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CFD STUDIES OF FLOW IN SCREW AND SCROLL COMPRESSORS

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ABSTRACT

An analysis of flow in rotary positive displacement machines has been carried out using a standard CFD package by separating out the motions of the fluid and the instantaneous volume in which it is contained. Examples are given of both twin screw and scroll compressors. Pressure-volume diagrams obtained from such analyses are compared with experimental results and good agreement shown. Longer term aims of such studies are to obtain a better understanding of the internal flow processes, especially where two component flow, such as oil/air mixtures are involved.

INTRODUCTION

This paper describes the first stages of a longer term project to obtain greater understanding of the nature of fluid flow within rotary positive displacement machines such as those of the screw and scroll type. The greatest need for such improved understanding is where there is more than one flow component as in oil injected compressors. In such cases, the assumptions normally made on oil and gas separation within the machines may thus be checked.

Flow of fluid in a rotary positive displacement compressor is unsteady, three-dimensional, turbulent and compressible. Its analysis may be simplified if a grid is defined to describe the instantaneous volume of fluid trapped in the machine and the motion is described first in terms of the fluid relative to the grid and then of the grid itself. By this means fluid motion through all compressors of this type may be analysed with standard numerical procedures.

The flow of the fluid relative to the grid may be calculated using a numerical procedure, but with the grid motion replaced by external body forces on the fluid. Although the grid is continuously in motion and changing in shape, its geometry may be defined at all times and its motion described by simultaneous translation and rotation. Thus the grid acceleration is calculated for each time step and included in the momentum equation terms while all other terms in the equation would be unchanged.

The result is the same as would be obtained if a general moving grid was built into the numerical procedure, but such a procedure would be more complex, and would require the use of more advanced CFD solvers than those currently in use.

Available CFD packages usually contain a preprocessor, which prepares a grid for the solver to integrate equations and a postprocessor which conveniently present results calculated. Despite their sofistication, commercial preprocessors can barely cope with any unusual shapes and, despite its simplicity, this appears to include screw and scroll compressor geometry. Our attempts to use commercial preprocessors to generate proper grids for screw and scroll compressor working chambers flow calculations failed. Finally we reprogrammed part of a general grid generator and applied it succesfully to produce grids to perform the calculations presented in this paper.

BASIC EQUATIONS

By use of mass averaged Navier-Stokes equations, the continuity equation may be written as:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho U_i)}{\partial x_i} = 0$$

where ρ and U_i are density and velocity, while the momentum equation may be expressed as:

$$\rho \frac{\partial U_i}{\partial t} + \rho U_j \frac{\partial U_i}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu S_{ij} - \rho \overline{u_i u_j} + \rho a_i\right]$$

$$S_{ij} = \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} - \frac{2}{3}\frac{\partial U_k}{\partial x_k}\delta_{ij}$$

where S_{ij} denotes for the strain, p the pressure, $\rho\overline{u_i u_j}$ the Reynolds stress, $k = \frac{1}{2}\rho\overline{u_i u_i}$ is the turbulent kinetic energy and μ is a fluid dynamic viscosity.

The total velocity is the sum of the transfer (grid point) velocity $\partial x_i/\partial t$ and the velocity relative to the grid U_i. Transfer acceleration is given as $a_i = \partial^2 x_i/\partial t^2$. Forces caused by the grid motion are implemented into the source terms in the same maner as other external body forces, as for example, gravity force.

In the absence of heat sources the transport equation for internal energy E is given as:

$$\rho \frac{\partial E}{\partial t} + \rho U_j \frac{\partial E}{\partial x_j} = p \frac{\partial U_i}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\frac{\mu}{Pr}\frac{\partial E}{\partial x_i} - \rho\overline{e u_j}\right] - \mu S_{ij}\frac{\partial U_i}{\partial x_j}$$

where Pr is Prandtl number.

The turbulence model implemented is a plane $k-\varepsilon$ model with no additional terms.

