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Design, Fabrication and Experimental Study of Heat Transfer Characteristics of a Micro Heat Pipe

R.A. Hossain [a,*], M.A.K Chowdhuri [b], C. M. Feroz [c]

[a] *Department of Mechanical Engineering,University of Alberta,Edmonton, AB, Canada, T6G 2G8.*

[b,c] *Department of Mechanical Engineering,Bangladesh University and Technology University, Dhaka 1000, Bangladesh.*

Abstract

In order to study the heat transfer characteristics, a micro heat pipe (MHP) of circular geometry having inner diameter 1.8 mm and length 150 mm is designed and fabricated. An experimental investigation is carried out also to investigate the performance of the MHP with different experimental parameters. These experimental parameters include inclination angle, coolant flow rate, working fluid and heat input. Inclination angle are varied from 30^0 to 90^0, where as coolant flow rate and heat input are varied from 0.3 lit/min to 1.0 lit/min and 0.612 W to 8.71W respectively. Three different types of working fluids are used; acetone, ethanol and methanol. For each working fluid, heat transfer characteristics are determined experimentally for different inclination angle and different coolant flow rate at different heat input. Acetone is proved to be better as working fluid. A correlation is also made for acetone to relate other experimental parameters for determination of heat transfer coefficient.

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Keywords: Micro Heat Pipe, Heat transfer characteristics, Inclination angle.

Nomenclature

Ae:	Surface area of evaporator	mm^2
m_c:	Coolant flow rate	lit/min
Q:	Heat Input	W
R:	Thermal Resistance	$^0C/W$
Tc:	Average condenser temp	^0C
Te:	Average evaporator temp	^0C
T_w:	Wall temperature	^0C
U:	Overall heat transfer coefficient	$W/m^2\,^0C$
X:	Axial distance	mm
θ:	Inclination angle	^0C

* Corresponding author. rhossain@ualberta.ca

1. Introduction

Overheating of integrated circuit (IC), microchip etc. is a potential threat to these electronic components. It is very important to facilitate optimum cooling of electronic components in a smaller electronic device because integrated circuit lifetime depends on it. An increasing market demand on powerful gadgets in smaller and smaller cabinets creates a trade off situation: either to enlarge the package to accept additional cooling or to sacrifice IC lifetime. This is a great challenge in thermal design management. Among other cooling techniques heat pipes emerge as the most appropriate technology and cost effective thermal design solution due to its excellent heat transfer capability, high efficiency and its structural simplicity. Due to the space constraint in most of personal computers and telecommunication devices, the size of heat pipes has to be carefully decided. Thus application of micro heat pipe (MHP) has been extended gradually. So investigation on MHP is indispensable for further development and improvement of its performance.

Besides electronic cooling, there are many other applications, where MHPs may be useful. For example, MHPs are interesting to be used in implanted neural stimulators, sensors and pumps, electronic wrist watches, active transponders transponders, self-powered temperature displays, temperature warning systems. MHPs are promising to cool and heat some biological micro-objects [1].

A heat pipe is a heat transfer mechanism that can transport large quantities of heat with a very small difference in temperature between the hot and cold interfaces. Heat pipes are pencil-sized metal tubes that move heat from one end of the tube to the other without the aid of a pump. Within the heat pipe, heat vaporizes a small amount of fluid at the pipe's hot end; the fluid travels to the other, slightly cooler end and condenses before returning to the hot end through a capillary wick, where it repeats the process. The device efficiently transfers large quantities of heat. The heat pipe can, even in its simplest form, provide a unique medium for the study of several aspects of fluid dynamics and heat transfer, and it is growing in significance as a tool for use by the practicing engineer or physicist in applications ranging from heat recovery to precise control of laboratory experiments. Today heat pipes are widely used in computer, telecommunication, and other various electronics equipment. It is a light weight device with no moving parts, silent in operation and having several hundred times the heat transport capacity as compared to the best metallic heat conductor like silver and copper. They are often called the "superconductors" of heat, as they possess an extra ordinary heat transfer capacity with almost no heat loss. Moreover, the thermal management problems of microelectronic components will worsen with further miniaturization. The size of microprocessors has been reducing day by day with the development of electronics. Consequently, the number of active semi-conductor devices per unit chip area has been increasing. In the last decade, the number of active semi-conductor devices per unit chip area has almost quadrupled [2]. The minimum feature size in microprocessors has reduced from 0.35 μm in 1990 to 0.25 μm in 1997 and which will go down further to 0.05 μm by the year 2012 [3]. This has increased the heat dissipation density for desktop microprocessors. As an example, heat flux has increased from 2W/cm2 for an Intel 486 microprocessor to almost 21 W/cm2 for the Intel P II 300-400 MHz microprocessors [4]. The current heat dissipation rates for some of desktop computers are approximately 25 W/cm2. It is expected that microprocessor chips for some of the next generation work stations will dissipate 50-100 W/cm2. Thus reduction in size also brings severe limitations to the conventional cooling techniques [5].

Development of efficient thermal management scheme is essential to dissipate these high heat fluxes and maintain suitable operating temperature of the device. Micro heat pipes are increasingly filling this role. To keep up with today's thermal solution challenges, micro heat pipes must improve efficiency and integrate remote heat transfer into thermal management solutions. Thus application of MHP has been extended gradually. Jin Zhang [6] studied the heat transfer and fluid flow in an idealized micro heat pipe. Yuichi et al. [7] experimentally confirmed steady-state heat transfer characteristics of flat MHP in detail and proposed a method for determining its maximum heat transfer rate. Moon et al. [8] studied performance of a triangular micro heat pipe mounted horizontally and found that its heat transport limit 6-15 W/cm2 for the operating temperature ranging from 45-80 oC. Zhuang et al. [9] compared the performance of heat pipes placed at different inclination angles in terms of maximum heat transfer capacity using three different structural wicks. It is found that all structures of wicks have little influence on the heat transfer capacity of heat pipes working with the aid of gravity. Under the condition of anti gravity, the structure of wicks has obvious influence on the heat transfer capacity of miniature heat pipes.

Moon et al. [10] experimentally investigated the thermal performance of micro heat pipe with polygon cross-sections. They showed that in the case of the MHP with a small sized equivalent diameter smaller than 2 mm, the effect of the pipe length on the thermal performance of the MHP could be large because of the pressure losses by friction at the vapor–liquid interface and the capillary limitation for returning of condensed liquid are significantly dominant as an increase of the pipe length. They also showed, the thermal performance of the triangular MHP tends to be increased according to the decrease of the pipe length. For this study the length of the pipe is chosen as close to specified by Moon et al. [10].

