Investigation of Refrigeration System Steam Ejector Performance Through Experiments and Computational Simulations

A thesis submitted by

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وَوَصَّيْنَا الإنسَانَ بِوالِدَيْهِ إِحْسَانًا ۖ حَمَلَتْه أَمْهُ كُرْرًا وَوَضَعَتْه كُرْرًا ۖ وَحَمْلَه وَفِصَالَه ثَلَاثَانَ شَهْرًا ۖ حِينًا إِذَا بَلَغَ أَشْدَه وَبَلَغَ أَرْبَعِينَ سَنَةَ قَالَ رَبَّي ۖ أَشْكُرَ بَعْمَتِكِ اللَّتِي أَنْعَمْتَ عَلَي ۔ وَعَلَى وَالِدَيْهِ وَأَنْ أَعْمَلَ صَالِحًا تَرْضِيَه وَأَصْلِحْ لِي فِي ذَرٰيَّتِي ۖ إِن ۖ أَنْتَ إِلَيْكَ وَإِنْ يُسْلِمُونَ صدق الله العظيم

Translation

And we have enjoined upon man, to his parents, good treatment. His mother carried him with hardship and gave birth to him with hardship, and his gestation and weaning [period] is thirty months. [he grows] until, when he reaches maturity and reaches [the age of] forty years, he says, "my lord, enable me to be grateful for your favor which You have bestowed upon me and upon my parents and to work righteousness of which You will approve and make righteous for me my offspring. Indeed, I have repented to you, and indeed, I am of the Muslims."
Dedication

Thee my Lord to be satisfied

And then

To Dad and Mum

To my brothers and my sister

To my family Hafssa, Oras,

Othman, Faidha (Meriam), Ryaheen, Abdulrahman,

Adnan, and Yousif
Abstract

Recently, vapour compression using ejectors has received more interest from researchers in the air conditioning and refrigeration field. Ejectors have the advantage of being extremely reliable with stable operation and no moving parts leading to essentially maintenance free operation. However, ejectors have very low efficiencies and this can be attributed to the low entrained mass flow rate of the secondary stream relative to the primary stream mass flow rate. The entrainment and mixing between primary and secondary streams is therefore a dominant feature which requires investigation. This thesis introduces and demonstrates new methods to characterise the mixing process in a vapour compression ejector operated with steam as the working fluid.

A new steam ejector refrigeration apparatus was built to visualise the mixing flow inside the ejector. New designs for the steam generator and the evaporator chamber were used in this apparatus. The evaporator chamber was developed in order to weigh the amount of water induced from the evaporator during the test time. The weighing method was demonstrated to be sufficiently accurate for this application. The highest percentage difference between the direct weighing method and a thermal method for deducing the secondary stream mass flow rate was 2.6 %, corresponding to about 0.3 % of the entrainment ratio.

A CFD simulation tool called Eilmer3 (developed by the Compressible Flow CFD group of the University of Queensland) was used to simulate the flow within the axisymmetric geometry of the steam ejector. The Eilmer3 computational simulations adopted an ideal gas model for the steam throughout the ejector. The distribution of static pressure within the ejector was accurately simulated for different primary and sec-
ondary stream conditions and different condenser pressures. The difference between experimental data and the Eilmer3 simulations of static wall pressure was typically less than 2%. The difference between experimental data and the Eilmer3 simulation of entrainment ratio was 2.9% at 130°C primary stream temperature and 2.6% at 120°C primary stream temperature for choked ejector operation. For ejector operation at unchoked conditions, differences between experimental results and the simulations were substantial with the critical condenser pressure typically underestimated in the simulation by around 8.8% at 270 kPa primary stream pressure and 15.4% at 200 kPa primary stream pressure.

To enable visualisation of the ejector flow, a transparent acrylic duct was optically designed using a ray tracing method. The machined and polished optical duct performed as expected in that it provided a field of view over the entire height of the duct. However, the planned schlieren visualisation could not proceed with the optical duct because of shadowing effects attributed to birefringence in the duct material. Nevertheless the optical duct was successfully used to gain photographic evidence of liquid water and ice building up within the ejector. The presence of liquid and solid forms of water within the ejector suggests the ideal gas model cannot generally be applied to the steam ejector flows even though a good degree of success was achieved in modelling the static pressure distribution and entrainment ratio under choked flow conditions.

A new technique for exploring the mixing region generated by the steam ejector nozzle was introduced. The new approach used a pitot tube to measure the pressure profile at different positions downstream of the nozzle exit within a mixing chamber with a relatively large rectangular cross section. A special pitot probe and traversing mechanism was designed and fabricated for this purpose. The experimental results demonstrate a shock train was established downstream of the nozzle exit. The experimental measurements indicate that pressure wave effects within the mixing jet have largely dissipated by the 70 mm downstream location. Eilmer3 simulations were also used to investigate the flow at the nozzle exit. Eilmer3 simulations duplicate experimental pitot pressure data at the first station downstream of the nozzle exit (1 mm) but are not consistent with the pitot pressure measurements at the other positions (25, 50, and 70 mm).

A momentum integral analysis of a control volume in the mixing chamber was at-
tempted. Using the Eilmer3 simulation at the nozzle exit, the momentum transport into the control volume was calculated. The momentum transport out of the control volume was the estimated using experimental data at the 70 mm station based on an ideal gas analysis and an assumed constant static pressure across the jet. To reflect possible condensation effects, values for the ratio of specific heat lower than $\gamma = 1.326$ were trialled in the analysis in an attempt to achieve the required momentum transport out of the control volume. However, even with $\gamma = 1.001$ the momentum transport out the control volume was too high, indicating that an inflow momentum transport contribution due to recirculation across the downstream station may be significant although it was not included in the analysis.

The schlieren method was used to visualize the steam flow at the exit from the nozzle inside the rectangular mixing chamber which was also used in the pitot pressure surveys. The schlieren arrangement followed Toepler’s method with one lens. An edge feature of the steam jet downstream of the nozzle was detected using an image analysis process applied to the video record from the schlieren visualisation. The stagnation conditions of the steam were 380 kPa and 144°C in this case, and the background pressure in the mixing chamber was 3 kPa.
Certification of Dissertation

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

GHASSAN FADIL LATTIF AL-DOORI

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Signature of Candidate

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Date

ENDORSEMENT

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Signature of Supervisor/s

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Date
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First of all I would like to thank My God, who helped me finished this thesis.

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<th>Description</th>
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<tr>
<td>$a$</td>
<td>Sonic speed, m/s</td>
</tr>
<tr>
<td>$A$</td>
<td>Area, m(^2)</td>
</tr>
<tr>
<td>$A_T$</td>
<td>Nozzle area ratio</td>
</tr>
<tr>
<td>$C_1$</td>
<td>Molecular weight correction factor</td>
</tr>
<tr>
<td>$C_2$</td>
<td>Temperature correction factor</td>
</tr>
<tr>
<td>$C_d$</td>
<td>Discharge coefficient nozzle</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Specific heat capacity at constant pressure, kJ/kg K</td>
</tr>
<tr>
<td>$C_v$</td>
<td>Velocity coefficient</td>
</tr>
<tr>
<td>$d$</td>
<td>Inside diameter of stainless steel pipe, m</td>
</tr>
<tr>
<td>$d^*$</td>
<td>Diameter of throat nozzle area, m</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter of throat ejector, m</td>
</tr>
<tr>
<td>$Di$</td>
<td>Inside diameter of copper tube, m</td>
</tr>
<tr>
<td>$Do$</td>
<td>Outside diameter of copper tube, m</td>
</tr>
<tr>
<td>$f$</td>
<td>Internal roughness for stainless steel, m</td>
</tr>
<tr>
<td>$h_{4,5,6,9,...}$</td>
<td>Enthalpy, kJ/kg</td>
</tr>
<tr>
<td>$H$</td>
<td>Height of the evaporator, m</td>
</tr>
<tr>
<td>$h_i$</td>
<td>Inside tube heat transfer coefficient, W/m(^2)K</td>
</tr>
<tr>
<td>$h_o$</td>
<td>Outside tube heat transfer coefficient, W/m(^2)K</td>
</tr>
<tr>
<td>$h_f$</td>
<td>Head losses, m</td>
</tr>
<tr>
<td>$I_{turb}$</td>
<td>Turbulence intensity</td>
</tr>
<tr>
<td>$g$</td>
<td>Acceleration of gravity, taken as 9.81 m/s(^2)</td>
</tr>
<tr>
<td>$Gr$</td>
<td>Grashof number</td>
</tr>
</tbody>
</table>
\( k \)  
Thermal conductivity, W/m K

\( L \)  
Length, m

\( L_m \)  
Conical mixing chamber length, m

\( L_n \)  
Secondary inlet length, m

\( L_p \)  
Parallel mixing chamber length, m

\( L_d \)  
Diffuser length, m

\( M \)  
Mach number

\( \dot{m}_p \)  
Primary mass flow rate, kg/hr

\( \dot{m}_s \)  
Secondary mass flow rate, kg/hr

\( N_p \)  
Primary ratio

\( N_s \)  
Compression ratio

\( Nu \)  
Nusselt number

\( P \)  
Pressure, kPa

\( P_c \)  
Condenser pressure, kPa

\( P_{co} \)  
Breakdown pressure, kPa

\( P_b \)  
Background pressure, kPa

\( P_e \)  
Nozzle exit pressure, kPa

\( P_{pit} \)  
Pitot tube pressure, kPa

\( Pr \)  
Prandtl number

\( Q_e \)  
Cooling load, W

\( Q_{ee} \)  
Power input to the evaporator element heater, W

\( Q_g \)  
Input heat to the steam generator, W

\( q \)  
Heat transfer from oil to water, W

\( R \)  
Gas constant, kJ/kg K

\( R_{1, 2, 3, ...} \)  
Heat resistance K/W

\( r_{1, 2, 3, ...} \)  
Radius of pipe and insulation, m

\( T \)  
Temperature, ºC

\( T_s \)  
Surface temperature, ºC

\( T_f \)  
Fluid temperature, ºC

\( u \)  
Velocity, m/s

\( V \)  
Velocity of steam inside the pipe, m/s

\( W \)  
Watt

\( W_p \)  
Pump work, W
NOTATION

\[ Wm \] Watt meter,
\[ y \] Distance, mm
\[ y^+ \] Dimensionless wall distance

Greek symbols

\[ \omega \] Entrainment ratio
\[ \rho \] Density, kg/m\(^3\)
\[ \rho_{\text{wall}} \] Wall density, kg/m\(^3\)
\[ \theta_m \] Mixing chamber angle
\[ \theta_d \] Diffuser angle
\[ \gamma \] Specific heat ratio
\[ \eta_N \] Nozzle’s isentropic efficiency
\[ \tau_{\text{wall}} \] Wall shear stress, kg/m \(\text{s}^2\)
\[ \mu \] Dynamic viscosity, Pa.s
\[ \mu_{\text{wall}} \] Wall dynamic viscosity, Pa.s
\[ \beta \] Volumetric thermal expansion coefficient, 1/K
\[ \nu \] Kinematic viscosity, m\(^2\)/s
\[ \mu_{\text{lam}} \] Laminar viscosity, m\(^2\)/s
\[ \mu_{\text{turb}} \] Turbulent viscosity, m\(^2\)/s
\[ \nu^* \] Friction velocity, m/s
Subscripts

\( o \) Stagnation condition
\( i \) Inside
\( c \) Cold
\( h \) Hot

Superscripts

\( * \) Critical
\( . \) Refers to flow rate
## Acronyms & Abbreviations

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>ASHRAE</td>
<td>American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational fluid dynamics</td>
</tr>
<tr>
<td>CMA</td>
<td>Constant mixing area</td>
</tr>
<tr>
<td>CPM</td>
<td>Constant pressure mixing</td>
</tr>
<tr>
<td>C.P.</td>
<td>Critical condenser pressure</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance of cycle</td>
</tr>
<tr>
<td>CV</td>
<td>Control volume</td>
</tr>
<tr>
<td>ESDU</td>
<td>Engineering sciences data unit</td>
</tr>
<tr>
<td>I.P.</td>
<td>Intersect point</td>
</tr>
<tr>
<td>LMTD</td>
<td>Log mean temperature difference</td>
</tr>
<tr>
<td>NXP</td>
<td>Nozzle exit position</td>
</tr>
<tr>
<td>SSR</td>
<td>Solid state relay</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

1.1 Applications of an ejector

Ejector or jet pumps are used extensively in the power generation, nuclear and chemical processing industries. Ejectors also have applications in distillation, vacuum evaporation and drying (for instance, soap drying). Ejectors have many advantages over conventional pumps or compressors such as extremely reliable and stable operation, the absence of moving parts, and the ability to handle highly corrosive vapours and fluids. Ejectors can be fabricated in wide range of sizes and can operate safely under hazardous conditions (ESDU86030, 1986).

Ejectors have a very low efficiency compared with normal pumping system but when a source of low grade or waste energy is available, an ejector may be cheaper to operate than a mechanical pump (Dandachi, 1990). Recently, ejectors have become of interest for researchers in air conditioning and refrigeration fields.

1.2 Ejector refrigeration systems

Cooling and heating systems typically use natural refrigerant fluids or artificial working fluids, such as chlorofluorocarbons (CFC) (e.g. R-12 and R-11), hydrochloroflu-
orocarbons (HCFC) (e.g. R-22 and R-123) and hydrofluorocarbons (HFC) (e.g. R-32 and R-134a) (ASHRAE, 2002). These working fluids can cause serious problems like depletion of the stratospheric ozone layer, production of large amounts of greenhouse gas emissions and global warming. Furthermore, refrigeration and air conditioning systems are normally driven by electricity, which strongly increases the demand for electricity and the consumption of fossil fuel. An alternative solution for this problem is the application of low-grade waste energy or solar energy. There are a number of types of air conditioning systems potentially powered by such energy including absorption, adsorption and desiccant coolers. However, these systems are typically large and expensive. Evaporative coolers are another possibility but these have a high level of water consumption and may not be effective at all locations. Ejectors can also be driven by waste-energy or solar power and through an increasing use of waste or solar energy in refrigeration systems, the relative demand for electrical energy sources can be decreased.

Figure 1.1 illustrates an ejector refrigeration cycle. The ejector provides the compression effect for the cycle.

### 1.3 Ejector operation

The geometry of a typical ejector is presented in Figure 1.2. The primary flow from the generator enters the ejector at position G and accelerates through a converging-diverging nozzle. The supersonic flow creates a low pressure at the nozzle exit position 1, lower than the evaporator pressure. Therefore, a flow of vapour is induced from the evaporator at E. The vapour flow from E is called the secondary flow or entrained flow and it has a low pressure and velocity at position 2. Mixing between the primary and secondary streams begins at the exit of the supersonic nozzle. However, mixing is sometimes considered to begin at point 3 downstream, for the convenience of thermodynamic modelling. The mixing process is usually considered to be complete at position 4. Shock waves occur in the supersonic mixing flow. After the shock compression process, the velocity of the mixture becomes subsonic and is further reduced in the diffuser section and at the same time the mixture is further compressed in the
Figure 1.1: (a) Ejector pump refrigeration system; (b) Illustrative p-h diagram of a closed cycle ejector based refrigeration system.
diffuser. Finally, the mixture leaves the diffuser to enter the condenser at position C. An important parameter in ejector design is the entrainment ratio which is defined as \( \omega = \frac{\dot{m}_s}{\dot{m}_p} \), where \( \dot{m}_s \) is the secondary mass flow rate and \( \dot{m}_p \) is the primary mass flow rate.

![Figure 1.2: Geometry of an ejector.](image)

Ejector refrigeration systems are promising because of their relative simplicity and low capital cost and the fact that the system can be powered by “low grade” energy (for example solar energy) instead of electricity. This means ejector-based refrigeration systems potentially have significant environmental benefits, especially when a renewable energy source such as solar energy can be used. The primary disadvantage with ejector-based refrigeration systems is the low coefficient of performance (COP). As illustrated in Figures 1.1(a) and 1.1(b), the refrigeration effect is \( \dot{m}_s(h_6 - h_9) \) and the power supply (heat supply) is \( \dot{m}_p(h_4 - h_2) \). The COP is defined to be

\[
COP = \frac{\text{refrigeration effect}}{\text{power supply + pump work}} \tag{1.1}
\]

and when the pump work is very small compared with the power supply (which is usually the case), the pump work can be neglected and the COP can be approximated as

\[
COP = \frac{\dot{m}_s(h_6 - h_9)}{\dot{m}_p(h_4 - h_2)} \tag{1.2}
\]

\[
COP = \omega \frac{(h_6 - h_9)}{(h_4 - h_2)} \tag{1.3}
\]

Clearly the entrainment ratio \( \omega = \frac{\dot{m}_s}{\dot{m}_p} \) is directly proportional to the COP, meaning that for given thermodynamic states of the system operation, an increase in the entrainment ratio will directly increase the COP.
1.4 Ejector design

Supersonic ejector design profoundly affects refrigeration system performance. Previous studies have presented varying degrees of success when attempting to computationally simulate and model the static pressure distributions within the mixing region of ejectors. Ejector design optimisation based on computational simulation or modelling is not yet a reliable strategy because of the variable performance of computational tools. The modest degree of success achieved so far with ejector computational simulation arises because of the complexity of the ejector flows which typically include strong turbulent mixing, high speed compressible vapour, strong pressure gradients and shock waves. To clarify the mixing and compression processes occurring in ejectors, additional experimental data are required for validation of thermodynamic modelling and numerical simulations.

1.5 Objectives of the dissertation

The present work targets the generation and interpretation of new data on the mixing and compression processes in a representative ejector apparatus. Steam is used as the working fluid because of its well-established thermodynamic and transport properties combined with its ease of handling in the laboratory environment. The main objectives of this study are to:

1. Design and commission an open steam ejector refrigerant cycle with a cooling capacity sufficient for visualisation purposes.

2. Confirm system performance over an appropriate range of operating conditions.

3. Design and commission a transparent mixing chamber suitable for visualisation of ejector flows.

4. Acquire new data which provides additional insight into the ejector mixing and compression processes.

5. Interpret the results with the aid of analytical and/or computational tools.
1.6 Overview of the Dissertation

This dissertation is organised as described in the following sections.

1.6.1 Chapter 2 Literature review

The literature review is presented in three main sections. The first section presents an overview of ejector design and operation, while section 2 presents an overview of experimental flow visualisation studies and section 3 discusses computational simulation results obtained for different ejector configurations.

1.6.2 Chapter 3 Overview of the design of the apparatus

This chapter presents the design of the steam ejector refrigeration apparatus and the measurement methods used to characterize the coefficient of performance of the system. A new design for the steam generator is introduced and is compared with a conventional boiler. To weigh the amount of water induced from the evaporator during test time, a new evaporator chamber is also introduced.

1.6.3 Chapter 4 System performance

Standard operating conditions for the ejector apparatus were established during commissioning of the rig. Methods and results for the calculation of COP, heat losses from the primary stream, and the heat gain by the evaporator from non-electrical sources are also presented. Measurement methods and results for the mass flow rates of the primary and secondary flows and the critical condenser pressure are also presented.

1.6.4 Chapter 5 Ejector results and discussion

This chapter presents the results of the experiments in the steam ejector configuration. Results demonstrate the effect of the primary stream conditions, the effect of condenser
pressure on pressure distributions in the mixing chamber and diffuser, and the effect of evaporator load on the performance of the apparatus. Comparison of the present experimental results with previous work is also presented.

1.6.5 Chapter 6 Computational simulation

Computational Fluid Dynamics (CFD) simulation of the ejector was also performed using a $k$-$\omega$ turbulence model and good agreement between the experimental data and the simulations for wall static pressure was achieved. Pressure and velocity maps identifying some features such as recirculation flow within the diffuser and mixing chamber, and reverse flow in the secondary inlet are also discussed.

1.6.6 Chapter 7 Mach 4 steam jet

A new method to characterise the steam flow in the mixing chamber was required. For this task, a special pitot tube apparatus was designed and fabricated. Pitot pressure measurements were performed with the pitot probe positioned at four locations from the lip of the nozzle: 1, 25, 50, and 70 mm. The experimental results were analysed with the aid of CFD simulations and a momentum integral analysis.

1.6.7 Chapter 8 Flow visualisation

Two new optical arrangements were designed to manage the refraction of light through the circular cross section of the mixing chamber. Tests were performed using the schlieren method in an effort to visualise the flow. Rearrangement of the Toepler's schlieren method with one lens was trialled in an effort to increase the sensitivity of the schlieren method.
1.6.8 Chapter 9 Conclusions and future work

The last chapter reports the overall conclusions of this study, and presents recommendations for future work.
Chapter 2

Literature review

2.1 Introduction

There are several ongoing attempts to investigate the low coefficient performance of ejectors compared to conventional pumping systems. Most efforts have adopted experimental and computational approaches, although analytical work has played a significant role in the development of ejector design approaches.

2.2 Ejector performance studies

2.2.1 Analytical work

A steam ejector refrigeration system was developed by Le Blanc and Parson as early as 1901 (ASHRAE, 1969). In the past, the design of the ejector relied greatly on empirical results. Flugel (1939) presented significant guidelines and recommendations for ejector design. The analysis was dependent on solving an energy equation with constants and quantities evaluated from several experimental studies. Keenan, Neumann and Lustwerk (1950) examined theoretical and experimental results for two types of ejector: the constant mixing area (CMA) ejector and the constant pressure mixing (CPM) ejector. The result from the Keenan et al. (1950) investigation was that CPM gives
a better coefficient of performance. One-dimensional constant area flow models have
been used to solve many supersonic ejector optimisation problems. The Mach number
for the primary nozzle and the ejector area ratio were found to have a significant
influence on performance parameters such as the compression ratio or the entrainment
ratio (Dutton and Carroll, 1986).

Huang et al. (1999) carried out a one-dimensional ejector performance analysis for ejector
operation. Referring to Figure 2.1, a number of assumptions were made including:

- A hypothetical throat occurs inside the constant-area section of the ejector cross
  section y-y, meaning the entrained flow reaches a choked condition at the y-y
  section.

- Primary and secondary flows start mixing at the hypothetical throat with a
  constant pressure.

- A shock will appear at cross section s-s, causing a sharp pressure rise.

Huang et al. (1999) studied 11 ejectors with R141b as the working fluid. The experi-
mental results were used to calculate the coefficients of the primary nozzle and other
parameters defined in the one-dimensional model. It was shown that the analytical
investigation can accurately predict the performance of the ejectors but the modelling
assumptions would not be applicable to all ejector configurations.

Figure 2.1: Schematic diagram of ejector (Huang et al., 1999).
Al-Ansary (2004) formulated a one-dimensional model of two phase flow in ejectors. The effect of adding a non-volatile liquid to the primary stream in an ejector was evaluated. A control volume analysis was used to develop a model based on a number of reasonable assumptions, such as:

- The volatile fluid is an ideal gas.
- The velocities of primary and secondary streams at the inlet are negligible.
- The size of the droplets of the non-volatile fluid is very small such that the two components have the same velocity and temperature at any given position. This means the flow is homogeneous at the exit.
- The ejector is adiabatic and all properties of the fluids are constant.

The following results were deduced from the analysis.

- A normal shock wave leads to compression of the mixture of primary and secondary flows at the end of the mixing section.
- The adding of a non-volatile liquid to the primary stream causes a slight decrease to the flow rate of gaseous components.
- The choking phenomenon is a function of the value of the entrainment ratio.
- The entrainment ratio is increased through adding a non-volatile liquid to the primary flow.

Pridasawas (2006) discussed the effect of the back pressure, also called the condenser pressure $P_c$, on the entrainment ratio. Three ejector operation regimes are associated with the condenser pressure, see Figure 2.2.

- The greatest COP can be obtained for critical mode operation, where the condenser pressure is lower than the critical pressure. In this case, choking occurs for both primary and secondary flows.
- The COP decreases when the condenser pressure is higher than the critical pressure. In this case, choking occurs only in the primary flow, not in the secondary flow. Therefore, the entrainment ratio decreases with increase \( P_c \).

- The COP drops to zero when the condenser pressure is higher than the limiting pressure of the ejector, \( P_{co} \). In this case, back-flow occurs in the secondary flow.

In Pridasawas (2006), many governing equations were used during the theoretical analysis, which adopted a number of assumptions such as the process in the ejector being adiabatic. Mixing and friction loss were calculated in the form of an isentropic efficiency. Furthermore, the refrigerant was assumed to have constant properties and steady flow along the ejector was also assumed.

Sun (1997) used an open steam jet refrigeration system to investigate the performance characteristics of the system. Boiler temperature range was from 95°C to 135°C while the evaporator temperature range was from 5°C to 15°C. The results show that the entrainment ratio first rose and then reduced thereafter when the boiler temperature increased. Evaporator temperature plays an important role in the ejector performance. An increase in the COP happens when there is an increase in the evaporator temperature.

Figure 2.2: Schematic illustration of operation regimes associated with the condenser pressure (Pridasawas, 2006).

### 2.2.2 Operating conditions

Sun (1997) used an open steam jet refrigeration system to investigate the performance characteristics of the system. Boiler temperature range was from 95°C to 135°C while the evaporator temperature range was from 5°C to 15°C. The results show that the entrainment ratio first rose and then reduced thereafter when the boiler temperature increased. Evaporator temperature plays an important role in the ejector performance. An increase in the COP happens when there is an increase in the evaporator temperature.
2.2 Ejector performance studies

Meyer, Harms and Dobson (2009) also examined a steam ejector which was setup in an open system configuration. The tests included a boiler temperature range between 85°C and 140°C, an evaporator temperature range from 5°C to 10°C, and a maximum condenser pressure of 5.63 kPa. The primary nozzle throat diameters used were 2.5, 3, and 3.5 mm. The results reveal that the steam ejector system can be operated at boiler temperatures below 100°C. Meyer et al. (2009) concluded that the key parameters that govern the functioning of the steam ejector are the evaporator temperature, boiler temperature, condenser pressure, and primary nozzle exit position.

Chunnanond and Aphornratana (2004) performed experiments using a closed steam ejector refrigeration system. The experiments were conducted over a boiler temperature range from 120°C to 140°C, an evaporator temperature range from 5°C to 15°C, and with three primary nozzle throat diameters: 2, 1.75, and 0.5 mm. From their results, it appears that the superheating of the primary flow does not have an impact on either the COP of the steam ejector or the critical condenser pressure. Furthermore, the results show that the COP of the system decreased when the primary pressure was increased at constant evaporator temperature.

2.2.3 Primary nozzle arrangement

Chang and Chen (2000) found that the performance of the ejector using a petal nozzle was better than ejector performance with a conical nozzle for large nozzle area ratios. Figure 2.3 illustrates the conical and petal nozzles. Both nozzles have the same throat and exit areas of 3.14 and 63.4 mm², respectively. The exit plane of the petal nozzle was formed by six lobes. The experiments were carried out at various evaporator temperatures, boiler temperatures, and condenser pressures. The results reveal that the entrainment ratio and compression ratio can be improved if the petal nozzle with large area ratio is used in the steam ejector.

Ma et al. (2010) investigated a novel steam ejector refrigeration system. The primary flow rate of the nozzle was controlled by a spindle placed in front of the nozzle inlet, as shown in Figure 2.4. The primary flow rate varied from 19.3 m³/hr to 51.3 m³/hr when the spindle position was changed from 2 mm to full capacity (30 mm). The results
demonstrate that the cooling capacity decreased when the spindle moved toward the nozzle because of the decrease in primary flow rate. The critical back pressure rose with the increase in the spindle position. The COP and cooling capacity increased with evaporator temperature. The maximum boiler temperature was less than 95°C and the optimum COP and entrainment ratio were found at a boiler temperature of about 90°C.
2.3 Flow visualisation studies

Throughout history, experimental flow visualisation has been an important instrument in fluid dynamics research and remains a very helpful tool for the understanding of complex flow phenomena and the validation of analytical and computational models. Up to this point, little work has been done in the area of flow visualisation within ejectors.

Fabri and Siestrunck (1958) utilised schlieren methods to visualise the flow in six nozzles for supersonic air ejectors operated over various ranges of pressure. Experimental results were compared with theoretical work and a number of the flow regimes were categorised. Matsuo, Sasaguchi, Tasaki and Mochizuki (1981) used schlieren methods to analyse the performance of a supersonic flow of air through a rectangular ejector without a secondary flow. Ejector operation with an exhaust or vacuum pump was investigated.

Hong, Alhussan, Zhang and Garris (2004) used a schlieren system and a high speed camera to investigate a new idea to improve the efficiency of the ejector by reducing the velocity of the primary flow by allowing it to expand through a rotor-vane. Schlieren visualisation was used to capture unsteady phenomena in the entrance region of the mixing chamber. Optical measurements using schlieren were more sensitive to changes of density perpendicular to the flow than in any other direction (Dvorak and Safarik, 2005). However, the shadowgraph and schlieren methods do not allow visual distinction between two interacting (primary and secondary) streams in an ejector (Bouhanguel et al., 2010).

