Calculation of clearances in twin screw compressors

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Calculation of clearances in twin screw compressors

Ermin Husak[1], Ahmed Kovacevic[2], Isak Karabegovic[1]

[1]University of Bihac, Technical faculty, Irfana Ljubijankića bb. 77 000 Bihac, Bosnia and Herzegovina,
[2]City, University of London, Department of Engineering, Centre for Compressor Technology, London, UK.

erminhusak@yahoo.com

Abstract. Clearances between rotating and stationary parts in a screw compressor are set to ensure the efficient operation and allow for thermal deformation without unwanted contacts. The change in clearances is caused by both pressure and temperature changes within the machine. If clearances are too large, the increased leakage flows will reduce efficiency. However, if the nominal clearances are too small, contacts between the rotating and stationary parts can occur as a consequence of rotor and casing deformations. In order to determine the operational clearances, a numerical analysis of deformation of screw compressor rotors and casing has to be performed. This paper discusses how the temperature of rotor and casing surfaces calculated from the one-dimensional chamber model in the SCORG could be used as a boundary conditions for a steady state thermal and structural analysis of a screw compressor solid parts. Deformations of rotors and casing under temperature load were calculated using a commercial Finite Element Analysis code ANSYS. Operational clearance are estimated from these deformations and some recommendations for further work are proposed.

1. Introduction

Screw compressors are positive displacement machines that comprise pair of helically geared rotors contained in a casing as shown in Figure 1. Relative motion of rotors and casing causes change in the volume of the compressor working chamber which increases the pressure and causes the compression process [1, 2].

Clearances must exist between rotating and stationary parts to allow for the relative motion between rotors and casing. That in turn provides leakage gaps between rotating and stationary elements. The compressed fluid leaks through these leakage gaps and influences the efficiency of a screw compressor. The size of clearances is affected by the deformation of structural elements caused by pressure and temperature.

If the compressor rotors deform due to the increase in temperature, there is a risk of contact between the rotating and stationary elements and therefore risk of damage or complete failure of a compressor. In order to avoid that contact, designers increase compressor clearances. This however causes higher leakage losses which increases the working chamber temperature and further deforms the rotors [3]. Therefore it is desired to minimize the assembly clearance.

Due the fast development of manufacturing technologies in the past several decades it is now possible to manufacture screw compressor parts with high precision. Screw compressor rotors can now be produced at an economic cost with tolerances as small as 5 micrometres while casing bores can be manufactured with repeatability of 2 micrometres [4]. This gave possibility to manufacture screw compressor with low level clearances and avoid previously mentioned malfunctions.

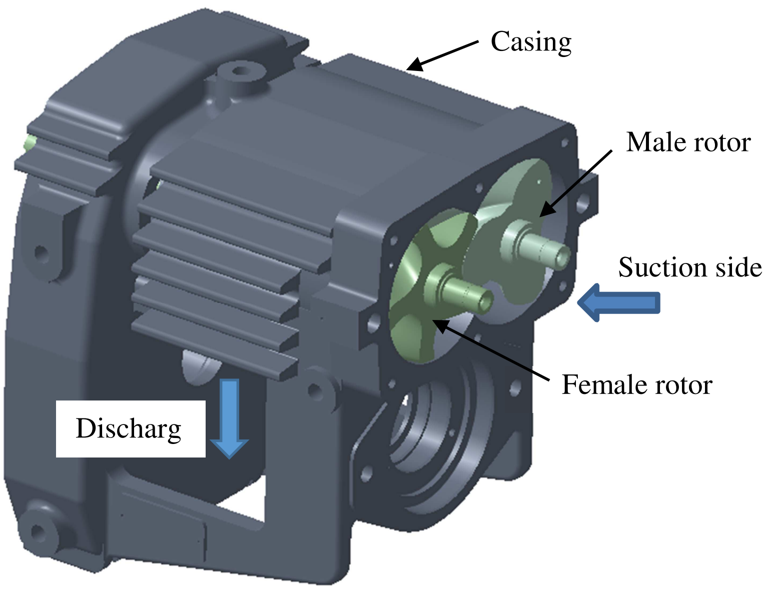

Fig. 1 3D model of the main casing and rotors of the oil free twin screw compressor

Despite the advanced manufacturing capabilities provide a deterministic framework to the design process of screw compressors the thermodynamic process during the operation of a screw compressor significantly affects the change in clearances. The increase in pressure and temperature will cause screw compressor parts to deform. Which of these two parameter will influence more on deformation depends mostly on the compressor type, i.e. is it oil free or oil injected [4].

Oil free screw compressors are mostly designed for low pressures of up to 3 bar but due to lack of cooling of the compressed gas, they usually have high discharge temperatures. In this case pressure loads are much less significant than the temperature loads.

