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An experimental study of the turbulent swirling flow and heat transfer downstream of an sudden expansion in a circular pipe with uniform heat flux

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선회류를 동반한 급확대 원관내에서의 열전달 특성에 관한 실험적 연구

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Summary

Experiments were carried out for the turbulent flow and heat transfer downstream of an abrupt circular pipe expansion. The uniform heat flux condition was imposed to the downstream of the abrupt expansion by using an electrically heated pipe.

Experimental data are presented for local heat transfer rates and local axial velocities in the tube downstream of an abrupt 3 : 1 expansion. Air was used as the working fluid. In the upstream tube, the Reynolds number was varied from 60,000 to 120,000 and the swirl number range (based on the swirl chamber geometry, i.e. L/d ratio) in which the experiments were conducted were L/d=0, 8 and 16. A uniform wall heat flux boundary condition was employed, which resulted in wall-to-bulk temperatures ranging from 24℃ to 71℃. As swirl strenght increased, the location of peak Nu numbers was observed to shift from 4 to 1 step heights downstream of the expansion. This upstream movement of the maximum nusselt number was accompanied by an increase in its magnitude from 2.2 to 8.8 times larger than fully developed tube flow values.

Nomenclature

d : upstream tube diameter
f : friction factor
D : downstream tube diameter
y : distance from the wall
u : local velocity
Tb : bulk temperature
$\overline{u}$: mean axial velocity in upstream tube
L : distance along the plenum chamber
Re : Reynolds number in upstream tube
R : radius of a downstream tube
k : thermal conductivity of air

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x : axial distance from expansion face

w : wall temperature

$\dot{m}$: mass flow rate

Introduction

Experimental research on the heat transfer in regions of seperated and reattached flows inside of pipes and ducts goes back at least to the work of Boetler et al., in 1948.

They reported maximum heat transfer coefficients near the point of reattachment about four times the fully developed flow values.

Thereafter, without swirl, flow through a sudden expansion produces mixing rates and subsequently, heat transfer coefficients which are substantially higher downstream of the expansion than those which would be obtained at the same Reynolds number in the entrance region of pipe. (Ede et. al., 1956, Krall and Sparrow, 1966, Zemanick and Dougall, 1970, Baughn et. al., 1984).

Swirl is also responsible for increased shear rates, greater turbulence production, and longer path lengths for a particular fluid particle so that the effect of swirl like the effect of the sudden expansion, is also to increase heat transfer rates significantly over those found in purely axial pipe flow. (Syred and Beer, 1974, Hay and West, 1975, Sparrow and Chaboki, 1984, Chang and Kwon, 1988).

Recently, Dellenback et al. (1987) reported an ambitious investigation of the flowfield and related heat transfer behavior for the present problem, incluiding swirl. (Habib and McEligot, 1982, Sultanian, 1984, Dellenback et. al., 1987).

However, many studies of heat transfer of the swirling flow or unswirled flow in a abrupt pipe expansion are widely carried out, the mechanism is not fully found evidently due to the instabilities of flow in a sudden change of the shape, appearance of turbulent shear layers in a recirculation region and secondary vortex near the corner.

Thus the purpose of this study is to obtain new accurate data throughout an experimental study of the swirling flow and heat transfer downstream of an abrupt expansion in a circular pipe with uniform heat flux.

Experimental Apparatus and Procedure

The entire experimental apparatus is shown schematically in Fig. 1.

An experimental test-rig was manufactured to permit a detailed interrogation of all flow variables. The rig incorporated a specially designed swirl generator, fitted to the inlet of a perspex circular pipe, enabling varying intensities of swirl flow to be stimulated over the Reynolds number range of $60\text{-}120\times10^3$ in the upstream tube.

An identical pipe, manufactured out of copper, enabled a constant heat flux to be applied at its outer surface, thereby permitting a corresponding investigation of the heat transfer phenomena.

1. Pressure and Velocity measurement

The acrylic test section used for the velocity and pressure measurements is a plexiglass cylinder of internal diameter 150mm and thickness of 5mm, lenght of 3,000mm.

