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IMPROVING SCREW COMPRESSOR DISPLACEMENT AND EFFICIENCY BY INCREASING ROTOR PROFILE DEPTH

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ABSTRACT

The effect of rotor profile depth on screw compressor performance has been examined for a number of main and gate rotor configurations, all at the same main rotor diameter. Both dry and oil-flooded operational modes were analysed. It was found that increasing the rotor profile depth by reducing the inner radius leads to enhanced flow rates, with improved both volumetric and adiabatic efficiencies. As a result, a new series of rotors has been developed incorporating this feature, which has already been integrated into numerous modern rotor designs.

Keywords: Screw compressor, Efficiency, Sustainability

NOMENCLATURE

A	interlobe area [m ²]
d	rotor depth [m]
D	rotor diameter [m]
L	length scale [m]
Q	volume flow [m ³ /min]
S	sealing line length [m]
η	efficiency
Indices	
l	leakage
t	theoretical
v	volumetric

1. INTRODUCTION

Screw machines are used in a variety of systems to compress or expand a wide range of fluids. To achieve optimal performance, each application requires a tailored design and operational mode. Consequently, there has been a steady development of software to obtain the best rotor profile for each case, when all aspects of both the rotor geometry and the working fluid properties are taken into consideration simultaneously. The foundational principles for this approach were initially published by Sakun in 1960 [1]. This was followed by Amosov et al. in 1977 [2], Rinder

in 1979 [3] and Xing in 2000 [6]. Other publications including some details of compressor design were made by O'Neill in 1993 [4] and Arbon in 1994 [5].

None of these publications, as well as any other literature source studied the influence of the number of lobes and their depth upon the rotor geometrical parameters and thermodynamic performance. This is done in generic form and for the first time presented in this paper. Moreover, these findings are then confirmed by Stosic, Smith and Kovacevic in [12] through a detailed calculation on the examples of the rotor profiles viable for engineering application in the screw compressor industry.

This is presented in Figure 1 through evolution of the rotors of the same outer diameters, 127 mm with 5/7, 10/12, 20/22 and 40/42 lobes in the main and gate rotors. The rotor displacement is the interlobe area multiplied by the rotor length. The sealing line length is the length of the curves which represent the most proximate position of the rotors between themselves and between the rotors and rotor housing.

The rotor interlobe area is proportional to the rotor circumference $D\pi$ multiplied by the rotor depth d which is a difference between the outer and root rotor diameters which may be presented as $A=D\pi d$ (1). The sealing line length may be fairly well approximated by the two rotor depths added to the rotor circumference, $S=2d+D\pi$ (2).

It is obvious from Figure 1, as well from equations (1) and (2) that the interlobe area is becoming smaller with the increase of the number of rotor lobes, and it becomes zero for the infinite number of lobes when the rotor lobe depth is zero. The sealing length becomes smaller with the increase of the lobe number and reduction of the rotor lobe depth, however, contrary to the interlobe area, it tends to the finite value $D\pi$ once d approaches zero. Therefore, it can duly be concluded that the reduction of the rotor depth will lead to the reduction of the compressor flow which will become zero for a certain rotor depth even before the rotor depth becomes zero. Moreover, it can be deduced that the opposite, the reduction of the rotor lobe number and increase of

the rotor depth will result with the improvement of the compressor performance.

