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HEAT TRANSFER COEFFICIENT MEASUREMENTS FOR SMOOTH AND ROUGH TUBES AT LOW REYNOLDS NUMBER

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ABSTRACT

Measurements of the heat flux; heat transfer coefficients are reported for the gas flow in the smooth and rough tubes. Results are presented for laminar, transition and turbulent flow of air for a Reynolds number range of 500 < Re < 5000 with a constant surface temperature of 90^0 C. The geometrical parameters are same for the smooth and rough tubes. The experimental results show that the rough tube heat flux and heat transfer coefficient has no effect in the laminar region but from transition to turbulent flow it is 10 – 15 %[1] higher than that of the smooth tube. So, the rough tube shows better heat transfer performance than the smooth tube.

1. INTRODUCTION

In the heat exchanger, the roughness has no effect on the laminar regime, i.e., the heat transfer coefficients in the smooth and rough tubes are same in the laminar flow. The performance of a conventional heat exchanger, which is an essential unit in the heat extraction and recovery systems, can be substantially improved by a number of augmentation techniques. The common thermal–hydraulic goals are to reduce the size of the heat exchanger required for a specified heat duty, to upgrade the capacity of an existing heat exchanger, and to reduce either the approach temperature difference for the process streams, or to reduce the pumping power. A preferred approach to the problem of increasing heat exchanger effectiveness, while maintaining minimum heat exchanger size and operational cost, is to increase the heat transfer exchange rate.

Many different methods have been considered to increase the rate of heat transfer in forced convection while reducing the size of the heat exchanger and effective energy savings [2, 3, 4]. Surface methods include any technique, which directly involve the heat exchanger surface. They are used on the side of the surface that comes into contact with a fluid of low heat transfer coefficient in order to reduce the thickness of the boundary layer and to introduce better fluid mixing. The primary mechanisms for thinning the boundary layer are increased stream velocity and turbulent mixing. Secondary re-circulation flows can further enhance convective transfer. Flows from the core to the wall reduce the thickness of the boundary layer and the secondary flows from the wall to the core promote mixing. Flow separation and reattachment within the flow channel also contribute to heat transfer enhancement.

Among the existing methods for enhancing heat transfer in a single-phase flows, the laminar flow in a round tube is one of them. Here the inner surface of the tube is roughened, with repeated or helical ribbings. It is well known that two or more of the existing techniques can be utilized simultaneously to produce an enhancement larger than that produced by only one technique. The combination of different techniques acting simultaneously is known as compound augmentation. This is an emerging area of interest and holds promise for practical applications. Interactions between different augmentation methods contribute to greater values of the heat transfer coefficients compared with the sum of the corresponding values for the individual techniques used alone. Preliminary studies in compound passive augmentation techniques are encouraging. One example is: rough tube wall with helical twisted tape.

This study reports an experimental investigation to see whether or not the heat transfer is enhanced by the multiplicative effects of a rough tube.

2. OBJECTIVES

Measurements of heat transfer coefficients are reported for a gas flow in smooth and rough tubes.
Results are presented for laminar, transition and turbulent air flow for a Reynolds number range of 500 < Re < 5000 with a constant surface temperature of $90\,^0$C.

The geometrical parameters are kept same for all types of tubes. The experimental results show that the rough tube heat transfer coefficient is higher than that of the smooth tube.

3. LITERATURE REVIEW

There are some literatures concerning the effect of surface roughness on heat and flow characteristics in small channels. Hence, a brief overview of what has been done in the past is presented below. As early as in the nineteenth century, Darcy [5], conducted careful pressure drop experiments on pipes of different materials and roughness, and established that the flow depended on the pipe roughness, pipe diameter, and slope. Nikuradse's [6] conducted exhaustive experiments to study the effect of roughness on flow characteristics in circular pipes. Their work established the effect of relative roughness (ε/D), on the flow characteristics. Since the relative roughness (ε/D), affects the flow characteristics, the same surface roughness value has different effects on large and small diameter tubes. The work available in the literature clearly indicates that the roughness affects the laminar to turbulent transition, including the flow and heat transfer characteristics. The present work is aimed at studying the effect of surface roughness on heat transfer in 10 mm diameter cupper tubes.

