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# THERMOHYDRAULIC ANALYSIS OF CONCENTRIC ANNULAR TUBE HAVING NARROW GAP

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**ABSTRACT** Convective heat transfer and single phase water flow in concentric annular tubes having narrow gap were numerically and experimentally investigated. The inner diameter of copper tube having 1mm wall thickness is fixed 18 mm. The rodes having diameter of 16, 17 ve 17.5 mm are concentrically inserted in the tube. The gap of the annular duct is fixed at 1, 0.5 and 0.25 mm. The numerical study was based on the Finite volume method. Reynolds Stress model is used for turbulent region. The Reynolds number was chosen in the range of 600-12000. The results are compared with the conventional correlations presented for macro annular ducts in literature. The results show that heat transfer and single phase water flow in mini/micro annular ducts are different from those of macro-sized annular ducts for turbulent regime as it is in good agreement in laminar regime. For laminar and turbulent flow, the ratios of experimental to numerical values are found about 0.97 -1.01 and 1.08 - 1.3 for friction factor and 1.05-1.17 and 0.97 - 1.24 for Nusselt number, respectively. For turbulent flow, the numerical and experimental values are found bigger about 9-36% and 18-36% for Nusselt number and 23-43% and 45-69% for friction factor than conventional results presented in literature for macro- sized annular duct, respectively.
**Keywords:** Annular, mini- micro channel, heat transfer, flow.

## DAR ARALIKLI EŞ EKSENLİ HALKA KESİTLİ BORUNUN TERMOHİDROLİK ANALİZİ

**ÖZET** Dar aralığa sahip eş eksenli halka kesitli borularda tek fazlı su akışı ve ısı transferi sayısal ve deneysel olarak incelenmiştir. 1 mm et kalınlıklı bakır borunun iç çapı 18 mm'dir. Boru içine eş eksenli olarak yerleştirilmiş içi dolu silindirin çapı 16,17 ve 17.5 mm'dir. Oluşan halka kanalın aralığı 1,0.5 ve 0.25 mm'dir. Sayısal çalışma sonlu hacimler yöntemine dayanmaktadır. Türbülanslı akışlar için Reynolds Stress modeli kullanılmıştır. Reynolds sayısı 600-12000 aralığında alınmıştır. Sonuçlar literatürdeki makro halka kanallar için sunulan geleneksel bağıntılarla karşılaştırılmıştır. Sonuçlar mini/mikro halka kanallardaki ısı geçişi ve tek fazlı su akışının laminer akışta uyumlu iken türbülanslı akış için makro kanallardakinden farklı olduğunu göstermiştir. Laminer ve türbülanslı akış için, deneysel değerlerin sayısal değerlere oranı sürtünme faktörü için yaklaşık 0.97 -1.01 ile 1.08 - 1.3 ve Nusselt sayısı için 1.05-1.17 ile 0.97 - 1.24 aralığında bulunmuştur. Bununla birlikte, türbülanslı akış için, sayısal ve deneysel değerler literatürde mevcut olan makro boyutlu halka kanal sonuçlarından Nusselt sayısı için yaklaşık 9-36% ile18-36%, sürtünme faktörü için yaklaşık 23-43% ile 45-69% oranında daha büyük bulunmuştur.
Anahtar kelimeler: Halka, mini- mikro kanal, ısı geçişi, akış.

## NOMENCLATURE

| | |
|---|---|
| $A$ | surface area of tube [m$^2$] |
| $C_p$ | specific heat [J/kg K] |
| $d$ | diameter of tube [m] |
| $f$ | Darcy friction factor |
| $k$ | thermal conductivity [W/m K] |
| $Nu$ | Nusselt number |
| $Pr$ | Prandtl number |
| $Q$ | heat transfer rate [W] |
| $Re$ | Reynolds number |
| $T$ | temperature [K] |
| $\vec{V}$ | velocity vector |
| $\bar{V}$ | mean velocity [m/s] |

