NUMERICAL SIMULATION OF COMBINED SCREW COMPRESSOR-EXPANDER MACHINES FOR USE IN HIGH PRESSURE REFRIGERATION SYSTEMS

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Abstract

Recent interest in natural refrigerants has created a new impetus for studies of CO_2 as a working fluid in vapour compression systems for refrigeration and air conditioning. Two major drawbacks to its use are the very high pressure differences required across the compressor and the large efficiency losses associated with the throttling process. One of potential advantages of screw machines over other types of positive displacement machine is their ability to perform both the compression and expansion functions simultaneously, using only one pair of rotors. A further feature is the use of the rotors which seal on both contacting surfaces so that the same profile may be used both for the expander and the compressor sections. By using the same profile for both, the compressor and expander rotors can be machined or ground in a single cutting operation and then separated by machining a parting slot in them on completion of the lobe formation. Computational Fluid Dynamics and Structure Analysis are used in this paper for investigation of fluid and solid interaction in such machines.

Key Words: Screw Compressor, Screw Expander, Computational Fluid Dynamics, Mathematical Modelling

Nomenclature

C	- turbulence model constants
f	- body force
i	- unit vector
I	- unit tensor
k	- kinetic energy of turbulence
m	- mass
$\dot{m}$	- source in the pressure correction equation
p	- pressure
P	- production of turbulent kinetic energy
q	- source term
$\dot{Q}$	- source in the energy equation
s	- control volume surface
t	- time
u	- displacement in solid
v	- fluid velocity
V	- volume
x	- spatial coordinate
z	- axial coordinate
Γ	- diffusion coefficient
ε	- dissipation of turbulent kinetic energy
ϕ	- variable
λ	- Lame coefficient
μ	- viscosity
η	- Lame coefficient
ρ	- density
σ	- Prandtl number
Δt	- time step used in the calculation

Indices

add	- added or subtracted
eff	- effective
rec	- receiver
T	- turbulent

Introduction

A significant amount of energy in developed countries is used in compressors for air-conditioning and refrigeration. Therefore, market demands require the rapid production of such machines, which are competitive both in efficiency and unit price. Apart from that, there is large interest in introducing natural fluids for refrigeration. One of these is CO_2. Despite the environmental advantages of CO_2 as a working fluid in vapour compression systems for refrigeration and air conditioning, there are two major drawbacks to its use. These are the very high pressure differences of up to 60 bars required across the compressor and the large efficiency losses associated with the throttling process. In this case, the throttle losses are so large that some form of power recovery from the expansion is essential to obtain an acceptable coefficient of performance, COP from a CO_2 system. To overcome the throttle losses, a number of proposals have been made for various types of positive displacement machine, mainly of the vane type, which combine compression with some recovery of work from the expansion process. However, how well they operate with high pressure differences across the vanes has not yet been confirmed.

For many years, the authors have been investigating the use of twin screw machines to fulfil both the expansion and compression processes when using more conventional halocarbon refrigerants. These have many potential advantages over other types of positive displacement machine. Unfortunately, when applied to CO_2 the huge bearing forces associated with the pressure distribution within them have hitherto made them appear to be unsuitable.

In this paper, it is shown how the rotor forces created by the compression and expansion processes can be partially balanced in order to eliminate the axial forces and reduce the radial bearing forces. The disadvantages of twin screw compressors for such high pressure applications are thereby reduced. The balanced rotor concept is also applicable to vapour compression systems using more conventional refrigerants and even for high pressure gas compression.

Apart from reducing pressure loads, the simultaneous compression and expansion processes reduce throttling losses in CO_2 vapour compression system plants significantly. It may also be used in non refrigeration applications to permit high pressure compression in a single stage system with reduced rotor loads, even where there is no system requirement to recover power from the expansion process. Analysis shows that the coefficient of performance will be improved by both these factors, recovering of the throttle loss and reducing of mechanical loses because of lower rotor loads and thus be increased by 72% from 2.79 to a more acceptable 4.8. However, these figures are based on idealised work input and output. In a practical system, allowance would have to be made for the compression and expansion efficiencies, which would reduce the expansion work and increase the compression work. Nonetheless, an overall gain in coefficient of performance over the ideal cycle with a throttle valve should still be achievable by this means. More information on this application can be found in *[8]*.

