A Comprehensive Model of Scroll Compressors Part I: Compression Process Modeling

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Part I: Compression Process Modeling

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This paper presents a detailed model for the compression process of a scroll compressor, which will be used to investigate the compressor's performance under different operating conditions and subject to design changes. The governing mass and energy conservation differential equations were developed for each compressor chamber. Models for the refrigerant flow in the suction and discharge processes, radial and flank leakage, and heat transfer between the gas and scroll wraps were combined with the conservation equations. The state of the refrigerant changes with a period of angle 2π, and thousands of step are used to solve the governing differential equations during each period. It is assumed that in each step the compressor is in steady state. Since the differential equations for the different chambers are coupled, all these equations are solved simultaneously using a nonlinear equation solver. A description of the corresponding computer code and some results are included in this paper. Verification of the compression process model can be referred to that of the overall model, which is described in Chen et al (2000).

INTRODUCTION

The working principal of a scroll machine has been known since the beginning of this century when it was invented by Creux (1905). But it was not until the mid 1970's that it became possible to manufacture a working pair of scrolls due to the very small tolerances required. Since then, the scroll compressor has gained more and more popularity. The scroll compressor's inherent characteristics such as few moving parts, low level of noise and vibration, high efficiency and high reliability were soon recognized and attracted a lot of research and development. Compressor modeling has emerged from the different methods to investigate compressors almost as a research field of its own. Just about every individual process occurring in a compressor has been modeled such as the dynamics of the moving parts including frictional losses, the compressor process, heat transfer between the refrigerant and compressor parts, internal flow and leakage, oil transport and distribution and pulsating flow at suction and discharge. However, for scroll compressors, no complete model that includes all of these processes in one model has been published so far.

This paper and a companion paper (Chen et al, 2000) present the development and validation of a comprehensive scroll compressor model, which comprises most of the above mentioned individual processes and combines them into one overall model. This theoretical approach has been supported by experimental investigations on a macroscopic basis, i.e. measurement of mass flow rate, power consumption and discharge temperature. The model developed for this compressor allows for variation of the scroll geometry with single symmetric sets of suction, compression and discharge chambers. However, the model could be extended to allow for a variation of the number of compression chambers and asymmetries.

GEOMETRY STUDY OF SCROLL COMPRESSOR

The geometry of the scroll compressor is one of the main factors influencing the efficiency of the compressor. In order to establish a working mathematical model for the compressor, geometric characteristics have to be known and thoroughly understood.

Definition of Various Compressor Chambers

As shown in figure 1, different compressor chambers are referred to by numbers. The suction chamber with its suction port located directly next to the suction line inlet is labeled *chamber 1*; the suction chamber connected via the channel line is called *chamber 2*. *Chamber 3* is the compression chamber that developed from *chamber 1* and *chamber 4* is the one that developed from *chamber 2*. As the compression chambers open up to the discharge region, *chamber 3* becomes *chamber 5* and *chamber 4* becomes *chamber 6*. The innermost portion is labeled *chamber 7*. If the pressures in *chamber 5* and *chamber 7* have equalized, both chambers are treated as one control volume of *chamber 8*, and if the pressures in *chamber 6* and *chamber 7* have equalized, the control volume will be labeled *chamber 9*. In the case that the pressures in all three *chambers of 5, 6* and *7* have equalized, the entire control volume will be treated as *chamber 10* (see Halm, 1997).

