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Transient Thermal Analysis of Screw Compressors, Part I: Use of Thermodynamic Simulation to Determine Boundary Conditions for Finite Element Analyses

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ABSTRACT

This paper is the first of a three part report on the transient thermal analysis of screw compressors. Knowledge of rotor-to-housing and rotor-to-rotor clearances is a fundamental first step in defining part dimensions that lead to a design with high efficiency and reliability, characteristics which must be balanced as they impose opposite demands on internal clearances. As reliability and performance are functional characteristics of the compressor, it is important that clearances be understood in the context of loaded operation. A proper assessment of the design can only be made if this understanding extends over the entire operating envelope of the compressor for both steady-state and transient operation.

A finite element analysis procedure has been developed to allow computation of rotor-to-rotor and rotor-to-housing clearances for a refrigerant screw compressor. In this first part of the report, we describe a model for calculating surface heat transfer coefficients and refrigerant bulk temperatures using a one-dimensional thermodynamic simulation. The result of this step is a set of heat transfer coefficients and gas temperatures defined on the rotor and housing surfaces. This information is then passed on to serve as boundary conditions in a finite element model. Development of the finite element modeling approach, analysis of rotor-to-housing clearances and analysis of the thermal effects on rotor-to-rotor clearances are discussed in Parts II and III of the report.

1. INTRODUCTION

One of the primary challenges in screw compressor design is achieving the low levels of clearance required for high efficiency while not allowing them to go so low as to result in damage or failure of the compressor. There are two major factors that make this task particularly difficult: variation in dimensions of the parts due to manufacturing variability and changes in the part caused by the pressure and temperature loads imposed during operation. This paper addresses the second of these issues, specifically the development of a process to compute the effects of thermal loads on operating clearances between the screw rotors and between the rotors and stationary housing parts.

Computations of temperatures of the rotor and housing parts and the resultant changes in size and shape of these parts are carried out using an ANSYS® finite element model. This requires that boundary conditions be defined on the surfaces of the model that are exposed to the fluid in and around the compressor. The approach chosen is to specify fluid temperature and heat transfer coefficients at the solid-fluid interfaces. Fluid properties are calculated using a proprietary thermodynamic simulation developed for use in the design and analysis of refrigeration screw compressors. This program computes the internal pressures and temperatures given details of the rotor geometry, clearances and flowpath. Using this model to supply boundary conditions for the finite element analyses requires the addition of two new capabilities: estimation of local heat transfer coefficients and a process for associating the information in the thermodynamic simulation with nodes in the finite element model. These topics are discussed in this paper. Finite element modeling and analysis results are reported in parts II and III of the report, Weathers, et al (2006) and Powell, et al (2006), respectively.

2. CALCULATION OF BOUNDARY CONDITIONS

An existing thermodynamic simulation of the refrigerant screw compressor serves as the basis for the process of determining appropriate boundary conditions for the finite element model. This program computes the pressure, temperature and mass in the volume – the working chamber – defined by the meshing screw rotors and adjacent stationary housing parts. Properties change as the compression process proceeds from inlet to discharge, the fluid being transported and compressed due to the reduction in the working chamber volume caused by the meshing of the rotor pair. At any time during this process, the properties of the fluid 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. The flutes in the rotors and the stationary housing parts (not shown) capture the fluid. As the rotors rotate, the volume of the chamber changes, at first increasing from the state shown in Figure 1 to a maximum value defining the physical displacement of the compressor. The volume then decreases to compress then discharge the refrigerant on the high pressure side to the left and on the bottom of the rotor pair as shown in the figure. 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. However, such a model was already developed by Kauder and Keller (1995). We decided to adopt the methods reported in the reference and develop modifications or adjustments as necessary as experience was gained in our oil-injected, refrigerant compressors.

