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Screw Compressors for Heat Pump Application

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SCREW COMPRESSORS

FOR HEAT PUMP APPLICATION

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1. INTRODUCTION

Many sectors of our present-day industrial life have become unthinkable without screw compressors. This applies in particular to the modern production of compressed air for which both screw compressors cooled by oil injection and oil-free screw compressor sets are being used in large numbers. But also in the chemical and petrochemical industries, the mining industry, the building-material industry and in many other industrial branches the screw compressor has made its way not only for the compression of air but also of many other gases. Its market share is increasing in all sectors.

The application of screw compressors for the compression of refrigerants is of comparatively recent date. During the early fifties when screw compressors were first used for industrial purposes - GHH STERKRADE have been manufacturing screw compressors since 1952 - the compression of air and process gases stood in the foreground. Small construction sizes in conjunction with comparatively high flow rates, the high degree of operational reliability at the desired pressures, and a wide control range have gradually gained the screw compressor a notable position also in the industrial refrigeration industry. As compared to refrigeration engineering, the requirements of air-conditioning plants - and there in particular the application of heat pumps - with regard to systems, apparatuses and compressors are more diversified. The question of how the screw compressor meets these demands and of its possibilities and limits with regard to heat pump application shall be answered by the following details.

2. MODE OF OPERATION OF SCREW COMPRESSORS

First, however, a few general remarks on the principle and mode of operation of screw compressors. As schematically shown in Fig. 1, screw compressors fall into the category of positive displacement machines among which, together with the Roots blowers, they form the group of double-shaft rotary machines.

The justification of this grouping results from the mode of operation of screw compressors, as illustrated in Fig. 2. This figure shows a rotor pair in four different positions. The profiled rotor pair itself consists of the so-called male rotor, which is usually driven and in this case has four "lobes", and of the female rotor, which in this case has six "flutes". The turning movement of the rotors brings about a change of the "interlobal spaces", i.e. the grooves between the rotor profiles, owing to their helical shape in such a way that they are increased on the one side - in this case on the underside of the rotor - and reduced on the other side - in this case at the top. An increase of the interlobal spaces means suction, while a reduction means compression.

The process itself comprises four phases: suction - transportation - compression - discharge. On the suction side one interlobal space of both the male and female rotors is filled with the gas to be compressed. In the course of rotation the two spaces increase until they are closed by the discharge-side face wall. Then there begins the phase of transportation, where first the interlobal spaces are also closed on the suction side, until a lobe of the male rotor plunges into the corresponding gas-filled flute of the female rotor.

With increasing "plunging depth" of the lobe in the flute, the pressure within the interlobal space increases until the rotors pass over the discharge-side outlet control ledges, thus establishing a connection between the interlobal space and the compression chamber. The volume enclosed in the interlobal spaces is then completely discharged as rotation of the rotors continues.

The well-known p-V diagram of compression is shown in Fig. 3. Suction, which is effected without any loss, is followed by compression until the rotor lobe tips pass over the outlet control ledges. The ratio between the volume now achieved and the suction volume is determined solely by the position of the control ledges in the machine. This ratio is therefore also called the "installed" volume ratio or "installed" pressure ratio, since with a given medium conversion into pressures is possible without difficulty. The screw compressor has no dead space.

On line 3 in the left diagram this installed pressure ratio corresponds to the discharge pressure, so that along this line the medium is discharged at p = 3 bar = const. During practical operation the actual pressure ratio may deviate from the installed ratio. With any such deviations, additional work as compared to the work required with coinciding pressure ratios has to be carried out. When the installed pressure ratio is too low (upper line), this additional work involves a sudden shock compression to the discharge pressure, and when it is too high (lower line), it causes a sudden expansion to the discharge pressure. This is marked in the diagrams by shaded areas.

The diagram on the right side of this figure, which applies to a screw compressor with an installed pressure ratio of 3, shows that the volumetric-efficiency and internal-efficiency curves are comparatively flat with deviating discharge pressures.

Further losses in screw compressors are caused, on the one side, by internal gaps - both between the rotors themselves and between rotors and casing - and, on the other side, by friction between the medium to be compressed and the casing and rotors. These internal gaps, whose effect can be compared to that of bypasses, mainly cause internal leakage losses. If, however, the rotors are turned more quickly, the absolute gap losses per unit of time remain constant, while the volume handled per unit of time increases. This means that from this point of view, high circumferential speeds are advantageous. An opposite effect, however, is produced by the gas-friction and whirl losses, which increase with rising circumferential speed. The cumulative curve of all losses, as shown in Fig. 4, reaches a flat, though noticeable minimum at such circumferential speed which is the optimum speed value for the respective conditions, i.e. at which the machine achieves the highest efficiency. From their particular mode of functioning it was deduced that screw compressors, just like piston compressors, are positive displacement machines. For this reason they also have similar partial-load characteristics. Fig. 5 is a comparison between the partial-load curves of screw compressors and turbo compressors of radial and axial design. While turbo compressors feature a typical interdependency of pressure and volume, and below a certain delivery volume, the so-called surge limit, enter an unstable range, screw compressors are able to compress a volume which is dependent practically only on the speed, to any pressure eligible within the range of the particular construction type.

