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IMPROVING THE CAPACITY AND PERFORMANCE OF AIR-CONDITIONING SCREW COMPRESSORS

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ABSTRACT

Traditional compressor design methods make it very difficult to make significant improvements to screw compressors required for refrigeration and air conditioning applications. The principles which determine the optimum design of such machines are reviewed and, in some cases, redefined. These include rotor profiling, configuration and proportions, clearance distribution and bearing clearances and it is shown that there are conflicting requirements for them when attempting to obtain optimum performance. The best result can thus only be obtained by consideration of all the relevant parameters in an optimisation procedure, which leads to rotor designs that vary according to the specific duty of a machine, rather than follow a standard pattern An outline description is given of a process modelling procedure which has been developed to optimise the design of a compressor by the use of multi variable minimisation methods that take simultaneous account of all these parameters

As a result of a collaborative project between industry and an academic institution, this procedure was applied to the design of a new refrigeration screw compressor, which was then built and tested. Its performance was then compared with that of the machine it was intended to replace. It is shown that the optimised design was up to 9% better than the original machine.

Keywords: Refrigeration, Screw Compressor, Design, Rotor, Optimization, Efficiency, Noise, Reliability

INTRODUCTION

Screw compressors, especially of the oil flooded type, are widely used in refrigeration and air conditioning systems. Because the market for these is highly competitive, continuing efforts are being made to produce more compact, efficient and cost effective machines. Due to the product maturity, large efforts are required, based on increasing attention to fine detail to make only small improvements, when using traditional design methods. However, better understanding of the operating principles of these machines, gained from more advanced analytical procedures, can be used to advantage to make more significant improvements to the final product.

Most importantly, new rotor generation methods have been developed, based on numerical procedures for optimisation of the entire design, when considering the simultaneous variation of a number of variables, which result in the best delivery and highest efficiency for the same rotor tip speed. Such an approach leads to rotor profiles that are unique for each application, rather than scaled versions of classical shapes.

Although the efficiency of screw compressors is mainly dependent on improved rotor profiles and clearance distribution, other items, such as the housing ports, the seals and lubrication system and especially the bearings must be designed to take full advantage of the potential which these confer if maximum performance gains are to be achieved.

DEVELOPMENT OF REFRIGERATION SCREW COMPRESSORS

Historically, the Swedish company, Svenska Rotor Maskiner (SRM), are the pioneers, who introduced the use of screw compressors into refrigeration systems during the early nineteen seventies. Consequently, a great number of these machines are still being produced under their licence, based on their continuing improvements to the original designs. The scientific basis of these improvements became known only much later, mainly as a result of independent studies by compressor manufacturers and universities. *Mossemann* of the Kuhlautomat, company, Berlin, Germany *1975,* was the first of these to bring his design work to the attention of the refrigeration community. *Sauls,* of the Trane, company of Lacrosse, WI, USA, wrote a series of papers between *1992* and *2002,* which defined important routes and suggested actions needed to produce better refrigeration screw compressors. *Fujiwara et al* of Hitachi, a Japanese compressor company and of Moroun University, Japan*, 1994,* published important work, which has been much used in the development of their refrigeration compressors.

Later *Fleming et al* of Strathclyde University, Glasgow, Scotland, *1998,* reviewed contemporary design methods and procedures resulting from their comprehensive work in screw compressor modelling and optimization, which paid special attention to refrigeration compressors. Recently *Xing* of Xi'an Jiaotong University, Xi,an, China, *2000,* published his book on screw compressors, in Chinese, from which a discerning examination will show almost all that is needed for up to date design of refrigeration screw compressors. The influence of his research group on the newest Far Eastern refrigeration compressor designs can already be detected.

The most recent developments in screw compressors and new procedures for their design are published by *Stosic et al* of City University, London, England*, 2005*. The authors of this paper have used the procedures, described in this work, to produce a family of new refrigeration screw compressors for industrial applications.

DESIGN REQUIREMENTS FOR REFRIGERATION SCREW COMPRESSORS

Refrigeration and air conditioning system compressors must be quiet, efficient, reliable and durable. To perform without causing excessive noise, compressors must run slowly. This implies that a refrigeration screw compressor must be built with small rotor and housing clearances to maintain low leakage under these conditions. The manufacture of rotors by grinding, especially with simultaneous measurement and control of the profile, today makes it possible to maintain a profile tolerance of 5 μm. This, in turn, enables the clearances between the rotors to be kept below 15 μm. With such small clearances, rotor contact is very likely and hence a profile must be generated which avoids damage or seizure should this occur. This is best achieved by the use of rotors with involute lobes and low contact stresses.

