Proceedings of 2002 ASME Congress
Symposium on the Analysis and Applications of Heat Pump and Refrigeration Systems
New Orleans, November 17-22 2002

IMECE2002-33367

NUMERICAL SIMULATION OF FLUID FLOW AND SOLID STRUCTURE IN SCREW COMPRESSORS

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Keywords: Computational continuum mechanics, Solid-Fluid Interaction, Screw Compressor, Mathematical modelling

ABSTRACT

Screw compressor performance is usually predicted by thermodynamic and fluid flow calculations based on a dimensionless flow model. However factors, which affect the power consumption, delivery and reliability of these machines, such as dynamic flow losses, leakage and oil flow drag, are not estimated accurately by such procedures. By the use of 3-D numerical modelling of the compressor fluid flow and structure deformation, these factors can be predicted more precisely and hence losses within the machine can be minimized at the design stage.

The authors have developed a CAD-CCM (Computer Aided Design to Computational Continuum Mechanics) interface program to transfer the screw compressor geometry directly into a commercial numerical solver, which simultaneously solves the compressor fluid flow and structural deformation of its parts. All necessary initial and boundary conditions, a procedure for moving of the mesh and additional functional features of the commercial solver are introduced through the user functions automatically generated by the interface program.

Results have been obtained for an oil-flooded screw compressor and some of them are presented in this paper. They are presented as overall compressor parameters in the form of power and volume flow. Additionally, pressure-time diagrams of the compression process, flow and pressure patterns in the compressor chambers and rotor deformation are illustrated.

INTRODUCTION

Screw compressors comprise a meshing pair of helical rotors on parallel axes, contained in a casing. Together, these form a succession of working chambers whose volume depends on the angle of rotation. An outline of the main elements of a screw compressor is presented in Figure 1, where it is shown how the two rotors are contained in the casing, by means of views from opposite ends and sides of the machine. The dark shaded portions show the enclosed region where the rotors are surrounded by the casing and compression takes place, while the light shaded areas show the regions of the rotors which are exposed to external pressure. The large light shaded area in Fig 1a) corresponds to the low pressure suction port. The small light shaded region between shaft ends B and D in Fig 1b) corresponds to the high pressure discharge port. Admission of the gas to be compressed occurs through the low pressure port. After a certain angle of rotation, the port is cut off and further rotation leads to a progressive reduction in the trapped volume. This causes the pressure of the contained gas to rise. The compression process continues until the rear ends of the passages between the rotors are exposed to the high pressure discharge port through which gas flows out at approximately constant pressure. The design parameter which influences screw compressor performance most strongly is the rotor profile. A difference in shape, which can hardly be detected by the eye, can effect significant changes in flow rates delivered and power consumption. Clearances between the rotors and between the rotors and the casing determine the leakage through the compressor and hence strongly influence both the volume flow rate and the power consumption.

Compressor rotors are today manufactured with very small clearances at an economic cost and, therefore, internal leakages have been reduced to a fraction of their values in earlier designs. Pressure differences between the rotor regions subjected to admission, compression and discharge cause rotor pressure loads and consequently rotor displacement and bending deformation. This increases clearances in areas where the pressure difference is the highest. Thus internal leakage is greatest in these regions. Therefore, rotor deflection becomes one of the significant parameters affecting compressor efficiency.

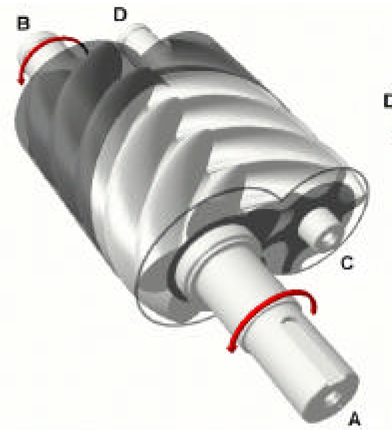
Other features of the design also strongly affect the overall compressor performance. Thus the shape and position of the suction and discharge ports influence the dynamic losses, which in turn affect efficiency. Dimensionless or quasi-steady mathematical models predict the overall effects of changes in these parameters on compressor overall behaviour fairly accurately. However, some effects cannot be taken into account by these models, especially if the influence of the local change in clearances caused by deformations induced by pressure and temperature fields in fluid is considered. Consequently, the simplified analytical mo dels currently in use are not sufficiently accurate to design screw compressors to obtain the maximum possible improvements.

