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Study on the Efficiency of a Convergent-Divergent Two-Phase Nozzle as a Motive Force for Power Generation from Low Temperature Geothermal Resources

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ABSTRACT

Two-phase nozzles could be used as energy conversion devices in geothermal total flow systems or binary fluid systems followed by a trilateral cycle for power generation. In this paper, the efficiency of such nozzles is investigated. Also, a profound research has been done on similar area in the past where mostly high pressure and high temperature energy resources were considered; so, the possibility of utilizing low temperature energy resources remains limited in the literature. In order to bridge the knowledge gap, the feasibility of utilizing low temperature resources for power generation is studied in this paper.

In this regards, experiments are carried out with the following conditions: a convergent-divergent nozzle is supplied with water at atmospheric pressure with various temperatures at / below 100ºC. This nozzle is connected to a tank that is evacuated by a vacuum pump. The driving force for water to flow through the nozzle is the pressure difference between atmosphere and vacuum pressure in the flash tank. As water is passed through the nozzle, the thermal energy is converted to kinetic energy as a motive force for power generation. The impulse force caused by the jet exiting the nozzle is measured and compared against the ideal case (i.e. isentropic expansion assumption) to calculate the thrust coefficient of the nozzle and evaluate the efficiency of the process. Also, the pressure and temperature profiles along the nozzle are obtained and compared against saturation pressure corresponding to measured temperatures. The results encourage the utilization of low temperature geothermal energy resources for power generation.

1. INTRODUCTION

Geothermal energy has been utilized for power generation through three conversion systems; the flashed steam system, the binary cycle system and the total flow system Austin and Lundberg (1979). A schematic of these energy conversion systems is shown in Fig. 1. In the flashed steam system, high temperature geothermal water is introduced directly to a separator where separation of vapor and liquid occurs. The extracted brine is injected back into the field and the vapor is passed through a turbo-generator for power generation. This system cannot work with high salinity water (>3% dissolved solids) since scaling of turbine components, corrosion and erosion could be caused by carryover of salts with the vapor. Since in a single flash system, at best, only approximately 10% of the thermal energy can be converted, the overall thermal efficiency is low.

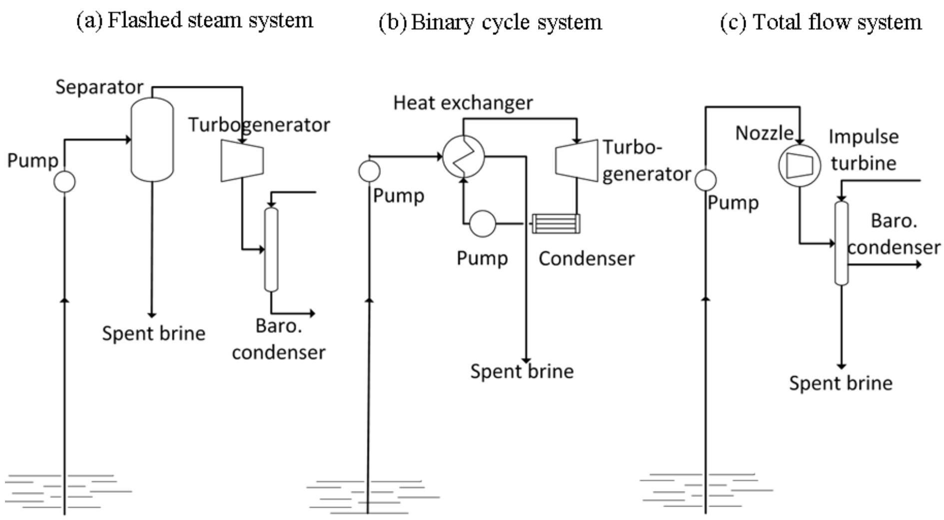

Figure 1- Schematic of (a) Flashed steam system (b) Binary cycle system (c) Total flow system

In order to utilize brines at relatively low temperatures, or brines containing large amounts of non-condensable gases or dissolved solids, the binary cycle system was introduced for conversion of hot water geothermal energy. In this method, a heat- exchanger is required to transfer the internal energy of the brine to a clean secondary fluid. The advantage is that the turbine is protected from the brine by using the secondary fluid. However there will still be scale formation in the brine side of the heat exchanger. Also, in order to optimize energy transfer per unit of exchanger area, the brine outlet temperature must be high. This results in a relatively low overall thermal efficiency as is the case for the flashed steam system.

In the total flow system, the entire wellhead product (liquid or liquid vapor mixture) is fed directly into an impulse or reaction turbine (a schematic of impulse and reaction turbines is shown in Fig. 2). This involves expansion of the fluid through converging-diverging nozzles to convert the enthalpy of the high-temperature fluid into kinetic energy in the form of low-temperature, high-velocity, streams of vapor- liquid mixture (nozzles are discussed in more detail in the next section). The advantage of this system is the potential for achieving higher utilization of thermal energy than either the flashed steam or binary method Alger (1975).

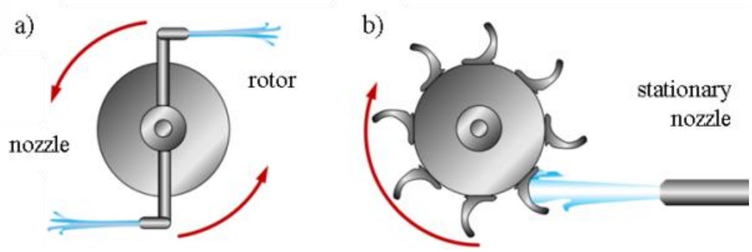

Figure 2- Schematic of (a) reaction turbine (b) impulse turbine

The total flow process involves the expansion of the fluid from a geothermal well (including water, vapor, and dissolved solids) through a single energy-conversion machine. Three characteristics of the total flow process are especially important:

1- It has the potential for the highest resource utilization efficiency because most of the available energy of the well head product is used.

2- The simplicity of the single-stage expansion is an advantage in dealing with the harsh chemistry of geothermal brines.

3- It broadens opportunities for successful exploitation of high-temperature/ high-salinity resources.

It is known that application of an efficient energy conversion device (turbine) in trilateral (topping, bottoming and/or standalone) thermal energy conversion cycles could increase the overall efficiency of many energy conversion systems (like Rankine cycles) by 10 to 50%. In the trilateral cycle, most of the available heat in the system is utilized which is not the case in the Rankine cycle. This is why trilateral cycles have higher cycle efficiencies (in terms of utilization of available heat), and why application of two-phase turbines, at least in a topping cycle, would ensure increased electrical power output Kris (2006).

1.1 Literature Review on Nozzles

The requirement of high velocity for efficient conversion of thermal energy to kinetic energy for momentum transfer in an impulse/reaction turbine dictates the use of converging-diverging nozzles for supersonic flow velocities. It is also most important to develop a highly efficient two-phase flow nozzle when adapting a turbo-type of machine as a two-phase flow turbine.

A nozzle is a device in which the available energy of a high pressure fluid is converted to kinetic energy in an expansion process. A flashing flow nozzle is one in which the incoming fluid is a slightly subcooled liquid or low-quality liquid-vapor mixture so that substantial phase change takes place during the expansion process Bunch et al. (1996).