From the above discussion it may be noted that a number of research works on micro heat pipes have been carried out but very limited research work has done so far to completely determine the performance of MHP. Again researches that include effects of different experimental parameters such as inclination angle, coolant flow rates, heat input and working fluid on the performance of MHP has not studied yet to the best of our knowledge. This study completely investigates the effect of all the above experimental parameters on the performance of MHP.

2. Design of Micro Heat Pipe

Since a micro heat pipe (MHP) generally refers to small heat pipe with a diameter of less than 3 mm, selection of diameter is an important design consideration for fabrication of MHP. Heat pipe of circular geometry is used in this experiment. In this study for MHP a copper tube of 2 mm outer diameter and 1.8 mm inner diameter is selected. This dimension is chosen because it fulfils the required range for MHP and its availability in local market as well. The reasons behind selecting copper tube are: The compatibility of copper with both the working fluid and the external environment is very high., Thermal conductivity of copper is also a higher than the other container materials such as stainless steel, mild steel etc, Copper facilitates the ease of fabrication, including weld ability, machine ability and ductility and also there are some other important properties required for heat pipe container like porosity, wet ability, strength to weight ratio - which is more important in spacecraft applications etc.

The heat pipe consists of three sections; evaporator section, adiabatic section and condenser section (Fig. 1). The total length of MHP is arbitrarily selected as 150 mm. This dimension is convenient to handle and can easily be used in practical applications. The detail dimension of

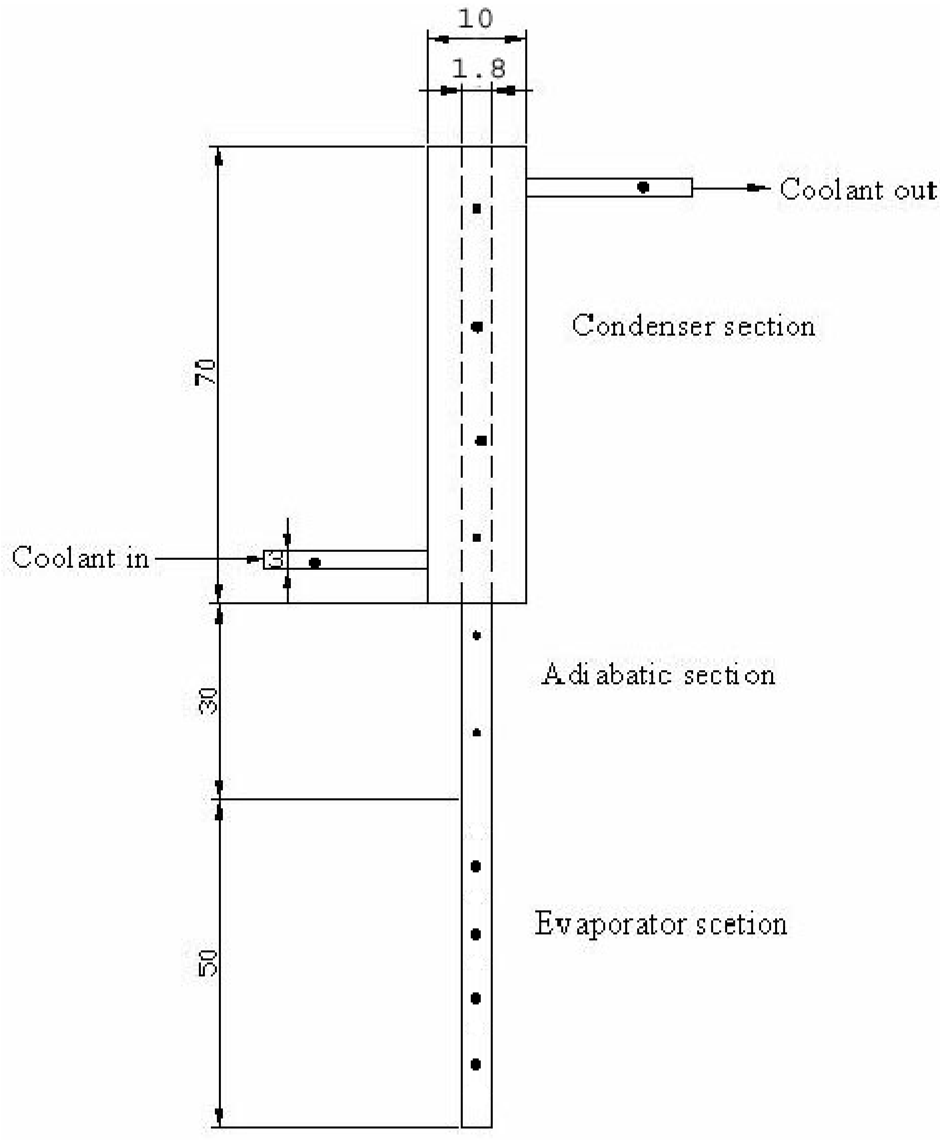

Figure 1: Schematic view of circular MHP.

MHP used in the experiment is summarized in Table 1.

Table 1: The detail dimension of MHP

Parameter	Dimension (mm)
Hydraulic diameter of pipe, d_h	1.8
Length of Heat Pipe, L	150
Length of evaporator section, L_e	50
Length of adiabatic section, L_a	30
Length of condenser section, L_c	70

Evaporator section is located at the bottom of the heat pipe. Heat is added to the heat pipe through evaporator section. Adiabatic section is located in between the evaporator and condenser section. This section is actually kept with heat pipe to distinguish evaporator section and condenser section. Condenser section is the uppermost part of the heat pipe. There is a water jacket around this section which is concentric with the container section. Water flowing through the jacket cools the condenser section of the pipe and thus takes away the latent heat of condenser of vapor.

