Augmentation of Heat Transfer in a Circular Pipe by Means of Twisted Vane Inserts

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Abstract

Heat transfer characteristics for swirl flow of water in a circular pipe are studied numerically. The pipe is fitted with heat resistant vanes. The performance of this pipe is compared with that of an empty pipe under similar conditions. The three-dimensional Navier-Stokes equations for incompressible Newtonian fluid flow are used. This investigation reveals that the swirl flow increases heat transfer. The effects of various parameters such as Reynolds number, inlet pressure, twisting angle, surface heat flux, surface temperature, pipe length etc. are also studied.

Keywords: Swirl Flow; Steady State; Incompressible Fluid; Heat Transfer; Uniform Heat Flux; Stationary Vanes

1.0. Introduction

Laminar flow heat transfer in a circular pipe has various applications such as heating of circulating fluids in solar collectors, heating or cooling of viscous fluids (oils, liquids in process industry). There are considerable challenges in designing a heat exchanger. The main purpose is to make the equipment compact and achieve a high heat transfer rate using minimum pumping power. Since the laminar flow heat transfer coefficients are predominantly low various types of heat transfer augmentation techniques are used. These augmentation techniques (passive, active or a combination of passive and active methods) are discussed in several published literatures. The active techniques demand external power input for the enhancement of heat transfer and do not possess as much potential as the passive techniques for being complex in design. In the passive techniques, no external power input is required. Inserts are used in the flow passage to increase the heat transfer rate. They are beneficial compared to active techniques because these techniques can be easily employed in an existing heat exchanger. They are also inexpensive.

Bergles (1985) discussed almost all types of heat transfer augmentation technique with external inserts up to 1985. Swirl flow can be generated by different kinds of swirl generators.

Recently injection, vanes (tangential, radial), twisted tape, axial blades, a helical screw, wire coil, ribs, fins, dimples and other types of passive heat transfer augmentation techniques are being used.

An extensive literature review on swirl flow was discussed by Gupta et al. (1984) and Bergles et al. (1996). In a swirl flow, there is a spiral motion in the tangential direction in addition to the axial and radial directions. Swirl flows can be categorized into two types: non-decaying (steady) and decaying (unsteady). In non-decaying swirl flow, the swirl is continuous in the flow direction, i.e. the flow maintains its characteristics throughout the whole length of the pipe. In decaying swirl flow, the swirl is introduced to the flow by inserting a flow guide in the entrance of the pipe only and the effect of swirl decays in the axial direction of flow. These flows can be generated by means of insertion of various swirl generators. According to Razgaitis et al. (1976) swirl flows may be classified into three groups depending upon characteristic velocity profiles: (i) curved, (ii) rotating and (iii) vortex flow. These velocity profiles are different, depending upon the particular flow geometry and swirl generation methods. Curved flow is produced by a stationary boundary. It causes a continual bending of the local velocity vector. Also, complex secondary flows with an appreciable velocity component normal to the instantaneous osculating plane are generated. Curved flows can be generated by inserting coiled wires, twisted tapes and helical vanes into the pipe, by coiling the tube helically or by making helical grooves in the inner surface of the duct. Curved flow is also called "continuous swirl flow". Rotating flow is generated by a rotating boundary, either confining the flow (as for a rotating tube) or locally influencing the flow field (as for a spinning body in a free stream). Vortex flow arises when a flow with some initial angular momentum is allowed to decay along the length of a tube. Vortex flow is also called, "decaying swirl flow". Decaying swirl flows are generated by the use of tangential entry swirl generators and guided vane swirl generators. According to Yapici et al. (1992) by using a single tangential inlet duct or more than one tangential entry, tangential entry of the fluid into a duct stream can be achieved. Guided vane swirl generators may be categorized into two types: radial guide vane and axial guide vane. Axial vane swirl generators consist of a set of vanes fixed at a certain angle to the axial direction of the duct, which give a swirling motion to the fluid. Commonly, the vanes are mounted on a central hub and they occupy space in an annular region. Even one single helical vane or twisted tape can be used as a means of generating decaying swirl flow. Radial guide vane swirl generators are generally mounted between two disks and the vanes are so constructed as to be adjustable to obtain the desired initial degree of the swirl. Radial generators are capable of generating much more intense swirls and they cause more complex velocity profiles than axial generators. Because the flow direction must change from radially inward to axial downstream which can occur either abruptly or by means of a fairing section. An inserted center body (deflecting element) can be used in radial generators whose function is to deflect the flow into the pipe as smoothly as possible.

