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Liquid Flooded Compression and Expansion in Scroll Machines – Part II: Experimental Testing and Model Validation

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Abstract

The use of liquid-flooding in the compression and expansion of non-condensable gas in scroll compressors and expanders enables the possibility of quasi-isothermal working processes. Liquid-flooded scroll machines were installed in a fully-instrumented Liquid-Flooded Ericsson Cycle test rig to conduct entire cycle performance tests. In addition, detailed compressor and expander performance data was obtained. Oil mass fractions of up to 92% and 76% were added to the gas entering the scroll compressor and expander respectively. The overall isentropic efficiency of the scroll compressor based on the shaft power with flooding was up to 73% and the volumetric efficiency was above 92%. For the expander, the best overall isentropic and volumetric efficiencies achieved were 66% and 105% respectively. The mechanistic model presented in the companion paper was validated against the experimental data for both the compressor and the scroll expander with good agreement, though the agreement is better for the scroll compressor.

Key words:

scroll compressors, scroll expanders, liquid flooding, isothermal compression, high efficiency

1. Introduction

In a four-component gas-phase refrigeration cycle, gas is compressed in the compressor, cooled at a constant high pressure in a heat exchanger, expanded in the expander, and finally, the low-pressure, cooled gas warms up at a constant low pressure, providing the cooling effect. The Liquid-Flooded Ericsson Cycle provides a few modifications on top of the basic configuration of the gas refrigeration cycle. While the Liquid-Flooded Ericsson Cycle also adds a regenerator to exchange heat between the hot and cold sides of the cycle, the most significant change is the addition of liquid loops for both the hot and cold sides of the cycle. The liquid loops allow for quasi-isothermal operation of the compressor and expander, which in the absence of pressure drop and other losses, yields the ideal Ericsson cycle which is composed of four processes: isothermal compression, isobaric heat rejection, isothermal expansion, and isobaric heat addition. If all components perform reversibly, the efficiency of the Ericsson cycle is equal to the Carnot cycle.

Prior experimental work on the Liquid-Flooded Ericsson Cycle (LFEC) has been conducted by Hugenroth (2006; 2007), who developed a first experimental prototype of

the LFEC system. During the course of the studies, it was determined that a relocation of the heat exchangers from the oil loops to the gas loop could provide improved performance (LFEC 2 configuration). The system presented here is an embodiment of the LFEC 2 modification. In his studies, Hugenroth noted that the experimental performance of the scroll machines was poor, and suggested from simplified cycle modeling that in order to achieve a COP of the system of 1.25, overall isentropic efficiencies of 87% would be required for all rotating machinery. The work presented here is a step towards achieving the goal of scroll machines with these extremely high efficiencies. Further analysis of the potential for redesign of the scroll compressor for liquid flooding is presented in Bell et al. (2012a).

2. Experimental Methods

The flooded scroll machines were installed in a cycle test stand as seen in Figure 1 to measure the performance of the Liquid-Flooded Ericsson Cycle. Both the scroll expander and the scroll compressor were the same model of automotive scroll compressor seen in Figure 2. While the primary goal of the testing was to provide system-level data on the performance of the LFEC, detailed data was also available on the performance of the flooded scroll compressor and flooded scroll expander. The system was charged with dry nitrogen as the working fluid, and alkyl-benzene refrigeration oil (Zerol 60) as the flooding liquid. Though other

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Nomenclature

X_d	Area correction factor
h	Specific Enthalpy (kJ kg^{-1})
M	Calculated value (varies)
$\dot{m}$	Mass Flow (kg s^{-1})
N	Rotational Speed (rev min^{-1})
p	Pressure (kPa)
s	Specific Entropy ($\text{kJ kg}^{-1} \text{K}^{-1}$)
T	Temperature (K)
$\overline{UA}_{amb}$	Ambient heat transfer conductance (kW K^{-1})
V_{disp}	Displacement Volume (m^3)
$\dot{W}$	Shaft Power (kW)
x	Mass Fraction
δ	Gap Width (-)
η	Efficiency (-)
ε	Effectiveness
ϵ	Absolute Uncertainty (varies)
ψ	Entrainment Fraction
ρ	Density (kg m^{-3})
σ	Ratio Down/Upstream Area
ω	Rotational Speed (rad s^{-1})
τ	Torque (N m)

Subscripts

amb	Ambient
$comp$	Compressor
$discharge$	Discharge
exp	Expander
f	Flank
g	Gas
$high$	High-side
i	Isentropic
in	Inlet
int	Internal
l	Liquid
low	Low-side
m	Mixture
$meas$	Measured
mix	Of the mixture
out	Outlet
r	Radial
$shell$	Shell
v	Volumetric
Υ	Parameter under consideration

Abbreviations

HX	Heat Exchanger
RPM	Revolution per minute

Figure 2: Photo of scroll compressor employed in this study.

fluid combinations could have been used to achieve superior cycle performance, nitrogen and alkyl-benzene oil are readily available, environmentally friendly and safe. Hugenroth (2006) provides analysis of LFEC system performance for several working fluid pairs.

