the heat transport.

Here, a fundamental distinction is made between three different layouts of mass flux to each other: parallel flow, counter-current flow, cross flow.

In practice, even the mixed flow types cross-parallel flow, cross-counter-current flow and multi cross-counter-current flow occur. For heat-engineering reasons, mostly counter-current flow heat exchangers are used.

Tubular heat exchanger

Here, two basic designs are distinguished, the tubular heat exchanger with fixed boiler tubes and the design with removable boiler tubes. An overview of the different designs is given in figure 5-23.

The tube routing in the shell chamber has various designs. The tubes running inside are straight, U-shaped or are designed as multiple spiral boiler tubes. With tubular heat exchangers in straight-tube design the fluid in the shell chamber can be led by means of baffle plates. The tubes can be arranged in different numbers of paths, i.e. as single flow, double and quadruple flow.

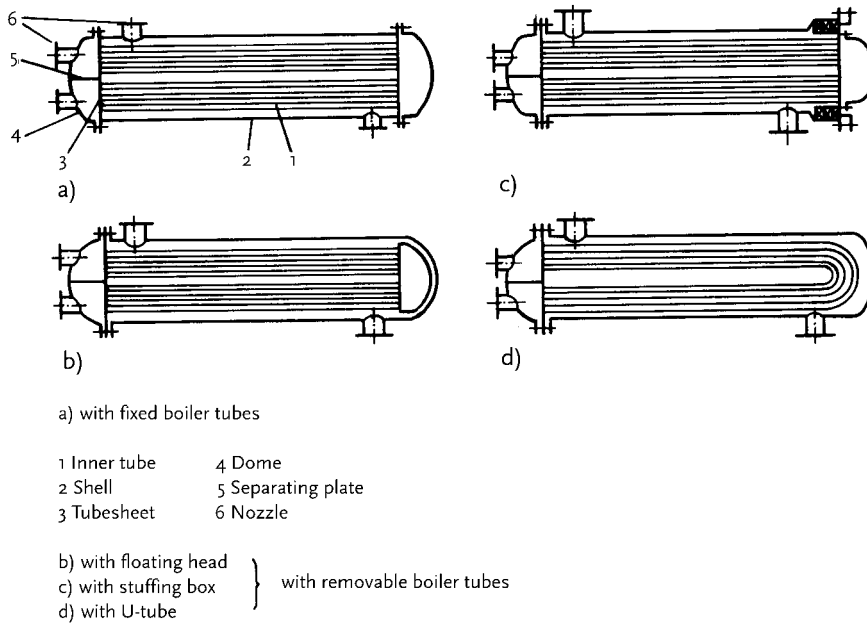


Figure 5-23. Basic design types of tubular heat exchangers [5.2]

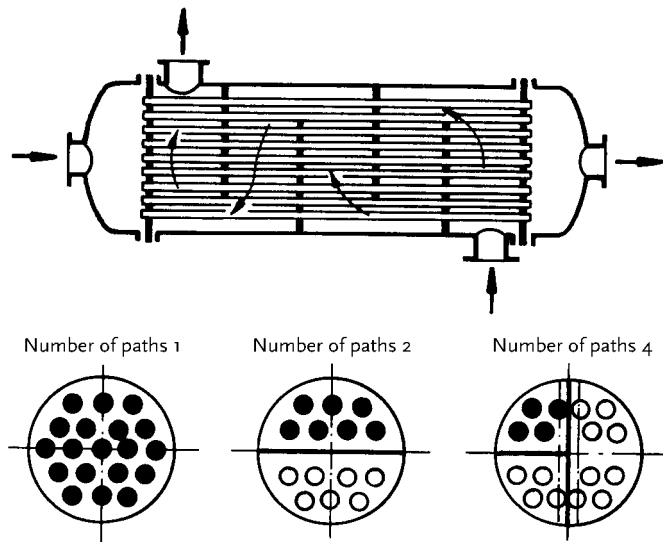


Figure 5-24. Tubular heat exchanger with straight pipes and fixed boiler tubes [5.8]

Medium guidance with tubular heat exchangers

Substances with higher viscosities are practically led through the shell chamber. Dirty or contamination-prone fluids are led through the tube chamber in fixed boiler tubes. When flowing in the shell chamber with removable boiler tubes, the tubes should be arranged square.

Safety heat exchangers with double boiler tubes

In devices of this specific design, the inner boiler tubes consist of two telescoped tubes. The safety chamber in the double boiler tube is filled with a special shut-off liquid connected with an expansion tank and an overpressure switch, (fig. 5-25). In case of a burst tube and a pressure rise in the shut-off liquid resulting from this, alarms can be triggered off via the overpressure switch, shut-off elements can be activated or machines can be stopped. Owing to this design possibly arising damages can be detected immediately and the undesired mixing of the two liquid flows in the heat exchanger can be prevented. These designs are especially applied in cases in which utmost safety and the adherence to the environmental protection regulations are of great importance.

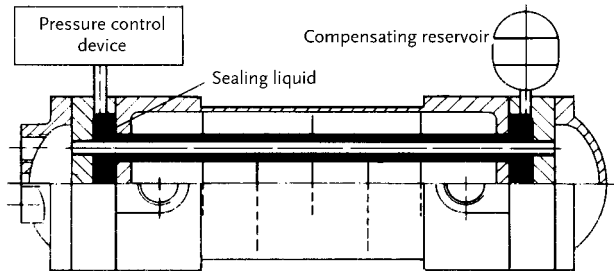


Figure 5-25. Safety heat exchanger (acc. to Funke)

Double-tube heat exchanger

For reasons of stability the shell diameter of tubular heat exchangers has to be smaller with increasing pressures. In a borderline case with extremely high pressure only one inner tube remains, i.e. only one double-tube heat exchanger. This design is especially suitable for high pressures or for special demands on the residence time. With standard components by means of series or parallel connection expandable heat exchangers can be manufactured for pure parallel or counter-current flow. Easy cleaning is an advantage, too (fig. 5-26).

Ribbed tube heat exchanger

For the heat transfer between media the thermal transmission coefficients of which show great differences, a unilateral enlargement of the transfer surface approximately in a ratio of the thermal transmission coefficients is reasonable. For this purpose, the outer surface of the tube is ribbed (fig. 5-27). Different designs such as helical type, gilled and longitudinal-fin tubes are used.

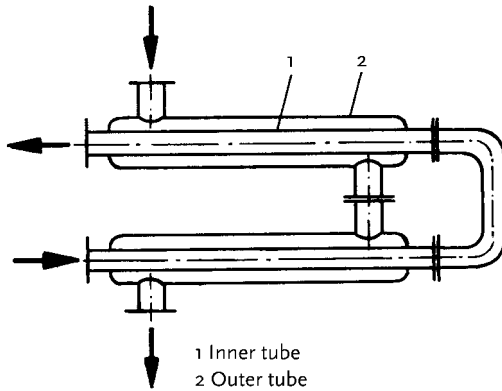


Figure 5-26. Double-pipe heat exchanger

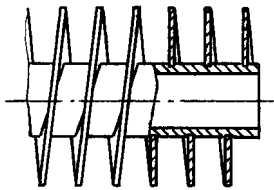
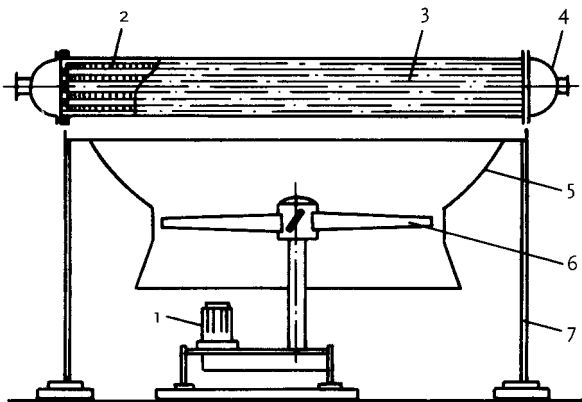


Figure 5-27. Gilled pipe



- | | |
|------------------------|---------------|
| 1 Fan drive | 5 Diffusor |
| 2 Ribbed tube | 6 Fan |
| 3 Ribbed tube register | 7 Steel frame |
| 4 Chamber | |

Figure 5-28. Air-cooled heat exchanger [5.2]

These heat exchangers are used e.g. in air-cooled condensers, air heaters and gas coolers (fig. 5-28).

Due to the generally increasing water shortage of the last decades and the environmental protection measures, these designs have gained importance. Owing to the fact that the specific heat capacity of air is about a quarter of the heat capacity of water, approximately four times the mass flow is required for the same cooling capacity with air.

Plate heat exchanger

In conventional designs, profiled plates are arranged in package-shape between two tripod or end plates and held together with screw joints (fig. 5-29). The plates have a corrugated profile. Usually the mass flows in these heat exchangers are counter-current flows. The heat transfer in these devices is very favorable. Further advantages are low construction volume, easy variation by installing more or less plates and easy cleaning. Their range of application includes the food, pharmaceutical and chemical industry.

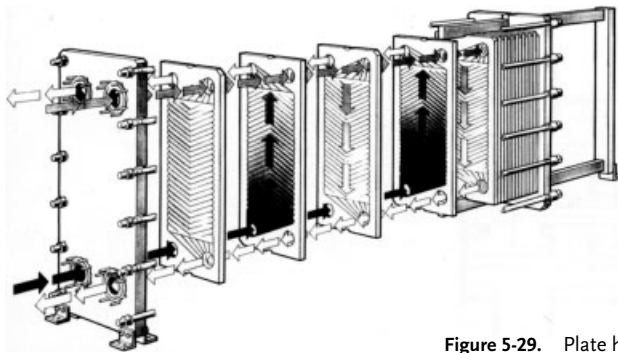


Figure 5-29. Plate heat exchanger (acc. to Sigri)

Helical-type heat exchanger

In principle, this special design for the heat transfer between liquids is a plate heat exchanger, too. A helical-type heat exchanger consists of two steel sheets helically wound around a middle part with the same distance (fig. 5-30). The liquids are guided in parallel or counter-current flow through channels without deviation. An

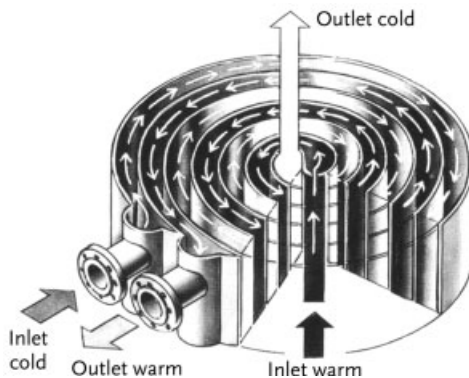


Figure 5-30. Helical type heat exchanger (acc. to Alfa-Laval)

outstanding feature of this design is its compactness. On the front faces, there are flat lids removable for cleaning and serving as a lateral sealing.

Graphite plate heat exchanger

The graphite heat exchangers (fig. 5-31), manufactured in chamber design, are plate heat exchangers as well. As a result of the material properties, graphite heat exchangers are manufactured in plate design and block design. With larger transition surfaces, the tubular design is applied, too. If graphite is mixed with chemical-resistant thermosetting plastics, chemical-resistant, thermal-shock-resistant materials of a high thermal conductivity are the result. Due to high resistance of graphite against aggressive chemical substances such as acids, lye and solvents, such devices are preferably used in circuits in which substances or fluids with similar properties occur.

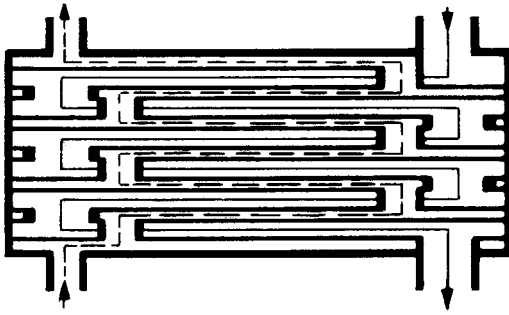


Figure 5-31. Graphite plate heat exchanger (acc. to QVF Glastechnik)

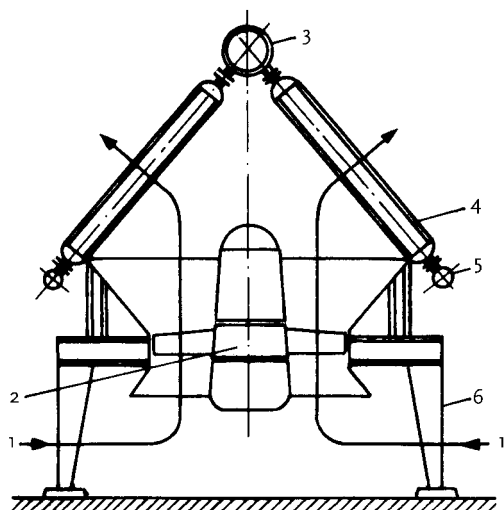
5.6

Condensers

From a physical point of view, condensing is the reversal of the evaporation process. By means of cooling vapors below the saturation temperature – the dew point – they are transferred to the liquid state. During the transformation from the vaporous to the liquid state, a vacuum arises as a result of the decrease in volume.

Condensing is aimed at the recovering of condensate from vapors or generating or maintaining vacuum in devices. The non-condensable gaseous particles are sucked by vacuum pumps. Due to the arrangement of condensers in front of the vacuum pumps, these are relieved and it is possible to apply designs of smaller dimensions. Therefore, condensers are very economical vacuum pumps for vapors.

Their designs are distinguished according to the used refrigerant. For instance, cooling water, cooling brines and air can be used as refrigerants. The devices cooled with liquid are called surface condensers or co-condensers, according to the kind of cooling. Air condensers are equipped with ventilators or axial-flow fans (fig. 5-32).



1 Air flow, 2 Axial fan, 3 Steam line, 4 Gilled cooler elements,
5 Condensate line, 6 Supporting frame

Figure 5-32. Air-cooled surface condenser in roof construction [5.4]

5.6.1

Surface condensers

According to their design, surface condensers belong to the indirect heat exchangers. They are mostly designed as tubular heat exchangers (fig. 5-33). Coiled-tube and double-tube heat exchangers as well as irrigation coolers are manufactured as condensers, too. With this design e.g. in the case of distillation, the pure condensate accrues separately from the liquid refrigerant. The condensate can drain automatically via a barometric downpipe (see section 5.6.3). In tubular heat exchangers the

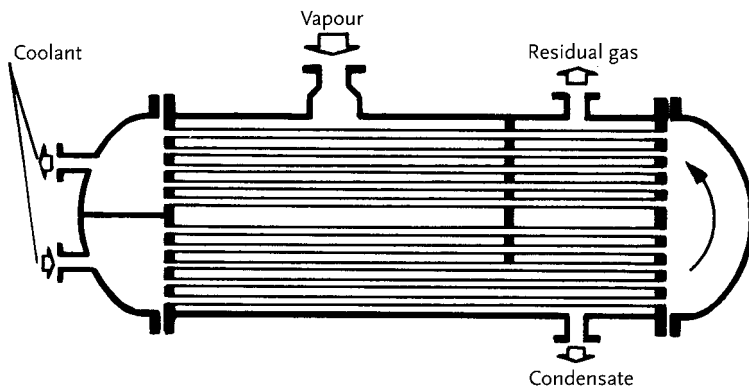


Figure 5-33. Liquid-cooled tubular surface condenser

vapor to be condensed is usually led through the shell chamber while the cooling medium flows through the tubes. Vapor and coolant are mostly guided in counter-current flow, i.e. the vapor enters the condenser from above, the refrigerant from below. The condensing chamber can also be under pressure. With devices of this design, high vacuums can be achieved and maintained.

5.6.2

Co-condensers

Co-condensers – also called injection condensers – belong to the group of contact heat exchangers. The entering vapors are directly mixed with injected cooling water. Nozzle or irrigation bottom constructions create large contact surfaces between vapor and coolant. The vapor gives its condensation heat to the coolant and condenses. The condensate accrues together with the coolant. For this reason, co-condensers are only suitable for condensing water vapor or worthless exhaust gases. Through suction of non-condensable gases, the vacuum is maintained. Co-condensers achieve more unfavorable vacuums than surface condensers, as for instance, the cooling water contains dissolved air and enters the condensation chamber as well. Condensers are classified into wet and dry co-condensers.

Wet co-condensers

These condensers work mainly according to the parallel-flow method, i.e. coolant and vapor flow through the condenser in the same direction (fig. 5-34). The non-condensing gases are usually sucked by a steam jet or liquid ring vacuum pump. Wet co-condensers are applied for smaller and medium vapor throughput capacities.

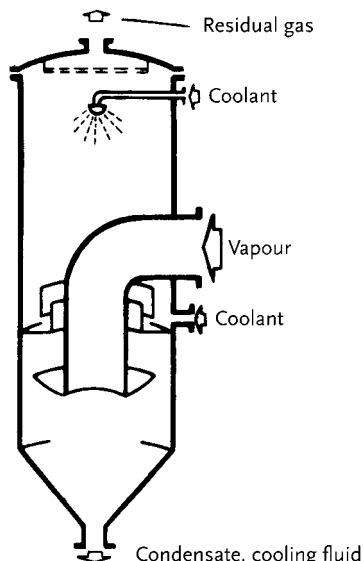


Figure 5-34. Wet co-condenser

Dry co-condensers

Dry co-condensers work according to the counter-current flow principle (fig. 5-35). The coolant flows via installed plates and irrigation bottoms from top to bottom. The vapor inlet is situated in the lower part of the co-condenser.

Non-condensing gases are sucked from the upper side of the condensing chamber by a dry vacuum pump, liquid ring pump or a jet vacuum pump. Coolants and condensate come down and leave the condenser continuously e.g. via a barometric downpipe. Dry co-condensers are used for higher throughput capacities.

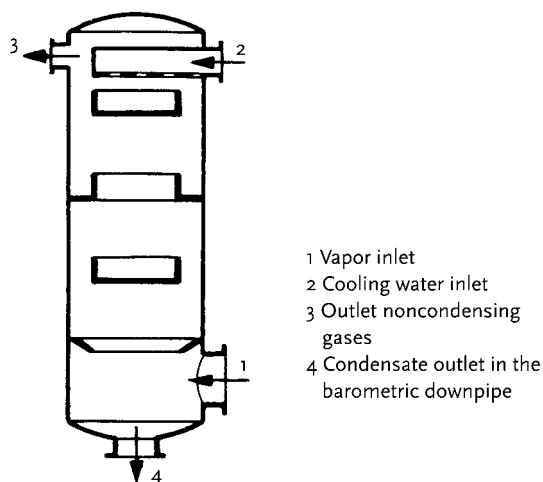


Figure 5-35. Dry barometric co-condenser with distribution bottoms [5.3]

5.6.3

Condensate discharge

If possible, condensers should be installed barometrically, because then the condensate or the condensate with the cooling water can drain automatically over the downpipe (fig. 5-36). If owing to the insufficient chamber height a barometrical installation is not possible, the evacuation mainly of smaller quantities and with discontinuous operation may occur over collecting vessels under vacuum. It is also possible to perform a continuous discharge at low heights by using intermediate vessels and special pump layouts (fig. 5-37).

With the condenser being installed barometrically, the required height of the liquid column in the downpipe depends on the pressure prevailing in the condenser (fig. 5-36). The height of the liquid column is calculated acc. to equation (5-6).

$$h = \frac{p_o - p}{\rho \cdot g_n}$$

(5-6)

h liquid column head [m]

p_o atmospheric air pressure [$\text{kg}/(\text{m} \cdot \text{s}^2)$]; [Pa (abs)]

p pressure in the condenser [$\text{kg}/(\text{m} \cdot \text{s}^2)$]; [Pa (abs)]

ρ liquid density [kg/m^3]

g_n normal gravity acceleration = $9.81 \text{ [m/s}^2\text{]}$

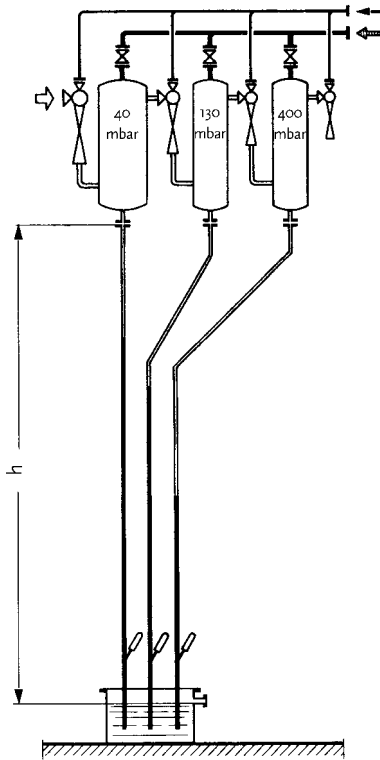


Figure 5-36. Barometric installation for co-condensation [5.9]

Since due to the draining water a pressure loss occurs in the downpipes and part of the non-condensing gases is swept away owing to the flow speed, a safety margin of about 0.50 m should be added to the calculated theoretical height of fall of the liquid.

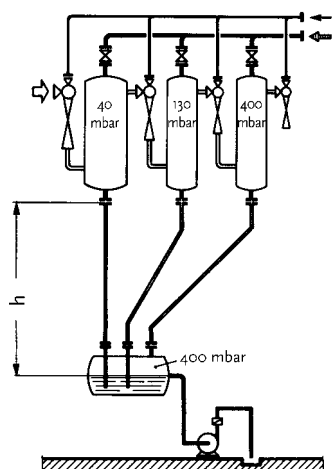


Figure 5-37. Semi-barometric installation with intermediate vessel and water scavenge pump [5.9]

5.6.4

Exhaust gas condenser

Owing to the direct assembly of an exhaust condenser on the liquid separator of the liquid ring vacuum pump, further, not yet condensed vapors can be recovered from the compressed gas (fig. 5-38), [5.10]. The condensate flows back into the separator towards the escaping gas flow. At lower temperatures, cooling water or brine is used as coolant for the exhaust condenser in order to ensure a low-emission discharge flow as dry as possible.

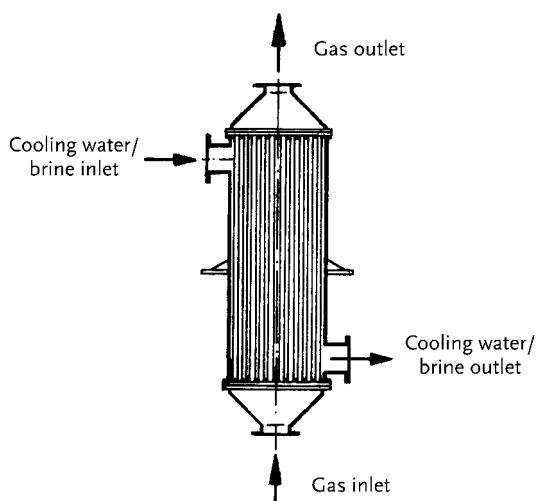


Figure 5-38. Exhaust gas condenser

5.7

Temperature controllers

The application of temperature controllers in coolant circuits of heat exchangers and condensers allows the optimal adaptation of cooling water or brine consumption to the operating conditions. The design and functional principle of a temperature controller without auxiliary energy is shown in figures 5-39a and 5-39b.



Figure 5-39 a. Temperature controller without auxiliary energy (acc. to Spirax-Sarco)

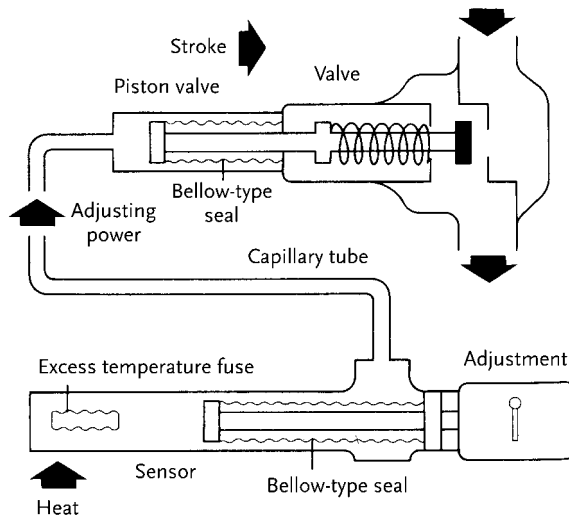


Figure 5-39 b. Function diagram

5.8

Flowmeters

The operating liquid flow of liquid ring machines or the quantity of a coolant of heat exchangers can be locally measured with liquid flowmeters and can be tele-transmitted electrically or pneumatically by the respective additional appliances. Fig. 5-40 shows a device working according to the suspended floater principle.

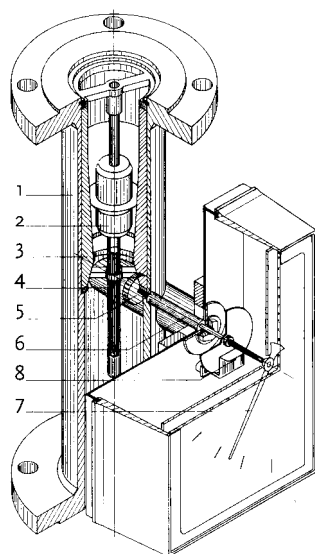


Figure 5-40. Flowmeter (acc. to Turbo-Werk Messtechnik)

1 armature, 2 suspend floater, 3 measuring cone,
4 bar magnet, 5 disc magnet, 6 pointer shaft,
7 pointer, 8 eddy current damping

5.9

Shut-off instruments

The most different designs of shut-off instruments are classified according to the movement direction of the shut-off elements with regard to the flow direction. Acc. to figure 5-41, a distinction is made between

- a) valve: the shut-off valve is moved in or against the flow direction,
- b) gate valve: the shut-off valve is moved vertically to the flow direction,
- c) butterfly valve: the shut-off valve is revolving and arranged vertically to the flow direction, the flow is almost parallel to the surface of the terminating element,
- d) plug valve: owing to the rotation, the shut-off valve is moved as a cylinder or cone diagonally to the flow direction.

Besides their purpose to shut-off or release flowing substances in pipeworks, these instruments are also used for controlling and throttling.

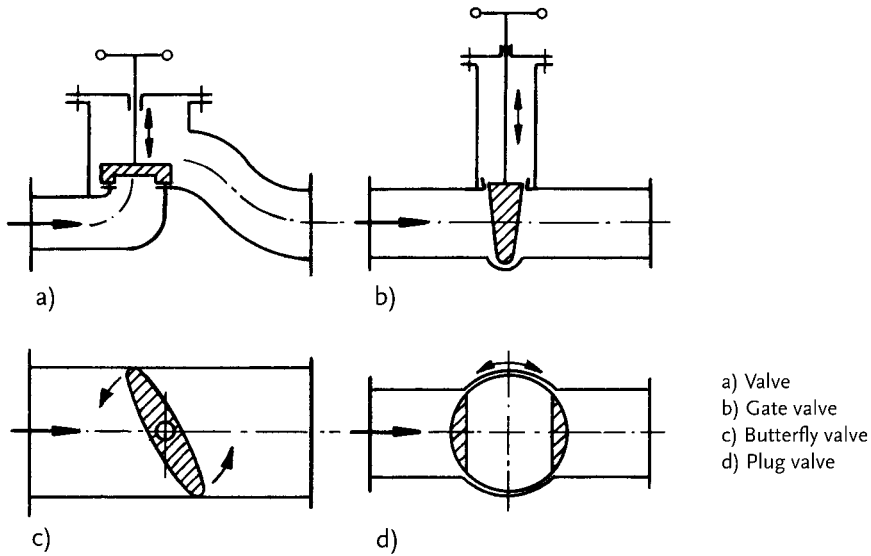


Figure 5-41. Schematic diagram of shut-off valves [5.2]

Therefore, shut-off valves are also designed with flat cones or parabolic throttling cones. With needle valves (fig. 5-42) pressure control in vacuum systems is possible by means of letting in very small gas flows.

With operation under high vacuum or in systems in which hazardous substances are transported and absolutely tight circuits are required, special shut-off instruments with metallic bellows or designs with permanent magnetic coupling are used in the technological processes, both on the gas side and on the liquid side (figures 5-43 and 5-44).

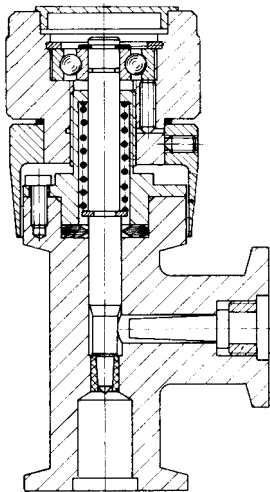


Figure 5-42. Needle valve (acc. to Balzers)

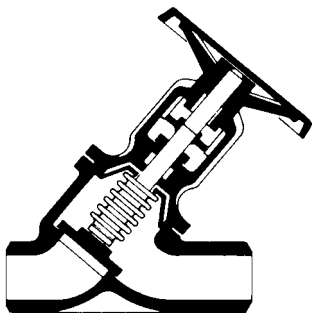


Figure 5-43. Slanted seat valve with metallic bellows and welding sockets (acc. to ARI-Armaturen)

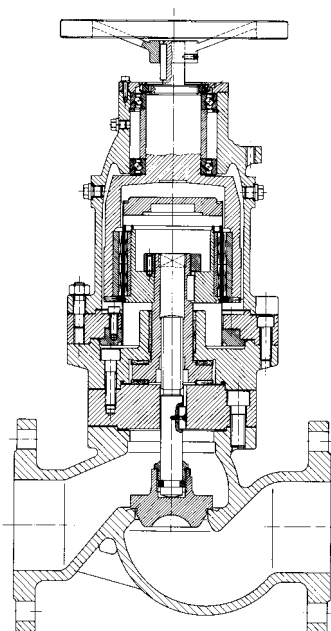


Figure 5-44. Straight-way type valve sealless design with permanent magnetic coupling (acc. to Lederle-Hermetic)

The special valves include also the electromagnetically operated valves. They are used e.g. as ventilation valves in sliding vane pumps or in automatic controls (see section 3.13.2). Figure 5-45 shows a valve of this design.

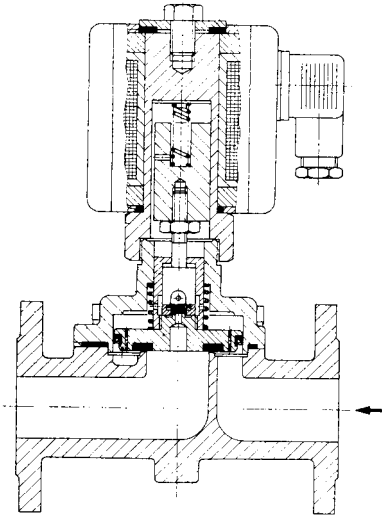


Figure 5-45. Solenoid 2/2 way valve (acc. to Buschjost)

5.10

Check valves and ball check valves

Check valves (fig. 5-46) and ball check valves (fig. 5-47) prevent the return flow of the flow medium.

They are installed in liquid ring vacuum pumps and compressors, in order to avoid the return flow of the conveyed gas and the operating liquid to the suction side after having stopped the machine. Compared to check valves, ball check valves dispose of lower pressure losses and are therefore used particularly in the high vacuum range.

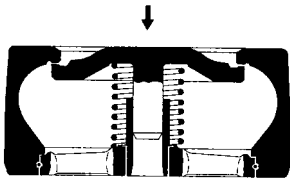


Figure 5-46. Check valve (acc. to Gestra)

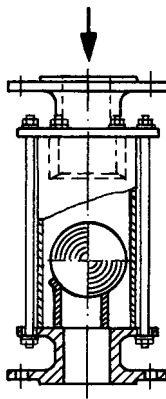


Figure 5-47. Ball check valve

5.11

Safety valves

With the installation of safety valves, exceeding the maximum permissible operating pressure in pipe systems and vessels shall be prevented. In plants with liquid ring machines being installed, these safety valves are used at high system pressures or in cases of existing vessels and devices being subject to the pressure vessel regulations, for example. There is a distinction made between mass-loaded, spring-loaded and pilot-controlled safety valves. Spring-loaded safety valves are partly manufactured with metallic bellows compensating counter-pressures (fig. 5-48).

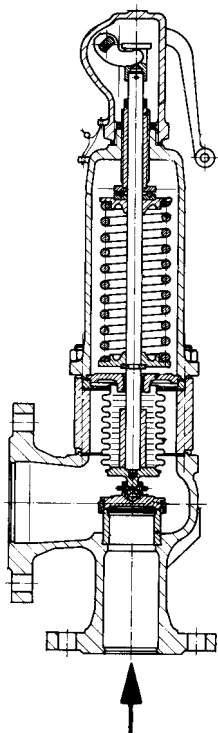


Figure 5-48. Full-stroke spring safety valve with metallic bellows (acc. to Leser)

5.12

Vacuum ventilation valves

Vacuum ventilation valves (fig. 5-49) are used for the prevention of an undesirable generation of vacuum in pipes, vessels and columns. The ventilation valves work with spring load and are usable as vacuum breakers and limiting valves, however not as control valves, as the cone does not dispose of control characteristics. These valves are controllable only to a limited extent, as owing to the increasing spring tension, they open slowly.

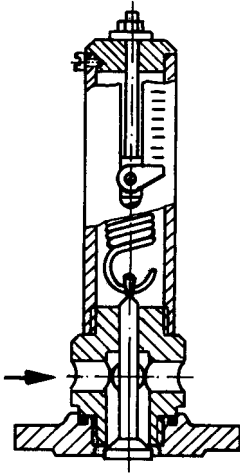


Figure 5-49. Automatic vacuum ventilation valve
(acc. to Mankenberg)

5.13

Flanges in vacuum technology

(Extract from DIN 28404, October 1986)

Application range and purpose

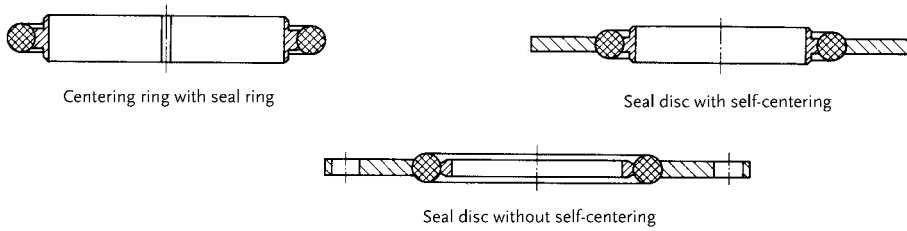
This standard is applicable for fixed, clamped and saddle flanges as well as for equipment used for plants in the range of rough, fine and high vacuum for vacuum-tight connections.

Flanges according to this standard determine the interfaces between the components and serve the assembling of components and vacuum plants running in rough, fine and high vacuum. The flanges are also used for the connection of pipes, with the external diameter of the pipe being taken from the standards listed in this standard. They can be installed in any position and are also suitable for overpressures of up to 1.5 bar.

This standard is connected with the international standard draft ISO/DIS 1609.4-1984 published by the International Organization for Standardization (ISO).

Examples of applications are shown in figure 5-50.

Examples for seal elements



Examples for flange connections

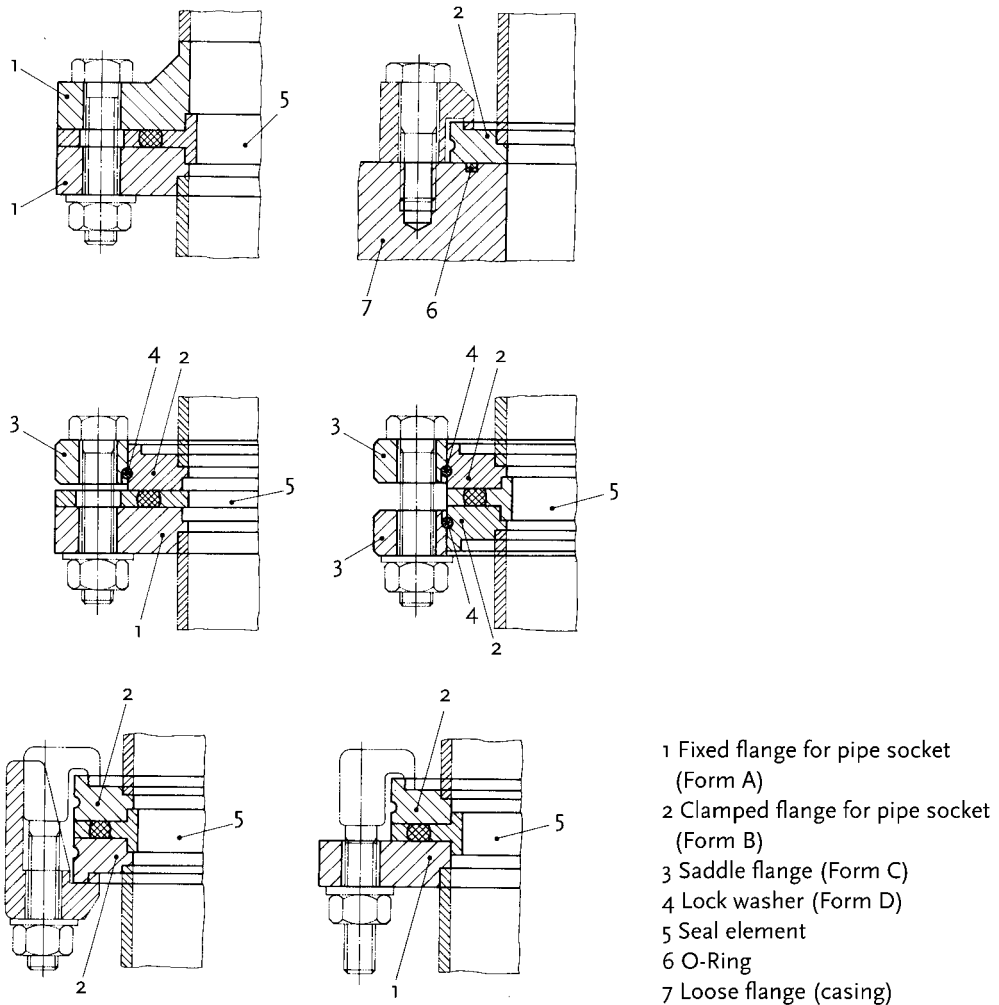


Figure 5-50. Application examples of sealing elements and flanges in vacuum technology

5.14

Fast flange connections, small flange connections in vacuum technology

(Extract from DIN 28403, September 1986)

Application range and purpose

This standard is applicable for small flange connections and their individual parts for vacuum plants used in rough, fine or high vacuum.

The small flange connections according to this standard are fast connections serving the easy and fast installation of vacuum plants. They can be installed in any position and are also suitable for overpressures of up to 1.5 bar. They serve as a connection of pipe sockets consisting of both metallic and non-metallic materials.

Dimensions, names

General tolerances: DIN 7168 – m

The small flange connections are not required to correspond to the illustrations, only the stated dimensions are to be adhered to.

This standard is set up in co-operation with PNEUROP.

An overview of dimensions and names is given in fig. 5-51.

5.15

Surface condition of sealing surfaces [5.11]

As a rule, the harder the sealing material in the surface area of the sealing surface is, the smaller the surface roughness of the respective flange surface has to be.

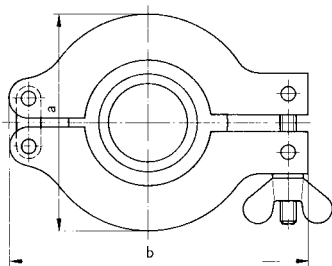
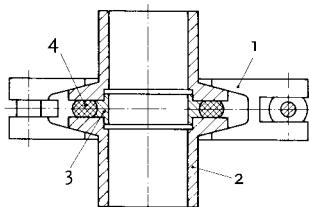
For metallic sealing materials, table 5-2 can serve as a basis for the recommended surface roughness of the flanges.

Table 5-2. Surface roughness of metallic seals

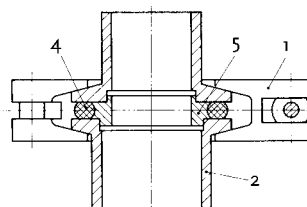
sealing material, metal	hardness HB [N/mm ²]	recommended max. surface roughness of flange R_t [μm]	sealing material, metal	hardness HB [N/mm ²]	recommended max. surface roughness of flange R_t [μm]
Lead	40	100	Iron	900	3.2
Gold	150	25	Monel	1100	3.2
Aluminium	200	25	Nickel	1200	3.2
Silver	300	12.5	Steel	1200	3.2
Copper	500	6.3	Titan	1300	3.2
Brass	700	6.3	Stainless steel	1600	1.6

With galvanic coatings in the area of sealing surfaces, the roughness of the basic material of the seal and the flange must be observed.

Form A
Small flange connection



Form B
Transition small flange connection



Top view like Form A

Designation of a complete small flange connection
of Form A, nominal width DN 25:

Connection DIN 28 403 – A 25

Designation of a complete transition small flange connection
of Form B, nominal width DN 20 to nominal width DN 25:

Transition DIN 28 403 - B 20 - 25

No.	Number of pieces Form		Standard designation	
	A	B	Designation block	Identification block 1)
1	1	1	Tension ring	DIN 28 403 – 1 – ...
2	2	2	Small flange	DIN 28 403 – 2 – ...
3	1	–	Centering ring	DIN 28 403 – 3 – ...
4	1	1	O-Ring	DIN 3771 – ...
5	–	1	Transition ring	DIN 28 403 – 5 – ... – ...
1) Include the nominal width DN in the designations for No. 1 to 3 and 5, e.g.: Transition ring DIN 28 403 – 5 – 20 – 25				

Material: The material has to be indicated by the manufacturer in his documents (e.g. steel).

Assembly dimensions

Nominal width DN Form		a	b	c	e	f	g
A	B	max.	max.	max.	max.	max.	max.
10 and 16	10 to 16	45	62	62	62	32	66
20 and 25	20 to 25	55	75	75	75	40	72
32 and 40	32 to 40	70	90	90	95	46	85
50	–	95	125	125	130	66	125

Dimensions in mm

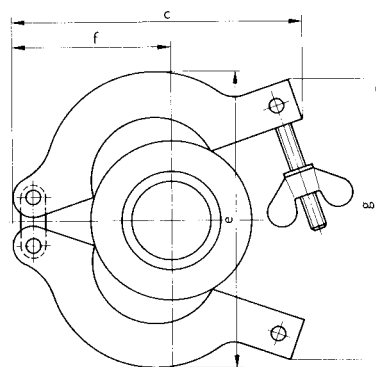


Figure 5-51. Sizes and names of fast flange connections

The layer thickness s should be larger or at least the same as the sum of the roughness.

$$s \text{ [}\mu\text{m]} > R_{t \text{ seal}} + R_{t \text{ flange}}$$

For seals made of several materials, the material of the seal surface is decisive for the required roughness height of the flanges (table 5-3).

Table 5-3. Surface roughness of soft packing

Material of the seal surface	recommended maximum surface roughness of the flange R_t , [μm]
PTFE non-sintered	100
Graphite expandate	100
PTFE sintered	100
Asbestos-containing or asbestos-free papers or yarns without significant binders	50
Asbestos-containing or asbestos-free cords or yarns e.g. with graphite impregnation	50
Asbestos-containing or asbestos-free seal plates	25

5.16

Sealing materials in vacuum technology

In separable connections for the range from rough to high vacuum, usually elastomere and thermoplastics are applied (see table 5-4). For higher demands on tightness even particularly profiled seals with thermoplastic-metal or precious metal alloys are used. In the ultra-high vacuum range, metal seals are used for the separable flange connections. Here, the “Conflat-sealing-method”, which uses copper as seal material, is known. For other kinds of flanges, also aluminum, gold, silver or iridium can be used as sealing material.

Especially in the range of ultra-high vacuum, permanent couplings are often preferred which are manufactured by melting or soldering of the respective metal-glass or metal-ceramics combinations applying the respective technologies. For special problems, liquid seals are used; in the ultra-high vacuum range, metals with a low melting point and low vapor pressure are used.

Table 5-4. Elastomere and thermoplastics for sealing in vacuum technology [5.12]

Abbreviation (DIN ISO 1629)	Explanation	trade name	Special characteristic properties	Application in vacuum technology
1. Rubber NR	natural rubber			flexible connection, thick-walled rubber hose
2. PVC soft	high polymeric polyvinylchloride		chemically resistant, low price	flexible connection, vac- uum hose for use in rough and fine vacuum
3. NBR	acrylic-nitrile- butadiene rubber	Perbunan N	resistant against oil, good mechanic properties, low price	as sealing in general high vacuum applica- tions; sealing ring; temperature range –25 °C to +80 °C
4. CR	polychlorine- butadiene	Neoprene	as 3	as 3
5. MVQ	silicon rubber	Siloprene	Thermally resistant up to maximum 150 °C	Rarely used at tempera- tures up to 150 °C
6. FPM	vinylidenfluorid- hexafluoro- propylene-co- polymerisate (fluorine rubber)	Viton	Thermally resistant from –10 °C to + 200 °C	in ultrahigh-vacuum technology mainly used as seal material between heatable flanges (sealing ring)
7. PTFE	polytetrafluoro- ethylene	Teflon Hostaflon Halon	Thermally resistant up to maximum 300 °C (no-load), very small gas emission	as bellows material in high and ultra-high vacuum
8. CFM (PCTFE)	polytrifluor- chloroethylene	Kel F	similar to 7, thermally resistant up to 50 °C under load	valve seal in valves of the cryogenic technology
9. –	Copolymer of tetrafluoroethylene Perfluoromethyl- vinylether	Kalrez	high thermal resistance (300 °C) and chemical resistance	O-rings, hoses, plates

5.17

Vacuum greases

In separable ground joints and taps as typically designed for glass apparatus in the range from rough to high vacuum, for the assembly a thin layer of a special grease is applied between two ground surfaces. With this grease layer better tightness is to be achieved. The applied greases must therefore be stria-free disposing of a low vapor pressure at the viscosity adapted to the particular application. Table 5-5 gives an overview of some applied vacuum greases.