Experiment on test sections with different diameters was conducted by Binnie et al. (1975). They examined the pressure and velocity distributions inside a convergent nozzle that discharges water downward under pressure. In the test section, a volute for the swirling flow was used. Results show two large departures from inviscid flow because of permanganate injection into the nozzle with a hyperdermic tube inserted through pressure tapping. The decay process of swirling flow in an axially symmetrical, cylindrical pipe in relation to the flow pattern was analyzed by Ito et al. (1978). They also derived an empirical equation for the dimensionless decrease in circulation. Hong et al. (1976) correlated heat transfer and pressure drop data for twisted-tape inserts. They used twisted tape with different twisted ratio to reduce the size of the heat exchanger. They showed that the friction factor is affected by tape twist only at high Reynolds numbers in accordance with analytical predictions. Akpinar et al. (2004) studied swirling flow generated by injection experimentally. The authors studied the effect of holes diameter, holes number and angle of injection on the heat transfer rate and the pressure drop. They found that use of injector to create swirling flow enhances heat transfer rate but it increases the pressure drop and requires more pumping power.

From the above discussion, it can be understood that very few numerical studies have been done on swirl flow. So, there are scopes for further numerical investigations. So, in the present study, a three-dimensional numerical code is developed and validated with theoretical data. The code is then used for numerical simulation to understand the heat transfer characteristics of swirl flow.

Nomenclature

D	Outer diameter of the pipe [m]
L	Length of the pipe [m]
Q	Heat transfer rate [W]
R	Inner radius of the pipe [m]
V	Radial velocity component [m/s]
W	Tangential velocity component [m/s]
d	Inner diameter of the pipe [m]
f	Friction factor
k	Thermal conductivity [W/m K]
q	Heat flux at the outer surface [W/m^2]
r	Local radius of the pipe [m]
$\dot{m}$	Mass flow rate [kg/s]
μ	Dynamic viscosity [Ns/m^2]

ρ	Density [kg/m^3]
C_p	Specific heat at constant pressure [J/kg K]
T_0	Outlet water temperature [^{0}C]
T_i	Inlet water temperature [^{0}C]
U(r)	Axial velocity component [m/s]
U_{max}	Maximum axial velocity of pipe [m/s]
U/ U_{max}	Velocity ratio
Re	Reynolds number

Table 1. Material properties.

Parameters	Water	Copper
Density, ρ (kg/m3)	997.13	8930
Dynamic viscosity, μ (Pa s)	8.9×10-4	–
Thermal conductivity, K (W/m K)	0.58	401
Specific heat at constant pressure, Cp (J/kg K)	4.18×103	380

2.0. Numerical models

The study is carried out to investigate the effects of twisted vanes on heat transfer characteristics of a circular pipe. Two models are considered. In one model, the pipe is fitted with heat resistant vanes and in another model, there are no vanes inside the pipe. In both models, the pipe is made of copper and the working medium is

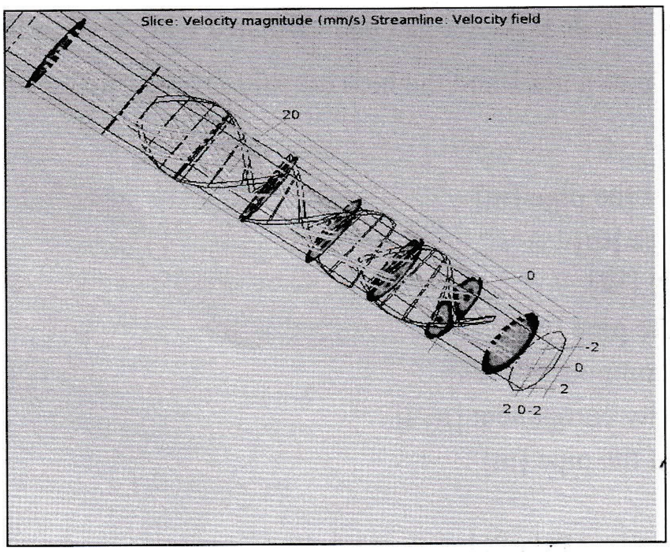

Fig. 1. Streamlines in the pipe fitted with twisted vanes.

water. The pipe for both models has a length of 42mm, an internal diameter of d = 3mm and an outer diameter of 3.5mm. Twisting angles of 22^0, 26^0, 30^0 and 35^0 are used to determine its effects on heat transfer. Pipe length is varied to obtain the L/d ratios of 11, 12 and 13. The initial temperature and pressure are 298 K and 1atm. respectively. The inlet velocity is varied from 0.01m/s to 0.5m/s to determine the effects of Reynolds Number on heat transfer characteristics. Inlet pressure and surface heat flux are varied from 100 Pa to 700 Pa and 2000 W/m^2 to 14000 W/m^2 respectively.