**Greek letters**

| | |
|---|---|
| $\rho$ | density [kg/m$^3$] |
| $\nu$ | kinematic viscosity [m$^2$/s] |

**Subscripts**

| | |
|---|---|
| $b$ | bulk |
| $c$ | cross- sectional |
| $w$ | wall |

| | |
|---|---|
| *i* | inner |
| *o* | outer |
| *r* | rod |
| *h* | hydraulic |
| *in* | inlet |
| *out* | outlet |

## INTRODUCTION

In HVAC systems, mini/micro channels are preferred because of their low thermal resistance, very high ratio of surface area to volume, low inventory, small volumes and lower total mass of working fluids (Chen et al., 2010). Thus, in the last decades, many researchers have investigated the phenomenon of the flow and heat transfer in mini/micro ducts. Kandlikar (2002) determined that the channels having $d_h$ > 3 mm, 200 μm < $d_h$ < 3 mm and 10 μm < $d_h$ < 200 μm are conventional/macro, mini and micro channels, respectively.

So many investigations on boiling and multiphase flow in mini/micro channels (Kandlikar, 2002; Balasubramanian and Kandlikar, 2005; Celata et al., 1993; Boye et al., 2007; Pehlivan et al, 2006) have been presented, but single-phase water flow in mini/micro channels are still investigated rare. Several comparisons of the Nusselt number and friction factor values obtained for mini/micro channels with conventional correlations presented in literature for macro sized channels were presented by many researchers (Sobhan and Garimella, 2001; Adams et al.,1998). They point out the differences between the results of the small and macro sized channels. Adams et al. (1998) investigated the heat transfer of single-phase turbulent water flow through circular microchannels having 0.76 and 1.09 mm diameter. They found that Nusselt number is higher than the values calculated from conventional Gnielinski equation. Thus, they modified Gnielinski correlation using their experimental findings. Caney et al. (2007) studied experimental flow and heat transfer during single-phase flow in a vertical mini channel to test of applicability of conventional correlations presented for macro sized channels. They found that the heat transfer results are in fair agreement with the conventional results as friction results are in good agreement.

Steinke and Kandlikar (2006) reviewed the studies investigating the single-phase friction factor in micro channels and compared with their experimental results. They concluded that conventional friction factor correlations are valid in laminar regime for the channel diameters considered. Lelea et al. (2004) numerically and experimentally studied the microchannel heat transfer and fluid flow for distilled water flowing with Re-number range up to 800 through 0.1, 0.3 and 0.5 mm diametered microtubes. They determined that the conventional theories including the entrance effects are applicable for the case considered. Parlak et. al (2011) experimentally and numerically investigated the viscous heating effect at single–phase laminer water flow in microtubes having diameters of 50-150 μm.

The above literature review clearly indicates that the studies for the convective heat transfer and single phase water flow in the mini/micro ducts are rare. Moreover, it can't be still reached a consensus. Thus, in this paper, the heat transfer and single phase flow in a horizontal annular duct having mini/micro gap are analyzed experimentally and numerically. The effects of the mini/micro gaps in the annular tubes on the heat transfer and single phase water flow will be examined in detail. The experimental and numerical results are compared with each other and the conventional results available for the macro annular ducts.

## MATHEMATICAL MODEL

The numerical solution is accepted as three-dimensional. It is conducted ¼ part of the geometry due to the axial symmetry. The flow is steady state, single phase and incompressible. According these assumptions, the continuity, momentum and energy equations for laminar regime are given as

$$\frac{\partial(\rho.u_i)}{\partial x_i} = 0 \tag{1}$$

$$u_j \frac{\partial(\rho.u_i)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu \frac{\partial u_i}{\partial x_j}\right) \tag{2}$$

$$C_p u_j \frac{\partial(\rho.T)}{\partial x_j} = \frac{\partial}{\partial x_j}\left(k \frac{\partial T}{\partial x_j}\right) \tag{3}$$

