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ABSTRACT

The refrigerants being used in vapor compression processes have specific thermodynamic properties, which are decisive for the performance of the compressor of the system. The Montreal and the Kyoto protocols initiated a discussion of alternative refrigerants, which lead to new requirements of the compressor.

In the paper, the relevant properties of the refrigerants are identified and compared. Furthermore, the characteristic design features of positive displacement compressors are discussed. By combining the information about the refrigerant properties and state of the art positive displacement compressors, advantageous compressor designs are recommended for the individual refrigerants and applications. The analysis focuses on the impact of the refrigerant propertied on the selection and the design of the compressor components, which have an impact on the thermodynamic compression process including the suction, the compression and the discharge of the gas.

INTRODUCTION

Vapor compression processes are dominating throughout the refrigeration technology. The technical realization of such processes requires basically four key components: a compressor, two heat exchangers - one on the high and one on the low-pressure side - and a throttling device. The compressor is the component with the major influence on the performance and the reliability of the entire refrigeration system. Compressors of the positive displacement type are widely used in these types of applications, while there are various specific compressor concepts known. Each design has characteristic features matching the requirement of a certain application and the choice of refrigerant in a more or less sophisticated way.

Due to environmental considerations, a number of alternative refrigerants for the traditionally applied CFCs have recently been introduced. The perspective of using these new fluids was to reduce or even avoid the harmful impact of CFC emissions on the ozone layer and the global warming. Some of these new refrigerants, like for example R134a, were developed to have a rather similar performance like the CFCs, which were to be replaced. Other alternatives like especially CO_2 have significantly different properties offering new opportunities and challenges at the same time.

Furthermore to minimize the overall environmental impact of refrigeration installations, a high efficiency and reliability of the entire refrigeration system is required. This is only achieved, if the best technology using the optimal refrigerant and optimally designed components including compressors is used for each application field.

The selection of a suitable compressor design for a certain application and refrigerant is performed in two steps. In the first step, the compressor type needs to be identified that has the highest potential to match the specific requirements given by the actual application and refrigerant. In the second step, the individual geometry parameters of the compressor have to be optimized to achieve the best possible performance of the entire thermodynamic compression process.

REFRIGERATION COMPRESSORS OF THE POSITIVE DISPLACEMENT TYPE

Refrigeration compressors pump the refrigerant from the low process pressure against the high process pressure. Today, positive displacement compressors and turbo compressors are applied. This paper does not refer to turbo compressors, but rather concentrates on positive displacement ones. Since the first realization of the vapor compression process, which basically used a reciprocating positive displacement pump as the compressor, a variety of other positive displacement mechanisms were introduced to the mentioned field of technology.

Compressor types

As shown in ***Figure 1,*** the relevant compressors of the positive displacement type may be classified into a group of compressors, which have reciprocating pistons and another group of compressors being characterized by rotating pistons or displacers.

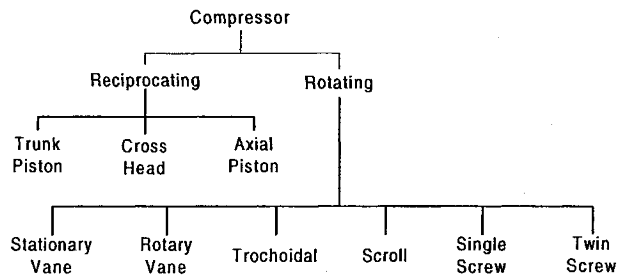

Figure 1 *Classification of compressors of the positive displacement type*

At all positive displacement machines the periodical movement of the piston or the displacer causes compression process including the suction, the compression and the discharge of the gas. Often, pressure driven self acting valves are applied for the timing of the suction and the discharge phase. Such valves ensure, that the pressure of the gas inside the compression chamber is adapting to the pressures of the refrigeration process. Furthermore, it is possible to design an in- and outlet with an opening that depends on the piston position. Choosing this concept, the chamber pressure at the opening to the suction or the discharge side may differ from the corresponding refrigeration process pressure, resulting in efficiency losses. On the other hand, the piston position controlled in- and outlet port allows bigger flow areas, what reduces the flow losses.