Rotor Configuration

Increasing the number of rotor lobes enables the same built-in volume ratio to be attained with larger discharge ports. Larger discharge ports decrease the discharge velocity and therefore reduce the discharge pressure losses, thereby increasing the compressor overall efficiency. Hence refrigeration compressors tend to be built with more lobes than the traditional 4-6 combination and 5-6 and 6-7 configurations are becoming increasingly popular. Also, the greater the number of lobes, the smaller the pressure difference between the two neighbouring working chambers. Thus, interlobe leakage losses are reduced. Furthermore, more lobes combined with a large wrap angle ensure multiple rotor contacts which reduce vibrations and thus minimize noise. However, more lobes usually mean less rotor throughput, which implies that refrigeration compressors are somewhat larger than their air counterparts. Also the leakage to delivery ratio is worse with more rotor lobes. Therefore, such compressors are less efficient. Additionally, more lobes increase the manufacturing cost.

Rotor Sealing Line Length and Blow-Hole Area

Since refrigeration compressors rotate relatively slowly, their rotor profiles must have the smallest possible blow-hole area if leakages are to be minimised. However, reduction of the blow-hole area is associated with increase in the sealing line length. It is therefore necessary to find the optimum profile shape which minimises the sum of both the blow-hole and sealing line leakage areas.

Rotor Proportions

A general feature of screw compressors is that the pressure difference through them causes high rotor loads and this is especially the case for low temperature refrigeration compressors, where these are large. Therefore, to maintain their rigidity and minimise deflection, the rotor profiles usually have a relatively small male rotor addendum in order to increase the female root diameter. This sometimes leads to very shallow and clumsy rotors. An alternative possibility is to increase the female rotor lobe thickness. This greatly increases its moment of inertia and thereby reduces the rotor deflection more effectively.

Rotor Wrap Angle

Increasing the rotor wrap angle is generally associated with reducing the interlobe sealing line and hence, with reduced leakage between the rotors. Contemporary trends in refrigeration screw compressor design are therefore towards larger wrap angles. However, on occasion, this has led to exceeding the limiting values and thereby reducing the compressor displacement.

Rotor Bearings

In some compressor designs, multiple cylinder roller bearings or multipoint ball bearings are located at the high pressure end of the rotors to withstand the large radial forces reliably over a long operating life. Frequently, two bearings are also employed for axial loads. Since only one axial bearing actually takes the load, the role of the other is mainly to prevent rotor bounce in the axial direction.

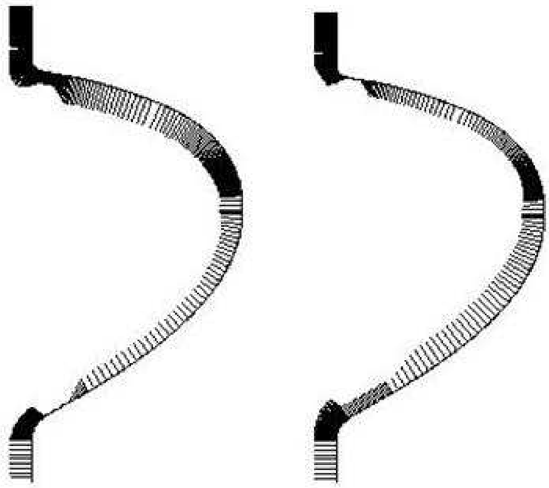
Rotor Clearance Distribution and Contact on the Lobe Flat Side

Oil flooded compressors have direct contact between their rotors. In well designed rotors, the clearance distribution will be set so that this is first made along their, so called, contact bands, which are positioned close to the rotor pitch circles. Since the relative motion between the contacting lobes in this region is almost pure rolling, the danger of their seizing, as a result of sliding contact, is thereby minimised. As shown in Fig 1, the contact band may be either on the rotor round flank a), or on the rotor flat flank b).