Table 5-5. Selection of vacuum greases and their properties [5.13]

Name	Manufacturer or supplier	t_{\max} [°C]	p_D (at t) [Pa]	Notes
High vacuum grease 5	1	≈ 40		Long-fiber, soft
20	1	≈ 40	$< 10^{-4}$	Short-fiber, medium
30	1	≈ 50	(300 K)	Short-fiber, soft
40	1	≈ 50		short-fiber hart
High vacuum grease 33	2	150		similarly to HV grease 30
81	2	150		from manufacturer 1
110	2	220		only for pre-vacuum
217	2	220		
Ramsay-grease, viscous	3	25	10^{-2} (20 °C)	ground joints up to 1 Pa
Ramsay-grease, soft	3	25	10^{-2} (20 °C)	for taps up to 1 Pa
L-H high vacuum grease P	3	25	u	for grounds and taps at low final pressure
L-H high vacuum grease R	3	30	u	As with L-H-HV grease P, however hydrocarbon-free vacuum
Lithelen*)	3	150	u	grounds and taps at low
Silicone high vacuum grease **)	3	150	u	final pressures and high operating temperature
Apiezon L	4	30	10^{-7} (25 °C)	only for temporary seals
Apiezon M	4	30	10^{-6} (25 °C)	
Apiezon N	4	30	10^{-5} (25 °C)	for conical seals
Apiezon T	4	110	10^{-6} (25 °C)	grease with a high melting point
Grease PB 1	5	–		light
Grease PB 2	5	(30)		for large seals
Grease PB 3	5	(30)		heavy
Lubriseal	6	30		for glass/metal
Vacuseal light	6	50	10^{-3} (25 °C)	
Vacuseal heavy	6	60	10^{-3} (25 °C)	
Celvacene light	7	–	10^{-4} (25 °C)	
Celvacene medium	7	–	$< 10^{-4}$ (25 °C)	
Celloseal	8	–	10^{-4} (25 °C)	soluble in chloroform
Cellogrease	8	–	$< 10^{-4}$ (25 °C)	
Silicone stopcock grease	9	200	10^{-5} (25 °C)	
Silicone high vacuum grease	9	200	10^{-5} (25 °C)	Up to -40 °C

*) The material contains lithium soaps. The consistency changes slightly with the temperature.

**) High molecular substances with silicon and oxygen as generating chain links. At high temperatures, their lubricity is poorer than with Lithelen. Temperatures above 200 °C cause polymerization with high gas discharge.

u not measurable

Manufacturers or suppliers: 1 VEB Technische Wachse Jena, Germany; 2 VEB Carl Zeiss JENA, Germany; 3 Leybold-Heraeus, Germany; 4 Edwards High Vacuum Ltd. and Shell Co., GB; 5 Comp. Générale de Radiologie, France; 6 Central Scientific Comp., USA; 7 Consolidated Vacuum Corp., USA; 8 Fischer Scientific Co., USA; 9 Dow Corning Corp., USA.

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- Spiralwärmeaustauscher, Company Edition Alfa-Laval Industrietechnik GmbH, Company Edition No. IB 626 59 T, 8711
- Korobon Graphit-Wärmeaustauscher, Company Edition QVF Glastechnik GmbH
- Temperaturregler und Sicherheitstemperaturbegrenzer, Company Edition Spirax Sarco GmbH, Company Edition No. TR-3/87
- Durchflußmesser Typen RE 580-585 mit Ganzmetall-Armatur, Company Edition Turbo-Werk Meßtechnik GmbH, Company Edition No. 01/1985

Betriebsanleitung Nadelventile EVN 010 H1 und EVN 010 H2, Company Edition Balzers AG, Company Edition No. BP 800035 BE, 7.80/DN 9187

ARI-Armaturen, Company Edition Albert Richter GmbH & Co. KG

2/2-Wege Ventile DN 15 bis 100 für neutrale, gasförmige und flüssige Medien, elektromagnetisch betätigt, Baureihe 84100, Company Edition FR. Buschjost GmbH & Co., Company Edition No. 750 1515.05.02.82

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6

Design of Vacuum Pumps and Pipework

6.1

Leakages in vacuum systems

The generic term “leak” subsumes all gas sources effectuating a gas flow into a vessel, regardless of the kind of gas (see section 4.1). According to the nature of the gas source, a distinction is made between real leaks and virtual leaks.

Real leaks are pores, fissures or cracks in the vessel wall through which the gas can flow from outside into the vessel. These leaks comprise permeation of gas through the wall material, too.

Virtual leaks are gas or vapor sources within a vessel, caused by gas discharge, desorption, substance evaporation, return flow from the vacuum generator or technological processes.

The gas flow power caused by a real or virtual leak is called leak rate.

In technological processes under vacuum, a difference is made between operating pressure and final pressure. The actual vacuum process takes place at *operating pressure*. The *final pressure* is the lowest pressure achievable by the applied vacuum pump.

Since vacuum pumps usually show a decreasing suction capacity in the range of its end pressure, the operating pressure, i.e. the suction pressure of the vacuum pump, should not be too near to the end pressure. When determining the operating point of a vacuum pump, the determination of the suction capacity occurs only after the definition of the suction pressure.

When defining the operating point, the safety margin is always to be added to the suction capacity of the pump, i.e. not in direction of lower suction pressures. Operation at low suction pressures usually requires a more expensive pump or pump set. According to the Boyle-Mariotte's Law, $p \cdot V = \text{constant}$, i.e. halving the pressure causes the doubling of the suction capacity.

When installing a vacuum pump, pipework of sufficient dimensions has to be considered, especially on the suction side, as with the pipework being too narrow, the effective suction capacity is determined only by the conductivity and no longer by the suction capacity of the pump (see section 1.4.6).

6.2

Evacuation time and suction capacity of the pump

If a plant is evacuated from the atmospheric pressure to a certain pressure, the period of time needed to reach the required operating pressure is important in technological processes. If there are no leakages and the gas discharge of inner surfaces can be excluded, the evacuation time can be calculated acc. to equation (6-1) on the simplified assumption of a constant suction capacity \dot{S} of the pump. For p_1 and p_2 the same pressure units have to be inserted in the formula.

$$t = 60 \cdot \frac{V}{\dot{S}} \cdot \ln \frac{p_1}{p_2} \quad (6-1)$$

- t evacuation time [min]
- V volume of a completely dry and tight plant to be evacuated [m^3]
- \dot{S} suction capacity of the vacuum pump [m^3/h]
- p_1 pressure at the beginning of the evacuation [mbar; Pa; Torr]
- p_2 pressure at the end of the evacuation [mbar; Pa; Torr]
- \ln natural logarithm

In case the suction capacity of the pump is not constant over the entire suction pressure range, the evacuation time has to be calculated gradually.

The curve for the values of a natural logarithm depending on the pressure ratio is given in figure 6-1.

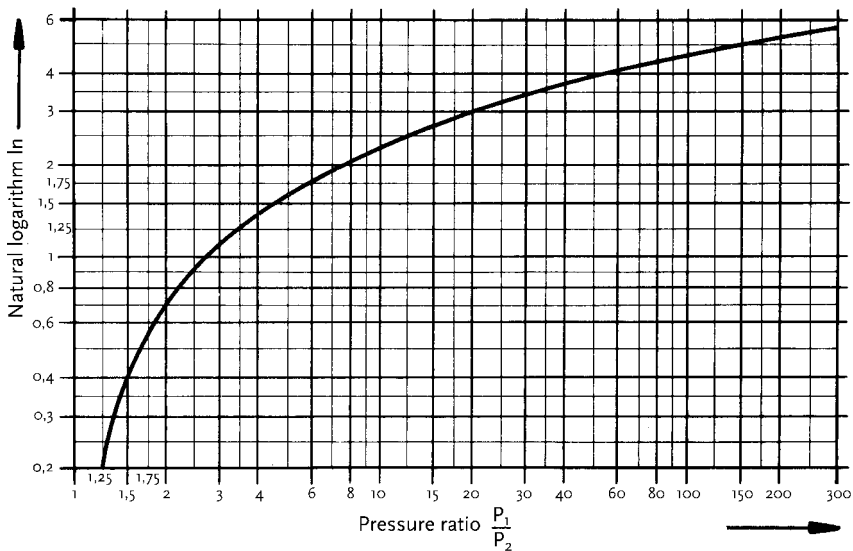


Figure 6-1. Projection of the natural logarithm depending on the pressure ratio

Example 6.1: [6.2]

For pumping out a vessel of $V = 0.50 \text{ m}^3$ to 10^{-2} Torr, a Roots vacuum pump with $\dot{S}_2 = 150 \text{ m}^3/\text{h}$ series installed with a sliding vane pump of $\dot{S}_1 = 10 \text{ m}^3/\text{h}$ is used. Since the Roots vacuum pump is activated only at 50 Torr, the evacuation is carried out in two sections. From 760 Torr to 50 Torr, the sliding vane pump works with $\dot{S}_1 = 10 \text{ m}^3/\text{h}$, and from 50 Torr to 10^{-2} Torr the Roots vacuum pump on the suction side is operated with $\dot{S}_2 = 150 \text{ m}^3/\text{h}$ from. With a presumed constant suction capacity of the vacuum pump over the determined pressure range, a gradual calculation is possible acc. to equation (6-1).

$$t = 60 \cdot \frac{V}{\dot{S}} \cdot \ln \frac{p_1}{p_2} \quad (6-1)$$

$$t_1 = 60 \cdot \frac{0.5}{10} \cdot \ln \frac{760}{50}$$

$$t_1 = 8.16 \text{ [min]}$$

$$t_2 = 60 \cdot \frac{0.5}{150} \cdot \ln \frac{50}{0.01}$$

$$t_2 = 1.70 \text{ [min]}$$

The total time for evacuation is therefore $t_1 + t_2 = 9.86 \text{ [min]}$.

6.2.1

Graphical determination of the evacuation time of vessels in the rough vacuum range

The evacuation time of vessels and vacuum systems can also be determined with good approximation acc. to the nomogram in fig. 6-2.

Example 6.2: (for figure 6-2) [6.3]

A vessel with a volume of $V = 2000 \text{ l}$ shall be evacuated by a rotary plunger pump with a suction capacity at the vessel of $\dot{S}_{\text{eff, max}} = 60 \text{ m}^3/\text{h} = 16.7 \text{ l/s}$ from pressure $p_{\text{start}} = 1000 \text{ mbar}$ (atmospheric pressure) to pressure $p_{\text{final}} = 10^{-1} \text{ mbar}$. The evacuation time is obtained from the nomogram in two steps:

- 1) Determination of τ : Through $V = 2000 \text{ l}$ (scale 1) and $\dot{S}_{\text{eff}} = 60 \text{ m}^3/\text{h} = 16.7 \text{ l/s}$ (scale 2), a straight line is drawn, and on the intersection of this straight line with scale 3, the value $\tau = 120 \text{ s} = 2 \text{ min}$ is to be read (note that the uncertainty of this method amounts to about $\Delta\tau = \pm 10 \text{ s}$, thus the relative uncertainty is about 10%).
- 2) Determination of t_p : The final pressure of the rotary pump is as stated by the manufacturer $p_{\text{end, p}} = 3 \cdot 10^{-2} \text{ mbar}$, the device is clean, and leaks are negligible ($q_{pV, \text{in}} = 0$); then $p_{\text{end}} - p_{\text{end, p}} = 10^{-1} \text{ mbar} - 3 \cdot 10^{-2} \text{ mbar} = 7 \cdot 10^{-2} \text{ mbar}$. Now a straight line is drawn through the point $\tau = 120 \text{ s}$ (scale 3), determined under (1), and the point $p_{\text{end}} - p_{\text{end, p}} = 7 \cdot 10^{-2} \text{ mbar}$ (scale 5), and the inter-

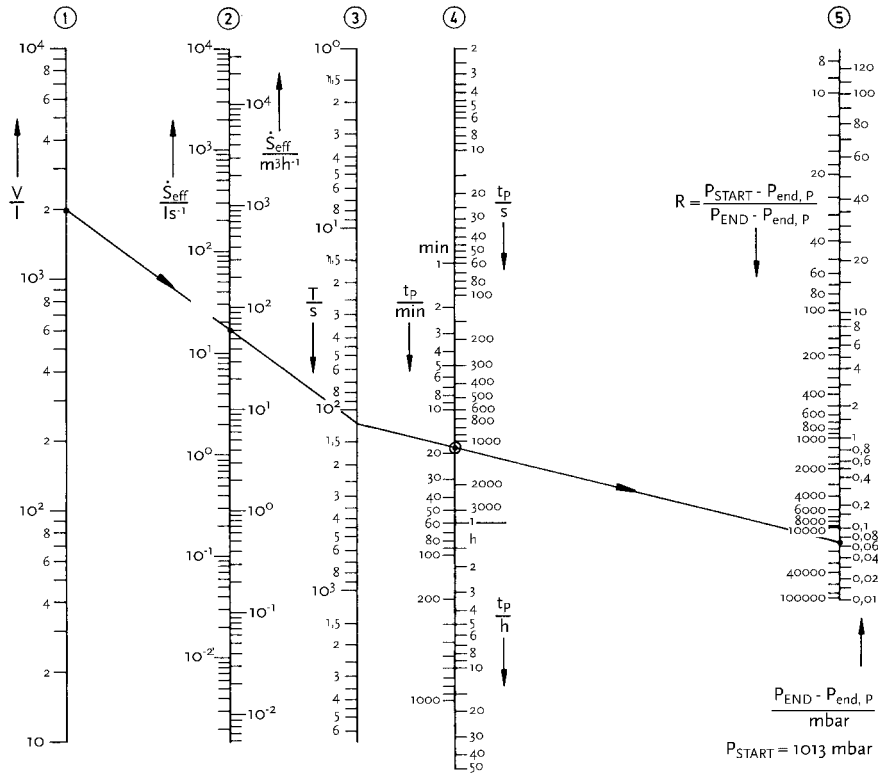


Figure 6-2. Nomograph for the determination of the evacuation time of a vessel in the rough vacuum range ($p \gtrsim 1$ mbar) [6.3]

Scale 1: Vessel volume V in litres.

Scale 2: Maximum of the effective suction capacity $\dot{S}_{\text{eff,max}}$ on the vessel in (left) litres per second or (right) cubic metre per hour respectively.

Scale 3: Time constant (e-value time) τ in seconds, $\tau = V/\dot{S}_{\text{eff,max}}$.

Scale 4: Evacuation time t_p in (right, above) seconds or (left, centre) minutes or (right, below) hours, respectively.

Scale 5: (right): Pressure p_{END} in millibar at the END of the evacuation time, if at the START of the evacuation time, the atmospheric pressure $p_{\text{START}} \approx p_N = 1013$ mbar was prevailing. The desired pressure p_{END} has to be reduced by the end pressure of the pump $p_{\text{end,p}}$, and the scale has to be entered at the differential value. If an inflow $q_{\text{pV in}}$ exists, the scale has to be entered at the value $p_{\text{END}} - p_{\text{end,p}} - q_{\text{pV in}}/\dot{S}_{\text{eff,max}}$.

On the left: Pressure reduction ratio $R = (p_{\text{START}} - p_{\text{end,p}} - q_{\text{pV in}}/\dot{S}_{\text{eff,max}})/(p_{\text{END}} - p_{\text{end,p}} - q_{\text{pV in}}/\dot{S}_{\text{eff,max}})$, if at the beginning of the pumping process, the pressure p_{START} prevails and the vessel is to be evacuated to the pressure p_{END} .

The dependence of the suction capacity on the pressure enters the nomograph and is expressed in the point $p_{\text{end,p}}$ on scale 5. If the pump pressure $p_{\text{end,p}}$ is low, compared with the pressure p_{END} which is intended to be achieved at the end of the evacuation process, this corresponds to the constant suction capacity \dot{S} or \dot{S}_{eff} during the entire pumping process.

section of this straight line with scale 4, $t_p = 1100 \text{ s} = 18.5 \text{ min}$ can be read. (Again the relative uncertainty of this method amounts to about 10%, so that the relative uncertainty of t_p will be about 15%). By adding a safety margin of 20%, the evacuation time is calculated with $t_p = 18.5 \text{ min} \cdot (1 + 15\% + 20\%) = 18.5 \text{ min} \cdot 1.35 = 25 \text{ min}$.

6.3

Determination of suction capacity of vacuum pumps from the leakage of the vessel [6.1]

The leakage \dot{Q}_L of vacuum plants is usually stated in $\text{mbar} \cdot \text{l/s}$ or in kg/h . When carrying out the planning, the leak rates of the systems are often determined on the basis of empirical values. In general, the conversion of the leak rates is usually carried out acc. to the equation (6-2) and (6-3) or table 6-1.

$$1 \frac{\text{kg}}{\text{h}} = 23.09 \cdot \frac{T}{M} \left[\frac{\text{mbar} \cdot \text{l}}{\text{s}} \right] \quad (6-2)$$

$$1 \frac{\text{mbar} \cdot \text{l}}{\text{s}} = 4.33 \cdot 10^{-2} \cdot \frac{M}{T} \left[\frac{\text{kg}}{\text{h}} \right] \quad (6-3)$$

T gas temperature [K]

M molar mass of the gas $\left[\frac{\text{kg}}{\text{kmol}} \right]$

For air at 20°C , the following is valid:

$$1 \frac{\text{kg}}{\text{h}} = 230 \left[\frac{\text{mbar} \cdot \text{l}}{\text{s}} \right]$$

and

$$1 \frac{\text{mbar} \cdot \text{l}}{\text{s}} = 4.3 \cdot 10^{-3} \left[\frac{\text{kg}}{\text{h}} \right]$$

The required suction capacity of the vacuum pump for the pumping of the leak gas flow can be calculated after the determination of the suction pressure acc. to equation (6-4).

$$\dot{S} = \frac{\dot{Q}_L}{p} \quad (6-4)$$

\dot{S} suction capacity $\left[\frac{\text{l}}{\text{s}} \right]$

\dot{Q}_L leakage $\left[\frac{\text{mbar} \cdot \text{l}}{\text{s}} \right]$

p suction pressure [mbar]

Table 6-1. Suction power units and their conversion [6.4]

Unit	mbar · l · s ⁻¹	mbar · m ³ · h ⁻¹	mol · s ^{-1 1)}	mol · h ^{-1 1)}	kg · h ⁻¹ air ²⁾
1 mbar · l · s ⁻¹	1	3.6	4.1 · 10 ⁻⁵	0.148	4.3 · 10 ⁻³
1 mbar · m ³ · h ⁻¹	0.278	1	1.14 · 10 ⁻⁵	4.1 · 10 ⁻²	1.2 · 10 ⁻³
1 mol · s ^{-1 1)}	2.44 · 10 ⁴	8.77 · 10 ⁴	1	3.6 · 10 ³	104
1 mol · h ^{-1 1)}	6.76	24.4	2.78 · 10 ⁻⁴	1	29 · 10 ⁻³
1 kg · h ⁻¹ air ¹⁾	230	840	9.6 · 10 ⁻³	34.5	1

1) assumption: 20 °C, ideal behavior

2) assumption: 20 °C, dry air, ideal behavior

6.3.1

Leak rate values in practice**Leak rates according to the vessel volume**

Table 6-2 shows the leak air flow to be expected depending on the total volume of the equipment and the kind of device and pipework joints. The data is based on empirical values. Here, shaft ducts are not considered. Concerning the seal tightness of vacuum plants, it can be said: the lower the pressure to be maintained in the plant, the higher the demand on tightness of the plant.

Table 6-2. Recommended values of air leak rate in vacuum apparatus and systems [6.5]

	Vacuum apparatus volume to be kept under vacuum in m ³									
	0.2	1	3	5	10	25	50	100	200	500
Vacuum apparatus and pipe connections	Leakage air flow in kg/h									
normally sealed,	0.15–0.3	0.5–1	1–2	1.5–3	2–4	4–8	6–12	10–20	16–32	30–60
predominantly flanged										
partly flanged, partly	0.1–0.2	0.25–0.5	0.5–1	0.7–1.5	1–2	2–4	3–6	5–10	8–16	15–30
welded										
predominantly welded	< 0.1	0.15–0.25	0.25–0.5	0.35–0.7	0.6–1.2	1–2	1.5–3	2.5–5	4–8	8–15
or executed with special seals										

Leak rates of seals

The leakage in vacuum systems can also be calculated by means of the leak rates on the seals used in the plants. Depending on the sealing quality, the calculations can be based on the leak rate standard values given in table 6-3. The stated leak rate values are applicable for suction pressures in the range of 535 [mbar (abs)] or below.

Table 6-3. Recommended values for leak rates depending on the quality and length of seals [6.1]

Quality of sealing	leak rate [kg air per h and m seal length]
Very good	0.03
good	0.10
standard	0.2

6.3.2

Determination of the leak rate by measuring on an existing plant [6.1]

On an existing vacuum system, the leak rate can be determined by measuring the pressure rise. After the evacuation of the plant and the shut-off of the vacuum pump, the pressure rise is measured e.g. in mbar [see equation 6-5].

$$\dot{Q}_L = \frac{\Delta p \cdot V}{t} \quad (6-5)$$

$$\dot{Q}_L \quad \text{leak rate} \quad \left[\frac{\text{mbar} \cdot \text{l}}{\text{s}} \right]$$

$$\dot{Q}_0 \quad \text{additional gas flow} \quad \left[\frac{\text{mbar} \cdot \text{l}}{\text{s}} \right]$$

$$\Delta p \quad \text{measured pressure rise} \quad [\text{mbar}]$$

$$V \quad \text{volume in the plant} \quad [\text{l}]$$

$$t \quad \text{measuring time} \quad [\text{s}]$$

In the case of plants the volume of which cannot simply be determined a second measuring is carried out after the normal measuring of the pressure rise. For the second measuring, an additional, exactly defined gas flow \dot{Q}_0 is let into the vacuum system. According to equation (6-5), the first measuring produces:

$$\dot{Q}_{L1} = \frac{\Delta p_1 \cdot V}{t_1}$$

while the second measurement produces:

$$\dot{Q}_{L2} + \dot{Q}_0 = \frac{\Delta p_2 \cdot V}{t_2}$$

After transposing both equations, equation (6-6) results.

$$\dot{Q}_{L1} = \frac{\dot{Q}_0 \cdot t_2}{\Delta p_2} \cdot \frac{1}{\frac{t_1}{\Delta p_1} - \frac{t_2}{\Delta p_2}} \quad (6-6)$$

Example 6.3: [6.2]

In an impregnation plant, an operating vacuum of 10^{-2} Torr is maintained by a pump arrangement with a suction capacity of 2000 m³/h. The leakage in the plant is to have max. 10% of the pump suction power. How large is the leak rate?

$$\dot{Q}_L = \frac{\Delta p \cdot V}{t} \quad (6-5)$$

With a leakage of 10%:

$$\dot{Q}_L = \frac{\Delta p \cdot V}{t} \cdot \frac{10}{100}$$

$$\dot{Q}_L = \frac{10^{-2} \text{ Torr} \cdot 2000 \cdot 1000 \text{ l}}{3600 \text{ s}} \cdot \frac{10}{100}$$

$$\dot{Q}_L = 0.56 \left[\frac{\text{Torr} \cdot \text{l}}{\text{s}} \right]$$

Example 6.4: [6.2]

A tube transmitter with a volume of 1 l is finished with a vacuum of 10^{-8} Torr. After a storage time of 5 years, the pressure in the valve must not exceed 10^{-5} Torr. What is the permissible leak rate?

$$\dot{Q}_L = \frac{\Delta p \cdot V}{t} \quad (6-5)$$

$$\dot{Q}_L = \frac{(10^{-5} - 10^{-8}) \text{ Torr} \cdot 1 \text{ l}}{5 \cdot 365 \cdot 24 \cdot 3600 \text{ s}}$$

$$\dot{Q}_L = 0.634 \cdot 10^{-13} \left[\frac{\text{Torr} \cdot \text{l}}{\text{s}} \right]$$

6.4**Determination of the pump suction capacity according to the apparatus volume**

As a rule, the suction capacity of vacuum pumps is designed in accordance with the gas and vapor amounts, leakage flows, evacuation time and the operating pressure accruing in the engineering processes. If gas and vapor amounts are not known, the pump suction capacity can be approximately determined according to empirical values depending on the vessel volume according to fig. 6-3. However, the values given in this diagram are only standard values useful for a rough planning. For vessel volumes below 100 l especially in the rough vacuum range, suction capacities lower than those given in figure 6-3 are sufficient.

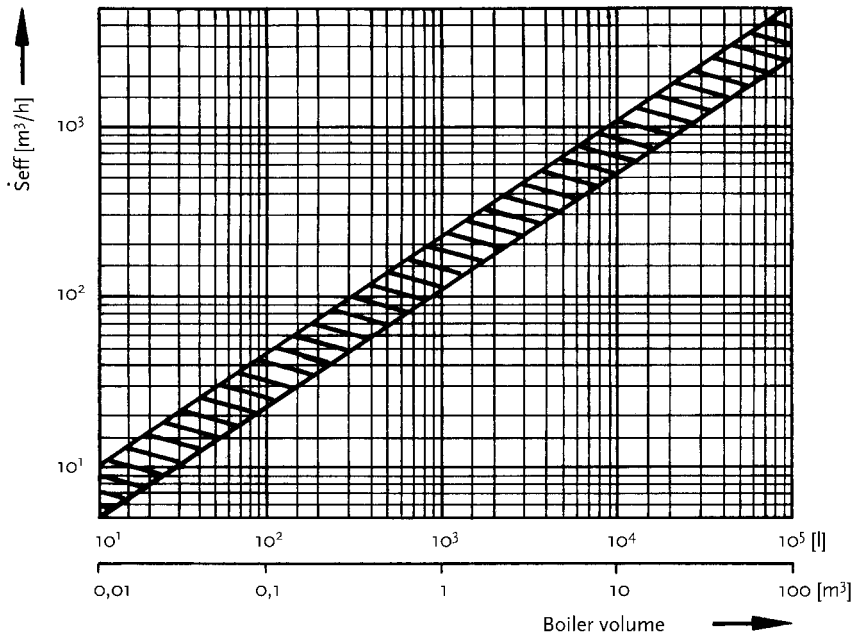


Figure 6-3. Recommended effective suction capacity \dot{S}_{eff} on a vacuum vessel for operation in rough vacuum (and fine vacuum), depending on the vessel volume (recommended values) [6.3]

6.5

Vacuum loss of vessels with different designs [6.6]

At the vacuum leak test and the leak rate measuring, the relevant vessel is evacuated and completely shut off against the vacuum pump. With suitable measuring methods, the arising pressure increase caused by the inflow of a leak air amount during a certain test time can be detected.

An arithmetic determination of the leak air inflow \dot{L}'_r or of the leak rate \dot{L}_r or of the pressure rise Δp can also be carried out with empirically determined formulas. In cases in which the gas permeability of the vessel walls and of the seals cannot be calculated, it is possible to determine the leak air rate approximately according to equation (6-7). In this case, the planned operating vacuum or test vacuum and vessel volume V_g are taken into consideration. The leak rate results from equation (6-8), the pressure rise for any period of time is calculated according to equation (6-9), and the pressure rise referring to one hour acc. to equation (6-10).

The maximum permissible pressure rise for a vessel to be tested is calculated acc. to equation (6-11), with f being a factor depending on the complexity of the vessel to be tested. For the maintenance of a certain operating vacuum a minimum essential suction flow \dot{S}_h is required which compensates the loss continuously and is responsible for the stable vacuum [see equation (6-12) or (6-13) resp.]. Therefore, when determining the suction flow \dot{S} , a vacuum pump should be dimensioned larger by a safety margin s . The safety margin can be assumed as $s=10$ to 6 in rough to fine vacuum, and as $s=6$ to 4 in fine to high vacuum [equation (6-14)].

For a vacuum pump with the given suction flow \dot{S} , it is possible acc. to equation (6-15) to determine the maximum pressure increase Δp permissible for a certain operating vacuum, taking the vessel volume, the safety factor and the vessel factor into consideration.

Leak air inflow

$$\dot{L}_r' = f' \cdot \sqrt{p'} \cdot \sqrt[3]{V_g^2} \quad \left[\frac{g}{h} \right] \quad (6-7)$$

Leak rate

$$\dot{L}_r = \frac{\dot{L}_r'}{\rho} \cdot p_b \quad \left[\frac{\text{mbar} \cdot \text{m}^3}{h} \right] \quad (6-8)$$

Pressure rise

$$\Delta p = p'' - p' = \frac{\dot{L}_r}{V_g} \cdot t \quad [\text{mbar}] \quad (6-9)$$

Pressure rise

$$\Delta p = \frac{f'}{1.225} \cdot \frac{\sqrt{p'}}{\sqrt[3]{V_g}} \quad \left[\frac{\text{mbar}}{h} \right] \quad (6-10)$$

Maximum permissible pressure rise

$$\Delta p_{\max} = f \cdot \frac{\sqrt{p'}}{\sqrt[3]{V_g}} \quad \left[\frac{\text{mbar}}{h} \right] \quad (6-11)$$

Suction stream (stable vacuum)

$$\dot{S}_h \geq \frac{\dot{L}_r}{p_i} \cdot f \quad \left[\frac{\text{m}^3}{h} \right] \quad (6-12)$$

Suction stream (stable vacuum)

$$\dot{S}_h \geq \frac{\Delta p \cdot V_g}{p_i} \cdot f \quad \left[\frac{\text{m}^3}{h} \right] \quad (6-13)$$

Suction stream in the main vacuum pump (nominal power of the pre-vacuum pump)

$$\dot{S} = \dot{S}_h \cdot s \quad \left[\frac{\text{m}^3}{h} \right] \quad (6-14)$$

Maximum permissible pressure increase referring to the suction stream of the available vacuum pump

$$\Delta p = \frac{\dot{S} \cdot p_i}{s \cdot f \cdot V_g} \quad \left[\frac{\text{mbar}}{\text{h}} \right] \quad (6-15)$$

$f = f' / 1.225$ is a factor depending on the complexity of the vessel to be tested

V_g vessel volume [m^3]

s safety margin; in the range of normal to fine vacuum $s = 10$ to 6 , in the range of fine to high vacuum $s = 6$ to 4

p' test vacuum [mbar]

p'' vacuum at the end of the test [mbar]

p_b barometer reading [mbar]

p_i operating vacuum [mbar]

ρ air density [g/m^3]; at 15°C and 1000 mbar = 1225 [g/m^3]

t period of time [h]

For different vessel sizes and test vacuum conditions, based on the factor $f = 1$ the hourly permissible pressure rise, i.e. the vacuum loss can be taken from the diagram in figure 6-4. In this case, the kind of vessel is taken into consideration by multiplying the calculated pressure rise value with the relevant factor f acc. to table 6-4.

Table 6-4. Correction factors f for the kind of vessel in figure 6-4 [6.6]

$f = 1$	Vessels with minimum number of seals (all high vacuum devices)
$f = 1 \dots 2$	Vessels with numerous joints or shut-off valves
$f = 2 \dots 3$	Vessels with plain bearing guides (cold)
$f = 3 \dots 4$	Vessels with plain bearing guides, heated or, as above, with numerous joints and shut-off valves
$f = 4 \dots 5$	Vessels with plain bearing guides, heated, with numerous joints and shut-off valves

Example 6.5: (see figure 6-4)

$V_g = 3.5 \text{ m}^3$, $p' = 10$ mbar:

$\Delta p = 2$ mbar/h for kind of vessel with $f = 1$

or

$\Delta p = 6$ mbar/h for kind of vessel with $f = 3$

or

$\Delta p = 10$ mbar/h for kind of vessel with $f = 5$

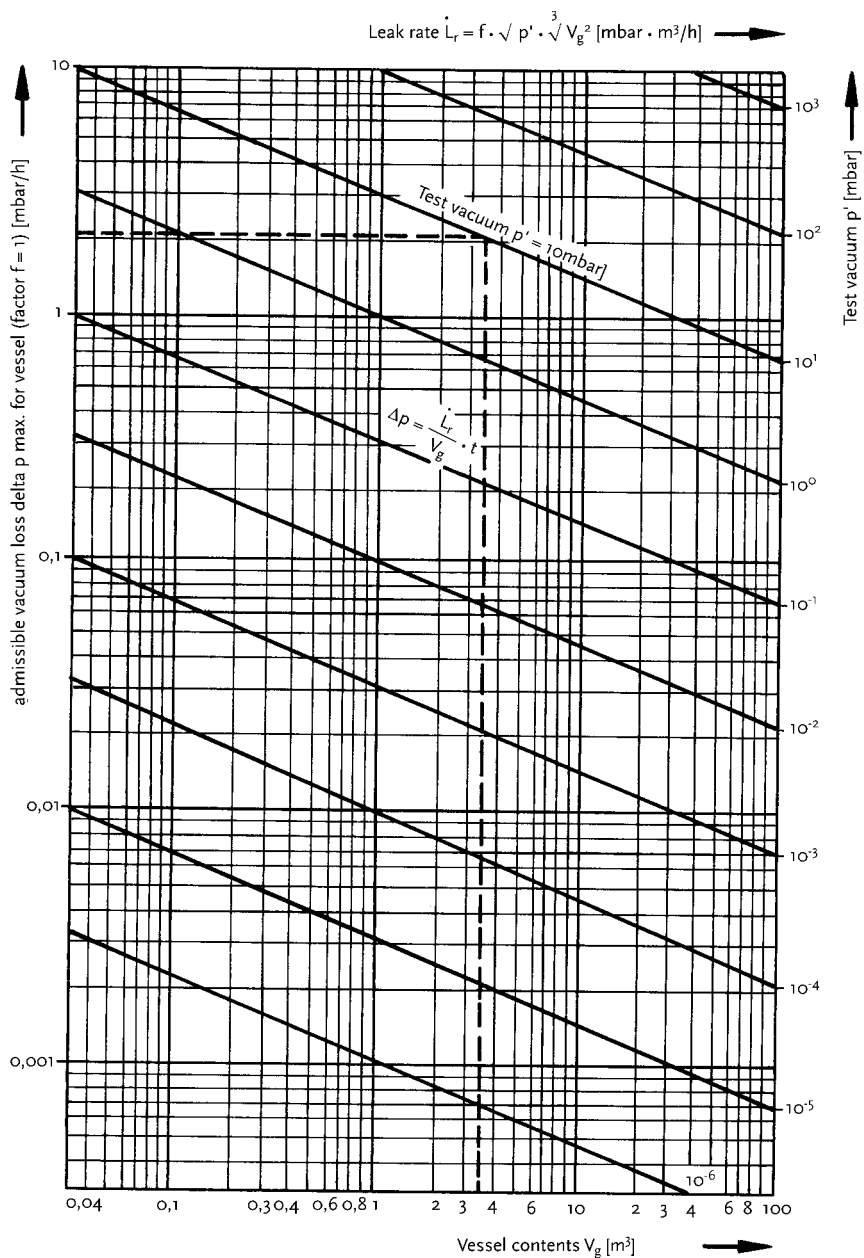


Figure 6-4. Permissible vacuum loss of vacuum vessels in the rough and fine vacuum range at the leak test [6.6]

6.6

Arithmetic determination of volume flows, mass flows and partial pressures

For the dimensioning of liquid ring vacuum pumps not only leak losses but of course also volume flows of the process gases and gas-vapor mixtures to be sucked have to be taken into consideration. The most important relationships for the determination of values of two-component mixtures (e.g. humid air) are described below (for gas laws and calculation examples, see also sections 1.4–1.9).

6.6.1

Calculation of gas-vapor mixtures [6.7]

Value to be calculated:

1. Mixture volume flow, \dot{V}_{mix} [m³/h]

known value:

- a) mass flow, gas or vapor \dot{m}_G , \dot{m}_{vap} [kg/h]

$$\dot{V}_{\text{mix.}} = \frac{\dot{m}_G \cdot R_G \cdot T_{\text{mix.}}}{p_G \cdot 100} \quad (6-16)$$

$$\dot{V}_{\text{mix.}} = \frac{\dot{m}_{\text{vap}} \cdot R_{\text{vap}} \cdot T_{\text{mix.}}}{p_{\text{vap}} \cdot 100} \quad (6-17)$$

with

$$R_i = \frac{R_0}{M_i}, \quad p_G = p_{\text{mix}} - p_{\text{vap}}$$

or

$$\dot{V}_{\text{mix.}} = \frac{\dot{m}_{\text{mix.}} \cdot R_{\text{mix.}} \cdot T_{\text{mix.}}}{p_{\text{mix}} \cdot 100} \quad (6-18)$$

with

$$R_{\text{mix.}} = w_G \cdot R_G + w_{\text{vap}} \cdot R_{\text{vap}}, \quad w_G = \frac{\dot{m}_G}{\dot{m}_{\text{mix.}}}, \quad w_{\text{vap}} = \frac{\dot{m}_{\text{vap}}}{\dot{m}_{\text{mix.}}},$$

- b) volume flow gas or vapor \dot{V}_G , \dot{V}_{vap} [m³/h]

$$\dot{V}_{\text{mix.}} = \dot{V}_G \cdot \left(1 + \frac{p_{\text{vap}}}{p_G}\right) \quad (6-19)$$

$$\dot{V}_{\text{mix.}} = \dot{V}_{\text{vap}} \cdot \left(1 + \frac{p_G}{p_{\text{vap}}}\right) \quad (6-20)$$

$$p_G = p_{\text{mix.}} - p_{\text{vap}}$$

c) molar flow, gas or vapor $\dot{n}_G, \dot{n}_{\text{vap}}$ [kmol/h]

$$\dot{V}_{\text{mix.}} = \frac{R_0 \cdot T_{\text{mix.}}}{p_{\text{mix.}}} \cdot (\dot{n}_G + \dot{n}_{\text{vap}}) \quad (6-21)$$

Value to be calculated:

2. Volume flow: gas or vapor $\dot{V}_G, \dot{V}_{\text{vap}}$ [m³/h]

known value:

a) mass flow, gas or vapor $\dot{m}_G, \dot{m}_{\text{vap}}$ [kg/h]

$$\dot{V}_G = \frac{\dot{m}_G \cdot R_G \cdot T_{\text{mix.}}}{p_{\text{mix.}} \cdot 100} \quad (6-22)$$

$$\dot{V}_{\text{vap}} = \frac{\dot{m}_{\text{vap}} \cdot R_{\text{vap}} \cdot T_{\text{mix.}}}{p_{\text{mix.}} \cdot 100} \quad (6-23)$$

b) mixture volume flow, \dot{V}_{mix} [m³/h]

$$\dot{V}_G = \frac{\dot{V}_{\text{mix}}}{1 + \frac{p_{\text{vap}}}{p_G}} \quad (6-24)$$

$$\dot{V}_{\text{vap}} = \frac{\dot{V}_{\text{mix.}}}{1 + \frac{p_G}{p_{\text{vap}}}} \quad (6-25)$$

Value to be calculated:

3. Partial pressure p_i [mbar]

known value:

a) partial pressure of the vapor stage, p_{vap} [mbar]

$$p_G = p_{\text{mix}} - p_{\text{vap}} \quad (6-26)$$

for saturated mixture:

$$p_{\text{vap}} = p_{\text{vap,s}}(T_{\text{mix.}})$$

b) mass flow, gas and vapor $\dot{m}_G, \dot{m}_{\text{vap}}$ [kg/h]

$$p_G = p_{\text{mix.}} \cdot \frac{\frac{\dot{m}_G}{M_G}}{\frac{\dot{m}_G}{M_G} + \frac{\dot{m}_{\text{vap}}}{M_{\text{vap}}}} \quad (6-27)$$

$$p_{\text{vap}} = p_{\text{mix.}} \cdot \frac{\frac{\dot{m}_{\text{vap}}}{M_{\text{vap}}}}{\frac{\dot{m}_{\text{G}}}{M_{\text{G}}} + \frac{\dot{m}_{\text{vap}}}{M_{\text{vap}}}} \quad (6-28)$$

c) molar flow, gas and vapor $\dot{n}_{\text{G}}, \dot{n}_{\text{vap}}$ [kmol/h]

$$p_{\text{G}} = p_{\text{mix.}} \cdot \frac{\dot{n}_{\text{G}}}{\dot{n}_{\text{mix}}} \quad (6-29)$$

$$p_{\text{vap}} = p_{\text{mix.}} \cdot \frac{\dot{n}_{\text{vap}}}{\dot{n}_{\text{mix}}} \quad (6-30)$$

or

$$p_{\text{G}} = p_{\text{mix.}} \cdot x_{\text{G}} \quad (6-31)$$

$$p_{\text{vap}} = p_{\text{mix.}} \cdot x_{\text{vap}} \quad (6-32)$$

$$\text{with } x_{\text{G}} = \frac{\dot{n}_{\text{G}}}{\dot{n}_{\text{G}} + \dot{n}_{\text{vap}}}, \quad x_{\text{vap}} = \frac{\dot{n}_{\text{vap}}}{\dot{n}_{\text{G}} + \dot{n}_{\text{vap}}}$$

Value to be calculated:

4. Gas resp. vapor mass flow $\dot{m}_{\text{G}}, \dot{m}_{\text{vap}}$ [kg/h]

known value:

a) volume flow gas resp. vapor $\dot{V}_{\text{G}}, \dot{V}_{\text{vap}}$ [m³/h]

$$\dot{m}_{\text{G}} = \frac{\dot{V}_{\text{G}} \cdot p_{\text{mix.}} \cdot 100}{R_{\text{G}} \cdot T_{\text{mix.}}} \quad (6-33)$$

$$\dot{m}_{\text{vap}} = \frac{\dot{V}_{\text{vap}} \cdot p_{\text{mix.}} \cdot 100}{R_{\text{vap}} \cdot T_{\text{mix.}}} \quad (6-34)$$

b) mixture volume flow \dot{V}_{mix} [m³/h]

$$\dot{m}_{\text{G}} = \frac{\dot{V}_{\text{mix.}} \cdot p_{\text{G}} \cdot 100}{R_{\text{G}} \cdot T_{\text{mix.}}} \quad (6-35)$$

$$\dot{m}_{\text{vap}} = \frac{\dot{V}_{\text{mix.}} \cdot p_{\text{vap}} \cdot 100}{R_{\text{vap}} \cdot T_{\text{mix}}} \quad (6-36)$$

c) mass flow of the gas resp. vapor \dot{m}_G , \dot{m}_{vap} [kg/h]

$$\dot{m}_G = \frac{R_{\text{vap}}}{R_G} \cdot \frac{p_G}{p_{\text{vap}}} \cdot \dot{m}_{\text{vap}} \quad (6-37)$$

$$\dot{m}_{\text{vap}} = \frac{R_G}{R_{\text{vap}}} \cdot \frac{p_{\text{vap}}}{p_G} \cdot \dot{m}_G \quad (6-38)$$

Legend of formulas

Symbol	Unit	Values
\dot{m}_{mix}	[kg/h]	mass flow of the mixture
\dot{m}_i	[kg/h]	mass flow of gas i
M_i	[kg/kmol]	molar mass of gas i
\dot{n}_i	[kmol/h]	molar flow of gas i
\dot{n}_{mix}	[kmol/h]	total molar flow of the mixture
p_{mix}	[mbar]	total pressure of the mixture
p_i	[mbar]	partial pressure of gas i
$p_{\text{vap},s}(T_{\text{mix}})$	[mbar]	boiling pressure from table of vapors depending on T_{mix}
R_0	[J/(kmol · K)]	general gas constant = $8.314 \cdot 10^3$ [J/(kmol · K)]
R_i	[J/(kg · K)]	gas constant of gas i
R_{mix}	[J/(kg · K)]	gas constant of the mixture
R_{vap}	[J/(kg · K)]	gas constant of water vapor = 461.5 [J/(kg · K)]
R_G	[J/(kg · K)]	gas constant of air = 287.05 [J/(kg · K)]
T_{mix}	[K]	mixture temperature (absolute)
\dot{V}_i	[m ³ /h]	volume flow of gas i
\dot{V}_{mix}	[m ³ /h]	mixture volume flow
w_i	–	mass share of gas i
x_i	–	molar share

6.7

Flow velocities of liquids, vapors and gases

The adequate dimensioning of the pipework is very important for the trouble-free functioning of a vacuum plant. This concerns the pipes carrying liquid as well as vapor and gas. Pipe joints are to be dimensioned and laid as short as possible and straight, considering the flow velocities (fig. 6-5). In long pipes, the pipe cross-sections may need to be enlarged out of consideration for the maximum permissible flow resistance. Table 6-5 gives an overview of customary velocities of different fluids.