Temperature loads in oil free screw compressors cause significant size and shape deformations which cause to clearance level changes. Typically, the interlobe and radial clearances on the discharges side of the compressor will reduce. To determine values of clearances under the working conditions especially under temperature changes, several steps have to be performed.

Typically, the first step is to perform calculations in order to determine the gas temperature and In turn to predict temperature distribution on the boundary of rotors and casing.

Second step is the thermal analysis which gives temperature distribution within rotors and casing. The third step is the structural analysis which gives deformations of rotors and casing. In order to perform such a process, due to complexity of screw compressor geometry simplification have to be taken into account. Different authors suggest different approaches to solve this problem.

Sauls et al, 2006 proposed to separately calculate temperature field in the compressed gas by CFD and then perform FEA analysis [5, 6, 7]. This process requires serious effort and time to perform calculation and transfer results between different software and it can last for months and is not practical for industrial use. Kovacevic et al, 2002 proposed to use Computational Continuum Mechanics (CCM). This method utilises the finite volume method in order to calculate both fluid flow and solid structure at the same time in the same solver. Using this method excellent results were reported [4]. The process is suitable for research but still requires significant effort to be implemented in industry.

The alternative is to use thermodynamic model to calculate gas temperatures and three dimensional analytical approaches to map these temperatures to the casing and rotors and determine local deformations. Such method proposed by Buckney et al, 2014 is applicable in industry, it is fast but takes into account some assumptions which cannot be always generally accepted [3].

It is important to emphasize that during the thermodynamic process in screw compressor heat will also transfer not just to the rotors and casing but to some other parts of the compressor too.

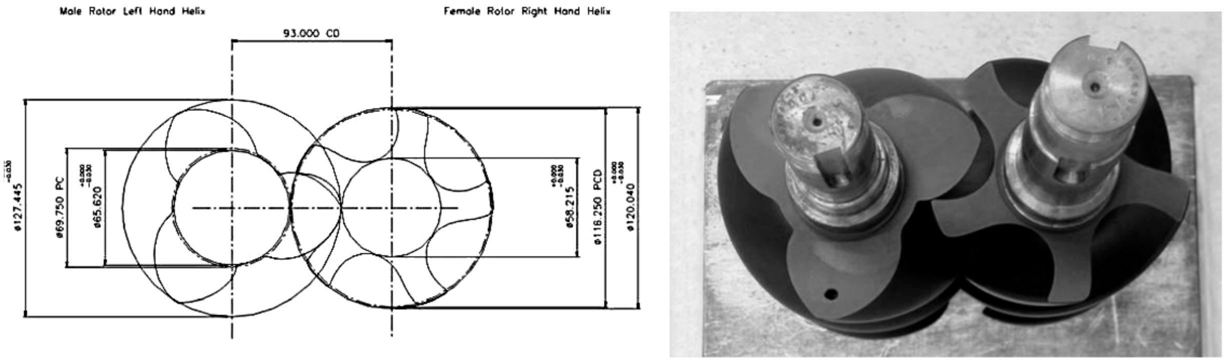

Fig. 2 Oil free screw compressor with 3-5 "N" rotors

This paper presents an integrated approach which uses results from the chamber model to calculate averaged temperatures of the rotors and casing which are then used as a boundary conditions for the steady state numerical analysis of screw compressor rotors and casing deformation using commercial solver ANSYS in order to estimate change in clearances. For this analysis oil free screw compressor with 3-5 "N" rotors showed in Figure 2 is used.

2. Thermal and structural analysis of screw compressor rotors and casing

Chamber models are often used to calculate compressor performance and determine temperature changes within the working chamber in time, i.e. with the rotation of rotors [1]. Such models are fast and reliable.

Screw compressors are positive displacement machines in which the change of pressure is caused by the change in volume. Due to relatively low speed, the influence of kinetic energy is often neglected. In such cases it is reasonable to assume that the pressure and temperature of the gas are the uniform for the entire chamber in one time instant. They change with rotation of the rotors in time as function of the change in volume. Therefore, pressure and temperature values calculated by use of chamber model can be assigned to the part of the rotors and casing depending on the rotation angle.

The computer code SCORG developed at City, University of London allows grid generation for CFD and calculation of preliminary thermodynamics. The grid generated in SCORG is exported to ANSYS – CFX for temperature and pressure calculations. State of fluid temperatures obtained from the chamber model and CFD simulation for one specific position is shown in the Figure 3.

