Influence of Manufacturing and Operational Effects on Screw Compressor Rotor Pair Clearances

Einfluss von Herstellen und Betriebliche Wirkungen auf der Spalte einer Schraubenverdichter Rotorpaar

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Abstract

This paper illustrates the effects of manufacturing variation and operational loads on rotor-to-rotor clearance in a refrigeration screw compressor. Rotor clearances are computed according to methods developed in the Fachgebiet Fluidenergiemaschinen at the University of Dortmund [1]. A program implementing these methods is used to compute the clearance distribution along the full three-dimensional rotor-to-rotor seal line. The effects of manufacturing variation and operational thermal and pressure loads on this clearance characteristic are described. Then, a thermodynamic simulation of the compressor is used to show how compressor performance is affected by the different factors.

1. Introduction

Leakage through the clearance between the rotors is one of the more important factors in screw compressor performance. The mutually generated, theoretical rotor pair has no clearance; each point on the male rotor profile contacts a specific point on the female profile, these points of contact occurring at a specific rotation of the rotors. Some clearance between the rotors is necessary to ensure reliable operation, so rotors are modified from their theoretical forms. This results in a leak path that lowers both capacity and efficiency.

Much effort is put into manufacturing individual rotors with accurate profiles in order to minimize the resulting clearance in mated pairs. While the rotor's profile accuracy plays a role, there are other manufacturing and operational issues that affect the actual running compressor seal line clearance. Accurate positioning of the rotors relative to each other depends on the bearings and housings; operational stresses such as bending and thermal deformation change the rotor shapes and thus further affect seal line clearances.

Ideally, we would have minimum variation due to the manufacturing processes and would adjust the design of the rotors to accommodate changes caused by operational loads. However, use of cost effective manufacturing techniques and, ultimately, limits to manufacturing technology ensure that we will have to deal with rotor variation at levels that will affect performance. Furthermore, refrigeration compressors such as considered here are required to operate over a wide range of pressure and temperature in steady state and transient modes. Hence, there is not a single “deformed” state to design to.

It is useful, however, to have an understanding as to the nature and magnitude of the various effects when designing a rotor pair. The intent of this report is to outline methods we use to deal with the issues noted and to provide examples taken from analyses of rotors used in refrigerant R-134a compressors.

Some definitions and an outline of the analysis procedure are provided in the Section 2 of the report. Sections 3 and 4 show examples of the effects of manufacturing variation and operation loads, respectively. Then, we illustrate results of performance calculations to quantify the seal line clearance variations in terms of changes in compressor efficiency in Section 5. The report concludes with a short summary in Section 6.

2. Clearance Analysis Procedure

The seal line is the locus of points at which the theoretical forms of the male rotor flank, female rotor flank and the profile generating rack are in contact. For an actual rotor pair designed for use in a compressor, the profiles are modified to create some separation between the rotor flanks everywhere along the seal line, with the exception of a specific contact band where the male rotor drives the female. Figure 1 shows the seal line as it appears when the line of sight is along the normal rack angle. Nomenclature used in the presentation of results is seen to the right in Figure 1, showing a single lobe pitch where the inlet end of the rotors (not visible) is to the top and the discharge end (S=1) is at the bottom.

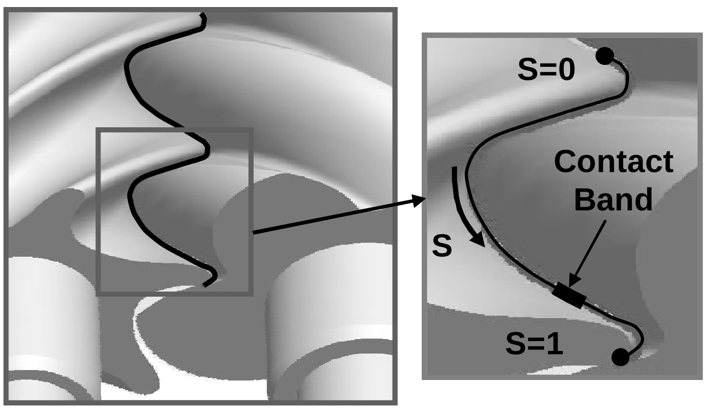

Fig. 1: Seal line view and definitions

A method of computing the intermesh clearance for a screw rotor pair whose lobes are defined by a locus of points for each rotor is reported in [1]. The method was provided us in the form of the *Intermesh Clearance* computer program (IMC) that allows for the profile to be defined by the nominal design form plus data given in an arbitrary number of files that define deviations from the nominal form. These deviations can be based on measurement of an actual rotor profile or can come from analysis of deformations due to operating loads and can vary from one end of the rotor to the other. The one limitation in the program is that all lobes are assumed to be identical and are always equally spaced circumferentially.

