Transient Thermal Analysis of Screw Compressors

Part I – Development of Thermal Model

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Abstract

This is the first of a two-part report on the transient thermal analysis of screw compressors. This work was undertaken to better understand rotor-to-housing and rotor-to-rotor clearances. Reliability and performance are functional characteristics of the compressor that are strongly dependent on the operating clearances. It is important that these be understood in the context of loaded operation. Development of a process using a one-dimensional thermodynamic simulation and a three-dimensional finite element model of the compressor is described in this part. Results from application of the analysis and comparisons to experimental results are presented in Part II.

Introduction

Screw compressors require small clearances for high efficiency, but if clearances get too small, damage or failure of the compressor can result. Changes in critical dimensions from their nominal design state result from manufacturing variability and changes in the parts caused by operational loads. This report documents development of a process for computing the effects of thermal loads on the operating clearances between the screw rotors and between rotors and stationary housing parts.

Calculation of deformation caused by thermal loads and analysis of the resulting clearances is carried out using a finite element model of the compressor rotor-housing assembly. This requires specifying temperature and heat transfer coefficient boundary conditions at fluid-solid interfaces. These boundary conditions are calculated using a proprietary thermodynamic simulation which computes internal pressures and temperatures given details of compressor geometry. This information is then used to calculate heat transfer coefficients according to the procedures described in this report. The final step in the preparation of the boundary condition information is associating the temperatures and heat transfer coefficients with nodes on the finite element model. Computation of the boundary conditions and associating them with the finite model are discussed in the remainder of this report.

Thermodynamic Simulation and Finite Element Model Boundary Conditions

The procedure for computation of heat transfer coefficients is based on the work of Kauder and Keller [1]. The process described here is also reported in [2]. A thermodynamic simulation of the refrigerant screw compressor is used to compute fluid properties to use as boundary conditions for a finite element model. The program computes pressure, temperature and mass in the volume – the working chamber – defined by the meshing screw rotors and adjacent stationary housing parts. The properties within the working chamber change during the compression process, but at any time during the process, fluid properties are assumed to be constant throughout the working chamber. Figure 1 shows a pair of screw rotors, illustrating the working chamber at a point early in the intake process.

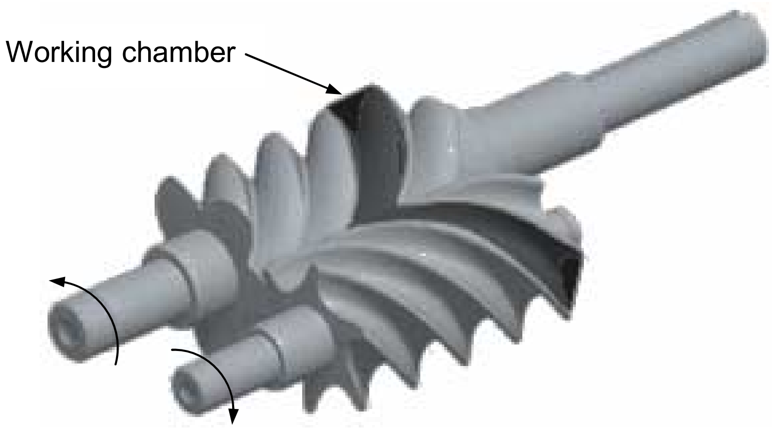


Fig. 1: Screw rotor pair and illustration of the working chamber

The simulation program provides much of the information needed to estimate surface heat transfer coefficients, but it did not have a model for carrying out the actual calculations. Such a model was reported in [1]; we decided to adopt the methods reported in the reference and modify the models as necessary as experience was gained in our oil-injected, refrigerant compressors.

