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Development of Liquid-Flooded Scroll Compressor and Expander Models

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ABSTRACT

The Liquid Flooded Ericsson Cooler (LFEC) is a gas refrigeration cycle that uses liquid flooding of both the compressor and the expander to achieve nearly isothermal compression and expansion processes as developed by Hugenroth et al. (2007). Efficiency of the LFEC strongly depends on the efficiencies of the compressor and the expander. It is hoped that the LFEC can be a commercially competitive cooling technology, but for these hopes to be realized, high efficiencies for the compressor and expander must be achieved. In order to optimize the design of the compressor and the expander, comprehensive models of both machines have been developed. Validation of the models and analysis of these machines are presented in two separate papers (Bell et al. 2008, Lemort et al. 2008). The models are mechanistic and allow investigation of the influence of design changes and operating conditions on performance.

1. INTRODUCTION

The model developed in this paper is an extension of earlier models that were developed by Halm (1997) and Chen (Chen et al. 2002, Chen et al. 2004) for scroll compressors that did not incorporate liquid flooding. The important contribution of this paper is the treatment of a mixture of gas and large fraction of oil. The first step in the analysis of a scroll compressor or expander is the calculation of the scroll chamber volumes as a function of the crank angle. In addition the derivatives of scroll chamber volumes with respect to the crank angle are necessary for mass and energy balances. The geometric model also predicts the evolution of the axial and radial leakage gaps, as well as the heat transfer areas.

Flooded compression and expansion processes are modeled by numerically solving the governing equations of conservation of mass and energy for each chamber within the scroll compressor or expander. These differential equations take into account boundary work, mass flows, and heat exchange and their solution yields the evolution of the mixture pressure and temperature in all of the chambers over one revolution. An overall model for either machine couples the solution to compression/expansion process to an overall network energy flow analysis that includes a set of lumped mass elements for which steady-state energy balance equations are established. The global model computes the mass flow rate displaced by the machine, the average discharge temperature and enthalpy, the shaft power and the efficiencies of the machine.

Experimental validation of these models has been conducted and is presented in two separate papers (Bell et al. 2008, Lemort et al. 2008). These two papers also describe modeling features unique to each machine. In the current study, the same automotive semi-hermetic scroll compressor is used as both the compressor and expander. Thus, the volumes of the scroll chambers will be the same but the definitions of the chambers will be different.

2. VOLUME CALCULATIONS

Analytic expressions for a scroll machine's chamber volumes as a function of crank angle have been previously derived. The basic equations describing the geometry of a scroll machine based on the involute of a circle were introduced by Creux (1905), and modern treatments of the scroll machine's geometry have been proposed by Halm

(1997) and Wang (2005). It is possible to use these expressions to accurately model the evolution of the volume in the suction and compression chambers of the compressor, but the discharge region of the compressor (suction region of the expander) is not amenable to analytic representation due to its complex geometry. Previous investigators modeled compressors that used simple tangent arcs to close the involute pairs that form each scroll in the discharge region. The scroll machine currently under investigation uses a set of two arcs and a line to close the involute pair as seen in Figure 1. Therefore numerical integration of the discharge region volume is required.

One of the critical parameters in matching compressors and expanders to their application is the volumetric ratio. This is given by the displacement volume of the compressor's suction chamber divided by the volume of the compression chamber right at the point where it is about to discharge into the discharge region. A volumetric ratio that is well matched to the operating pressure ratio will minimize under- or over- compression and expansion losses.

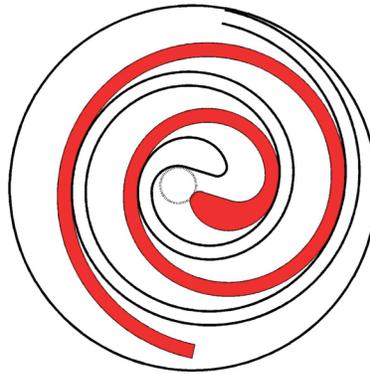


Figure 1: Scroll Machine Geometry

1.1 Identified Scroll Parameters

An optical scan of the compressor's orbiting scroll was obtained and an involute curve was fit to the points obtained from the scan. The thickness of the scroll is defined by $t=r_b(\phi_{i0}-\phi_{o0})$. Using the scan and the measured thickness, the scroll geometric parameters can be obtained, the values of which can be found in Table 1. The definitions of these geometric parameters follow those from Halm (1997).

