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Rotor clearance design and evaluation for an oil injected twin screw compressor

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Abstract. Designing twin screw compressors to safely operate at higher than normal temperatures poses a challenge as the compressor must accommodate larger peak thermal distortions while ideally maintaining efficiency at nominal operating conditions. This paper will present a case study of an oil injected compressor tested at elevated discharge temperatures with original and revised clearances. The local thermal distortions occurring within the compressor during operation were estimated using a procedure developed by the authors - thermodynamic results from a chamber model were used to approximate component temperature distributions that are then used to predict possible thermal distortions and the resulting affect on clearance gaps. The original and revised clearance designs are evaluated and performance penalties incurred due to the modifications are discussed.

1. Introduction

End users of twin screw compressors occasionally demand reliable compressor operation over a greater operating range than the original design envelope. If for example the allowable cooling oil temperature can be increased, this can reduce the cooling requirements and the overall plant cost. This poses a challenge as the compressor design needs to accommodate higher peak operating temperatures while ideally maintaining efficiency at nominal operating conditions.

In one sense the clearances in oil injected machines are easier to manage than in oil free machines as temperatures are usually kept within much lower limits however the lack of timing gears and the direct rotor to rotor drive does add more complexity to clearance analysis. Rotor contact is of course necessary for one rotor to drive the other; the objective is to control precisely where it is possible for contact to occur. Best practice for direct drive clearance design ensures only rolling contact at the pitch radius of each rotor as presented by Stosic et al [1].

Deviations from nominal design clearances, whether due to manufacturing and assembly variations or due to operational distortions can result in a shift in the relative rotation between the main and gate* rotors [3]: this further distorts the interlobe clearance distribution and must be considered in clearance evaluation for direct drive, oil injected compressors.

* The adopted rotor terminology is consistent with [1]. The main and gate rotors can also be referred to as the male and female rotors respectively, which is often the case.

This paper will present a case study of an oil injected compressor that was tested at elevated discharge temperatures. The focus of this work is on potential thermal distortion and other factors affecting clearance distributions have been omitted from the current analysis. The local thermal distortions occurring within the compressor during operation will be estimated using a procedure developed by the authors [2] whereby thermodynamic results from a chamber model are used to approximate component temperature distributions based on available fluid boundary temperatures [4]. This is then used to predict possible thermal distortions and the resulting affect on clearance gaps.

2. Case study

The case study presented in this paper is based on historical data available in Howden Compressors Ltd. The compressor is a Howden WCVTA510/193/26 compressor, this is currently the largest Howden compressor with rotors just over 510mm in diameter. In this case the compressor has a built in volume ratio of 2.6.

In addition to air testing at contract pressure ratio and temperature the compressor was tested at an elevated discharge temperature by increasing the pressure ratio to 11 and then limiting injection oil cooling by increasing oil temperature and restricting flow. Due to power restrictions on the testing of this large compressor the speed was reduced from the contract speed of 1400rpm to 750rpm during this overload testing. After testing at up to a discharge temperature of 120°C (from suction at 20°C), tear down inspection revealed that rotor to rotor contact had occurred at the root of the main rotor as highlighted in Figure 1.

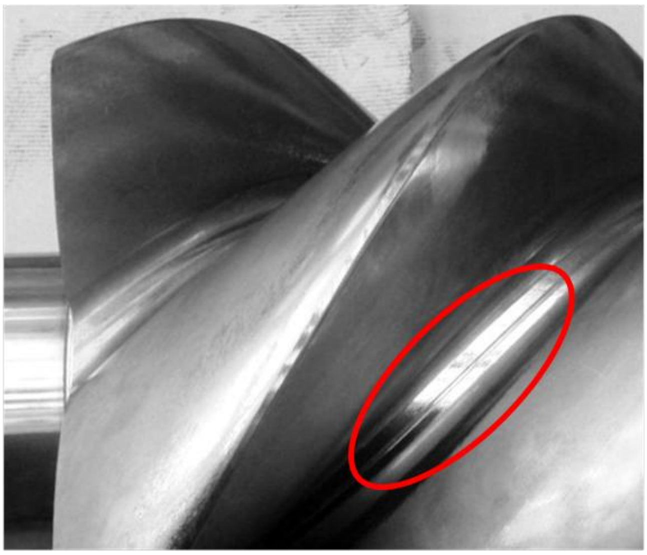

Figure 1. Main rotor with original interlobe clearance distribution showing rooting after testing at 120 degrees C.