But for the screw compressor, too, there is a limitation on the performance map which makes itself felt especially in the low speed range.

As pointed out before, the volumetric efficiency deteriorates with decreasing circumferential speeds. The internal leakage losses which remain constant per unit of time increase relative to the decreasing delivery volume. The already compressed gas which flows back to the suction side is hot and heats up the incoming medium so that the compression temperatures rise as the speed falls. As a result, thermal expansion of the rotors becomes also more pronounced. At a certain temperature and a predetermined cold rotor clearance, contacting of the rotors would occur, which would make the machine inoperable. For this reason a certain compression temperature must not be exceeded, though this level will differ, of course, depending on the construction size, clearances in cold state and materials used and on whether or not a cooling system is provided for the rotors.

The figure shows such temperature limit curves. It is apparent that in the case of low speeds, i.e. small delivery volumes, the pressures that can be achieved are not quite as high as in the case of high speeds.

3. SCREW COMPRESSORS FOR REFRIGERANTS, COOLED BY OIL INJECTION

For industrial refrigeration purposes mainly such screw compressors are employed where oil is injected into the compression chamber upon completion of the suction phase.

This oil has the task to remove part of the compression heat, to seal the gaps between the rotors and between rotors and casing, to lubricate the bearings and the shaft passage seal, to adjust the capacity control slide, and to lubricate the gearbox, where applicable. Thanks to the cooling function of the oil, the compressor discharge temperatures can be kept low. Its sealing function prevents compressed gas from flowing back through gaps from the discharge end to the suction side. Both these effects ensure that high pressure ratios can be combined with low discharge temperatures.

Fig. 6 shows the dependence of the compressor discharge temperature on the actual pressure ratio, both for a screw compressor cooled by oil injection and for dry-running piston or screw compressors for R 22 refrigerant duty. With screw compressors cooled by oil injection it is possible even at high pressure ratios to keep the temperature at not much more than 80° C.

Fig. 7 represents a comparison between measured efficiency values of screw compressors cooled by oil injection and piston compressors. As can be seen, screw compressors offer considerable advantages at pressure ratios below 6, this being mainly due to the fact that there occur neither valve losses nor any other losses occasioned by dead spaces. The decisive precondition for such efficiencies as well as for the high quality and performance of the screw compressor lies in the manufacturing accuracy of rotors and casings. For the manufacture of the rotor profiles alone, more than 30 individual dimensions have to be considered.

Fig. 8 gives a survey of the points where measurements must be taken. The manufacturing tolerances of diameters, radii, widths and lengths lie within the range of a few hundredths of a millimeter, just like those of the admissible clearances between male and female rotors perpendicular to the helices. Even the backlash between male and female rotors from the base to the tip of the profile has to be kept continuously increasing in the range between 0.01 mm to 0.05 mm for small rotor diameters (82 mm dia.) and between 0.03 mm to 0.09 mm for medium diameters (255 mm dia.). This entails additional limitations with regard to the manufacturing tolerances, since to ensure a correct backlash pattern, manufacture has to follow either the upper or the lower tolerance limit.

The influence of accurate clearances on the volumetric efficiency and power requirement of a screw compressor was determined empirically by a large number of measurements. Roughly generalizing, it can be said that a mean increase of the clearance by one hundredth of a millimeter costs about 1% in efficiency.

With the screw compressor, high-precision machining of the profiles can therefore be regarded as the key factor for a competitive machine.

The lubricating effect of the injected oil permits torque transmission between the rotors without any mutual contact, so that there is no need for a synchronizing gearbox to divide the power between the rotors.

Thus, the mechanical layout of screw compressors cooled by oil injection is extremely simple; there only exist small and purely rotating masses which permit vibration-free operation without costly foundations.

Owing to the attenuating effect of the injected oil, the noise level is comparatively low. Fig. 9 shows the measured curves of a direct-driven ammonia compressor without noise hood. The values are below NR 85, reaching a peak value at approx. 2,000 c/s, which is 10 times the basic frequency of the male rotor of the screw compressor.