The expansion of fluids and gases through nozzles has been the subject of intensive study with literally hundreds of papers written on the subject. Austin and Lundberg (1974) investigated almost inclusively the expansion of superheated and high-quality steam, liquids, and gases, with very little emphasis given to low quality (<20%) steam, or two-phase flow. Goodenough (1927) studied widely in this area and concluded that nozzle coefficients dropped to 0.9 when the quality dropped to 80%. He also constructed an analytical expression which predicts a continually decreasing nozzle coefficient with decreasing quality. This work is known as "conventional wisdom" indicating that two-phase flow of low-quality steam through nozzles will not efficiently convert thermal energy to kinetic energy. However, some other investigations (Maneely (1962; Neusen (1962; Starkman et al. (1964)) show that conclusions based solely on the "conventional wisdom" could be misleading, and higher nozzle coefficients could be achieved depending on the flow rate, inlet pressure and backpressure on the nozzle.

Some experimental studies were done previously (Brown (1961; El-Fiqi et al. (2007; Maneely (1962; Mutair and Ikegami (2010; Neusen (1962; Starkman et al. (1964)) where the main interest was the determination of mass flow rate for choked two-phase flow conditions. The experimental data indicated that the nozzles tested were typically operating overexpanded (ambient pressure higher than ideal) and inefficiently, because of the shock waves occurring inside the nozzles. Thus, the thrust coefficients and inferred efficiencies were typically low for the operating conditions studied.

Flashing flow nozzles are of obvious interest for energy systems using high pressure hot geothermal water. Such nozzles are used in many total-flow turbine designs including the Hero turbine Comfort III (1978). They are also used in two phase ejectors proposed for flash system use Kornhauser (1990). Other flashing flow nozzle applications include two phase ejectors for desalination plants Leigh and Kaye (1970), two phase ejectors for refrigeration systems Kornhauser (1990; Menegay and Kornhauser (1996) and total-flow turbines for refrigeration systems Hays and Brasz (1996). Menegay (1991) and Menegay and Kornhauser (1996) used flashing flow nozzles in tests of a refrigeration system incorporating a two-phase ejector with a flashing flow motive nozzle. In later tests, the researchers attempted to improve nozzle efficiency by seeding the motive flow with small bubbles. The method of seeding the flow has been patented by Kornhauser and Menegay (1994). Comfort (1977) showed that, in the case of a turbo-type two-phase flow turbine, a two-phase flow with a low slip and a small droplet size has a higher efficiency. Moreover, a method of remixing flows Alger (1978), a method of injecting steam into initially subcooled hot water Ikeda and Fukuda (1980) and a method of swirling inlet subcooled hot water Hijikata et al. (1985) resulted in increased efficiency. Akagawa et al. (1988) and Ohta et al. (1993) are among the few researchers who measured flashing flow nozzle efficiency. They showed that the optimum pressure profile was influenced by inlet subcooling and by the divergence angle. The values of the thrust coefficient decrease with increasing subcooling under a constant divergence angle and an increase in subcooling lowers the nozzle efficiency. The maximum value of the thrust coefficient was obtained at a divergence angle of 6°. Also, the maximum thrust coefficient could be obtained at the optimum back pressure. They also showed that nozzle efficiency could be increased by placing thin wires just upstream the throat. In this way, thin wires could disturb the flow; therefore, stimulating a flashing inception, thereby shortening the delay time for vapour formation and increasing the nozzle efficiency. Another of their findings was that the maximum nonequilibrium pressure drop at the throat decreases with increasing inlet subcooling. The installation of wires is effective in lowering the maximum non-equilibrium pressure drop at the throat. Last but not least, the magnitude of the thrust increases with decreasing back pressure, because the thrust is the product of the mass flow rate and the velocity of the flow at the nozzle exit. The reduction in back pressure increases the adiabatic temperature drop and the mass flow rate is unaffected by the back pressure. Bunch (1996)found out that the efficiency of a nozzle could be increased by increasing the inlet quality with increasing total mass of seeding bubbles, and decreasing size of those bubbles.

Since most of the available geothermal resources all over the world and specifically in Australia are at low temperature and of low quality, there is strong interest in the investigation of harvesting such energy. However, further attention and research is required in this area. Therefore, the authors of the present paper have conducted experiments to enhance understanding of these phenomena and to investigate the possibility of utilizing the energy of low temperature, low quality mixtures in geothermal resources.

2. ANALYSIS OF TWO-PHASE FLOW NOZZLE

Superheated steam nozzles can be designed quite satisfactorily using perfect gas principles, with corrections extending the analysis into the high-quality two-phase region. In the low-quality region, such simplicity cannot properly model the flow since the complex interphase mass, momentum, and energy transfer mechanisms must be taken into account. Han et al. (2010) investigated two-phase flow of refrigerants in short tube orifices as a one-dimensional model.

Fig. 3 shows a schematic of a two-phase flow nozzle. For some geothermal resources, especially lower-temperature ones, the wellhead fluid that enters the nozzle is a saturated or compressed liquid. At the vena contracta, the fluid stream accelerates and pressure energy is converted to kinetic energy due to the sudden contraction. After the throat, the fluid cycles into the two-phase state.

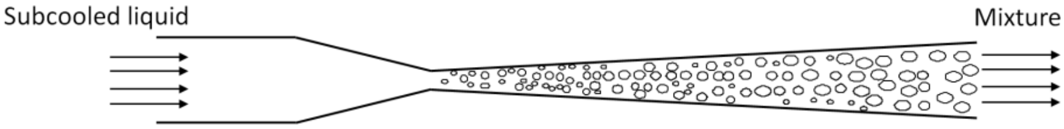

Figure 3- Schematic of a two-phase flow nozzle

The following assumptions are applied in order to obtain the main governing equations:

1- The flow is steady state and adiabatic

2- The flow is horizontal and is not affected by gravity or any other force including surface tension so that the partial pressure of the vapour phase is equal to that of the liquid phase.

Based on these assumptions, the continuity, momentum, and energy equations are given as

$$\dot{m} = \left[\frac{u_{l,e}}{\vartheta_{f,e}}(1-\alpha_e) + \frac{u_{g,e}}{\vartheta_{g,e}}\alpha_e\right]A_e \quad (1)$$

$$F_{exp} = \dot{m}(1-x_e)u_{l,e} + \dot{m}x_e u_{g,e} \quad (2)$$

$$h_{in} - h_e = (1-x_e)\frac{{u_{l,e}}^2}{2} + x_e\frac{{u_{g,e}}^2}{2} \quad (3)$$

$$h_e = h_{f,e} + x_e h_{fg,e} \quad (4)$$

$$x_e = \frac{\dot{m}_{g,e}}{\dot{m}} = \frac{1}{1+\left(\frac{u_{l,e}}{u_{g,e}}\right)\left(\frac{\vartheta_{g,e}}{\vartheta_{l,e}}\right)\left(\frac{1-\alpha_e}{\alpha_e}\right)} \quad (5)$$

where $\dot{m}, \dot{m}_{g,e}, u_{l,e}, \vartheta_{f,e}, \alpha_e, u_{g,e}, \vartheta_{g,e}, A_e, F_{exp}, x_e, h_{in}, h_{f,e}, h_{fg,e}$ and h_e are total mass flow rate, mass flow rate of vapor at exit, liquid velocity at exit, specific volume of liquid at exit, void fraction at exit, vapour velocity at exit, specific volume of vapour at exit, exit area, experimentally measured impulse (estimated thrust), quality of the mixture at exit, specific enthalpy of liquid at inlet, specific enthalpy of liquid at exit, the latent heat of evaporation at exit pressure, and specific enthalpy of the mixture at exit, respectively. When the inlet temperature, the inlet pressure, the back pressure, the estimated thrust, the total mass flow rate and exit area of the nozzle are known, the values of $u_{l,e}, u_{g,e}, a_e$ and x_e can be determined from the above equations with the assumption of thermal equilibrium at the nozzle exit.

