# RETROFIT 'N' ROTORS FOR EFFICIENT OIL-FLOODED SCREW COMPRESSORS

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**ABSTRACT**

The market for twin screw compressors is highly competitive, especially in compressed air and refrigeration systems, and new designs are continually being introduced which are more efficient and cost effective than their predecessors. However, because of the high cost of development of new machines, manufacturers seek to maintain their existing designs for as long as possible. Closer study of many of the older designs has shown that in the majority of cases, all that is required to bring them up to date is to change the rotor profile to one of more up to date type.

This paper describes the design of a family of highly efficient screw compressor rotors based on rack generated profiles which were used to replace standard asymmetric rotors in sizes between 102 and 204 mm and thereby maximise the compressor efficiencies. They were developed with the aid of a software package which takes into account almost every aspect of thermodynamic and geometric modelling. All the definitive dimensions of the original rotors, such as centre distance, outer and root diameters, rotor length and average clearance were retained. However the favourable features of the new 'N' rotors 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.

Experimental tests showed that in all cases the introduction of the new rotors increased the compressor displacement by more then 6.5 % and raised the adiabatic efficiency by not less than 2.5 %. This led to an increased power input requirement of 2.6 %. However, the existing drive motors were sufficient to meet this additional need.

# INTRODUCTION

Screw compressors are basically simple rotary positive displacement machines, capable of high speed operation over a wide range of operating pressures and flow rates at high efficiencies. Their spectacular increase in use during the past twenty five years is mainly due to the development of machine tools capable of maintaining linear tolerances of less than 10μm through high accuracy profile milling or grinding. Rotors can thus be manufactured with interlobe clearances of 30-50 mm at an economic cost, thereby reducing internal leakages to a small fraction of their values in earlier designs. Consequently, twin screw compressors are not only smaller than other types but also more efficient. Typically, for oil flooded air compression applications, the volumetric efficiency of these machines now exceeds 90 % and the specific power input has been reduced to values which were regarded as unattainable only a few years ago.

Despite the popularity of these machines, public knowledge of the scientific basis of rotor profile generation is still limited. Three textbooks, published in Russian, cover this topic well. Sakun, 1960, gives full details of the generation of circular and elliptic profiles, as well as a Russian asymmetric profile named SKBK. Andreev, 1961, makes a contribution in the field of rotor tool profile generation, while Amosov et al, 1977, was the first to demonstrate how to generate the SRM unsymmetric profile. In textbooks published in German: Rinder, 1979, presents a method based on gear theory to reconstruct the SRM unsymmetric profile. Konka, 1988, published some engineering aspects of screw compressors. There are two textbooks in English which deal with screw compressors; namely: O'Niell, 1993, on industrial compressors and Arbon, 1994, on rotary twin shaft compressors. A large number of patents have been published on various aspects of screw compressors and especially their profiles. The most significant of them were filed by SRM, a Swedish compressor company who were the pioneers in research, development and design of screw compressors. These are the symmetric profile, 1946, the unsymmetric, 1970 and the 'D' profile, 1982. The last two of these became the industrial standards in screw compressor profile generation.

The overwhelming majority of screw compressor manufacturers became licensees of SRM, and a considerable number still remain as such. However, as time passed, many of the larger companies decided to perform their own profile and compressor design. Small and medium companies, who could not afford either to become licensees or develop their own profiles are limited to manufacture compressors based on expired rotor profile patents and are thus out of touch with more recent developments.

The search for new profiles has been both stimulated and facilitated by recent advances in mathematical modelling and computer simulation. These analytical methods may be combined to form a powerful tool for process analysis and optimisation and thereby eliminate the earlier approach of intuitive changes, verified by tedious trial and error testing. As a result, this approach to the optimum design of screw rotor lobe profiles has substantially evolved over the past few years and is likely to lead to further improvements in machine performance in the near future.