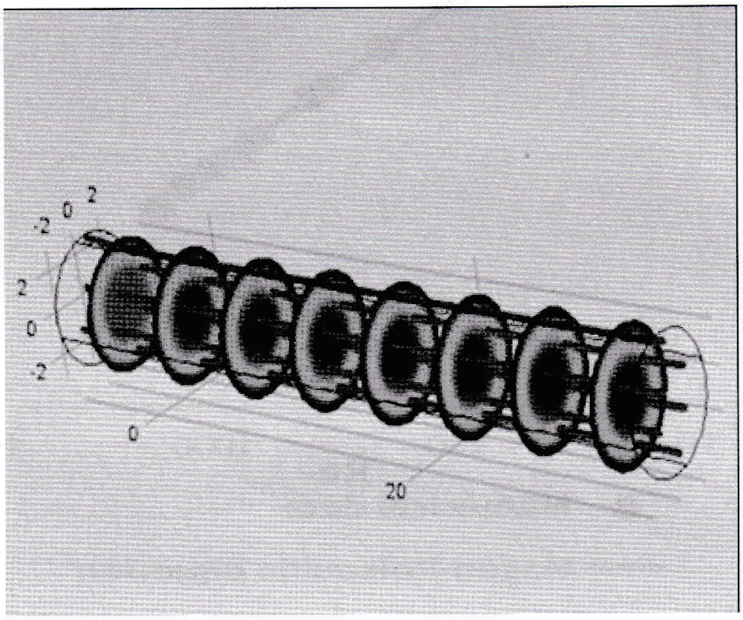

Fig. 2. Streamlines in steady state condition in different sections of the pipe without any vane.

3.1. Mathematical modeling

The governing equations are:

$$\frac{\partial U}{\partial x}+\frac{\partial V}{\partial r}+\frac{V}{r}=0 \quad (1)$$

The amount of heat carried away by the water,

$$Q = \dot{m}C_p(T_0-T_i) \quad (2)$$

Heat supplied to the outer surface of the pipe,

$$Q = (\pi DL)*q \quad (3)$$

Heat transfer efficiency

$$= \frac{\text{Amount of heat carried away by the water}}{\text{Heat supplied to the outer surface of the pipe}}$$