Table 1: Scroll Geometric Parameters

r_b [m]	r_o [m]	ϕ_{i0} [rad]	ϕ_{is} [rad]	ϕ_{ie} [rad]	ϕ_{o0} [rad]	ϕ_{os} [rad]	h [m]
0.003522	0.006405	0.1983	4.7	15.5	-1.125	1.8	0.03289

3. MIXTURE PROPERTIES

For a homogenous mixture, the mixture specific volume is given by

$$v_m = x_l v_l + x_g v_g \quad (1)$$

The mixture specific internal energy u_m , mixture specific enthalpy h_m , mixture specific entropy s_m , and mixture constant pressure specific heat $c_{p,m}$ are defined in a similar way.

The homogeneous void fraction is the fraction of the mixture's volume that is gas. The void fraction α is given by

$$\alpha = \frac{x_g v_g}{x_l v_l + x_g v_g} \quad (2)$$

The mixture thermal conductivity is therefore defined as

$$k_m = (1 - \alpha)k_l + \alpha k_g \quad (3)$$

and the mixture viscosity is given as (Levy, 1999)

$$\mu_m = \left(\frac{x_l}{\mu_l} + \frac{x_g}{\mu_g} \right)^{-1} = \frac{\mu_l \mu_g}{\mu_g x_l + \mu_l x_g} \quad (4)$$

The mixture Prandtl and Reynolds Numbers are based on mixture properties and are given by

$$\text{Pr}_m = \frac{\mu_m c_{p,m}}{k_m} \quad (5)$$

$$\text{Re}_{D_h} = \frac{4\dot{m}}{\pi \mu_m D_h} = \frac{\rho_m U_m D_h}{\mu_m} \quad (6)$$

For the compression and expansion work chambers, there is no primary flow path, but the Reynolds number can still be defined based on the wall speed of the orbiting scroll and the hydraulic diameter of the chamber given by $D_h = 4V/A$.

As proposed by Hugenroth et al. (2007), a useful non-dimensional group that allows characterization of the performance of the flooded scroll machines is the ratio of capacitance rates of liquid and gas entering the compressor. This ratio, referred to as C_{ratio} , is given by

$$C_{ratio} = \frac{\dot{m}_{liquid} c_{liquid}}{\dot{m}_{gas} c_{p,gas}} \quad (7)$$

4. TWO-PHASE MASS FLOW

Numerous models are available to describe the flow of a two-phase mixture of liquid and gas through nozzles, orifices, and other flow elements. With respect to the pressure differential characteristic to suction, discharge and leakage flows, the model should account for the compressibility of the gas. Based on momentum conservation, Chisholm (1983) derived an equation for calculating the mass flow rate of a compressible flow of gas-liquid mixtures through an orifice. According to this equation, for a given pressure differential, the mass flow rate is given by:

$$\dot{m} = C_d A \sqrt{\frac{2 \int_{p_{low}}^{p_{high}} v_e dP}{v_{low}^2 - \sigma v_{high}^2}} \quad (8)$$

where σ is the ratio of downstream to upstream areas. The integral in Equation 7 is obtained using numerical integration. For suction, discharge and leakage flows, the downstream area can be treated as being infinitely smaller than that of the upstream area, and thus $\sigma \approx 0$.