Figure 3 shows the fluid temperatures next to the rotors and casing surfaces for one of rotor positions predicted by 3D CFD. This temperature distribution will change with the rotational angle. Similar results are achieved by a multi-chamber thermodynamic model built in SCORG shown in Fig 3a. The results are very similar and therefore with the high confidence the results of mapping from thermodynamic results within SCORG are used for calculation of rotor and casing deformations by FEA. Note the difference in temperature scale in Fig 3 and 3a.

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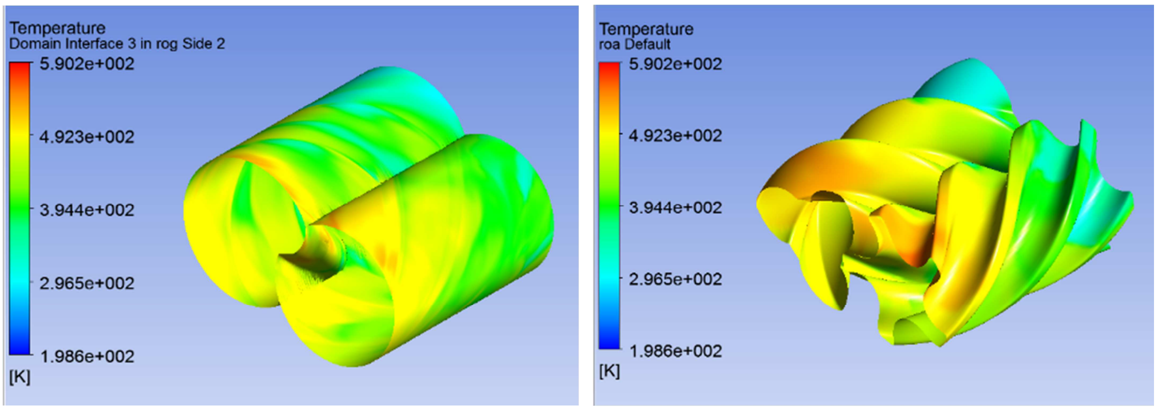

Fig. 3 Fluid temperature calculated by CFD near the casing (left) and rotors (right)

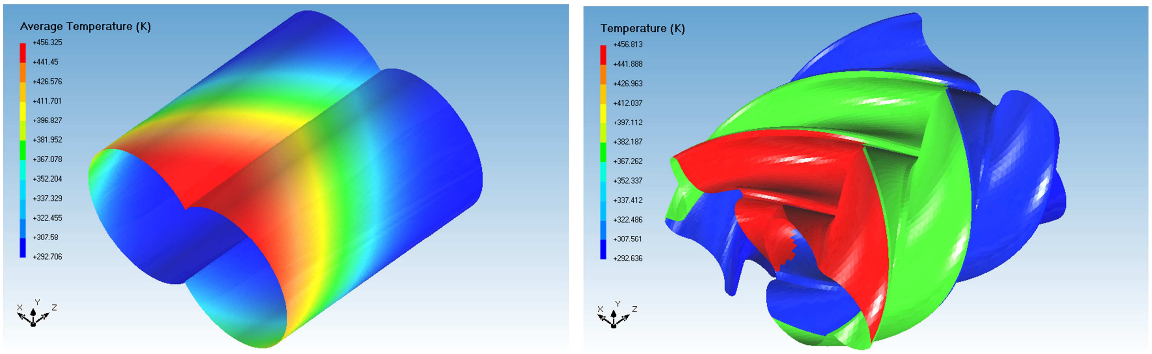

Fig. 3a Fluid temperature calculated by chamber model near the casing (left) and rotors (right)

Due to the cyclic operation of the rotors and the inertia of thermal conductivity, the local rotor surface temperatures will average over time and remain unchanged for the steady state operating conditions. These averaged temperatures are used to set boundary conditions as shown in Figures 4 and 5 [3, 5].

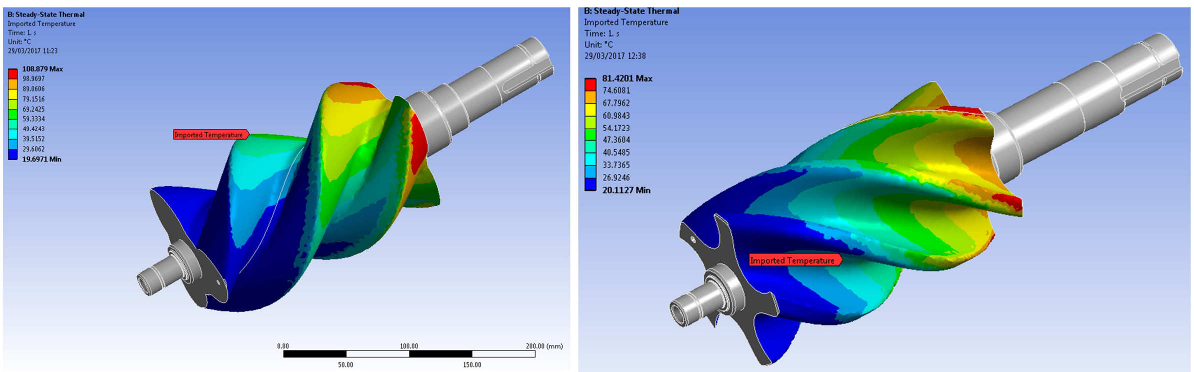

Fig. 4 Averaged temperature distribution on male rotor (left) and female rotor (right)

The average temperatures are than used as boundary conditions in the steady state thermal analysis of rotors and the casing performed by the FEA solver ANSYS.