Rayleigh and Mie scattering are imaging techniques based on light scattering by small particles. Researchers have used these methods to visualise the flow in supersonic air ejectors. Porcar and Prenel (1976) visualised shock waves in supersonic air ejectors by using Rayleigh scattering with diffuse water droplets inside the ejector without a secondary flow. Desevaux (2001a) used a laser sheet method to discriminate visually between the secondary and primary flows before they finished mixing. Visualisation was successful when the entrainment ratio was less than 0.3 and poor when the entrainment ratio was more than 0.3. Furthermore, the laser sheet techniques have been
used in qualitative studies when water nanodroplets were formed in an induced ejector (Desevaux, 2001a). For this purpose a TSI atomiser (model 9306A) was used to produce fluorescent doped droplets of diameter 0.1-2 $\mu$m. Figure 2.5 shows the experimental rig including a TSI atomiser and the data acquisition system used by Desevaux (2001b). Desevaux (2001b) continued investigating the formation of water nanodroplets in an induced air ejector. The water nanodroplets occurred when the induced air was moist. An analysis of the light scattered by the droplets generated by condensation within the flow revealed that their mean diameter did not exceed 0.1$\mu$m.

Desevaux and de Sousa (2004) conducted a visualised air flow experiment inside a supersonic ejector. The laser tomography visualisation was compared with a computational fluid dynamic (CFD) simulation. Zero secondary flow and free entrainment of induced air were examined through visualisation.

Marynowski et al. (2009) presented visualisation results for droplet condensation with moist air supplied to the ejector. The results were compared with visualisation from CFD. Figure 2.6 illustrates reasonable agreement between experimental visualisation and CFD simulations for three values of primary stagnation pressure $P_1$ with the free entrainment operating mode of the ejector. However, experimental visualisation for the condensation region appears slightly longer than the length obtained from the CFD
Rayleigh scattering and Mie scattering techniques were improved by Bouhanguel et al. (2010) to visualise supersonic air flow inside the ejector. Figure 2.7 illustrates the experimental arrangement using a laser beam from a CW argon laser with 120 mW of power in the green line ($\lambda_0=514\,\text{nm}$) as a light source. The camera had 500x582 effective pixels.
2.4 Computational fluid dynamics simulations

Computational fluid dynamics (CFD) was used by Riffat, Gan and Smith (1996) to simulate ejector performance. A commercial CFD software package was used to analyse ejector performance with different types of refrigerant and various geometries for the primary nozzle. However, Riffat et al. (1996) modelled the refrigerant as an incompressible flow to avoid a convergence difficulty with the computational method. Clearly this is an invalid assumption in the context of supersonic ejector design.

Riffat and Everitt (1999) designed a steam ejector to provide air conditioning for a small vehicle. The ejector system was manufactured and assessed on a laboratory controlled test rig. CFD was used to simulate the steam ejector and the working fluid was modelled as compressible. The results from the CFD illustrated no signs of shocks and a little reversed flow close to the entrance and the mixing chamber. In contrast,
2.4 Computational fluid dynamics simulations

the experimental results indicated the presence of a shock.

Computational fluid dynamic simulations were used by Rusly (2004) to simulate and design an ejector working with R141b. Simulations included three different flow fields related to operating conditions and the ejector entrainment ratio. The best performance of the ejector was achieved when operating in an over expanded state and the entrained flow was choked. However, a weak oblique shock wave was observed in the ejector simulations, especially in the constant area section filling the centre part of the tube slightly off the wall.

Bartosiewicz, Aidoun, Desevaux and Mercadier (2005) studied the performance of six well-known turbulence models, applied to supersonic air ejectors using CFD simulations. The shock strength, shock location and the average pressure recovery were the focus of the simulations. Supporting experiments were also performed and centreline pressure was measured throughout the ejector using a capillary probe. The results were compared with the measurements of Desevaux and Aeschbacher (2002). The best model was the $k-\omega$-SST model which demonstrated the best performance in stream mixing. In addition, the CFD simulations showed some ability to simulate the different operational modes of a supersonic ejector, ranging from the on-design point, with maximum flow rate (choking), to off-design, in which the secondary flow rate drops to zero (complete malfunctioning).

Normally, ejectors are classified as either constant mixing area (CMA) ejectors or constant pressure mixing (CPM) ejectors, as illustrated in Figure 2.8. The flow phenomena and performance of both types have been assessed using the CFD code FLUENT, operated with the $k-\epsilon$ turbulence model, and an ideal gas as the working fluid (Pianthong et al., 2007)

Sriveerakul et al. (2007a) compared CFD and experimental data. They studied a wide range of operating conditions and different geometries (such as a change in the primary nozzle arrangement) and they measured the performance of the steam ejector. The CFD simulation results were validated with data obtained from experimental work. In subsequent work, Sriveerakul et al. (2007b) continued to carry out the CFD simulations for flow phenomena inside the steam ejector. Figure 2.9 shows the entrainment ratio
change with back pressure. The operation of an ejector can be classified into three regions as shown in Figure 2.10: choked flow, unchoked flow and reversed flow. Despite increasing the back pressure, points A, B and C in the ejector still operated within the choked regime because the back pressure did not exceed the critical point value. Points D and E were unchoked and reversed flow respectively because the downstream pressure was increased more than the critical value.
2.4 Computational fluid dynamics simulations

Figure 2.10: Filled contours of Mach number: effect of downstream pressure. All operating points A, B, C, D and E correspond to those shown in Figure (2.9) (Sriveerakul et al., 2007b).

Hemidi et al. (2008) presented comparisons between CFD and experimental measurements. A wide range of pressure operating conditions was tested and they focused on the different behaviour of two turbulence models: $k-\varepsilon$ and $k-\omega$-SST. Figure 2.11 illustrates the differences between the two turbulence models, and how results vary depending on the primary pressure. The difference between the two models became very small when the primary pressure was increased. In contrast to the work of Bartosiewicz et al. (2005), good validation outcomes were obtained for the supersonic air ejector using the $k-\varepsilon$ model for both on-design and off-design conditions: Figure 2.11(b). The $k-\omega$-SST model gave similar results to the $k-\varepsilon$ model for the on-design conditions but over predicted the results at off-design conditions as shown in Figure 2.11(a) where the entrainment ratio was very low. Simulation at off-design conditions appears more sensitive to condenser pressure $P_c$.

Hemidi et al. (2009) continued to perform CFD using data obtained from earlier experiments reported in Hemidi et al. (2008) and focused primarily on off-design operation because it is a more complex situation. Figure 2.12 depicts the stream function for
the secondary flow near to the primary nozzle exit with different condenser pressure $P_c$ at off-design operation for the case of $P_1 = 6$ bar, and a secondary pressure of 1 bar. As the condenser pressure was increased, a flow detachment occurred close to the wall of the mixing chamber for the $k-\epsilon$ simulation. When the condenser pressure was further increased, this detachment became a large recirculation zone for $P_c = 1.25$ bar. When $P_c$ reached 1.3 bar, the recirculation claimed the majority of the area and the secondary flow rate fell to zero. However, in the case of the $k-\omega$-SST model, no such recirculation zone appeared and because the experimental configuration is more likely to have a recirculation zone, it is probable that the $k-\epsilon$ model performed better.
The performance of a steam ejector was simulated using the $k$-$\epsilon$ turbulence model by Varga, Oliveira and Diaconu (2009). The different ejector efficiencies (nozzle, suction, mixing and diffuser) were examined. The results indicated that nozzle efficiency can be deemed independent of operating conditions, and that this efficiency was only slightly affected by nozzle diameter. Suction efficiency was considered constant before the back pressure reached the critical value, and the suction efficiency dropped when back pressure went beyond the critical value. Other efficiencies increased with increasing back pressure until the critical value was reached.

Yang et al. (2012) investigated the coefficient of performance of the steam ejector using CFD simulation for five different nozzle structures: conical, elliptical, square, rectangular and cross-shape nozzles as shown in Figure 2.13. CFD simulation results were validated with experimental work conducted elsewhere (Sriveerakul et al., 2007a). In the CFD simulations, three turbulence models were tested: realizable $k$-$\epsilon$, standard $k$-$\epsilon$, and RNG $k$-$\epsilon$. The results show that the realizable $k$-$\epsilon$ model produced a wall static...
pressure closer to the experimental results than the other two models tested. The CFD simulation results also demonstrate that the entrainment ratio and critical back pressure of the rectangular nozzle were 7.1% and 21.3% lower respectively than for the conical nozzle, and the entrainment ratio and critical back pressure of the elliptic nozzle were 7.9% and 21.3% lower respectively than the conical nozzle. Relative to the performance of the conical nozzle, the square nozzle enhanced the entrainment ratio by about 2% and decreased the critical back pressure by about 2.1% while the cross-shape nozzle improved the entrainment ratio by about 9.1% and decreased the critical back pressure by about 6.4%.

![Nozzle shapes tested by Yang et al. (2012) (all dimensions in mm).](image)

Figure 2.13: Nozzle shapes tested by Yang et al. (2012) (all dimensions in mm).
Ruangtrakoon, Thongtip, Aphornratana and Sriveerakul (2013) explored the effect of primary nozzle geometries on the coefficient of performance of the steam ejector. Tests were performed at a fixed evaporator temperature of 7.5°C, boiler temperature range from 110°C to 150°C, and eight diameters of primary nozzle throat from 1.4 to 2.6 mm. A Mach number of 4 was used for six nozzles and two nozzles used a Mach number of 3 and 5.5. The simulations tested two turbulences models, $k$-$\omega$-$SST$ and realizable $k$-$\varepsilon$. The simulation results demonstrated that the $k$-$\omega$-$SST$ model was more accurate than the realizable $k$-$\varepsilon$ model when compared with experimental values obtained from the literature (Ruangtrakoon, Aphornratana and Sriveerakul, 2011).

2.5 Conclusion

Researchers have made a wide variety of assumptions to obtain solutions to the equations governing ejector operation. Such assumptions are not necessarily valid in all configurations and operating conditions.

Researchers have performed experiments with both open and closed cycles to determine the COP and a number of studies have focused on the nozzle shape and the ejector geometries.

Further contributions can be made in the experimental visualisation of ejector flows. Furthermore, until now, flow visualisation has not been used for steam ejectors.

CFD simulations have generally been successful for choked ejector operation, but when the secondary stream is not choked, significant differences in certain CFD models have been revealed.
Chapter 3

Overview of the design of the apparatus

3.1 Introduction

In an effort to understand the low coefficient of performance of the steam ejector, a new facility was built for operation in the university laboratory environment. The intention was to establish a facility which could be used in this work, and in future programs to help researchers to visualise the primary and secondary streams within the ejector, and also to visualise the mixing region between two streams as this region is particularly important for understanding the behaviour of the flow inside the ejector. This chapter will focus on the design of the steam ejector apparatus and the measurement methods used to characterise its performance. The steam ejector including all the parts such as the primary nozzle, the plenum chamber with the secondary inlet, the mixing chamber and diffuser will all be discussed. Furthermore, a new design of steam generator and evaporator chamber were used in this apparatus and will also be covered.
3.2 Apparatus layout

The apparatus was developed to evaluate the steam ejector performance and visualise the flow and mixing region between the primary and the secondary streams inside the mixing chamber. This apparatus was designed as a relatively large size rig for a university laboratory environment. A cooling capacity of about 3500 W was targeted to make this apparatus of a similar capacity to conventional air conditioning systems. The rig consists of an ejector, steam generator, evaporator, condenser, pumps, valves, and measurement equipment needed to characterize system performance.

A schematic diagram and photograph of the rig setup appears in Figures 3.1 and 3.2 respectively. The apparatus was built in various workshops and laboratories at the University of Southern Queensland.

Two key design features in the present apparatus which are not commonly seen in other systems are:

- Steam generation and supply without a large steam reservoir resulting in the ability to safely work with high pressure steam because the mass flow rate of steam through the system is very small (about 13 kg/hr if the system works at full capacity).

- An accurate method for determining mass flow rates based on a weighing method for both the secondary and the primary flows.

3.3 Steam ejector

The steam ejector represents the heart of the refrigerant cycle because it provides the compression effect. The steam ejector should be made from a material that can withstand the structural and thermal stresses imposed by the steam temperature and pressure. The choice of the material of the ejector should take into consideration issues such as machinability, cost, strength and resistance to wear and corrosion.
Overview of the design of the apparatus

Figure 3.1: Schematic diagram of the steam ejector system.

The steam ejector was designed with reference to Johannesen (1951), Molyneux (1963), ESDU86030 (1986), ESDU92042 (1992), and Frank (2003). The ejector was designed to entrain 5.1 kg/hr, at an evaporator temperature of 7°C and a pressure of 1.22 kPa when primary supply conditions were a mass flow rate of 12.7 kg/hr, at a temperature of 130°C, and a pressure of 270 kPa. This set of parameters is consistent with the production of a 3500 W cooling capacity. Figure 3.3 shows the ejector which includes nozzle, spacer, spacer holder, mixing chamber and diffuser. The steam ejector was fabricated at the University of Southern Queensland, except for the nozzle which was fabricated by Spray Nozzle Engineering (Melbourne, Australia).
Figure 3.2: Photograph of the steam ejector system.
Figure 3.3: (a) Schematic illustration of the steam ejector (b) Section A-A compression fitting (c) Section T-T threaded adapter.
The ejector is divided into four sectors for the purpose of design as illustrated in Figure 3.4.

- Nozzle sector from 0 to 1.
- Mixing chamber sector from 1 to 2.
- Parallel mixing chamber sector from 2 to 3.
- Diffuser sector from 3 to 4.

Several assumptions are used to formulate a one-dimensional theoretical model used in the design process:

1. The flow inside the ejector is one-dimensional and steady.
2. There is no heat exchange through the external wall of the ejector.
3. Mixing between primary and secondary flows is complete before entering the diffuser.
4. The entrained flow (the secondary flow) reaches a sonic velocity \( M = 1 \) at the hypothetical throat section H-H and the static pressure of both streams at this position is the same.

![Figure 3.4: Illustration of steam ejector sectors considered in design.](image-url)
### 3.3.1 Primary nozzle

A convergent-divergent nozzle was used as the primary nozzle which delivered the high energy steam to the ejector. The maximum mass flow rate passes through the nozzle when its throat is at the sonic condition \((M = 1)\) and the nozzle is said to be choked. The mass flow conservation through the nozzle can be written

\[
\dot{m}_{\max} = \rho^* A^* u^* = \rho_1 A_1 u_1
\]  

(3.1)

The secondary mass flow rate can be calculated from the cooling load by equation 3.2 when the enthalpy of the evaporation \(\Delta h\) is known

\[
Q_e = \dot{m}_s \Delta h
\]  

(3.2)

where

\[
\dot{m}_s = \text{Secondary mass flow rate}
\]

\[
\Delta h = \text{Enthalpy of evaporation} = 2484.85 \text{kJ/kg. (Moran and Shapiro, 1995)}
\]

\[
Q_e = \text{Cooling load} = 3500 \text{W}
\]

Thus \(\dot{m}_s = 1.41 \times 10^{-3} \text{kg/s} = 5.1 \text{ kg/hr.}\)

ESDU86030 (1986) recommends the correction of the mass flow rate by using two correction factors: the molecular weight correction factor \(C_1\) for gases other than air and the temperature correction factor \(C_2\) for gases at more than 20°C. Figure 3.5 illustrates the variation of the molecular weight correction factor \(C_1\) with the molecular weight of the gas. For present work with steam, the molecular weight is 18 and hence \(C_1 = 0.8\) from Figure 3.5, also \(C_2 = 1\) because the secondary flow temperature is less than 20°C. The air-equivalent secondary mass flow rate \(\dot{m}_{s,\text{air}}\) can be calculated

\[
\dot{m}_s = \dot{m}_{s,\text{air}} C_2 C_1 = 6.375 \text{ kg/hr}
\]  

(3.3)
Furthermore the mass flow rate of primary flow $\dot{m}_{\text{max}} = \dot{m}_p$ is given by

$$\dot{m}_s = \omega \dot{m}_p$$

(3.4)

where

$\omega$  
Entrainment ratio of the steam ejector

Estimates for the entrainment ratio can be obtained using ESDU86030 (1986) (see Figure 3.6) based on values of the primary ratio $N_p$ and compression ratio $N_s$, which are defined as

$$N_p = \text{primary ratio} = \frac{\text{primary pressure}}{\text{condenser pressure}}$$

$$N_s = \text{compression ratio} = \frac{\text{condenser pressure}}{\text{evaporator pressure}}$$

In the present design work, the required pressures are taken as:

Primary pressure = 270 kPa.

Condenser pressure = 5.5 kPa.
Evaporator pressure = 1.227 kPa.

and hence the required ratios are

\[ N_p = 49.09 \]
\[ N_s = 4.48 \]

From Figure 3.6, the entrainment ratio expected with these conditions is 0.5, and hence the required value of primary mass flow rate is \( \dot{m}_p = 12.75 \) kg/hr.

![Figure 3.6: Variation of compression ratio \( N_s \) with primary pressure ratio \( N_p \) and entrainment ratio \( \omega \) (ESDU86030, 1986).](image)

From equation (3.1) combined with isentropic relationships, the throat area can be written as (Huang et al., 1999)

\[
A^* = \frac{\dot{m}_p \sqrt{RT_o}}{P_o \sqrt{\gamma \eta N \left( \frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}}}^{\frac{\gamma+1}{\gamma-1}}}
\]  \hspace{1cm} (3.5)

The ratio of specific heats is taken as \( \gamma = 1.326 \) for steam.
The nozzle’s isentropic efficiency $\eta_N$ is defined as (Varga et al., 2009; Pridasawas, 2006).

$$\eta_N = \frac{h_{\text{enter}} - h_{\text{noz,exit}}}{h_{\text{enter}} - h_{\text{noz,exit,ise}}}$$ (3.6)

Huang et al. (1999), Cizungu, Mani and Groll (2001), Selvaraju and Mani (2004), and Zhu, Cai, Wen and Li (2007) identified the isentropic nozzle efficiency as $\eta_N \approx 0.95$.

Under these conditions the throat area expression becomes

$$A^* = \frac{\dot{m}_p \sqrt{RT_o}}{0.654 P_o}$$ (3.7)

In this design, stagnation conditions of primary flow were: pressure of 270 kPa and temperature of 130°C and hence $A^* = 8.5529 \times 10^{-6}$ m$^2$.

The area ratio $A_T$ can be estimated from standard ESDU86030 (1986) results presented in Figure 3.7. Figure 3.7 illustrates the variation of the nozzle area ratio $A_T$ and primary pressure ratio $N_p$ for different values of the entrainment ratio. In the present design, the required area ratio was estimated to be 17.4.

Kroll (1947) and Molyneux (1963) recommended a 10° included angle for the diverging section of the nozzle because at higher angles the flow may separate from the wall of the nozzle. This value of 10° was adopted in the present design. ESDU92042 (1992) recommended that the geometry for the convergent section of the nozzle should be a circular-arc profile with a radius of at least 0.3 $d^*$ to decrease the friction losses. In the present design, the arc radius was actually 1.8 mm (well in excess of this factor 0.3 $d^*$) as shown in Figure 3.8. Furthermore, the transition from the convergent section to the divergent section should be smooth without any change of curvature or discontinuity of slope.

Molyneux (1963) recommended the minimum area of steam pipe at the nozzle inlet (point 0 in Figure 3.4) be given by

$$A_{\text{pipe}} \geq 3 A^*$$ (3.8)

In the present design, the values of $A_{\text{pipe}}$ achieved was actually $7.854 \times 10^{-5}$ m$^2$ (well in excess of the factor of 3 in area).
Figure 3.7: Variation of nozzle area ratio $A_T$ with primary pressure ratio $N_p$ for different values of entrainment ratio $\omega$ (ESDU86030, 1986).

From nozzle throat area $A^*$, the nozzle area ratio $A_T$ and the nozzle divergence angle, the length of the diverging part of the nozzle can be calculated. The primary nozzle was made from stainless steel (316) and was manufactured by Spray Nozzle Engineering (Melbourne, Australia).

Figure 3.8 shows the cross section of the nozzle, and for more information and details see Appendix A.
Figure 3.8: (a) Overall dimensions of the nozzle, all dimensions (mm) (b) Throat profile details.
3.3.2 Secondary inlet

It is important for the design to minimise loss in the secondary inlet in order to enhance the performance of the ejector. The secondary flow moves into the mixing chamber via an annular gap between the primary nozzle and the body of ejector and this region is called the secondary inlet. The secondary inlet should be very smooth with no sharp constrictions or expansions. The best approach would be to have the secondary flow pass through a plenum chamber and thence into the mixing chamber via a bellmouth, but this arrangement may not be feasible if there is insufficient space. Fabrication cost may also prevent this arrangement. The secondary inlet can have a conical shape when bellmouth entry is not possible, and the best cone full angle lies between 20° and 40° (ESDU86030, 1986). However, the conical configuration gives slightly higher losses and the inlet should be as short as possible to reduce the friction losses (Kastner and Spooner, 1950; Johannesen, 1951). Figure 3.4 illustrates the inlet length $L_n$. The plenum chamber was made from stainless steel (316) at the University of Southern Queensland, and more details on the plenum chamber and secondary inlet chamber are presented in Appendix A.

3.3.3 Mixing chamber

Normally, mixing chambers are classified as either constant area mixing, or constant pressure mixing. In the case of constant pressure mixing, a converging cross sectional area merged with a short parallel section in generally used. ESDU86030 (1986) advises the use of constant pressure mixing for steam as the primary flow and a gas as the secondary flow. Constant area mixing is recommended when steam is the primary flow and a liquid is the secondary flow ESDU86030 (1986), but the basis for these recommendations is not clear.

Flugel (1939), Johannesen (1951) and ESDU86030 (1986) give recommendations about the mixing chamber length. The distance from nozzle lip to the start of the diffuser should be in the range of 5 to 10 $D$ where $D$ is the diameter of the parallel section. The half angle of the conical section of the mixing chamber $\theta_m$ (as defined in Figure 3.4) should be within the range from 2° to 10°.
In the parallel section, shock waves occur. Flugel (1939), Johannesen (1951) and ESDU 86030 (1986) recommended the length of the parallel section should be between 2 and 4\(D\). However ASHRAE (1983) suggested the length of parallel section to be 3 to 5\(D\); Chang and Chen (2000) suggested the length of parallel section to be 5\(D\).

In this study, the total mixing chamber length used was about 9\(D\), and the parallel section length was 3\(D\), where \(D= 25.4\text{ mm}\). Two materials have been used for the mixing chamber: one of them aluminium and the other transparent perspex (Acrylic). Figure 3.9 shows the converging conical section and the constant area section of the mixing chamber made from transparent perspex (Acrylic). The optical design of the perspex mixing chamber is discussed in chapter 8 and more details on the mixing chamber construction can be found in Appendix A.
Overview of the design of the apparatus

Figure 3.9: Photographs of the perspex mixing section. (a) Converging section of the mixing chamber (b) Parallel section of the mixing chamber (c) Assembled mixing chamber.

3.3.4 Diffuser

When the two streams reach the end of the mixing chamber, the mixed flow is subsonic and passes into the diffuser, reducing the velocity and increasing the pressure. The subsonic diffuser has a conical configuration. Johannesen (1951) and ESDU86030 (1986) recommended that the half angle ($\theta_d$ in Figure 3.4) should be in the range from $3^\circ$ to $4^\circ$ and in no case should it be greater than $7^\circ$. If the mixing chamber is actually short, producing a highly non-uniform flow, a smaller included angle should be utilised. Previous researchers recommended that the area ratio of the diffuser ($A_4/A_3$) should
be no more than 5 (Johannesen, 1951; ESDU86030, 1986).

In this study, the half angle $\theta_d$ and the area ratio used were $3.3^\circ$ and 2 respectively. This diffuser was made from brass at the University of Southern Queensland. Figure 3.10 presents a photograph of the diffuser and more details on its dimensions can be found in Appendix A.

![Figure 3.10: Diffuser made from brass.](image)

### 3.4 Steam generator

Most previous researchers working on steam ejector systems including Eames, Aphornratana and Haider (1995), Sun (1997), Aphornratana and Eames (1997), Eames, Sun, Worall and Aphornratana (1999), Sriveerakul et al. (2007a) and Meyer et al. (2009) have all used conventional steam generators (boilers). These devices consist of a pressure vessel and element heaters as shown in Figure 3.11. However, for the present work a different steam generator has been designed. Table 3.1 presents some features for both steam generator options.
### Table 3.1: Features for both steam generator options.

<table>
<thead>
<tr>
<th>Present steam generator</th>
<th>Conventional steam generator</th>
</tr>
</thead>
<tbody>
<tr>
<td>• Elevated pressure is established inside the copper tube only. The copper tube works as a pressure vessel.</td>
<td>• A pressure vessel system is used, see Figure 3.11.</td>
</tr>
<tr>
<td>• Relatively small amount of energy stored in the system leading to improved operational safety. For example, the present ejector is designed to work with a primary flow rate of 13 kg/hr, and if a leak occurs, the amount of the steam released will be about 0.32 kg (all the mass of the steam inside the copper tube), if the steam generator is working with full capacity.</td>
<td>• Relatively large amount of energy stored in the system representing a potential hazard. For example, for an ejector primary stream capacity of 13 kg/hr, that means at least 12 kg of water should be in the pressure vessel to keep the water level higher than the element heater. If a leak occurs, the full mass of the water may be released as steam if the steam generator is working with full capacity.</td>
</tr>
<tr>
<td>• Suitable for fabrication in the university laboratory environment.</td>
<td>• Pressure vessel fabrication required.</td>
</tr>
<tr>
<td>• Relatively inexpensive to fabricate.</td>
<td>• Expensive to fabricate.</td>
</tr>
<tr>
<td>• Can produce both saturated and superheated steam, like a water tube boiler.</td>
<td>• Can produce saturated steam only, like a fire tube boiler.</td>
</tr>
<tr>
<td>• No extra system is needed if superheated steam is required.</td>
<td>• Extra superheated element if superheated steam is required.</td>
</tr>
</tbody>
</table>

![Figure 3.11: Schematic boiler as used by Sriveerakul et al. (2007a).](image)

Table 3.1: Features for both steam generator options.

- Elevated pressure is established inside the copper tube only. The copper tube works as a pressure vessel.
- Relatively small amount of energy stored in the system leading to improved operational safety. For example, the present ejector is designed to work with a primary flow rate of 13 kg/hr, and if a leak occurs, the amount of the steam released will be about 0.32 kg (all the mass of the steam inside the copper tube), if the steam generator is working with full capacity.
- Suitable for fabrication in the university laboratory environment.
- Relatively inexpensive to fabricate.
- Can produce both saturated and superheated steam, like a water tube boiler.
- No extra system is needed if superheated steam is required.

- A pressure vessel system is used, see Figure 3.11.
- Relatively large amount of energy stored in the system representing a potential hazard. For example, for an ejector primary stream capacity of 13 kg/hr, that means at least 12 kg of water should be in the pressure vessel to keep the water level higher than the element heater. If a leak occurs, the full mass of the water may be released as steam if the steam generator is working with full capacity.
- Pressure vessel fabrication required.
- Expensive to fabricate.
- Can produce saturated steam only, like a fire tube boiler.
- Extra superheated element if superheated steam is required.
Figure 3.12 illustrates the arrangement of the steam generator used in the present work. This steam generator uses thermal oil (Mobiltherm 605), with a copper tube coil carrying the water immersed in thermal oil, element heaters and a stirrer.

The Mobiltherm 605 is a high quality heat transfer medium made from solvent refined mineral oil (ExxonMobil, 2001). The thermal oil works as heat transfer medium between the electrical heating elements and the copper tube. Its features include (ExxonMobil, 2001):

- Highly resistant to thermal cracking and chemical decomposition reaction.
- Non toxic in the event of leakage.
- Maximum pour point -6°C.
- Minimum flash point 212°C at atmospheric pressure.

For the sizing of the steam generator, the length of the copper tube depends on a number of reasonable assumptions such as

- Constant oil temperature along the length of the copper tube.
- Natural convection between the oil and the copper tube.

For the oil-side heat transfer coefficient calculation, natural convection was assumed and the following empirical equation (Kreith and Bohn, 2000) was used

$$\text{Nu} = 0.53(\text{GrPr}^2)^{0.25}$$

where

$$\text{Nu} \quad \text{Nusselt number} = \frac{h_o D_o}{k}$$

$$\text{Gr} \quad \text{Grashof number} = \frac{g \beta (T_s - T_f) D_o^3}{\nu^2}$$

$$\text{Pr} \quad \text{Prandtl number} = \frac{\mu C_p}{k}$$
In the present work, the above values were \( \text{Nu} = 15.6 \), \( \text{Gr} = 2 \times 10^7 \), and \( \text{Pr} = 0.194 \) and hence the outer heat transfer coefficient was \( h_o = 14.07 \, \text{W/m}^2\text{K} \).