2.1. Description of system

Oil and gas are adiabatically mixed at state point 21. Simultaneously the gas is compressed and the oil is pumped from state point 22 to state point 23, at which point the oil-gas mixture passes into the hot heat exchanger at state

point 29 and is cooled to state point 30. The mixture is cooled against an ethylene glycol-water temperature bath. After exiting the hot heat exchanger, the two-phase mixture enters into the hot-side separator (state point 26) where the oil and gas are separated into oil (state point 31) and gas (state point 32) phases. The oil is then expanded from high pressure (state point 24) to low pressure (state point 25) in a hydraulic expander to generate electrical power. The expanded oil at state point 19 is then mixed back into the hot gas stream. The hot gas exiting the separator then enters the regenerator (state point 8) where it is cooled to state point 9.

After exiting the regenerator, the cooled gas (state point 4) is mixed with cool oil (state point 3) to state point 5. This two-phase mixture enters the expander (state point 6) where it is expanded to state point 7. After the expansion process, the two-phase mixture passes into the cool heat exchanger, where the mixture is heated to state point 17, providing the cooling capacity of the Ericsson cycle. The heated two-phase mixture enters into the cool separator at state point 14, and is separated into oil (state point 12) and gas streams (state point 13). The oil stream is pumped up from low pressure (state point 1) to high pressure (state point 2), and then mixed back into the gas stream. The gas exiting the cold separator enters the regenerator where it is warmed from state point 11 to state point 10.

2.2. Measurement Devices

Both the compressor and expander were configured with motor controllers so that the rotational speed could be accurately controlled. The motor controllers were able

Figure 1: Schematic for Ericsson Cycle test rig.

to maintain the rotational speed within 1 revolution per minute, and the rotational speeds were output to the data acquisition system. The compressor was run at a constant rotational speed of 3500 RPM to simulate the small amount of slippage in AC induction motors operating at 60 Hz. Practical flooded compression systems would also likely run at a constant rotational speed near 3500 RPM. The rotational speed of the expander was varied in order to control the pressure ratio of the test rig. The speed ratio S (ratio of compressor to expander speed) was fixed at 2, 3, and 4 in order to investigate the impact of expander rotational speed on performance. Both compressor and expander were configured with rotary torque cells with a full-scale range of 22.6 N·m in order to measure the shaft power delivered or produced by the scroll machines.

The hydraulic expander and hydraulic pump were also configured with motor controllers in order to control the amount of oil being circulated through the hot and cold oil loops. As the hydraulic machines are positive displacement devices, the amount of oil delivered should be nearly proportional to the rotational speed (barring variation in the volumetric efficiency with operating conditions). In reality, bubbles of gas entrained in the oil will decrease the delivered mass flow rate of oil. The hydraulic pump had a displacement of $17.21 \text{ cm}^3 \text{ rev}^{-1}$ ($1.05 \text{ in}^3 \text{ rev}^{-1}$) and the hydraulic expander had a displacement of $18.03 \text{ cm}^3 \text{ rev}^{-1}$ ($1.10 \text{ in}^3 \text{ rev}^{-1}$). The hydraulic expander and pump were outfitted with the same torque cell as the compressor and expander.

Once the cycle reaches steady-state operation, the mass flow rate of the gas is the same through both the compressor and the expander as long as there is perfect separation and no gas solubility in the oil in the oil separators. The gas flow rate was measured with a Coriolis mass flow meter as shown in Figure 1. After all the tests were completed, the system was opened, and the tubes delivering the gas to the gas flow meter were found to be completely dry, which confirms the assumption that the separators provide good phase separation.

The temperature at all points of the cycle were initially measured with 4-wire Pt100 RTDs. Unfortunately all the RTDs had been installed perpendicular to the tubes with a large length of the RTDs exposed into the ambient. However, RTDs should be installed fully immersed in a T-junction. Thus, significant inaccuracies were found in the temperature measurements when using RTDs. In order to correct the temperature measurements at the compressor and expander, the incorrectly installed RTDs were replaced with T-type thermocouples.

The pressures in the system were measured with pressure transducers, with full scale ranges of 0-17.23 bar [gage] (0-250 psig) for the low pressure measurements, and 0-34.47 bar [gage] (0-500 psig) for the high pressure measurements. The low-pressure state-points are those with indices 1, 7, 10, 11, 12, 13, 14, 16, 17, 19, 20, 21, 22 and 25. The atmospheric pressure was measured with a mercury barometer.

The system was run until quasi-steady-state operation

Figure 3: Temperature differences downstream of hot-side components.

was achieved. Due to the very large amount of thermal mass in the system, it was never possible to achieve true steady-state operation. Once quasi-steady-state operation was achieved, experimental data were acquired for no less than 8 minutes, after which all experimental data acquired were averaged and used in the analysis which follows.