Different expressions for the mixture effective volume have been proposed according the flow pattern (Chisholm 1983, Morris 1991). In the case that the gas and liquid phases are separate, the effective volume is given by

$$v_e = [x_g v_g + K(1 - x_g)v_l] \left[x_g + \frac{1 - x_g}{K} \right] \quad (9)$$

where the slip ratio is given by

$$K = \left(\frac{v_g}{v_l} \right)^{0.4/4} \quad (10)$$

In the case of flow with liquid entrainment in the gas phase, Chisholm (1983) proposed to calculate the mixture's effective specific volume by a momentum balance on a differential flow element. In the flow entrainment formulation of the effective mixture specific volume, the effective specific volume is given as

$$v_e = [x_g v_g + K_e(1 - x_g)v_l] \left[x_g + \frac{1 - x_g}{K_e} \right] \quad (11)$$

and the effective slip ratio is given by

$$K_e = \left[\psi + \frac{(1-\psi)^2}{K-\psi} \right]^{-1} \quad (12)$$

where ψ is the fraction of liquid that travels in the gas phase at the gas velocity. The entrainment slip ratio K is evaluated from

$$K = \psi + (1-\psi) \sqrt{\frac{1 + \psi[(1-x_g)/x_g]v_l/v_g}{1 + \psi(1-x_g)/x_g}} \sqrt{\frac{v_g}{v_l}} \quad (13)$$

In the case that there is no entrainment (the oil and gas flow separately), the liquid entrainment model reduces to the separated flow model. In the case that the oil travels at the same velocity as the gas phase, the homogeneous flow model is obtained. In this case, the effective specific volume is given by Equation 1.

The two-phase discharge coefficient C_d is obtained by applying the two-phase flow model presented by Morris (1991), which yields a nominal value for C_d of 0.77. At each step of the rotation, the instantaneous flow rates between all of the control volumes are determined using this model. These flow rates are then used in the integration of the pressure and temperature differential equations.

5. LEAKAGE

As in any scroll compressor or expander, the leakage paths allow higher pressure gas to leak back to lower pressure compression chambers. This leakage will tend to decrease machine efficiency due to the exergetic destruction associated with this flow. In a typical scroll compressor or expander, the oil mass fraction is small, but for the flooded compression and expansion processes, the oil mass fraction is quite large. Therefore the leakage losses in the scroll machine will be dependent on the leakage gap width as in a typical scroll machine, as well as the leakage mixture composition. Treatment of the leakage gaps for the compressor and the expander are described in the companion papers. From these results, it is shown that the scroll expander leakage gap is dependent on the machine rotational speed.

The model considered for treating the flank leakage flow in the expander was proposed by Huagen et al (2004). In this model, the leakage is treated as a two-phase layer flow. The gas mass flow rate is described by classical equations for the isentropic flow through a nozzle (Halm, 1997). From these equations, the gas velocity C_g at the nozzle throat can be computed. The gas flow rate can be written as

$$\dot{m}_g = \alpha A_{leak} C_g / v_g \quad (14)$$

The oil leakage mass flow rate is given by

$$\dot{m}_l = (1-\alpha) A_{leak} \frac{C_g}{v_l K} \quad (15)$$

The slip ratio is estimated by

$$K = 0.4 + 0.6 \left(\frac{v_g}{v_l} + 0.4 \frac{x_l}{1-x_l} \right)^{\frac{1}{2}} \left(1 + 0.4 \frac{x_l}{1-x_l} \right)^{\frac{1}{2}} \quad (16)$$

6. HEAT TRANSFER

Heat transfer in the scroll machine is complex, and is simplified by condensing the many elements of the compressor and expander into lumped masses for which steady-state conservation of energy is imposed. For the machine considered in this study, two lumped masses were employed: one that contains the scrolls and another that is the shell. Further description of the heat transfer processes particular to the compressor or expander is available in the respective companion paper.

6.1 Suction and Discharge Heat Transfer

The scroll machine under investigation is a semi-hermetic automotive compressor. As the gas enters the machine, the gas exchanges heat with the shell of the machine, and then the gas passes into the suction region. For the

compressor, the suction region is a channel that allows gas to flow to the opposite side of the compressor and equalize the suction chamber pressures for balanced compression. For the expansion process, the gas flows through the scroll machine in the opposite direction, and thus the inlet for the expander is what would normally be the discharge plenum of the compressor. For both machines, the suction and discharge heat transfer coefficients can be obtained from the classic Dittus-Boelter turbulent pipe-flow correlation given by

$$h_c = 0.023 \frac{k_m}{D_h} \text{Re}_{D_h}^{0.8} \text{Pr}_m^{0.4} \quad (17)$$