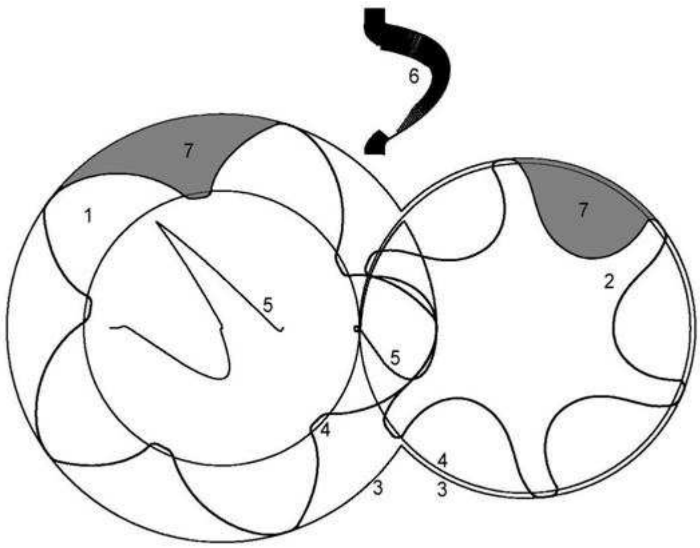
The traditional approach is to maintain a high, so called, positive gate rotor torque, which ensures round flank contact. What is not widely appreciated is that there are significant advantages to be gained by maintaining a negative gate rotor torque to ensure that contact, when it occurs, will be on the flat face. The reason for this can be understood by examination of the sealing line lengths, shown as item 5 in Fig 2. Here it can be seen clearly that the flat flank sealing line is much longer than that of the round flank. Thus, minimising the clearance on the flat flank will reduce the interlobe leakage more than minimising the round flank clearance. Also, negative gate torque is achieved by making the gate rotor lobes thicker and the main rotor lobes correspondingly thinner. The displacement is thereby increased. Thus both these effects lead to higher compressor flows and efficiencies.

Displacement of Bearing Centres

One additional design aspect, which though important, is not widely appreciated is that the pressure loads will tend to push the rotors apart from their design position in the casing, as a result of the clearances within the bearings. If these are not taken fully into account, the resulting displacement will cause contact between the rotor tips and the casing, when their rotor clearances are small and the pressure loads are high. To counter this, the bearing centre distance must be smaller than that of the rotor housing. To maintain the rotor interlobe clearance as small as possible, the bearing centre distance must be even further reduced.



a) Rotor contact on the round face, b) Contact on the flat face
Figure 1 Rotor interlobe clearance distribution



1-main, 2-gate, 3-rotor external and 4-pitch circles, 5-sealing line, 6-clearance distribution, 7- rotor displacement area
Figure 2 Screw compressor rotors

Also, if the bearing centres are set to be the same as those of the rotors, the clearance between the rotors and housing will be smaller at the low pressure side of the rotors and larger at the high pressure side. Since leakage is caused by the pressure difference, this displacement creates the least favourable rotor position for efficient compressor operation. The bearing centre distances must therefore be arranged to maintain a uniform clearance between the rotors and the housing.

Optimizing the Rotor Profile and Compressor Operational Parameters

Even the simple analysis of rotor behaviour in refrigeration compressors, given here, shows that there are conflicting requirements for desirable rotor characteristics. This implies that only a simultaneous optimization of all the variables involved in the design process will lead the best possible compressor performance.

In the designs describe in this paper, a full multivariable optimization of screw compressor geometry and operating conditions was performed to establish the most efficient compressor design for a given duty. This was achieved by the use of a computer software package, developed by the authors, based on a Box constraint simplex method, which provides the general specification of the lobe segments in terms of several key parameters and which can generate various lobe shapes.

The full rotor and compressor geometry, like the rotor throughput cross section, rotor displacement, sealing lines and leakage flow cross section, as well as suction and discharge port coordinates are calculated from the rotor transverse plane coordinates and rotor length and lead. These are later used as input parameters for calculation of the screw compressor thermodynamic process. The compressor geometry must be recalculated for any variation of the input parameters. The compressor built-in volume ratio is also used as an optimization variable. Computation of the instantaneous cross-sectional area and working volume can thereby be carried out consecutively for increments of the rotation angle.

The procedure started with determination of the suction pressures, which were selected, in this case, to maintain standard air conditioning and refrigeration evaporation temperatures of +5, –15 and –35^{o}C, respectively, while the discharge pressures were set to corresponded to standard condensation temperatures of 30, 40 and 50^{o}C. The NIST (National Institute for Standards and Technology) property routines, which include virtually all known pure refrigerants and their mixtures, were used to enable real fluid thermodynamic properties to be calculated. These also permit the calculation of fluid mixtures, which can be arbitrarily selected to create new customised mixtures of refrigeration fluids.

A mathematical model of the thermodynamic and fluid flow process is contained in the package, as well as models of associated processes encountered in real machines, such as variable fluid leakages, oil flooding or other fluid injection, heat losses to the surroundings, friction losses and other effects. All these are expressed in differential form in terms of an increment of the rotation angle. The numerical solution of these equations enables the screw compressor flow, power and specific power and compressor efficiencies to be calculated. Nine optimization variables were used in the calculation carried out. These comprised the rotor radii, defined by four rotor profile parameters, the built–in volume ratio, the compressor speed and the oil flow, temperature and injection position, since the compressor considered was oil flooded.. The Box constrained simplex method was then used to find the local minima. It stochastically selects a simplex, which is a matrix of independent variables and calculates the optimization target. In the case of the examples given, this was the compressor specific power. This was later compared with those of previous calculations and then minimized.