The type of contact observed and the fact that this standard compressor had not presented any problems while operating within normal temperature limits points to thermal distortion of the rotors being the most likely cause of this rotor contact. Based on these test findings a revised interlobe clearance design was proposed. The revised clearance design was implemented on a different compressor at a later date and though teardown inspection revealed some very localised rotor contact, related to the tip sealing strip, there was no contact over the main part of the root and the clearance modification was approved for further use. The analysis performed in the remainder of this paper has been performed retrospectively in order to evaluate the original and revised clearance designs in more detail.

3. Clearance analysis procedure

3.1. Thermal analysis

The fluid temperature inside the compressor, assumed to be homogeneous throughout any given compression chamber, can be calculated with the use of well establish chamber models developed for twin screw compressors [1]. The overload test condition, with a discharge temperature of 120° C was modelled to determine how the temperature varied throughout the compression cycle for that particular duty. All modelling was done to replicate the test conditions where the overload discharge temperature was achieved by compressing air. It is worth noting that the intended gas for this compressor is actually a hydrocarbon mixture which has a lower isentropic index than air. The same discharge temperature is targeted regardless of the gas being used and for the time being it is assumed that the difference in temperature distributions on the rotors are not significantly different between these operating conditions.

Using a procedure developed by the authors [2], the fluid temperature from the compression cycle was mapped onto the surfaces of the rotors and the casing bores. These fluid boundary conditions at the surface of the compressor components can be used for further thermal analysis to determine the temperature distributions within the metal components of the compressor. Figure 2 shows the temperature of the fluid boundary on the main and gate rotor surfaces. The left image shows the instantaneous fluid temperature for a particular rotor position – this highlights how different compression chambers are exposed to different points in the compression cycle which are at different temperatures. In the right hand image these instantaneous temperatures has been averaged over one full rotation for each respective rotor in order to show the local average boundary temperature. In addition the temperature has also been averaged over each transverse plane of the rotors as this allows potential thermal distortions to be approximated analytically.