Table 6-5. Standard value for flow velocities of liquids, vapors and gases [6.8]

Fluid	v [m/s]
<i>Vapor</i>	
low pressure pipe up to 10 bar	15...20
medium pressure pipe for steam machines 10...15 bar	15...20
medium pressure pipe, general, 10 ... 40 bar	20...40
high pressure pipe over 40 bar	30...60
exhaust vapor pipe	15...30
degasser pipe	10...30
<i>Water</i>	
drinking water and process water long distance pipe,	up to 3
drinking water and process water main pipe	up to 2
drinking water and process water local pipes	0.5...0.8
cooling water suction pipe	0.7...1.5
cooling water discharge pipe	3
feed water inflow pipe	0.5...1
feed water discharge pipe	1.5...3
condensate suction pipe, without air discharge	0.5...1
condensate discharge pipe	1.5...3
hot water and warm water long distance pipe	1...2.5
hot water and warm water local pipes	0.8...2
water turbine pipe, long and flat	1...3
water turbine pipe, steep, with small diameters	2...4
water turbine pipes, steep, with large diameters	3...7
process water pipe – length longer than 100 x DN	up to 20
process water pipe – length shorter than 100 x DN	20...30
mine water discharge pipe	1...1.5
ash drainage pipe, suction side	0.5...1
ash drainage line, discharge side	1.7...2.5
purge water	1...2
coal slurry	1...1.5
chemicals, liquid	3...5
<i>Gases</i>	
town gas local network, domestic supply up to 0.02 bar	up to 1
town gas local network, low pressure pipe up to 0.05 bar	3...8
town gas local network, medium pressure pipe up to 1 bar	3...10
town gas local network, high pressure pipe higher than 1 bar	10...25
industrial gas low pressure pipe up to 1.05 bar	3...20
industrial gas medium pressure pipe up to 2 bar	5...25
industrial gas high pressure pipe higher than 2 bar	20...60
gas and gas mixtures 200...300 bar (chem. high pressure plants)	8...12
gas and gas mixtures up to 700 bar (chem. high pressure plants)	5...8
grid gas high pressure	25...60
grid gas medium pressure	10...20
flue gas collector suction pipe	15...25

Table 6.5 Continued

Fluid	v [m/s]
flue gas collector discharge pipe	20...30
hydrogen, up to 20 bar	12...15
compressed air pipe, network	12...25
suction pipe for piston compressor, suction capacity up to 2000 m ³ /h	up to 12
suction pipe for piston compressor, suction capacity over 2000 m ³ /h	up to 20
discharge pipe for piston compressor, suction capacity up to 2000 m ³ /h	up to 20
discharge pipe for piston compressor, suction capacity above 2000 m ³ /h	up to 30
fan suction pipe, up to 0.01 bar	8...12
fan discharge pipe, up to 0.01 bar	15...20
fan suction pipe, over 0.01 bar	up to 20
fan discharge pipe, over 0.01 bar	25...30
centrifugal compressor suction pipe	18...25
centrifugal compressor discharge pipe	25...30
vacuum pump suction pipe	10...30
dust extraction pipe	over 20
oxygen pipes, rust-proof steel pipe	up to 25
oxygen pipes, non-rustproof steel pipe	up to 10
pneumatic conveying pipe for granular substances	over 20
pneumatic conveying pipe for dusty substances	over 15
pneumatic conveying pipe for explosive substances	over 20
<i>Oils, petrol</i>	
heavy-oil suction pipe, heated	0.5...0.8
heavy-oil discharge pipe, heated	1...1.5
long-distance oil pipe, middle oil	1...2
lubricating oil, delivery pipe	1.5...2
lubricating oil, drain pipe	up to 1
petrol-, benzene-, gas oil suction pipes	0.5...0.8
petrol-, benzene-, gas oil discharge pipes	1...1.25

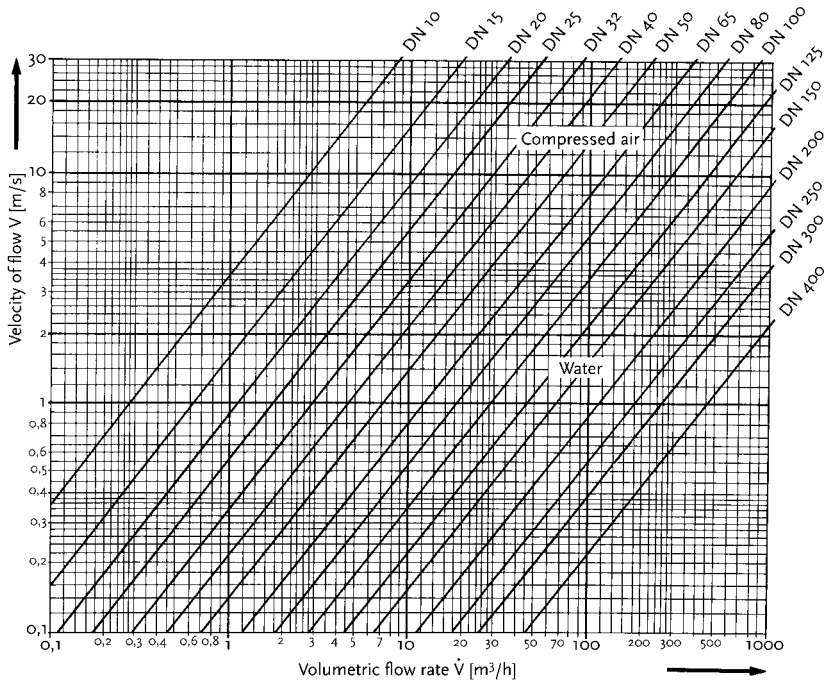


Figure 6-5. Velocities in pipelines

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7

Assembly and Testing of Vacuum Pumps and Systems

7.1

Installation of machines and devices

When installing machines and devices, all layout and base drawings, as well as instructions of the manufacturers are to be adhered to. Before carrying out the assembly work, the foundations should have hydrated. After the installation, the machines are to be aligned with a spirit level and must then be reinforced with a thin cement mortar. The anchor bolts should be tightened only after the cement has been cured. Components and equipment exposed to a thermal expansion stress during operation due to heating must, according to this demand, be mounted movably on gliding bolsters.

During the assembly of the components, important control checks must be carried out. This comprises the examination of the as-delivered condition of the equipment at the building site, control measurements on the assembled components and function tests. The function test includes the visual check regarding the project-conformity of the assembly, the idle speed control of rotating and other moving plant components, the mechanical and chemical cleaning, the leak and pressure tests, as well as the drying of the equipment. For assembly purposes it is useful to apply pre-manufactured equipment in order to keep the on-site production costs low.

7.2

Pipework

7.2.1

General notes regarding installation

Pipes are used for the transport of gaseous, liquid and also fine-grained solid substances. They are essential connecting elements within a plant or between different plants. Depending on the respective engineering process, further components, such as expansion joints, measuring, control and regulation facilities, components for draining and aeration, pipe supports etc. are integrated in the pipe systems. Pipes are mostly subject to pressure and temperature loading. To achieve favorable flow

conditions, efforts are made to lay the pipes with equal cross-sections if possible, and with thin transition pieces if required. The seals protruding to the inside influence the flow behavior negatively; therefore a workmanlike installation is required. Air and water pockets are to be avoided; possibly ventilation and draining taps are to be provided for. Principally, pipes are to be laid with a slight gradient, i.e. as follows:

- vapor pipes with gradient after the discharge point,
- water pipes with gradient after the discharge point,
- compressed air pipes with gradient after the discharge port,
- condensate pipes with gradient after the condensate collector,
- vacuum pipes with gradient towards the condenser.

7.2.2

Cleaning of the pipework

Before the initial start-up of the machines and plants, the inside of the pipes and components must be cleansed from adhering rust, scale and other contaminants. The cleaning methods to be applied depend on the future engineering process and on the purpose of the flow media. The following cleaning methods are known:

- mechanical cleaning,
- blowing out with air,
- flushing with water,
- flushing with vapor,
- blowing out with vapor and
- chemical treatment by pickling.

Decisive factors for the choice of the cleaning method are often the plant units integrated in the pipe system such as liquid or gas pumps, measuring devices, control equipment as well as the appliances installed after the plant component. The cleaning of the pipes is already carried out during the installation, for example with rotary steel brushes.

Before the cleaning process starts, all installed valves and fittings, control elements, measuring facilities, pumps etc. which could be damaged or destroyed by contaminants accruing during the cleaning process, must be removed from the pipework and replaced with suitable adapters.

The pipes to be cleaned have to be equipped with provisional pipe ends for the discharge of the contaminants and the blown-out media, which lead outside or into collecting vessels. When blowing-out by means of compressed air, the discharge of the loose contaminants takes place with the help of the kinetic energy of the air. Flushing with water has to occur from the highest to the lowest point and the speed of the flushing medium has to be chosen almost equal to the speed of the operating medium.

Cleaning by blowing-out with vapor is based on the pulse effect of rapidly flowing vapor and on the erosion effect caused by temperature fluctuation that effectuates the chipping of the contaminants from the inside wall of the pipe. Blowing-through with vapor is only to be carried out after the pipework has been warmed up to the operating

temperature by the inflow of vapor and when this temperature has decreased by approximately 100–200 °C after cutting off the vapor supply. During the following blowing-out, the operating temperature will be reached again and then, when blowing-through, cleaning can be carried out with the maximum vapor mass.

Chemical cleaning allows the removal of rust, scale, dyes and other contaminants without residues. This process called pickling is only reasonable and economical when all the pipes of the circuit can be included. Usually, the recirculation method is used, for which the pickling liquid is being recirculated by a pump after one single filling of the pipework. Pipework, components and devices to be cleaned according to the pickling method, have to be degreased before.

7.2.3

Characterization of the pipework according to the flow media

(Extract from DIN 2403, March 1984)

This standard is applicable for the characterization of non-underground pipework according to the flow media.

A clear characterization of the pipes according to the flow media is indispensable in the interest of safety, proper repair and effective fire-fighting. It must point out to dangers in order to avoid accidents and health injuries.

Table 7-1. Allocation of the colors to the flow media (acc. to DIN 2403, March 1984)

Flow medium	Group	Color	Color symbol acc. to DIN 6164 Part 1	Nearest color pattern in the RAL Color Register RAL840HR
Water	1	green	23 : 7 : 3	RAL 6018
Water vapor	2	red	7 : 8 : 3	RAL 3000
Air	3	grey	18 : 1 : 3	RAL 7001
Flammable gases	4	yellow	2 : 6 : 1	RAL 1021
		or yellow	2 : 6 : 1	RAL 1021
		with supplementary color red	7 : 8 : 3	RAL 3000
Non-flammable gases	5	yellow	2 : 6 : 1	RAL 1021
		with supplementary color black	N : 0 : 9,5	RAL 9005
		or black	N : 0 : 9,5	RAL 9005
Acids	6	orange	5 : 5 : 1	RAL 2003
Lye	7	purple	11 : 2 : 4	RAL 4001
Flammable liquids	8	brown	3 : 5 : 4	RAL 8001
		or brown	3 : 5 : 4	RAL 8001
		with supplementary color red	7 : 8 : 3	RAL 3000
Non-flammable liquids	9	brown	3 : 5 : 4	RAL 8001
		with supplementary color black	N : 0 : 9,5	RAL 9005
		or black	N : 0 : 9,5	RAL 9005
Oxygen	0	blue	17 : 5 : 2	RAL 5015

Pipes must be clearly identified with labels according to the flow media.
In the case of color identification:

- either labels or stickers in the group color acc. to the table 7-1 are to be used,
or
- color rings in the group color are to be fixed,
or
- the pipe is to be painted with the respective group color over its entire length,
or
- labels, stickers or color rings are to be fixed on the points important for operation, e.g. at the beginning, end, branches, wall ducts and fittings.

Flow media transported through the pipework are subdivided into 10 groups according to their general properties the colors of which are defined in table 7-1.

7.3

Leakage tests and pressure tests of devices and pipework in the overpressure range

The functional reliability of plants as well as of mechanical and electrical components is often only given when leakages in walls of devices or vessels are only very small or do not exist at all. In practice, there are basically two different possibilities to detect leakages. A test regarding the *overall leakage* of components without leakage localization is carried out or the *individual leakages* and the size of the leaks are determined. Both possibilities are used in overpressure plants and in vacuum plants. In chemistry and general process engineering, the demands on the tightness of plants have grown significantly. The most important reasons for this are the environmental protection requirements, the workplace safety, the process optimization and the improvement of the quality of the manufactured products. In the case of overpressure plants, leakages lead directly to emissions, which may exceed the permissible limit values set by the "TA Luft", while in the case of vacuum plants this occurs indirectly. In overpressure plants, leakages of hazardous substances may lead to the endangering of man.

Pressure losses caused by leakages in overpressure plants, or rather pressure increase in vacuum plants or undesirable penetration of gases or vapors can slow down the process run and influence product quality in a negative way. Undesirable pressure fluctuations must be compensated by a larger pump power which causes an increase in energy consumption as well.

7.3.1

The leak test

By means of a leak test, the tightness of the pipework, especially of the joints arising from the assembly, is to be checked. This test is also called pre-pressure test. Two methods are used here:

- the diffusion test and
- the test with internal over pressure.

The *diffusion method* uses the physical principle of the diffusion of liquid substances through the smallest pores and orifices existing in solid materials. Due to the application of humidity-absorbing substances (e.g. chalk paint) on the outer pipe wall leakage points can be made visible. The diffusion times depend on the liquid media used and on the wall thickness of the parts to be checked.

In the leakage test based on the method of *internal overpressure*, the verification of the tightness is produced with the help of the pressure of a gaseous substance in the pipe. With the application of foaming agents onto the outside wall or to the joints, the existing leakages become visible. On leak points, strong bubble formation occurs. Apart from air or nitrogen, even halogenated hydrocarbons or e.g. helium can be used. Then the detection of leakages is carried out by means of leak detectors (see section 7.4.2).

7.3.2

The pressure test

The pressure test, also called main pressure test, is carried out both with gaseous and liquid flow substances. Prior to the pressure test, all pipework elements dismantled for cleaning must be reassembled. Fittings lying in the pipework section to be checked must be in working order and in open position. The general functionality of the pipework, the devices and fittings is to be verified at a slightly increased operating pressure.

In pressure tests, the normally arising operating pressure is increased to the determined test pressure. Depending on specific demands, the test primarily comprises the reaching of the maximum pressure during the pressure test, the maximum duration of the holding time under pressure and the control of the assembly joints. If air is used for the pressure test, the gas pipes that have already been in operation should undergo a gas analysis. In order to avoid the development of gas mixtures, inert gases are often used for pressure tests.

If the pressure test is carried out with water, the filling of the pipework has to occur at the lowest point of the pipe system. Before the filling process, vent taps are to be opened. The test pressure is to be generated only after bubble-free water is emitted from the ventilation.

The pressure tests have to be carried out taking the applicable safety regulations into account. With occurring leakages, the pressure increase must be stopped, the pipes have to be decompressed and the damage repaired.

Selection of the test method

The overpressure methods have the advantage that their application allows for leak tests on thin-walled devices that might not bear an underpressure test and would be dented due to the atmospheric pressure existing outside. The pressure test with liquids is mostly used as mechanical stress test. The measuring of pressure drop and pressure increase are integral methods with a leakage localization being impossible. The methods are used for an estimation of the integral leak rates, before a measuring with more expensive methods is carried out to localize the leakage. Ultrasound

measuring can only be applied in the overpressure range. The influence of other sound sources prevents a precise locating of leaks.

In the following a range of principle test methods according to the overpressure method used in practice is described [7.1].

Overpressure leak detection by measuring the pressure drop

The test item with a known volume is filled with a test gas until the desired pressure is reached. On the filled and closed vessel, the occurring pressure drop is measured over a determined period of time. This kind of measurement is called integral leak test, since it determines all leakages together. With the application of special differential pressure gauges respectively chosen, the determination of the pressure drop is easy.

Overpressure leak detection by bubble test

In the case of components or devices with small dimensions being tested, the test item filled, for example, with air up to the test pressure can be examined for leakages by immersing it into water and observing the escaping air bubbles. The leak size can be easily determined e.g. by means of the respective arrangement of a measuring cylinder filled with water over the leak.

Overpressure leak detection by soap bubble test

Using this method, leak-suspect spots on the devices or the plant are being coated with a soap solution and the test item is being pressurized from inside, e.g. with air under test pressure. In specialist shops, adequate agents are even available as sprays under the names of Nekal and Ergantol. The detection sensitivity corresponds more or less to the bubble test under water.

Overpressure leak detection by liquid pressure-testing

The component to be checked is filled with water or another liquid instead of air and then pressure-tested. The pressure-test pressure should be adapted to the maximum permissible device pressure. After this test has been carried out, the test item usually has to be dried. For the detection of possibly existing smaller leaks, more precise leak detection methods are applied.

Overpressure leak detection with chemical processes

If ammonia instead of air is used as pressure gas in a test item, the Ozalid paper (blueprint paper) fixed on the outside of the test item turns dark at the leaking spots. Instead of using Ozalid paper, the surfaces can be wrapped around with gauze bandages soaked with Ozalid. This method is used especially in cases in which later on, the plants are anyway operated with ammonia. With the application of ammonia there is a generally higher corrosion risk.

Overpressure leak detection with halogen-containing gases

For overpressure leak detection with a halogen-containing gas, the device or the vessel is filled with test gas or test gas-air mixture under a slight overpressure. By

means of a sniffer valve, the leak-suspect spots are sniffer-tested from outside thus enabling the exact localization of the leakage (see sections 7.4.2 and 7.4.4). Freon 12 (Freon CCl_2F_2), known from the refrigerating engineering, has made a start as test gas. Freon is non-toxic, cheap and generally available. For overpressure leak detection with halogen-containing gases, commercial halogen detectors are available. For the detection of leaks occurring in rough vacuum or liquid vessels, so-called “barrel leak detectors” can be used. Devices equipped with a small propane gas storage canister in lamp-shape design are guided along the leak-suspect spots and show the existence of halogen-containing gas from a leak by the changing color of the propane gas flame from blue to greenish. This way, coarse leaks are more or less localized. These indicating instruments are frequently used in refrigerating engineering.

Overpressure leak detection with helium

The safest proof method among all overpressure methods is the mass- spectrometric leak detection with a helium leak detector. The sensitivity of this method is first and foremost limited by the helium content in the atmospheric air. The helium escaping through a leak is absorbed by a sniffer valve together with the atmospheric air and then led into the ion source of a mass spectrometer. In the case of devices working according to the principle of the carrier gas sniffer (see section 7.4.2, figure 7-3), the respond time is much shorter compared to designs with a diaphragm sniffer, even in long pipes. A helium leak detector is characterized by the features of a mass spectrometer, with its help and by means of the suction capacity of the pump system, the partial pressure of the test gas is measured (see section 7.4.2, figure 7-2).

7.4

Leak detection methods on components and plants in the range of vacuum and overpressure

For the detection and localization of leakages on vessels and devices, the test unit should always be checked for leakages according to the method corresponding to the purpose of the test unit. Therefore, pressure vessels should be checked according to the overpressure method while a vacuum vessel is to be checked according to the vacuum method. Just like in the range of overpressure, leak detection methods have been developed in the vacuum range, too. Test methods using air or water are well-known and show a generally sufficient sensitivity; in vacuum systems, however, they are practically inapplicable.

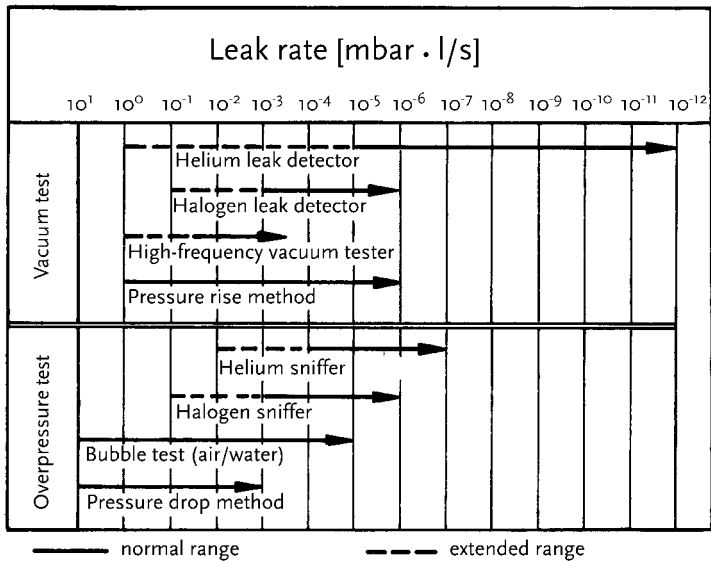
In accordance with today's demands, both for the integral test and for the leak localization, gas-specific proof systems such as halogen and helium leak detectors are used.

An overview of leak detection methods and leak detectors with details concerning leak rates is given in table 7-2.

Leak tests with test gases are partly carried out on components and devices in the overpressure range, too, namely as a rule, according to the overpressure test method.

In this section, a range of customary test methods is described.

Table 7-2. Ranges of leak rates of customary leak detection methods and leak detectors [7.1]



7.4.1

The leak detection

During the leak detection, the individual test sections should be carried out according to the flow chart in fig. 7-1 and to the following steps [7.2]:

1. Leak detection (integral) – yes/no-statement,
2. Leak detection (localization of (a) leaking spot(s),
3. Rework or rejects (for mass products),
4. Repeated leak test,
5. Test item accepted.

7.4.2

Leak detectors

Leak detectors detect the existence of test gases that enter from outside through a leak into a test unit or escape through a leak from the test unit into the atmosphere or enter through a leak into a vacuum plant, for example. In the case of a leakage entering from outside into the test item, proof is delivered in the vessel. In the case the gas escapes through a leak into the atmosphere, the localization of the leakage can occur according to the sniffer leak detection method. In cases in which the test gas enters a vacuum device through a leak and is sucked by a vacuum pump, proof by means of sniffing is possible in the exhaust pipe.

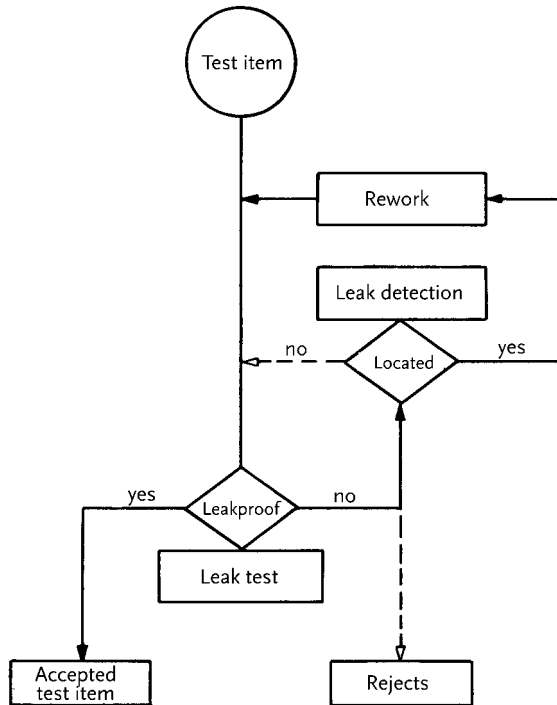


Figure 7-1. Flow chart leak test and leak detection [7.2]

Leak detectors for halogenated hydrocarbons

The halogen leak detector consists of a measuring cell installed in a so-called sniffer in the case of overpressure leak detection (see figures 7-7 and 7-9). For vacuum leak detection (figure 7-2), it is integrated into probes designed for different applications. In the detector, halogen-containing substances on a platinum anode heated to 800 °C catalytically cause a drastic increase of the ion current. The measuring signal led over an amplifier is displayed visually and acoustically.

Halogen leak detectors are rather inexpensive and even relatively sensitive devices enabling however only a qualitative measuring of leak rates. Usually, in the ambient atmosphere, there are no disturbing substances and measuring results remain undistorted.

Up to now, halogen substances were used as test gases, e.g. the refrigerant R 12 acc. to DIN 8960. Due to their detrimental environmental effects and new legal regulations, conventional chlorofluorocarbons (CFC) are substituted by ecologically tolerable chlorine-free fluorocarbons or by other new refrigerants. Owing to this adjustment, new halogen leak detectors and refrigerant-sniffer leak detectors have been developed in the meantime that work according to the principle of a quadrupole mass spectrometer and can selectively proof the existence of different gases.

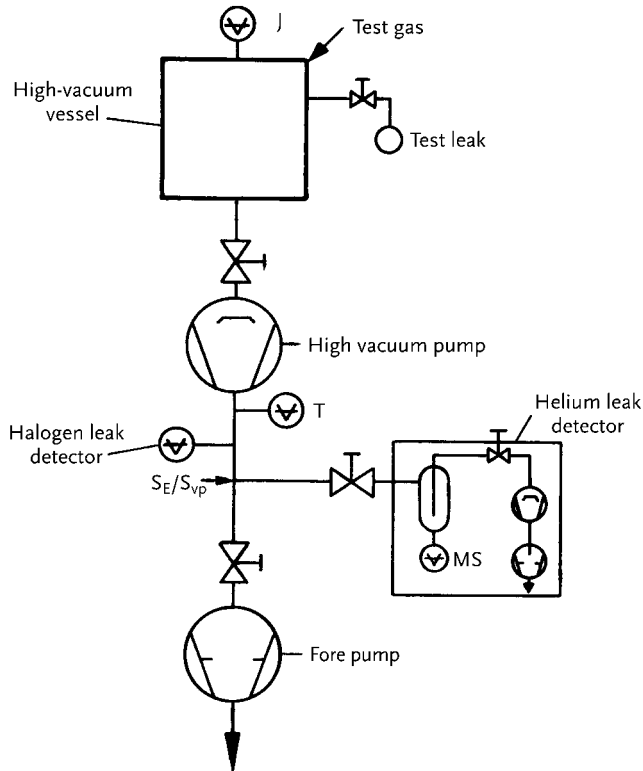


Figure 7-2. Different possibilities of leak detection on a vacuum system. [7.3]

Helium leak detectors

The best results with regard to quality and quantity and the highest sensitivity can be achieved with the application of a helium leak detector. As the name indicates, this is a leak detector responding to helium as test gas. The helium leak detector uses the principle of a magnetic mass spectrometer as test method. The individual gases of a gas mixture are separated according to their specific molecular weight after ionization through electron bombardment of the magnetic forces. The mass spectrometer of a helium leak detector is set to helium (mass 4) and therefore suited to separate helium strictly from other masses. In atmospheric air, helium exists in a concentration of 5 ppm, in usual engineering operations, however, practically not at all. Due to this fact, nil reports are not to be expected. Since the proof of helium is so specific, it is almost unquestionable. Helium as inert gas is an absolute environmentally friendly test medium for leak detection purposes that is neither a risk for persons working with it nor a problem with regard to contamination or corrosion of the components to be tested.

Positive properties of helium are [7.4]:

- non-toxic,
- non-flammable,
- inert, i.e. no reactions with other substances,
- non-condensable in the entire range of technical applications,
- thermally stable,
- with only 5 ppm contained in the air, i.e. extremely low cross interferences,
- in a mass spectrometer, easily separable from neighboring masses, thus no influence on measuring results by other kinds of gas,
- quite inexpensive, e.g. it is not more expensive than halogen test gases.

Since a mass spectrometer can only be operated with a total pressure of $< 1 \cdot 10^{-4}$ mbar, these leak detectors are equipped with an own pump system consisting of a turbo-molecular pump and a forepump [7.3]. Such a leak detector, just like any other leak detecting tube, can be connected to a vacuum system as shown in figure 7-2.

Helium sniffer

The sniffer method is primarily used when the system has to be checked temporarily at overpressure (figures 7-7, 7-9), i.e. if there are not enough vacuum pumps, or if, for technical reasons, these pumps cannot be connected to the system. These leak detectors suck the helium escaping through a leak together with atmospheric air through a sniffer and feed it to the ion source of a mass spectrometer by means of a hose of several meters in length (figure 7-3). The pressure stage between the atmospheric pressure and operating pressure of the leak detector is shown in form of a gas-permeable diaphragm or a fine capillary.

The gas-air quantity flowing in is adjusted to the maximum permissible gas flow of the leak detector.

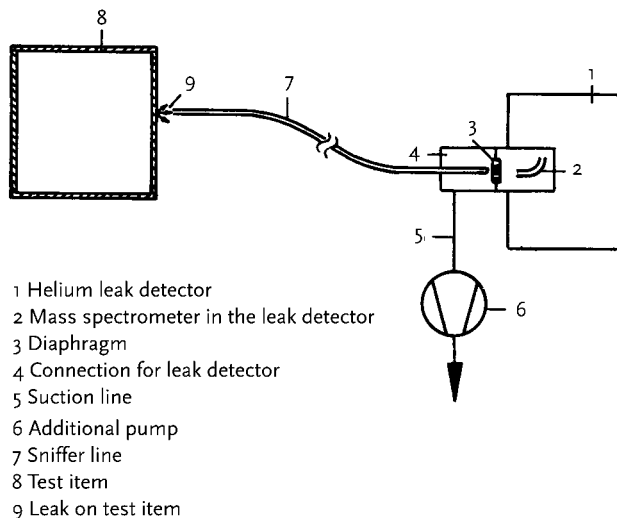


Figure 7-3. Principle of the carrier gas sniffer for helium [7.1]

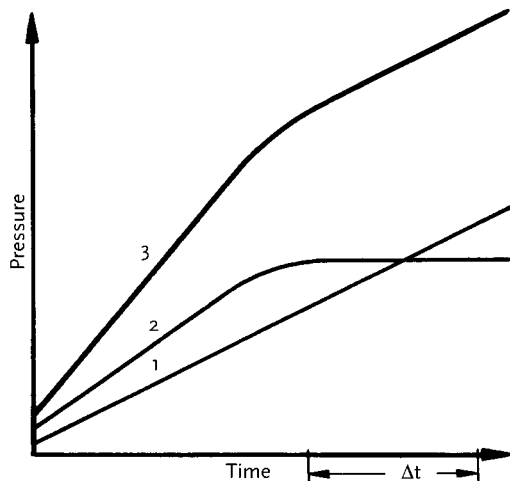
The application of a small additional pump (figure 7-3) considerably reduces the time-consuming measuring process required in the case of large pipe lengths. With this method, working according to the carrier principle, the additional pump sucks a relatively large gas-air quantity through the sniffer pipe part of which is fed to the leak detector via a throttle or diaphragm. Thus, part of the helium contained in the carrier-gas air reaches the helium leak detector equipped with the mass spectrometer.

A sniffer for fluorinated hydrocarbons (Freon) can be designed in the same way as the helium carrier-gas sniffer described above. In this case, the sucked gas-air quantity flows immediately through the detector. Halogen sniffers are used e.g. when in refrigerating engineering, systems filled with Freon have to be checked for leakages.

7.4.3

Integral leak test

A leakage in a vacuum system is easily detected in case the pressure to be expected is not reached. Whether it is a leakage or gas exhaust can be detected by monitoring the behavior of the pressure rise after having closed the valves towards the pump (see figure 7-4). More exact statements with regard to this are possible by means of



- 1 Pressure increase in case of a leak
- 2 Pressure increase due to other effects (evaporation, degassing, desorption)
- 3 Pressure increase of 1 and 2; delta t is the range in which an integral leak rate determination is possible by the pressure rise method also in case of a pressure increase of 1 and 2.

Figure 7-4. Pressure increase in a vacuum vessel after switching off the pump (schematically) [7.4]

determining the gas composition with a partial pressure gauge taking into consideration the gases typical for the penetrating air.

For the detection of leakages in practice, the so-called integral leak control method is used, too.

The integral leak control consists in filling or surrounding the component or system to be checked with test gas or a test gas-air mixture from inside acc. to figure 7-5 or from outside acc. to figure 7-6, and in measuring the test gas quantity escaping from the system or rather entering the system with a leak detector connected to the system. This method is absolutely safe and free from interpretation errors. A further test method for the integral leak control is possible with the help of a test gas sniffer according to the layout shown in figure 7-7. The test unit filled with the test gas is surrounded by a test shell. This test is also called shell test.

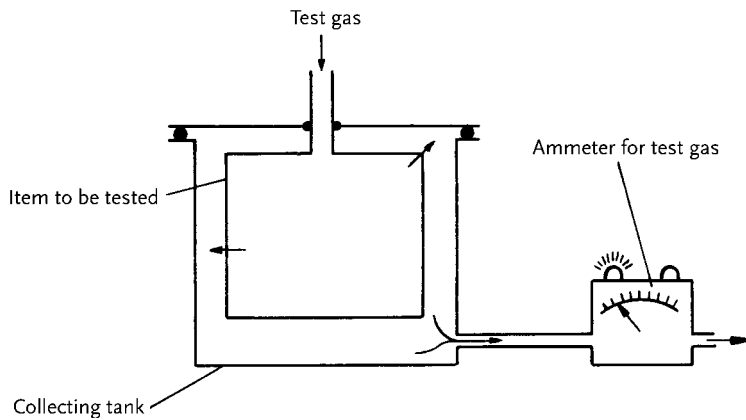


Figure 7-5. Integral leak test from inside to outside [7.3]

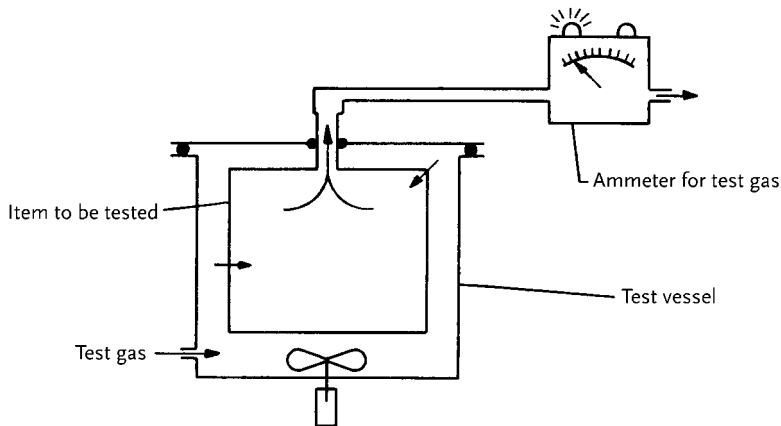


Figure 7-6. Integral leak test from outside to inside [7.3]

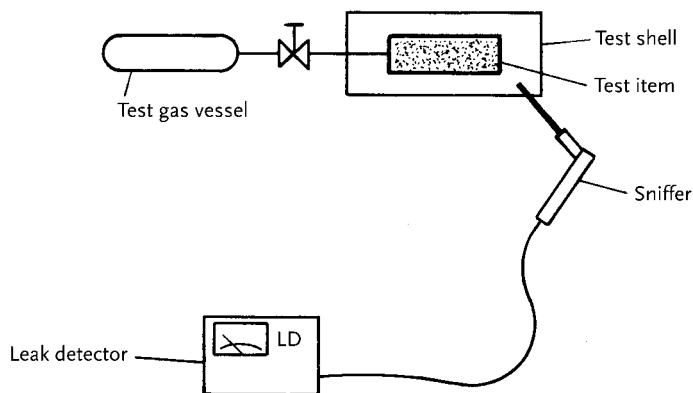


Figure 7-7. Integral leak test according to the principle of the conveying gas sniffer [7.3]

7.4.4

Leak localization on test units under vacuum or with test gas overpressure

Basically, there are two methods for the localization of leaks. A distinction is made between leak tests on components and devices with underpressure and those filled with test gas being under overpressure.

According to the layout shown in figure 7-8, the test unit under vacuum is sprayed with test gas from outside. By means of a suitable pump system, the test gas entering the test unit through a leak is fed to a test gas detector which indicates the existence of a test gas. In the case of components under test gas overpressure,

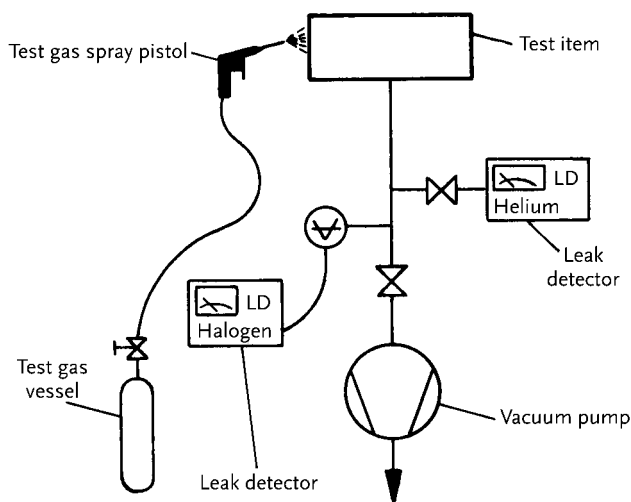


Figure 7-8. Scheme of leak detection by spraying of test gas [7.3]

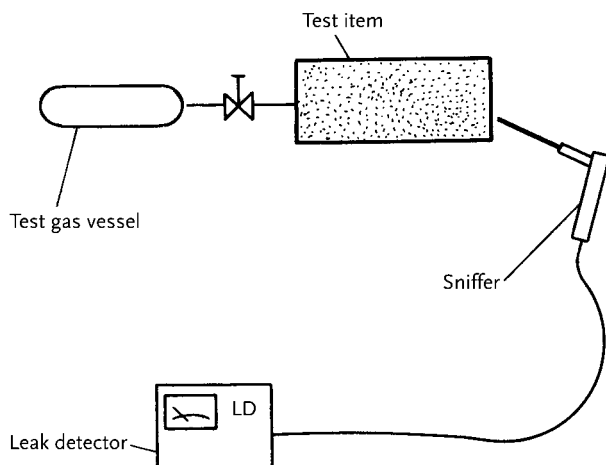


Figure 7-9. Leak detection with a sniffer [7.3]

leak detection can be carried out with a sniffer as shown in figure 7-9. In this case, the escaping test gas is sucked by a probe and led to the test gas detector. With both methods a rather exact detection and localization of leakages is possible.

7.4.5

Leak test methods with helium leak detectors on vacuum plants [7.4]

Vacuum leak detection with helium leak detectors in partial flow operating mode. Figure 7-10 shows the design of a vacuum plant equipped with a helium leak detector in the partial flow operating mode. The maximum permissible test gas quantity is supplied to the leak detector through a metering valve. Before carrying out the measuring, the detection limit and the maximum response time for the helium leak detector are determined by a test leak with a known leak rate. The test leak should be

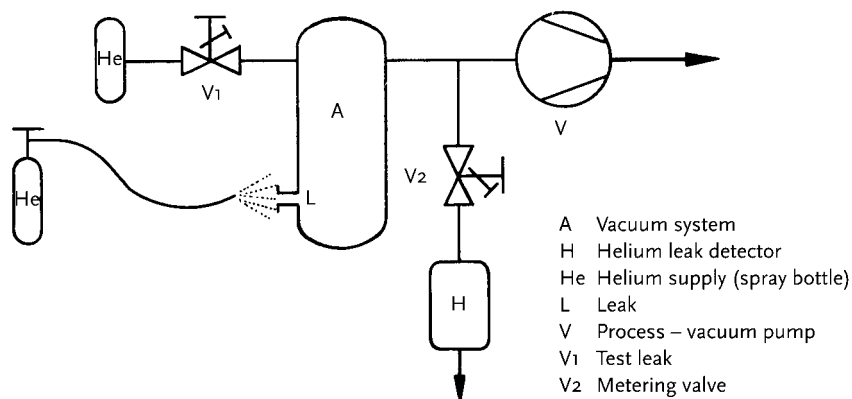


Figure 7-10. Scheme of vacuum leak detection with a helium leak detector in partial flow operation mode [7.4]

integrated into the vacuum plant with as large a distance from the leak detector as possible. During the test, the leak-suspect spots on the components to be checked are sprayed with helium. The leak detector records the helium penetrating the device through a leak.

Vacuum leak detection with helium sniffer. The plant shown in the figure 7-11 disposes of a helium leak detector with a helium sniffer. The sniffer sucks small amounts of gas from the exhaust gas flow and supplies them to the mass spectrometer of a helium leak detector. The helium is sprayed onto leak-suspect spots of the plant. The helium entering the plant through leaks is sucked by the process vacuum pump, conveyed to the discharge side and recorded by the leak detector installed there. This method allows leak tests and the detection of larger leaks in the rough vacuum range without connection to the vacuum circuit and without interruption of the production process.

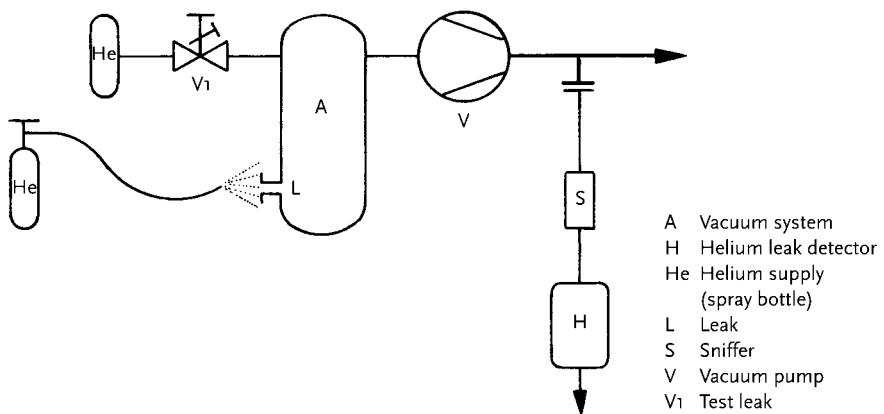


Figure 7-11. Scheme of vacuum leak detection with a helium sniffer [7.4]

Overpressure leak detection with helium sniffer. This detection method using a helium sniffer can be regarded as a refined test method of the known leak test on test units immersed in water. The test unit is filled with helium or a helium-air mixture instead of air. A helium sniffer is applied for the localization of leaks and is led along the leak-suspect spots (figure 7-12). With the method of helium sniffers, water, oil and gas leakages are detected easily and fast. With a helium sniffer, also very large leakages can be detected.

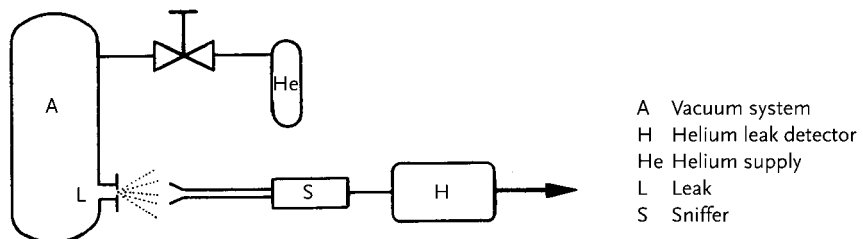


Figure 7-12. Scheme of overpressure leak detection with a helium sniffer [7.4]

7.4.6

Test leak

The test leak shown in figure 7-13 consists of a stainless steel vessel filled with helium. The diffusion leak arranged in the cylindrical vessel is connected to the joint flange (DN 10) via a shut-off valve. The filling nozzle is situated at the end of the reservoir. The test leak is connected to the helium leak detector with the flange (DN 10). Since helium diffuses quite well through glass, the diffusion leak (specified leak) is made of a thin-walled glass pipe. Test leaks are used for the examination of the operational availability, for recalibration and for the sensitivity control of leak detectors. They are also used for the determination and to measuring of the response time. Test leaks of this design are manufactured for the range from $1 \cdot 10^{-7}$ [(mbar · l)/s] up to some 10^{-10} [(mbar · l)/s].

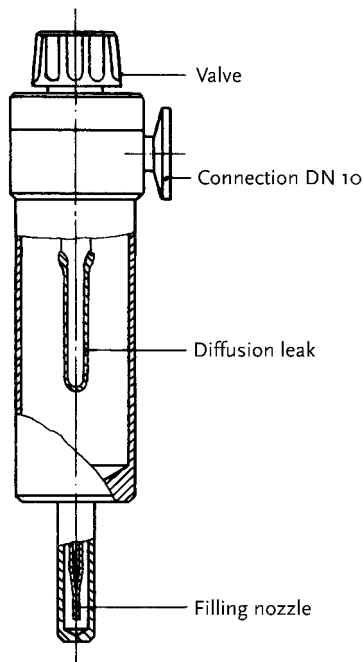


Figure 7-13. Helium test leak [7.3]

7.5

Acceptance and performance tests on liquid ring machines

7.5.1

Acceptance rules

Procedures regarding acceptance tests and technical conditions for the delivery of liquid ring compressors are determined in the VDMA standard sheet 4368, July

1981. The standard DIN 1945, Part 1 refers to the thermodynamic acceptance and performance tests on displacement compressors, however, the definitions of this standard are not unreservedly applicable to liquid ring compressors. The same is valid in the general sense for VDI 2045, Sheet 1. Therefore, the VDMA standard sheet 4368 contains deviating rules for liquid ring compressors going even beyond the DIN 1945, Part 1. An equally valid standard is the DIN 1952 regarding flow metering with standard nozzles, orifices and Venturi nozzles (VDI flow-metering rules).

The acceptance rules particularly for liquid ring vacuum pumps were recorded in DIN 28431, January 1987. This standard determines the procedures of acceptance tests and technical conditions for liquid ring vacuum pumps. The absolute suction pressure for these liquid ring vacuum pumps shall exceed 1 mbar. This standard includes specific data about the measuring of the volume flow rate referring to the suction state and the power consumption as well as the indications concerning the conversion of measuring units to the agreed conditions.

7.5.2

Similar experiment on liquid ring vacuum pumps

(Extract from DIN 28431, January 1987)

An examination of liquid ring vacuum pumps under similar conditions can be carried out, if for the conversion of the performance data the following laws for different operating conditions are applied.

The correct application presupposes:

- geometric similarity of all elements influencing the flow,
- absence or arithmetical elimination of thermodynamic effects influencing the performance, such as heating, cooling, evaporation and condensation,
- adherence to operating limits, such as maximum rotational speed and minimum suction pressures.