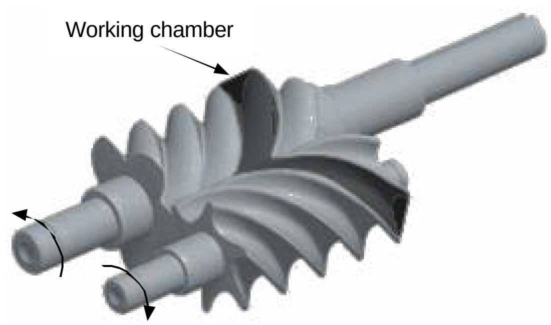


Figure 1 Rotor pair and working chamber

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

Examples of how these models are used for the screw compressor problem follow in the remainder of this section. Heat transfer coefficients for the surfaces of the screw rotor lobes are determined by employing the relationship for flow in a pipe as documented in Section 2.1. Section 2.2 contains an example of using the surface models for heat transfer at the compressor housing surfaces. The work reported is based on the approach taken in Kauder and Keller (1995). This work was presented to the project team in more detail in a seminar conducted by Kauder, Janicki and von Unwerth (2001).

2.1 Heat Transfer Coefficients on Rotor Housing Surfaces

Estimation of the heat transfer coefficient for the working chamber surfaces is 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:

$$\alpha = \frac{\dot{q}}{A \cdot \Delta T} \qquad (1)$$

This equation is used to solve for the heat flux for known values of α, which is computed using the definition of the Nusselt number, Nu:

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

After the suggestion of Kauder and Keller (1995), the calculation of the Nusselt number for the flow in the working chamber is adapted from that for flow in a pipe:

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

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

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

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

The viscosity ratio at the right hand side of equation (3) is assumed to be 1.0 after the proposal in the reference. However, due to the application of the models to a real gas environment in the refrigeration compressor and because the computational effort is low, the Prandtl number is retained as a variable.

Evaluation of equations 2, 3 and 4 for the working chamber requires selection of a characteristic length, L, hydraulic diameter, d_h, and velocity, c. Following the referenced analysis approach, the velocity relative to the stationary housing surfaces forming the working chamber is the vector sum of an axial velocity and the rotor tangential velocity. The axial velocity is determined from 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 velocity and the lead or wrap angle of the helical rotors. The calculation of the reference velocity is illustrated in Figure 2. The axial component c_z of the working chamber velocity is:

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

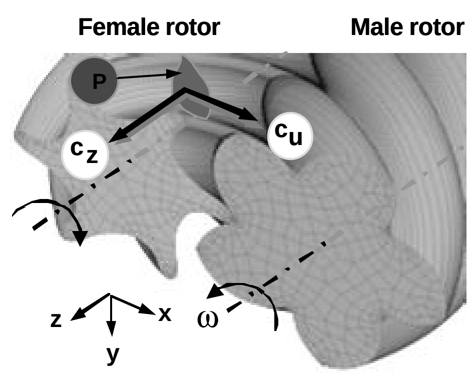


Figure 2 Reference velocity

The tangential velocity is computed using a reference radius r_m. This radius bisects the chamber cross sectional area (the 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).

We decided to use the overall rotor length for the reference length (L) term in equation 3. With this, the Nusselt number varies with the fluid properties for a given geometry and rotational speed as the Reynolds and Prandtl numbers are the only variables in the equation. These vary only with fluid properties as a result of the assumptions and simplifications introduced.

The thermodynamic simulation is set up as an initial value problem, solving a set of simultaneous differential equations to compute what happens to the flow as it traverses the compressor. At any point in time, 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 the fluid properties are uniform throughout the volume. This means that the fluid temperature and the calculated heat transfer coefficients will be the same at all surfaces enclosing the chamber at the particular time and rotor rotation angle being analyzed. If we consider one point on the stationary housing forming the outer boundary of the 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 part of the compression process seen by 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.