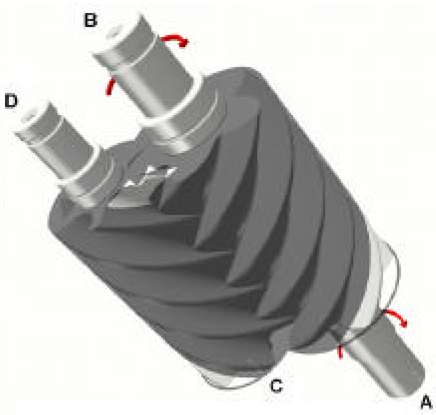
A full 3-D numerical calculation of the compressor fluid flow and solid structure is therefore required to determine the maximum reduction in clearances possible to improve screw compressor performance without contact between the rotors and their casing. By this means, rotors may be manufactured with minimum working clearances and screw compressors may be made smaller as well as more efficient.

A 3-D Computational Continuum Mechanics (CCM) method can simultaneously combine fluid flow and structure behaviour to determine the effects of changes in the compressor geometry on internal heat and fluid flow and vice versa. Such an approach can produce reliable predictions only if calculated over a substantial number of grid points. Hence, a high computer potential and capacity is needed in order to use such procedures to analyse a screw compressor.

Apart from the authors' publications [Kovacevic et. al 1999, 2000 and 2001], there is hardly any reported activity in the use of CFD 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.

Demirdzic and Peric successfully applied finite volume calculations of 3-D flows to complex curvilinear geometries. Based on that, Ferziger and Peric [1996] published a book on finite volume methods for fluid dynamics. 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.





a) Top Front View **b) Bottom Rear View** **c) Photograph of analysed rotors**

Figure 1 Twin Screw Compressor Rotors and Casing Outline

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], 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.

A pre-processing interface has been developed by the authors in order to generate a 3D 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.

DISCRETISATION OF SCREW COMPRESSOR GEOMETRY

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.

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. More details of the different grid types and the relative advantages of each grid system are given by Thompson et al [1999]. Block structured grids allow easier grid generation for complex geometries. Two basic topology types are used here for the screw compressor grid generation, namely polyhedral blocks a) and O grids b) as presented in Figure 2.

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. These are determined by the rotor profile coordinates and their derivatives and are obtained by means of the rack generation procedure described in detail by [Stosic 1998].