Here, the energy conversion coefficient is expressed as follows Akagawa et al. (1988)

$$\eta_s = \frac{x_e u_{g,e}^2 + (1-x_e)u_{l,e}^2}{u_{es}^2} \quad (6)$$

where u_{es} represents both liquid and vapor and is the isentropic velocity of the fluid at the exit of the nozzle. In order to evaluate the performance characteristics of a nozzle, a thrust coefficient is used. The thrust coefficient, C_T, according to Akagawa et al. (1988) can be defined as

$$C_T = \frac{x_e u_{g,e} + (1-x_e) u_{l,e}}{u_{es}} \tag{7}$$

The value of η_s is in general larger than C_T^2 because:

$$\eta_s - C_T^2 = \frac{x_e(1-x_e)}{u_{es}^2}(u_{g,e} - u_{l,e})^2 \geq 0 \tag{8}$$

Under the conditions of $u_{l,e} = u_{g,e}$ or $x_e = 0$ or $x_e = 1$, the value of η_s is equal to that of C_T^2.

The ideal condition theoretically is the isentropic condition which is not achievable in the real world. The isentropic case has been chosen as a reference for comparison of the experimental results and a Isentropic-Homogeneous Expansion (IHE) model is adopted Starkman et al. (1964) in order to analyse the results , based on the following assumptions:

1. The velocities of the two phases are equal ($u_l = u_g$)
2. Thermal equilibrium exists
3. The expansion in the nozzle is isentropic
4. Property data correspond to those of a static, equilibrium two-phase system with plane interfaces.

In order to clarify the thermodynamics of the system, a T-S diagram is shown in Fig.4. In this figure, the isentropic condition is presented with the end point 'es' to which the entropy of the fluid remains constant. In this case, the fluid leaves the nozzle with a minimum value of enthalpy and the enthalpy drop is converted to kinetic energy. Therefore, the isentropic case would offer the optimum situation for fully converting thermal energy to kinetic energy. In the isenthalpic case (end point shown as 'eh' in the Fig.4), the enthalpy of the fluid will not change during the expansion, and therefore no conversion of thermal energy to kinetic energy is expected. The real case (end point displayed as 'e' in the Fig.4) lies somewhere between these two limiting conditions, where part of the thermal energy is converted to kinetic energy, and part of it is associated with phase change inside the nozzle.

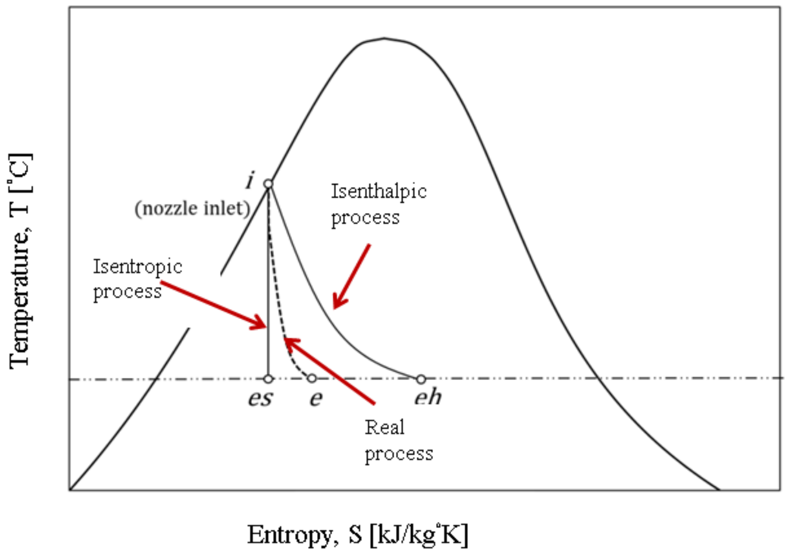

Figure 4- T¬–S diagram for subcooled/saturated liquid feeding to a two phase flow nozzle with possible scenarios as the most ideal (isentropic), fully irreversible (isenthalpic) and a real case as shown.

The ideal output for the total flow system is obtained from the isentropic enthalpy drop, (h_i-h_{es}). Since this occurs entirely in the nozzle, the nozzle exit velocity (neglecting inlet velocity and the change in potential energy) is given by

$$u_{es} = [2(h_i - h_{es})]^{1/2} \tag{9}$$

The isentropic enthalpy at the nozzle exit is obtained through calculating the quality of the isentropic mixture at the exit as

$$x_{es} = \frac{S_i - S_{f,e}}{S_{fg,e}} \tag{10}$$

where $S_i, S_{f,e}$ and $S_{fg,e}$ are the entropy of the fluid at the inlet, the entropy of the saturated liquid at exit conditions and the entropy difference of the saturated liquid and vapour at exit conditions, respectively. Now, h_{es} is obtained from

$$h_{es} = h_{f,e} + x_{es} h_{fg,e} \tag{11}$$

By considering the first assumption of the IHE model (both vapor and liquid have the same velocity in each section, i.e. $u_{l,e} = u_{g,e} = u_e$), C_T defined in Eq. (7) may be written as

$$C_T = \frac{u_e}{u_{es}} \tag{12}$$

where u_e is the velocity of the mixture at exit. Eq. 11 may also be written as

$$C_T = \frac{F_{exp}}{\dot{m}u_{es}} \tag{13}$$

indicating that the thrust coefficient could also be defined as the ratio of the measured impulse (estimated thrust) $F_{exp}$to the theoretical value of the thrust F_{th} calculated from the IHE model. It is noted that because of uncertainties involved in measuring thrust, in the current experiments the impulse force is measured which is theoretically equivalent to the thrust.

Starkman et al. (1964)mentioned that the IHE model provides acceptable agreement between the critical flow and pressure distribution values and the corresponding experimental data for initial qualities greater than approximately 5 percent. Below this quality, the measured data can diverge rapidly from the predicted values.

