# STUDY OF PRESSURE DROP AND HEAT TRANSFER CHARACTERISTICS OF MINI-CHANNEL HEAT SINKS

**Raed Shakir**
**raed-sh@utq.edu.iq**
**University of Thi-qar- College of engineering-Department of Petroleum and Gas engineering**

## ABSTRACT

Predictions on micro-scale of heat sinks beneath the conditions of heat flux was implemented to study the properties of heat transfer. A copper plate with dimensions of (50mm* 50mm) and (10 mm) in thickness, was subsequently developed as a square plate heat sinks along with 25 mini-channel. Each channel has (1 mm) height and (1 mm) width using a machine on wire of electrical evacuation. Effects of various heat sinks factors, such as the mini-scale of complex sinks and heat flow of heat transfer properties were considered. An iteration process by excel software was used to obtain the wall and the fluid temperatures, the Reynolds number at the inlet and outlet of the channels, the coefficient of heat transfer, and the pressure drop. The predictions were implemented for the Reynolds number and pressure drop in the ranges of (1014. 23–3272.56) and (1.90–11.66) mbar, respectively as well as the results for inlet and outlet Nusselt number ranges (4.27 - 5.75) and (3.92 - 4.92) respectively. The studied mechanism of heat transfer for micro-scale of complex geometry showed that the micro-scale arrangement has a substantial effect on the enhancement of heat flow of transmission parameters, so that the results of these investigations are expected to be used in the design of mini-heat exchangers used in electronic devices. Air data were obtained using suitable heat transfer equations which are very comparable to real lab tests under steady-state conditions.

**Keywords: Mini-Channel, Coefficient of Heat Transfer, Pressure Drop, Heat Flow Properties**

## NOMENCLATURE

| | |
|---|---|
| a | air |
| h | heat |
| $D_p$ | Pressure drop (mbar) |
| $H_c$ | height of channel (mm) |
| $K_c$ | copper thermal conductivity ($W/m^2.K$) |
| T | Temperature (K) |
| $T_a$ | Temperature of air (K) |
| $T_h$ | Temperature of heat (K) |

| | |
|---|---|
| $T_w$ | Temperature of wall (K) |
| $q_h$ | Heat flux ($W/m^2$) |
| V | The velocity of flow (m/sec) |
| W | The width of test piece (mm) |
| $W_c$ | The width of channel (mm) |
| $W_{cell}$ | The unit cell of base area ($mm^2$) |
| $W_s$ | The solid of channel (mm) |
| Z | The direction of flow |
| $\epsilon fin$ | Thermal efficiency (-) |
| $\omega_{ht}$ | Coefficient of heat transfer ($W/m^2.K$) |
| $\rho_{air}$ | Air density ($Kg/m^3$) |

## INTRODUCTION

There is a growing demand for higher performance and high reliable heat dissipation mechanisms for electronic devices, which are generally required. Traditional scaling of heat sinks through for a specific geometries on micro-channels had been employed by many manufacturers. Researches on using micro channels are required for better understanding of the associated heat transfer. The experimental results of the current work agree with the findings of the experimental study presented by (Naphon & Khonseur, 2009) for Reynolds number more than (800). It was noted that heat transfer character tics depend on the height, width, and length of the channel, laminar or turbulent flows as well as fully developed or developing flows; and the results are roughly work in line with the current study findings. The heat transfer properties of gaseous flows in a mini-channel of heat sink and a micro-tube heat sink was studied by (Hong & Asako, 2007)**;**(Chein & Chuang, 2007)**;** (Chen, 2007);(Hong et al., 2008). The flow heat achievement of mini-channel with/without thermoelectric case, by employing Nano-fluids as coolants, has been investigated by (Chein & Huang, 2004)**;** (Chein & Chen, 2005)**;** (Chein & Chuang, 2007); (Lineykin & Ben-Yaakov, 2007). (Jeng & Tzeng, 2007) empirically investigated the heat transfer characteristics and (pressure drop for a square array of pin-fins on rectangular shape channel.. (Lie et al., 2007) studied heat flow for FC-72 that employed to a mini of pin-fins for silicon chip. **(**Shakir, 2020) predicated a study concerning the enhancement of boiling heat transfer for the micro-channel of the heat sink by using R113 as coolants. (Shakir, 2021) numerically reported the R113 stream and heat transfer in an inline-pin-fins channel by using the iteration method via uniform heat flux on the boundary conditions**.** (Shakir, 2022) studied (Dp) and heat transmission of a mini-scale on a square shape. (Shakir, 2022) reported an investigation of boiling-phase heat transmission for mini-scale of complex geometry to electronic and electrical bundles as indicated above.
Several investigations were reported related to heat transmission of heat sink by a lot of mini-channel, However, there are still continuous fields to be covered to study heat

transmission on micro-channel of complex geometry. In this work, conduction heat transmission properties, wall temperature, fluid temperature, and fluid characteristics were investigated in details in presence of a micro-channel of complex geometry.

