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A. Kovacevic
London City University

N. Stosic
London City University

I. K. Smith
London City University

FLUID- SOLID INTERACTION IN SCREW COMPRESSORS

Ahmed Kovacevic, University Research Fellow
Nikola Stosic, Chair in Positive Displacement Technology
Ian K. Smith, Professor of Applied Thermodynamics

Centre for Positive Displacement Compressor Technology
City University, London EC1V 0HB, UK; Tel: +44 20 7040 8780; Fax: +44 20 7040 8566
E-Mail: a.kovacevic@city.ac.uk

ABSTRACT

Pressure and temperature differences within screw compressors cause the various components to distort. The analytical methods currently used to predict the distortion of the components are insufficiently accurate to account for the complexity of the processes within these machines. Accordingly, in order to avoid rotor contact and seizure, the design clearances normally selected are larger than those required to satisfy manufacturing tolerances. The aim of this study was therefore to use full 3-D numerical modelling of solid-fluid interaction within the machine to determine minimum safe clearances at the design stage and thereby maximise the compressor efficiency.

To this end, the authors have developed an independent stand-alone CAD-CCM (Computational Continuum Mechanics) interface program in order to transfer the screw compressor geometry directly into a commercial CCM solver. The interface program employs a rack-generating procedure and an analytical transfinite interpolation method to obtain a fully structured, block oriented, hexahedral 3-D numerical mesh of the screw compressor rotors and their working chambers. All necessary initial and boundary conditions, a deformation procedure for the moving mesh and additional functional features for the commercial solver are introduced through the user functions automatically generated by the interface program. By this means, the interaction between fluid flow and deformed solid body are predicted for screw compressor.

Results obtained for a dry air compressor are presented in this paper and it is shown, by means of flow and distortion diagrams and pressure-angle diagrams of the compression processes, how rotor profile clearances vary in the machine.

INTRODUCTION

Screw compressors are currently designed with the aid of mathematical models, based on the assumption of dimensionless or quasi-steady flow through rigid components. However, pressure changes within the working chamber during the compression process lead to large loads on the rotors, which cause them to bend and deform. This increases clearances in areas where the pressure difference is the highest. Thus internal leakages become greater in these regions. In addition, the effect of thermal expansion is to maximise rotor enlargement in the regions of highest temperature. This in turn reduces rotor interlobe clearances towards the inlet and discharge ends of the compressor.

Improvements in manufacturing methods now enable the various compressor components to be manufactured to such close tolerances that rotor deflection has become a significant parameter affecting the clearances within the machine and hence must clearly be accounted for at the design stage in order to minimise internal leakage without leading to seizure. Consequently, the simplified analytical models currently in use are not sufficiently accurate to design screw compressors to obtain the maximum possible improvements. The next stage in improving these machines must therefore be based on full 3-D numerical calculation of the fluid-solid interaction within. The best means of achieving this is with the aid of analyses based on computational continuum mechanics (CCM) procedures.

Apart from the authors' publications Kovacevic et. al [1999], [2000] and [2001], there is hardly any reported activity in the use of CCM for screw compressor studies. This is mainly because the existing grid generators and the majority of solvers are still unable to cope with the problems associated with both, the screw compressor geometry and the physics of the compressor process. Also, difficulties in obtaining simultaneous calculation for solid and fluid domains in order to evaluate fluid-solid interaction have contributed to the lack of publications in this area.

A pre-processing interface has been developed by the authors in order to generate a 3-D numerical grid for the purpose of simultaneous calculation of the compressor structure and its fluid flow. The interface employs a rack generation procedure to produce rotor profile points and analytical transfinite interpolation between them with adaptive meshing to obtain a fully structured 3-D numerical mesh of both solid compressor elements and fluid flow areas. This grid is directly transferable to a CCM (Computational Continuum Mechanics) code. The grid accounts for simultaneous moving, stretching and sliding of the rotor domains and results in robust calculations within domains of significantly different geometrical ranges. Some changes needed be made within the solver functions to enable accurate and faster calculations. These include a means to maintain constant pressure at the inlet and outlet ports and interaction between solid and fluid domains.