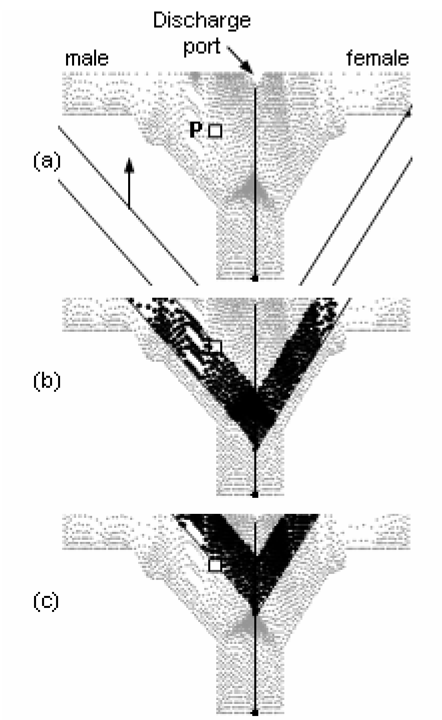


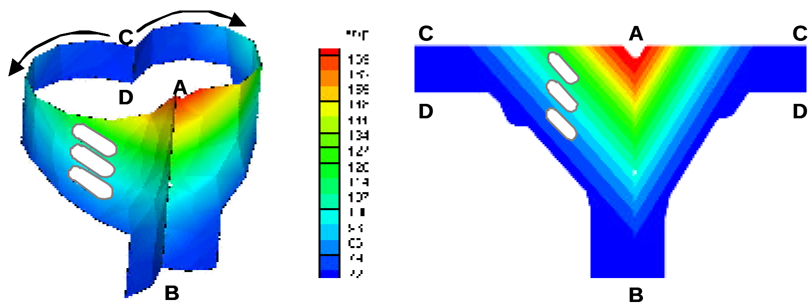
Figure 3. Rotor housing

Figure 3 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 white, square symbol designated “P” in Figure 3a. The illustrations show the rotor housing in an “unwrapped” view. The gray points in 3a are actually locations of nodes on the inner surface of the rotor housing from the finite element model used for the analysis. The housing surfaces are unwrapped around the high pressure cusp, the dark vertical line in the center (see Figure 4 for an illustration of the physical and unwrapped views compared). The inlet or low pressure end of the housing it at the bottom of the illustration; the high pressure end, with the discharge port as indicated, is at the top. The pairs of oblique lines are the unwrapped projections of the rotor lobe tips with the male rotor on the left, the female on the right. The working chamber we follow through the process exists between these lines. As the rotors rotate and the compression process proceeds from intake through to the discharge, the working chamber in Figure 3 moves from the bottom towards the top.

The inlet port for the compressor analyzed includes a section of the housing above the rotors that is relieved – a very large radial clearance. This is the white area with no symbols in the plots in Figure 3. Figure 3a shows a point in the process where the working chamber is filling as the rotors are fully in this area. After some rotor rotation, the working chamber has been carried to the position shown in Figure 3b. Here, the trailing lobes (lower lines) have passed the edge of the inlet port – the boundary between the open area and the area defined by the gray symbols. The chamber volume has reduced and the refrigerant is somewhat compressed. In the location shown in 3b, the leading lobe of the male rotor has arrived at the point P. We begin to track the fluid conditions seen at P at this point. As mentioned, the properties within the working chamber vary with time as the compression process progresses, but the properties are uniform throughout the chamber at any particular time during this process. All of the nodes on the housing that are within the working chamber are re-plotted in the figure with the larger, black symbols. All fluid properties and the resultant heat transfer coefficient are the same at each of these points for this rotor orientation. Figure 3c shows the process farther along in the compression phase. Now, the trailing lobe has reached point P. The trailing lobe in this case is in the same location as the leading lobe in 3b; thus, point P has seen one entire cycle of property variation and is now beginning to repeat the cycle.