The IMC program computes clearance between the rotor flanks at numerous points along the seal line. To illustrate the characteristics for a particular analysis, clearance is plotted against distance along the seal line (S = S/Stotal), as shown in Figure 2. This form will be used throughout the report to display results from the clearance calculations. Presented this way, the area under the clearance vs. S curve is the average intermesh clearance.

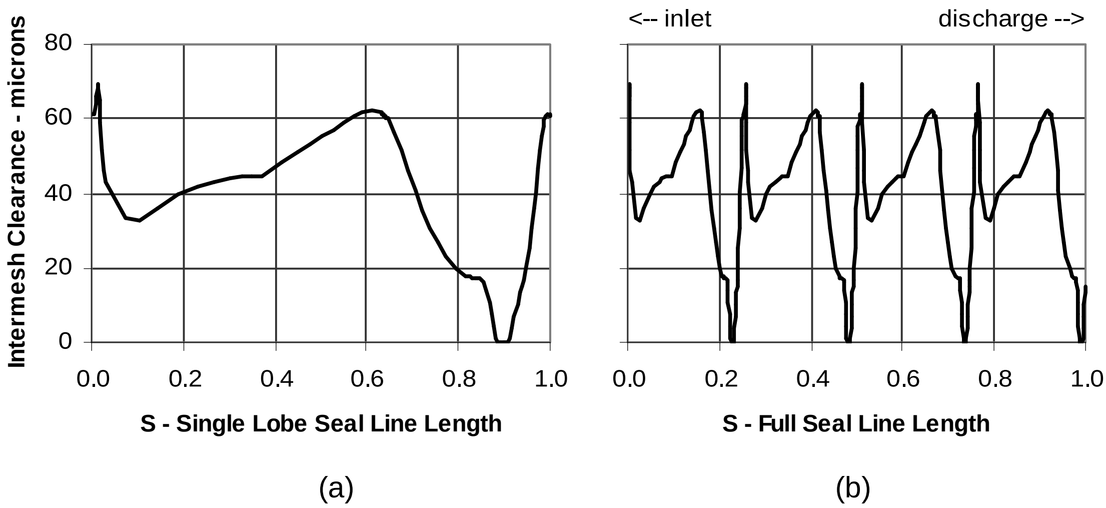

Fig. 2: Method of Displaying Intermesh Clearance (S = 0 is inlet end of rotor)

(a) Single lobe pitch

(b) Complete seal line

Figure 2, as with all other examples in this report, represents the clearance characteristic for a single rotor angular orientation. As the rotors rotate, the patterns in the figures would move to the right. This detail is not included in the analyses here.

The effects of several sources of deviation from nominal conditions on the clearance characteristics of the rotor pair designed for use in compressors applied in R134a water chillers are illustrated in the remainder of this report. Manufacturing variation effects are reviewed in the next section. Here, we provide an example of using actual rotor profiles measured on a coordinate measuring machine (CMM) to determine the intermesh clearance for a rotor pair to be tested. There are also examples of the effects of rotor centerline misalignment caused by manufacturing variation in bearings and housings. Following this is a section illustrating the effects of thermal and pressure induced deformation. In each case, the clearance characteristic calculated using the IMC program and the efficiency computed using the Trane screw compressor thermodynamic simulation are presented. The efficiency computations are based on the area average of the overall clearance characteristics. This clearance value can vary over the length of the rotor if the deviations are defined in three dimensions; this factor is represented in the calculations.

3. Effect of Manufacturing Variation on Intermesh Clearance

As-assembled rotor intermesh clearance will vary as a result of manufacturing variations in the individual rotors and due to variation in the position of the rotors relative to each other caused by manufacturing variation in bearing internal clearance and location of the bearings in the housings. It has been our practice to measure all of the parts that go into a compressor tested in our laboratory. This information is used to calculate the rotor intermesh and rotor-to-housing clearances. This process is described in [3].

