Performance mapping vs'design parameters for screw compressors and other displacement compressor types

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Zusammenfassung

Der für Turbomaschinen gewöhnliche Parameter "Spezifische Drehzahl", ist für Verdrängermaschinen nicht geeignet. Fördervolumen und Drehzahlen sind für Verdrängerverdichtern von spezifischen Verdrängervolumen, Hauptläuferdurchmesser und Umfangsgeschwindigkeiten bestimmt. Die Abhängigkeit zwischen den Konstruktionsparametern und den thermodynamischen Parametern die die Leistungsfähigkeit beeinflussen, wird für trockene Verdichtern bestimmt. Diese Parametern sind dimensionslos, und damit auch für beliebige Verdichterbauarten und Hauptrotordurchmessern gültig. Die Abhängigkeit zwischen Spalthöhe und Hauptläuferdurchmesser wird auch beobachtet. Kennfelder (Fördervolumen—Drehzahl) wird dann für beliebige Bauarten bestimmt. Begrenzungen von Maschinenelementen und erforderlichem Wirkungsgrad, sind auch in diesen Kennfeldern eingeschlossen. Der Einfluß von Hauptparametern wie Zähnezahl für Haupt- und Nebenrotor, Zahntiefe und Läuferlänge/Durchmesser, wird für Schraubenverdichtern auch bestimmt. Die für die verschiedenen Fördervolumen und Anwendungen geeignete Verdichterbauarten können damit bestimmt werden.

Abstract

For positive displacement type machines, the "specific speed" parameter used for turbomachinery, will not be a relevant performance parameter. Capacity and shaft speed for displacement compressors will be given by specific displacement, rotor diameter and tip speed. For dry (non-injected) compressors, relations between major design parameters and thermodynamic parameters determining compressor performance and optimum tip speed, have been established. These parameters are size-independent and represent any geometry concept. Scale influence due to clearance variation with rotor size will also be included. Regions of operation (capacity—shaft speed) could then be mapped for arbitrary concepts. Limitations set by mechanical components and efficiency requirements have been considered. For screw compressors, influence of major geometry parameters like number of male and female lobes, lobe depth and length/diameter ratio could be studied. Feasible types of compressors for different capacity regions and applications could then be determined.

Symbols/Symbole

A	[–]	leakage loss parameter	Leckageverlustparameter
A	[m^2]	area	Fläche
a	[m/s]	sonic velocity	Schallgeschwindigkeit
B	[–]	throttling loss parameter	Drosselverlustparameter
c	[m/s]	local velocity	Strömungsgeschwindigkeit
D	[m]	rotor diameter	Rotordurchmesser
e	[m]	eccentricity	Excentrizität
L	[m]	leakage path or rotor length	Spaltlänge oder Läuferlänge
m	[–]	number of male lobes	Zähnezahl der Hauptrotor
n	[s^{-1}]	shaft speed	Drehzahl
p	[Pa]	pressure	Druck
q	[–]	number of rotor cells in contact	Anzahl den Arbeitsräumen in Behrührung
R	[J/(kg K)]	gas constant	Gaskonstante
s	[–]	$s = (\nu - 1)/\nu$	
T	[K]	thermodynamic temperature	thermodynamische Temperatur
u	[m/s]	peripheral speed (tip speed)	Umfangsgeschwindigkeit
V	[m^3]	displacement (volume/revolution)	Verdrängervolumen (Volumen pro Umdrehung)
V/D^3	[–]	specific displacement	Spezifische Verdrängervolumen
$\dot{V}$	[m^3/s]	capacity (suction rate of volume flow)	Eintrittsvolumenstrom
Z	[–]	throttling coefficient	Drosselkoeffizient
α	[–]	exponent in clearance-size relation	Exponent des Spalthöhenverhältnisses
α	[–]	discharge coefficient	Durchflußbeiwert
δ	[m]	rotor clearance	Spalthöhe
η	[–]	efficiency	Wirkungsgrad

θ	[–]	polytropic/isentropic power ratio	Polytropisches/Isentropisches Leistungsverhältnis
κ	[–]	isentropic exponent	Isentropenexponent
ν	[–]	polytropic exponent	Polytropenexponent
Π	[–]	pressure ratio	Druckverhältnis
ρ	[kg/m^3]	local gas density	Dichte
σ	[–]	$\sigma = (\kappa - 1)/\kappa$	
τ	[–]	relative power loss	spezifischer Leistungsverlust
ϕ	[rad]	open port angular interval	Intervall des Drehwinkels, für offenen Eintritt b.z.w. Auslaß
ψ	[–]	isentropic nozzle flow coefficient	Isentropischer Durchflußkoeffizient

Subscripts/Indizes

i, j	rotor cell number	Arbeitsraum (Index)
ij	leakage path number	Leckageweg
k	in- or outlet	Eintritt oder Auslaß
F	female rotor	Nebenläufer
M	male rotor	Hauptläufer
o	reference (datum) or inlet	Referenz oder Eintritt
in	inlet	Eintritt
leak	leakage	Leckage
max	maximum	Maximum
min	minimum	Minimum
opt	optimum	optimal
out	outlet	Auslaß
s	isentropic	isentrop
thr	throttling	Drosselung
vol	volumetric	Liefergrad

1 Introduction

In compressor design, detailed analysis and/or prototype testing are the methods normally used for evaluation of new compressor concepts. Computer simulation, as well as testing with a great number of parameters, seldom gives the overall view that should be required for rough comparison of various geometry concepts, however. By testing or by refined analysis, a proposed concept could be evaluated "at one point", i.e. for one set of values for all design parameters. Much work will then be required for optimization of the design. During the early "conceptual" design phase, it is essential to develop simplified methods for determination of the feasible range of operation and for indication of optimum operating conditions, for each concept. By performance mapping based on simplified analytical relations, various geometry concepts could be compared more quickly. The best solutions for different regions within the entire capacity range specified for a new range of compressors could then be determined. This kind of analysis should be completed before going too far in detailing the proposed design.