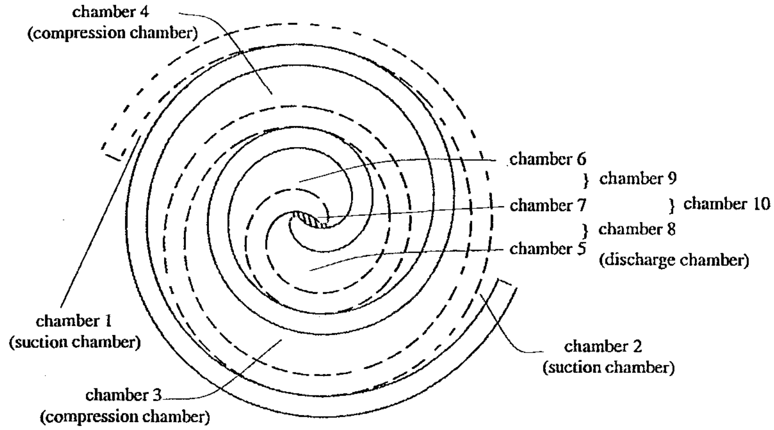

Figure 1. Definition of various compressor chambers

Calculation of Volumes of Various Compressor Chambers

Expressions for areas and volumes of different compressor chambers as a function of the orbiting angle were derived (Halm, 1997), since these areas and volumes and their derivatives determine the compression process. In addition, change of the volume with respect to the orbiting angle was calculated.

It should be noted that as the compression process proceeds, at a certain orbiting angle, compression chamber 3 and 4 will connect to the discharge chamber 5 and 6 and thus becomes the discharge region. This orbiting angle is called the discharge angle θ_d. At the angle θ_d, the compression process has just terminated and the discharge process will begin. The discharge angle is determined by the involute ending angle φ_e and the outer involute starting angle φ_{os} of the scroll. For the specific investigated scroll compressor, the discharge angle has the following form:

$$\theta_d = \varphi_e - 3\pi - \varphi_{os} \tag{1}$$

Discussion of the Chamber Volumes

The calculated volumes of the suction chambers, compressor chambers and discharge region are plotted as a function of the orbiting angle θ in figure 2.

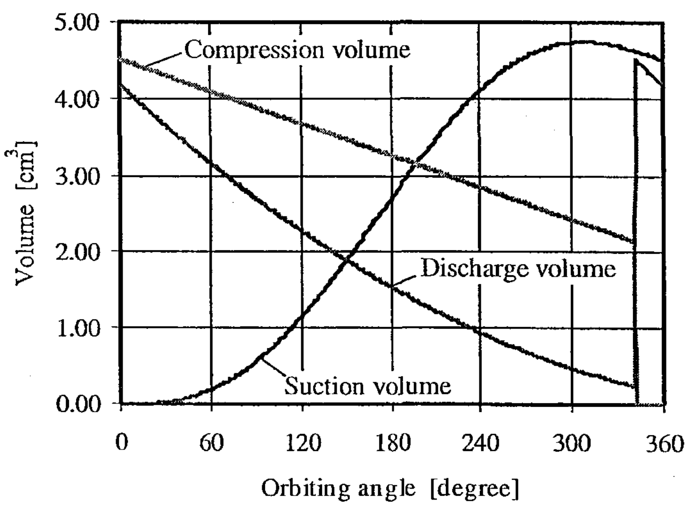

Figure 2 Suction, compression and discharge volume vs. orbiting angle

It can be seen that the suction volume increases until it reaches a maximum and then decreases until the suction chamber is closed. The change of the suction volume is positive initially and decreases afterwards and becomes negative, reflecting the decrease in suction volume as the suction chambers become more and more closed. The compression chambers exist only when the orbiting angle θ is less than the discharge angle θ_d of one entire shaft revolution. The change of compression chamber volume with respect to the orbiting angle is constant. It should be noted that the discharge region volume has a sudden volume increase at the discharge angle θ_d, due to the fact that the compression chambers open to the discharge region at θ_d and become part of the discharge region and therefore the integral boundaries change at this point.

Compressor Chamber Transition

The existence and number of compressor chambers changes during each revolution of the scroll. For example, compression chambers 3 and 4 connect to the discharge chamber and become the discharge region after the discharge angle. 9 distinct cases cover all the situations of the existence of the compressor chambers during one revolution, as shown in table 1. This classification facilitates and speeds up the compression process modeling.