In 1995, in recognition of the needs of compressor manufacturers, a Centre for Positive Displacement Compressor Technology was established at City University, London, to carry out research and provide a service to manufacturing companies in all aspects of compressor design and development. Ongoing work on rotor profile generation soon led to the development of a new family of shapes which were classified as ‘N’ profiles. An early request was to design a suitable twin screw compressor which, would be competitive in performance but not too difficult to manufacture by a company with only limited experience in close tolerance machining and assembly. This was recognised as an opportunity to try the 'N' profile in practice, since its theoretically desirable features enabled the rotors to be designed with relatively large clearances while still maintaining a high efficiency. Subsequently, a family of ‘N’ rotors was designed with a 4/6 lobe configuration in the main and gate rotors for standard rotor diameters of 102, 128, 163 and 204 mm. These were intended to replace those in an existing family of compressors and thereby improve both their effectiveness and efficiency.

## THE DESIGN OF ‘N’ PROFILE ROTORS

As rotor clearances become smaller, internal leakage rates are reduced. Further improvements in compressor performance are then only possible by the introduction of more refined design principles. The main requirement is to improve the rotor profiles so that flow area through the compressor is maximised while the leakage path is minimised and internal friction due to relative motion between the contacting rotor surfaces is made as small as possible. The design must therefore start with the choice of a good rotor profile. However, in addition, the designer must be able to predict accurately the effects on performance of changes in the relative proportions of its elements and in such parameters as clearance, oil or fluid injection position and rate, rotor diameter and proportions and speed.

Although it may appear that the principles of rotor profiling are well established, there is scope for substantial improvements. The most promising approach seems to be the method of rack profile generation. This enables rotors to be constructed, which are both lighter and stronger with higher throughput and lower contact stress than existing designs. The latter parameter enables a lower viscosity fluid than oil to be used for lubrication.

The optimum rotor profile has a large flow cross section area, a short sealing line and a small blow-hole area. The larger the cross section area the higher the flow rate for the same rotor sizes and rotor speeds. Shorter sealing lines and a smaller blow-hole reduce leakages. Higher flow and smaller leakage rates both increase the compressor volumetric efficiency, which is the rate of flow delivered as a fraction of the sum of the flow plus leakage. This in turn increases the adiabatic efficiency because less power is wasted in the compression of gas which is recirculated internally.

The optimum choice between blow hole and flow areas depends on the compressor duty since for low pressure differences the leakage rate will be relatively small and hence the gains achieved by a large cross section area may outweigh the losses associated with a larger blow-hole. Similar considerations determine the best choice for the number of lobes since fewer lobes imply greater flow area but increased pressure difference between them.

