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Testing of experimental set up for determination of heat transfer coefficient of working fluids

Sonage B.K* and Mohanan.P

Department of Mechanical Engineering, National Institute of Technology Karnataka, surthakal, Mangalore, Karnataka-575025 India.
*Corresponding author - Phone: 09860511099, E-mail:sbasvaraj@rediffmail.com

***Abstract*-Alternative working fluids with enhanced thermo physical properties are to be developed to meet future challenges in the field of energy conservation and saving. Heat transfer coefficient and pressure drop characteristics plays a vital role in development of new coolants. To study heat transfer and pressure drop performance of different working fluids a test set up is required. Development and testing of set up required for determining forced convective heat transfer and pressure drop at constant wall temperature in turbulent flow regime is presented in this paper. 100 observations are taken and results are compared with well known equations to assess performance of developed set up. Maximum error of 5.33% in mass flow rate measurement, 60% in heat transfer coefficient determination and 8.33% in pressure drop measurement is observed when tested with water. Developed set up can be used for assessment of heat transfer and fluid flow performance of different working fluids with error correction.**

***Keywords*- heat transfer coefficient, pressure drop, experimental setup, turbulent flow, constant wall temperature.**

Nomenclature

Re	Reynolds number
Nu	Nusselt number
Pr	Prandtl number
h	Heat transfer coefficient, W/m^2K
K	Thermal conductivity, W/mK
Cp	Specific heat, J/Kg K
f	Friction factor
L	Length, mm
V	Velocity, m/s
A	Area, m^2
ΔT_m	Logarithmic mean temperature difference

Greek symbols

μ	Dynamic viscosity, Kg/ms
ρ	Density, Kg/m^3
ε	Relative roughness

Subscript

w	wall
i	inlet
0	outlets

1. Introduction

Use of working fluid is integral part of heat exchange process. High speed electronic and computer applications have high power density to be dissipated to sink. In such applications one major factor, in heat transfer dissipation and enhancement, is the poor thermal characteristics of heat transfer fluids. If lower specific heat fluids are to be used, requirement of mass flow rate will be more which increases pumping power and inventory of working fluid. High viscous fluid increases thermal dissipation and thereby increasing required pumping power. This created need to develop alternative working fluids. In most of heat exchangers turbulent fluid flow with heat transfer encounters. There is no theoretical method available to calculate forced convective heat transfer coefficient of different working fluids for turbulent fluid flow, while one should rely on empirical correlations or experimental work. To study different aspects of convective heat transfer coefficient an experimental set up is required to determine heat transfer coefficient and pressure drop of different working fluids. The experimental set up has been developed to study heat transfer and pressure drop characteristics of different working fluids which is presented in this paper. Following experimental studies have been assumed as basis for development of present experimental set up.

Darcy–Weisbach equation, Colebrook equation, Haaland explicit equation are famous for calculating friction factor for fluid flow trough smooth and rough pipes, While Avci and Karagoz[1] have introduced a

novel formulation for friction factor which is presented in equation (1)

$$f = \frac{6.4}{\{\ln(Re) - \ln[1 + 0.01 Re\epsilon(1 + 10\sqrt{\epsilon})]\}^{2.4}} \quad \text{(Eq.1)}$$

Lazarus Godson et al [2]. have presented a experimental set up which consists of tube in tube type of heat exchanger to study convective heat transfer coefficient under laminar and turbulent flow conditions. They used logarithmic mean temperature difference method to determine h on working fluid side.

LiDong Huang et al[3] have developed a test set up to study convection through vertical tube at high Reynolds number.

M.N. Pantzali et al.[4] performed convective heat transfer experiments on nanofluid with set up consisting of miniature plate heat exchanger with modulated surface

B. Farajollahi et al.[5] have studied heat transfer to nanofluid with shell and tube type of heat exchanger.

M. Haghshenas Fard et al [6] have developed a heat transfer loop to study two phase and single phase flow of nanofluid at constant wall temperature.