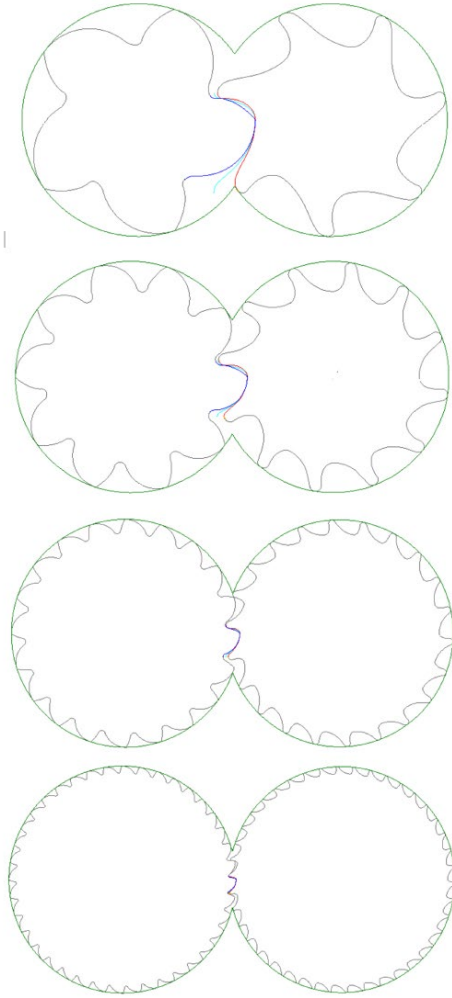


Figure 1. Evolution of screw compressor rotors with the number of their lobes, from top: 5/7, 10/12, 20/22 and 40/42

This is quantified in the present study by analysing the influence of the rotor profile depth upon the rotor geometrical characteristics and their effect on the compressor performance which was estimated by the use of SCORPATH (Screw Compressor Optimal Rotor Profiles And THermodynamics), a proprietary, experimentally-validated software package that includes nearly all facets of geometric and thermodynamic modelling. In this case, the rotor design was based on the rack principle, leading to the development of the authors' unique 'N' profiles, as detailed in Stosic and Hanjalic, 1997 [7]. This was employed to ascertain the optimal rotor dimensions and rotational speed, as well as the volume ratio and the configuration and location of the suction and discharge ports. Further details can be found in Hanjalic and Stosic, 1997 [8]. Further improvements were made by Stosic et al, 1997 [9].

However, none of these addressed the generation of rotor profiles to improve fluid displacement and, thereby both the volumetric and adiabatic efficiencies.

To optimise the design, the rotor profile must maximise the machine displacement while minimising the length of the sealing line. This is best understood by consideration of the following: If L is a characteristic length, the theoretical flow is dependent on the entrained volume, which is proportional to L^3 and the leakage flow is proportional to both the length of the sealing line and the clearances, which increase with size, i.e. to L^2 . If Q describes the volume flow, then the volumetric efficiency, η_v can be expressed as in Equation (1):

$$\eta_v = \frac{Q}{Q_t} = \frac{Q_t - Q_l}{Q_t} = 1 - \frac{Q_l}{Q_t} \approx 1 - \frac{L^2}{L^3} = 1 - \frac{1}{L} \quad (3)$$

where Q_t is the theoretical flow and Q_l is the leakage flow.

However, if the clearances are constant, the leakage losses are reduced by its inverse squared and the volumetric efficiency increases with size, as shown in Equation (2).

$$\eta_v = \frac{Q}{Q_t} = \frac{Q_t - Q_l}{Q_t} = 1 - \frac{Q_l}{Q_t} \approx 1 - \frac{L}{L^3} = 1 - \frac{1}{L^2} \quad (4)$$

The actual volumetric efficiency is likely to fall between these two extreme cases.

The adiabatic efficiency of a compressor is significantly influenced by its volumetric efficiency. Assuming all other factors remain constant, it can be inferred that adiabatic efficiency is directly proportional to volumetric efficiency. When analysing equations (1) and (2), it is crucial to identify which rotor dimension should be designated as the scale length L .