For the turbulent regime

$$\frac{\partial(\rho.\bar{u}_i)}{\partial x_i} = 0 \tag{4}$$

$$u_j \frac{\partial(\rho.\bar{u}_i)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left((\mu + \mu_t) \frac{\partial \bar{u}_i}{\partial x_j}\right) \tag{5}$$

$$C_p u_i \frac{\partial(\rho.T)}{\partial x_i} = \frac{\partial}{\partial x_j}\left((k + k_t) \frac{\partial T}{\partial x_j}\right) \tag{6}$$

where ρ, T, k, $k_t$, P, μ , $\mu_t$, $u_i$ and $\bar{u}_i$ are fluid density, temperature, thermal conductivity, turbulent thermal conductivity, pressure, dynamic viscosity, turbulent dynamic viscosity, i-axis velocity component and its time-averaged value, respectively. For Reynolds Stress model (Launder et al., 1975), the turbulent kinetic energy and its dissipation rate are given as

$$\frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \frac{1}{2}(P_{ii} + G_{ii}) - \rho\varepsilon(1 + 2M_t^2) + S_k \tag{7}$$

$$\frac{\partial(\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] C_{\varepsilon 1}\frac{1}{2}(P_{ii} + C_{\varepsilon 3}G_{ii})\frac{\varepsilon}{k} - C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k} + S_\varepsilon \tag{8}$$

where turbulent viscocity, $\mu_t$

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{9}$$

where $C_\mu$=0.09, $\sigma_k$=0.82, $\sigma_\varepsilon$ =1.0, $C_{\varepsilon 1}$=1.44, $C_{\varepsilon 2}$ =1.92, $S_\varepsilon$ and $S_k$ is a user-defined source terms.

Table. 1 shows the physical model and dimensions of the geometry. The inner diameter of the tube made of copper with 1 mm wall thickness ($d_i$) is fixed 18 mm. The diameter of the rod placed coaxially ($d_r$) is 16, 17 and 17.5 mm, respectively. Thus, the gaps occurred between the rod and tube wall are 1, 0.5 and 0.25 mm. The lengths of tube and rod (*L*) are 250 mm.

The assumptions for the study are:

- Steady state, incompressible and single phase water flow
- The Reynolds number in the range of 600-12000
- The constant thermophysical properties of fluid
- The water having constant temperature at the inlet of tube ($T_{in}$= 349 K).
- The uniform velocity calculated to Reynolds number studied at the inlet of tube ($u_{in}$=constant)
- The adiabatic condition on the rode surface
- No- slip conditions on the rod and tube walls (u,v,w=0).
- The relative pressure is zero at the outlet of tube.
- The liquid boiling of n-pentan ($C_5H_{12}$) in a pool is considered as surrounding medium in order to obtain the maximum heat transfer coefficient, in other words; the minimum thermal resistance at the outer surface of tube ($T_{sat}$=309 K)
- The outside heat transfer coefficient which depends on the outer surface temperature of tube
- Symmetry condition on the symmetric surfaces

In this study, the computations were done for the Reynolds number which ranges from 600 to 12000. Reynolds Stress model (Launder et al., 1975) was used for the solution of turbulent region. The RSM is the most accurate turbulence model due to be eliminated the isotropic eddy-viscosity hypothesis, thus the RSM closes the Reynolds-averaged Navier-Stokes equations by solving transport equations for the Reynolds stresses together with an equation for the dissipation rate (Fluent Users Guide, 2001). Simple algorithm was applied with finite volume method approach (Patankar, 1980). Segregated method was used for all governing equations due to the incompressible flow. GAMBIT was used for the structures of geometry and grid. Numerical solutions are obtained by using FLUENT 6.1.22.

