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THREE-DIMENSIONAL MODELLING OF SOLID-FLUID INTERACTION AS A DESIGN TOOL IN SCREW COMPRESSORS

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1. Introduction

Screw compressors are rotary positive displacement machines of simple design capable of high speed operation over a wide range of operating pressures and flow rates with high efficiencies. They are reliable and compact and consequently they comprise the majority of all positive displacement compressors now sold and of those currently in operation. Compressor rotors are today manufactured with very small clearances at an economic cost and, therefore, internal leakages have been recently reduced to a fraction of their values in earlier designs. However, since screw compressor rotors are heavily loaded by the pressure differences across them, their deformation is of the same order of magnitude as the clearances between them and the casing. Thus distortion may significantly influence the compressor performance.

A full 3-D numerical calculation of the interaction between the solid structure of a screw compressor and fluid flow through it 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.

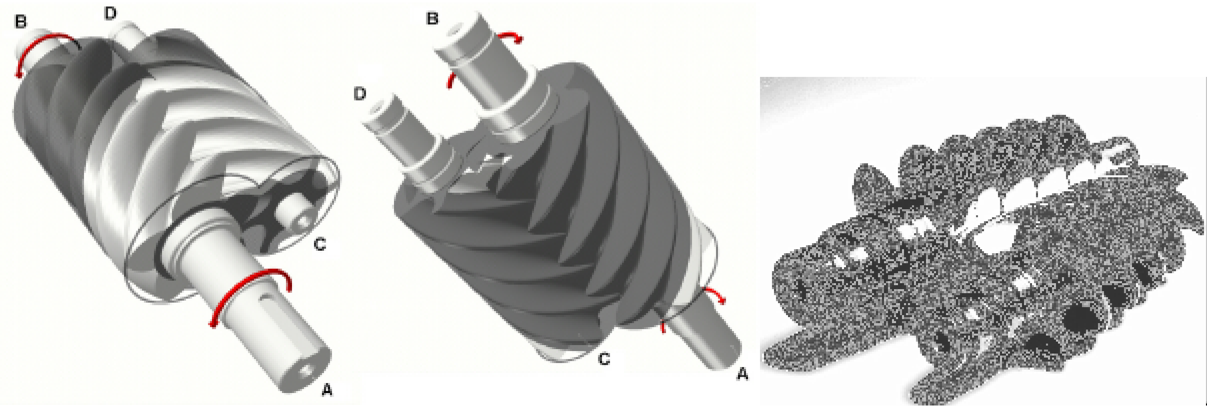
A pre-processing interface has been developed by the authors in order to generate a 3-D numerical grid for the purpose of simultaneous calculation of the compressor structure and its fluid flow. The interface employs a rack generation procedure to produce rotor profile points and analytical transfinite 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.

Results obtained by use of the interface are applied to a commercial CMM solver Comet and given in this paper for an oil-flooded air screw compressor. The effects of the change in working clearances are shown through their effect on overall compressor parameters such as the torque, volume flow, forces, efficiencies and compressor specific power. Additionally, pressure time diagrams of the compression process, the flow and temperature patterns in the compressor chambers and rotor deformations are provided which gives more detailed insight into the results obtained. All calculated results could then be employed to improve the design of screw machines.

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 1b) corresponds to the suction low pressure port. The small light shaded region between shaft ends B and D in Fig 1a) corresponds to the discharge high pressure 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.

Pressure differences between rotor regions subjected to admission, compression and discharge cause rotor pressure loads and consequently rotor displacement and rotor bending deformation. This results in increased compressor clearances which are most pronounced in areas where the pressure difference is the highest. The highest pressure differences in the regions with the largest clearances, maximises the internal leakages. Therefore, rotor deflection becomes one of the most significant parameters affecting compressor efficiency.



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

Figure 1. Twin Screw Compressor Rotors and Casing Outline

However, other features of the design also strongly affect the overall compressor performance. 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 inducted by pressure and temperature fields in fluid is considered. Consequently, the simplified analytical models currently in use are not sufficiently accurate to design screw compressors to obtain the maximum possible improvements from the close manufacturing tolerances now achievable with contemporary numerically controlled machine tools.