If the outer diameter of the main rotor is fixed, increasing the number of lobes will lead to an increase in the inner, root rotor diameter and consequently reducing the throughput. In the extreme case, as the number of lobes approaches infinity, the throughput will approach zero, while the sealing line will still maintain a finite value. Therefore, for a specific rotor size, decreasing the number of lobes will result in a greater rotor profile depth compared to a configuration with a higher number of lobes. This indicates that, when assessing the impact of size on volumetric efficiency, the rotor profile depth, defined as the difference between the outer and inner radii of the rotor, serves as a more accurate indicator of the rotor length scale than the outer diameter or rotor length. If the centre distance is decreased for rotors with a specified main rotor outer diameter, the rotor profile depth will increase. Consequently, rotor profile depth should be regarded as the preferred criterion for profile design aimed at producing rotors with substantial displacement and high efficiency. However, this may not hold true when it is critical to preserve other design parameters, such as rotor rigidity to minimize deflection and ensuring adequate space for bearings within the centre distance.

2. ROTOR PROFILES AND PERFORMANCE CALCULATION

A study of the effect of rotor depth variation was carried out using SCORPATH. In all cases analysed, the outer diameter of the main rotor was consistently set at 128 mm. All other profile elements, with the exception of the centre distance, were maintained constant, ensuring that clearances and blow-hole areas remained uniform across all configurations. For oil-free operation, both 3/4 and 3/5 configurations were selected as the primary and gate lobe numbers, whereas for oil-flooded operation, the configurations were designated as 4/5, 4/6, and 5/6. The inlet pressure and temperature were standardized at 1 bar and 20°C, respectively, while the outlet pressures were established at 3.5 bar for oil-free compressors and 8 bar for oil-flooded compressors, using air as the working fluid. All intermediate parameters were adjusted through performance calculations. In every instance, the rotors were generated as outlined in reference [10], and the compressor performance was evaluated based on the modelling principles detailed in reference [11].

3. PRESENTATION OF RESULTS AND DISCUSSION

Each of the five rotor configurations was designed with both standard and deeper profiles. The deeper profiles were created by decreasing their centre distances, ensuring that all parameters remained constant except for the main rotor profile addendum, which was modified in each instance to keep the main rotor outer diameter at 128 mm.

In the plots and performance diagrams, the profiles with nominal centre distance are illustrated with the thin dashed line while the profiles with reduced centre distance and deeper interlobes are represented by thick full lines.

Flow and specific power in diagrams are represented as functions of speed.

3/4 Configuration

The 3/4 rotor are presented in Figure 2. Their overall rotor dimension measures for the centre distance 90 mm is 198 mm, while the rotor depth is 30 mm. The displacement is 1.84 l/rev for the length-to-diameter ratio (L/D) 1.7. The reduced centre distance 81 mm features the overall rotor dimension 193.5 mm and rotor depth of 31.5 mm. This configuration has displacement of 1.88 litres per revolution. The compressor performance, characterized as oil-free machines operating within a pressure range of 1 to 3.5 bar, is presented in Figures 3 and 4.