6.2 Scroll-Gas Heat Transfer

In the scroll machine, the scrolls are in general at a different temperature than the gas that is in contact with them. Thus as a result there will be heat transfer between the scrolls and the gas in the chambers. For the compressor, mechanical losses will tend to heat the scrolls resulting generally in heat transfer from the scrolls to the gas, but this will depend on the exact operating conditions. The calculation of the local heat transfer coefficient at the scroll surface is obtained by adjusting the Dittus-Boelter turbulent pipe flow correlation with correction factors for the oscillating nature of the flow and the increase in heat transfer from the spiral shape of the flow path. The adjusted heat transfer coefficient is obtained using correction factors from Tagri (1962) and Jang and Jeong (2006) and is given by

$$h_c = 0.023 \frac{k_m}{D_h} \left(1.0 + 1.77 \frac{D_h}{r_{avg}} \right) (1 + 8.48 [1 - \exp(-5.35 St)]) \text{Re}_{D_h}^{0.8} \text{Pr}_m^{0.4} \quad (18)$$

where the Strouhal number St is given by $St = f A_{max} / U$ and in the case of the work chamber, $f = \omega / 2\pi$, $U = r_o \omega$, $A_{max} = r_o$, and $St = (2\pi)^{-1}$.

7. CONSERVATION LAWS

To obtain expressions for the derivatives of temperature and pressure of a given control volume, conservation of mass and energy must be expressed in differential form. If the assumption is made that the mixture composition is invariant throughout the scroll machine, then the mixture properties can be uniquely defined using the independent variables temperature, pressure, and mixture liquid fraction.

Conservation of mass and energy for a control volume can be given as:

$$\begin{aligned} \frac{dm_{CV}}{dt} &= \sum \dot{m} \\ \frac{dE_{CV}}{dt} &= \dot{Q} + \dot{W} + \sum \dot{m}h \end{aligned} \quad (19)$$

If these conservation laws are then converted into derivatives of temperature and pressure with respect to the crank angle θ , a system of coupled differential equations results as seen in Equation 20

$$\begin{aligned} \frac{dT}{d\theta} &= - \frac{PdV - \dot{Q} / \omega + \sum [\dot{m} / \omega (u_{CV} - h)]}{\rho V \frac{du}{dT}} \\ \frac{dP}{d\theta} &= \frac{\frac{d\rho}{dT} PdV - \frac{d\rho}{dT} \dot{Q} / \omega - \frac{du}{dT} dV \rho^2 + \sum \left[\dot{m} / \omega \left(\frac{du}{dT} \rho + \frac{d\rho}{dT} u_{CV} - \frac{d\rho}{dP} h \right) \right]}{\rho V \frac{du}{dT} \frac{d\rho}{dP}} \end{aligned} \quad (20)$$

The remaining property derivatives can be simplified if the assumption is made that the gas is ideal and the liquid is incompressible, as was assumed in the modeling presented here. Otherwise the derivatives can be evaluated with numerical differentiation.

8. SOLUTION METHOD

The volumes are calculated by numerical integration, and thus the geometric model is decoupled from that of the thermodynamic calculations. This allows for more efficient model execution, as the scroll chamber volumes can be calculated once and then repetitively used in the thermodynamic model.

Once the volumes have been calculated, the temperature and pressure of the scroll machine chambers at the starting angle are initialized with guess values. Then for each iteration step, the mass flows are calculated between all of the chambers, as well as the instantaneous heat transfer between the scrolls and the scroll chambers. The temperature and pressure at the next step are obtained with the forward Euler method by simultaneously applying

$$\begin{aligned} T_{i+1} &= T_i + \Delta\theta \left. \frac{dT}{d\theta} \right|_i \\ P_{i+1} &= P_i + \Delta\theta \left. \frac{dP}{d\theta} \right|_i \end{aligned} \quad (21)$$

Additional accuracy, and/or less restrictive requirements on $\Delta\theta$ for solution stability could be obtained by using implicit or higher order solution methods, but the forward Euler solver gives sufficient accuracy and stability for the current application.