## THEORITICAL MODELING

Estimates of heat transfer from the flow parameters of are plotted as air temperature and wall temperatures at specific values of system pressure. More than 500 equations are used to predict properties of heat transmission, such as air transmission temperature, temperature on the heat sink. A prepared program written by Excel software to analyze data by iteration method. The specification states that only a single to allocation of heat flux can exist to solid-air interface. As seen from figure (1), the 2-Dimensional array, the main effect on the wall line; the first D array was perpendicular to the airflow, while the second D array was parallel to the airflow. The thermal conductivity is given by (Shakir, 2020) and (Shakir, 2021),

$$\delta^2 T/\delta y^2 + \delta^2\, T/\delta z^2 = 0 \qquad (1)$$

where (y) is perpendicular on airflow axis, ( T) is the temperature on the wall of copper. The heat transfer for equation (1) is calculated by (Shakir, 2020) and (Shakir, 2021):

$$T_{i,j} = \delta y^2\left(T_{i+1,j} + T_{i-1,j}\right) + \delta z^2\left(T_{i,j+1} + T_{i,j_1}\right)/2\ (\delta y^2 + \delta z^2) \qquad (2)$$

As noted in figure(1), the energy balances for each cell must be achieved, (Holman, 2012) as,

$$q_h W_{cell} = \omega_{ht}(T_w - T_a)\left[W_c + 2\eta_{fin} H_c\right] \qquad (3)$$

Where ($W_{cell}$) is a base area of a unit cell and is given by,

$$W_{cell} = W_C + W_S \qquad (4)$$

The symbol $\eta_{fin}$ is the fin efficiency, (Holman, 2012), which can be given by,

$$\eta_{fin} = \tanh(\epsilon_{fin} H_C)/\epsilon_{fin} H_C \qquad (5)$$

where $\epsilon_{fin}$ is the fin Parameter, (Holman, 2012) so that,

$$\epsilon_{fin} = \sqrt{2\omega_{ht}/K_C W_S} \qquad (6)$$

The value for $T_w$ at mini and micro-scale can be calculated by (Holman, 2012),

$$T_W = T_{th} - \frac{q_h\, L_{th}}{K_C} \qquad (7)$$

While $T_a$ at mini and micro-scale is calculated by (Holman, 2012),

$$\mathrm{T_a} = \mathrm{T_{in}} + \frac{\mathrm{q_h\,W\,Z}}{\mathrm{M_a C_{pa}}} \tag{8}$$

The calculation of Nusselt number can be calculated by(Holman, 2012),

$$\mathrm{NU_{x,3}} = \mathrm{NU_{x,4}} \frac{\mathrm{NU_{fd,3}}}{\mathrm{NU_{fd,4}}} \tag{9}$$

where$(\mathrm{NUx_{,4}})$, $(\mathrm{NUfd_{,3}})$,and are calculated by liquid properties respectively.

The Prandtl number ($P_r$) can be given by(Holman, 2012),

$$P_r = \frac{C_p\,\mu_a}{k_a} \tag{10}$$

İn addition, Reynolds number is calculated according to (Holman, 2012) as,

$$R_e = \rho_a V D_h / \mu_a \tag{11}$$

where$D_h$ is calculated by

$$D_h = A_h / P_h \tag{12}$$

For the mini-channel at the laminar bounary condition, the associated Reynold number (f laminar Re) is calculated based on (Holman, 2012) as,

$$f_{laminar} R_e = 24(1 - 1.3553 + 1.946\beta^2 - 1.7012\beta^3 + 0.95641\beta^4 - 0.2537\beta^5) \tag{13}$$

For the mini-channel at the turbulent boundary conditions, the associated Reynold number ($f_{\text{turbulent}}$ Re) is calculated based on (Holman, 2012) as,

$$f_{\text{turbulent}} = 0.0791\, Re^{-0.025} \tag{14}$$

## PERDICTION SETUP

### The Loop Test

A graph of the prediction set-up is presented in figure (2). The prediction of the test loop consists of a cooling of air flow loop and acquisition data system. Air is utilized as a working liquid. Loop of the wind tunnel is a rectangular air duct made of acrylic of (20 X 20) cm and a length of (150) cm. Theduct is isolated by Aero-flex compliant of (10 mm) thick on sheet of a metal. Air on the wind tunnel is discharged via air compressor to the wind tunnel, directed to the test segment and move to discharge to ambient. Air flow rate is measured by a flow-meter (0.3%) precision on whole scale. The pressure drop on the metering section was measured by a differential pressure sensor with an accuracy of (0.03%) of full scale. Tap of pressure is fixed at two locations on the wall, one for downstream and the other for upstream on test segment. A K-type thermo-couples, with accuracy of (0.3%) on full scale, is used to measure air temperature. So, the inlet and outlet temperatures for air are measured by (2) thermocouples via probes (1mm), the diameter on the probes respectively extending into the channel via which the air flows. All thermocouple probes are pre-calibrated with a dry box temperature for calibrator by (0.03°C).

