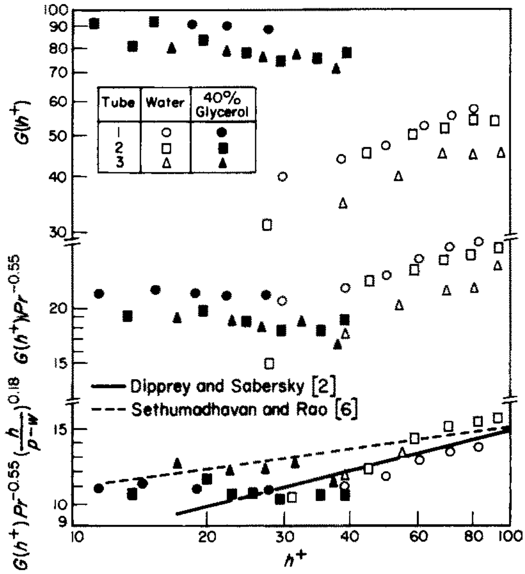

FIG. 4. Variation of $G(h^+)$ with (h^+).

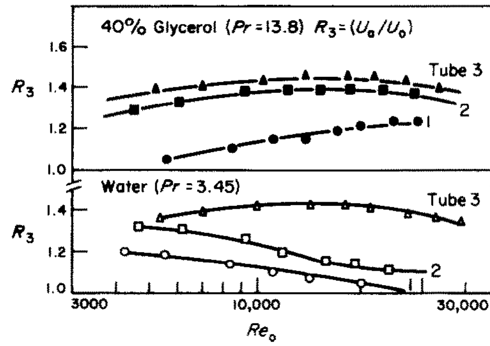

FIG. 5. Variation of R_3 with Re_o.

enhancement in heat duty for water ranged from 15 to 30% only for tubes 1 and 2, but it varies from 35 to 45% for tube 3. However, for 40% glycerol, tubes 2 and 3 give 35–50% enhanced heat duty and tube 1 gives only 10–20%. Obviously tube 3 (with a coiled ribbon pitch of 11 mm) gives the best performance compared to other tubes, and it gives an increased heat duty of 45 and 50% for water and 40% glycerol, respectively, around a Reynolds number of 15,000–20,000.

CONCLUSIONS

Turbulent flow friction factors in smooth tubes, roughened by helically coiled ribbons are 2.2–5.0 times greater, and Nusselt numbers are 1.3–2.2 times higher than those for the smooth tube over the range, $3000 < Re < 25{,}000$, and correlations are proposed for f and Nu.

Friction and heat transfer results were also analysed in terms of roughness momentum and heat transfer functions, and suitable correlations are suggested for $R(h^+)$ and $G(h^+)$.

Tube performance was evaluated on the basis of heat duty per unit pumping power, and tube 3 (p = 11 mm, α = 79°) giving 45–50% enhanced heat duty was identified as the most efficient tube for water and 40% glycerol used as test liquids.

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A combined convection correlation for vertical downward cooling flow in a natural circulation loop

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INTRODUCTION

SEVERAL empirical and analytical correlations for combined free- and forced-convection flow through vertical tubes without connection to a natural circulation loop have been derived [1, 2]. Holman and Boggs [3] derived an empirical correlation for the combined convection of upward turbulent flow in a vertical heating tube of a natural circulation loop. It can be seen from their results that the correlation is quite different from that of a single heater and a pure forced convection. A similar phenomenon was also observed by Creveling *et al.* [4] for the cooling jacket in a toroidal natural circulation loop and by Bau and Torrance [5] for the heater

NOMENCLATURE

C_p specific heat of loop fluid [J kg^{-1} °C^{-1}]
D inside diameter of the loop [m]
g gravitational acceleration, 9.8 m s^{-2}
h combined free and forced convection in the cooling tube [W m^{-2} °C^{-1}]
k thermal conductivity of the loop fluid [W m^{-1} °C^{-1}]
L_c length of the cooling tube [m]
Nu Nusselt number, hD/k
Pr Prandtl number
T_b bulk fluid temperature in the cooling tube [°C]
T_w inner wall temperature in the cooling tube [°C].

Greek symbols
β thermal expansion coefficient [°C^{-1}]
μ viscosity of the loop fluid [kg m^{-1} s^{-1}]
ρ density of the loop fluid [kg m^{-3}].

in a U-loop. It seems that the combined convections are dependent on the loop shape and operating mode.

The correlation for the combined convection of laminar downward flow in a vertical cooling tube of a rectangular natural circulation loop, however, still cannot be found in the literature. Zvirin *et al.* [6] applied the combined convection correlation for downward cooling fluid flowing vertically without connection to a natural circulation loop [1] to the analysis of the cooler in the rectangular loop studied and found that a large deviation from the experimental estimation existed. This is probably attributed to the effect of loop friction. But, little work has been done on this.

