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NEW DESIGN AND ROTOR RETROFIT TO IMPROVE CAPACITY AND PERFORMANCE OF REFRIGERATION SCREW COMPRESSORS

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ABSTRACT

As manufacturing methods improve, screw compressor components can be made more accurately and hence with smaller operating clearances. This gives the opportunity both to improve their efficiency and reduce manufacturing costs. However, to obtain the full benefit of these developments, improved design procedures are needed to model the processes within these machines more precisely, than those based on the simplified assumptions traditionally used. Such methods need to be based on the numerical solution of the differential equations describing the instantaneous heat and fluid flow through the compressor, using real working fluid properties, combined with multi variable optimisation procedures to determine the optimum rotor profile for each application. This is particularly the case for compressors in refrigeration and air conditioning systems, where the traditional approximations of integral relationships for the compression process and the working fluid being modelled as a perfect gas, are both very gross. Such analytical procedures take a long time to develop and are best done by academia in collaboration with industry. An example of the results of such a cooperative activity in screw compressor development is given in this paper, where improved design procedures, based on advanced software packages, were used both for rotor retrofitting and completely new compressor designs. Details are given of the results of both these functions in two specific cases.

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

1. INTRODUCTION

Screw compressors are widely used in refrigeration and air conditioning systems, where the market is highly competitive, especially for oil-flooded machines. Consequently, compressor manufacturers must pay attention to every detail in the design to maximize efficiency while minimizing the cost. Major improvements have already been made by cumulative small gains, using traditional design procedures. However, these have reached their limit. Further advances are still possible, but only by adopting new and more advanced methods. These can lead to further small but still significant gains and a better final product.

The main area in which such improvements are possible is in rotor profile design, based on multi variable optimisation of all the component features which define the shape and determine the compressor operating characteristics. By this means modern rotor profiles are uniquely defined for each application and achieve increased compressor capacity and higher efficiencies for the same diameters and tip speeds.

Although the efficiency of screw compressors is mainly dependent on the rotor profile and clearance distribution, other items, such as the housing ports, the seals and the 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. When all these items are taken into consideration simultaneously, the resulting compressor design will be unique for each application and will be stronger, with lighter rotors and a higher displacement. Consequently the machine will be more compact and hence cheaper, as well as being more efficient.

For some applications, the design of a complete new machine may be too costly to be justified. In those cases, Retrofit rotor design can be used to advantage. By this means the compressor capacity and efficiency can still be significantly improved, while retaining the same rotor configuration, centre distance and outer rotor diameters but modifying all the other profile parameters.

Improved screw compressor design procedures, undertaken over a long period by university research workers, resulted in user friendly software packages which are now being implemented by industry. Some details are given here of these procedures and their accompanying program packages, and how these were successfully applied to refrigeration systems both for a retrofit and a completely new machine design.

2. DEVELOPMENT OF REFRIGERATION SCREW COMPRESSORS

Historically, SRM introduced screw compressors for refrigeration systems in the early nineteen seventies and the majority of them are still being produced under their licence. From the outset, improvements were continually introduced to make these machines more compact, reliable and efficient. However, details of these improvements were published only much later, mainly by compressor manufacturers and university research workers. *Mossemann of Kuhlautomat, 1975*, [3] was the first of these to bring his design work to the attention of the refrigeration community. *Sauls of Trane*, wrote a series of papers between 1992 and 2002, of which [4] is a typical example, which defined important routes and suggested actions needed to produce better refrigeration screw compressors. *Fujiwara et al of Hitachi and the Muroran Institute of Technology, 1994*, [2] published their important work on screw compressors, which has been much used in the development of their refrigeration compressors. Later *Fleming et al of Strathclyde University, 1998*, [1] reviewed contemporary design methods and procedures resulting from their previous comprehensive work in screw compressor modelling and optimization, which paid special attention to refrigeration compressors. Recently *Xing of Xi'an Jiaotong University 2000*, [7] published his book on screw compressors in Chinese, a careful examination of which can lead to the deduction of almost all that is needed for the up to date design of refrigeration screw compressors. The influence of his research group can already be detected in the newest Far Eastern refrigeration compressor designs.

The most recent developments in screw compressors and new procedures in screw compressor design have been summarised by *Stosic et al of City University, London, 2005* [6]. The authors of this paper have used these procedures for new compressor designs, as well as for rotor retrofits to existing refrigeration screw compressors with the aim of improving performance in a series of industrial applications.