M. Chandrasekar et al [7] have highlighted importance of calming section and riser section in their study related with heat transfer and friction factor in circular pipe with wire coil inserts to nanofluid.

Yurong He et al. [8] have performed convective heat transfer experiments on set up with straight circular pipe at constant heat flux condition for both forms of heat transfer i.e heating and cooling of nanofluid.

K.B. Anoop et al. [9] have developed experimental set up and investigated convective heat transfer in nanofluid in the developing region at constant heat flux condition.

2. Working of experimental setup

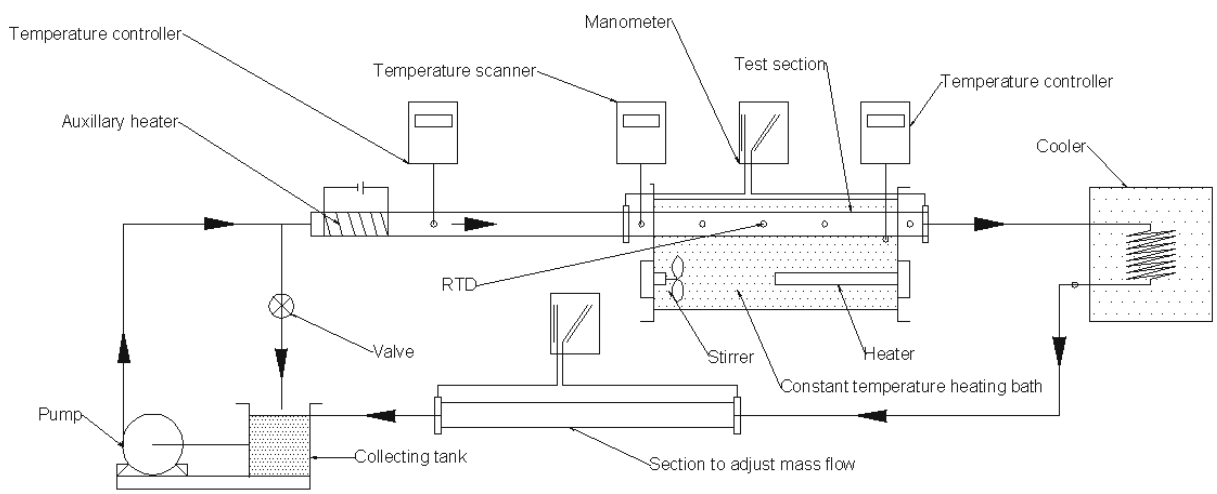

Figure 1. Experimental set up to determine heat transfer coefficient and pressure drop

An experimental setup, as shown in figure 1, is developed to determine inner surface heat transfer coefficient at constant wall temperature condition. The experimental set up consists of test section, constant temperature heating bath, cooler, collecting tank, pump and auxillary heater etc. Test section is made up of stainless steel tube with inner diameter 4.93mm and outer diameter of 6.23mm and length 0.5m. A constant temperature heating bath is made up of stainless steel and insulated from outer side with fiber glass insulation. A heater is mounted inside the tank to heat the tank fluid i.e. water. Stirrer is mounted inside the tank to stir the tank fluid continuously to avoid built up of temperature gradient inside the tank fluid. A cooler is used to cool the hot fluid coming out of the test section. A centrifugal pump is used to circulate the fluid in the

heat transfer loop. Auxiliary heater is wound around the tube to heat the fluid flowing through the tube to inlet temperature.

A fluid is pumped through test section mounted in a constant temperature heating bath via auxiliary heater and delivered to collecting tank via cooler. In auxiliary heater fluid is heated to inlet temperature which is maintained constant by controlling power to heater by temperature controller by taking feedback from Resistance Temperature Detector (RTD) mounted at outlet section of heater. Test section is mounted in a constant temperature heating bath in such a way that it will get completely

immersed in a bath fluid. The temperature of the bath is maintained constant by another temperature controller. Thus constant wall temperature condition of heat transfer is achieved by maintaining the temperature of the fluid around the test section constant. A heated fluid coming out of the test section is cooled in a cooler to recirculate the same fluid. The cooled fluid is discharged to collecting tank.