2.3. Measurement of oil flow rate

Initially, it was desired to use the energy balance over the hot and cold heat exchangers to determine the hot and cold oil flow rates, but it was found that at certain points in the system, there is a significant amount of thermal non-equilibrium. For instance, the temperature differences between the outlet of the hot heat exchanger and the inlet to the hot separator, which represents a distance of approximately one meter, range between +2K and -2K, as seen in Figure 3. The differences are due to the differences in flow pattern and inter-facial heat transfer inside the flat-plate heat exchanger. Thus, it was not possible to use the energy balance over the hot-side heat exchanger to back out the oil flow rate. Figure 4 shows that the same problem is also manifested at the cold-side heat exchanger, where the temperature differences range up to ± 7 K. The run numbers are consistent with the run numbers of the experimental data presented below.

Figure 3 also shows little difference in temperature between the outlet of the compressor and the inlet of the hot heat exchanger, and because the temperatures are equal to within experimental uncertainty, this suggests that the phases are in thermal equilibrium. The outlet of the expander exhibits better thermal equilibrium than that of the cold heat exchanger, but not quite as good as that of the scroll compressor.

Therefore, it was required that an energy balance over the compressor and expander be used to back-calculate the oil flow rate. For each machine, the shaft power is given

Figure 4: Temperature differences downstream of cold-side components.

by

$$\dot{W}_{shaft, meas} = N \frac{2\pi}{60} \tau \quad (1)$$

where the rotational speed N is given from the motor controller in rev min^{-1} , and the value of the measured torque τ is taken to be positive for the compressor and negative for the expander. Thus, the energy balance is given by

$$\dot{W}_{shaft, meas} = \begin{bmatrix} \overline{U}A_{amb} (T_{shell} - T_{amb}) \\ -\dot{m}_l (h_{l, out} - h_{l, in}) \\ -\dot{m}_g (h_{g, out} - h_{g, in}) \end{bmatrix} \quad (2)$$

where the value of the shell temperature T_{shell} is based on the inlet temperature for the compressor and the outlet temperature for the expander. The inlet and outlet enthalpies of liquid and gas are known from measurements of inlet and outlet temperature and pressure. Thus, the only remaining parameter needed to calculate the mass flow rate of oil is the overall shell-ambient heat transfer. Both compressor and expander are entirely covered with approximately 1 cm of foam insulation, and from a simplified network heat transfer analysis, the value of $\overline{U}A_{amb}$ is approximated as 1 W K^{-1} . Thus, the oil flow rate can be obtained from Equation 2.

Figures 5 and 6 show the oil mass flow rates calculated by a number of different methods, which demonstrates the challenges of measuring the oil mass flow rates through each oil loop. Three different methods are used to calculate the oil flow rate. The first, used as the reference oil mass flow rate, is based on the energy balance on the scroll compressor or expander. The second is based on calculating the oil flow rate using the density of the oil and the displacement rate of the hydraulic expander or pump and assuming 100% volumetric efficiency. The final method of calculating the oil flow rate is based on an energy balance over the heat exchanger of the loop. The scroll-machine energy balance method was ultimately selected because

Figure 5: Oil mass flow rates through the hot loop calculated by three different methods (compressor energy balance, hot HX energy balance, and hydraulic expander displacement rate).

Figure 6: Oil mass flow rates through the cold loop calculated by three different methods (expander energy balance, cold HX energy balance, and pump displacement rate).

it generated a data set with the most sensible physical trends.

2.4. Data Reduction

The parameters which are not directly measured must be calculated based on experimental data. The volumetric effectiveness is defined based on the displacement volume of the scroll machine. In the expander, the displacement volume is equal to the compressor suction volume divided by the built-in volume ratio. Thus the displacement volumes of the compressor and expander are 104.8 cm³ and 65.5 cm³ respectively. The volumetric effectiveness can be defined as

$$\varepsilon_v = \frac{\dot{m}_{m,meas}}{\dot{m}_{m,ideal}} = \frac{\dot{m}_{m,meas}}{\rho_{m,in} V_{disp} \frac{N}{60 \text{ s min}^{-1}}} \quad (3)$$

The volumetric effectiveness for both compressor and expander is ideally 1.0. In the compressor the volumetric effectiveness will generally be less than 1.0 since leakage and other losses will tend to decrease the amount of flow through the compressor. On the other hand, leakage in the expander will tend to increase the amount of mass flow, resulting in a flow rate that in general is greater than the ideal flow rate, depending on the magnitude of the suction pressure losses. The density $\rho_{m,in}$ is calculated as the homogeneous mixture density based on upstream temperature and pressure measurements as described in the companion paper (Bell et al., 2012b). This mixture density model assumes that both the liquid and gas phases travel at the same velocity, a good assumption here because the working chambers are enclosed by the scroll wraps.