3. EXPERIMENTAL ARRANGEMENT

3.1 Description of Experiment

A test facility was designed to enable experimental study of the efficiency of two-phase flow nozzles under low pressure conditions. Fig. 5 shows a simple schematic of the experiment set up. Water was heated, in a tank of 100 litre capacity, to the boiling point at atmospheric pressure and introduced into the flash tank where an absolute pressure of 5kPa was maintained. This low pressure was initially achieved by connecting a vacuum pump to the flash tank. The pressure in the tank was then maintained at a low value by running cooling water through a condenser coil with a total surface area of 16.84 m^2.

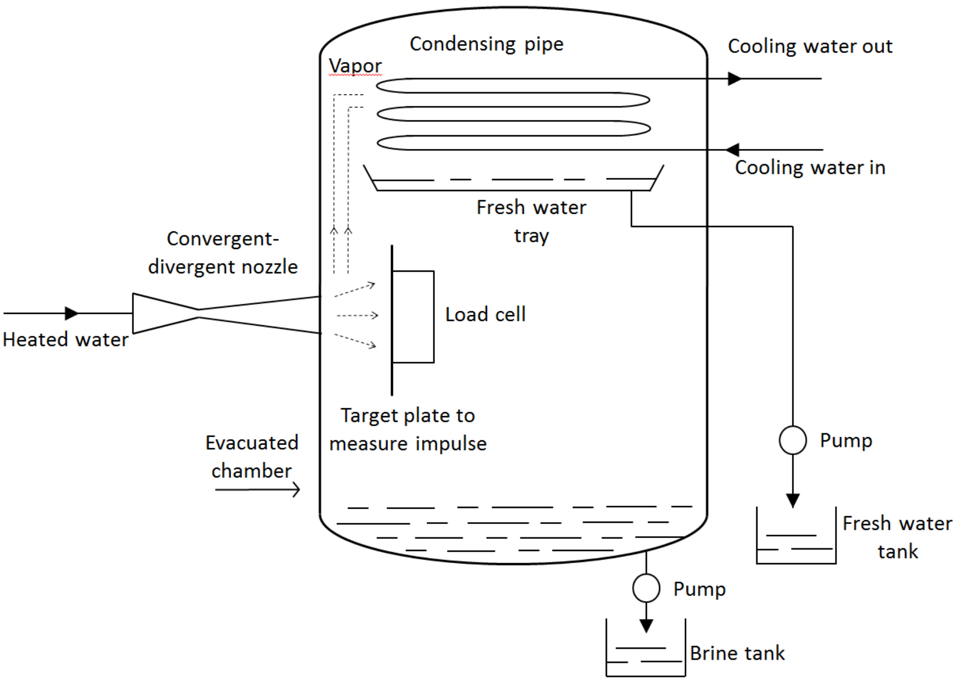

Figure 5- A simplified schematic of the experiment set up

Before hot water entered the inlet pipe, any solid impurities were filtered to prevent clogging of the nozzle. The hot water moved rapidly from the atmospheric pressure region to a very low pressure region through the converging-diverging nozzle. At the nozzle throat, the fluid was accelerated due to the sudden contraction and, therefore experienced a sudden pressure drop. As a result of experiencing a sudden pressure drop, the fluid flash evaporated. This created a high velocity jet of liquid and vapor mixture at a lower temperature which impinged on a target plate. In order to assist analysis of the process, the temperature and pressure profiles along the nozzle were measured using pitot tubes and thermocouples as shown in Fig. 6.

It was assumed that a homogenous mixture of the liquid and vapour flowed out of the nozzle at very high velocity, and as a consequence, the jet produced an impulse force ($\dot{m} \times u_e$) on the target where an equal reaction force (thrust) also acted on the stationary nozzle.

In order to capture the impulse force, a target plate was installed in the vacuum tank and located in front of the nozzle exit. By installing a load cell at the back of the target plate, the impulse force was measured. The liquid portion of the mixture eventually fell to the bottom plate to be collected in the brine tank, whereas the vapor rose in the chamber. The vapor was then condensed on the surface of the condenser and collected in the freshwater tank.

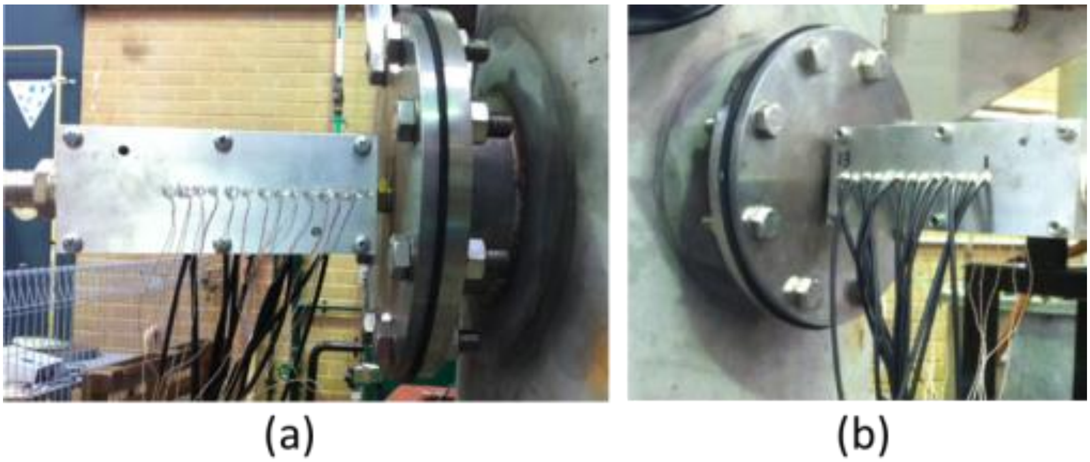

Figure 6- (a) measuring temperature along the nozzle by T-type thermocouples, (b) measuring pressure along the nozzle using static pitot tubes

A main focus of this experimental study was to achieve the maximum impulse force caused by the jet exiting the nozzle.

Fig. 7 depicts a detail drawing of the nozzle. This geometry is adopted from Ohta et al. (1993) where higher pressure feed water was investigated. The convergent and divergent angles were 28 and 6 degrees, respectively. The throat diameter was 3.5 mm and the exit diameter was 14.8 mm.

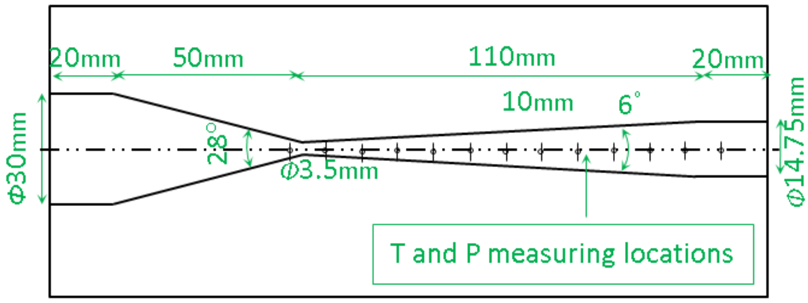

Figure 7- Cross section of the nozzle used in the experiments

3.2 Experimental Procedure, Data Collection and Sample Calculations

In each experiment, water was heated to the desired temperature (60, 80 or 100°C) in a tank. At the same time the flash tank was evacuated using a vacuum pump. Low pressure was maintained throughout the experiment by the provision of proper sealing of the system and by flow of cold water through the condenser tubes. The load cell was calibrated before water was introduced into the system. Then, the hot water was introduced to the flash tank through the stationary nozzle wherein the energy conversion occurred. Thirteen measuring points at 10mm increment were allocated for measuring the temperature and pressure along the nozzle. Temperature was measured and recorded by T-type thermocouples connected to a DataTaker DT-800. In order to measure pressure, all the points were connected through pneumatic pipes of 0.3 mm diameter to a manifold where a pressure gauge and a pressure transducer were installed. The procedure for taking the pressure measurement along the nozzle was as follows: Only one valve at each time was opened; the corresponding pressure was read; the valve was closed, and the next one opened. The impulse produced on the target plate by the jet exiting the nozzle was measured by a load cell and recorded by a Vishay strain indicator.