Figure 3 shows the computed clearance characteristics for a nominal rotor and for a rotor whose profile is measured on a CMM. In this figure, as in all of the figures in the remainder of this report, the thin black line shows the clearance distribution from a rotor pair with nominal lobe profiles with their shafts perfectly aligned and separated by the nominal rotor design center distance. The bold line shows the case computed for comparison. In Figure 3, this would be the representation of the clearances based on measured lobe profiles.

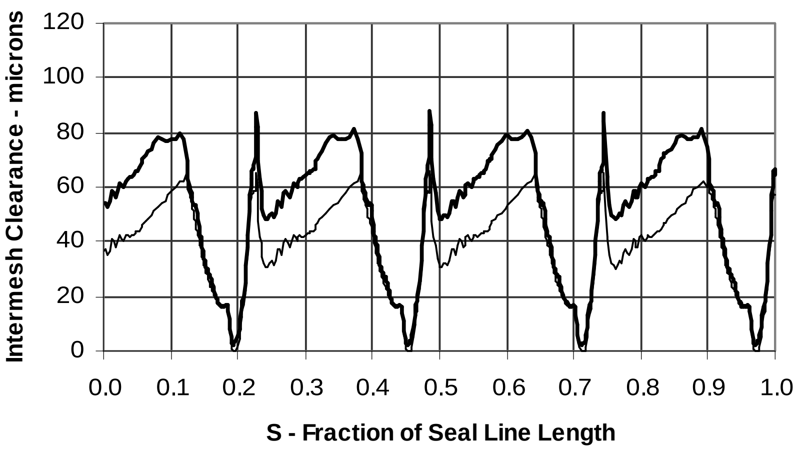

Fig. 3: Comparison of clearances of measured rotor pair compared to baseline

The average clearance across the entire rotor length for the baseline case is 39 microns while the actual rotor pair's clearance is 52 microns. With the tolerances we specify for the lobe profiles, extremes of intermesh clearance are a minimum of 18 microns and a maximum of 59 microns.

Another manufacturing variation is illustrated in Figure 4. Here, we examine the effect of having non-parallel rotor axes. The rotor axes are skewed by having variation in center distance between the inlet end bearings and the discharge end bearings. Variations in the center distance are due to variation in the bearing bores machined into the housings and in the variation in clearance of the bearings. Figure 4(a) shows the case where the inlet end of the rotors is at a maximum center distance (high clearance bearings and maximum allowable separation of the bores) and the discharge is at a minimum center distance (tight bearings, minimum allowable bore separation). The case of minimum center distance at the inlet end bearings and maximum center distance at the discharge end bearings is shown in 4(b).