3. Experimental Procedure

The schematic diagram of the experimental setup is shown in Fig. 2. At first working fluid is poured to the evaporator section of MHP. The amount of working fluid is selected to maintain a charge ratio, which is the volume of the evaporator filled by the working fluid to the total volume of the evaporator to 0.9 because previous studies on MHP show that, this value of charge ratio is convenient. In other words the evaporator is 90 percent filled by the working fluid. Since no vacuum pump is used, the remaining portion of MHP may contain air. Ni-Cr thermic wires having width of 1.5 mm, thickness of 0.1 mm and total resistance of 16 Ω are wound around the

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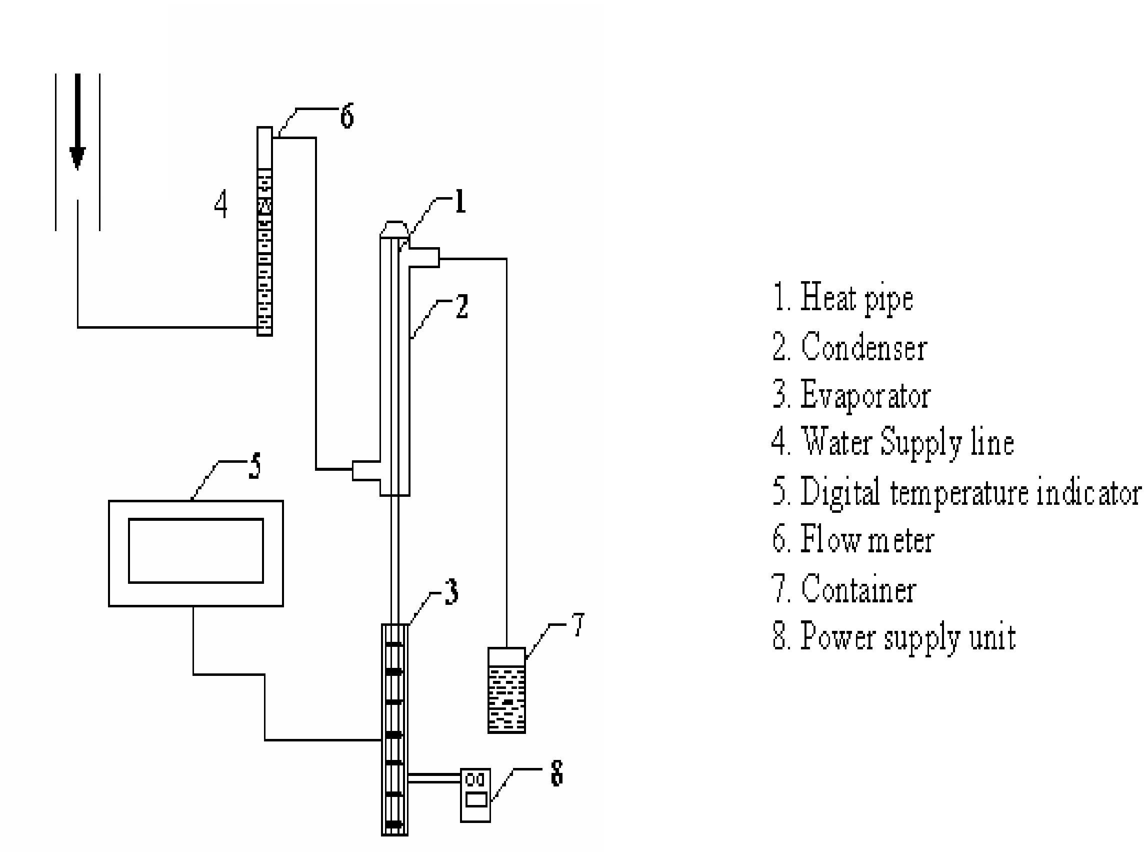

Figure 2: Schematic diagram of the experimental setup.

evaporator wall maintaining equal spacing of 1.5 mm. For electrical insulation in the evaporator section, insulation tape is used. The heat added to the evaporator section of MHP is processed in the electric method by using the AC power supply (variac). To minimize heat losses, evaporator section is covered with glass wool.

The condenser section is cooled by a constant temperature water coolant, circulating in an annular space between the copper tube and the jacket. The water coolant is taken from supply line trough pipe and the flow is controlled by the flow meter. The inlet and outlet coolant temperatures are measured. Twelve calibrated *K* type (Cooper-Constantan, Φ 0.18 mm) thermocouples are used. Ten are attached at the wall of each MHP to measure the wall temperature; four units at the evaporator section, two units at the adiabatic section and four units are at the condenser section. Remaining two are used to measure the inlet and outlet temperature of the condenser water. The thermocouples are attached at the wall surface-using adhesive. Temperatures are measured by a digital thermometer.

An input power to the heater in the evaporator section is increased by using a variac from 0.612 W to 8.71 W. Ammeter and voltmeter were used to measure the voltage and current in the heater circuit so that a relation can be made between the variac reading and actual heat transfer from the heater to the heat pipe. The measurements are made under a steady state condition at each input power. To understand the effects of inclination as well as the change of coolant flow rate in the condenser, the same procedure is followed at each fixed inclination angle and coolant flow rate.

4. Math Model

This paper is focused on the experimental works only. The performance and other mathematical analysis are not presented here. However, interested reader may referred to Reay et al.[11] and Valeri and Roger [12].

The thermal resistance, *R* (^{0}C/W) is defined as

$$R = \frac{T_e - T_c}{Q} \tag{1}$$

and the overall heat transfer coefficient, U (W/m^2 ^{0}C) is by

$$U = \frac{Q}{A_e(T_e - T_c)} \tag{2}$$

5. Results and Discussion

Experiment is carried out with MHP for different working fluids; methanol, ethanol and acetone at different inclination angles, $30^0 \leq \theta \leq 90^0$ for various heat input, $0.612\text{W} \leq Q \leq 8.71\text{W}$ and coolant flow rates, 0.3 lit/min ≤