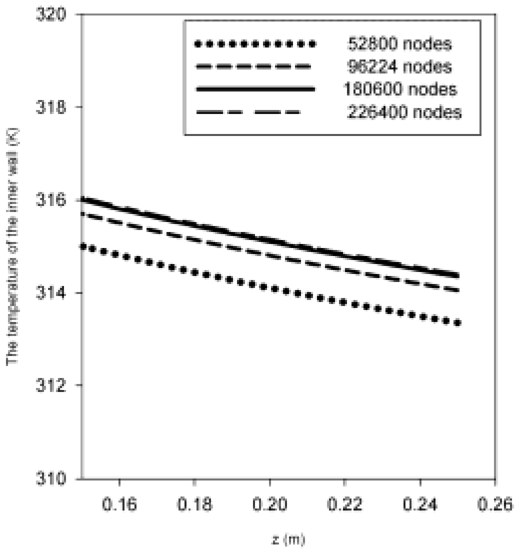
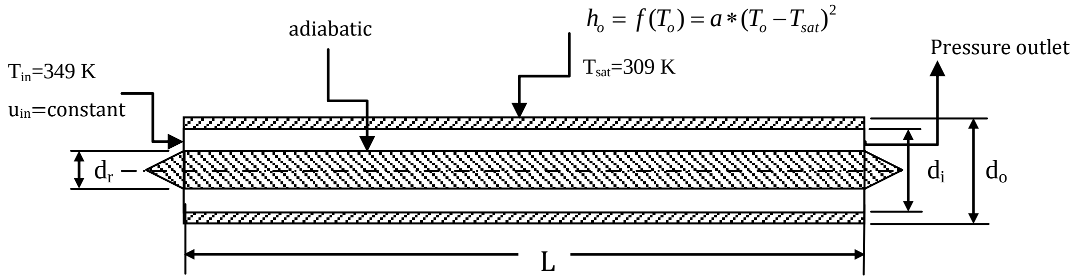



**Fig. 1.** The grid independency study.

For grid independence, four different cases varied the number of nodes from 52800 to 226400 are examined. Fig.1 shows the temperature of the inner tube wall along tube length for all the studied grid structures. As is seen from this figure, the wall temperature doesn't change after the case of 180600 nodes. It is found that after 180600 nodes the relative errors between the fine and coarse grid solution are found 1.5% for Nusselt number (*Nu*) and 2% for friction factor (*f*), respectively. Thus, the grid structure of 180600 nodes was accepted for the numerical calculations.

**Table1.** Dimensions of the annular duct

($d_o$ = 20 mm, $d_i$ = 18 mm, L = 250 mm)




| $d_r$ (mm) | 16 | 17 | 17.5 |
|---|---|---|---|

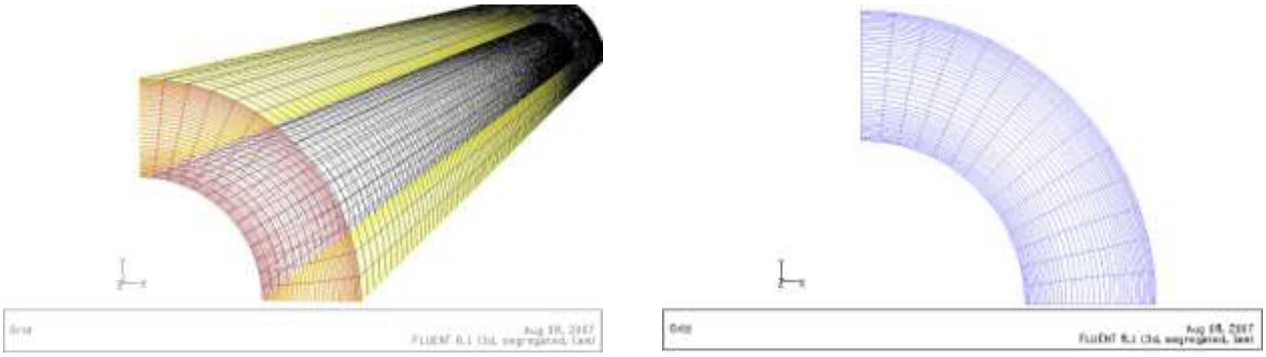


**Fig. 2.** The grid structure of the solution domain (Ugurlubilek, 2007).

Fig. 2 shows the grid structure of the computational solution region. It was performed more frequent nodes near the walls due to the importance of the boundary layer.