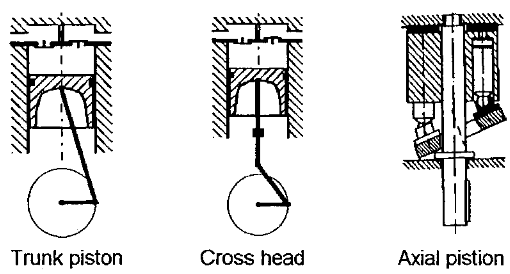

Figure 2 *Compressors with reciprocation pistons*

Reciprocating compressors are as trunk piston, cross head or axial piston machines. The different designs, which are shown in ***Figure 2***, differ principally just in the design of the driving mechanism of the pistons. The piston of trunk piston compressors is directly connected to the connection rod putting a radial force on the piston. Compressors with a cross head have no radial piston forces as they are compensated by the driving mechanism. Axial piston compressors are designed as wobble or swash plate compressors, where the variation of plate angle may allow controlling the displacement of the compressor. Reciprocating compressors are normally equipped with self-acting pressure driven valves at the gas in- and outlet. At compressors for higher capacities, piston rings seal the cylinders whereas small hermetic compressors rely on reducing the narrow gap between piston and the cylinder wall.

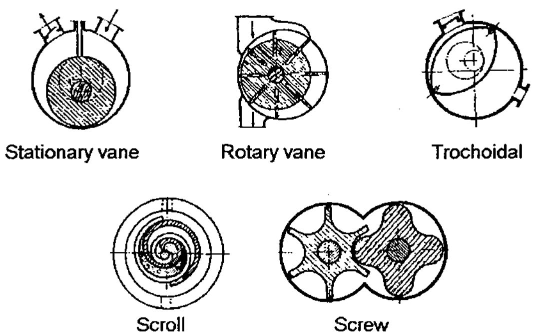

Figure 3 *Compressors with rotating pistons*

The refrigeration compressors with rotating pistons are shown in ***Figure 3***. Typically for all of them is, that the periodical chamber volume variation is achieved by rotating or orbiting displacers, which are arranged single or in pairs in their housing.

Compressors with rotating displacers are normally designed with an inlet controlled by the piston position and an outlet, which may either be controlled also by the piston position or a self-acting pressure controlled valve. Normally, no particular sealing elements are applied, and the compressors components forming the compression chamber are manufactured with high accuracy. Oil is lubricating the compressor's drive and reducing the gas leakage.

Thermodynamic compression process and its efficiency

To compress the gas from the suction pressure p_s to the discharge pressure p_d, the gas at suction conditions enters the compression chamber, until its maximum volume is reached. Reducing the chamber volume again to its minimum value, the gas pressure is increasing until reaching the high pressure. At this moment a pressure controlled discharge valve opens to allow the discharging of the gas from the chamber. If the outlet is controlled by the piston position, the chamber pressure will equalize with the high pressure as soon as the chamber is connected to the discharge line. In this paper "discharge" and "suction line" refers to the internal flow paths (pipes) in the compressor

For the described compression process, a real compressor requires the indicated compression work w_i, which is equivalent to the ring integral of the real process pressure of an entire process. Due to throttling losses during suction and discharge, compression chamber leakage and heat transfer effects, the work w_i exceeds the isentropic compression work w_t. The isentropic compression work is defined as the work required to compress the gas from point 1 at constant entropy up to point 2 at the discharge pressure. Point 1 is defined in a way, that the ideal compressor has the same suction volume $V_{g,id}$ as the real compressor. The real and the ideal isentropic compression are shown in ***Figure 4***, in a p,V-diagram of a compression process.

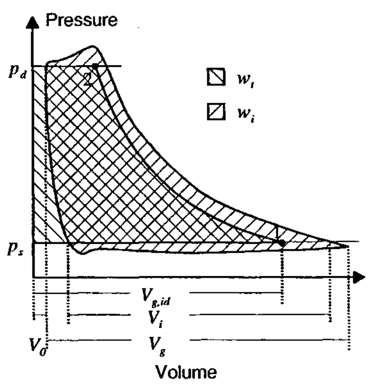

Figure 4 *Theoretical and real indicated compression process*

The isentropic compression process efficiency may be calculated from the diagram as $\eta_i = w_t / w_i$. It is defined as the ratio between the theoretical work w_t and the indicated work w_i. The volumetric efficiency of a positive displacement compressor $\lambda = V_i / V_g \cdot \eta_{th}$ is calculated using the volume V_i that is actually filled with refrigerant at the end of the suction process, the geometric compression chamber volume V_g and the thermometric efficiency η_{th}. The actual filled volume V_i is mainly influenced by a possible re-expansion of the gas from a clearance volume and the suction port closing at the end of the suction process. The thermometric efficiency η_{th} represents the reduction of the discharged refrigerant mass due to leakage effects and a specific volume increase of the refrigerant in the suction line of the compressor due to heat transfer. The thermometric efficiency η_{th} can not be derived from a p,V-diagram.