$m_c \leq$ to 1.0 lit/min. To keep the volume of the paper minimum only representative curves (similar nature curves are not shown) are given. Fig. 3 shows the distribution of wall temperature along the length of MHP for different coolant flow rate where as Fig. 4 is drawn wall temperature profile for different heat input.. From this figure it is seen that in evaporator section temperature remains same, which indicates uniform heating. After that temperature drops in adiabatic section and in condenser section. Again it indicates coolant flow rate has a little effect on wall temperatures. From Fig. 4, it is seen that the nature of the curve in different section of MHP is similar as Fig. 3. Wall temperature is higher for higher heat input in each section of MHP. From Fig. 5, it is seen that effect of coolant flow rates is insignificant on thermal resistance. From Fig. 5 it is clear that thermal resistance exhibited a decreasing trend as the heat input is increased. It could be mentioned here, as the heat input is raised, heat transfer rate is increased due to the increase of vapor density; this is consistent with the result found by Jon H. B. [13] and Kim, K. S. [14]. Thermal resistance decreases slightly with inclination angle up to 50^0 at constant heat input. After that sharp decrease is visible up to 70^0 and then increases (Fig. 6). For this working fluid and heating condition MHP will work better at inclination angle 70^0. Again, coolant flow rate has a little effect on thermal resistance. Overall heat transfer coefficient is higher for high heat input (Fig. 7). Fig. 8 shows the effect of inclination angle on overall heat transfer coefficient. It is evident that overall heat transfer coefficient is maximum at $\theta = 70^0$ for MHP with ethanol as working fluid when Q = 3.67 W. Fig. 9 depicts the variation of overall heat transfer coefficient with heat input for MHP with ethanol as working fluid. The overall heat transfer coefficient is increased with the increase of heat input. Again this figure also shows overall heat transfer coefficient is better for inclination angle 70^0. To find the effect of working fluid, Fig.10 is plotted. This figure is plotted for inclination angle 70^0 and heat input is 3.67 W. From the figure it is quite clear that performance of MHP with acetone is better than MHP with methanol or ethanol.

An empirical equation is developed as described in Appendix-A for MHP with acetone as working fluid. The same relation can be developed for MHP with other two working fluids. The developed equation is

$$U/U_{\max} = 0.695\,(Q/Q_{\max})^{0.785}\,(m_c/m_{c\,\max})^{-0.114}\,(1+\sin\theta)^{.0634} \quad (3)$$

The relation is valid for

Q = 0.612 W to 8.71 W, m_c = 0.3 lit/ min to 1.0 lit/min and $\theta = 30^0$ to 90^0

The relation is developed for the specified range of different parameters used in this experiment. For this reason U_{max}, Q_{max} and m_{cmax} terms are used; which could be defined as the maximum values of the range of corresponding parameters. Further studies can be carried on to generalize this relation.

Then some of the experimental data are correlated with the developed equation and presented in Fig. 11. From this figure it is clear that the developed equation can correlate the experimental data within ± 7% error only.

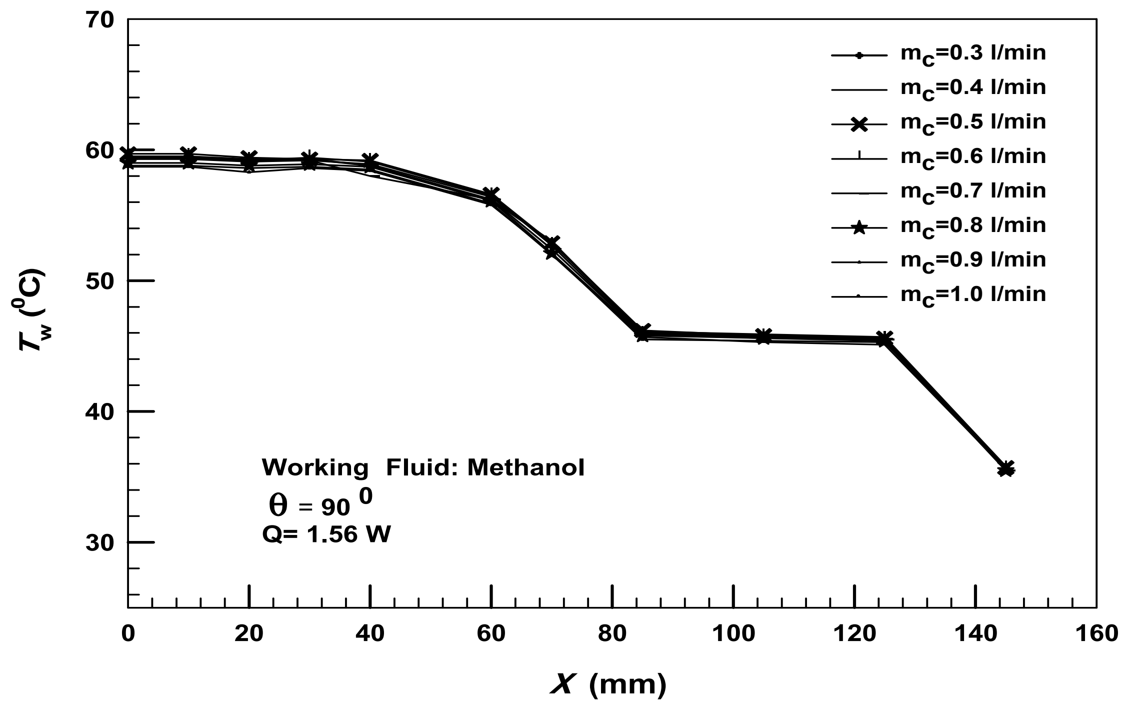

Figure 3: Wall temperature distribution along the length of MHP with methanol for various coolant flow rates when $\theta = 90^0$ and Q= 1.56 W.