The similarity law goes:

If the pressure ratios

$$\Theta = \frac{p_2}{p_1} \quad (7-1)$$

and the pressure coefficients

$$K = 10^2 \cdot \frac{p_1}{\frac{\rho_L}{2} \cdot u^2} \quad (7-2)$$

of different operation points are the same, then the isothermal efficiencies

$$\eta_T = \frac{1}{3.6 \cdot 10^4} \cdot \frac{p_1 \cdot \dot{V} \cdot \ln \frac{p_2}{p_1}}{P} \quad (7-3)$$

and the impeller utilization

$$\lambda = \frac{1}{60} \cdot \frac{\dot{V}}{\frac{\pi}{4} d_{\text{imp}}^2 \cdot B \cdot n} \quad (7-4)$$

are the same, too.

Here the definitions are:

K	pressure coefficient
p_1	suction pressure [mbar]
P_2	discharge pressure [mbar]
u	circumferential speed of the impeller [m/s]
B	impeller width [m]
d_{imp}	impeller diameter [m]
n	rotational speed [l/min]
P	power consumption [kW]
\dot{V}	suction volume flow [m ³ /h]
η_T	isothermal efficiency
λ	impeller utilization
Θ	pressure ratio
ρ_L	operating liquid density [kg/m ³]

Terms

Impeller utilization: ratio of the suction volume flow to the product of the cylinder volume of the impeller and the rotational speed.

Relative humidity: ratio of the absolute and to the maximum humidity

7.5.3

Acceptance test for liquid ring vacuum pumps

(Extract from DIN 28431, January 1987)

Conditions for the acceptance test

- Kind of gas: ambient air
Ambient air is the air at environmental conditions on the test site. The air is regarded as dry if it expands after a throttle before entering the liquid ring vacuum pump.
- Kind of operating liquid: water ($\rho_L = 1000 \text{ kg/m}^3$)

The values to be agreed upon for the acceptance test and the maximum variances are given in tables 7-3 and 7-4. In table 7-5, measuring values are summarized which must at least be determined during the experiments.

Table 7-3. Acceptance test. Values to be agreed upon (acc. to DIN 28431, January 1987)

Measured variable	Formula symbol	Unit
suction pressure	p_1	bar, mbar
inlet temperature	t_1	°C
outlet pressure	p_2	bar, mbar
rotational speed	n	1/min
volume flow of operating liquid	\dot{V}_L	m ³ /h
inlet temperature of operating liquid	t_L	°C

Table 7-4. Acceptance test – maximum variances (acc. to DIN 28431, January 1987)

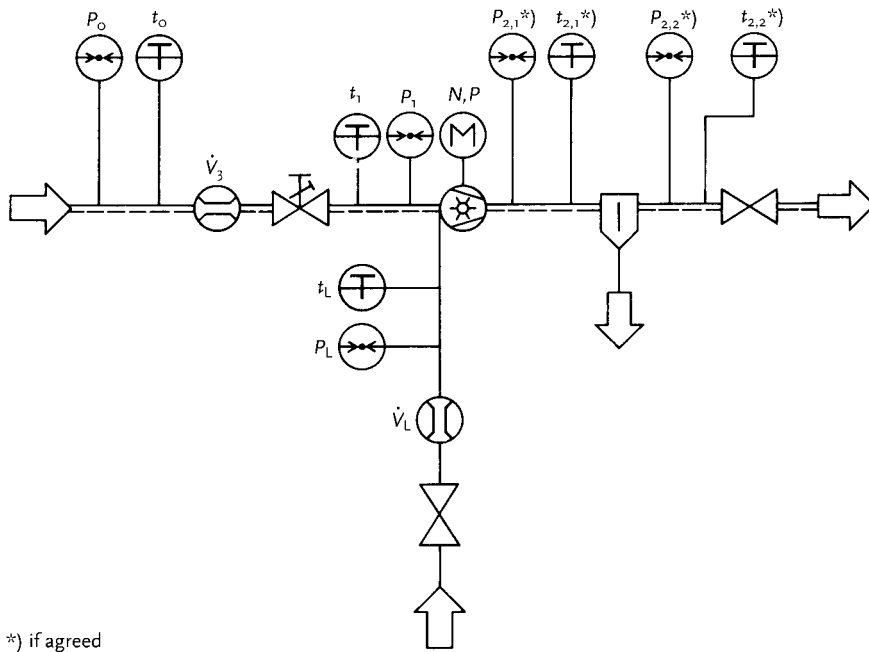
Measured variable	Formula symbol	Unit	Maximum variances ¹⁾ with impeller diameters of	
			≤ 500 mm	> 500 mm
Suction volume flow	\dot{V}	m ³ /h	$\begin{smallmatrix} 0 \\ -10 \end{smallmatrix} \%$	$\begin{smallmatrix} 0 \\ -10 \end{smallmatrix} \%$
power consumption	P	kW	$\begin{smallmatrix} +10 \\ 0 \end{smallmatrix} \%$	$\begin{smallmatrix} +5 \\ 0 \end{smallmatrix} \%$

1) Smaller variances can be agreed upon

Table 7-5. Acceptance test – Compilation of measuring values (acc. to DIN 28431, January 1987)

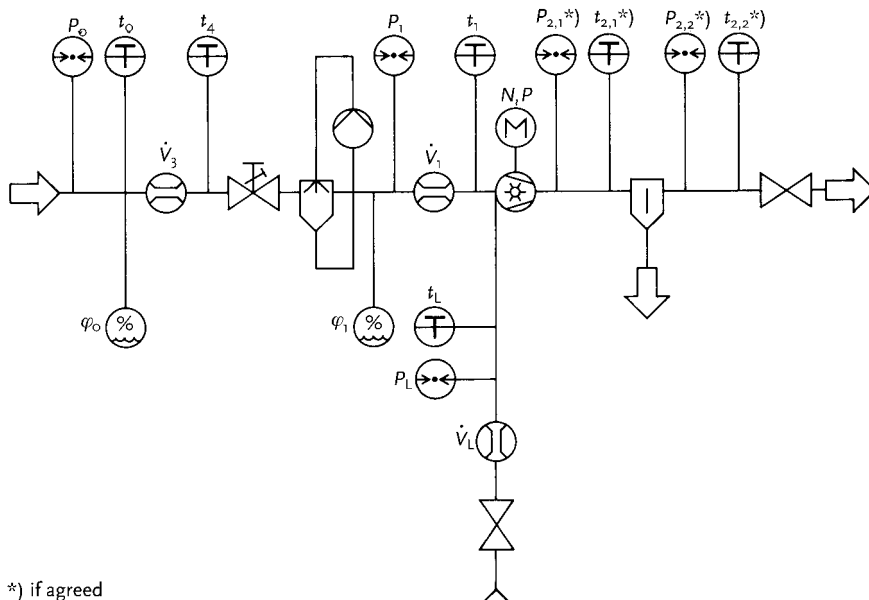
No.	Subject	Formula symbol	Unit
1	date	–	–
2	atmospheric pressure	p_0	bar, mbar
3	gas suction pressure	p_1	bar, mbar
4	gas outlet pressure	p_2	bar, mbar
5	temperature on site	t_0	°C
6	gas inlet temperature	t_1	°C
7	gas outlet temperature	t_2	°C
8	operating liquid inlet temperature	t_L	°C
9	operating liquid inlet pressure	p_L	bar, mbar
10	operating liquid volume flow	\dot{V}_L	m ³ /h
11	power consumption	P	kW
12	rotational speed	n	1/min
13	suction volume flow (measuring value)	\dot{V}_3	m ³ /h
14	suction volume flow (arithmetic or measuring value)	\dot{V}_1, \dot{V}_4	m ³ /h
15	relative humidity of the atmosphere	ϕ_0	–
16	relative humidity in front of the liquid ring vacuum pump	ϕ_1	–
17	temperature of gas in front of the throttle	t_4	°C

The structure of a test arrangement for the acceptance with *ambient air* is shown in figure 7-14 and for the acceptance with *ambient air, measured with air saturated with water-vapor* is shown in figure 7-15. The suction volume flow of a liquid ring vacuum pump increases during the pumping of saturated air and water-vapor mixtures compared to the pumping of dry air.



*) if agreed

Figure 7-14. Acceptance test arrangement – Volume flow rate measured with ambient air (acc. to DIN 28431, January 1987, graphical symbols acc. to DIN 28401)



*) if agreed

Figure 7-15. Acceptance test arrangement – Volume flow rate measured with ambient air, testing with water vapor saturated air (acc. to DIN 28431, January 1987, graphical symbols acc. to DIN 28401)

Upon the entering of humid air into the suction chamber of a liquid ring vacuum pump, the vapor share is more or less condensed. Owing to this condensation, the suction volume flow of the vacuum pump increases.

A test with saturated air delivers proof of a larger suction volume flow caused by the influence of the condensation of the vapor share.

In *simulation tests*, the suction volume flow and the power consumption are measured with the gas or gas-vapor mixture intended for the individual application and with the operating liquid at the agreed suction pressure and discharge pressure.

7.6

Electrical components and cables

Besides the basic demands on the safety-relevant design of the plants regarding the protection of man, plant and environment, even the problem of fire and explosion protection are in the focus of project engineers. The electrical fittings must be carried out in accordance with the official regulations (ex-protection). Attention has to be paid to a faultless electrical grounding. Electrical heating elements may be switched on only after the heating chambers being provided with the corresponding liquid fillings. Thus, a blow-out of the electric resistance or a damage of the seals due to overheating is avoided.

Explosion means exothermal reactions with a significant pressure and temperature rise. There is a danger of explosion if the prerequisites for the occurrence of an explosive mixture, of the corresponding concentration of oxygen and of an ignition source of sufficient energy coincide in place and time. In the case of chemically unstable substances, explosions are possible even without air or oxygen, if exothermal reactions are triggered off by an ignition. Explosion-prone plants have to be designed in such a way that explosion is prevented when at least one of the prerequisites leading to explosions cannot occur. Primary explosion protection comprises measures that prevent the development of explosive mixtures or rather the escape of inflammable liquids and gases. Measures that prevent the ignition of explosive mixtures by eliminating completely or reducing as much as possible the probability of the activation of ignition sources are called secondary explosion protection (see section 9.2). Plants are to be planned and designed in such a way that the consequences of a possible explosion are minimal.

7.7

Insulation

In all pipes, fittings and vessels where heat is generated, conveyed or stored, a part of it escapes into the environment. In order to keep the heat losses low and to reduce the heat transfer to the environment, these components are wrapped up in an insulating layer with high insulation effect. For the selection of the insulating material, factors such as heat conductivity, thermal resistance, volume mass, volume stability

and strength have to be considered. Heat losses are reduced by about 90–95% owing to a careful selection and professional performance of the thermal insulation, which reduces the operating costs significantly. A distinction is made between heat protection and cold protection. Heat protection prevents a flow substance in the pipe from cooling down. Nowadays, insulating material for heat protection are mostly products made of synthetic mineral fibers. Cold protection means the insulation of a pipe or a plant the temperature of which is below the temperature of the ambient air. With cold protection, the warming up of the flow medium in the pipe or plant is prevented. In this case, the temperature gradient is directed from the outside to the inside. Cold insulation is subject to a diffusion process, which requires a careful performance of the cold protection wrappings. A defective cold insulation causes the development of condensate, a temperature rise in the pipes, higher capacities of the refrigerating machines, soaking-through symptoms and the early destruction of the insulating material owing to ice formation. Nowadays, mostly foams, such as polystyrene and polyurethane foam, are used for cold protection.

The thickness of the insulating layer is usually determined from an economic point of view. Heat losses are diminishing with an increasing thickness of the insulating layer. The operational requirements include e.g. the maintenance of a certain temperature gradient of the heat carrier in a pipe, a certain surface temperature and a certain heat loss as well as the avoidance of condensate formation in the case of cold water pipes and refrigerating pipes and the non-freezing of pipes. When planning and designing pipes and plants, the distances from walls, vessels and pipes among themselves is to be chosen in such a way that insulation can be carried out professionally with a protective sheathing.

Generally, insulations are only to be mounted when the tightness of the system is proved. Vacuum drying and evaporation devices should be insulated for thermal efficiency and operational safety reasons, namely particularly the directly vapor-heated part as well as the walls of the vacuum chamber in order to avoid the formation of condensates.

7.8

Putting into operation

With the putting into operation, the so-called start-up, machines and plants are transferred to the operating mode determined according to the project after a successful completion of the assembly. It has to be proved that the equipment meets the requirements the project is based on and that all determined operating points can be reached during continuous operation. Actually, a well-prepared start-up of plants begins already with the projecting of the plant, since the design adequate for start-up has to be planned already in this phase. Before putting into operation, the operator should get acquainted with the specific conditions of the plant, the operating mode of the machines and the operating conditions to be expected. Instructions and conditions to be observed in order to enable the trouble-free operation are given in the operation and maintenance instructions of the manufacturers. It is a com-

mon practice that the operator's staff in charge of operation and maintenance is trained in the handling of the plant components and their operational modes by the experts of the plant designers and machine manufacturers.

Before starting-up rotary machines, a control of the direction of rotation of the driving machine should be carried out, namely with the coupling being released or declutched. Usually, arrows indicating the direction of rotation are attached on these machines. In case of closed systems such as submersible pumps, the pump filled with liquid works in both directions of rotation for a short time while the shut-off valve on the pressure side is closed. The correct direction of rotation is given when the manometer on the pressure side shows the highest pressure. In the case of machines with grease-lubricated roller bearings or with oil-lubricated bearings, the appropriate and sufficient supply of lubricants should be checked. In systems installed and operated outdoors, possibly required protective covers, heating facilities and insulation as well as the availability e.g. of the suitable lubricants for operation in winter are to be checked. Supply lines for cooling, heating and lubrication facilities or for shaft seals should be checked for their proper casing and functioning.

Before the start-up, liquid ring pumps or machines operated with liquid must be filled with liquid and ventilated as far as required. In the case of radial centrifugal pumps with longer pressure lines, non-return valves are usually installed after the pump, in order to avoid a back-flow after stopping the machine. After the non-return valve, a control valve should be installed for the adjustment of the operating point of the pump.

Pumps working according to the displacement principle, such as piston pumps, rotary pumps etc., should be checked for a safety overflow valve or another instrument for the protection of the pump being installed between the pump and the shut-off valve on the pressure side. The function of this safety instrument has to be controlled. Displacement pumps must not be put into operation with a closed shut-off valve on the pressure side, as both the pump and the driving machine may be badly damaged if the safety valve does not open. Shut-off valves in the suction pipe must always be kept open; they are only intended for shutting-off. Due to their flow resistance, fully opened and mainly partially closed shut-off valves on the suction side increase the specific suction work in any case and can lead to cavitation with detrimental effects in the pumps.

For temporary corrosion protection, manufacturers sometimes equip the pump units with a preservative. Before the start-up, the anticorrosive agents, depending on kind and quantity, may have to be removed. The relevant instructions are contained in the operating instructions. When putting vacuum plants into operation, the complete plant will be operated under vacuum for a while without feeding material. After a trouble-free trial run, the production is increased depending on the technological process e.g. gradually until the intended performance is achieved.

7.9

Closing down

Plants and devices closed down for a longer period have to be evacuated and dried completely by the repeated heating under vacuum. A total cleaning is also recommended if due to residual material and humidity accruing in the vessels corrosion may develop. A complete evacuation and drying is to be carried out before changing standards and transport. In these cases, the closing of all vessel and pipe orifices with false covers and the possible insertion of pouches containing drying agents is recommendable. Pumps equipped with conventional shaft seals that suck liquid or gas from a vacuum vessel have to be supplied with sealing liquid also during a standstill, in order to avoid the penetration of air on the shaft seal. This measure is not required for machines with sealless, hermetic design (see also section 3.17.2). Media to be pumped that solidify or crystallize in the machine during the standstill have to be removed. Afterwards the relevant components and pipes must be cleaned and flushed. Rotary machines in stand-by mode or intended for intermittent operation should be put into operation for a short time in regular periods during a long standstill to check their functioning.

Basically, even these stand-by machines have to be supplied by all these auxiliary devices during a standstill as they cannot be put into operation only in the moment of their start-up. These devices include e.g. heating facilities, pressurized seal liquid systems for seals and supply systems for emergency cases.

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8

Materials, Surface Treatment and Safety-at-work in Vacuum Engineering

8.1

Criteria for the selection of materials

In vacuum engineering, the selection of materials requires the consideration of a lot of material parameters apart from the common rules of apparatus construction and mechanical engineering. For vacuum engineering purposes, especially in the high and ultra-high vacuum ranges, materials with a gas discharge rate as low as possible are to be applied. Gas discharge rate means the complexity of permeation and gas flow from the inside of the material as well as desorption of the material surface (see also section 1.4.7). Here, attention has to be paid to the gas discharge rates of elastomere, too. In the case of highly heated components, gases bound inside may diffuse or vapors may escape owing to the saturation vapor pressure of the metals (see also section 1.5.5), as is practiced for example, during the heating process of plants in ultrahigh vacuum. Since the saturation vapor pressure of the selected materials in higher vacuums sometimes determines the achievable maximum vacuum, also in alloys the component with the highest vapor pressure has to be taken into consideration. This applies also to the selection of filler material for soldering and welding, since for the improvement of the flow characteristics (e.g. zinc for brazing), substances with the behavior of the saturation vapor pressure unfavorable for the application in the vacuum range are added. The selection of the coating material for the galvanic surface finish has to be adapted out of consideration for its vapor pressure. Improperly applied coating layers involve the risk of cavities with permanent leaks developing between the base material and the coating. Therefore, galvanic layers should never be provided for on sealing surfaces.

When the material is determined, attention is to be paid even to the fact that different materials tending to local-element formation should not be combined in order to avoid corrosion.

Corrosion elements often occur on valuable metal constructions (electronic components, fittings, measuring devices, etc.) with metals of different electrochemical potentials being in contact, thus causing contact corrosion. With existing humidity, the heterogeneity of a lot of metal materials (e.g. alloys, contaminations, material inhomogeneities) causes the formation of corrosion elements.

Therefore, in the case of high-alloyed steels, for example, contact with other metals should be avoided, especially with ferrous materials.

8.2

Surface treatment

8.2.1

Vacuum hygiene

The perfect functioning of a vacuum pump or vacuum device is also influenced by a precise, carefully and extremely clean assembly. A thorough removal of all foreign substances from the vacuum system prior to its use can reduce the evacuation time considerably. Influenced by the heat supply in vacuum, impurities lead to the decomposition of metal oxides under the discharge of oxygen, or at higher pressures, e.g. at atmospheric pressure, to the sorption of gases and then at lower pressures again to their slow desorption. Dirt particles such as sand or chips can come off of the surface at the beginning of the evacuation and disturb the functioning of the downstream pumps and devices. Oils, greases or other organic residues can emit gases and vapors into the vessel being under vacuum for a long time. Through annealing under inert gas (hydrogen, nitrogen, argon), oxide films can be mostly eliminated and solder flux can be removed. Rough surfaces are a permanent gas discharge source, why in high vacuum and ultrahigh vacuum plants with polished inner surfaces are preferred. Rough surfaces bind approximately ten times more gas than glossy-polished surfaces [8.2]. Polishing leads to a reduction of the real surface.

The assembly of vacuum components and vacuum plants should be carried out under cleanest conditions, not only in high and ultrahigh, but also in fine and rough vacuum. In the ranges of very high vacuum, sweat from hands causes long-lasting gas discharge, why here non-fraying gloves are required for assembly work. Since a lot of solvents increase the gas discharge of elastomere, non-fraying cloth or just water should be used for the cleaning of seals and O-rings.

8.2.2

Corrosion and corrosion protection [8.1]

Metallic and non-metallic materials can be exposed to damaging influences. Figure 8-1 gives an overview of possibly occurring mechanical, chemical and electrochemical wear and tear. Corrosive influences can occur on components of machines and plants exposed to the atmosphere or coming into contact with fluids used in engineering processes.

Commonly used terms, such as corrosion, erosion, cavitation and wear and tear are briefly explained in the following.

Corrosion is the undesirable destruction of materials coming from the surface as a result of chemical or electrochemical reactions between the materials and their environments.

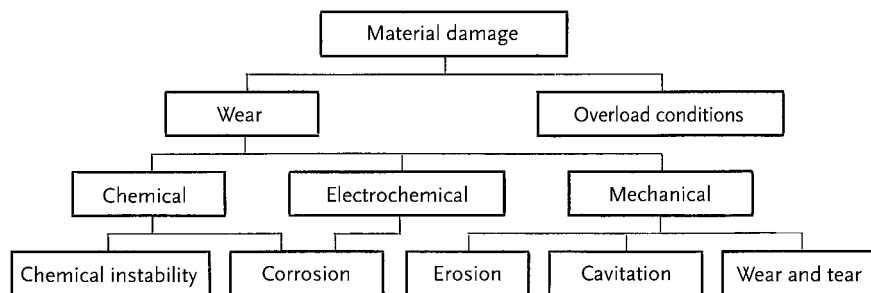


Figure 8-1. Chart of the most important influential factors damaging materials [8.1]

Erosion is the destruction of materials coming from the surface as a result of mechanical action. It is caused by flowing gases, vapors and liquids containing solid particles or gases and vapors with liquid particles.

Cavitation is the destruction of materials owing to implosion caused by degassing or vapor generation as a result of the pressure decrease in flowing liquids (see sections 1.8.8 and 3.11).

Wear and tear arises from mechanical stress. Through this, the metal surface undergoes a modification through material abrasion. Since in practice often more than one influencing factor given in fig. 8-1 is contemporarily acting on the material, it is sometimes rather complicated to detect a clear reason for the destruction of the material. Therefore, it is possible that the application of a certain corrosion protection method increases or reduces the mechanical destruction as well. For this reason, the term “corrosion protection” should not be narrowed down too strictly.

Absolutely corrosion-resistant metals or alloys do not exist. There are only corrosion-resistant materials that are resistant to certain aggressive media. Mostly high-quality alloys are concerned, the application of which is usually worthy of consideration for components, machines or plants with particularly important functions. Chemical instability (instability of the medium) is the destruction of non-metallic materials that are not included in the general term “corrosion”, for instance, solvent aggression on plastic.

Basic protection measures against corrosion include the corrosion-resistant design of components, protective measures on the material, the selection of the suitable material, as well as measures influencing the aggressive medium.

A distinction is made between *active* and *passive* corrosion protection. In the case of active corrosion protection, the protection measures intervene actively in the corrosion process. This is carried out in cases in which the protection by means of a surface treatment process is not possible or not desirable. Known methods are the removal of corrosion stimulants and the application of inhibitors. In the case of passive corrosion protection, cover layers are applied to prevent the metal-decomposing reaction.

In practice, the methods of passive corrosion protection have a broader range of application. In the case of passive protection methods, the pre-treatment of the sur-

faces in the form of chemical or mechanical processes is of great importance. Owing to imperfect surface cleaning, life and quality of the protective coatings are reduced.

Corrosion protection by coatings is sub-divided into *non-metallic* and *metallic coatings*. In practice, non-metallic coatings play a significant role. They comprise surface treatment processes such as chemical passivation, oxidation, phosphatizing and material protection by means of painting and plastic or elastomere coatings.

For temporary or occasional corrosion protection of components and plant components, temporary corrosion protection agents, such as greases, waxes and wax emulsions are used. The protective quality and duration of antirust oils is significantly lower, because of their low film thickness and viscosity. Strippable lacquers and organic- chemical passivation agents can only be used in special cases.

The advantages of metallic coatings are their good protection features and the high stability of the coating. Even with thin layers, this method achieves a high corrosion resistance. Examples are electroplating, electroless metallization, molten metallization, metal spraying, diffusion process and cladding.

8.2.3

Treatment of metal surfaces for corrosion protection by means of inorganic coats

(Extract from DIN 50902, July 1975)

This standard is connected with the recommendations ISO/R 2079-1973 and 2080-1973, issued by the International Organization for Standardization (ISO).

In this standard, terms concerning the treatment of metal surfaces for corrosion protection by means of inorganic layers, including the pre-treatment are listed and explained. Terms in the general sense also determined in other standards are provided with the code of the respective standard.

This standard does not include the corrosion protection of metal surfaces by means of painting material and plastic coatings.

Layer, coating

Covering of base material consisting of one or more layers. It can be formed by means of coating or the modification of the material composition on the surface (see also DIN 8580). Layer and coating are terms that are difficult to differentiate between.

Protective coating

Coating significantly reducing the velocity of corrosion of the base material in case of corrosive stress or preventing corrosion completely (see also Cover Layer DIN 50900 Part 1).

Transition layer. Layer consisting of metal compounds, the development of which is formed owing to reactions of the base materials with a medium on the surface.

Diffusion layer. Layer which is formed through diffusion of a metal or non-metal into the base material.

Pre-treatment of the surface

Cleaning

Generic term for the removal of undesirable substances from the surface, e.g. degreasing, removal of rust, descaling and the removal of drawing compounds (see also DIN 8580).

Mechanical treatment

Brushing. Treatment of the surface with brushes, e.g. made of metal wire, natural fiber or synthetic fiber.

Shot-blasting. Treatment with a blasting abrasive shot onto the surface.

Grinding. Abrasion of the surface with granular abrasives.

Polishing. Levelling out the surface with polish.

Thermal treatment

Flame blasting. Removal of undesirable substances (e.g. rust or scale) from the surface through quick, brief heating with a flame-blasting burner (see also DIN 8522).

Reduced glowing. Bright annealing. Removal of oxides from surfaces through gases with a reducing effect at a high temperature (see also DIN 17014 Sheet 1).

Flame priming. Burning of a transition layer, before applied to the surface (see also DIN 8522).

Physical, chemical and electrolytic treatment

Degreasing. Removal of fat- and oil-containing contaminants from the surface, preferably through solvents or emulsifying media also under the simultaneous influence of ultrasound.

Pickling. Removal of oxides (e.g. scale or rust) and other metal compounds from the surface by means of chemical or electrolytic treatment.

Cleaning by pickling. Short staining to activate the surface.

Burning. Staining of copper materials with niter-containing solutions.

Chemical/electrolytic polishing and glazing. Chemical/electrolytic treatment of metals to achieve a glossy and/or smooth surface.

Drying. Removal of liquid from the surface.

Coating

Application of an adherent layer (see also DIN 8580).

Enamelling

Formation of one or more layers on a surface through melting a preferably vitreous congealing mass of oxide composition.

Thermal spraying

Formation of a layer, e.g. of metals, oxides, carbides, borides, silicates, through hurling the layer material in a finely distributed, melted state onto the material surface (see also DIN 32530).

Vapor deposition

Formation of a layer on the material surface through condensation of the layer metal evaporating in the vacuum.

Electroless metal separation from aqueous solutions

Formation of a metallic layer from an aqueous solution through autocatalytic reduction.

Electrolytic metal separation

Formation of a metallic layer through cathodic metal separation from an electrolyte.

Note: The electrolytic separation of metals is generally called “galvanizing”, while the continuous coating of steel is also called “electroplating”.

Hot-dipping

Formation of a layer through dipping a workpiece into the melt of the layer metal.

Application soldering

Formation of a layer through soldering (see also DIN 8505).

Tin plating. Formation of a layer through melting and wiping tin onto the surface of a base material.

Homogenous lead-coating. Formation of a layer through melting lead, usually firmly bound to the base material by means of intermediate alloy (see also IN 28058).

Cladding. Formation of a layer more corrosion-resistant than the base material e.g. through roller cladding, explosion cladding, application soldering (see also DIN 1910).

Modification of the material compositions on the surface

Definition see DIN 8580.

Blueing

Formation of a thin oxide layer on the steel surface through oxidizing treatment of the surface, resulting in a blue interference color.

Nitration

Formation of a nitrogen-enriched layer on the steel surface through annealing in nitrogen-emitting media (see also DIN 17014, sheet 1).

Boronizing

Formation of a boron-enriched layer on the steel surface through annealing in gases or salt melts emitting boron.

Siliconization

Formation of a silicone-enriched layer on the steel surface through annealing in gases or salt melts emitting silicone.

Alitizing

Formation of an aluminum-enriched diffusion layer on steel through annealing in aluminum-containing powders or through annealing of materials previously provided with an aluminum layer.

Chromizing

Formation of a chromium-enriched diffusion layer on steel through annealing in chromium-separating gases.

Sherardizing

Formation of an iron-zinc diffusion layer on steel through annealing in a mixture of zinc dust and sand.

Black-finishing

Formation of a thin strongly adhering dark-brown to black oxide layer on steel through the treatment of iron workpieces with hot, alkaline salt solutions (see also DIN 50938).

Phosphatizing

Formation of a layer mainly consisting of phosphates through treatment of metals with phosphate-containing solutions (see also DIN 50942).

Oxalizing

Formation of a layer mainly consisting of oxalates through treatment of metals with acid, oxalate-containing solutions.

Chromatizing

Formation of a layer mainly consisting of chromium compounds through treatment with acid or alkaline solutions containing hexavalent chromium (see also DIN 50941).

Anodic oxidizing, anodizing

Formation of an oxide layer on metallic workpieces, preferably light metal, in electrolytes through anodic current.

8.2.4**Formulas for chemical or electrolytic pickling and electrolytic polishing of metals**

Metal parts that are not to be treated mechanically, or on which undesirable residues remain after mechanical cleaning, are better treated or re-treated chemically. Rough surfaces can be refined and smoothed by mechanical or electrolytic polishing.

For some metals, a selection of pickles for the removal of oxides and other metal compounds from surfaces by means of chemical and electrolytic pickling is shown in tab. 8-1. An overview of formulas for the electrolytic polishing of some metals is shown in Table 8-2.

Table 8-1. Selected pickling solutions for some metals [8.2]

Metal	Pickling solutions
Ag	concentrated HNO_3 , or H_2SO_4
Al	10% NaOH (saturated with NaCl), pickling time 15...50 s In case of black coloring due to Cu-content, re-pickling in 20...30% HNO_3 -solution, rinsing in water
Au	aqua regia
Be	alkali hydroxide solution, H_2SO_4 , HCl
Cu	a) <i>pre-dip</i> : 1 l 65% HNO_3 , 10 cm^3 fuming 37% HCl Dip Cu parts for 2...3 s, then rinse with H_2O immediately b) <i>bright-dip</i> : 500 cm^3 65% HNO_3 , 500 cm^3 conc. H_2SO_4 , 10 cm^3 fuming HCl of 37%, 5 g soot Dip Cu parts for 2...3 s, then rinse with H_2O immediately c) <i>matt-dip</i> : 650 cm^3 HNO_3 (density 1.4 g/cm^3), 350 cm^3 conc. H_2SO_4 (density 1.83 g/cm^3), 120 g ZnSO_4 dip tecn. Cu parts into hot pickling bath of 80 °C, rinse in running H_2O and dry in ethyl alcohol
Fe	a) <i>scale pickling</i> : 10...20% arsenic-free H_2SO_4 b) <i>fine pickling</i> : 50% HCl with Ferrocleanol additive (5 cm^3 per 10 l)
Ir	hot aqua regia
Nicosil	10% HCl + 10% HNO_3 (1: l) 2...5 min in movement
Mo	a) hot chromium sulphuric acid at 90 °C b) <i>bright pickling</i> : anodically in cold chromium sulphuric acid
Ni	a) 30...50% HNO_3 b) <i>de-oxidation</i> : 70 °C hot 10% HNO_3
Pd	strong pickling: in tecn. HNO_3 , light pickling: in HCl or H_2SO_4
Pt	hot aqua regia (1 vol.-share HNO_3 + 3 vol.-shares HCl)
Rh	a) hot H_2SO_4 b) solution of NaClO
Ta	a) HF, possibly boiling in HF b) melted KOH
Ti	a) HF b) conc. HCl
W	a) conc. HF + conc. HNO_3 (1:1) Rinsing down in boiling distilled H_2O , then rinsing down 3 times for 3 min. in cold distilled H_2O , then in a pure ethyl alcohol or methyl alcohol. Drying at 90 °C b) anodically in 13% NaOH solution

Table 8-2. Formula for the electrolytic (anodic) polishing of some selected vacuum-engineering metals [8.2]

Metal	Electrolytic composition [cm ³]	J [Ampere/dm ²]	U [Volt]	t° [°C]	t
Al	45 perchlorine acid 800 ethanol 155 dist. water	200...400	100...200	35	8...12s
	165 perchlorine acid 785 acetic acid anhydride 50 dist. water *)	3...5	50...100	50	10...15 min
Cu	670 orthophosphorus acid 100 H ₂ SO ₄ conc. 270 dist. water	10	2...2.2	22	
Fe	530 orthophosphorus acid 470 dist. water	0.6	0.5...2.0	15...20	8...10 min
Mo**)	35 H ₂ SO ₄ conc. 140 dist. water		12	50	
Ni	210 perchlorine acid 790 acetic acid anhydride	18	22	20	
Ta	90 H ₂ SO ₄ conc. 10 HF	10		35...45	9 min
Ti	10 perchlorine acid 200 acetic acid anhydride	100		5	4 min
W	100 NaOH 900 dist. water	3...6		20	20...30 min
Zr	80 perchlorine acid 910 glacial acetic acid 10 dist. water	1...15	12...18	20...30	45s
CrNi-steel 18/8	133 glacial acetic acid 25 g CrO ₃ 7 dist. water	90...250	20	18	4...6 min

*) +4...5 g Al per 1 l electrolyte

**) only for sintered Mo

8.2.5

Paint coats

Still today, the majority of corrosion protection measures in almost all branches of industry are carried out in the form of organic coatings. Usually coatings of paint and lacquers are concerned.

The paint coat material mostly consists of binding agents (film builder), extenders (pigments, fillers), solvents and thinning agents, as well as auxiliary substances (anti-sedimentation agents and skin protectants). These coating systems consist of several coatings adapted to each other. A distinction is made between ground coat, priming coat, and top coat. Ground coats have to create a durable binding between

the metal surface and the other coatings. Priming coats are the links between ground coat and top coat. They must properly adhere to the ground coat and cover its color completely. Top coats protect the coats underneath from the direct influence of the atmosphere or from other influences. The number of coats and the coat thicknesses are to be determined according to the location of installation and the kind of atmospheric load. Table 8-3 shows empirical values concerning coat thicknesses under atmospheric load.

Table 8-3. Minimum layer thicknesses under atmospheric load [8. 3]

Kind of coating	Layer thicknesses in μm	
	plain steels and low-alloy steels	aluminum materials and metallic protective coatings
Shop primers	15...20	–
Ground coats	60	20
Supply coats for systems for load kinds I, It, Ig,	90	50
If	120	50
entire system at load kinds I, It, Ig,	140	100
If	180	100

I	normal industry atmosphere without additional load
If	industry atmosphere with brief additional load
If _w	condensate, spray water, neutral saline solution
If _s	acids
If _A	alkali
If _O	oils, fats, aliphatic hydrocarbons
If _{L1}	aromatic hydrocarbons
If _{L2}	chlorine-hydrocarbons, high-grade alcohols $\geq C_4$, ketones
If _{L3}	low-grade alcohols $< C_3$
Ig	industry atmosphere with permanent gaseous load
Ig _s	acid gases
Ig _A	alkaline gases
It	industry atmosphere with permanent thermal loads
It ₁₀₀	dry heat up to 100 °C
It ₁₅₀	dry heat up to 150 °C
It ₅₀₀	dry heat up to 500 °C

For special demands, chemically resistant and thermally resistant coatings are applied. The protective film formed by the coating on the surface of the workpiece should dispose of the following properties [8.1]:

- long working life,
- high adhesive strength,
- high elasticity,

- low porosity (water and air permeability),
- high wear-resistance,
- high swelling incapacity.

All these demands can be better fulfilled, the better the substrate is prepared.

Residual fats and oils as well as humidity may impair the adhesive strength of the protective coats. It is theoretically and practically proven that the pre-treatment of the substrate is the most important of all influencing factors affecting the functions of organic protective coats.

About 90% of the coating damages are caused by insufficient preparation of the substrate [8.3]. Other influencing factors, such as the kind of coat binding agent applied, the sequence and performance of the work etc., are of minor importance.

Table 8-4 shows some clue values for the durability of protective coatings depending on the substrate preparation. The values are optimal durability guidelines to be expected in the case of proper pre-treatment.

Table 8-4. Durability of coats depending on the kind of substrate pre-treatment [8.1]

Methods of substrate pre-treatment	Quality of substrate pre-treatment	Durability in years
Without substrate pre-treatment	painting on undamaged mill scale	3...4
manual derusting	removal of peeling-off rolling scale and flash rust	4
mechanical derusting	like manual derusting	5...6
flame derusting	removal of mill scale, scale and rust	7...8
pickling	removal of mill scale, scale and rust, completely shiny derusting	6...8
blasting	like pickling	8...12

8.3

Health and safety protection at the workplace during maintenance and operation of vacuum plants

8.3.1

Danger through implosion [8.4]

Owing to atmospheric pressure being effective from outside, vessels, pipes and other components in vacuum systems are exposed to stress working from one side. In the case of metal vessels with insufficient wall thicknesses or with improper design, there is the danger of the vessel walls being dented. Vacuum plants made of glass or consisting of glass components have to be assembled particularly tension-free in order to avoid shattering. Glass vessels and glass devices should dispose of constant wall thicknesses, if possible, to avoid inner tensions. For larger and more complicated components, a visual tension control before the assembly is advisable. The destructions and glass splinters arising from the implosion accidents that

release a considerable amount of energy may cause severe injuries. Therefore, glass vessels are usually enclosed in a protective cage made of wire mesh or perforated sheet metal, in order to keep together the splinters in case of implosion. Works on glass devices should always be carried out wearing protective goggles, face masks and protective gloves as well as wrist protections. In practice, larger vacuum vessels are made of metal or steel according to the requirements. Taking the external atmospheric pressure into consideration, vacuum chambers are preferably manufactured with vaulted bottoms. Glass vessels are usually made with vaulted walls designed as balls, elongated pistons or as bells. As far as flat walls are used for steel or metal vessels, they must be reinforced with ribs accordingly.

For the calculation and design of vessels under vacuum, the general calculation methods of the container construction are applicable. In the case of vacuum chambers it has to be assumed that the external pressure of 1 atmosphere [1 bar (abs)] reacts to the surface.

8.3.2

Auxiliary materials for operation and maintenance of vacuum pumps and plants [8.2]

Coolants and refrigerants

In vacuum engineering, the substances carbonic acid and nitrogen are also used as coolants and refrigerants e.g. for the operation of cold traps. Carbonic acid (CO_2) in its solid state of aggregation sublimates at -78.5°C , the evaporation temperature of liquid nitrogen (N_2) is at -195.81°C . In the case of longer inhalation, 4–5 vol.-% CO_2 are toxic, 8.0 vol.-% are even fatal. Liquid nitrogen evaporates at room temperature causing breathing difficulties in case of higher concentration.

Therefore, these substances should only be handled in rooms that are thoroughly aired out. Due to its high density, the gaseous carbonic acid always accumulates on the floor. A direct physical contact with both refrigerants is to be avoided by means of suitable protective clothing and goggles, as direct contact may lead to burn-like injuries. Carbonic acid in the solid state of aggregation is often used in mixtures with methyl alcohol, ethyl alcohol or acetone, as the eutectic points of these refrigerant mixtures are below the sublimation point of solid carbonic acid. Since the substances mixed with the carbon acid are flammable products, these refrigerant mixtures should not be dealt with near the open fire.

Detergents and solvents

Glass devices are mostly cleaned with acids and lye, either concentrated or diluted. The following substances or mixtures of substances are known:

- hydrochloric acid,
- chromium sulphuric acid (hot, *in statu nascendi*),
- hydrofluoric acid,
- aqua regia (mixture of hydrochloric and nitric acid),
- nitric acid,
- sulphuric acid,

- potash alkali,
- caustic soda.

In order to avoid burns of the skin or the mucous membranes, protective glasses, face masks, gloves and protective clothing should be worn. The inhalation of acid and lye vapors or gas mixtures of the generated reaction products can be avoided e.g. by the installation of gas and smoke exhausters or by the application of respiratory filters.

For the degreasing of components, organic solvents are often used, such as

- ether,
- petrol,
- benzene,
- acetone,
- tetrachlorohydrocarbon,
- and trichloroethylene.

Drying is partly carried out using pure ethyl alcohol or methyl alcohol.

According to the latest knowledge (Report 29 of the Senate Commission/DFG for the control of working substances detrimental to health, dated Sept 1st 1993) and with regard to the classification according to the MAK-values (maximum concentration at the workplace) the substances mentioned above can partially be assigned to carcinogenic working substances and to the substances detrimental to unborn life during pregnancy [8.5]. Substances of this kind have to be handled taking precautionary measures in view of the health-damaging effects. Limiting values of maximum concentrations at the workplace and biological tolerance values for working substances such as diverse gases, vapors or suspended matter can be taken from the lists annually published by the Deutsche Forschungsgemeinschaft (German Research Association) DFG) [8.5].

The properties of harmless detergents and solvents and the regulations regarding their handling can be gathered from the respective product descriptions. Media of this category are usually flammable or explosive. An extensive overview with the most important data of a multitude of materials used in the chemical industry and the general process engineering are provided by the "Handbuch der gefährlichen Güter" (Manual of Hazardous Goods) [8.6].

Mercury

In vacuum technology, mercury is used both as motive fluid for pumps and as a liquid for manometers. Mercury is the only metal which is liquid at room temperature and generates toxic vapors. It forms alloys (amalgams) with many metals. Using a mercurimeter, a special measuring gauge, it is possible to measure the concentration of mercury vapors in the air (up to 0.10 mg/m³). Inhalation of mercury vapors leads to intoxication and the inhaled vapors remain in the human body.

The handling of these substances requires utmost care. Work has to be carried out in thoroughly aired rooms using extractor hoods, if possible, and a respirator filter. Physical contact with mercury should be avoided.

Owing to the high density of mercury, glass vessels may be filled only up to one third. Considering the danger of the glass vessel to break, collecting trays should be arranged under mercury-filled vessels. Large quantities of mercury are usually stored in closed steel vessels. Spilled metal can be collected with a pair of mercury pliers. Mercury residues that cannot be collected are covered with zinc powder or iodine carbon. Owing to the danger of splashing, works with mercury should be carried out on smooth tables with high edges and collecting trays put underneath.

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9

Explosion Protection and Explosion-proof Electrical Equipment

9.1

General

The VDE regulations for firedamp-proof and explosion-proof electrical equipment are based on the harmonized European Standards from EN 50014 to EN 50020. These standards were adopted by the Technical Bureau of CENELEC. As members of CENELEC are obliged to take over the European standards without any deviation to their respective national standards, explosion-proof motors certified according to these standards are permitted for free trade in the member states.

The European Standards from EN 50014 to EN 50020 are given in table 9-1.

Table 9-1. European Standards for electrical equipment for explosion-proof areas

Number	VDE regulation	Title
EN 50014	DIN EN 50014/VDE 0170/0171 Part 1	General requirements
EN 50015	DIN EN 50015/VDE 0170/0171 Part 2	Oil immersion 'o'
EN 50016	DIN EN 50016/VDE 0170/0171 Part 3	Pressurized packaging 'p'
EN 50017	DIN EN 50017/VDE 0170/0171 Part 4	Sand filling 'q'
EN 50018	DIN EN 50018/VDE 0170/0171 Part 5	Flameproof enclosure 'd'
EN 50019	DIN EN 50019/VDE 0170/0171 Part 6	Increased safety 'e'
EN 50020	DIN EN 50020/VDE 0170/0171 Part 7	Intrinsic safety 'i'

Besides the adopted European Standards, the VDE regulations for flame-proof and explosion-proof electrical equipment, VDE 0170 /0171/1.69, were applicable until 05/01/1988. Electrical equipment already used acc. to VDE 0170 /0171/1.69, is permitted to be operated even after 05/01/1988 [9.1].

9.2

Danger of explosion and measures to prevent the ignition of explosion-prone atmospheres

Wherever flammable substances exist, are processed or transported, there is the danger of explosion, i.e.

- if these substances are fine-distributed as gases, vapors, fogs or dusts,
- if their concentration in the air is within certain ranges, and
- if their explosive quantity is endangering.

However, explosions can occur only if all these conditions are fulfilled contemporaneously and an ignition source exists. Possible ignition sources that ignite explosive mixtures thus triggering off explosions are for instance:

- hot surfaces on electrical equipment
- electrically or mechanically generated sparks
- glowing parts
- flames and hot gases
- static electricity
- lightning strike
- electromagnetic waves in certain frequency ranges
- ultrasound
- adiabatic compression
- flowing gases and chemical reactions.

The development of explosive atmosphere may be prevented or limited through appropriate measures. There is, for instance, the possibility to substitute flammable substances for substances that do not develop explosive mixtures. Amongst other things, this may occur by adding inert gases or by appropriate aeration. A further possibility is the removal of ignition sources from explosion-prone areas, if feasible [9.2].