4. PERFORMANCE CRITERIA

The primary considerations for assessing the effectiveness of augmented surfaces are economic: relating to initial development cost, capital cost, operating cost, and maintenance cost. Reliability and safety factors are also important. The relationship between the thermal and hydraulic performances must also be considered.

Major process operational variables include the rate of heat transfer, pumping power, pressure drop, and heat flow rate and fluid velocity. Webb [7] proposed a broad range of performance evaluation criteria for single-phase flows in tubes to obtain the optimum surface geometry. Three performance objectives considered are increased heat duty, reduced surface area and reduced pump power.

A fixed geometry criterion is used for smooth tubes with augmented tubes of equal length to compare the increased heat duty for the constant surface temperature heat exchanger. The pumping power of the augmented tube exchanger would naturally be greater for the augmented surface tube due to the higher friction. Alternatively, the pumping power could be kept constant by reducing the tube-side velocity.

A fixed flow area criterion for heat exchangers having constant diameter tubes, e.g., shell and tube exchanger was proposed. For the constant pumping power, the tube length and possibly the flow rate would be reduced. Augmented tubes are used to obtain reduced pumping power with constant heat duty and flow rate.

In most cases, a heat exchanger is sized for a specific thermal duty with a specified flow rate. In these situations, the previously mentioned criteria do not apply. This is accomplished using a greater number of tubes in parallel or by using the same number of larger diameter tubes. It must be noted that the preferred size of specific roughness geometry is dependent on the operational Reynolds number. As the Reynolds number increases, the preferred roughness size becomes smaller.

As preliminary design guidance to the selection of a technique, the heat transfer efficiency can be evaluated based on the power consumption per unit mass of the fluid. The criterion i_E [8] is defined as the ratio between the heat transfer coefficients for the tube using the heat transfer promoter to the value for a smooth tube at the same level of power. The criterion i_E can be defined as follows:

$$i_E = \frac{Nu_R / Nu_S}{(f_R / f_S)^{0.291}} = f(\mathrm{Re}_S) \tag{1}$$

where Nu_R and Nu_S stand for the rough and smooth tube Nusselt numbers, and f is the friction factor for both tubes. When the corrugated tube is used, i_E =1.0-1.2 for Re> 2000.

When an enhanced tube is considered for the replacement of a smooth one, there are many possible effects on the performance. The design constraints imposed on the exchanger flow rate and velocity cause key differences among the possible Performance Evaluation Criteria (PEC) on the basis of the first law analysis. The increased friction factor due to augmented surfaces may require a reduced velocity to satisfy a fixed pumping power (or pressure drop) constraint. However, if the mass flow rate is reduced, it is possible to maintain a constant flow frontal area at a reduced velocity. In many cases the heat exchanger flow rate is specified and a flow rate reduction is not permitted. Despite of the fact that a large number of possible PEC can be defined. The PEC suggested by Webb and Bergles [9] characterize almost all the PEC and some of them are shown below. The equations are developed for tubes of different diameters, and heat transfer and friction factors, based on the presentation format of performance data for enhanced tubes [10]. The relative equations for single-phase flow inside enhanced tubes are

$$A_{\bullet} = N_{\bullet} L_{\bullet} D_{\bullet} \tag{2}$$

where A_* is the dimensionless heat transfer surface area (A_R/A_S), N_* is the ratio of number of tubes ($N_{t,R}/N_{t,S}$), L_* is the dimensionless tube length (L_R/L_S), D_* is the dimensionless tube diameter (D_R/D_S).

$$P_{\bullet} = W_{\bullet} \Delta p_{\bullet} = \frac{f_R}{f_S D_{\bullet} L_{\bullet} N_{\bullet} u_{m\bullet}{}^3} = \frac{W_{\bullet}{}^3 L_{\bullet}}{N_{\bullet}{}^2 D_{\bullet}{}^5} \frac{f_R}{f_S} \tag{3}$$

where P_* is the dimensionless pumping power (P_R/P_S), W_* is the dimensionless mass flow rate (W_R/W_S), $\Delta P_*$is the dimensionless pressure drop ($\Delta P_R/\Delta P_S$), u_{m*} is the dimensionless flow velocity (u_{mR}/u_{mS}).