After data collection, derived results were calculated. In this section, a sample calculation based on the thermodynamics of the system (Fig. 4) is presented.

By measuring the temperature of point ‘i’ (inlet feed water) and assuming the state is saturated liquid, other thermal properties of water may be found using tables of water thermal properties. Since there is yet no vapour generated at this point, thermal properties will be the ones assigned to saturated liquid, as follows:

$$S_i = S_{f,i} \tag{14}$$

$$h_i = h_{f,i} \tag{15}$$

$$\vartheta_i = \vartheta_{f,i} \tag{16}$$

By knowing the pressure in the flash tank (evacuated chamber), the quality of the mixture at the nozzle exit for an isentropic process is calculated as follows

$$S_{es} = S_i \tag{17}$$

$$S_{es} = S_{f,e} + x_{es} S_{fg,e} \tag{18}$$

from equations (14), (17) and (18), the quality of the mixture at point 'es' shown in Fig. 4 is obtained from

$$x_{es} = \frac{S_{f,i} - S_{f,e}}{S_{fg,e}} \tag{19}$$

For the real case, entropy of the mixture at the exit of the nozzle will be higher than the isentropic case (refer to Fig.4). The isentropic efficiency of the cycle indicates how far the real case differs from the isentropic case. For this purpose, the quality of the mixture at the exit of the nozzle, x_e, is required. In order to obtain x_e, the measured impulse force, caused by the jet of mixture after exiting the nozzle and impinging on the target plate, Equation 20 is employed:

$$u_e = \frac{F_{exp}}{\dot{m}} \tag{20}$$

where u_e is the velocity of jet at the nozzle exit, $\dot{m}$ is the total mass flow rate and F_{exp} is the experimentally measured impulse on the target plate.

$$h_e = h_{in} - \frac{u_e^2}{2} \tag{21}$$

$$x_e = \frac{h_e - h_{f,e}}{h_{fg,e}} \tag{22}$$

Then, the isentropic efficiency is calculated from

$$\eta_s = \frac{h_i - h_e}{h_i - h_{es}} \tag{23}$$

which will be equal to 0 for an isenthalpic (or constant enthalpy) process and equal to 1 for an isentropic process. For the real case, the process is shown by the 'i-e' line for which η_s will be between 0 and 1.

As discussed in the previous section, the thrust coefficient factor C_T, is calculated from

$$C_T = \frac{F_{exp}}{\dot{m} u_{es}} \tag{24}$$

which from equations (9) and (12) equals

$$C_T = \frac{u_e}{u_{es}} = \sqrt{\frac{h_i - h_e}{h_i - h_{es}}} \tag{25}$$

The thrust coefficient, C_T, and the isentropic efficiency of the cycle are then related by the following equation

$$\eta_s = {C_T}^2 \tag{26}$$

which confirms Equation (8) under the assumption of no slip velocity at the exit.

In Table 1 the experimental conditions and corresponding calculated results are presented.

Table 1- experimental conditions and corresponding calculation results

Experiment	Feed water temp. (°C)	Back pressure (kPa)	$\dot{m}$ (kg/s)	F_{exp} (N)	u_e (m/s) Calc. from Eq. 20	h_e (kJ/kg) Calc. from Eq. 21	x_e (-) Calc. from Eq. 22	η_s (-) Calc. from Eq. 23	C_T (-) Calc. from Eq. 25
exp. A	60.5	5.8	0.093	2.8	30.0	218.9	0.029	0.241	0.491
exp. B	80.0	7.7	0.075	5.2	69.2	306.0	0.056	0.352	0.593
exp. C	96.0	8.8	0.072	7.6	105.7	371.7	0.079	0.420	0.648

3.3 Sources of Measurement Error

Temperature was measured at different positions including: at the inlet to the nozzle, at the exit of the nozzle, condenser inlet and outlet, and at thirteen points along the nozzle. T-type thermocouples were used in the experimental set-up with accuracy of ±0.5^{o}C. These temperatures were recorded by a DataTaker as shown in Fig. 8.

Figure 8- DataTaker used for taking the temperature data

The reaction force created by the mixture fluid exiting from the nozzle was approximated by measuring the impulse force on a target plate attached to a load cell. A circular plate of 150mm diameter was installed 20 mm downstream of the nozzle exit and perpendicular to the stream flow in order to capture the impulse force caused by the mixture. The load cell was mounted at the back of the plate from which the value of the force was logged through a digital Vishay strain indicator and recorder as shown in Fig. 9. To ensure the reliability of the readings, the strain indicator was calibrated at the beginning of each run by different weights varying between the ranges of 7 and 400 grams. Then the correlation between force (m×g) and readings from the strain indicator was determined allowing the data from the strain indicator to be converted to the reaction force.

Figure 9- Vishay strain indicator used for taking the impulse force

Measurement of the pressure along the nozzle was done using a pressure transducer and a dial pressure gauge. A PTX/PMP 1400 series was used with typically ±15% accuracy. All the pressure tubes along the nozzle were connected to a manifold as presented in Fig. 10 with an individual valve incorporated in each line.

Figure 10- Manifold used for measuring the pressure at each location along the nozzle

4. RESULTS AND DISCUSSION

4.1 Calculated Results

In Table 1, a summary of experimental conditions and corresponding calculated results is presented. The following conclusions are based on the results shown in the table:

- By increasing the inlet temperature, the nozzle efficiency increases considerably.
- By increasing the inlet temperature, the thrust coefficient increases considerably.
- As the inlet temperature increases, the quality of mixture leaving the nozzle increases by a factor of 2.7 in the experimental regime.
- The impulse force caused by the jet increases substantially by a factor of 2.7 when the feed water temperature varies from 60°C to 96°C.

4.2 Pressure and Temperature Profiles

In the range of our experiments, by increasing the inlet temperature, the mass flow rate decreases slightly by a factor of 0.7. The cause of slight change could be the square root effect that appears in the relationship between mass flow rate and pressure drop.