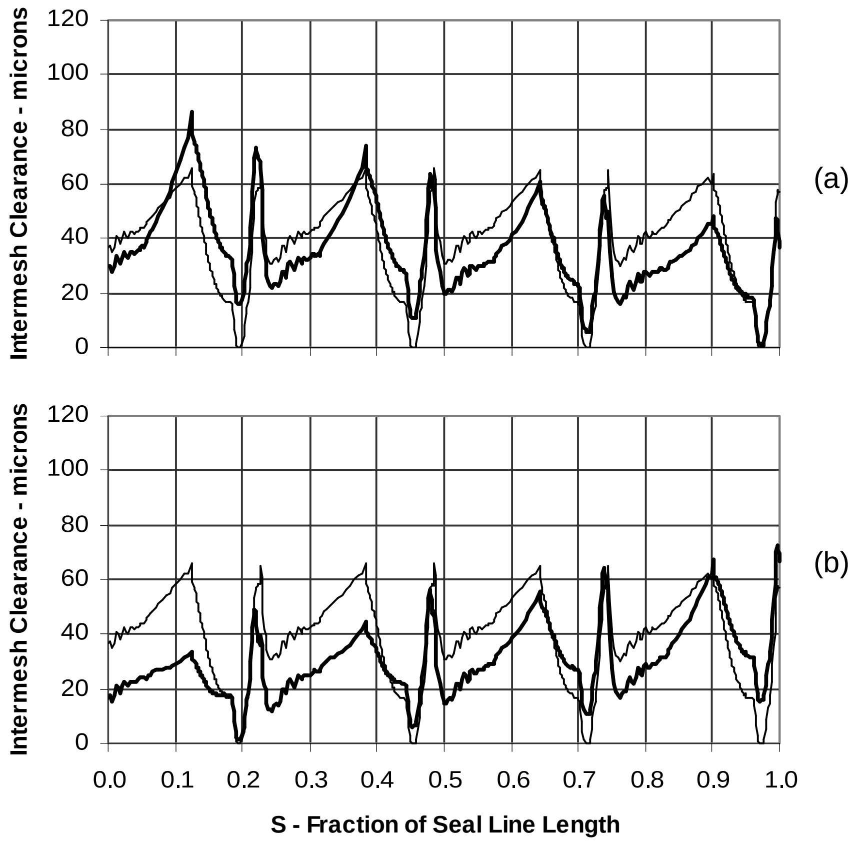

Fig. 4: Effect of skewing of the rotor axes relative to each other

(a) Center distance increased at inlet end, decreased at discharge end

(b) Center distance decreased at inlet end, increased at discharge end

For the case illustrated in 4(a), the inlet end center distance is 25 microns greater than nominal while at the discharge end, the center distance is 33 microns less than nominal. For the case in 4(b), the center distance is 36 microns below nominal at the inlet end and 20 microns above nominal at the discharge end.

The average clearances computed for cases 4(a) and 4(b) are 36 microns and 30 microns, respectively. The effect isn't symmetrical because the clearance characteristic is not symmetric along the rotor axes due to use of asymmetric rotor profiles and since the inlet end and discharge end bearings are not the same distance from the end planes of the rotors.

The skewing of the rotors also effects the rotor-to-rotor contact. For parallel axes, there are three or four contact zones, depending on the rotation of the rotors. This has an effect on the running quality of the rotor pair which influences compressor noise and vibration. Discussion of these effects is beyond the scope of this report; however, such rotor meshing effects are reviewed in references [4] and [5].

This example illustrates the value of using a tool such as the IMC program. Here, the three dimensional nature of the clearance characteristic as affected by axis alignment is clear. As reported in [3], we have a process that allows us to correlate measured performance with known clearance characteristics of the compressor tested. In the case of the intermesh clearance, the original work was carried out using a program that was only capable of computing average clearance based on the average rotor center distance and a single profile shape with no provision for variation along the length of the rotor. So the question arises as to what effect the three dimensional nature of intermesh clearance has on performance.

The Trane screw compressor simulation model was improved to allow specification of the details of the intermesh clearance characteristic. We used this program to compute the effects of the clearance cases generated for this report; results of these calculations are detailed in Section 5.

4. Effect of Operational Loads on Intermesh Clearance

Pressure and thermal loads on the rotors result in bending along the rotor axes and a general deformation of the profile shape, respectively. In addition, housing thermal deformations result in changes in the rotor center distance. Figures 4(a), (b) and (c) illustrate the effects these loads have on the computed intermesh clearance characteristics.