$$Q_{\bullet} = W_{\bullet} \varepsilon_{\bullet} \Delta T_i^{\bullet} \tag{4}$$

where, ε_* is the ratio of heat exchanger effectiveness ($\varepsilon_R/\varepsilon_S$), ΔT_i^* is the dimensionless inlet temperature difference between the hot and the cold fluids ($\Delta T_{i,R}/\Delta T_{i,S}$)

$$\Delta p_{\bullet} = \left(\frac{f_R}{f_S}\right) \frac{L_{\bullet}}{D_{\bullet}} u_{m\bullet}{}^2 \tag{5}$$

5. EFFECT OF SURFACE ROUGHNESS

Roughing the surface is known to provide more eddies and thus can enhance heat transfer. To quantify the roughness effects, the heating data for the smooth and rough surfaces are obtained for various fluid levels. The results show that the rough surface does have better heat transfer performance than that of the smooth surface. Surface roughness has an effect on the pipe friction only if the flow is turbulent.

A laminar sub-layer exists very close to the wall. If the surface roughness protrudes beyond this layer, it has an effect, otherwise the pipe is said to be smooth. Surface roughness increases the friction loss because a form drag is superimposed on the skin friction drag. This happens because the fluid close to the surface cannot follow the shape of the surface. The effect is shown on the following diagram.

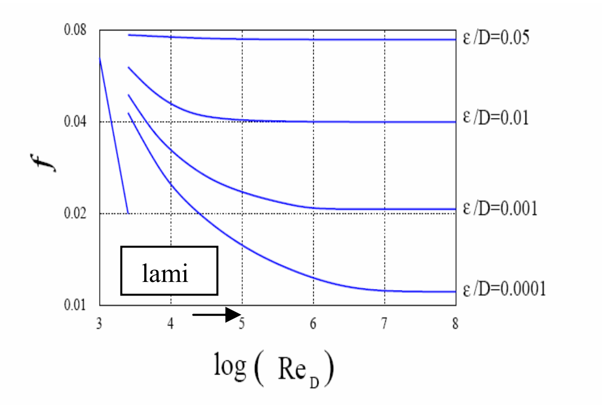


Fig 1. For different roughness effect inside the tubes.

There are three points to note about surface roughness:

- Roughness has no effect in the laminar flow regime,
- Roughness increases the friction factor in the turbulent regime,
- For a rough tube with a sufficiently large Reynolds number, the friction factor is independent of Reynolds number. This is referred to as the fully rough regime.

6. HEAT TRANSFER IN CHANNEL FLOW

The Nusselt number is a dimensionless parameter that represents the heat transfer coefficient at a surface where heat transfer by convection takes place. It is a very important number for convection problems and it is defined as

$$Nu = \frac{hd_h}{k} \tag{6}$$

where, d_h is the hydraulic diameter, *h* is the convection heat transfer coefficient, *k* is the thermal conductivity of the fluid.

When the cross-section of the tube is not circular, the hydraulic diameter is defined as

$$d_h = \frac{4 * volume\ of\ channel}{\text{Total wetted surface}} \tag{7}$$

This diameter can be used in Eq. (6).

6.1 Turbulent Channel Flow with Low Reynolds Number

For the turbulent flow with a small Reynolds number, Gnielinski [14] proposes an expression

$$\overline{Nu}_D = \frac{(f/8)(\mathrm{Re}_D - 1000)\mathrm{Pr}}{1 + 12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3} - 1)} \tag{8}$$

where *f* is the friction factor, and for smooth tubes

$$f = (0.90 \ln \mathrm{Re}_D - 1.64)^{-2} \tag{9}$$

This correlation is valid for 0.5 < Pr < 2000 and 2300 < $\mathrm{Re}_D < 5*10^6$.

Other correlations are available for the estimation of the shell side heat transfer coefficients [Perry; Donohue] [11]. The Wilson method uses the exponent of the Reynolds number from these correlations to determine the leading coefficient in a correlation of the form of [13], estimating the inside and outside coefficients. The flow regime should be determined before the appropriate correlations can be used.