$$= \frac{\dot{m}Cp(T0-Ti)}{(\pi DL)*q} \quad (4)$$

$$\text{Reynolds Number, Re} = \frac{\rho vd}{\mu} \quad (5)$$

3.2. Code validation

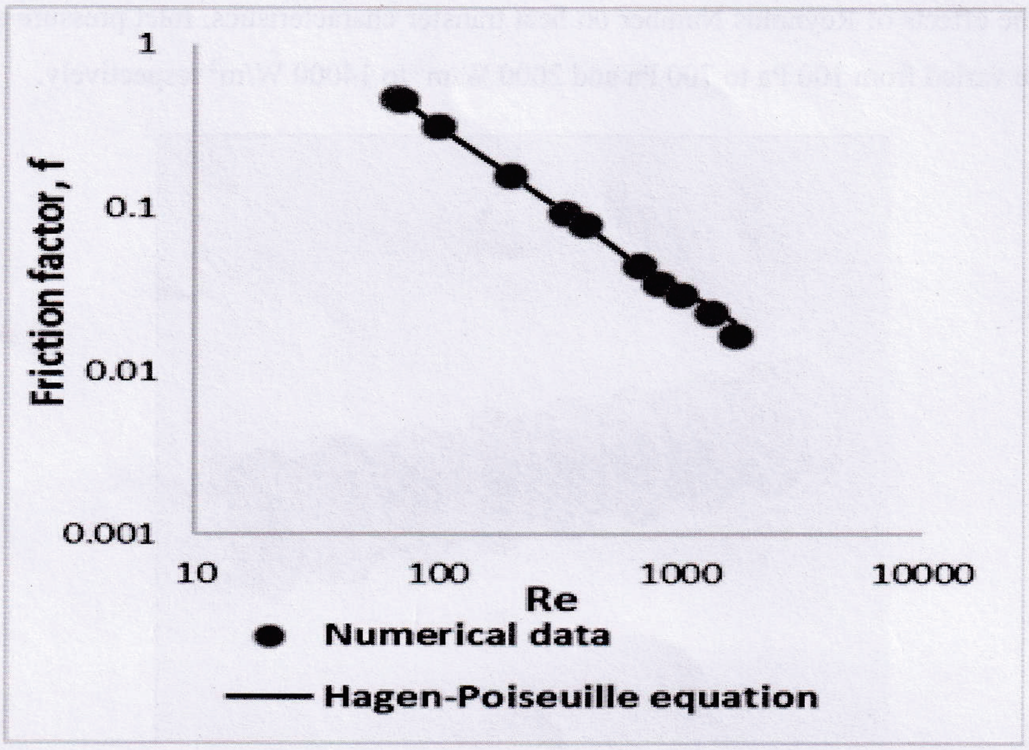

Fig. 3. Numerical and theoretical friction factor in a straight, circular pipe without vanes.

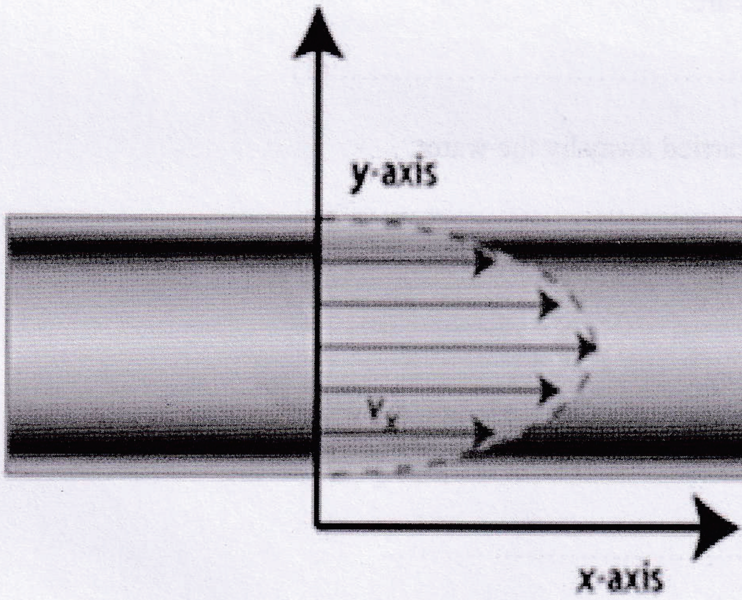

Fig. 4. Theoretical velocity profile for laminar flow.

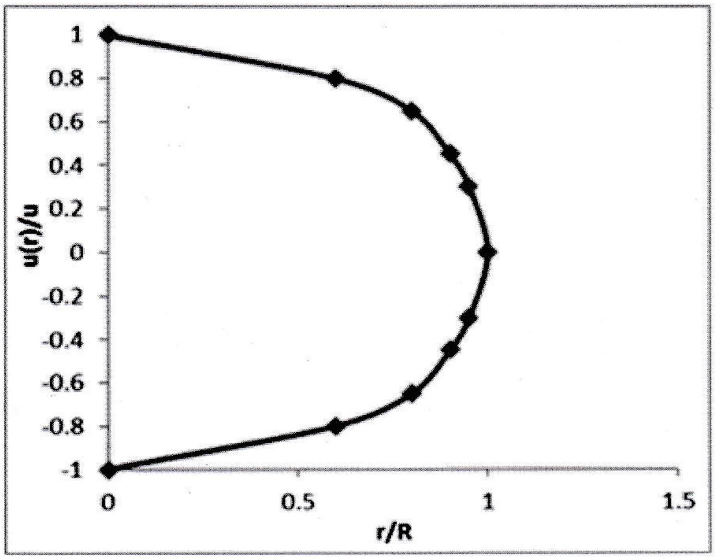

Fig. 5. Velocity profile obtained from numerical study.

To validate the numerical code, laminar flow friction factors in a straight, circular pipe without vanes are calculated from measured pressure drops and flow rates and compared with those given by the Hagen–Poiseuille equation. The comparison shown in Fig. 3 reveals a good agreement between numerical and theoretical results. Furthermore, for fully developed laminar flow, the velocity profile is parabolic as shown in Fig. 4. Fig. 5 shows the velocity profile obtained from the study and it has similar parabolic shape.