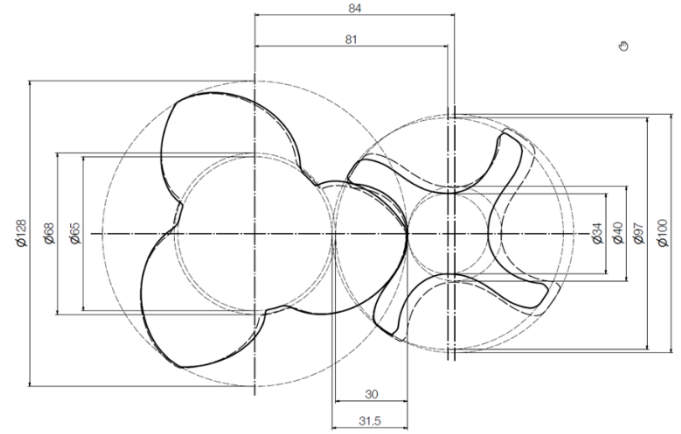


Figure 2. 3/4 rotors

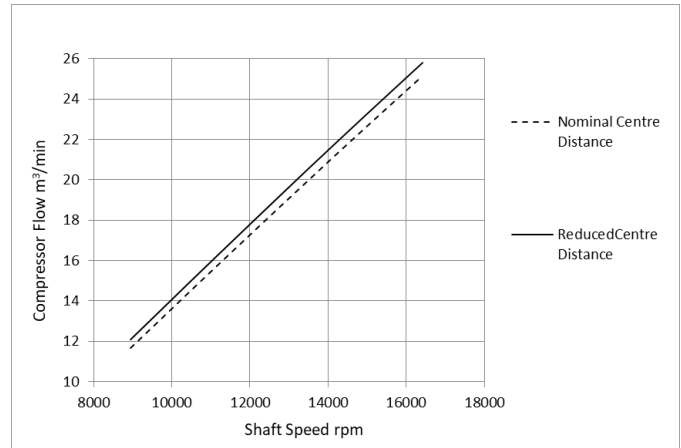


Figure 3. 3/4 rotors: flow as function of shaft speed

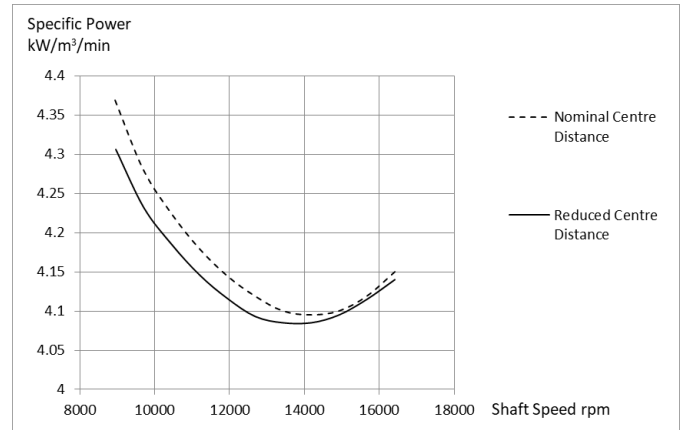


Figure 4. 3/4 rotors: specific power as function of shaft speed

3/5 Configuration

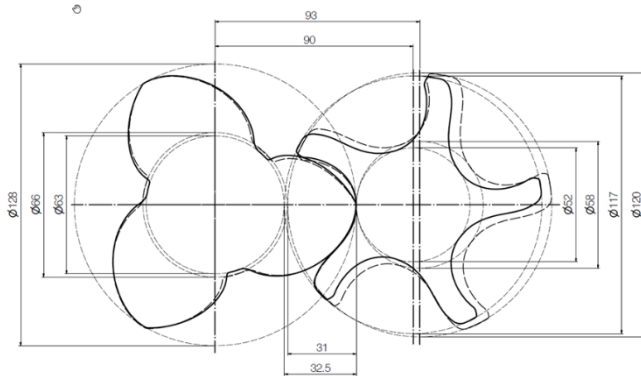


Figure 5. 3/5 rotors

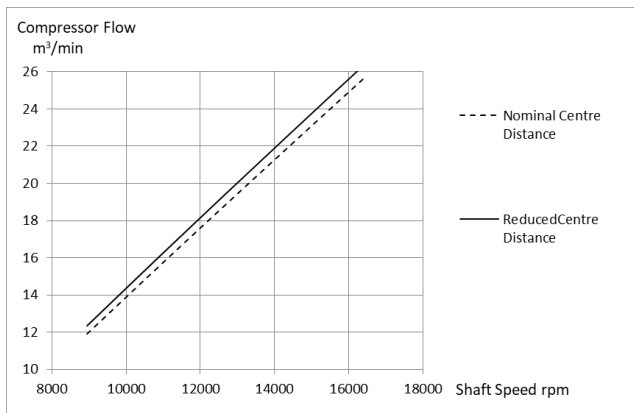


Figure 6. 3/5 rotors: Flow in function of shaft speed

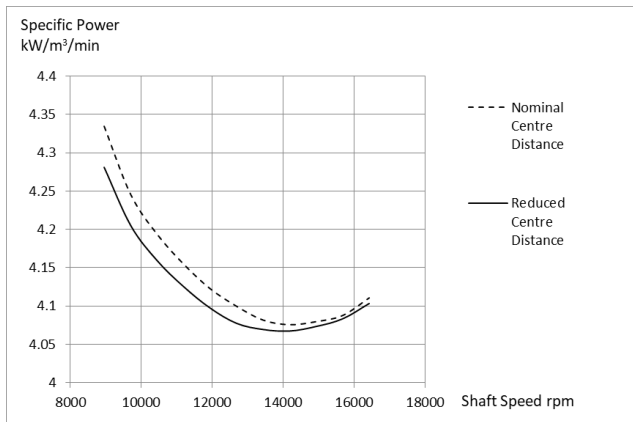


Figure 7. 3/5 rotors: Specific power in function of shaft speed

The 3/5 profiles plot is illustrated in Figure 5. It features a nominal centre distance of 93 mm, a main rotor outer diameter of 128 mm, and a gate rotor outer diameter of 120 mm, resulting in an overall rotor dimension of 217 mm. The main rotor inner diameter measures 66 mm, while the gate rotor inner diameter is

58 mm, with a rotor depth of 31 mm. The aspect ratio (L/D) is 1.7, and the displacement is 1.92 litres per revolution.

Additionally, the rotors with a reduced centre distance of 90 mm, maintaining a main rotor outer diameter of 128 mm and a gate rotor outer diameter of 117 mm, leading to an overall rotor dimension of 212.5 mm are also presented in this figure. The main rotor inner diameter is 63 mm, the gate rotor inner diameter is 52 mm, and the rotor depth is 32.5 mm, with the same aspect ratio of 1.7 and a displacement of 1.96 litres per revolution.