Table 1 Definition of the cases of the scroll compressor chamber configurations

Case	Scroll Compressor *Chambers*										Involute Angle θ	Transition Condition
	1	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*		
1	x	x	x	x	x	x	x				$\theta \le \pi$	Pressures of *chamber 5, 6* and *7* are different
2	x	x	x	x	x				x			Pressures of *chamber 6* and *7* become the same, which are treated as *chamber 9*
3	x	x	x	x		x		x				Pressures of *chamber 5* and *7* become the same, which are treated as *chamber 8*
4	x	x	x	x						x		Pressures of *chamber 5, 6* and *7* become the same, which are treated as *chamber 10*
5	x	x	x	x						x	$\pi \le \theta \le \theta_d$	From $\theta = \pi$, the starting cross section of suction *chamber 2* is nearly closed. Pressure of *chamber 5, 6* and *7* are the same, which are treated as *chamber 10.*
6	x	x			x	x	x				$\theta \ge \theta_d$	Pressure distribution of *case 6, 7, 8,* and *9* is the same as that of *case 1, 2, 3,* and *4,* respectively. However, from the discharge angle $\theta = \theta_d$, compression *chamber 3* and *4* become discharge chamber 5 and *6*, respectively.
7	x	x			x				x			
8	x	x				x		x				
9	x	x								x		

Notes: Symbol "x" means that chamber exists in that case.

COMPRESSION PROCESS MODEL

To calculate the temperature, mass, and pressure of the refrigerant in various compressor chambers as a function of the orbiting angle θ, the discharge temperature/enthalpy, mass flow rate, and compression work for a certain suction state of the refrigerant, differential equations governing the compression process and models of the real gas were developed. In addition, other models had to be found and supplemented into the compression process model to reflect correctly the suction gas mass flow and heating, discharge process, leakage, and heat transfer of refrigerant in each of the compressor chambers.

Differential Equations Governing the Compression Process

The change of temperature, mass, and pressure in any of the above described chambers with respect to orbiting angle θ can be calculated from an equation based on the first law of thermodynamics for an open control volume in conjunction with a mass balance and an equation of state (Halm, 1997). The temperature change of the refrigerant as a function of the orbiting angle can be written as:

$$\frac{dT}{d\theta} = \frac{1}{mC_v}\left\{-T\left(\frac{dP}{dT}\right)_v\left[\frac{dV}{d\theta} - \frac{v}{\omega}\left(\dot{m}_{in} - \dot{m}_{out}\right)\right] - \sum\frac{\dot{m}_{in}}{\omega}(h - h_{in}) + \frac{\dot{Q}}{\omega}\right\} \tag{2}$$

and the mass balance is:

$$\frac{dm}{d\theta} = \sum\frac{\dot{m}_{in}}{\omega} - \sum\frac{\dot{m}_{out}}{\omega} \tag{3}$$

Equation (2) represents a first order differential equation with the independent variables of temperature T and mass m_{CV} which needs to be integrated numerically. Together with the mass balance equation (3) and equation of state of the refrigerant, it provides the desired relation to calculate the thermophysical properties in each compression chamber as a function of the orbiting angle θ. In order to evaluate equation (2), models that predict the mass flow rates into and out of each chamber as well as the heat transfer rate are required. Also, expressions for the change of pressure with respect to temperature at constant specific volume, $(dP/dT)_v$ and for the enthalpies h and h_{in} are required. These expressions can be derived from the equation of state.

Thermodynamic Properties

Refrigerant R22 is used for the investigated scroll compressor. For the real gas model, virial equations of state for R22 are used as described by Baehr and Tillner-Roth (1995). The relation of pressure and enthalpy as a function of temperature and density are given and the change of pressure and enthalpy with respect to temperature can be found by calculating the partial derivatives. In addition, specific heat at constant specific volume or pressure is also given.

Suction Gas Mass Flow

The suction volume increases with increasing orbiting angle θ until it reaches a maximum and decreases afterwards. During the increase of volume, gas is drawn into the chamber. The pressure in the suction chamber drops below the pressure of the suction gas due to expansion of the gas. After the suction volume has reached its maximum, compression of the gas starts, leading to a pressure rise in the suction chamber and a flow back towards the inlet. This effect leads to more refrigerant trapped in the suction chamber than a quasi-static process would predict and is referred to as the supercharging effect, increasing the volumetric efficiency of the compressor. The flow rate of the suction gas is calculated using the flow equation for isentropic flow of a compressible ideal gas, corrected by a flow factor as described by Fox and McDonald (1992):

$$\dot{m} = \psi A_s \sqrt{2 P_h \rho_h} \sqrt{\frac{\kappa}{\kappa - 1}\left[\left(\frac{P_l}{P_h}\right)^{\frac{2}{\kappa}} - \left(\frac{P_l}{P_h}\right)^{\frac{\kappa+1}{\kappa}}\right]} \tag{4}$$