7. EXPERIMENTAL SETUP

* 1 Water tank
* 2 Water pump
* 3 Water flow meter
* 4 Room air
* 5 Pressurized air
* 6 Air flow meter
* 7 Test section
* 8 Data logger
* 9 Monitor

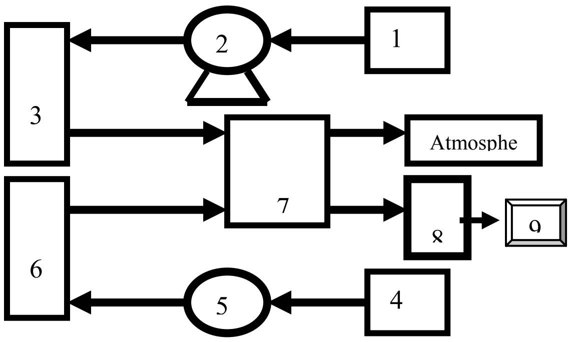


Fig 2. Experimental Setup

The test section together with a schematic drawing of the flow loop is shown in Fig. 2.

-One Energy brand single-pass shell-and-tube heat exchanger covered with a detachable foam-insulating jacket.

-Four K-type thermocouples are used to monitor the inlet and outlet temperatures from each side of the heat exchanger.

-Two flow meters Model NP-G25 and G26 F02-700

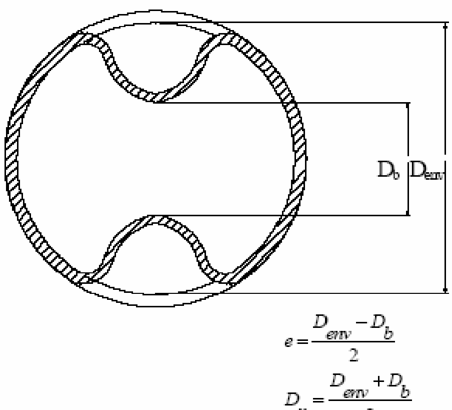
188-G for airflow and Model KLK-4EA for water flow. The air flow meters have a nominal flow range of 2.5 to 40 lpm. For water, a flow range of 10 to 100 lpm is equipped with the flow transmitter model.

-A 15-kW auxiliary electric water heater is equipped with a water re-circulation pump. Temperature controller controls this heater. The pump specification is Flow (m^3/hr)-1.2 to 4.2, Head (m)-56 to22, Max Head (m)-64.00, Speed (rpm)-2900, Weight (kg)-22 Manufactured by LOWARA (SV206F07T).

The prescribed condition of, uniform surface temperature is obtained by flowing hot water and insulating the outer wall of the water tube. The heated test section is one meter long and it is preceded to the development approach section of about 300mm. The whole length of the heat transfer section is thermally insulated to minimize the heat exchange with the environment. The fluid temperatures have been measured through K-type Nickel-Cobalt thermocouples. Three thermocouples in each side probes directly immersed in the fluid, measure the inlet and outlet temperatures. The bulk temperature at exit location has been calculated from the power supplied by the hot water to the tube. The data acquisition software is provided to update the data coming from all the channels and to plot them as a function of time on a screen. In these conditions, the effect of the variation of the fluid properties with temperature is assumed to be negligible. Air is used as the working fluid. The Reynolds number range investigated is 500 < Re < 4500.

8. TESTED TUBE HEAT EXCHANGERS

Different tubes heat exchangers are used here. Among the tubes one is rough tube. It is characterized by an internal helical ridging corresponding to an external smooth tube, made of copper. One rough tube geometry tested in the present study shows a single helix ridging and it is obtained from a tube having an external diameter of 12.73 mm and a wall thickness of 1 mm. The tube is shown in Fig. 3. Tube shows a very regular wall profile; in particular, two helix do not cross along the same generatrix of the cylindrical envelope surface. The geometric parameters usually used to describe enhanced tubes are Fig. 3, the bore diameter D_b , the envelope diameter D_{env} , the ridge depth *e*, the pitch *p*, [16]).



$$e = \frac{D_{env} - D_b}{2}$$

$$D_n = \frac{D_{env} + D_b}{2}$$

Fig 3. Cross-section of rough tube geometry

A relevant non-dimensional parameter introduced by Withers and Habdas [17] for this kind of geometry is the severity, defined as follows: s = e^2/ p*D_n, where $D_n$is the nominal tube diameter.