NEW COMPRESSOR DESIGN FOR IMPROVED PERFORMANCE

Following the principles described, a refrigeration screw compressor was designed using a 5/6 rotor configuration with the main rotor outer diameter of 196 mm and an L/D ratio of 1.35,. The rotors, presented in Fig 3 were optimized for R-407C as the working fluid, in an air conditioning application. The compressor is semi-hermetic and designed to run at 3000 or 3600 rpm. It is presented in Fig. 4 and it is expected to replace the old design which is presented in Fig. 5. A photograph of the new compressor is given in Fig. 6.

The compressor was tested in an industrial experimental test rig, containing R-407C. The test rig schematic diagram is shown in Fig 7. It includes only one cooler-condenser and no evaporator. Refrigerant vapour is compressed to the discharge pressure by the compressor and the oil is separated either within the compressor itself, if it is integrated with the oil separator, or inside a stand alone separator, which is a part of the test rig. After that, the discharge gas is divided into two streams, one of which goes to the cooler where it is cooled, partially condensed and throttled down to the suction pressure and then mixed with the second stream which has also been throttled to the suction pressure. By this means, only the heat generated by the compressor motor is rejected by the condenser.

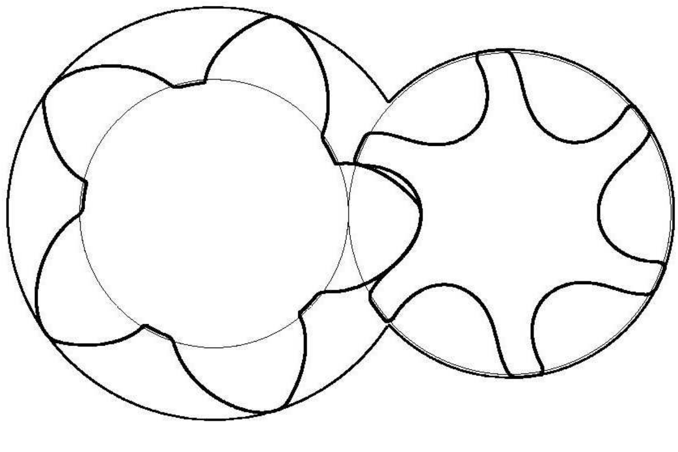


Figure 3 Screw compressor rotors optimised for air conditioning and light refrigeration duty

This results in a fairly compact test rig. Moreover, such a test rig is far more flexible and controllable than the ordinary refrigeration plant allowing a wide range of suction and discharge pressures to be used.

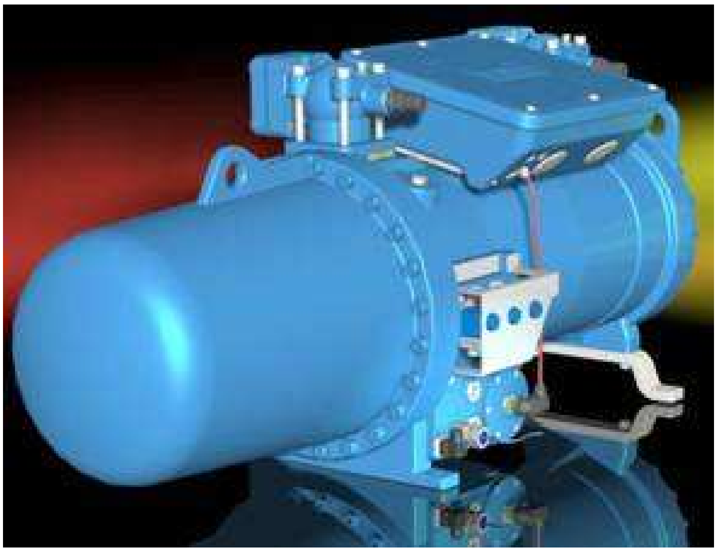


Figure 4 New Design CXHO-140 428

The compressor delivery is measured at the compressor suction by an orifice plate. For that purpose the orifice pressure drop is measured, as well as the orifice pressure and temperature. Suction and discharge flange pressures and temperatures are also measured. Compressor motor power was estimated by use of current and voltage measured on the variable frequency converter, which takes account of all electrical losses. Specific power was calculated as a ratio of the compressor power and flow.