IMPACT OF REFRIGERANT PROPERTIES ON THE COMPRESSOR

Presently, there are a number of alternative refrigerants for vapor compression processes under discussion, and a final globally accepted commitment to a set of favorable fluids for the different applications has not emerged yet. As stated before, the fluid properties of the refrigerants differ significantly.

Table 1 Condensation and evaporation temperatures of various standardized rating points

Standard	Condensation/evaporation temp. in ° C LBP	MBP	HBP
ISO	40,0/-35,0	45,0/-10,0	-
ASERCOM	40,0/-35,0	-	50,0/5,0
CECOMAF	55,0/-25,0	55,0/-10,0	55,0/5,0
ARI	40,6/-31,7	48,9/-6,7	54,4/7,2
ASHRAE	54,4/-23,3	54,4/-6,7	54,4/7,2

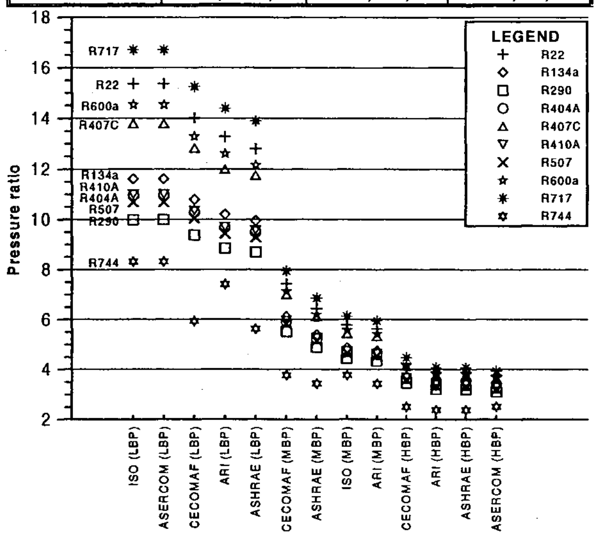

Figure 5 Pressure ratio for various refrigerants at various rating points

Looking for more environmentally benign refrigerants for vapor compression processes, the CFCs are phased out due to their Ozone Depletion Potential (OPD) and a high Global Warming Potential (GWP).

The HCFCs alternatives have a lower ODP than the CFCs, but they are only regarded as an interim solution. Furthermore, the application of HFCs is critically discussed in some parts of the world, as these fluids still have a significantly high GWP. Natural refrigerants with no ODP and only negligible GWP may have other unfavorable properties like toxicity and flammability. The ongoing dynamic discussion of various technical options changes the focus rather quickly on different scenarios. This brings about that the main goal of choosing the optimal refrigeration system and compressor design from an environmental point of view is not always fulfilled.

The suitability of a compressor for a certain application depends – as stated before - on the selected refrigerant. ***Table 1*** shows the condensation and evaporation temperatures for various standardized rating points for applications with low (LBP), medium (MBP) and high (HBP) back pressure (evaporation temperature).

The individual pressure ratios for different refrigerants at these rating points can be seen in ***Figure 5***, while for R744 (CO_2) a high pressure of 10 MPa is assumed in all calculations, if the temperature of the rating point leads to a super critical pressure.

<u>**Impact of suction gas temperature increase and pressure drop on the compression chamber filling**</u>

A temperature increase or a pressure decrease of a refrigerant results in an increase of the specific volume reducing the volumetric efficiency of the compressor. The volumetric efficiency again influences the capacity and the efficiency of the compressor.

On the path of refrigerant through a compressor to the compression chamber, both effect occur in parallel. Heat transfer phenomena have only a significant impact on the specific volume of the suction gas, before it enters the compression chamber, while the compression process itself is to short in time and to low in temperature to allow substantial heat transfer. The pressure drop of the suction gas occurs between the inlet of the suction line and the pressure inside the compression chamber at the end of the suction phase.