For the steam-side heat transfer coefficient calculation, the copper tube was considered in two sections. For the first section, the Reynolds number was calculated according to the mean properties of liquid water from ambient temperature to 100°C, and for the second tube section, average steam properties from 100°C to 150°C were used. In this study the pipe flow Reynolds number was 1125, and hence the Nusselt number was 3.66 at constant temperature (Kreith and Bohn, 2000). The length of each section of the copper tube was then calculated using

\[
L = \frac{q_i}{(\pi D_i h_i LMTD)}
\]  

(3.10)
where

\[
L = \text{Length of copper tube} = 30 \, \text{m}
\]

\[
q = \text{Heat transfer from oil to water} = 10,000 \, \text{W}
\]

\[
D_i = \text{Inside diameter of copper tube} = 0.0106 \, \text{m}
\]

\[
h_i = \text{Inside heat transfer coefficient} = 231.66 \, \text{W/m}^2\text{K}
\]

\[
LMTD = \text{Log mean temperature difference} = 44.4 \, \text{K}
\]

According to this approach, the total required length of copper tube was calculated as 30 m. The tube was fabricated as a helical coil with outside diameter of 260 mm, the distance between successive turns was 22.7 mm and 37 turns were used. The copper tube which was used was type A with an outer diameter of 12.7 mm and a wall thickness of 1.02 mm. Two pieces of copper tube 18 m and 12 m in length were silver brazed to make them one piece in the mechanical workshop. The coil was proof tested after reshaping and joining at 7 bar in the laboratory.

The idea of this design is that the steam will be generated inside the copper tube only. In this design the copper tube works as a pressure vessel, and water is introduced into the coil by a dosing pump after the thermal oil reaches the set temperature. The steam generator has the capability to supply a flow rate between 0.001 and 13.5 kg/hr at temperatures up to 135 °C and pressures of more than 400 kPa. It is insulated by rock wool insulation of 100 mm thickness with a heavy duty foil sheet.

Four screw-in immersion elements heaters of type 2”BSP Incoloy Sheath 2623 supplied by Stokes Synertec Australia, were used in the steam generator and each one has a capacity of 3750 W and is rated at 31 kW/m² with a length of 847 mm. This type of element is intended for oil use only.

An oil stirrer was also developed. It consisted of a 12 DC motor with a speed controller driving a paddle on a vertical rod supported with two high temperature ball bearings. The stirrer was designed to improve the heat transfer coefficient between elements and oil. Further details about the steam generator and stirrer can be found in Appendix A.
3.5 Evaporator

The evaporator was designed for a 3500 W cooling load. The shape of the evaporator was arranged to increase the evaporative area instead of using a circulation pump to improve the rate of evaporation. The inner diameter of the evaporator was 280 mm with a total height of 270 mm. It was made from galvanised steel sheet with a thickness of 1 mm for the wall and 2 mm for the base. A heater element type 2388-10B with a rating of 3600 W and a single phase from Stokes Synertec Australia was used and filled most of the base of the evaporator. The heater element was mounted about 15 mm above the base of the evaporator. The evaporator was insulated by armaplex insulation with a thickness of 10 mm to reduce the external heat gain by the evaporator. Further details and dimensions on the evaporator can be found in Appendix A.

3.6 Evaporator load control

A single phase half controlled rectifier was used to control the load of the evaporator. At the same time, a Watt meter was used to measure the electrical power. A single phase half controlled rectifier has the capability to control the load with a range from zero to full capacity of the element heater. The rectifier and Watt meter were tested in the electrical laboratory in the Faculty of Engineering at the University of Southern Queensland. Errors were within the range from 0.1 % to 0.22 % when the load was varied from 400 W to 3500 W. For more details see Appendix A

3.7 Evaporator chamber

A special evaporator chamber was developed for this system in order to weigh the amount of water induced from the evaporator during the test time. The weighing method was thought to be more accurate than other methods of comparable cost in this application. The evaporator chamber included a ViBRA balance type CJ-15KE, the evaporator, flexible ducts and two valves as illustrated in Figure 3.13. The main idea of the chamber was to create the same environmental pressure inside and outside
of the evaporator to let the balance work normally without any effect from the lifting force on the evaporator which would otherwise come from the pressure differential between the inside and outside of the evaporator.

Figure 3.13: Evaporator chamber. (a) Schematic illustration of arrangement; (b) Photograph of device.
3.8 Condenser

The mixture of primary and secondary flows from the diffuser of the ejector was passed into the condenser. It included two coils made from copper tube type B, 19.05 mm outside diameter and 1.02 mm wall thickness. The coils were fixed inside a shell with a diameter of 300 mm and height of 920 mm. A pressure transducer and a pressure gauge were mounted on the top of the condenser and a thermocouple well 350 mm deep was positioned in the middle top of condenser. A sight glass was mounted on the side of the condenser for visual identification of the level of water condensation. A centrifugal pump for the laboratory water was used to cool the condenser system. The condenser was designed to reject heat at a rate of about 13.5 kW. For more details see Appendix A.

3.9 Measurement arrangement

The instrumentation in this rig comprised thermocouples, pressure transducers and balances. Five thermocouples of type K were used, two of them were mounted in the inlet and outlet of the circulation water in the condenser, and one each for the condenser temperature, the primary flow temperature, and the secondary flow temperature. Sixteen pressure transducers were used, 13 of them were placed along the ejector, one in the condenser and the evaporator, and the last one was on the delivery line of the primary flow. Figure 3.14 illustrates the measurement equipment on the apparatus.

3.9.1 Pressure transmitter

Two types of pressure transducers were used in this apparatus. Wika model 10-A in two ranges were used: one range for the high pressure (primary steam pressure) region, and the other range for lower pressure as appropriate for the ejector, evaporator and condenser regions.

Kulite transducers (model IS-XTL-190) were used for the pitot tube data acquisition
3.9 Measurement arrangement

because of their high frequency response. Selected details of these transducers are presented in Table 3.2.

Table 3.2: Specifications of pressure transducers (Wika, 2011; Kulite, 2011).

<table>
<thead>
<tr>
<th>Type</th>
<th>Pressure range (bar)</th>
<th>Non-Linearity (%)</th>
<th>Operating temperature range (°C)</th>
<th>Accuracy (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wika 10-A</td>
<td>0 - 1</td>
<td>0.5</td>
<td>-30 ··· +80</td>
<td>0.5</td>
</tr>
<tr>
<td>Wika 10-A</td>
<td>1 - 7</td>
<td>0.5</td>
<td>-30 ··· +80</td>
<td>0.5</td>
</tr>
<tr>
<td>Kulite</td>
<td>0 - 0.7</td>
<td>0.5</td>
<td>-55 ··· +175</td>
<td>0.5</td>
</tr>
<tr>
<td>IS-XTL-190</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
3.9.2 Balance

Two ViBRA type CJ-15KE precision tuning-fork balances were used in this rig to weigh the mass of water delivered to the primary and secondary streams. This balance promises excellent performance in any environment: humid, dust oil splashed and dark conditions. Maximum load of 15,000 g, readability 1 g, non-linearity ± 1, repeatability 1 g, and with external weight calibration.

3.10 Data acquisition

All pressure transducers and thermocouples were wired to a National Instruments Compact Data Acquisition (cDAQ) system. A programme was written by specialist engineers in the Faculty of Engineering at the University of Southern Queensland using National Instruments LabView which has interfaces via the NI-DAQ drivers and USB. It consists of chassis NIcDAQ 9178 with a number of signal conditioning amplifiers and analogue to digital conversion modules. The first group of two NI 9223 4-channel (± 10 V, 16 Bit simultaneous analogue input) modules was connected to the Kulite pressure transducers and sampled at a frequency of about 1 MHz. The second group of modules included one NI 9205 32-Channel (± 10 V to ± 200 mV, 16-Bit analogue input) module which was used for the Wika pressure transducers and two sets of NI 9219 4-channel universal analogue input both of them operating at 100 Hz to convert thermocouple readings to digital values representing degrees Celsius or volts.

3.11 Electrical arrangement

An enclosure was developed containing the main power switch, other switches, circuit breakers, contactors for all electrical components of the apparatus and an emergency switch. The steam generator had four heater elements, each one with a capacity of 3750 W (3 phase, 440 volt). To control these elements, an advanced digital temperature controller and solid state relay (SSR) were used. The temperature controller OMRON type E5CN-HQ2m-500 was arranged to work with the SSR to control the temperature.
3.12 Conclusion

A new apparatus has been developed for research on steam ejectors at the University of Southern Queensland featuring a steam generator system and a novel method for measuring secondary stream mass flow rate. The system is of an open type and this is expected to facilitate future work using particle image velocimetry and similar techniques where seed particles will be injected into one of the streams.
Chapter 4

System performance

4.1 Introduction

The coefficient of performance of the apparatus is a very important indicator of system effectiveness. After completing the design and fabrication of the apparatus, it was necessary to confirm system performance before commencing the main tests. This chapter will present the calculation of the coefficient of performance of the steam ejector apparatus, measurement methods of the mass flow rates of the primary and the secondary streams, the estimation of heat losses from the main delivery line of the steam, and the heat gained by the evaporator. Identification of the critical condenser pressure will also be described.

4.2 Performance of the open cycle

The coefficient of performance of the open cycle steam ejector can be calculated from equation (4.1). Figure 4.1 illustrates the open cycle of the steam ejector used in this work.

\[
COP = \frac{Q_e}{(Q_g + W_p)}
\]  (4.1)

where
4.2 Performance of the open cycle

Figure 4.1: Illustrative of p-h diagram of the open cycle steam ejector.

\[ \text{COP} = \frac{\dot{m}_s(h_6 - h_9)}{\dot{m}_p(h_4 - h_3)} \] (4.2)

\( \text{COP} \) = Coefficient of performance

\( Q_e \) = Cooling load in the evaporator

\( Q_g \) = Input heat to the steam generator

\( W_p \) = Pump work

In the present study the pump work was about 24 W, while the steam generator heat input was about 7800 W. Therefore, the pump work represents about 0.3% of the steam generator heat input and hence the pump work can be neglected in the calculation of COP.

The COP for the present case (an open cycle) can be calculated as
\[ COP = \omega \frac{(h_6 - h_9)}{(h_4 - h_3)} \] (4.3)

This method for the calculation of the COP for the open system agrees with the approach of Sun (1997) and Meyer et al. (2009).

To accurately deduce the COP, it is important to accurately define both the entrainment ratio \( \omega \) from measurements of the mass flow rates and the enthalpy differences from measurements of temperatures and pressures at the key points. In the following section, the impact of measurement methods and system settings is assessed. The impact of likely heat losses within the primary steam pipe, the heat gain within the evaporator, and the electrical noise in the instrumentation is also assessed.

### 4.2.1 Measurement of mass flow rate

1. The primary mass flow rate was calculated by two methods:

   - **Weight method.**
     
     The weight of the water tank was measured before the start of the test and after the finish of the test
     
     \[ \dot{m}_p = \frac{\text{mass before test} - \text{mass after test}}{\text{test time}} \] (4.4)

   - **Dosing pump method.**
     
     The dosing pump (GRUNDFOS, model DDA 17-7) was used to supply water to the system generator. It has the capability to change the water flow rate from 0.001 to 17 litre/hr, with a maximum pressure of 700 kPa.
     
     The display of volumetric flow rate on the dosing pump makes its operation very convenient. Volumetric flow rates were converted to mass flow rates by taking the density of water as 998 kg/m\(^3\). Details for the application of the dosing pump are presented in Appendix B. Calibration of the dosing pump is discussed in Appendix C.

The differences between the two methods of the mass flow rate measurement are presented in Table 4.1. From the results presented in Table 4.1, the mean
4.2 Performance of the open cycle

The difference between the two methods is 0.5%. This value is very small and will not have a large impact on the calculated COP of the system.

Table 4.1: Comparison of primary mass flow rate measurement by two methods.

<table>
<thead>
<tr>
<th>Weight method $\dot{m}_p$ (kg/hr)</th>
<th>Dosing pump $\dot{m}_p$ (kg/hr)</th>
<th>Percentage error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>9.92</td>
<td>10</td>
<td>0.8</td>
</tr>
<tr>
<td>11.432</td>
<td>11.5</td>
<td>0.59</td>
</tr>
<tr>
<td>12.15</td>
<td>12.2</td>
<td>0.41</td>
</tr>
<tr>
<td>12.66</td>
<td>12.7</td>
<td>0.315</td>
</tr>
<tr>
<td>13.1</td>
<td>13.2</td>
<td>0.75</td>
</tr>
<tr>
<td>13.65</td>
<td>13.7</td>
<td>0.36</td>
</tr>
</tbody>
</table>

2. The secondary mass flow rate was also calculated by two methods:

- **Weight method.**
  
  This method is basically the same as that for the primary mass flow rate.

- **Electrical cooling load method.**
  
  The secondary mass flow rate can be calculated if $h_6$ and $h_9$ are known and the net heating power delivered to the water in the evaporator is measured. If the net heating power is taken as the power input to the evaporator element heater measured by the Watt meter $Q_{ee}$, then the $\dot{m}_s$ can be determined from the following expression.

  $$\dot{m}_s = \frac{Q_{ee}}{h_6 - h_9}$$

(4.5)

The non-electrical component of heating power in the evaporator is assessed in Section 4.2.4. Table 4.2 presents a comparison between the two methods of secondary stream mass flow measurement. The magnitude of the difference generally increases with a decrease in the load. One possible contribution is the size of the flexible hose which connects the evaporator and the evaporator chamber (see Figure B.1). This hose may not have been sufficiently large to ensure equilibrium pressure conditions over short time scales, and this may have led to the observed fluctuations in the scale readings during the test time. The
highest percentage difference between the two methods was 2.6%. This difference does not have a large influence on the value of the calculated COP of the system.

Table 4.2: Comparison of secondary mass flow rate measurement by two methods.

<table>
<thead>
<tr>
<th>Load (Watt)</th>
<th>Weight method $\dot{m}_s$ (kg/hr)</th>
<th>Electrical load $\dot{m}_s$ (kg/hr)</th>
<th>Percentage error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3500</td>
<td>5.064</td>
<td>5.112</td>
<td>+0.93</td>
</tr>
<tr>
<td>3000</td>
<td>4.25</td>
<td>4.3</td>
<td>+1.16</td>
</tr>
<tr>
<td>2500</td>
<td>3.64</td>
<td>3.6</td>
<td>-1.11</td>
</tr>
<tr>
<td>2000</td>
<td>2.98</td>
<td>2.92</td>
<td>-2.0</td>
</tr>
<tr>
<td>1000</td>
<td>1.49</td>
<td>1.452</td>
<td>-2.6</td>
</tr>
</tbody>
</table>
4.2 Performance of the open cycle

4.2.2 Heat loss from primary stream within the stainless steel pipe

Heat loss from the primary stream within the stainless steel pipe (between the temperature measurement position and the nozzle inlet) was estimated using detailed engineering methods as presented in Appendix B. For a measured steam pressure of 270 kPa and temperature of 150°C, the total heat loss from the pipe was estimated at 9.4 W, resulting in a decrease of the temperature of the steam of about 1.8°C. Figure 4.2 illustrates the details of the stainless steel pipe for the purposes of heat transfer calculation. The resulting heat loss represents 0.12% of the total energy entering the nozzle. This result does not have a large impact on the deduced COP of the system.

![Diagram of stainless steel pipe within insulation and concentric copper pipe](image1)

![Cross section of region 1 and region 2](image2)

Figure 4.2: (a) Schematic of stainless steel pipe within insulation and concentric copper pipe; (b) Cross section of region 1 and region 2.
4.2.3 Head losses from primary stream within the stainless steel pipe

A pressure drop of the steam flow within the stainless steel pipe from the measurement station to the nozzle inlet of 0.38 Pa was determined from equation (4.6).

\[ p = \rho g h_f \]  

\[ h_f = f(L/d)\left(\frac{V^2}{2g}\right) \]  

For more details on this calculation, see Appendix B.

The value of pressure drop relative to the stagnation pressure amounts to 0.00014%. This result is very small relative to other pressure measurement uncertainties. Thus, this value was not included in the analysis process.

4.2.4 Non-electrical heat gain by the evaporator

The evaporator was insulated with 10 mm thick armaflex insulation to keep the temperature of water between 6 and 14°C during electrical heating element operation and the arrangement is shown in Figure 4.3. Table 4.3 presents the estimated non-electrical heat gain by the evaporator for the three cases. Details on the calculation methods are presented in Appendix B.
The results presented in Table 4.3 show that the amounts of non-electrical heat gain were very minimal compared with the electrical heat gain. The magnitude of these values do not have a large impact on the deduced COP of the system so corrections will not be included in the analysis.

Table 4.3: Estimates of non-electrical heat gain by the evaporator.

<table>
<thead>
<tr>
<th>Evaporator temperature</th>
<th>Electrical heat gain (W)</th>
<th>Non-electrical heat gain (W)</th>
<th>Non-electrical to electrical (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>14°C</td>
<td>3500</td>
<td>6.5</td>
<td>0.18</td>
</tr>
<tr>
<td>10°C</td>
<td>2500</td>
<td>8.3</td>
<td>0.33</td>
</tr>
<tr>
<td>6°C</td>
<td>1200</td>
<td>11.7</td>
<td>0.97</td>
</tr>
</tbody>
</table>
4.2.5 Pressure transducer electrical noise

Pressure transducers and associated amplifiers were tested in the laboratory at University of Southern Queensland to determine their noise characteristics. Signals from the pressure transducers and amplifiers were recorded over a period of more than ten minutes by the data acquisition system. The electrical noise was very small, amounting to about 0.004% of the typical pressure values.

4.3 Preliminary test

Before performing the main tests, a number of preliminary tests were performed to evaluate parameters such as the velocity coefficient, the discharge coefficient of the nozzle, and the nozzle exit position. The measurement of pressure fluctuations during the test time, and the identification of the critical condenser pressure of the steam ejector system was also achieved in these tests.

4.3.1 Examination of the nozzle parameters

Two performance characteristics are generally used to characterise the primary nozzle for practical applications: the effectiveness of converting pressure energy to kinetic energy, and the mass flow capacity. The effectiveness of energy conversion and mass flow capacity are measured in terms of the velocity coefficient ($C_v$) and the nozzle discharge coefficient ($C_d$) respectively and these are defined in the studies of Grey and Wilsted (1948) and Sheshagiri, Sridhara and Paranjpe (1969) as

$$C_v = \frac{\text{Actual velocity}}{\text{Theoretical velocity}}$$

$$C_d = \frac{\text{Actual mass flow rate}}{\text{Theoretical mass flow rate}}$$

From the present work, the mass flow rate was measured at 12.8 kg/hr for stagnation conditions of 270 kPa and 130°C. The actual velocity at the throat was calculated from the equation

$$V_{\text{actual}} = \frac{\dot{m}_{\text{actual}}}{\rho^* A^*} = 442.2 \text{ m/s}$$
where

\[ A^* = \text{Throat area} = 8.5529 \times 10^{-6} \text{ m}^2 \]

\[ \rho^* = \text{Density at the throat area} = 0.94 \text{ kg/m}^3 \]

The theoretical velocity was estimated from equation (4.11) when the Mach number at the throat was sonic.

\[ a = \sqrt{\frac{\gamma RT^*}{\rho^*}} \quad (4.11) \]

With \( \gamma = 1.326 \), \( R = 461.5 \text{ J/kgK} \), and temperature at the throat = 346.5 K, the theoretical velocity at the throat is \( V_{\text{theoretical}} = 460.48 \text{ m/s} \) and the effectiveness of energy conversion is \( C_v = 0.96 \).

The theoretical mass flow rate at the nozzle throat can be determined from equation (4.12)

\[ \dot{m}_p = \frac{A^* P_0}{\sqrt{T_0}} \sqrt{\frac{\gamma}{R}} \left( \frac{2}{\gamma + 1} \right)^{\frac{(\gamma + 1)}{(\gamma - 1)}} \quad (4.12) \]

Using \( \gamma = 1.326 \), \( R = 461.5 \text{ J/kgK} \), and stagnation temperature = 403 K, the theoretical mass flow rate is calculated at 12.95 kg/hr. The actual (measured) mass flow rate was 12.8 kg/hr, giving the discharge coefficient of the nozzle \( C_d = 0.98 \). Grey and Wilsted (1948) estimated \( C_v \) and \( C_d \) for a nozzle with a half cone angle of 5\(^\circ\) to be 0.945 and 0.975, respectively. Sheshagiri et al. (1969) determined that \( C_d = 0.96 \) during experimental tests on a number of conical nozzles. Geatz (2003) calculated \( C_d = 0.9915 \) from an analytical solution.

### 4.3.2 Nozzle exit position (NXP) effect on COP

The position of the nozzle exit can have a strong influence on the system COP. Movements in nozzle exit position NXP can cause an increase or decrease in the COP and cooling load. ESDU86030 (1986) does not provide precise recommendations on the optimal NXP for constant pressure mixing chambers. In the present work, four positions of the nozzle were tested: 6, -4, -18 and -32 mm. These distances refer to the distance from the nozzle lip to the start of the mixing chamber, as shown in Figure
3.4. Experiments were conducted by setting a primary stream pressure of 270 kPa and temperature of 130°C and evaporator temperatures of 6, 10 and 14°C. COP results are shown in Figure 4.4. For each condition, the highest COP occurred at NXP = -32 mm (which was the test limit) and values varied with evaporator load as indicated in Table 6.3. The tests reveal that the COP increases when the primary nozzle is retracted (moved upstream) because this results in an increase in the area available to the secondary flow. As a result, more entrained mass has access to the mixing chamber, and at the same time, the pressure at the diffuser exit decreases. The minimum NXP that can be reached is -32 mm because there is a physical limit inside the ejector (a spacer between the steam supply and the nozzle inlet). However, moving the primary nozzle into the mixing chamber reduced the COP: it reduces the area available for secondary flow and at the same time, the pressure at the diffuser exit increases. These results agree with ESDU86030 (1986) and Chunnanond and Aphornratana (2004). Further details on the experimental work can be found in Appendix B.

Figure 4.4: Effect of nozzle exit position on COP for primary stream stagnation conditions of 130°C and 270 kPa.
Table 4.4: Variation of COP with evaporator load for NXP = -32 mm.

<table>
<thead>
<tr>
<th>Evap. Temp. 6°C</th>
<th>Load (W)</th>
<th>COP</th>
</tr>
</thead>
<tbody>
<tr>
<td>1200</td>
<td>0.175</td>
<td></td>
</tr>
<tr>
<td>Evap. Temp. 10°C</td>
<td>2500</td>
<td>0.34</td>
</tr>
<tr>
<td>Evap. Temp. 14°C</td>
<td>3500</td>
<td>0.485</td>
</tr>
</tbody>
</table>

4.3.3 System stability during the test time

The stability of the pressure and other readings during the test time could have an impact on the evaluation of the system performance. Figure 4.5(a) presents the primary stream pressure during the test time for three primary stream pressures: 270 kPa, 200 kPa and 143 kPa. The reading of the pressure transducer at the nominal condition of 270 kPa fluctuated between a maximum pressure of 271.35 kPa and a minimum pressure of 267.8 kPa, indicating about 0.6% fluctuation during the time period. Figure 4.5(b) shows the corresponding temperatures of primary stream. Similar results are presented in Figures 4.6 and 4.7 at different locations within the system and results are summarized in Tables 4.5 to 4.9. In Table 4.5, the fluctuations ranged from ±0.6% at 270 kPa to ±3% at 200 kPa respectively. These variations are within an acceptable range, and do not have a large impact on the COP. Tables 4.7 and 4.8 indicate the variation of the evaporator pressure and evaporator water temperature, respectively. The highest value of the variation was -4.4% and -3.1% for pressure and temperature, respectively. The evaporator pressure and temperature were influenced by fluctuations in the primary stream conditions. The evaporator always had the lowest pressure and temperature point in the cycle. The variation of -4.4% is approaching the limits of acceptability. Condenser pressure variation during the time test is presented in Table 4.9. The highest percentage variation was -1.19%.

There are 13 pressure transducers placed along the ejector. Figure 4.8(a) illustrates the configuration of the ejector, while Figure 4.8(b) shows the variation of pressure along the ejector. The error bars in this figure indicate the maximum and minimum pressure values recorded for the 100 samples during a 10 second interval. Figure 4.8(c) displays
the corresponding results for 6000 samples in 10 minutes. There is not a significant
difference between these results: the fluctuation range of all the pressure transducers
was from 0.28 % to 0.75 %.

Figure 4.5: (a) Fluctuations of the primary stream pressure during the test time for
nominal primary stream pressures of 270 kPa, 200 kPa and 143 kPa; (b) Corresponding
fluctuations of the primary stream temperature during the test time. Result for NXP =
-32 mm.
Figure 4.6: (a) Fluctuations of the evaporator pressure during the test time for three loads 3500 W, 2500 W and 1200 W. (b) Corresponding fluctuations of the secondary flow temperature during the test time. Result for NXP = -32 mm.
Figure 4.7: Fluctuations of the condenser pressure during the test time for NXP = -32 mm, and conditions corresponding to Figure 4.6.
Table 4.5: Variation of primary stream stagnation pressure during the test time.

<table>
<thead>
<tr>
<th>Case</th>
<th>Pressure (kPa) (%)</th>
<th>Pressure (kPa) (%)</th>
<th>Pressure (kPa) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max</td>
<td>271.35 +0.6</td>
<td>202 +3</td>
<td>147.6 +2.6</td>
</tr>
<tr>
<td>Mean</td>
<td>269.6</td>
<td>196</td>
<td>143.8</td>
</tr>
<tr>
<td>Min</td>
<td>267.8 -0.6</td>
<td>190 -3</td>
<td>140.29 -2.4</td>
</tr>
</tbody>
</table>

Table 4.6: Variation of primary stream stagnation temperature during the test time.

<table>
<thead>
<tr>
<th>Case</th>
<th>Temperature (°C) (%)</th>
<th>Temperature (°C) (%)</th>
<th>Temperature (°C) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max</td>
<td>130.539 +0.19</td>
<td>120.99 +0.6</td>
<td>110.74 +0.44</td>
</tr>
<tr>
<td>Mean</td>
<td>130.34</td>
<td>120.39</td>
<td>110.3</td>
</tr>
<tr>
<td>Min</td>
<td>130.1 -0.24</td>
<td>119.78 -0.61</td>
<td>109.5 -0.8</td>
</tr>
</tbody>
</table>

Table 4.7: Variation of evaporator pressure for different loads during the test time as indicated in Figure 4.6(a).

<table>
<thead>
<tr>
<th>Case</th>
<th>Pressure (kPa) (%)</th>
<th>Pressure (kPa) (%)</th>
<th>Pressure (kPa) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max</td>
<td>1.9 +2.1</td>
<td>1.665 +3.0</td>
<td>1.419 +3.2</td>
</tr>
<tr>
<td>Mean</td>
<td>1.86</td>
<td>1.616</td>
<td>1.375</td>
</tr>
<tr>
<td>Min</td>
<td>1.79 -3.7</td>
<td>1.56 -3.4</td>
<td>1.314 -4.4</td>
</tr>
</tbody>
</table>

Table 4.8: Variation of evaporator temperature for different loads during the test time as indicated in Figure 4.6(b).

<table>
<thead>
<tr>
<th>Case</th>
<th>Temperature (°C) (%)</th>
<th>Temperature (°C) (%)</th>
<th>Temperature (°C) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max</td>
<td>14.08 +0.15</td>
<td>10.09 +0.2</td>
<td>6.45 +0.15</td>
</tr>
<tr>
<td>Mean</td>
<td>13.93</td>
<td>9.89</td>
<td>6.3</td>
</tr>
<tr>
<td>Min</td>
<td>13.8 -0.13</td>
<td>9.73</td>
<td>6.1                -0.2</td>
</tr>
</tbody>
</table>

Table 4.9: Variation of condenser pressure during the test time.

<table>
<thead>
<tr>
<th>Case</th>
<th>Pressure (kPa) (%)</th>
<th>Pressure (kPa) (%)</th>
<th>Pressure (kPa) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max</td>
<td>5.05 +0.39</td>
<td>4.33 +0.23</td>
<td>4.2 +0.2</td>
</tr>
<tr>
<td>Mean</td>
<td>5.03</td>
<td>4.32</td>
<td>4.117</td>
</tr>
<tr>
<td>Min</td>
<td>4.97 -1.19</td>
<td>4.3</td>
<td>4.09 -0.65</td>
</tr>
</tbody>
</table>
Figure 4.8: (a) Geometric arrangement of the ejector; (b) Pressure variation with distance along the ejector with error bars indicating the temporal pressure variations recorded for 100 samples during 10 second; (c) Corresponding results for the case of 6000 samples during 10 minutes, primary stream pressure of 270 kPa and evaporator load of 1200 W.
4.3 Preliminary test

4.3.4 Critical condenser pressure of the steam ejector

To estimate the critical condenser pressure of the steam ejector, three parameters were monitored: evaporator temperature, evaporator pressure, and condenser pressure. The procedure below was used to evaluate the critical condenser pressure.