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho U_j k)}{\partial x_j}=P_k-\rho\varepsilon+\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]$$

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial(\rho U_j\varepsilon)}{\partial x_j}=C_{\varepsilon 1}P_k\frac{\varepsilon}{k}-\rho C_{\varepsilon 2}\frac{\varepsilon^2}{k}+\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]$$

where ε is the turbulent energy dissipation, $P_k = 2\mu_t S_{ij}S_{ij}$ for turbulence energy production, $\mu_t = \rho C_\mu k^2/\varepsilon$ is a turbulent viscosity and σ is a turbulent Prandtl number. Reynolds stress and turbulent heat flux are given as:

$$\rho\overline{u_i u_j}=-\mu_t S_{ij} \qquad \rho\overline{e u_j}=\frac{\mu_t}{Pr}\frac{\partial E}{\partial x_i}$$

Constants of $k-\varepsilon$ turbulence model are: $C_\mu = 0.09, \sigma_k = 1, \sigma_\varepsilon = 1.3, C_{\varepsilon 1} = 1.44, C_{\varepsilon 2} = 1.92$

Standard wall functions are implemented on all walls, and a standard constant pressure boundary condition was used at all flow faces.

NUMERICAL SOLUTION PROCEDURE

The standard 3d finite volume colocated grid numerical procedure "Fastest" [1], [3] was employed to obtain numerical solutions. The force terms arising from the grid motion were added to the momentum equation source terms. This part of the numerical procedure was most conveniently introduced through user defined functions. The SIMPLE pressure procedure was adapted to account for fluid compressibility by means of an equaton of state with a pressure correction correction transport equation of a mixed convection-diffusion type [2].

CALCULATION OF SCREW COMPRESSOR FLOW

Fig 1 presents a screw rotor mesh generated by the adopted grid preprocessor. Fig 2 shows how the male and female rotor passages were created by the same preprocessor. One block, containing 2420 control volumes, was constructed for each passage and the multigrid procedure was used to obtain a final solution. Initially, two grids of control volumes were tried and the second grid used contained 38720 control volumes.

The full multigrid procedure required less than one minute per unit time step, using a P-166MHz workstation, and a full cycle calculation was performed in less than six hours. The results obtained were ploted by the "Fastest" postprocessor in the form corresponding to experimental results which were used to validate numerical calculations. Absolute velocity vectors and velocity vectors relative to the grid are presented Fig 3.

The screw compressor flow thus calculated has been partially validated experimentally and comparative results in the form of a pressure-volume plot are shown in Fig 4.

CALCULATION OF SCROLL COMPRESSOR FLOW

An outline of a scroll compressor configuration is shown in Fig 5 together with the grid used to describe it. The grid is comprised of two blocks, namely: the discharge passage with 384 control volumes and the interspiral chmaber with 600 control volumes. In Fig 6 the interspirale block is presented in the third grid comprising 38400 control volumes. Computer performances were very similar to those obtained for the screw compressor.

Velocity vectors in the grid coordinate system and the absolute velocity values for the grid in motion are given in Fig 7. Finaly, a pressure history for the scroll compressor is given in Fig 8.

CONCLUSIONS

As may be seen, the agreement between the only quantity measured, the pressure history in the screw compressor chamber and its estimated counterpart was reasonably good. It is hoped that further development of this technique will enable the flow characteristics within any type of rotary positive displacement machine to be modelled accurately and thereby lead to new insights into flow within them. This should be a useful tool in compressor design and, hopefully, lead to designs with improved performance. Also a more detailed measurments of velocity field would be appreciated for a thorough comparison with calculated data.