Table 9-2. Comparative concentration values of some gases and vapors in ppm = cm³/m³, ppm · 10⁴ = vol.-% [9.3]

Substance	Limit of perception	MAK-values 1986	lethal after 5–10 min.	lowest ignition limit
Hydrogen	odorless	non-toxic	non-toxic	4.0 · 10 ⁴
Carbon oxide	Odorless in pure state	30	0.50 · 10 ⁴	12.5 · 10 ⁴
Ammoniac	50	50	0.50 · 10 ⁴	15.0 · 10 ⁴
Hydrogen sulphide	2	10	0.08 · 10 ⁴	4.3 · 10 ⁴
Hydrocyanic acid	1	10	0.02 · 10 ⁴	5.4 · 10 ⁴
Benzene	30	5	2.00 · 10 ⁴	1.2 · 10 ⁴
Acetylene			50.00 · 10 ⁴	1.5 · 10 ⁴
Acetaldehyde	1	50		4.0 · 10 ⁴
Diethyl-ether	0.3	400		1.7 · 10 ⁴
Methyl-chloride		50	30.00 · 10 ⁴	7.1 · 10 ⁴
Carbon disulfide	1	10	0.20 · 10 ⁴	1.0 · 10 ⁴

Measures to prevent the development of explosive atmospheres are called “*primary explosion protection*”. Preventing dangers is better than any “*secondary protection*”. Mostly, the requirements of primary explosion protection in areas where people stay are already fulfilled through the implementation of health protection rules (see table 9-2). For most gases and vapors, the MAK-values (maximum concentration at the workplace) that have to be obeyed lie considerably below the lowest ignition limit.

9.3

Zoning of explosion-prone areas

Explosion-prone areas are subdivided into the zones 0, 1, 2, 10 and 11 (see table 9-3). These danger zones differ from each other in frequency and duration of the occurrence of explosive mixtures. The zoning determines the ignition protection class and with this the design of the electrical equipment. The decision whether an outdoor or indoor area is or is not to be regarded as explosion-prone in terms of regulations or standards is exclusively taken by the competent supervisory authority, e.g. by the TÜV or the Trade Supervisory Office.

9.4

Classification of explosion-proof electrical equipment into the main groups I and II [9.2]

Explosion-proof electrical equipment is subdivided into the two main groups I and II according to their application (see table 9-4).

- Group I: Electrical equipment for fiery mines in coal mining.
In areas where fire damp and other gases may be released, electrical equipment is to be designed acc. to the design rules of the groups I and II and has to be checked for the respective explosive mixture of group II, and be marked for both groups.
- Group II: Electrical equipment for explosion-prone areas except for fiery mines.

Within group II, there is a further subdivision into the groups II A, II B and II C for electrical equipment of the ignition protection class “d” and “i”. In the case of the ignition protection class “flameproof enclosure”, the groups II A, II B and II C differ from each other in the limiting width of the clearances measured according to the ignition disruptive capacity of flammable gases and vapors. In the case of ignition protection class “intrinsic safety”, these three groups differ in the minimum ignition current.

Table 9-3. Explosion-endangered areas [9.2]

Flammable substances	Zone acc. to VDE 0165/9.83	Explanations	Examples	Permissible electrical equipment	
				General	Motors
Gases, vapors and fogs	Zone 0	Areas in which a permanent, frequent and long-lasting explosive atmosphere exists	Usually this comprises only the inside of containers or the inside of apparatuses.	In zone 0, only electrical equipment especially certified for this is to be used.	The application of explosion-proof motors is not permitted, regardless of their protection class. Only supervisory authorities can make exceptions.
	Zone 1	Areas in which the occasional occurrence of an explosive atmosphere is to be expected.	They comprise, amongst others: The closer surroundings of zone 0, the inside of apparatuses, the closer areas around: feeding ports filling and discharge facilities fragile apparatus or pipes made of glass or ceramics.	Electrical equipment must be explosion-proof under one of the ignition protection classes acc. to EN 50014.	In zone 1, explosion-proof motors have to be used in ignition protection class “flameproof enclosure”, or “increased safety”, or “pressurized packaging”.
	Zone 2	Areas in which an explosive atmosphere is to be expected only rarely.	This may include, amongst others: areas surrounding zones 0 and 1. Areas around flange connections with flat seals in pipe-work in closed rooms.		In zone 2, explosion-proof motors can be used. Moreover, squirrel cage induction motors of protection class IP 54 can be used without prototype approval certificates, if the maximum surface temperature fulfils the requirements of VDE 0165/9.83, section 6.3 and the extractor fans and fan hoods comply with EN 50014.

Table 9-3. Continued

Flammable substances	Zone acc. to VDE 0165/9.83	Explanations	Examples	Permissible electrical equipment	
				General	Motors
Dusts	Zone 10	Areas in which a dangerous explosive atmosphere exists for a longer time or occurs frequently.	Usually, this comprises only the inside of apparatuses (mills, dryers, mixers, pipelines, silos, etc.).	In zone 10, only electrical equipment especially certified for this is to be used.	The application of explosion-proof motors is not permitted. Only supervisory authorities can make exceptions.
	Zone 11	Areas in which a dangerous explosive atmosphere caused by swirling dust deposits is to be expected occasionally.	Amongst others, this comprises: areas surrounding apparatuses in case of dust being emitted through leaks (e.g. mills).	In zone 11, explosion-proof motors of protection class "flameproof enclosures" or "increased safety" can be used, if they comply at least with protection class IP 54. Furthermore, in zone 11, motors without prototype approval certificates can be used, if they fulfill at least protection class IP 54, the maximum surface temperature complies with VDE 0165/9.83, section 7.1, and the data plate contains the details "Zone 11/surface temperature \leq ... °C". The surface temperature has to be stated only if 80 °C are exceeded.	

Note: For firedamp-proof equipment, the erection regulations VDE 0118 are applicable.
Areas of mining below ground are not subdivided into zones.

Table 9-4. Allocation of groups, protection and temperature classes

	Electrical equipment for areas susceptible to firedamp	Electrical equipment for explosion hazardous areas
Group:	I	II
Protection class:		
Flameproof enclosure	d	d ¹⁾
Increased safety	e	e
Pressurised packaging	p	p
Intrinsic safety	i	i ¹⁾
Oil immersion	o	o
Sand filling	q	q
Temperature class:	—	T 1 to T 6 for all types of protection
Maximum surface temperature of equipment fluid in °C: ²⁾	150°C at places where coal dust can deposit or 450°C if coal dust deposits are excluded.	

1) additionally classified in II A, II B, II C.

2) For type of protection "e" also in interior of equipment.

9.5

Ignition protection classes

The ignition protection classes are decisive for the design of explosion-proof equipment. For electric machines, there are the classes "flameproof enclosure", "increased safety" and "pressurized packaging" (see table 9-5). Due to high costs for the supply of protective gas and supervision, the protection class "pressurized packaging" is mainly used in large machines and plants.

Table 9-5. Ignition protection classes [9.2]

Ignition protection classes	Principle of protection classes	Typical applications
d	Flameproof enclosure All components capable of igniting flammable mixtures are enclosed in explosion-proof casings. An explosion inside the casing cannot penetrate outwards. On the outer components of the flameproof enclosure, certain surface temperatures are not exceeded.	<i>Motors, slip ring case, switching stations, transformers,</i>
e	Increased safety The equipment is designed so that inadmissibly high temperatures, sparks or electric arcs do not occur. Additional mechanical, electric or thermal protective measures improve safety, so that even in the case of disturbances, there is no danger of ignition.	<i>Motors with squirrel cages¹⁾, clamps, measuring devices</i>
p	Pressurized packaging All components capable of igniting flammable mixtures are enclosed in casings flushed with protective gas under a slight overpressure. Explosive mixtures cannot penetrate the casing during operation.	<i>Motors, switching stations, transformers</i>
o	Oil immersion All components capable of igniting flammable mixtures are under oil. An explosive atmosphere above the oil surface or outside the casing is not ignited.	<i>Transformers, static frequency converters, power switches</i>
q	Sand filling All components capable of igniting flammable mixtures are enclosed in casings filled with sand-like bulk material. An electric arc occurring under the filling may not ignite the explosive atmosphere surrounding the casing.	<i>Switching stations, condensers, fuses, resistances</i>
i	Intrinsic safety The energy in an intrinsic safety circuit is kept so low that igniting sparks, electric arcs or temperatures cannot occur.	<i>Measuring and control equipment</i>

1) In the case of slip ring motors together with ignition protection class "flameproof enclosure".

9.6

Temperature classes [9.2]

Group I

The maximum surface temperature of electrical equipment of group I is

- 150 °C on spots where coal dust may deposit or
- 450 °C if the danger mentioned above is excluded, e.g. through insulation against dust.

Group II

For electrical equipment of group II, all ignition protection classes are subdivided into temperature classes from T1 to T6 according to the maximum permissible surface temperature of the equipment and according to the ignition temperature of the flammable substances (comp. table 9-6).

Table 9-6. Temperature classes

Temperature class	Maximal surface temperature of the equipment °C
T1	450
T2	300
T3	200
T4	135
T5	100
T6	85

In table 9-7, the classification into temperature classes of flammable gases is given, while table 9-8 shows the assignment of dusts. The ignition protection class, the group and the temperature class enable the classification of explosion-proof electrical equipment and its fields of application. The numerical sequence of the symbols for group II and temperature classes are chosen in a way that the equipment fulfilling the requirements of a certain group or temperature class, fulfils the requirements of lower groups or temperature classes as well.

Table 9-7. Groups and temperature classes of flammable gases and vapors

Group Temperature classes									
T1	T2		T3		T4		T5		T6
Substance	Ignition temperature, °C	Substance	Ignition temperature, °C	Substance	Ignition temperature, °C	Substance	Ignition temperature, °C	Substance	Ignition temperature, °C
IIA									
acetone	540	i-amyl acetate	380	petrol		acetaldehyde	140		
ethane	515	n-butane	385	gasoline	1)				
ethylacetate	460	n-butyl alcohol	340	special benzene	1)				
ethylchloride	510	cyclohexanon	430	diesel fuels	1)				
ammoniac	630	1,2-dichloroethane	440	fuel oil	1)				
benzol	555	acetic acid anhydride	330	n-hexane	240				
acetic acid	485								
carbon oxide	605								
methane	595								
methanol	455								
methylchloride	625								
naphthalene	520								
phenol	595								
propane	470								
toluol	535								
IIB									
Town gas (coal gas)	560	ethyl alcohol	425	hydrogen	270	ethylene ether	180		
		ethylene	425	sulphide					
		ethylene oxide	440						
IIC									
Hydrogen	560	acetylene	305					Carbon disulphide	95

1) The ignition temperature depends on the composition and is between 220 and 300 °C, in special cases over 300 °C.

Table 9.8. Temperature classes of dusts



Temperature classes						
T2			T3		T4	
Substance	Ignition temperature, °C	Glowing temperature, °C	Substance	Ignition temperature, °C	Glowing temperature, °C	Substance
woods fibers			cotton	560	350	
			cellulose	500	370	
			sawdust, wooden chips	410	310	
			cork	470	300	
			paper	540	300	
			peat	360	295	
foodstuff animal feed	cocoa	580	cereals	370	280	
	wheat flour	480	concentrated feedstuff	520	295	
			milk powder	440	340	
			starch	350	290	
			tobacco	450	300	
			tea	510	300	
			dust of sugar beet chips	460	290	
coals and other natural products			brown coal	380	225	
			coal	550	245	
technical chemical products	polyvinylchloride mixture	530	lamine	510	330	india rubber
metals	magnesium	610	aluminum	530	280	bronze
	zinc	570	iron	310	300	
			manganese	330	285	

9.7

Standardized symbols for electrical equipment in explosion-prone areas acc. to EN 50014 to EN 50020

Electrical equipment must dispose of permanent and well visible markings according to the certified data. In table 9-9, the symbols according to the European Standards are listed.

Table 9-9. Symbols for explosion protection and firedamp protection [9.2]

EN 50014 and VDE 0170/0171, part 1/5.78				
	General symbol	Ignition protection class	Group	Temperature class
Firedamp protection	EEx I			
Explosion protection	EEx II			
Pressurized packaging		d		
Oil immersion		o		
Sand filling		q		
Flameproof enclosure		p		
Increased safety		e		
Intrinsic safety		i		
Electrical equipment for fiery mines			I	
Electrical equipment for “e”			II	
for explosion-prone areas except for fiery mines	for “d”		IIA	
			IIB	
			IIC	
Maximum surface temperature of the equipment in °C	450			T1
	300			T2
	200			T3
	135			T4
	100			T5
	85			T6
Common symbol	For group I:		For group II:	
				

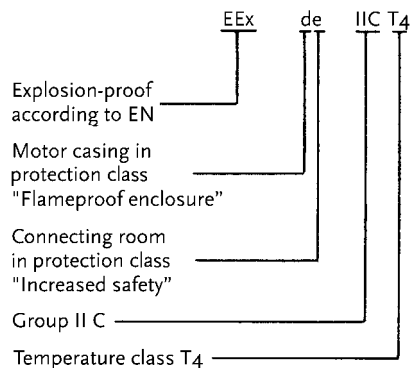
9.8

Examples of explosion protection symbols [9.2]

The identification system may only be used for electrical equipment that corresponds to the European Standards from EN 50014 to EN 50020.

Example 9.1

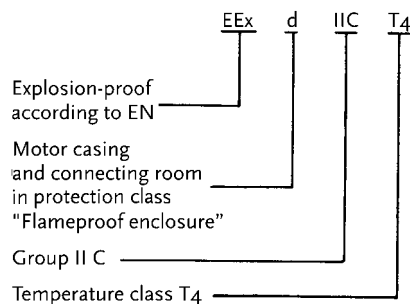
Three-phase A.C. motor of ignition protection class “flameproof enclosure” for group II C, temperature class T4.



Note: If more than one ignition protection class is used for electrical equipment, the symbol of the main ignition protection class is to be indicated first, then the symbols of the other ignition protection classes used have to follow. In example 9.1, the ignition protection class “e” of the terminal box is stated after the main ignition protection class.

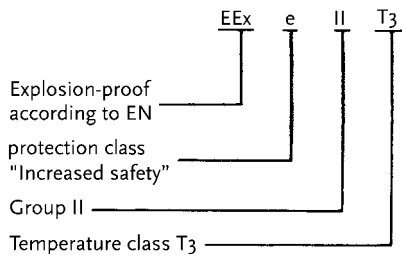
Example 9.2

Three-phase A.C. motor as in example 9.1, however with the terminal box in ignition protection class “flameproof enclosure”.

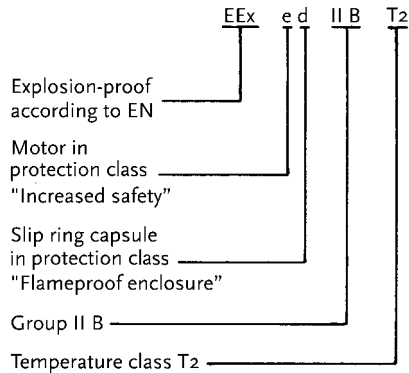


Example 9.3

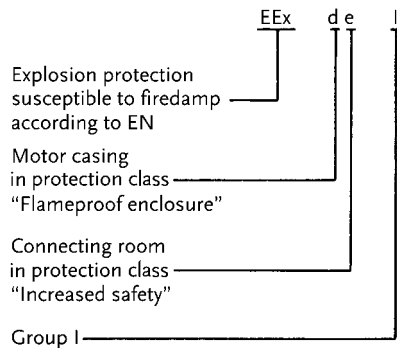
Three-phase A.C. motor in ignition protection class “increased safety” for group II, temperature class T3.

**Example 9.4**

Three-phase A.C. motor with slip ring motor in ignition protection class “increased safety” for group II, temperature class T2; slip ring bush in ignition protection class “flameproof enclosure”.

**Example 9.5**

Firedamp-proof three-phase A.C. motor in ignition protection class “flameproof enclosure” for group I.





9.9

Comparison of symbols for explosion protection and firedamp protection according to the old and new standard

Table 9-10. Symbols for explosion protection and firedamp protection acc. to the old and new standard [9.2]

VDE 0170/0171/1.69										EN 50014 and VDE 0170/0171, part 1/5.78			
		General symbol	Protection class	Explosion class	Ignition group	General symbol	Ignition protection class	Group	Temperature class				
		(Sch)											
		(ex)											
Firedamp protection			d			EEx I	d						
Explosion protection			p			EEx II	–						
Flameproof enclosure			o				o						
Plate protection enclosure			–				q						
Oil immersion			f				–						
Sand filling			–				p						
External aeration			e				e						
new: Pressurized packaging			i				i						
Increased safety			s				–						
Intrinsic safety								I					
Special protection class													
Electrical equipment for fiery mines				–									
Electrical equipment for explosion-prone areas except for fiery mines		For “e” For “d”		1 2 3n				II IIA IIB IIC					
Permissible temperature of equipment in °C		360			G1				–				
		240			G2				–				
at 40 °C ambient temperature		160			G3				–				
		110			G4				–				
		80			G5				–				

Table 9.10 Continued

VDE 0170/0171/1.69					EN 50014 and VDE 0170/0171, part 1/5.78			
	General symbol	Protection class	Explosion class	Ignition group	General symbol	Ignition protection class	Group	Temperature class
Maximum surface temperature of equipment in °C		450		–				T1
		300		–				T2
at 40 °C ambient temperature		200		–				T3
		135		–				T4
		100		–				T5
		85		–				T6
Examples of symbols								
Motor in ignition protection class “flameproof enclosure”	(Ex) d 3n G4				EEx de IIC T4 ^{*)}			
Motor in ignition protection class “increased safety”	(Ex) e G3				EEx e II T3			

^{a)} In this example:
d = Ignition protection class “flameproof enclosure” for the motor casing
e = Ignition protection class “increased safety” for the terminal box

9.10

Protection classes acc. to DIN IEC 34, part 5/VDE 0530, part 5 [9.4]

Depending on the operating and environmental conditions, the adequate selection of the suitable protection class prevents the detrimental effects of water, foreign bodies and dust as well as the contact of rotating components inside the motor or of components being live (comp. table 9-11). The protection classes of electrical machines are indicated in the form of a symbol consisting of two code letters and two code numbers, and possibly of an additional code letter.

IP	(International Protection) Code letter for the degrees of protection against contact and the penetration of foreign bodies and water.
0 to 6	1 st code number for degrees of protection against contact and against the penetration of foreign bodies.
0 to 8	2 nd code number for degrees of protection against the penetration of water (no oil protection).
W, S, and M	additional letters for special protection classes.

Table 9-11. Protection classes of electric motors

Motor	Protection degree	1 st code number protection against contact	Protection against foreign bodies	2 nd code number protection against water
Inside cooling	IP 21	contact with fingers	medium-size solid impurities with more than 12 mm Ø	vertical dripping water
	IP 22			dripping water up to 15° to the vertical line
	IP 23			spray water, up to 60° to the vertical line
Surface cooling	IP 44	contact with a tool or similar	small solid impurities with more than 1 mm Ø	spray water from all directions
	IP 54	complete protection against contact	damaging dust deposits	jet water from all directions
	IP 55			jet water from all directions
	IP 56			heavy sea ¹⁾ , strong jet
	IP 65	complete protection against contact	protection against penetration of dust	jet water from all directions
	IP 67			motor under water at determined pressure and time conditions

1) In heavy seas (temporary flooding), only the non-aired version 1PB is possible.

Special protection classes

W for weather-protected machines. The additional letter W is placed between the code letter IP and the protection class code numbers, e. g. IP W 23.

It is applicable for machines for “the application under determined weather conditions and with additional protection measures or facilities”.

S and M, for protection against water

For special applications (such as open, draft-ventilated machines on deck of a ship, with the orifices for the inlet and outlet of the air being closed during the standstill), the code number may be followed by a letter indicating whether the protection against the damaging penetration of water during a standstill of the machine (letter S) or in operation (letter M) has been verified and controlled. In this case, the protection class for both operating modes of the machine has to be indicated, e. g. IP 55 S/ IP23 M.

With the additional letters missing, the protection class has been observed in both cases, i.e. with the machine running and at a standstill.

9.11**Motor power rating [9.5]**

The power rating is applicable for continuous operation acc. to VDE 0530, part 1 at a frequency of 50 Hz, at a coolant temperature (CT) 40 °C and an altitude of installation of up to 1000 m above mean sea level.

In the case of deviant conditions, the permissible power has to be determined according to the tables 9-12 and 9-13. These tables are unreservedly applicable for normal three-phase A.C. asynchronous motors and for motors in the ignition protection class “flameproof enclosure”. In the case of motors in the ignition protection class “increased safety” the power is not only influenced by the winding temperature of the insulation material class, but also by the t_E time.

In the case of contemporaneously deviating coolant temperatures and installation altitudes, the factors for the permissible power must be multiplied. Coolant temperature and installation altitude are rounded up to 5 °C and 500 m.

Table 9-12. Motor power rating depending on coolant temperature (CT)

CT, [°C]	Permissible power in % of power rating
10	116
15	113
20	111
25	108
30	106
35	103
40	100
45	96
50	92
55	87
60	82

Table 9-13. Influence of the installation altitude on the motor power rating

Installation altitude above mean sea level in m	Permissible power in % of power rating
1000	100
1500	97
2000	94
2500	90
3000	86
3500	82
4000	77

9.12

Three-phase A.C. motors in VIK-design [9.2]

For the chemical industry and power stations, electric motors have always been developed and manufactured according to additional specifications adapted to the particular operating conditions.

In June 1975, the “Industrial Power Plant Association” (abbr. VIK, Vereinigung Industrielle Kraftwirtschaft, e. V., Essen) published the “Technical requirements for three-phase A.C. asynchronous motors”, in order to secure the standard design of motors and their quality standards for certain fields of application.

The VIK requirements to be agreed upon when placing an order are applicable for single-speed

- explosion-proof three-phase A.C. motors in the ignition protection class “flameproof enclosure”
- explosion-proof three-phase A.C. motors in the ignition protection class “increased safety”
- three-phase A.C. motors in a normal, non-explosion-proof version.

Among other things, the following items are specified in the technical requirements:

- conditions of installation
- operating mode
- power and voltage
- mechanical protection class
- coating, corrosion protection
- materials
- terminal box with terminals and inlet parts
- storage and shelf-life
- lifetime lubrication, grease life
- re-lubricating devices, re-lubrication period, storing greases
- stator winding, insulating material class
- starting features, torques, pick-up current, pick-up apparent power, number of starts
- rotation direction
- noise level
- running quality and balancing
- explosion protection, temperature class, group, warming-up time t_E
- performance and test plate.

9.13

"ATEX 100a" according to EU-Directive 94/9/EG Application for liquid ring vacuum pumps

In the Cenelec-member states the regulations according to ATEX 100a, applicable for the purposeful application of new devices and protection systems in explosion-prone areas, are in force since July 1, 2003. All the equipment used before this date can continue to be used according to the previous approval.

The directive 97/53EG dated 09/11/1997 is based on article 118a of the directive 94/9/EG. Article 118a subdivides areas into zones and determines which devices and protection systems can be used in the zones. The original subdivision into zones as per DIN VDE 0165, Part 101, has been completely taken over and is now valid according to IEC 79-10. A further overview of explosion-prone areas and the subdivision of zones are given in appendix I of 97/53/EG. The best-known combustible gases are listed in the appendix of EN 50014.

According to article ATEX 100a, devices are subdivided into equipment groups and categories. The subdivision into equipment groups and categories is shown in table 9-14.


Table 9-14. Equipment groups and categories

Equipment group I (mining)						
	Category M1 very high level of protection			Category M2 high level of protection		
Sufficient safety	by means of 2 protective measures / in case of 2 faults			has to be switched off in case of Ex-atmosphere		
Equipment group II (other explosion-prone atmospheres)						
	Category 1* very high level of protection		Category 2* high level of protection		Category 3* normal level of protection	
Sufficient safety	by means of 2 protective measures / in case of 2 faults		in case of frequently occurring equipment faults / in case of 1 fault		during trouble-free operation	
Application in	zone 0	zone 20	zone 1	zone 21	zone 2	zone 22
atmosphere:	G	D	G	D	G	D
G=gas, D=dust						

*) II (1) G associated electrical apparatus – installation in the safe area

The CE-sign and declaration of conformity of the manufacturer are conditions for the free trade in Europe.

The marking according to equipment group and category for electrical equipment as per ATEX 100a can be classified as follows:

 II 2 G EEx ib IIC T4.

According to the EU-Directive, explosion protection is also required for non-electrical components of aggregates operated in the ex-range.

Fig. 9-1 shows a liquid ring vacuum pump with permanent magnetic coupling and Fig. 9-2 shows a liquid ring vacuum pump with canned motor with an example for ex-protection marking.

The monitoring systems necessary according to the safety and health requirements of the EG-prototype approval certificate have to be installed even on the pump component, in order to trigger off the emergency cut-out in case of malfunction or to arrange for adequate safety measures in good time.

Depending on the existing categories and zones in the explosion-prone areas, on the pump the monitoring of the liquid flow, the liquid level, the temperature and possibly of the gas suction and compression pressure are to be checked and guaranteed on the pump.

The liquid ring machines according to fig. 9-1 and fig. 9-2 are applicable in the explosion-prone areas according to the categories 3 and 2 in zone 2 and zone 1 respectively. For the liquid ring vacuum pumps with permanent magnetic coupling

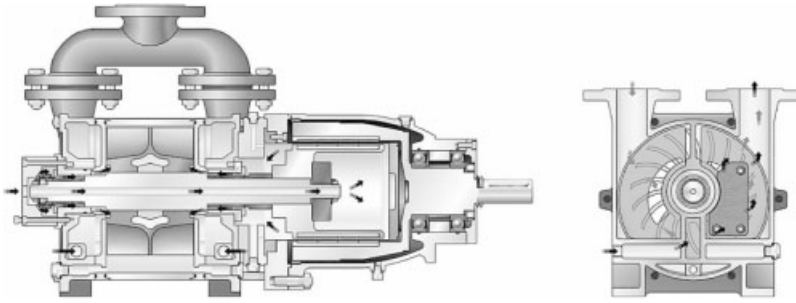


Figure 9-1. Liquid ring vacuum pump with permanent coupling
Explosion protection according to EC-prototype approval certificate as per EU-Directive:
94/9/EG ATEX 100a Ex II 2 G EEx c T5-T6 (company drawing acc. to Lederle-Hermetic).

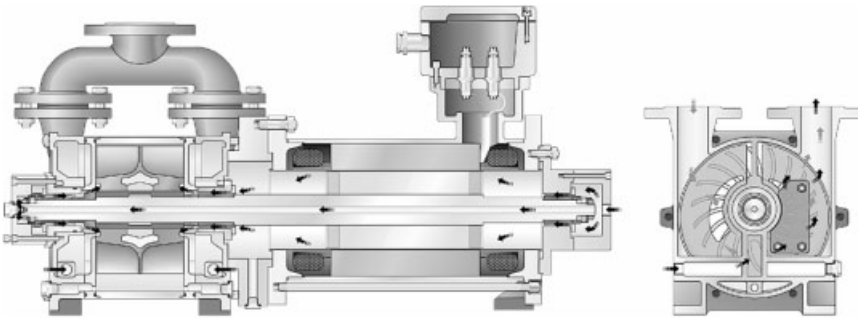


Figure 9-2. Liquid ring vacuum pump with canned motor
Explosion protection according to EC-prototype approval certificate as per EU-Directive:
94/9/EG ATEX 100a Ex II 2 G EEx de IIC T5-T6/EEx c T5-T6 (company drawing acc. to Lederle-Hermetic).

(fig. 9-1), additional test certificates for the application according to category 1 for zone 0 exist. These machines can be used in cases in which the conveyed gas-vapor mixtures in pump workrooms require explosion protection acc. to category 1.

9.14

Regulations outside of CENELEC member states [9.2]

Explosion-proof three-phase A.C. motors in the ignition protection class “flameproof enclosure” can be adapted to the regulations of countries that do not belong to the CENELEC members. For non-European countries belonging to the English-speaking area, it is usually required to supply the flameproof-enclosed motors with a terminal box in ignition protection class “flameproof enclosure”. The ignition protection class “increased safety” is not recognized in all of these countries. An example is shown in the table 9-15.

Table 9-15. Three-phase A.C. motors for explosion-prone areas acc. to the American National Electrical Code (NEC 500)

Flammable substances	Symbols and ranges	Permissible three-phase A.C. motors
gases vapors and fogs	class 1 division 1 (corresponds to zones 0 and 1 acc. to VDE 0165/9.83)	Explosion-proof three-phase A.C. motors in ignition protection class “d” as a minimum. EEx d IIB T. For acetylene and hydrogen: EEx d IIC T4
	class 2 division 2 (similarly to zone 2 acc. to VDE 0165/9.83)	Three-phase A.C. motors registered for divi- sion 1. Moreover, surface-cooled three-phase A.C. motors with squirrel cages (not explosion- proof), if they do not generate igniting sparks and if their surface temperature does not exceed the range of the required temperature class. With non-sparking ventilation.

9.15**Electric motors for explosion-prone areas acc. to the American NEC-Rules [9.6]**

In the USA, the safety of electrical plants is handled acc. to the NEC-Rules (NEC = National Electric Code).

Explosion-prone operating areas are defined as “Hazardous Locations”. These areas are subdivided into three classes, namely into class I, II and III. Each class is again subdivided into two danger zones, i.e. division 1 and division 2. The ignition temperature of gases and the glowing temperature of dusts determine the classification under the groups A to G that correspond more or less to the EN-temperature classes.

9.15.1**Classes and hazardous locations**

Class I – Locations acc. to NEC-article 500-5 are areas in which flammable gases or vapors occur or may occur in quantities sufficient to develop explosive or flammable mixtures.

Class I, Division 1 Locations are areas in which inflammable concentrations of flammable gases or vapors

- may exist under normal operating conditions;
- often occur on the occasion of repair and maintenance works or in case of leakages;

- escape in the case of a breakdown or malfunction of devices or in the case of failures in the process flow contemporaneously causing a breakdown of electrical equipment.

Class I, Division 2 Locations are areas,

- in which volatile flammable liquids or vapors are handled, processed or used that are normally kept in closed containers or pipework, where they escape from only in the event of damage or in case of improper functioning.
- in which accumulations of explosive concentrations of gases or vapors are usually prevented by means of induced ventilation, the breakdown of which, however, may be a risk, or which dispose of overpressure ventilation by means of fresh air effectively secured against breakdown, bordering on Class I Division 1 locations, from which hazardous gas or vapor mixtures can only pass if inside overpressure does not exist.

Class II Locations acc. to NEC 500-6 are areas in which combustible dust creates a source of danger.

Class II, Division 1 Locations are areas, in which

- under normal operating conditions, combustible dust exists in the air in quantities that form explosive or inflammable mixtures.
- explosive and inflammable dust-air-mixtures arise from mechanical breakdown or abnormal operating behavior of machines and devices and that might be ignited owing to the contemporaneous breakdown of electrical equipment, safety devices or for other reasons.
- combustible, electrically conducting dust in dangerous quantities exists.

Class II, Division 2 – Locations are areas with a quantity of combustible dust in the air usually insufficient to create explosive or inflammable mixtures and where accumulations of dust are not able to disturb normally operating devices; but where combustible dust resulting from a rare disturbance of the process flow may float in the air and where accumulations of combustible dusts on, in or near electrical equipment are sufficient to impair their heat discharge or that might get explosive owing to a failure of the equipment.

Class III – Locations acc. to NEC 500-7 are areas containing highly-inflammable fibers or fluffs usually floating in the air in such small quantities that an explosive mixture cannot develop.

Class III, Division 1 Locations are areas in which highly-inflammable fibers or materials arise through combustible fluffs, where they are being processed, manufactured, used or handled.

Class III, Division 2 Locations are areas in which highly inflammable fibers are stored or handled. Exception: Plants for treatment and processing.

9.15.2

Group classification

The classification into groups corresponds more or less to the classification into explosion classes of IEC and EN, as the comparison in tables 9-16 and 9-17 shows.

Table 9-16. Classification of substances for Class I

Acc. to NEC	Acc. to EN
<i>Group A</i> acetylene	II C
<i>Group B</i> hydrogen industrially generated gases, e.g. town gas	II C II B
<i>Group C</i> ethyl-ether ethylene cydopropane	II A II B II A
<i>Group D</i> alcohol acetone petrol benzol butane hexane natural gas propane	II A II A II A II A II A II A II A II A

Table 9-17. Classification of substances for Class II

Group E	metal dust
Group F	coal dust
Group G	flour, starch or cereal dust

9.15.3

Temperatures for Class I and Class II in “hazardous locations”**Temperatures for Class I**

Up to now, gases and vapors of similar kind and similar explosion features have been classified under one group. The substance in a group disposing of the lowest ignition temperature is decisive for the limiting temperature of a group (comp. table 9-18).

Table 9-18. Allocation of the limiting temperature acc. to the lowest ignition temperature of a substance in a group

Class I Group	Limiting temperature of the group in °C	Temperature class acc. to EN
A	280	T3
B	120	T3
C	180	T4
D	280	T3

Lately, motors are to indicate details concerning the highest operating temperature on the warmest spot instead of indicating the group. Similarly to those in EN 50014, they are classified according to temperature classes (see table 9-19).

Table 9-19. Determination of the limiting temperature acc. to the Operating Temperature Code

Class I, limiting temperature in °C	Operating temperature code	Temperature class acc. to EN 50014
450	T1	T1
300	T2	T2
280	T2A	–
260	T2B	–
230	T2C	–
215	T2D	–
200	T3	T3
180	T3A	–
165	T3B	–
160	T3C	–
135	T4	T4
120	T4A	–
100	T5	T5
85	T6	T6

Temperatures for Class II

The operating temperature or temperature class shown on the name plate shall be lower than the ignition temperature of the specific dust and by no means higher than the limiting temperature of the group (comp. table 9-20).

Table 9-20. Allocation of the limiting temperature acc. to the group at different operating conditions

Class II Group	Limiting temperature in °C	
	at rated power	at abnormal operating conditions
E	200	200
F	150	200
G	120	165

9.15.4

Application of motors according to American regulations

Fig. 9-3 gives an overview of the application of motors acc. to European Standards “EN” in explosion-prone areas as per American regulations.

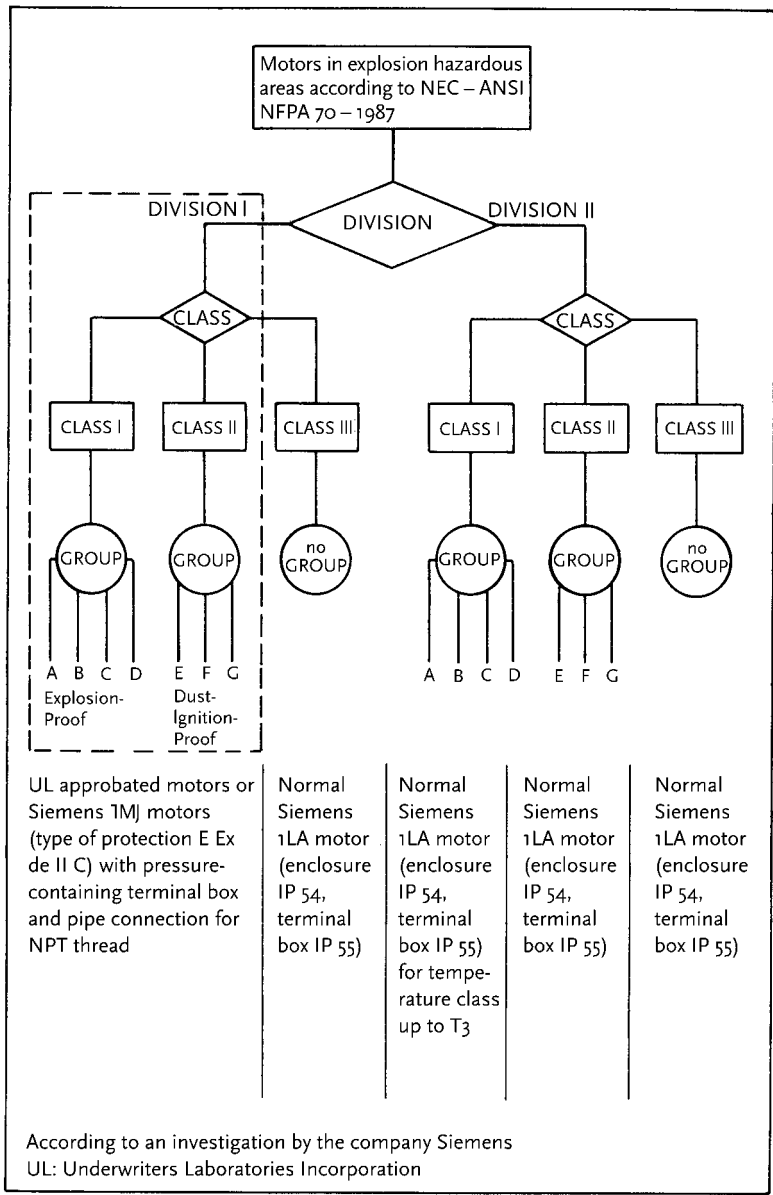


Figure 9-3. Hazardous zones and classes of explosion-prone areas acc. to NEC

9.15.5

Identification of motors

According to NEC 500-3, approved motors are to be marked with the following data:

- class
- group
- operating temperature or temperature class at CT 40 °C.

9.15.6

Protection classes acc. to NEMA in comparison to IEC [9.7]

Tables 9-21 and 9-22 show the most common protection classes acc. to NEMA as well as the fulfillment criteria acc. to IEC.

Table 9-21. Protection class for open machines

Protection class acc. to NEMA, MG 1-1.25	Protection degree acc. to IEC (or VDE, DIN, EN)
drip-proof machine	IP 12
splash-proof machine (water protection 100° against vertical axis)	IP 23
semi-guarded machine	not planned for standard motors
guarded machine	IP 22
drip-proof guarded machine	IP 22
open externally ventilated machine	not planned for standard motors
open type ventilated machine	IP 23 or IP 44
weather-protected, type I	IPW 23
weather-protected, type II	IPW 24

Table 9-22. Protection class for enclosed machines

Protection class acc. to NEMA, MG 1-1.26	Protection class acc. to IEC (or VDE, DIN, EN)
totally enclosed, non-ventilated machine	IP 44 or higher, without fan (self-cooling)
totally enclosed, fan-cooled machine	IP 44 or higher (intrinsic cooling)
explosion-proof machine	E Ex d II A, B or C (flameproof enclosure) IP 44 or higher
dust-ignition-proof machine	IP 44 or higher, terminal box in IP 54 or higher (for fulfillment of NEMA requirements, non-sparking fan is required)
water-proof machine	IP 46 or higher

9.16

Internationally common power supply systems

Table 9-23 gives an informative compilation of the usually existing systems of public power supply in different countries. In the industrial range of application, higher voltages deviating from those given in the table are frequently used.

Table 9-23. Frequencies and voltages of power supply systems in different countries

Country	Frequency (Hz)	Voltages (V)
<i>European countries</i>		
Albania	50	220/380
Austria	50	220/380
Belarus	50	220/380-660
Belgium	50	220/380-127/220-220
Bulgaria	50	220/380
CIS	50	220/380-660
Czech Republic	50	220/380
Denmark	50	220/380
Finland	50	220/380-500 – 660
France	50	127/220-220/380-500 -380/660-525/910
Germany	50	220/380-500
Greece	50	220/380-127/220
Great Britain	50	240/415
Hungary	50	220/380
Iceland	50	127/220-220/380
Ireland	50	220/380
Italy	50	127/220-220/380
Luxembourg	50	220/380
Netherlands	50	220/380-660
Northern Ireland	50	230/400 – Belfast 220/380
Norway	50	230-220/380-660
Poland	50	220/380
Portugal	50	220/380
Romania	50	220/380
Slovakia	50	220/380
Spain	50	220/380
Sweden	50	127/220-220 – 220/380
Switzerland	50	220/380-500 – 230/400
Ukraine	50	220/380-660
Yugoslavia	50	220/380
<i>Non-European countries</i>		
Afghanistan	50	220/380
Algeria	50	127/220-220/380
Angola	50	220/380
Argentina	50	220/380

Table 9-23. Continued

Country	Frequency (Hz)	Voltages (V)
Australia	50	240/415 – Western Australia 254/440
Bahamas	60	115/200-120/208
Bahrain	50	230/400
Bangladesh	50	230/400
Barbados	50	110/190-120/208
Belize	60	110/220-220/440
Benin	50	220/380
Bermudas	60	115/230
Bolivia	50	110/220-220/380 (exception)
	60	220/380-480
Brazil	60	110/220-220/440-127/220 – 220/380
Burma	50	230/400
Burundi	50	220/380
Cambodia	50	120/208 – Phnom Penh: 220/380
Cameroon	50	127/220-220/380
Canada	60	120/240-460 – 600
Chile	50	220/380
Colombia	60	110/220-150/260-440
Congo	50	220/380
Costa Rica	60	120/208-120/240-127/220 – 254/440-277/480
Cuba	60	120/240-220/380-277/480 – 440
Cyprus	50	240/415
Dominican Republic	60	120/208-120/240 -480
Ecuador	60	120/208-127/220
Egypt	50	110/220-220/380
El Salvador	60	110/220-120/208-127/220 – 220/440-240/480- 254/440
Ethiopia	50	220/380
Gabon	50	220/380
Gambia	50	230/400
Ghana	50	127/220-220/380
Guatemala	60	120/208-120/240-127/220 – 277/480-480 – 550
Guinea	50	220/380
Guyana	50	110/220 Georgetown
	60	110/220-240/480
Haiti	50	220/380 Jacmel
	60	110/220
Honduras	60	110/220-127/220-277/480
Hong Kong	50	200/346
India	50	220/380-230/400-240/415
Indonesia	50	127/220-220/380-400
Iraq	50	220/380
Iran	50	220/380
Israel	50	230/400
Ivory Coast	50	220/380

Table 9-23. Continued

Country	Frequency (Hz)	Voltages (V)
Jamaica	50	110/220-440
Japan	50	100/200-400 Hokkaido, Northern Honshu
	60	110/220-440 Southern Honshu, Shikoku, Kyushu
Jordan	50	220/380
Kenya	50	240/415
Korea (North)	60	220/380
Korea (South)	60	100/200-220/380-440
Kuwait	50	240/415
Laos	50	220
Lebanon	50	110/190-220/380
Liberia	60	120/208-120/240
Libya	50	127/220-220/380
Madagascar	50	127/220-220/380
Malawi	50	220/380
Malaysia	50	240/415
Mali	50	220/380
Mauritania	50	220
Mauritius	50	240/415
Mexico	60	127/220-440
Mongolia	50	220/380
Morocco	50	115/200-127/220-220/380 – 500
Mozambique	50	220/380
Namibia	50	220/380
Nepal	50	230/400-220/440
New Zealand	50	230/400
Nicaragua	60	110/220-120/240-127/220 – 220/440-254/440
Niger	50	220/380
Nigeria	50	240/415
Oman	50	220/380-240/415
Pakistan	50	230/400
Panama	60	120/208-120/240-254/440 – 277/480
Paraguay	50	220/380-220/440
People's Republic of China	50	127/220-220/380 – 1140 in mines
Peru	60	220-220/380-440
Philippines	60	110/220-440
Puerto Rico	60	120/208-480
Qatar	50	240/415
Rwanda	50	220/380
Saudi Arabia	50	220/380-240/415 (occasional)
	60	127/220-220/380-480
Senegal	50	127/220-220/380
Sierra Leone	50	220/380
Singapore	50	240/415
Somalia	50	220-220/440
South Africa	50	220/380-500 – 550/950

Table 9-23. Continued

Country	Frequency (Hz)	Voltages (V)
Sri Lanka	50	230/400
Sudan	50	240/415
Surinam	60	115/230-127/220
Swaziland	50	220/380
Syria	50	115/200-220/380-400
Tahiti	50	127/220
Taiwan	60	110/220-220/440
Tanzania	50	230/400
Thailand	50	220/380
Togo	50	127/220-220/380
Trinidad	60	110/220-120/240-230/400
Tunisia	50	115/200-220/380
Turkey	50	220/380 – Parts of Istanbul: 110/190
Uganda	50	240/415
United Arab Emirates	50	220/380-240/415
Uruguay	50	220
USA	60	120/208-120/240-277/480 – 600
Venezuela	60	120/208-120/240-208/416 – 240/480
Vietnam	50	220/380
Yemen (North)	50	220/380
Yemen (South)	50	230/400
Zaire	50	220/380
Zambia	50	220/380-415 – 550
Zimbabwe	50	220/380

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10

Appendix

10.1

International system of units (SI) [10.1]

10.1.1

SI basic units

Table 10-1. SI basic units

Unit	SI base unit	
	Name	Unit symbol
length	meter	m
mass	kilogram	kg
time	second	s
electric current	Ampere	A
thermodynamic temperature	Kelvin	K
quantity of substance	mole	mol
luminous intensity	candela	cd

10.1.2

Derived SI units

Table 10-2. Examples for derived SI units expressed as basic units

Unit	SI base unit	
	Name	Unit symbol
area	square meter	m ²
volume	cubic meter	m ³
speed, velocity	meter per second	m/s
acceleration	meter per second square	m/s ²
wave number	reciprocal meter	m ⁻¹
mass density	kilogram per cubic meter	kg/m ³

Table 10-2. Continued

Unit	SI base unit	
	Name	Unit symbol
specific volume	cubic meter per kilogram	m^3/kg
current density	ampere per square meter	A/m^2
magnetic field density	ampere per meter	A/m
amount-of-substance concentration	mole per cubic meter	mol/m^3
luminance	candela by square meter	cd/m^2

Table 10-3. Derived SI units with particular names

Unit	Basic SI unit			
	Name	Unit symbol	Expressed by other SI units	Expressed by SI basic symbols
frequency	hertz	Hz		s^{-1}
force	newton	N		$\text{m} \cdot \text{kg} \cdot \text{s}^{-2}$
pressure, stress	pascal	Pa	N/m^2	$\text{m}^{-1} \cdot \text{kg} \cdot \text{s}^{-2}$
energy, work, quantity of heat	joule	J	$\text{N} \cdot \text{m}$	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-2}$
power, energy flow	watt	W	J/s	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-3}$
quantity of electricity	coulomb	C		$\text{s} \cdot \text{A}$
electric potential, electric voltage	volt	V	W/A	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-3} \cdot \text{A}^{-1}$
electric capacity	farad	F	C/V	$\text{m}^{-2} \cdot \text{kg}^{-1} \cdot \text{s}^4 \cdot \text{A}^2$
electric resistance	ohm	Ω	V/A	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-3} \cdot \text{A}^{-2}$
electric conductance	siemens	S	A/V	$\text{m}^{-2} \cdot \text{kg}^{-1} \cdot \text{s}^3 \cdot \text{A}^2$
magnetic flux	weber	Wb	$\text{V} \cdot \text{s}$	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-2} \cdot \text{A}^{-1}$
magnetic flux density	tesla	T	Wb/m^2	$\text{kg} \cdot \text{s}^{-2} \cdot \text{A}^{-1}$
inductance	henry	H	Wb/A	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-2} \cdot \text{A}^{-2}$
Celsius temperature	degree Celsius	$^{\circ}\text{C}$		K
luminous flux	lumen	lm		$\text{cd} \cdot \text{sr}$
luminous intensity	lux	lx	lm/m^2	$\text{m}^{-2} \cdot \text{cd} \cdot \text{sr}$

Table 10-4. Derived SI units with particular names permitted to protect human health

Unit	Name	Unit symbol	Expressed by other SI units	Expressed by SI basic units
activity (of a radionuclide)	becquerel	Bq		s^{-1}
Specific transferred energy	gray	Gy	J/kg	$\text{m}^2 \cdot \text{s}^{-2}$
dose equivalent	sievert	Sv	J/kg	$\text{m}^2 \cdot \text{s}^{-2}$

Table 10-5. Examples for derived SI units expressed with the help of particular names

Unit	SI unit		
	Name	Unified symbol	Expressed by SI base unit
dynamic viscosity	Pascal second	$\text{Pa} \cdot \text{s}$	$\text{m}^{-1} \cdot \text{kg} \cdot \text{s}^{-1}$
moment of force	Newton meter	$\text{N} \cdot \text{m}$	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-2}$
surface tension	Newton per meter	N/m	$\text{kg} \cdot \text{s}^{-2}$
heating flow density	watt per square meter	W/m^2	$\text{kg} \cdot \text{s}^{-3}$
heat capacity, entropy	joule per Kelvin	J/K	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-2} \cdot \text{K}^{-1}$
specific heat capacity, specific entropy	joule per kilogram- Kelvin	$\text{J}/(\text{kg} \cdot \text{K})$	$\text{m}^2 \cdot \text{s}^{-2} \cdot \text{K}^{-1}$
specific energy	joule per kilogram	J/kg	$\text{m}^2 \cdot \text{s}^{-2}$
thermal conductivity	watt per meter Kelvin	$\text{W}/(\text{m} \cdot \text{K})$	$\text{m} \cdot \text{kg} \cdot \text{s}^{-3} \cdot \text{K}^{-1}$
energy density	joule per cubic meter	J/m^3	$\text{m}^{-1} \cdot \text{kg} \cdot \text{s}^{-2}$
electric field strength	volt per meter	V/m	$\text{m} \cdot \text{kg} \cdot \text{s}^{-3} \cdot \text{A}^{-1}$
electric charge density	coulomb per cubic meter	C/m^3	$\text{m}^{-3} \cdot \text{s} \cdot \text{A}$
electric flux density	coulomb per square meter	C/m^2	$\text{m}^{-2} \cdot \text{s} \cdot \text{A}$
permittivity	farad per meter	F/m	$\text{m}^{-3} \cdot \text{kg}^{-1} \cdot \text{s}^4 \cdot \text{A}^2$
permeability	henry per meter	H/m	$\text{m} \cdot \text{kg} \cdot \text{s}^{-2} \cdot \text{A}^{-2}$
molar energy	joule per mole	J/mol	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-2} \cdot \text{mol}^{-1}$
molar entropy, molar heat capacity	joule per mole-Kelvin	$\text{J}/(\text{mol} \cdot \text{K})$	$\text{m}^2 \cdot \text{kg} \cdot \text{s}^{-2} \cdot \text{K}^{-1} \cdot \text{mol}^{-1}$
ion dose rate (x-rays and y-rays)	coulomb per kilogram	C/kg	$\text{kg}^{-1} \cdot \text{s} \cdot \text{A}$
Specific transferred energy per second	gray by second	Gy/s	$\text{m}^2 \cdot \text{s}^{-3}$

10.1.3

Additional SI units

This class comprises two units: the SI unit of the plane angle (radian) and the one of solid angle (steradian).