The performance of oil-free compressors utilizing these rotors, operating within a pressure range of 1 to 3.5 bar, is presented in Figures 6 and 7.

4/5 Configuration

The 4/5 profiles plot is shown in Figure 8. This configuration has a nominal centre distance of 89 mm, a main rotor outer diameter of 128 mm, and a gate rotor outer diameter of 101 mm, resulting in an overall width of the compressor rotors assembly of 203.5 mm. The main rotor inner diameter is 77 mm, while the gate rotor inner diameter is 50 mm, with a rotor depth of 24.5 mm. The aspect ratio (L/D) is 1.55, and the displacement is 1.54 litres per revolution.

The rotors with a reduced centre distance of 86 mm, which maintain a main rotor outer diameter of 128 mm and a gate rotor outer diameter of 98 mm, leading to an overall width of the compressor rotors assembly of 199 mm are also presented in this figure. The main rotor inner diameter is 74 mm, the gate rotor inner diameter is 44 mm, and the rotor depth is 27 mm, with a displacement of 1.58 litres per revolution.

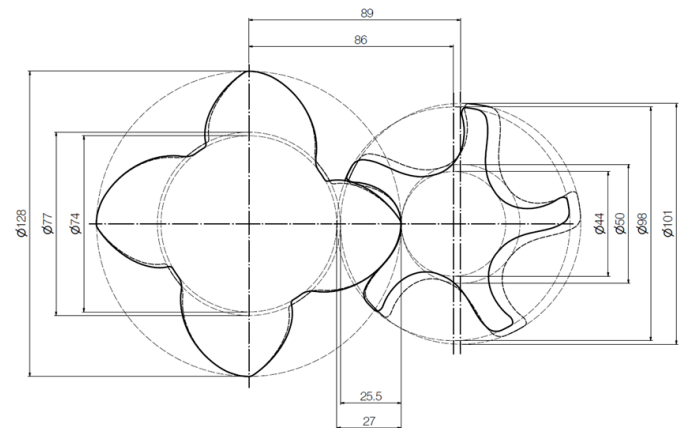


Figure 8. 4/5 rotors

The performance of oil-flooded compressors with these rotors, operating between 1 and 8 bar, is illustrated in Figures 9 and 10.