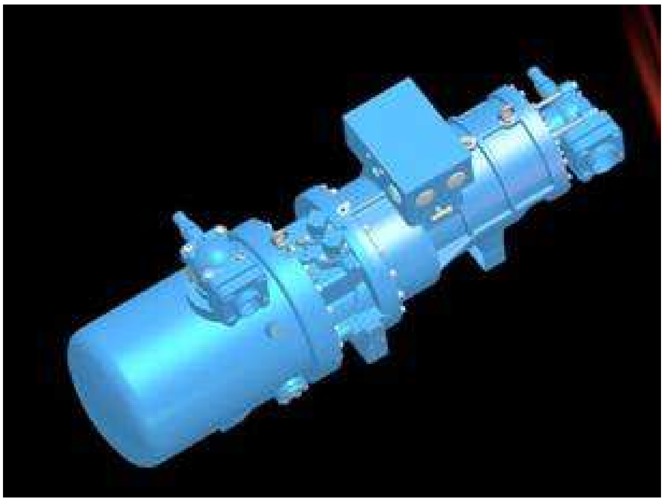


Figure 5 Old Design CTSH8-120 360 Y

The refrigeration capacity was then estimated for an equivalent whole plant operating between the same suction and discharge pressures. The coefficient of performance, COP was thereby calculated as the ratio of the plant refrigeration capacity to the compressor motor input power.

Figure 6 New compressor in the industrial experimental rig

Table 1 Comparison of compressor performance

		New Design CXHO-140 428			**Old Design CTSH8-120 360 Y**		
Tc °C	**Te** °C	**Qo** W	**P** W	**COP**	**Qo** W	**P** W	**COP**
	-5	295600	68300	4.33	248200	58090	4.27
30	0	361900	70500	5.13	298120	61030	4.88
	5	439100	72600	6.05	356180	64260	5.54
	-5	262100	83000	3.16	219650	68740	3.20
	0	322900	85900	3.76	264900	72530	3.65
40	5	393700	89300	4.41	317500	76500	4.15
	10	475500	93100	5.11	378490	80080	4.73
	12.5	520900	95100	5.48	412420	81650	5.05
	-5	221300	102100	2.17	191440	82930	2.31
50	0	275300	105100	2.62	231610	86150	2.69
	5	337900	108600	3.11	278370	89820	3.10
	10	410400	112300	3.65	332670	93320	3.56
	12.5	450600	114200	3.95	362910	94920	3.82
	0	248500	116800	2.13	214910	93390	2.30
55	5	306900	120200	2.55	258160	97390	2.65
	10	374300	123700	3.03	308350	101890	3.03
	12.5	411700	125300	3.29	336290	104230	3.23

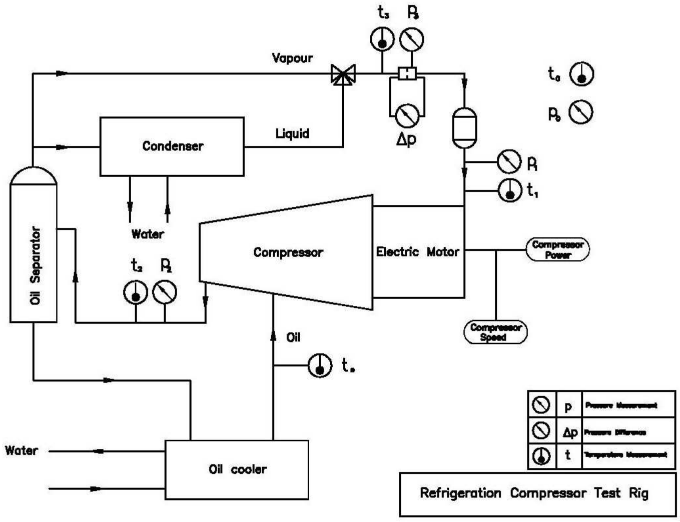


Figure 7 Layout of the industrial test rig

The results of the tests are given in Table 1, from which it can be seen that the new compressor performed almost 9 % better than the reference one.

CONCLUSION

Efficient operation of refrigeration screw compressors is highly dependent on the rotor profile and clearance distribution. Nonetheless, other compressor components, such as the housing ports, bearings, seals and the lubrication system must be properly designed and every detail counts in obtaining the best result. It is widely, but erroneously, believed that beyond this, little can be done to improve screw compressor efficiencies in refrigeration systems. In fact there is still some scope for the design of a better product by the use of optimization procedures. Thus the use of new rotor generation procedures and rotor and compressor design optimization for a specified compressor duty will result in a specialized compressor design with stronger but lighter rotors, higher displacement and more compact and efficient machines for each application. Based on this, new advanced refrigeration compressors have been designed and built which are distinctly superior to their predecessors.

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