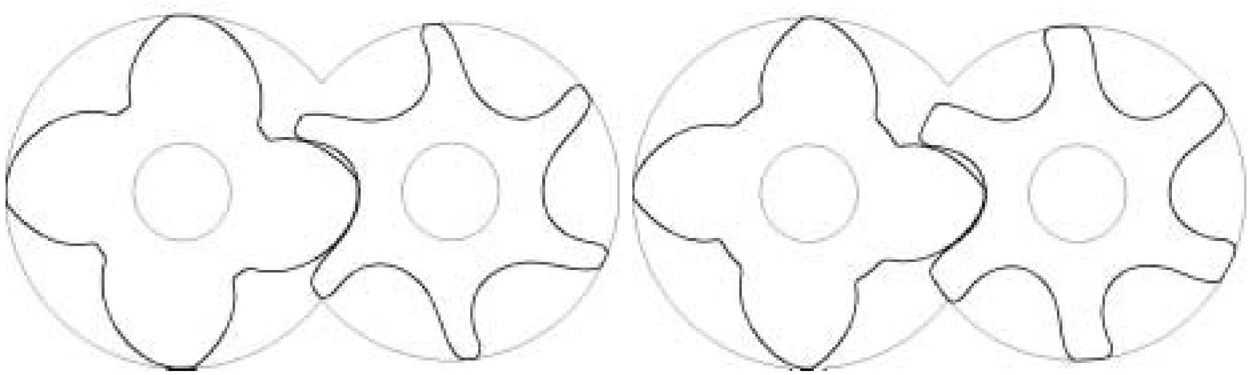

Figure 1: Dry air screw compressor rotor profiles (left – Case 1; right – Case 2)

A dry air compressor was chosen for analysis in this paper. The same housing is used as the casing for two sets of compressor rotors. Both are based on 'N' profiles with a 4/6 lobe configuration. The rotor outer diameters are 143 and 138 mm for the male and female rotors respectively and their centre axes are 108 mm apart. The rotor length to diameter ratio is 1.77. Both rotor sets are presented in Fig 1. They can be distinguished only by the male and female rotor lobe thickness. The female rotor lobes are substantially bigger for the second set of rotors. Despite their similarity, the second set of rotors has a 4% greater displacement and a 2% longer sealing line than the first. This difference in geometry was expected to result in more flow and slightly better performance for the second set of rotors. The most pronounced difference was expected in the female rotor torque and in the distortions of female rotor in the area of the discharge port.

NUMERICAL SOLUTION OF THE FLUID-SOLID INTERACTION

The density of the compressor working fluid changes with both pressure and temperature. The compressor flow and compressor parts structure is fully described by the mass averaged conservation equations of continuity, momentum, energy and space, which are accompanied by the turbulence model equations and an equation of state, as it is, for example, given by Ferzinger and Peric [1995]. In the case of a multiphase flow, the concentration equation is added to the system. The numerical solution of such a system of partial differential equations is then made possible by inclusion of constitutive relations in the form of Stoke's, Fick's and Fourier's law for the fluid momentum, concentration and energy equations respectively and Hooke's law for the momentum equations of a thermo-elastic solid body.

The resulting system of partial differential equations is discretised by means of a finite volume method in the general Cartesian coordinate frame. This method enhances conservation of governing equations while at the same time enables a coupled system of equations for both metal and fluid regions to be solved simultaneously. Demirdzic and Peric [1990] set the guidelines for successful finite volume calculation of 3-D flows in complex curvilinear geometries. Demirdzic and Muzaferija [1995] showed a possibility of simultaneous application of the same numerical methods in fluid flow and structural analysis within moving frames on structured and unstructured grids.