The impact of a temperature increase or a pressure decrease on the specific volume differs for the various refrigerants, if comparing a certain rating point. In ***Figure 6*** the refrigerant mass inside the compression chamber is shown as a function of the superheat defined as the difference between the evaporation temperature and the gas temperature inside the compression chamber at the end of the suction phase. ***Figure 7*** shows the refrigerant mass inside the compression chamber as a function of the difference between the evaporation pressure and the chamber pressure at the end of the suction.

When comparing the figures, it is obvious that the reduction of the refrigerant mass inside the compression chamber due to a temperature increase is more significant for these kinds of fluids, which lead to low process pressures. Such a refrigerant is R600a. On the other hand it is common sense that these low-pressure fluids are more sensible to the same pressure drop, than refrigerants with a higher pressure at the same rating point. Refrigerants leading to higher process pressures are R410A and especially R744.

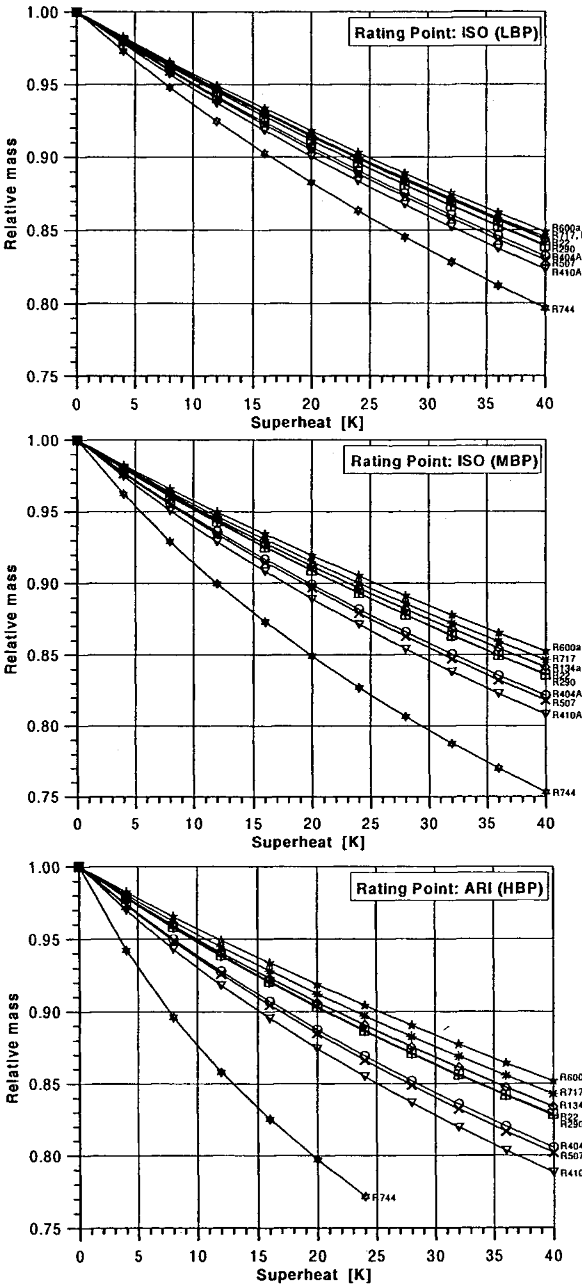

Figure 6 *Refrigerant mass inside the compression chamber as a function of the temperature increase until the end of the suction process at various rating points*