## The Mini-Channel of Test Section

The drawing scheme of test segment is illustraeted by figure(3). It's obvious that the width and length of heat sink are 50 mm and 50 mm respectively. The mini-channel of heat sink has supplemented via a wire of electric evacuation machine. Mini-scale of complex geometry has (25) open channels and (25) soild channels. AC power has the power supply on plate kind so the back surface on test segment was isolated by standard (15 mm) solid resistant heat of micaboard, criterion (10 mm) of Aero-flex board, and standard (10 mm) thick of acrylic board. Two T-type thermocouples of a diameter of (1 mm) were employed to measure of temperature of wall for micro-scale. The wall is drilled to insert the thermocouples on back face and fixed by using a distinct adhesive employed on wall back face.

## The Conditions of Procedure

Predictions had conducted by various heat fluxes values and by various flow rates values of air entering to test section. On the predictions, air flow rate was increased on a slight basis so the heat input to mini-scale was kept stable on heat sink and input heat to mini-scale of walls on complex geometry had set to attain required scale via employing an electrical heater. The provided energy had set on a scale current and voltage had provided to the heaters. The provided voltage and current to the heaters had scaled via digital clamp meter . A sensible of heat on steady state was obtained via air flow shall be found by energy equilibrium. In this investigation, At each temperature location as well (Dp) across to test segment had listed many times. Data was collected and executed by empolying the acquisition data system. The uncertainty of the used devices are: such as airflow meter (±0.03); the temperature of the thermocouple (±0.01); as well as the differential of a pressure transducer (±0.04).... etc.

## RESULTS AND DISCUSSION

The iteration method models by multiple flow rates inlets, and heat applies that suggested in this work are novel, comparable models to be employed as reference criteria are presently unavailable in a literature review. The validation of the simulation had been done by implementing a lot of heat transfer equations as well as energy balance equations for the heat sink types. The temperature of the wall as well as the temperature of the fluid at the inlet and at outlet locations are calculated by employing the equations of energy balance and equations of heat transfer that given in this work. figure (4) shows the variation of the cofficient of heat transfer for inlet and for outlet postions versus Reynolds number of air for (1 mm) in width and (1 mm) in hight. As air flow rate raises this lead to bigger rate of heat transfer applied. So, the increase of the rate of heat transfere is lower than that of air mass flow rate. Cofficient of heat transfer tends to increase as the air mass flow rate increaseses. By increasing the rate of heat transfer due to surface roughness and bigger surface area of heat sink, the cofficient of heat transfer of a channel with (M= 1.5 g/sec) are higher than those of (M=0.5, 0.75 ; and 1 g/sec) as well as the coefficient of heat-transfer on the inlet as well at outlet locations on channel are observed to increase with Re increasing with outlet value importantly smaller than an inlet value for all prediction data. The values of data that concluded for a constant heat flux are inclusived. The influences of the conduction of wall are noted to importantly affect the consequences. It is importantly to noted that the present study has relied on equations of heat transfer to predict all data. Figure (5) presents the variation of the Nu versus Re of air on (1 mm) width-channel; (1 mm) height. Accordingly, all test were predicted, a rate of heat transfere depends on rate to cooling capacity of air. So, the Nu increases as the (Re) of air increases due to greater heat transfer area as well as greater surface roughness, coefficient of heat transfer on heat surface for (M=1.5 g/sec) is greater than that corresponding when (M=0.5; 0.75 and 1 g/sec), as noted from figure(5). Because of the bigger superficies of the area of heat transfer, so bigger mass flow on heat sink makes Nu bigger than that corresponding for other mass flow. Figure(6) shows the variation of the coefficient of heat transfer at inlet and at outlet locations against to temperature of fluid for a square channel. The cofficient of heat transfer at outlet tends to increase as the temperature of fluid increases as well as cofficient of heat transfer at inlet tends to decrease as the temperature of fluid increases due to the dependency of the wall temperature on the temperature of the fluid, which means the relationship was proportional between each other. So, this parameter has a slight influence on the heat sink temperature with (M=1.5 g/sec) being bigger than those of (M=0.5, 0.75 and 1 g/sec). It is significant to note that the present study has relied on heat flow as well as mass flow to guess all data on the prediction process. Figure(7) shows the difference of pressure drop against fluid temperature on inlet and on outlet locations for mini-scale of heat sink. As compare to the traditional scale. The property of flow over on mini-scale has fully high and complicated due to the bigger surface roughness and bigger surface force so the expressions of mean surface roughness. Generally, the surface roughness has performed for the traditional tubes. As well as mini-scale of heat sink, so these data still demand and can be investigated. In addition, a style and a magnitude of roughness of mini-scale superficies had important influence on differences of pressure drop as seen in Figure(7). in addition, the pressure drop is reasonably constant in first of (3) points region data, but increases rapidly on flow developing to the end of flow data region. A pressure drop was measured between an inlet and outlet locations and represented the contributions via the pressure drop due to the flow of the mini-channels as well as an inlet loss and an exit loss. So, (Dp) data results presented in Figure(8), an inlet loss as well an outlet loss component had deducted via scaled data accepted to method of the equation. So, the pressure drop at inlet and at outlet locations was increased as the Re increases. In addition, pressure drop at outlet data is higher than at inlet data that due to the temperatures at outlet locations bigger that at inlet locations. The intended data consequences via this method for airflow at the (4) air flow rates was included on Figure(8) It has noted that airflow can be seen uniform divided among of mini-scales of complex geometry as well on (Dp) that crossed for any single channel represented to total (Dp). So, an intended data results are approved to be better than theother reviewed data measurements. Figure(9) shows the variation of the temperature of thermocouples with Re.As expected, the temperature of thermocouples has decreased to growing (Re). Due to (Re) on inlet as well on outlet positions depends to temperature of airflow as indicated by the the temperature at the outlet location which was higher than the temperature at the inlet location. It was important to highlight the fact that thermocouples are very close to a heat source in this study , but it can be chosen for any location on the heat sink. At last, the data obtained good consequences by employing a smart iteration method.