This mathematical scheme is accompanied by boundary conditions for both the solid and fluid parts. A special treatment of the compressor fluid boundaries was introduced in the numerical calculation. The compressor was positioned between two relatively small suction and discharge receivers. By this means, the compressor system becomes separated from the surroundings by adiabatic walls only. It communicates with its surroundings through the mass and energy sources or sinks placed in these receivers to maintain constant suction and discharge pressures. The solid part of the system is constrained by both Dirichlet and Neuman boundary conditions through zero displacement in the restraints and zero tractions elsewhere. Connection between the solid and fluid parts are explicitly determined if the temperature and displacement from the solid body surface are boundary conditions for the fluid flow and vice versa.

The numerical grid is applied to the commercial CCM solver to obtain distribution of pressure, temperature, velocity and density fields through out the fluid domain and deformations and stresses of the solid compressor elements. Based on the solution of these equations, integral parameters of screw compressor performance were calculated in the form of force reactions on restraints and torque together with volume and mass flows.

GRID GENERATION FOR FSI

An appropriate numerical grid must be generated as a necessary preliminary to a CCM calculation. The grid must define both the stationary and moving parts of the compressor. The rotors form the most complex part of the screw compressor grid and are the most important components since it is within the rotor interlobe chambers that the compression process occurs.

Many authors extensively discussed contemporary grid generation methods. The most detailed textbooks are Liseikin [1999] and Thompson et al [1999]. Adequately applied, the grid generation they describe, accompanied by an appropriate CCM solver, lead to the successful prediction of screw compressor fluid-solid interaction. Such an approach resulted in the algebraic grid generation method, which employs a multi parameter adaptation. This is given in detail by the authors in Kovacevic et. al

[2000] and [2001], where an interface, which transfers the screw compressor geometry to a CFD solver, is also described and compressor suction flow is given as a working example.

The compressor spatial domain is replaced by a grid, which contains discrete volumes. A composite grid, made of several structured grid blocks is patched together and based on a single boundary fitted co-ordinate system. Block structured grids allow easier numerical simulation of fluid flow and structural analysis for complex geometries. The grid generation for compressor rotors starts with the definition of their spatial domains inside the rotors, representing metal and outside the rotors, representing fluid. The rotor profile coordinates and their derivatives determine compressor domain borders. These are obtained by means of the rack generation procedure described in detail by Stosic [1998]. The grid components define all connections between the rotors and the housing and contain the interlobe, tip and blow-hole leakage paths. Domains of the fluid around the rotors and the rotors themselves are simultaneously generated in a single, fully structured block. This allowed a change of interlobe and radial clearances to be accounted for in the calculation of flow change due to deformations of the rotors. The grid calculation is based on an algebraic transfinite interpolation procedure with a static multi parameter adaptation on boundaries. This includes stretching functions to ensure grid orthogonality and smoothness.

RESULTS AND DISCUSSION

Grids and other control parameters generated by interface were applied to a commercial CMM solver, Comet, and the results obtained are given in this paper for the two different profiles of dry air screw compressor already described. The overall compressor parameters such as the torque, volume flow, forces, efficiencies and compressor specific power were calculated. The pressure-time diagrams of the compression process, the flow, pressure and temperature field patterns in the compressor chambers and rotor deformations are provided. These give more detailed insight into the results obtained. All calculated results can be employed to improve the design of screw machines.

The compressor flow is presented through velocity patterns in Figure 2. Two diagrams in Figure 2 are given in the cross section along female rotor axis. The rotors in Case 1 give more recirculation of the fluid within the working chamber and greater velocity in some regions of discharge port. Due to a 2% longer sealing line, leakage flows through clearances are also greater.