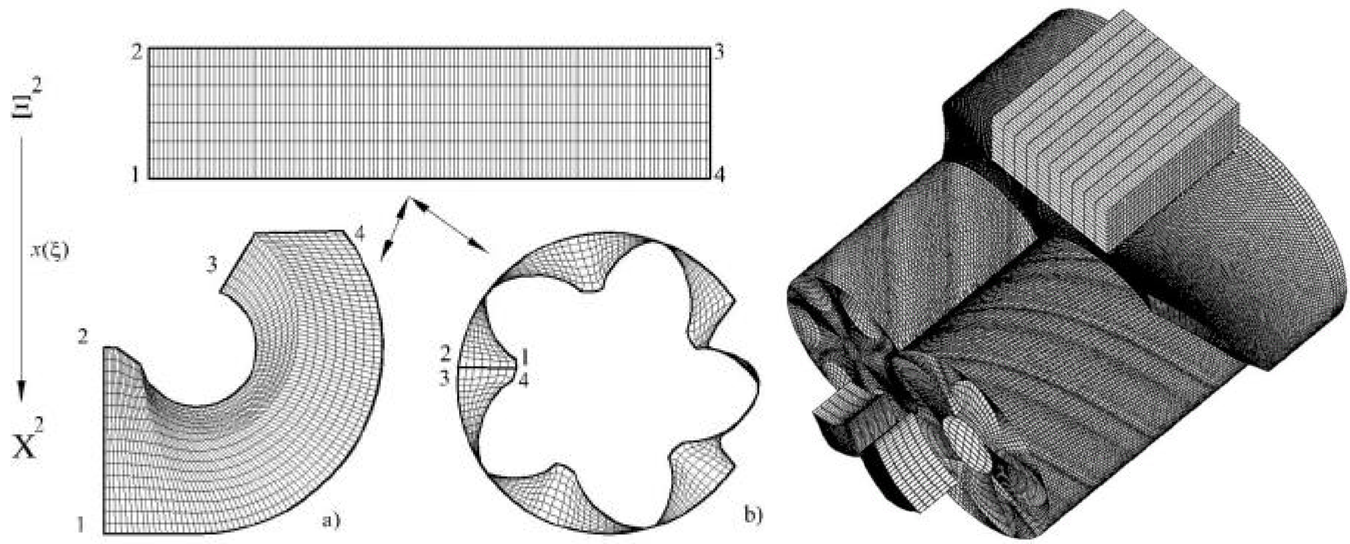


Figure 2 Left - Patterns of grid topology in a screw compressor
Right – Numerical mesh of the screw compressor domains

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. More information about analytical grid generation methods can be found in Liseikin [1999].

A detailed description of the procedure for discretisation of screw compressor domain is explained by Kovacevic et al [2000] and [2001].

Numerical mesh for the test case in this study comprises of 513,617 numerical cells of which 162,283 cells represent a solid part of both rotors, another 189,144 cells are mapped on the fluid parts of the rotors while the rest are numerical cells of suction and discharge domains which include both, the ports and openings.

NUMERICAL SOLUTION OF FLUID FLOW AND SOLID BODY DEFORMATION

The density of the compressor working fluid varies with both pressure and temperature. The compressor flow and structure is fully described by the mass averaged equations of continuity, momentum, energy and space conservation, which are accompanied by the turbulence model equations and an equation of state, as for example, given by Ferzinger and Peric [1995]. In the case of multiphase flow, the concentration equation is added to the system. The solution of such 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.

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.

The resulting system of partial differential equations is discretised by means of a finite volume method in the general Cartesian coordinate system. This method enhances conservation of governing equations while at the same time enables coupled system of equations for both solid and fluid parts to be solved simultaneously. Connection between the solid and fluid parts is in this manner 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.

PRESENTATION OF THE CALCULATED RESULTS

Results obtained by use of the interface were applied to a commercial CMM solver, Comet, and given in this paper for an oil-flooded air compressor. A personal computer, with Athlon 800 MHz processor and 1 GB RAM, has been used for calculation. Compressor movement was simulated through 24 time steps for one interlobe rotation giving the overall number of 120 time steps for the full rotation of the male rotor. Time step length was synchronised with compressor speed of 5000 rpm. Error reduction for 4 orders of magnitude required approximately 50 outer iterations for each time step, which took approximately 30 minutes of the computer time. The overall compressor parameters such as the torque, volume flow, forces, efficiencies and compressor specific power were calculated. Additionally, pressure-time diagrams of the compression process, the flow and pressure 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 Figures 3 and 4. Figure 3 shows two cross sections perpendicular to rotor axes, one through the suction and oil injection ports and another one close to discharge. In Figure 4, a remarkable clearance gap flow can be noticed in the cross section between rotors. Figure 5 presents the pressure field and oil concentration in the same cross section as in Figure 3, while Figure 6 gives distribution of pressure in an axial cross section of the compressor working chamber.