4.0. Results and Discussion

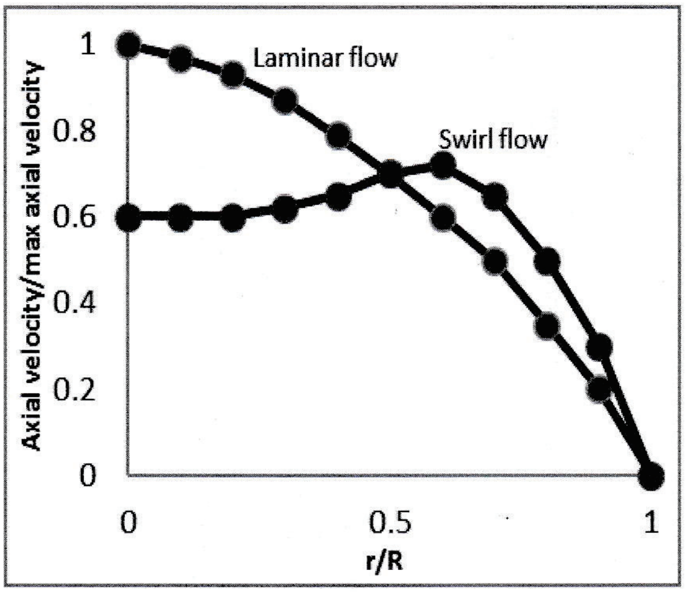

Fig. 6. Radial velocity distribution for x/d=5.

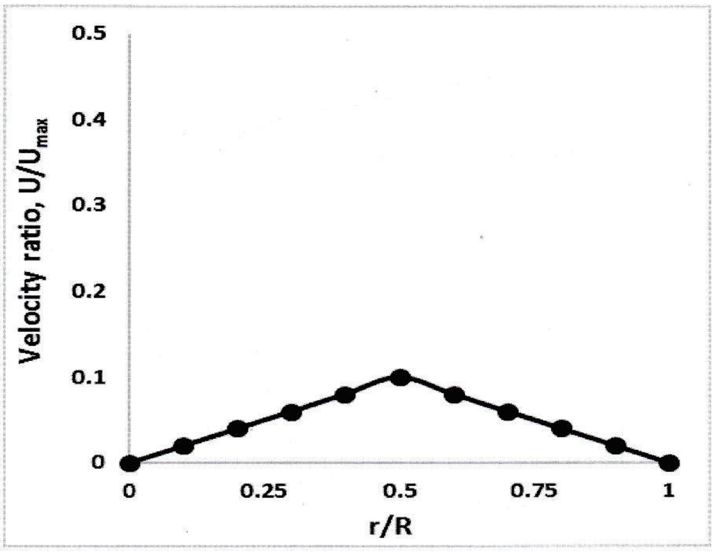

Fig. 7. Tangential velocity distribution for x/d=5.

The axial velocity distribution in the pipe is presented in Fig. 6, where it is revealed that the fully-developed velocity distribution of laminar flow is altered due to the introduction of the swirl. This is due to the destabilizing effect of the swirl. The actual fully-developed profile of laminar flow will be recovered if the vanes are completely removed and the swirl completely disappears. Fig. 7 shows the distribution of the tangential velocity. The tangential velocity is measured along the tangent of the spiral vanes. Due to the presence of this velocity flow rate is increased for swirl flow. This trend of tangential velocity was also reported by Chang et al. (1995) and Bali (1998) for turbulent swirling flow.

4.1 Effects of Swirl Flow

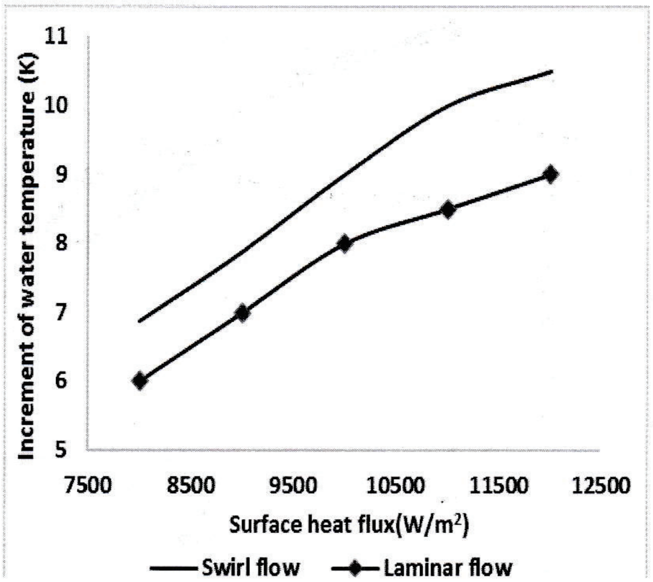

Fig. 8. Increment of water temperature with respect to surface heat flux.