For rotary displacement type machines, there is virtually an infinite number of degrees of geometrical freedom. It is then very important to identify critical design parameters and to establish general relations between these parameters and compressor performance. Non-dimensional, i.e. size-independent formulations should be preferred for generality.

2 Concepts for rotary displacement compressors

For turbo compressors, with open channel flow in blade cascades, enthalpy rise ("head") and hence the stage pressure ratio will be related to the tip speed and the volume flow. The well established "specific speed" parameter, based on the similarity laws for dynamic fluid machinery, then gives good information on performance and shaft speed—size—capacity relations.

For the case of positive displacement compressors, however, fluid is trapped in closed rotor cavities and forced towards the discharge side. The head is then not primarily related to the peripheral speed or the volume flow, and the pressure—capacity characteristic will be stiff. The "specific speed" will then not be a relevant performance parameter for displacement type compressors. As pointed out, for rotary displacement machines there is virtually an infinite number of degrees of geometrical freedom, fig.1. Over a long period of time, a great number of geometry concepts have been proposed, for pumps, compressors and internal combustion engines. Classifications of such concepts, according to geometry (e.g. number of moving elements, rotor profile curve type etc.) have been established /9/. Nothing about performance can be predicted from these "concept catalogues", however. Comparisons of performance for some specific types of fluid expanders (radial turbines, and screw type displacement machines) have been presented, though /3/, /7/, /8/. The method that will be described in this paper was developed to make possible early performance predictions for arbitrary displacement machine concepts, and has been utilized for evaluation and comparison of rotary compressor principles.

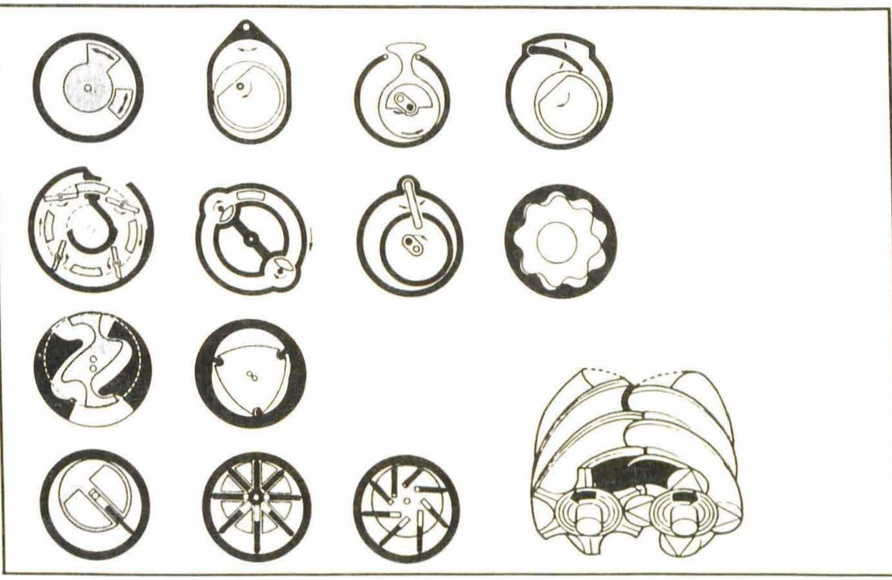

Figure 1. Some examples of rotary displacement machine concepts, as listed according to /9/.

Bild 1. Einige Rotationsverdichterbauarten, aus Ref. /9/.

3 Basic parameters and performance relations

The efficiency of displacement machines will to a great extent be determined by internal leakage losses, and throttling losses particularly in outlet and inlet ports. The fluid velocity in the machine, which is proportional to the peripheral speed of the rotor(s), will then be of major importance for the efficiency. Concerning mechanical design, shaft speed and rotor diameter (machine size) are of primary interest. Like for most types of rotary machines, smaller units should be operated at higher shaft speeds, as the optimum peripheral speed will usually be fairly size-independent. The shaft speed will then be critical particularly for smaller units, considering limitations for rolling-element bearings, driver (electric motor or engine) speed and drive gear. Very simple kinematics give the following fundamental relations:

$$\dot{V} = V\,n \tag{1}$$

$$u = \pi\,D\,n \tag{2}$$

or, solving for shaft speed and rotor diameter:

$$n = \sqrt{\frac{(V/D^3)\, u^3}{\pi^3\, \dot{V}}} \sim \frac{u^{3/2}}{\dot{V}^{1/2}} \tag{3}$$

$$D = \sqrt{\frac{\pi\, \dot{V}}{(V/D^3)\, u}} \sim \frac{\dot{V}^{1/2}}{u^{1/2}} \tag{4}$$