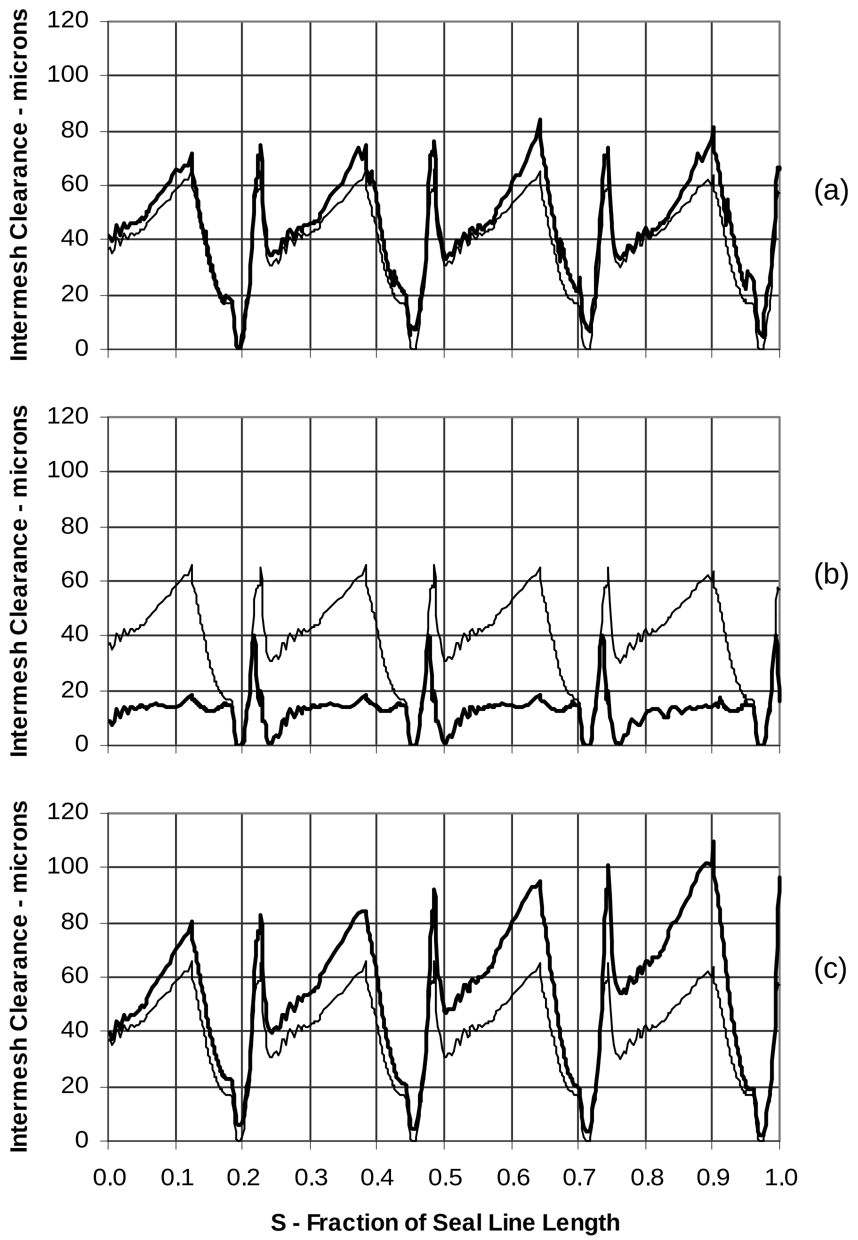

Fig. 5: Effect of operating loads on intermesh clearance characteristics

(a) Rotor bending due to pressure loads

(b) Thermal deformation of rotors only

(c) Rotor bending and thermal deformation of rotors and housings

The process of computing the deformations due to thermal loads is described in reference [6]. The approach used was developed at the University of Dortmund as noted in the citations provided in the reference.

Examination of the characteristics in Figure 5 shows that bending alone has little effect on the intermesh clearance; the average is only 6 microns greater than the nominal case. Thermal effects, on the other hand, are quite large. In the case of thermally deformed rotors, Figure 5(b), the rotors would run at zero backlash. The average intermesh clearance is 13 microns. However, when the effect of thermal deformation of the housings, which increases center distance at both ends of the rotor, is added, the clearance characteristic changes significantly. This is the case shown in Figure 5(c). The average clearance is 54 microns.

Finally, we will look at the situation where the rotor axes are skewed as in cases 4a and 4b with the rotor bending and rotor and housing thermal deformations used in case 5c. These cases are illustrated in Figure 6.