The temperature, pressure, and saturation pressure profiles along the nozzle for three different flow conditions (mainly different inlet temperatures and slight variation in back pressure) are shown in Figures 11-13. For the sake of better visualization, the convergent-divergent nozzle geometry profile is also shown. In these figures, the nozzle throat defines the datum coordinate of the horizontal axis. Based on these figures, the measured pressure profile is close to the saturation pressure profile for the measured temperature at each measurement point except before the throat where the liquid is subcooled (refer to Fig. 11). For higher inlet temperatures, the pressure downstream of the throat is higher (eg for inlet temperature of 100ºC, the pressure near the throat is approximately 68kPa; whereas in the case of inlet temperature of 60 ºC the pressure near the throat is approximately 20kPa). This could be explained by the higher velocity of the mixture in the diverging section for the first case (with higher inlet temperature) compared to that for the latter case. In Fig. 14, the pressure profiles of the nozzle for three different conditions are presented to facilitate comparison. It can be seen that increase of inlet temperature will result in higher pressure downstream the throat leading to lower mass flow rate ($\dot{m}$). This is explained by the observation that with higher inlet temperature, the rate of phase change is higher, the velocity is higher, and pressure drop is lower. Lower pressure drop in the nozzle implies higher pressure downstream of the throat.

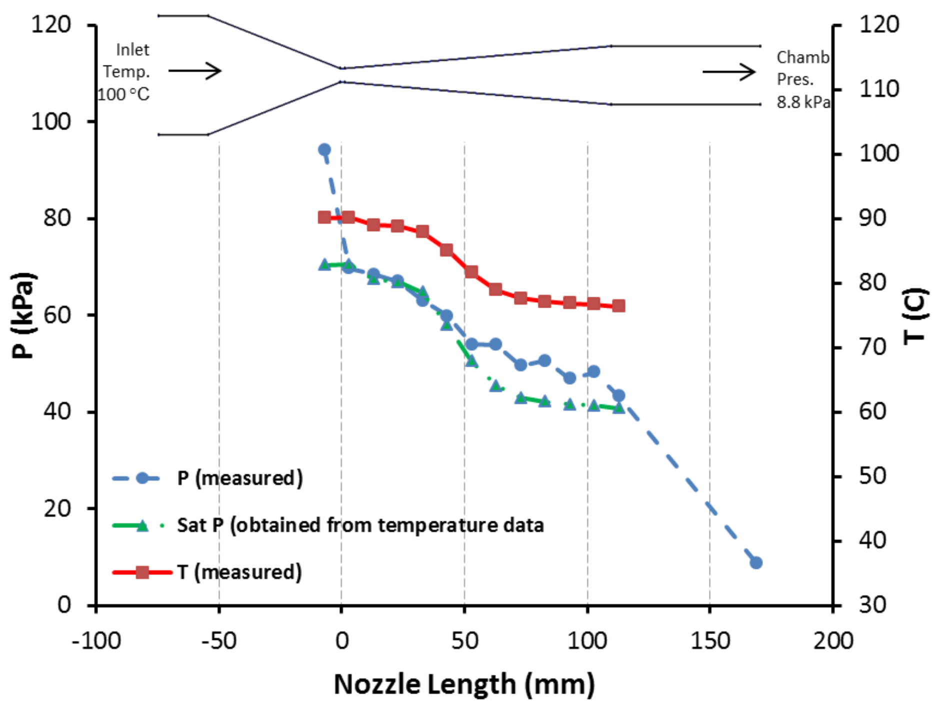

Figure 11. Pressure and temperature profiles along the nozzle- inlet temperature 100°C, back pressure 8.8 kPa

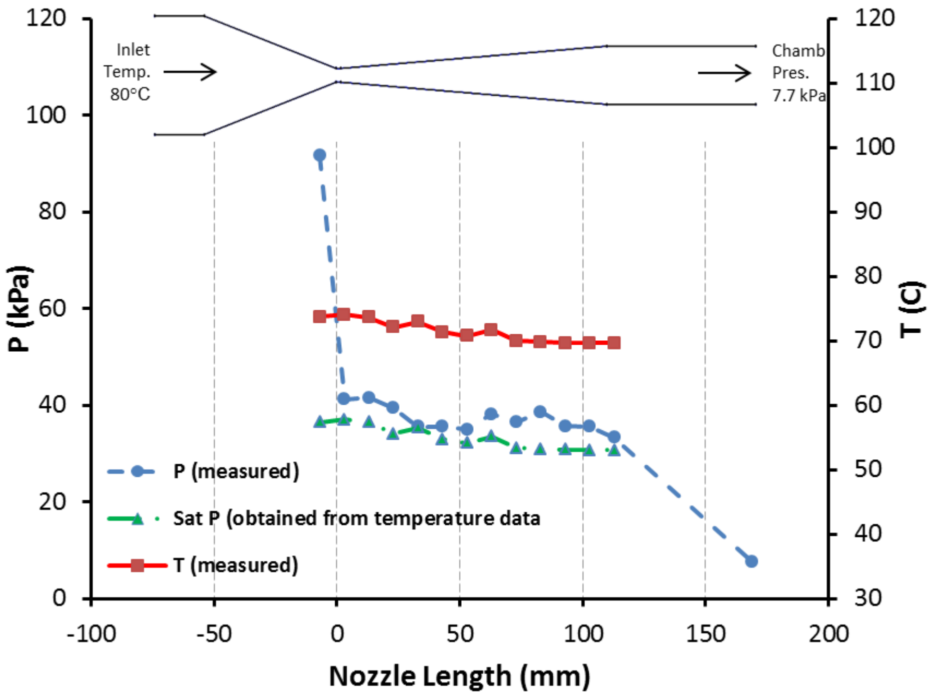

Figure 12. Pressure and temperature profiles along the nozzle- inlet temperature 80°C, back pressure 7.7 kPa

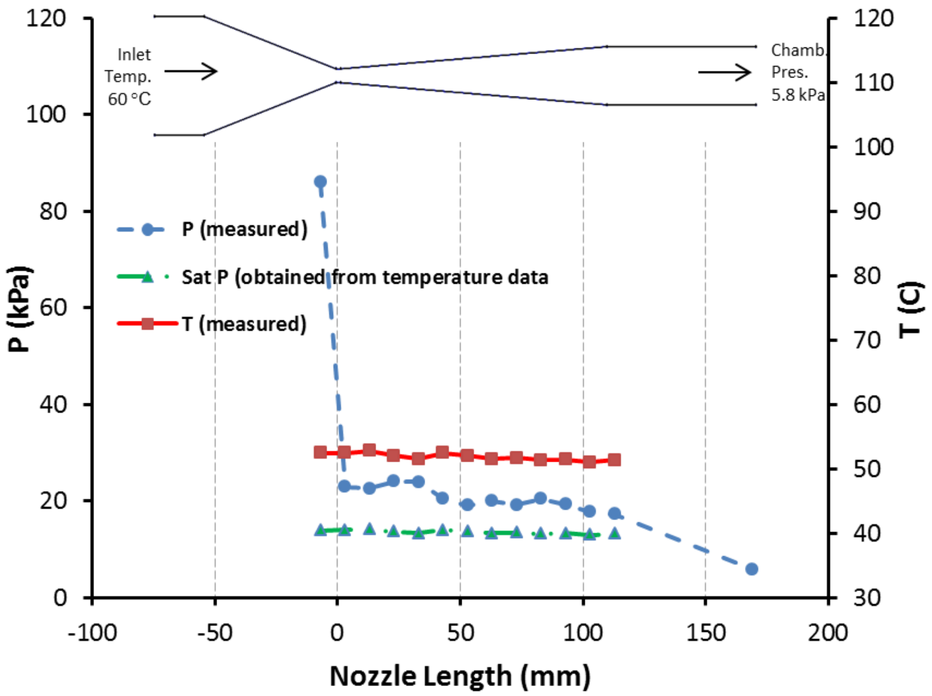

Figure 13. Pressure and temperature profiles along the nozzle- inlet temperature 60°C, back pressure 5.8 kPa

