Screw Rotors with Partial Length Contact.

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Abstract. Tribology theory suggests manufacturing screw rotors with partial helical contact along the longitudinal axis of the intermeshing rotors may significantly influence their lubrication characteristics. This paper considers the theoretical effects of using this geometry feature for multiphase flow machines. Modelling of machines that include partial contact along with other relevant aspects such as oil port location, has further been investigated using ANSYS CFX and some results of these simulations are presented and discussed.

Notation.

\[ h_{\text{min}} = \text{Minimum elastohydrodynamic film thickness [in]} \]
\[ \alpha = \text{Pressure Viscosity coefficient [in}^2/\text{lbf]} \]
\[ \mu_0 = \text{Dynamic Viscosity at operational temperature [lbf s/in}^2] \]
\[ V_e = \text{Entraining velocity [in/s]} \]
\[ \rho_n = \text{Normal relative radius of curvature [in]} \]
\[ X_r = \text{Load sharing factor, no units} \]
\[ w_{N_r} = \text{Normal unit load [lbf/in]} \]
\[ E_r = \text{Reduced modulus of elasticity [lbf/in}^2] \]
\[ k = \text{Experimentally defined constant, no units} \]
\[ S = \text{Experimentally defined constant, no units} \]
\[ W_{N_r} = \text{Normal operating load [lbf]} \]
\[ L_{\text{min}} = \text{Minimum length of contact [in]} \]

1 Introduction.

Fluid is injected into screw compressors to serve three functions; provide a separating film between rolling and sliding contacting surfaces, act as a cooling medium for the compressed fluid, and to seal the gaps between the rotors to reduce leakage.

The sealing and cooling brought about by fluid injection is extremely advantageous, but the lubricating film is essential to allow reliable operation as it removes friction and wear between the rotor surfaces.

There are three regimes in standard lubrication theory. Hydrodynamic or full film lubrication is the condition when the load carrying surfaces are separated by a relatively thick film of lubricant.
Elastohydrodynamic lubrication occurs when a lubricant is introduced between surfaces that are in rolling contact. In this lubrication regime, the load causes the surfaces to elastically deform during the hydrodynamic action by magnitudes in the order of the actual film thickness. Boundary lubrication is the condition when the fluid films are negligible and there is considerable asperity contact. Viscosity is probably the single most important physical characteristic in describing a lubricant and often defines which of these lubrication regimes occurs.

For two phase screw compressors, the lubrication is elastohydrodynamic in nature, Zaytsev\cite{1}. This is the best operational regime as a high viscosity fluid required for full hydrodynamic lubrication would induce high frictional forces between the rotor tip and the housing. This would require extra power input to overcome these losses. A low viscosity fluid would cause boundary lubrication where there is metal to metal contact which leads to coefficients of friction approximately 100 times more than those experienced in elastohydrodynamic lubrication, Hamrock\cite{2}. In addition to frictional losses; wear and the significant risk of contact damage such as scuffing, makes screw compressor operation in the boundary lubrication regime undesirable.

Elastohydrodynamic lubrication is an established branch of tribology with proven formulas that explain the behaviour of materials operating under this regime. Jacobson\cite{3} described a method for calculation of the minimum elastohydrodynamic film thickness based on a linear contact pattern. This initial theory has been further developed and applied to helical gear contact by Davis\cite{4}. Screw rotors are a specialised gear form and as involute geometry is normally employed in rotor profile design at the contact band it is felt that the equations presented by Davis are the most relevant for screw compressor lubrication theory. The Davis formula is presented in equation 1 below with the constituent parameters defined in the notation section.

\[
h_{\text{min}} = \frac{1.63 \alpha^{0.54} (\mu_0 V_r)^{0.7} \rho_n^{0.43}}{(X, W_N)^{0.13} E_r^{0.03}}
\]  

This report initially assesses this formula for the specific case of rotors with partial helical contact and demonstrates that rotors incorporating this geometrical feature theoretically offer a wider range of options in terms of the fluid used to lubricate the rotors whilst maintaining the desired regime of lubrication. ANSYS simulations are then presented to demonstrate the temperature distribution of the lubricant within a screw compressor. Finally, the partial contact feature is introduced into the models to gain an overview of the wider performance issues that partial helical contact may induce.