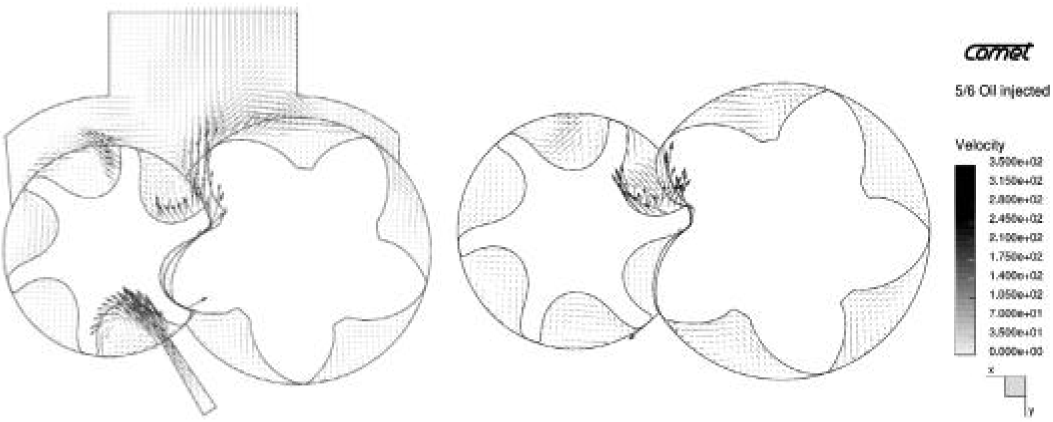


Figure 3 Velocity vectors in the two compressor cross sections

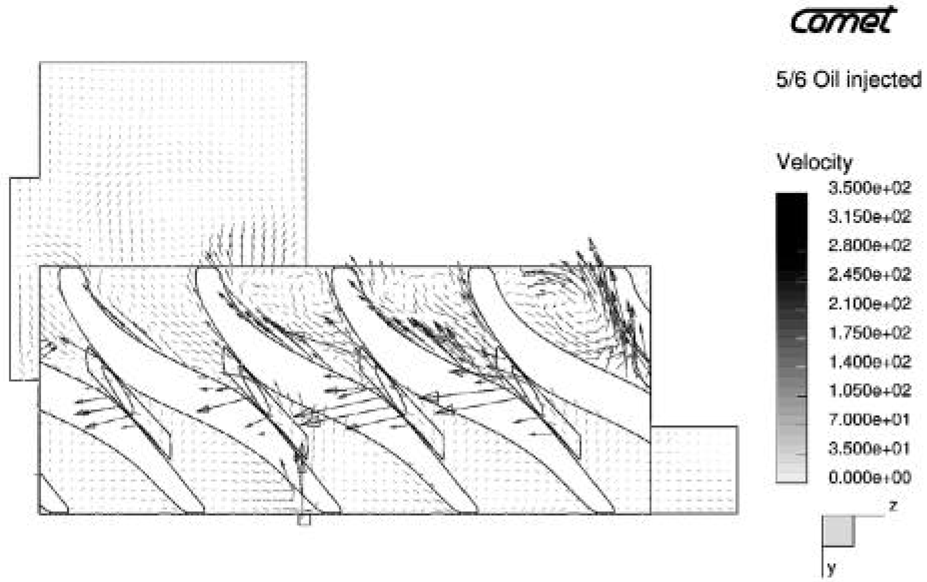


Figure 4 Velocity vectors in the compressor axial section

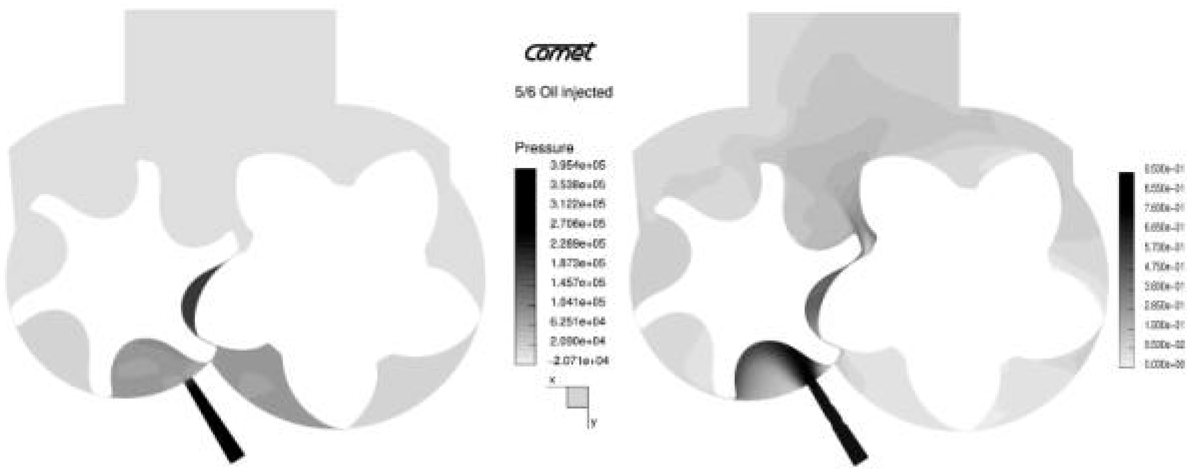


Figure 5 Pressure (left) and oil distribution (right) in the cross section through the oil injection port

Figure 6 Pressure field in the axial cross section between rotors

The calculated pressure history in one of the rotor interlobes is presented in Figure 7 as a function of the shaft rotation angle and compared with its measured values. Good agreement is maintained not only for the compression process, but also for the pressure fluctuation during discharge. Pressure fluctuations in the experimental compressor are measured by "ENDEVCO" piezoelectric transducers conveniently positioned in the housing at the male rotor side to cover as much as possible of a compressor process, as presented in figure 7.

Calculated compressor flow and power are presented in Figure 8 and compared with the experimental data. The obtained agreement between them is very reasonable.