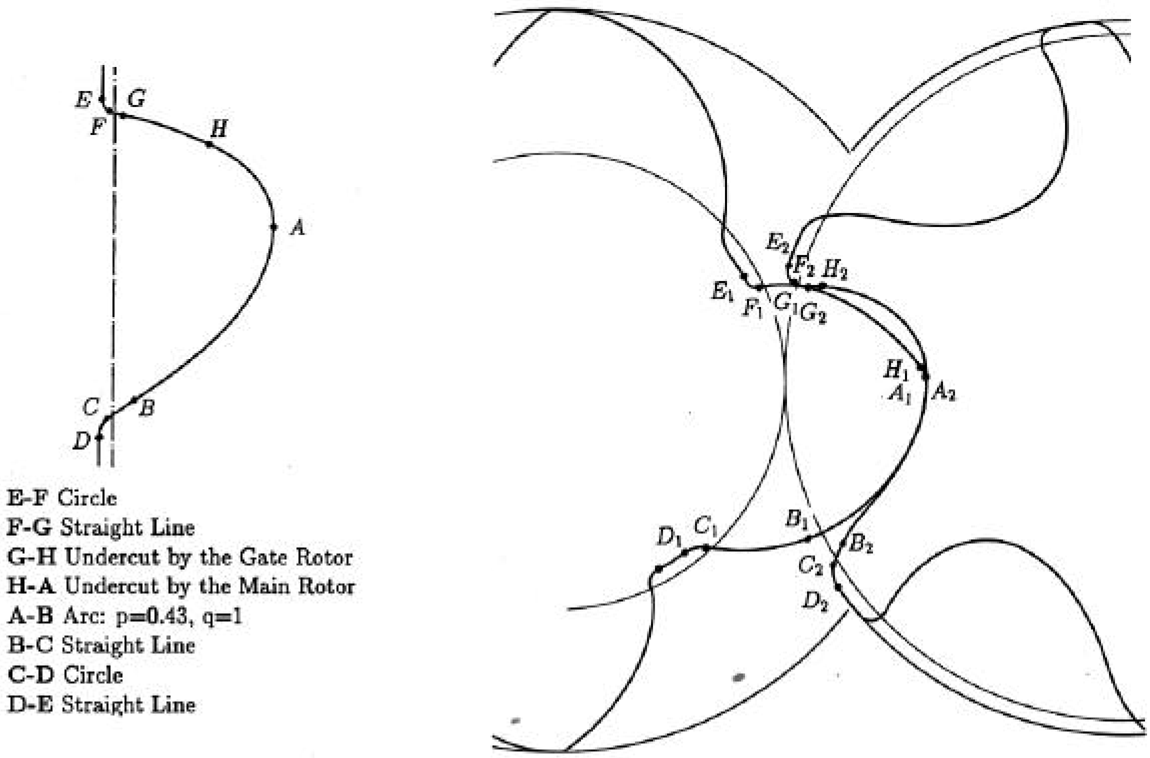


Fig. 1 Details of ‘N’ Rotors

As precise manufacture permits rotor clearances to be reduced, despite oil flooding, the likeliehood of direct rotor contact is increased. Hard rotor contact leads to deformation of the gate rotor, increased contact forces and ultimately rotor seizure. Hence the profile should be designed so that the risk of seizure is minimised.

A pair of a rack generated rotors, (Stosic 1996) is shown in Fig 1. The selection and distribution of generating curves on a rack which was used to create these rotors give a larger cross section area with stronger gate rotor lobes than any other known screw compressor rotor. Additional favourable features characterize these rack generated rotors: they maintain a seal over the entire contact length while maintaining a small blow-hole and the two contact bands, close to the pitch plane are straight lines on the rack which generate involutes on the rotors. Hence the relative motion between the rotors is insensitive to small variations in the rotor centre distance. ‘N’ rotors have already been used for a number of successful compressor designs. The high efficiencies obtained on tests confirm their validity. A full scale presentation of ‘N’ rotors is given in Stosic and Hanjalic, 1997, and one of the applications of ‘N’ rotors is given in Stosic et al 1997.

Although advanced rotor profiles are a necessary condition for a screw compressor to be efficient, all other components must be designed to take advantage of their potential if the full performance gains are to be achieved. A screw compressor, especially of the oil flooded type, which operates with high pressure differences, is heavily loaded by axial and radial forces which must be transferred to the housing by the bearings.