As compared to piston compressors, the screw compressor has the further advantage of high-speed rotation, from which there result extremely favourable ratios between throughput volume and construction size and thus a compact design. As will be explained later in more detail, screw compressors cooled by oil injection can be equipped with a control device for infinitely variable adjustment with high partial-load efficiencies, which is of particular interest with regard to their application in refrigeration and air-conditioning plants. They furthermore offer the possibility of side-stream admission, a speciality of which only brief mention can be made in this context. Another feature worthy of reference consists in their insensitivity to a certain amount of condensates in the medium handled, so that condensate proportions of up to several per cent can be coped with without any difficulty.

A brief summary of its main characteristic features shows the following advantages of the screw compressor cooled by oil injection for application in refrigeration and air-conditioning plants:

simple and extremely compact design,
high efficiency,
vibration-free operation at a low noise level,
high operational reliability,
little maintenance,
possibility of side-stream admission,
insensitivity to condensate portions in the medium handled,
efficient volume control.

3.1 Construction Types

Screw compressors cooled by oil injection can be supplied with or without a control system. Since, however, it must be possible with any refrigeration and air-conditioning plant to achieve partial loads in conjunction with high efficiencies, a non-controlled compressor will have to be employed in such cases where adjustable-speed drivers are available. The main field of application will here consist in heat pumps driven by combustion engines.

Fig. 10 illustrates the design of a non-controlled compressor, which is based on a construction series of air compressors used with success for many thousands of applications. The suction nozzle is arranged at the top, whereas the discharge nozzle leads vertically downwards. The driver acts on the male rotor which has four lobes while the female rotor has six, but contrary to the compressed-air construction series, drive in this case is effected from the discharge side. In this way partial compensation of axial thrust produced by the gas forces is achieved. Since the load on the discharge-side axial bearings is reduced thereby, the service life of these bearings is correspondingly longer.

Both the axial and radial bearings are of antifriction type, which offers the advantage of comparatively simple lubrication - neither pre-lubrication nor a separate oil pump being needed - as well as of low bearing losses.

Furthermore, it is easier with antifriction bearings than with journal bearings to minimize the clearance between the rotors, this being absolutely necessary to achieve high efficiencies.

Controlled screw compressors are well-known from industrial refrigeration engineering. Their applicability with regard to heat pumps will probably be centered on the air-conditioning of large housing estates or public installations as well as on the recovery of heat from industrial processes. The main characteristic features of a controlled screw compressor as compared to a non-controlled machine, consists in the capacity control slide arranged below male and female rotor which is designed as an axially movable part of the casing in the area of the rotor bores. Its mode of functioning can be seen from Fig. 11. When the capacity control slide is moved, its suction-side end clears an opening through which part of the gas sucked into the interlobal spaces is returned to the suction nozzle via an annular duct. During this phase, the two helices filled with incoming gas are just about closed on the suction side, so that no compression work is performed on the returning medium. By further opening the capacity control slide, delivery can be reduced down to zero. While the actuator of the control slide works hydraulically, adjustment can also be effected through an electric servo motor. Just as with other forms of control, signals produced by pressure and temperature can be used as input variables.

Fig. 12 represents the cross-section of a controlled refrigeration screw compressor suited for air-conditioning and heat pump applications. The driver acts on the male rotor from the discharge side of the compressor. The suction nozzle is arranged centrically on the casing for a radial suction flow. The discharge nozzle can be mounted at the side of the casing either on the right-hand or left-hand lower part. On the side of the coupling, the driving shaft is sealed from the atmosphere by a mechanical (contact-ring) seal with oil seal chamber. An additional standstill seal can also be provided, so that the mechanical seal can be dismantled, if necessary, without opening up the compressor.

The axial and radial bearings are arranged upstream of the shaft passage seal. Forces in axial direction are absorbed by axial antifriction bearings. In order to relieve these bearings of load, in particular during operation at high pressures, the free end of the male rotor is subjected to the discharge pressure so that it acts as a balancing drum.

3.2 Limits Of Application

The limits of application of the above described screw compressors with oil injection cooling for heat pump application are mainly determined, on the one hand, by the maximum pressures admissible for the compressor casing and, on the other, by the load acting on the bearings.

For refrigeration compressors in R 22 service, a condensation temperature of 60°C necessitates service pressures of 25 bar. On the basis of the most unfavourable inspection specifications this requires a test pressure of 42 bar, and the casings are dimensioned accordingly. With other types of refrigerants a service pressure of 25 bar means that the condensation temperature is 80° C, such as with R 12, or 120° C, such as with R 114.

The higher bearing loads for heat pump application as compared to industrial refrigeration service can be clearly seen from Fig. 13.

Referred to normal industrial refrigeration service, this diagram illustrates the increase in the bearing load of the male rotor of a refrigeration compressor as a function of the evaporation temperature, with R 22 used as medium and the condensation temperature of the heat pump amounting to 60° C. Even at an evaporation temperature of 0° the load on all bearings is by more than 50% higher than with normal industrial refrigeration applications.