- The condenser pressure was gradually increased through a decrease of the cooling water mass flow rate via a valve (valve c) installed on the discharge line of the cooling water pump as shown in Figure 3.1. Figure 4.9(a) illustrates the variation of condenser pressure with time.

- The pressure and the temperature of the evaporator were monitored during this process. Figure 4.9 illustrates results from a representative experiment. Figure 4.9(b) presents the evaporator pressure as a function of time. Figure 4.9(c) shows the evaporator temperature as a function of time.

It is observed that the evaporator pressure in this case did not start to increase until 2.125 minutes, while the evaporator temperature did not start to increase until 2.197 minutes (Figure 4.9(b) and 4.9(c)). The time was specified by using an intersection point (I.P.) between the two lines fitted to the data as illustrated. When the condenser pressure increases, Figure 4.9(a), there is initially no change in the pressure of the evaporator until about 2.125 minutes. The critical condenser pressure was identified by specifying the time when the pressure and the temperature of the evaporator started to increase. The intersection points between time and the condenser pressure curve represent the critical condenser pressure (C.P.) as illustrated in Figure 4.9(a).

The condenser pressure curve Figure 4.9(a) is used to identify the pressures corresponding to these times. Since the time at which the evaporator pressure and temperature start to rise appear slightly different, the range for the critical condenser pressures can be specified in this case as either 6.61 kPa or 6.64 kPa. Therefore the mean critical condenser pressure in this case is at 6.625 kPa.
Figure 4.9: (a) Condenser pressure change during the test time; (b) Evaporator pressure change during the test time; (c) Evaporator temperature change during the test time
The steam ejector system performs largely as intended, the instrumentation and measurement systems which have been developed appear fit for purpose. Table 4.10 summarizes the variability of quantities considered in this chapter for three cooling load cases with primary stream conditions of 270 kPa, and 130 °C. Measurement uncertainties, systematic errors, and system stability issues may all contribute to a calculated COP which is not accurate. However, the most significant contribution is likely to arise from the uncertainty in the measured secondary mass flow rate. Based on results presented in Table 4.10, it is anticipated that entrainment ratios and COPs reported in the present work will be accurate to within ± 3%.

<table>
<thead>
<tr>
<th>Feature</th>
<th>Type</th>
<th>Evaporator temperature (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>6°C</td>
</tr>
<tr>
<td>Primary mass flow rate</td>
<td>MU</td>
<td>0.6</td>
</tr>
<tr>
<td>Secondary mass flow rate</td>
<td>MU</td>
<td>2.48</td>
</tr>
<tr>
<td>Heat loss from primary stream</td>
<td>SE</td>
<td>0.12</td>
</tr>
<tr>
<td>Head loss from primary stream</td>
<td>SE</td>
<td>0.00014</td>
</tr>
<tr>
<td>Non-electrical heat gain by the evaporator</td>
<td>SE</td>
<td>0.97</td>
</tr>
<tr>
<td>Pressure transducer electrical noise</td>
<td>SS</td>
<td>0.004</td>
</tr>
<tr>
<td>Primary stagnation pressure</td>
<td>SS</td>
<td>0.6</td>
</tr>
<tr>
<td>Evaporator pressure</td>
<td>SS</td>
<td>3.5</td>
</tr>
<tr>
<td>Condenser pressure</td>
<td>SS</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Where

MU = Measurement uncertainty

SE = Systematic error

SS = System stability
Chapter 5

Ejector results and discussion

5.1 Introduction

This chapter provides experimental results from the steam ejector including the effect of parameters such as evaporator temperature, condenser pressure, and evaporator load. A comparison of present results with previous experiments with steam ejectors will also be described in this chapter.

Figure 5.1 illustrates the variation of the primary stream pressure, the evaporator pressure, and the condenser pressure during the test time. An increase in the condenser pressure eventually leads to an increase in the evaporator pressure without any change in the primary stream pressure, even in the reverse flow region. This is because there still is a high pressure difference between the primary pressure and the condenser, meaning the primary nozzle remains choked.
Figure 5.1: Illustrative variations of pressure during apparatus operation. Eventually the evaporator pressure increases in response to an increase in the condenser pressure whereas no change of the primary stream pressure occurs during the transition from choked flow to reverse flow regions.

5.2 Operating regimes

The experimental work was carried out over five evaporator loads 1000, 2000, 2500, 3000 and 3500 W with each test performed over different primary stream pressures of 143, 200, 270 kPa, and temperatures of 110, 120, 130°C; the condenser pressure range was varied from 3.5 to 7.0 kPa. The primary nozzle exit was positioned at NXP = 6 mm. Data from the experiments are presented in Figures 5.2, 5.3, and 5.4 in which three operating regions can be identified:

(a) Choked flow where the primary and secondary flows are choked;

(b) Unchoked flow where the primary flow is choked and secondary flow is unchoked.

The secondary flow rate reduced from maximum flow rate until zero flow rate
without any reverse flow to the evaporator; and

(c) Reversed flow where the primary flow is choked and most of the primary flow rate is reversed flow to the evaporator via the secondary inlet region.

Figure 5.2(a) presents the COP of the apparatus at primary stream conditions of 270 kPa, 130°C and 14°C evaporator temperature (with choked secondary stream operation). The greatest COP was obtained at 0.385 for the choked mode of operation. The COP and cooling load are independent of the condenser pressure until it reaches the critical pressure (Huang, Jiang and Hu, 1985; Chunnanond and Aphornratana, 2004). In this case, the critical pressure was 6.9 kPa. After this critical pressure is reached, the COP becomes dependent on the condenser pressure. An increase in pressure of the condenser led to a decrease in COP and cooling load. If the condenser pressure increases beyond the breakdown pressure, it causes reverse flow into the evaporator and the COP drops rapidly to zero. The breakdown pressure was measured in this case to be about 7.5 kPa.

The temperature of the evaporator rose rapidly from 14°C to 20°C in the unchoked region due to the decrease in the entrainment ratio with increasing condenser pressure as shown in Figure 5.2(b). The increased evaporator temperature arose because the heater element continued to be supplied at 3000 W.

Figure 5.3(a) illustrates the COP of the system at primary stream conditions of 200 kPa, 120°C and 10°C evaporator temperature (with choked secondary stream operation). At these conditions, the COP obtained was 0.291 up to a critical condenser pressure of 5.15 kPa as indicated clearly in Figure 5.3(b). The COP was only 0.161 and the critical pressure was 3.8 kPa for primary stream conditions of 143 kPa and 110°C with 6°C evaporator temperature during choked secondary stream operation, as shown in Figure 5.4.

The results presented in Figures 5.2, 5.3, and 5.4 show that the COP and cooling load are constant until the critical condenser pressure. However, COP decreased when there is an increase in the condenser pressure beyond the critical pressure. Reverse flow occurs when the condenser pressure is increased more than the breakdown pressure.
Figure 5.2: Experimental results over a range of condenser pressures for an evaporator load of 3000 W with primary stream conditions of 270 kPa and 130°C (a) COP of steam ejector; (b) Evaporator temperature variation.
Ejector results and discussion

Figure 5.3: Experimental results over a range of condenser pressures for an evaporator load of 2000 W with primary stream conditions of 200 kPa, 120°C (a) COP of steam ejector; (b) Evaporator temperature variation.
Figure 5.4: Experimental results over a range of condenser pressures for an evaporator load of 1000 W with primary stream conditions of 143 kPa and 110°C (a) COP of steam ejector; (b) Evaporator temperature variation.
5.3 Effect of primary stream conditions

Figure 5.5 illustrates the comparison between the COP of three primary stream pressures of 270 kPa, 200 kPa, and 143 kPa at 3000 W constant evaporator load, with a primary nozzle exit position of NXP = 6 mm. The results show that an increase in primary pressure causes a decrease in the COP and the cooling load coupled with an increase in the critical condenser pressure. The variation of the COP of the present apparatus is consistent with the experiments of Huang et al. (1985), Eames, Aphornratana and Sun (1994), and Chunnanond and Aphornratana (2004). Results similar to those of Figure 5.5 were obtained for different operating conditions with primary stream pressures of 200 kPa and 143 kPa and cooling loads of 2000 W and 1000 W, as shown in Figures 5.6 and 5.7. The results from Figures 5.5, 5.6, and 5.7 show that the COP decreased when there was an increased primary pressure. Further, there is an increased COP when the evaporator temperature increased (the equivalent of an increase in evaporator load). Pridasawas (2006) interpreted this phenomenon as the primary mass flow rate increases with an increase in the primary pressure, while the secondary mass flow rate maximum can be reached at choked conditions through the secondary inlet section. Therefore, an increase in the primary pressure leads to a decrease in the COP of the system by virtue of a reduced entrainment ratio.
5.3 Effect of primary stream conditions

Figure 5.5: COP variation with condenser pressure for primary stream pressures of 270, 200, and 143 kPa and corresponding temperatures of 130°C, 120°C, and 110°C with 3000 W cooling load.

Figure 5.6: COP variation with condenser pressure for primary stream pressures of 270, 200, and 143 kPa and corresponding temperatures of 130°C, 120°C, and 110°C with 2000 W cooling load.
Chunnanond and Aphornratana (2004) suggested that when the steam ejector operates with a low primary stream pressure, a smaller mass flow rate passes through the nozzle with low velocity causing what they described as a smaller expanded wave angle, due to smaller injected momentum, and resulting in a longer entrainment region inside the mixing chamber. They suggested that a longer entrainment region leads to a higher secondary mass flow rate into the mixing chamber as shown in Figure 5.8, causing the system to produce a higher COP at lower primary stream pressures. However, their argument is not consistent with the present results as discussed below.

Figure 5.8: Entrainment and mixing process in the mixing chamber (a) Long entrainment duct; (b) Short entrainment duct (Chunnanond and Aphornratana, 2004).
5.3 Effect of primary stream conditions

Table 5.1 presents a comparison between the secondary and the primary flow rates at constant evaporator temperature 14°C (3000 W cooling load) with different primary pressures in the present work. As can be seen, there is no change in the secondary flow rate, even though there is an increase in the entrainment ratio $\omega$ with a decrease in the primary flow rate. This result is consistent with the study of Pridasawas (2006) and inconsistent with Chunnanond and Aphornratana (2004) because the present results indicate that there is no change of the secondary flow rate. There was change in the primary flow rate and this caused an increased COP. Moreover, the momentum increased with an increase in the primary pressure causing a higher condenser pressure as shown in Figure 5.9. For the primary stream pressure of 270 kPa, the static pressure rise within the diffuser appears delayed spatially relative to the primary stream pressure of 143 kPa and this delay is related to the momentum difference between the primary stream in these cases.

Table 5.1: Secondary and primary stream mass flow rates for three different primary stream temperatures with an evaporator temperature of 14°C.

<table>
<thead>
<tr>
<th>Entrainment ratio $\omega$</th>
<th>Secondary mass flow rate $\dot{m}_s$ (kg/hr)</th>
<th>Primary mass flow rate $\dot{m}_p$ (kg/hr)</th>
<th>Primary temperature ($^\circ$C)</th>
<th>Primary pressure (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.33</td>
<td>4.3</td>
<td>12.96</td>
<td>130</td>
<td>270</td>
</tr>
<tr>
<td>0.38</td>
<td>4.3</td>
<td>11.316</td>
<td>120</td>
<td>200</td>
</tr>
<tr>
<td>0.44</td>
<td>4.3</td>
<td>9.77</td>
<td>110</td>
<td>143</td>
</tr>
</tbody>
</table>
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Figure 5.9: Static pressure along the ejector wall for primary stream pressures of 270 kPa, 200 kPa and 143 kPa, and corresponding temperatures of 130°C, 120°C, and 110°C for an evaporator temperature of 14°C and a 4.2 kPa condenser pressure.

5.4 Effect of condenser pressure on pressure distributions

It is difficult to specify the location of the shocks along the ejector except in the pressure profile at 3.6 kPa condenser pressure and the primary stream pressure of 270 kPa (Figure 5.10). In this case, a sudden increase in pressure can be observed between 300 mm and 450 mm (before the end of the parallel section of the mixing chamber). The second series of oblique waves usually appear in this region (Matsuo et al., 1981; Dutton and Carroll, 1988; Ruangtrakoon et al., 2013). The details and computational simulation of the second series of oblique shock waves can be found in Chapter 6.

The static pressure results from experiments for different operating conditions are presented in Figures 5.10, 5.11, and 5.12. From these results it can be seen that the static pressure rose gradually along the ejector rather than with a sudden increase. The
lowest pressure point was at a position close to 170 mm in each case. This location corresponds to the hypothetical throat section H-H shown in Figure 3.4. This result was explained by Fabri and Siestrunck (1958) and Munday and Bagster (1977) by considering the secondary stream initially at vapour pressure $P_s$ and when expanded to section H-H, reaching $P_{HH}$, a pressure at or below the critical value for sonic flow. Section H-H was defined by Munday and Bagster (1977) as the “effective cross-section area” in which the secondary flow reaches sonic flow and mixing begins between primary flow and secondary flow. The mixing causes an increase in the static pressure after cross section H-H, as illustrated in Figures 5.10, 5.11, and 5.12. The transition between the parallel section of mixing chamber and the diffuser was defined by Chunnanond and Aphornratana (2004) as the “shocking position” which creates a compression effect beyond the cross section H-H, resulting in more compression achieved in the rest of the diffuser.

The results indicate that when the secondary flow reached the sonic flow at the hypothetical throat section H-H or effective cross-section area, there was increased pressure by a shock wave at the shocking position, and a further compression effect at the diffuser.

![Graph](image)

Figure 5.10: Static pressure along the ejector wall for primary stream pressure of 270 kPa, and corresponding temperature of 130°C for an evaporator temperature of 14°C and 3.6 kPa, 4.4 kPa, 5.5 kPa, and 6.8 kPa condenser pressures.
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Figure 5.11: Static pressure along the ejector wall for primary stream pressure of 200 kPa, and corresponding temperature of 120°C for an evaporator temperature of 14°C and 3.6 kPa, 4.3 kPa, 5.4 kPa, and 6.1 kPa condenser pressures.

Figure 5.12: Static pressure along the ejector wall for primary stream pressure of 143 kPa, and corresponding temperature of 110°C for an evaporator temperature of 14°C and 3.7 kPa, 4.2 kPa, and 4.8 kPa condenser pressures.
5.5 Effect of evaporator load on COP

Figure 5.13 illustrates the effect of evaporator load on the coefficient of performance of the ejector. The COP rises with increasing evaporator load. The primary nozzle exit position was at NXP = -32 mm, resulting in choked secondary stream conditions. The primary stream flow rate was maintained constant for different primary stream temperatures of 130°C, 120°C and 110°C. The results presented in Figure 5.13 show that the secondary mass flow rate increased with evaporator load, and caused an increase in the COP with a decrease in the temperature of the primary stream.

Data on the static pressures for different evaporator loads 3000 W, 2000 W, and 1000 W (corresponding to temperatures 6, 10, and 14°C) are presented in Figure 5.14. Increased pressures in the ejector are associated with an increase in the COP due to a higher amount of secondary flow being drawn into the mixing chamber. A high temperature of the evaporator led to operating the cycle at a higher condenser pressure, similar to the study of Eames et al. (1995).

Figure 5.13: Experimental results displaying the effects of evaporator load on the COP of the ejector for condenser pressures sufficiently low to achieve choked secondary stream conditions.
5.6 Comparison with previous works

In this section, the experimental results obtained in this study are compared with previous works on the steam ejector. Comparisons for 130°C primary stream temperature and 15°C evaporator temperature are presented in Figure 5.15, for 130°C primary stream temperature and 10°C evaporator temperature in Figure 5.16, and for 120°C primary stream temperature and 10°C evaporator temperature in Figure 5.17.

Figure 5.15 presents a comparison between the COP of the present steam ejector with the previous work of Sun (1997) and Chunnanond and Aphornratana (2004). The highest values of the COP and the critical pressure reached by Sun (1997) were 0.46 and 5 kPa, respectively. Chunnanond and Aphornratana (2004) achieved higher values than Chunnanond and Aphornratana (2004) for the COP and the critical pressure: 0.49 and 5.4 kPa, respectively. In the present work, the COP was slightly higher than Chunnanond and Aphornratana (2004) at 0.519. In addition, there was a significant difference in the critical pressure, being about 6.9 kPa in the present work whereas Chunnanond and Aphornratana (2004) achieved 5.4 kPa. The primary nozzle exit
5.6 Comparison with previous works

Position in the present work was NXP = 6 mm. Chunnanond and Aphornratana (2004) and Sun (1997) carried out their experiments at similar operating conditions to this study except that the primary nozzle area ratios were 12 and 16, respectively. The area ratio used in this study was 18.

The differences between the present results and those of Sun (1997) and Chunnanond and Aphornratana (2004) primarily arise for two reasons:

1. Differences between the primary nozzle area ratios and;
2. Differences in the capacities of the rigs.

Figure 5.15: COP results from previous studies compared with those of the present work for a primary stream temperature of 130°C and an evaporator temperature of 15°C.
Figure 5.16 shows the comparison between the COP and critical pressure at primary stream temperature 130°C and evaporator temperature 10°C. The maximum COP and critical pressure reached by Eames et al. (1994) were 0.289 and 4.9 kPa, respectively. The result from Sun (1997) was slightly lower than Eames et al. (1994). However, in the present work the COP was 0.33 and critical pressure was 6.6 kPa.

![COP results from previous studies compared with those of the present work for a primary stream temperature of 130°C and an evaporator temperature of 10°C.](image)

Figure 5.16: COP results from previous studies compared with those of the present work for a primary stream temperature of 130°C and an evaporator temperature of 10°C.
5.6 Comparison with previous works

Figure 5.17 illustrates the COP as a function of the critical pressure at 120°C primary stream temperature and 10°C evaporator temperature. There is a slight difference between the present result and the study from Eames et al. (1994) in COP of about 2.7%. However, there is a considerable difference in critical pressure between the result from Eames et al. (1994) which was at 3.8 kPa and this present work, which was at 5 kPa. Conversely, Kehnmonyi (1996) demonstrated a lower COP and the higher critical pressure reached 7.3 kPa.

![Figure 5.17: COP results from previous studies compared with those of the present work for a primary stream temperature of 120°C and an evaporator temperature of 10°C.](image)

Figure 5.17: COP results from previous studies compared with those of the present work for a primary stream temperature of 120°C and an evaporator temperature of 10°C.
5.7 Conclusion

The present apparatus can produce a cooling load of about 3500 W at 14°C, 2500 W at 10°C, and 1200 W at 6°C with different condenser pressures.

This cycle has a higher COP compared with previous work by about 5.6% at a primary stream pressure of 270 kPa and 15°C evaporator temperature, 12.4% at a primary stream pressure of 270 kPa and 10°C evaporator temperature, and 2.7% at a primary stream pressure of 200 kPa, and 10°C evaporator temperature. In addition, the apparatus has a higher critical condenser pressure in comparison with previous work, by about 21.7% for a primary stream pressure of 270 kPa, and 25.7% at a pressure of 200 kPa.

If the steam ejector system works with a constant evaporator temperature such as 14°C, and the primary stream pressure is increased from 143 kPa to 270 kPa, the COP will decrease from 0.48 to 0.39 and the cycle could be operated at higher condenser pressure like 6.9 kPa, rather than 5 kPa.
Chapter 6

Computational simulation

6.1 Introduction

Experiments make an important contribution to understanding and optimising thermo-fluids systems. Global parameters such as stagnation pressure and temperature, mass flow rate, and heat transfer provide substantial insight into system performance. However, when more complete details are required such as the mixing process inside a steam ejector, experimental data are relatively costly.

Computational Fluid Dynamics (CFD) simulation can provide an alternative approach when experiments are dangerous, complex or expensive. CFD uses a numerical method to solve differential equations which represent the flow inside the ejector. In the present work, a CFD program called Eilmer3 was used to simulate the flow within the axisymmetric geometry of the ejector. Eilmer3 is an integrated collection of programs for the simulation of transient compressible flow, especially with supersonic speed, in two or three dimensions. Eilmer3 is part of the large collection of compressible flow solvers that has been developed by Compressible Flow CFD Group at the University of Queensland. The program depends on the specification of the geometric flow domain, the initial flow field, and boundary conditions.
6.2 Building and running simulation

Eilmer3 is coded in C / C++ for the core solver and modules. The python language is used to interpret an input script which defines the flow domain and the initial and boundary conditions. The Lua language is used for user-defined boundary conditions. The program has been developed in a Linux environment, though the program can also be built in different environments including MacOS-X and MS-Windows.

Establishing a simulation in Eilmer3 follows three steps:

1. Create the geometry definition, mesh, initial flow state, and boundary conditions using a Python script.

2. Run the simulation code and save the flow data (solutions) at various times according to the specifications in the Python script.

3. Reformat the data from the simulation to produce a file appropriate for the visualisation. Typically Paraview or GNU-Plot are used for this purpose.

6.3 Turbulence models

There are many turbulence models in the literature which have been applied to ejector flows. For example Bartosiewicz et al. (2005) compared the Reynolds stress model (RSM) with other models. The realizable $k$-$\epsilon$ was used by Sriveerakul et al. (2007a) while Hemidi et al. (2008) tested two models: the $k$-\omega-SST and the $k$-$\epsilon$. Chan, Jacobs and Mee (2012) used Eilmer3 simulations with the $k$-\omega model to test five cases including a 2D flat plate, a backward-facing step, an axisymmetric hollow cylinder, a turbulent boundary layer, and mixing of coaxial jets. A good agreement between simulations and experimental results was obtained for all of these test cases. In this study, the $k$-\omega model as implemented in Eilmer3 by Chan et al. (2012) was used.
6.4 \( k-\omega \) Turbulence model

Wilcox (2006) introduced the \( k-\omega \) model used in this work. The Favre-averaged equations for mass, momentum, energy conservation and the equations defining the \( k-\omega \) model are as follows.

- Mass conservation:

\[
\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial}{\partial x_i} (\bar{\rho} \bar{u}_i) = 0 \tag{6.1}
\]

- Momentum conservation

\[
\frac{\partial}{\partial t} (\bar{\rho} \bar{u}_i) + \frac{\partial}{\partial x_i} (\bar{\rho} \bar{u}_j \bar{u}_i) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} [\bar{t}_{ij} + \bar{\rho} \tau_{ij}] \tag{6.2}
\]

- Energy conservation:

\[
\frac{\partial}{\partial t} \left[ \bar{\rho} \left( \tilde{c} + \frac{\bar{u}_i \bar{u}_i}{2} + k \right) \right] + \frac{\partial}{\partial x_j} \left[ \bar{\rho} \tilde{h} \left( \tilde{h} + \frac{\bar{u}_i \bar{u}_i}{2} + k \right) \right] = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu}{Pr_L} + \frac{\mu_T}{Pr_T} \right) \frac{\partial \tilde{h}}{\partial x_j} + \left( \mu + \frac{\rho k}{\omega} \right) \frac{\partial k}{\partial x_j} \right] + \frac{\partial}{\partial x_j} [\tilde{u}_i (\bar{t}_{ij} + \bar{\rho} \tau_{ij})] \tag{6.3}
\]

- Molecular and Reynolds-Stress tensors:

\[
\bar{t}_{ij} = 2\mu \bar{S}_{ij}, \quad \bar{\rho} \tau_{ij} = 2\mu_T \bar{S}_{ij} - \frac{2}{3} \bar{\rho} k \delta_{ij}, \quad \bar{S}_{ij} = S_{ij} - \frac{1}{3} \frac{\partial \bar{u}_k}{\partial x_k} \delta_{ij} \tag{6.4}
\]

- Eddy viscosity:

\[
\mu_T = \frac{\bar{\rho} k}{\bar{\omega}}, \quad \bar{\omega} = \max \left[ \omega, C_{lim} \sqrt{\frac{2\bar{S}_{ij} \bar{S}_{ij}}{\beta^*}} \right], \quad C_{lim} = \frac{7}{8} \tag{6.5}
\]
• Turbulence kinetic energy:

\[
\frac{\partial}{\partial t} (\bar{\rho} k) + \frac{\partial}{\partial x_j} (\bar{\rho} \tilde{u}_j k) = \bar{\rho} \tau_i \frac{\partial \tilde{u}_i}{\partial x_j} - \beta^* \bar{\rho} k \omega + \frac{\partial}{\partial x_j} \left[ \left( \mu + \sigma^* \bar{\rho} k \right) \frac{\partial k}{\partial x_j} \right]
\] (6.6)

• Specific dissipation rate:

\[
\frac{\partial}{\partial t} (\bar{\rho} \omega) + \frac{\partial}{\partial x_j} (\bar{\rho} \tilde{u}_j \omega) = \frac{\omega}{k} \bar{\rho} \tau_i \frac{\partial \tilde{u}_i}{\partial x_j} - \beta \bar{\rho} \omega^2
\] 
\[+ \sigma_d \frac{\bar{\rho}}{\omega} \frac{\partial \omega}{\partial x_j} \frac{\partial \omega}{\partial x_j} + \frac{\partial}{\partial x_j} \left[ \left( \mu + \sigma \bar{\rho} k \right) \frac{\partial \omega}{\partial x_j} \right]
\] (6.7)

• Closure coefficients:

\[
\alpha = 0.52, \quad \beta = \beta_o f_\beta, \quad \beta^* = 0.09, \quad \sigma = 0.5, \quad \sigma^* = 0.6, \quad \sigma_{do} = 0.125 \quad (6.8)
\]

\[
\beta_o = 0.0708, \quad Pr_T = 0.8888, \quad \sigma_d \begin{cases} 
0 \leq \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \\
\frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} > 0
\end{cases} \quad (6.9)
\]

\[
f_\beta = \frac{1 + 85 \chi_\omega}{1 + 100 \chi_\omega}, \quad \chi_\omega = \left| \frac{\Omega_{ij} \Omega_{jk} \hat{S}_{ki}}{(\beta^* \omega)^3} \right|, \quad \hat{S}_{ki} = S_{ki} - 0.5 \frac{\partial \tilde{u}_m}{\partial x_m} \delta_{ki} \quad (6.10)
\]

The \( k-\omega \) model of Wilcox (2006) represents an improvement in modelling as compared with previous studies. An additional two features were included in this model, namely: a cross diffusion term and a built-in stress-limiter modification. Cross diffusion improved the original \( k-\omega \) model sensitivity to the free stream value of \( \omega \). The eddy viscosity became a function of \( k \) and \( \omega \) as a result of the built-in stress-limiter modification. The model’s capability to predict supersonic and hypersonic separated flow has been demonstrated using this modification (Chan, Jacobs, Nap, Mee and Kirchhartz, 2011).

Wilcox (2006) indicated that all the improvements to the \( k-\omega \) model represent a substantial enhancement of its range of applicability.

• The current version produces the same accuracy of the 1988 version when it is used to simulate an attached boundary layer.
6.5 Recommendation to use $k$-$\omega$ model in Eilmer3

- The current version’s prediction of free shear layer spreading rates better approximates experimental data as compared to the 1988 version.

- Without any compressibility modification, the current model can give a good simulation of shock-separated flows.

- The current model has the ability to simulate flows from transonic to hypersonic regimes without any compressibility modification.

Chan et al. (2011) indicated that the turbulence kinetic energy $k$ and the specific dissipation rate $\omega$ can be deduced from values of turbulence intensity and turbulent-to-laminar viscosity ratio, as shown in equations (6.11) and (6.12). In this study, the turbulence kinetic energy $k$ and the specific dissipation rate $\omega$ used were $0.135 \text{m}^2/\text{s}^2$ and $14.76 \text{l/s}$ respectively.

$$k = 1.5(I_{turb}u_{\infty})^2 \quad (6.11)$$

$$\omega = \rho \frac{k}{\mu_{lam}} \left( \frac{\mu_{lam}}{\mu_{turb}} \right) \quad (6.12)$$

6.5 Recommendation to use $k$-$\omega$ model in Eilmer3

When establishing grids and specifying flow properties for turbulent CFD simulations, the $y^+$ value, the number of cells within the boundary layer, and the cell aspect ratio are given emphasis. Chan et al. (2011) presented some recommendations for these parameters when using Eilmer3.