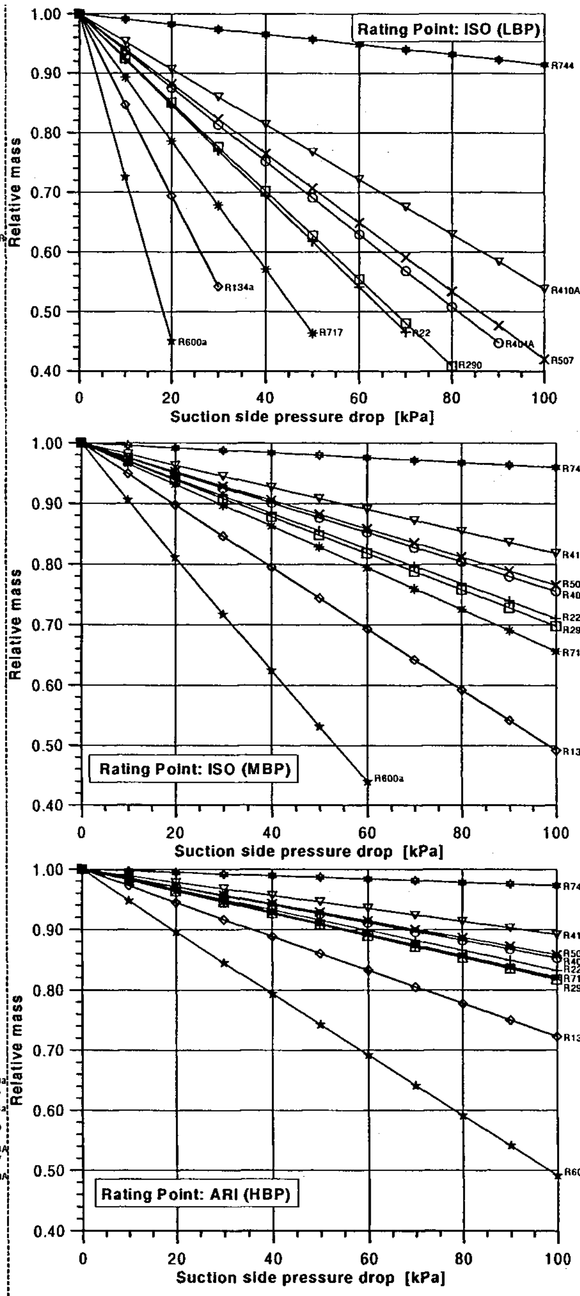

Figure 7 *Refrigerant mass inside the compression chamber as a function of the suction gas pressure drop at various rating points*

It is obvious that the temperature increase and the pressure drop of the suction gas having entered the compression chamber should be minimized. On the other hand, from practical considerations, it is an optimization between suction gas superheat and pressure drop. When reducing the flow area of the suction line of a compressor, superheat will decrease and pressure drop will increase. When minimizing the occurring pressure drop of the suction gas, compressors of the rotating type are favorable. The superheat may be limited by short, direct and insulated suction line

line to the compression chamber. Hermetic type compressors, where the suction gas is used for motor cooling should possibly be equipped with alternative cooling concepts, if the applied refrigerant is extremely sensible to superheat.

Impact of pressure ratio and pressure difference on the compression process efficiency

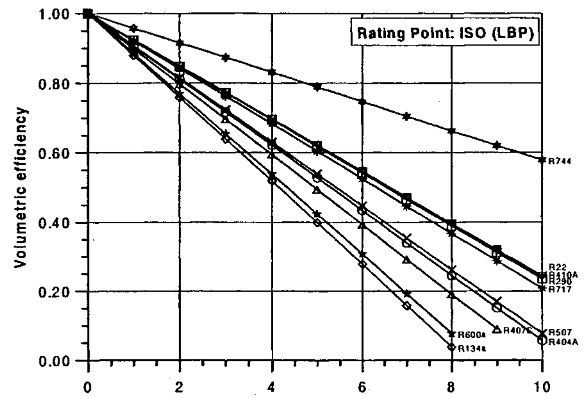

Figure 8 Volumetric efficiency as a function of the clearance volume of the compression chamber

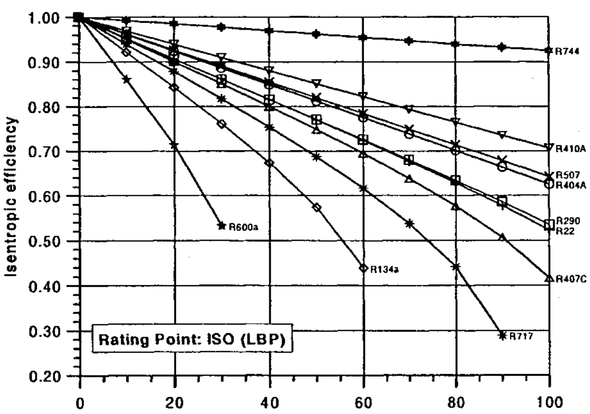

Figure 9 Isentropic efficiency as a function of the pressure drop of the suction and the discharge line

The analysis of the impact of different refrigerant properties on a compressor's performance depends on the operation conditions. To show the trend of the refrigerant impact when applying high, medium and low-pressure refrigerants for certain applications, two sets of calculations were performed.

