An integrated model for the performance calculation of Screw Machines

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ABSTRACT

There is a need to develop improved analytical procedures in order to improve performance, reduce noise emission and reduce the manufacturing costs of screw compressors. Most mathematical models, used by industry for screw compressor performance estimation and optimisation, are based on quasi one dimensional calculation of the governing flow equations in a control volume. Despite being fast and accurate for general compressor performance calculations, these do not take full account of flow losses induced in the compressor inlet and outlet ports although the effects of these are significant. Three dimensional fluid flow calculations take account of these phenomena but require significant time and effort to be performed due to the complex geometry handling procedures and large numerical meshes required. An alternative approach is to combine both methods in a single integrated procedure.

This paper describes how a three dimensional Computational Fluid Dynamics model of flow in the compressor ports has been integrated with a one dimensional mathematical model of flow in the working chamber through a common integral management system. By this means, solutions may be obtained faster than with a full 3D approach with results that are more accurate than from a 1D model. The transmission of the boundary conditions from one region to the other has been established through the user coding of a CFD system. The methods described are of considerable scope and can be applied, not only to screw compressors but also to any other type of twin rotor rotational machine with parallel axes, such as gear pumps, vacuum pumps and roots blowers. A comparison of measured and predicted pressure oscillations in the discharge port of an industrial oil injected compressor is given as an example of the use of this procedure.

NOMENCLATURE

A	- area	V	- volume
m	- Mass in the chamber	v	- *velocity*
$\dot{m}$	- mass flow	P	- pressure
U	- Internal energy	T	- temperature

ϕ	- transported property	t	- time
ρ	- density		

Subscripts

Cc	- compression chamber	CcD	- compression to discharge
Sc	- suction chamber	in	- compressor inlet
Dc	- discharge chamber	out	- compressor outlet
SCc	- suction to compression		

1 INTRODUCTION

Early designs of screw compressors were based on the assumption of an ideal gas in a leak proof working chamber undergoing a compression process which could reasonably be approximated in terms of pressure-volume changes by the choice of a suitable value of the exponent "n" in the relationship pV^n = Constant. The advent of digital computing made it possible to model the compression process more accurately and, with the passage of time, ever more detailed models of the internal flow processes have been developed, based on the assumption of one-dimensional non-steady bulk fluid flow and steady one dimensional leakage flow through the working chamber. Together with suitable flow coefficients through the passages, and an equation of state for the working fluid, it was thus possible to develop a set of non-linear differential equations which describe the instantaneous rates of heat and fluid flow and work across the boundaries of the compressor system. These equations can be solved numerically to estimate pressure-volume changes through the suction, compression and delivery stages and hence determine the net torque, power input and fluid flow, together with the isentropic and volumetric efficiencies in a compressor. In addition, the effects of oil injection on performance can be assessed by assuming that any oil passes through the machine as a uniformly distributed spray with an assumed mean droplet diameter. Such models have been refined by comparing performance predictions, derived from them, with experimentally derived data. A typical result of such modelling is the suite of computer programs described by *Stosic et al, 2005*. Similar work was also carried out by many other authors such as *Fleming and Tang 1998* and *Sauls, 1998.* Despite their speed and relatively accurate results, these models neglect some important flow effects that influence compressor performance, mainly in the suction and discharge ports.

Screw compressor performance can be estimated more precisely by the use of a three dimensional Computational Fluid Dynamics (CFD). Nonetheless there are few publications available that describe its successful application in this field. *Kovacevic et al* published a number of papers between 1999 and 2005 which described 3D numerical analysis of the entire machine domain. These were followed by a monograph on CFD applied to screw machines by *Kovacevic et al, 2006,* which gives a comprehensive overview of the methods and tools used for the analysis of flow in these machines.

A number of commercial CFD software packages are currently available which can both analyse the flow through screw machines and easily integrate the results with CAD systems. However the moving, stretching and sliding mesh required to map the working chamber cannot be produced within their grid generator packages. Additionally, the time required for calculation of the flow through the entire machine by use of these codes is excessively long.