6.5.1 Dimensionless wall distance, $y^+$

$y^+$ is a non-dimensional normal distance from the wall and is relevant for the simulation of wall-bounded flow. Wilcox (2006) defined it as
\[ y^+ = \frac{y v^* \rho_{\text{wall}}}{\mu_{\text{wall}}} \]  
\[ v^* = \sqrt{\frac{\tau_{\text{wall}}}{\rho_{\text{wall}}}} \]  
\[ \tau_{\text{wall}} \approx \mu_{\text{wall}} \left( \frac{\partial u}{\partial y} \right)_{y=0} \]

In the context of CFD, \( y^+ \) values refer to the size of the first cell adjacent to the wall. Several researchers including Roy and Blottner (2003), ESI CFD (2004), Wilcox (2006) Versteeg and Malalasekera (2007), and Jiyuan, Yeoh and Liu (2008) recommended that \( y^+ < 1 \) for the first cell when using the \( k-\omega \) model. Different models require different settings. This value indicates there is at least one cell within the viscous sublayer.

The present work used a value of \( y^+ = 0.28 \) for the fine mesh. Table 6.1 presents the values of \( y^+ \) for three mesh resolutions: coarse, medium, and fine, and this table also presents the number of cells for each case. In each case, the value of \( y^+ \) was taken on the parallel section of the mixing chamber.

<table>
<thead>
<tr>
<th>Resolution</th>
<th>Number of cells</th>
<th>( y^+ )</th>
<th>Maximum aspect ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse</td>
<td>4958</td>
<td>1.81</td>
<td>30</td>
</tr>
<tr>
<td>Medium</td>
<td>19832</td>
<td>0.62</td>
<td>34.9</td>
</tr>
<tr>
<td>Fine</td>
<td>44622</td>
<td>0.28</td>
<td>34.6</td>
</tr>
</tbody>
</table>

6.5.2 Number of cells within boundary layer

The number of cells within the boundary layer must be sufficiently high to allow resolution of the boundary layer profile in order to simulate a shock separated boundary layer. Roy and Versteeg and Malalasekera (2007) recommended the use of a minimum number of cells between 10 to 20, while Boyce and Hillier (2000) used 90 cells to resolve the boundary layer.
6.5.3 Maximum cell aspect ratio

Jiyuan et al. (2008) and Fluent (2006) recommended the use of a cell aspect ratio of less than 5 in both boundary layers and non-boundary layers, but they did not specify a maximum cell aspect ratio for either of these regions. Hirsch (1988) and (2007) observed that aspect ratios higher than 100 and of the order of 1000 are not suitable for the first cells adjacent to a wall since high cell aspect ratios can cause degradation of computation quality. However, an aspect ratio of less than 5 is unfavourable especially for viscous simulations. Chan et al. (2011) presented a study of the simulation of an axisymmetric cylinder. Three types of mesh were used: coarse, medium and fine with different aspect ratios. Chan et al. (2011) determined that for aspect ratios higher than 600, the simulations became unstable and gave incorrect results. Therefore, it was recommended that the maximum value for the aspect ratio of cells near the wall should be no more than 600.

In this study, the maximum aspect ratio of cells of the ejector was around 35. Actual values of aspect ratio for the different grids are presented in Table 6.1.

6.6 Simulated ejector configuration

The geometry of the calculation domain of the modelled steam ejector was taken from the experimental configuration presented in Chapter 3. In this study the ejector was divided into 14 blocks to form the shape of the steam ejector as shown in Figure 6.1. Boundary conditions were applied to the two inlets and the one outlet of the ejector. ‘Subsonic inlet’ boundary conditions were specified at locations where the high pressure and temperature primary stream enters the ejector through a converging-diverging nozzle and where the low pressure and temperature secondary stream enters the ejector through a plenum chamber. An ‘extrapolate outlet’ boundary condition was specified in a single outlet. Chan et al. (2011) recommended the use of the ‘extrapolate outlet’ boundary condition in preference to a ‘fixed pressure outlet’ condition since ‘extrapolate outlet’ boundary condition proved to yield more accurate results.
Nine additional blocks were used to control the pressure at the downstream end of the ejector as illustrated in Figure 6.2. The throat region represented by blocks 20, 21 and 22 worked as a valve to change the pressure inside the ejector while maintaining a largely supersonic stream departing from the flow domain as suitable for the ‘extrapolate outlet’ condition. Values listed in Table 6.2 were used to specify the flow conditions in the present simulation. Primary and secondary stream conditions refer to the specified inlet boundary conditions whereas the condenser pressure refers to the range of pressures achieved along the wall towards the downstream end of block 13 through the constricting effect of the downstream throat region.

Table 6.2: Ejector operation conditions specified in the simulations.

<table>
<thead>
<tr>
<th></th>
<th>Pressure (kPa)</th>
<th>Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary stream</td>
<td>143, 200, and 270</td>
<td>110, 120, and 130</td>
</tr>
<tr>
<td>Secondary stream</td>
<td>0.9, 1.2, and 1.5</td>
<td>6, 10, and 14</td>
</tr>
<tr>
<td>Condenser pressure</td>
<td>3.5 to 6.6</td>
<td></td>
</tr>
</tbody>
</table>
6.7 Grid of the ejector in CFD

Computations were performed using three different levels of discretization of the flow domain: coarse, medium, and fine resolutions as presented in Table 6.3. Figure 6.3 illustrates a comparison between the simulated static pressure results for the three different grid resolutions and experimental data. Differences in the wall static pressure along the ejector for the different meshes are evident in Figure 6.3. The differences between the coarse mesh simulation and the experimental result is about 10.7%, this percentage difference decreased to about 1.5% for the medium, and about 1% for fine mesh. Because of this mesh independence, the fine grid was used for the present simulation. Figure 6.4 shows the fine grid within the ejector.
Table 6.3: Number of cells in the axial and radial directions for coarse, medium, and fine resolution cases.

<table>
<thead>
<tr>
<th>Block number</th>
<th>Resolution</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Coarse</td>
<td>Medium</td>
<td>Fine</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>x axis</td>
<td>r axis</td>
<td>x axis</td>
<td>r axis</td>
<td>x axis</td>
</tr>
<tr>
<td>0</td>
<td>23</td>
<td>4</td>
<td>46</td>
<td>8</td>
<td>69</td>
</tr>
<tr>
<td>1</td>
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<td>45</td>
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<td>90</td>
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<td>86</td>
<td>4</td>
<td>172</td>
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<td>258</td>
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<td>172</td>
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<td>20</td>
<td>8</td>
<td>40</td>
<td>12</td>
<td>60</td>
</tr>
</tbody>
</table>
Figure 6.3: Comparison of simulated static pressure and experimental data for the three types of grids: coarse, medium, and fine.

Figure 6.4: Fine grid used throughout the ejector with a focus region at the nozzle exit.
6.8 Simulation time

The Eilmer3 solver marches solutions forward in time from given initial and boundary conditions. Therefore, it is important to confirm that steady ejector operating conditions are reached by the end of the simulation period. The Eilmer3 simulation has the capability to monitor any point during a transient simulation within the ejector. Figure 6.5 shows the time to reach steady-state pressure at three points: (1) primary nozzle exit, (2) ejector surface at primary nozzle exit location, (3) ejector surface at exit. By the end of the simulation period, static pressures within the ejector have remained within 0.2% of their final values for a period of 1 ms. (The value 1 ms corresponds approximately to the flow transit time through the ejector.) Therefore it is concluded that the simulation time used in the present work is adequate.

Figure 6.5: Temporal variation of static pressure during the simulations: primary nozzle exit, ejector surface at primary nozzle exit location, and ejector surface at exit.
6.9 Validation of Eilmer3 simulations

6.9.1 Validation of static pressure

This section presents a comparison of CFD (Eilmer3) and experimental static pressures within the ejector. A selection of operating conditions were chosen for validation of Eilmer3.

Figures 6.6, 6.7, and 6.8 show the comparison between wall static pressure along the ejector from the two approaches, the experimental data and the CFD simulation. The comparisons were made when the operating conditions of the ejector were at: (1) primary stream conditions of 270 kPa stagnation pressure, 130°C stagnation temperature; (2) 6, 10, and 14°C evaporator temperature; (3) different condenser pressures; and (4) the primary nozzle exit position at NXP of 6 mm. A similar range of conditions were explored for the primary stream conditions of 200 kPa and 120°C, as illustrated in Figures 6.9, 6.10, and 6.11.

Referring to Figures 6.6 to 6.11, the wall static pressure profile along the ejector from the experiment and Eilmer3 simulation are seen to be similar to those illustrated by Huang et al. (1985), Eames et al. (1999), Chunnanond and Aphornratana (2004), and Sriveerakul et al. (2007a). The present study demonstrates that the Eilmer3 simulation results are consistent with experimental data for the pressure distribution along the ejector.

Table 6.4 presents a comparison between the static pressure distributions along the ejector with different turbulence models. Sriveerakul et al. (2007a) compared their experimental data with commercial CFD software using FLUENT 6.0. Yang et al. (2012) compared their simulations to the experiments of Sriveerakul et al. (2007a) with three types of turbulence models. In the present work Eilmer3 simulations used the $k$-$\omega$ turbulence model and results are compared with the present experimental data. Eilmer3 has lowest percentage error: less than 2%. The values of error reported in Table 6.4
were obtained by calculating the average values of the errors for all pressure transducers along the ejector. Equation (6.16) was used to calculate the error percentage.

\[
\text{Error } \% = \frac{\text{CFD's static pressure} - \text{Experimental static pressure}}{\text{Experimental static pressure}} \times 100 \quad (6.16)
\]

In this case used data presented in Figure 6.8

Figure 6.6: Comparison between Eilmer3 and experimental results of static pressure along the ejector for primary stream conditions of 270 kPa, 130°C, with an evaporator temperature of 14°C, and a condenser pressure of 6.1 kPa.
Figure 6.7: Comparison between Eilmer3 and experimental results of static pressure along the ejector for primary stream conditions of 270 kPa, 130°C, with an evaporator temperature of 10°C, and a condenser pressure of 6 kPa.

Figure 6.8: Comparison between Eilmer3 and experimental results of static pressure along the ejector for primary stream conditions of 270 kPa, 130°C, with an evaporator temperature of 6°C, and a condenser pressure of 5 kPa.
Figure 6.9: Comparison between Eilmer3 and experimental results of static pressure along the ejector for primary stream conditions of 200 kPa, 120°C, with an evaporator temperature of 14°C, and a condenser pressure of 5 kPa.

Figure 6.10: Comparison between Eilmer3 and experimental results of static pressure along the ejector for primary stream conditions of 200 kPa, 120°C, with an evaporator temperature of 10°C, and a condenser pressure of 4.8 kPa.
6.9 Validation of Eilmer3 simulations

Figure 6.11: Comparison between Eilmer3 and experimental results of static pressure along the ejector for primary stream conditions of 200 kPa, 120°C, with an evaporator temperature of 6°C, and a condenser pressure of 4.8 kPa.

Table 6.4: Comparison of different turbulence models with Eilmer3 simulation.

<table>
<thead>
<tr>
<th>Author</th>
<th>Turbulence model</th>
<th>$P_c$ (kPa)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sriveerakul et al. (2007a)</td>
<td>realizable $\kappa$-$\epsilon$</td>
<td>5</td>
<td>-8</td>
</tr>
<tr>
<td>Yang et al. (2012)</td>
<td>realizable $\kappa$-$\epsilon$</td>
<td>4.7</td>
<td>-6</td>
</tr>
<tr>
<td>Yang et al. (2012)</td>
<td>standard $\kappa$-$\epsilon$</td>
<td>4.4</td>
<td>-12</td>
</tr>
<tr>
<td>Yang et al. (2012)</td>
<td>RNG $\kappa$-$\epsilon$</td>
<td>4.5</td>
<td>-10</td>
</tr>
<tr>
<td>Eilmer3</td>
<td>$\kappa$-$\omega$</td>
<td>5</td>
<td>-2</td>
</tr>
<tr>
<td>Experimental data of this study</td>
<td>-</td>
<td>5</td>
<td>-</td>
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</tbody>
</table>
6.9.2 Validation of entrainment ratio

Figures 6.12 presents the comparison between the CFD simulation results and the experimental data for entrainment ratio at primary stream conditions of 270 kPa, 130°C, with an evaporator temperature of 14°C, and different condenser pressures. The primary nozzle exit position was at NXP of 6 mm.

For this condition, the maximum values of the experimental and simulated entrainment ratios were 0.33 and 0.34, respectively, the equivalent of about 3% difference. These differences are similar to those presented by Sriveerakul et al. (2007a), which was 0.53 for experimental and 0.54 for simulated entrainment ratios. Pianthong et al. (2007) also presented a comparison between experimental and simulated entrainment ratios with the difference amounting to about 5%. Figure 6.12 also illustrates the comparison between experimental and simulation results of the ejector performance at the critical pressure. The experimental and simulated critical pressure were 6.8 and 6.2 kPa, respectively. Pianthong et al. (2007) obtained a difference in critical pressure of 13.5% between experimental and simulation.

One of the possible reasons for this difference between the experimental results and the computational simulations, may come from assuming an ideal gas rather than modelling a real, condensing gas in the simulations.

A similar range of conditions were explored for the primary stream conditions of 200 kPa and 120°C, as illustrated in Figure 6.13. Differences between maximum experimental and simulated entrainment ratios were around 2.6% and in this case, the experimental and simulated entrainment ratios were 0.38 and 0.39, respectively. The difference in critical condenser pressure between the simulations and experiments was 15.4% at a primary stream pressure of 200 kPa and 14°C evaporator temperature.
Figure 6.12: Comparison between Eilmer3 simulations and experimental results for the entrainment ratio at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and different condenser pressures.

Figure 6.13: Comparison between Eilmer3 simulations and experimental results for the entrainment ratio at primary stream conditions of 200 kPa, 120°C with an evaporator temperature of 14°C and different condenser pressures.
6.10 Simulated pressure and velocity distributions

In this section, simulated pressure and velocity distributions will be presented for three different ejector operation cases: choked, unchoked, and reversed flow.

6.10.1 Choked flow

Figure 6.14 shows the static pressure distribution for the steam ejector operating at the choked condition with primary stream conditions of 270 kPa, 130°C, with 14°C evaporator temperature, and condenser pressure of 3.88 kPa. The first series of oblique shock waves is called a “diamond wave” by Sriveerakul et al. (2007b) and Ruangtrakoon et al. (2011). The diamond wave can be investigated from the variation of the static pressure at the centreline of the ejector as shown in Figure 6.15. At a certain distance from the beginning of the diffuser, a second series of oblique shock waves appear as shown in Figure 6.15 and such waves have also been observed previously (Matsuo et al., 1981; Dutton and Carroll, 1988; Ruangtrakoon et al., 2013).

Additional simulation results are presented in Figure 6.16 for the axial velocity profile for five sections along the ejector: 1, 78, 192.5, 270, and 430 mm distance from the nozzle exit corresponding to: (1) two sections in the converging section of the mixing chamber, (2) one section in the middle plane of the parallel section of the mixing chamber, and (3) two sections in the diffuser region. Figures 6.17 and 6.18 show the separation of the flow at the internal wall of the diffuser and the case of recirculation flow between main core of the flow and the wall of the diffuser. One possible reason for this phenomenon is the low condenser pressure leading to a continuous high speed flow, as shown in Figure 6.16. The axial velocity at the end of the diffuser was 450 m/s (the fifth section) corresponding to a Mach number of nearly 1.
6.10 Simulated pressure and velocity distributions

Figure 6.14: Pressure map from Eilmer3 simulations at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 3.88 kPa.

Figure 6.15: Centreline pressure from Eilmer3 simulations at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 3.88 kPa.
Figure 6.16: Simulated velocity profiles for five sections along the ejector at primary stream conditions of 270 kPa, 130 °C with an evaporator temperature of 14 °C and condenser pressure of 3.88 kPa. AA-location is the separation flow at the wall of the diffuser. Each velocity increment corresponds to 100 m/s.

Figure 6.17: Velocity map from Eilmer3 simulation of the ejector at primary stream conditions of 270 kPa, 130 °C with an evaporator temperature of 14 °C and condenser pressure of 3.88 kPa.

Figure 6.18: Flow recirculation on the diffuser wall at section AA for conditions and location indicated in Figure 6.17.
Simulated results at a higher condenser pressure of 5.88 kPa are illustrated in Figures 6.19 and 6.20. Ejector operation is still within the choked regime but the secondary train (series) of oblique shock waves moved backward into the parallel section of mixing chamber due to the increase in the condenser pressure (Ruangtrakoon et al., 2013). When this occurs the flow separation observed in Figures 6.17 and 6.18 is eliminated: no such separated flow region is observed in Figure 6.19. Figure 6.21 demonstrates the simulated velocity at the fifth section decreased to about 160 m/s, corresponding to a Mach number of less than 0.4 at the end of diffuser. Within the choked flow regime, there is no change in the entrainment ratio as presented in Figure 6.12. Figures 6.22 and 6.23 illustrate the secondary flow direction at the secondary inlet and the axial velocity at the end of the entrance reached about 113 m/s with no recirculation zone present.

Figure 6.19: Pressure map from Eilmer3 simulations at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 5.88 kPa.

Figure 6.20: Centreline pressure from Eilmer3 simulations at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 5.88 kPa.
Figure 6.21: Simulated velocity profiles for five sections along the ejector at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 5.88 kPa. Each velocity increment corresponding to 100 m/s.

Figure 6.22: Velocity map from Eilmer3 simulation of the ejector at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 5.88 kPa.
6.10 Simulated pressure and velocity distributions

Figure 6.23: Velocity map from Eilmer3 simulation of the secondary inlet at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 5.88 kPa.

6.10.2 Unchoked flow

In the unchoked ejector flow regime, the primary flow is still choked but the secondary flow is unchoked (secondary velocity is less than the sonic condition) resulting in a decrease in the secondary mass flow rate. Unchoked flow comes from an increase in the condenser pressure higher than the critical value. A secondary series of shock waves moving farther upstream causes a decrease in the axial velocity as shown in Figure 6.26. Simultaneously, the secondary flow separates at the conical mixing chamber wall as presented in section CC in Figure 6.26. Figures 6.27 and 6.27 illustrate the position of the recirculation of the secondary flow. Pianthong et al. (2007) observed recirculation in the ejector throat when there is an increase in condenser pressure to higher than critical pressure in the CFD simulation. However, it is very difficult and expensive to visualise the recirculation of the secondary flow in the experiment. Therefore, the Eilmer3 simulation is very useful in depicting the flow inside the ejector.
Figure 6.24: Pressure map from Eilmer3 simulations at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 6.5 kPa.

Figure 6.25: Centreline pressure from Eilmer3 simulations at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 6.5 kPa.

Figure 6.26: Simulated velocity profiles for five sections along the ejector at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 6.5 kPa. CC-location is the separation flow at the wall of the mixing chamber. Each velocity increment corresponds to 100 m/s.
6.10 Simulated pressure and velocity distributions

Figure 6.27: Velocity map from Eilmer3 simulation of the ejector at primary stream conditions of 270 kPa, 130 °C with an evaporator temperature of 14 °C and condenser pressure of 6.5 kPa.

Figure 6.28: Flow recirculation on the mixing chamber wall at section CC for conditions and location indicated in Figure 6.27.
6.10.3 Reverse flow

Simulation of the ejector in the reverse flow state is shown in Figures 6.29 and 6.30. The secondary oblique shock wave train has moved further toward the primary nozzle. This shock train causes disturbance in the primary jet core until the expanded wave cannot be produced (Ruangtrakoon et al., 2013). The increase in condenser pressure causes the primary stream to be forced back to the entrance of the secondary flow. Thus the primary flow goes back to the evaporator, which causes the entrainment ratio into drop to zero. Figures 6.31 and 6.32 show the decrease in the axial velocity in fourth and fifth positions, and reversed flow in first and second positions. Results from the simulation at the secondary inlet position are presented in Figure 6.33. This indicates the completely reverse flow in the secondary inlet (block 4), and reverse flow and recirculation in the conical mixing chamber (block 7) as presented in Figure 6.34.

Figure 6.29: Pressure Eilmer3 simulated at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 6.9 kPa.
Figure 6.30: Centreline pressure Elmer3 simulated at primary stream conditions of 270 kPa, 130 °C with an evaporator temperature of 14 °C and condenser pressure of 6.9 kPa reverse flow.

Figure 6.31: Simulated velocity profiles for five sections along the ejector at primary stream conditions of 270 kPa, 130 °C with an evaporator temperature of 14 °C and condenser pressure of 6.9 kPa. DD and EE locations are the reverse flow at the secondary inlet region. Each velocity increment corresponds to 100 m/s.
Figure 6.32: Velocity map from Eilmer3 simulation of the ejector at primary stream conditions of 270 kPa, 130°C with an evaporator temperature of 14°C and condenser pressure of 6.9 kPa.

Figure 6.33: Reversed flow profile of section DD at secondary inlet region for conditions and location indicated in Figure 6.32.

Figure 6.34: Reversed flow profile of section EE at the conical mixing chamber for conditions and location indicated in Figure 6.32.
6.11 Conclusion

A computational fluid dynamic simulation tool called Eilmer3 was used to simulate the steam ejector using the $k$-$\omega$ turbulence model and an ideal gas model for the steam.

Eilmer3 simulations are in good agreement with experimental data for wall static pressure along the ejector when the ejector was operated with a primary stream pressure of either 200 or 270 kPa, and evaporator temperatures were 6, 10, and 14°C.

The differences between the CFD simulations and experimental data of entrainment ratio for the primary steam pressures 200 and 270 kPa with an evaporator 14°C were 2.6% and 2.9% respectively.

The differences between the CFD simulations and experiments in the value of the critical condenser pressure were 8.8% at a primary stream pressure of 270 kPa and 15.4% at a primary stream pressure of 200 kPa, when the evaporator temperature was 14°C in both cases.

Eilmer3 simulations allowed the visualisation of the recirculation inside the diffuser at low condenser pressure, and the visualisation of a recirculation zone in the mixing chamber which appeared when the condenser pressure increased. At the higher condenser pressures, the recirculation zone in the diffuser was removed.

Eilmer3 simulations also reveal the reverse flow in the secondary inlet after the condenser pressure exceeded the breakdown pressure.

Eimer3 simulations also show the first and secondary series shock waves.
Chapter 7

Mach 4 steam jet

7.1 Introduction

Steam ejectors have a very low coefficient of performance compared to normal pumping systems (Dandachi, 1990). Researchers have focused on the design and geometry of ejectors (Yuan-Jen and Yau-Ming, 2000; Pianthong et al., 2007; Varga et al., 2009) and have also developed methods to visualise the flow inside ejectors. Chandrasekhar, Krothapalli and Baganoff (1991) investigated the performance characteristics of an under-expanded, multiple jet ejector using a schlieren method. Using a laser sheet method Desevaux, Prenel and Hostache (1994), (2001a), and (2001b) visually distinguished between the supersonic primary flow and the subsonic secondary flow for an air-air ejector in the region before complete mixing was achieved.

This chapter introduces a new approach for investigating the mixing region generated by a steam ejector nozzle. Rather than attempting to visualise the flow field, a new approach using a special pitot probe designed and fabricated at University of Southern Queensland was used for this study. A comparison between experimental results and CFD simulations using Eilmer3 will also be presented.
7.2 Experimental arrangement

The experimental refrigerator steam ejector with cooling capacity 3.5 kW described in Chapter 3 was used for this work including a modified form of mixing chamber described in Section 7.2.1. A photograph of the rig setup is shown in Figure 7.1. Different parameters were varied such as the flow rate of the primary flow from 0.001 to 13.5 kg/hr, pressures from 10 to 400 kPa, and temperature between 100 and 145°C. The condenser pressure ranged from 3 to 7 kPa.

![Figure 7.1: Photograph of the steam ejector system with mixing chamber.](image)

7.2.1 Mixing chamber

The arrangement consisted of the primary nozzle, the mixing chamber, a bell mouth inlet, and the diffuser. These items were introduced in Chapter 3, except the mixing chamber and the bell mouth inlet.

- The mixing chamber had windows for observation of the pitot tube traverse during testing and for visualisation work which will be discussed in Chapter 8. It was designed to fit in the same location of the mixing chamber discussed in Chapter 3. The mixing chamber was fabricated with a rectangular cross section area of 106 mm (in width) and 100 mm (in height). The top and bottom of the mixing chamber were made from U-channel and the two sides were made from transparent acrylic (perspex) 10 mm thick. The perspex windows were fastened by ten bolts and sealed by rubber between the windows and the U-channel.
- The bell mouth inlet to the diffuser was made from a plastic tube which was reshaped in the laboratory. The bell mouth shape allows an increased amount of steam to be drawn into the diffuser with a reduced pressure drop.

At the top of the mixing chamber, four holes were made to allow the pitot probe to be positioned at different distances from the nozzle exit: 1, 25, 50, and 70 mm. All holes were aligned with the centreline of the nozzle. Holes not in use were plugged by a bolt, nut, and gasket to prevent any leakage into the system. Figures 7.2 and 7.3 illustrate the steam ejector nozzle with bell mouth inlet relative to the mixing chamber and the cross section of the mixing chamber at section AA, respectively. Figure 7.4 shows a photograph of the mixing chamber including nozzle, pitot tube, and the bell mouth inlet.

Figure 7.2: Schematic diagram showing the mixing chamber.
7.2 Experimental arrangement

Figure 7.3: Schematic diagram showing the mixing chamber at cross section AA.

Figure 7.4: Photograph of the mixing chamber with windows, nozzle, pitot tube and bell mouth.

7.2.2 Pitot tube probe

Head (1949) and Gery (1952) presented significant recommendations for pitot pressure measurements. The pitot tube was unaffected by condensation during tests on condensing air through a hypersonic wind tunnel. McBride and Sherman (1971) used pitot pressure measurements for the flow of a two component mixture of gases, one inert and one condensable. The measurements were made at the exit of a nozzle with Mach number of 25. Pitot pressure results for tubes having an internal diameter of more than 0.125 inch were consistent, while lower pressures were measured for pitot tubes with less than 0.008 inch internal diameter. As a consequence, researchers have
tended to assume that pitot pressure is insensitive to condensation (McBride and Sherman, 1971).

In this study, the pitot tube was designed to traverse the mixing chamber smoothly. Figure 7.5 shows the pitot probe and traversing gear. The pitot tube was made from stainless steel tube with 1.7 mm outside diameter and 0.85 mm inside diameter. Figure 7.6 illustrates the sealing method for the sliding support. An O-ring was used to seal the sliding support. The pitot tube traversed through the mixing flow and was driven by a DC motor. The DC motor was fixed on the top of the slider. Simultaneously, a potentiometer was connected with the base of the DC motor to give an indication of the pitot probe position. A solid bar of the same diameter as the pitot tube was fixed below the pitot probe to minimise changes of the back pressure which could occur when the pitot tube is moved through the mixing jet.

![Figure 7.5: Overall arrangement of traverse mechanism.](image-url)
7.2 Experimental arrangement

7.2.3 Pressure and temperature

High natural frequency pressure transducers from Kulite (model IS-XTL-190) were used to measure the pitot pressure and the back pressure in the mixing chamber. The pressure transducers operating range was from 0 to 70 kPa. The pressure transducer used to measure the primary flow was from Wika (Model A-10) with a pressure range from 0 to 700 kPa. The stagnation temperature of the primary flow and the condenser temperature were measured using type K thermocouples. The pressure transducers and thermocouples were wired to a National Instruments Compact Data Acquisition (cDAQ) as described in Chapter 3.

7.2.4 Operating procedure

The steam generator supplied superheated steam to the nozzle with pressures of 270 kPa and 300 kPa at a temperature of 144°C. The steam was delivered to the nozzle through an insulated stainless steel pipe. No secondary flow from the evaporator was introduced in these experiments. The flow within the mixing chamber entered the diffuser via a bell mouth, then the steam left the diffuser to enter the condenser. All the
data were taken when the apparatus reached a steady operating state, which occurred after about 15 minutes of operation. Pitot pressure surveys were performed at four positions: 1, 25, 50, and 70 mm downstream from the nozzle exit. At each position, three tests were performed to gauge the repeatability of the measurements. The pitot tube survey in any particular test commenced from a point just below the centreline of the nozzle core and continued upwards to a location about 5 mm above the nozzle lip. This strategy was adopted to reduce the possibility of condensing droplets of water entering the pitot tube during the survey.

7.3 System performance during the test time

The stability and repeatability of the system operation during test time was evaluated. In this study, the pressure regulator on the primary line was not used. This enabled the operation at superheated steam conditions. Removing the pressure regulator from the system led to some variation in the primary stream pressure. As a consequence, the pitot pressure and the static pressure in the mixing chamber also changed. Additional variation in the mixing chamber static pressure arose due to changes in blockage effects as the pitot tube was traversed across the jet. These variations were minimised through the addition of a vertical section below the pitot tube, and this addition provided the jet with a relatively constant blockage effect, as illustrated in Figure 7.5.