If X denotes the axial distance along the test section and d denotes the internal diameter of the test pipe then X/d locates a station measurement position in non-dimesnional form on the test section. The

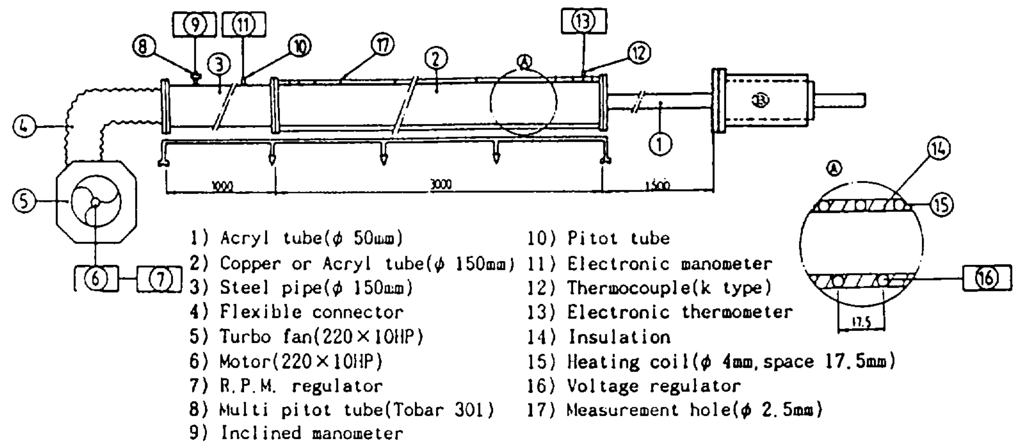

Fig. 1. Schematic diagram of experimental apparatus

inlet is located at X/d=0 and the outlet located at X/d=60.

The unheated upstream tube(inside diameter 50mm) and the expansion face were the same plexiglass components used in the flowfield measurements.

The swirl generator is located upstream of the test section and is comprised of a plenum chamber and a swirl generator. The plenum chamber is a plexiglass cylinder of outer diameter 200mm and thickness of 5mm, length of 1,500mm. The plenum chamber is attached to the test section by wooden flanges and the generator. The section of the swirl generator is shown on Fig. 2. To create swirl, a swirl generator is placed within the plenum chamber. The generator consists of a 165.5mm outside diameter plexiglass cylinder with a 6mm wall thickness and 240mm length. The disign of the generator is based on that used by Chaboki(1983), producing a near perfect axisymmetric swirl.

The pressure distrubution along the length of the test section is obtained by a series of pressure tappings located on the test pipe connected to a multimanometer by plastic

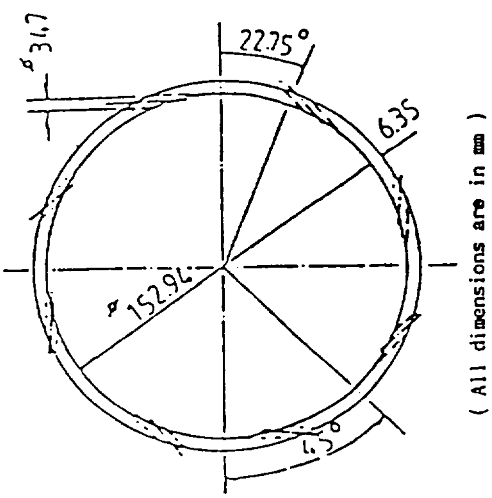

Fig. 2. A cross section of the swirl generator

tubing.

The blower is located downstream of the test section which produces the air suction. For friction factor measurement an identical apparatus is used and the pressure gradients are obtained along the length of the test section and substituted into the friction factor relation described later.

The static pressure signals were then conveyed to the multimanometer by plastic tubing. Air is drawn through the generator and through the tangentially cut holes, resulting in the generator of swirl.

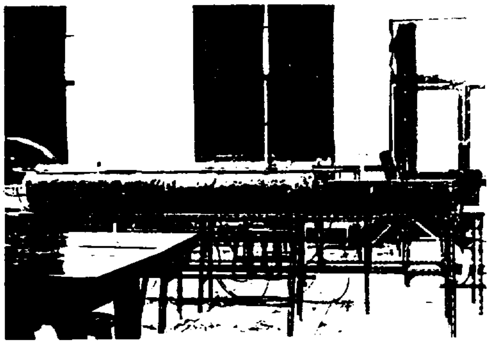

Fig. 3. Photograph of the test tube mounted with heating coil

2. Fluid Flow Measurement

The torbar 301 pressure gauge is also located between the test section outlet and the blower. The torbar 301 gauge enables the correct Reynolds number flow to be ascertained by connecting to and inclined manometer and obtaining the pressure difference in mm WG required for a particular Reynolds mumber flow. The flow is adjusted by a control valve located on the blower. The flow rate was measured downstream of the test section by a self averaging pitot tube, the torbar 301. A honeycomb mesh was used to control the flow characteristic at the inlet of the impact tube.