A prerequisite for success in the highly competitive market of screw machines is the ability to design, analyse and produce machines quickly. These activities need to be automated for use by design engineers in industry. A management suite like that elaborated in *Kovacevic et al, 2006* was developed to integrate tools for the design and manufacture of screw machine components in a user friendly environment suitable for industrial use. It manages both geometric and non geometric information transfer between several software components and has been given the name DISCO (Design Integration for Screw Compressors). There is still the need to reduce computational time and increase the accuracy of the results when calculating flows in compressor ports and specific procedures for such calculations need to be developed.

2 INTEGRATION OF 1D AND 3D MODELS

2.1 1D MODEL OF THE COMPRESSION CHAMBER

Any compressor can be generally represented by three flow domains which complete a working cycle of the machine. As shown in Figure 1 the compression chamber in which the compression process occurs is connected to the suction and discharge chambers through which the compressor communicates with the environment.

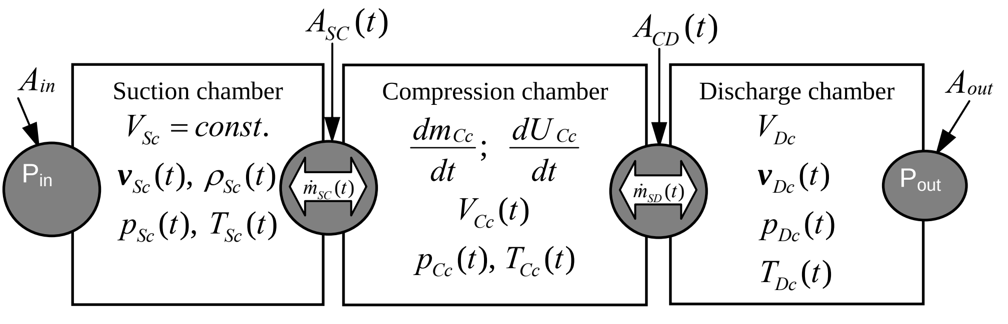


Figure 1 Schematic representation of a compressor working process

In positive displacement machines such as screw compressors, where internal energy changes are significantly larger than kinetic energy effects, the compression process can be quite accurately described by a set of differential equations for internal energy and mass conservation, such as those of *Stosic et al, 2007*. These must be closed with equations that define leakage flows, liquid injection, heat exchange as well as the mass and energy contributions from the inlet and outlet flows in order to be able to solve them. An equation of state must also be included to establish the relationship of the working fluid thermodynamic properties such as pressure, temperature and volume.

The inlet and outlet flows are the means by which a compressor chamber exchanges energy and mass with its surroundings. These occur through openings which generally change both in size and shape with time. More than one of them can be connected to the compression chamber at the same time. In the mathematical models mentioned earlier, these flows are introduced through the enthalpy and mass contributions to the fluid in the compression chamber. If all three chambers, namely suction, compression and discharge are simulated by a quasi one dimensional thermodynamic model, as discussed by *Stosic et al, 2005*, both the mass and energy flow estimates are based on the assumption of adiabatic flow through the suction and discharge cavities.