CFD analysis has been used here as the means of modelling the pressure and temperature changes within the machine. This has been developed further by determining how the solid components deform as a result of these changes and hence to determine how the fluid-solid interaction resulting from these changes altered their performance. The results of such numerical simulation are presented in the paper.

Background

Screw compressors are reliable and compact machines and consequently they comprise the majority of all positive displacement compressors now sold and of those currently in operation. One of the main reasons for their success is the advance in manufacturing techniques, which enable compressor rotors to be manufactured with very small clearances at an economic cost. Internal leakages have thereby been reduced to a fraction of their values in earlier designs. However, pressure differences across screw compressor rotors impose heavy loads on them and create rotor deformation, which is of the same order of magnitude as the clearances between the rotors and the casing. Consequently the working pressure differences at which twin screw machines can operate reliably and economically are limited. Current practice is for a maximum discharge pressure of 85 bar and a maximum difference between suction and discharge of 35 bar. *Rinder, [7]*, presented a comprehensive analysis of these effects and *Arbon, 0*, gave a good review of current trends in the design, manufacture and use of high pressure screw compressors. CO_2 (R744) in refrigeration cycles requires both maximum pressures and pressure differences beyond these limits. Accordingly, hitherto screw compressors have not been considered for this purpose.

Other types of positive displacement compressors are used today for compression in CO_2 cycles. Typically, these are single and two-stage reciprocating compressors. A vane compressor study was presented in *[4]*. In such applications the authors concentrate on either the thermodynamic aspects of the CO_2 cycle or mechanical design aspects of its compressors. In considering twin screw machines capable of operation over the pressure range required in CO_2 refrigeration systems, the following factors must be taken into account

Screw compressors and expanders are rotating positive displacement machines widely used in industry for air compression, refrigeration and process gas applications. These essentially consist of a pair of meshing helical lobe rotors contained in a casing. Rotors and housing together form a series of working chambers in which either compression or expansion process occur. Screw machines function as a result of the volume change when their rotors rotate in the casing.

The compression process is thus achieved by positioning of the low pressure suction port where the trapped volume is largest and the high pressure discharge port at a position where the trapped volume is reduced in size, thereby causing the pressure to rise within the machine, as shown in Figure 1a.

An important feature of such a machine is that if the direction of rotation of the rotors is reversed, then gas will flow into the machine through the high pressure port and out through the low pressure port and it will act as an expander, as shown in Figure 1b.The machine will also work as an expander when rotating in the same direction as a compressor, provided that the suction and discharge ports are positioned on the opposite sides of the casing to those in the compressor, since this is effectively the same as reversing the direction of rotation relative to the ports. When operating as a compressor, mechanical power must be supplied to the shaft to drive it. When acting as an expander, it will rotate automatically and the power generated will be supplied externally through the drive shaft.