Suction Gas Heating

As the suction gas flows from the accumulator to the suction chamber through the connecting pipe, the gas is heated by the pipe. This process is modeled using a heat transfer coefficient for turbulent flow in a constant cross section pipe of constant temperature. The heat transfer coefficient can be calculated by the following relation (Incropera and Dewitt, 1996):

$$h_c = 0.23 \frac{k}{d_p} \mathrm{Re}^{0.8}\, \mathrm{Pr}^{0.4} \tag{5}$$

The outlet temperature $T_{s,o}$ of the refrigerant can be calculated by

$$T_{s,o} = T_{pipe} - (T_{pipe} - T_{s,i}) \exp\left(- \frac{\pi\, d_p L_p h_c}{\dot{m} C_p} \right) \tag{6}$$

Discharge Process

The compression chamber remains closed until the orbiting angle becomes equal to the discharge angle θ_d. Shortly after opening up to the discharge region, the opening is so small that the flow is restricted and the pressures in the former compression chamber and discharge region do not equalize immediately. As long as the pressures have not equalized, one has to distinguish between the two regions of different pressure. Each region will be treated as an individual control volume and flow rates between these chambers have to be calculated. The flow rate between *chamber 5/6* and *chamber 7* is calculated using an isentropic flow model for an ideal compressible gas, equation (4). *Chamber 7* is always connected to the discharge port, and depending on the pressure in that chamber and on the discharge pressure, compressed gas will be discharged through the port and the check valve. As the discharge process progresses, the pressures in *chamber 5/6* and *chamber 7* will eventually equalize, and from that point the entire region of *chamber 5/6* and *chamber 7* can be treated as one control volume, namely *chamber 10,* as described by Halm (1997).

The discharge port is closed by a check valve to prevent back flow into the compressor during shutdown. The valve is modeled with only one degree of freedom. If the pressure in the discharge chamber exceeds the discharge pressure P_d, the valve opens. The distance y which the valve opens is determined with a static force balance according to:

$$y = (P - P_d)\frac{d^2}{4}\pi\frac{1}{C_{valve}} \tag{7}$$

The maximum distance that the valve can travel is determined by the valve stop. The flow area is then calculated by

$$A_{dis} = y\pi\, d \tag{8}$$

The flow through the discharge port is determined using the equation for isentropic compressible flow, equation (4). In this model, the dynamics of the valve motion have been neglected.

Leakage

There are two different paths for leakage in a scroll compressor. One is the path that is formed by a gap between the bottom or the top plate and the scrolls. This kind of leakage is called radial leakage. Another path is formed by a gap between the flanks of the two scrolls and is called flank or tangential leakage. In order to calculate the leakage rate from a higher pressure chamber to one with low pressure, the width of the flow path needs to be evaluated. The radial flow area between two chambers is given by the following:

$$A_{in} = \delta_r \int_{\varphi_k}^{\varphi_{k+1}} L_o d\varphi = \frac{1}{2}\delta_r r_b (\varphi_{k+1}^2 - \varphi_k^2) \tag{9}$$

where A_{in} is the effective flow cross section for the in-flowing refrigerant, φ_k and φ_{k+1} are the involute angles of the end points of the separating vane and δ_r is the gap height. For out flowing refrigerant

$$A_{out} = \delta_r \int_{\varphi_k}^{\varphi_{k+1}} L_i d\varphi = \frac{1}{2}\delta_r r_b (\varphi_{k+1}^2 - \varphi_k^2 - \varphi_{i0}\varphi_{k+1} + \varphi_{i0}\varphi_k) \tag{10}$$

where A_{out} is the effective flow cross section for the out-flowing refrigerant. The flow is modeled as an isentropic flow of an ideal compressible gas, equation (4).