The specific displacement V/D^3 will then be a dimensionless geometry parameter, independent of machine size, but characteristic for one geometry class or concept. For compressors with eccentric motion and non-rotating rotors, like the scroll type in fig. 2, these equations will be modified as follows (for ordinary rotating machines, the eccentricity parameter will equal unity; $2e/D = 1$):

$$n = \sqrt{\frac{[V/(2e)^3]\, u^3}{\pi^3\, \dot{V}}} \tag{5}$$

$$D = \frac{1}{(2e/D)} \sqrt{\frac{\pi\, \dot{V}}{[V/(2e)^3]\, u}} \tag{6}$$

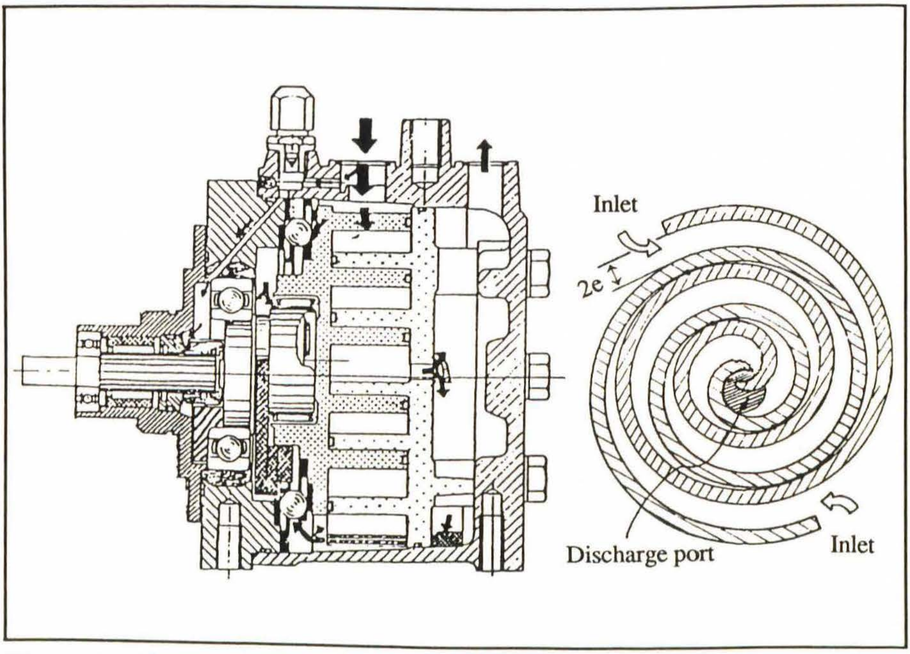

Figure 2. Scroll compressor, eccentric motion. /6/, /10/.
Bild 2. Spiralverdichter, mit translatorischer Kreisbewegung. /6 /, /10/.

Using these relations, shaft speed—capacity performance maps could be plotted for any geometry concept, with tip speed and rotor diameter (size) as parameters, fig 3. All relations will be represented as straight lines in a log-log graph. Dependent on its specific capacity, a concept is then found to be of low speed or high speed type.

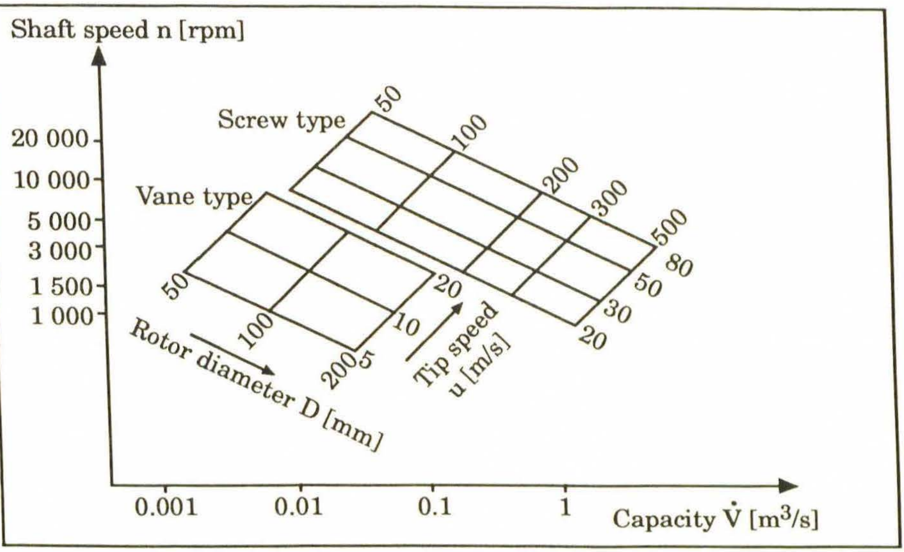

Figure 3. Shaft speed—capacity maps for some geometry concepts.
Bild 3. Kennfelder Drehzahl—Fördervolumen, für einige geometrischen Bauarten.