The energy efficiency of the scroll machines is defined based on the overall isentropic efficiency of the machine, given by

$$\eta_{i,comp} = \frac{\dot{m}_{m,meas}(h_{out,s} - h_{in})}{\dot{W}_{shaft,meas}} \quad (4)$$

where the enthalpies and entropies are based on mixture properties as described in the companion paper (Bell et al., 2012b). Similarly, the overall isentropic efficiency of the expander can be defined by

$$\eta_{i,exp} = \frac{-\dot{W}_{shaft,meas}}{\dot{m}_{m,meas}(h_{in} - h_{out,s})} \quad (5)$$

Internal isentropic efficiencies, which have the effect of partially decoupling the losses that impact volumetric effectiveness (primarily leakage) from the other losses impacting the machine power, can also be defined. The internal isentropic efficiency is the efficiency that would be achieved if the volumetric effectiveness were equal to unity and there were no external heat transfer. Thus, the internal isentropic efficiency can be seen as a measure of the non-leakage losses. The decomposition of the overall isentropic efficiency into the internal isentropic efficiency and the volumetric effectiveness has no exact physical meaning. However, in the case of the expander, it allows for the partial decoupling between leakage losses and other

losses. The internal isentropic efficiency for the compressor can be defined as

$$\eta_{i,comp,int} = \frac{\eta_{i,comp}}{\varepsilon_{v,comp}} \quad (6)$$

and for the expander this same term can be defined as

$$\eta_{i,exp,int} = \varepsilon_{v,exp} \eta_{i,exp} \quad (7)$$

2.5. Measurement Uncertainty

An understanding of the measurement uncertainty is critical to analyze the data obtained. In order to calculate the measurement uncertainty of each of the calculated parameters, the total uncertainty of a given calculated value M is calculated by

$$\epsilon_M = \sqrt{\sum_i \left(\frac{\partial M}{\partial x_i} \epsilon_i \right)^2} \quad (8)$$

based on the uncertainties of each measured values x_i with absolute uncertainty ϵ_i given in Table 1. Tables 2 and 4 present the calculated values of the experimental uncertainties. The uncertainties were calculated by a numerical sub-routine that utilizes numerical derivatives in order to calculate the necessary partial derivatives and the absolute uncertainty of each parameter.

Table 1: Uncertainties of experimentally measured parameters

Parameter	Units	Uncertainty
T	K	± 0.5
p_{low}	kPa	± 2.24
p_{high}	kPa	± 4.48
$\dot{m}_g$	kg s ⁻¹	$\pm 1.0\%$
τ	N m	± 0.0452
N	rev min ⁻¹	1.0

3. Experimental Results

As shown in Figure 7, the volumetric effectiveness of the scroll compressor is constant at a high value near 95% over the range of experimental data obtained. Thus, the addition of a large amount of oil does not appear to have a significant negative impact on the delivered mass flow rate of the compressor. On the other hand, the expander volumetric effectiveness (larger effectiveness values indicate a decrease in volumetric performance) increases significantly as the expander speed is slowed. The poor volumetric effectiveness of the expander at low rotational speed is due to the fact that as the expander runs more slowly, leakage plays a more dominant role. In the limit that the rotational speed of the expander is zero, leakage will entirely dominate the mass flow rate and the expander will behave as a throttling valve.

Figure 7: Experimental volumetric effectiveness of scroll machines.

Figure 8 shows similar trends for the overall isentropic efficiency for the compressor. For the compressor, as the amount of oil injected increases, the irreversibilities also increase, which ultimately result in a larger amount of compressor power. The companion paper (Bell et al., 2012a) provides a more detailed treatment of the irreversibilities present in the scroll compressor. This treatment shows that the majority of the irreversibilities are due to suction and discharge pressure drops.

In the case of the expander, the decrease in overall isentropic efficiency with increasing oil mass fraction cannot be readily visualized in Figure 8, because of the large impact of the leakage losses on the overall performance. The internal efficiency allows for partial decoupling of leakage losses and other losses. Figure 9 indicates that the internal efficiency as a function of the oil mass fraction shows similar trends for both the compressor and the expander. The increase of the liquid flooding appears to be detrimental to both machines' performance. This is due to the fact that conventional automotive scroll compressors, which were not appropriately designed for liquid flooding, were used.

The fundamental goal of liquid flooding is to approach isothermal compression and expansion processes. As shown in Figure 10, the ratio of the high to low temperature for the compressor and the expander both approach 1.0 as the oil mass fraction increases, indicating that both processes become more isothermal. In the compressor, the high temperature is the outlet temperature, and in the expander, the high temperature is the inlet temperature. The difference in slope for the scatter plots for both compressor and expander is due to the difference in pressure

Figure 8: Experimental overall isentropic efficiency of scroll machines.

Figure 9: Experimental internal isentropic efficiency of scroll machines.

ratios experienced by the two machines. Since the test stand is relatively large for the delivered cooling capacity with significant piping and a large number of fittings, the pressure drop between the compressor and expander is quite large. As a result, the imposed pressure ratio on the compressor will always be higher than the pressure ratio imposed on the expander. In the limit of no pressure drops

Figure 10: Experimentally measured temperature ratios for compressor and expander.