On the assumption that a non-controlled compressor is used for R 12 in a bivalent system for the production of heating power and hot water with a direct drive and based on the resulting service conditions, the evaporation temperatures common for heat pumps can be realized without any reservations. For controlled compressors an inlet temperature of +5° C in R 22 duty at a condensation temperature of 60° C marks a certain limit. In both cases, the screw compressors have a practically unlimited service life provided the instructions of the manufacturer of the antifriction bearings concerning regular replacement after about 40,000 hours are adhered to.

3.3 Power At Coupling, Heating Power, Performance Numbers

The efficiencies of the non-controlled compressor type are shown in Fig. 14.

Even where the volume ratio is adapted to the respective conditions, efficiencies decrease as the pressure ratios rise, owing to greater gap and outlet losses.

The diagram on the left shows the prevalent conditions when the male rotor is driven direct at 2,950 rpm. The efficiencies are represented for four compressor types at different levels, depending on their rotor diameter and thus their delivery volume. It is obvious that smaller rotor diameters will always go hand in hand with lower efficiencies, since the ratio of gap volume to throughput volume becomes more unfavourable. The diagram to the right shows the efficiencies that would be achieved if the driver were to act on the female rotor, a potential design which is not available at present. If the female rotor were driven at 2,950 rpm, the lobe ratio of 4 to 6 between male and female rotors would result in a male rotor speed of 4,425 rpm with a consequently higher throughput. Driving the female rotor would thus offer the advantage of a higher performance ratio, and furthermore of higher efficiencies, since the circumferential speeds achieved with direct drive of male and female rotors would permit operation of the compressor at almost optimum speeds.

Realization of the above design is, however, not without difficulties, since it is not easy to transfer the high driving torques through the comparatively thin journal of the female rotor. The profile meshing and roll-off conditions cannot be called really favourable either with the profiles commonly used at present, but the above described advantages show that it might be well worth while to tackle these difficulties.

The efficiencies of controlled screw compressors - in R 22 service and with a condensation temperature of 60° C - can be seen from the diagram at the bottom. Owing to larger rotor diameters the values are correspondingly higher. The diagram shows, however, that the increase in efficiency becomes smaller, the larger the rotor diameter. The reason for this is that at constant speed the circumferential speeds rise with increasing rotor diameter, so that the dynamic losses more and more absorb the gain in volumetric efficiency.

The figure which interests the manufacturer of heat pumps most, is the heating power which can be achieved as a function of the power at coupling, as illustrated for non-controlled and controlled compressors in Fig. 15. As can be seen, there is a very narrow gradation, which can be achieved in the case of non-controlled machines by drive input via the female rotor, and in the case of controlled machines by the selective use of two types of one and the same construction size, which merely differ by the rotor length and thus by the delivery volume. The respective powers at coupling range between only a few kW and far more than 1,000 kW with the larger compressors.

The ratio of heating power output to coupling power input - i.e. the effective performance number - is shown in Fig. 16.

Such performance numbers have been determined for the media R 12, R 22 and R 114 at two different condensation temperatures each, as a function of the evaporation temperature.

The differing efficiencies of the small and large types cause a certain spreading of the performance numbers, which were therefore combined to form several groups where the lower limit represents the performance numbers of small compressors and the upper limit those of large machines.

4. PARTIAL-LOAD BEHAVIOUR

For the handling of partial loads by non-controlled machines the driving speed has to be changed. The curve for power at coupling versus heating power in such a case is shown in Fig. 17, which illustrates that when the speed is reduced, the power at coupling initially decreases more than the heating power; in this range the performance number rises over the design value.

It is only below 50% heating power that the performance number starts to become lower, until finally in the minimum speed range the power at coupling merely covers the machine losses. In order to achieve such a behaviour - which may be advantageous from the point of view of the heat pump manufacturer - the compressor is rated such that at the maximum circumferential speed, in this case 50 m/s, the required nominal heating power is just obtained. This circumferential speed is already so high that the compressor has passed beyond its optimum efficiency. When its speed is therefore decreased in the partial-load range, the efficiency and thus also the performance number initially improve until the optimum circumferential speed is reached. Following this, with decreasing speed and thus also circumferential speed, the machine losses rise again. A different behaviour is shown by a machine where the capacity is adjusted by means of a control slide. This will be explained with the aid of two examples.