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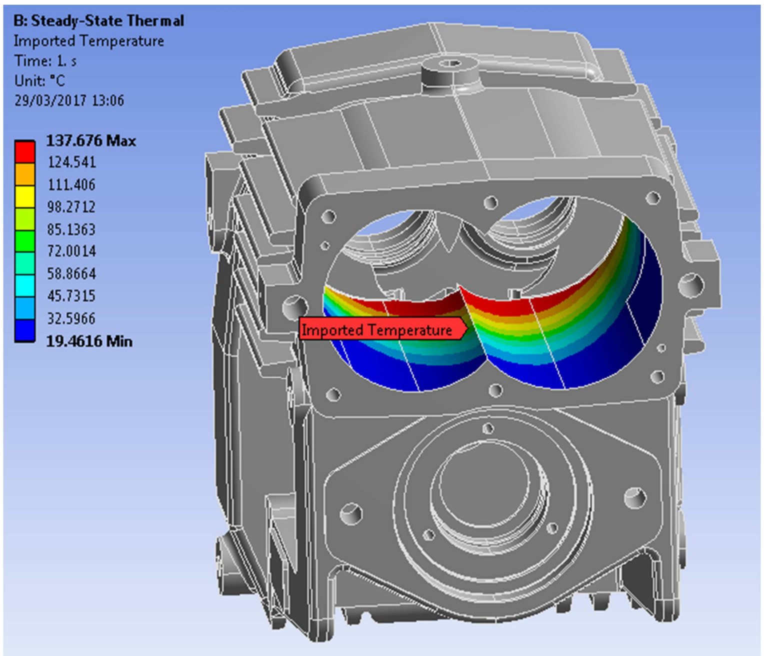

Fig. 5 Averaged temperature distribution on the casing

Figure 6 shows the resulting temperature distribution on male and female rotors after the steady state thermal analysis is performed in ANSYS. Despite the high gas temperature which reach 180˚C, the resulting temperatures of the rotor surfaces wary from around 20 to 82 ºC for the female rotor and 20 to 110 ºC for the male rotor. Temperature load for the casing are vary from around 20 to 138 ºC but the distribution over surfaces is completely different for the rotors and for the casing. Figure 7 shows temperature loads on the casing.