## CONCLUSIONS

The characteristics of heat transfer in the mini-channel of the heat sink are studied as predictions only in this work. The main conclusions that can be drawn from the results of this work can be listed as:

- The flow was laminar and turbulent for all prediction data tests.
- The flow was developing for all prediction data tests.
- For all guess data tests coefficient of heat transfer at inlet significantly higher than at outlet location due to the influences of wall conduction are seen to significantly influence the consequences.
- For all guess data tests pressure drop at outlet higher than at inlet location due to high wall and fluid tempertures at outlet.

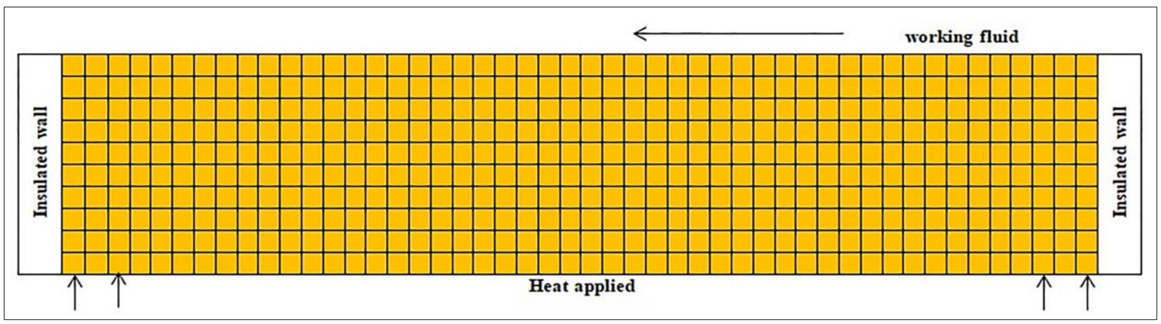



Fig.1. Wall Conduction of copper test Piece (Shakir, 2020); (Shakir, 2021).