4 Optimum performance

When designing a new type of machine, the optimum tip speed will not be known in advance. The performance analysis could then be further extended by relating fluid flow/thermodynamic losses to design parameters and determining optimum tip speed. Rotary compressors are usually clearance-sealed, i.e. without seal elements between rotors and casing, but could be sealed, cooled and lubricated by injected oil or water (not covered by this paper, for simplicity). The two major tip speed dependent loss components for dry (non-injected) compressors will be internal leakage losses in clearances, and throttling losses in outlet and inlet ports. For a given concept, the gas velocity at different locations within the machine will be proportional to rotor tip speed. As the internal leakage flow is fairly independent of tip speed, and primary flow (i.e. capacity) is proportional to tip speed, the leakage loss (dependent on leakage/primary flow ratio) will be reduced with increased speed. On the other hand, throttling losses are (approximately) proportional to fluid velocity squared and will hence increase with tip speed. The optimum tip speed for any concept will then be found from these two loss components /1/, fig. 4.

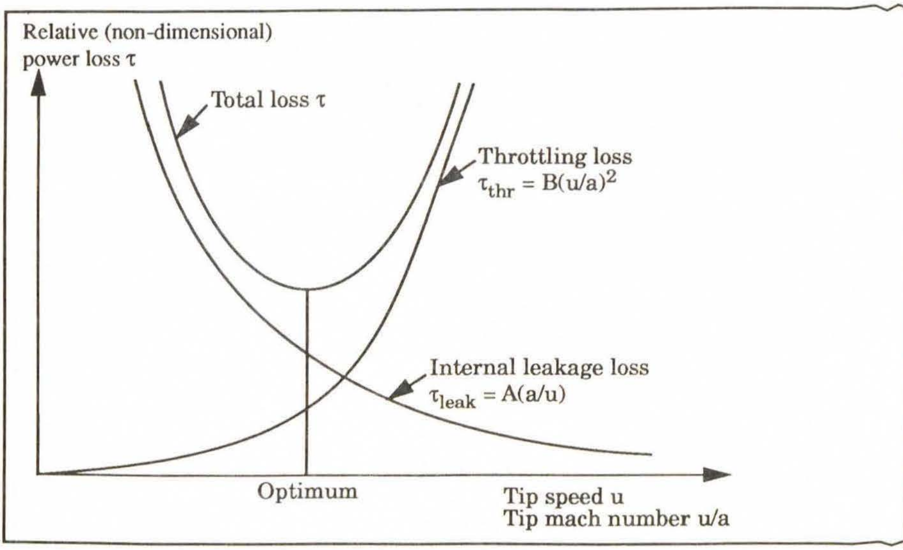

Figure 4. Major speed-dependent losses for non-injected displacement compressors.
Bild 4. Hauptsächliche geschwindigkeitsabhängigen Verluste für trockene Verdrängerverdichtern.

The loss components could be expressed by non-dimensional (size-independent) coefficients A and B, which are functions of geometry parameters.

The leakage loss factor A depends on the specific displacement V/D^3, relative leakage path lengths L_{ij}/D, internal pressure ratios Π_{ij} and Π_j (which are in fact geometry dependent), and relative clearance δ/D. Leakage will be proportional to clearances at running conditions, i.e. influenced by thermal expansion as well as shaft and rotor deflections. For a first rough estimation and comparison of performance for different concepts, cold clearances could be utilized. The throttling loss factor B depends on specific displacement, relative port area A_k/D^2, and stage pressure ratio Π_o.

For screw compressors, all geometry parameters like specific displacement, relative leakage path lengths and relative port areas will be related to the rotor profile geometry parameters and the rotor length/diameter ratio. Primary rotor profile parameters are number of male and female lobes, and relative lobe depth. The rotor profile curve type is also of importance for the specific displacement, the leakage path lengths and possible blowhole area. Approximately, the specific displacement for a screw compressor will be:

$$V/D^3 = (L/D)\, m \quad (A_M/D^2 + A_F/D^2) \tag{7}$$

where A_M and A_F represent the cross sectional areas of one male and one female rotor groove.

For a compressor with eccentric motion, e.g. the scroll type, loss factors A and B will also be influenced by the eccentricity e/D.

In the following analysis, second-order influence, i.e. losses on losses, have been neglected. The relative leakage power loss should be based on lost compression work for internal leakage /1/. Using equations for one-dimensional isentropic nozzle flow, the leakage mass flow between an up-stream rotor cell "i" and a downstream rotor cell "j" (fig 5) will be:

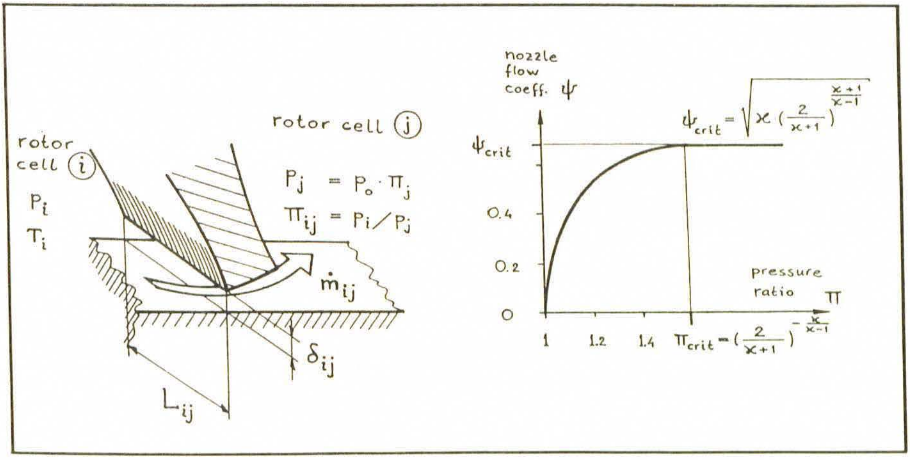

Figure 5. Internal leakage mass flow between rotor cells.
Bild 5. Leckage (Massendurchfluß) zwischen Arbeitsräumen.