Estimation of the heat transfer at the screw rotor and housing surfaces is based on application of well-known heat transfer models for flow in a pipe and for free or forced convection at simple plain surfaces or fins. Surface models adopted are:

Cylindrical shell
Upper side of a flat plate
Lower side of a flat plate
Vertical flat surface

Heat transfer coefficients for the working chamber surfaces are based on the model for turbulent flow in a pipe. The definition of the non-dimensional heat transfer coefficient α as used in the modeling is:

$$\alpha = \frac{\dot{q}}{A \cdot \Delta T} \tag{1}$$

The goal of the analyses is to determine the temperatures in the compressor parts; doing this requires knowing the heat fluxes at the fluid-solid interfaces. The role of equation 1 is to relate the heat flux and fluid-solid temperature difference. Used in this way, it is necessary to provide a value for the heat transfer coefficient α. This is done using the definition of the Nusselt number, Nu:

$$\mathrm{Nu} = \frac{\alpha \cdot L}{\lambda} \quad \text{or} \quad \alpha = \frac{\mathrm{Nu} \cdot \lambda}{L} \tag{2}$$

The Nusselt number is computed as follows:

$$\mathrm{Nu} = 0.0235 \cdot \left(\mathrm{Re}^{0.8} - 230\right) \cdot \left(1.8 \cdot \mathrm{Pr}^{0.3} - 0.8\right) \cdot \left[1 + \left(\frac{d_h}{L}\right)^{2/3}\right] \cdot \left(\frac{\eta_{FL}}{\eta_w}\right)^{0.14} \tag{3}$$

Where the Reynolds number (Re) and Prandtl number (Pr) are defined as:

$$\mathrm{Re} = \frac{c \cdot L \cdot \rho}{\eta} \tag{4a}$$

$$\mathrm{Pr} = \frac{\eta \cdot C_p}{k} \tag{4b}$$

This is the way in which the Nusselt number is computed for use in the screw compressor analyses as reported in [1].

Evaluation of equations 2, 3 and 4 for the working chamber requires a velocity, c, a characteristic length, L, and a hydraulic diameter, d_h. The velocity relative to the stationary housing surfaces is taken as the vector sum of an axial velocity and the rotor tangential velocity. The axial velocity is the rate at which the end of the working chamber sealed by the meshing of the rotors moves axially as the rotors rotate, determined by the rotor rotational speed and the lead or wrap angle of the helical rotors. Definitions for the calculation of the reference velocity are illustrated in Figure 2.

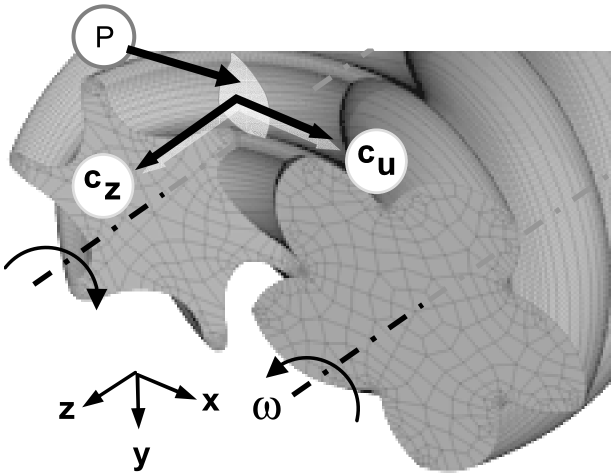


Fig. 2: Definitions for working chamber reference velocity calculations

The axial component c_z of the working chamber velocity is:

$$c_z = \frac{L_r}{\Phi_r} \cdot 2\pi \cdot n \tag{5}$$

The tangential velocity is computed using a reference radius r_m. This radius divides the chamber cross sectional area (plane P in Figure 2) into two equal areas, one between the rotor root and r_m and the other between r_m and the outer diameter of the rotor lobe. The tangential velocity associated with this radius is:

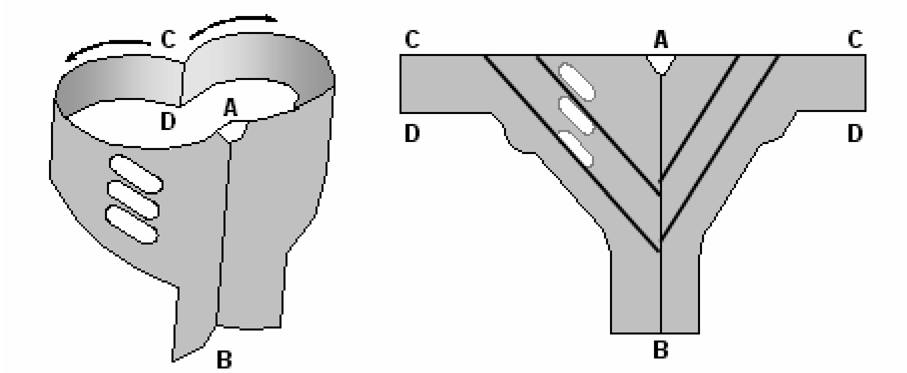
$$c_u = r_m \cdot \omega \tag{6}$$

This leads to the calculation for the fluid velocity in the chamber to be used for heat transfer coefficient calculations at the housing wall:

$$c = \sqrt{c_z^2 + c_u^2} \tag{7}$$

The hydraulic diameter, defined as 4 times the perimeter of a surface divided by its planar area, in equation 3 is computed for the cross sectional area, P in Figure 2. This hydraulic diameter is used as the reference length for the Reynolds number calculation (equation 4a).

The thermodynamic simulation computes the fluid properties in the working chamber as the compression process proceeds from the intake phase through to the discharge of the vapor on the high pressure side. The working chamber has a particular size (volume) and location depending on the rotational orientation of the two rotors. The assumption in the model is that while the fluid properties change continuously during the compression process, they are uniform throughout the working chamber volume at any instant of time. This means that the fluid temperature and the calculated heat transfer coefficients will be the same over all of the surfaces enclosing the chamber.



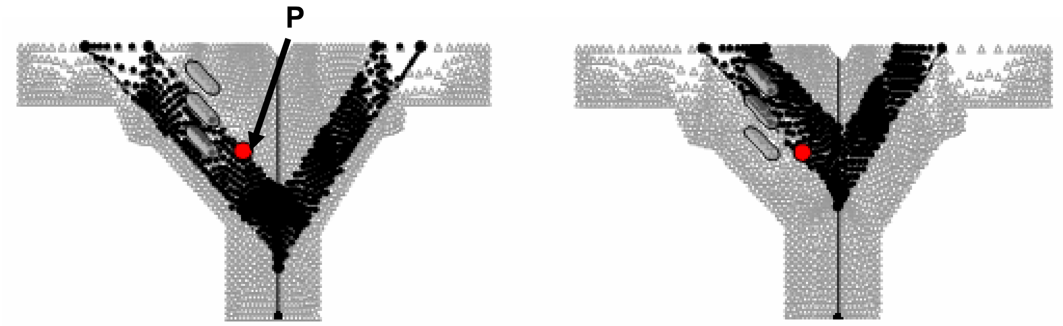
If we consider one point on the stationary housing forming the outer boundary of the

Fig. 3: Development of the unwrapped view of the rotor housing

chamber, we see that it will experience a variation in properties during the time it is within the boundaries of the chamber – a period of time equivalent to the passage of one rotor lobe. The problem, then, is one of associating the appropriate part of the compression process to each point on the housing, then averaging the temperature and heat transfer coefficient over that period to arrive at values for use as boundary conditions in the finite element modeling.

Assigning the averaged properties to specified points on the rotor housing requires associating it with the period in the compression process during which it is exposed to the working chamber defined by the rotor lobes. This is best illustrated using an "unwrapped" view of the rotor housing. The unwrapping is shown in Figure 3. The actual geometric form of the rotor housing surfaces that make up the stationary boundary of the working chamber is shown in the view on the left in the figure. The three elongated holes are the unloader ports on the male rotor side of the housing and the notch near point A is the radial discharge port. We imagine cutting the housing along the low pressure cusp marked CD in the figure. The housing is then unwrapped in the directions shown by the arrows and "flattened" to arrive at the unwrapped view seen on the right side in the figure.