In the first example, Fig. 18, hot water is to be produced utilizing the heat of groundwater, further ensuring that evaporation and condensation temperatures, and therefore also pressures, remain constant during partial-load operation, as shown by the temperature graph of the first diagram. The compressor is to have an installed volume ratio of about 3.5 in the nominal load point. When the capacity control slide is now opened on the suction end to return part of the intake volume to the suction side, it moves at the same time towards the discharge side, first passing over the radial part of the outlet port in the casing, as a result of which the installed volume ratio changes only slightly. It follows the parabolic course shown in the second diagram. Once the radial part of the outlet port is closed - in the example referred to this is at approx. 65% partial load - there is no further change on this port even if the throughput volume continues to decrease, since the axial part of the outlet port is independent of any slide position. As a consequence the volume ratio decreases proportionally to the reduction of the delivery volume.

The behaviour of the power at coupling and the heating power is demonstrated in the third diagram. With rising partial load the first decisive influence is exerted by the throttling losses of the deflected flow on the suction side which affect the performance to an increasing, but on the whole not particularly high degree. The effective performance number drops only slightly. When, however, the radial outlet port is closed, the losses caused by the differing volume ratio must also be taken into account. From here on, the required effective power decreases only very slightly as the useful heat becomes less, so that the performance number deteriorates.

The screw compressor manufacturer has a certain latitude available as regards the division of the outlet port into radial and axial portions so that it can be varied depending on the respective purpose of application to achieve the most favourable partial-load behaviour.

The second example, Fig. 19, is based on the task of utilizing heat from the ambient air for a floor heating system. In this case partial load means a reduction of the heat quantity in response to a rise in the open-air temperature. When the air temperature rises, followed by the evaporation temperature, the condensation temperature and, as a result, also the differential between these two temperatures, drops. This leads to smaller pressure ratios. An ideal screw compressor for this application should therefore be able to change the volume ratio in just the way corresponding to the changing thermodynamic conditions.

For the example chosen here, an installed volume ratio of 5 in the nominal duty point would be required. The actually installed ratio, however, is only 4.5, with the radial outlet port closing already at approx. 80% partial load, after which the volume ratio of the compressor drops rapidly. The curves marking the optimum and real volume ratios show that this compressor was optimized for partial loads in the range of about 70%.

The behaviour of the power at coupling and heating power is demonstrated in the diagram on the right. Over a wide load range the power at coupling decreases considerably more than the heating power, i.e. the performance number improves. This is due to the fact that the effective power is mainly dependent on the differential between condensation and evaporation temperatures, whereas the heating power is solely determined by the condensation temperature. As the temperature graph shows, the temperature differential drops more than the condensation temperature by itself. This explains why the performance number rises in spite of a decrease in the compressor efficiency at low partial loads.

5. FUTURE DEVELOPMENTS

The performance diagrams show that for most heat pump applications, small screw compressors are fully adequate, and that for many applications - in particular heating of self-contained houses for one family - the performance of available screw compressors frequently is even too high. For the future there may therefore result the task of designing smaller screw compressors especially for heat pump application. Since, however, small screw compressors with the usually required direct drive have only very low circumferential speeds and since in addition, the relative gap widths and thus their influences are high, not only the question of compressor design but also of the efficiency will be in the foreground of interest.

Improving the efficiency is being approached by several methods at the same time: As mentioned before, the size of the gaps between the rotors and between rotors and casing considerably influence the efficiency of screw compressors. An attempt is therefore being made to achieve minimum clearances by an even more stringent precision in machining, and to determine through extensive test runs the particular oil quantities and circumferential speeds which produce optimum efficiencies.

Fig. 20 shows the result of a measuring test run series with an air compressor cooled by oil injection, where both the front-face gap at the discharge side and the clearances between profile lobes and flutes were varied. The success scored by narrow clearances is quite obvious. Practical realization of such knowledge in industrial fabrication is, however, no easy problem for the shop engineers.

Fig. 21 represents the specific power requirement at a given discharge pressure as a function of circumferential speed and injected oil quantity. On the basis of such measurements the optimum range of application of a certain type is determined.

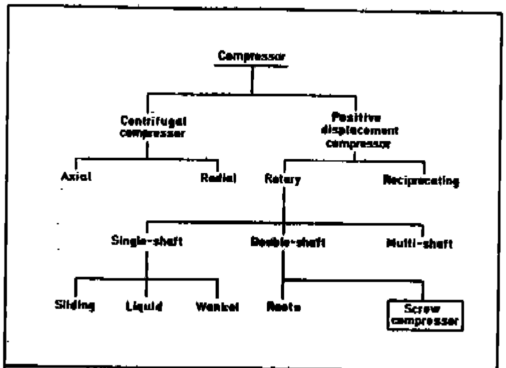
Development work does not only extend to improvements in the sector of fabrication and to optimization of the service conditions, but also to the profile itself.