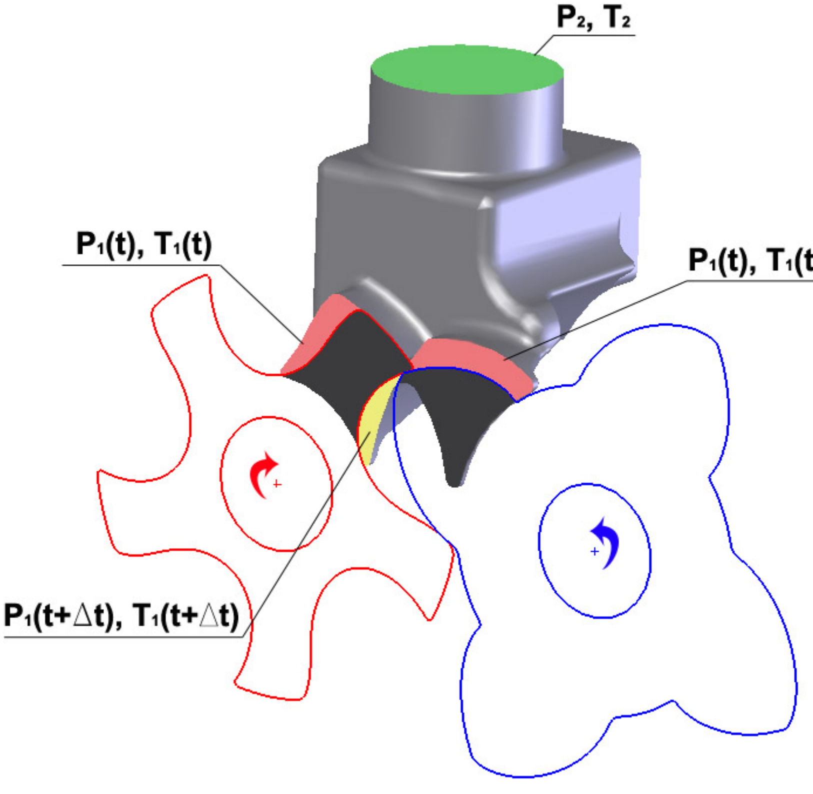


Figure 2 3D domain of the screw compressor discharge chamber

However, if a one dimensional model of the compression chamber needs to be integrated with a three dimensional model of the suction and discharge chambers the exchange of mass and energy must be calculated by the summation of the boundary flows that occur in the three dimensional domains. Once the integrated flows are added to those of the one-dimensional model in the compression chamber, they can be used to calculate the thermodynamic properties in that chamber in the form of pressure, temperature and density, as shown in Figure 1. The solution of the one dimensional model will then be obtained by integration of the differential equations, typically, using the Runge-Kutta 4th order method, or its equivalent, as described by *Stosic et all, 2005* or by *Mujic et al, 2007.*

The derived values of pressure and temperature in the compression chamber are used later as boundary conditions for the three dimensional models of the suction and discharge chambers.

2.2 3-D MODEL OF SUCTION AND DISCHARGE CHAMBERS

A three dimensional Computational Fluid Dynamics (CFD) model has been used for calculating flows through the stationary suction and discharge chambers of a screw machine. These components are usually specified in three dimensions by a CAD system. The discharge port geometry of an oil injected screw compressor with 4 lobes in the male rotor and 5 female rotor lobes is shown in Figure 2. The finite volume method was used for solving the governing PDE equations of momentum, energy and mass conservation and was closed by the constitutive relations, an equation of state and equations for turbulent flow models. The mathematical model was extended for oil injection in the compressor, with an equation for dispersed phase mass concentration to allow for the calculation of both the oil distribution and heat exchange. All the equations of the three dimensional flow model of the screw compressor port geometry can be expressed in the following common form:

$$\frac{d}{dt}\int_V \rho\phi dV + \int_S \rho\phi(\mathbf{v}-\mathbf{v}_s)\cdot d\mathbf{s} = \int_S \Gamma_\phi \operatorname{grad}\phi \cdot d\mathbf{s} + \int_S \mathbf{q}_{\phi S}\cdot ds + \int_V q_{\phi V}\cdot dV \qquad (1)$$

Descriptions of the transported properties of fluid and the source terms Γ_ϕ, $\mathbf{q}_{\phi s}$ and $\mathbf{q}_{\phi V}$ are given in Table 1. These definitions were used in a commercial CCM solver to obtain the distribution of pressure, temperature, velocity and the density fields throughout the fluid domain. The integral parameters of screw compressor performance can then be derived from the solution of these equations.

The discharge chamber is connected to its surroundings by two boundaries, as shown in Figure 2. Index 1 defines the pressure boundary connection to the compression working chamber, while index 2 defines the area of the pressure boundary through which the compressor displaces fluid to the discharge reservoir. All the other boundaries are set to be no slip walls.

The size of the discharge flange which connects the discharge chamber with the receiver is constant. The pressure and the temperature are set to be constant across the boundary, as indicated in Figure 2 by T_2 and p_2.