Figure 11: Data flow diagram for model.

in the system, the two curves should come closer to each other. There will still be some difference in slope due to differences in scroll machine efficiency, manifesting itself as a difference in the outlet temperature.

4. Model Validation

After developing detailed simulation models for both the liquid-flooded scroll compressor and the liquid-flooded scroll expander, it is necessary to validate the models using experimental data as well as tune several parameters that are difficult to estimate directly. The scroll machine models operate as shown in the schematic in Figure 11, where all the parameters listed as model input parameters must be estimated, tuned, or correlated based on operating conditions.

Table 2: Compressor Experimental Data (including experimental uncertainties of calculated parameters)

Run	T_{22} K	p_{22} kPa	p_{23} kPa	N RPM	T_{amb} K	T_{23} K	x_l -	$\dot{W}_{shaft}$ W	$\dot{m}_m$ kg s ⁻¹	ε_v -	η_i -
1	310.0	236.7	719.1	3500	294.8	318.4	0.908±0.012	2584±17	0.155±0.021	0.931±0.013	0.598±0.010
2	306.9	234.9	714.1	3500	295.2	321.8	0.826±0.014	2423±17	0.086±0.007	0.967±0.013	0.655±0.010
3	302.9	235.0	711.3	3500	295.5	328.7	0.704±0.014	2343±17	0.052±0.002	0.977±0.013	0.687±0.010
4	315.7	385.3	1054.5	3500	295.7	327.1	0.857±0.015	3590±17	0.160±0.016	0.941±0.011	0.642±0.009
5	312.6	390.6	1049.0	3500	295.9	332.1	0.745±0.016	3404±17	0.095±0.006	0.960±0.011	0.686±0.009
6	307.4	390.0	1042.3	3500	296.0	339.4	0.599±0.016	3292±17	0.062±0.002	0.963±0.011	0.715±0.009
7	321.7	524.0	1330.3	3500	296.2	335.2	0.813±0.016	4385±17	0.166±0.015	0.950±0.011	0.668±0.008
8	317.9	534.3	1323.5	3500	296.5	340.4	0.685±0.018	4193±17	0.104±0.006	0.958±0.011	0.701±0.008
9	312.3	537.1	1317.2	3500	296.7	347.5	0.530±0.018	4054±17	0.072±0.003	0.957±0.010	0.727±0.009
10	311.6	224.1	739.1	3500	297.6	320.3	0.914±0.011	2657±17	0.153±0.020	0.921±0.013	0.587±0.009
11	308.6	225.4	737.3	3500	297.2	324.2	0.834±0.013	2518±17	0.086±0.007	0.966±0.013	0.645±0.009
12	301.8	223.4	735.0	3500	297.3	337.5	0.637±0.011	2414±17	0.041±0.001	0.972±0.013	0.685±0.010
13	324.3	441.4	1218.5	3500	302.8	337.1	0.839±0.015	4055±17	0.159±0.014	0.942±0.011	0.658±0.008
14	317.0	421.8	1177.4	3500	300.8	338.3	0.736±0.015	3794±17	0.098±0.006	0.960±0.011	0.693±0.008
15	309.1	409.6	1160.8	3500	298.5	343.8	0.593±0.014	3668±17	0.064±0.002	0.962±0.011	0.716±0.009
16	326.1	531.1	1436.0	3500	305.7	340.6	0.814±0.014	4695±17	0.166±0.013	0.945±0.011	0.673±0.008
17	322.5	544.1	1417.3	3500	305.1	346.3	0.686±0.016	4478±17	0.104±0.005	0.955±0.010	0.704±0.008
18	314.8	512.4	1361.2	3500	304.4	351.9	0.544±0.013	4225±17	0.070±0.002	0.956±0.010	0.726±0.009
19	322.6	277.5	895.7	3500	295.9	332.3	0.896±0.014	3057±17	0.152±0.020	0.919±0.012	0.619±0.009
20	320.0	284.5	893.6	3500	296.5	336.8	0.809±0.018	2908±17	0.089±0.008	0.947±0.012	0.668±0.009
21	314.8	285.4	891.0	3500	296.7	343.6	0.681±0.020	2827±17	0.055±0.003	0.954±0.012	0.697±0.010
22	327.8	395.6	1177.5	3500	296.9	339.7	0.860±0.016	3878±17	0.159±0.018	0.925±0.011	0.652±0.009
23	324.7	405.3	1174.0	3500	297.3	344.9	0.753±0.019	3719±17	0.097±0.008	0.944±0.011	0.691±0.008
24	318.5	406.5	1167.4	3500	297.4	352.1	0.609±0.021	3626±17	0.063±0.003	0.946±0.011	0.714±0.009
25	332.9	534.2	1496.9	3500	297.4	347.1	0.820±0.017	4817±17	0.166±0.016	0.931±0.011	0.674±0.008
26	329.1	547.1	1491.0	3500	297.6	352.9	0.698±0.021	4636±17	0.105±0.007	0.943±0.010	0.707±0.008
27	322.9	550.5	1478.8	3500	297.8	360.4	0.547±0.022	4503±17	0.072±0.003	0.942±0.010	0.730±0.009

4.1. Compressor Model Validation and Model Tuning

Tuning of the compressor model is carried out in a two-step process. First the mass flow rate is tuned based on leakage and suction pressure drop parameters using approximate values for the mechanical losses and the ambient heat transfer. Then, the shaft power is tuned based on mechanical losses, discharge pressure drop and external heat transfer parameters.