9. MECHANICAL LOSSES

In the absence of a mechanistic model for mechanical losses, it is possible to correlate the mechanical losses to the gas compression power and obtain a relationship for the mechanical efficiency. This correlation is constructed by first running the model using input state points for which experimental data is available, with no mechanical losses or heat transfer. The residual between the model predicted shaft power and the experimental shaft power is then assumed to be due to mechanical losses. A correlation of the form

$$\dot{W}_{loss} = a_{loss} + b_{loss} \dot{W}_{gas} \quad (22)$$

is fit to the data and the coefficients a_{loss} and b_{loss} are then obtained. With the addition of heat transfer and mechanical losses, the gas compression power can be obtained by subtracting the heat transferred to the gas from the total power. For the compressor the rotational speed was held at a constant 3500 RPM, but for the expander the rotational speed was varied and thus the mechanical losses are a function of both gas compression power and rotational speed. This can be described by expressing the constant losses a_{loss} as a function of the expander speed ω and a fictitious friction torque T_{loss} .

$$a_{loss} = \omega T_{loss} \quad (23)$$

10. MODEL CLOSURE

Once the pressure and temperature have been determined for one revolution, the energy balance for the compressor or expander scrolls and shell are obtained. The energy balance for these elements is a summation of the heat fluxes interacting with the lumped masses. For the shell and scroll energy balances, the mechanical losses must be taken into account. Once the mechanical losses have been calculated, 70% of the heat generated from the mechanical losses is added to the scroll lumped mass and the other 30% is added to the shell lumped mass. Finally the temperatures of the shell and scroll lumped masses are obtained by solving the system of heat balances in Equation 24 for the shell and the scroll temperatures.

$$\begin{aligned} \sum_{scroll} q(T_{shell}, T_{scroll}) &= 0 \\ \sum_{shell} q(T_{shell}, T_{scroll}) &= 0 \end{aligned} \quad (24)$$

These temperatures are obtained by applying Newton's Method to this system of coupled non-linear equations. After solving for T_{shell} and T_{scroll} , the outlet enthalpy from the scroll set can be determined by calculating the mean discharge enthalpy over one revolution from

$$h_{disc} = \frac{\sum_{discharge} \dot{m}h}{\sum_{discharge} \dot{m}} \quad (25)$$

Solving for the temperature as a function of scroll set discharge enthalpy gives the temperature at the outlet of the scroll set. Further heat transfer in the discharge region will heat or cool the discharging flow, and will result in discharge temperature T_d at the outlet of the scroll machine. In the case of overexpansion or undercompression there will be back flow as the work chamber opens to the discharge pressure. The shaft power of the machine is then

given by

$$\dot{W}_{shaft} = \dot{m}(h(T_d, P_d) - h_{suct}) - \dot{Q}_{amb} \quad (26)$$

11. EXPERIMENTAL METHODS

The flooded scroll machines were installed in a cycle test rig as seen in Figure 2 to measure flooded Ericsson cycle performance. With this setup, sufficient control was available to run a variety of state points for the compressor and expander. The compressor suction pressure was changed by adjusting the charge level in the rig. The rotational speed was imposed using variable frequency motor controllers, and the high pressure was imposed by the rotational speed of the expander. Some control over the suction temperature was available, but in general, the suction temperature was dependent on the other inputs. The oil flow rates on the hot and cold sides were calculated using energy balances over the compressor and expander respectively, and the oil flow rates were controlled by adjusting the rotational speeds of the motor and pump.

The gas flow is determined using a Coriolis mass flow meter, and both oil separator gas outlet flows were seen to be very dry and assumed to be liquid-free. Due to thermal non-equilibrium for the heat exchanger flows, the liquid mass flows are calculated from energy balances over the compressor and expander. The mass flow of the aqueous ethylene glycol mixture used as a heat transfer fluid in the heat exchangers is also calculated using a Coriolis mass flow meter. The shaft torque is measured with a torque cell and the rotational speed is given from the VFD drive.