This process occurs at every point on the housing and each point sees one-fifth of one complete rotation of the male rotor for this case of a 5-lobed male. The location of the working chamber at any point in the process is identified by the angular orientation of the leading lobe of the male rotor. The thermodynamic simulation produces a table of fluid properties with this rotation angle serving as the independent variable. The data is imported into an Excel® workbook where the computation of the heat transfer coefficients as described earlier in this section is carried out. Now, we have a table of rotor rotation angle and corresponding pressures, temperatures and heat transfer coefficients that need to be assigned to particular points on the housing. The process is that just discussed and illustrated in Figure 3. The workbook has a function that computes the male rotor rotation angle required to bring the leading lobe of the male rotor to the location of the point – point P in our illustration. This is the rotation of Figure 3b. The properties for the fluid from this rotation angle through to the rotation 72° later (for this 5 lobe rotor) are then averaged. These are the properties assigned to point P to be used as boundary conditions for the finite element model.

Figure 4 shows results from the thermodynamic simulation displayed on physical (4a) and unwrapped (4b) rotor bore surfaces. The low pressure cusp – the line of intersection of the housing bores – lies along CD. The housing surface is cut here and unwrapped as shown by the arrows shown in 4a. The unwrapped, flattened surface that results is shown in Figure 4b. The three openings shown on the male rotor side of the housing are unloader ports. Unloading for this compressor is described in Sauls (1998).



(a)
Physical Surface

(b)
Unwrapped Surface

Figure 4 Fluid temperatures at housing surface.

Contours of fluid temperature at the rotor bore surfaces computed, averaged and applied as discussed earlier in this section, are shown in the figure (scale in °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 bounding the working chamber at the housing surface.

The example illustrated in Figure 3 showed the housing points identified by coordinates of nodes from a finite element model. In its current state, the workbook used for computing heat transfer coefficients and assigning them to housing locations uses a selected set of nodes from the housing surface in the finite element model. The resulting table of coordinates and properties is written to a text file. A macro was developed within ANSYS to read this table and interpolate at actual node locations selected for the finite element model, meaning that the boundary condition assignment process does not have to be repeated outside of the ANSYS environment if the finite element model is modified.

2.2 Heat Transfer Coefficients on Rotor Surfaces

The process of computing temperatures and heat transfer coefficients for the rotor surfaces is the same as for the rotor bore surfaces. It was suggested by Kauder and Keller (1995) that the fluid velocity for use in calculations be the projection of the axial velocity c_z (equation 5) in the direction of the rotor helix. This adjustment can be made to arrive at a velocity c on which to base the Reynolds number. The representative length and diameter factors are the same as used in Section 2.1.

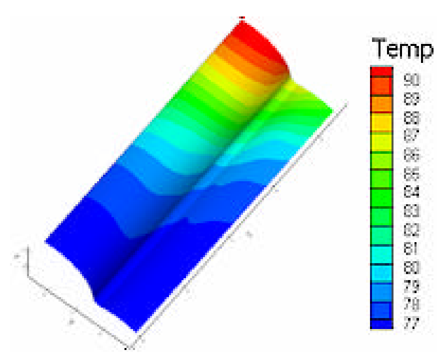


Figure 5. Male Rotor Temperatures

The pattern of boundary conditions computed for the rotor surfaces is illustrated in Figure 5. Here, a single male rotor passage is shown without the actual rotor helix – a zero wrap-angle representation. Plotted on the surface are contours of temperature (scale in °F). Again, the conditions are averaged over a period of one revolution. The shape of the pattern on the surface is dictated in part by the rotor mesh pattern. A point on the rotor inside of the working chamber will remain exposed to the conditions in that chamber until it reaches the point of meshing with a corresponding point on the opposite rotor. At this point, it is one of a locus of points that define the sealing line between the active working chamb er and another chamber on the low pressure side of the process. Further rotation moves the point into this other chamber. Points on the rotor profile near the inlet end of the rotors (lower left in Figure 5) are almost always exposed to inlet conditions, hence the large area at the nearly uniform low temperature. At the discharge end, however, a point will be exposed to conditions of the higher pressure end of the compression process and to the conditions during the discharge process. They are exposed to higher temperatures for a longer period of time, but do cycle back to exposure to inlet conditions for at least a short period of a rotor revolution. The highest temperature assigned to a rotor profile point is always less than the discharge gas temperature due to this averaging.