It is therefore timely to apply a more complex analytical procedure, such as a 3-D Computational Continuum Mechanics (CCM) method, which simultaneously combines 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. all 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 (FSI) have contributed to the lack of publications in this area.

Demirdzic and Peric set the guidelines for successful finite volume calculation of 3-D flows in complex curvilinear geometries, Based on which, 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. 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. all 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.

2. 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]. 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].

3. Numerical solution of fluid-solid interaction

The density of the compressor working fluid varies with both pressure and temperature. Therefore compressor flow and structure is fully described by the mass averaged equations of continuity, momentum, energy and space conservation of compressible fluid, which are accompanied by the turbulence model equations and an equation of state, as for example, given by Ferziger 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 mad 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 equation of a thermo-elastic solid body.

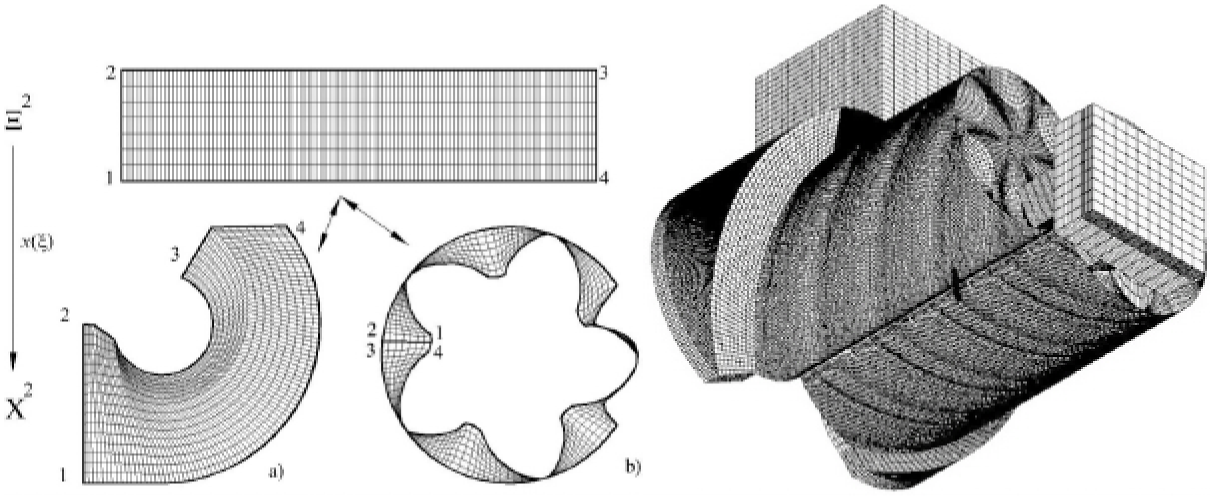


Figure 2. Left - Patterns of grid topology in a screw compressor
Right – Numerical mesh of the screw compressor domains

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 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 the zero displacement in the restraints and zero tractions elsewhere.

The resulting system of partial differential equations is discretised by means of 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.

4. Presentation of the calculated results

The calculated pressure history in one of the rotor interlobes is presented in Figure 3 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 the compressor discharge.

The pressure difference acting on the compressor rotors deforms real rotors to an extent which is of the same order of magnitude as the compressor clearances. Such deflections are presented in Figure 4 indicating that the main displacement occurred 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 overall clearances resulting in higher leakages and thereby reducing the compressor efficiency.