REFERENCES

[1] Schäfer M, Stošić N, 1994: Parallel multigrid calculations of turbulent flows in complex geometries *Proc. World CFD User Conference Basle, Switzerland*

[2] Stošić N, 1982: Numerical modelling of internal flows of compressible fluid, *PhD Thesis, University of Sarajevo*

[3] — 1995: Fastest: Parallel Multigrid Solver for Flows in Complex Geometries: User Manual *Lehrstuhl für Strömungsmechanik, Universität Erlangen-Nürnberg, Germany*

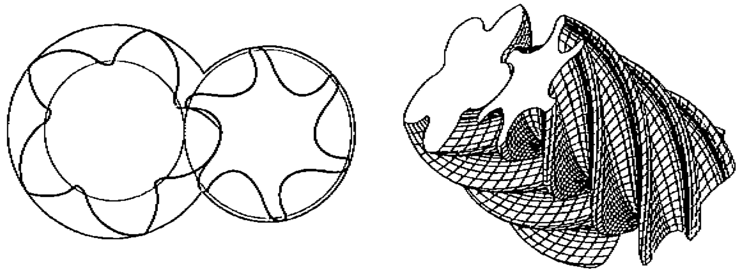

Fig 1 Screw rotor mesh

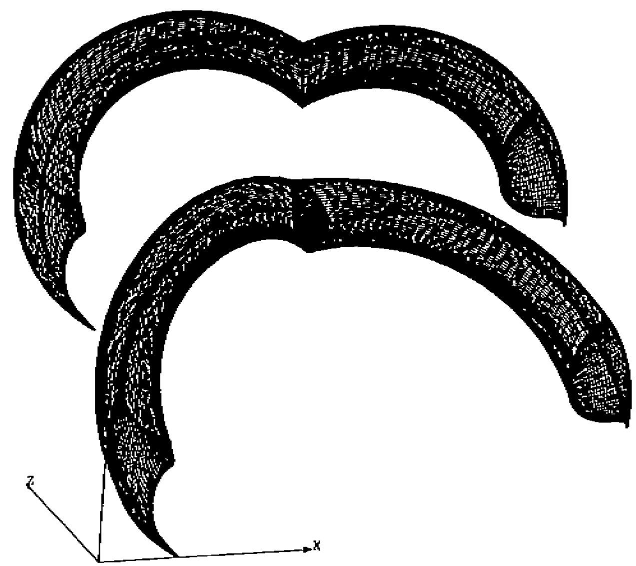