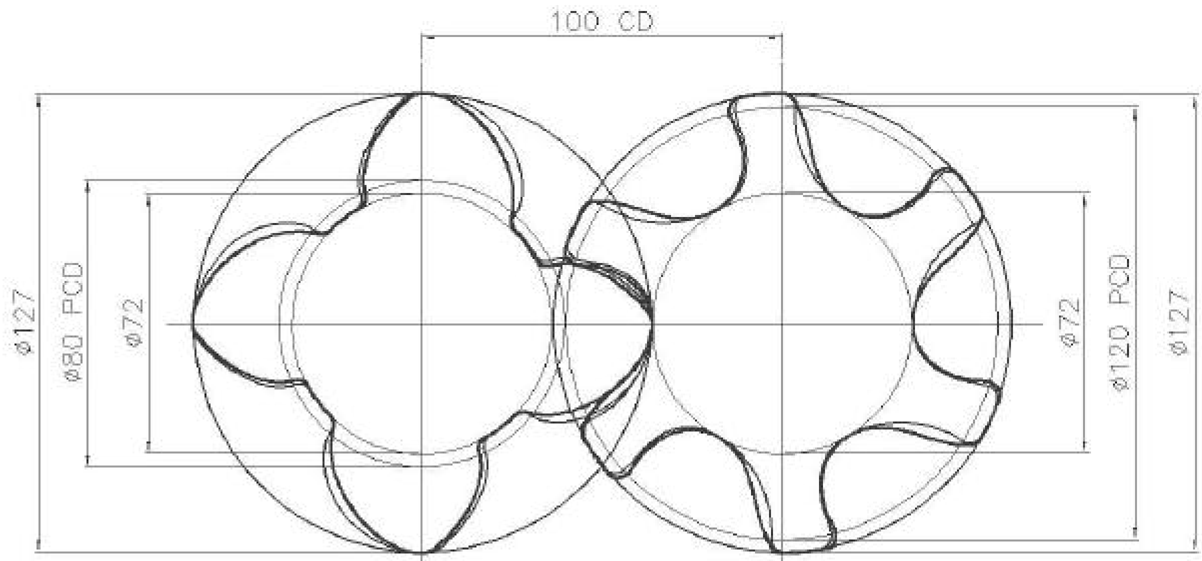


Fig. 2 'N' Rotor Replacement (bold) for Asymmetric Rotors (light)

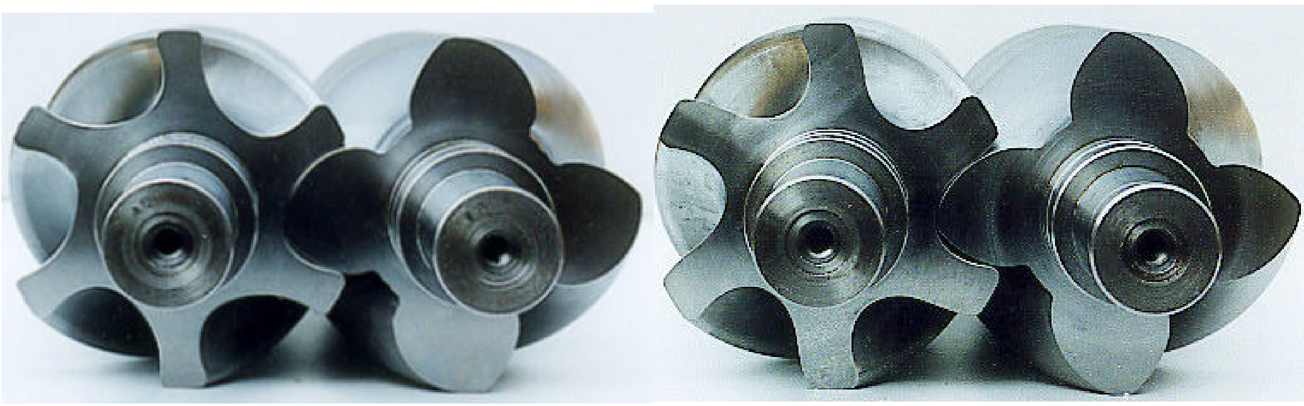

Fig. 3 Comparison of Asymmetric Rotors (left) and 'N' Rotor Replacement (right)

Fig.4 Compressor with Replacement Rotors installed

In the present case, apart from the rotors, the manufacturer was satisfied with their current compressor design and it was decided that the rotors would be designed within the existing rotor envelope, retaining their size, centre distance, outer and root diameters, rotor length and average clearance. However, the clearance distribution was modified to exclude the possibility of rotor seizure under conditions of reduced centre distance and thermal distortion. Since ‘N’ rotors allow the clearance to be smaller than in alternative types, this option was left open for further rotor refinements. Maintaining equal clearances in both the new and old rotors enabled a reliable comparison to be made between the two rotor designs.