For the flank leakage, the in- or out-flowing refrigerant flow areas are simply given by the product of the gap width δ_f and the height of the scroll vanes h:

$$A_{in} = A_{out} = h\delta_f \tag{11}$$

The gap sizes δ_r and δ_f are functions of the operating conditions, particularly of the ratio of discharge pressure and suction pressure P_d / P_s. Correlation of the gap size was provided by the scroll compressor manufacturer.

Heat Transfer

As the refrigerant is transported through the scroll compressor, it experiences heat transfer from the scrolls and the bottom and top plate. For this model, the convective heat transfer coefficient has been approximated with a correlation for a spiral plate tube heat exchanger. The curved shapes of the spiral plate heat exchanger with its rectangular cross section represent the crescent shaped chambers in a scroll compressor chamber. The correlation for the spiral heat exchanger is as follows (see Kakac and Shah, 1987):

$$h_c = 0.23\frac{k}{D_h}\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}\left(1.0 + 1.77\frac{D_h}{r_{aver}}\right) \tag{12}$$

The temperature distribution of the scroll is non-uniform and it is assumed that the temperature along the scroll wraps is linear with the involute angle φ of the scrolls (Jang and Jeong, 1999). The heat exchange rate between the refrigerant and the scroll walls / plates in any chamber can be calculated by an integral method according to the following equation:

$$\dot{Q} = h_c \int_A [T(\varphi) - T(k, j)]\, dA \tag{13}$$

IMPLEMENTATION OF COMPRESSION PROCESS MODEL

The compression process involves solution of the first order differential equations (2) and (3) in a step by step procedure. For this purpose, the Euler method as described by Conte and DeBoor (1980) is used to solve the first order differential equations so that at any orbiting angle θ_j

$$T(k,j) = T(k,j-1) + \left(\frac{dT}{d\theta}\right)_{\theta=\theta_{j-1}} \Delta\theta \tag{14}$$

$$m(k,j) = m(k,j-1) + \left(\frac{dm}{d\theta}\right)_{\theta=\theta_{j-1}} \Delta\theta \tag{15}$$

where $\frac{dT}{d\theta}$ and $\frac{dm}{d\theta}$ are the functions on the right hand sides of equation (2) and (3).

Since there are several compressor chambers, the solutions to several differential equations have to be solved simultaneously. The step by step procedure starts at orbiting angle $\theta = 0^o$, which corresponds to the step $j = 0$. It also starts with initial guess values of temperature and pressure for chambers 1,2,3,4,5,6 and 7, and the initial guess values of the average temperature of the scrolls. By applying the compressor geometrical model, the volume and the change of volume with respect to the orbiting angle of each chamber at the current orbiting angle can be calculated. Based on the values of temperature, pressure, and the calculated volume, mass and density for each chamber are calculated. The process proceeds by using the CASES routine to determine the case of the compressor chamber configuration (see Table 1) at the current orbiting angle, so that the existence of compressor chambers can be determined. If the case of the current step is changed from that of the last step, the REASSIGN routine needs to be called to reassign the properties of the compressor chambers for the case transition. The suction gas mass flow model and then discharge process model are applied to calculate the mass flow into the suction chambers and out of the discharge region, respectively. The leakage model is called to calculate the losses of the mass flow due to the two kinds of compressor leakage. The suction gas heating model and the heat transfer model are called to calculate the heat transfer of the suction gas through the suction pipe and the average heat transfer between the refrigerant in each chamber and the scroll wraps and plates. The calculated results obtained above are substituted into equations (2) and (3) to get the temperature and mass change with respect to the orbiting angle. By applying equations (14) and (15), the temperature and pressure of each chamber can be calculated for the orbiting angle of the next step. If the calculation has not gone through all the steps, it proceeds to the next step at $\theta_{j+1} = \theta_j + \Delta\theta$ and returns to the beginning. Once the calculation has gone through all the steps, the convergence of the compression process calculation is checked. The convergence criteria for the compression process are that the pressures and temperatures for all compressor chambers at $\theta = 2\pi$ equal to the value for $\theta = 0$ within specified tolerance (see Halm, 1997). If the criteria are not met within the tolerance, the initial guess values have to be corrected and the calculation for the compression process needs to be repeated. It was found that a successive substitution for determining pressures and temperatures at $\theta = 0$ provides a fast and accurate iteration scheme.