Table 10-6. Additional SI units

Unit	SI unit	
	Name	Unit symbol
plane angle	radian	rad
solid angle	steradian	sr

Table 10-7. Examples for derived SI units expressed with the help of additional units

Unit	SI unit	
	Name	Unit symbol
angular velocity	radian per second	rad/s
angular acceleration	radian per second square	rad/s ²
radiant intensity	watt per steradian	W/sr
radiant density	watt per square meter steradian	W · m ⁻² · sr ⁻¹

10.1.4

Decimal multiples and parts of SI units**Table 10-8.** SI prefixes

Factor	Name	Prefix symbol	Factor	Name	Prefix symbol
10 ¹⁸	exa	E	10 ⁻¹	deci	d
10 ¹⁵	peta	P	10 ⁻²	centi	c
10 ¹²	tera	T	10 ⁻³	milli	m
10 ⁹	giga	G	10 ⁻⁶	micro	μ
10 ⁶	mega	M	10 ⁻⁹	nano	n
10 ³	kilo	k	10 ⁻¹²	pico	p
10 ²	hecto	h	10 ⁻¹⁵	femto	f
10 ¹	deka	da	10 ⁻¹⁸	atto	a

10.1.5

Units outside the International System of Units**Table 10-9.** Units used together with the International System of Units

Name	Unit symbol	Conversion factor
minute (time)	min	1 min = 60 s
hour	h	1 h = 60 min = 3600 s
day	d	1 d = 24 h = 86400 s
degree (angle)	°	1° = (π/180)rad
minute (angle)	'	1' = (1/60)° = (π/10800)rad
second (angle)	"	1" = (1/60)' = (π/648000)rad
litre	l, L	1 l = 1 dm ³ = 10 ⁻³ m ³
metric ton	t	1 t = 10 ³ kg

Table 10-10. Units used together with the International System of Units with their relations to the SI units being determined by experiment

Name	Unit symbol	Definition
electron volt	eV	a)
unified atomic mass unit	u	b)

a) The electron volt is equal to the kinetic energy an electron gains passing through a potential difference of 1 V in vacuum:

$$1 \text{ eV} = 1.6021892 \cdot 10^{-19} \text{ J.}$$

b) The standardized atomic mass unit is equal to 1/12 of the mass of an atom of the nuclide ^{12}C :

$$1 \text{ u} = 1.6605655 \cdot 10^{-27} \text{ kg.}$$

Table 10-11. Units temporarily used apart from the International System of Units

Name	Unit symbol	Value in SI units
nautical mile		1 nautical mile = 1852 m
knot		1 nautical mile per second = (1852/3600) m/s
ångström	Å	1 Å = 0.1 nm = 10^{-10} m
are	a	1 a = 1 dam ² = 10 ² m ²
hectare	ha	1 ha = 1 hm ² = 10 ⁴ m ²
barn	b	1 b = 100 fm ² = 10 ⁻²⁸ m ²
bar	bar	1 bar = 0.1 MPa = 10 ⁵ Pa
Gal	Gal	1 Gal = 1 cm/s ² = 10 ⁻² m/s ²
curie	Ci	1 Ci = 3.7 · 10 ¹⁰ Bq
roentgen	R	1 R = 2.58 · 10 ⁻⁴ C/kg
rad	rad	1 rad = 1 cGy = 10 ⁻² Gy
rem	rem	1 rem = 1 cSv = 10 ⁻² Sv

Table 10-12. CGS units with particular names

Name	Unit symbol	SI equivalent
Erg	erg	1 erg = 10 ⁻⁷ J
dyne	dyn	1 dyn = 10 ⁻⁵ N
poise	P	1 P = 1 dyn · s/cm ² = 0.1 Pa · s
stokes	St	1 St = 1 cm ² /s = 10 ⁻⁴ m ² /s
gauss	Gs, G	1 Gs equals 10 ⁻⁴ T
oersted	Oe	1 Oe equals (1000/4π) A/m
maxwell	Mx	1 Mx equals 10 ⁻⁸ Wb
stilb	sb	1 sb = 1 cd/cm ² = 10 ⁴ cd/m ²
phot	ph	1 ph = 10 ⁴ lx

Table 10-13. Other units to be generally refused

Name	Relation to SI units
fermi	1 fermi= 1 fm= 10^{-15} m
metric karat	1 metric karat= 200 mg= $2 \cdot 10^{-4}$ kg
torr	1 torr= (101325/760) Pa
physical atmosphere (atm)	1 atm= 101325 Pa
kilopond (kp)	1 kp= 9.80665 N
calorie (cal)	
micron (μ)	1 μ = 1 μ m= 10^{-6} m
X-unit	1 X-unit= $1.002 \cdot 10^{-4}$ nm
stere (st)	1 st= 1 m ³
gamma (γ)	1 γ = 1 nT= 10^{-9} T
γ	1 γ = 1 μ g= 10^{-9} kg
λ	1 λ = 1 μ l= 10^{-6} l= 10^{-9} m ³

10.2

Units of measurement and their conversion

Table 10-14. Conversion of common units [10.2]

Force				
	N	dyn	kp	P
1 N	= 1	10^5	0.101972	101.972
1 dyn	= 10^{-5}	1	$0.101972 \cdot 10^{-5}$	$1.10972 \cdot 10^{-3}$
1 kp	= 9.80665	$9.80665 \cdot 10^5$	1	10^3
1 P	= $9.80665 \cdot 10^{-3}$	980.665	10^{-3}	1
Pressure, mechanical tension				
	Pa = N/m ²	bar	μ bar	kp/m ² = mmWS
1 Pa = 1 N/m ²	= 1	10^{-5}	10	0.1019716
1 bar= 10^5 Pa= 10^6 dyn/cm ²	= 10^5	1	10^6	$1.019716 \cdot 10^4$
1 μ bar= 1 dyn/cm ²	= 10^{-1}	10^{-6}	1	$1.019716 \cdot 10^{-2}$
1 kp/m ² = 1 mmWS	= 9.80665	$0.980665 \cdot 10^{-4}$	$0.980665 \cdot 10^2$	1
1 kp/mm ²	= $0.980665 \cdot 10^7$	98.0665	$0.980665 \cdot 10^8$	10^6
1 at= 1 kp/cm ²	= $0.980665 \cdot 10^5$	0.980665	$0.980665 \cdot 10^6$	10^4
1 atm= 760 Torr	= $1.01325 \cdot 10^5$	1.01325	$1.01325 \cdot 10^6$	$1.033227 \cdot 10^4$
1 Torr	= $1.333224 \cdot 10^2$	$1.333224 \cdot 10^{-3}$	$1.333224 \cdot 10^3$	13.59510

Table 10-14. Continued

	kp/mm²	at	atm	Torr
1 Pa = 1 N/m ²	= $1.019716 \cdot 10^{-7}$	$1.019716 \cdot 10^{-5}$	$0.986923 \cdot 10^{-5}$	$0.750062 \cdot 10^{-2}$
1 bar = 10^5 Pa = 10^6 dyn/cm ²	= $1.019716 \cdot 10^{-2}$	1.019716	0.986923	$0.750062 \cdot 10^3$
1 μ bar = 1 dyn/cm ²	= $1.019716 \cdot 10^{-8}$	$1.019716 \cdot 10^{-6}$	$0.986923 \cdot 10^{-6}$	$0.750062 \cdot 10^{-3}$
1 kp/m ² = 1 mmWS	= 10^{-6}	10^{-4}	$0.967841 \cdot 10^{-4}$	$0.735559 \cdot 10^{-1}$
1 kp/mm ²	= 1	10^2	96.7841	$0.735559 \cdot 10^5$
1 at = 1 kp/cm ²	= 10^{-2}	1	0.967841	$0.735559 \cdot 10^3$
1 atm = 760 Torr	= $1.033227 \cdot 10^{-2}$	1.033227	1	760
1 Torr	= $1.359510 \cdot 10^{-5}$	$1.359510 \cdot 10^{-3}$	$1.315789 \cdot 10^{-3}$	1

Energy, work, quantity of heat

	J	kWh	kcal	kpm
1 J = 1 Nm = 1 Ws	= 1	$2.77778 \cdot 10^{-7}$	$2.38844 \cdot 10^{-4}$	0.101972
1 kWh	= $3.6 \cdot 10^6$	1	$8.5984 \cdot 10^2$	$3.67089 \cdot 10^5$
1 kcal	= $4.1868 \cdot 10^3$	$1.16300 \cdot 10^{-3}$	1	$4.26935 \cdot 10^2$
1 kpm	= 9.80665	$2.72407 \cdot 10^{-6}$	$2.34228 \cdot 10^{-3}$	1
1 PSh	= $2.64779 \cdot 10^6$	0.735499	$6.32416 \cdot 10^2$	$2.70000 \cdot 10^5$
1 MeV	= $1.6021 \cdot 10^{-13}$	$0.44999 \cdot 10^{-19}$	$3.8263 \cdot 10^{-17}$	$1.63358 \cdot 10^{-14}$
1 erg	= 10^{-7}	$2.77778 \cdot 10^{-14}$	$2.38844 \cdot 10^{-11}$	$1.01972 \cdot 10^{-8}$

	PSh	MeV	erg
1 J = 1 Nm = 1 Ws	= $3.77673 \cdot 10^{-7}$	$0.62422 \cdot 10^{13}$	10^7
1 kWh	= 1.35962	$2.24719 \cdot 10^{19}$	$3.60000 \cdot 10^{13}$
1 kcal	= $1.58124 \cdot 10^{-3}$	$2.6135 \cdot 10^{16}$	$4.1868 \cdot 10^{10}$
1 kpm	= $3.70370 \cdot 10^{-6}$	$6.1215 \cdot 10^{13}$	$9.80665 \cdot 10^7$
1 PSh	= 1	$1.6528 \cdot 10^{19}$	$2.64779 \cdot 10^{13}$
1 MeV	= $6.0503 \cdot 10^{-20}$	1	$1.6021 \cdot 10^{-6}$
1 erg	= $3.77673 \cdot 10^{-14}$	$0.62422 \cdot 10^6$	1

Power, energy flow, heat flow

	W	kW	PS	kpm/s
1 W	= 1	10^{-3}	$1.35962 \cdot 10^{-3}$	0.101972
1 kW	= 10^3	1	1.35962	101.972
1 PS	= 735.499	0.735499	1	75
1 kpm/s	= 9.80665	$9.80665 \cdot 10^{-3}$	$1.3333 \cdot 10^{-2}$	1
1 erg/s	= 10^{-7}	10^{-10}	$1.3596 \cdot 10^{-10}$	$1.0197 \cdot 10^{-8}$
1 cal/s	= 4.1868	$4.1868 \cdot 10^{-3}$	$5.6923 \cdot 10^{-3}$	0.42694
1 kcal/s	= $4.1868 \cdot 10^3$	4.1868	5.6923	$4.2694 \cdot 10^2$
1 kcal/h	= 1.1630	$1.1630 \cdot 10^{-3}$	$1.5812 \cdot 10^{-3}$	0.11859

Table 10-14. Continued

	erg/s	cal/s	kcal/s	kcal/h
1 W	$= 10^7$	0.238846	$2.38846 \cdot 10^{-4}$	0.859845
1 kW	$= 10^{10}$	$2.38846 \cdot 10^2$	0.238846	$8.59845 \cdot 10^2$
1 PS	$= 7.35499 \cdot 10^9$	175.67	0.17567	$6.3241 \cdot 10^2$
1 kpm/s	$= 9.80665 \cdot 10^7$	2.3423	$2.34228 \cdot 10^{-3}$	8.43228
1 erg/s	$= 1$	$2.38846 \cdot 10^{-8}$	$2.38846 \cdot 10^{-11}$	$0.859845 \cdot 10^{-7}$
1 cal/s	$= 4.1868 \cdot 10^7$	1	10^{-3}	3.6
1 kcal/s	$= 4.1868 \cdot 10^{10}$	10^3	1	$3.6 \cdot 10^3$
1 kcal/h	$= 1.1630 \cdot 10^7$	0.277	$0.277 \cdot 10^{-3}$	1

Frequently occurring units commonly used up to now

Physical unit	Unit usual up to now	Standard unit	Conversion factor
torque	kpm	Nm	9.80665
thermal conductivity	cal/cm · s · grad	W/(cm · K)	4.1868
	kcal/m · h · grd	W/(m · K)	1.163
specific heat	cal/g · grd	J/(kg · K)	$4.1868 \cdot 10^3$
		J/(g · K)	4.1868
thermal transmission coefficient	cal/cm ² · s · grd	W/(cm ² · K)	4.1868
	kcal/m ² · h · grd	W/(m ² · K)	1.163
dynamic viscosity	Poise (P)	Pas = N · s/m ²	10^{-1}
kinematic viscosity	stokes (St)	cm ² /s	1
		m ² /s	10^{-4}
magnetic flux	maxwell (M, Mx)	Wb	10^{-8}
induction flux		nWb	10
magnetic flux density,	gauss (G, Gs)	T	10^{-4}
induction		mT	10^{-1}
magnetic field strength	oersted (Oe)	A/m	$10^3/4 \cdot \pi$
		A/cm	$10/4 \cdot \pi$

Conversion example:

$$100 \text{ cal/g} \cdot \text{grd} = 4.1868 \cdot 100 \text{ J/(g} \cdot \text{K)} = 418.68 \text{ J/(g} \cdot \text{K)}$$

Table 10-15. Conversion of English and American units [10.2]

Units of length

1 inch ¹⁾	in	0.02540 m
1 foot ¹⁾ = 12 in	ft	0.30480 m
1 yard = 3 ft	yd	0.91440 m
1 mile (Statute) = 1760 yds		1.6093 km
1 nautical mile	sm	1.853 km
1 mil	10^{-3} inch	$25.4 \cdot 10^{-3}$ mm
1 micro-inch ¹⁾	10^{-6} inch	$25.4 \cdot 10^{-6}$ mm
1 fathom = 6 ft		1.8288 m

1) Acc. to DIN 4890, to be used for industrial purposes. There is a slight deviation between English and American units.

Table 10-15. Continued

Unit surface area

1 square inch ¹⁾	sq in	$6.4516 \cdot 10^{-4} \text{ m}^2$
1 circular inch (Brit.)	$\pi/4$ sq in	$5.067107 \cdot 10^{-4} \text{ m}^2$
1 square foot ¹⁾	sq ft	$9.29030 \cdot 10^{-2} \text{ m}^2$
1 square yard ¹⁾	sq yd	0.836127 m^2
1 acre		4046.85 m^2
1 mile of land = 640 acres		2.589979 km^2

1) determined by using 1 inch = 25.4 mm acc. to DIN 4890

Unit volume

1 cubic inch ¹⁾	cu in	$16.3871 \cdot 10^{-6} \text{ m}^3$
1 cubic foot ¹⁾	cu ft	$2.83168 \cdot 10^{-2} \text{ m}^3$
1 cubic yard ¹⁾	cu yd	0.764555 m^3
1 gallon (US) ⁴⁾ = 4 quarts = 8 pints	gal	$3.78543 \cdot 10^{-3} \text{ m}^3$
1 gallon (US) ³⁾	gal	$4.41 \cdot 10^{-3} \text{ m}^3$
1 gallon (GB) ⁴⁾ = 4 quarts = 8 pints	gal	$4.54609 \cdot 10^{-3} \text{ m}^3$
1 bushel (US) ³⁾	bu	$3.5239270 \cdot 10^{-2} \text{ m}^3$
1 bushel (GB) ³⁾ = 8 gallons	bu	$3.6368686 \cdot 10^{-2} \text{ m}^3$
1 barrel (US) ³⁾		0.1192414 m^3
1 barrel (petrol) (US) ⁴⁾		0.158762 m^3 or 0.1635 m^3
1 barrel (GB) ³⁾		0.163656 m^3
1 register ton ²⁾ = 100 cu ft		2.8317 m^3
1 pint (GB) ⁴⁾	Pt	$5.68229 \cdot 10^{-4} \text{ m}^3$
1 pint (US) ⁴⁾	pt	$4.7317 \cdot 10^{-4} \text{ m}^3$
1 quart (GB) ⁴⁾		$1.13646 \cdot 10^{-3} \text{ m}^3$
1 quart (US) ⁴⁾		$9.4633 \cdot 10^{-4} \text{ m}^3$

1) by using 1 inch = 25.4 mm acc. to DIN 4890

2) volume units

3) measures for dry substances

4) measures for liquids

Volume per unit of time

1 scfm =	$1.6957 \text{ m}^3/\text{h}$
1 standard cubic feet per minute (at 14.696 psia and 60 °F)	(at 1013.25 mbar and 15 °C)

Table 10-15. Continued

Mass units¹⁾

1 pound (Av.)	lb	0.45359237 kg
1 hundredweight (GB) = 112 lbs	cwt	50.80235 kg
1 (short) hundredweight (US) = 100 lbs	cwt	45.35924 kg
1 long (gross) ton (GB) = 2240 lbs	ltn	1016.05 kg
1 short (net) ton (US) = 2000 lbs	shtn	907.185 kg
1 ounce (Av.)	oz	0.0283495 kg
1 grain	gr	$0.06480 \cdot 10^{-3}$ kg
1 carat (metric)		$0.2 \cdot 10^{-3}$ kg
1 pennyweight	dwt	$1.5552 \cdot 10^{-3}$ kg

1) GB and US units "mass",
Av. = Avoirdupois (commercial weight).

Specific weight

1 lb. force/ft ³	16.01847 kp/m ³
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Force units

1 grain	grf	0.00063546 N
1 dram	dmf	0.017376 N
1 poundal	pdl	0.13825 N
1 ounce	ozf	0.27801 N
1 pound	lbf	4.4482 N
1 slug = 1 gee pound = 32.174 lb		143.12 N
1 short ton	shtn	8896.4 N
1 long ton	ltn	9964.0 N

Pressure, mechanical tension¹⁾

	N/m ²	bar	kg/m ²
1 lbf/sq in (psi) ¹⁾	$6.89476 \cdot 10^3$	$6.89476 \cdot 10^{-2}$	$7.0307 \cdot 10^2$
1 lbf/sq ft	$4.7876 \cdot 10$	$4.7876 \cdot 10^{-4}$	4.88243
1 lbf/sq yd	5.3201	$5.3201 \cdot 10^{-5}$	0.5425
1 long tonf/sq ft (GB)	$1.0725 \cdot 10^5$	1.0725	$1.09367 \cdot 10^4$
1 long tonf/sq in (GB)	$1.5444 \cdot 10^7$	$1.5444 \cdot 10^2$	$1.57488 \cdot 10^6$
1 short tonf/sq in (US)	$1.37894 \cdot 10^7$	$1.37894 \cdot 10^2$	$1.40613 \cdot 10^6$
1 inch of water	$2.4899 \cdot 10$	$2.4899 \cdot 10^{-4}$	$2.539 \cdot 10$
1 inch Hg	$3.8638 \cdot 10^3$	$3.8638 \cdot 10^{-2}$	$3.45315 \cdot 10^2$
1 grf/sq in	$0.98497 \cdot 10^2$	$0.98497 \cdot 10^{-3}$	$1.00439 \cdot 10$
1 ozf/sq yd	$3.313 \cdot 10^{-1}$	$3.313 \cdot 10^{-6}$	$3.38 \cdot 10^{-2}$
1 ozf/sq in=osi	$4.3092 \cdot 10^2$	$4.3092 \cdot 10^{-3}$	$4.3942 \cdot 10$

Table 10-15. Continued

	atm	torr	at
lbf/sq in (psi) ¹⁾	$6.8046 \cdot 10^{-2}$	$5.17151 \cdot 10$	$7.0306 \cdot 10^{-2}$
lbf/sq ft	$4.725 \cdot 10^{-4}$	$3.591 \cdot 10^{-1}$	$4.882 \cdot 10^{-4}$
lbf/sq yd	$5.251 \cdot 10^{-5}$	$3.991 \cdot 10^{-2}$	$5.425 \cdot 10^{-3}$
long tonf/sq ft (GB)	1.0585	$8.045 \cdot 10^2$	1.09367
long tonf/sq in (GB)	$1.524 \cdot 10^2$	$1.158 \cdot 10^6$	$1.57488 \cdot 10^2$
short tonf/sq in (US)	$1.3609 \cdot 10^2$	$1.0342 \cdot 10^5$	$1.40613 \cdot 10^2$
inch of water	$2.458 \cdot 10^{-3}$	1.867	$2.539 \cdot 10^{-3}$
inch Hg	$3.342 \cdot 10^{-2}$	$2.547 \cdot 10$	$3.45315 \cdot 10^{-2}$
grf/sq in	$9.7209 \cdot 10^{-3}$	7.387	$1.00439 \cdot 10^{-3}$
ozf/sq yd	$3.271 \cdot 10^{-6}$	$2.485 \cdot 10^{-3}$	$3.38 \cdot 10^{-6}$
ozf/sq in=osi	$4.2529 \cdot 10^{-3}$	3.2322	$4.394 \cdot 10^{-3}$

1) 1 ksi= 1000 psi

Energy, work and quantity of heat

Unit	Joule	kpm	kWh	kcal	PSH
1 ft lbf	1.35582	0.138255	$3.7650 \cdot 10^{-7}$	$3.2379 \cdot 10^{-4}$	$5.12056 \cdot 10^{-7}$
1 yd lbf	4.06746	0.41476	$1.12886 \cdot 10^{-6}$	$9.7147 \cdot 10^{-4}$	$1.53617 \cdot 10^{-6}$
1 ft tonf (US)	2716.4	277	$7.5456 \cdot 10^{-4}$	0.64878	$1.0259 \cdot 10^{-3}$
1 ft tonf (GB)	3037.1	309.7	$8.4365 \cdot 10^{-4}$	0.72538	$1.147 \cdot 10^{-3}$
1 HPh ¹⁾	$2.6845 \cdot 10^6$	$2.73745 \cdot 10^5$	$7.45476 \cdot 10^{-1}$	$6.41104 \cdot 10^2$	1.01387
1 Btu ²⁾	$1.05550 \cdot 10^3$	$1.07599 \cdot 10^2$	$2.93019 \cdot 10^{-4}$	$2.51996 \cdot 10^{-1}$	$3.98518 \cdot 10^{-4}$
1 Ctu ³⁾	$1.8991 \cdot 10^3$	$1.9367 \cdot 10^2$	$5.2743 \cdot 10^{-4}$	0.45359	$7.1733 \cdot 10^{-4}$

1) HPh= horsepower per hour

2) 1 British thermal unit= 0.55557 Ctu

3) 1 centigrade thermal unit= 1.800 Btu

Power

Unit	kpm/s	PS	kW	kcal/s	ft.lbf/s
1 ftlbf/s	0.138255	$1843.4 \cdot 10^{-6}$	$1355.4 \cdot 10^{-6}$	$23.93 \cdot 10^{-6}$	1
1 HP	76.0913	1.01455	0.74597	0.17828	550.369
1 Btu/s	107.590	1.43465	1.05486	0.25210	778.265
1 Ctu/s	193.678	2.58238	1.8987	0.45379	1400.877

Specific heat

1 Btu/(lb · °F)	1.00041cal/(g · degree)	4.1886 kJ/(kg · K)
1 Ctu/(lb · °C)	1.00041cal/(g · degree)	4.1886 kJ/(kg · K)

Table 10-15. Continued

Heat transmission coefficient

1 Btu/(sq ft · h · °F)	$1.356 \cdot 10^{-4} \text{ cal}/(\text{cm}^2 \cdot \text{s} \cdot \text{degree})$	5.681 W/(m ² · K)
1 Ctu/(sq ft · h · °C)	$1.356 \cdot 10^{-4} \text{ cal}/(\text{cm}^2 \cdot \text{s} \cdot \text{degree})$	5.681 W/(m ² · K)

Thermal conductivity

1 Btu/(ft · h · °F)	$4.134 \cdot 10^{-3} \text{ cal}/(\text{cm} \cdot \text{s} \cdot \text{degree})$	1.731 W/(m · K)
1 Ctu/(ft · h · °C)	$4.134 \cdot 10^{-3} \text{ cal}/(\text{cm} \cdot \text{s} \cdot \text{degree})$	1.731 W/(m · K)

10.3

Summary of physical and technical units

Table 10-16. Physical constants (selection)

speed of light in vacuum	c_0	$= 2.99792458 \cdot 10^8 \text{ m/s}$
normal gravitational acceleration	g_n	$= 9.80665 \text{ m/s}^2$
gravitation constant	γ	$= 6.672 \cdot 10^{-11} \text{ m}^3/(\text{kg} \cdot \text{s}^{-2})$
general (molar) gas constant	R_0	$= 8.3144 \cdot \text{mol}^{-1} \cdot \text{K}^{-1}$ $= 83.14 \text{ mbar} \cdot \text{l} \cdot \text{mol}^{-1} \cdot \text{K}^{-1}$ $= 8.314 \cdot 10^3 \text{ Pa} \cdot \text{m}^3 \cdot \text{kmol} \cdot \text{K}^{-1}$
normal temperature	T_n	$= 273.15 \text{ K}$
normal pressure	p_n	$= 101325 \text{ N} \cdot \text{m}^{-2}$
normal molar volume	V_{mn}	$= 22.41383 \text{ m}^3/\text{kmol}$
Avogadro constant	N_A	$= 6.022045 \cdot 10^{23}/\text{mol}$
Loschmidt constant	N_L	$= 2.686754 \cdot 10^{25}/\text{m}^3$
Boltzmann constant	k	$= 1.380662 \cdot 10^{-23} \text{ J/K}$
radiation constant	σ	$= 5.67032 \cdot 10^{-8} \text{ W}/(\text{m}^2 \cdot \text{K}^4)$
Magnetic field constant	ϵ_0	$= 8.85418782 \cdot 10^{-12} \text{ F/m}$
Magnetic field constant	μ_0	$= 1.25663706144 \cdot 10^{-6} \text{ H/m}$
Faraday constant	F	$= 9.648456 \cdot 10^4 \text{ C/mol}$
electronic elementary charge	e	$= 1.6021892 \cdot 10^{-19} \text{ C}$
Planck constant	h	$= 6.626176 \cdot 10^{-34} \text{ Js}$
electron rest mass	m_e	$= 9.109534 \cdot 10^{-31} \text{ kg}$ $= 5.4858026 \cdot 10^{-4} \text{ u}$
electron specific charge	e/m_e	$= 1.7588 \cdot 10^{11} \text{ C} \cdot \text{kg}^{-1}$
hydrogen atom mass	m_h	$= 1.6736 \cdot 10^{-27} \text{ kg}$
proton rest mass	m_p	$= 1.6726485 \cdot 10^{-27} \text{ kg}$ $= 1.007276470 \text{ u}$
neutron rest mass	m_n	$= 1.6749543 \cdot 10^{-27} \text{ kg}$ $= 1.008665012 \text{ u}$
Angular momentum quantum	h	$= 1.0545887 \cdot 10^{-34} \text{ Js}$
atomic mass unit	u	$= 1.6605655 \cdot 10^{-27} \text{ kg}$

Table 10-17. Important physical and technical units, their formula symbols¹⁾ and SI units

Unit	Formula symbol	SI unit
decay coefficient	δ	s^{-1}
coefficient of absorption	α	–
admittance	Y	S
radioactive substance activity	A	Bq
specific activity	a	Bq/kg
equivalent dose	\dot{D}_q, H	J/kg
equivalent capacity, equivalent rate	D_q	W/kg
work	W, A	J
relative atomic mass	A_r	–
propagation rate of a wave	c	m/s
specific radiation	M, M_e	W/m ²
Avogadro constant	N_A, L	Mol ⁻¹
luminous intensity	E, E_v	lx
light exposure	H, H_v	lx · s
acceleration	a	m/s ²
kinetic quantity	p, I	kg · m/s ²
bending moment	M_b	N · m
reactive power	Q, P_q	W
Reactive impedance	X	Ω
Boltzmann constant	k	J/K
power of a lens	n	–
focal distance	f	m
specific caloric value (upper thermal value)	H_0	J/kg
Celsius temperature	θ	°C
Compton wave length	λ_C	m
damping factor	θ	–
damping coefficient	α	m ⁻¹
time (period)	t	s
expansion	ε	–
density, mass density ($\rho = m/V$)	Q, ρ	kg · m ⁻³
dielectric constant	ε	F/m
diffusion coefficient	D	m ² /s
electric dipole moment	p, p_e	C · m
moment of momentum	L	kg · m/s ²
torque	M	N · m
rotation speed	n	s ⁻¹
pressure	p	Pa

Pressure can be identified more exactly by means of indices.

Recommended are

absolute pressure

p_{abs}

(In vacuum technology, the absolute pressure is always stated. Therefore, the 'abs' index can be omitted)

total pressure

p_t

Table 10-17. Continued

Unit	Formula symbol	SI unit
partial pressure of gas component i	p_i	
Instead of i, chemical sum formula can be used, e.g. p_{H_2} , p_{O_2}		
saturated vapour pressure	p_s	
vapour pressure	p_d	
normal pressure	p_n	
residual total pressure	p_r	
residual gas pressure	p_{rg}	
residual vapour pressure	p_{rd}	
ultimate pressure	p_{end}	
ambient pressure	p_{amb}	
overpressure	p_o	
linkage, electric	Θ	A
linkage, magnetic	Ψ_m	Wb
elastic modulus	E	N/m ²
amount of electricity	Q	C
elementary charge	e	C
emissivity	ε	—
energy	E, W	J
energy, specific, internal	u	J · kg ⁻¹
enthalpy	H	J
enthalpy, specific	h	J/kg
entropy	S	J/K
entropy, specific	s	J · kg ⁻¹ · K ⁻¹
gravitational acceleration	g	m/s ²
Faraday constant	F, g_f	C/mol
fine structure constant	α	—
electric field constant	ε_0	F/m
magnetic field constant	μ_0	H/m
field excitation, electric	E	V/m
field excitation, magnetic	H	A/m
area	A, S	m ²
surface density	σ	C/m ²
electric flux	Ψ , Ψ_e	C
magnetic flux	Φ	Wb
electric flux density	D	C/m ³
magnetic flux density	B	T
frequency	f, ν	Hz
general gas constant (molar)	R_0	kJ · kmol ⁻¹ · K ⁻¹
specific gas constant (special) for an individual gas	R	kJ · kmol ⁻¹ · K ⁻¹
gas flow rate	q	—
velocity	v, u	m/s
weight, force	G, F_G	N
gravitation constant	f, G	N · m ² /kg ²
surface tension	σ , γ	N/m

Table 10-17. Continued

Unit	Formula symbol	SI unit
half-life	$T_{1/2}$	s
net calorific value	H_u	J/kg
height	h	m
acoustic impedance	Z_a	$\text{Pa} \cdot \text{s}/\text{m}^3$
complex impedance	Z	Q
pulse	p, I	$\text{kg} \cdot \text{m}/\text{s}$
induction, magnetic	B	T
inductance	L	H
isentropic exponent	κ	–
electric capacity	C	F
atomic number	Z	–
compressibility	κ, χ	m^2/N
compression module	K	N/m^2
compression ratio	K, k	–
condensation coefficient	σ_k	–
force	F	N
angular frequency	ω	s^{-1}
electric charge	Q	C
charge coverage	σ	C/m^2
charge density	ρ, η	C/m^3
length	L	m
thermal coefficient of linear expansion	α, α_1	K^{-1}
power	P	W
property of conducting electricity	γ', σ, κ	S/m
susceptance, electric	G	S
susceptance, magnetic	\mathcal{A}	H
luminance indicatrix	L, L_v	cd/m^2
luminous efficiency	η	lm/W
light velocity (in empty space)	c_0	m/s
quantity of light	Q/Q_v	$\text{lm} \cdot \text{s}$
light intensity	I, I_v	cd
luminous flux	Φ, Φ_v	lm
magnetisation	M, H_i	A/m
mass	m	kg
m_a, m_M, m_e : mass of an atom, molecule, electron,	$m_u = 1.66 \cdot 10^{-27} \text{ kg}$	= atomic mass unit
mass per unit of time	\dot{m}	kg/s
surface-related absorbed mass	$\tilde{\mu}$	kg/m^2
molar mass	M_{molar}	kg/kmol
mass flow rate	q_m	$\text{kg} \cdot \text{s}^{-1}$
mass flow density	I	$\text{kg}/(\text{m}^2 \cdot \text{s})$
mass number	A	–
relative molecular mass, $M_r = m_M/m_u$	M_r	–
electromagnetic moment	m	$\text{A} \cdot \text{m}^2$
magnetic moment of one particle	μ	$\text{A} \cdot \text{m}^2$
reverberation time	T	s

Table 10-17. Continued

Unit	Formula symbol	SI unit
suction capacity rate	\dot{S}_n	m^3/s
surface	A, S	m^2
surface tension	σ, γ	N/m
ordinal number of an element	Z	—
cycle period	T	s
permeability	μ	H/m
permeance	\mathcal{A}	H
permeance conductivity	\bar{q}_{perm}	$\text{m}^2 \cdot \text{s}^{-1}$
phase angle	ϕ	rad
Planck quantum of action	h	$\text{J} \cdot \text{s}$
Poisson number	μ, ν	—
electric polarisation	P	C/m^2
magnetic polarisation	J, B_i	T
chemical potential	μ	J/mol
electric potential	ϕ	V
electric potential difference	U	V
Poynting vector (amount)	S	W/m^2
proton number	Z	—
pV flow rate	q_{pV}	$\text{N} \cdot \text{m} \cdot \text{s}^{-1} = \text{Pa} \cdot \text{m}^3 \cdot \text{s}^{-1}$
pV value	pV or $p \cdot V$	$\text{Pa} \cdot \text{m}^3$
cross-section, cross-section surface	S, q	m^2
radius	r	m
volume	V, τ	m^3
space charge density	ρ, η	C/m^3
three-dimensional angle	Ω, ω	sr
reactance	χ	Ω
reflection coefficient	X	—
coefficient of friction	μ, f	—
reluctance	R_m	H^{-1}
Rydberg constant	R_∞	m^{-1}
saturation ratio	β	—
suction power	\dot{Q}	$\text{Pa} \cdot \text{m}^3 \cdot \text{s}^{-1}$
suction capacity	\dot{S}	m^3/h
sound absorption coefficient	α	—
sound reduction	R	(dB)
sound pressure	p	Pa
sound pressure level	L_p	(dB)
sound energy density	w, E	J/m^3
sound energy flux	q	m^3/s
sound velocity	c	m/s
specific acoustic impedance	Z_s, Z	$\text{Pa} \cdot \text{s}/\text{m}$
sound intensity	J	W/m^2
sound power	P_a, P	W
acoustic power level	L_p, L_w	(dB)

Table 10-17. Continued

Unit	Formula symbol	SI unit
sound particle velocity	v	m/s
apparent power	S, P_s	W
admittance	$ Y $	S
impedance	$ Z $	Ω
shear module	G	N/m ²
shear stress	τ	N/m ²
oscillation period	T	s
electric voltage	U	V
magnetic voltage	V	A
Stefan-Boltzmann constant	σ	W/(m ² · K ⁴)
amount of substance	ν	kmol
amount of substance flow	q_ν	kmol · s ⁻¹
photometric radiation equivalent	K	lm/W
radiant energy	Q_e, W	J
radiant energy density	W, u	J/m ³
radiant flow	Φ_e, P	W
radiant flow density	φ, ψ	W/m ²
flow susceptance ($L = W''^{-1}$)	L	q/ Δp
flow resistance ($W = L''^{-1}$)	W	$\Delta p/q$
electric current density	J, S	A/m ²
electric current strength	I	A
electric susceptibility	χ_e, χ	—
magnetic susceptibility	χ_m, κ	—
particle number	N	—
particle number density	n	m ⁻³
surface-related absorbed number of particles	\tilde{n}	m ⁻²
number of particles per unit of time	\dot{N}	s ⁻¹
thermodynamic temperature,	T, Θ	K
temperature (in Celsius degrees)	t, ϑ	°C
temperature susceptance	a	m ² /s
inertial torque	J	kg · m ²
speed of light in vacuum	c	m/s
V evaporation temperature per 1 litre liquid	h_v	J/l
doubling of temperature = Sutherland constant	T_d	K
ratio of specific or molar heat capacities of gases	κ	—
$\kappa = C_{p, \text{spez.}} / C_{V, \text{spez.}} = C_{p, \text{molar}} / C_{V, \text{molar}}$		
electric displacement	D	C/m ³
dynamic viscosity	η	Pa · s
kinematic viscosity	ν	m ² /s
volume	V, τ	m ³
specific volume	ν	m ³ /kg
volume expansion coefficient	α_v, γ	K ⁻¹
volume flow rate	q_v	m ³ · s ⁻¹
heat transition coefficient (thermal transmittance)	k	J · m ⁻² · s ⁻¹ · K ⁻¹ = W · m ⁻² · K ⁻¹

Table 10-17. Continued

Unit	Formula symbol	SI unit
heat capacity	C	J/K
specific heat capacity	c	J/(kg · K)
–, at constant pressure	c_p	J/(kg · K)
–, at constant volume	c_v	J/(kg · K)
heat capacity ratio	κ	–
thermal conductivity	λ	W/(m · K)
quantity of heat	Q	J
quantity of heat per time unit	\dot{Q}	J/s
heat flow	Φ, \dot{Q}	W
heat flow density	q	W/m ²
thermal transmission coefficient	α	J · m ⁻² · s ⁻¹ · K ⁻¹ = W · m ⁻² · K ⁻¹
length of path	s	m
free length of path	l	m
main free path	\bar{l}	m
wave length	λ	m
intrinsic impedance	Z, Γ	Ω
wave number	σ	m ⁻¹
electric resistance	R	Ω
magnetic resistance	R_m	H ⁻¹
specific electric resistance	ϱ	$\Omega \cdot m$
plane angle	α, β, γ	rad
three dimensional angle	Ω, ω	sr
angular acceleration	α	rad/s ²
angular velocity	ω, Ω	rad/s
efficiency	η	–
time, period	t	s
decay constant	λ	s ⁻¹
tensile stress	σ	N/m ²

1) Acc. to the examples of DIN 1304, indices in formula symbols of physical and technical units are set straight; as far as they symbolize a unit or a letter substituting a number, they can be set in italics.