$$\dot{m}_{ij} = \alpha_{ij}\, L_{ij}\, \delta_{ij}\, \psi_{ij}(\Pi_{ij})\, p_i/\sqrt{R\,T_i} \tag{8}$$

where, neglecting all secondary losses, the upstream pressure p_i will be:

$$p_i = p_o\, \Pi_j\, \Pi_{ij} \tag{9}$$

The internal pressure ratios Π_j and Π_{ij} will be determined by geometry as well as by overall stage pressure ratio Π_o. The upstream temperature T_i, assuming polytropic compression with the polytropic exponent ν will be:

$$T_i = T_o\, (\Pi_j\, \Pi_{ij})^s \quad ; \qquad s = (\nu - 1)/\nu \tag{10}$$

Using equations for polytropic recompression of all internal leakage and putting the equations in non-dimensional form, we then get the following expression for the leakage loss, where a is the sonic velocity at reference conditions:

$$\tau_{leak} = \frac{\pi\, \sigma \sum_{j=1}^{q-1} \sum_{i=j+1}^{q} \alpha_{ij}\, (L_{ij}/D)\, (\delta_{ij}/D)\, \psi(\Pi_{ij})\, \Pi_j^{1+s/2}\, \Pi_{ij}^{1-s/2}\, (\Pi_{ij}^{s}-1)\, (2e/D)}{\sqrt{\kappa}\; s\, \eta_{vol}\, (V/D^3)\, (\Pi_o^{\sigma}-1)} \cdot \frac{a}{u} \tag{11}$$

Average values over the compression cycle should be used for the different leakage path lengts L_{ij}/D, the clearance discharge coefficients α_{ij} and the internal pressure ratios Π_{ij}, Π_j. These average values could be rough estimates, or could be based on detailed simulations of the compression cycle. It should be accounted for, that some of the leakage paths are active during part of the compression cycle only. The volumetric efficiency η_{vol} is an empirical constant, that could be estimated, or neglected, as a first approximation.

Neglecting external heat transfer, as well as heating due to internal losses, the polytropic exponent ν could be approximated by the specific heat ratio κ.

The actual stage pressure ratio Π, considering throttling losses in in- and outlet, will approximately be:

$$\Pi = \Pi_o \left(1 + \frac{\Delta p_{in}}{p_o} + \frac{\Delta p_{out}}{p_{out}}\right) \tag{12}$$

The throttling losses in outlet and inlet ports could be estimated roughly, e.g. by using empirical throttling coefficients Z:

$$\Delta p_k = Z_k \, \rho_k \, c_k^2/2 \tag{13}$$

The relative throttling power loss will then be:

$$\tau_{thr} = \frac{{\eta_{vol}}^2 \, (\kappa-1)(V/D^3)^2}{2\,\pi^2\,(2\,e/D)^2(\Pi_o^{\sigma}-1)} \left[Z_{in}\,\Pi_o^{s}\,\frac{(2\pi/\phi_{in})^2}{(A_{in}/D^2)^2} + Z_{out}\,\Pi_o^{-2/\nu}\,\frac{(2\pi/\phi_{out})^2}{(A_{out}/D^2)^2}\right] \left(\frac{u}{a}\right)^2 \tag{14}$$

Port areas and throttling coefficients should be average values for the compression cycle. The port opening angle ϕ accounts for, that ports are open during part of the compression cycle only.

The total speed dependent loss (dimensionless) for arbitrary concepts could now be expressed as:

$$\begin{aligned} \tau = {} & A(V/D^3, L_{ij}/D, \Pi_j, \Pi_{ij}, \Pi_o, \alpha_{ij}, \kappa, \nu, \eta_{vol}, \delta/D, e/D) \cdot a/u \; + \\ & + B(V/D^3, A_k/D^2, \phi_k, Z_k, \Pi_o, \kappa, \nu, \eta_{vol}, e/D) \cdot (u/a)^2 \end{aligned} \tag{15}$$

The first term represents internal leakage power loss, the second represents throttling power loss, and a is the sonic velocity of the fluid.

Optimum tip speed u_{opt} will then be determined as follows, and is a function of the design parameters in dimensionless form, which have typical values for each geometry class, as described above.