A simultaneous optimization of area correction term $X_{d,inlet}$ and leakage gap widths was carried out in order to minimize the error in total mass flow rate. To carry out the optimization and minimize the number of optimization parameters, the flank leakage gap width was imposed to be equal to the radial gap width ($\delta_f = \delta_r$). Therefore, the two independent parameters to tune the mass flow rate are the area correction term $X_{d,inlet}$ and the radial leakage gap width δ_r . During compressor operation, gas leaks from the higher pressure chambers to the lower pressure chambers. Since the compressor operates at a uniform rotational speed of 3500 rev min⁻¹, the compressor leakage gap widths are assumed to be constant. If compressor experimental data were to be available over a range of rotational speeds, the leakage gap widths could be determined as a function of rotational speed.

Different flow models are used for the different flow paths. For both the flank and radial leakages, the flow is taken to be entirely gas, with frictional flow, as described

in the companion paper (Bell et al., 2012c). The entrained flow model that is described in the companion paper is applied to the primary flow paths, with the ratio of downstream to upstream areas σ set to be zero, as shown in Table 3. An entrainment factor ψ of 0.4 is used, as recommended by Chisholm (1983). That is, 40% of the liquid is assumed to travel in the gas phase at the gas velocity.

The model can then be run for each of the 27 experimental data points for a range of leakage gap widths and area fractions. For each set of tuning parameters, the mean absolute error of the mixture mass flow prediction can be given by

$$MAE = \frac{\sum_{i=1..27} \left| \frac{\Upsilon_{model,i}}{\Upsilon_{exper,i}} - 1 \right|}{27} \times 100 \quad (9)$$

where Υ is the variable that is under consideration, here the mass flow rate. Figure 12 shows the results from tuning the mass flow rate. The model error was evaluated at a range of gap widths and area correction terms, and the contours were interpolated from the coarse grid data. The mean absolute error of the flow prediction is at a minimum for a gap width of approximately 12 μ m and suction area correction term of 0.4. Thus, these values are used in the shaft power tuning which follows. The terms of the mechanical loss model must also be determined. After the mass flow rate tuning parameters δ_r and $X_{d,inlet}$ have been

Figure 12: Tuning of mass flow by altering gap width and area correction term.

obtained, the flooded compressor model can be run with some guess for the mechanical losses. After the model converges, the “mixture” compression power can be obtained as the internal power required to compress the mixture of oil and gas to the discharge pressure, less the internal heat transfer amount, or

$$\dot{W}_{mix} = \dot{m}_m(h_{disc} - h_{suct}) - \overline{\dot{Q}_{gas}} \quad (10)$$

where $\overline{\dot{Q}_{gas}}$ is positive if the mean heat transfer over one rotation is to the gas from the lumped mass, and the enthalpies are based on mixture properties. The difference between the experimentally measured shaft power and the “mixture” compression power is therefore due to mechanical losses (assuming all the flow irreversibilities have been properly captured). Finally, the mechanical losses can be regressed as a function of the “mixture” compression power.

When no correction is made to the $X_{d,discharge}$ for the flow through the discharge port ($X_{d,discharge} = 1.0$), the mechanical losses are dependent on the oil-mass-fraction. In this compressor design, the dominant flow path of the oil-gas mixture does not bring it into contact with the bearing surfaces, so the oil mass fraction shouldn’t have a significant impact on mechanical losses. There might be an impact on oil-film frictional forces, though this effect was not considered. As a consequence, the discharge port discharge coefficient $X_{d,discharge}$ was tuned to decrease the oil-mass-fraction dependence of the mechanical losses. Further study of the two-phase flow through the discharge port is needed but is beyond the scope of this paper.

Figure 13 shows the results of the tuning process for the shaft power of the scroll compressor. There is a family of near-optimal solutions found by altering the area correction term, which distributes the total amount of irreversibility required between discharge port pressure drop,

Figure 13: Tuning of shaft power by altering ML and area correction term.

under-compression losses, and mechanical losses. From this analysis, a constant mechanical loss was selected, with the value of 0.35 kW, or a constant loss torque of 0.00096 kN·m, and a discharge port area correction factor of 0.5, that is, the discharge port is treated as being 50% as large as the physical port. This pairs of parameters is the minimum MAE value obtained from the actual model output without the interpolation required for the contour development. Over the range of experimental data, the model-predicted mechanical efficiency ranged from 84% to 93%.