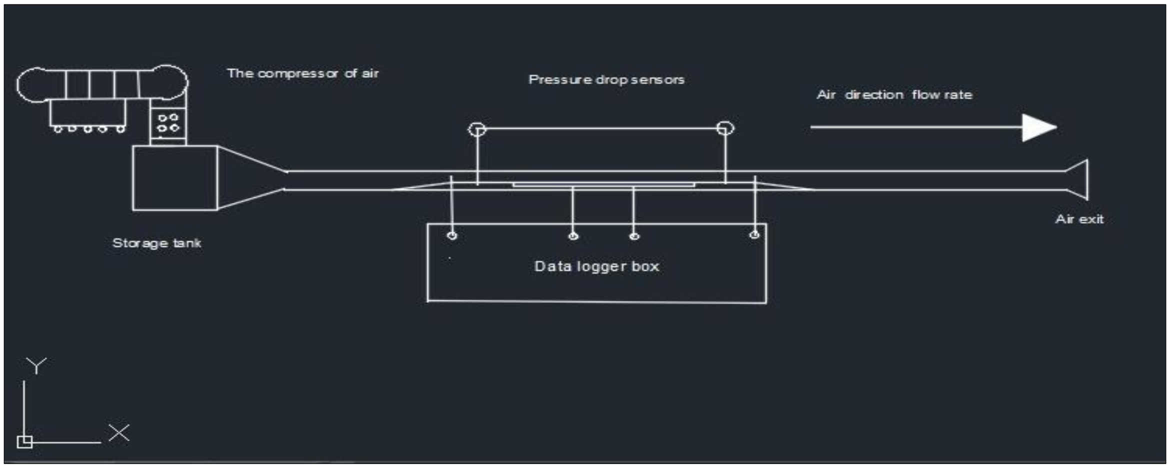



Fig.2. Prediction Rig

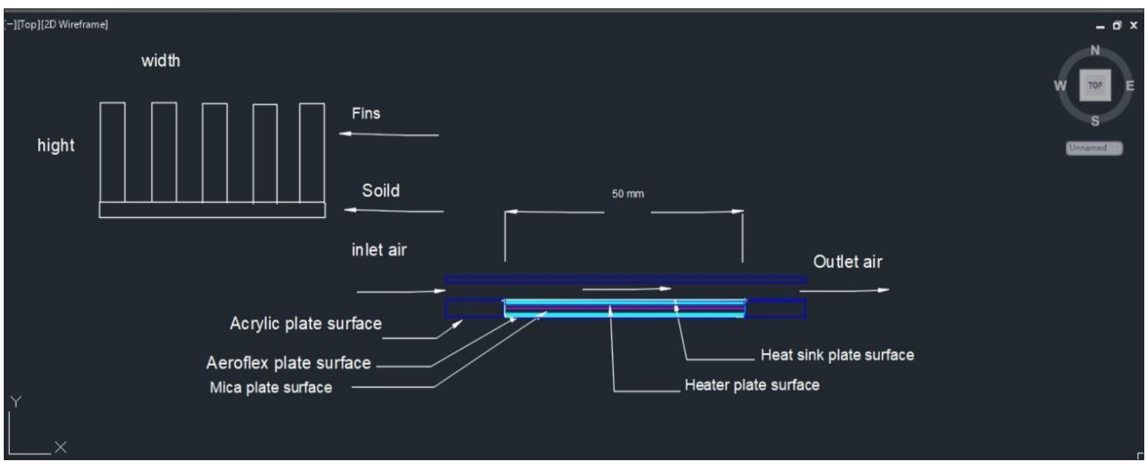
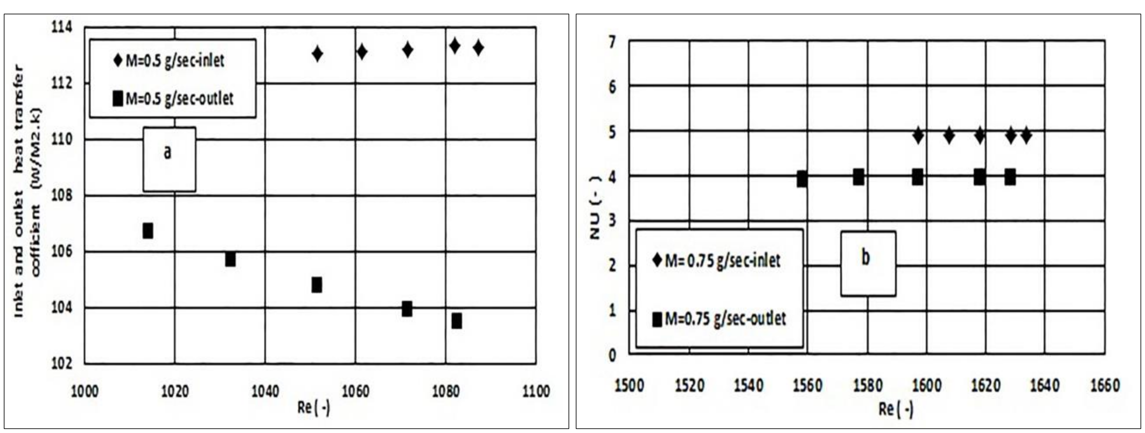



Fig.3. Test section& test piece



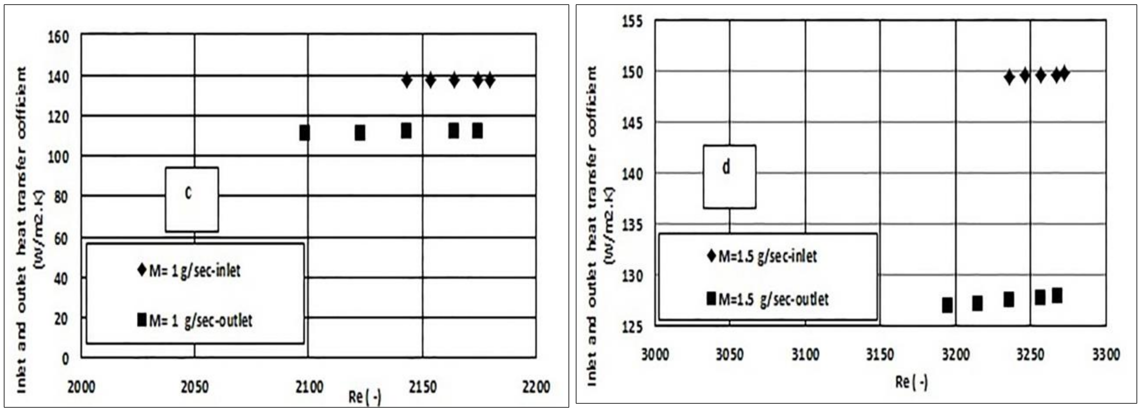



Fig.4. Alteration to coefficient of heat transfer versus (Re) for air
(a) M=0.5 g/sec (b) M=0.75 g/sec (c) M= 1 g/sec (d) M=1.5 g/sec