Derivation gives optimum tip speed:

$$\frac{\partial \tau}{\partial u} = 0 \quad \Rightarrow \quad \begin{cases} u_{opt} = a \sqrt[3]{\dfrac{A}{2B}} & (16) \\ \tau_{min} = 3 \sqrt[3]{A^2 \, B/4} & (17) \end{cases}$$

Losses should rather be expressed by the (isentropic) efficiency η_s:

$$\eta_s = \frac{1}{\theta + \tau} = \frac{1}{\theta + A(a/u) + B(u/a)^2} \quad (18)$$

where θ is the ratio between polytropic and isentropic compression power:

$$\theta = \frac{\sigma}{s} \cdot \frac{\Pi_o^s - 1}{\Pi_o^\sigma - 1} \quad (19)$$

As the leakage loss factor A is linear in relative clearance δ/D, we can now derive expressions for optimum tip speed, optimum efficiency $\eta_{s,opt}$, and efficiency η_s at non-optimum tip speed, related to the relative clearance δ/D:

$$\text{optimum performance:} \quad \Rightarrow \quad \begin{cases} u_{opt} = \text{const} \cdot (\delta/D)^{1/3} & (20) \\ \eta_{s,\,opt} = \dfrac{1}{\theta + 3\,(A^2B/4)^{1/3}} = \dfrac{1}{\theta + \text{const} \cdot (\delta/D)^{2/3}} & (21) \end{cases}$$

efficiency at non-optimum speed u:

$$\eta_s = \frac{1}{\theta + \text{const} \cdot (\delta/D)\,(a/u) + B \cdot (u/a)^2} \quad (22)$$

where all constants are functions of the non-dimensional design parameters, with specific values for each geometry concept. Efficiency versus tip speed can be represented in the graph according to fig. 6. If the minimum acceptable efficiency level η_{min}, as well as the relative clearance δ/D are specified, the acceptable speed range (i.e. capacity range for one size of machine) can then be determined from the graph. This is of interest, when optimizing the number of machine sizes to cover a specified capacity range. There will obviously be a trade-off between manufacturing costs (number of sizes to be used) and operating costs (efficiency). It should be pointed out, however, that the efficiency used for this analysis represents only the speed-dependent loss components. This efficiency value could therefore not be directly compared to measured performance.

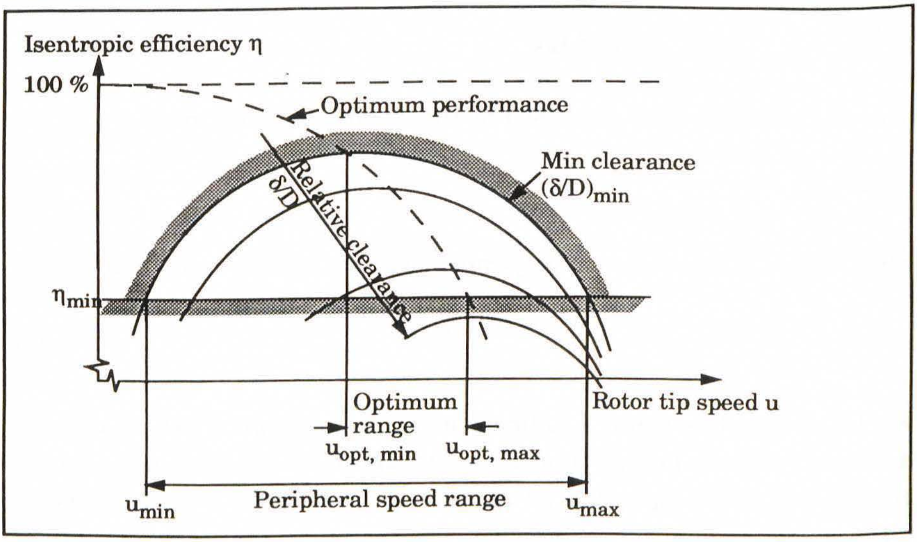

Figure 6. Isentropic efficiency versus peripheral speed for one geometry concept, with relative clearance δ/D as parameter.

Bild 6. Gütegrad im Vergleich der Umfangsgeschwindigkeit mit relativer Spalthöhe als Parameter, für eine Verdichterbauart.

5 Performance maps

To plot the performance map for one geometry class, however, shaft speed versus capacity will be more interesting as a basis for the choice of concept and for the detailed design work. Using previous relations, we get:

$$n_{opt} \sim (V/D^3)^{1/2} \cdot (\delta/D)^{1/2} \cdot \dot{V}^{-1/2} \tag{23}$$

As there is usually a scale influence on clearances, the assumption of relative clearance δ/D as a parameter, independent of machine size, will not be very realistic. If instead using absolute clearance as a size independent parameter, the optimum shaft speed expression will be modified as follows:

$$n_{opt} \sim (V/D^3)^{4/5} \cdot \delta^{3/5} \cdot \dot{V}^{-4/5} \tag{24}$$

The most realistic assumption should be to assume a power relationship between clearance δ and machine size (rotor diameter) D:

$$\delta = \delta_o \cdot (D/D_o)^{\alpha} \tag{25}$$

Within the diameter range of interest, $\alpha = 1/3$ represents fairly well the size influece for the grades of tolerance according to ISO. The optimum shaft speed expression will then be modified as follows:

$$n_{opt} \sim (V/D^3)^{\frac{4-\alpha}{5+\alpha}} \cdot \frac{\delta_o^{\frac{3}{5+\alpha}}}{D_o^{\frac{3\alpha}{5+\alpha}}} \cdot \dot{V}^{\frac{\alpha-4}{5+\alpha}} \tag{26}$$

All expressions for optimum and non-optimum performance, based on dimensionless design parameters that represent any geometry concept, can now be used to plot the shaft speed—capacity map for an arbitrary geometry class. Constant relative or absolute clearance, as well as a functional size-clearance relationship can be used. Efficiency levels, as well as constant diameter (machine size) lines should be included. When using logarithmic scales, straight lines will be obtained except for the efficiency level curves. One example of a performance map plot is presented in fig. 7. This map has been computed for screw geometry with 4:6 lobes and compression of air from atmospheric pressure to 3 bar, assuming a functional clearance—rotor diameter relationship. It should be noted, that the efficiency used for this optimization only represents the speed dependent loss components. Loss components of more constant nature, like mechanical losses in bearings, gears and seals, have not been included.