For laminar flow, the maximum local temperature is occurring at the pipe outlet and is greater than that of swirl flow. This is due to the fact that for laminar flow the water layer adjacent to the pipe has higher temperature than that of swirl flow. So, this layer absorbs the highest amount of heat. But for swirl flow different layers of water comes in contact with the pipe at different times. No single layer remains in contact with the pipe at all times. So, no water layer of swirl flow can absorb the same amount of heat as that of laminar flow. Swirl increases the heat transfer due to its stirring effect on water. That is why the average water temperature of swirl flow is greater than that of laminar flow as observed from Fig. 8. It shows that for low heat flux, difference between laminar and swirl flow for the increment of water temperature is low. The difference is higher for high heat flux.

4.2 Effects of Surface Temperature and Surface Heat Flux

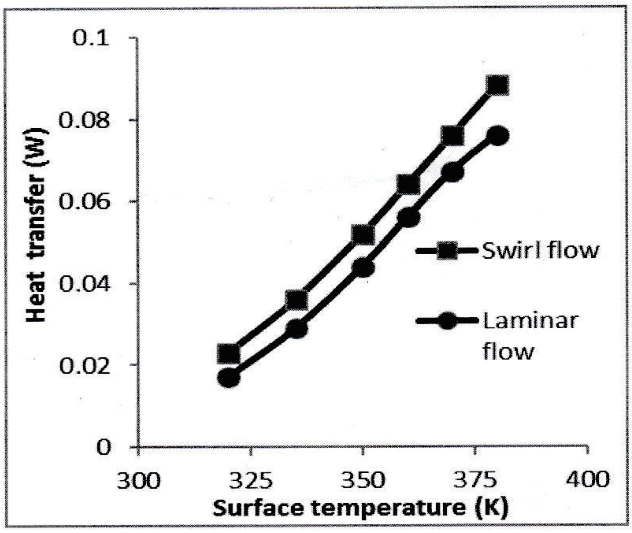

Fig. 9. Heat transfer with the variation of surface temperature.

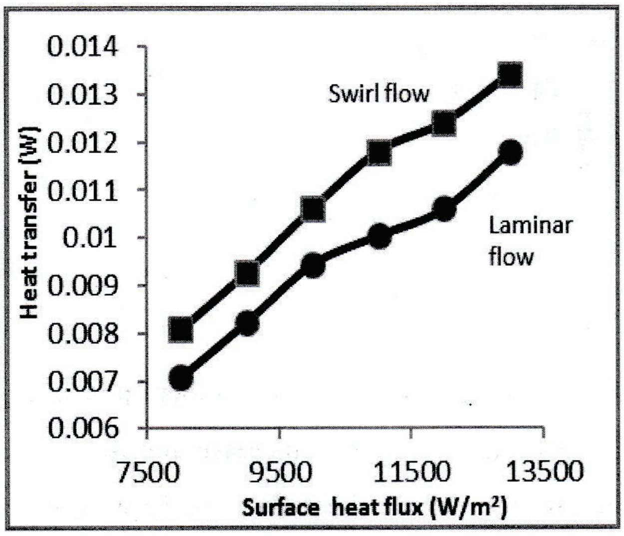

Fig. 10. Heat transfer with surface heat flux.

With the increase of surface temperature and surface heat flux outlet water temperature increases which indicates an increase in heat transfer as shown in Fig. 9 and Fig. 10. In Fig. 9 for different constant surface temperature heat transfer varies almost linearly. However, in swirl flow heat transfer is always higher than that of laminar flow. As the mass flow rate and C_p of water are constant, heat transfer is proportional to temperature increase.

4.3. Effects of Reynolds Number

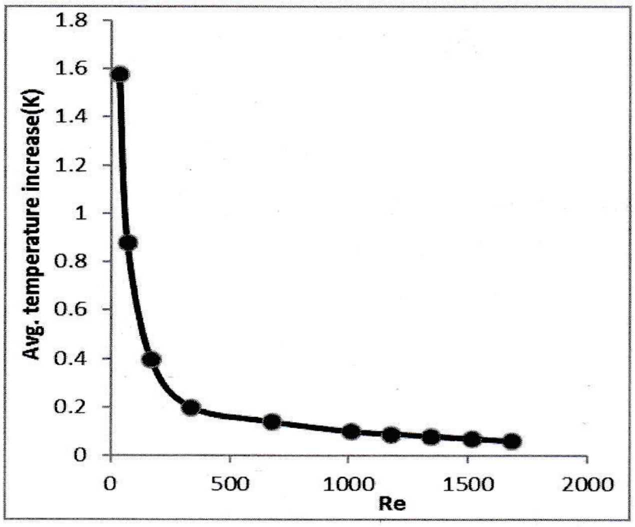

Fig. 11. Effects of Reynolds Number on flow temperature.