The most important requirements for refrigeration and air conditioning system compressors are that they must be quiet, efficient, reliable and durable. To perform without causing excessive noise, a compressor must run slowly. Speed variation is gradually becoming the preferred method of controlling the capacity of refrigeration compressors and this also requires good efficiency at low speeds. This implies that a refrigeration screw compressor must be built with small rotor and housing clearances to maintain low leakage and consequent high efficiency under these conditions. The manufacture of rotors by grinding, especially with simultaneous measurement and control of the profile, today makes it possible to maintain profile tolerances of 5 μm , which, in turn, enables the clearances between the rotors to be kept under 15 μm . With such small clearances, rotor contact is very likely and hence a profile must be generated which avoids damage or seizure should hard contact occur. This is best achieved by the use of rotors with involute lobes and low contact stresses.

Increasing the number of 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 contact. This reduces vibration and thus minimizes 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 decreases as the number of rotor lobes is increased. Therefore, such compressors are less efficient. Also, more lobes increase the cost of manufacture.

To minimise the higher leakage rates associated with the relatively slow rotational speeds of refrigeration compressors, their rotors are profiled for the smallest possible blow-hole area. 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.

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, thereby reducing the compressor displacement.

Another means of minimising compressor noise is to maintain a relatively large female rotor torque and thereby avoid its reversal during the compression cycle. Rotor contact is thereby made only on one pair of surfaces at all times, thus preventing what is commonly known as rotor rattle. Unfortunately, inducing a high positive female torque simultaneously results in weaker female rotor lobes and reduced rotor displacement. Consequently, when female rotors are designed for high positive torque, flow capacity is reduced and rotor contact forces become high. This results in reduced volumetric, adiabatic and mechanical efficiencies. An alternative approach is to aim for, so called, negative torque, which is to keep the rotors in permanent contact, but on their flat side. Since the sealing line on the rotor flat side is longer, this approach simultaneously results in decreased interlobe leakage, a thicker female rotor lobe and, consequently, increased rotor displacement. The net effect of this technique is to achieve higher compressor flows and efficiencies.

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 rotors are regularly profiled with 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.

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 works, the role of the other is usually to prevent rotor bounce in the axial direction.

One additional important design aspect is not widely appreciated. This is that if bearing clearances are not taken into account during the design process, small rotor clearances and high pressure loads together cause contact due to the resulting rotor displacement. This contact may occur between the rotor tips and the housing unless the bearing centre distance is smaller than that of the rotor housing. To maintain the rotor interlobe clearance as small as possible, the bearing centre distance must be still further reduced.

3. OPTIMIZATION OF REFRIGERATION COMPRESSORS

As shown in the last section, even a simple analysis of rotor behaviour in refrigeration compressors shows that a number of desirable rotor characteristics lead to conflicting design requirements. This implies that simultaneous optimization of all the variables involved in the design process must be performed to obtain the best possible compressor performance.

Full multivariable optimization of screw compressor geometry and operating conditions was performed, for the design studies described here, to establish the most efficient compressor designs for their given duties. This was achieved with a computer 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, including the throughput cross section, displacement, sealing line lengths and leakage flow cross section areas were calculated from the rotor transverse plan coordinates and rotor length and lead, together with the suction and discharge port coordinates. These were used later as input parameters for calculation of the screw compressor thermodynamic process. The compressor geometry had to be recalculated for any variation of the input parameters. The compressor built-in volume ratio was also used as an optimization variable. Computation of the instantaneous cross-sectional area and working volume was then calculated in terms of the rotation angle.

The procedure started with the determination of the suction pressures, which were maintained at the standard air conditioning and refrigeration evaporation conditions of +5, -15 and -35 °C respectively, and the discharge pressure corresponding to standard condensation temperatures of 30, 40 and 50 °C. All the compressors were oil flooded. A choice of working fluids was possible, determined by the use of a set of real fluid property subroutines. This included the possible use of premixed fluid mixtures as well as the creation of customised mixtures of refrigeration fluids and has become possible by use of the NIST (National Institute for Standards and Technology) property routines. These include virtually all known pure refrigerants and their mixtures.