The boundaries of the connection with the compression chamber, defined by index 1 change both shape and size as the rotor lobes pass over the port. Usually, at any time, two working chambers, separated by a rotor wall, are in direct contact with the port. The pressure and temperature values for that boundary region of the three dimensional domain are calculated from the one dimensional model of the compression chamber as denoted with index 1. It is necessary to calculate the values for only one of the compression chambers in contact with the discharge chamber with the 1D model. By advancing the other chamber in time by *Δt,* the immediately preceding interlobe values can then be used for the other working chamber.

Table 1 Terms in the generic transport equation

Equation	ϕ	Γ_ϕ	$\mathbf{Q}_{\phi S}$	$q_{\phi V}$
Momentum	v_i	μ_{eff}	$\left[\mu_{eff}\left(\mathrm{grad}\,\mathbf{v}\right)^{\mathrm{T}} - \left(\frac{2}{3}\mu_{eff}\mathrm{div}\,\mathbf{v} + p\right)\mathbf{I}\right]\cdot\mathbf{i}_i$	$f_{\mathrm{b},i}$
Energy	e	$\frac{k}{\partial e/\partial T} + \frac{\mu_t}{\sigma_T}$	$-\frac{k}{\partial e/\partial T}\frac{\partial e}{\partial p}\cdot\mathrm{grad}p$	$\mathbf{T}$: grad $\mathbf{v}$ + h
Oil conc.	c_i	$\rho D_{i,\mathrm{eff}}$	0	s_{ci}
Space	$\frac{1}{\rho}$	0	0	0
Turbulent kin. energy	K	$\mu + \frac{\mu_t}{\sigma_k}$	0	$P - \rho\varepsilon$
Dissipation	ε	$\mu + \frac{\mu_t}{\sigma_\varepsilon}$		$C_1 P\frac{\varepsilon}{k} C_2\rho\frac{\varepsilon^2}{k} - C_3\rho\varepsilon\,\mathrm{div}\,\mathbf{v}$

In establishing the mathematical model for the suction and discharge chambers three dimensional domains, the following simplifications have been made:

i) heat exchange through the compressor walls is neglected

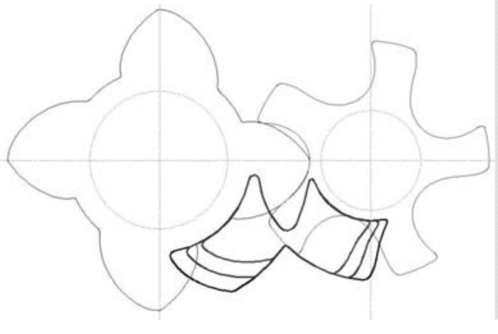
ii) the pressure and temperature are uniformly distributed at the boundaries in contact with the compression and discharge chambers.

A detailed review of the tools and methods used for the analysis of screw machines is given by *Kovacevic et al, 2006*.

3 PREDICTIONS OF AEROACOUSTIC NOISE BY USE OF THE INTEGRATED METHOD

Noise identification and analysis in screw compressors and similar types of rotating machine have been an area of significant interest for a long time. This section gives some details of an ongoing research and development programme, which uses these integrated simulation methods to estimate and reduce sound emission.

Pressure fluctuations in the discharge port affect not only the aero acoustics in that domain but also the noise generated mechanically by rotor rattling. Figure 3 shows some preliminary measurements of gas pulsations in an industrial compressor for different sizes of discharge ports, where differences in pressure oscillations in the port area can be seen. It is believed that correct porting can decrease the level of noise and increase the performance of the machine. A one dimensional thermodynamic method presented by *Mujic et al, 2007* can be used to predict the pressure oscillations as a function of the shape of the port and the cross sectional area of the connecting flange. These predictions agree reasonably well with the measured results. However, this model does not take into account the shape of the discharge chamber which affects the higher harmonics of pressure fluctuations. These play a significant role in noise generation and may also change the flow losses in the ports. In one dimensional flow models, the latter are either neglected or estimated from empirical relationships.