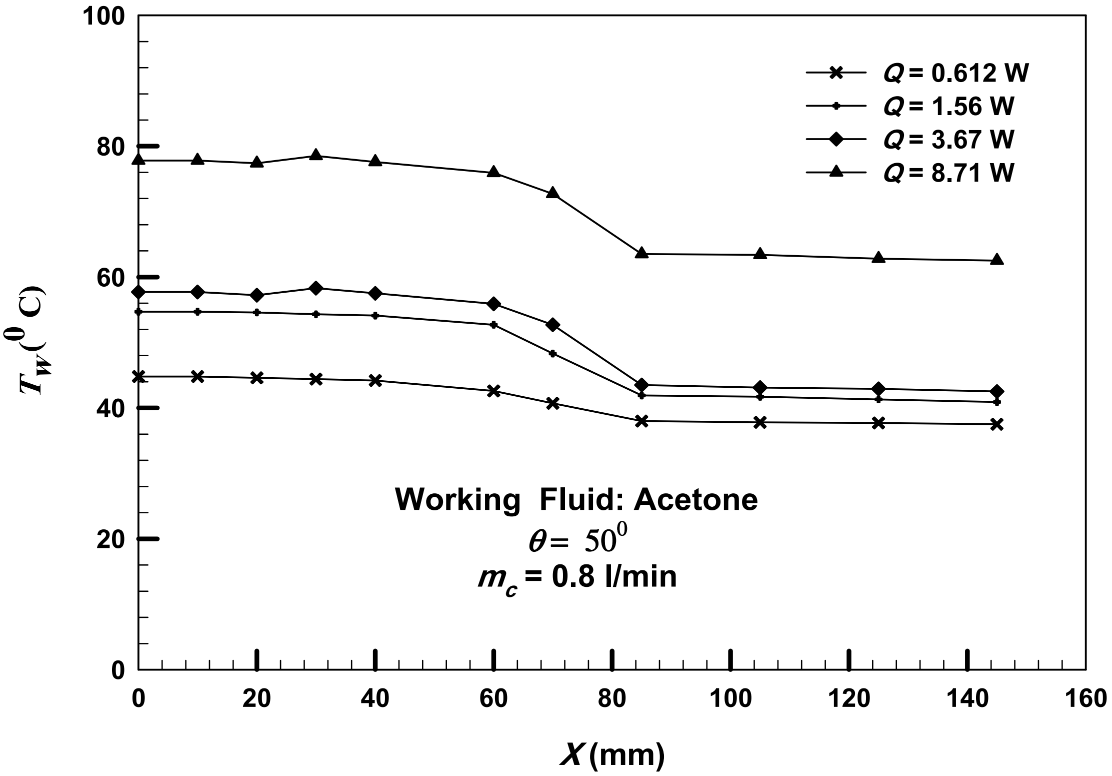

Figure 4: Wall temperature distribution along the length of MHP with acetone for various heat inputs when $\theta = 50^0$ and $m_c = 0.8$ lit/min.

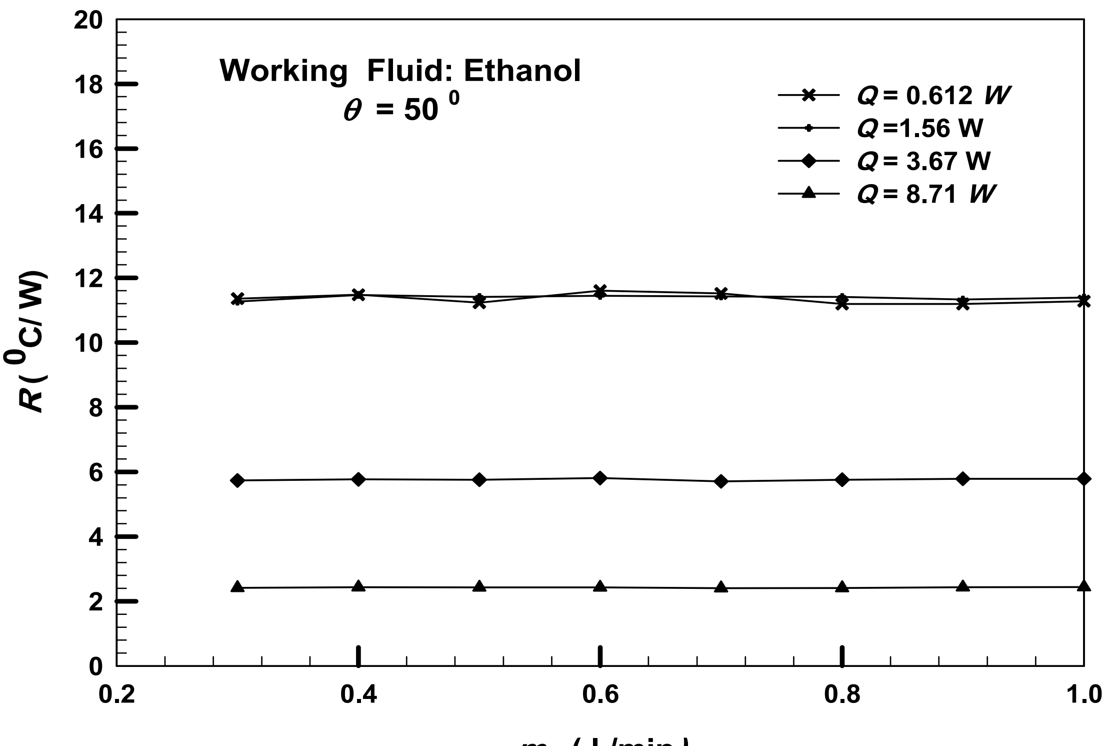

Figure 5: Effect of coolant flow rate on thermal resistance of MHP with ethanol for various heat inputs when $\theta = 50^0$.

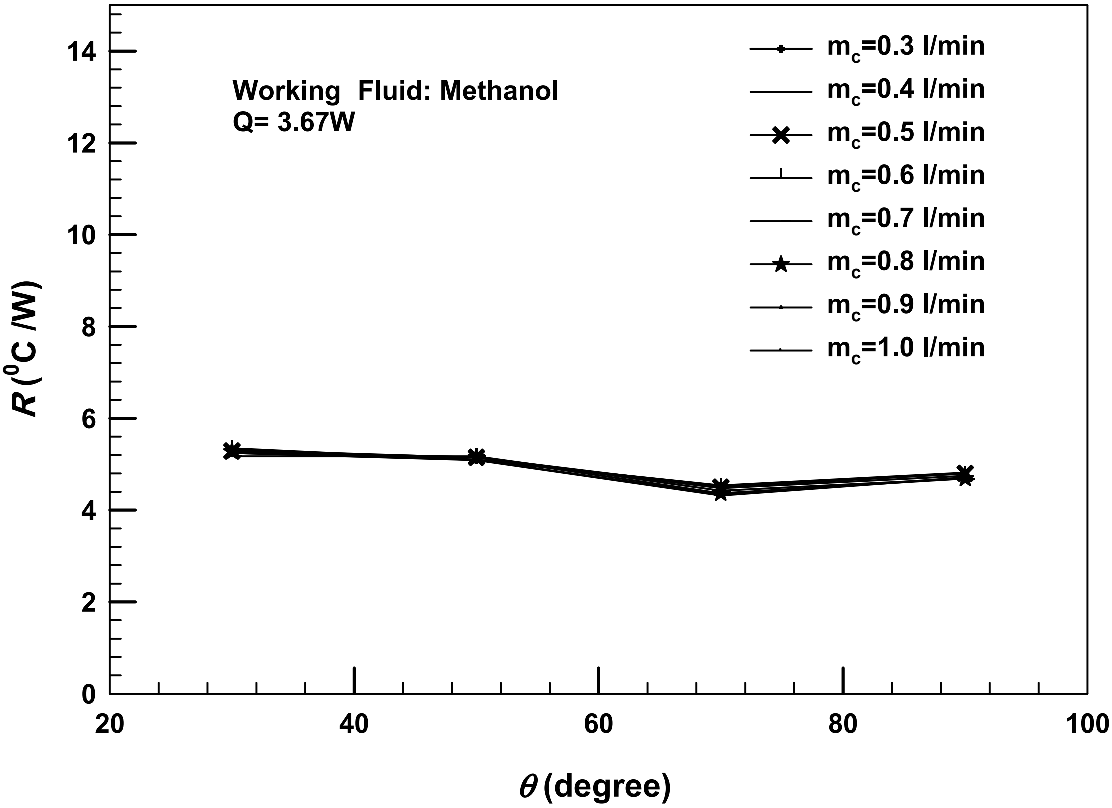

Figure 6: Effect of inclination angle on thermal resistance of MHP with methanol for various coolant flow rate when Q =3.67 W.