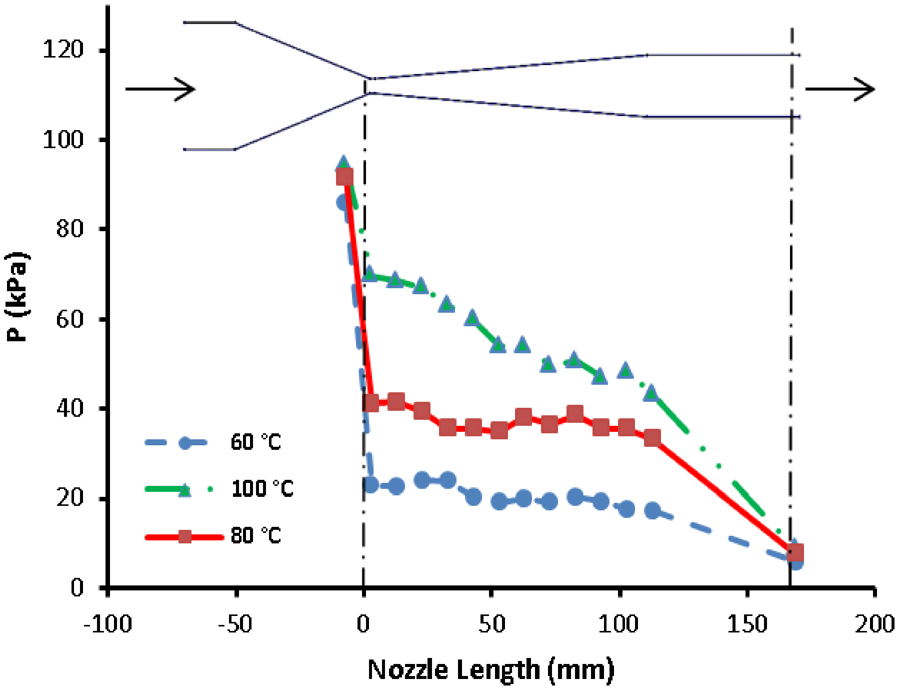

Figure 14. Pressure profiles along the nozzle for three different conditions

As can be seen from Figures 11 to 14, the fluid experiences a large pressure drop at the throat. This agrees well with similar results reported from previous investigations Ohta et al. (1993), as shown in Fig. 15 which presents the pressure profile along the nozzle for the experiments of Ohta et al. (1993) with similar experimental arrangements and approximately similar nozzle geometry, but with higher inlet and higher back pressures. As can be seen from Figs. 11-14 the fluid experiences a high pressure drop at the nozzle throat.

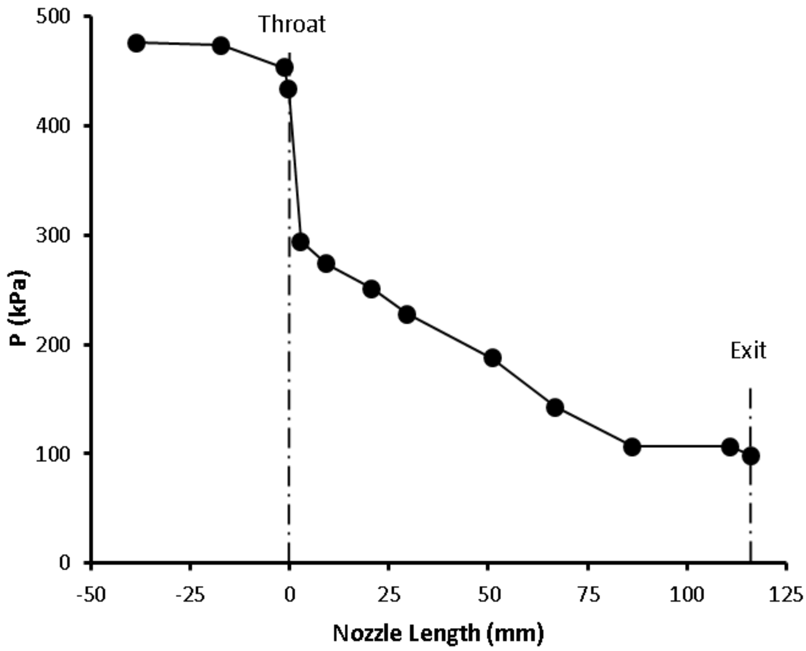

Figure 15. Pressure profile along the nozzle from Ohta et al. (1993) with water as working fluid- inlet subcooling temperature 1.5°C, back pressure 100.4 kPa

4.3 Thrust Coefficient and Isentropic Efficiency of the Nozzle

In Fig. 16, the thrust coefficient, C_T, and isentropic efficiency, η_s, of the nozzle as percentages as described in Section 3.2 and calculated in Table 1 are depicted against feed water temperature for different experimental conditions. The thrust coefficient and isentropic efficiency of the nozzle are calculated from Eqs 25 and 23, respectively. The calculations demonstrate that Eq. 26 is valid for the assumption of no slip velocity, i.e. $u_l = u_g$.

It is observed that by increasing the feed water temperature and keeping the back pressure similar (note that the inlet condition is at atmospheric pressure, and a slight change in back pressure around 3kPa does not influence the results), the efficiency of the nozzle is increased. The trend that this figure is shows, suggests that by having similar experimental conditions, the maximum nozzle efficiency probably could be achieved by introducing the feed water at a temperature of approximately 130°C.