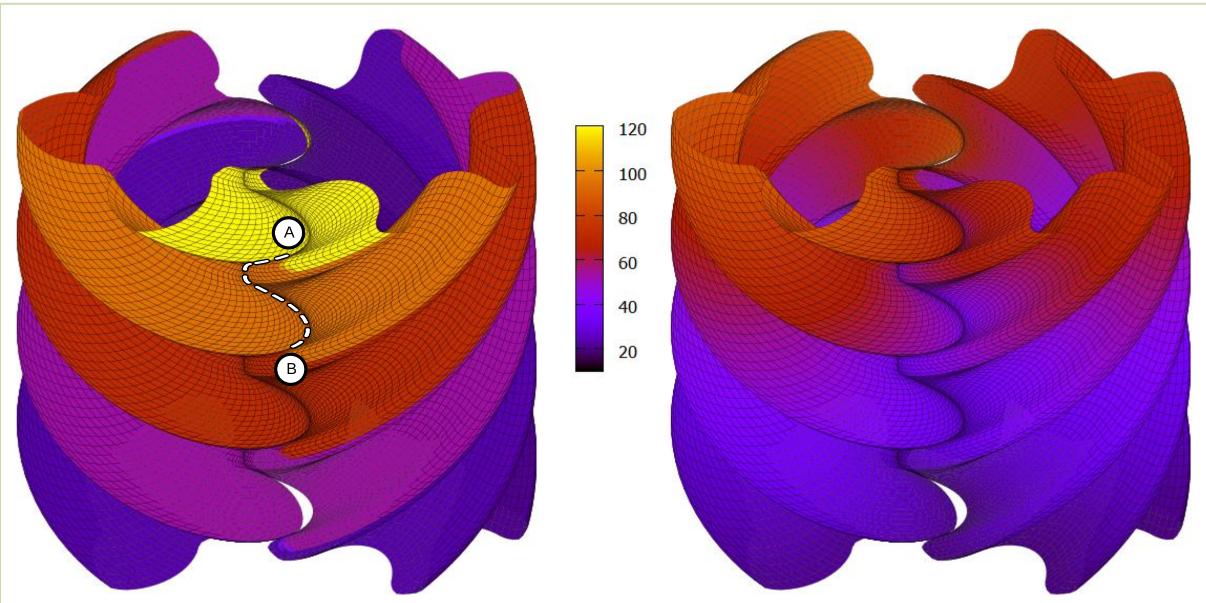

Figure 2. Instantaneous (left hand side) and averaged (right hand side) fluid boundary temperature mapped onto the surfaces of the main and gate rotors. Line AB highlights the interlobe sealing line for a single compression chamber.

Table 1. Average fluid boundary temperature across outlet plane.

Component Description	**Temperature (°C)**
Main rotor	86.8
Gate rotor	73.2
Casing	69.4

The peak temperatures occur at the outlet plane of the rotors therefore the analysis presented in this work will be based on the average fluid boundary temperatures presented in Table 1. It is clear that the averaged fluid boundary temperatures at critical locations on the compressor are actually much lower than the peak fluid discharge temperature of 120°C.

It is interesting to note that the average boundary temperature is higher on the main rotor than on the gate rotor, this is related to the number of lobes on each rotor. Consider the 4 compression chambers at the outlet end of the main rotor shown in the left hand figure. The gate rotor is also exposed to these 4 chambers but only over an angle of 240 degrees. For the other 120 degrees the gate rotor is exposed to 2 additional compression chambers at a lower temperature.

It is not possible to calculate the actual local temperature on any given component within the scope of this analysis however the average fluid boundary temperature gives a very good indication of the maximum temperature that the compressor components can reach. This results in a conservative approach that most likely overestimates actual thermal distortions due to heating by the compression fluid alone. Knowing the temperature of the rotors and casing, local clearances on a transverse section of the compressor can be evaluated. To generalise: rotor thermal distortions act to reduce the interlobe clearance while casing thermal distortions act to increase the interlobe clearance; if both rotors and casing were at the same temperature the net effect would be no change to the interlobe clearances.

3.2. Clearance presentation
Due to the 3D nature of the interlobe sealing line it can be difficult to present clearance information clearly so Figure 3 has been included in an attempt to make the clearances a bit more intuitive. This presents the same clearances mapped onto each of the rotors as vectors normal to the transverse rotor curve (actually showing the magnitude of clearances normal to the 3D rotor surface). This approach is very intuitive and clearly shows how clearances relate to difference areas on the rotor profiles. The difficulty with this clearance presentation is comparing small differences in similar clearance distributions. Magnifying the vectors quickly distorts the apparent clearance gap due to the curvature of the profiles. On the other hand, plotting local clearances on a linear axis such as ‘relative position on sealing line’ makes it difficult to determine where the clearances actually occur on the rotors.