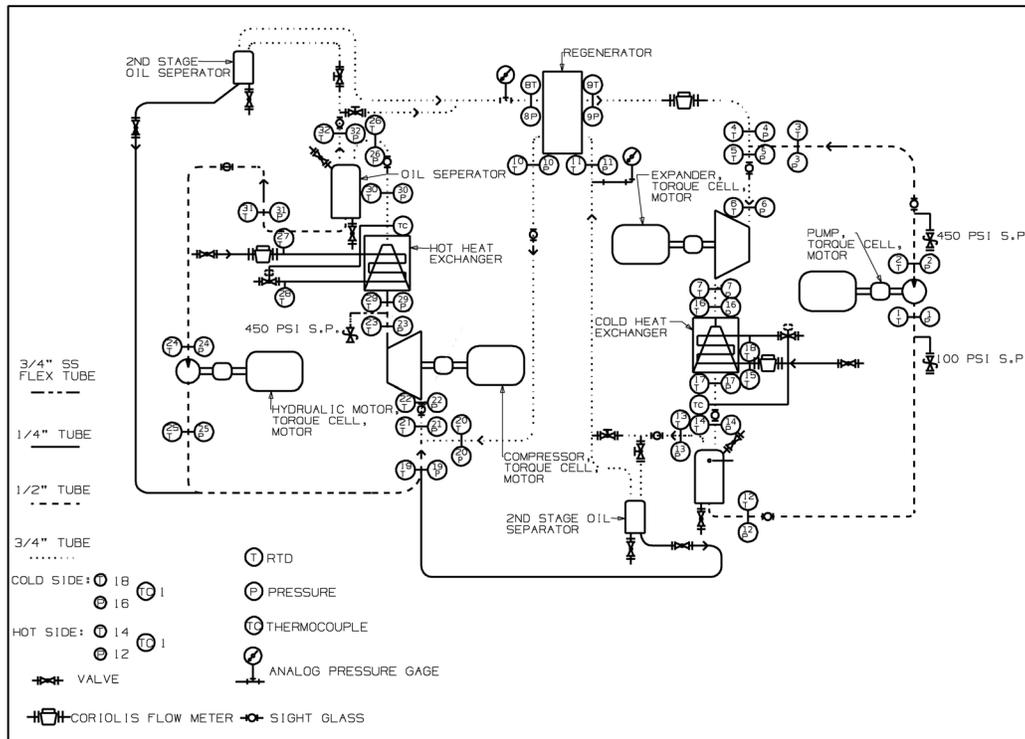


Figure 2: Schematic for Ericsson Cycle Test Rig

In theory, the energy balance used to obtain the oil flow rates could have been carried out over the heat exchangers as well. In addition, the heat exchanger energy balance requires four temperature measurements and the compressor heat balance only two. The dominant source of uncertainty in the heat balance is temperature measurement, so the fewer temperature measurements needed the better. One of the major challenges with using the heat exchanger energy balance is that the two phase oil/gas flow is not in thermal equilibrium. For instance, the difference in temperature between the outlet of the hot heat exchanger and the inlet to the hot side separator, which is only half a

meter downstream ranged from +2K to -2K. For the compressor, the outlet temperature does not appear to suffer from the same problems, likely due to the discharge plenum which helps to mix the gas and liquid phases.

The experimental studies were carried out using Zerol 60, an alkyl-benzene refrigeration oil, as the flooding agent and Nitrogen as the working gas. While other working gases, namely Argon would provide improved cycle performance, the use of nitrogen in the test rig was advantageous in terms of supply and operation.

12. RESULTS

While model validation is available in the companion papers, a few results are presented here. Figure 3 and Figure 4 show the scroll machine pressure-volume relationship for one revolution of the scroll compressor and expander respectively. A few features should be noted, especially the suction pressure drop for both machines and the maladjustment of the current machines' volumetric ratios for the imposed pressure ratio. Clearly the off-the-shelf scroll compressor is not optimal for the application but exhibits fairly good performance. In addition, the compressor's discharge region overpressure due to the inability to discharge the mixture sufficiently quickly results in excess compression power.