The fluids used in this case were R-717 and R-134A. The centre distance and male rotor outer diameters were kept constant for all compressors at 90 and 135 mm respectively. Rack generated profile rotors in 4/6, 5/6 and 6/7 configuration were used to show how optimization permits both, better delivery and higher efficiency for the same tip speed. It was thereby shown that a different rotor design is required for each application. Four distinctive rotor profiles were obtained for different applications, namely, a 5/6 rotor for air conditioning and light refrigeration, a 6/7 rotor for heavy refrigeration duty, while both a 4/6 and a 5/6 were optimised to meet both these duties. As a result a series of optimized rotors was calculated and the data obtained were used for the further compressor design.

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 presented. These include 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. A Box constrained simplex method was 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 minimum compressor specific power. This was later compared with the results of previous calculations and then their minimization was performed.

4. ROTOR RETROFIT FOR IMPROVED PERFORMANCE

Improvement of the compressor performance can be achieved by replacing the old rotors by the new ones with better characteristics, while all other compressor characteristics remain the same. New highly efficient rotors, based on rack generated profiles, were used here to replace standard SRM D type rotors in sizes between 160 and 312 mm to maximise the compressor efficiencies. The screw compressors chosen for retrofit were standard open compressors with both, a capacity sliding valve and a 'variable' built-in volume sliding valve, each independent of the other.

Table 1: Experimental Comparison of Compressor Performance with Retrofit Rotors

Standard Rotors			
Evaporation/Condensation Temp	-15/30°C	-35/35°C	
Shaft Speed [rpm]	2920	2920	
Refrig Capacity [kW]	626	216	
Motor Power [kW]	178	156	
COP	3.523	1.383	
Optimised Rotors			
Evaporation/Condensation Temp	-15/30°C	-35/35°C	0/35°C
Shaft Speed [rpm]	2920	2920	2920
Refrig Capacity [kW]	669	243	1187
Motor Power [kW]	182	168	245
COP	3.671	1.486	4.98
Improvement Old/New	4.2 %	7.5 %	-

The new rotors were developed with the aid of the software design package described in the last section. All the definitive dimensions of the original rotors, such as centre distance, outer and root diameters, rotor length and average clearance were retained. Additionally, the favourable features of the new rotor designs permit their clearances to be reduced, if required, with no increase in manufacturing cost. All the other compressor components

remained unchanged and no changes of any kind were required in the housing and bearing design. A comparison between the new and old rotors is presented in Fig. 1.

One set of experimental results for the rotor retrofit in a 4/6-204 mm unit is presented in Table 1, obtained on the industrial test rig with ammonia. The refrigeration compressor, which was integrally mounted above the oil separator, is shown in Fig 2.

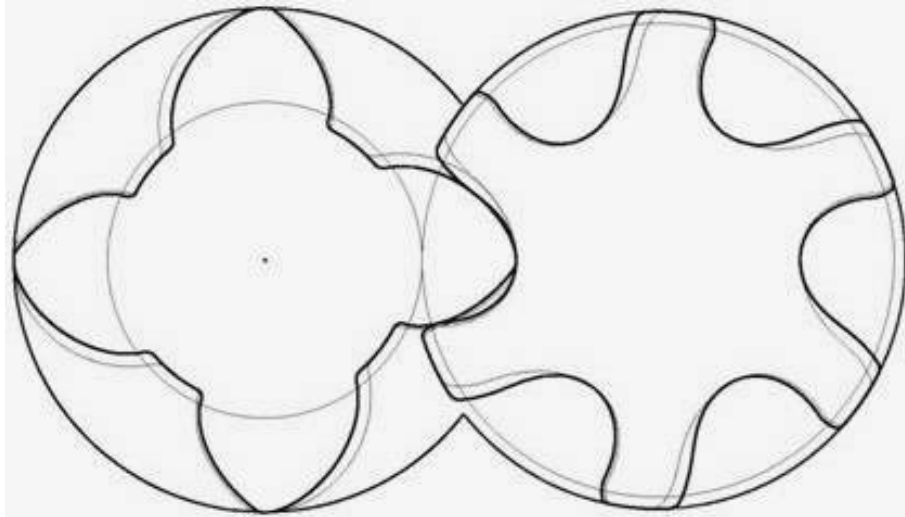


Fig. 1 Retrofit rotors optimised for general refrigeration duty, bold line compared with the old rotors, light line



Fig. 2 Compressor and prototype rotors in the industrial premises

5. NEW COMPRESSOR DESIGN FOR IMPROVED PERFORMANCE

A refrigeration screw compressor was designed from first principles based on a 5/6 rotor configuration with the main rotor outer diameter of 106 mm and an L/D ratio of 1.55. This is presented in Fig 3. The compressor is semi-hermetic with a gear multiplier for coupling with a 60 or 90 kW motor for operation at 6000 and 9000 rpm respectively and cooled by superfeed vapour or liquid injection. A photograph of the compressor is presented on the left of Fig 4. The liquid pipe for the motor cooling can be identified from it.