These flow-measuring devices were connected via tygon tubing to a micromanometer capable of measuring pressure differences.

3. Temperature Measurement

The copper heat transfer test section was of the same dimensions as the acrylic test section used for the velocity and pressure measurements.

The copper electrodes were connected to a variable transformer in series with a 240V AC regulator, allowing an adjustable, and stable, AC voltage to be dialed into the electrodes.

Voltages used in the experiments ranged 240V AC, providing a total power to the heated tube of 1,750W. This, in turn, gave heat fluxes of 0.011 W/cm^2.

The local tube wall temperatures were measured with thermocouples (0.08mm C-A) epoxied to the back side of the Intrex through small holes in the tube wall. There were 15 calibrated thermocouples located at different axial positions along the tube. Also, measurement of the radial temperature fields could then be accomplished by installing the probe in the traversing mechanism.

4. Data reduction

The Standard formula for calculating velocity from velocity pressure is

$$u = 1.291\sqrt{Pv} \qquad (1)$$

The measured pressure distributions were used to evaluate local apparent friction factors.

The friction factor was obtained by local application of its equation of definition

$$f = (-dp/dx)D/1/2\rho\,\bar{u}^2 \qquad (2)$$

where $\rho\bar{u}^2 = (\dot{m}/1/4\pi D^2)^2/\rho$, and ρ is the local density at the point of interest.

Heat transfer tests were performed in a horizontal copper tube by passing AC current in the tube wall.

Local heat transfer coefficients were evaluated from the basic definition

$$dq = \dot{m}\ Cp\ dTb \qquad (3)$$

$$h = \frac{\dot{m}\ Cp\ dTb}{2\pi rdx\,(Tw-Tb)\,mean} \qquad (4)$$

The local heat flux is designated by q, the local wall temperature by Tw and the local bulk temperature by Tb.

A dimensionless representation was made in terms of the local Nusselt number

$$Nu = \frac{hD}{k} \qquad (5)$$

Here, D is the downstream tube diameter , k is the thermal conductivity of air at the local bulk temperature.

The local convective heat flux q was determined by correcting the electric power input for axial conduction in the tube wall and for losses through the insulation that surrounds the test section. The value of Tb at any axial station X was found from an energy balance on the fluid, using a control volume which extended from X=0 to X=X and integrating the aforementioned local q values to obtain the heat input to the control volume. Values of the local wall temperature Tw were available from the experimental data.

Results and Discusson

1. Axial velocity distributions

Fig. 4 shows the axial velocity variation with radial distance along the pipe for no swirl flow and Re=100,000. For Re=100,000, it can be seen that the axial velocity distribution rise from tube wall to tube centre and as y/R>0.65 rises rapidly.

Fig. 5 shows the axial velocity variation with radial distance along the pipe for swirl flow and Re=80,000 with a concentric expansion for L/d=0. At L/d=0 the flow pattern resembles that at S=1.23, Re=100,000 by Dellenback(1987). It can be seen that with increasing axial distance the highest axial velocities move toward the tube wall.