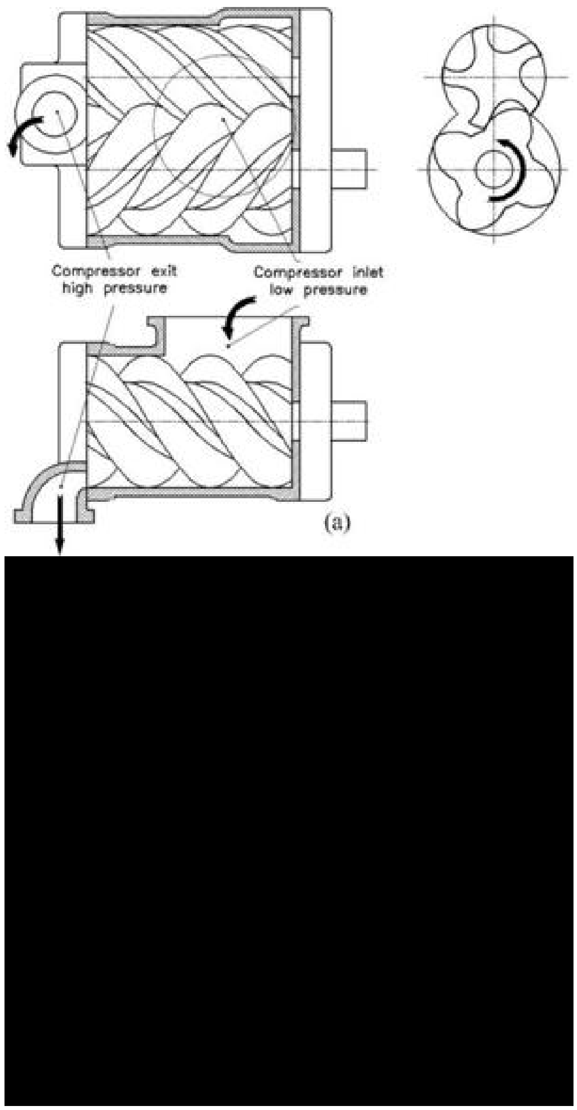

Figure 1 Screw compressor (a) and Screw expander (b) working principles

If the rotor profile is generated to form a seal on both contacting surfaces, then the expansion and compression functions will be performed with equal efficiency regardless of the direction of rotation and hence, more significantly, the same pair of rotors may be used for simultaneous expansion and compression in the same machine. There are two methods by which this may be achieved. The first, which is cheaper but more limited in scope, is to carry out both functions in a single chamber, containing only three ports, as shown in Figure 1a. The second is to divide the chamber into separate expander and compressor sections with independent inlet and exit ports for each function, as shown in Figure 1b. Their common feature is that, for both the rotors and the casing, the compression and expansion sections can be machined or ground in a single cutting operation.