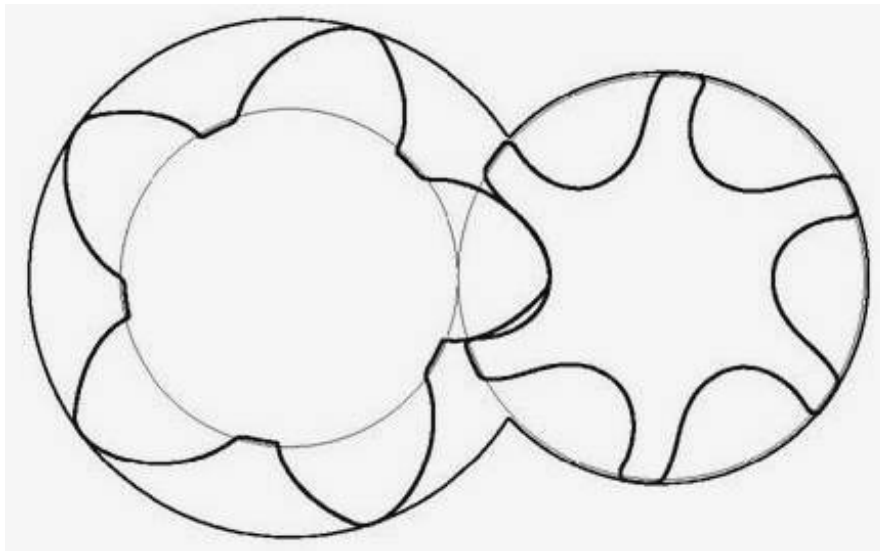


Fig. 3 Screw compressor rotors optimised for air conditioning and light refrigeration duty