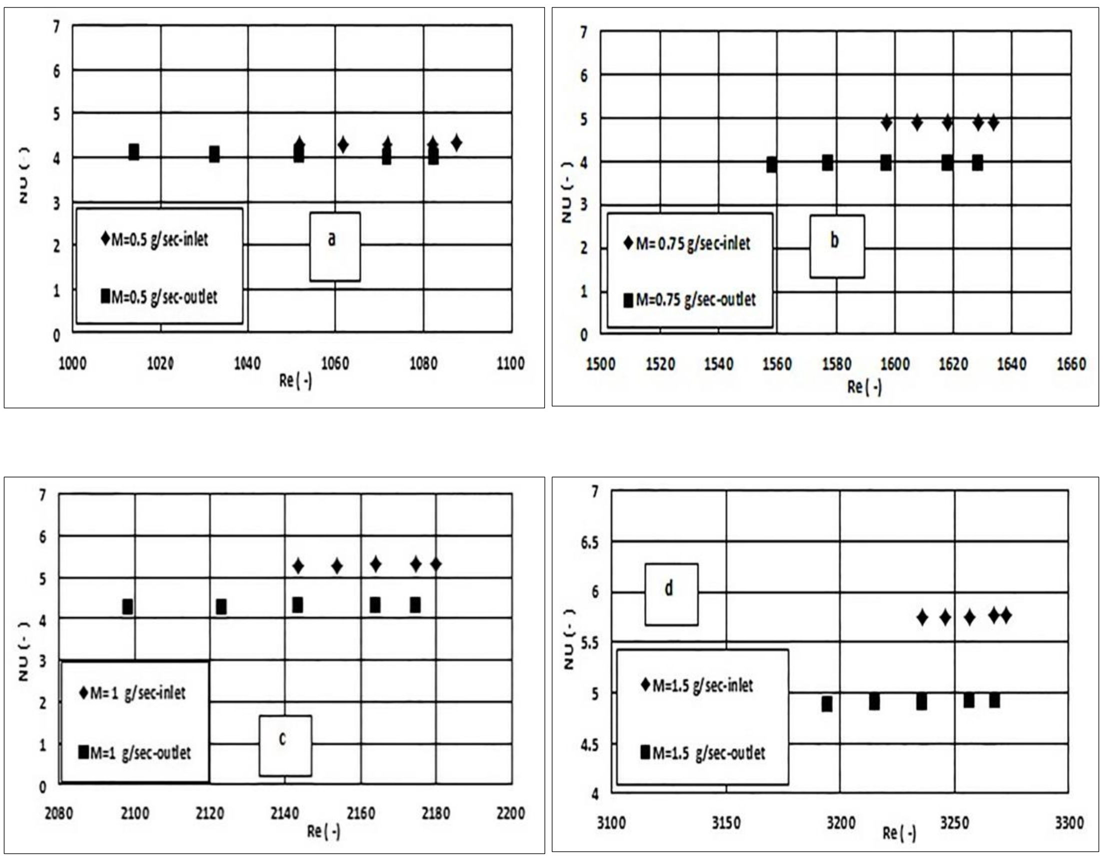



Fig.5. Variation to (Nu) with (Re) to (a)M=0.5 g/sec (b) M=0.75 g/sec (c) M=1 g/sec (d) M=1.5 g/sec

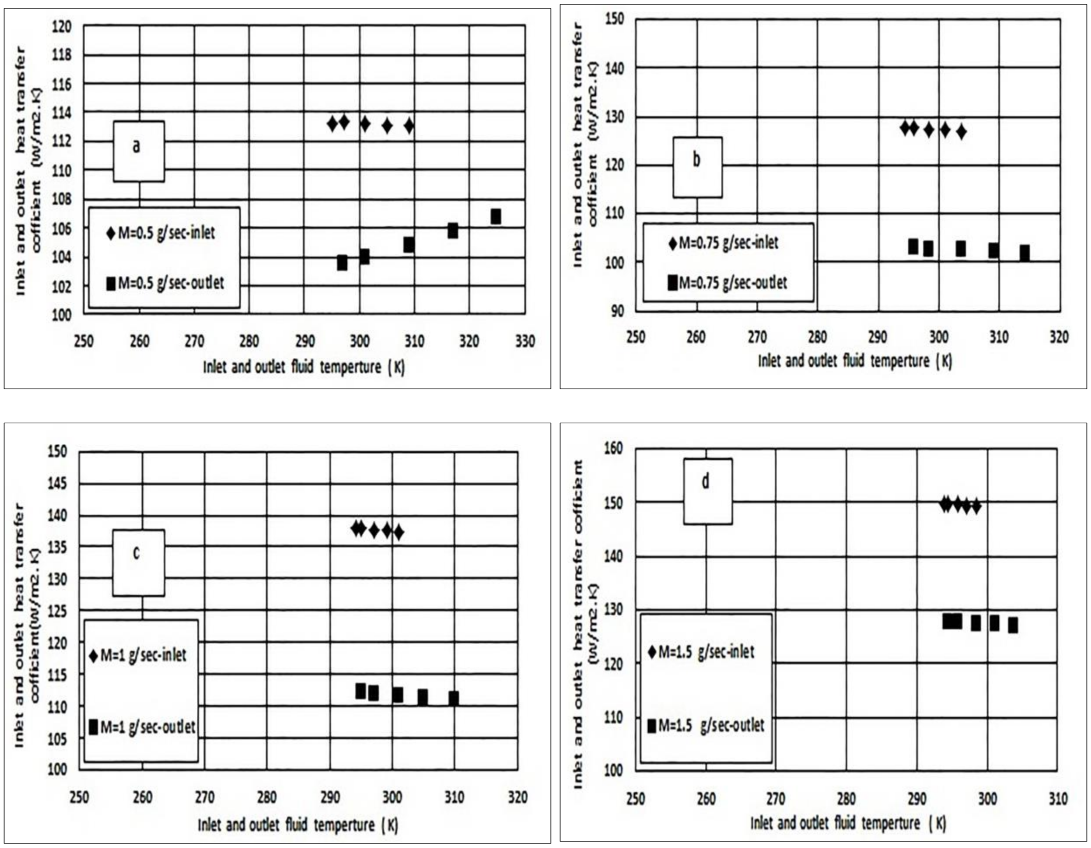
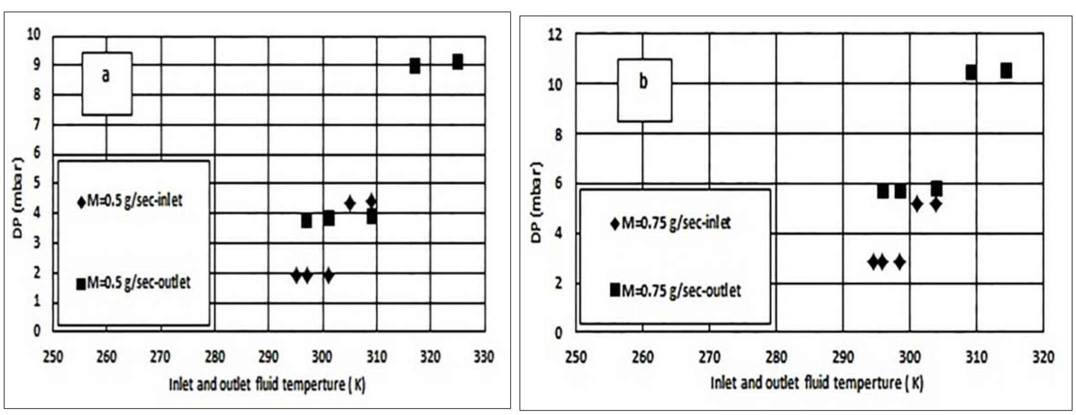