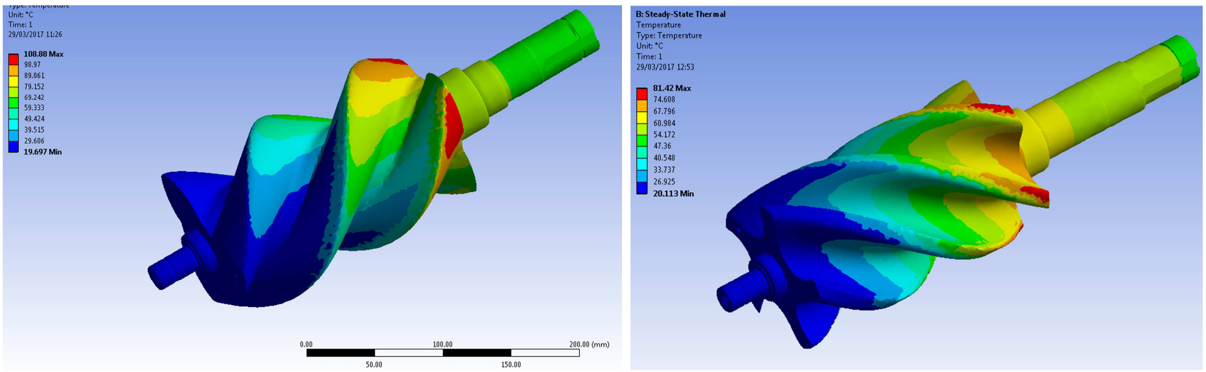

Fig. 6 Temperature loads on female (left) and male (right) rotors

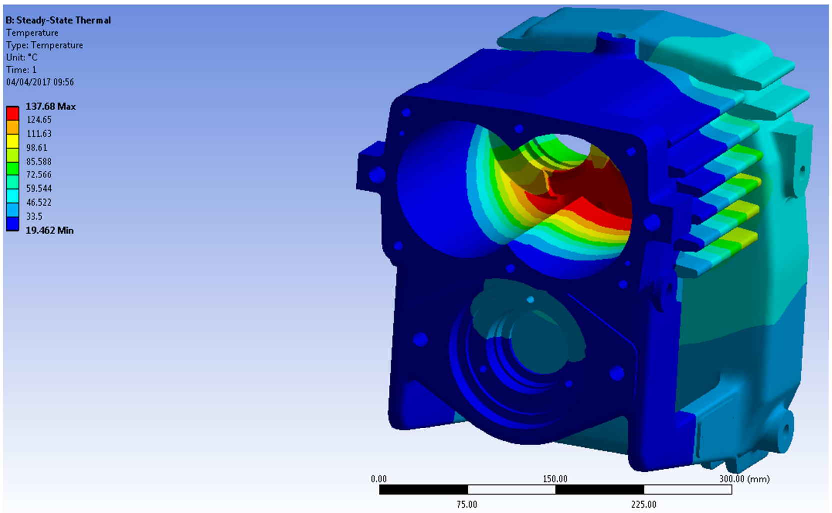

Fig. 7 Calculated temperature distribution on the casing

The casing has the highest temperature in areas close to the discharge port. For the purpose of verifying the calculated temperature loads, the experiment with the thermal imaging of this screw compressor has been performed. Figure 8 shows the cold state of the outside of the screw compressor casing.

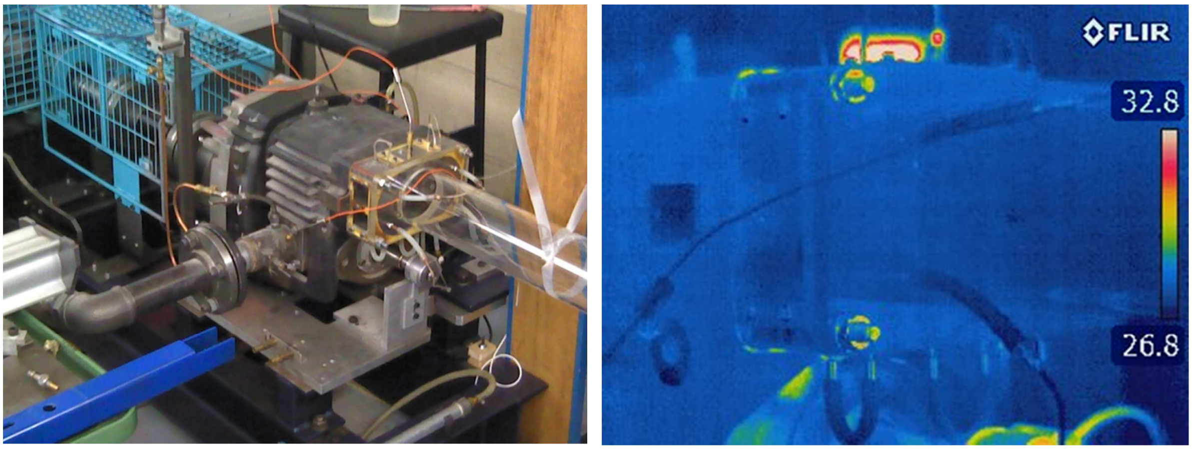

Fig. 8 Cold state of casing

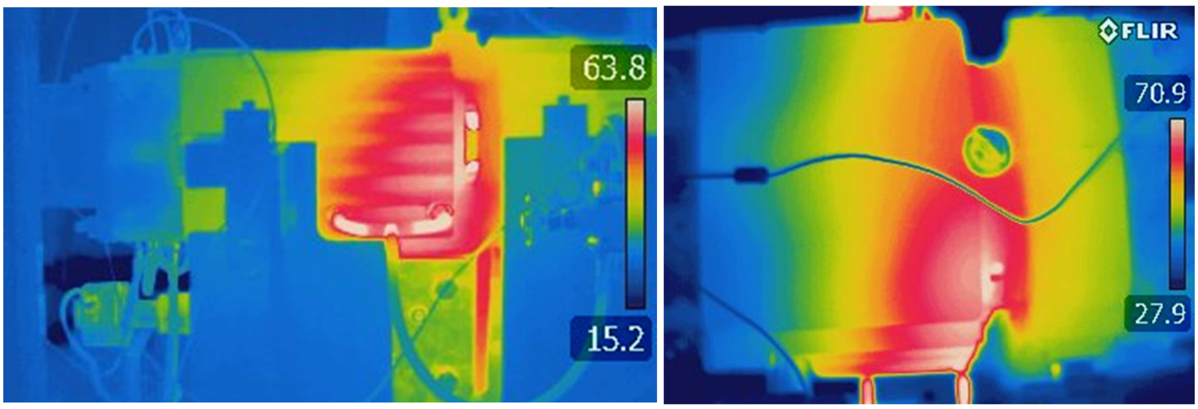

Fig. 9 Hot state of casing

Figure 9 shows measured temperatures on the casing during the screw compressor operation. It can be notices that the results of the temperature distribution on the outside of the compressor obtained by the numerical analysis and thermal imaging (Figures 7 and 9) are almost identical.