One interesting feature of the pattern on the male rotor is the fairly large difference in average temperature along the profile near the discharge end (upper right end in Figure 5). This is due to the nature of the rotor meshing resulting from the conjugate generation of the two profiles relative to each other. For this particular profile, the meshing results in the "round side", the left side in Figure 5, being exposed to the high pressure discharge gas for a fairly long period of time. The "straight side," on the right side in the figure, however, forms the high pressure end of a working chamber on the low pressure side for a correspondingly long period of time. Hence, even at the discharge end of the rotors, average rotor surface temperatures on the straight side are quite low.

3. TRANSIENT ANALYSES

An important application of the procedure is in studying what happens to clearances during transient operation. This could be the transition from the idle state to running operation, from an unloaded to a loaded state or from one operating condition to another as the air-conditioning unit adjusts to a changing environment. Two approaches are taken to supply and use boundary conditions for the finite element modeling. First, the simulation is simply run at the operating conditions of the end state. The initial conditions for the parts can be defined as being at a uniform temperature as if the compressor had been idle for a sufficient period of time for temperatures to have equalized at the level of the local ambient temperature. Or, the temperature of the parts can be as defined from a previous solution.

The second approach is really assembling a set of solutions that proceed from one condition to the next along a defined path of operating conditions…successive implementation of the simple first approach just described. This is done when there is a long transient which contains discreet events that change the "direction" of the unit operation. For example, we might go from an idle state to the compressor running in an unloaded state at a relatively low operating pressure ratio. Demand can result in a call to load the compressor, resulting in operation at a new and generally higher pressure ratio. Changes in the number of fans running in an air-cooled water chiller can also perturb the operating conditions.

To follow a path such as this, the thermodynamic simulation is run at conditions representing the key transition points. The finite element solution starts with an assumed initial state for all metal temperatures. The boundary conditions – fluid properties – associated with the end point of the first step are input and the solution is run for a specified period of time as defined by the unit's transient characteristic. At the end of this time step, the solution contains the computed transient variation in temperatures and resulting part deformations for this step. The solution at this point is then used as the initial state for a solution step defined by putting the boundary conditions fro m the next step in operating conditions onto the model. The process is repeated as often as necessary to simulate the transient path.

The process is described in more detail in Weathers (2006). Mechanics of modeling and an example of rotor to housing clearances computed during transient operation are provided in that report. Typical results of such an analysis are shown here in Figure 6. The clearance between a fixed point on the compressor housing and the rotating rotor is followed during the course of the operating transient. This example illustrates a key finding arising from the analyses: the critical issue in clearance control and hence operating reliability, is found in transient operation.

Differences in thermal inertia of the rotors and housings lead to different expansion rates, the result being changes in clearances. As the imposed operating conditions stabilize and the compressor parts approach their steady state temperatures, the clearance rises from the minimum, returning to almost the same level as at the start of the transient, even though the operating conditions are significantly different.