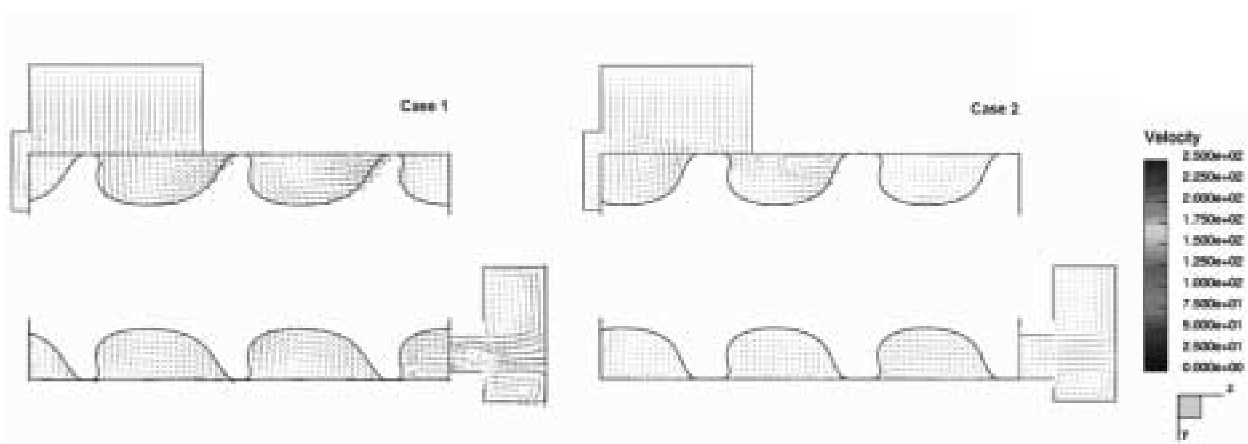

Figure 2: Flow patterns in the axial cross section of dry air screw compressor

The pressure field for the rotors in Case 1 is given in the left part of Figure 3 in which the boundary pressures on the rotors and the distribution of pressure in the suction and discharge ports are shown. In the right diagram the temperature field is presented along the cross section through both, the male and female rotor axes of the compressor with Case 1 rotors. Both the solid and fluid flow regions are included in that figure. The temperature in the fluid paths is presented with darker shades according to the rise of pressure in the rotor interlobes. The darker the shade, the higher is the temperature. The temperature field in the solid rotor domains changes along the rotor axes as a consequence of the surrounding fluid temperature. The temperature in a cross section perpendicular to the rotor axis is almost constant. The maximum rotor temperature appears at the discharge side of the compressor with the value approximately half the temperature difference between those of the discharge and suction of the air. Such a distribution of temperatures causes the rotors to enlarge in the regions closer to discharge as presented in Figure 4 where rotor distortions are presented on a deformed grid.