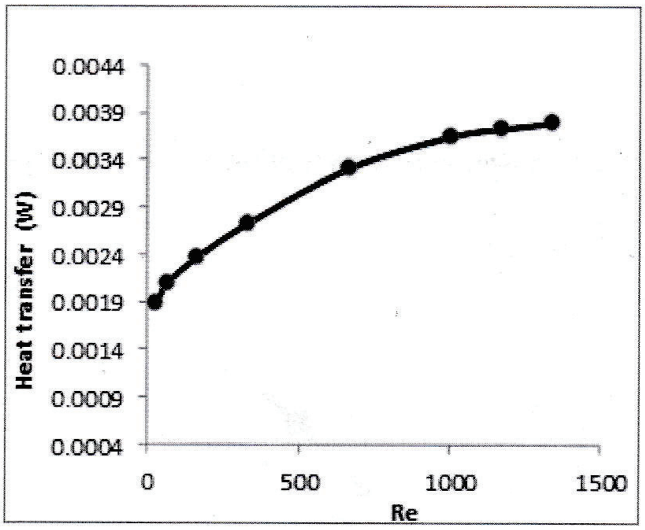

Fig. 12. Effects of Reynolds Number on heat transfer.

The effects of Reynolds Number on temperature increase and heat transfer are shown in Fig. 11 and Fig. 12. The inlet temperature and surface heat flux are kept constant at 298 K and 2000 W/m^2 respectively. With the increase of Reynolds Number, the outlet water temperature decreases because the water has less time to flow through the pipe and absorb heat. This results in a decrease of the temperature difference. However, heat transfer increases with the Reynolds number due to an increase of mass flow rate. The increase of mass flow rate is greater than the decrease of temperature difference. As a result, heat transfer increases.

4.4. Effects of Inlet Pressure

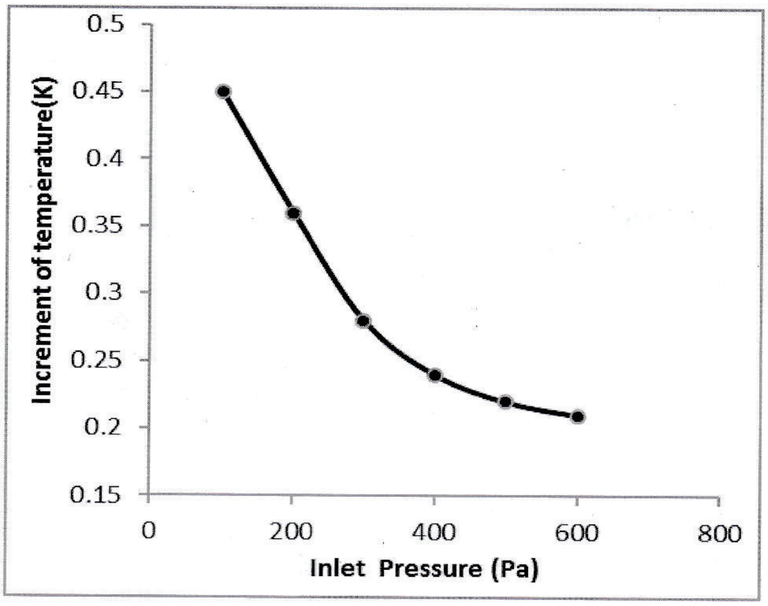

Fig. 13. Effects of inlet pressure on temperature increase.