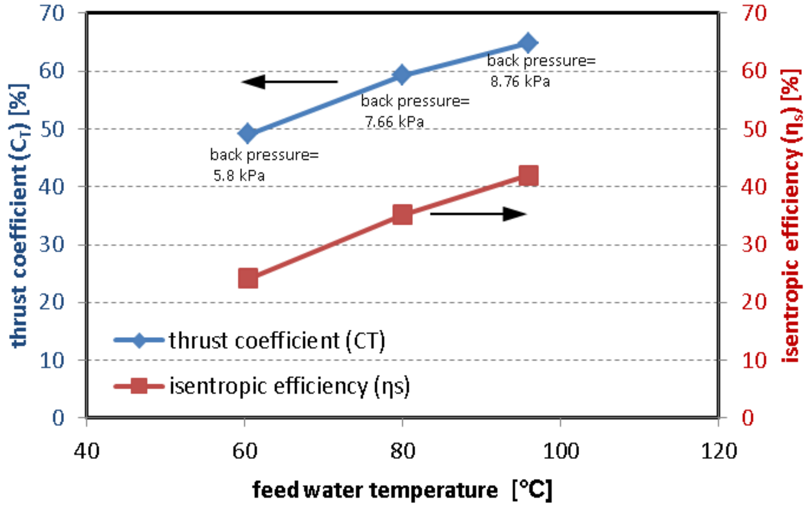

Figure 16. Nozzle isentropic efficiency and thrust coefficient vs. feed water temperature

Fig. 17 compares the thrust coefficients as a percentage vs. back pressure in different experiments of Alger (1975). In the same figure results of the current work are presented. It should be noted that the data presented by Alger (1975) refer to experiments where the fluid is at a higher state of energy at its entrance to the nozzle compared to the present work. The experiments conducted by Alger (1975) were considered quality of 10 to 15 percent steam; whereas, in the present work, the fluid had a quality of zero as it was substantially subcooled (up to 40°C) at its entrance to the nozzle. This explains the lower thrust coefficient achieved in the current work. Despite having a quality of the mixture equal to zero at the nozzle entrance, a thrust coefficient of 60 percent is still achievable. This result encourages utilization of low temperature, low quality resources for energy harvest.

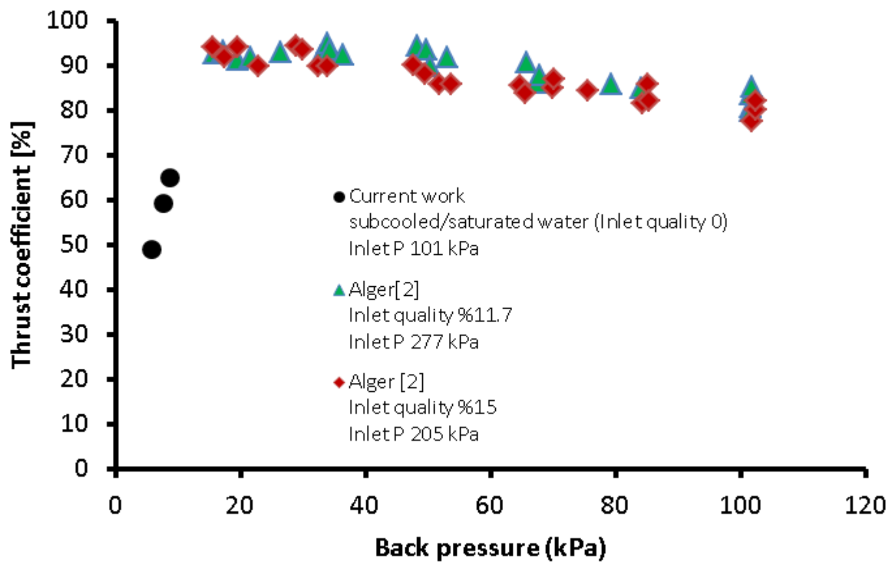

Figure 17. Thrust coefficient percentage vs. back pressure (kPa)

The influence of inlet quality on nozzle efficiency was investigated by Bunch et al. (1996) where the feed fluid was seeded with small bubbles. Fig. 18 depicts thrust coefficient (as a percentage) vs. nozzle inlet quality, for two sets of experiments by Bunch et al. (1996), in which R134A was used as the working fluid. The results show that as the nozzle inlet quality decreases, the thrust coefficient of the nozzle decreases. Thrust coefficients are generally considered to decrease markedly when water droplets are present Austin and Lundberg (1974). However, the work presented in Alger (1975) shows that high performance nozzles with thrust coefficients up to and probably beyond 0.94 can be designed for inlet quality (vapour mass fraction) as low as 12 percent (whereas, in the present case the inlet quality is zero (subcooled / saturated liquid)). However, with almost zero quality a 60 percent thrust coefficient is achievable which encourages further utilization of such resources.

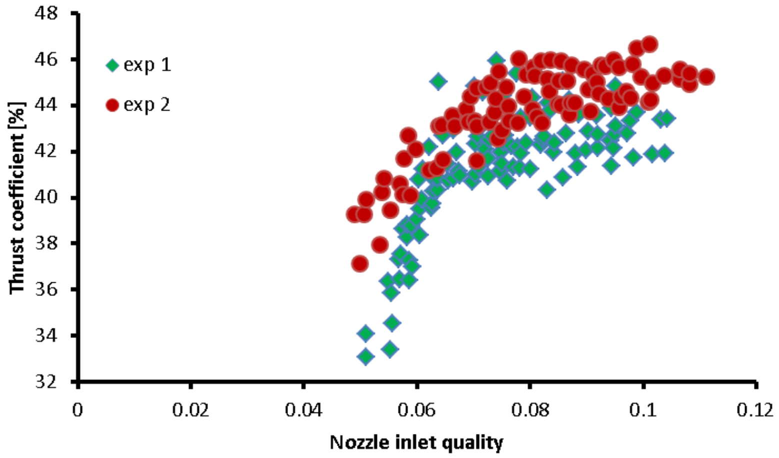

Figure 18. Thrust coefficient percentage vs. Nozzle inlet quality for two sets of experiments of Bunch et al. (1996)

The low efficiency of the nozzle indicates that less vapour is produced than expected. The assumption of the calculations is that a homogeneous mixture is present along the nozzle; however, a slug of liquid enters the nozzle at higher than saturation pressure. Also, there is likely to be some heat transfer between the nozzle and the environment indicating a non-adiabatic process. Therefore, not all the expected heat will evaporate liquid, leading to a less efficient nozzle.

Under-expanded situations when the flow has not had the chance to fully equilibrate, result in a pressure at the exit that is higher than the chamber pressure. Since this condition does not represent a complete expansion (all available thermal energy has not been converted into kinetic energy) the nozzle is inefficient when operating in this mode. Also, for operating conditions where no shock wave or expansion wave is present, the highest thrust value could be achieved.

5. CONCLUSION AND FUTURE WORK

Experiments on two-phase flow in a converging-diverging nozzle have been carried out under various fluid conditions. Results indicate that with feed water temperatures as low as 60°C, with no bubbles in the inlet flow, thrust coefficients as high as 50 percent could be achieved. This demonstrated that nozzles, whether stationary or rotary, could be used to generate force as a means of generating mechanical power using low temperature heat sources as low as 60°C and with reasonable efficiencies. This encourages application of this energy conversion method to better utilization of heat whether using geothermal, solar or waste heat resources. Further research could enable achievement of higher nozzle efficiencies. Possibilities for future research include:

- Change of chamber pressure ranging from minimum saturation pressure, which could be achieved by a good condensation facility, to atmospheric pressure.
- Experiments with different nozzle lengths (from the throat to the exit opening).
- Experiments with different nozzle geometries (different ratios of the throat area to the exit area).
- Experiments with other fluids that are commonly being adopted for power producing devices, in which renewable energy or waste heat sources are applied (for example isopentane).
- Investigating the effects of putting transverse wires upstream of the throat.
- Further to discussion in Section 4.3, it is encouraged to experiment with feed water temperatures close to 130°C to in order to investigate the possibility of achieving the maximum nozzle efficiency.

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