7.3.1 Primary flow stability

A typical variation of primary stream stagnation pressure is presented in Figure 7.7. Figures 7.8 and 7.9 show how the pitot pressure and back pressure were affected by the variation of stagnation pressure illustrated in Figure 7.7. In this test, the pitot probe was mounted in the centreline of the nozzle (and was not traversed) and the static pressure probe was mounted in the top middle of the chamber. The data was recorded for a period of 10 seconds. The primary stream stagnation pressure changed from 324 kPa to 278.5 kPa, a variation of around 14%. Consequently, the pitot tube pressure also changed by about 14% as illustrated in Figure 7.8. During this same
period, the mixing chamber pressure changed in the opposite direction by about 11.2% as shown in Figure 7.9. A normalised form of pressure $P_{\text{pit}}/P_o$ is used to present results as the nozzle exit pitot pressure follows the stagnation pressure ($P_o$). However, since the mixing chamber pressure does not follow $P_o$, some variability in mixing jet parameters will remain unaccounted for in the results.

![Figure 7.7](image1.png)

*Figure 7.7: Primary stream stagnation pressure change during a period of 10 s.*

![Figure 7.8](image2.png)

*Figure 7.8: Pitot pressure change during the period of 10 s identified in Figure 7.7.*

![Figure 7.9](image3.png)

*Figure 7.9: Mixing chamber static pressure change during the period of 10 s identified in Figure 7.7.*
7.3.2 Repeatability of survey data

One primary stream operating condition (270 kPa, 144°C) is considered to illustrate the repeatability of the survey results for the four positions of the pitot tube. The traverse of the mixing region took about five seconds. Figures 7.11, 7.12, 7.13, and 7.14 present repeated measurements from traversing the nozzle flow at 1, 25, 50, and 70 mm from the nozzle exit. The pitot probe traverse in any test started from a point which was 3.5 mm below the centreline of the nozzle and continued upwards to a position 5 mm above the nozzle exit as illustrated in Figure 7.10. Therefore, the data presented in the lower portion of these figures is actually a mirror of the data from the positions above the centreline to give the impression of the mixing jet. The data in Figure 7.11 for the 1 mm distance are consistent to better than 0.5% over the three experiments. For the locations of 25, 50, and 70 mm, the data from the repeated experiments in Figures 7.12, 7.13, and 7.14 are consistent to within about 4%, 4%, and 2% respectively.

Figure 7.10: Normalised pitot pressure from a point 3.5 mm below the centreline of the nozzle to a location 5 mm above the nozzle lip at 1 mm downstream from the nozzle lip, at a primary stream stagnation pressure of 270 kPa and temperature of 144°C.
7.3 System performance during the test time

Figure 7.11: Normalised pitot pressure at 1 mm distance from nozzle lip at a primary stream stagnation pressure of 270 kPa and temperature of 144°C.

Figure 7.12: Normalised pitot pressure at 25 mm distance from nozzle lip at a primary stream stagnation pressure of 270 kPa and temperature of 144°C.
Figure 7.13: Normalised pitot pressure at 50 mm distance from nozzle lip at a primary stream stagnation pressure of 270 kPa and temperature of 144°C.

Figure 7.14: Normalised pitot pressure at 70 mm distance from nozzle lip at a primary stream stagnation pressure of 270 kPa and temperature of 144°C.
7.4 Over-expanded jets

In the present work, the static pressure in the mixing chamber was higher than the estimated static pressure at the nozzle exit $P_e$. Thus, the flow formed an over-expanded jet. The general over-expanded jet structure results in a flow containing a series of oblique waves and expansion waves as illustrated in Figure 7.15. Anderson (1990) classified two wave reflection scenarios:

1. If waves are incident on a solid boundary, they will reflect as the same type. The axis of the jet effectively forms an inviscid solid boundary and therefore expansion waves reflect as expansion waves and compression waves reflect as compression waves when they reach the jet axis.

2. If waves are incident on a free boundary, they will reflect as the opposite type. An expansion wave reflects from the jet edge as a compression waves and a compression waves reflects as an expansion waves.

Figure 7.15: Structure of the flow near the nozzle exit of over-expanded convergent-divergent nozzle (Oosthuizen and Carscallen, 1997).
7.5 Results and discussion

Figures 7.16 and 7.17 show the pitot pressure survey data normalised by the stagnation pressure at the two different stagnation pressures of 270 kPa and 300 kPa for the four positions of the pitot probe from nozzle exit: 1, 25, 50, and 70 mm.

A more detailed view of the data from the experiment at 270 kPa stagnation pressure at location 1 mm distance from the nozzle exit is presented in Figure 7.18. The shear layer thickness and shock effects on the pitot pressure data are illustrated in Figure 7.18. The upper edge of the shear layer was defined by the first measurable deviation from the background value. The inner edge of the shear layer is taken to coincide with the sharp change in the pitot pressure which actually arises due to shock wave effects at the nozzle exit. Thus, the total thickness of the shear layer is equal to 1.7 mm.

Figure 7.19 presents results at the flow conditions of Figure 7.18 for the survey location 25 mm downstream of the nozzle lip. In this case, the shear layer thickness was identified as 4 mm. At the inner edge of the shear layer, expansion waves cause a slight increase in the pitot pressure towards the centreline. Closer to the centreline a decrease in the pitot pressure is registered due to a shock wave as illustrated in Figure 7.22.

Figure 7.20 illustrates a further increase in the shear layer thickness at position 50 mm, reaching a thickness of 7 mm with a further increase of the shear layer thickness up to the location at 70 mm downstream from the nozzle lip: a value of about 12.5 mm as presented in Figure 7.21.

The shear layer thickness results from Figures 7.18, 7.19, 7.20, and 7.21 are presented in Figure 7.22, which is interpreted as a jet flow shape of steam at the primary stream conditions of 270 kPa and 144°C. The shape of jet flow is consistent with studies of Anderson (1990) and Oosthuizen and Carscallen (1997).
Figure 7.16: Normalized pitot pressure data for four positions 1, 25, 50, and 70 mm downstream of the nozzle at 270 kPa stagnation pressure, 144°C stagnation temperature and 3 kPa back pressure. Each horizontal increment corresponds to 0.01.

Figure 7.17: Normalized pitot pressure data for four positions 1, 25, 50, and 70 mm downstream of the nozzle at 300 kPa stagnation pressure, 144°C stagnation temperature and 3 kPa back pressure. Each horizontal increment corresponds to 0.01.
Figure 7.18: Estimation of the shear layer thickness at 1 mm downstream from the nozzle exit for the primary stream pressure of 270 kPa and 3 kPa back pressure.

Figure 7.19: Estimation of the shear layer thickness at 25 mm downstream from the nozzle exit for the primary stream pressure of 270 kPa and 3 kPa back pressure.
7.5 Results and discussion

Figure 7.20: Estimation of the shear layer thickness at 50 mm downstream from the nozzle exit for the primary stream pressure of 270 kPa and 3 kPa back pressure.

Figure 7.21: Estimation of the shear layer thickness at 70 mm downstream from the nozzle exit for the primary stream pressure of 270 kPa and 3 kPa back pressure.
Figure 7.22: Shape of jet flow interpreted from the pitot pressure data.
7.6 Simulation using Eilmer3

7.6.1 Simulation of mixing chamber configuration

The mixing chamber was divided into nine blocks to form the shape of the mixing chamber as shown in Figure 7.23. To control the pressure inside the mixing chamber, three additional blocks were created at the end of the mixing chamber (B9, B10, and B11) as illustrated in Figure 7.24. A medium density mesh similar to that adopted in Chapter 6 was used in this study as illustrated in Figure 7.25. Primary stream inlet conditions and back pressure conditions imposed by the outlet geometry are presented in Table 7.1.

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<td>144</td>
</tr>
<tr>
<td>Back pressure</td>
<td>3.0 to 3.4</td>
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Table 7.1: Ejector operating conditions used in the simulation.

Figure 7.23: Geometry of the axisymmetric nozzle and mixing chamber with arrangement of blocks (denoted B0 to B8).

Figure 7.24: Geometry of the axisymmetric nozzle and mixing chamber with arrangement of blocks including blocks at the outlet which are used to control the back pressure (denoted B9 to B11).
Figure 7.25: Mesh used to simulate the free jet inside the mixing chamber.

7.6.2 Comparison of Eilmer3 simulations with pitot data

In this section, comparisons between experimental data and CFD simulations using Eilmer3 will be presented. Figure 7.26 illustrates the comparison between the normalized pitot pressure in the experiments (deduced from $P_{pit}/P_o$ on the assumption that $\gamma = 1.326$) and Eilmer3 simulations at the 1 mm distance from the nozzle exit. Figure 7.27 presents the comparison between the Mach number in the experiments and Eilmer3 simulations at the 1 mm distance from the nozzle exit. The simulations of Eilmer3 are consistent with the experiment results for the primary stream pressure 270 kPa. This demonstrates that Eilmer3 is accurately simulating the flow within the nozzle. Figure 7.28 shows the map of simulated static pressure near the nozzle exit; it depicts clearly the shock wave and shear layer at a location 1 mm downstream from the nozzle exit, similar to the experimental results. Moreover, Figure 7.29 presents the map of axial velocity profile at location 1 mm from nozzle lip. A similar level of consistency between simulations and experimental data was obtained at this station for the primary stream pressure of 300 kPa.

Figures 7.30, 7.31, and 7.32 present the comparison between experimental results and the CFD simulation of the normalised pitot pressure profile within the jet at 25, 50, and 70 mm respectively for the conditions: 270 kPa stagnation pressure, 144°C stagnation temperature, and 3 kPa back pressure. The results presented in Figures 7.30, 7.31, and 7.32 show that the CFD simulation is not consistent with the experimental data. At the 50 mm station (Figure 7.31), values from the CFD simulation exceed the normalised pitot pressure on the centreline of the jet flow by a factor of 2.

One possible reason for the difference between CFD simulation and experimental data is the use of an ideal gas model in CFD instead of a real condensing gas. Another
reason that could cause the difference is the shape of mixing chamber used in CFD simulation. A circular cross section shape of the mixing chamber was used in the CFD simulations, while the actual cross section of the mixing chamber was rectangular.

Clifton and Cutler (2007) compared results from a CFD simulation (CFD code Vulcan) with data from a coaxial mixing jet experiment performed by Cutler, Diskin, Drummond and White (2006). The results show that the CFD simulation and experiment were in agreement close to the nozzle exit and that this agreement disappeared at downstream locations.

Figure 7.26: Comparison of measured and simulated pitot pressure ratio profile across the steam jet at 1 mm distance from nozzle lip at 270 kPa stagnation pressure and 3 kPa back pressure.
Figure 7.27: Comparison of measured and simulated Mach number across the steam jet at 1 mm distance from nozzle lip at 270 kPa stagnation pressure and 3 kPa back pressure.

Figure 7.28: Map of simulated static pressures near the nozzle exit.
Figure 7.29: Map of simulated axial velocities near the nozzle exit.

Figure 7.30: Comparison of measured and simulated pitot pressure ratio profile across the steam jet at 25 mm distance from nozzle lip at 270 kPa stagnation pressure and 3 kPa back pressure.
Figure 7.31: Comparison of measured and simulated pitot pressure ratio profile across the steam jet at 50 mm distance from nozzle lip at 270 kPa stagnation pressure and 3 kPa back pressure.

Figure 7.32: Comparison of measured and simulated pitot pressure ratio profile across the steam jet at 70 mm distance from nozzle lip at 270 kPa stagnation pressure and 3 kPa back pressure.
7.7 Momentum integral analysis

Given the lack of agreement between the experimental results and the computational simulation at stations downstream of the nozzle (x = 25, 50, and 70 mm), further investigation of the jet mixing profiles was undertaken. In this section, an attempt is made to apply a momentum internal analysis to the mixing chamber flow. Figure 7.33(a) shows the layout of the mixing chamber including the control volume (CV). The control volume extends to the downstream location of 70 mm and upstream behind the nozzle exit to the solid wall where there is no momentum transport across the boundary. Figure 7.26 demonstrates that the experimental data and CFD simulations are consistent at the first station immediately downstream of the nozzle. Therefore the data from the CFD simulation was used to calculate the momentum transport into the CV at the nozzle exit according to

\[ \int \rho u^2 dA \]  

(7.1)

When this integration was performed, the value obtained was 3.96 N.

At the downstream location (x = 70 mm) the measured pitot pressure profile suggests that the static pressure at this location is reasonably uniform since wave effects are not apparent. At this station the pitot pressure measured at the locations furthest from the centreline was also within about 4% of the value measured on the wall of the duct and this offers further confirmation that the static pressure is reasonably uniform across the mixing jet at x = 70 mm. The experimental data for \( \frac{P_{\text{pitot}}}{P_{\text{static}}} \) were used to calculate a Mach number profile from the Rayleigh pitot formula assuming two different values for the specific heat ratio \( \gamma = 1.001 \) and 1.326) as presented in Figure 7.34. From the Mach number profile, the momentum can be deduced since

\[ \rho u^2 = \gamma PM^2 \]  

(7.2)

Profiles of the momentum flux for the two ratio of specific heat cases were obtained by smoothing Mach number profile (Figure 7.34) and applying the scaling of the equation (7.1) as presented in Figure 7.35. When the integration specified in Equation (7.1) was performed, a value of 4.66 N was obtained with \( \gamma = 1.001 \) and 4.78 N was obtained with \( \gamma = 1.326 \).
The conservation of linear momentum equation for the x-direction can be expressed

\[ \text{momentum transport into the CV + net force on CV} = \text{momentum transport out of the CV} \quad (7.3) \]

Since the static pressure at station 2 is higher than station 1 and the region of the CV to the left of the nozzle inlet in Figure 7.33, the net force on the CV is most likely to be negative (acting in the negative x-direction). Thus, the momentum out of the CV should be less than the momentum into the CV, but the present momentum integral analysis on the experimental data at the 70 mm station indicates this cannot be achieved even with \( \gamma = 1 \). Although using a lower value for \( \gamma \) for the deduction of \( \rho u^2 \) from the pitot pressure measurement improved the momentum integral analysis, it is not possible to conclude that condensation is playing an important role. This is because the pitot survey was performed over the central region of the mixing duct and therefore it does not provide data on the likely recirculation at station 2 which could cause an additional component of momentum transport into to CV at this station.
Figure 7.33: (a) Illustration of the nozzle and mixing chamber arrangement including the control volume used in the analysis; (b) Illustration of pressure forces applied to the control volume.
Figure 7.34: Mach number distributions at $x = 70\text{ mm}$ deduced from the pitot pressure measurement using an ideal gas assumption.

Figure 7.35: Momentum flux distributions at $x = 70\text{ mm}$ deduced from the pitot pressure measurement using an ideal gas assumption.
7.8 Conclusion

A special pitot probe was designed and fabricated to investigate the mixing region generated by a steam ejector nozzle.

In this case, a pressure regulator at the primary supply was not used in order to achieve superheated steam conditions. There is some variation in the main primary pressure as a consequence of removing the pressure regulator. The variations of primary steam pressure and the pitot pressure were about 14\%, while the variation of mixing chamber pressure was about 11.2\%.

The repeatability of the pitot pressure measurements for four locations of the pitot probe: 1, 25, 50, and 70 mm were 0.5\%, 4\%, 4\%, and 2\% respectively.

The shear layer thickness deduced at the four measurement locations: 1, 25, 50, and 70 mm were 1.7, 4, 7, and 12.5 mm respectively.

The shape of steam jet flow was interpreted from the pitot pressure data.

The CFD simulations using Eilmer3 were not consistent with the experimental data except at the location 1 mm downstream of the nozzle exit.

The momentum integral analysis demonstrated the influence of possible condensation effects through the use of different values for the ratio of specific heats, but it is not possible to conclude condensation effects did influence the measurements because of possible unaccounted momentum transport in a recirculation zone at the 70 mm station.
Chapter 8

Flow visualisation

8.1 Introduction

Visualisation of ejector flows has been attempted previously but most work has focussed on the air jet. Fabri and Siestrunck (1958) used a schlieren method to visualise the flow in supersonic air ejectors operated over different ranges of conditions. Hong et al. (2004) utilised a schlieren method and a high speed camera to capture unsteady phenomena in the entrance region of the mixing chamber of an air ejector. Bouhanguel et al. (2010) used Rayleigh scattering and Mie scattering to visualise supersonic air flow inside the ejector by using a laser sheet from a CW argon laser with 120 mW of power in the green line ($\lambda_o=514$ nm). However, until the present time, flow visualisation has not been applied to a steam jet ejector.

This chapter presents an arrangement of Toepler’s schlieren method for visualisation of the steam jet from the nozzle inside the mixing chamber. Two new designs of lenses that mange the refraction of light in the perspex mixing chamber are also presented. The performance of the lens arrangements will be described. The detection of density gradient effects in a steam jet flow will be presented. A comparison of the density gradients in representative steam and air flows will also be discussed.
8.2 Lens design and commissioning

8.2.1 Overview of optical arrangements

In conventional visualisation methods such as shadowgraph, schlieren, and background oriented schlieren, a circular cross section pipe containing the test fluid is not suitable. With a circular pipe, incoming parallel rays are dispersed due to refraction on passing through the outer and inner surfaces of the section. In order to avoid this problem, two types of correcting lenses were investigated in this work.

1. An ordinary perspex pipe with two lenses was investigated as a way to correct the refraction effects as shown in Figure 8.1.

2. A tube arrangement with integral lenses was investigated as second option for refraction effects correction as shown in Figure 8.2.

![Figure 8.1: Cross section of lenses separate to the pipe: two lenses with one pipe.](image1)

![Figure 8.2: Cross section of lenses as integral to the pipe.](image2)
8.2.2 Overview of ray-tracing design

Figure 8.3(a) shows the geometry for a perspex pipe with separate correcting lenses. The pipe construction is axiymetric so it is sufficient to consider a quarter of the geometry. The ray of light $Z_1-Z_2$ enters the internal surface of the pipe parallel to the centre plane and refraction through the perspex interface occurs according to Snell’s law between the steam medium and the perspex material.

The light ray $Z_2-Z_3$ is also refracted as it leaves the outer surface of the pipe and Snell’s law between perspex material and the air is also applied. The ray $Z_3-Z_4$ impinges on the surface of the lens and the design processes requires identification of the angle of the lens surface such that the ray $Z_4-Z_5$ will again be parallel to the centre plane. The distance $h$ has a very strong effect on the size of the correction lens.

Figure 8.3(b) illustrates the geometry required for the design of the integral lens. The light ray $Z_1-Z_2$ impinges on the internal surface of the pipe parallel to centre plane. The orientation of the outer surface of the lens is then designed to redirect the light ray so it is parallel to the centre plane when it leaves the outer surface.

Additional details on the theory and design of the two arrangements are presented in Appendix D.

![Figure 8.3: Ray-tracing method for two of the optical arrangements: (a) Ordinary pipe with two lenses; (b) Pipe with integral lenses.](image-url)
8.2.3 Performance assessment

The configuration consisting of an ordinary perspex pipe with two correcting lenses requires careful alignment and precise position control in the schlieren visualisation apparatus. The integral lenses configuration requires less effort in alignment in the schlieren system but is considerably more expensive to machine. The integral lens arrangement was trialled extensively in the present work.

The success of the optical design method is demonstrated in Figure 8.4 which shows a piece of paper inside the integral lens arrangement and demonstrates a clear view across the full height of the duct.

Unfortunately, neither optical arrangement was entirely satisfactory for the schlieren visualisation work. Machining the optical devices from either round perspex or flat perspex appeared to introduce shadow-effects when these components were positioned within the schlieren system. It is assumed that these effects arise due to birefringence in the perspex material.

For the schlieren visualisation reported in the following section, a rectangular mixing chamber was used as described in Chapter 7. This rectangular mixing chamber also used perspex to enable visualisation, but in this case, no machining of the optical
surfaces was performed.

8.3 Schlieren visualisation arrangement

8.3.1 Method overview

Toepler’s method is presented in Figure 8.5, where \( L \) represents the light source with lower point \( M \) and upper point \( N \). Settles (2001) indicates that light source edges must have at least one sharp edge limited by a knife or an opaque mask. \( K \) is a single schlieren lens that is corrected for a spherical and chromatic aberration (Schardin, 1947). The schlieren object is represented by \( S \). \( l \) is the distance between light source and the \( K \) lens and \( l_1 \) is the distance between the \( K \) lens and knife edge. The lens \( K \) should produce an image from the light source at position \( L' \). The knife edge represented at \( B \) is adjusted parallel to the upper boundary of the image of the light source, and \( a' \) is the distance between knife edge and upper boundary of the image of the light source. The camera lens \( O \) focuses the schlieren objective \( S \) upon the image plane \( P \). The illumination should be uniform everywhere when there is no deflecting effect on the light that passes through the schlieren object \( S \). However, if there is any deflection in the light that occurs in the schlieren object \( S \), the image will appear either brighter or darker, depending upon whether the light bent upward or downward relative to the knife edge.

![Figure 8.5: Arrangement of Toepler’s schlieren method (Schardin, 1947).](image-url)
8.3 Schlieren visualisation arrangement

8.3.2 System sensitivity

Schlieren sensitivity or contrast sensitivity is the rate of change of image contrast with respect to refraction angle (Settles, 2001). Schardin (1947) found that the sensitivity of the Toepler’s schlieren method depends on the following:

- The distance \( s \) between schlieren object and the knife edge.
- The extent to which the knife edge cuts off the image of the light source

\[ \Delta a = \epsilon s \] \hspace{1cm} (8.1)

where \( \Delta a \) is the extent of change, and \( \epsilon \) represents the smallest visible angle of deflection. Figure 8.6 illustrates the schlieren arrangement used in the present work, and the quantities \( s \) and \( \epsilon \) are included in this figure.

![Figure 8.6: Arrangement of the schlieren method in the present work.](image)

Holder and North (1953) recommended adjusting the sensitivity of the schlieren method by a change in the height of the light source, \( \Delta a \). A convenient method is to use a rectangular source of light, which can change the direction of the axis of the optical system so that the height of the source can be adjusted to any value between its length and its width.
8.3.3 Experimental arrangement

The experiment arrangement used the following features:

- **Illumination.** Light source consisting of one tungsten halogen bulb: 60/55 W, 12 VDC supply.

- **Optics.** One condenser plano-convex lens of 50 mm diameter and 200 mm focal length; and one plano-convex lens of 125 mm diameter by 1000 mm focal length was used.

- **Camera.** A high speed camera, Olympus (model: i-speed 3). This camera has the capability to reach 150,000 frames per second with a memory size 8GB. However, in this case, 2000 frames per second were used. A 95 mm diameter lens with 600 mm focal length was used with the camera.

- **Miscellaneous hardware.** The slit, filter, and knife edge are essential hardware for the system. The slit used in this experiment was made from two razor blades separated by 0.6 mm with additional masking to give a width of 1 mm. A red filter was used in this set-up and the knife edge was made from a razor blade.

- **Schlieren object.** For the test results reported in section 8.4, the schlieren object was the Mach 4 jet created within the rectangular mixing chamber described in the Chapter 7.

Figure 8.6 illustrates the single field lens schlieren arrangement used in this work. Figure 8.7 shows the camera side, which consists of camera, lens, and knife edge. Figure 8.8 presents the schlieren object side, which consists of 1000 mm lens, slit, condenser lens, filter, and light source.
8.3 Schlieren visualisation arrangement

Figure 8.7: Camera, 600 mm focal lens, and knife edge mounted on slide rail with camera monitor.

Figure 8.8: Lamp, filter, condenser lens, slit, and 1000 mm focal lens mounted on slide rail with Schlieren object.


8.4 Schlieren visualisation results

Figure 8.9 presents the schlieren image of an over-expanded supersonic air jet emerging from the nozzle which was built for the steam ejector. Oblique shock waves can be clearly seen in the image. The shock waves created by a pitot tube under nominally identical flow conditions are presented in Figure 8.10.

Figures 8.11(a) and 8.11(b) illustrate the schlieren image for steam nozzle at stagnation pressure of 380 kPa and stagnation temperature of 144°C, with a mixing chamber pressure of 3 kPa. It is difficult to visually distinguish the upper edge of the steam flow in Figure 8.11(a) but it is possible to do so through additional image analysis. Figure 8.11(b) includes a line which represents the upper edge of the steam jet. To identify this edge feature through image analysis, different zones can be identified, and one such analysis zone is illustrated in Figure 8.11(b). By comparing the intensity of the light within the analysis zone before and after the jet reached the steady operating conditions shown, it is possible to define some structure. Figure 8.12 presents the relative change of the intensity associated with a slight change in the jet operating conditions. Figure 8.12 demonstrates a strong feature at about \( y = 9 \) mm corresponding to the edge feature annotated in Figure 8.11(b).

Figure 8.9: Schlieren image of air discharged from the nozzle within the mixing chamber for stagnation conditions of 380 kPa and 25°C.
Figure 8.10: Schlieren image of air discharged from the nozzle within the mixing chamber for stagnation conditions of 380 kPa and 25°C with a pitot tube positioned close to the nozzle exit.
Figure 8.11: (a) Schlieren image of the steam nozzle within the mixing chamber for stagnation conditions of 380 kPa and 144°C, with mixing chamber pressure of 3 kPa; (b) Same photograph as part (a) but annotated showing the analysis zone and the observed edge effect.
Figure 8.12: Normalized brightness of the Schlieren image within the analysis zone (defined in Figure 8.11(b)) for steam flow at stagnation conditions of 380 kPa and 144°C with a mixing chamber static pressure of 3 kPa.
8.5 Assessment of density gradients

Visualisation of the present steam jet using schlieren methods was known to be a challenging undertaking because of the very low density gradients. This is confirmed with the calculation presented below. Steam flow stagnation conditions were taken as 380 kPa and 144°C so from equation 8.2 using $R = 461.5 \text{J/kg K}$, the stagnation density can be calculated.

$$\rho_o = \frac{P_o}{RT_o} \quad (8.2)$$

This gives the stagnation density of steam as $\rho_o = 1.97 \text{kg/m}^3$

The density at the nozzle exit can then be calculated using

$$\frac{\rho_o}{\rho} = \left(1 + \frac{\gamma - 1}{2} M^2\right) \frac{1}{\gamma - 1} \quad (8.3)$$

where

$$\gamma = 1.326$$

$M =$ Mach number at nozzle exit = 4.2

This gives the value of $\rho = 0.0309 \text{kg/m}^3$ at the nozzle exit. The density of steam in the mixing chamber at locations away from the high speed jet can also be estimated from equation 8.2 using a pressure of 3 kPa and temperature of 25°C. This gives a density value of 0.0218 kg/m$^3$ in the background region of the mixing chamber. Taking the radius of the nozzle (6.75 mm) as a representative spatial distance over which the density gradient will develop, the density gradient can be estimated as

$$\frac{\Delta \rho}{\Delta y} = \frac{0.00908}{0.00675} = 1.346 \text{kg/m}^4 \quad (8.4)$$

A similar calculation for air supplied to the nozzle at stagnation conditions of 380 kPa, and 25°C yields $\rho_o = 4.44 \text{kg/m}^3$, $\rho = 0.1025 \text{kg/m}^3$, at the nozzle exit, and 0.035 kg/m$^3$ in the background region of the mixing chamber. From which, the density gradient in the case of air is estimated as

$$\frac{\Delta \rho}{\Delta y} = \frac{0.0675}{0.00675} = 10 \text{kg/m}^4 \quad (8.5)$$
Thus, the estimated air density gradient is more than seven times larger than the steam density change.

The CFD simulations also demonstrate that the density gradients in the steam jet are almost an order of magnitude lower in magnitude than those generated with air at similar pressure but with an ambient stagnation temperature.

Figures 8.13 and 8.14 illustrate the density maps for steam and air flows for stagnation conditions of 380 kPa and 144°C for steam; and 380 kPa and 25°C for air. The results presented in Figure 8.13 show the difficulty in diagnosing the shape of flow patterns in steam, whereas Figure 8.14 more clearly presents the density-dependent flow patterns in an air flow.

Figure 8.13: Density map of steam at stagnation conditions of 380 kPa and 144°C with a back pressure of 3 kPa.

Figure 8.14: Density map of air at stagnation conditions of 380 kPa and 25°C with a back pressure of 3 kPa.
8.6 Visualisation of ice formation

Although the optical-designed mixing chamber was not used successfully in the schlieren application, it did demonstrate ice forming inside the mixing chamber as shown in Figure 8.15.

Before the perspex mixing chamber was used, it was exposed to post-machining annealing (stress-relief) to minimise stress crazing which could occur during machining. The stress relief procedure recommended by Luc (2005) was followed:

1. Heat up the acrylic from room temperature to 83°C in about 2 hours.

2. Hold the temperature at 83°C for about 1 hour, the time depends on the thickness of the acrylic.

3. Cool down the acrylic at a rate of about 10°C per hour.

Luc (2005) recommends all of these processes to be completed in a nitrogen environment and this was followed in the present work.