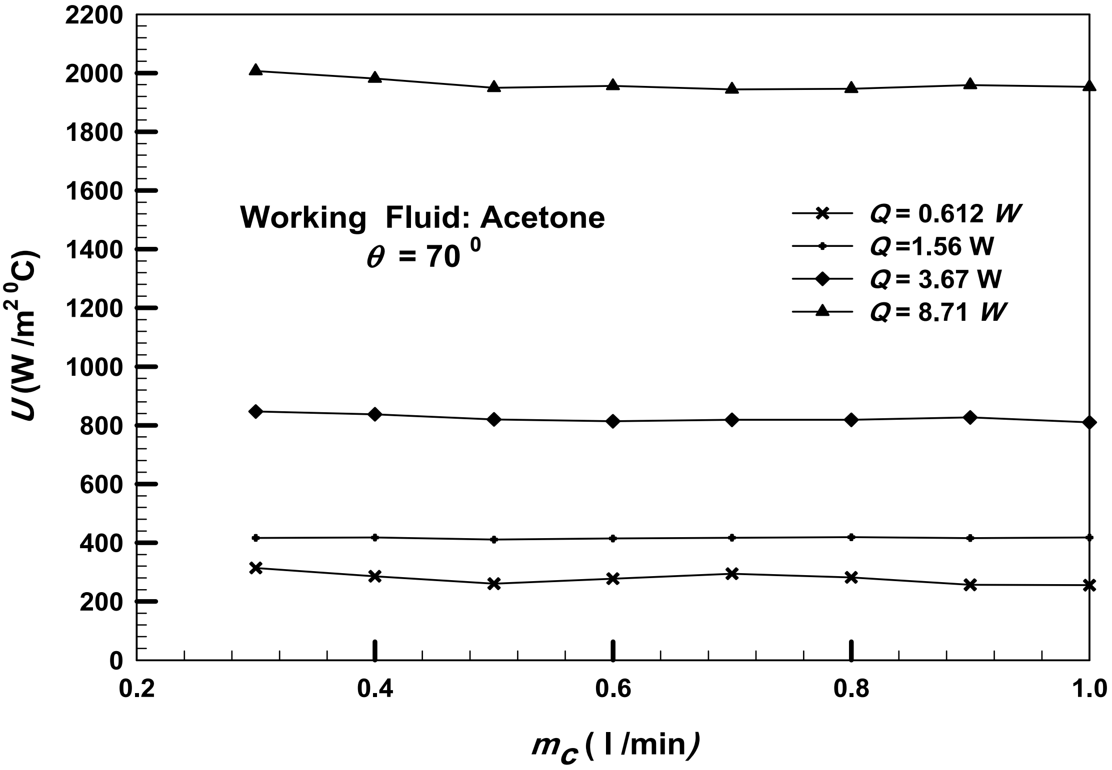

Figure 7: Effect of coolant flow rate on overall heat transfer coefficient of MHP with acetone for various heat inputs when θ =70^0.

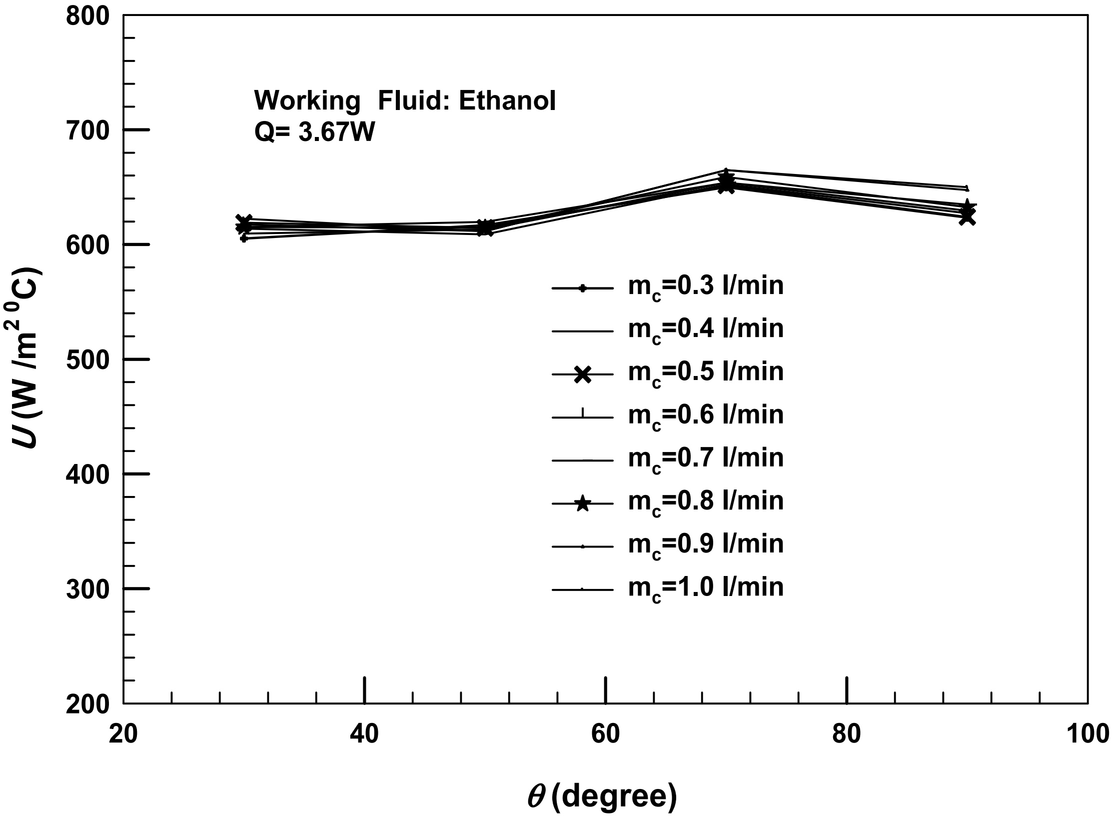

Figure 8: Effect of inclination angle on overall heat transfer coefficient of MHP with ethanol for various coolant flow rate when Q =3.67W.

