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Improving Screw Compressor Displacement and Efficiency by Increasing the Rotor Profile Depth

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1. Introduction

An investigation on the influence of rotor profile depth on screw compressor performance has been carried out for a range of main/gate rotor configurations, all with the same main rotor diameter. Both dry and oil flooded operation were considered.

Volumetric efficiency is the ratio between the real and the theoretical volumetric displacement. The theoretical displacement is the product of the rotor void cross sectional area and its length. Thus, assuming that the rotor shape does not change, it is proportional to the cube of the rotor length. However, the real displacement is the difference between the theoretical displacement and the internal displacement losses. These losses are caused by leakage, the pressure loss in admission and the temperature rise of the working fluid on admission to the working chamber. Of these effects, leakage comprises the major part and this is proportional to the leakage path cross section area and the square root of the pressure difference across it. The leakage area is the product of the sealing line length and the magnitude of its clearance, both of which may be regarded as proportional to the rotor length scale.

There are two limiting leakage rates. The first is that the flow loss is proportional to the rotor length squared, while the other is to assume the clearance size to be constant, in which case it, is proportional only to the rotor length.

Consequently, if the clearances are proportional to the rotor size, then the leakage losses are proportional to rotor size square. Hence the volumetric efficiency increases with size, as shown in Equation (1).

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

where Q is flow, Q_t is theoretical flow and L is the scale length.

However, if the clearances are not dependent of the rotor size, but constant for all sizes, the leakage losses are proportional to the size and volumetric efficiency increases with the square of the rotor length scale as shown in Equation (2).

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

The real volumetric efficiency will probably be somewhere between these two limiting cases.

There is no doubt that compressor adiabatic efficiency is heavily dependent upon its volumetric efficiency. By keeping all other parameters the same, it can be assumed that the adiabatic efficiency will be proportional to the volumetric efficiency.

In evaluating equations (1) and (2) it is necessary to determine which rotor dimension should be assigned as the value of the scale length L. If the main rotor outer diameter is fixed, then, as the number of lobes is increased, the inner, root rotor diameter is increased and the throughput will be reduced. In the limit, when the number of lobes tends to infinity, the throughput will tend to zero, while the sealing line will still have a finite value. It follows that for a given rotor size, reducing the number of lobes will result in the rotor profile depth being larger than in the case of a higher number of lobes. This means that the rotor profile depth, which is the difference between the rotor outer and inner radii, is a more reliable measure of the rotor scale length than the rotor outer diameter, or the rotor length, on which to base the effect of size on volumetric efficiency.

If the centre distance is reduced for rotors of a given main rotor outer diameter, the rotor profile depth will rise. It follows that rotor profile depth should be taken as the preferred criterion for profile design to generate rotors with large displacement and high efficiency, but it may not be the case when it is essential to maintain other design parameters, such as rotor rigidity to minimise deflection.

2. Presentation and Review of Analysed Profiles

In all cases, the rotors were generated and their performance calculated by use of the proprietary screw compressor design program suite as described in ref [1], while the compressor performance was calculated according to modelling principles given in ref [2].

For oil free operation 3/5 configurations was taken as the main and gate lobe numbers, while for oil-flooded operation, these were taken to be 4/5, 4/6 and 5/6. The inlet pressure and inlet temperature were 1 bar and 20°C in all cases, while the outlet pressures were taken as 3.5 bar and 8 bar respectively, for oil-free and oil-flooded compressors with air as the working fluid.

Each of the four rotor configurations assumed was designed with both nominal and deeper profiles. The deeper profiles were generated by reducing the centre distances so that all parameters were kept constant other than the main rotor profile addendum which was adjusted, in each case, to maintain the main rotor outer diameter at 128 mm as follows:

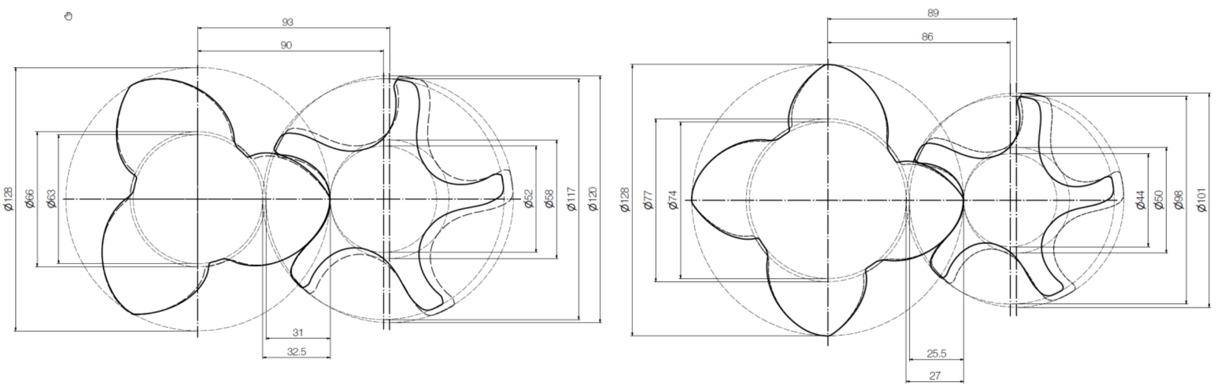

Figure 1: 3/5 and 4/5 rotors

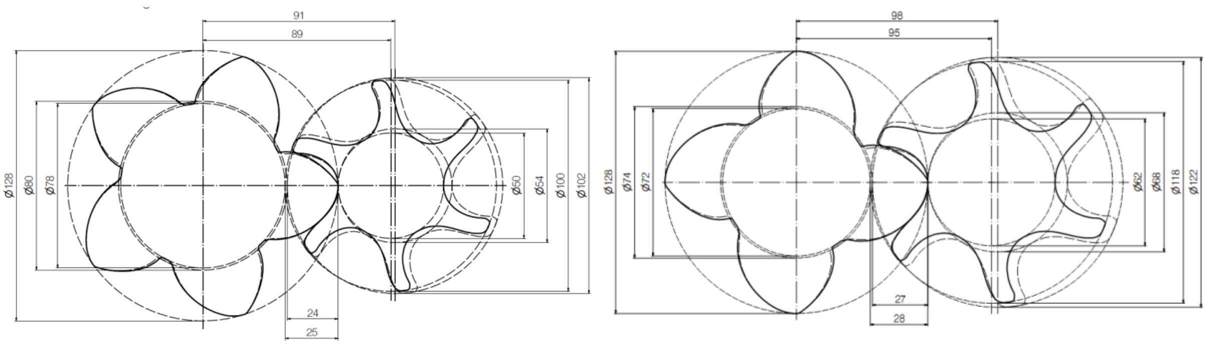

Figure 2: 5/6 and 4/6 rotors

The 3/5 profiles plot is presented in Figure 1. Its overall dimension is 217 mm and rotor depth 31 mm, L/D=1.7 and displacement 1.92 lit/rev for nominal centre distance of 93 mm, while for the reduced centre distance 90 mm, the rotor overall dimension is 212.5 mm, rotor depth 32.5 mm, L/D=1.7 and displacement 1.96 lit/rev. Average flow and specific power of oil free compressors with these rotors operating between 1 and 3.5 bar were improved 2.9% and 0.6% respectively.

The 4/5 profiles plot is also presented in Figure 1. The rotor overall dimension for nominal centre distance 89 mm, is 203.5 mm, rotor depth 24.5 mm, L/D=1.55, displacement 1.54 lit/rev, while the rotors with the reduced centre distance 86 mm have overall dimension 199 mm, rotor depth 27 mm, L/D=1.55, and displacement is 1.58 lit/rev. Average flow and specific power of oil flooded compressors with these rotors operating between 1 and 8 bar were improved 4.2% and 1.4% respectively.

The 4/6 profiles are presented in Figure 2. For the nominal centre distance 98 mm, the rotor overall dimension is 223 mm, rotor depth 28 mm, L/D=1.55 and displacement is 1.61 lit/rev. For the reduced centre distance 95 mm, the overall is 218 mm, rotor depth 27 mm, L/D=1.55 and displacement 1.64 lit/rev. Average flow and specific power of oil flooded compressors with these rotors operating between 1 and 8 bar were improved 3.1% and 1.4% respectively.

The 5/6 profiles are presented in Figure 2. For the nominal centre distance 91 mm, the rotor overall is 206 mm, rotor depth 22 mm, L/D=1.55 and displacement 1.48 lit/rev, while for the reduced centre distance of 89 mm, the rotor overall is 203 mm, rotor depth 25 mm L/D=1.55 and displacement 1.52 lit/rev. Average flow and specific power of oil flooded compressors with these rotors operating between 1 and 8 bar were improved 3.5% and 1.4% respectively.

3. Discussion of Results

It was shown, in all cases that the rotor inner diameter decreases and rotor depth increases with reduction of the centre distance. This increases the flow rates and the volumetric and adiabatic efficiencies. The deeper rotor profiles show an advantage in all cases, with improvements of capacity and specific power respectively up to 2.9% and 0.6% for dry compressors and up to 4.5% and 1.4% for oil flooded compressors.

It is also shown that rotors with fewer lobes are more efficient than the rotors with more lobes. Thus, the flow was 7.1% higher and specific power was 5.1% lower for the 4/5 than for the 5/6 rotors. Apart from their smaller overall dimension of the rotors with a lower number of lobes require less lobes to be produced, 9 for the 4/5 rotors and 11for the 5/6.

Conclusion

A comparative analytical study was carried out to determine the effect of increasing the rotor profile depth of compressors with 3/5, 4/5, 4/6 and 5/6 configuration, with nominal and increased rotor profile depth for each case. It was found that in all cases, profiles with a greater depth deliver a larger flow with higher volumetric and adiabatic efficiencies than their counterparts of the same size with a smaller profile depth. In all cases, the greater profile depth for the same rotor outer diameter was obtained both by assuming smaller centre distances and a smaller number of lobes.

References

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