Once the thermal steady state analysis has been performed it is possible to start with structural analysis of the screw compressor rotor and casing under the calculated temperature loads.

Computation of rotors and casing deformation has been carried out by Finite Element Method (FEM) in ANSYS software. Numerical analysis for each part is performed separately. Temperature loads are taken from the steady state thermal analysis performed in the previous step. Rotors are restrained at bearings and casing is restrained in the position of the fixing screws.

Figure 10 shows deformations of the rotors and casing under the temperature loads. Deformations are enlarged to be visible in the figures. The rotor deformations are gradually increasing from the suction side to the discharge side which means that the largest deformations are on the discharge side where temperature field has the highest values. The maximum deformations of 100µm and 80µm are observed on the male and female rotors respectively. The deformations of the casing have much higher values in some parts of the casing which reach more than 200 µm. Within the rotor bores, the deformation is smaller. However the rotor centres on the discharge side have also moved. This means that the clearances set in the assembly process have changed.

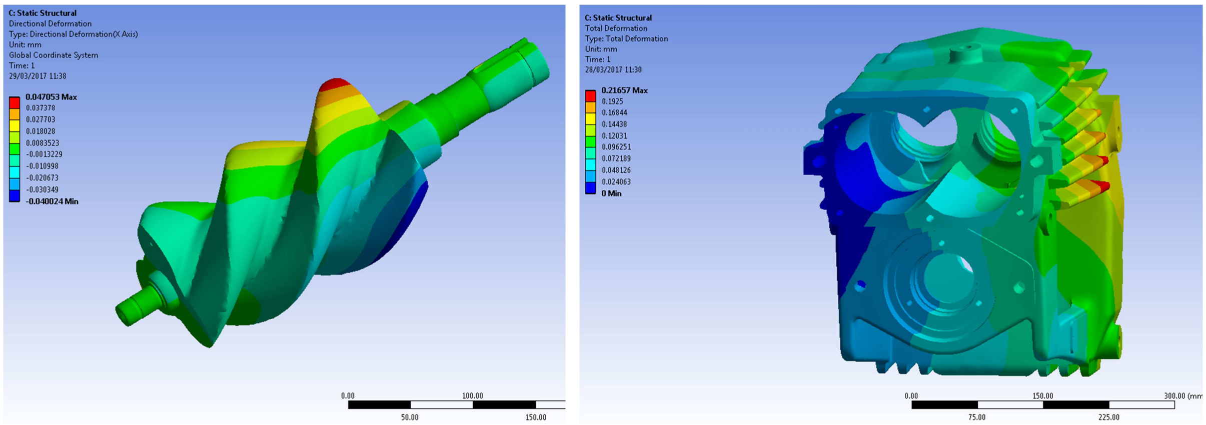

Fig. 10 Enlarged deformations of rotors and casing

If only the rotor deformations are taken into account, a contact will occur between the rotors and the casing as well as in between the rotors. However, in reality both, the screw compressor rotors and casing will deform. Therefore the deformation of the casing needs to be included is the analysis. Deformations of the casing are significantly different than the rotors deformations.

The casing is not deformed symmetrically around the axis like in the rotors case. Casing expansion will make some free space for the expansion of the rotors. Due to the slight change in the position of rotor axes the contact will be avoided but the clearances will be affected.

3. Analysis of clearances

Using the obtained results, it is now possible to determine displacements of rotor and the casing and calculate the change in clearances. In order to visualise this change the authors have decided to use five characteristic locations identified by the in Figure 11. The change in radial clearances is examined along the top and bottom of the rotors and in the cusp between two rotor bores as shown in the Figure 11.