In Fig. 2 a drawing is given of the new and old rotors of the 127 mm compressor and a photograph of the ‘N’ and Asymmetric rotors is given in Fig 3. The increase in displacement for the new rotors is 5.4 % and at the same time the sealing line length increase is 4.5% relative to the old ones. A better torque distribution on the new rotors ensured a lower contact stress between the rotors and minimised rattling. Both features enhanced the rotor reliability. The rotors were manufactured and directly installed in an existing compressor. An overall view of the compressor is presented in Fig 4.

## PERFORMANCE MEASUREMENTS WITH NEW AND OLD ROTORS

A compressor was tested on the standard test rig for screw compressor acceptance tests. The following parameters were measured directly.

$p_0$ - Atmospheric pressure [mmHg]
$p_1$ - Suction pressure [bar]
$p_2$ - Discharge pressure [bar]
$p_3$ - Orifice plate pressure [bar]
Dp - Orifice plate pressure difference [Pa]
$t_1$ - Suction temperature [$^oC$]
$t_2$ - Discharge temperature [$^oC$]
$t_3$ - Orifice plate temperaure [$^oC$]
$t_o$ - Oil injection temperature [$^oC$]
n - Compressor speed [rpm]
N - Power [kW]

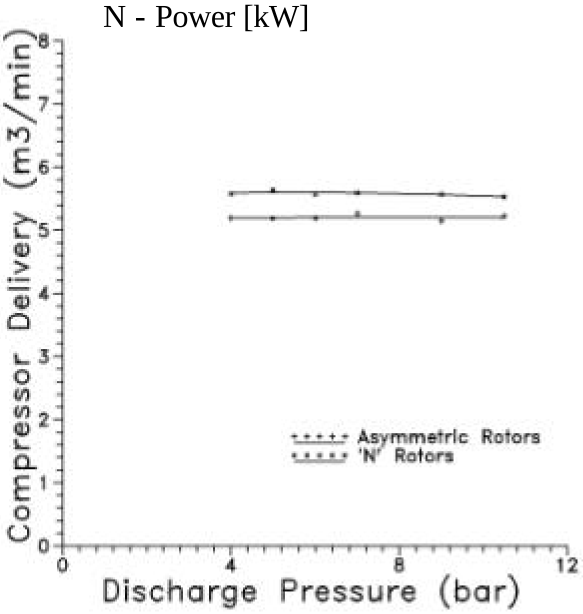

Fig. 5 Compressor Flow

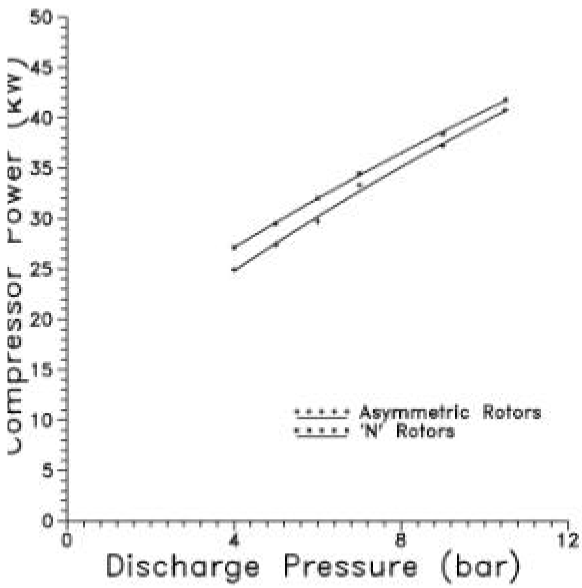

Fig. 6 Compressor Power

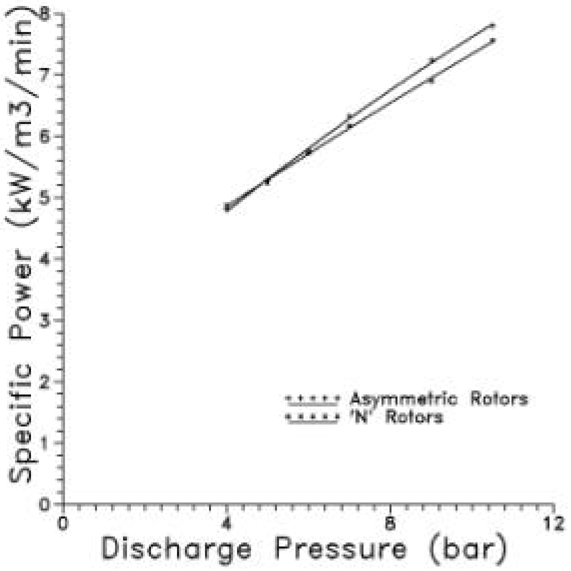


Fig. 7 Specific Power