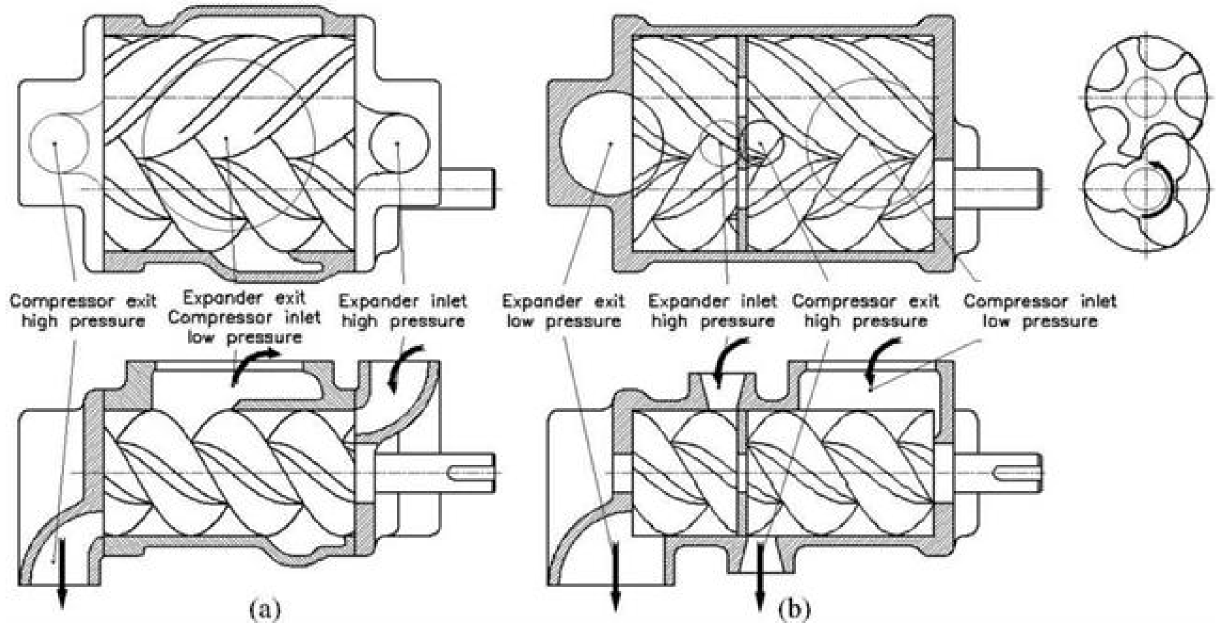

Figure 2 Combined compressor-expander with single chamber (a) and with split chambers (b)

A balanced rotor two-chamber combined compressor expander is considered in this paper, as shown in Figure 2b. A fair amount of oil is injected in the system to seal, cool and lubricate rotors. Because of that, temperature of the rotors is kept at a low level similar to the temperature of the main fluid. For this application, the most important feature of the two chamber arrangement is the layout of the porting by which the fluid enters and leaves this combined compressor-expander. In the proposed arrangement, the axial load on the compressor rotors is balanced by that on the expander side. Additionally, the radial loads on the compressor and expander sections are located almost opposite each other near the centre of the rotors. This reduces the radial loads on the bearings significantly and hence the mechanical friction losses become smaller. This is of particular importance when considering the design of refrigeration compressors for use with natural refrigerants such as CO_2, where the sustainable bearing load limits the maximum permissible operating pressure difference. Additionally, in the combined machine, the deformation of the rotors can be substantially smaller. Together, the reduced rotor deformations and the stronger female rotor increase the range of pressure for which a screw machine can be used for CO_2 refrigeration applications.

3-D CCM Calculation

Improvements in both, computational speed and numerical methods over the last thirty years have greatly increased the scope and power of Computational Fluid Dynamics, CFD and Computational Continuum Mechanics, CCM in engineering design. Consequently, vendors of CFD and CCM software packages have developed facilities for their use in a wide range of engineering applications. Because none of the standard packages were capable of analysing the complex geometry and processes within screw machines, designers of them have hitherto made little or no use of them.

A CCM method simultaneously enables calculation of the 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 on sliding and stretching moving mesh.

Grid Generation

The authors have developed an automatic numerical mapping method for arbitrary screw compressor geometry, as explained in *[5]*, which was later used for the analysis of the processes in screw compressors. This method took advantage of the work done by Peric and Demirdzic, *[3]* and *[2]*, who showed that by the use of moving frames on structured and unstructured grids, a common numerical method could be used for the simultaneous solution of fluid flow and structural analysis. On that basis, authors have developed an interface program called SCORG (Screw COmpressor Rotor Geometry Grid generator), which also enables a grid, generated by the program, to be directly transferred to a commercial CFD or CCM code through its own pre-processor. A number of commercial numerical solvers are currently available, of which the authors decided to employ COMET of Star CD for screw machine calculations. That code offers the possibility to calculate both the fluid flow and solid structure simultaneously by the application of the Computational Continuum Mechanics, CCM.

The interface employs a novel procedure to discretise rotor profiles and to adapt boundary points for each particular application. An analytical transfinite interpolation method with adaptive meshing is used to obtain a fully structured 3-D numerical mesh, which is directly transferable to a CFD code. This was required to overcome problems associated with moving, stretching and sliding rotor domains and to allow robust calculations in domains with significantly different ranges of geometry features.

Governing Equations

The density of the compressor working fluid changes in screw machines with both pressure and temperature. The compressor flow and the structure of compressor parts 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 in [3]. In the case of a multiphase flow, like in this case where oil is injected in the working chamber, the concentration equation, is added to the system. The numerical solution of such a system of partial differential equations is then made 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.