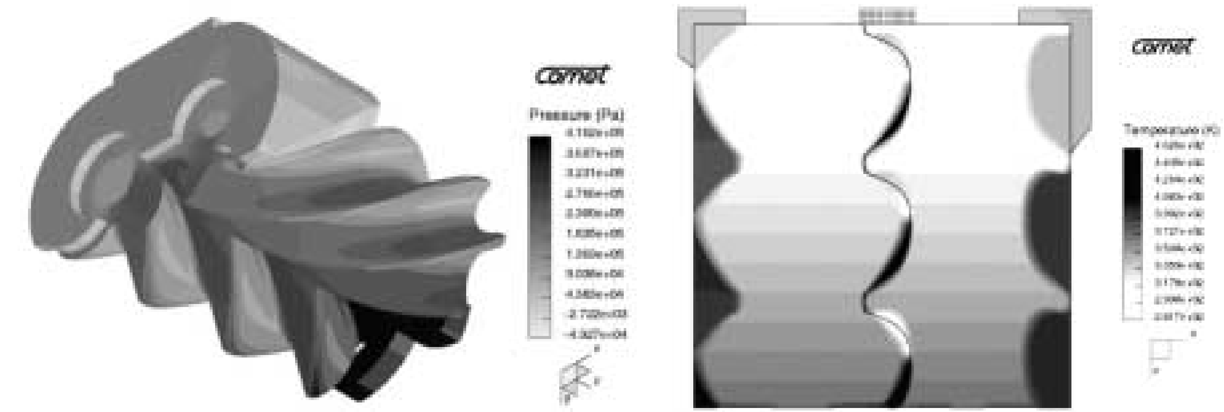

Figure 3: Pressure and temperature distribution in dry air screw compressor

The amount of displacement corresponds to the shade of grey. Darker shades indicate higher displacements and vice versa. The maximum displacement is of the order of 10[ths] of micrometers. This indicates a reduction of the interlobe clearance. It may be seen from Figure 4 that the distortion of the female rotor is greater than the distortion of the male rotor in both cases and that rotor distortion is very similar in both Cases 1 and 2.

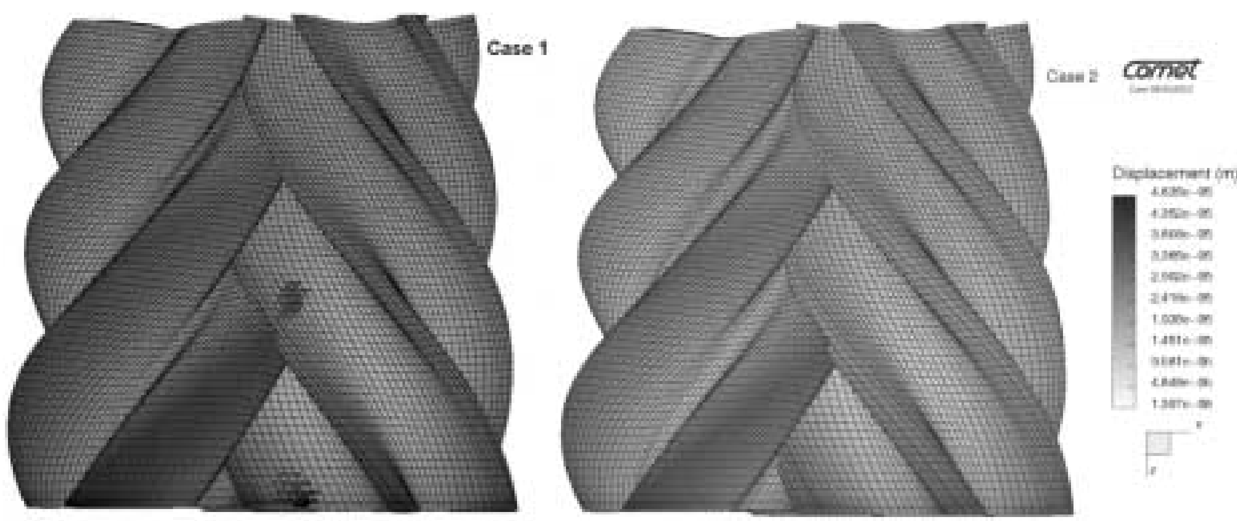

Figure 4: Rotor deformations caused by temperature dilatations

The calculated pressure history in one of the rotor interlobes is presented in Figure 5 as a function of the shaft rotation angle for both Case 1 and Case 2. Comparison of the measurements and simulation model presented by the authors [2001] show good agreement with similar case to this, not only for the compression process, but also for the pressure fluctuation during discharge.