Figure 8 shows the impact of the clearance volume ratio on the volumetric efficiency of a compressor. The calculation is performed for the conditions at the ISO LBP rating point, while pressure drop of the suction and discharge gas as well as suction gas superheating are neglected. The high pressure for R744 is 10 MPa. The volumetric efficiency of a compressor depends on the size of the clearance volume and is decreasing with a rising pressure ratio, which depends on the rating point and on the refrigerant. A re-expansion does not only reduce the volumetric efficiency of the compressor, but it also affects the mechanical performance, as it results in an increase of the average process pressure. Therefore, an optimized compressor especially designed for a high compression ratio should have a minimized clearance volume. Compressors with rotating pistons are favorable for such applications, as a re-expansion from the clearance volume, if it occurs at all, does not reduce the volumetric efficiency of the compressor.

The impact of a pressure drop of the suction and the discharge gas on the compressor's isentropic efficiency is shown in ***Figure 9***. The calculation is performed for the ISO LBP rating point, while suction gas superheating is neglected. The high pressure for R744 is 10 MPa. The pressure losses of the suction line are assumed to be throughout the simulation 50% of these of the discharge line. Obviously, it is especially important to reduce pressure loss when applying low-pressure refrigerants.

Impact of compression chamber leakage

The performance of compressors suffers from any leakage occurring from the high-pressure side back to lower pressure /1/. Leakage occurs at the valves, if being applied, and the gaps of the compression chamber. While leakage at the valves is less critical and may be reasonably limited with an appropriate valve design, the amount of leakage of the gaps of the compression chamber depends highly on the compressor concept. Furthermore, it depends on the compressor design, whether the gas leaks directly to the suction side or to a succeeding compression chamber with a lower pressure: Leakage at the piston/cylinder gap of reciprocating compressors with the suction pressure in the crankcase, takes the gas back to the suction side, while the internal leakage of compressors with rotating pistons is directed to a following compression process.

Leakage out of the compression chamber back to the suction side is equivalent to a total loss of the energy, which was required to compress the gas to that pressure, when the leakage occurs. Leakage directly into a succeeding compression chamber has an even worse impact. As the additional disadvantage the energy absorbed by the compressed gas is partly transported over to the succeeding compression process. This energy transfer results in an increase of discharge temperature of the gas, being equivalent to a decrease of the isentropic efficiency. This kind of leakage occurs at compressors with rotating pistons, while the actual amount of leakage depends on the pressure difference at the relevant gap. Stationary vane compressors for example have the full difference between the process pressures at the gap between the vane and the piston, while at screw and scroll compressors the leakage is conducted to a succeeding compression chamber with a pressure above the suction pressure.

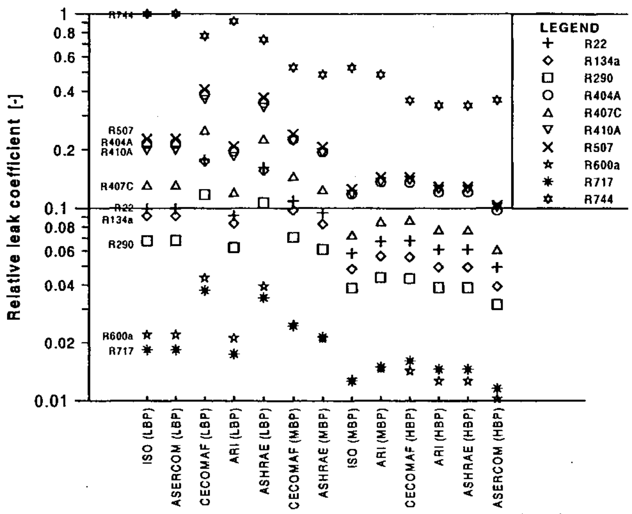

Figure 10 Coefficient for the compression chamber leakage as a function of the evaporation temperature

The refrigerant mass m leaking though a narrow gap depends in the following way on the refrigerant properties and the gap geometry /2/:

$$m = C_{ref} \cdot \frac{2 \cdot U \cdot h^3}{l} \quad \text{with } C_{ref} = \frac{p_1^2 - p_2^2}{\mu \cdot p_1 \cdot v_1}$$

The circumference U, the height h and the length l in the dimension of the leakage flow characterize the gap's geometry. The leak coefficient C_{ref} describes the impact of the refrigerant properties. It is calculated with the pressure p_1, the specific volume v_1 and the dynamic viscosity μ before the gap and the pressure p_2 after the gap. In ***Figure 10*** C_{ref} is shown for different rating points related to the highest occurring value. Refrigerants with a large C_{ref}, require a minimization of the compressor's leakage gap to achieve a competitive performance. Reciprocating compressors with piston rings and a suitable stroke-to-bore ratio offer the best perspective to minimize the internal leakage /3, 4/.