The heavier lines in the unwrapped view are the projections of the rotor lobe tips shown for a particular angular orientation. As the rotors rotate, the lobe-tip projections in the unwrapped view will move up, in the direction from B to A, the inlet and discharge ends of the high pressure cusp, respectively. Figure 4 illustrates the process of assigning average properties to a particular point on the housing. Consider a representative location on the housing of the male rotor, indicated by the larger symbol designated “P” in the figure. The many smaller



(a) (b)

Fig. 4: Assignment of fluid properties to finite element model nodes

symbols show locations of the finite element model nodes on the surface of the rotor housing. Figures 4a and 4b show two orientations of the rotors, one lobe pitch apart. The darker symbols highlight the nodes that are exposed to the working chamber pressure and temperature during the part of the compression at each of these two rotor orientations. With the 5 lobe rotor of this example, each of the housing points sees one-fifth of the compression process in a repeating cycle. The particular part of the process seen by each point is determined by the process shown graphically in Figure 4. Each node is assigned a pressure, temperature and heat transfer coefficient that are averages over the part of the cycle to which it is exposed.

Contours of fluid temperature at the rotor bore surfaces computed, averaged and applied as discussed above, are shown in Figure 5 (temperature scale °F). Contours of pressure and heat transfer coefficient would look very much the same. As expected from the averaging process, areas of constant properties align with the helical orientation of the projection of the rotor tips at the housing surface bounding the working chamber. The highest temperatures are found at the discharge port.

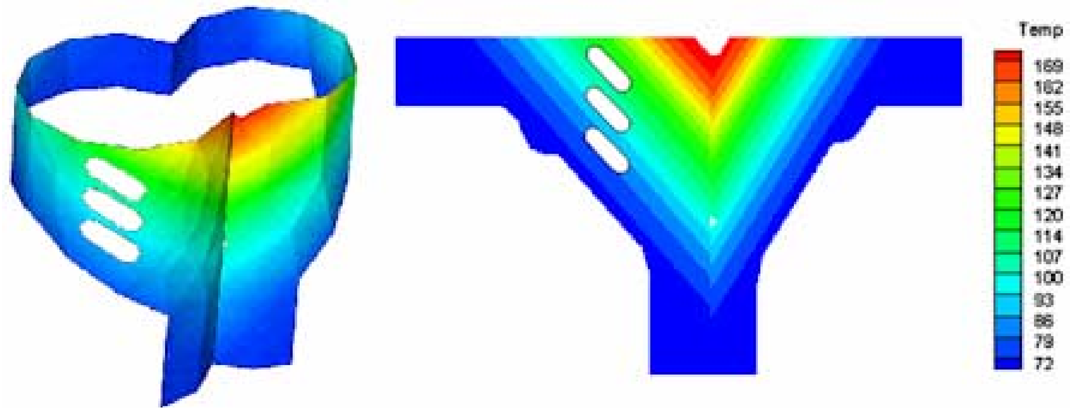


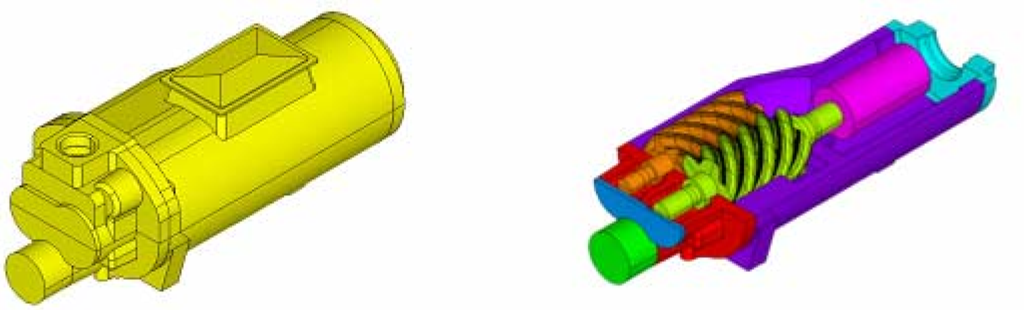
Fig. 5: Calculated fluid temperatures on rotor housing surfaces

The boundary conditions computed with the thermodynamic simulation are written to a table which is loaded into the finite element model. Boundary conditions are transferred to the model for the analysis. If the node locations in the finite element model are changed, the boundary conditions can be interpolated from the original table so that the thermodynamic simulation does not have to be re-run.