10.4**National and international standards, recommendations and regulations****Table 10-18.** Standards and guidelines in vacuum technology

National standards and norms

DIN	Title
1301	Units Part 1: Unit names, unit symbols Part 2: Submultiples and multiples for general use Part 3: Conversions of units no longer to be used
1302	General mathematical symbols and terms
1303	Vectors, matrices, tensors; symbols and terms
1304	General formula symbols
1305	Mass, force, weight load; terms
1306	Density; terms
1310	Composition of gaseous and liquid mixtures and mixed crystals; basic concepts
1311	Part 1 – Theory of vibrations, kinematic terms
1313	Physical quantities and equations; terms, spelling
1314	Pressure; basic concepts, units
1315	Angles; terms, units
1324	Electric field; terms
1325	Magnetic field; terms
1338	Letters, numbers and figures in mathematical compositions
1339	Units of magnetic quantities
1341	Heat transfer; basic terms, units, parameters
1342	Viscosity of Newtonian liquids
1343	Standard temperature and pressure
1345	Thermodynamics; formula symbols, units
1355, Part 1	Time; Calendar, numbering of weeks, date, time
1357	Units of electrical parameters
1945, Part 1	Displacement compressor; thermodynamic acceptance and performance test
1952	Flow rate measuring with standard nozzles, orifices and Venturi-nozzles (VDI rules of flow rate measurement)
2403	Identification of pipes according to the fluid conveyed
2429, Part 2	Graphical symbols for technical drawings; pipelines; functional depiction
5492	Formula symbols in fluid mechanics
5496	Thermal radiation
19227, Part 1	Graphical symbols and code letters for measuring, controlling, regulation in process engineering; symbols for functional depiction
24901, Part 4	Graphical symbols for technical drawings in mechanical engineering; compressors, fans; depiction in flow diagrams
28004, Part 1	Flow diagram of process engineering plants; terms, kinds of flow diagrams, information content
28004, Part 2	Flow diagrams of process engineering plants; graphical design
28004, Part 3	Flow diagrams of process engineering plants; graphical symbols
28004, Part 4	Flow sheets and diagrams of process plants; acronym

Table 10-18. Continued

DIN	Title
28400	Vacuum technology; terms and definitions Part 1 – General terms Part 2 – Vacuum pumps Part 3 – Vacuum measuring instruments Part 4 – Vacuum coating technology Part 5 – Vacuum drying and lyophilization Part 6 – Analysis techniques for surface coating Part 7 – Vacuum metallurgy Part 8 – Vacuum systems, components and equipment
28401	Vacuum technology; graphical symbols, overview
28402	Vacuum technology; quantities, symbols, units, overview
28403	Vacuum technology; quick release couplings; clamped type couplings
28404	Vacuum technology; flanges; dimensions
28410	Vacuum technology; Mass-spectrometric measuring instruments for partial pressure; terms, parameters, operating conditions
28411	Vacuum technology; acceptance specifications for mass-spectrometric leak-detectors, terms
28416	Vacuum technology; calibration of vacuum measuring instruments in the range of 10^{-3} to 10^{-7} mbar, general method: pressure reduction by continuous flow
28417	Vacuum technology; measurement of $p\dot{V}$ throughput acc. to the volumetric method at constant pressure
28418	Vacuum technology; standard method for vacuum gauge calibration by direct comparison with a reference gauge Part 1 – General basics Part 2 – Ionisation vacuum gauge Part 3 – Thermal conductivity gauges
28426	Vacuum technology; acceptance specifications for rotary piston vacuum pumps Part 1 – Rotary plunger vacuum pumps and sliding vane vacuum pumps for the ranges of rough and fine vacuum Part 2 – Roots vacuum pumps for the range of fine vacuum
28427	Vacuum technology; acceptance specifications for diffusion pumps and vapour jet vacuum pumps for motive fluid vapour pressures smaller than 1 mbar
28428	Vacuum technology; acceptance specifications for turbomolecular pumps
28429	Vacuum technology; acceptance specifications for getter ion pumps
28430	Vacuum technology; rules for the measurement of steam jet vacuum pumps and steam jet compressors; motive fluid: water vapor
28431	Acceptance specifications for liquid ring vacuum pumps
30600	Graphical symbols, registration, description
32830, Part 10	Graphical symbols; design rules for graphical symbols in technical product documentation; supplementary advice (currently in the planning stage)
45635, Part 13	Noise measurement on machines; airborne noise measuring, enveloping surface method, compressors incl. vacuum pumps (displacement-, turbo- and jet compressors)
66034	Kilopond – Newton, Newton – Kilopond; conversion tables
66036	Horsepower – Kilowatt, Kilowatt – Horsepower; conversion tables

Table 10-18. Continued

DIN	Title
66038	Torr – millibar, millibar – torr; conversion tables
66039	Kilocalorie – watt hour, watt hour – kilocalorie; conversion tables
VDI	Title
2045, sheet 1	Acceptance and performance tests for compressors (VDI Compressor rules); test procedure and guarantee comparison
VDMA	Title
4368	Liquid ring compressors Thermodynamic acceptance and performance tests
VDE	Title
0410	Rules for electric measuring instruments
International Standards and Norms	
ISO	Title
1000	SI-units and recommendations for the use of their multiples and of certain other units
2533	Standard atmosphere
PNEUROP	Title
5697	Vacuum Pumps, Rules of Acceptance, Part II (Fluid Entrainment Pumps)
5608	Vacuum Pumps, Rules of Acceptance, Part III (Turbomolecular Pumps)
5615	Vacuum Pumps, Rules of Acceptance, Part IV (Getter Ion Pumps)
6601	Application of National Standards for Acceptance and Power Measurement of Steam Jet Vacuum Pumps and Steam Jet Compressors
6602	Vacuum Pumps, Rules of Acceptance: Part I – (Oil-sealed rotary vacuum pumps – Roots Pumps)
6606	Vacuum flanges and junctions; dimensions
661200	Acceptance Specification and Performance Tests for Liquid Ring Vacuum Pumps
661700	Practical advice on vacuum measurement

Table 10-19. Meaning and explanation of frequently used abbreviations

Abbreviation	Meaning, explanation
ABS	American Bureau of Shipping. Ship classification association; country of origin: USA
AD	Working Association for Pressure Vessels
AISI	American Iron and Steel Institute
ANSI	American National Standards Institute. Publishes rules and standards in almost all specialized fields (not only electrical engineering). For motors, ANSI took over the majority of American NEMA and UL rules. ANSI C 52.1 corresponds to NEMA-MG 1.
API	American Petroleum Institute: American Institute for crude oil
AS	Australian Standard: Australian regulations (partially adapted to IEC)
ASME	The American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
AWS	American Welding Society
AWWA	American Water Works Association
BAM	(Bundesanstalt für Materialforschung) Federal Authority for Materials Research, Berlin
BP	Boiling point= boiling temperature
BS	British Standard: Regulations of Great Britain partially adapted to IEC
BSI	British Standards Institution
BV	Bureau Veritas: Ship classification association; Country of origin: France
BVS	(Berggewerkschaftliche Versuchstrecke) Mining trade-union test course, Dortmund-Derne
CEB	Corresponds to BEC, Comité Electrotechnique Belge, Brussels: Belgian Committee for Electrical Engineering
CEI	Comitato Elettrotecnico Italiano: Italian Committee for Electrical Engineering
CEI	Corresponds to IEC, Commission Electrotechnique Internationale, Geneva
CEN	Comite Europeen de Normalisation
CENELEC	European Committee for electrical engineering standards (General Office in Brussels)
CF	conflat (-flange)
CGPM	Comptes rendus des seances de la Conference Generale des Poids et Mesures : General Conference on Weights and Measures
CGS	Unit system based on three units: centimetre, gram, second
CIPM	Proces-Verbaux des seances du Comite International des Poids et Mesures : International Committee for Weights and Measures
CSA	Canadian Standards Association: Canadian Supervisory Authority publishes regulations and grants licences.
DEMKO	Danmark Elektriske Materielkontrol: Danish supervisory authority for electronics publishes regulations and grants licences
DIN	German industrial norms
DN	Diameter nominal (former NW, Nennweite= nominal width)
DSRK	DDR (DDR – former German Democratic Republic) Ship inspection and classification
DVGW	German Association for Gas and Water

Table 10-19. Continued

Abbreviation	Meaning, explanation
DVS	German Welding Association
EEMAC	Electrical and Electronic Manufacturers Associations of Canada: Canadian union of electric and electronic products (former CEMA), similar to NEMA
EN	European Norm
FP	Fusing point: melting point= melting temperature
FV	Fine vacuum
GB	Gas ballast
GGVE	Hazardous Goods Regulation, German Railways
GL	German Lloyd: Ship classification association Country of origin: Federal Republic of Germany
GV	Rough vacuum
HV	High vacuum
IEC	International Electrotechnical Commission: All large industrial countries participate in this commission. The IEC recommendations are partly taken over directly to national standards, or national standards are mostly adapted to the IEC recommendations.
IMO	(former IMCO) International Maritime Organisation
IPTS	International practical temperature scale
IS	Indian Standard: partly adapted to IEC
ISO	International Organisation for Standardisation
ISO/DIS	ISO/Draft International Standard (Draft of an international document, released for temporary use)
IUPAC	International Union of Pure and Applied Chemistry
IUPAP	International Union of Pure and Applied Physics
IVC	International Vacuum Congress
JIS	Japanese Industrial Standard: Japanese regulations
KEMA	Keuring van Elektrotechnische Materialen: Dutch Testing Agency for electronic products (also gives CSA approvals for European manufacturers).
KF	small flange
KTA	Committee for nuclear technology
LED	light emitting diode
LF	light flange: clamp flange
LHe	liquid helium
LN	German Air and Space Operations
LN ₂	liquid nitrogen
LRS	Lloyd's Register of Shipping, ship classification association; County of origin: Great Britain
MSS	Manufacturers Standardisation Society of the Valve and Fittings Industry
NBN	Belgian Standards, published by Belgian Institute of Standardisation (partly adapted to IEC)
NEC	National Electrical Code= ANSI C1-: American rule of erection
NEMA	National Electrical Manufacturers Association; national association of manufacturers of electric products in the USA
NEMA-MG1	NEMA Standards Publication-Motors and Generators: NEMA-regulations for motors and generators= ANSI C52.1...

Table 10-19. Continued

Abbreviation	Meaning, explanation
NEMKO	Norges Elektriske Materiellkontroll: Norwegian supervisory authority for electrical equipment (publishes standards and gives approvals).
NEN	Nederlands Norm: Dutch standard
NF	Norme Française: French standard
NFPA	National Fire Protection Association: American Association for fire protection
NK	Nippon Kaiji Japan Marine Association
NPT	USA Standard Taper Pipe Threads: Standards for conical pipe threads; corresponds to USA S B 2.1
NTP	Normal Temperature Pressure: at standard temperature and pressure, usually not used here; see (T_n , p_n)
NV	Det Norske Veritas: ship classification association, country of origin: Norway
OCMA	Oil Companies Materials Association: Association of mineral oil companies
OFHC	Oxygen free high conductivity copper
ÖVE	Austrian Electrical Engineers' Association: ÖVE regulations correspond mostly to VDE and IEC.
PF	Pneurop flange
PNEUROP	European association of manufacturers of compressors, vacuum pumps and pneumatic tools
PRS	Polski Pejestre Statkow: ship classification association; Country of origin: Poland
PTB	German physical technical authority, Braunschweig
REGO	Richtlijnen voor de samenstelling en de beproeving van Elektrisch materieel in verband met Gasontploffingsgevaar, Nederland: Guidelines on Manufacturing and Test of electrical equipment for application in explosion-prone areas, Netherlands.
RID	Reglement International concernant le Transport des Marchandises Dangereuses par Chemin de Fer
RIN _a	Registro Italiano Navale: ship classification association; Country of origin: Italy
RUSSR	Register of Shipping of the USSR
SABS	South African Bureau of Standards
SEMKO	Svenska Elektriska Materielkontrollanstalten: Swedish Supervisory Authority for electric engineering products publishing standards and granting licences
SEN	Svensk Standard
SEV	Swiss Electrical Engineers' Association
SI	Système International d'Unites: International Unit System
SIS	Swedish Standardization Committee
(T_n , p_n)	Volume unit for gas quantity in the standard state, at standard temperature T_n and standard pressure p_n [e.g. m^3 (T_n , p_n)], used here instead of m^3 (NTP)]
TP	triple-point (temperature)
TRAG	Technical rules for acetylene devices and calcium carbide storage
TRB	Technical rules for pressure vessels
TRbF	Technical rules for flammable liquids
TRD	Technical rules for steam boilers
TRG	Technical rules pressurised gases
TRgA	Technical rules for hazardous materials

Table 10-19. Continued

Abbreviation	Meaning, explanation
TRGL	Technical rules for gas high-pressure pipes
TRT	Technical guidelines, tanks
UHV	Ultra-high vacuum
UIC	Union Internationale des Chemins de Fer
UL	Underwriters' Laboratories, Inc.: American national testing agency for fire insurance (executes tests of electrical engineering products, publishes relevant regulations, etc.).
UNI	Ente Nazionale Italiano die Unificazione: Italian Bureau of Standardization
UTE	Union Technique de l'Electricite: French electrical Engineers' Association
UVV	accident prevention
VBG	Alliance of Industrial Trade Associations
VDE	Association of German Electrical Engineers
VDI	Association of German Engineers
VDMA	Association of German Manufacturers of Plants and Machinery
VdTÜV	Association of German Technical Inspection Agency

Table 10-20. Symbols for American and British types of threads

American thread types

NC	American National Coarse Thread Series
NF	American National Fine Thread Series
NEF	American National Extra Fine Thread Series
N	American National 8-, 12- and 16 Pitch Series 8N, 12N, 16N
NS	Special Threads of American National Form
NH	American National Hose Coupling and Fire Hose Coupling Thread
NPT	American Standard Taper Pipe Thread
NPTF	American Standard Taper Pipe Thread, Dryseal
NPTR	American Standard Taper Pipe Thread for Railing Fittings
ANPT	Army-Navy-Aeronautical Pipe Thread- Specification AN-P-363
NPS	American Standard Straight Pipe Thread
NPSC	American Standard Straight Pipe Thread for Pipe Couplings
NPSF	American Standard Internal Straight Pipe Thread, Dryseal
NPSH	American Standard Straight Pipe Thread for Hose Couplings and Nipples
NPSI	American Standard Intermediate Internal Straight Pipe Thread, Dryseal
NPSL	American Standard Straight Pipe Thread for Locknuts and Locknut Pipe Thread
NPSM	American Standard Straight Pipe Thread for Mechanical Joints,
ACME	Acme Threads, Acme-C Centralising Acme-G General Purpose
STUB-ACME	Stub Acme Threads
NGO	American National Gas Outlet-Thread

Table 10-20. Continued

American thread types

V	A 60°-“V” Thread with truncated crests and roots The theoretical “V” form is usually flattened several thousandths of an inch to the user’s specifications			
SB	Manufacturers Stovebolt Standard Thread Classes of Tolerances NC-NF-NEF-N-NS: Class 1 Class 2 Class 3 Class 4 loose fit free fit medium fit close fit			

British thread types

BSW	British Standard-Withworth Coarse Thread Series
BSF	British Standard Fine Thread Series
BSPT	British Standard Taper Pipe Thread
BSPP	British Standard Pipe (Parallel) Thread
WHIT	Whitworth Standard Special Thread
BA	British Association Standard Thread

Table 10-21. Acceptance rules (selection) [10.3]

Regulations	Requirements and proofs
Acceptance AD – Explanatory leaflet A4	Permissible materials: AD-explanatory leaflets, series W Material acceptance: AD explanatory leaflets, series W
Acceptance DIN 2470 Part 1	Final acceptance: Manufacturer’s plant Proof: Acceptance test certificate DIN EN 10205-3.1 B
Acceptance DIN 2470 Part 2	Permissible materials: Group WG 4 DIN 3230 Part 5 Test group: PG 2 DIN 3230 Part 5 Final acceptance: Nominal diameters > DN 200: Acceptance test certificate DIN EN 10205-3.1 C Nominal diameters ≤ DN 200: Acceptance test certificate DIN EN 10205-3.1 B
Factory acceptance DIN 3545	Shut-off valves with tested components: DIN EN 10205-3.1 B System and non-system requirements and tests for condensate traps, acc. to the tests procedures defined in DIN 3230 Part 3. Final acceptance: Manufacturer’s plant Proof: Acceptance test certificate DIN EN 10205-3.1 B
Technical Rules for Steam Boilers (TRD)	a) Material acceptance: Officially recognized experts (TÜV). Proof: Acceptance test certificate DIN EN 10205-3.1 A b) Material acceptance: Manufacturer’s plant Proof: Acceptance test certificate DIN EN 10205-3.1 B c) Final acceptance: Officially recognized experts (TÜV) Proof: Acceptance test certificate DIN EN 10205-3.1 A d) Final acceptance: Manufacturer’s plant Proof: Acceptance test certificate DIN EN 10205-3.1 B

Table 10-21. Continued

Regulations	Requirements and proofs
Technical Rules for flammable liquids (TRbF 301)	<p>Permissible materials: DIN 3230 Part 6 to be adhered to</p> <p>Final acceptance:</p> <p>Nominal diameters > DN 200: Acceptance test certificate DIN EN 10205-3.1 A</p> <p>Nominal diameters \leq DN 200: Acceptance test certificate DIN EN 10205-3.1 B</p> <p>Fittings with tested components:</p> <p>(VdTÜV-Explanatory leaflet 1065): Acceptance test certificate DIN EN 10205-3.1 B</p>
Lloyd's Register of Shipping	<p>a) Material acceptance: Lloyd's Register of Shipping Proof: Acceptance test certificate of Lloyd's Register of Shipping</p> <p>b) Material acceptance: Manufacturer's plant Proof: Acceptance test certificate DIN EN 10205-3.1 B</p> <p>c) Final acceptance: Lloyd's Register of Shipping Proof: Acceptance test certificate of Lloyd's Register of Shipping</p>
American Bureau of Shipping	<p>a) Material acceptance: American Bureau of Shipping Proof: Acceptance test certificate of American Bureau of Shipping</p> <p>b) Material acceptance: Manufacturer's plant Proof: Acceptance test certificate DIN EN 10205-3.1 B</p> <p>c) Final acceptance: American Bureau of Shipping Proof: Acceptance test certificate of American Bureau of Shipping</p>
Det Norske Veritas	<p>a) Material acceptance: Det Norske Veritas Proof: Acceptance test certificate of Det Norske Veritas</p> <p>b) Material acceptance: Manufacturer's plant Proof: Acceptance test certificate DIN EN 10205-3.1 B</p> <p>c) Final acceptance: Det Norske Veritas Proof: Acceptance test certificate of Det Norske Veritas</p>
German Lloyd	<p>a) Material acceptance: German Lloyd Proof: Acceptance test certificate of German Lloyd</p> <p>b) Material acceptance: Manufacturer's plant Proof: Acceptance test certificate DIN EN 10205-3.1 B</p> <p>c) Final acceptance: German Lloyd Proof: Acceptance test certificate of German Lloyd</p>
Bureau Veritas	<p>a) Material acceptance: Bureau Veritas Proof: Acceptance test certificate of Bureau Veritas</p> <p>b) Material acceptance: Manufacturer's plant Proof: Acceptance test certificate DIN EN 10205-3.1 B</p> <p>c) Final acceptance: Bureau Veritas Proof: Acceptance test certificate of Bureau Veritas</p>

Table 10-21. Continued

Regulations	Requirements and proofs
USSR-Register of Shipping	Material acceptance: Manufacturer's plant Proof: Acceptance test certificate DIN EN 10205-3.1 B Final acceptance: USSR-Register of Shipping Proof: Acceptance test certificate USSR-Register of Shipping
Registro Italiano Navale	Material acceptance: Manufacturer's plant Proof: Acceptance test certificate DIN EN 10205-3.1 B Final acceptance: RINA Proof: Acceptance test certificate RINA
Indian Boiler Regulations	Material acceptance: expert authority entrusted by the customer Proof: Acceptance test certificate of the entrusted expert authority Final acceptance: Expert appointed by the customer Proof: Acceptance test certificate of the appointed expert
Final acceptance acc. to the customer's rules	a) Final acceptance: expert appointed by the customer Proof: acc. to the customer's rules b) Final acceptance: Manufacturer's plant Proof: Acceptance test certificate DIN EN 10205-3.1 B



















10.5

Graphical symbols used in the vacuum and process technology

Table 10-22. Graphical symbols used in vacuum technology
(extract from DIN 28401, November 1976)

	Vacuum pump ¹⁾ , general		Axial vacuum pump		Condensation trap with reservoir
	Positive displacement vacuum pump, general		Gas ring vacuum pump		Sorption trap
	Positive displacement vacuum pump, oscillating		Turbo molecular pump		Test dome
	Piston vacuum pump		Fluid entrainment vacuum pump		Metering valve
	Diaphragm vacuum pump		Diffusion pump	Connections and pipes	
	Positive displacement vacuum pump, rotating		Adsorption pump		
	Rotary plunger vacuum pump		Getter pump		Flange connection, general
	Sliding vane vacuum pump		Evaporation pump		Flange connection, screwed
	Rotary piston vacuum pump		Sputtering ion pump		Small flange connection
	Liquid ring vacuum pump		Cryogenic pump		Clamped flange connection
	Roots vacuum pump	Vacuum pump accessories			Threaded tube connection
	Turbo vacuum pump, general				Ball-and-socket joint
	Radial vacuum pump		Baffle, general		Socket joint
			Baffle, cooled		Taper-ground joint

Table 10-22. Continued

	Change in the pipe diameter		Linear motion leadthrough without flange		Vacuum gauge, operating and indicating device for gauge head
	Intersection of two ducts		Rotary/linear motion leadthrough		Vacuum gauge, recording (writing)
	Cross-over of two ducts		Rotary transmission leadthrough		Vacuum gauge with analog indicator
	Branch-off point		Electrical cable duct		Vacuum gauge with digital indicator
	Collection of ducts	Measuring and measuring instruments 2)			Flow measurement
	Mobile line (e.g. compensator, connecting tube)				
	Linear motion leadthrough with flange		Vacuum (for designation of vacuum)		
			Vacuum measurement, vacuum gauge head		

Vacuum pumps 1)
The higher pressure is at the side of the contraction.

2) The graphical symbols ∇ depend on the position. The agreed meaning requires that the point of the angle shows downwards.

Table 10-23. Graphical symbols for components in pipework
(extract from DIN 2429, Part 2, January 1988)






































Shut-off valves			Pressure reducing valve The bigger triangle shows the side with the lower pressure.		Actuating drive with spring
	Shut-off valve general		Pressure reducing valve with injection		Actuating drive with diaphragm
	Angle shut-off valve general	Actuating drives			Actuating drive with weight
	Three-way valve general		Actuating drive with rotating system general		Actuating drive with float
	Four-way valve general		Actuating drive with electric motor	Drain fittings	
	Gate valve		Actuating drive with hydraulic motor with 1 direction of flow		Steam trap general The black space shows the outlet side
	Shut-off valve		Actuating drive with hydraulic motor with 2 directions of flow		without showing the kind of connection
	Isolation valve general		Actuating drive with pneumatic motor with 1 direction of flow		flanged
	Shut-off ball valve		Actuating drive with pneumatic motor with 2 directions of flow		welded
	Plug cock		Actuating drive with thrust system general		mortised
	Butterfly valve		Actuating drive with piston	Expansion bellows	
	Diaphragm valve		Actuating drive with electromagnet		Compensator general
	Hose clamp		Actuating drive, the auxiliary energy is the fluid of the pipe line		Bellow expansion joint
Fittings with progressive adjustment characteristic			Actuating drive, manual, general		Sliding collar

Table 10-23. Continued














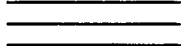


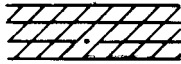



	Lyra compensator		Spray nozzle		Flow limiter with throttling disk
Other piping parts			Shower		Flow limiter with pressure recovery
	Jet nozzle		Sprinkler nozzle		Mixing nozzle

Table 10-24. Graphical symbols for flow diagrams of process engineering plants
(extract from DIN 28004, Part 3, May 1988)

	Check valve		Heat exchanger with crossing flow lines		Centrifugal pump
	Non-return valve		Heat exchanger without crossing of flow lines		Positive displacement pump general
	Air relief valve, relief valve protection		Cooling tower general		Gear pump
	Sight glass		Fluid filter general, filtering apparatus general		Screw pump
	Strainer		Liquid filter general		Helical rotor pump
	Vessel with dished ends		Gas filter general, air filter general		Reciprocating piston pump
	Vessel with conic ends		Separator general		Diaphragm pump
	Spherical vessel		Strainer apparatus general, rack general		Liquid jet pump
	Column general, vessel with baffles general		Pump general		Fan general

Table 10-24. Continued

	Radial fan		Drive machine with expansion of fluid, turbine		Pipe heated or cooled
	Axial fan		Coupling general		Pipe with jacket or protection tube
	Drive machine general		Facility for heating or cooling, general		Pipe heated or cooled and insulated
	Electric motor general		Vessel with external electrical heating		Pipe insulated

10.6

Graphical symbols and call letters for measuring control and regulation (MCR) in process engineering

(extract from DIN 19227, sheet 1, September 1973)

Supplementary data to table 10-25.

to (1):

Letters already determined as “additional letters” shall not be used as subsequent letters.

to (2):

For additional letters, also minuscules are permitted.

to (3):

The letters A, B, C, D, I, J and Z of group I are subject to later standardization.

to (4):

For practical reasons, the letter H has been included in group I of the table in accordance with ISO. It stands for the effects caused by man (not automatically occurring).

to (5):

The letters N, O and Y are reserved for free utilization by the user. This means that one of these letters may be allocated to a frequently repeating measured parameter in a plant if this measured parameter is not contained in the table of letters.

For an individually occurring, not allocated measured parameter, the letter X shall be used.

to (6):

Quality parameters are e. g.: concentration, pH value, conductivity, thermal power, Wobbe-index, flash point, chrominance, refractive index, consistency.

to (7):

Parameter composed of several other parameters, as far as it cannot be depicted by call letters.

to (8):

Term see DIN 1305.

to (9):

According to ISO, the letter J is meant as additional letter for "Scanning".

to (10):

The distinction whether it concerns control or regulation becomes clear in the flow diagram or the respective chart (for open or closed action function, see DIN 19226, Edition May 1968, part 1.2 and 1.3).

to (11):

According to ISO, the letter Q is not only meant as additional letter, but also as subsequent letter for "Integrating" or "Summating".

to (12):

Registration is the collective term for output with memory function, regardless of the kind of storage.

to (13):

According to ISO, T is meant as subsequent letter for the use of transducers (Transmitting).

to (14):

Upper limit value, mean value and lower limit value of the measured parameter are marked with plus, oblique or minus following the subsequent letters Q, S, Z, A. In the general sense, the signs + and – may also be used for the marking of the end positions 'open' or 'closed' or for switching positions 'ON' or 'OFF'. Switching commands may be shown on the MCR-circle.

ISO provides for the marking of the upper and lower limit values with the letters H (high) and L (low). They may be written inside or outside of the MCR-circle.

Table 10-25. Call letters for MCR-technology (measuring, control, regulation)

Call letter	Group 1: Measurement category or other input parameter as first letter	as additional letter (1) (2)	Group 2: Function as subsequent letter sequence: O, I, R, C, S, Z, A
A	_____ (3)		Limit value signal, alarm
B	_____ (3)		
C	_____ (3)		
D	Density	Difference	Automatic regulation, automatic continuous control (10)
E	Electrical sizes		_____ (1)
F	Flow, flow rate	Ratio	_____ (1)
G	Distance, length, position		
H	Manual input, manual intervention (4)		
I	_____ (3)		Indication
J	_____ (3)	_____ (9)	_____ (1)
K	Time		
L	Level (also of separating layer)		
M	Humidity		
N	Disposable (5)		
O	Disposable (5)		
P	Pressure		
Q	Quality categories (analysis, material property) (except for D, M, V) (6)	Integral, sum	Sight signal, yes/no statement, (not alarm) _____ (1) (11)

Table 10-25. Continued

Call letter	Group 1: Measurement category or other input parameter	Group 2: Function
	as first letter	as subsequent letter sequence: O, I, R, C, S, Z, A
R	Radiation categories	Registration (12)
S	Velocity, speed, frequency	Circuit, non-continuous control
T	Temperature	_____ (13)
U	Multi variables (7)	
V	Viscosity	
W	Weight, mass (8)	
X	Other sizes (5)	
Y	Disposable (5)	
Z	_____ (3)	
+		Emergency intervention, control by trigger at:
/		upper limit value (14)
		intermediate value (14)
-		lower limit value (14)

Difference pressure measurement, display and control in measuring station

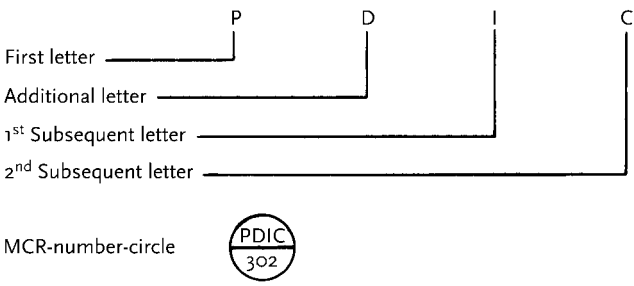


Figure 10-1. Sequence of letters written in a MCR (measuring, control, regulation) circle

Supplements to figure 10-1.
 Call letters are written in the upper part of the “measuring, control regulation circle”. The sequence results from table 10-25.
 The output and operation location (i. e. communication node of man and MCR device) is determined as follows:

Output and operation location	Identification in the circle
measuring or setting place	Call letters not underlined
central measuring station	Call letters continuously underlined (see examples)
local measuring panel, measuring sub-station	Call letters may be underlined twice if a distinction to the central measuring station is to be made.

In the MCR-circle, besides call letters, also position identification and MCR-position numbers are indicated (see fig. 10-1 and table 10-26).

Table 10-26. Examples of application

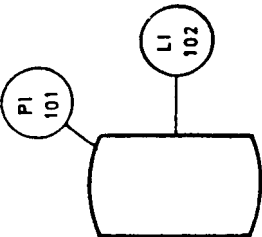
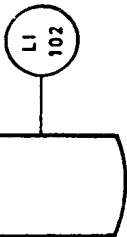
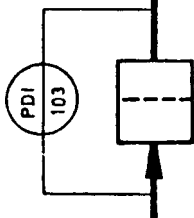
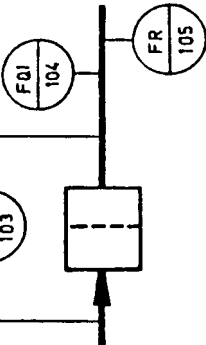
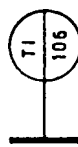
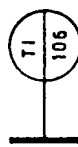
MCR-equipment (measuring, control and regulation)	In text	Identification in flow diagram	Remarks
Pressure measurement Local indication	PI 101		e.g. pressure gauge
Level measurement Local indication	LI 102		e.g. level glass
Differential pressure measurement Indication in measuring station	PDI 103		e.g. volumeter with connected pulse counter in measuring station
Flow quantity measurement Registration in measuring station	FQI 104		e.g. with standard orifice as sensor
Flow rate measurement Registration in measuring station	FR 105		
Temperature measurement Indication in measuring station	TI 106		

Table 10-26. Continued

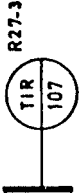
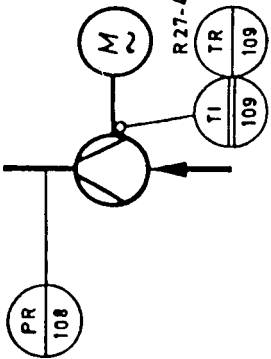
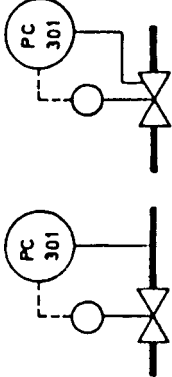
MCR-equipment (measuring, control and regulation)	In text	Identification in flow diagram	Remarks
Temperature measurement Indication and registration in measuring station	TIR 107		e.g. on six-color recorder R 27. Recording position 3 and separate indicator
Pressure measurement Registration in central measuring station	PR 108		
Measurement of bearing temperature Indication in sub-station, additional registration in central measuring station	TIR 109		
Pressure control, local	PC 301		If no special discharge port is required, e.g. for reduction stations, the diagram on the right can be used.

Table 10-26. Continued

MCR-equipment (measuring, control and regulation)	In text	Identification in flow diagram	Remarks
Flow control Registration of control variable, alarm when reaching the lower fault area and adjustment of command variable in measuring station	FRCA-302		
Pressure control Indication of control variable and adjustment of command variable local, additional alarm in measuring station when reaching the upper fault area	PA + 303 PIC 303		e.g. two point pressure control of a compressor

10.7

Physical call values of liquids and gases [10.4]

Table 10.27. Thermodynamic properties of liquids.

Substance	Formula	Molar mass kg/kmol	Density ρ at 20 °C kg/m ³	Boiling tem- perature t_b at 1013 mbar °C	Evaporation enthalpy r at 1013 mbar kJ/kg	Specific heat capacity c_p at constant pres- sure at 20 °C kJ/kg · K	Thermal con- ductivity λ at 20 °C W/m · K	Dynamic viscosity η at 20 °C mPas	at 50 °C mPas
acetone	C ₃ H ₆ O	58.08	791	56.25	523	2.156	0.160	0.325	0.249
acetonitrile	C ₂ H ₃ N	41.05	783	81.65	800	2.173	–	0.26	0.272
formic acid	CH ₂ O ₂	46.02	1220	100.65	494	2.169	0.224	1.791	1.031
aniline	C ₆ H ₇ N	93.13	1022	184.35	448	2.068	0.195	4.40	1.85
benzaldehyde	C ₇ H ₆ O	106.12	1046	178.85	371	–	–	1.60	–
benzene	C ₆ H ₆	78.11	879	80.15	394	1.729	0.144	0.649	0.436
benzyl alcohol	C ₇ H ₈ O	108.14	1045	205.45	467	1.972	0.160	5.584	2.574
benzyl chloride	C ₇ H ₇ Cl	126.58	–	179.4	–	1.432	0.138	1.38	–
bromine	Br ₂	159.81	3140	58.75	187	0.452	0.125	0.996	0.762
bromine-benzene	C ₆ H ₅ Br	157.01	1495	156.2	231.8	0.963	0.112	1.124	0.80
<i>n</i> -butanol	C ₄ H ₁₀ O	74.12	810	117.75	590	2.345	0.155	2.951	1.42
chlorobenzene	C ₆ H ₅ Cl	112.56	1106	131.75	325	1.298	0.129	0.80	0.58
chlorobutane	C ₄ H ₉ Cl	92.57	897	50.8	–	1.888	0.116(2)	0.44	0.328
chloropentane	C ₅ H ₁₁ Cl	106.59	872	108.30	–	–	0.116	–	–
chloropropane	C ₃ H ₇ Cl	78.54	890	46.45	–	1.650	0.119(2)	0.329	–
<i>m</i> -chlorotoluol	C ₇ H ₇ Cl	126.58	–	161.6	–	1.214	–	0.877	0.616
cyclohexanol	C ₆ H ₁₂ O	100.16	952(7)	161.15	452	1.750(7)	0.136(7)	68.02	11.81
decane	C ₁₀ H ₂₂	142.29	730	174.15	276	2.173	0.126	0.364	0.237
dichloroacetic acid	C ₂ H ₂ O ₂ Cl ₂	128.95	1552	194.4	323	–	–	–	–
1,1-dichloroethane	C ₂ H ₄ Cl ₂	98.96	1174	57.25	309	1.276	–	0.48	–
1,2-dichloroethane	C ₂ H ₄ Cl ₂	98.96	1258	83.45	324	1.260	0.135	0.84	0.584
1,1-dichloroethylene	C ₂ H ₂ Cl ₂	96.95	1250	31.7	272	1.172	–	0.467	–

Table 10-27. Continued

Substance	Formula	Molar mass kg/kmol	Density ρ at 20 °C kg/m ³	Boiling tem- perature t_b at 1013 mbar °C	Evaporation enthalpy r at 1013 mbar kJ/kg	Specific heat capacity c_p at constant pres- sure at 20 °C kJ/kg · K	Thermal con- ductivity λ at 20 °C W/m · K	Dynamic viscosity η at 20 °C mPas	at 50 °C mPas
dichloromethane	CH ₂ Cl ₂	84.93	1336	39.85	329	1.156	0.141	0.442	0.346
diethylether	C ₄ H ₁₀ O	74.12	714	34.55	360	2.336	0.138	0.243	0.183
diethylsulfide	C ₄ H ₁₀ S	90.18	837	90	—	2.010	—	0.446	0.334
dimethylsulfide	C ₂ H ₆ S	62.13	850	35.9	427	1.897	—	—	—
1,4-dioxane	C ₄ H ₈ O ₂	88.11	1034	101.35	405.3	1.721	0.163	1.26	0.779
diphenyl	C ₁₂ H ₁₀	154.21	1040(7)	255.25	310	1.235(7)	—	—	1.49(5)
dodecane	C ₁₂ H ₂₆	170.34	751	216.35	257	2.173	0.143	1.489	0.922
eicosan	C ₂₀ H ₄₂	282.56	—	343.85	204	—	—	—	3.276
acetic acid	C ₂ H ₄ O ₂	60.05	1049	117.95	406	1.997	0.161	1.21	0.792
acetic acid anhydride	C ₄ H ₆ O ₃	102.09	1082	138.85	385	—	0.170	0.905	0.62
ethanol	C ₂ H ₆ O	46.07	789	78.35	846	2.395	0.173	1.201	0.701
ethylbenzene	C ₈ H ₁₀	106.17	866	136.15	340	1.738	0.131	0.679	0.482
ethylene glycol	C ₂ H ₆ O ₂	62.07	1115	197.25	812	2.357	0.256	20.41	—
ethylene oxide	C ₂ H ₄ O	44.05	896	10.35	580	1.972	—	0.27	—
fluorobenzene	C ₆ H ₅ F	96.10	1024	85.35	—	1.524	0.137	0.598	0.428
formamide	CH ₃ NO	45.04	1112	210.5	—	2.382	0.352(4)	3.764	1.84
glycerine	C ₃ H ₈ O ₃	92.09	1260	289.85	825	2.366	0.286	14.99	—
heptadecane	C ₁₇ H ₃₆	240.47	—	302.5	—	—	—	—	2.144
heptane	C ₇ H ₁₆	100.20	684	98.45	317	2.198	0.124	0.413	0.302
1-heptanol	C ₇ H ₁₆ O	116.20	822	175.9	440	2.358	0.137	7.011	2.681
hexadecane	C ₁₆ H ₃₄	226.45	775	286.85	227	2.144	—	3.506	1.841
hexane	C ₆ H ₁₄	86.18	659	68.75	335	2.227	0.137	0.308	0.229
1-hexanol	C ₆ H ₁₄ O	102.18	820	157.05	496.4	2.173	0.143	5.427	2.16
hydrogen fluoride	HF	20.01	987	19.55	1290	3.85	—	0.256(1)	—
isobutanol	C ₄ H ₁₀ O	74.12	806	107.85	578	2.311	0.135	3.901	1.62
isopentanol	C ₅ H ₁₂ O	88.15	810	130.6	502	2.345	0.151	4.301	1.851
isopropanol	C ₃ H ₈ O	60.10	785	82.25	670	2.496	0.141	2.432	1.063
isopropylbenzol	C ₉ H ₁₂	120.19	862	152.45	313	—	0.125	—	—

Table 10-27. Continued

Substance	Formula	Molar mass kg/kmol	Density ρ at 20 °C kg/m ³	Boiling tem- perature t_b at 1013 mbar °C	Evaporation enthalpy r at 1013 mbar kJ/kg	Specific heat capacity c_p at constant pres- sure at 20 °C kJ/kg · K	Thermal con- ductivity λ at 20 °C W/m · K	Dynamic viscosity η at 20 °C mPas	at 50 °C mPas
iodine	I ₂	253.81	4937(7)	184.35	171	0.214(7)	—	—	—
iodobenzene	C ₆ H ₅ I	204.1	1832	188.25	195	0.775	0.101	1.705	1.121
methanol	CH ₃ O	32.04	792	64.65	1100	2.495	0.202	0.584	0.396
methyl ethyl ketone	C ₄ H ₈ O	72.11	803	79.65	433	2.219	0.145	0.323	0.31
monochloroacetic acid	C ₂ H ₃ O ₂ Cl	94.50	—	189.5	266	—	—	—	3.146
naphthalene	C ₁₀ H ₈	128.17	1145(7)	217.95	314	1.281(7)	—	—	0.65(6)
nitrobenzol	C ₆ H ₅ NO ₂	123.11	1174	210.9	396	1.465	0.149	2.01	1.241
nonadecane	C ₁₉ H ₄₀	268.52	—	329.95	211	—	—	—	2.861
nonane	C ₉ H ₂₀	128.26	718	150.85	288	2.177	0.137	0.301	0.199
octadecane	C ₁₈ H ₃₈	254.50	—	316.25	216	—	—	0.713	0.492
octane	C ₈ H ₁₈	114.23	703	125.65	301	2.186	0.131	0.546	0.389
1-octanol	C ₈ H ₁₈ O	130.23	829	179.75	410	2.219	0.137	8.933	3.217
pentadecane	C ₁₅ H ₃₂	212.42	768	207.65	233	2.151	—	2.863	1.567
pentane	C ₅ H ₁₂	72.15	626	36.05	358	2.273	0.136	0.227	0.188
1-pentanol	C ₅ H ₁₂ O	88.15	814	137.85	515	2.315	0.156	5.10	2.001
phenol	C ₆ H ₆ O	94.11	1071(7)	181.85	511	1.294(7)	—	11.41	3.421
phosgene	COCl ₂	98.92	1388	7.65	247	1.02(1)	—	—	—
propanol	C ₃ H ₈ O	60.10	804	97.25	754	2.345	0.157	2.231	1.126
propionic acid	C ₃ H ₆ O ₂	74.08	993	140.85	419	2.165	0.150	1.094	0.738
1,3-propyleneglycol	C ₃ H ₈ O ₂	76.10	1053	214.45	—	—	0.199	—	—
mercury	Hg	200.63	13546	356.20	295.60	—	—	—	—
sulphur	S ₂	64.12	2062(7)	444.6	293	0.350(7)	—	—	0.163(6)
carbon disulphide	CS ₂	76.13	1262	46.25	351	1.00	0.170	0.367	0.293
styrol (vinylbenzene)	C ₈ H ₈	104.15	907	145.15	—	1.717	—	0.74	—

Table 10-27. Continued

Substance	Formula	Molar mass kg/kmol	Density ρ at 20 °C kg/m ³	Boiling tem- perature t_b at 1013 mbar °C	Evaporation enthalpy r at 1013 mbar kJ/kg	Specific heat capacity c_p at constant pres- sure at 20 °C kJ/kg · K	Thermal con- ductivity λ at 20 °C W/m · K	Dynamic viscosity η at 20 °C mPas	at 50 °C mPas
1,1,2,2-tetrachloro- ethane	C ₂ H ₂ Cl ₄	167.88	1600	146.2	230	1.118	0.114	1.77	1.132
tetrachloroethylene	C ₂ Cl ₄	165.83	1621	121.15	209	0.879	–	0.904	0.657
tetrachloromethane	CCl ₄	133.82	1594	76.55	195	0.850	0.106	0.961	0.654
tetradecane	C ₁₄ H ₃₀	198.39	763	253.55	241	2.186	0.145	2.319	1.324
thiophene	C ₄ H ₄ S	84.14	1065	84.1	–	1.46	0.137(3)	0.664	0.474
thiophosgene	CSCl ₂	114.99	–	73.5	–	–	–	–	–
toluol	C ₇ H ₈	92.14	867	110.65	356	1.717	0.141	0.586	0.419
tribromomethane	CHBr ₃	252.77	2890	150.50	–	0.536	–	2.109	1.50
trichloroacetic acid	C ₂ HO ₂ Cl ₃	163.40	1620(7)	195.6	–	–	–	7.506	–
1,1,1-trichloroethane	C ₂ H ₃ Cl ₃	133.40	1338	74	–	1.076	–	0.84	–
trichloroethylene	C ₂ HCl ₃	131.39	1463	87.25	242	0.950	0.128	0.60	0.447
trichloromethane	CHCl ₃	119.38	1490	61.15	254	0.992	0.128	0.563	0.424
tridecane	C ₁₃ H ₂₈	184.37	756	235.45	247	2.160	0.146	1.865	1.11
1,2,3-trimethyl- benzene	C ₉ H ₁₂	120.19	894	176.05	333	1.787	–	–	–
undecane	C ₁₁ H ₂₄	156.31	740	195.95	266	2.173	0.141	1.176	0.757
water	H ₂ O	18.02	998.3	100	2257	4.182	0.600	0.89	0.547
<i>m</i> -xylene	C ₈ H ₁₀	106.17	866	139.15	343	1.717	0.132	0.615	0.445
<i>o</i> -xylene	C ₈ H ₁₀	106.17	881	144.45	348	1.733	0.134	0.807	0.556
<i>p</i> -xylene	C ₈ H ₁₀	106.17	861	138.35	339	1.700	0.133	0.643	0.456

(1) at 0 °C

(2) at 4.4 °C

(3) at 12 °C

(4) at 40 °C

(5) at 70 °C

(6) at 100 °C

(7) applicable for the solid state

Table 10-28. Thermodynamic properties of gases

Substance	Formula	Molar mass kg/kmol	Standard density ¹⁾ kg/m ³	Specific heat capacity c_p at 25 °C kJ/kg · K	Gas constant R J/kg · K
acetaldehyde	C ₂ H ₄ O	44.05	(1.97)	1.239	188.74
acetylene	C ₂ H ₂	26.04	1.17	1.687	319.27
formic acid methylester	C ₂ H ₄ O ₂	60.05	(2.68)	–	138.45
ammonia	NH ₃	17.03	0.77	2.093	488.19
argon	Ar	39.95	1.78	–	208.11
benzene	C ₆ H ₆	78.11	(3.49)	1.047	106.43
bromomethane	CH ₃ Br	94.94	(4.24)	0.452	87.57
hydrogen bromide	HBr	80.91	3.64	0.360	102.75
butane	C ₄ H ₁₀	58.12	2.70	1.700	143.04
butadiene-(1.2)	C ₄ H ₆	54.09	(2.41)	1.482	153.70
butadiene-(1.3)	C ₄ H ₆	54.09	(2.41)	1.474	153.70
butene-(1)	C ₄ H ₈	56.11	(2.50)	1.595	148.17
chlorine	Cl ₂	70.91	3.17	0.477	117.24
chlorodifluoromethane	CHClF ₂	86.47	(3.86)	0.844	96.14
chloroethane	C ₂ H ₅ Cl	64.51	(2.88)	1.315	128.87
chloroethene (vinyl- chloride)	C ₂ H ₃ Cl	62.50	(2.79)	0.858	133.02
chloromethane	CH ₃ Cl	50.49	2.31	0.808	164.66
chlorotrifluoroethene (-ethylene)	C ₂ ClF ₃	116.48	(5.20)	0.722	71.37
hydrogen chloride	HCl	36.46	1.63	0.795	228.03
cyan bromide	CNBr	105.94	(4.73)	0.440	78.47
cyan chloride	CNCl	61.48	(2.74)	0.729	135.23
cyan fluoride	CNF	45.02	(2.01)	0.946	184.67
cyan iodide	CNI	152.94	(6.83)	0.314	54.36
hydrogen cyanide	HCN	27.03	(1.21)	1.327	307.58
cyclobutane	C ₄ H ₈	56.11	(2.50)	1.287	148.17
cyclopropane	C ₃ H ₆	42.08	(1.88)	1.331	197.57
dichlorodifluoromethane	CCl ₂ F ₂	120.91	(5.40)	0.721	68.76
dichlorofluoromethane	CHCl ₂ F	102.92	(4.59)	0.593	80.78
dichlorotetrafluoroethane	C ₂ Cl ₂ F ₄	170.92	(7.63)	0.729	48.64
dicyan	C ₂ N ₂	52.04	(2.32)	1.097	159.76
diethylether	C ₄ H ₁₀ O	74.12	(3.31)	1.553	112.16
difluormethane	CH ₂ F ₂	52.03	(2.32)	1.097	159.79
dimethylacetylene	C ₄ H ₆	54.09	(2.41)	1.440	153.70
dimethylamine	C ₂ H ₇ N	45.08	(2.01)	1.531	184.42
dimethylether	C ₂ H ₆ O	46.07	(2.06)	–	180.46
2.2-dimethylpropane	C ₅ H ₁₂	72.15	(3.22)	1.687	115.23
dinitrogen tetroxide	N ₂ O ₄	92.02	(4.11)	0.858	90.34
dinitrogen monoxide	N ₂ O	44.01	1.97	0.879	188.91
ethane	C ₂ H ₆	30.07	1.35	1.754	276.48
ethanol	C ₂ H ₆ O	46.07	(2.06)	1.595	180.46

Table 10-28. Continued

Substance	Formula	Molar mass kg/kmol	Standard density ¹⁾ kg/m ³	Specific heat capacity c_p at 25 °C kJ/kg · K	Gas constant R J/kg · K
ethylene	C ₂ H ₄	28.05	1.26	1.553	296.39
ethylacetylene	C ₄ H ₆	54.09	(2.41)	1.507	153.70
ethylamine	C ₂ H ₇ N	45.08	(2.01)	–	184.42
ethylene oxide	C ₂ H ₄ O	44.05	(1.97)	1.097	188.74
ethylmercaptane	C ₂ H ₆ S	62.13	(2.77)	1.170	133.81
fluorine	F ₂	38.00	1.70	0.825	218.78
fluoroethane	C ₂ H ₅ F	48.06	(2.15)	1.227	172.99
fluoromethane	CH ₃ F	34.03	1.52	1.099	244.31
hydrogen fluoride	HF	20.01	0.98	1.444	415.49
formaldehyde	CH ₂ O	30.03	(1.34)	1.172	276.85
furan	C ₄ H ₄ O	68.07	(3.04)	0.961	122.13
helium	He	4.00	0.18	5.200	2078.5
hydrogen iodide	HJ	127.91	5.79	0.226	64.99
ketene	C ₂ H ₂ O	42.04	(1.88)	1.143	197.76
carbon dioxide	CO ₂	44.01	1.97	0.846	188.91
carbon oxide	CO	28.01	1.25	1.038	296.82
krypton	Kr	83.80	3.74	–	99.21
air	–	28.96	1.29	1.005	287.08
methane	CH ₄	16.04	0.72	2.227	518.32
methanol	CH ₄ O	32.04	(1.43)	1.407	259.48
methylacetylene	C ₃ H ₄	40.06	(1.79)	1.516	207.53
methylamine	CH ₅ N	31.06	(1.39)	1.665	267.67
methylmercaptane	CH ₄ S	48.11	(2.15)	1.054	172.81
2-methylpropane	C ₄ H ₁₀	58.12	2.59	1.666	143.04
methylpropylether	C ₄ H ₁₀ O	74.12	(3.31)	–	112.16
neon	Ne	20.18	0.90	–	412
phosgene	COCl ₂	98.92	(4.42)	0.615	84.04
propane	C ₃ H ₈	44.10	2.01	1.671	188.52
propanol	C ₃ H ₈ O	60.10	(2.68)	1.496	138.33
propadiene	C ₃ H ₄	40.06	(1.79)	1.474	207.53
propylene	C ₃ H ₆	42.08	1.91	1.520	197.57
oxygen	O ₂	32.00	1.43	0.913	259.81
sulphur dioxide	SO ₂	64.06	2.92	0.607	129.78
sulphur hexafluoride	SF ₆	146.05	(6.52)	0.641	56.92
sulphur trioxide	SO ₃	80.06	(3.57)	0.632	103.84
hydrogen sulphide	H ₂ S	34.08	1.54	1.005	243.95
nitrogen	N ₂	28.01	1.25	1.038	296.82
nitrogen dioxide	NO ₂	46.01	(2.05)	0.808	180.69
nitrogen monoxide	NO	30.01	1.34	0.971	277.04
tetrachlorodifluoroethane	C ₂ Cl ₄ F ₂	203.83	(9.10)	–	40.78
tetrafluoromethane	CF ₄	88.00	(3.93)	0.758	94.47
toluene	C ₇ H ₈	92.14	(4.11)	1.126	90.23

Table 10-28. Continued

Substance	Formula	Molar mass kg/kmol	Standard density ¹⁾ kg/m ³	Specific heat capacity c_p at 25 °C kJ/kg · K	Gas constant R J/kg · K
1,1,1-trifluoroethane	C ₂ H ₃ F ₃	84.04	(3.45)	0.929	98.92
trifluoromethane	CHF ₃	70.02	(3.13)	0.837	118.73
trimethylamine	C ₃ H ₉ N	59.11	(2.64)	1.552	140.65
water	H ₂ O	18.02	(0.80)	1.879*	461.37
hydrogen	H ₂	2.02	0.09	14.34	4115.84
xenon	Xe	131.30	5.86	–	63.32

1) Density at 0 °C and 1013 mbar. Values in brackets are calculated for (hypothetic) ideal gas in this state (values in brackets are not reliable)

* if saturated

10.8

Tables and diagrams

Table 10-29. Pressure conversion table [10.5]

The table serves the conversion of the unit *Torr* used in vacuum technology up to now into the nowadays common unit mbar and vice versa.