Table 3: Scroll Compressor Model Parameters

Parameter	Units	Value
δ_r	μm	12
δ_f	μm	12
$X_{d,inlet}$	-	0.4
$X_{d,discharge}$	-	0.5
$\dot{W}_{ML}$	kW	0.35
$\overline{UA}_{amb}$	kW K ⁻¹	0.001
σ	-	0

The end result of the tuning process is a set of identified parameters which can be used to accurately predict the performance of the scroll compressor with flooding. The parameters in Table 3 were obtained for the compressor under study.

Figure 14 shows a parity plot for model predicted parameters compared with the experimental data. For the compressor, the model is able to accurately capture the physical effects occurring during the compression process. The total mass flow rate passing through the compressor is accurately predicted, with a mean absolute error of 0.81% and a maximum absolute error of 2.32%. The shaft power is also well-predicted, with a mean absolute error of 1.20% and maximum absolute error of 2.76%. The compressor

Figure 14: Parity plot for model and experimental data for scroll compressor.

discharge temperature is predicted within an absolute error band of 1.1 K. The full set of experimental data, as well as the results of some additional tests conducted, can be found in the work of Bell (2011).

4.2. Expander Model Validation and Model Tuning

The tuning of the expander model is carried out in a similar way to the compressor model tuning. First the mass flow rate is tuned based on internal leakage and pressure drop parameters, and then the shaft power is tuned based on mechanical loss parameters. Table 5 gives the parameters for the scroll expander model.

4.2.1. Prediction of the mass flow rate

Tuning of the total mass flow rates for the scroll expander indicated that the leakage gap widths are varying with the operating conditions. This variation of flank gap width with operating conditions could be explained by both scroll deformation and by the presence of oil adhering to the scroll walls and reducing the effective flow passage to the clearances. The best agreement between model predictions and experimental measurements for the mass flow rate and the mechanical power was found by tuning the flank gap rather than the radial gap. The radial gap was imposed to be $12 \mu\text{m}$ for consistency with the value identified for the compressor.

A flank leakage has been identified for each test by adjusting it in order to bring the calculated gas flow rate within 1% of the measured one. It appears that the flank leakage gap decreases with increasing the rotational speed,

the pressure ratio over the machine and the oil mass fraction. A first order correlation has been proposed:

$$\delta_f [\mu\text{m}] = \left(\begin{array}{c} 143.9235 - 52.7104 \frac{N}{N_{\max}} \\ -11.5330x_l - 68.0729 \frac{r_p}{r_{p,\max}} \end{array} \right) \quad (11)$$

where $r_p = p_6/p_7$ is the pressure ratio of the expander, $N_{\max} = 1736 \text{ rev min}^{-1}$ and $r_{p,\max} = 2.23$. Prediction of the flank leakage by this correlation is given in Figure 15. The correlation can predict the gap widths of 25 of the 27 points within an absolute error of 10%.

4.2.2. Tuning of Mechanical Losses

In the case of the expander, it was found that the mechanical losses can be correlated to the speed of the expander by the following relationship involving a mechanical loss torque, as proposed by Yanagisawa et al. (2001):

$$\dot{W}_{\text{loss}} = \omega \tau_{\text{loss}} \quad (12)$$

Table 5: Scroll Expander Model Parameters

Parameter	Units	Value
δ_r	μm	12
δ_f	μm	Varied
$X_{d,\text{inlet}}$	-	1.0
$X_{d,\text{discharge}}$	-	0.5
τ_{loss}	kN m	0.00070
$\overline{U}A_{\text{amb}}$	kW K^{-1}	0.001

Figure 16 presents the results of the validation for the scroll expander with liquid flooding. With respect to the mass flow rate prediction, the mean absolute error is 1.5%, and the maximum absolute error is 3.5%. The shaft power is slightly poorer predicted, with a mean absolute error of 2.9% and a maximum absolute error of 7.3%. The discharge temperature of the expander is predicted within an absolute error band of 2.0 K. It can be observed that the quality of the validation of the expander model is lower than that of the compressor. This is largely due to the fact that the relative uncertainties of the model input parameters, particularly the oil mass fraction, are quite large for the expander. For that reason it is more challenging to develop a mechanistic model that properly captures the physical effects occurring during the working process.