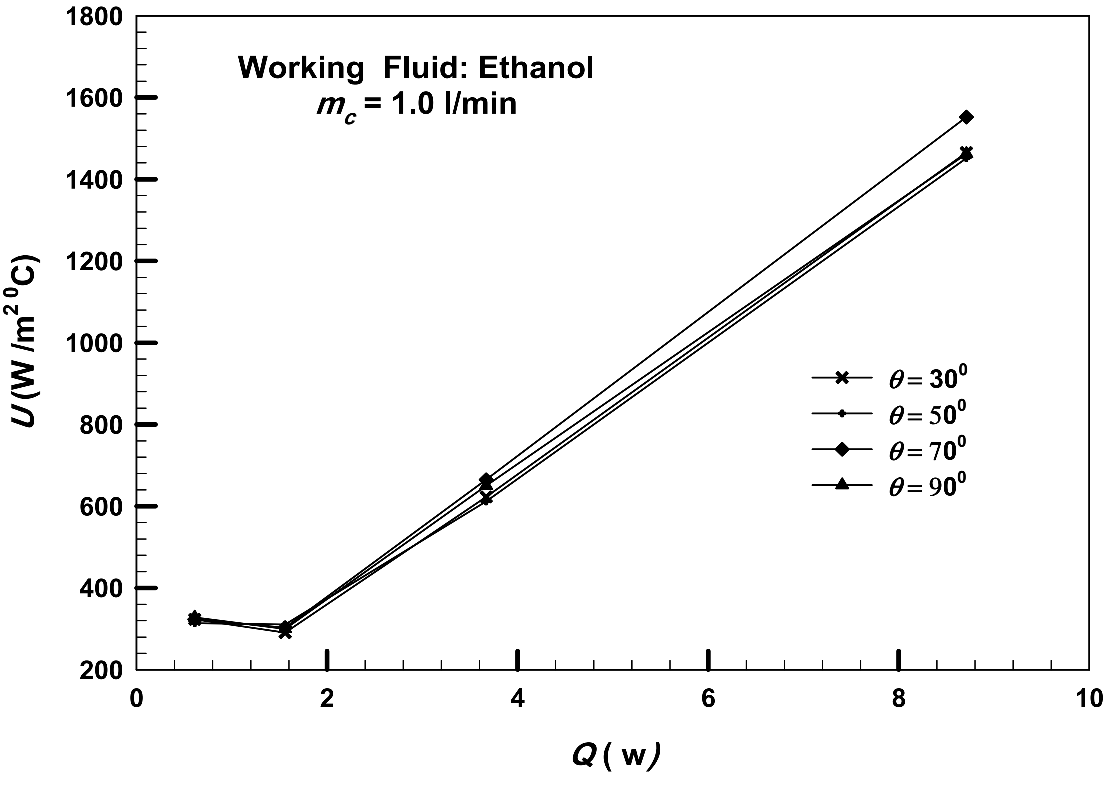

Figure 9: Effect of heat input on overall heat transfer coefficient of MHP with ethanol for various inclination angles when m_c = 1.0 lit/min.

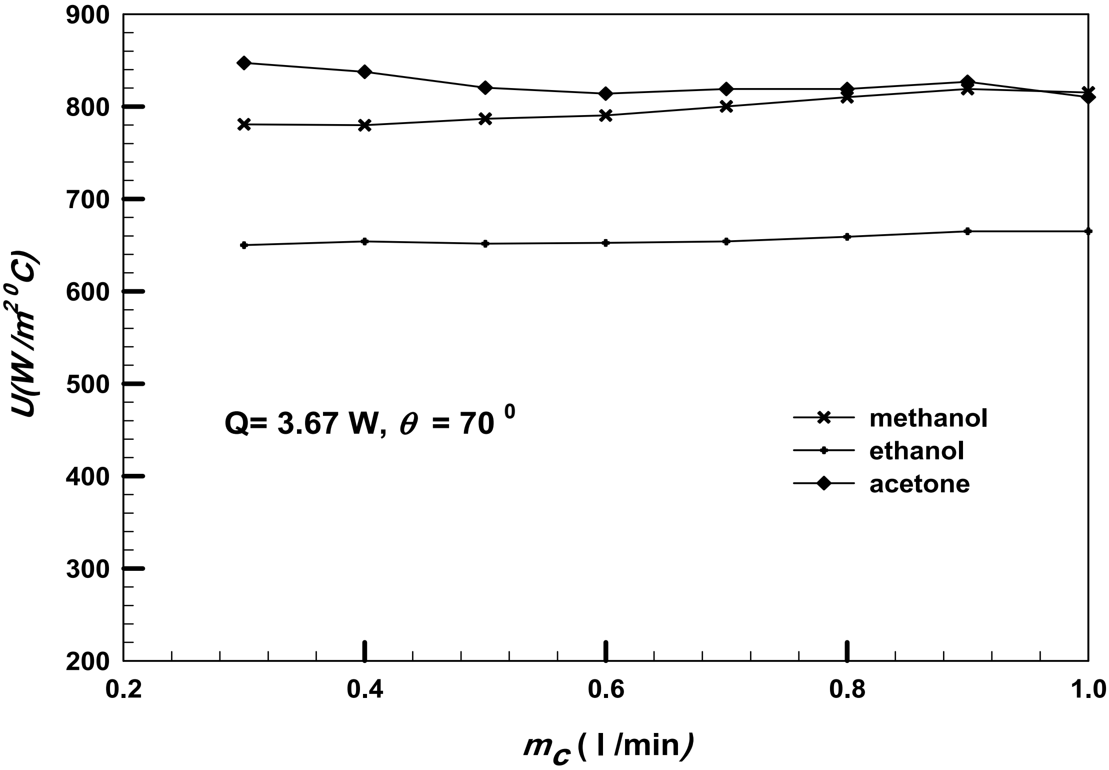

Figure 10: Variation of overall heat transfer coefficient of MHP with m_c for different working fluid when Q=3.67W and $\theta = 70^0$.