Four RTDs are mounted on the outer surface of the test section, two RTDs are mounted at inlet and outlet of test section respectively, and another two RTDs are mounted at inlet and outlet of the cooler to measure the temperature at the respective locations. Inverted U tube water filled manometer is mounted to measure pressure drop across the test section. A mass flow rate of the fluid is adjusted and measured by adjusting the pressure drop across the section used for the same.

3. Results and Discussion

Observations are noted at steady state condition for inlet temperature range 20-40°C, surface temperature range 50-80°C and Reynolds number range 2000 to 15000 for water. The data obtained from experimental work is reduced to calculate heat transfer coefficient and pressure drop. As theoretical methods to determine 'h' for turbulent flow is not available one should rely on experimental work. For determining heat transfer coefficient experimentally mass flow rate, temperature at inlet, outlet and surface is required.

Mass flow rate measurement

Mass flow rate is measured by weighing the collected fluid in a flask in a particular time. Three such measurements are taken for single mass flow rate and average mass flow rate is worked out. Also mass flow rate is calculated from pressure drop across the test section by *Blasius correlation* [10] *and Darcy's friction factor formula* [11] for confirmation. Friction factor is calculated in terms of Reynolds number of fluid flow by *Blasius correlation* (equation2)

$$f = 0.3164Re^{-0.25} \quad \text{(Eq.2)}$$

With observed value of pressure drop and friction factor, Reynolds number is calculated by Darcy's friction factor formula (Equation3) from

$$\Delta P = \frac{fLV^2}{2gD} \quad \text{(Eq.3)}$$

Velocity and mass flow rate of fluid flow is calculated from Reynolds number (Equation4)

$$Re = \frac{\rho V D}{\mu} \quad \text{(Eq.4)}$$

All the thermophysical properties required for above calculations are taken at bulk mean temperature. A graph of error in measured value and calculated value is as shown in figure 2. The graph indicated the calculated value of m deviates from measured value by 5.66%. The deviation remains constant for whole range of measurement.

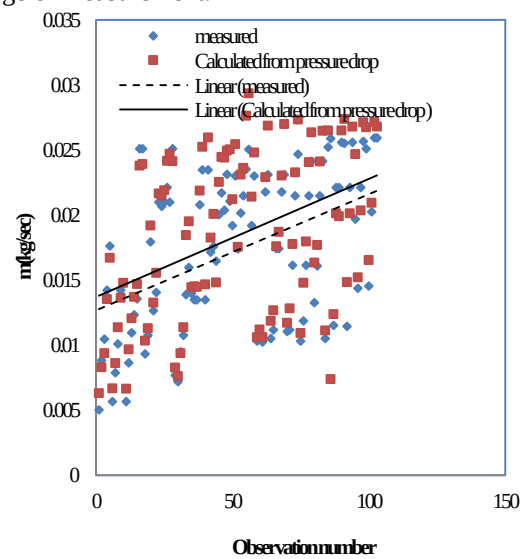

Figure 2. Graph indicating error in measured and calculated mass flow rate

Heat transfer coefficient calculation

Heat transfer coefficient is calculated at constant wall temperature condition and steady state condition. With measured value of m and temperatures, 'h' is calculated by energy equation (Equation 5) assuming the temperature of the fluid flowing through the test section vary logarithmically

$$mC_p(T_o - T_i) = hA_s(\Delta T_m) \quad \text{(Eq.5)}$$

The variation of h as per Reynolds number is plotted and shown in figure 3. Thermophysical properties of working fluid depends on temperature of fluid and h is function of ρ, v, D, μ, C_p, K hence h varies as per velocity and temperature. As velocity increases Reynolds number increase so h increases, which is observed in figure 3. As theoretical methods to determine 'h' for turbulent flow is not available one should rely on empirical correlations for comparision. Experimental h is compared with that calculated from Gnielinsky equation [11] (Equation 6) and comparison indicated a maximum deviation of 60% for the considered range of Re.