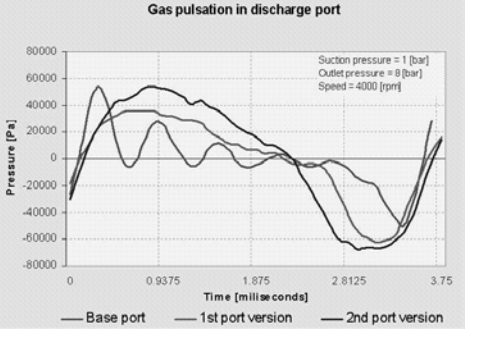


Figure 3 Pressure oscillation measurements in a compressor discharge port

It was decided that a 1D analysis of the pressure and temperature change in the compression chamber, integrated with a 3D CFD model, solved in the novel commercial CFD code StarCCM+, would be a suitable procedure for analysing this case. Since the 1D chamber model runs as a user subroutine in the CFD software it allows easy mapping of the boundary conditions and fast overall calculations. The 3D CAD model wa produced directly from the design management suite for screw compressors called DISCO. A polyhedral numerical mesh was, in this case, automatically generated from the 3D model of the discharge port, as shown in **Figure 4**.

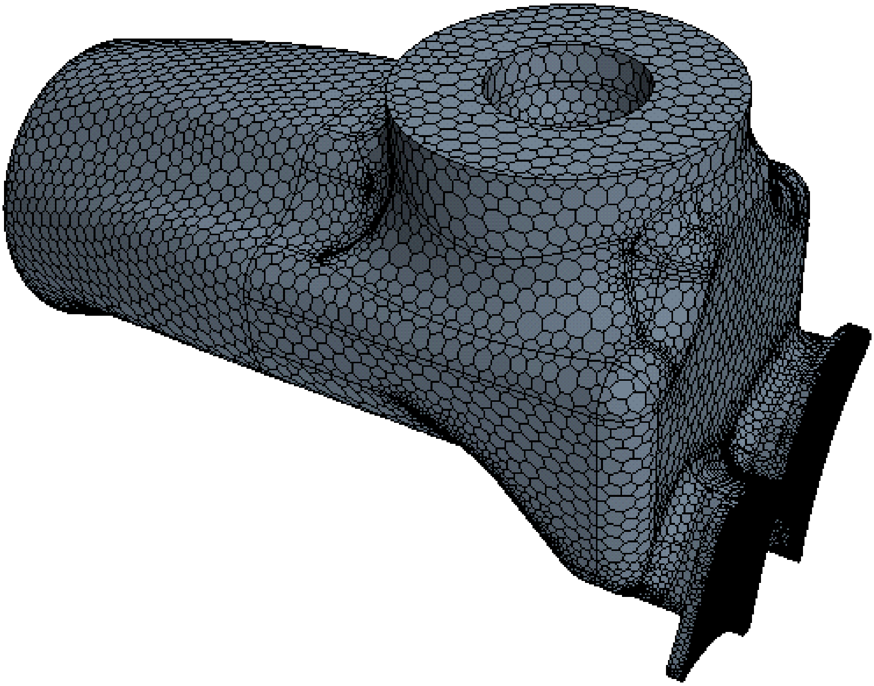


Figure 4 Numerical mesh of the screw compressor discharge chamber

As shown in Figure 5, the stationary computational grid consists of polyhedral grid elements. By the use of polyhedral grid elements it is possible to discretise the solution domain with fewer elements than with any other type of numerical grid. This reduces computational time without affecting the accuracy. Local grid refinements were performed in parts of the numerical grid, where needed. The model of the compressor discharge chamber used in this example has slightly more than 100000 control volumes.

The convergence of the procedure was assessed by checking the difference of pressure in the control domain over two consecutive cycles. The calculation continued until that difference was reduced by three orders of magnitude. By using such a procedure and a time step corresponding to 1 degree of the male rotor angle rotation, only 4 cycles were required to establish a periodic condition. Approximately 5 hours per cycle were required on a computer with a single Pentium IV 2 GHz processor. Therefore for the converged solution 20 hours of computation time were required.