Morini and Spiga (2007) and Koo and Kleinstreuer (2004) found that viscous dissipation effect has a major effect only for $d_h$< 50µm for water flow. Celata et al. (2006) also said that this limit is $d_h$<100 µm. In this study, the hydraulic diameter ranges of annular tube are considered 2 mm ≤ $d_h$ ≤ 500 µm. Thus in this paper, viscous dissipation effect is neglected.
At the surrounding medium, the liquid n-pentan having a saturated temperature of 309 K boils as a result of the heat transfered from the water ($T_{in}$=349 K) flowing through the tube. The outside heat transfer coefficient as a function of the outside temperature of tube ($T_o$) is given as

$$h_o = f(T_o) = \mathrm{a} * (T_o - T_{sat})^2 \tag{10}$$

where a and $T_{sat}$ are a constant and the saturated temperature of n-pentan, 131.006 and 309 K, respectively (Ugurlubilek, 2007). In this study, Eq. (10) was coded by C ++ programme and it was numerically calculated interpreting into the Fluent 6.1.22 as a user-defined function.

## EXPERIMENTAL STUDY

The main purpose of this study is to investigate applicability of the numerical software and conventional macro-sized annular duct correlations to mini/micro annular ducts. Thus, because of the lasting and costly experiments, the tests are conducted for sufficiently geometry instead testing of all geometry calculated numerically by Fluent. For that purpose, the annular ducts having gaps of 1 and 0.25 mm are only tested in the experiments. The flow and heat transfer data obtained are compared with results of numerical analysis and traditional correlations for macro-sized annular tubes.

A schematic view of the experimental set up was shown in Fig. 3. It consists of a test tank into which two tubes were fitted as condencer and evaporator. These pipes pass through the test tank in radyal direction. One of those is evaparator pipe and it is set at the bottom of tank. It is immersed into liquid n-pentan in depth about 200 mm. Other one is condenser pipe and it condences n-pentan vapoured in process of experiment. For all the experiments, n-pentan boils at atmospheric pressure and its saturated temperature of 309 K. The technical drawing of the test tank of stainless stell (A321) is also detailed in Fig. 4.

The investigated region in this study is the evaporator copper tube having 18 mm inner diameter, 250 mm length and 1 mm wall thickness. The annular flow was occured in this tube. In the experiments, two different rods having 16 and 17.5 mm diameter and 250 mm length are placed in the tube. Thus, the gaps occured in annular duct are 1 and 0.25 mm.

The water having temperature about 280-300 K pumped from the water chiller group passes through rotameters into tubes of test tank. Water flow rates can be adjusted with sensitive valves on rotameters. Water is heated to 349 K before entering the evaporator tube. Thus, the saturated liquid n-pentan in test tank begins to boil on the outer surface of evaporator tube at temperature of about 309 K. In the meantime; the saturated n-pentan vapored in test tank condences on the outer surface of condencer tube.

In the experiments, evaporator tube was used as the test tube. After water flow rate and temperature are fixed to the desired values, the system remains in thermal balance adjusting water flow rate passed through condencer tube and runs up to the steady state regime. At the moment, the mesaured values are recorded by data logger. Water output from two tubes returns to storage tank in chiller group. Then water cooled in the evaporator of the chiller group is recirculated by the pump through all the system.