All these equations are conveniently written in a form of the following generic transport equation:

$$\begin{aligned}&\frac{d}{dt}\int_V \rho\phi dV + \int_S \rho\phi(\mathbf{v}-\mathbf{v}_s)\cdot d\mathbf{s} = \\ &= \int_S \Gamma_\phi \mathrm{grad}\,\phi \cdot d\mathbf{s} + \int_S \mathbf{q}_{\phi S}\cdot d\mathbf{s} + \int_V \mathbf{q}_{\phi V}\cdot dV\end{aligned} \tag{1}$$

Here ϕ stands for the transported variable, e.g. Cartesian components of the velocity vector in fluids v_i, enthalpy h, etc. Γ_ϕ is diffusion coefficient. The meaning of source terms, $\mathbf{q}_{\phi S}$ and $\mathbf{q}_{\phi V}$ in all transport equations is given in Table 1.

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 solid and fluid regions to be solved simultaneously.

Boundary Conditions

This mathematical scheme is accompanied by boundary conditions for both the solid and fluid parts. A novel treatment of the compressor fluid boundaries was introduced in the numerical calculation, as presented in [6]. 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. Mass source, which exists in the pressure correction equation, is calculated as the difference between the cell pressure p from the previous iteration and the prescribed pressure in the receiver, p_{rec} as:

$$\dot{m}_{add} = \left(\frac{dm}{dt}\right)_{p=const} \approx \frac{p_{rec} - p}{p_{rec}} \frac{V\,\rho}{\Delta t} \quad (2)$$

The volume source for the receiver cells which exists in the energy equation is calculated from the mass source as:

$$\dot{Q}_{add} = h_{add} \left(\frac{dm}{dt}\right)_{p=const} = \dot{m}_{add}\, h_{add} \quad (3)$$

Enthalpy of the fluid added to or subtracted from a cell is calculated on the basis of thermodynamic values achieved in previous iteration.

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.

Table 1 Terms in the generic transport equation (1)

Equation	ϕ	Γ_ϕ	$\mathbf{q}_{\phi S}$	$\mathbf{q}_{\phi V}$
Fluid Momentum	v_i	μ_{eff}	$\left[\mu_{eff}\left(\text{grad}\,\mathbf{v}\right)^{\text{T}} - \left(\frac{2}{3}\mu_{eff}\,\text{div}\,\mathbf{v} + p\right)\mathbf{I}\right]\cdot\mathbf{i}_i$	$f_{\text{b},i}$
Solid Momentum	u_i	η	$\left[\eta\left(\text{grad}\,\mathbf{u}\right)^{\text{T}} + \left(\lambda\,\text{div}\,\mathbf{u} - 3K\alpha\Delta T\right)\mathbf{I}\right]\cdot\mathbf{i}_i$	$f_{\text{b},i}$
Energy	e	$\frac{k}{\partial e/\partial T} + \frac{\mu_t}{\sigma_T}$	$-\frac{k}{\partial e/\partial T}\frac{\partial e}{\partial p}\cdot\text{grad}p$	$\mathbf{T}$: grad $\mathbf{v}$ + h
Concentration	c_i	$\rho D_{i,\text{eff}}$	0	s_{ci}
Space	$\frac{1}{\rho}$	0	0	0
Turbulent kinetic energy	K	$\mu + \frac{\mu_t}{\sigma_k}$	0	$P - \rho\varepsilon$
Dissipation	ε	$\mu + \frac{\mu_t}{\sigma_\varepsilon}$	0	$C_1 P\frac{\varepsilon}{k} - C_2\rho\frac{\varepsilon^2}{k} - C_3\rho\varepsilon\,\text{div}\,\mathbf{v}$

Presentation and Discussion of the Results

A personal computer, with Athlon 1800 MHz processor and 1.5 GB memory was used for calculation. Compressor operation was simulated through 24 time steps for one interlobe rotation giving the overall number of 120 time steps for a full rotation of the male rotor. The time step length was synchronised with the compressor speed of 5000 rpm. Error reduction of 4 orders of magnitude was obtained after approximately 50 outer iterations at each time step, which took approximately 10 minutes of the computer time for one time step. The overall compressor parameters such as the torque, volume flow, forces, efficiencies and compressor specific power were then calculated. Additionally, pressure-time diagrams of the compression process, the flow and pressure patterns in the compressor chambers and rotor deformations are provided.