2 Partial contact.

Now the importance of viscosity in generalised lubrication has been established, let us consider this parameter in more detail with reference to equation 1 and screw rotor contact. In the formula, viscosity is the most significant term, with an exponent of 0.7.

The pressure viscosity coefficient \( \alpha \) (alpha) defines the “thickening” that occurs under hydrodynamic loads and is also predominantly reliant on the dynamic viscosity of the fluid. AGMA 925\cite{5} defines the coefficient as:

\[
\alpha = k \mu_0^s
\]  

For Polyalkylene Glycol (PAG), a synthetic oil predominantly used in compressors, typical values for the constant \( k \) is 0.0047 and the exponent \( s \) is 0.1572.

Screw compressors generally use a lubricant with an ISO viscosity grade of between 32 and 68, with ISO VG 46 being the most common. This grade has a kinematic viscosity of 46 Centistokes at 40 degrees Celsius. For all film thickness calculations, the dynamic viscosity is calculated by multiplying the kinematic viscosity by the fluid density. Viscosity follows an exponential relationship with temperature as defined by Reynolds\cite{6}. A graph is presented in figure 1 to demonstrate just how sensitive the VG 46 oil viscosity is to temperature variation.
Consider the oil lubrication of a screw air compressor. The oil is injected at approximately 20 degrees Celsius. The compression process raises the temperature of the oil/air mixture which is typically discharged in the range of 40-60 °C higher than suction. Referring to figure 1 this order of temperature variation creates a large range of potential viscosities within the screw compressor.

A modified set of screw rotors are shown in figure 2 with partial contact along the longitudinal axis of the intermeshing rotors. The rotor set is manufactured to ensure there are defined areas of contact and non-contact for the threaded section of the rotors.

For a simplified example, partial contact is applied to an oil lubricated air compressor with the first half of the rotor body length in contact and the second half with a small separation gap. It is presumed the contact is restricted to areas where the lubricant temperature remains at, or below, 30 degrees Celsius. We will compare this directly to a situation for a rotor with the same operational parameters minus the partial contact. For this case there is full contact all the way to the high temperature outlet end where a lubricant temperature of 60 degrees is assumed.

With reference to formula 1, factors that include speed, profile geometry, material and loading will be constant for both situations. The normal unit load $W_{Nr}$, which is shown in equation (3) is defined as the normal operating load, $W_N$, divided by the minimum contact length $L_{min}$.

$$W_{Nr} = \frac{W_N}{L_{min}} \quad (3)$$
The pressure viscosity coefficient in equation (1) is replaced with the AGMA term described in equation (2). This leaves a simple relationship that describes the minimum film thickness with the contact length and the viscosity as the only variables. All other terms remain consistent.

For the conditions we have assumed, the rotors with partial helical contact will have a minimum film thickness 3.6 times bigger than that for the same set of rotors with full length contact. This is due to contrasting significance within the theory of the terms for the contact length and the viscosity. The contact length exponent is only 0.13 in direct comparison to the combined viscosity term (from equations 1 and 2) of 0.8572. At the cold end, shown in figure 2(a) the rotors are designed to contact as normal. Low temperatures facilitate stable film formation and adequate lubrication. At the hot end (figure 2(b)) the rotors are kept separate. Contact is avoided where the lubricating film is at its thinnest and film breakdown is most likely.

This method provides the option of using lubricants with lower viscosity whilst still maintaining a thick enough film for elastohydrodynamic lubrication. This perhaps offers the opportunity to lubricate with fluids that reduce the viscous drag forces within the machine or have improved cooling capabilities for more economical operation. It may also, in some cases, allow process fluid lubrication where the working fluid is in a liquid phase for all, or part of the compression/expansion process.