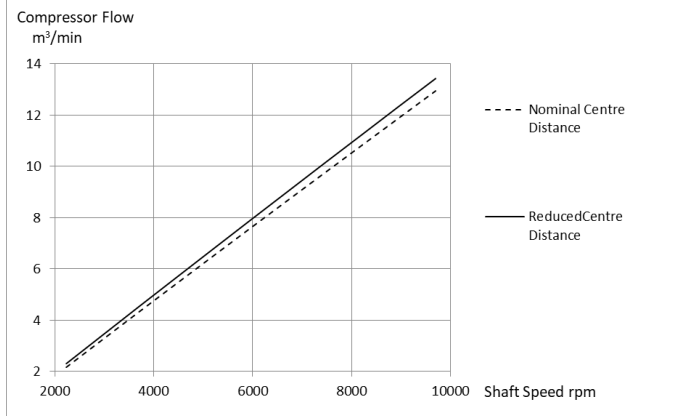


Figure 9. 4/5 rotors: flow as function of shaft speed

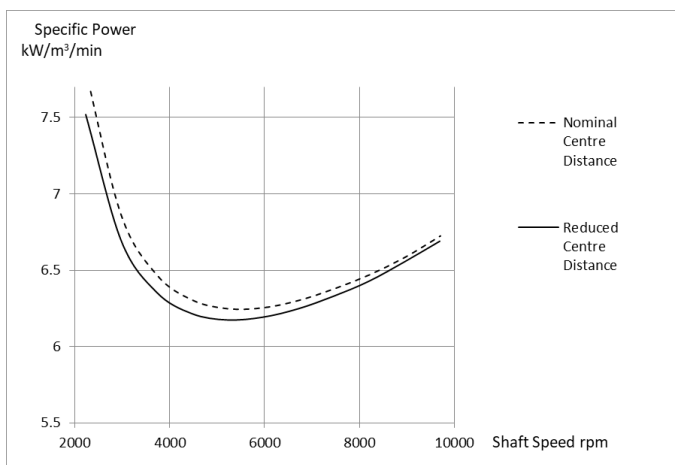


Figure 10. 4/5 rotors: specific power as function of shaft speed

4/6 Configuration

The 4/6 profiles are illustrated in Figure 11, featuring a nominal centre distance of 98 mm, a main rotor outer diameter of 128 mm, and a gate rotor outer diameter of 122 mm. The overall rotor dimension is 223 mm, with a main rotor inner diameter of 74 mm and a gate rotor inner diameter of 68 mm. The rotor depth measures 28 mm, with an L/D ratio of 1.55 and a displacement of 1.61 litres per revolution.

In contrast, with a reduced centre distance of 95 mm, the main rotor outer diameter remains at 128 mm, while the gate rotor outer diameter is 118 mm. The overall rotor dimension is 218 mm, with a main rotor inner diameter of 72 mm and a gate rotor inner diameter of 62 mm. The rotor depth is 27 mm, maintaining an L/D ratio of 1.55 and a displacement of 1.64 litres per revolution.