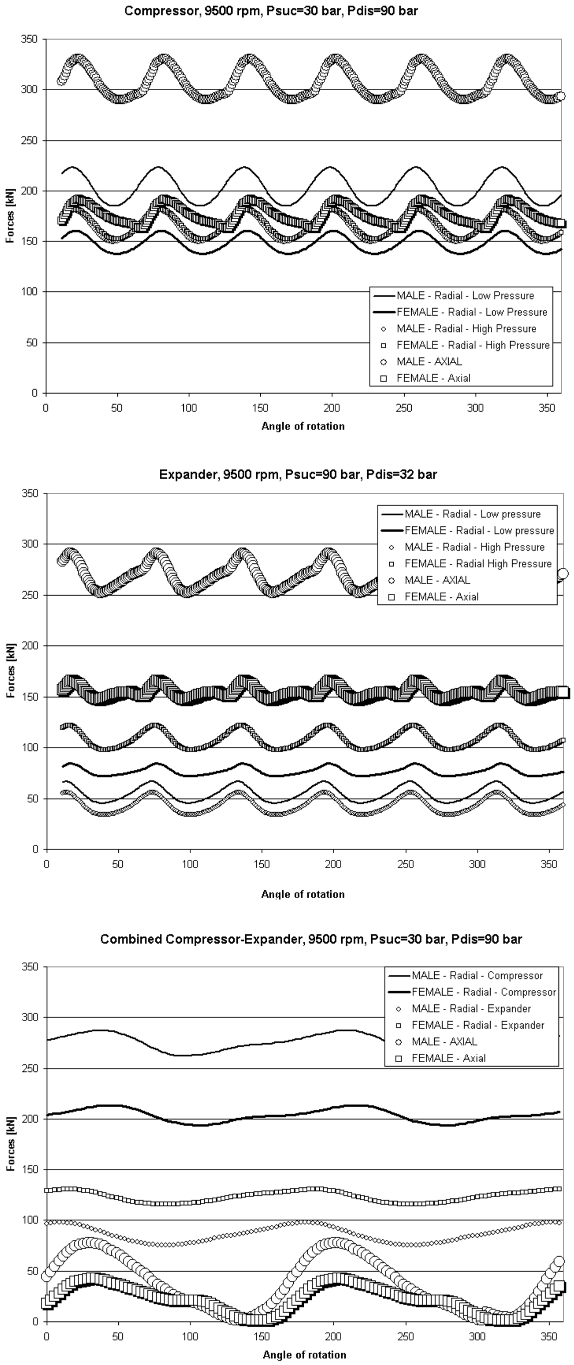

Figure 3 Radial and axial forces for compressor, expander and combined compressor-expander

A balanced rotor two-chamber combined compressor expander is considered here for high pressure CO_2 application. Gears are not used to synchronize the rotors since a fair amount of oil is injected in the system to seal, cool and lubricate rotors. Because of that, temperature of the rotors is kept at a low level similar to the temperature of the main fluid.

The most important feature of the two chamber arrangement shown in Figure 2, for this application, is the layout of the porting by which the fluid enters and leaves this combined compressor-expander. By this means, the axial load on the compressor rotors is balanced by that on the expander side as shown in Figure 4. Additionally, the radial loads on the compressor and expander sections are located almost opposite each other near the centre of the rotors.