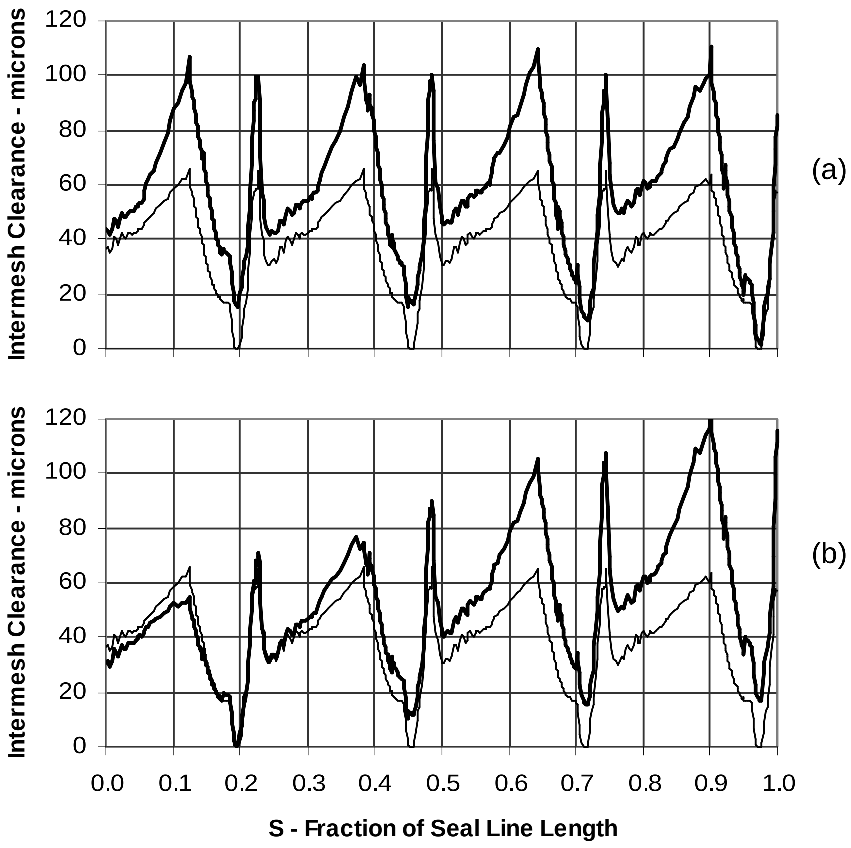

Fig. 6: Effect of operating pressure and thermal loads with skewed axes

(a) Center distance increased at inlet end, decreased at discharge end

(b) Center distance decreased at inlet end, increased at discharge end

The average clearances for cases 6a and 6b are 60 microns and 53 microns, respectively. Inspection of Figures 5 and 6 shows that the skewing of the rotor axes results in a greater clearance variation along the rotor axis than does the rotor thermal deformation.

Results of the calculations in Sections 3 and 4 are summarized in Section 5 along with the calculated effects of the various clearance characteristics on compressor efficiency.

5. Effect of Intermesh Clearance on Compressor Performance

A summary of intermesh clearances for the cases run for this report is provided in Table 1. The average clearance is the area under the clearance vs. relative seal line length curve. Clearances labeled Inlet and Discharge show the averaged clearance over one lobe pitch at the inlet or discharge end, respectively.

The baseline, min and max cases represent nominal, maximum and minimum material states of the profiles for a pair at nominal center distance with their axes aligned. Case 3 is the example of using measurement data to define the profile shapes. In these cases, the average clearance over one lobe pitch is the same, regardless of where it is measured. For the other cases, the axial variations of profile shape and/or center distance result in a variation from inlet to discharge.

Clearance Case	Case Description	Intermesh Clearance - Microns		
		Average	Inlet	Discharge
Baseline	Nominal	39	39	39
Min	Max material	18	18	18
Max	Min material	59	59	59
3	CMM data	52	52	52
4a	Skew: wide inlet	36	45	27
4b	Skew: wide discharge	30	22	38
5a	Rotor bending	45	42	46
5b	Rotor thermal	13	13	12
5c	5b plus housing thermal	54	48	59
6a	4a plus 5c	60	61	57
6b	4b plus 5c	53	35	68

Table 1: Clearances for Sample Cases

We computed compressor efficiency for 18 cases. There was one case each for the first 4 entries in Table 1 where the intermesh clearance input was the average value listed. Two calculations were made for the remaining cases: the first was based only on the average clearance, while the second was run with the actual clearance characteristic which included the clearance variation along the rotor axis. Results are shown in Figure 7.