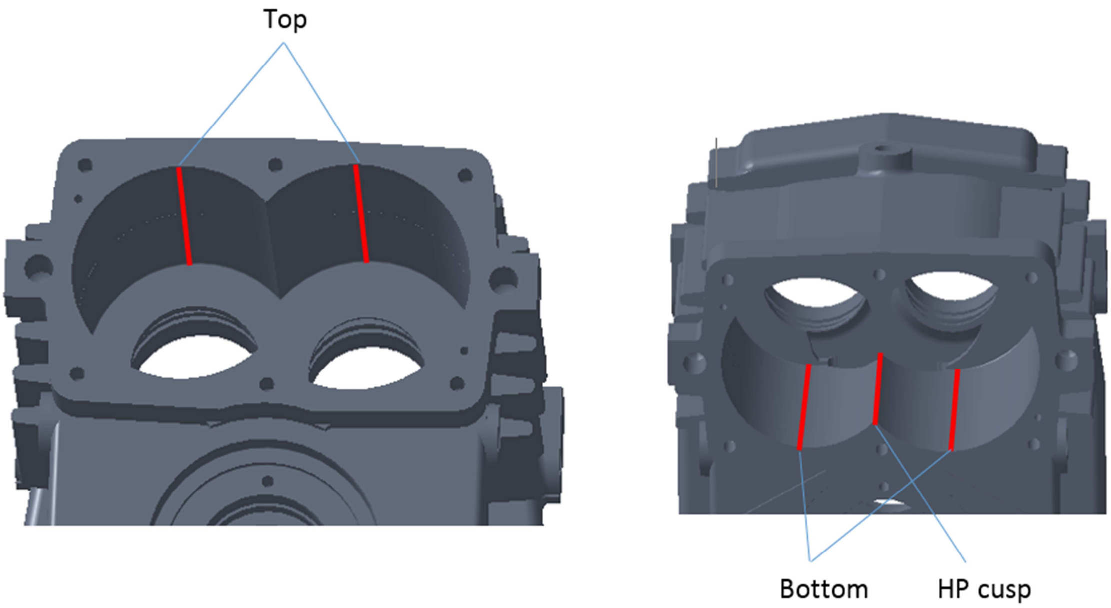

Fig. 11 Analysed positions of radial clearances

The assembly value of the radial clearance for this type of compressor is 180 µm. The length of the rotors and the line for observing the changes of the clearances is around 220 mm except for the HP cusp which is somewhat shorter due to the location of the radial discharge port. In the calculation of clearances the points on the rotor with maximum radius in each axial position along rotors are taken in consideration. It is considered that there are no other significant disturbances to the clearance distribution like for example the movements in bearings.

Compressor rotors expend due to the increase in temperature which causes reduction of radial clearances towards the discharge. The displacement values are subtracted from the total value of the predetermined clearances of 180 µm in the points along the rotors.

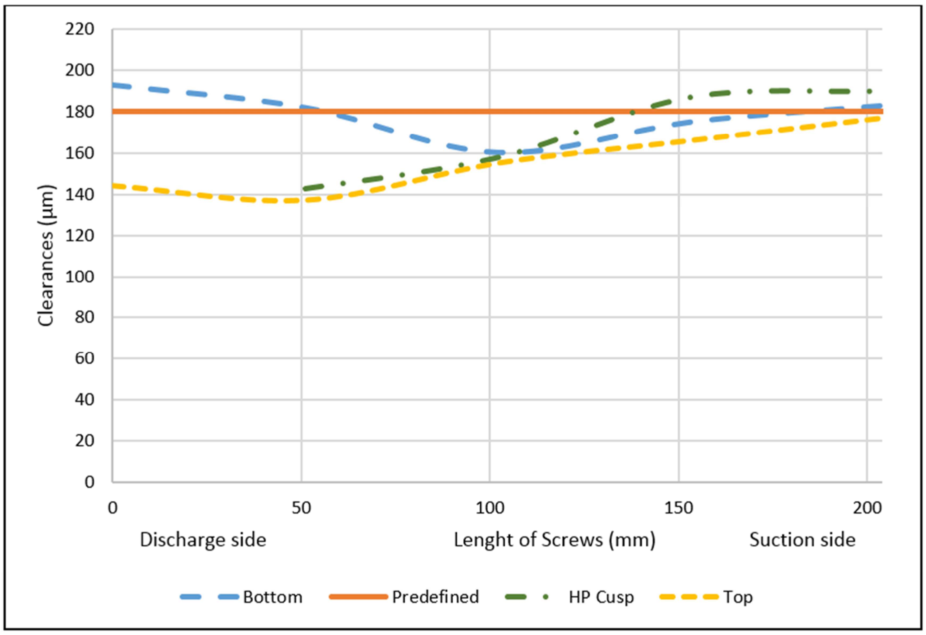

Fig. 12 Change of female rotor – casing clearances

Determining the displacement of the points on the casing from FEA results is significantly more difficult since the casing deformation is asymmetric. Firstly, it is necessary to determine the new position of bearing centres due to displacements in the casing. Secondly, the positions of points along the pre-defined lines relatively to new rotor axis has to be calculated. The radial clearances along the defined lines are calculated by adding or subtracting values of rotor and casing deformations. The results are shown in the Figures 12 and 13.