2.1 Manufacture of rotors.

It is important to consider the practicality of introducing partial contact into the manufacturing process of screw rotors.

The most economical way to produce high volumes of rotors is to cut from a solid metal bar using a specialised formed tool. A diagram of the tool cutting a rotor is shown in Figure 3.

The tool makes one or more passes along the longitudinal axis of the bar while the tool spins about its centre. At the same time the metal bar rotates about its longitudinal axis. The profile on the tool cuts the required rotor profile shape while the relative rotation of the bar and the movement of the tool creates the helix for the rotor thread.

Consider a rotor that has been manufactured in this way without any relief applied. To create the extra gap as shown in Figure 2(b), it is now a simple case of making a small extra rotational movement of the rotor whilst taking a final cut. This rotation pushes the drive side of the rotor towards the cutting tool and the non-drive away from the tool, creating a cut that removes material only from the drive side of the rotor. Rather than the tool progressing the full distance along the rotor, the tool is stopped at the required distance into the rotor and is then withdrawn. This results in the addition of the required relief in one single extra cut per flute.

Figure 3. Technique for rotor manufacture.
A small amount of material is removed from the drive side of the rotor while the inner diameter and non-drive side of the profile is unaffected. Davey[7] calculated a formula to remove precise amounts of material using this method and demonstrated that it was possible to accurately remove material in the order of 5 to 20 microns range from the drive flank. This relatively small profile change leads to minimal change in the size of the leakage gap between the rotors. This small change should therefore be the target adjustment that is applied. Importantly, as the cutting technique to create partial helical contact is so simple, the cycle time for rotor manufacture will also not be increased by more than a few minutes, so the added cost to manufacture is small.

3 Simulations.

Section 2 has described an idealised example. An understanding of the true oil temperature and distribution within screw machines is required to assess how realistic these assumptions are.

Screw rotors are housed in a solid casing, a closed system where non-invasive measurement techniques are impossible. The rotation speed and small clearance gaps between the rotors make it impractical to include any form of thermal sensor on the rotor surface. Essentially, the only practical method to calculate the oil temperature in operation, and therefore assess the viability of partial contact components is through computer simulation.

Computational Fluid Dynamics (CFD) is an established method for understanding the behaviour of positive displacement screw machines. The basic theory behind simulation of multiphase flows has been previously reported by Papes[8] and Rane[9].

A CFD Eulerian multiphase model has been set up and run in ANSYS CFX solver. The rotor geometry is shown in figure 4 looking down from above the discharge (view B). As viewed in the diagram the female is rotating in a clockwise direction and the male in an anti-clockwise direction.

The rotors have an ‘N’ type generated profile (Stosic[10]) with 3 male lobes and 5 female lobes. The centre distance is set at 93 mm, with a male outer diameter of 127.323 mm and a female diameter of 120.262 mm. The male rotor has a length to diameter ratio of 1.6 and a wrap angle of 285°. The rotors have a uniform flank to flank interlobe clearance gap of 80 microns and a 100 micron gap between the profile tip/root diameters. The clearance gap to the housing is also set at 100 microns. There is a single oil port of 5 mm diameter located 32 mm axially from the discharge end of the rotors and offset 47 mm from the female centre line. Oil is delivered directly onto the female rotor at 19.85° Celsius at a volume fraction of 10%. The rotors are run at 8000 RPM, have an inbuilt volume ratio of 1.8, with suction pressure and temperature of 19.85° Celsius and 1.0 Bar respectively. The discharge pressure is
set at 3.0 Bar. The rotors are meshed using SCORG, which is a structured mesh generation and CFD analysis setup tool specifically for rotary positive displacement machines. The basic grid generated is shown below in figure 5 and uses a casing to rotor conformal pattern which has been shown to offer the most accurate output for multiphase simulations, Rane\cite{9}.