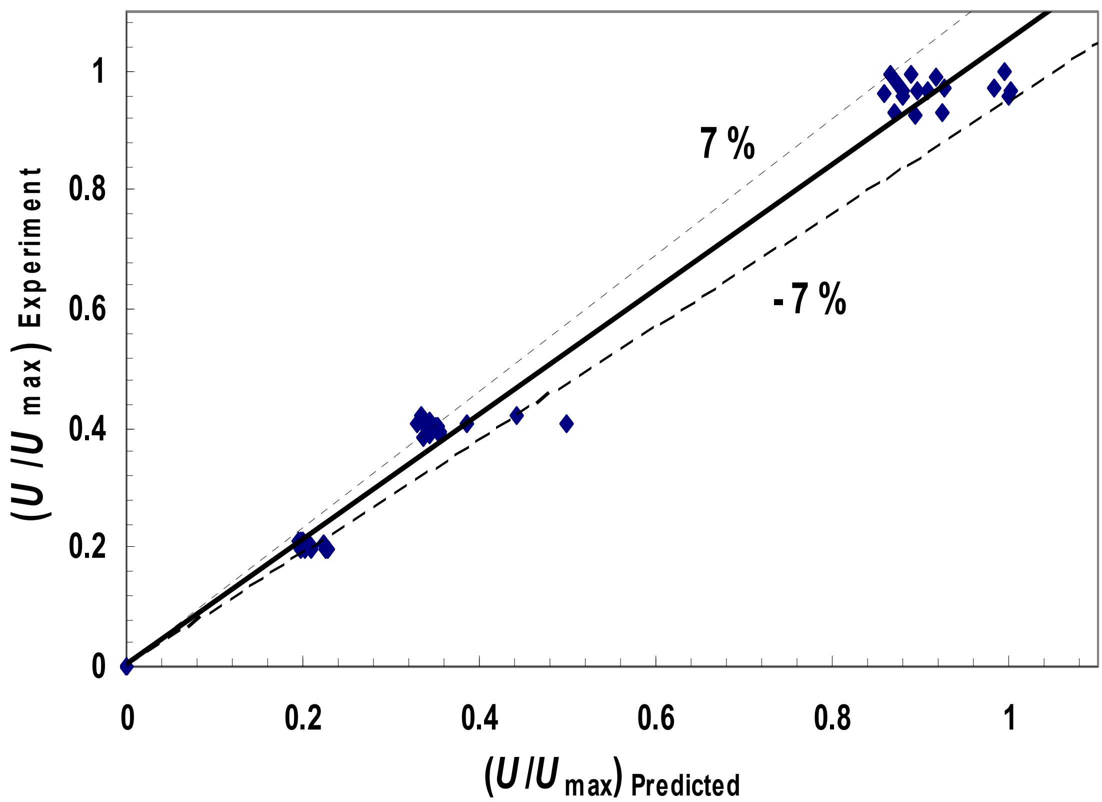

Figure 11: Variation of results obtained from empirical formula and those of from experiment.

6. Uncertainty Analysis

A precise method of estimating uncertainty in experimental results has been described by Kline and McClintock [15]. If the result R is a given function of the independent variables X_1, X_2, X_3,…………..X_n andt G_1,G_2,G_3,…………..G_n be the uncertainty in the independent variables given with the same odds.

Then the uncertainty in the result having this same odd is given in Eq. (4)

$$G_R = \left[\sum_{i=1}^{n} \left(\frac{\partial R}{\partial X_i} \times G_i \right)^2 \right] \qquad (4)$$

Using Eq. (4) and the uncertainty of primary measurands, the uncertainty of R and overall heat transfer coefficient, U are calculated as 9.11 % and 0.2 % respectively.

7. Conclusions

The experiment of investigating thermal performance of MHP is done by varying the angle of inclination, coolant flow rates, working fluids and heat inputs. From the results the following conclusion can be made.

- Coolant flow rate has an insignificant effect on the performance of MHP.
- Performance of MHP depends upon angle of inclination. Better performance is found for an inclination angle of 70^0.
- Heat input has significant effect on the performance of MHP. It is found that overall heat transfer coefficient is higher for higher heat input.
- It is observed that for the same heat input and inclination angle MHP with acetone as working fluid performs better.

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Appendix- A

Correlation Procedure

In order to develop an empirical formula, selection of experimental parameters is extremely important. In this study experimental parameters are heat input, angle of inclination, coolant flow rate and working fluid. Because Acetone shows better performance as a working fluid over methanol and ethanol and to avoid futher difficulties, only MHP with acetone as working fluid is considered for correlation purpose. So no term for working fluid is considered in the equation. Eq. (A.1) is assumed considering the above experimental parameters with some unknown co-efficient C, a, b and c:

$$\frac{U}{U_{\max}} = C\left(\frac{Q}{Q_{\max}}\right)^{a}\left(\frac{m_c}{m_{c\,\max}}\right)^{b}\left(1+\sin\theta\right)^{c} \tag{A.1}$$

This equation can be written as

$$\ln\frac{U}{U_{\max}} = \ln C + a\ln\left(\frac{Q}{Q_{\max}}\right) + b\ln\left(\frac{m_c}{m_{c\,\max}}\right) + c\ln\left(1+\sin\theta\right) \tag{A.2}$$

Eq. (A.2) is subjected to experimental data and has obtained many equations. Those equations are needed to solve simultaneously. They may be arranged in matrix form as given below:

$$\begin{bmatrix} A_i \\ A_{i+1} \\ A_{i+2} \\ A_{i+3} \end{bmatrix} = \begin{bmatrix} 1 & M_i & N_i & S_i \\ 1 & M_{i+1} & N_{i+1} & S_{i+1} \\ 1 & M_{i+2} & N_{i+2} & S_{i+2} \\ 1 & M_{i+3} & N_{i+3} & S_{i+3} \end{bmatrix} \begin{bmatrix} x \\ a \\ b \\ c \end{bmatrix} \tag{A.3}$$

where, $A = \ln\left(\frac{U}{U_{\max}}\right)$, $M = \ln\left(\frac{Q}{Q_{\max}}\right)$, $N = \ln\left(\frac{m_c}{m_{c\,\max}}\right)$,

$S = \ln\left(1+\sin\theta\right)$ and $x = \ln C$

Equation (A.3) is solved by Reduced Gaussian Elimination Method with the help of computer programming and coefficient 'C', 'a', 'b' and 'c' are determined. Computer programming language 'C' is used in this case.