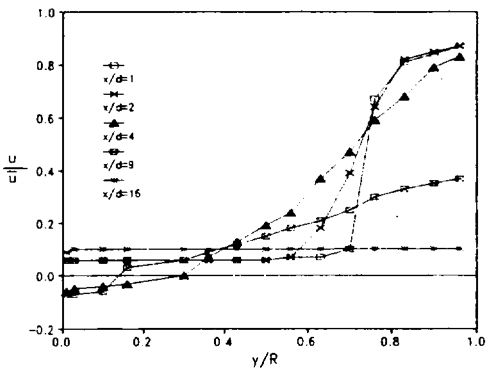

Fig. 4. Axial velocity profiles for Re=100000 with a concentric expansion

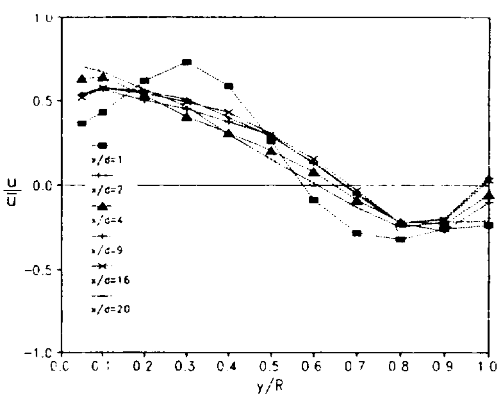

Fig. 5. Axial velocity profiles for Re=80000 with swirling flow at L/d=0

2. Friction factor

Fig. 6-1 shows the friction factor distributions with axial distance along the pipe as Reynolds number varies from 60,000 to 120,000 in unswirled flow. For all Reynolds number, friction factor of the test tube showed a constant value from x/d=20 regardless of the inlet flow boundary condition.

Fig. 6-2 shows the friction factor distributions with axial distance along the pipe as Reynolds number varies from 60,000 to 120,000, swirling flow at L/d=8, heat flow=1.75kW. For all Reynolds number, the friction factor of the test tube decreased rapidly at x/d=4 regardless of the inlet flow boundary condition in swirled flow. The value of friction factor showed a slight increase in the entrance region of pipe when the Reynolds number was larger than any other Reynolds numbers. It shows a gradient increase up to x/d=4 but a sharp decrease between x/d=4 and x/d=15, showing a constant consistency.

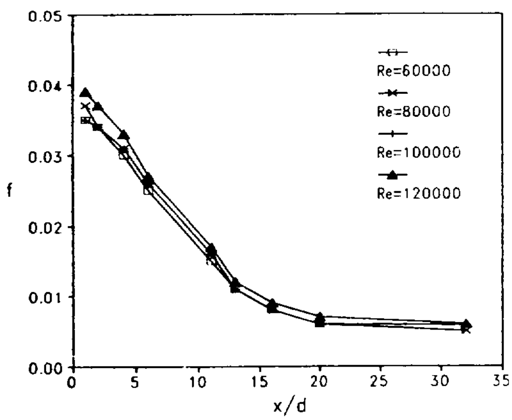

Fig. 6-1. Friction factor distribution for various Reynolds numbers with a concentric expansion

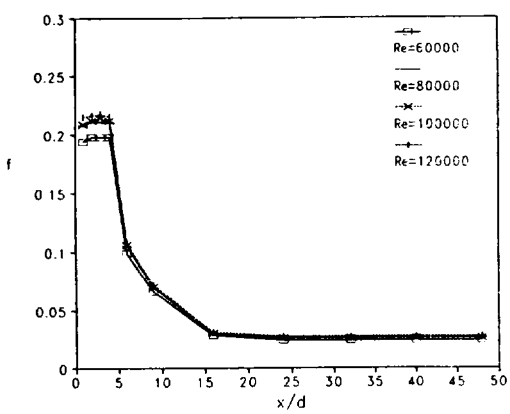

Fig. 6-2. Friction factor distribution for various Reynolds numbers with swirling flow at L/d=8, Heat flow=1.75kw

3. Temperature distributions

Fig. 7-1 and 8-1 show the wall and fluid bulk temperature variation with for no swirl flow as Reynolds number varies from 80,000 to 120,000. For all Reynolds number, the wall temperature showed a curve of parabolic variation at $24 \le x/d \le 59$. The bulk temperature showed a minimum value at x/d=6~9, but it showed a linear distribution of increase. In addition, the values of Tw, Tb showed a greater increase at Re=80,000 than any other Reynolds numbers.

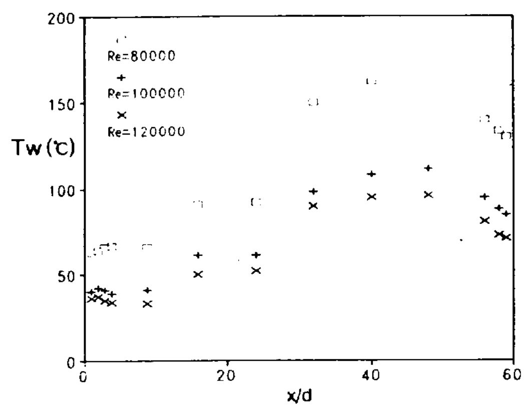

Fig. 7-1. Distribution of wall temperature along the test tube with the abrupt expansion