Figure 5. Rotor mesh in SCORG.

There are 8 radial divisions for both the male and female rotor with 60 circumferential divisions per female lobe. This pattern is generated for the full length of both rotors creating 71400 discrete quad elements.

4 Simulation Analysis.

Results of the simulation are presented for the surface oil temperature as the machine is rotated through a single flute/lobe combination. Figure 6(b) shows a plot of the oil temperature values for a horizontal line on the surface of the female rotor labelled AA in 6(a). They are ordered, starting with the highest recorded temperature time step as shown in figure 6(a). Subsequent time steps are plotted for four further instances, each representing a female rotation of 14.4\(^\circ\). This gives a full rotation through a single female flute.

Figure 6(a) is presented from the side-on viewpoint looking through an invisible male rotor (View A in figure 4) to show the oil temperature between the rotors in the areas in which contact between the male and female would occur. The oil temperature is also presented from above the discharge port for the matching rotational position (view B in figure 4). Analysis of figure 6(b) shows the range of temperatures where contact occurs is in the order of 40 to 140\(^\circ\) Celsius. As the cycle progresses, remnants of the high temperature oil mainly stay within the same flute after discharge has occurred and are forced backwards along that flute. The pattern of high temperature oil appears to follow the interlobe seal line between the rotors, suggesting the oil travels towards the suction end and is subsequently partially blocked by the next advancing male lobe. The leakage into the low pressure flute directly beneath is added to residuals of the high temperature oil from the previous discharge cycle. The oil in the adjacent flute preceding the discharge flute therefore remains at a relatively low temperature in comparison to the two flutes thereafter. Effectively, the flute contains and then guides the hotter oil around the helix of the rotor. Consider this effect for the simulated rotors and compare directly to a rotor set with a male wrap angle of 300\(^\circ\) and 5 lobes meshed with 6 female flutes as shown in figure 7. For the high wrap/flute combination, the hot oil would naturally flow away from the line of contact between the rotors in preceding flutes leaving a longer ‘cold’ contact region.

A further simulation has been completed with the oil injection port moved axially to the centre of the rotor body length. Results are presented in figure 8 for the same rotation position as that shown in figure 6a.
Figure 6(a). Hottest surface oil temperatures. Reference zero position.

Figure 6(b). Surface temperatures along line AA.

Figure 7. Geometry comparison 3/5, 285° wrap rotor set to 5/6, 300° wrap rotor set.
Figure 8. Oil temperature with central oil port at reference zero position.

In this instance, there is a reduction in the spread of the hottest oil. This shows the porting location is an important variable when considering partial contact design.

Positive aspects of using the partial contact feature have been studied and highlighted. By introducing additional clearance, there will also be a negative impact on performance due to a bigger interlobe leakage path. This also requires further investigation to provide a balanced overview. Rane\cite{11} has researched the concept of modelling rotors with variable profiles and a feature for meshing such geometries is available in the pre-modelling software package SCORG. Compressor models evolved from the original representation, but with partial contact have subsequently been created. In each case the female rotor has been adjusted by rotating the drive side of the profile to create a gap at the pitch line as shown in figure 9.

Figure 9. Method of applying clearance.

This modification is consistent with the method of manufacture described in section 2.1. A number of female profiles with extra clearance have been generated in SCORG (figure 10a) and imported into ANSYS CFX (figure 10b) for simulation.

The female rotors have the original profile shape from the suction end to the axial mid-section. At this location there is a step change created by the profile rotation. The rotor continues from the mid-section to the discharge end with uniform rotor profile clearance created by the applied adjustment. The male rotor geometry stays unchanged. The mesh density also remains unaltered. Simulations have been carried out for female rotors with 100, 300 and 500 microns additional clearance at the pitch line on the drive side and the results are presented in table 1.