Mass Flow Rate / Average Discharge Temperature and Enthalpy / Compression work:

The mass flow rate is calculated as the amount of mass that is discharged from the discharge region divided by the time it takes for one entire revolution. Then, the mass flow rate is given by

$$\dot{m} = rps \sum_{j=1}^{MAXN} \dot{m}_{dis,j} \Delta\theta \tag{16}$$

The average enthalpy of the discharged refrigerant can be calculated by the following equation:

$$h_{dis} = \frac{\sum_{j=1}^{MAXN} \dot{m}_{dis,j} h_{dis,j} \Delta\theta}{\sum_{j=1}^{MAXN} \dot{m}_{dis,j} \Delta\theta} \tag{17}$$

The rate of work $\dot{W}_{compression}$ done on the refrigerant during the compression process can be calculated from an energy balance on the refrigerant across the pump assembly, i.e. the two scrolls only. The compression work can be calculated by

$$\dot{W}_{compression} = \dot{m}(h_{dis} - h_{suc}) - \dot{Q}_{average\_alum} - \dot{Q}_{average\_steel} \tag{19}$$

$\dot{Q}_{average\_alum}$ and $\dot{Q}_{average\_alum}$ are the average heat transfer rates from the aluminum scroll and steel scroll to the refrigerant, respectively, and are calculated by

$$\dot{Q}_{average\_alum} = rps \sum_{j=1}^{MAXN} \dot{Q}_{alum,j} \frac{\Delta\theta}{\omega} \tag{20}$$

$$\dot{Q}_{average\_steel} = rps \sum_{j=1}^{MAXN} \dot{Q}_{steel,j} \frac{\Delta\theta}{\omega} \qquad (21)$$

Coupling with the Overall Compressor Model

To solve the compression process model, the average temperature of the scrolls need to be known, which requires solving the overall compressor model. At the same time, the calculation results of the average discharge temperature and enthalpy of the refrigerant obtained from the compression process model are unknowns for solving the overall compressor model. Thus, the compression process model and the overall compressor model are coupled and need to be solved simultaneously. The overall compressor model is discussed by Chen et al (2000).

CALCULATION RESULTS

The detailed compression process model was used to calculate the refrigerant properties in the suction, compression, and discharge chambers as a function of the orbiting angle. The suction state of the refrigerant is at $497.40\,kPa$ and $10^{o}C$ and the discharged pressure is $1354\,kPa$. The pressure as a function of the orbiting angle was plotted in figure 3 for the refrigerant in suction chamber 1 until the refrigerant is discharged through chamber 7. From figure 3, it can be seen that for the orbiting angle of 0 to 360^{o}, initially the pressure in the suction chamber 1 is slightly lower than the suction line pressure and just before the suction chamber is entirely closed, the pressure in the suction chamber rises due to a decrease of suction pocket volume and flow resistance at the small suction port. It can be seen that the refrigerant stays in compression chamber 3 until chamber 3 becomes chamber 5 at the orbiting angle of 680.4^{o}. Chamber 5 becomes chamber 7 at the orbiting angle of 846^{o}. Refrigerant in chamber 7 keeps being discharged until the orbiting angle reaches 1054.8^{o} when chamber 5 is open to chamber 7. Shortly after chamber 5 opens to chamber 7, the pressure of the refrigerant in chamber 7 decreases because the pressure of chamber 5 is lower. Since the volume of chamber 7 decreases afterwards, the pressure of chamber 7 goes up again and is fully discharged after the orbiting angle of 1206^{o}.

The temperature of the refrigerant as a function of the orbiting angle was plotted in figure 4. It can be seen that the temperature of the refrigerant keeps increasing until the orbiting angel reaches 730.8^{o}. Afterwards, the temperature of the refrigerant decreases due to the heat transfer from the refrigerant to the compressor scroll wraps. It should be noted that there is a sudden temperature decrease for the refrigerant in chamber 7 at the orbiting angle of 1054.8^{o} when chamber 5 opens to chamber 7 due to the fact that temperature of chamber 5 is lower. Due to the further compression of the refrigerant and heat transfer between the refrigerant and the scrolls, temperature of the refrigerant in chamber 7 rises again and goes down afterwards.