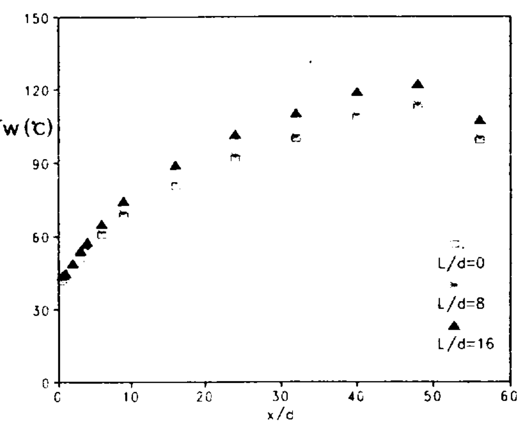

Fig. 7-2. Distribution of wall temperature along the expansion for Re=60000 with swirl

Fig. 7-2 and Fig. 8-2 show the wall and fluid bulk temperature variation with axial distance along the pipe for swirl flow. It shows that the wall temperature initially rises rapidly, then reduces, until the distribution of Tw becomes linear with axial distance, whereas the fluid bulk temperature exhibits a linear increase with axial position virtually from the pipe inlet. The temperature distributions give some qualitative suggestions concerning the phenomenon being studied, ie., heat transfer effectiveness downstream of an abrupt expansion.

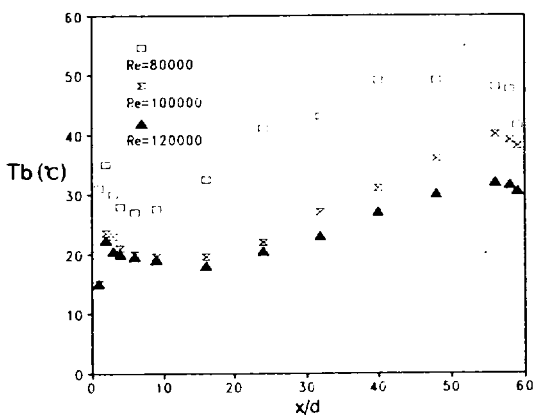

Fig. 8-1. Distribution of bulk temperature along the test tube with the abrupt expansion

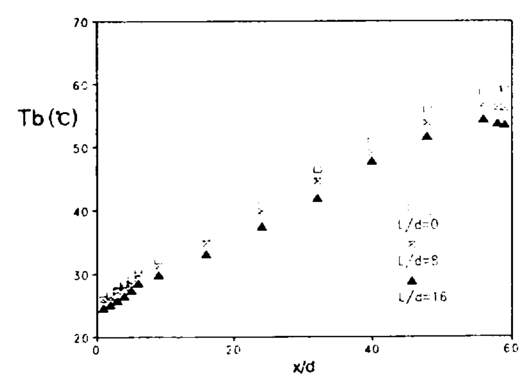

Fig. 8-2. Distribution of bulk temperature along the test tube expansion for Re=80000 with swirl

Nusselt number is inversely proportional to the wall-to-bulk temperature difference for a fixed heat flux and constant properties.

For all Reynolds number, the value of Tb showed a minimum value at the pipe inlet, it also exhibited a linear increase with axial distance along the pipe. The value of Tb showed a greater increase at L/d=0 than any other L/d ratios. It is seen that an effect of swirl is to increase the Tb.

4. Nusselt Number variations

Fig. 9-1 and 9-2 show the measured axial variations in normalized Nusselt number as a function of swirl strength for nominal Reynolds of 60,000 and 80,000, respetively. The peak Nusselt numbers increase consistently in magnitude and move upstream with increasing swirl strength.

This upstream migration of Nu_m is a direct result of the shortening reattachment length. The shortening of the recirculation region causes shear rates and hence production of turbulence kinetic energy to increase with consequently higher heat transfer rates. In Figs 9-1 and 9-2, the Nu/Nu_{db} ratio showed a greater increase at L/d=0 than any other L/d ratios. These features are also demonstrated by Dellenback (1987).

In Figs 9-1 and 9-2, the Peak Nu/Nu_{db} occurred at 1 step height and it showed a greater increase at Re=60,000 than at Re=80,000.