Fig.6. Alteration to cofficient of heat transmission versus fluid temperture for (a) M=0.5 g/sec (b) M=0.75 g/sec (c) M=1 g/sec (d) M=1.5 g/sec



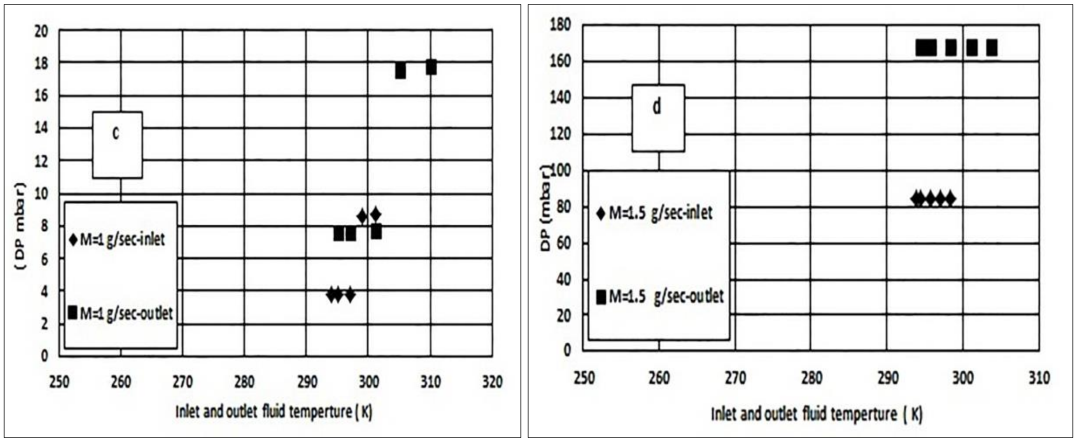



Fig.7.Alteration to (Dp) versus fluid temperture on (a)M=0.5 g/sec(b) M=0.75 g/sec (c) M=1 g/sec (d) M=1.5 g/sec