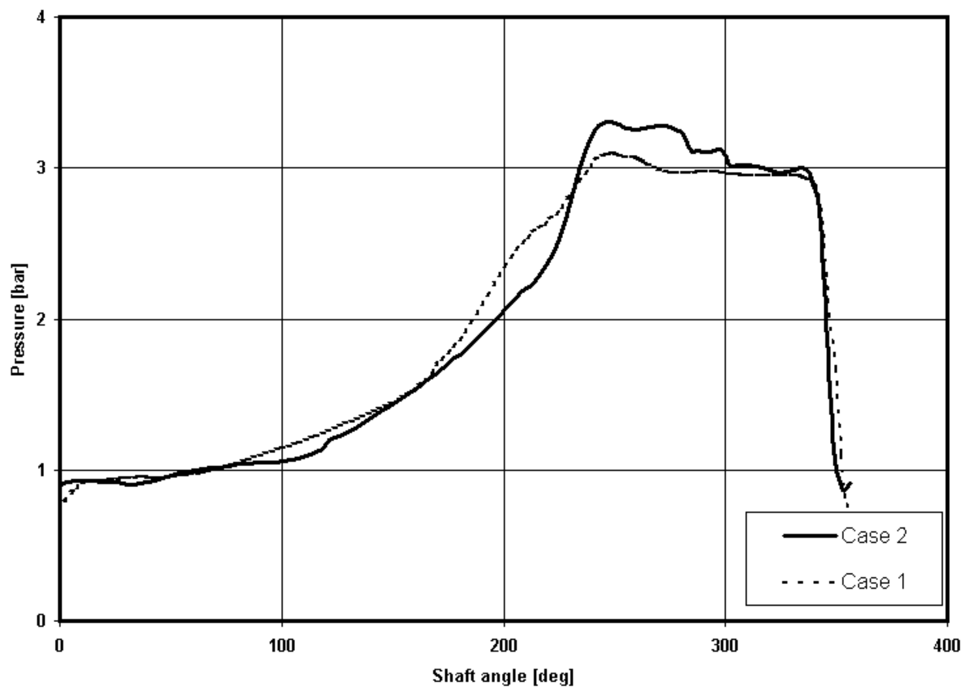

Figure 5: Comparison of p-α diagrams for two rotor profiles

The pressure difference acting on the compressor rotors deforms real rotors by an amount, which is of the same order of magnitude as that of the compressor clearances. Such deflections are presented in Figures 6 to 8. These indicate that the main displacement occurs somewhere in the middle of the female rotor which is significantly weaker than the male one. Figure 6 shows the displacements in the vertical plane for both cases. The maximum rotor displacement for this application is around two micrometers for Case 1 and approximately 30% less for Case 2.

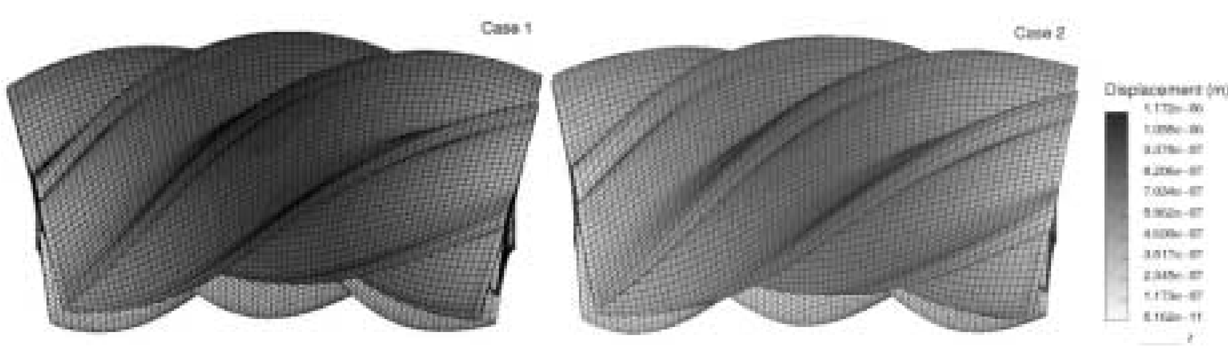

Figure 6: Rotor deformations in vertical plane due to pressure difference

Figure 7 shows a view of both pairs of rotors along the rotor axis looking from the discharge side while in Figure 8 the rotors are shown in the plane parallel to both rotor axes. The rotors in Case 1 have a greater deformation in that plane than these of Case 2.