$$Nu = \frac{(f/8)(Re-1000)Pr}{1+12.7\sqrt{f/8}(Pr^{2/3}-1)}\left(1+(D/L)^{2/3}\right) \quad \text{(Eq.6)}$$

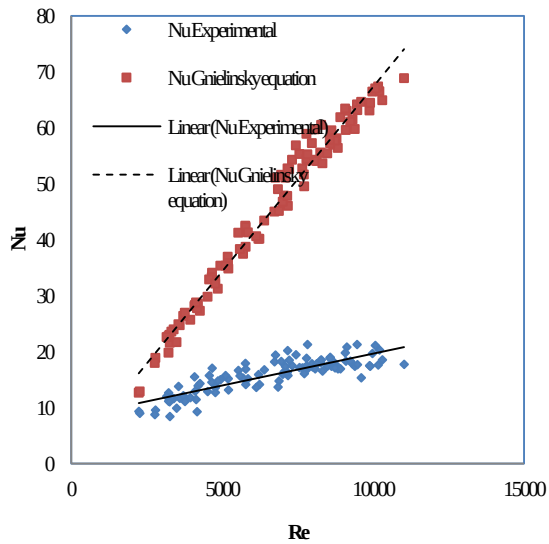

Figure 3. Reynolds number verses Nusselt number

Pressure drop calculation

In any heat transfer augmentation method heat transfer alongwith pressure drop is studied, as pressure drop increases pumping power also increases. Based on pumping power and heat transfer, a judicious decision can be made about overall benefits of augmentation methods. Presure drop is measured with air filled inverted U tube manometer for each mass flow rate. Pressure drop is calculated from Blasius correlation (equation 1). Calculated pressure drop and measured pressure drop are plotted as per Reynolds number and the plot is as shown in figure 4. The deviation goes on increasing as per Re and the maximum deviation observed is 8.33%

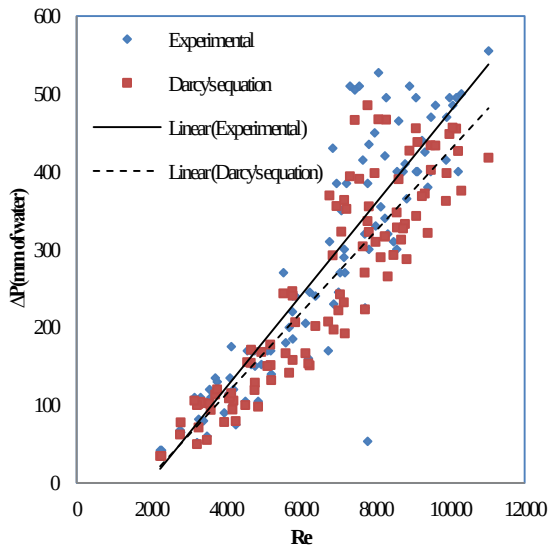

Figure 4 Pressure drop verses Reynolds number

4. Conclusion

Experimental set up to determine heat transfer coefficient and pressure drop at constant wall temperature condition of convective heat transfer has been developed. Set up is tested with water as fluid flowing through it for Reynolds number range 2000 to 15000 of turbulent flow regime. Test Results are compared with existing correlations. Results shows that developed set up gives steady error of 5.66% in mass flow rate measurement, 60% error in heat transfer coefficient when compared with that calculated from Gnienlinsky equation and 8.33% error in pressure drop measurement when compared with that calculated from Darcy-Weisbach friction factor formula. Further improvements are required to reduce the error in determination of 'h'.

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