Unfortunately, fracture of the optical mixing chamber was initiated in the region where ice was formed as shown in Figure 8.16. It is deduced that this failure is related to the thermal stress in the perspex material.

Figure 8.15: Photograph showing ice forming inside the mixing chamber. Primary stream operating conditions were 270 kPa and 144°C.
8.7 Conclusion

The success of the optical design method was proved. The integral lens design demonstrates a clear view across the full height of the parallel mixing chamber.

Two type of shock waves emerging from the supersonic flow of air from the steam nozzle were visualised using the schlieren technique: oblique shock waves through over-expanded supersonic flow and shock waves created by a pitot tube.

The flow of ambient temperature air through the nozzle is considerably easier to visualize than the steam because density gradients are shown to be about seven times larger in the air flow case. Nevertheless, the successful visualisation of an edge feature of the steam flow was demonstrated using an image analysis process for pictures taken by the schlieren method.

Ice forming inside the mixing chamber was visualised.
Chapter 9

Conclusions and future work

9.1 Summary

Ejectors have many advantages compared with conventional compressors including extremely reliable and stable operation, no moving parts, and maintenance free operation. However, ejectors have very low efficiencies when compared with conventional compressors. Recently, ejectors have been studied by researchers in the air conditioning and refrigeration field with a view to understanding their low coefficients of performance. The present work was motivated by a similar goal and a new apparatus was devised and built to enhance the understanding of the COP and to visualise the flow inside the ejector. Furthermore, new designs for the steam generator and the evaporator chamber were used in this apparatus. A computational fluid dynamics program called Eilmer3 was used to simulate the flow through the axisymmetric geometry of the ejector. A new approach for investigating the mixing region generated by the ejector nozzle was also introduced, and using this method the pitot pressure profile within the flow generated by the ejector nozzle was measured. Schlieren methods were also used in an effort to visualise the steam flow inside the mixing chamber.
9.2 Conclusions

9.2.1 Ejector

The apparatus developed for this study was operated with 3500 W as the maximum cooling load. Primary stream temperatures were 130, 120, and 110°C and the evaporator temperatures were 6, 10, and 14°C. The results showed that:

- With a primary stream pressure of 270 kPa, a high evaporator temperature of 15°C and a condenser pressure lower than the critical condenser pressure of 6.9 kPa, a high COP of 0.52 was obtained at the primary nozzle exit position NXP=-32 mm. The COP was 0.385 at NXP=6mm when operating conditions were a primary stream pressure of 270 kPa and evaporator temperature of 14°C.

- Relative to other steam ejector studies reported in recent literature, the present apparatus has improved the COP by 5.6% and 12.7% at a primary stream temperature of 130°C, and evaporator temperatures of 15 and 10°C, respectively.

- Relative to other steam ejector studies reported in recent literature, the present apparatus has improved the critical condenser pressure by 12.7% at a primary stream pressure of 270 kPa, and by 25.7% at 200 kPa.

9.2.2 Computational simulation

- Eilmer3 simulations using an ideal gas model for the stream have accurately reproduced the ejector wall static pressure and the entrainment ratio of the system under choked operating conditions but some significant differences exist for unchoked operation. The results showed:

  1. CFD simulations of wall static pressure have a good agreement with experimental results for primary stream pressure of 270 kPa, evaporator temperatures of 14, 10, and 6°C, and different condenser pressures.

  2. CFD simulation results were consistent with experimental results at a primary stream pressure of 200 kPa and evaporator temperatures of 14, 10,
Conclusions and future work

and 6°C;

(3) There is a difference between the entrainment ratio in the CFD simulation and experiment results of 2.9% for a primary stream pressure of 270 kPa and an evaporator temperature of 14°C;

(4) A slightly smaller difference exists between the entrainment ratio in the CFD simulation and experiment results of 2.6% for primary stream pressure 200 kPa and evaporator temperature of 14°C; and

(5) The differences in critical condenser pressure between the CFD and experiments were 8.8% at a primary stream pressure of 270 kPa and 15.4% at a primary stream pressure of 200 kPa, when the evaporator temperature was 14°C in both cases.

- Eilmer3 simulations enhance the understanding the flow in the two streams downstream of the nozzle.

(1) CFD simulations reveal a recirculation inside the diffuser at low condenser pressure. Another recirculation in the converging conical section of the mixing chamber was also revealed when there was an increase in the condenser pressure higher than the critical pressure. Totally reversed flow at the secondary inlet section appeared when the condenser pressure increased to more than the breakdown pressure.

(2) CFD simulations demonstrate the occurrence of a first series of shock waves associated with the nozzle flow and a second series of shock waves in the parallel section of the mixing chamber and diffuser.

9.2.3 Mach 4 steam jet

This study investigated the supersonic mixing flow generated downstream of the nozzle exit using a special pitot tube and traversing apparatus which was established for this purpose. Results showed the following:

- From experiments, the shear layer thickness was obtained for four positions of the traversing probe as follows:
(1) 1.7 mm shear layer thickness at location 1 mm from the nozzle exit;
(2) 4 mm shear layer thickness at location 25 mm from the nozzle exit;
(3) 7.1 mm shear layer thickness at location 50 mm from the nozzle exit; and
(4) 12.5 mm shear layer thickness at location 70 mm from the nozzle exit.

- The flow near the nozzle exit was interpreted from the pitot survey station results to be in the form of an over-expanded jet.

- CFD simulations were used to investigate the flow inside the mixing chamber. The comparison between the CFD and experimental results yielded a good agreement at the position 1 mm downstream of the nozzle exit, but not at the other positions. The possible reasons for the differences include:
  (1) The use of an ideal gas assumption in the CFD simulations rather than a real (condensing) gas.
  (2) The use of a hydraulic diameter in axisymmetric simulations instead of the actual rectangular cross section of the mixing chamber.

- A momentum integral analysis was applied to the mixing chamber results. Possible condensation effects were simulated by reducing the value of $\gamma$ used in the analysis but a possible component of momentum transport which was not measured could also explain the results.

### 9.2.4 Flow visualisation

The outer circular cross section of the test section is not appropriate for schlieren visualisation, therefore a new design for two lenses was applied to manage any refraction. The optical design method was shown to be successful and the duct with integral lens design allowed good optical access to the upper and lower edges of the internal diameter of the duct. The optical duct demonstrated the following.

- Liquid water and ice were observed within the optical duct. Ice formed on the conical contraction region of the mixing chamber.
The optical duct was not suitable for schlieren visualisation because of birefringence effects.

Cracks formed in the optical duct most likely because of the thermal stress.

Toepler’s schlieren method was used to visualise the steam flow within the rectangular mixing duct used for the pitot pressure measurements.

- Oblique shock waves generated by air flow through the steam nozzle were successfully visualised with operating conditions: jet stagnation pressure of 380 kPa and background pressure of 3 kPa.

- Successful detection of an edge feature of the steam flow through the nozzle with operating conditions: jet stagnation pressure of 380 kPa and temperature of 144°C, and background pressure of 3 kPa. Detection was achieved using an image analysis process for pictures taken by the schlieren method.

9.3 Future work

- The visualisation of the ejector flow should be further explored by using a laser sheet technique. Seed particles added to the secondary flow or primary flow can be used to distinguish between the secondary and primary flows when a laser light is used to illuminate the particles.

- The measurement of the temperature of the steam along the ejector should be performed using the tuneable diode laser absorption spectroscopy (TDLAS) technique. This technique has not been used for measurement of flow temperature inside a steam ejector at this time. The temperature measurement will provide a more direct indication of the thermodynamic state of the steam at the nozzle exit and within the mixing region downstream.

- Improvement of the schlieren method is possible by increasing the light source power and decreasing the slit size. Lenses which avoid any spherical and chromatic aberration should be used. This will aid the visualisation of the primary and secondary flows.
9.3 Future work

- A pitot tube which can be traversed in two directions inside the mixing chamber would be useful. Additional pitot pressure measurement stations are also needed. Additional pressure transducers along the mixing chamber will also aid in the understanding of the steam flow within the mixing chamber.

- Investigate the flow inside mixing chamber with a three dimensional calculation using the Eilmer3 simulation.
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Appendix A

A.1 Primary nozzle and spacer

Figure A.1: Primary nozzle, all dimensions in mm.
Figure A.2: Nozzle spacer, all dimensions in mm.

Figure A.3: Nozzle holder, all dimensions in mm.
A.2 Plenum chamber

Figure A.4: Plenum chamber, all dimensions in mm.
A.3 Secondary inlet and mixing chamber 1

Figure A.5: Secondary inlet, all dimensions in mm.
Figure A.6: Mixing chamber with parallel section, all dimensions in mm.
Figure A.7: Diffuser dimensions with pressure transducers holes, all dimensions in mm.
A.5 Steam generator and stirrer

Figure A.8: Steam generator and stirrer, all dimensions in mm.
A.6 Evaporator

Figure A.9: Evaporator details, all dimensions in mm.
A.7 Evaporator controller load

Figure A.10: (a) Single phase half controlled rectifier; (b) Evaporator element heater.
Figure A.11: Condenser details, all dimension in mm.
A.9 Electrical arrangement

Figure A.12 shows the front panel and the internal components of the electrical box. Figures A.13 and A.14 show the control diagram and power diagram respectively.

Features include:

- The temperature controller OMRON type E5CN-HQ2m-500 has features suitable for many types of thermocouples B, E, J, K, L, N, output voltage range 10.2 V DC to 13.8 V DC and accuracy 0.1%.

- Solid-state relay (SSR) specification has load current 55 A, operating voltage range 42 V AC to 600 V AC and control voltage range 12 V DC.

- Variable speed control for stirrer output 12 V DC.

- Cut off thermostat with capillary and bulb was used to cut off electrical power on the SSR if the temperature of the thermal oil becomes hotter than the temperature controller setting.

- Three pumps (one dosing pump and two circulation pumps) are used and controlled from the electrical box.

- Evaporator load is controlled by contactor and switch which delivers the power supply to single phase half controlled rectifiers.
Figure A.12: (a) Front panel with switches and temperature controller; (b) Internal side of enclosure.
Figure A.13: Diagram of the control circuit.

Figure A.14: Drawing of the power circuit.
Appendix B

Calculations and other supporting details on system performance.

B.1 Heat loss from primary stream within the stainless steel pipe

Heat losses from the stainless steel pipe between the temperature measurement positions up to the inlet nozzle were calculated as below. Figure 4.2 illustrates all details of the pipe.

\[
Q = Q_1 + Q_2 \quad \text{(B.1)}
\]

\[
Q_1 = \frac{(T_h - T_c)}{(R_1 + R_2 + R_3 + R_4)} \quad \text{at region 1} \quad \text{(B.2)}
\]

\[
Q_2 = \frac{(T_h - T_c)}{(R_1 + R_2 + R_5 + R_6 + R_7)} \quad \text{at region 2} \quad \text{(B.3)}
\]

where,

\[
Q = \text{Heat (W)}
\]

\[
T_h = \text{Primary temperature (steam inlet temperature) = 130°C}
\]

\[
T_c = \text{Ambient temperature = 20°C}
\]
Heat loss from primary stream within the stainless steel pipe

\[ h_{ci} = \text{Inside heat transfer coefficient if the fluid is steam} = 10000 \text{ W/m}^2\text{K} \quad (\text{Kreith and Bohn, 2000}) \]

\[ h_{co} = \text{Outside heat transfer coefficient} = 15 \text{ W/m}^2\text{K} \quad (\text{Kreith and Bohn, 2000}) \]

\[ r_{1,...,5} = \text{Radius of pipe and insulation (m)} \]

\[ k_{ss} = \text{Stainless steel thermal conductivity at 150°C} = 16.2 \text{ W/m.K} \quad (\text{Kreith and Bohn, 2000}) \]

\[ k_{in} = \text{Insulation thermal conductivity} = 0.04 \text{ W/m.K} \quad (\text{Holman, 2002}) \]

\[ k_c = \text{Copper thermal conductivity} = 400 \text{ W/m.K} \quad (\text{Kreith and Bohn, 2000}) \]

\[ k_a = \text{Air thermal conductivity} = 0.024 \text{ W/m.K} \quad (\text{Kreith and Bohn, 2000}) \]

\[ L = \text{Length of stainless steel pipe} = 0.06 \text{ m inside insulation and 0.3 m inside copper tube} \]

\[ R_1 = \text{Convection resistance between steam and inside stainless steel pipe} = 1/(2h_{ci}\pi r_1 L_{1,2}) \]
\[ = 0.053 \text{ K/W} \]

\[ R_2 = \text{Conduction resistance through stainless steel} = 0.0078 \text{ K/W, ln}(r_2/r_1)/(2\pi k_{ss} L) \]

\[ R_3 = \text{Conduction resistance through insulation} = 122 \text{ K/W, ln}(r_3/r_2)/(2\pi k_{in} L) \]

\[ R_4 = \text{Convection resistance between outside insulation and air} = 4.42 \text{ K/W, 1/(2h}_{co}\pi r_3 L) \]

\[ R_5 = \text{Conduction resistance through air between outside stainless steel and inside copper tube} = 11.9 \text{ K/W, ln}(r_4/r_2)/(2\pi k_a L) \]

\[ R_6 = \text{Conduction resistance (copper pipe)} = 0.0002 \text{ K/W, ln}(r_5/r_4)/(2\pi k_c L) \]

\[ R_7 = \text{Convection resistance between outside of copper tube and air} = 4.42 \text{ K/W, 1/(2h}_{co}\pi r_5 L) \]

The total heat losses from the pipe equal 9.432 W
B.1.1 Head losses from primary stream within the stainless steel pipe

The pressure drop through the 10.6 mm inside diameter pipe was calculated from equation B.4. The percentage of pressure drop to stagnation pressure is equal to 0.00014 %

\[ p = \rho g h_f \]  \hspace{1cm} (B.4)
\[ h_f = f(L/d)\left(\frac{V^2}{2g}\right) \]  \hspace{1cm} (B.5)

where,

\[ p = \text{Pressure} = 0.381 \text{ Pa} \]
\[ \rho = \text{Density of steam at 130°C} = 1.496 \text{ kg/m}^3 \]
\[ h_f = \text{Head losses} = 0.026 \text{ m} \]
\[ f = \text{Internal roughness for stainless steel} = 4.6 \times 10^{-5} \text{ m} \]
\[ L = \text{Length of stainless steel pipe} = 0.36 \text{ m} \]
\[ d = \text{Inside diameter of stainless steel pipe} = 0.0106 \text{ m} \]
\[ V = \text{Velocity of steam inside the pipe} = 17.7 \text{ m/s} \]
\[ g = \text{Acceleration due to gravity} = 9.81 \text{ m/s}^2 \]

B.2 Non-electrical heat gain by the evaporator

The evaporator was insulated with 10 mm thick armaflex insulation to keep the low temperature of water between 6 to 14°C within the system as shown in Figure 4.3.

\[ Q = Q_1 + Q_2 \]  \hspace{1cm} (B.6)
\[ Q_1 = \frac{(T_h - T_c)}{(R1 + R2 + R3 + R4)} \text{ at wall} \]  \hspace{1cm} (B.7)
\[ Q_2 = \frac{(T_h - T_c)}{(R7 + R5 + R6 + R7)} \text{ at base} \]  \hspace{1cm} (B.8)

where
B.3 Experimental work procedure

\[ Q = \text{Heat (W)} \]

\[ T_h = \text{Ambient temperature} = 20^\circ\text{C} \]

\[ T_c = \text{Water temperature inside evaporator} = 6 - 14^\circ\text{C} \]

\[ h_{ci} = \text{Inside heat transfer coefficient} = 15 \text{W/m}^2\text{.K} \]

\[ h_{co} = \text{Outside heat transfer coefficient} = 15 \text{W/m}^2\text{.K} \]

\[ r_1, \ldots, 5 = \text{Radius of evaporator and insulation (m)} \]

\[ k_{ga} = \text{Galvanised steel thermal conductivity} = 18 \text{W/m.K} \]

\[ k_{in} = \text{Insulation thermal conductivity (armafflex)} = 0.039 \text{W/m.K} \]

\[ H = \text{Height of evaporator} = 0.27 \text{m} \]

\[ R_1 = \text{Convection resistance between water vapour inside evaporator and galvanised steel} = \frac{1}{(2h_{ci} \pi r_1 H)} = 0.975 \text{K/W} \]

\[ R_2 = \text{Conduction resistance through 1 mm galvanise steel} = 0.000217 \text{K/W}, \ln\left(\frac{r_2}{r_1}\right)/(2\pi k_{ga} H) \]

\[ R_3 = \text{Conduction resistance through insulation} = 0.97 \text{K/W}, \ln\left(\frac{r_3}{r_2}\right)/(2\pi k_{in} H) \]

\[ R_4 = \text{Convection resistance between outside insulation and air} = 0.245 \text{K/W}, \frac{1}{(2h_{co} \pi r_3 H)} \]

\[ R_5 = \text{Conduction resistance through 2 mm galvanise steel (base)} = 0.00157 \text{K/W}, \Delta x/k_{ga} (\pi r_1^2) \]

\[ R_6 = \text{Conduction resistance through 10 mm insulation (base)} = 0.725 \text{K/W}, \Delta x/k_{ga} (\pi r_1^2) \]

\[ R_7 = \text{Convection resistance between outside of insulation (base) and air} = 0.943 \text{K/W}, \frac{1}{(2h_{co} (\pi r_1^2))} \]

B.3 Experimental work procedure

The experimental work was executed in batches because the capacity of condenser is limited. The procedure below was used to run the system.

\textbf{step 1} Switch on the heaters of steam generator and stirrer.
step 2 Set the temperature of oil to 160°C by the temperature control in front panel of electrical box to get temperature of steam 130°C, oil = 150°C to get steam = 120°C, and oil = 140°C to get steam = 110°C.

step 3 Run vacuum pump.

step 4 Run scale inside the evaporator chamber. More details about this procedure can be found in Section B.3.1.

step 5 Fill the evaporator using distilled water to keep the system clean. Details about the procedure is specified in Section B.3.2.

step 6 Run dosing pump.

step 7 Monitor the gauge pressure mounted at top of steam generator until it reaches the desired pressure.

step 8 Open then close immediately the drain valve twice to warm the pipe before pushing the steam to the nozzle ejector.

step 9 Open the main valve to allow the steam to pass through the nozzle.

step 10 Switch on the evaporator load. The loads used during the tests are 1000 W, 1200 W, 2000 W, 2500 W, 3000 W, and 3500 W, depending on NXP of the system.

step 11 Monitor the pressures and temperatures through screen of the computer via the Data Acquisition system until it reaches a steady state. This task takes about 15 minutes.

step 12 After reaching the steady state, start recording the data and mass flow rate during a 10 minute run.

step 13 Change the condenser pressure and repeat step 12.

step 14 Change the primary pressure and repeat step 12.
B.3.1 Evaporator chamber

The weighing method was considered more accurate than other methods that could be implemented with a comparable budget. A special evaporator chamber was designed. It consists of scale, evaporator, flexible hoses, artificial finger and two valves. To run the scale inside the evaporator chamber, the following procedure is followed:

**Step 1** Close valve 1 and 2, as shown in Figure B.1

![Figure B.1: Evaporator chamber.](image)

**Step 2** Run the vacuum pump and monitor the evaporator from the perspex window in front of the evaporator chamber, as shown in Figure B.2
Calculations and other supporting details on system performance.

Step 3 Once the evaporator is lifted higher than the pan of the scale, press the on/off button of the scale through the artificial finger. The artificial finger is designed to run the scale from outside of the evaporator chamber. The scale does not work unless the pan of the scale is clear at switch on. Figure B.3 illustrates the push button of the scale and the space between the evaporator and scale. The display of the scale shows a null reading.
**B.3 Experimental work procedure**

**Step 4** Open valve 2 gradually until the evaporator is reseated on the pan of the scale, as shown in Figure B.4

![Figure B.4: Evaporator chamber.](image)

---

**B.3.2 Fill evaporator with water**

The easiest way to fill the evaporator with water is using the process below. This process can be used even during the system run.

**Step 1** Connect a hose to valve 1 and close valve 2.

**Step 2** Put the hose inside beaker draw up the water, as shown in Figure B.5.

**Step 3** Open valve 1 and monitor the water level. Monitor the scale reading so as not to exceed 15000 g.

**Step 4** Close valve 1 and open valve 2.
B.4 Dosing pump

The dosing pump GRUNDFOS model DDA 17-7 was used in the present study. The dosing pump is suitable for non-abrasive, and non-flammable liquids. The dosing pump works with 240 V, single phase, and 24 W (Grundfos, 2010). Figure B.6 shows photograph of the dosing pump during operation. Some applications of the dosing pump are listed below:

1. Waste water treatment
2. Boiler water treatment
3. Cooling and processing water treatment
4. Chemical industry
Figure B.6: Dosing pump photograph.
Appendix C

Instrument calibration

C.1 Pressure transducers calibration

There are three types of pressure transducers used in the apparatus:

1. Low pressure transducer, type Wika model A-10
2. Low pressure transducer, type Kulite model IS-XTL-190
3. High pressure transducer, type Wika model A-10

All low pressure transducers were calibrated in the laboratory at the University of Southern Queensland, using a pneumatic dead-weight tester Budenberg model 550 series. This worked in conjunction with a special adaptor model 24 vacuum adaptor piston/cylinder unit mounted upside down together with annular weights to generate a negative pressure (vacuum). Figure C.1 illustrates the model 24 adaptor. This tester is suitable for calibration of the two low pressure transducers.

The dead-weight tester Budenberg model 580 series was used to calibrate the high pressure transducer type Wika model A-10 as shown in Figure C.2. Figures C.3, C.4, and C.5 illustrate representative results for the calibration of the three types of pressure transducers. Table C.1 presents the resultant calibration equations for each of the pressure transducers.
Figure C.1: Photograph of the pneumatic dead-weight tester model 550 series (Budenberg).

Figure C.2: Photograph of the hydraulic dead-weight tester model 580 series (Budenberg).
Figure C.3: Calibration of the Kulite pressure transducer - representative result.

Figure C.4: Calibration of the Wika low pressure range transducer - representative result.
Figure C.5: Calibration of the Wika high pressure range transducer - representative results.
Table C.1: Calibration of Kulite and Wika pressure transducers. $V_{\text{meas}}$ = measured voltage, L.P. lower pressure, and H.P. high pressure.

<table>
<thead>
<tr>
<th>Serial Number</th>
<th>Type</th>
<th>Pressure</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Y99-60</td>
<td>Kulite</td>
<td>L.P.</td>
</tr>
<tr>
<td>2</td>
<td>Y99-61</td>
<td>Kulite</td>
<td>L.P.</td>
</tr>
<tr>
<td>3</td>
<td>110263XV</td>
<td>Wika</td>
<td>H.P.</td>
</tr>
<tr>
<td>4</td>
<td>1102HF3B</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>5</td>
<td>1102HF3C</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>6</td>
<td>1102HF3D</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>7</td>
<td>1102HF3E</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>8</td>
<td>1102HF3F</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>9</td>
<td>1102HF3G</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>10</td>
<td>1102HF3H</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>11</td>
<td>1102HF3I</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>12</td>
<td>1102HF3J</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>13</td>
<td>1102HF3K</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>14</td>
<td>1102HF3L</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>15</td>
<td>1102HF3M</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>16</td>
<td>1102HF3N</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>17</td>
<td>1102HF3O</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
<tr>
<td>18</td>
<td>1102HF3P</td>
<td>Wika</td>
<td>L.P.</td>
</tr>
</tbody>
</table>
C.2 Thermocouple calibration

An ice bath with a heater and mercury thermometer was used to calibrate the thermocouples with errors of less than 1°C for all the thermocouples. A typical calibration result is shown in Figure C.6.

![Figure C.6: Calibration of thermocouple - representative result.](image)

C.3 Dosing pump calibration

The GRUNDFOS dosing pump was calibrated locally. The following steps were followed.

**Step 1** A measuring beaker was filled with water. \( V_1 = 635 \text{ ml} \).

**Step 2** The suction hose was inserted in the measuring beaker, as shown in Figure
C.7.

Step 3 Start the calibration process in the “Setup and then Calibration” in the menu as shown in Figure C.8

Figure C.7: Dosing pump with suction hose from beaker.

Figure C.8: Calibration LCD display for calibration on dosing pump.
Step 4 The dosing pump automatically executed 200 dosing strokes and displayed the factory calibration value 312 ml, as shown in Figure C.9.

![Calibration](image)

Figure C.9: Factory calibration in LCD display.

Step 5 The suction hose was removed from the measuring beaker and the remaining volume was measured $V_2 = 312$ ml.

Step 6 From $V_1$ and $V_2$, the actual dosed volume was calculated $V_d = V_1 - V_2$

$V_d = 635 - 312 = 323$ ml.

Step 7 The value $V_d$ was applied in the calibration menu, as shown in Figure C.10. The pump is calibrated.

![Calibration](image)

Figure C.10: Actual dosed volume in LCD display.
Appendix D

Two correcting lens designs

D.1 Lens design

Two optical arrangements were designed and tested in an effort to find a configuration which would correct the refraction effects that arise as rays pass through a clear acrylic (perspex) tube with a circular inner dimension. This appendix describes the design process for these two cases.