Fig 2 Grid of a screw compressor working chamber, with the main and gate rotors

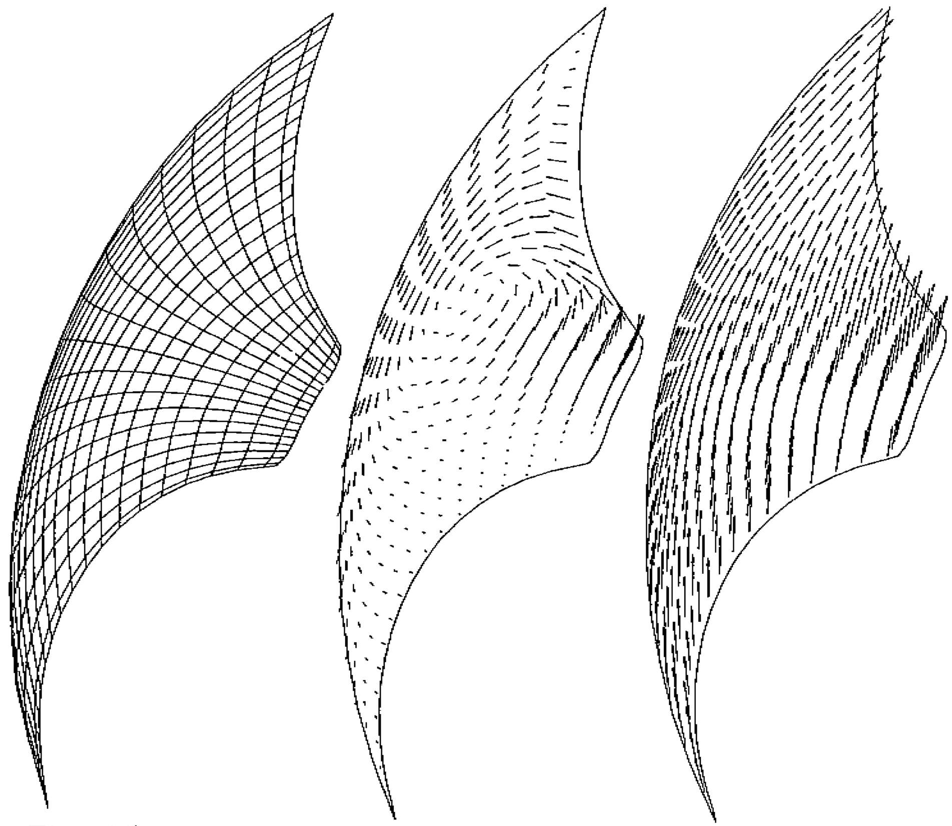

Fig 3 The intersection of the main rotor interlobe with calculated velocity vectors in the relative (grid) coordinate system and in the absolute coordinate system

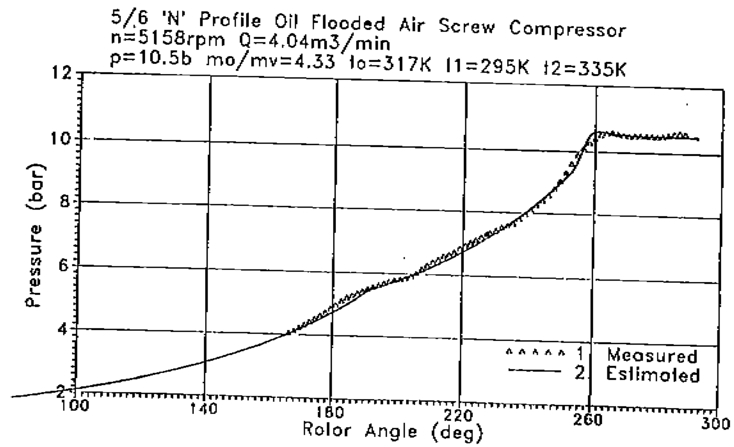

Fig 4 Measured and estimated pressure change within the screw compressor working volume

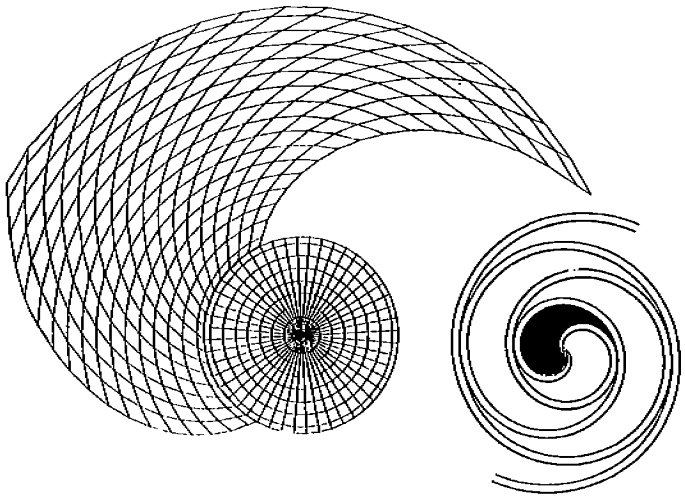

Fig 5 Outline of a scroll compressor profile with views of grid blocks of a scroll compressor