To fill this gap, an experiment was carried out in the present study to derive the empirical correlation for the combined convection of a downward laminar cooling flow in the cooler of a rectangular natural circulation loop subject to different loop frictions.

EXPERIMENTAL SETUP

The vertical cooling tube (test section) was designed in a double-pipe heat exchanger which consisted of an annular stainless-steel jacket with coolant flowing upwards and was connected to a rectangular loop, as shown in Fig. 1. Heating is provided directly by a constant and uniform heat flux by an evenly-wound electrical heating ribbon attached tightly on the tube surface; while cooling is provided by the cooling jacket using water as the coolant to remove heat from the loop.

The loop was all made of stainless steel and had a rectangular shape with a 28 mm inside diameter and a height of 1500 mm. The lengths of the cooling tube L_c and the heating section L_h are 480 and 600 mm, respectively, and the total loop length is 4500 mm. The relative height between the cooling tube and the heater measured from their centres, ΔZ, is 960 mm.

Water was used in the loop as the loop fluid. Electrical heat was supplied to the heater through a variable transformer (range: 0–1500 W) and measured by a wattmeter. To reduce the heat loss to the ambient, the heater was insulated with 5 cm thick calcium silicate. To prevent heat loss, fibreglass insulation was installed over the outside surface of the cooler and the connecting pipes. The total heat loss was found to be less than 10% of the total heat input rate.

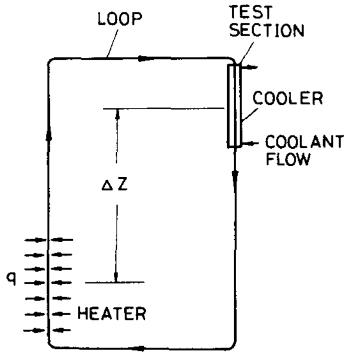

FIG. 1. Schematic diagram of a rectangular natural circulation loop.

To measure the temperature distribution, T-type thermocouples were installed at the inlet and outlet of the cooling tube and along the loop and both were recorded by a HP-3054 DL data logger with uncertainties of ± 0.5°C. To determine the total cooling rate in the cooler Q_c, the coolant mass flow rate was measured by a rotameter with an uncertainty of ± 0.003 kg s^{-1}. Since the temperature difference of the coolant flow across the cooling jacket was small, a HP quartz thermometer with an uncertainty of ± 0.04°C was used to measure the coolant temperatures at the inlet and outlet and hence the temperature drop across the cooling jacket so that the total cooling rate could be accurately determined.

The loop friction as well as loop circulation rate can be varied by adjusting a gate valve installed in the loop to produce three effective lengths, L_{ec} = 12.1, 24.6 and 37.0 m which were determined from conventional piping equations and referred as low, medium and high loop frictions, respectively. The total heating rate in the heater was varied between 225 and 1400 W in the experiment.

RESULT AND DISCUSSION

According to Holman and Boggs [3] and Creveling *et al.* [4], the combined convection in both laminar and turbulent regions may have the following correlation:

$$\frac{Nu\,Gr}{Pr} = a\,Re^b \tag{1}$$

where Gr is the Grashof number defined in the cooler as $D^3\rho^2 g\beta(T_b - T_w)/\mu^2$, Re is the Reynolds number at steady state, and a and b are the constants to be determined experimentally.

The values of $Nu\,Gr/Pr$ and Re were determined for each test run by the relations which were derived from the energy balance to the cooler

$$\frac{Nu\,Gr}{Pr} = \frac{D^3\rho^2 g\beta}{\pi\mu^3 C_p L_c} Q_c \tag{2}$$

and

$$Re = \frac{4Q_c}{\pi D\mu C_p \Delta T} \tag{3}$$

where ΔT is the inlet and outlet temperature difference of the loop fluid in the cooler, Q_c the total cooling rate; all of which are determined at steady states.

The present experiment was run for three coolant flow rates: 10, 20 and 29 l min^{-1} which are in the turbulent flow region with Reynolds number ranging from 2400 to 7000. It was found in the experiment that the combined convection is independent of the coolant flow rates in this turbulent flow region. This is reasonable since the thermal resistance induced by the combined convection in the inner wall, which is usually in the laminar region, dominates.