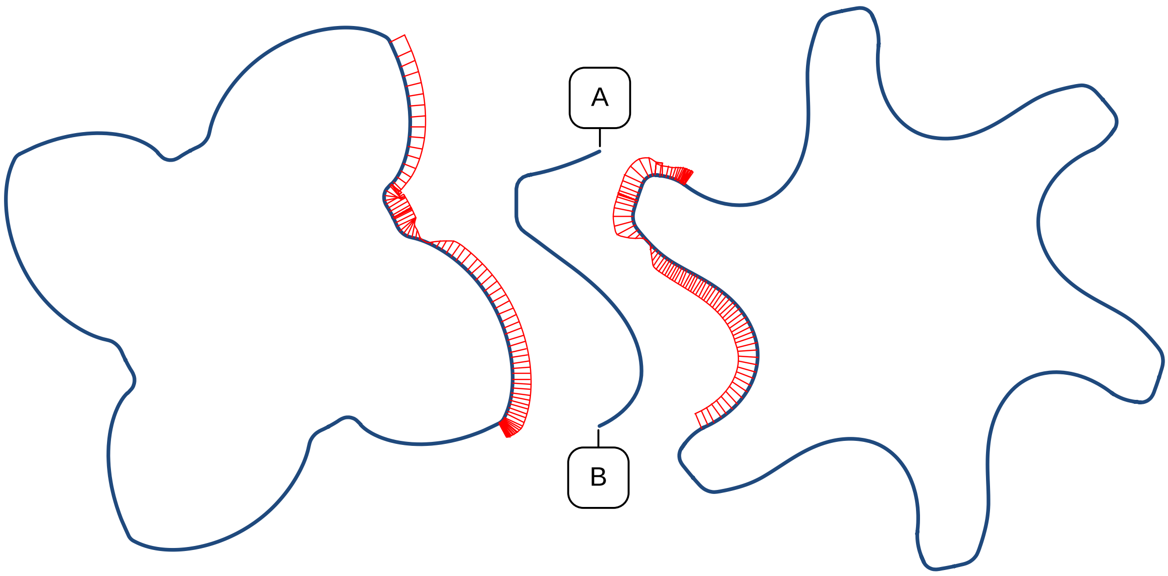

Figure 3. Transverse cross-section of main and gate rotors showing local interlobe clearances. The magnitude of the vectors represents the clearance *normal* to the 3D rotor surface. Line AB represents the conjugate rack, common to both rotors. The limits of this rack segment are the beginning and end of the sealing line for a single compression chamber, as previously highlighted in Figure 2.

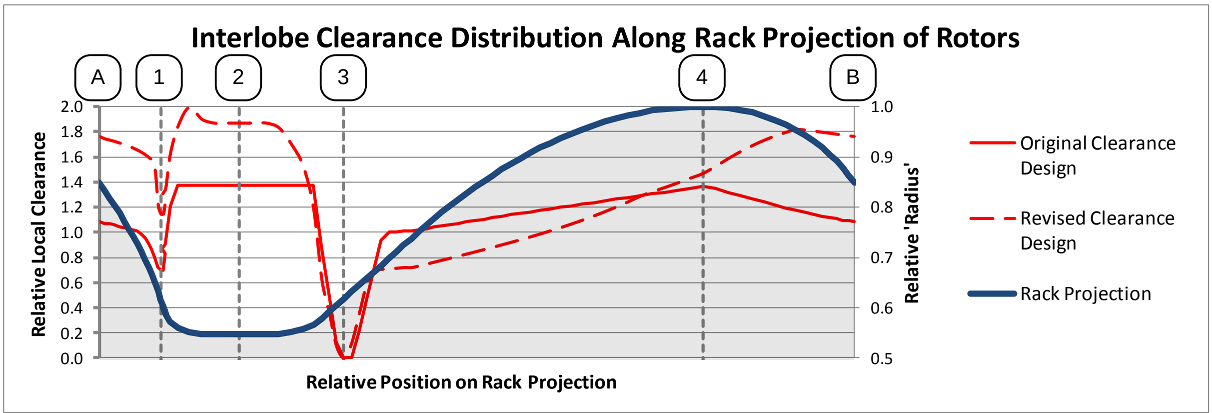

Figure 4. Interlobe clearance distribution along rack projection of rotors comparing original and revised clearance designs. Key clearance locations are highlighted along the common horizontal axis. The secondary axis shows the radius for the corresponding main rotor to give a better indication of where local clearances are located.

Table 2. Key clearance locations.

Location on rack	Description
A	Limit of sealing line for single chamber
1	Pitch radius on straight (undercutting) side of rotors
2	Root of main rotor / tip of gate rotor
3	Pitch radius on round (non-undercutting) side of rotors
4	Tip of main rotor / root of gate rotor
B	Limit of sealing line for single chamber

The compromise that has been taken is to plot the local clearances on the axis of a rack projection along which the rack would translate. Thus, the magnitude of the local clearances can be simply graphed and superimposed against the somewhat familiar shape of the rack as presented in Figure 4.

Rather than put a numerical value on the horizontal axis vertical lines have been added to identify key points along the length of this sealing line projection, details of which are provided in Table 2.