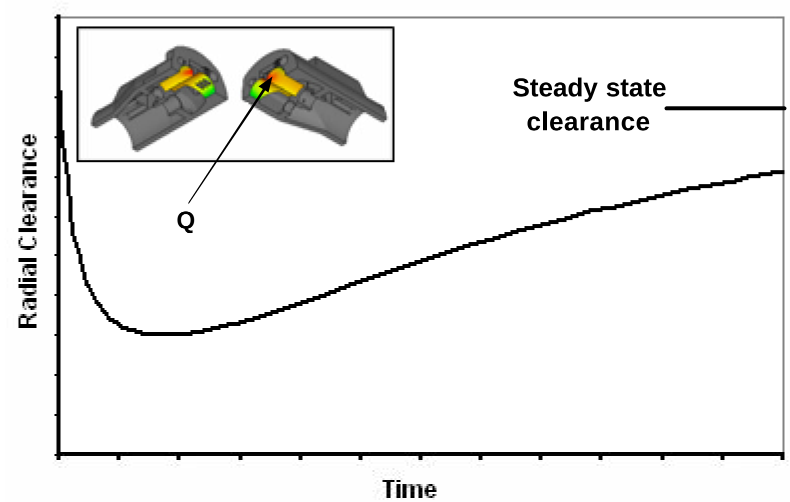


Figure 6. Example of Transient Analysis Results
Plot of clearance at point Q on male rotor bore

4. CONCLUSIONS

A model for the effects of thermal loading on screw compressor clearances during transient and steady state operation has been developed, following the work of Kauder and Keller (1995). Adaptation of their methods for computing finite element model boundary conditions to refrigeration screw compressor analyses is reviewed in this report.

The process uses a proprietary thermodynamic simulation for the compression process. Geometric and fluid properties used in the model are available for the computation of the heat transfer coefficients. The information is made available in tabular form to the finite element modeling step, Weathers, et al (2006). Execution of this detailed simulation and more discussion of results are found in this reference and the third part of this report in Powell, et al (2006).

The significant benefit of this work documented here along with the documentation in the other two parts of this report is the insight into the clearances during transient operation. Parts at steady state, regardless of operating temperature levels, tend towards the same clearances, often clearances that are not much different than they are 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 illustrated in the sample case in Section 3.

NOMENCLATURE

A	Area	m^2	α	Heat transfer coefficient	J/ sec-m^2-°K
c	Velocity	m/sec	ß	helix angle	rad
Cp	Const. pressure specific heat	J/kg-°K	η	Dynamic viscosity	kg/m-sec
d_h	Hydraulic diameter	m	λ	Thermal conductivity	J/sec-m-°K
k	Thermal conductivity	J/m-°K	ρ	Density	kg/m^3
L	Characteristic length	m	?	wrap angle	rad
n	Rotational speed	sec^{-1}	ω	Angular velocity	sec^{-1}
Nu	Nusselt number	-			
Pr	Prandtl number	-	**Subscripts**		
$\dot{q}$	Heat flow	J/sec	Fl	Fluid	
r	Radius	m	r	rotor	
Re	Reynolds number	-	u	tangential	
T	Temperature	°K	W	Wall	
x, y, z	Coordinate system directions	m	z	axial	

REFERENCES

Weathers, B., Powell, G. and Sauls, J., 2006, Transient Thermal Analysis of Screw Compressors, Part II -- Transient Thermal Analysis of a Screw Compressor to Determine Rotor-to-Housing Clearances, *Proceedings of the 2006 International Engineering Conference at Purdue*, Purdue University.

Powell, G., Weathers, B. and Sauls, J., 2006, Transient Thermal Analysis of Screw Compressors, Part III -- Transient Thermal Analysis of a Screw Compressor to Determine Rotor-to-Rotor Clearances, *Proceedings of the 2006 International Engineering Conference at Purdue*, Purdue University.

Kauder, K. and Keller, G., 1995, Wärmeübergangsrandbedingunen für Schraubenmaschinen, *Schraubenmaschinen – Forschunsberichte des FG Fluidenergiemaschinen Nr. 13*, Universität Dortmund: 5-19.

Kauder, K., Janicki, M. and von Unwerth, T., 2001, Workshop: Simulation of screw-type machines, *Private communication*, University of Dortmund.

Sauls, J., Design Of An Unloader System For Moderate Capacity Refrigeration Screw Compressors, *Proceedings of the 1998 International Engineering Conference at Purdue*, Purdue University: 583-588.

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