In the experiments, water flow rates passed through the two tubes, water temperatures at the inlet and outlet of these tubes, the liquid and vapour n-pentan temperatures, the pressure difference of evaporator tube are measured. Reynolds number (Re), Darcy friction factor (f) and Nusselt number (Nu) are calculated using the measured values. Water mass flow rate regulated by rotameter (VEBMLW Prüfgerate-Werk) varies from

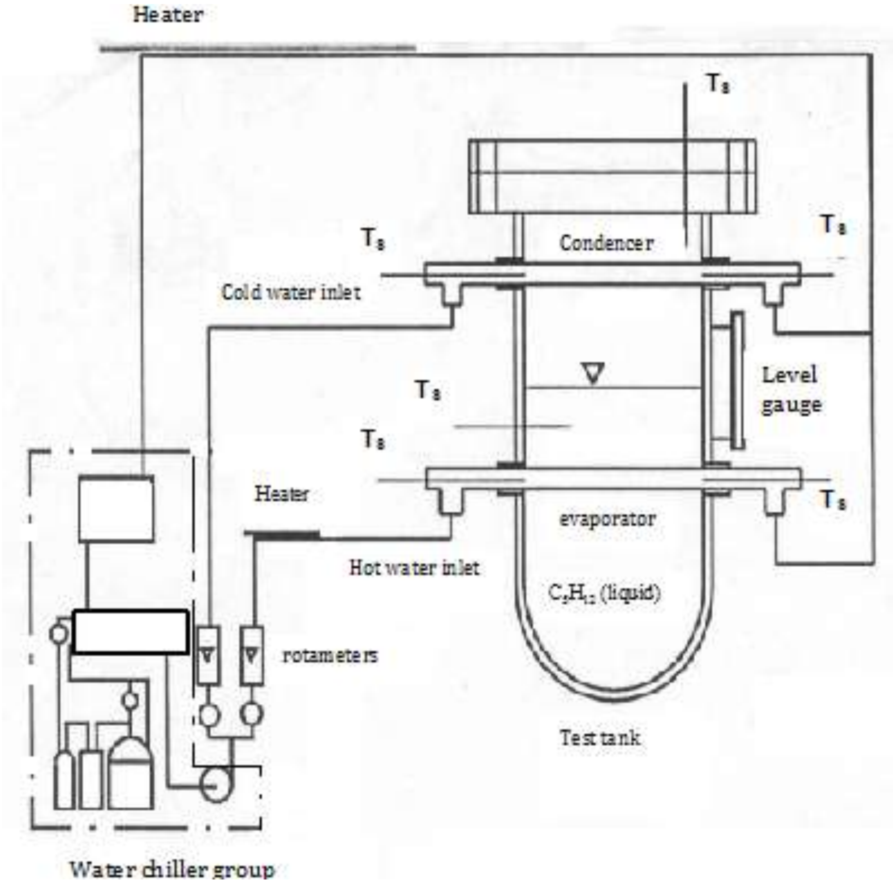



**Fig.3.** Shematic view of the experimental set up.

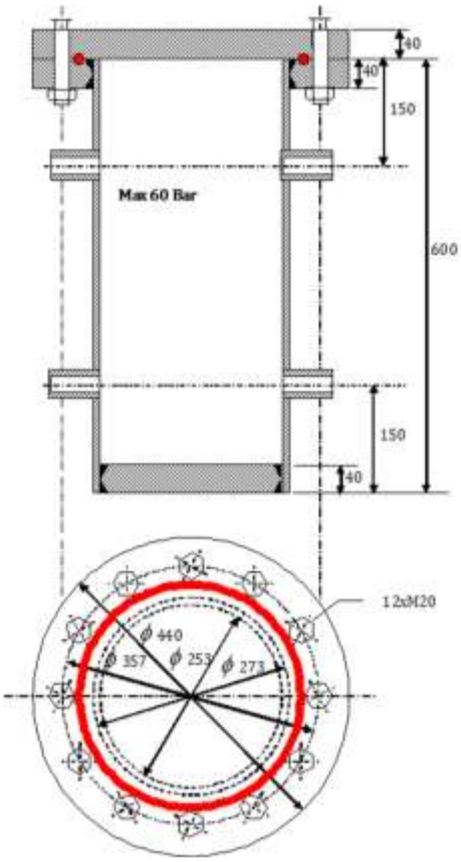



**Fig. 4.** Stainless stell (AISI 321) test tank.

0.0064 kg/s to 0.128 kg/s which correspond to Reynolds numbers from 600 to 12000. The inlet and outlet water pressures and temperatures of tubes are measured by pressure transmitters with accuracy ± % 0.25 (Omega PX1006L1-500DV) and NiCr-NiAl thermocouples with accuracy about 40 μV/ºC (Omega HH-K-30 PFA), respectively. All probes are connected to a data logger with true 6½-digit (22-bit) resolution (Keithley 2700 DMM). General view photos of the experimental set up are also shown in Fig.s 5 and 6.

**Fig.5.** General view of the experimental set up.

(a)

(b)

(c)

(d)

**Fig. 6.** Photos of the experimental set up: (a) Test tank, (b) General view(c) water chiller, (d) pump assembly.

**DATA REDUCTION**

For the annular duct, the hydraulic diameter is shown as

$$d_h = d_i - d_r \tag{11}$$

The Reynolds number is given by

$$Re = \bar{V} d_h / v \tag{12}$$

Friction factor (f) and Nusselt number (Nu) are two major quantities investigated for this study. The friction factor (*f*) is calculated by presure drop (ΔP) occurred across the length of the test tube and given as

$$f = \frac{\Delta P}{(L/d_h)(\rho \bar{V}^2/2)} \tag{13}$$

where $\bar{V}$ is mean velocity of water in test tube. The all thermophysical properties of water are taken at the bulk temperature given as

$$T_b = (T_{in} + T_{out})/2 \tag{14}$$