From the development of dry-running screw compressors it is a well-known fact that the number of rotor lobes, for instance, has a considerable influence on the suitability of the machine for different pressure level ranges. It is further known that at high pressure ratios the outlet cross-sections become very small, so that short rotors at the same angle of contact have lower outflow losses than rotors with a high L/D ratio.

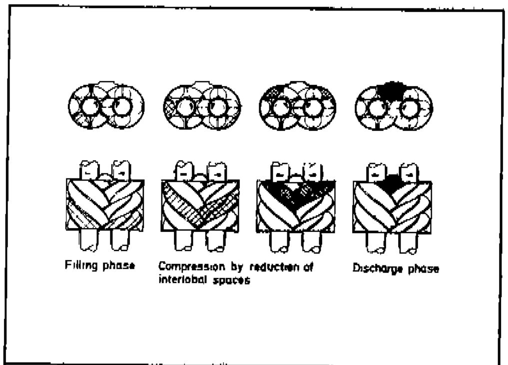
These two examples alone show that all the geometric parameters have also to be thoroughly studied, such as:

- shape of profile and number of lobes,
- angle of contact,
- length/diameter ratio,
- installed pressure ratio.

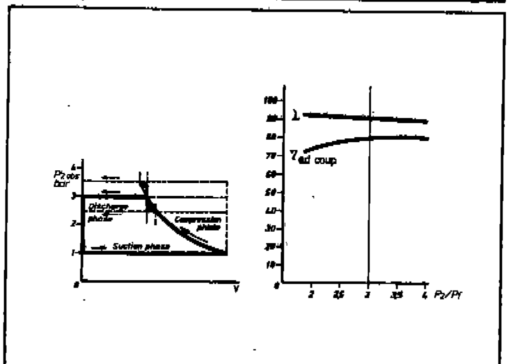
All this, however, must be considered under the aspect that a given enclosed space must be able to yield an acceptable delivery volume if the machine is not to become too expensive. The aim therefore must be to find a reasonable compromise, such as is required in all engineering sectors.



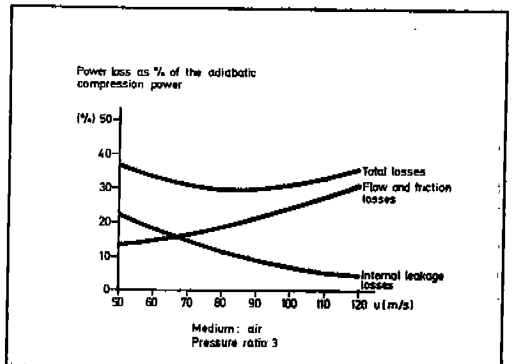
GHH STERKRADE | Fig. 1 Main classification of compressors | M.SM E 223a



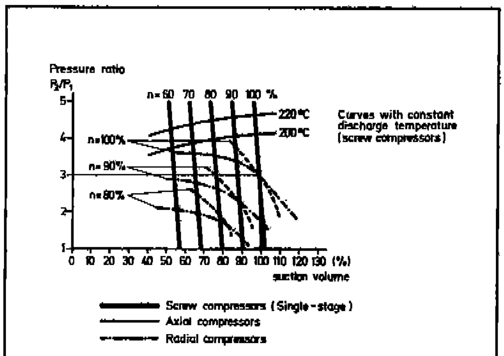
GHH STERKRADE | Fig. 2 Compression phases - normal construction type | M.SM E 103



GHH STERKRADE | Fig. 3 P-v diagram as well as volumetric-efficiency and internalefficiency diagram of a screw compressor | M.SM E 131



GHH STERKRADE | Fig. 4 Proportion of various power losses referred to the overall power requirement of a screw compressor | M.SM E 130



GHH STERKRADE | Fig. 5 Characteristic curves of various compressor types | M.SM E 132