Finite Element Model

A finite element analysis is used to solve the convective heat transfer problem for the compressor to calculate metal temperatures. Once metal temperatures are available, the thermal growth problem is solved and clearances calculated.

Figure 6a shows an external view of the screw compressor model while Figure 6b shows an internal view with the top half of the compressor removed.



(a) (b)

Fig. 6: Screw compressor finite element model

Extensive use was made of the finite element program's macro tools during model generation, problem set-up and solution, and post processing of the results. One important application was in the construction of models of the screw rotors and shafts. A generic macro was written, allowing the user to create screw rotors and shafts by reading a geometry file in a standard format. The geometry file contains information about points on the rotor profile, rotor lead and length, shaft dimensions, and bearing locations. Given this information, a hex dominant mesh can be created. An example of a rotor mesh is shown in Figure 7.

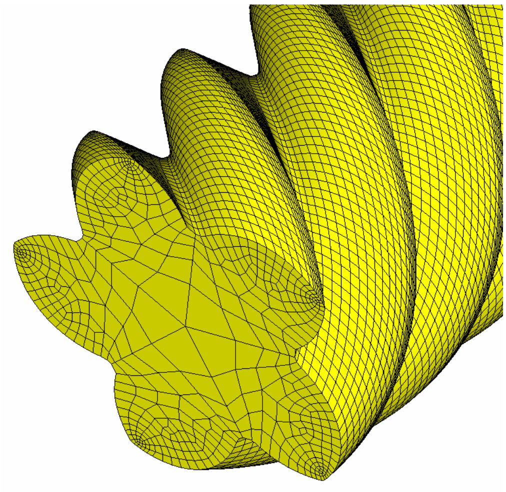


Fig. 7: Hex dominant mesh on rotor

Having quadratic elements on the surface of the screw rotors allows predicted displacements on the screw rotor surface to be organized in a very efficient way. It is much more difficult to organize this information using a triangular surface mesh.

Thermal Analysis

The thermal analysis proceeds in a straightforward fashion. Both steady-state and transient thermal problems can be solved. For steady-state cases, the thermodynamic simulation is run at the operating conditions of the steady-state run. The boundary conditions from this single run are loaded to the finite element model. The metal temperatures at the start of the solution can be ambient or can have values from the solution at a different operating condition. The finite element solution is transient as metal temperatures are recomputed based on the new boundary conditions until they converge to their new values.

In the case of a transient solution, heat transfer coefficients and bulk temperatures are updated each time step. A history of operating conditions, defined by inlet pressure and temperature and discharge pressure, is taken from compressor tests results or is assumed. A sampling of the operating conditions is taken and the boundary conditions are determined for each point. The conditions for the first point are loaded into the finite element model and a solution is found for the metal temperatures corresponding to these inputs. Then, the second set of boundary conditions (representing another increment in time) is loaded in and the metal temperatures are computed again. This process proceeds until the final operating state is computed. The result is a series of solutions that define the transient response of the housing to the operating condition vs. time history.

Displacement Solution

Metal temperatures predicted by the thermal analysis are used as a thermal boundary condition for the displacement solution. Gravity, rotary inertia of the rotating parts, and internal pressure effects are neglected. Rigid body motion is prevented by constraining a few nodes on the feet of the compressor. Displacements are computed for both rotors and the two housing parts in the model of the compressor assembly. Figure 8 shows an example of the clearances between the discharge end faces of the rotors and the stationary bearing housing. The clearances shown are computed for steady state operation. Red indicates the lower clearances, with yellow to green to blue representing progressively higher levels of rotor face to bearing housing clearance.