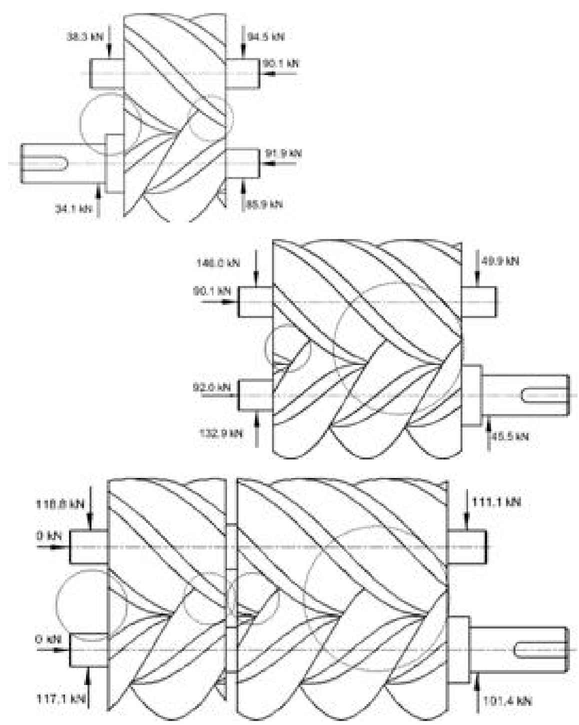

Figure 4 Position of radial and axial forces for compressor, expander and combined compressor-expander

This reduces the radial loads on the bearings significantly and hence the mechanical friction losses become smaller. This is of particular importance when considering the design of refrigeration compressors for use with natural refrigerants such as CO_2, where the sustainable bearing load limits the maximum permissible operating pressure difference.

Additionally, in the combined machine, the deformation of the rotors is substantially smaller as can be seen from Figure 5. Deformations presented in this figure are magnified 5000 times. Together, the reduced rotor deformations and the stronger female rotor, increase the range of pressure for which a screw machine can be used for CO2 refrigeration applications.

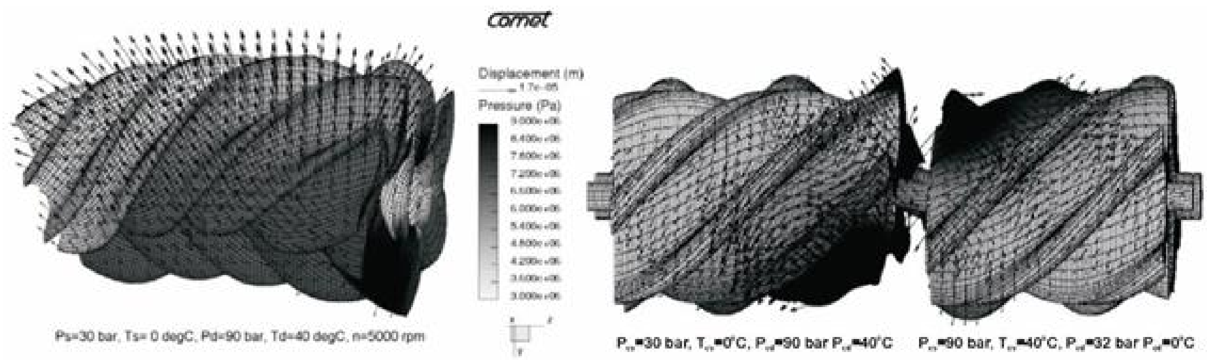

Figure 5 Deformations of the compressor rotors (left) and combined compressor-expander rotors (right)

Conclusions

Ability of screw rotors for simultaneous compression and expansion in the same pair of rotors leads to the possibility of use of such machines in vapour compression refrigeration and air conditioning systems with CO2 as a working fluid.

A full 3-D CFD fluid flow and structure simulation has been carried out to determine how pressure and temperature change during the process and what is their influence to internal distortion within the high pressure screw compressor-expanders. Preliminary results show reduction in both radial and axial forces and thereby further reduction in mechanical losses. Additional increase in efficiency of a system is due to partial recovery of the energy through the expander part of the machine.

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