The local heat transfer coefficients, and therefore the local Nusselt number, have been obtained by considering a heat transfer surface equal to the surface of a cylinder having a diameter D_{env}. The diameter D_{env}. has also been used as the characteristic length of the problem, in accordance with other investigators. Richards et al. [18] used both the bore diameter and the envelope diameter to reduce their experimental data, and they drew opposite conclusions according to the characteristic length chosen. The envelope diameter can in practice be easily determined, while on the contrary the other dimensions, like the bore diameter, the nominal diameter D_n, or the volume based diameter result are more difficult to be measured, as already pointed out by Richards et al. [18]. The geometric variables mentioned in Fig. 4 are often not exhaustive to describe the various geometries since different manufacturing techniques are currently in use to produce them. For example, the tubes show the same values of ridge depth, pitch and diameter, and hence severity, but the profile of the corrugation is different in Fig. 4.

For Rough Tube Geometry

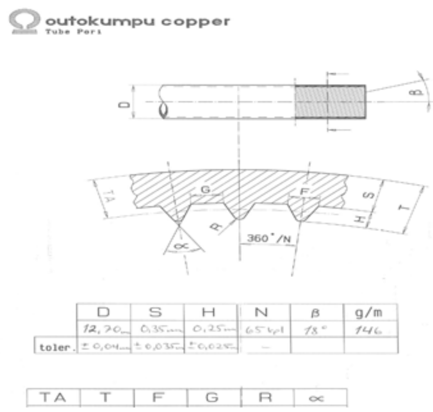


Fig 4. Rough tube surface is showing with different dimensions.

9. RESULTS AND DISCUSSION

Experiments are made using the smooth and rough tubes. The heat flux, heat transfer coefficient, and Nusselt number are obtained experimentally for lower Reynolds numbers at a constant surface temperature.

For theoretical heat fluxes, we add different flow regions: The laminar entrance region that follows Hausen's correlation [12], and the low turbulent flow region that follows Gnielinski correlation [14].

However for the developed flow everything remains unchanged and we compared these theoretical results with the experimental results. The surface roughness has no effect on the laminar flow case. However, the surface roughness in the transition and turbulent regions affects the heat transfer. The difference between measured heat flux and the theoretical value is about ± 5 %. Heat flux for rough tube is 5 ~ 10 %[1] higher than that of the smooth surface tube.

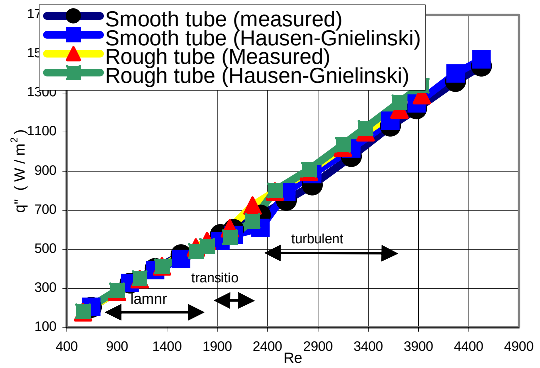


Fig 5. Measured and calculated heat fluxes as a function of Re, L = 1m, $d_i = 10$ mm and 10.48 mm.

The measured heat transfer coefficient for the rough and smooth tubes is shown in Fig. 6.

The higher values of e/d_i, e / p (where e is roughness height and p is pitch length) enhance the heat transfer rate significantly compared with the smooth tube.

The effect of the roughness on the heat transfer is mainly due to the increasing disturbances in the laminar sub-layer [19]. In the transition and turbulent regions, the heat transfer coefficient of the rough tube is 10 ~15 % [1] higher than that of the smooth tube.

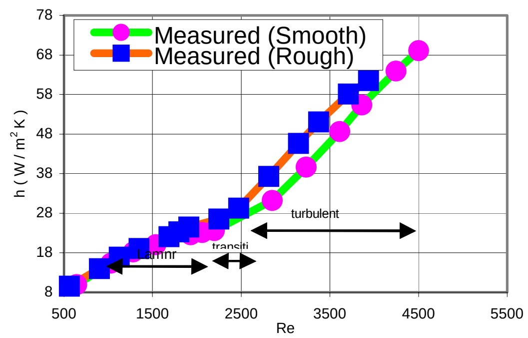


Fig 6. Measured average heat transfer coefficient $\bar{h}$ as a function of Re in laminar, transition and turbulent flows.