Figure 5 shows a comparison of the gas pulsations predicted by the three dimensional model with the experimental values for three different compressor speeds and two discharge pressures. As can be seen, the combined model is able to capture the higher pulsation harmonics and predict the amplitude of pressure oscillations for the compressor fundamental frequency reasonably accurately. The amplitudes of the higher harmonics are in some cases captured with more success than in others. It is believed that the differences are due to the assumption of a constant boundary condition at the outlet of the discharge chamber. This practically neglects the influence of the rest of the discharge system on the gas pulsations.

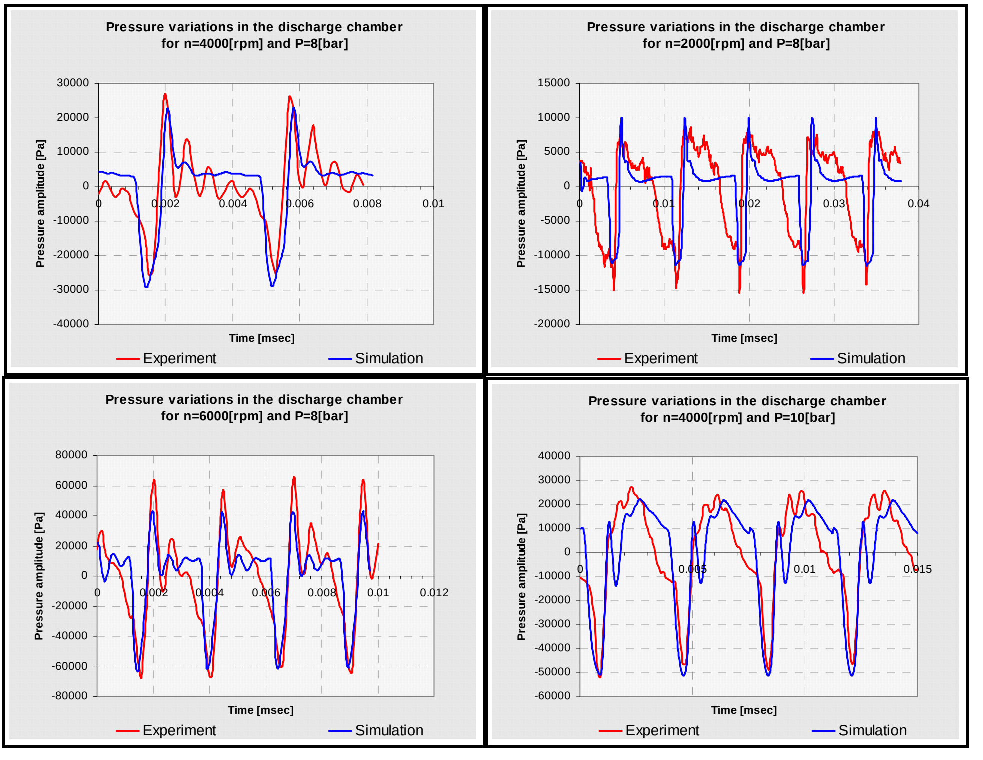


Figure 6 Gas pulsations calculated by three dimensional model

The results obtained from this model are of sufficient accuracy to evaluate the level of pressure fluctuations in the suggested compressor port. For a further improvement in the accuracy of the results, the boundary conditions between the 1D and 3D models need to be checked and improved. Additionally, for more accurate capture of the higher harmonics, the boundary condition for the connection of the discharge chamber with the reservoir needs to account for the remainder of the system.

4 CONCLUSIONS

New methods and tools for the analysis and development of screw compressors can extend the scope of their development and use for a long time. Moreover, these tools can be used for other types of machine with similar configurations, such as expanders, gear pumps, multiphase pumps and vacuum pumps.

Integrating three dimensional analysis of flow in the suction and discharge chambers with a one dimensional model of flow in the compression chamber offers the possibility for faster and more accurate results when optimising compressor ports. The results obtained are encouraging but further improvements in the method are still required.

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