D.2 Case one: Separate tube and lens

1. From Snell’s law, the refraction angle inside the perspex (cross section of tube) is calculated. Snell’s law was applied between steam and perspex internal wall of tube, as shown in Figure D.1.

\[ n_s \sin \theta_{i1} = n_p \sin \theta_{r1} \quad (D.1) \]

\[ \theta_{r1} = \sin^{-1} \left( \frac{n_s \sin \theta_{i1}}{n_p} \right) \]

where

\[ n_s = \text{Steam refractive index} = 1.000261 \]

\[ n_p = \text{Perspex refractive index} = 1.49 \]

\[ \theta_i = \text{Incident angle} \]
Figure D.1: Light ray passing through perspex tube and perspex lens.

$$\theta_r = \text{Refraction angle}$$

2. $\theta_{i2}$ is calculated from the sine rule

$$\frac{\sin \theta_{i2}}{r_1} = \frac{\sin \beta}{r_2}$$  \hspace{1cm} (D.2)
3. Snell’s law between the outer surface of the perspex tube and the air is applied.

\[ n_p \sin \theta_2 = n_a \sin \theta_{r2} \quad (D.3) \]

\[ \theta_{r2} = \sin^{-1} \left( \frac{n_p \sin \theta_2}{n_a} \right) \]

4. \( \alpha_2 \) can be calculated,

\[ \phi_4 + \alpha_1 + \pi/2 - \phi = \pi \quad (D.4) \]

\[ \pi/2 - \phi + \alpha_2 = \pi/2 \quad (D.5) \]

\[ \phi = \alpha_2 \]

which can be substituted into equation D.4

\[ \phi_4 + \alpha_1 + \pi/2 - \alpha_2 = \pi \quad (D.7) \]

\[ \alpha_2 = \alpha_1 - (\pi/2 - \phi_4) \quad (D.8) \]

let \( \pi/2 - \phi_4 = aa \)

\[ n_a \sin \alpha_1 = n_p \sin \alpha_2 \]

\[ n_a \sin \alpha_1 = n_p \sin(\alpha_1 - aa) \quad (D.10) \]

\[ n_a \sin \alpha_1 = n_p(\sin \alpha_1 \cos aa - \cos \alpha_1 \sin aa) \quad (D.12) \]

From subtraction trigonometry formulas

\[ n_a \sin \alpha_1 = n_p(\sin \alpha_1 \cos aa - \cos \alpha_1 \sin aa) \quad (D.12) \]

Dividing equation D.10 by \( \cos \alpha_1 \)

\[ \alpha_1 = \tan^{-1} \left( \frac{n_p \sin aa}{n_p \cos aa - n_a} \right) \quad (D.13) \]

\[ \theta_4 = \theta_3 + \theta_{r2} \quad (D.14) \]

\[ \theta_2 = 180 - (\beta + \theta_{l2}) \quad (D.15) \]

\[ \theta_3 = \theta + \theta_2 \quad (D.16) \]

\[ \alpha_2 = \alpha_1 - (90 - \theta_4) \quad (D.17) \]

where

\[ n_a = \text{Air refractive index} = 1.00 \]
5. The coordinates $x$ and $y$ of each point on the outer surface of the separate lenses are calculated. The slope of the line is dependent on the values of the coordinates of the point as shown in Figure D.2, where $h$ the distance between lens and tube.

\[ x_{c2} = r_2 \cos \theta \]
\[ y_{c2} = r_2 \sin \theta \]

- \[ \tan \alpha_2 = \frac{y_{l2} - y_{l1}}{x_{l2} - x_{l1}} \quad \text{(D.18)} \]

\[ y_{l2} = y_{c2} - x_{c2} \tan \theta_2 + x_{l2} \tan \theta_4 \quad \text{(D.19)} \]

From equations D.18 and D.19 the coordinates ($x$ and $y$) of all points on lens surface can be deduced.

\[ x_{l2} = \frac{y_{l1} + x_{l1} \tan \alpha_2 - y_{c2} + x_{c2} \tan \theta_4}{\tan \alpha_2 + \tan \theta_4} \quad \text{(D.20)} \]
\[ y_{l2} = (x_{l2} - x_{c2}) \tan \theta_4 + y_{c2} \quad \text{(D.21)} \]

Finally, the shape of the lenses are defined. The manufactured result is shown in Figures D.3(a) and D.3(b).
Figure D.3: (a) Photograph of the separate pipe with two lenses; (b) Photograph of the separate pipe with two lenses showing visualisation across the full height of the pipe.
D.3 Case two: Integral tube and lens

1. A light ray within the tube which is incident on the inner surface must emerge from the outer surface of the lens parallel to the initial ray as illustrated in Figure D.4. With reference to Figure D.4, Snell’s law can be applied.

\[ n_s \sin \theta_i = n_p \sin \theta_r \]

\[ \theta_r = \sin^{-1}\left(\frac{n_s \sin \theta_i}{n_p}\right) \]

Figure D.4: Light ray passing through the integral lens: definition angles.
The geometry gives the following results

\[ \phi + \phi_4 + \pi/2 - \alpha_1 = \pi \]  
\[ \phi + \pi/2 - \alpha_2 = \pi/2 \]  
\[ \phi = \alpha_2 \]  

which can be substituted into equation D.23

\[ \alpha_2 + \phi_4 + \pi/2 - \alpha_1 = \pi \]  
\[ \alpha_2 = (\pi/2 - \phi_4) + \alpha_1 \]
\[ \pi/2 - \phi_4 = aa \]
\[ \alpha_2 = (aa + \alpha_1) \]
\[ n_p \sin \alpha_1 = n_a \sin \alpha_2 \]  
\[ n_p \sin \alpha_1 = n_a \sin(aa + \alpha_1) \]
\[ n_p \sin \alpha_1 = n_a(\sin aa \cos \alpha_1 + \cos aa \sin \alpha_1) \]  

dividing equation D.26 by \( \cos \alpha_1 \)

\[ n_p \tan \alpha_1 - n_a \cos aa \tan \alpha_1 = n_a \sin aa \]
\[ \alpha_1 = \tan^{-1} \left( \frac{n_a \sin aa}{n_p - n_a \cos aa} \right) \]

2. The coordinates \( x \) and \( y \) of each point on the outer surface are now calculated.

The slope of the line is dependent on the values of the coordinates of the point as shown in Figure D.5.

\[ - \tan \alpha_2 = \frac{y_{l2} - y_{l1}}{x_{l2} - x_{l1}} \]
\[ y_{l2} = y_{l1} - x_{l2} \tan \alpha_2 + x_{l1} \tan \alpha_2 \]
\[ \tan \theta_4 = \frac{y_{l2} - y_{c2}}{x_{l2} - x_{c2}} \]
\[ y_{l2} = y_{c2} - x_{l2} \tan \theta_4 + x_{c2} \tan \theta_4 \]
from equation D.31 and D.33

\[ x_{l2} = \frac{y_{l1} + x_{l1} \tan \alpha_2 - y_{c2} + x_{c2} \tan \theta_4}{\tan \alpha_2 + \tan \theta_4} \]  \hspace{1cm} (D.34)

\[ y_{l2} = (x_{l2} - x_{c2}) \tan \theta_4 + y_{c2} \]  \hspace{1cm} (D.35)

where

\[ x_{c2} = r_i \cos \theta \]

\[ y_{c2} = r_i \sin \theta \]

Finally, the shape of the lens can be defined as shown in Figure D.6
Two types of the lenses were fabricated in workshop at University of Southern Queensland. All lenses were polished by using NOUVS plastic polish. This polish consists of 3 stages: a heavy scratch remover, a fine scratch remover, and a plastic clean and shine.
Appendix E

Eilmer3 scripts for steam ejector.

E.1 Python.py file

# Actual ejector_test.py
# 11 December 2011 with low initial pressure 500 kpa
# Water as a working fluid.
# Input SubsonicInBC(inflow)
# Output is ExtrapolateOutBC()

gdata.title = "Flow through an ejector."
print gdata.title

# Parameters that we are likely to change
m = 3; n = 3 # cell refinement control

nx0 = int(23*m); nx1 = int(12*m); nx3 = int(33*m); nx4 = int(45*m);
xn7 = int(86*m); nx10 = int(41*m); nx13 = int(116*m); nx15 = int(20*m)
ny0 = int(4*n); ny6 = int(2*n); ny1 = int(8*n); ny2 = int(3*n)

# Define x locations of each pressure transducer
# (starting at transducer 1)
xt1 = 0.0074
xt2 = 0.0729
xt3 = 0.10434
xt4 = 0.12636
xt5 = 0.14929
xt6 = 0.17326
xt7 = 0.19822
xt8 = 0.22313
xt9 = 0.24822
xt10 = 0.2765
xt11 = 0.3065
xt12 = 0.35191
xt13 = 0.39365
xt14 = 0.43578
xt15 = 0.47761
xt16 = 0.51946

# Accept defaults for water - treated as ideal gas for now
select_gas_model(model='ideal gas', species=['H2O'])

#==================specify the boundary conditions==============

#--------initial flow--------
p_ini= 500.0
u_ini= 10.0
v_ini= 0.0
T_ini= 283.0

Itub_ini= 0.002  #turbulence intensity for initial flow
tur_lam_ratio_ini = 1.0 #(Mt/ML)turbulent-to-laminar
#viscosity ratio for initial flow
rho_ini = p_ini / (461.5 * T_ini)  # Uses R from steam

#--------secondary flow--------
p_sec= 1950.0
u_sec= 20.0
T_sec= 286.8

Itub_sec= 0.02  #turbulence intensity for secondary flow
tur_lam_ratio_sec = 1000.0 #(Mt/ML)turbulent-to-laminar
#viscosity ratio secondary flow
rho_sec = p_sec / (461.5 * T_sec)  # Uses R from steam

#--------Primary flow--------
p_pri = 3.12e5
u_pri = 30.0
T_pri = 418.0
Itub_pri = 0.01  #turbulence intensity for primary flow
tur_lam_ratio_pri = 1000.0  #M/M1 turbulent-to-laminar
# viscosity ratio primary flow
rho_pri = p_pri/(461.5 * T_pri)  # Uses R from steam

# Sutherland’s viscosity law
S= 1064.0 ; T_ref= 350.0; mu_ref=1.12e-5  #steam sutherland’s
# constant from visocus fluid flow pp29 White(1991)

#=Estimate turbulence intensity and turbulent-to-laminar viscosity ratio====
# firstly initial flow
Sutherland_mu_ini= mu_ref * (T_ini / T_ref)**1.5 * (T_ref + S)/(T_ini + S)
#secondary Sutherland’s viscosity
# turbulent kinetic energy law = 1.5 *(Itub *u)**2  where
#:Itub= turbulence intensity & u = velocity
tke_ini = 1.5 * (Itub_ini * u_ini)**2  # turbulent kinetic energy for primary flow
omega_ini = rho_ini * tke_ini /(tur_lam_ratio_ini * Sutherland_mu_ini)

# secondary secondary flow
Sutherland_mu_sec= mu_ref * (T_sec / T_ref)**1.5 * (T_ref + S)/(T_sec + S)
#secondary Sutherland’s viscosity
# turbulent kinetic energy law = 1.5 *(Itub *u)**2  where
#:Itub= turbulence intensity & u = velocity
tke_sec = 1.5 * (Itub_sec * u_sec)**2  #turbulent kinetic energy for secondary flow
omega_sec = rho_sec * tke_sec /(tur_lam_ratio_sec * Sutherland_mu_sec)

# finally primary flow
Sutherland_mu_pri= mu_ref * (T_pri / T_ref)**1.5 * (T_ref + S)/(T_pri + S)
# primary Sutherland’s viscosity
# turbulent kinetic energy law = 1.5 *(Itub *u)**2  where
#:Itub= turbulence intensity & u = velocity
tke_pri = 1.5 * (Itub_pri * u_pri)**2  # turbulent kinetic energy for primary flow
omega_pri = rho_pri * tke_pri / (tur_lam_ratio_pri * Sutherland_mu_pri)

print "XXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXXX"
print "initial Sutherland viscousity = ",Sutherland_mu_ini
print "secondary Sutherland viscousity = ",Sutherland_mu_sec
print "primary Sutherland viscousity = ",Sutherland_mu_pri
print "density of initial flow",rho_ini
print "density of secondary flow",rho_sec
print "density of primary flow",rho_pri
print "turbulent kinetic energy for initial flow",tke_ini
print "turbulent kinetic energy for secondary flow",tke_sec
print "turbulent kinetic energy for primary flow",tke_pri
print "initial omega = ",omega_ini
print "secondary omega = ",omega_sec
print "primary omega = ",omega_pri

#============Initial and inflow conditions=============  #initial = ExistingSolution('ejegeometry', '.', 23,9999)
initial=FlowCondition(p=p_ini,u=u_ini,v=v_ini,T=T_ini, tke=tke_ini,omega=omega_ini)
# initial conditions
inflow_sec = FlowCondition(p=p_sec, T=T_sec,tke= tke_sec, omega=omega_sec)
# secondary conditions
inflow_pri = FlowCondition(p=p_pri, T=T_pri,tke= tke_pri, omega=omega_pri)
#primary conditions

# The following Nodes will be rendered in the SVG file.

# Node group A
a0 = Node(0.0, 0.0)
a1 = Node(0.042, 0.0)
a2 = Node(0.1015, 0.0)
a3 = Node(0.2565, 0.0)
a4 = Node(0.3315, 0.0)
a5 = Node(0.5415, 0.0) # end the ejector
a6 = Node(0.600, 0.0)
a7 = Node(0.650, 0.0)
a8 = Node(0.675, 0.0)
a9 = Node(0.700, 0.0)

# Nodes group B
b0 = Node( 0.0000000, 0.005000,label="B0")
b1 = Node(0.03855000, 0.005000,label="B1") # First point
b2 = Node(0.04020000, 0.0035000,label="B2")# Second point
centre_bc1= Node(0.03855000, 0.00335000, label="centre_B1")# Centre curve1
b3= Node(0.042000, 0.0016000) # Second point (Throat of the primary nozzle)
centre_bc2= Node(0.04200052, 0.00340124, label="centre_B2")# Centre curve2
b4 = Node(0.1015, 0.00675)
b5 = Node(0.2565, 0.0058)
b6 = Node(0.3315, 0.0058)
b7 = Node(0.5415, 0.0065)
b8 = Node(0.600, 0.0062)
b9 = Node(0.650, 0.0062)
b10 = Node(0.675, 0.0060)
b11 = Node(0.700, 0.0070)

# Nodes group C
c0 = Node(0.0, 0.010)
c0a = Node(0.01476000, 0.010)
c1 = Node(0.045800, 0.00981479)
c2 = Node(0.1015, 0.0071)
c3 = Node(0.2565, 0.0075)
c4 = Node(0.3315, 0.008)
c5 = Node(0.5415, 0.009)
c6 = Node(0.600, 0.0087)
c7 = Node(0.650, 0.0087)
c8 = Node(0.675, 0.0075)
c9 = Node(0.700, 0.0090)

# Node group D
d2x = 0.0268000
d3x = 0.0458000
d4x = 0.1015
d5x = 0.2565
d6x = 0.3315
d7x = 0.5415
d0 = Node(0.000000, 0.0446000)
d2 = Node(d2x, 0.0446000) # First point
d3 = Node(d3x, 0.0282500) # Second point
centre_d1= Node(0.05057707, 0.05301610, label="centre_D1")# Center curve1
d4 = Node(d4x, 0.0185)
d5 = Node(d5x, 0.0127)
d6 = Node(d6x, 0.0127)
d7 = Node(d7x, 0.025)
d8 = Node(0.600, 0.025)
d9 = Node(0.650, 0.025)
d10 = Node(0.675, 0.0115)  # throat of the system
d11 = Node(0.700, 0.016)

# Nodes group E
e0x = 0.000000
e1x = 0.0268000
e0 = Node(e0x, 0.0480000)
e1 = Node(e1x, 0.0480000)

# Block-0
north0 = Polyline([Line(b0, b1), Arc(b1, b2, centre_bc1), Arc(b2, b3, centre_bc2)])
east0 = Line(a1, b3)
south0 = Line(a0, a1)
west0 = Line(a0, b0)

# Block-1
east1 = Line(c0a, d2)
south1 = Line(c0, c0a)
west1 = Line(c0, d0)

# Block-2
north2 = Line(e0, e1)
east2 = Line(d2, e1)
south2 = Line(d0, d2)
west2 = Line(d0, e0)

# identify history cells in block 2
dcelldx2 = nx1/(e1x-e0x)  # number of cells per unit length in block 2
hn_2_t1 = int(dcelldx2*(xt1-e0x))

# Block-3
north3 = Line(b3, b4)
east3 = Line(a2, b4)
south3 = Line(a1, a2)

# Block-4
north4 = Polyline([Arc(d2, d3, centre_d1), Line(d3, d4)])
east4 = Line(c2, d4)
south4 = Polyline([Line(c0a, c1), Line (c1, c2)])

# identify history cells in block 4
dcelldx4 = nx4/(d4x-d2x) # number of cells per unit length in block 4
hn_4_t2 = int(dcelldx4*(xt2-d2x))

# Block-5
east5 = Line(a3, b5)
south5 = Line(a2, a3)

# Block-6
east6 = Line(b5, c3)
south6 = Line(b4, b5)
west6 = Line(b4, c2)

# Block-7
north7 = Line( d4, d5)
east7 = Line( c3, d5)
south7 = Line(c2, c3)

# identify history cells in block 7
dcelldx7 = nx7/(d5x-d4x) # number of cells per unit length in block 7
hn_7_t3 = int(dcelldx7*(xt3-d4x))
hn_7_t4 = int(dcelldx7*(xt4-d4x))
hn_7_t5 = int(dcelldx7*(xt5-d4x))
hn_7_t6 = int(dcelldx7*(xt6-d4x))
hn_7_t7 = int(dcelldx7*(xt7-d4x))
hn_7_t8 = int(dcelldx7*(xt8-d4x))
hn_7_t9 = int(dcelldx7*(xt9-d4x))

# Block-8
east8 = Line(a4,b6)
south8 =Line(a3,a4)

# Block-9
east9 = Line(b6, c4)
south9 = Line(b5, b6)

# Block-10
north10 = Line(d5, d6)
east10 = Line(c4, d6)
south10 = Line(c3, c4)

# identify history cells in block 10
dcelldx10 = nx10/(d6x-d5x) # number of cells per unit length in block 10
hn_10_t10 = int(dcelldx10*(xt10-d5x))
hn_10_t11 = int(dcelldx10*(xt11-d5x))

# Block 11
east11 = Line(a5, b7)
south11 = Line(a4, a5)

# Block 12
east12 = Line(b7, c5)
south12 = Line(b6, b7)

# Block 13
north13 = Line(d6, d7)
east13 = Line(c5,d7)
south13 = Line(c4, c5)

# identify history cells in block 13
dcelldx13 = nx13/(d7x-d6x) # number of cells per unit length in block 13
hn_13_t12 = int(dcelldx13*(xt12-d6x))
hn_13_t13 = int(dcelldx13*(xt13-d6x))
hn_13_t14 = int(dcelldx13*(xt14-d6x))
hn_13_t15 = int(dcelldx13*(xt15-d6x))
hn_13_t16 = int(dcelldx13*(xt16-d6x))

# Block 14
east14 = Line(a6, b8)
south14 = Line(a5, a6)

# Block 15
east15 = Line(b8,c6)
south15 = Line(b7, b8)

# Block 16
north16 = Line(d7,d8)
east16 = Line(c6, d8)
south16=Line(c5, c6)

# Block-17
east17 = Line(a7, b9)
south17 = Line(a6, a7)

# Block 18
east18 = Line(b9, c7)
south18 = Line(b8, b9)

# Block 19
north19 = Line(d8, d9)
east19 = Line(c7, d9)
south19 = Line(c6, c7)

#Block20
east20 = Line(a9, b11)
south20 = Polyline([Line(a7, a8), Line(a8, a9)])

#Block21
east21 = Line(b11, c9)
south21 = Arc3(b9, b10, b11)

#Block22
north22 = Arc3(d9, d10, d11)
east22 = Line(c9, d11)
south22 = Arc3(c7, c8, c9)

print "cell number for transducer 1 = ", hn_2_t1
print "cell number for transducer 2 = ", hn_4_t2
print "cell number for transducer 3 = ", hn_7_t3
print "cell number for transducer 4 = ", hn_7_t4
print "cell number for transducer 5 = ", hn_7_t5
print "cell number for transducer 6 = ", hn_7_t6
print "cell number for transducer 7 = ", hn_7_t7
print "cell number for transducer 8 = ", hn_7_t8
print "cell number for transducer 9 = ", hn_7_t9
print "cell number for transducer 10 = ", hn_10_t10
print "cell number for transducer 11 = ", hn_10_t11
print "cell number for transducer 12 = ", hn_13_t12
print "cell number for transducer 13 = ", hn_13_t13
print "cell number for transducer 14 = ", hn_13_t14
print "cell number for transducer 15 = ", hn_13_t15
print "cell number for transducer 16 = ", hn_13_t16

# Define the blocks, boundary conditions and set the discretisation.

# Inlet Nozzle (primary flow)
blk0 = Block2D(make_patch(north0, east0, south0, west0),
nni=nx0, nnj=ny0,
fill_condition=initial,
label="block0",
cf_list=[RobertsClusterFunction(1, 1, 1.08),
    RobertsClusterFunction(0, 1, 1.04),
    RobertsClusterFunction(1, 1, 1.08),
    RobertsClusterFunction(0, 1, 1.04)],
bc_list=[FixedTBC(308), SlipWallBC(),
    SlipWallBC(), SubsonicInBC(inflow_pri)],
hcell_list=[(1,1)])

# Plenum chamber (secondary flow)
blk1 = Block2D(make_patch(south2, east1, south1, west1),
nni=nx1, nnj=ny1,
fill_condition=initial,
label="block1",
cf_list=[RobertsClusterFunction(1, 0, 1.08),
    RobertsClusterFunction(1, 1, 1.04),
    RobertsClusterFunction(1, 0, 1.08),
    RobertsClusterFunction(1, 1, 1.04)],
bc_list=[SlipWallBC(), SlipWallBC(),
    FixedTBC(308), SubsonicInBC(inflow_sec)])

# Plenum chamber (secondary flow)
blk2 = Block2D(make_patch(north2, east2, south2, west2),
nni=nx1, nnj=ny2,
fill_condition=initial,
label="block2",
cf_list=[RobertsClusterFunction(1, 0, 1.08),
    RobertsClusterFunction(1, 1, 1.04),
    RobertsClusterFunction(1, 0, 1.08),
    RobertsClusterFunction(1, 1, 1.04)],
bc_list=[SlipWallBC(), SlipWallBC(),
    FixedTBC(308), SubsonicInBC(inflow_sec)]]
RobertsClusterFunction(1, 1, 1.04),
RobertsClusterFunction(1, 0, 1.08),
RobertsClusterFunction(1, 1, 1.04)],
bc_list=[FixedTBC(308), FixedTBC(308),
SlipWallBC(), SubsonicInBC(inflow_sec)],
hcell_list=[(hn_2_t1, ny2-1)])

# Outlet nozzle
blk3 = Block2D(make_patch(north3, east3, south3, east0),
ni=nx3, nj=ny0,
fill_condition=initial,
label="block3",
cf_list=[RobertsClusterFunction(1, 1, 1.08),
RobertsClusterFunction(0, 1, 1.04),
RobertsClusterFunction(1, 1, 1.08),
RobertsClusterFunction(0, 1, 1.04)],
bc_list=[FixedTBC(308), SlipWallBC(),
SlipWallBC(), SlipWallBC()],
hcell_list=[(1, ny0)])

# Mixing chamber1
blk4 = Block2D(make_patch(north4, east4, south4, east1),
ni=nx4, nj=ny1,
fill_condition=initial,
label="block4",
cf_list=[RobertsClusterFunction(0, 1, 1.08),
RobertsClusterFunction(1, 1, 1.04),
RobertsClusterFunction(0, 1, 1.08),
RobertsClusterFunction(1, 1, 1.04)],
bc_list=[FixedTBC(308), SlipWallBC(),
FixedTBC(308), SlipWallBC()],
hcell_list=[(hn_4_t2, ny1-1)])

# Mixing chamber2
blk5 = Block2D(make_patch(south6, east5, south5, east3),
ni=nx7, nj=ny0,
fill_condition=initial,
label="block5",
cf_list=[RobertsClusterFunction(1, 0, 1.08),
None,
Eilmer3 scripts for steam ejector.

```python
RobertsClusterFunction(1, 0, 1.08),
RobertsClusterFunction(0, 1, 1.04]),
bclist=[SlipWallBC(), SlipWallBC(),
SlipWallBC(), SlipWallBC()])

blk6 = Block2D(make_patch(south7, east6, south6, west6),
nxi=nx7, nny=ny6,
fill_condition=initial,
label="block6",
cf_list=[RobertsClusterFunction(1, 0, 1.08),
None,
RobertsClusterFunction(1, 0, 1.08),
RobertsClusterFunction(1, 1, 1.04]),
bclist=[SlipWallBC(), SlipWallBC(),
SlipWallBC(), FixedTBC(308)])

blk7 = Block2D(make_patch(north7, east7, south7, east4),
nxi=nx7, nny=ny1,
fill_condition=initial,
label="block7",
cf_list=[RobertsClusterFunction(1, 0, 1.08),
RobertsClusterFunction(0, 1, 1.04),
RobertsClusterFunction(1, 0, 1.08),
RobertsClusterFunction(1, 1, 1.04]),
bclist=[FixedTBC(308), SlipWallBC(),
SlipWallBC(), SlipWallBC()],
hcell_list=[(hn_7_t3,ny1-1),(hn_7_t4,ny1-1),(hn_7_t5,ny1-1),
(hn_7_t6,ny1-1),(hn_7_t7,ny1-1),(hn_7_t8,ny1-1),
(hn_7_t9,ny1-1)])

# Group Constant area region
blk8 = Block2D(make_patch(south9, east8, south8, east5),
nxi=nx10, nny=ny0,
fill_condition=initial,
label="block8",
cf_list=[None,
None,
None,
None],
bclist=[SlipWallBC(), SlipWallBC(),
```
SlipWallBC(), SlipWallBC()

hcell_list=[(1,1)]

blk9 = Block2D(make_patch(south10, east9, south9, east6),
nni=nx10, nnj=ny6,
fill_condition=initial,
label="block9",

cf_list=[None,
None,
None,
None],

bc_list=[SlipWallBC(), SlipWallBC(),
SlipWallBC(), SlipWallBC()],
	hcell_list=[(1,1)]

blk10 = Block2D(make_patch(north10, east10, south10, east7),
nni=nx10, nnj=ny1,
fill_condition=initial,
label="block10",

cf_list=[None,
RobertsClusterFunction(0, 1, 1.04),
None,
RobertsClusterFunction(0, 1, 1.04)],

bc_list=[FixedTBC(308), SlipWallBC(),
SlipWallBC(), SlipWallBC()],
	hcell_list=[(hn_10_t10,ny1-1),(hn_10_t11,ny1-1)]

# Group Expansion region (Diffuser region)

blk11 = Block2D(make_patch(south12, east11, south11, east8),
nni=nx13, nnj=ny0,
fill_condition=initial,

cf_list=[None,None,None,None],

bc_list=[SlipWallBC(), SlipWallBC(),
SlipWallBC(), SlipWallBC()],

label="block11")

blk12 = Block2D(make_patch(south13, east12, south12, east9),
nni=nx13, nnj=ny6,
fill_condition=initial,

cf_list=[None,None,None,None],

bc_list=[SlipWallBC(), SlipWallBC(), SlipWallBC(), SlipWallBC()],
label="block12"
)

blk13 = Block2D(make_patch(north13, east13, south13, east10),
            nni=nx13, nnj=ny1,
            fill_condition=initial,
            cf_list=[None,
                     RobertsClusterFunction(0, 1, 1.04),
                     None,
                     RobertsClusterFunction(0, 1, 1.04)],
            bc_list=[FixedTBC(308), SlipWallBC(), SlipWallBC(), SlipWallBC()],
            hcell_list=[(hn_13_t12,ny1-1),(hn_13_t13,ny1-1),(hn_13_t14,ny1-1),
                         (hn_13_t15,ny1-1),(hn_13_t16,ny1-1)],
            label="block13")

# Group Spacer region (for control flow)

blk14 = Block2D(make_patch(south15, east14, south14, east11),
               nni=nx15, nnj=ny0,
               fill_condition=initial,
               bc_list=[SlipWallBC(), SlipWallBC(), SlipWallBC(), SlipWallBC()],
               label="block14")

blk15 = Block2D(make_patch(south16, east15, south15, east12),
               nni=nx15, nnj=ny6,
               fill_condition=initial,
               bc_list=[SlipWallBC(), SlipWallBC(), SlipWallBC(), SlipWallBC()],
               label="block15")

blk16 = Block2D(make_patch(north16, east16, south16, east13),
               nni=nx15, nnj=ny1,
               fill_condition=initial,
               cf_list=[None,
                        None,
                        None,
                        RobertsClusterFunction(0, 1, 1.04)],
bc_list=[FixedTBC(308), SlipWallBC(), SlipWallBC(), SlipWallBC()],
label="block16")

# Group reducer
blk17 = Block2D(make_patch(south18, east17, south17, east14),
nni=nx15, nnj=ny0,
fill_condition=initial,
bc_list=[SlipWallBC(), SlipWallBC(), SlipWallBC(), SlipWallBC()],
label="block17")

blk18 = Block2D(make_patch(south19, east18, south18, east15),
nni=nx15, nnj=ny6,
fill_condition=initial,
bc_list=[SlipWallBC(), SlipWallBC(), SlipWallBC(), SlipWallBC()],
label="block18")

blk19 = Block2D(make_patch(north19, east19, south19, east16),
nni=nx15, nnj=ny1,
fill_condition=initial,
bc_list=[FixedTBC(308), SlipWallBC(), SlipWallBC(), SlipWallBC()],
label="block19")

# Group Throoting system region to controll pressure inside the ejector
blk20 = Block2D(make_patch(south21, east20, south20, east17),
nni=nx15, nnj=ny0,
fill_condition=initial,
bc_list=[SlipWallBC(), ExtrapolateOutBC(), SlipWallBC(), SlipWallBC()],
label="block20")

blk21 = Block2D(make_patch(south22, east21, south21, east18),
nni=nx15, nnj=ny6,
fill_condition=initial,
bc_list=[SlipWallBC(), ExtrapolateOutBC(), SlipWallBC(), SlipWallBC()],
label="block21")
blk22 = Block2D(make_patch(north22, east22, south22, east19),
               nni=nx15, nnj=ny1,
               fill_condition=initial,
               bc_list=[FixedTBC(308), ExtrapolateOutBC(),
                        SlipWallBC(), SlipWallBC()],
               label="block22")

identify_block_connections()
# Do a little more setting of global data.
gdata.dimensions=2
gdata.axisymmetric_flag = 1
gdata.viscous_flag = 1
gdata.turbulence_flag = 1
gdata.turbulence_model = "k_omega"
gdata.flux_calc = ADAPTIVE
gdata.max_time = 10.0e-3 # seconds
gdata.max_step = 7000000
gdata.cfl= 0.4
gdata.cfl_count=3
gdata.dt = 1.0e-12
gdata.dt_plot = 0.1e-3
gdata.dt_history = 1.0e-4
sketch.xaxis(-0.01, 0.8, 0.1, -0.005)
sketch.yaxis( 0.0, 0.04, 0.005, -0.015)
sketch.window(-0.011, -0.006, 0.8, 0.2, 0.02, 0.02, 0.2, 0.7)

## E.2 Shell Scripts

#!/bin/sh
# SteamEjector_run.sh
# Exercise the Navier-Stokes solver for the SteamEjector
e3prep.py --job=SteamEjector --do-svg
time e3shared.exe --job=SteamEjector --run
e3post.py --job=SteamEjector --tindx=all --vtk-xml
#!/bin/bash -l
#PBS -S /bin/bash
#PBS -N ejector
#PBS -q workq
#PBS -l select=3:ncpus=8:NodeType=medium:mpiprocs=8 -A q1087
#PBS -l walltime=100:00:00
echo "---------------------------------------------"

echo "Begin MPI job..."
date
cd $PBS_O_WORKDIR
mpirun -np 23 $HOME/e3bin/e3mpi.exe --job=
SteamEjector --run --max-wall-clock=2000000 > LOGFILE

echo"End MPI job."
date