where $T_{in}$ and $T_{out}$ are temperatures of water at inlet and outlet of test tube. In experiments, heat transfer coefficients of inner and outer side of test tube, $h_i$ and $h_o$, were obtained by "Wilson Plot" method (1915). This method makes possible to find $h_i$ and $h_o$ without the need for measuring of the wall temperatures of the tube. Heat transfer rate, Q, is calculated from

$$Q = mC_p(T_{in} - T_{out}) \tag{15}$$

The overall heat transfer coefficient, U, is given as

$$U = \frac{Q}{A \Delta T_{LMTD}} \tag{16}$$

where A and $\Delta T_{LMTD}$ are surface area of test tube and logarithmic mean temperature difference, respectively. Log mean temperature difference is given as

$$\Delta T_{LMTD} = \frac{\Delta T_{out} - \Delta T_{in}}{\ln\left(\frac{\Delta T_{out}}{\Delta T_{in}}\right)} \quad (17)$$

where $\Delta T_{in}$ and $\Delta T_{out}$ are the temperature differences between the water and medium at inlet and outlet of test tube, respectively. Firstly; heat transfer rate (Q) and log mean temperature difference ($\Delta T_{LMTD}$) are calculated by Eq.s (15) and (17) measuring these temperature for different mass flow rate studied. Secondly; the overall heat transfer coefficient (U) is calculated from Eq. (16) using the values obtained from Eq. (15) and Eq. (17). Then a curve based on the variation of U depend on fluid velocities studied is plotted. Finally; $h_i$ and $h_o$ can be found from the slope of this curve and the point of intersection with axis of the ordinate, respectively (Rose, 2004). After finding the heat transfer coefficient for inside of test tube, the average Nusselt number is given as

$$Nu = h_i d_h / k \quad (18)$$

## RESULTS AND DISCUSSION

In this study, convective heat transfer and single phase water flow in concentric annular tube having narrow gap with 250 μm, 500 μm and 1 mm are numerically and experimentally investigated for 600<Re<12000. Numerical and experimental data are compared with the values calculated by the conventional Nusselt and friction factor correlations used for macro annular ducts. Friction factor for fully developed laminar flow is computed as (Heat Atlas, 1993)