In addition, comparison of the unswirled flow result from Figs. 9-1 and 9-2 demonstrates that larger enhancements in heat transfer rates over straight pipe flow occur at lower Reynolds numbers. This feature is simply rationalized by the observation that convection heat trasfer behavior in separated flow is commonly found to depend on $Re^{2/3}$

Nu_{db} : fully developed Nusselt number for turbulent pipe flow represented by Dittus-Boelter corelation.

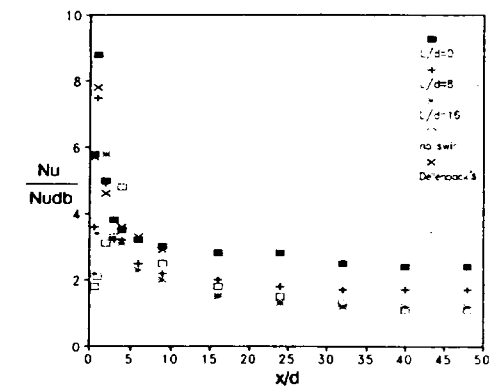

Fig. 9-1. Distribution of Nu/Nu_{db} along the test tube abrupt expansion for Re=60000 with swirl

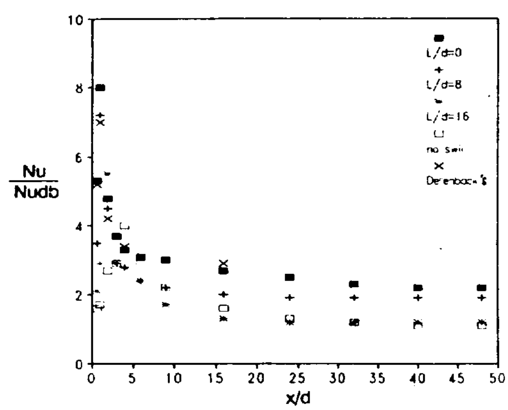

Fig. 9-2. Distribution of Nu/Nu_{db} along the test tube abrupt expansion for Re=80000 with swirl

Conclusions

Experiments were carried out for the turbulent swirling flow and heat transfer downstream of an abrupt circular pipe expansion.

The uniform heat flux condition was imposed to the downstream of the abrupt expansion by using an electrically heated pipe. The following conclusions are drawn from the test data.

1. The location of the peak Nu/Nu_{db} showed at the point of 1 step height for Re =60, 000, 80, 000 in the swirling flow abrupt concentric expansion, whereas it showed at the point of 4 step heights for same Reynolds number in unswirling flow expansion.

2. Axial velocity increased rapidly at r/R= 0. 65 in the abrupt concentric expansion turbulent flow through the test tube in unswirled flow, whereas it showed that with increasing axial distance the highest axial velocities move toward the tube wall in the case of the swirling flow abrupt expansion.

3. Friction factor of the test tube showed a constant value at x/d=13 and regardless of the inlet flow boundary condition in unswirled flow, whereas it showed a gradient increase up to x/d=4 but a sharp decrease between x /d=4 and x/d=15, showing a constant consistency.

4. In the turbulent swirling flow abrupt concentric expansion, the wall temperature and bulk temperature showed a minimum value at the pipe inlet, and the bulk temperature also exhibited a linear increase with axial distance along the pipe but the wall temperature initially rises rapidly.

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〈국문초록〉

선회류를 동반한 급확대 원관내에서의 열전달 특성에 관한 실험적 연구

실험데이터는 급확대비 3 : 1 팽창의 시험관에서의 실험결과를 나타내고 있으며, 실험에 이용된 동작유체로써 공기가 사용되었다. 입구관에서 레이놀즈수는 60,000으로부터 120,000까지 변하게 하였고, 스월강도는 0으로부터 16까지 변화되게 하였다. 균일한 열 플럭스 경계조건이 사용되었는데, 그 결과 관벽온도 및 체적온도는 24℃로 부터 71℃까지에 걸쳐 나타났다. 플롯상에 국소 Nusselt수는 최대 열전달점에서 정점을 이루는 모습을 보여 주고 있다. 스월강도가 0으로부터 최대값으로 증가되었을때 최고 Nusselt수의 위치는 시험관에서 4로부터 1스텝하이트로 변경되는것이 조사되었다. 이러한 최대 Nusselt수의 상류부 이동은 완전 발달된 유동에서의 값보다 2.2배에서 8.8배나 많은 그의 크기를 증가시킨다고 할수 있다.