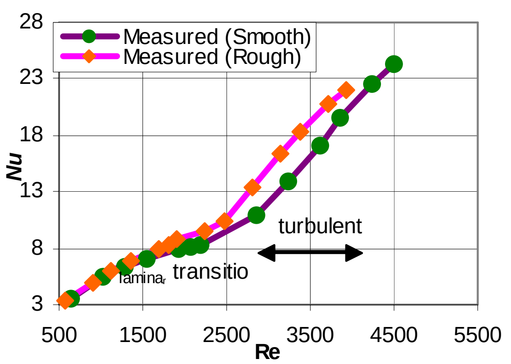


Fig 7. Variation of average $\overline{Nu}$ with Re at laminar, transition and low turbulent flows.

In Figure.7, the laminar, transition and turbulent flow results are presented in dimensionless forms.

10. CONCLUSIONS

The heat transfer characteristics subjected to the gas flow in the smooth and rough tubes have been evaluated experimentally. Results are presented for the laminar, transition and turbulent flows of air in a Reynolds number range of 500 < Re < 5000 with a constant surface temperature of 90 ^{0}C. The experimental results show that the rough tube heat transfers coefficient is 10 ~ 15 % [1] higher than that of the smooth tube.

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12. NOMENCLATURE

Symbol	Meaning	Unit
A_{in}	Inside area of the copper tube	(m^2)
A_{out}	Outside area of the copper tube	(m^2)
A_*	Dimensionless heat transfer surface area (A_R / A_S)	(-)
c_p	Specific heat	(J / kg. K)
D_{in}	Inside diameter of the copper tube, meter	(m)
D_{out}	Outside diameter of the copper tube	(m)
d_h	Hydraulic diameter of tubes	(m)
D_n	Nominal diameter	(m)
e	Roughness of the tube	(m)
f	Friction factor	(-)
G	Conductance	(W/ m)
$\bar{h}$	Average heat transfer coefficient	(W/ m^2. K)
$\bar{h}_w$	Heat transfer coefficient of water	(W/ m^2. K)
$\bar{h}_a$	Heat transfer coefficient of air	(W/ m^2. K)
i_E	Performance Evaluation Factor	(-)
k_w	Thermal conductivity of water	(W/ m.K)
k_a	Thermal conductivity of air	(W / m.K)
L	Length of tube, meter	(m)
L_*	Dimensionless tube length (L_R / L_S)	(-)
$\dot{m}_a$	Mass flow rate of air	(kg/s)
$\dot{m}_w$	Mass flow rate of water	(kg/s)
Nu	Nusselt number	(-)
p	Pitch length	(m)
Pr	Prandlt number of air	(-)
$\dot{q}$	Heat transfer	(W)
q''	Heat flux	(W / m^2)
Re	Reynolds number	(-)
s	Severity, $(e^2 /(p * D_n))$	(-)
$T_{i,w}$	Water inlet temperature	(^0C)
$T_{o,w}$	Water outlet temperature	(^0C)
T_{ave}	Average temperature	(^0C)
$T_{i,a}$	Air inlet temperature	(^0C)
$T_{o,a}$	Air outlet temperature	(^0C)
T_s	Surface temperature	(^0C)
ΔT	Temperature difference	(^0C)
ΔT_{lm}	Log-Mean Temperature Difference (LMTD)	(-)
ΔT_i	Dimensionless inlet temperature difference	(^{0}C)
$\dot{V}_a$	Volume flow rate of air	(lt./min)
$\dot{V}_w$	Volume flow rate of water	(lt./min)
v	Velocity)	(m/s
Greek Symbols		
ν	Kinematics viscosity	(m^2 / s)
μ	Viscosity of used in the operation	(Ns/ m^2)
ρ	Density	(kg/ m^3
ε	Effectiveness	(-)
Subscripts		
d	Developed flow	
n	Nominal	
S	Smooth / surface	