COMPRESSORS FOR VARIOUS REFRIGERANTS AND APPLICATIONS

The efficiency of the compression process is influenced by the thermodynamic properties of the applied refrigerant. To obtain the maximum performance of a vapor compression process, a compressor design has to be selected, which offers the best perspective to match the requirements given by the application and the selected refrigerant. In ***Table 2*** recommendations for the selection of the compressor concept are given sorted by application and capacity. The results shown in the table are based on the refrigerant issues discussed in the paper. Some further features, like for example the discharge temperature, the volumetric capacity, the oil miscibility/solubility, gas pulsations and safety aspects, which are also influencing the selection of the refrigerant and compressor design, were disregarded here. Furthermore, there are arguments for a final compressor's selection, which are not based on thermodynamic or technological aspects. In many cases alternative combinations of application, refrigerant and compressor are applied, but sometimes this may result in disadvantages regarding the compressors performance.

Additionally to the selection of the compressor concept, the design characteristics need to be optimized for each focused application. It is for example possible to modify the number and the geometry of the cylinders as well as the design of the suction and discharge lines of a reciprocation compressor to influence the impact of pressure losses, heat transfer and leakage on the compression process /4/.

Table 2 refers to compressor designs, as they are known as the state of the art. In the table, reciprocating compressors (recips) are treated as one family and it is depending on the application, in what specific design they are applied. For mobile applications, where a capacity variation by the adjustment of the stroke volume is used, axial piston compressors are preferred. Nevertheless, with a rising number of cylinders for the same stroke volume, the harmful impact of leakage on the compressor's performance is increasing.

Out off the group of compressors with rotating pistons, the focus is on screw, Scroll and stationary vane compressors (vane), which may also be designed as swing compressors. Basically it is possible to use rotary vane and Trochoidal instead of stationary vane compressors, as they have a comparable performance. They were not explicitly included here, as they have a lower mechanical efficiency than stationary vane compressors and/or a rather comprehensive design.

***Table 2** Recommendation of compressors for various refrigerants and applications*

	Application								
	Low temperature			**Medium temperature**			**Air Conditioning**		
	Compressor power input in kW			Compressor power input in kW			Compressor power input in kW		
Fluid	< 1	≤ 10	>10	< 2	≤ 20	> 20	< 3	≤ 30	> 30
R22, R134a	recip	recip	screw	recip, vane	recip	screw	vane	scroll	screw
R290	recip	recip	screw	recip	scroll	screw	vane	scroll	screw
R404A; R507	recip	recip	screw	recip	recip	screw	recip	scroll	screw
R407C	recip	recip	screw	vane, recip	recip	screw	vane	scroll	screw
R410A	recip	recip	screw	recip	recip	screw	recip	scroll	screw
R600a, R717	vane	vane	screw	vane	vane	screw	vane	scroll	screw
R744	recip	recip	recip	recip	recip	recip	recip	recip	recip

CONCLUSION

The suitability of the state of the art positive displacement compressors being applied for vapor compression refrigeration processes is different for the various refrigerants and applications. To achieve the maximum system performance, it is compulsory to apply a compressor that matches the specific requirements given by the refrigerant and application. The refrigerant features with a significant impact on the compressor's performance are process pressures as well as the temperature increase and the pressure drop of the suction gas before the compression starts.

Compressors with reciprocating pistons are advantageous, if the refrigerant tends to lead to high internal leakage, while compressors with rotating pistons are especially favorable for applications, where a pressure drop of the suction gas and a high pressure ratio result in a decrease of the volumetric and energetic efficiency.

REFERENCES

/1/ Eweis, M.: Reibungs- und Undichtigkeitsverluste an Kolbenringen. Forschungsheft 371. VDI-Verlag GmbH, Berlin 1935

/2/ Bartmann, L: Leckverluste im Zylinder eines Kältekompressors. Kältetechnik-Klimatisierung, 22 (1970) Heft 4, S.121 ff,

/3/ Süß, J.; Kruse, H.: Efficiency of the Indicated Process of CO_2-Compressors. International Journal of Refrigeration, Vol. 21, No.3, 1998

/4/ Süß, J.: Untersuchungen zur Konstruktion moderner Verdichter für Kohlendioxid als Kältemittel, DKV-Forschungsbericht Nr.59, Stuttgart 1998