It is shown in Fig. 2 that the combined convection correlations obey equation (1) for different loop friction which was represented by the parameter Y defined as

$$Y = \frac{\Delta Z L_h}{D L_{ec}} \tag{4}$$

where ΔZ is the relative height between the cooling tube and the heater measured from their centres. The results are, for laminar flow with $400 < Re < 2000$

$$\frac{Nu\,Gr}{Pr} = \begin{cases} 22.45Re^2, & \text{high friction } (Y = 0.556) \\ 12.78Re^2, & \text{medium friction } (Y = 0.83) \\ 8.92Re^2, & \text{low friction } (Y = 1.70). \end{cases} \tag{5}$$

Equation (5) shows that the exponents of the correlations are 2.0, which is close to 2.56 for the laminar flow in a toroidal loop [4], and 3.0 for the turbulent flow in a rectangular loop [3].

It is noticeable that the combined convection is extensively affected by the loop friction as is clearly shown in Fig. 2. This is due to the fact that for a given cooling rate (or total heating rate at steady state) the loop circulating flow rate as well as the temperature difference, ΔT, across the cooler may vary with loop friction. Higher friction will reduce the flow rate and increase ΔT to cause a variation of the combined convection. This indicates that one cannot ignore the effects of loop friction in deriving the combined convection correlation for cooling or even heating fluid in a natural circulation loop. Ignorance of this effect would cause serious errors as was observed [6].

If we introduced a new dimensionless group $(Nu\,Gr/Pr)Y$ containing a dimensionless parameter Y for loop friction, then it is very interesting to note that a new correlation can be obtained (see Fig. 3)

$$\frac{Nu\,Gr}{Pr}Y = 10.38Re^2. \tag{6}$$

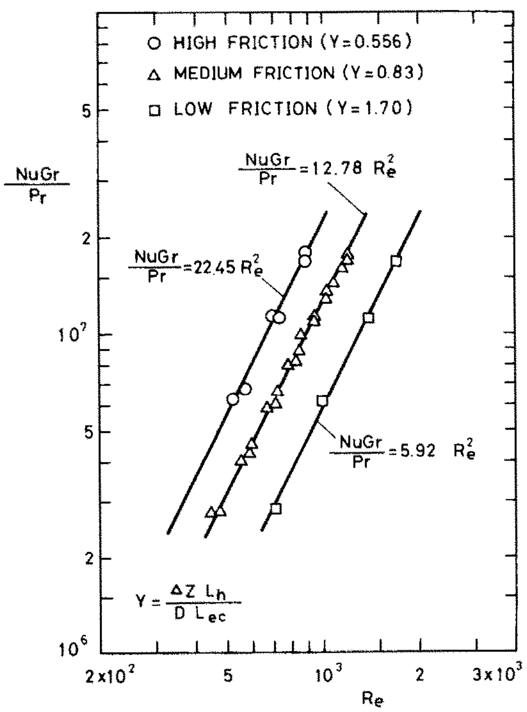

FIG. 2. Combined free- and forced-convection correlation for different loop frictions, $Nu\,Gr/Pr$ vs Re.

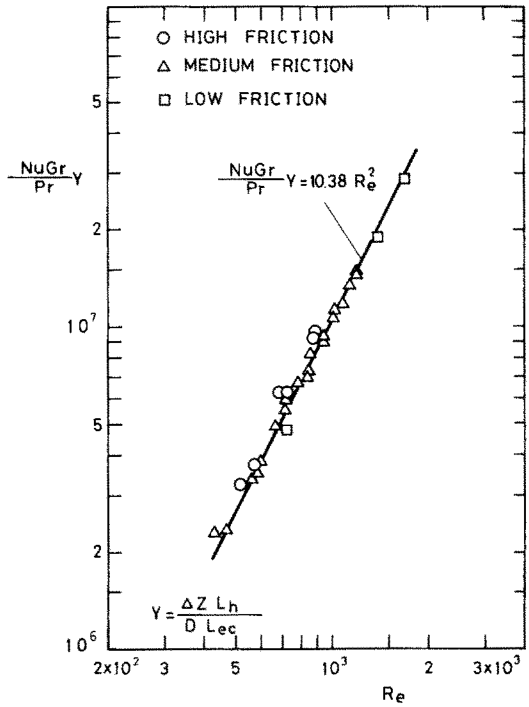

FIG. 3. Combined free- and forced-convection correlation, $(Nu\,Gr/Pr)Y$ vs Re.

This is a new correlation for laminar combined free and forced convection for cooling fluid flowing downward vertically in tubes connected to a rectangular natural circulation loop. As mentioned previously, the above correlation is independent of the Reynolds number of the coolant flow rate which ranges from 2400 to 7000 in the present experiment.

CONCLUSION

The empirical correlations for the laminar combined free and forced convection of a vertical cooling downward flow in a rectangular natural circulation loop is derived in the present study. The experiments show that the loop friction plays an important role in the combined convection and should be taken into account. A correlation using the dimensionless group $(Nu\,Gr/Pr)Y$ containing a dimensionless parameter Y for loop friction and Reynolds number Re is then derived.

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