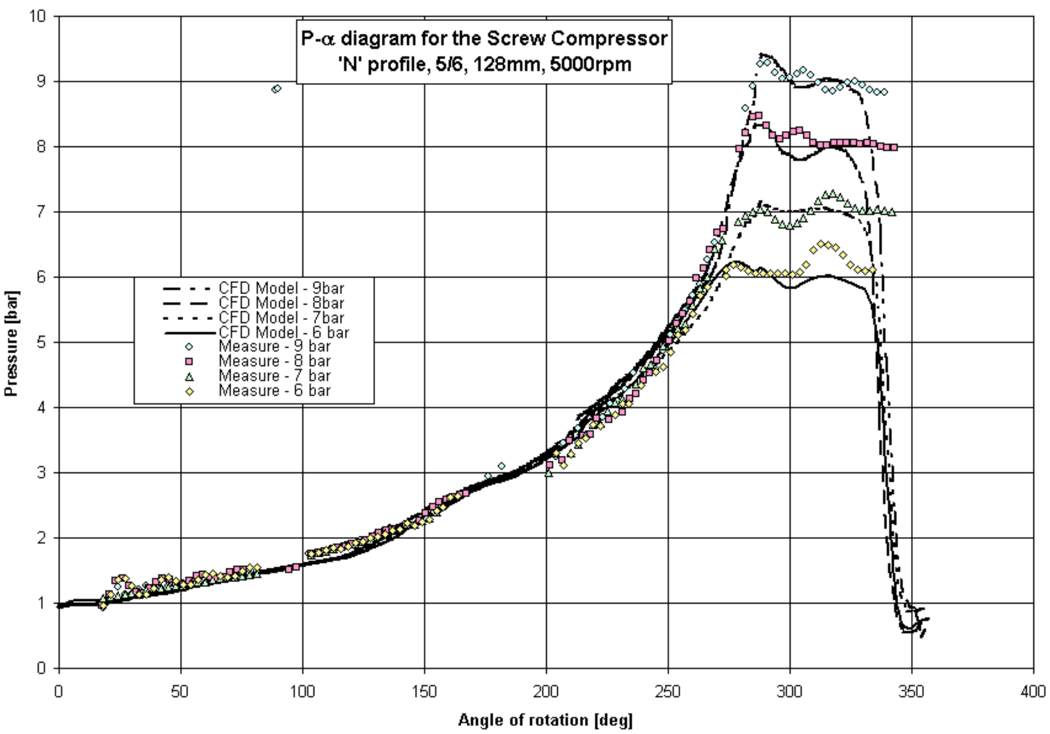


Figure 7 Pressure-rotation angle diagrams for various discharge pressures

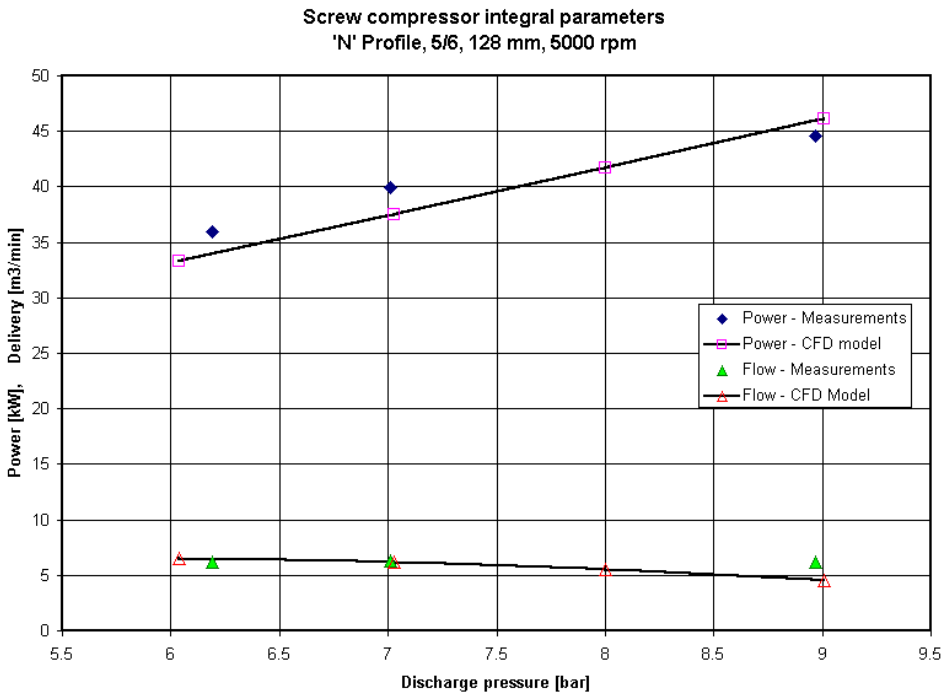


Figure 8 Comparison of integral results obtained by measurements and modelling

The pressure difference acting on the compressor rotors deforms real rotors by an amount which is similar in size to the compressor clearances. Such deflections are presented in Figures 9 and 10 indicating that the main displacement occurs somewhere in the middle of the gate rotor which is significantly weaker than the main one. Consequently, additional clearance caused by the rotor bending increases the compressor interlobe and end face clearances resulting in higher leakages and thereby reduced compressor efficiency. Calculations showed small deformations caused by the pressure and temperature gradients within the compressor, not larger then few micrometers which are an order of magnitude smaller then the size of the smallest cell face and still substantially smaller then the mesh deformations due to rotor movement. In order to make the enlarged clearances visible, the rotor displacements in Figures 9 and 10 are magnified 10000 times.