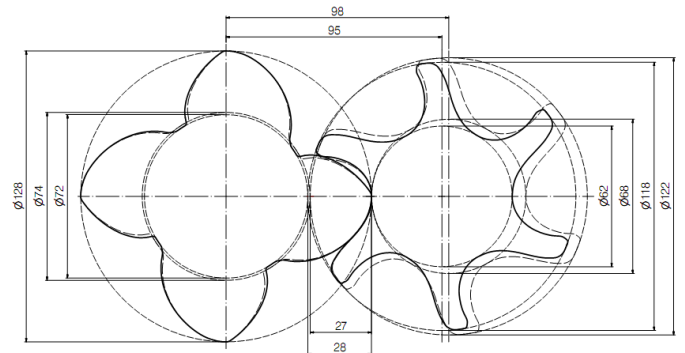


Figure 11. 4/6 rotors

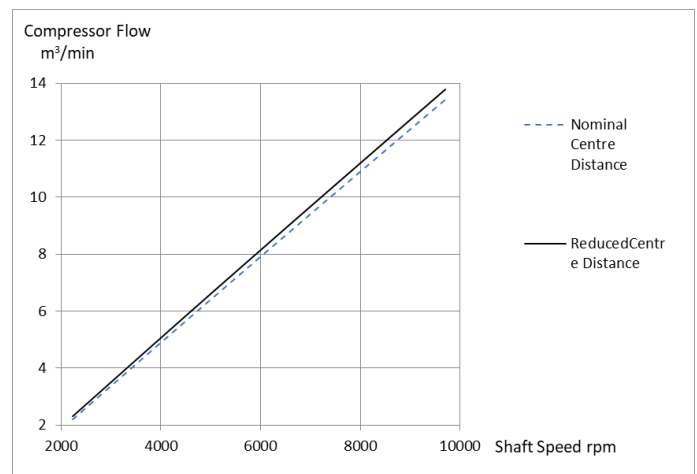


Figure 12. 4/6 rotors. Flow as function of shaft speed

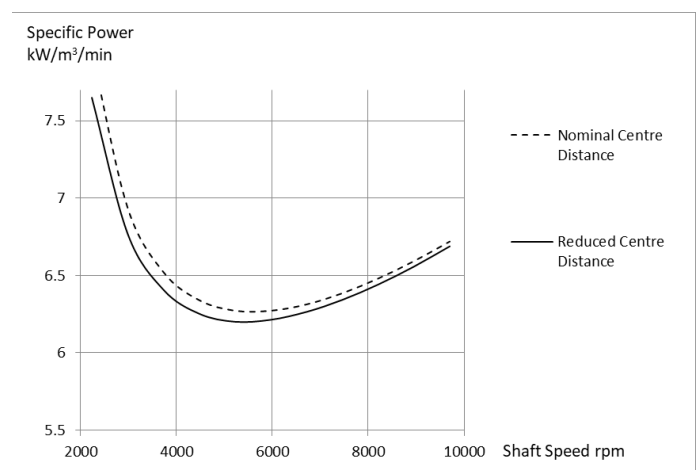


Figure 13. 4/6 rotors. Specific power as function of shaft speed

The performance of 4/6 rotors in an oil-flooded compressor operating within a pressure range of 1 to 8 bar is presented in Figures 12 and 13.

5/6 Configuration

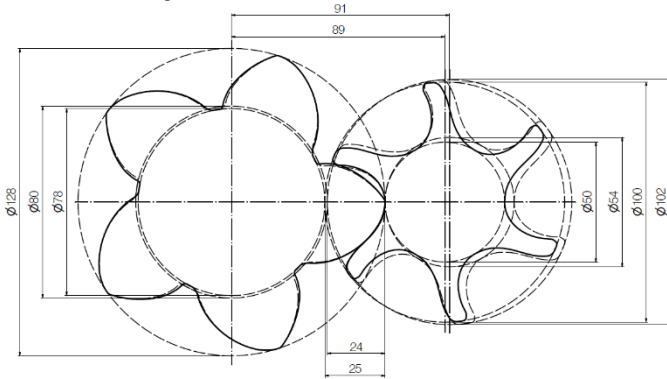


Figure 14. 5/6 rotors

The 5/6 profiles are illustrated in Figure 14, featuring a centre distance of 91 mm, a main rotor outer diameter of 128 mm, a gate rotor outer diameter of 102 mm, an overall rotor dimension of 206 mm, a main rotor inner diameter of 80 mm, a gate rotor inner diameter of 54 mm, a rotor depth of 22 mm, an L/D ratio of 1.55, and a displacement of 1.48 litres per revolution. With a reduced centre distance of 89 mm, the main rotor outer diameter remains at 128 mm, while the gate rotor outer diameter decreases to 100 mm, resulting in an overall rotor dimension of 203 mm. The main rotor inner diameter is 78 mm, the gate rotor inner diameter is 50 mm, the rotor depth is 25 mm, maintaining the L/D ratio of 1.55, and the displacement increases to 1.52 litres per revolution.