The compressor power is calculated by integrating pressure with respect to compressor volume. Specific power, which is the compressor power per unit flow, as well as compressor efficiencies are a final measure of the influence of reduction in operational clearances upon the compressor process. Their values are compared for two different discharge pressures, 6 and 7 bar and for three different rotor deflections quantified by a relative increase of 50%, 65% and 100% in the compressor clearance:

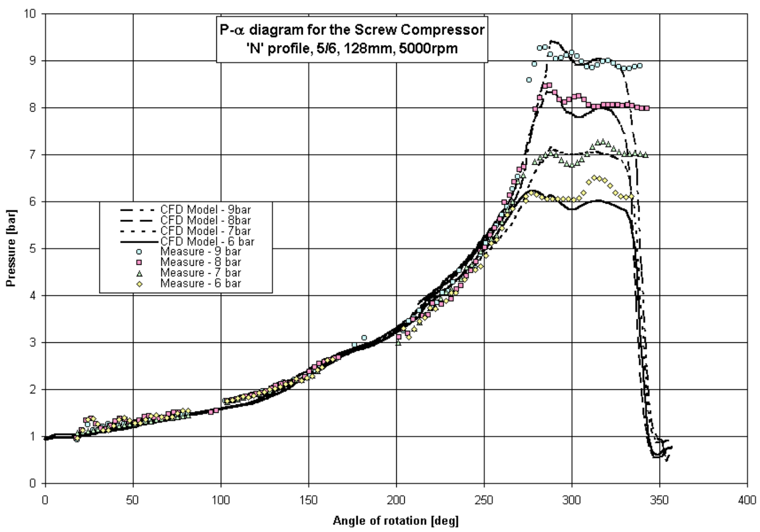


Figure 3. Pressure-rotation angle diagrams for various discharge pressures

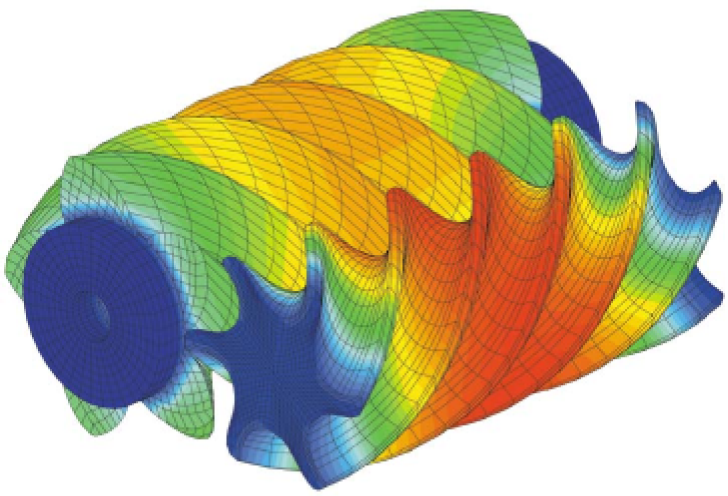


Figure 4. Rotor deformation

Table 1. Influence of rotor deflection upon compressor performance

Discharge pressure 6 bar

Clearance %	P kW	Flow m3/min	Psp KW/m3/min	ηv %
50	34.057	7.46	4.565	99.2
65	33.323	6.523	5.108	86.5
100	32.827	5.384	8.467	71.4

Discharge pressure 7 bar

Clearance %	P kW	Flow m3/min	Psp kW/m3/min	ηv %
50	37.72	7.458	5.058	99.2
65	37.49	6.176	6.07	81.9
100	36.9	4.842	7.623	63.9

5. Conclusion

Screw compressor elements, especially their rotors are heavily loaded by pressure forces and they deform during normal compressor operation. Working clearances therefore become larger, thereby increasing internal leakages. This leads to a deterioration in the compressor performance. A full 3-D calculation is performed to quantify the interaction of the compressor structure and its fluid flow. The effects of the change in working clearances are presented through the overall compressor parameters such as are the torque, volume flow, forces, efficiencies and compressor specific power. Additionally, pressure time diagrams, the flow and temperature patterns in the compressor chambers and rotor deformations are provided which show more details of the results obtained. It can be seen from the results presented that there rotor deflection has a significant influence on the compressor efficiency. This influence should, therefore, be taken into account in the compressor design. The results thus obtained could be employed to improve the design of screw machines.

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