The original and revised clearance distributions are both presented in Figure 4. In all cases the relative rotation between the rotors is adjusted in order to provide zero clearance at location 3: the pitch radius on the round side of the rotors. This is where the driving force is transferred from the main to the gate rotor. In the revised clearance design the main difference is that the clearance in the main rotor root has been increased and to a lesser extent also at the main rotor tip. Notice that the clearance on the round flank of the rotors (between locations 3 and 4) has actually been decreased on the revised clearance design – this clearance will not significantly reduce during operation because the gap is maintained due to the nearby contact at the pitch point (location 3), therefore this clearance was deemed to be unnecessarily large. Conversely, on the straight flank (from 4 to B, and from A to 1) the gap is adversely affected by thermal distortions at location 3 which are transferred over to the straight flank. Therefore it was necessary to increase the clearances on the straight flank.

To quantify how these clearance changes impact the leakage area through the interlobe gap the local clearances where integrated over the length of the 3D sealing line. The total leakage area is 108.5mm^2 for the original design is and 142.7mm^2 for the revised design, an increase of 31.5%.

4. Results

4.1. Analysis of clearance distortions

The following results show the predicted operational clearance distributions for the original and revised designs respectively. As previously discussed, a similar temperature increase in both the rotors and casing results in little net change in the operational clearances, this case has been presented with the longer dashed lines on each figure, the trend is not far off the original clearance.

A second case has also been presented where the rotors do not move apart due to casing thermal expansion. This case is quite possible in the event that sudden temperature changes cause the rotors to heat up faster than the casing or alternatively the outlet end bearings might be situated remotely from the rotor outlet plane in a location where the casing temperature is much lower than estimated in Table 1. Unfortunately the distance between the two dashed curves for the different operational scenarios results in a large band of uncertainty but it does serve to highlight 1) where rotor contact is most likely to occur for a given clearance design, and 2) where clearances can be further reduced without compromising reliability.

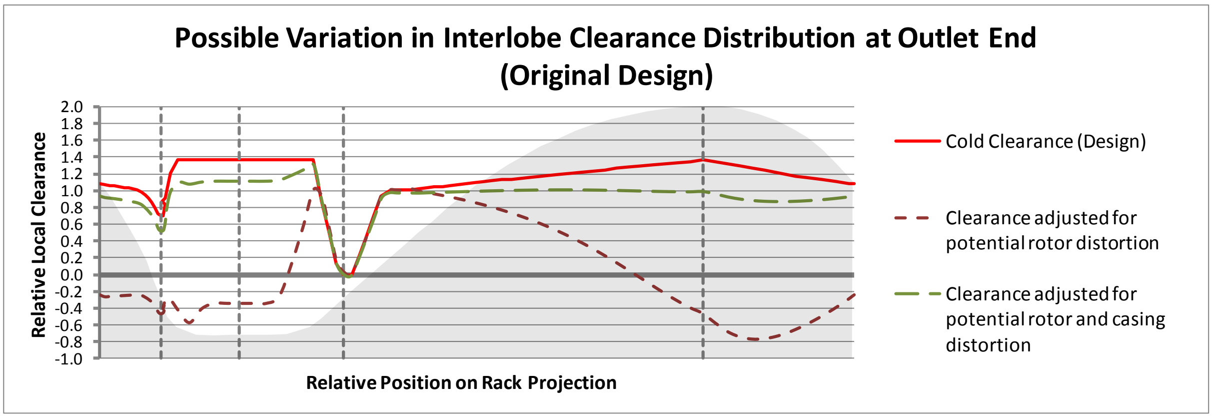

Figure 5. Possible variation in interlobe clearance distribution at outlet end (original design).