4.3. Conclusion

In this paper experimental data has been presented for scroll compressors and expanders with oil flooding. In addition, the predictions of the simulation models that were developed for the given oil-flooded scroll compressor and expander have been validated and tuned using this experimental database. The most important conclusions of this work are:

Table 4: Expander Experimental Data (including experimental uncertainties of calculated parameters)

Run	T_6 K	p_6 kPa	p_7 kPa	N RPM	T_{amb} K	T_{16} K	x_l -	$\dot{W}_{shaft}$ W	$\dot{m}_m$ kg s ⁻¹	ε_v -	η_i -
1	290.4	633.9	368.7	1736	294.8	286.6	0.758±0.077	-372±8	0.059±0.019	1.054±0.015	0.555±0.016
2	289.8	644.4	345.9	1736	295.2	281.5	0.617±0.096	-491±8	0.039±0.010	1.080±0.013	0.619±0.015
3	290.1	649.6	330.6	1734	295.5	274.2	0.428±0.158	-570±8	0.027±0.007	1.099±0.013	0.657±0.017
4	289.8	935.2	559.7	1736	295.7	284.4	0.661±0.079	-585±8	0.068±0.016	1.149±0.014	0.575±0.012
5	290.4	942.7	537.5	1735	295.9	279.6	0.475±0.101	-697±8	0.046±0.009	1.196±0.013	0.605±0.012
6	291.1	944.0	519.5	1733	296.0	272.5	0.272±0.150	-776±8	0.034±0.007	1.221±0.013	0.632±0.015
7	291.0	1182.8	729.7	1734	296.2	284.7	0.581±0.079	-715±8	0.074±0.014	1.230±0.014	0.554±0.010
8	292.4	1186.4	711.3	1733	296.5	280.8	0.377±0.096	-809±8	0.052±0.008	1.284±0.014	0.573±0.010
9	293.1	1188.7	694.5	1734	296.7	274.2	0.180±0.133	-891±8	0.041±0.007	1.313±0.014	0.597±0.013
10	290.8	670.8	346.2	1155	297.6	286.8	0.769±0.079	-384±5	0.057±0.020	1.401±0.022	0.504±0.012
11	290.8	679.7	330.7	1156	297.2	282.9	0.620±0.107	-446±5	0.038±0.011	1.472±0.018	0.508±0.011
12	290.2	687.7	308.3	1156	297.3	275.0	0.415±0.189	-500±5	0.025±0.008	1.483±0.017	0.514±0.016
13	295.5	1107.5	614.4	1155	302.8	289.7	0.605±0.093	-584±5	0.065±0.015	1.660±0.021	0.442±0.008
14	292.5	1082.3	567.6	1155	300.8	281.9	0.429±0.128	-653±5	0.045±0.010	1.677±0.019	0.464±0.010
15	291.2	1075.4	536.5	1155	298.5	273.1	0.220±0.181	-710±6	0.033±0.008	1.679±0.019	0.483±0.014
16	293.8	1308.6	727.4	1160	305.7	287.7	0.586±0.102	-703±6	0.074±0.018	1.672±0.023	0.448±0.009
17	295.2	1297.8	709.8	1156	305.1	283.4	0.335±0.139	-745±6	0.049±0.010	1.776±0.020	0.448±0.010
18	290.6	1259.0	651.9	1155	304.4	271.2	0.147±0.216	-801±6	0.037±0.009	1.745±0.020	0.476±0.016
19	299.7	814.6	411.9	863	295.9	296.0	0.722±0.069	-367±4	0.057±0.014	1.893±0.024	0.381±0.007
20	300.7	822.9	398.5	863	296.5	293.2	0.530±0.061	-412±4	0.036±0.005	2.004±0.021	0.382±0.006
21	301.2	826.2	383.2	863	296.7	286.8	0.285±0.111	-448±4	0.025±0.004	2.053±0.022	0.392±0.008
22	301.6	1075.9	556.2	864	296.9	296.8	0.629±0.068	-476±4	0.060±0.011	2.030±0.023	0.363±0.006
23	302.6	1081.6	542.5	864	297.3	293.4	0.410±0.060	-526±4	0.040±0.004	2.139±0.022	0.369±0.005
24	302.8	1081.9	525.6	864	297.4	285.9	0.154±0.109	-567±4	0.029±0.004	2.192±0.023	0.382±0.008
25	302.7	1373.9	725.0	864	297.4	296.9	0.548±0.066	-600±4	0.066±0.010	2.129±0.023	0.354±0.005
26	304.0	1374.7	711.9	863	297.6	293.2	0.308±0.057	-651±4	0.046±0.004	2.247±0.023	0.361±0.005
27	304.2	1370.1	694.8	863	297.8	285.6	0.058±0.099	-694±4	0.035±0.004	2.311±0.024	0.376±0.007

Figure 15: Prediction of the flank gap width of the scroll expander with operating parameters.

Figure 16: Parity plot for model and experimental data for scroll expander.

- Automotive scroll compressors and automotive scroll

compressors operated as expanders can operate with

large amounts of oil flooding with a relatively modest decrease in performance, even when using off-the-shelf compressors with minor modifications.

- The performance of the scroll expander is significantly decreased at lower speeds. This effect is stronger than the decrease in performance due to oil flooding.
- The mechanistic simulation models presented for the flooded scroll machines has been validated using experimental data, and for both machines, the mixture mass flow rate and shaft power can be predicted to within mean absolute errors bands of 4.0%.
- The validated models can be used to conduct parametric studies to evaluate changes in design parameters on compressor and expander performance.

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