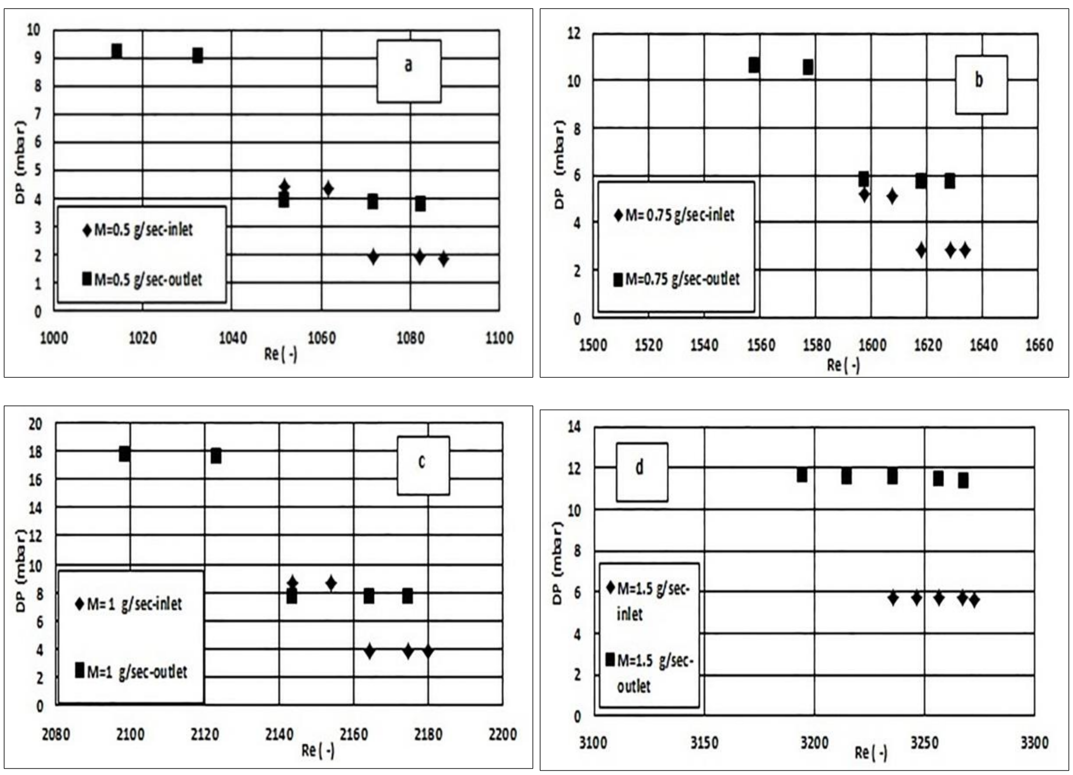



Fig.8. Alteration to (Dp) versus (Re) for (a)M=0.5 g/sec (b) M=0.75 g/sec (c) M=1 g/sec (d) M=1.5 g/sec

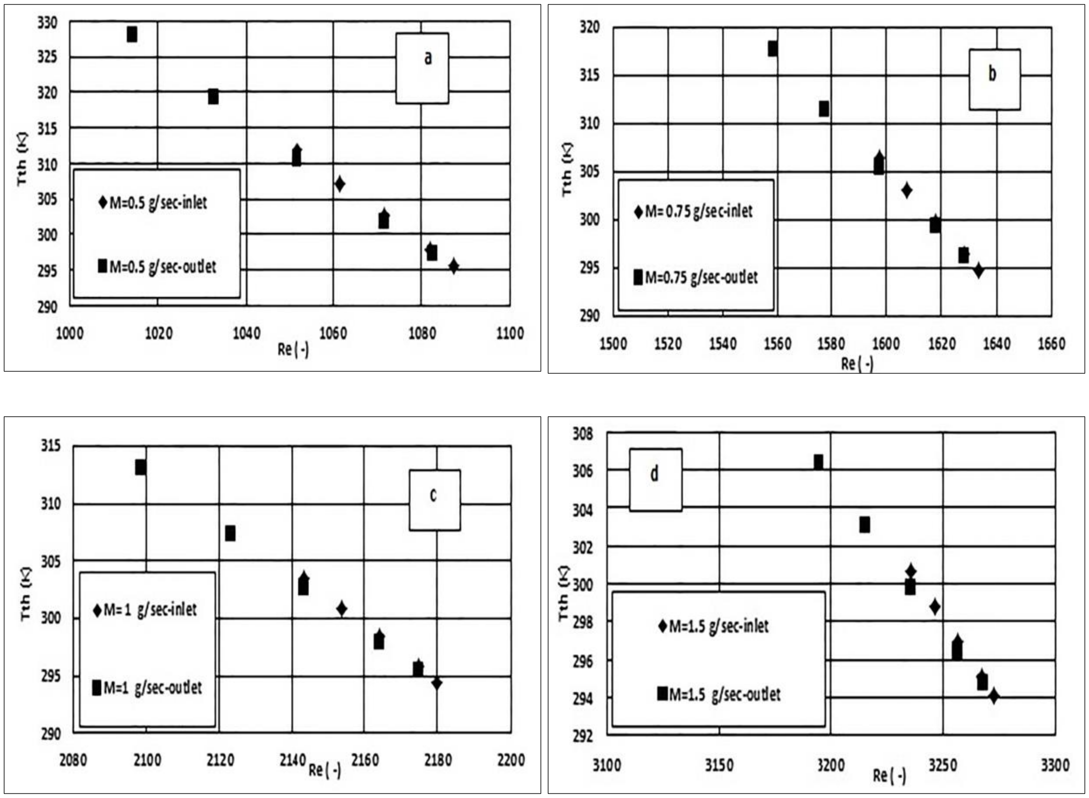



Fig.9**.**Variation to Thermmocouple temperture with air Reynolds number for (a)M=0.5 g/sec (b) M=0.75 g/sec (c) M=1 g/sec (d) M=1.5 g/sec

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