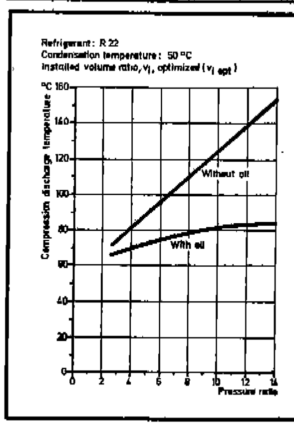


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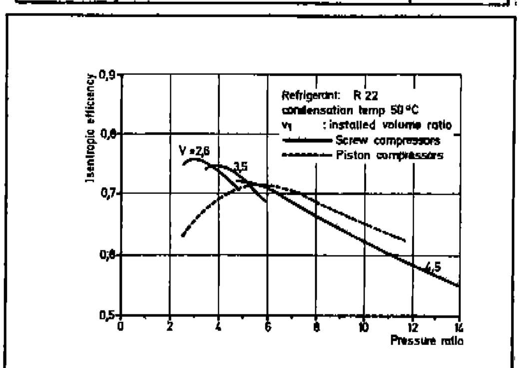
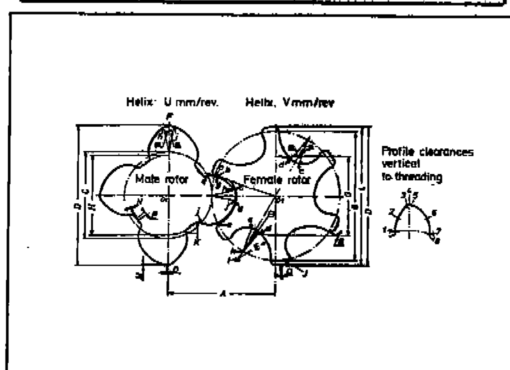
Fig. 6

Compression discharge temperature as a function of the pressure ratio.

M.SM E 260



GHH STERKRADE | Fig. 7 Measured efficiecies of screw and piston-type compressors for refrigerants | MSM E 261



GHH STERKRADE | Fig. 8 Asymmetrical profile of screw compressor rotors with more than 30 individual dimensions | M.SM E 238

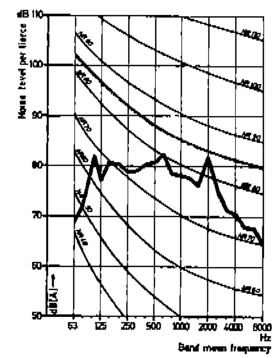


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Fig.:9

Noise-level analysis of a GHH screw compr., measured at 1m distance around the set, without noise hood mean value.

Type	SKK 204 D
Speed n:	2,940 rpm
Refrigerant:	NH₃
Suction pressure P_0:	2,65 bar
Discharge pressure P_c:	11,2 bar

M.SM
E 262

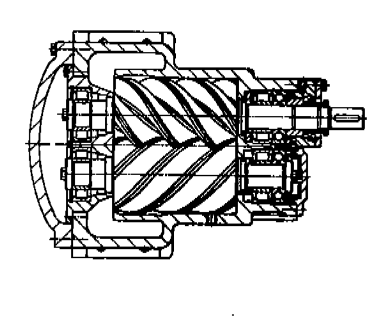

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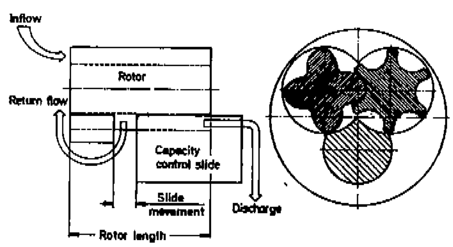
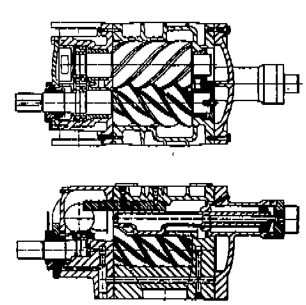
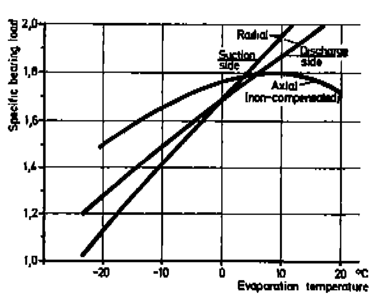
Fig.:10
Non-controlled screw compressor for refrigerants
M.SM
E263



GHH STERKRADE
Fig.: 11
Schematic diagram of delivery volume control with shortened capacity control slide.
M.SM
E 118



GHH STERKRADE
Fig.: 12
Refrigeration screw compressor with capacity control slide
M.SM
E 264



Specific reference values: $t_{compr} = -10\,°C$, $t_{cond} = 35\,°C$

GHH STERKRADE
Fig.: 13
Specific bearing load of male rotor in R 22 service ($t_K = 60\,°C$)
M.SM
E 265