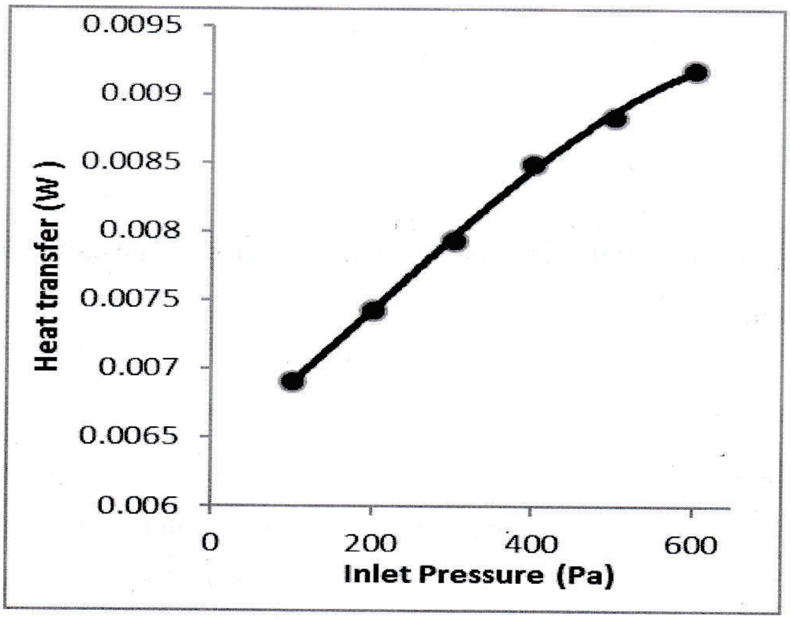

Fig. 14. Effects of inlet pressure on heat transfer.

The effects of inlet pressure are similar to the effects of Reynolds number. This is due to the fact that with the increase of inlet pressure, velocity increases. As a result, flow rate increases and thus heat transfer increases.

4.5. Effects of Twisting Angle

The effects of twisting angles from 22^0 to 35^0 are studied. As shown in Fig. 15, the heat transfer increases with increasing twisting angle. The increase is negligible at first, but after 25^0, heat transfer increases rapidly. With the increase of twisting angle, the tangential velocity increases at nominal constant values of axial velocity.

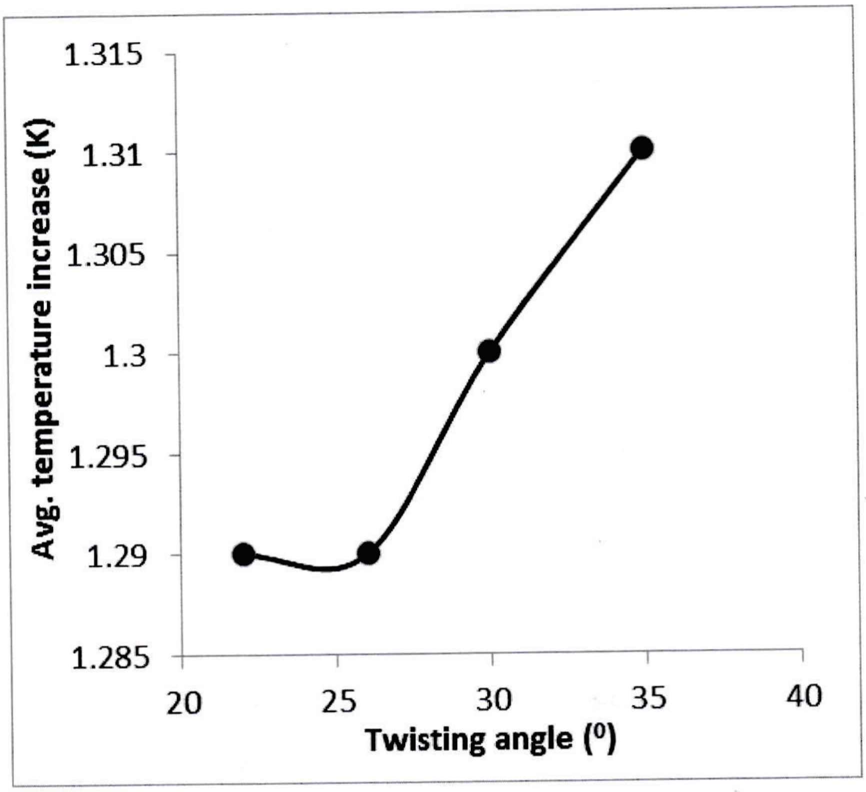

Fig. 15. Effects of twisting angle on average temperature increase.

5.0. Conclusion

The present study is a numerical investigation of heat transfer characteristics of laminar swirling flow through a pipe with continuous vanes. The swirl changes the usual parabolic velocity profile of fully developed laminar flow in the pipe. It is found that swirl flow increases the amount of heat transfer for both surface heat flux and surface temperature. The heat transfer increases with Reynolds Number due to the increase of mass flow rate. The effects of inlet pressure follow a similar pattern. The increase of twisting angle increases the heat transfer due to the increase of stirring effect of the swirl.

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