The clearance increase in the simulations is exaggerated in reference to the operational target change described in section 2.1. Within simulations, the results are extremely reliant on the mesh density. For extremely small changes in the order of 10 microns as desired would require an exceedingly fine mesh that would make the run time for a multiphase simulation prohibitive. There is no guarantee that even with the refined mesh the results would be distinguishable from each other at
this infinitesimal level of change. Large changes have therefore been instigated to achieve a
generalised overview of the consequences of partial helical contact.

![Figure 10(a). Partial contact geometry SCORG.](image)

![Figure 10(b). Partial contact geometry ANSYS.](image)

Adding the extra clearance creates extra leakage and reduced performance; these effects are summarised in table 1. An increase in clearance of 100 microns leads to an increase in seal line area of over 25%. For the case when the clearance is limited to the target 10 microns the seal line area increase is lowered to 3%, but is still significant. A more complicated cutting solution may be required that does not increase the clearance gap for the full rotor flank and limits the seal line area increase to less than 1%. It is feasible to achieve this by withdrawing the cutting tool slightly from the workpiece, applying a rotation and using the tip section of the tool to make a cut which removes material only from the contact band on the rotor. This cutting technique is more complicated than that proposed by Davey [7] and would require development, but would allow the application of the partial contact geometry without impacting to a significant degree on the leakage gap.

|                      | Standard | 0.1mm extra clearance | 0.3mm extra clearance | 0.5mm extra clearance |
|----------------------|----------|-----------------------|-----------------------|-----------------------|
| Interlobe seal line area suction (mm²) | 18.9     | 18.9                  | 18.9                  | 18.9                  |
| Interlobe seal line area discharge (mm²) | 18.9     | 25.8                  | 38.2                  | 50.8                  |
| Air mass displacement decrease % | 0        | 1.9                   | 4.2                   | 7.1                   |
| Power consumption increase % | 0        | 0.22                  | 0.51                  | 0.61                  |

The cold contact as shown in Figure 7 is limited to the extremity of the suction end. At this location there is intermittent rather than constant contact as the rotors come in and out of mesh. This further restricts the practical length of contact between the rotors when a partial contact geometry is applied. It is therefore necessary to ensure the contact between the rotors extends for a reasonable section independent of their rotational position so the possibility of localised point contact is eliminated.

Importantly for all simulations, the pattern of hot oil flow remained relatively unaffected for varying clearance gaps. Due to mesh size limitations, to gain a proper understanding of the negative aspects of partial contact will require experimental testing of prototype rotors including this geometry.
5 Conclusions.

Tribology theory for elastohydrodynamic lubrication suggests a partial helical contact pattern, where contact is allowed in regions with lower temperature lubricant and avoided in regions of relatively high temperature lubricant, may be beneficial. Applying basic assumptions for the contact geometry and the oil temperature suggest a threefold increase in the film thickness is attainable.

To assess this assumption, ANSYS CFX simulations have been carried out to monitor the temperature pattern of the oil between contacting rotors. The results show an oil temperature difference in the order of 40 – 140° Celsius may be realised along the contacting length of the rotors. After discharge occurs, the residual hot oil stays mainly within the same flute. The rotor helix then naturally guides the hotter oil around the rotor and away from the line of contact between the rotors in preceding flutes. This behaviour improves the potential of using partial contact in rotor sets with more flute/lobe combinations and higher wrap angles. Partial helical contact has been added to the simulations and as expected, there is a decrease in performance due to increased leakage. Modern manufacturing techniques are able to produce rotors with precise profile modifications of magnitudes as low as 10 microns. Although useful in terms of oil temperature distribution CFD is unable to completely describe the desired situation with adjustments of this magnitude. Practical experimentation is required to fully validate the predicted improvements.

Partial helical contact offers a promising method to increase the thickness and stability of lubricating films in screw compressors and expanders. This thickening effect may allow lubrication with alternative fluids not normally conducive to avoiding boundary lubrication, wear and machine failure. In the best eventuality, this geometry adaptation may allow working process fluid lubrication within screw machines.

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