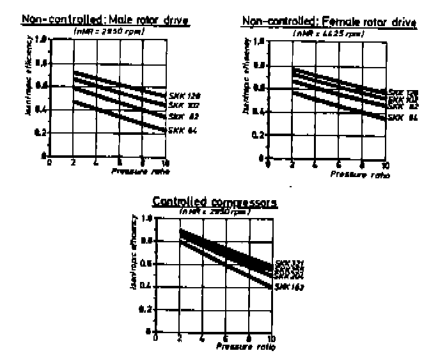


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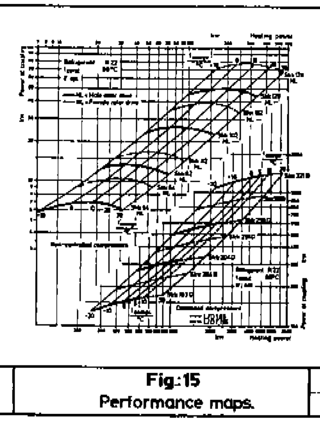
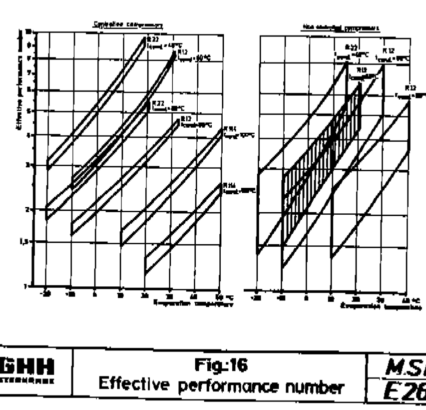
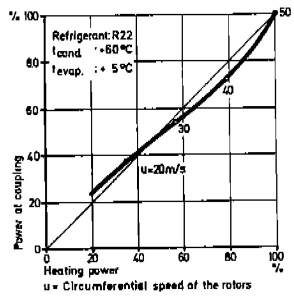
Fig.:14
Efficiency as a function of the pressure ratio
M.SM
E266



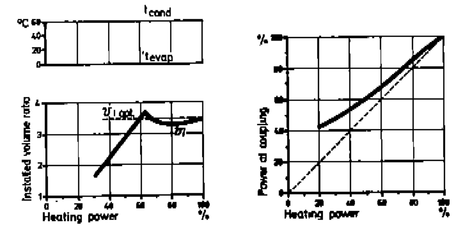
GHH STERKRADE
Fig.:15
Performance maps.
M.SM
E267



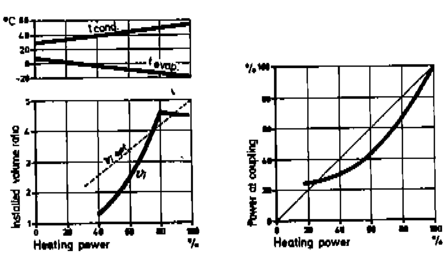
GHH STERKRADE
Fig.:16
Effective performance number
M.SM
E268



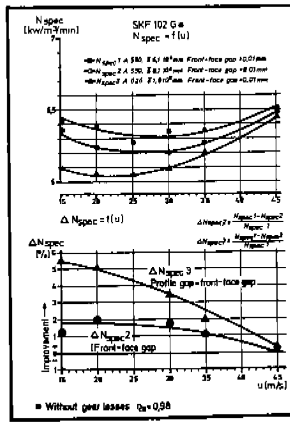
GHH STERKRADE | Fig: 17 Partial-load behaviour with speed control | M.SM E 269



GHH STERKRADE | Fig: 18 Partial-load behaviour with capacity control slide - heating with groundwater | M.SM E 270



GHH STERKRADE | Fig: 19 Partial-load behaviour with capacity control slide - heating with outside air | MSM E 271



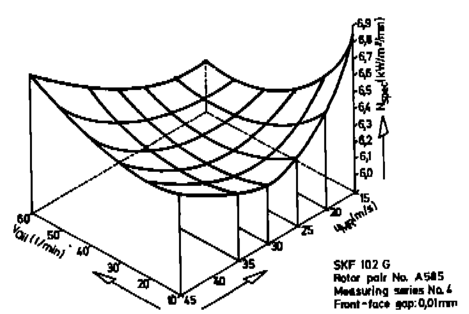
GHH STERKRADE

Fig.: 20

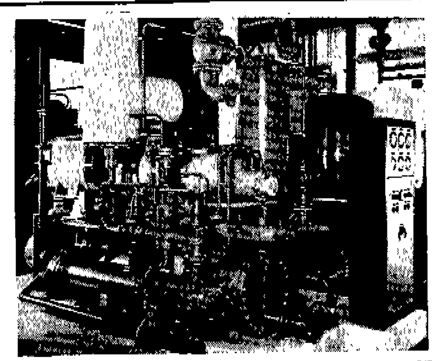
Influence of gap widths on the specific power requirement.

SKF 102 G
Mech. No. 51 249
P_1 = 1 bar (Converted)
t_2 = 80 °C

M.SM E 254



GHH STERKRADE | Fig: 21 Specific power requirement as a function of circumferential speed and injected oil quantity. | MSM E 255



GHH STERKRADE | Fig. 22 Refrigeration screw compressor in a brewery NH_3; t_0 = - 10 °C; t_c = 35 °C; Q_0 = 658 000 kcal/h | M.SM E 273