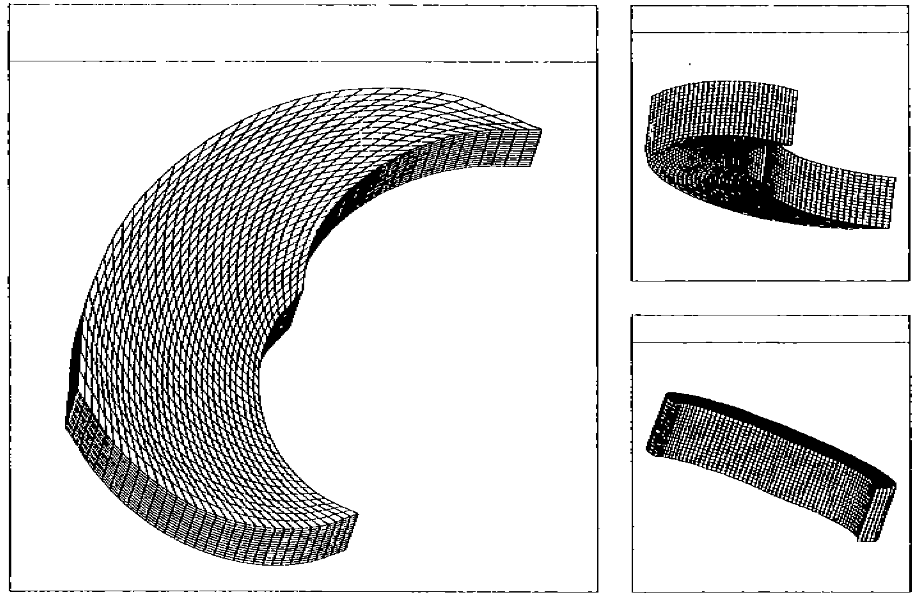

Fig 6 Axonometric view of a scroll compressor working chamber

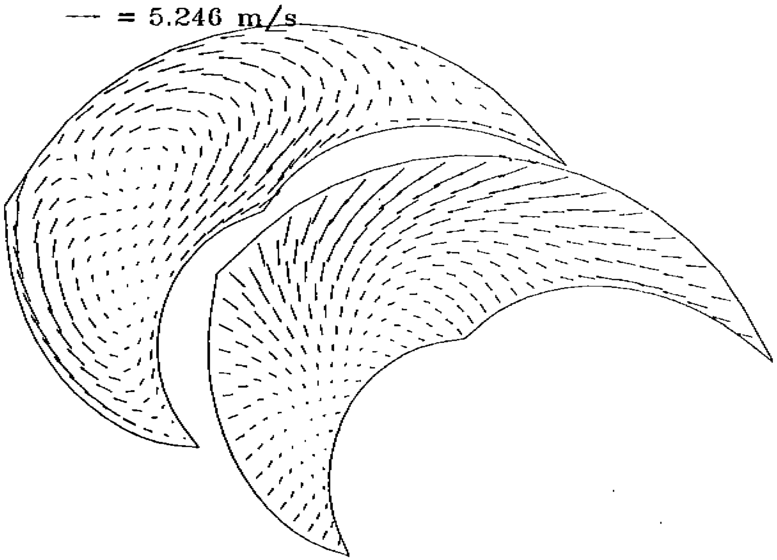

Fig 7 Calculated velocity vectors in relative (grid) coordinate system and for the stationary (absolute) coordinate system

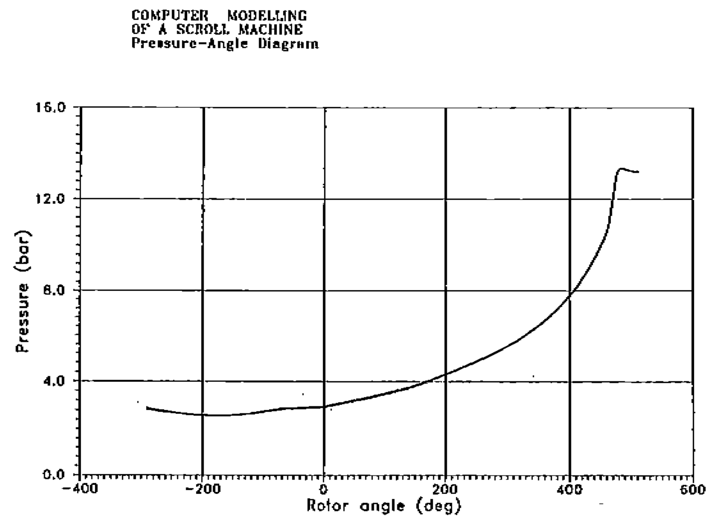

Fig 8 Calculated pressure changes in a scroll compressor