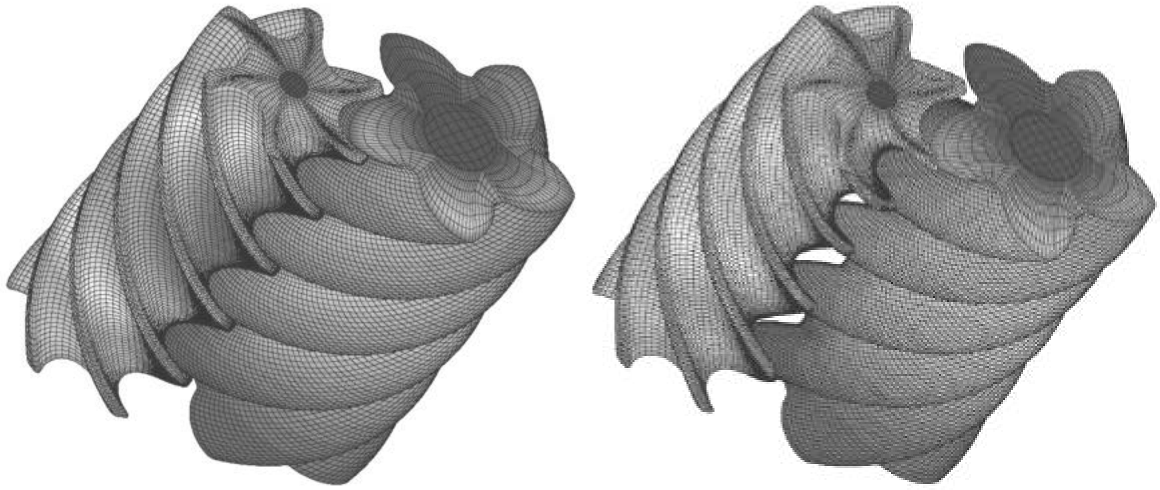


Figure 9 Rotor deformation in the plane of the screw compressor axes

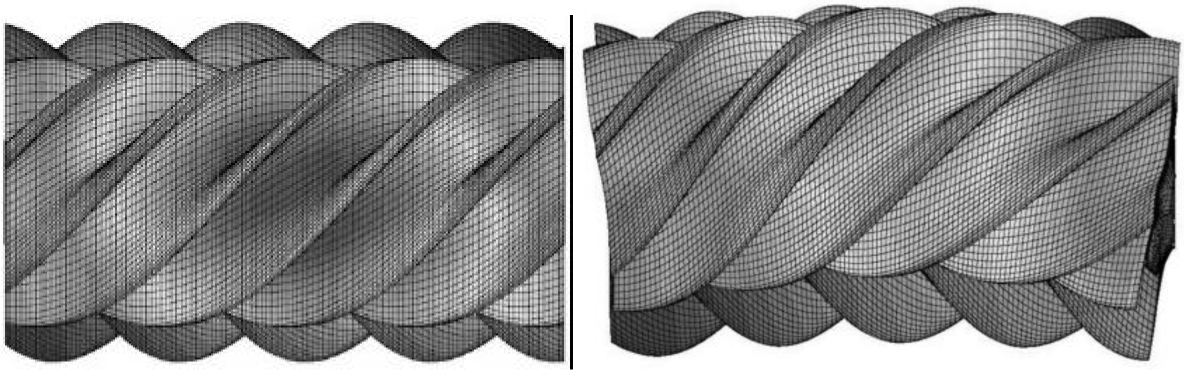


Figure 10 Rotors in their initial and deformed states in the vertical plane

CONCLUSIONS

Simultaneous three dimensional calculation of screw compressor fluid flow and structural deformation has been performed to obtain reliable data for compressor design. The effects of the change in working clearances are presented through the overall compressor parameters such as power and volume flow and additionally through pressure time diagrams.

Flow patterns in the compressor chambers and rotor deformation have been included which show more details of the results obtained. It can be seen from them that the rotor deflection caused by working pressure and temperature change has a significant influence on the compressor efficiency. This should, therefore, be taken into account in compressor design.

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