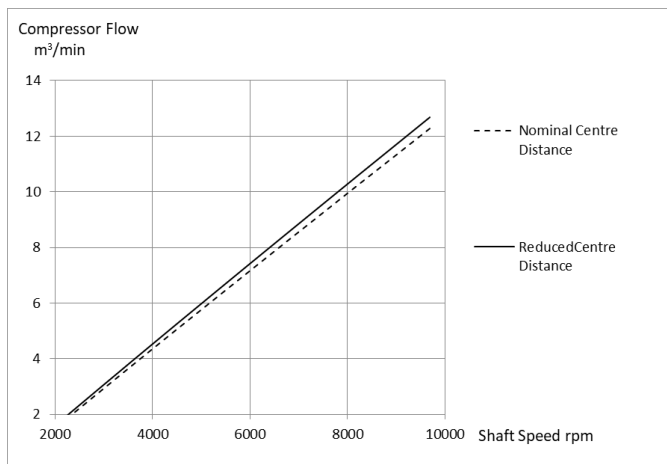


Figure 15. 5/6 Rotors: Flow as function of shaft speed

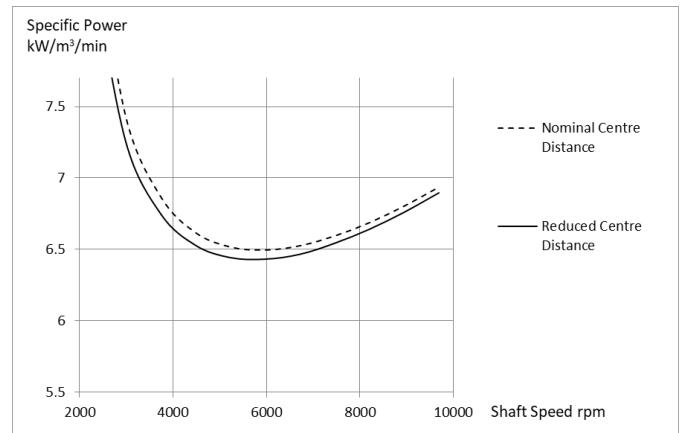


Figure 16. 5/6 Rotors: Specific power as function of shaft speed

Their performance in an oil-flooded compressor operating between 1 and 8 bar is presented in Figures 15 and 16.

It can be seen from the performance graphs that compressors with deeper rotor profiles consistently outperform those with shallower ones and lead to capacity gains of up to 2.3% for dry compressors and 4.5% for oil-flooded compressors. Also, specific power improvements of up to 0.7% for dry compressors and 2.5% for oil-flooded compressors were thereby obtained. Furthermore, it can be seen that compressors with fewer lobes are more efficient, with flow increases of 6.25% and specific power reduced by 5.5% for those with 4/5 rotors compared to those with a 5/6 configuration. Moreover, compressors with fewer lobes also have a smaller overall width and require fewer lobes to be manufactured. Thus, for instance, those with 4/5 rotors have only 9 lobes, while those with 5/6 rotors require 11.

4. CONCLUSION

A generic study on influence of the number of lobes and consequently rotor depth, presented for the first time in this paper, dully confirms that the flow and specific power of a screw compressor are reduced with the increase of number of their lobes and reduction of the lobe depth. It can be deduced that the increase of the rotor lobe depth will improve the compressor performance

This is then confirmed in a comparative analytical study to assess the impact of increasing rotor profile depth in screw compressors with configurations of 3/4, 3/5, 4/5, 4/6, and 5/6, which revealed that deeper profiles consistently yield greater flow rates, together with improved volumetric and adiabatic efficiencies than their shallower counterparts of the same size. In all instances, achieving greater profile depth for a given rotor outer diameter was obtained both by assuming smaller centre distances and a smaller number of lobes.

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