Figure 12 shows the change of the radial clearances between the female rotors and the casing on the Top and Bottom side of the casing as well as of the HP cusp. The largest change in clearances is on the discharge side indicated with lengths close to 0, while the deformations on the suction side are almost not existent. The different regions along the rotors have different variation in clearances values. At the Top of the rotor bore there is a 40 µm reduction of clearances on the discharge side between the rotor and the casing, which then increases for few microns towards the mid rotor position after which it reduces towards the suction. In the Bottom position, there is an increase of the clearances on the discharge side followed by a decrease in the middle and then again increase on the suction side.

The HP (high pressure) cusp line is on the same side as the Bottom line, the change in clearances is significantly different because of the growth of HP cusp. The clearances are reduced close to the discharge side but increase beyond the nominal clearances toward the suction side.

Figure 13 shows the clearances change between the male rotor and the casing. The three positions marked with lines are analysed. The changes in the clearances have a similar character to that in Figure 12 but with significantly different values. The clearance value at the discharge side at HP cusp position, according calculations, is reduced 80 µm. At the Top position, the critical region is reduced by over 60 µm. At the suction side there is almost no significant clearances change.

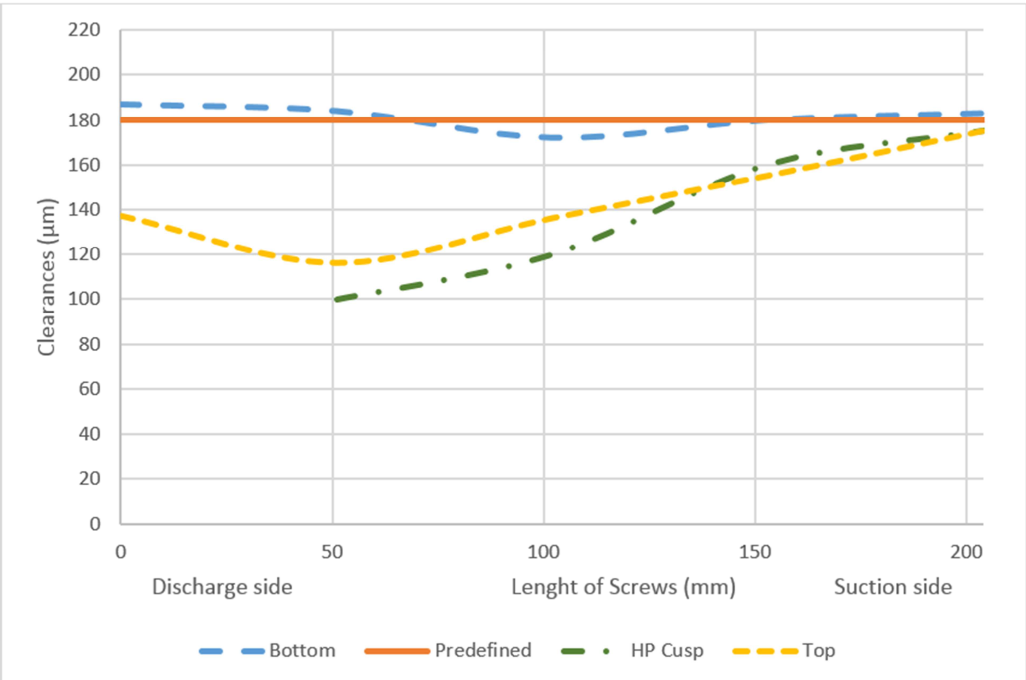

Fig. 13 Change of male rotor – casing clearances

The analysis shows that different regions have different levels of deformations. Deformations in the suction zone are significantly lower than in discharge zone which influences clearances to change differently along the rotors and casing.

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4. Conclusion

Changing the clearances value affects the efficiency and reliability of screw compressors. It is therefore of the utmost importance to establish an efficient method of determining this change. The major factor that affects changes of clearances is an oil free compressor is the internal temperature of the compressed gas.

In order to calculate the change in radial clearances between rotors and the casing it was necessary to link several calculation tools. The multi-chamber model built in SCORG program was used to obtain the average temperature on the surfaces of rotors and casing which are then imported in ANSYS to calculate deformation of structural elements.

The radial clearances between the rotor and the casing are calculated from their relative position and presented along five lines identified in the rotor bores. The largest reduction in radial clearance was recorded between the male rotor and the high pressure cusp on the discharge side. The nominal clearance of 180 μm reduced in that area by 80 μm. The work is continuing on further validation of this fast and reliable method for analysis of the clearance change and their effect on the performance of screw compressors.

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