The pressure values to be converted are given in the middle column.

The conversion of the pressure values < 1 and > 100 up to 1000 can also occur with the help of the table.

Regarding the values < 1, the comma needs only to be moved accordingly to the left, and in values > 100 and divisible by 10, the comma is to be moved one decimal place to the right. In the case of values not divisible by 10, the conversion by the addition of two corresponding numbers is possible.

The 4th digit of each table value is rounded up or down.

Examples:

- Value to be converted (middle column):
30 Torr \triangle 40 mbar (right column)
- Value to be converted (middle column):
30 mbar \triangle 22.5 Torr (left column)
- Middle column considering the altered decimal places:
450 Torr \triangle 600 mbar (right column)
- 450 mbar \triangle 337.5 Torr (left column)
- 0.8 Torr \triangle 1.067 mbar (right column)
- 0.8 mbar \triangle 0.6 Torr (left column)
- 763 Torr (700 + 63) \triangle 933.3 + 83.99 = 1017.29 mbar (right column)
- 763 mbar (700 + 63) \triangle 525 + 47.25 = 572.25 Torr (left column)

Torr		mbar	Torr		mbar
0.75	1	1.33	42.00	56	74.66
1.50	2	2.67	42.75	57	75.99
2.25	3	4.00	43.50	58	77.33
3.00	4	5.33	44.25	59	78.66
3.75	5	6.67	45.00	60	79.99
4.50	6	8.00	45.75	61	81.33
5.25	7	9.33	46.50	62	82.66
6.00	8	10.67	47.25	63	83.99
6.75	9	12.00	48.00	64	85.33
7.50	10	13.33	48.75	65	86.66
8.25	11	14.67	49.50	66	87.99
9.00	12	16.00	50.25	67	89.33
9.75	13	17.33	51.00	68	90.66
10.50	14	18.66	51.75	69	91.99
11.25	15	20.00	52.50	70	93.33
12.00	16	21.33	53.25	71	94.66
12.75	17	22.66	54.00	72	95.99
13.50	18	24.00	54.75	73	97.33
14.25	19	25.33	55.50	74	98.66
15.00	20	26.66	56.25	75	99.99
15.75	21	28.00	57.00	76	101.3
16.50	22	29.33	57.75	77	102.7
17.25	23	30.66	58.50	78	104.0
18.00	24	32.00	59.25	79	105.3
18.75	25	33.33	60.00	80	106.7
19.50	26	34.66	60.75	81	108.0
20.25	27	36.00	61.50	82	109.3
21.00	28	37.33	62.25	83	110.7
21.75	29	38.66	63.00	84	112.0
22.50	30	40.00	63.76	85	113.3
23.25	31	41.33	64.51	86	114.7
24.00	32	42.66	65.26	87	116.0
24.75	33	44.00	66.01	88	117.3
25.50	34	45.33	66.76	89	118.7
26.25	35	46.66	67.51	90	120.0
27.00	36	48.00	68.26	91	121.3
27.75	37	49.33	69.01	92	122.7
28.50	38	50.60	69.76	93	124.0
29.25	39	52.00	70.51	94	125.3
30.00	40	53.33	71.26	95	126.7
30.75	41	54.66	72.01	96	128.0
31.50	42	56.00	72.76	97	129.3
32.25	43	57.33	73.51	98	130.7
33.00	44	58.66	74.26	99	132.0
33.75	45	60.00	75.01	100	133.3
34.50	46	61.33	150.0	200	266.6

Table 10-29. Continued

Torr		mbar		Torr		mbar	
35.25	47	62.66	225.0	300	400.0		
36.00	48	63.99	300.0	400	533.3		
36.75	49	65.33	375.0	500	666.6		
37.50	50	66.66	450.0	600	799.9		
38.25	51	67.99	525.0	700	933.3		
39.00	52	69.33	600.0	800	1067		
39.75	53	70.66	675.1	900	1200		
40.50	54	71.99	750.1	1000	1333		
41.25	55	73.33					

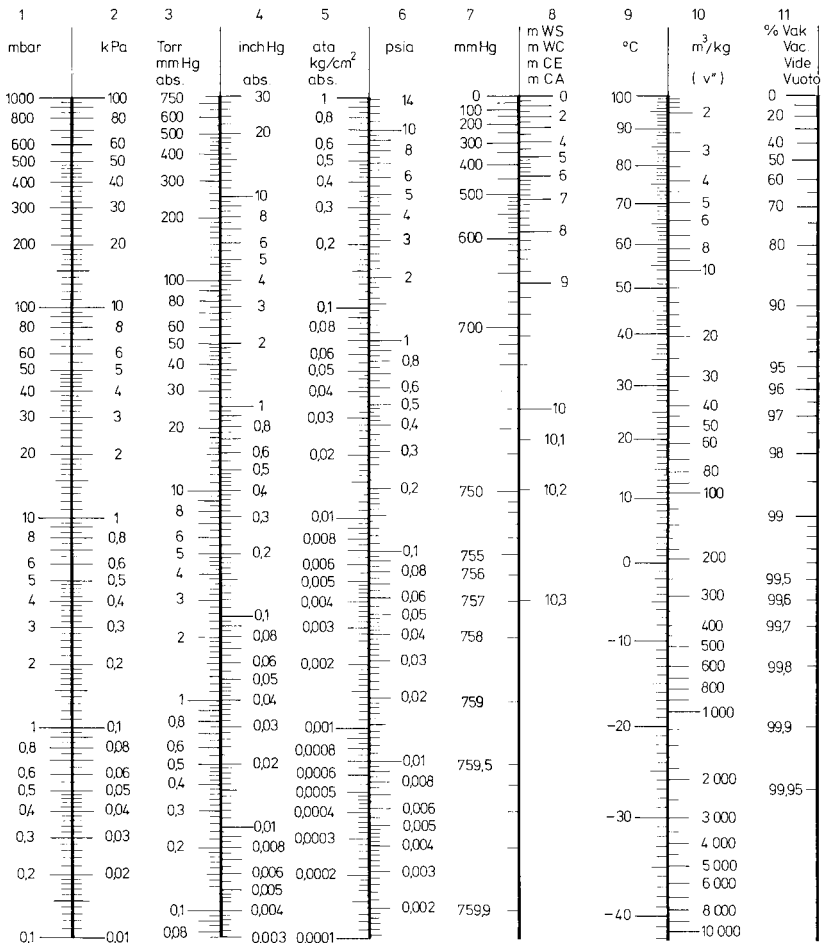


Figure 10-2. Vacuum measuring units [10.5]
Columns 1 and 2: pressure units acc. to SI-system, Columns 3 to 8: Formerly used units,
Columns 1 to 6: pressure units, absolute (abs), Columns 7 and 8: measuring units for under-
pressure used until now, Column 9: boiling temperature of water depending on its vapor
pressure, Column 10: corresponding specific volume of water vapor, Column 11: vacuum in %

Table 10-30. Temperature calculations scales Kelvin, Celsius, Fahrenheit and Rankine [10.4]

K	°C	°F	°R	K	°C	°F	°R
0.00	-273.15	-459.67	0.00	1500.00	1226.85	2240.33	2700.00
25.00	-248.15	-414.67	45.00	1525.00	1251.85	2285.33	2745.00
50.00	-223.15	-369.67	90.00	1550.00	1276.85	2330.33	2790.00
75.00	-198.15	-324.67	135.00	1575.00	1301.85	2375.33	2835.00
100.00	-173.15	-279.67	180.00	1600.00	1326.85	2320.33	2880.00
125.00	-148.15	-234.67	225.00	1625.00	1351.85	2465.33	2925.00
150.00	-123.15	-189.67	270.00	1650.00	1376.85	2510.33	2970.00
175.00	-98.15	-144.67	315.00	1675.00	1401.85	2555.33	3015.00
200.00	-73.15	-99.67	360.00	1700.00	1426.85	2600.33	3060.00
225.00	-48.15	-54.67	405.00	1725.00	1451.85	2645.33	3105.00
250.00	-23.15	-9.67	450.00	1750.00	1476.85	2690.33	3150.00
275.00	1.85	35.33	495.00	1775.00	1501.85	2735.33	3195.00
300.00	26.85	80.33	540.00	1800.00	1526.85	2780.33	3240.00
325.00	51.85	125.33	585.00	1825.00	1551.85	2825.33	3285.00
350.00	76.85	170.33	630.00	1850.00	1576.85	2870.33	3330.00
375.00	101.85	215.33	675.00	1875.00	1601.85	2915.33	3375.00
400.00	126.85	260.33	720.00	1900.00	1626.85	2960.33	3420.00
425.00	151.85	305.33	765.00	1925.00	1651.85	3005.33	3465.00
450.00	176.85	350.33	810.00	1950.00	1676.85	3050.33	3510.00
475.00	201.85	395.33	855.00	1975.00	1701.85	3095.33	3555.00
500.00	226.85	440.33	900.00	2000.00	1726.85	3140.33	3600.00
525.00	251.85	485.33	945.00	2025.00	1751.85	3185.33	3645.00
550.00	276.85	530.33	990.00	2050.00	1776.85	3230.33	3690.00
575.00	301.85	575.33	1035.00	2075.00	1801.85	3275.33	3735.00
600.00	326.85	620.33	1080.00	2100.00	1826.85	3320.33	3780.00
625.00	351.85	665.33	1125.00	2125.00	1851.85	3365.33	3825.00
650.00	376.85	710.33	1170.00	2150.00	1876.85	3410.33	3870.00
675.00	401.85	755.33	1215.00	2175.00	1901.85	3455.33	3915.00
700.00	426.85	800.33	1260.00	2200.00	1926.85	3500.33	3960.00
725.00	451.85	845.33	1305.00	2225.00	1951.85	3545.33	4005.00
750.00	476.85	890.33	1350.00	2250.00	1976.85	3590.33	4050.00
775.00	501.85	935.33	1395.00	2275.00	2001.85	3635.33	4095.00
800.00	526.85	980.33	1440.00	2300.00	2026.85	3680.33	4140.00
825.00	551.85	1025.33	1485.00	2325.00	2051.85	3725.33	4185.00
850.00	576.85	1070.33	1530.00	2350.00	2076.85	3770.33	4230.00
875.00	601.85	1115.33	1575.00	2375.00	2101.85	3815.33	4275.00
900.00	626.85	1160.33	1620.00	2400.00	2126.85	3860.33	4320.00
925.00	651.85	1205.33	1665.00	2425.00	2151.85	3905.33	4365.00
950.00	676.85	1250.33	1710.00	2450.00	2176.85	3950.33	4410.00
975.00	701.85	1295.33	1755.00	2475.00	2201.85	3995.33	4455.00
1000.00	726.85	1340.33	1800.00	2500.00	2226.85	4040.33	4500.00
1025.00	751.85	1385.33	1845.00	2525.00	2251.85	4085.33	4545.00
1050.00	776.85	1430.33	1890.00	2550.00	2276.85	4130.33	4590.00
1075.00	801.85	1475.33	1935.00	2575.00	2301.85	4175.33	4635.00
1100.00	826.85	1520.33	1980.00	2600.00	2326.85	4220.33	4680.00

Table 10-30. Continued

K	°C	°F	°R	K	°C	°F	°R
1125.00	851.85	1565.33	2025.00	2625.00	2351.85	4265.33	4725.00
1150.00	876.85	1610.33	2070.00	2650.00	2376.85	4310.33	4770.00
1175.00	901.85	1655.33	2115.00	2675.00	2401.85	4355.33	4815.00
1200.00	926.85	1700.33	2160.00	2700.00	2426.85	4400.33	4860.00
1225.00	951.85	1745.33	2205.00	2725.00	2451.85	4445.33	4905.00
1250.00	976.85	1790.33	2250.00	2750.00	2476.85	4490.33	4950.00
1275.00	1001.85	1835.33	2295.00	2775.00	2501.85	4535.33	4995.00
1300.00	1026.85	1880.33	2340.00	2800.00	2526.85	4580.33	5040.00
1325.00	1051.85	1925.33	2385.00	2825.00	2551.85	4625.33	5085.00
1350.00	1076.85	1970.33	2430.00	2850.00	2576.85	4670.33	5130.00
1375.00	1101.85	2015.33	2475.00	2875.00	2601.85	4715.33	5175.00
1400.00	1126.85	2060.33	2520.00	2900.00	2626.85	4760.33	5220.00
1425.00	1151.85	2105.33	2565.00	2925.00	2651.85	4805.33	5265.00
1450.00	1176.85	2150.33	2610.00	2950.00	2676.85	4850.33	5310.00
1475.00	1201.85	2195.33	2655.00	2975.00	2701.85	4895.33	5355.00
				3000.00	2726.85	4940.33	5400.00
				3025.00	2751.85	4985.33	5445.00

Table 10-31a. Water vapour temperature table (saturated steam), vacuum range [10.5]

 θ temperature, h'' specific enthalpy of dry saturated water vapor, p vapour pressure, Δh_v evaporation heat, v'' specific volume of water vapor

θ °C	ρ mbar	v'' m ³ /kg	h'' kJ/kg	Δh_v kJ/kg
-74	0.001411	651700	2349.3	2851.2
-73	0.001647	561000	2351.6	2851.0
-72	0.001920	483600	2353.9	2850.7
-71	0.002236	417500	2356.1	2850.5
-70	0.002598	361100	2358.4	2850.3
-69	0.003015	312600	2360.6	2850.0
-68	0.003495	271100	2362.8	2849.8
-67	0.004044	235400	2365.0	2849.6
-66	0.004672	204700	2367.2	2849.3
-65	0.005391	178300	2369.4	2849.1
-64	0.006212	155500	2371.6	2848.8
-63	0.007149	135800	2373.8	2848.5
-62	0.008215	118700	2376.0	2848.3
-61	0.009429	103900	2378.1	2848.0
-60	0.01080	91100	2380.3	2847.7
-59	0.01237	79900	2382.4	2847.5
-58	0.01414	70300	2384.5	2847.2
-57	0.01614	61800	2386.7	2846.9
-56	0.01841	54500	2388.8	2846.6
-55	0.02097	48000	2390.9	2846.4

Table 10-31a. Continued

θ °C	ρ mbar	v'' m ³ /kg	h'' kJ/kg	Δh_v kJ/kg
-54	0.02385	42400	2393.0	2846.1
-53	0.02711	37500	2395.1	2845.8
-52	0.03077	33200	2397.2	2845.5
-51	0.03488	29400	2399.3	2845.2
-50	0.03949	26100	2401.3	2844.9
-49	0.04467	23200	2403.4	2844.6
-48	0.05047	20600	2405.5	2844.4
-47	0.05696	18300	2407.5	2844.1
-46	0.06422	16300	2409.6	2843.8
-45	0.07232	14600	2411.6	2843.5
-44	0.08136	13000	2413.6	2843.2
-43	0.09144	11600	2415.7	2842.9
-42	0.1026	10400	2417.7	2842.6
-41	0.1151	9312	2419.7	2842.3
-40	0.1289	8347	2421.7	2842.0
-39	0.1443	7489	2423.7	2841.7
-38	0.1614	6726	2425.7	2841.4
-37	0.1803	6046	2427.7	2841.1
-36	0.2012	5441	2429.7	2840.8
-35	0.2244	4900	2431.7	2840.6
-34	0.2500	4416	2433.7	2840.3
-33	0.2783	3985	2435.7	2840.0
-32	0.3095	3598	2437.6	2839.7
-31	0.3438	3252	2439.6	2839.4
-30	0.3816	2942	2441.6	2839.1
-29	0.4233	2663	2443.5	2838.8
-28	0.4691	2413	2445.5	2838.6
-27	0.5194	2188	2447.4	2838.3
-26	0.5746	1986	2449.4	2838.0
-25	0.6351	1804	2451.3	2837.7
-24	0.7014	1640	2453.3	2837.5
-23	0.7741	1492	2455.2	2837.2
-22	0.8536	1358	2457.1	2836.9
-21	0.9407	1237	2459.1	2836.7
-20	1.035	1129	2461.0	2836.4
-19	1.139	1030	2462.9	2836.2
-18	1.252	940.9	2464.8	2835.9
-17	1.375	859.9	2466.8	2835.7
-16	1.509	786.5	2468.7	2835.4
-15	1.656	719.8	2470.6	2835.2
-14	1.815	659.3	2472.5	2835.0
-13	1.988	604.2	2474.4	2834.7
-12	2.176	554.2	2476.3	2834.5
-11	2.380	508.6	2478.3	2834.3

Table 10-31a. Continued

θ °C	ρ mbar	v'' m ³ /kg	h'' kJ/kg	Δh_v kJ/kg
-10	2.601	467.1	2480.2	2834.1
-9	2.841	429.3	2482.1	2833.9
-8	3.101	394.8	2484.0	2833.7
-7	3.383	363.3	2485.9	2833.5
-6	3.688	334.5	2487.8	2833.3
-5	4.017	308.2	2489.7	2833.1
-4	4.373	284.2	2491.6	2832.9
-3	4.758	262.1	2493.5	2832.7
-2	5.173	242.0	2495.4	2832.5
-1	5.622	223.5	2497.3	2832.4
0	6.105	206.6	2501.6	2501.6
1	6.566	192.6	2503.4	2499.2
2	7.054	179.9	2505.2	2496.8
3	7.574	168.2	2507.1	2494.5
4	8.128	157.3	2508.9	2492.1
5	8.718	147.2	2510.7	2489.7
6	9.345	137.8	2512.6	2487.4
7	10.01	129.1	2514.4	2485.0
8	10.71	121.0	2516.2	2482.6
9	11.47	113.4	2518.1	2480.3
10	12.27	106.4	2519.9	2477.9
11	13.11	99.91	2521.7	2475.5
12	14.01	93.84	2523.6	2473.2
13	14.96	88.18	2525.4	2470.8
14	15.97	82.90	2527.2	2468.5
15	17.03	77.98	2529.1	2466.1
16	18.16	73.38	2530.9	2463.8
17	19.36	69.09	2532.7	2461.4
18	20.62	65.09	2534.5	2459.0
19	21.95	61.34	2536.4	2456.7
20	23.36	57.84	2538.2	2454.3
21	24.85	54.56	2540.0	2452.0
22	26.41	51.49	2541.8	2449.6
23	28.07	48.62	2543.6	2447.2
24	29.82	45.93	2545.5	2444.9
25	31.65	43.40	2547.3	2442.5
26	33.59	41.03	2549.1	2440.2
27	35.63	38.81	2550.9	2437.8
28	37.78	36.73	2552.7	2435.4
29	40.04	34.77	2554.5	2433.1
30	42.41	32.93	2556.4	2430.7
31	44.91	31.20	2558.2	2428.3
32	47.53	29.57	2560.0	2425.9
33	50.28	28.04	2561.8	2423.6

Table 10-31a. Continued

θ °C	ρ mbar	v'' m ³ /kg	h'' kJ/kg	Δh_v kJ/kg
34	53.18	26.60	2563.6	2421.2
35	56.21	25.24	2565.4	2418.8
36	59.39	23.97	2567.2	2416.4
37	62.74	22.76	2569.0	2414.1
38	66.24	21.63	2570.8	2411.7
39	69.91	20.56	2572.6	2409.3
40	73.74	19.55	2574.4	2406.9
41	77.77	18.59	2576.2	2404.5
42	81.98	17.69	2577.9	2402.1
43	86.38	16.84	2579.7	2399.7
44	90.99	16.04	2581.5	2397.3
45	95.82	15.28	2583.3	2394.9
46	100.8	14.56	2585.1	2392.5
47	106.1	13.88	2586.9	2390.1
48	111.6	13.23	2588.6	2387.7
49	117.3	12.62	2590.4	2385.3
50	123.3	12.05	2592.2	2382.9
52	136.1	10.98	2595.7	2378.1
54	150.0	10.02	2599.2	2373.2
56	165.1	9.159	2602.7	2368.4
58	181.4	8.381	2606.2	2363.5
60	199.2	7.679	2609.7	2358.6
62	218.3	7.044	2613.2	2353.7
64	239.0	6.469	2616.6	2348.8
66	261.5	5.948	2620.1	2343.9
68	285.6	5.476	2623.5	2338.9
70	311.6	5.046	2626.9	2334.0
72	339.6	4.656	2630.3	2329.0
74	369.6	4.300	2633.7	2324.0
76	401.8	3.976	2637.1	2318.9
78	436.4	3.680	2640.4	2313.9
80	473.5	3.409	2643.8	2308.8
82	513.2	3.162	2647.1	2303.8
84	555.7	2.935	2650.4	2298.7
86	601.0	2.727	2653.6	2293.1
88	649.4	2.536	2656.9	2288.4
90	701.0	2.361	2660.1	2283.2
92	756.0	2.200	2663.4	2278.0
94	814.6	2.052	2666.6	2272.8
96	876.8	1.915	2669.7	2267.5
98	943.0	1.789	2672.9	2262.2
100	1013.2	1.673	2676.0	2256.9

Table 10-31b. Water vapor temperature table (saturated steam)

Temperature range 100 °C–300 °C [10.5]

 θ temperature, h'' specific enthalpy of dry saturated water vapor, p vapor pressure, Δh_v evaporation heat, v'' specific volume of water vapor

θ °C	K T	P bar	v'' m ³ /kg	h'' kJ/kg	Δh_v kJ/kg
100	373.15	1.0133	1.673	2676.0	2256.9
101	374.15	1.0500	1.618	2677.6	2254.3
102	375.15	1.0878	1.566	2679.1	2251.6
103	376.15	1.1267	1.515	2680.7	2248.9
104	377.15	1.1668	1.466	2682.2	2246.3
105	378.15	1.2080	1.419	2683.7	2243.6
106	379.15	1.2504	1.374	2685.3	2240.9
107	380.15	1.2941	1.331	2686.8	2238.2
108	381.15	1.3390	1.289	2688.3	2235.4
109	382.15	1.3852	1.249	2689.8	2232.7
110	383.15	1.4327	1.210	2691.3	2230.0
111	384.15	1.4815	1.173	2692.8	2227.3
112	385.15	1.5316	1.137	2694.3	2224.5
113	386.15	1.5832	1.102	2695.8	2221.8
114	387.15	1.6362	1.069	2697.2	2219.0
115	388.15	1.6906	1.036	2698.7	2216.2
116	389.15	1.7465	1.005	2700.2	2213.4
117	390.15	1.8039	0.9753	2701.6	2210.7
118	391.15	1.8628	0.9463	2703.1	2207.9
119	392.15	1.9233	0.9184	2704.5	2205.1
120	393.15	1.9854	0.8915	2706.0	2202.2
121	394.15	2.0492	0.8655	2707.4	2199.4
122	395.15	2.1145	0.8405	2708.8	2196.6
123	396.15	2.1816	0.8162	2710.2	2193.7
124	397.15	2.2504	0.7928	2711.6	2190.9
125	398.15	2.3210	0.7702	2713.0	2188.0
126	399.15	2.3933	0.7484	2714.4	2185.2
127	400.15	2.4675	0.7273	2715.8	2182.3
128	401.15	2.5435	0.7069	2717.2	2179.4
129	402.15	2.6215	0.6872	2718.5	2176.5
130	403.15	2.7013	0.6681	2719.9	2173.6
131	404.15	2.7831	0.6497	2721.3	2170.7
132	405.15	2.8670	0.6319	2722.6	2167.8
133	406.15	2.9528	0.6146	2723.9	2164.8
134	407.15	3.041	0.5980	2725.3	2161.9
135	408.15	3.131	0.5818	2726.6	2158.9
136	409.15	3.223	0.5662	2727.9	2155.9
137	410.15	3.317	0.5511	2729.2	2153.0
138	411.15	3.414	0.5364	2730.5	2150.0
139	412.15	3.513	0.5222	2731.8	2147.0

Table 10-31b. Continued

θ °C	K T	P bar	v'' m ³ /kg	h'' kJ/kg	Δh_v kJ/kg
140	413.15	3.614	0.5085	2733.1	2144.0
142	415.15	3.823	0.4823	2735.6	2137.9
144	417.15	4.042	0.4577	2738.1	2131.8
146	419.15	4.271	0.4346	2740.6	2125.7
148	421.15	4.510	0.4129	2743.0	2119.5
150	423.15	4.760	0.3924	2745.4	2113.2
152	425.15	5.021	0.3732	2747.7	2106.9
154	427.15	5.293	0.3551	2750.0	2100.6
156	429.15	5.577	0.3380	2752.3	2094.2
158	431.15	5.872	0.3219	2754.5	2087.7
160	433.15	6.181	0.3068	2756.7	2081.3
162	435.15	6.502	0.2924	2758.9	2074.7
164	437.15	6.836	0.2789	2761.0	2068.1
166	439.15	7.183	0.2661	2763.1	2061.4
168	441.15	7.545	0.2540	2765.1	2054.7
170	443.15	7.920	0.2426	2767.1	2047.9
172	445.15	8.311	0.2317	2769.0	2041.1
174	447.15	8.716	0.2215	2770.9	2034.2
176	449.15	9.137	0.2117	2772.7	2027.3
178	451.15	9.574	0.2025	2774.5	2020.2
180	453.15	10.027	0.1938	2776.3	2013.1
182	455.15	10.496	0.1855	2778.0	2006.0
184	457.15	10.983	0.1776	2779.6	1998.8
186	459.15	11.488	0.1702	2781.2	1991.5
188	461.15	12.010	0.1631	2782.8	1984.2
190	463.15	12.551	0.1563	2784.3	1976.7
192	465.15	13.111	0.1499	2785.7	1969.3
194	467.15	13.690	0.1438	2787.1	1961.7
196	469.15	14.289	0.1380	2788.4	1954.1
198	471.15	14.909	0.1324	2789.7	1946.4
200	473.15	15.549	0.1272	2790.9	1938.6
202	475.15	16.210	0.1221	2792.1	1930.7
204	477.15	16.893	0.1173	2793.2	1922.8
206	479.15	17.598	0.1128	2794.3	1914.7
208	481.15	18.326	0.1084	2795.3	1906.6
210	483.15	19.077	0.1042	2796.2	1898.5
212	485.15	19.852	0.1003	2797.1	1890.2
214	487.15	20.651	0.09646	2797.9	1881.8
216	489.15	21.475	0.09283	2798.6	1873.4
218	491.15	22.324	0.08936	2799.3	1864.9
220	493.15	23.198	0.08604	2799.9	1856.2
222	495.15	24.099	0.08286	2800.5	1847.5
224	497.15	25.027	0.07982	2800.9	1838.7
226	499.15	25.982	0.07691	2801.4	1829.8

Table 10-31b. Continued

θ °C	K T	P bar	v'' m ³ /kg	h'' kJ/kg	Δh_v kJ/kg
228	501.15	26.965	0.07412	2801.7	1820.8
230	503.15	27.976	0.07145	2802.0	1811.7
232	505.15	29.016	0.06889	2802.2	1802.5
234	507.15	30.086	0.06643	2802.3	1793.2
236	509.15	31.186	0.06408	2802.3	1783.8
238	511.15	32.317	0.06182	2802.3	1774.2
240	513.15	33.478	0.05965	2802.2	1764.6
242	515.15	34.672	0.05757	2802.0	1754.9
244	517.15	35.898	0.05558	2801.8	1745.0
246	519.15	37.157	0.05366	2801.4	1735.0
248	521.15	38.449	0.05181	2801.0	1724.9
250	523.15	39.776	0.05004	2800.4	1714.6
252	525.15	41.137	0.04833	2799.8	1704.3
254	527.15	42.534	0.04669	2799.1	1693.8
256	529.15	43.967	0.04511	2798.3	1683.2
258	531.15	45.437	0.04360	2797.4	1672.4
260	533.15	46.943	0.04213	2796.4	1661.5
265	538.15	50.877	0.03871	2793.5	1633.6
270	543.15	55.058	0.03559	2789.9	1604.6
275	548.15	59.496	0.03274	2785.5	1574.7
280	553.15	64.202	0.03013	2780.4	1543.6
285	558.15	69.186	0.02773	2774.5	1511.3
290	563.15	74.461	0.02554	2767.6	1477.6
295	568.15	80.037	0.02351	2759.8	1443.6
300	573.15	85.927	0.02165	2751.0	1406.0

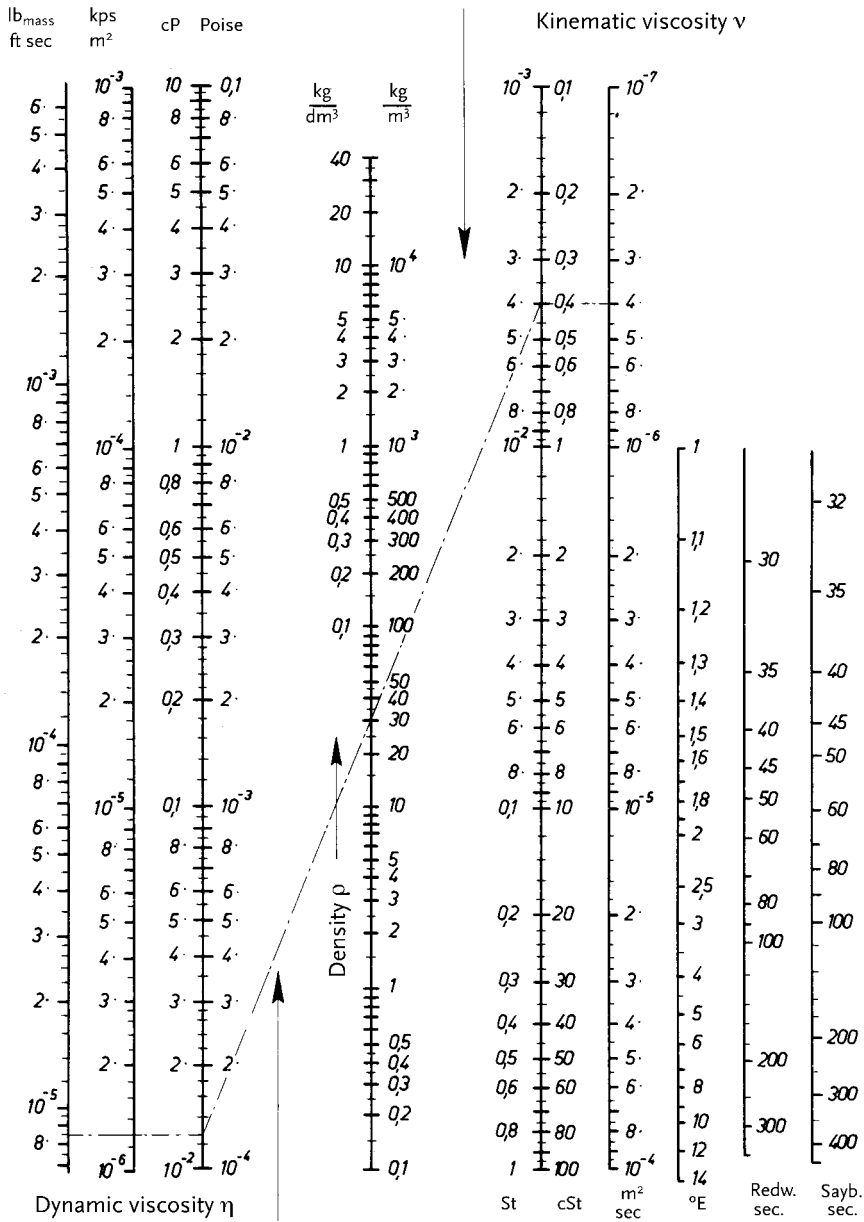


Figure 10-3 a. Conversion of different viscosity values [10.6]

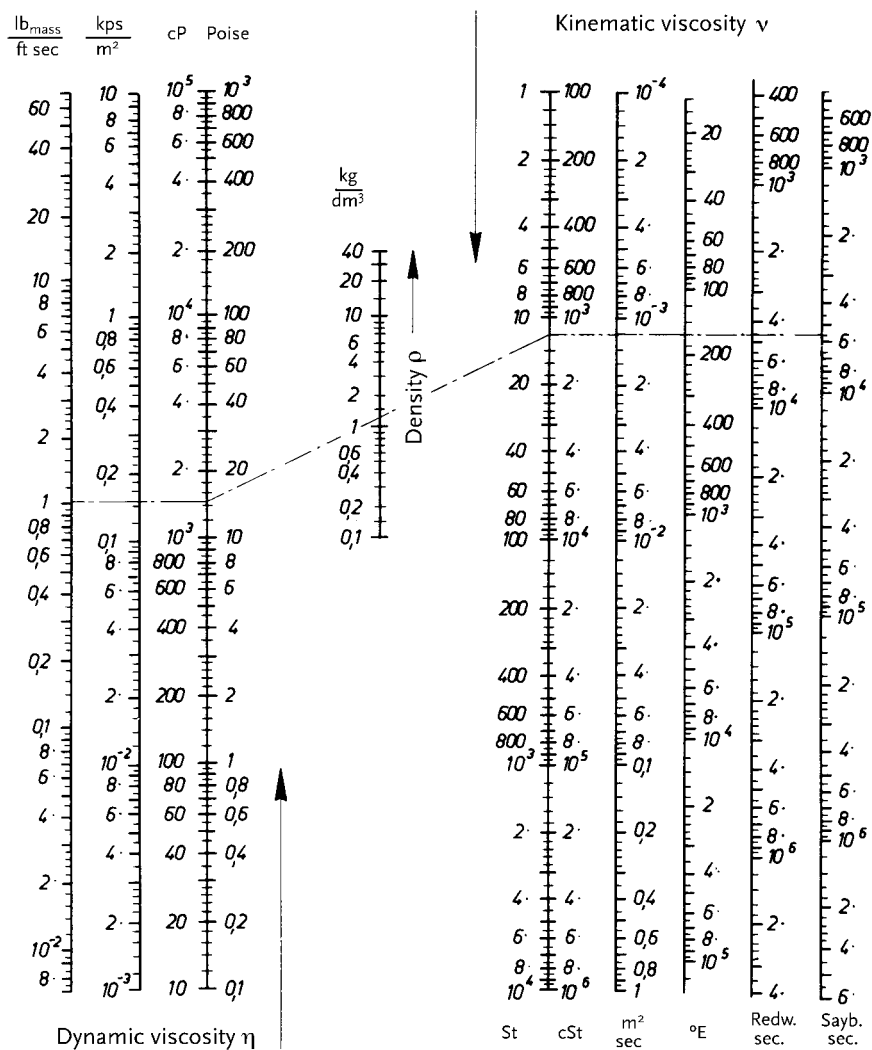


Figure 10-3 b. Conversion of different viscosity values [10.6]

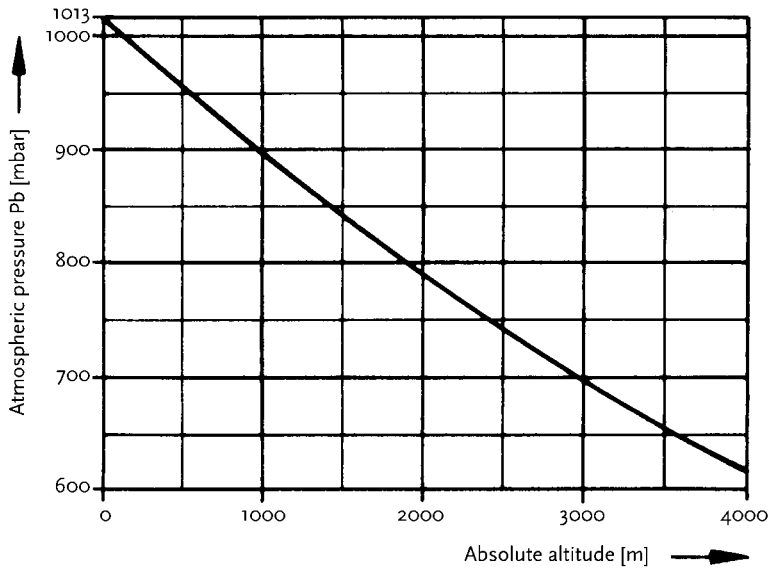


Figure 10-4. Atmospheric pressure and its dependence on altitude, referring to 1013 mbar (abs) (physical atmosphere) at 0 m standard zero level

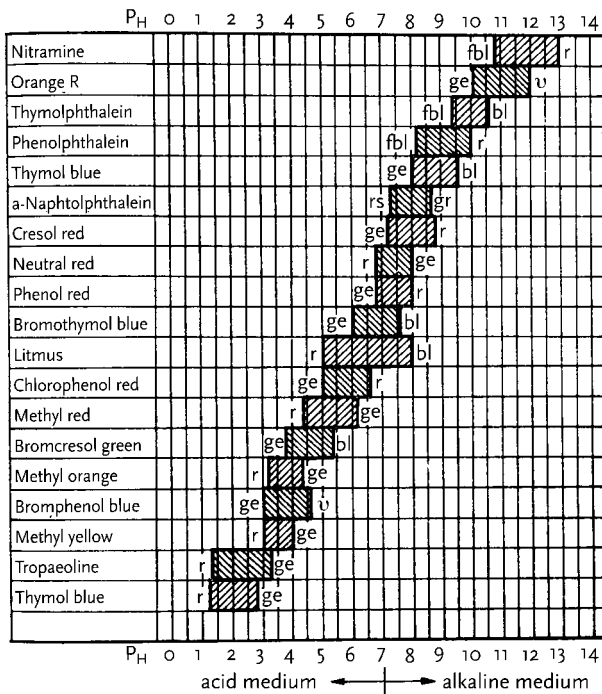


Figure 10-5. Ph-ranges of color changing of several indicators [10.7]
bl blue, cl colorless, ye yellow, gr green, r red, p pink, v violet

Table 10-32. Indices often used in physics and engineering with definite meaning (extract from DIN 1304)

Index ¹⁾	Legend	Index ¹⁾	Legend
abs	absolute	n	standard value
as	asynchronous	N	nominal value
at	atomic	opt	optic
ax	axial	p	constant pressure, isobar
b	base, basic, bending, blind	par	parallel
chem	chemical	pot	potential
d	damping	q	blind
dyn	dynamic	rad	radial
eff	effective	red	reduced
el	electric	rel	relative
ela	elastic	rev	reversible
en	energetic	s	apparent
eq	equivalent	sig	signal
hyd	hydraulic	sph	spherical
indi	indirect	stat	static, stationary
indu	induced	syn	synchronous
k	short circuit	th	thermal
kin	kinetic	tot	total
lin	linear	trv	transversal
long	longitudinal	V	constant volume
mag	magnetic	v	loss
max	maximal	vis	visual
min	minimal	w	active
mod	modulated	zul	permissible

1) Indices consisting of more than one letter can be shortened to the first letter if misunderstandings are not to be expected.

Table 10-33. The Greek alphabet

A α	B β	$\Gamma\gamma$	$\Delta\delta$	E ϵ	Z ζ
alpha	beta	gamma	delta	epsilon	zeta
H η	$\Theta\theta$	I ι	K κ	$\Lambda\lambda$	M μ
eta	theta	iota	kappa	lambda	my
N ν	$\Xi\xi$	O \omicron	$\Pi\pi$	P ρ	$\Sigma\sigma$
ny	xi	omikron	pi	rho	sigma
T τ	$\Upsilon\upsilon$	$\Phi\phi$	X χ	$\Psi\psi$	$\Omega\omega$
tau	ypsilon	phi	chi	psi	omega

Table 10-34. Mathematical symbols

...	and so on
=	equal
≡	identical
≠	not equal
≈	approximately
△	corresponding
<	less than
>	more than
≤	less than or equal,
≥	more than or equal
≪	low against
≫	large against
+	plus
−	minus
·, ×	multiplied by
:/−	divided by
%	percent
‰	per mille
	parallel
⧻	non-parallel
⊥	rectangular
≡	congruent
~	proportional
∠	angle
Σ	sum
√	square root
$\sqrt[n]{}$	n-root
π	pi ($\pi = 3.14159$)
Δf	delta f, difference of two function values
log	logarithm, general
lg	decade logarithm, logarithm based on 10
ln	natural logarithm, logarithm based on e

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