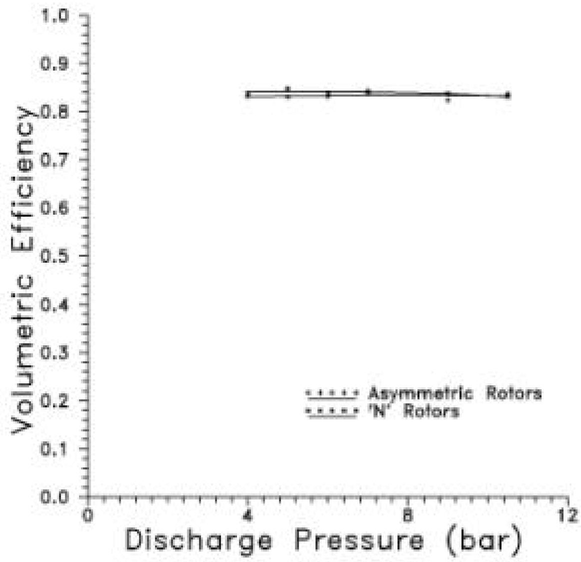


Fig. 8 Volumetric Efficiency

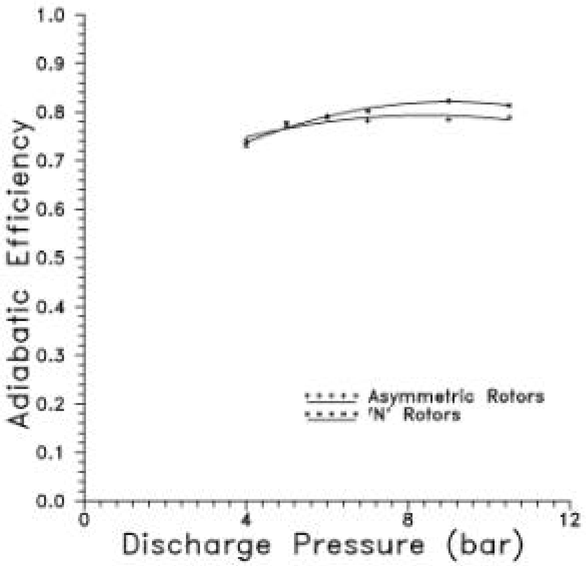


Fig. 9 Adiabatic Efficiency

Measured values were used to calculate compressor flow, power and specific power, volumetric and adiabatic efficiency. The test results are presented as compressor pressure characteristics in Figs 4 - 8 respectively. As may be observed, the compressor delivery is higher 6.5 % and the minimum specific power input with the 'N' rotors is at least 2.5 % less than that with the old ones in the same compressor.

## CONCLUSIONS

The 'N' profile in 4-6 configuration ensures a higher delivery and a better efficiency for the same clearances and tip speed, than the asymmetric profile in an oil flooded air screw compressor. The successful execution of this new compressor development has been used as an opportunity to publicise the advantages of contemporary modelling, design and manufacturing methods as well as some of the principles on which they are based. The high efficiencies obtained on test confirm their validity. It should be noted though that attention to the improvement of every detail of design such as the ports, the oil injection system and the bearings would all additionally contribute to the potential for improvement offered by the novel high throughput involute rotors.

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