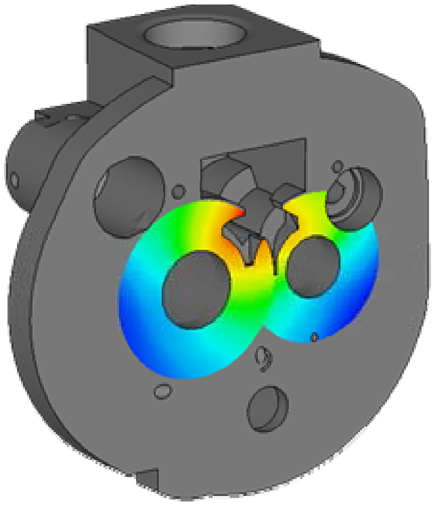


Fig. 8: Rotor-to-housing end clearance

The notable features of the clearance pattern are its non-uniformity over the face of the housing and the higher levels in the highest pressure regions around the discharge port. Having analyses such as this should allow the development of design alternatives to reduce these tendencies.

Clearances between the rotors and the stationary housing parts are computed within the finite element model as a post-processing step using macros developed specifically for this purpose.

Clearances between the rotors are computed in a separate program, developed by the team at FG Fluidenergiemaschinen at the University of Dortmund. The methods used in the analysis are described in [3]. Rotor profile displacements caused by the thermal loads as computed in the finite element model are saved to a file which is read by the rotor clearance program. In addition, changes in the position of the rotors relative to each other, the misalignment of the two shafts, are determined for computed displacements of the housing parts. This information is then entered into the rotor clearance analysis program.

Summary

The analyses introduced here are similar to those reported in [1] and [4]. This work was done in cooperation with authors of these references and adapted to use for refrigeration compressors and the software tools available to us in our design department.

The basic approach to using a one-dimensional thermodynamic simulation and a complex finite element model of the compressor assembly is introduced in this part of the report. Boundary conditions can be generated quickly with the thermodynamic model. This advantage is used to generate numerous boundary condition definitions along the path of a compressor transient. Running the finite element analysis from one set to the next allows approximation of actual transient operation. This is a significant benefit of this work, gaining insight into the clearances during transient operation. Assemblies at steady state, regardless of operating temperature levels, tend towards clearances that are not much different from clearances in the cold assembled state. However, large excursions from this clearance condition are predicted during transients as the differing thermal inertias of rotors and housings result in different rates of thermal expansion leading to the clearance variations.

There is one area in this analysis process yet to be completed. The finite element analysis generates information about the heat exchange between the housings and the fluid in the heat flux calculations made for the temperature boundary conditions provided. This heat flux should be identical to that in the thermodynamic simulation which generated the boundary conditions. If not, the computed heat flux should be used for another iteration on the calculation of the boundary conditions, this process being repeated until the input and computed heat fluxes are the same.

Results are reviewed in Part II of this report. There, measured and computed surface temperatures are compared. Results of selected clearance analyses are also presented.

Acknowledgements

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Nomenclature

A	Area	m^2
c	Velocity	m/sec
Cp	Const. press. specific heat	J/kg-°K
d_h	Hydraulic diameter	m
k	Thermal conductivity	J/m-°K
L	Characteristic length	m
n	Rotational speed	sec^{-1}
Nu	Nusselt number	-
Pr	Prandtl number	-
$\dot{q}$	Heat flow	J/sec
r	Radius	m
Re	Reynolds number	-
T	Temperature	°K
x, y, z	Coord. system directions	m

α	Heat transfer coeff.	J/ sec-m^2-°K
β	Helix angle	rad
η	Dynamic viscosity	kg/m-sec
λ	Thermal conductivity	J/sec-m-°K
ρ	Density	kg/m^3
φ	Wrap angle	rad
ω	Angular velocity	sec^{-1}

Subscripts

Fl	fluid
r	rotor
u	tangential
w	wall
z	axial

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