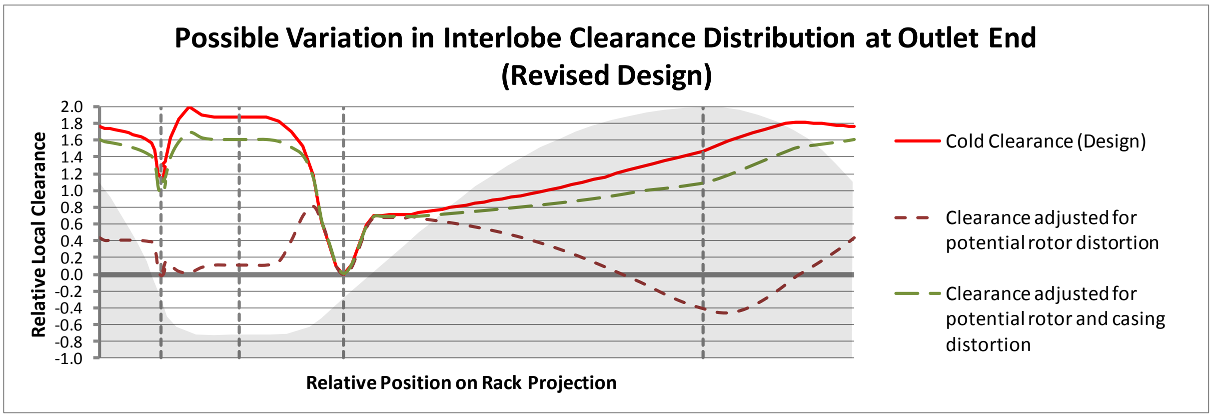

Figure 6. Possible variation in interlobe clearance distribution at outlet end (revised design).

This analysis supports the revised clearance in the root of the main rotor which shows contact cannot now occur there due to thermal distortion alone. The chance of contact at the tip of the main rotor has been significantly improved however the analysis suggests contact is still possible under extreme circumstances.

The most problematic clearance area is on the straight (undercutting) flank of the rotors – since distortions are transferred from the round contacting flank these clearances vary more than in other areas. What the figures do not show is that this represents a long part of the sealing line path due to the effect of undercutting, so this has a large effect on the overall leakage area. This area also experiences faster relative sliding motions between the rotor surfaces so contact should certainly be avoided. With these considerations in mind, there is probably still some scope to slightly reduce the clearances on both flanks just above the pitch locations.

4.2. Performance predictions

The models were also run at the nominal air test duty which is closest to the designed operating conditions: with a speed of 1400rpm, a pressure ratio of 5 and a discharge temperature maintained at approximately 90°C. The modelled deviation in the volume flow due to the revised design is compared with the deviation measured on test in Table 3.

Table 3. Performance penalty with revised interlobe clearance at pressure ratio of 5

	change in volumetric flow
Modelled	-0.5%
Measured on Test	-1.7%

The difference between the model result and the test result was higher than expected suggesting that the model is slightly underestimating the effect of leakages through the interlobe gap. Unfortunately as the test results were obtained using two completely different compressors the effect of other manufacturing and assembly tolerances can't be ruled out for the test results.

Importantly, the resulting change in flow on test was small enough that the compressor was still within normal operating tolerances in both cases. With the reduced flow there was a similar reduction in shaft power so there was very little change in overall efficiency. It is to be expected there would be some detrimental effect on performance at the nominal operating condition by increasing the rotor clearances to allow the compressor to operate at higher, non-standard, operating temperatures. This same design approach could be applied to investigate how far clearances could potentially be reduced for performance optimisation over a narrow operating band.

5. Conclusions

Results from a compressor model have been utilised to determine the fluid boundary temperature distribution on the surface of the rotors and casing. Estimated component temperatures were then used to evaluate how operational clearances are affected by thermal distortions.

Local interlobe clearance distributions have been superimposed against a rack projection of the rotors in order to compare small changes in local clearance while relating back to the location on the rotors.

Two interlobe clearance designs have been analysed to evaluate how suitable they are for operation at an elevated rotor discharge temperature of 120°C in an oil injected compressor with direct rotor drive. While there is a fairly large uncertainty on the predicted clearance deviations the results do confirm that the revised clearances should prevent rotor to rotor contact at the root of the male rotor. This is where contact was initially present on test, and later shown to be alleviated due to the revised clearance design.

Performance test results suggest that the increase in interlobe clearance has resulted in a 1.7% drop in flow however the overall flow was still within normal test tolerances. The change in flow predicted by the model was different from that measured on test.

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