Pressure as a function of the volume ratio of the refrigerant was plotted in figure 5. The volume ratio was defined as the volume of the chamber where the refrigerant is over the maximum volume of suction chamber 1. Pressure drop can be seen for the refrigerant in chamber 7 at a certain chamber volume due to the opening between chamber 7 and 5. Pressure then goes up as the volume decreases.

SUMMARY

A detailed compression process model of a scroll compressor was developed. The model is based on an approach using the first law of thermodynamics for open control volumes to calculate the instantaneous refrigerant states as a function of the orbiting angle. Individual processes within the compression process have been identified and models for these processes were developed. Refrigerant leakage and heat transfer with the scroll wrap were considered in developing the model. The model's main outputs at this point are the mass flow rate, the discharge temperature and work input. This model allows for the investigation of the influence of heat transfer, leakage and compressor geometrical parameters on the compressor's performance and can be used to improve the compressor's design in the future.

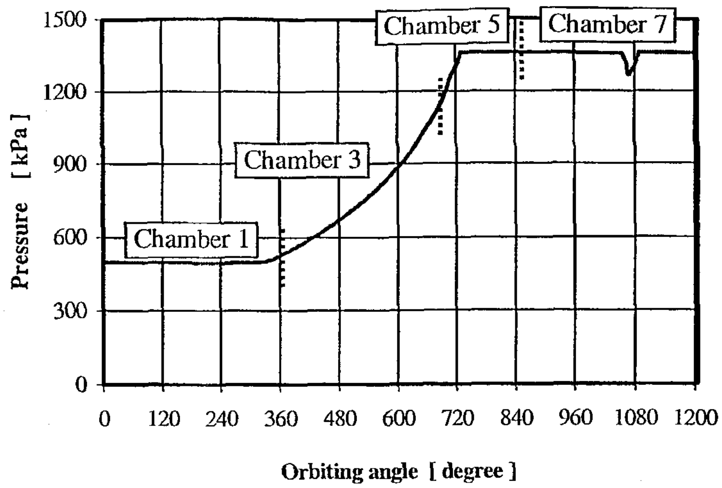

Figure 3 Pressure of the refrigerant as a function of the orbiting angle

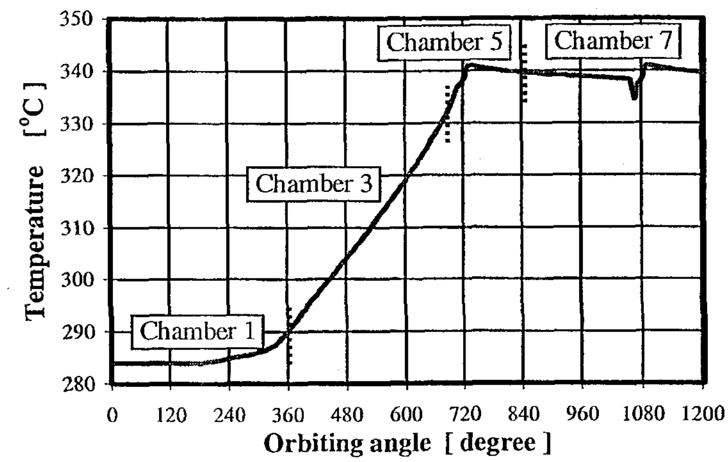

Figure 4 Pressure of the refrigerant as a function of the orbiting angle

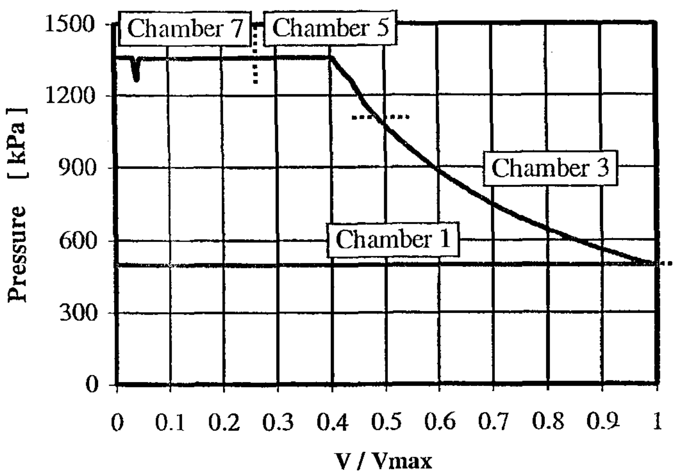

Figure 5 P-V diagram of the refrigerant

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NOMENCLATURE

Symbol	Description
A_{dis}	Discharge flow area
A_{in}	Effective flow cross section for the in-flowing refrigerant
A_{out}	Effective flow cross section for the out-flowing refrigerant
A_s	Flow area
C_p	Specific heat at constant pressure
C_v	Specific heat at constant specific volume
C_{valve}	Spring constant of the valve
d	Diameter of the discharge port
D_h	Hydraulic diameter
d_p	Inner suction pipe diameter
h	Height of scroll vanes
h	Specific enthalpy of the refrigerant in the control volume
h_c	Convection coefficient
h_{dis}	Average specific enthalpy of gas as discharged from pump assembly
$h_{dis,j}$	Discharge refrigerant enthalpy at the orbiting angle θ_j
h_{in}	Specific enthalpy of the refrigerant flowing into the control volume