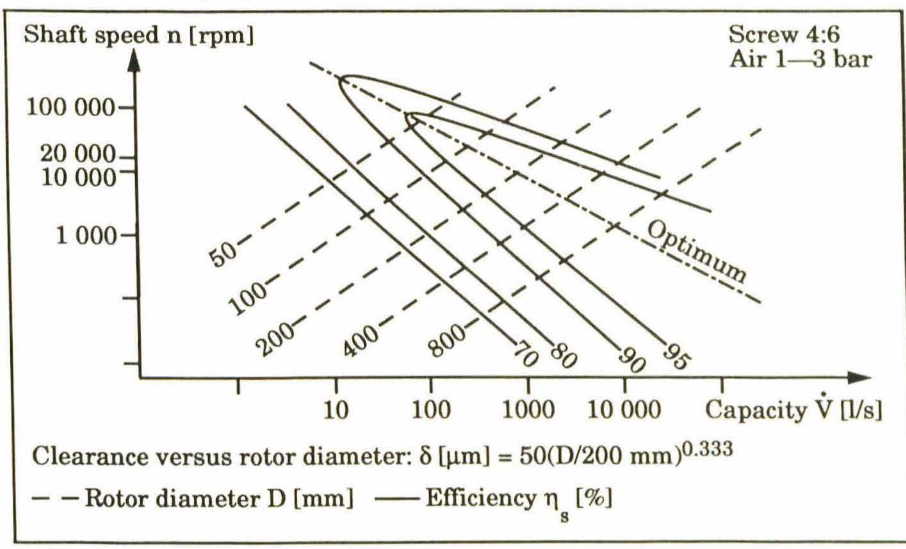

Figure 7. Performance map shaft speed—capacity for one geometry class, with constant rotor diameter lines and constant efficiency levels indicated.

Bild 7. Kennfeld Drehzahl—Fördervolumen für eine Verdichter-bauart, mit Hauptläuferdurchmesser als Parameter und bezogenen Linien konstanten Gütegrades.

As pointed out, during the conceptual design phase, simplified expressions that give rough estimates only of the geometry parameter mean values, could be utilized. In this way, performance maps accurate enough for a first comparison of different concepts, could be computed rather quickly. Parameters like leakage path lengths and internal pressure ratios (rotor cavity volume ratios) will vary during the compression cycle, however, and are functions of the rotor turning angle (or time). For a refined analysis, these parameters could be determined by means of a more detailed computer simulation for each type of geometry concept studied. One example of a detailed simulation model for the screw type compressor has been presented in ref. /2/, /5/. For performance mapping however, the simulation model should be formulated so that non-dimensional output parameters will be obtained. In this way, the non-dimensional output parameter values from the simulation could be used as input parameters for the calculation of a general performance map for the concept studied.

6 Performance map boundaries

After having computed the general performance map for one specific concept as described, the feasible range of operation for this concept could be determined by specifying the minimum acceptable efficiency. The maximum deviation from optimum conditions and hence the width of the capacity range will then be determined. So far, the general performance map for the concept has been based on pure kinematic and thermodynamic relations only. To continue with the mechanical design, limitations set by mechanical components and by rotor size/manufacturing should now be introduced.

The shaft speed will be limited by the drive system to be used. For instance, when using standard AC motors of 2-pole or 4-pole type, corresponding shaft speed lines, i.e. 3 000 (3 600) and 1 500 (1 800) r.p.m. for 50 Hz (60 Hz), should be introduced in the map. These lines indicate direct electric drive conditions. However, for small rotary units, mechanical transmissions for speed increase are most often used. Transmissions could be of gear or pulley type. As an example, when using a single-stage offset cylindrical gear, the gear-ratio should normally be limited to approximately 10:1. From this maximum gear ratio and the electric motor speed, the maximum shaft speed of the compressor unit will be defined.

If rolling-element bearings are required to minimize friction losses and to simplify the lubrication system, another limit will be defined by the maximum acceptable bearing speed. This limit will be expressed by the $n \cdot D$ product for the bearing, which is dependent on type of bearing as well as of lubrication.

Another limit representing maximum acceptable rotor diameter, i.e. unit size, should also be introduced.

This limit depends primarily on manufacturing accuracy and cost, and will therefore be specific for each geometry concept, and also dependent on the manufacturing method used for the rotor. This is not a well-defined specific limit, but a rough estimation should usually be possible to define.

After introduction of all these boundaries, the performance map will look like the example in fig. 8.