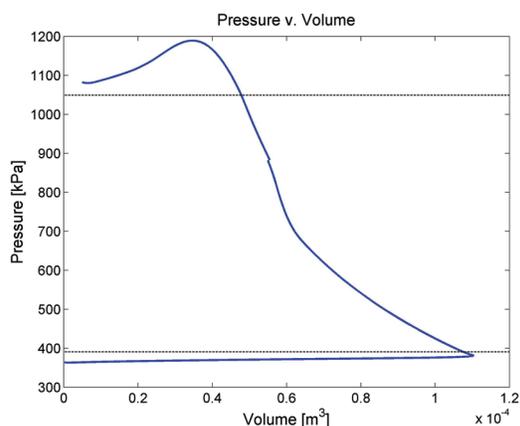


Figure 3: P-V plot for Compressor
 $T_s=40\text{C}$, $P_s=391\text{ kPa}$, $P_d=1049\text{ kPa}$, $C_{ratio}=5.72$

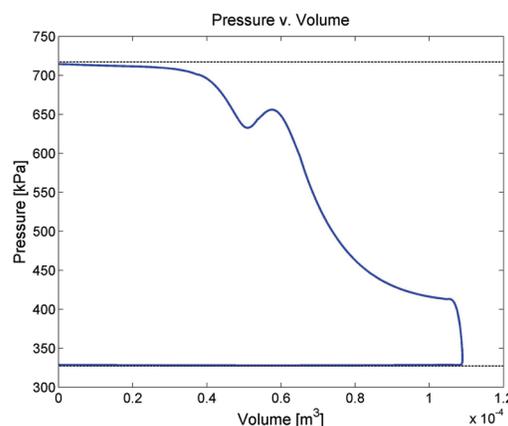


Figure 4: P-V plot for Expander
 $T_s=19\text{C}$, $P_s=717\text{ kPa}$, $P_d=327\text{ kPa}$, $C_{ratio}=3.05$

12. CONCLUSIONS

Detailed flooded scroll compressor and scroll expander models have been developed. The scroll machine model consists of the following sub-models:

- A volume calculation routine which permits numerical calculations of scroll chamber volumes as a function of the crank angle
- A two-phase mass flow model to calculate the instantaneous mass flow rates between scroll chambers
- An adjusted heat transfer coefficient model suitable for calculating the heat transfer between gas and scrolls
- A differential equation solver that allows the temperature and pressure in the control volumes to be calculated as a function of crank angle
- An overall solver function that obtains the temperatures of the lumped masses and calculates the relevant performance parameters

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NOMENCLATURE

<i>Variable</i>	<i>Definition</i>	<i>Units</i>	<i>Variable</i>	<i>Definition</i>	<i>Units</i>
A	Area	m ²	Re	Reynolds Number	-
C	Velocity	m/s	s	Specific Entropy	kJ/kg-K
C _d	Discharge Coefficient	-	St	Strouhal Number	-
C _{ratio}	Capacitance Rate Ratio	-	t	Time	s
c _p	Specific Heat	kJ/kg-K	T	Torque	N.m
D _h	Hydraulic Diameter	m	T	Temperature	K
E	Energy	kJ	u	Specific Internal Energy	kJ/kg
G	Mass Flux	kg/m ² -s	V	Volume	m ³
h	Specific Enthalpy	kJ/kg	x	Mass Fraction	-
h _c	Local Heat Transfer Coefficient	kW/m ² -K	α	Void Fraction	-
k	Thermal Conductivity	kW/m-K	φ	Involute Angle	rad
K	Slip ratio	-	v	Specific Volume	m ³ /kg
m	Mass	kg	ψ	Entrainment Fraction	-
\dot{m}	Mass Flow	kg/s	ρ	Density	kg/m ³
P	Pressure	kPa	σ	Ratio Down/Upstream Area	-
Pr	Prandtl Number	-	μ	Viscosity	Pa-s
			ω	Rotational Speed	rad/s

<i>Subscript</i>	<i>Definition</i>
CV	For the given control volume
e	Effective
g	Gas
i	Iteration Index
i0	Inner Involute Initial
l	Liquid
m	Mixture
oe	Outer Involute Ending
os	Outer Involute Starting
o0	Outer Involute Initial
shell	Shell Lump Mass
scroll	Scroll Lump Mass

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