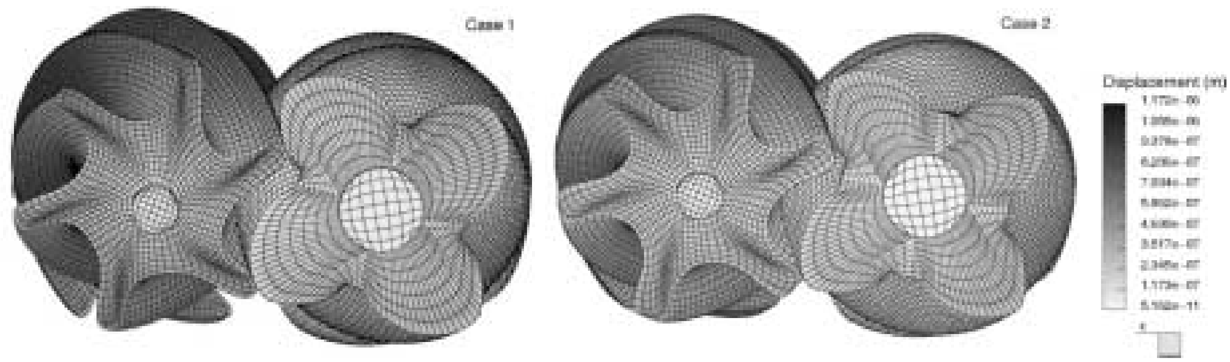

Figure 7: View from discharge side on rotors deformed by pressure difference

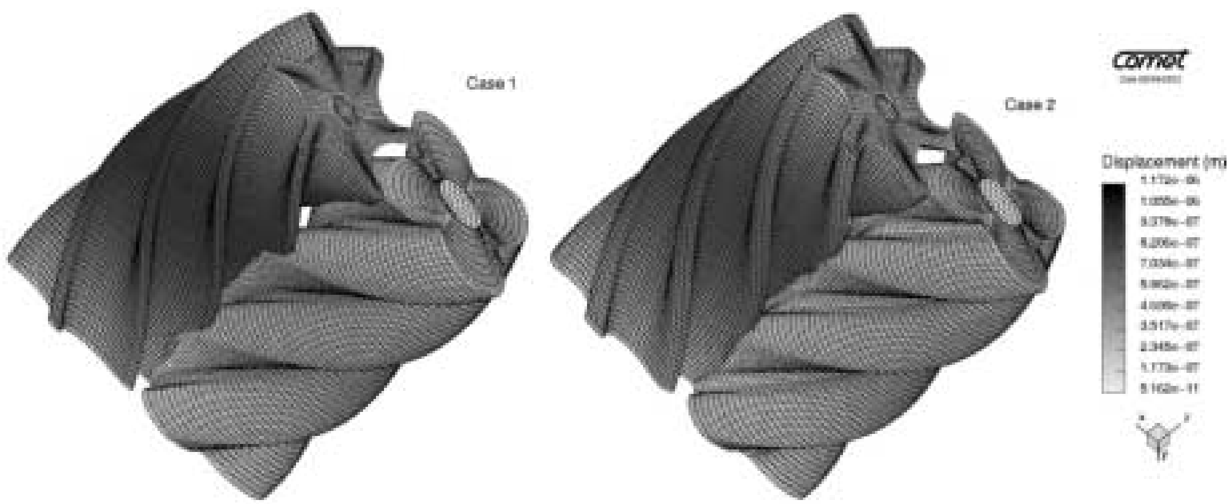

Figure 8: Horizontal plane visualization of rotor deformations caused by pressure difference

CONCLUSIONS

Screw compressor elements, especially their rotors are heavily loaded by pressure forces and temperature dilatations. These cause them to deform during normal operation. The working clearances therefore vary and usually become larger, thereby increasing internal leakage. This leads to deterioration in the compressor performance. A full 3-D calculation has been performed to quantify the interaction of the compressor structure and its fluid flow. The effects of the change in working clearances are compared for two different rotor profiles of air compressor and presented through flow and distortion diagrams and pressure-angle diagram of the working process. It can be seen from the results presented that the rotor profile affects the rotor deflection and has a significant influence on the compressor performance. This should, therefore, be taken into account in the compressor design.

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