h_{suc}	Specific enthalpy of the suction gas
k	Conductivity
L_p	Length of the pipe
m	Mass of the refrigerant
$m(k, j-1)$	Mass of the refrigerant in the k-th chamber at the orbiting angle θ_{j-1}

Symbol	Description
$\dot{m}$	Mass flow rate
$\dot{m}_{dis,j}$	Discharge mass flow rate in units of mass per radiant at the orbiting angle θ_j
$\dot{m}_{in}$	Mass flow rate into control volume
$\dot{m}_{out}$	Mass flow rate out of control volume
P	Pressure of the refrigerant
P_d	Discharge pressure of the refrigerant
P_h	Pressure of the refrigerant on the high pressure side
P_l	Pressure of the refrigerant on the low pressure side
Pr	Prandtl number
$\dot{Q}$	Heat flow rate into the control volume
$\dot{Q}_{alum,j}$	Heat transfer rate from the wall/plate of the aluminum scroll to the refrigerant at the orbiting angle θ_j
$\dot{Q}_{average\_alum}$	Average heat transfer rate from the wall/plate of the aluminum scroll to the refrigerant
$\dot{Q}_{average\_steel}$	Average heat transfer rate from the wall/plate of the steel scroll to the refrigerant
$\dot{Q}_{steel,j}$	Heat transfer rate from the wall/plate of the steel scroll to the refrigerant at the orbiting angle θ_j
r_{aver}	Average radius of compression chamber curvature
r_b	Radius of the basic circle of the scroll

Symbol	Description
Re	Reynolds number
rps	Number of rotations of the orbiting scroll per second
T	Temperature of the refrigerant
$T(k,j)$	Temperature of the refrigerant in the k-th chamber at the orbiting angle θ_j
$T(k,j-1)$	Temperature of the refrigerant in the k-th chamber at the orbiting angle θ_{j-1}
$T(\varphi)$	Temperature of the scroll at the involute angle φ
T_{pipe}	Temperature of the pipe wall
$T_{s,i}$	Temperature of suction pipe inlet
$T_{s,o}$	Temperature of suction pipe outlet
V	Volume of the control volume
$\dot{W}_{compression}$	Compression work
y	Discharge valve opening

Symbol	Description
δ_f	Flank leakage gap size
δ_r	Radial leakage gap size
ρ_h	Density of the refrigerant on the high pressure side
θ	Scroll orbiting angle
θ_d	Discharge angle
$\Delta\theta$	Step of the scroll orbiting angle
φ_e	Involute ending angle
φ_{i0}	Inner involute initial angle
φ_k	Involute angle at the k-th point of conjugacy
φ_{k+1}	Involute angle at the (k+1)-th point of conjugacy
φ_{os}	Outer involute starting angle
ψ	Flow factor
ω	Angular speed of compressor shaft