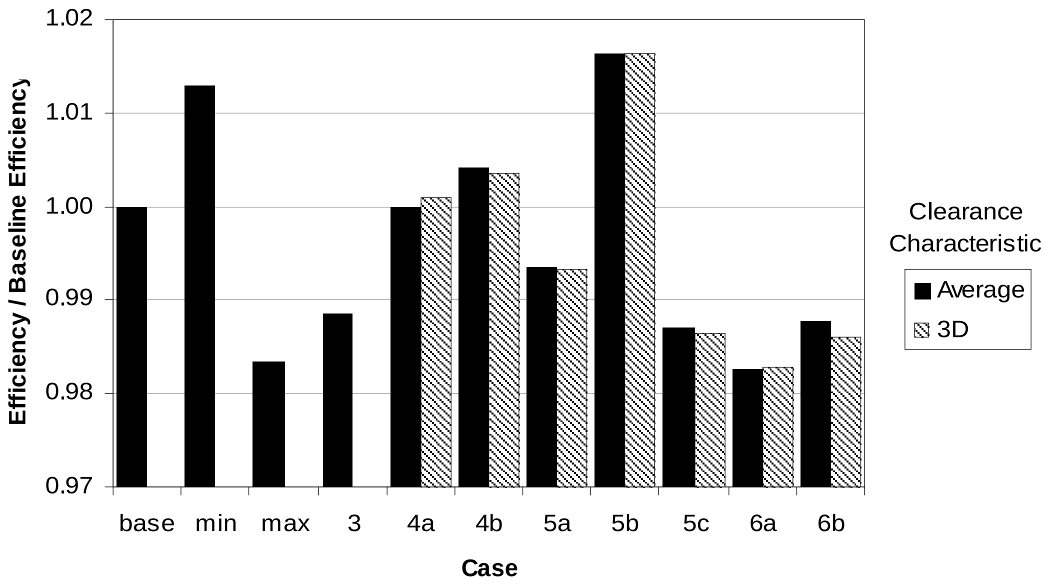

Fig. 7: Effect of clearance on compressor efficiency

Isentropic efficiencies are calculated using the simulation program and all cases are compared to the baseline making it is easy to see the percentage change in efficiency resulting from other clearance cases. Compressor efficiency is an important factor in the cost and performance of large water chillers. The cost of heat exchanger surface required to change chiller performance by 1% is substantial. Regulatory agencies and market forces determine what levels of water chiller performance must be offered. If a chiller can be designed with a more efficient compressor, heat exchanger cost savings can be realized. Conversely, costs increase if less efficient compressors are used. Changes in compressor efficiency on the order of 1% are significant.

The cases labeled base, min and max in Figure 7 show the targets for production rotors. Case 3 is an actual rotor pair whose profiles had been measured on a CMM. This rotor pair applied in an otherwise nominal compressor would result in an efficiency loss of just over 1%.

The remaining cases in the comparison show some interesting results. Cases 4a and 4b are those with misaligned rotor axes. Clearance details for these cases are shown in Figure 4. It is at first somewhat surprising that case 4a was not better than 4b, since the clearances near the higher pressure discharge end are lower. However, looking at Table 1, we can see that the average clearance of 4a is 20% above that of 4b and this offsets the effect of the tighter mesh near the discharge. Another factor, not evident in the clearance charts, is that during the discharge phase in the compression process, the length of the active seal line is constantly getting smaller. This too tends to dampen the effect of clearance magnitude.

The rotor bending effect, case 5a, is less than 1%, but still of significance. The greatest impact on efficiency from operational factors is from thermal deformations. These are illustrated in cases 5b and 5c (details in Figure 5). Considering rotor deformation alone (case 5a) results in a substantial reduction in intermesh clearance. In fact, a second contact on the trailing or “straight” side of the profiles is established. This effect alone would result in a nearly 2% higher efficiency. However, all of the compressor parts are subject to thermal deformation and in the case of the housings where the bearings are mounted, this results in an increase in center distance at both ends. When this effect is included along with rotor bending, case 5c, we get an actual operational effect of 1.5% efficiency reduction. Cases 6a and 6b complete the analyses by adding the extremes of axes misalignment. These cases compare to case 5c much as cases 4a and 4b compare to the *base* case.

A final observation from the analyses is the relatively small difference in results between cases computed using average clearance and those using the more detailed clearance variation details. Expectations were that factors that resulted in larger clearance at the discharge end would have greater impact on performance. However, use of actual tolerances and design details (especially the difference in distance of the suction and discharge bearings from the respective ends of the rotor body) resulted in various compensating factors. In addition, the continuous reduction in the length of the seal line during the discharge process tends to limit the effect of increased clearance.

6. Summary

Use of the IMC program to analyze the complex clearance characteristics of rotors as affected by manufacturing and operational factors has provided us with insights as to how compressor performance is affected and identifies possibilities for performance improvement. Profile tolerances and operational effects have the greatest impact on clearance and efficiency. The combination of thermal analysis and assembly modeling used with the IMC program capabilities allows us to quantify the effect of each factor and thus provide guidance for selection of improvement projects.

Rotor bending can be controlled through selection of profile form and rotor length. However, since these factors affect other leak paths and port details, selection of design parameters should be part of an overall optimization process. Thermal effects are difficult to control. Material selection and, to some extent, housing geometries can be considered to reduce center distance change that leads to higher clearances. Finally, improved manufacturing processes, which include designing the parts for manufacturing, can reduce the losses due to the need to accommodate profile form and housing variation.

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