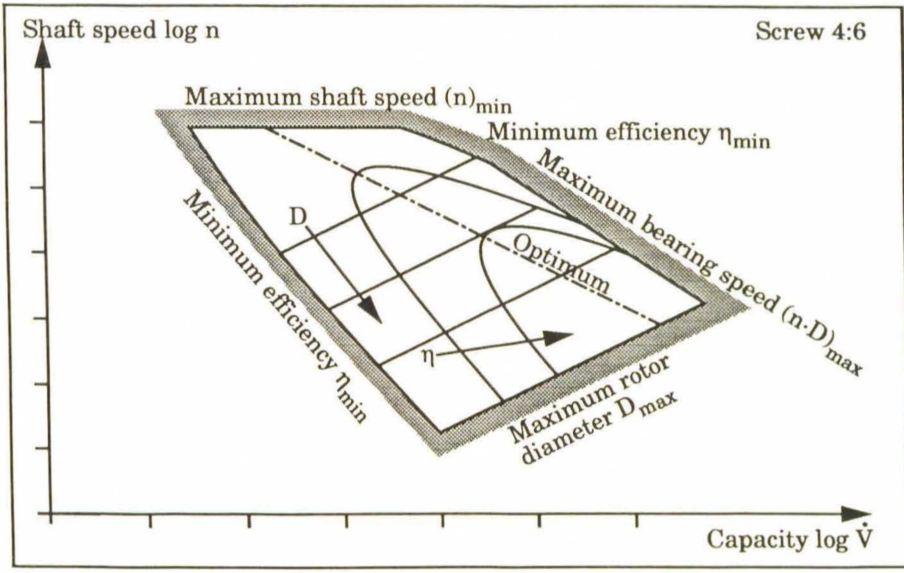

Figure 8. Performance map shaft speed—capacity for one geometry class, with various boundaries introduced.
Bild 8. Kennfelder Drehzahl—Fördervolumen für eine Verdichterbauart, mit verschiedenen Begrenzungen.

7 Comparison of various concepts

Finally, after having established the performance maps for a number of concepts of interest, a comparison should be performed. All maps for the different concepts studied should be plotted in a common graph. One example for air compressors is presented in fig. 9. From this type of diagram, it can easily be seen which concepts should be best adapted for different capacity regions. If a wide range of capacity is to be covered, different concepts must certainly be used for low capacity and high capacity units. For the high capacity end, compact, "high-speed" machines like the screw compressor, with a high specific displacement and "small" diameter should be preferred. For very low capacity, "low-speed" concepts should be preferred to minimize mechanical problems. The scroll compressor is particularly feasible for low capacity. The mechanical sliding speed of the scroll will be determined by eccenter radius and shaft speed, while the sealing point velocity and hence the gas velocity will be determined by scroll radius and shaft speed. Gas velocity could then be high, despite a low mechanical speed and low shaft speed. In the lowcapacity end, unit size/diameter is usually of less importance.

For the intermediate capacity range, concepts like tooth compressors, and screw compressors with few lobes, are more interesting in order to avoid high shaft speed.

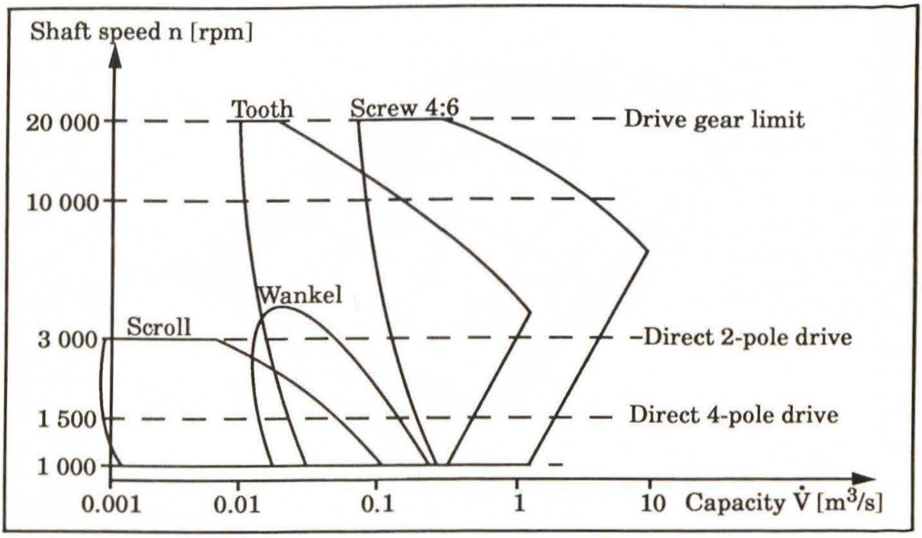

Figure 9. Example of a common performance map for a number of different geometry concepts.
Bild 9. Gemeinsame Kennfelder einiger Verdichterbauarten.

It must be remembered, however, that this type of analysis should only be used as a rough guide for selection of concept candidates for further evaluation and detailed design. Many important issues, like for instance bearing loads and seal configuration, have not been covered by this simplified analysis. As in all design, the performance/cost ratio must be evaluated and compared to existing technology. The type of analysis described in this paper could be used iteratively though, and boundaries as well as the level of refinement of the analysis could be changed for each iteration loop.

8 References

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