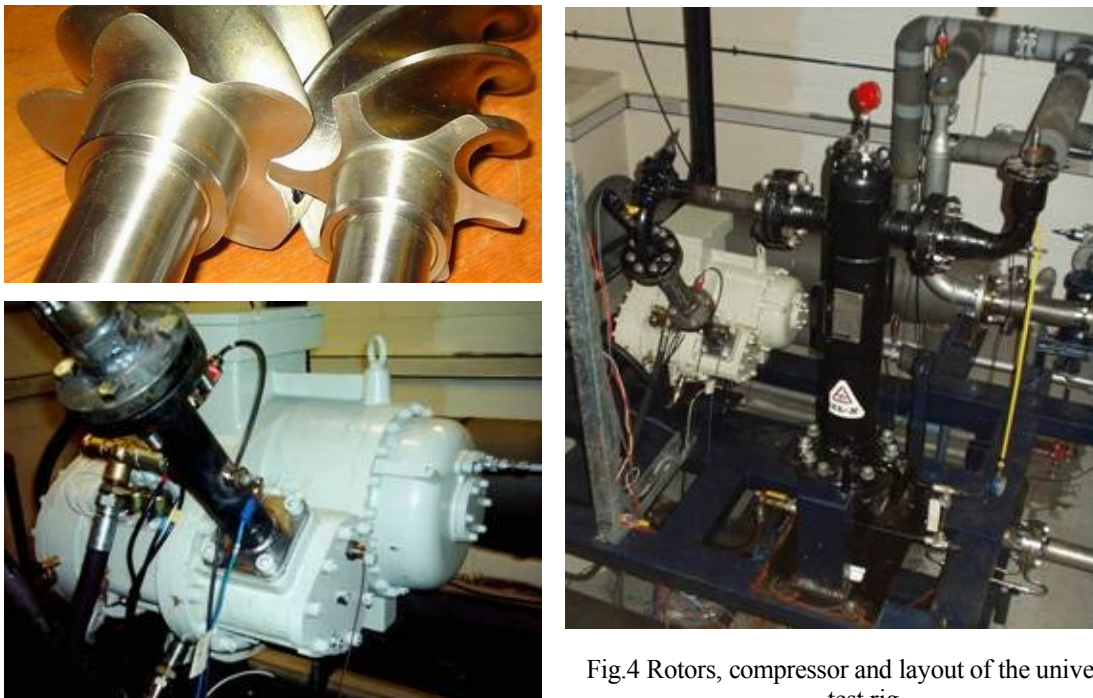


Fig.4 Rotors, compressor and layout of the university test rig

The compressor was tested simultaneously both on the industrial and the university experimental facilities. A schematic diagram of a refrigeration fluid gas rig installed at the university is shown in Fig 4. This test rig 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 absorbed by the condenser. This results in a fairly compact test rig. Moreover, such a test rig is far more flexible and controllable than an ordinary refrigeration plant allowing a wider range of suction and discharge pressure to be used virtually independently of the condenser and evaporator temperatures.

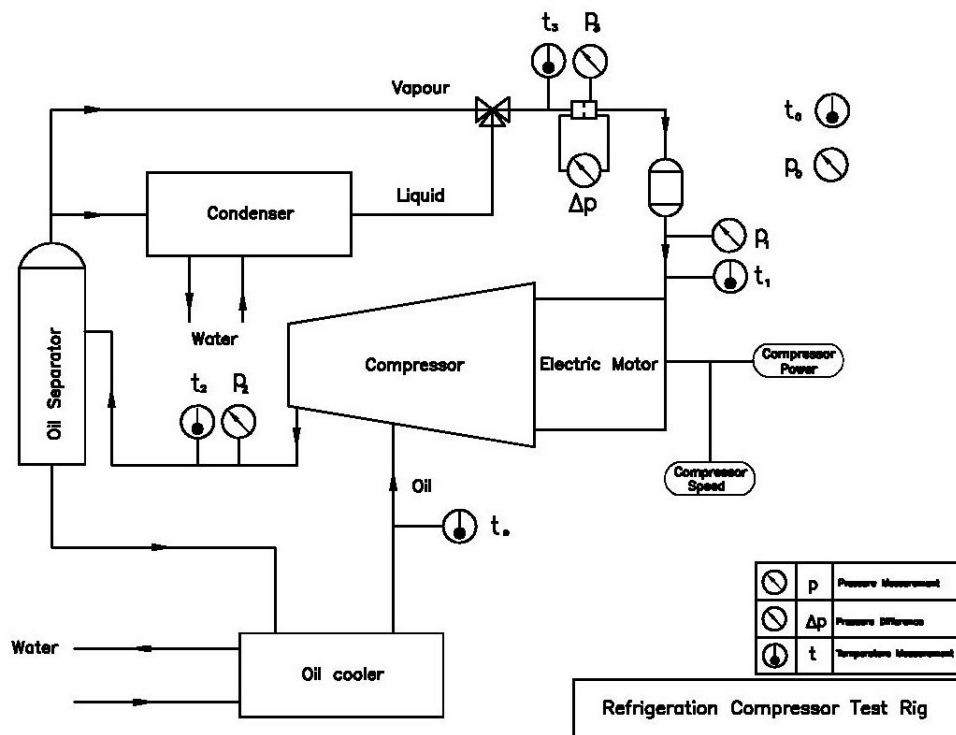


Fig. 5 Layout of the university test rig

Table 2 Comparison of compressor performance

New Compressor			Reference Compressor			COP Improvement: New/Reference [%]
Rotor Speed [rpm]	COP	Refrig Capacity [kW]	Rotor Speed [rpm]	COP	Refrig Capacity [kW]	
6011	4.20	210	5898	4.25	194	-1.1
5046	4.37	172	4998	4.19	149	4.3
4046	4.46	136	3990	3.88	120	15
3064	4.31	100	3003	3.67	86.4	17
1985	3.89	63.6	2016	3.02	52.0	29

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 is estimated by measuring the current and voltage

supply to the variable frequency converter. This takes account of all electrical losses. Specific power is calculated as the ratio of the compressor power to the flow rate.

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

As can be seen from Table 2, the maximum improvement in the COP obtained with the new compressor was nearly 30 % more than that of the reference compressor.

6. CONCLUSION

Efficient operation of refrigeration screw compressors is highly dependent on the rotor profile and clearance distribution. However, other compressor components, such as housing ports, bearings, seals and the lubrication system must be properly designed and every detail counts in obtaining the best results. 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 meet the continuing market requirement for improved products.

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