$$f = \varphi\, 64/Re \quad (19)$$

where $\varphi$ is a constant depending on geometry of the flow cross section ($d_i/d_r$). For turbulent flow, Blasius type formula increased about 10 % is recommended as (Rohsenow et al., 1998)

$$f = 0.3482\,\mathrm{Re}^{-0.25} \quad (20)$$

For laminar flow, Nusselt number is given as (Heat Atlas, 1993)

$$Nu = Nu_m \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_i}\right)^{0.11} \quad (21)$$

where Nu is the average Nusselt number improved according to the change of thermophysical properties of fluid. *Pr* and $Pr_i$ are Prandtl numbers of fluid at the bulk temperature and the inner surface temperature of the tube, respectively. $Nu_m$ is given as (Martin,1990)

$$Nu_m = (Nu_1^3 + Nu_2^3 + Nu_3^3)^{1/3} \quad (22)$$

where

$$Nu_1 = 3.66 + 1.2(d_i / d_o)^{0.5} \quad (23)$$

$$Nu_2 = f_g \sqrt[3]{\mathrm{Re}\,\mathrm{Pr}\, d_h / L} \quad (24)$$

$$f_g = 1.615\{1 + 0.14(d_i / d_o)^{1/3}\} \quad (25)$$

$$Nu_3 = \left[\frac{2}{1 + 22\,\mathrm{Pr}}\right]^{1/6} (\mathrm{Re}\,\mathrm{Pr}\, d_h / L)^{1/2} \quad (26)$$

For turbulent flow, Nusselt number given by Petukhov and Roizen (1968) is

$$Nu/Nu_{pipe} = 1 - 0.14(d_i/d_o)^{0.6} \quad (27)$$

where $Nu_{pipe}$ is the Nusselt number improved according to the change of thermophysical properties of fluid as with laminar flow and also calculated by Eq. (21), where $Nu_m$ for turbulent flow is given as (Heat Atlas, 1993)

$$Nu_m = 0.012(\mathrm{Re}^{0.87} - 280)\,\mathrm{Pr}^{0.4}[1 + (d_i / L)^{2/3}] \quad (28)$$

In this study, the Nusselt number and friction factor are presented depending on Reynolds number in the range of 600-12000 and the hydraulic diameter of the annular duct ($d_h$ = 0.5, 1 and 2 mm). The numerical and experimental results are compared with obtained from those conventional correlations of macro annular ducts given in Eq.s (19)-(28).

Fig.s 7- 9 show the results of friction factor for all the studied diameter. It is clear from these figures that there is a good agreement between the numerical data and the correlation results in laminar region. In turbulent regime, although similarity characteristics of the curves, numerical results are greater about 23-43% than the conventional results. However; the data obtained from conventional correlations are less about 31-41% than those of experimental data. Similar results was also obtained in previous studies presented by Hrnjak and Tu (2007) and Bucci et al. (2003). A review study related to the results of experimental studies for micro channels having different cross sections, Reynolds numbers and hydraulic diameters as shown in Fig. 10 was performed by Steinke and Kandlikar (2006). In their study, Blasius equation is used as conventional friction factor correlation. Here, a nondimensional characteristics, Poiseuille number was defined as

$$C^* = (fRe)_{exp}/(fRe)_{theory} \quad (29)$$

It was seen from Fig. 10 that experimental data for micro channels diverge from conventional values in the case of Re > 3000. In Fig.11, the results presented by Bucci et al.(2003) show clearly this case. It was also seen from this figure that $C^*$ values are close to unity in laminar region while it sharply increases in transitional region. These figures support the diverge shown in Fig.s 7 to 9.

In Fig. 12, it was given the ratio of experimental to numerical values of friction factor versus Reynolds number for $d_h$=2 and 0.5 mm. For laminar and turbulent regions, this ratio is between 0.97 -1.01 and 1.08 - 1.3, respectively. For laminar and turbulent regime, the ratio of $f_{exp}$ /$f_{num}$ increases with decreasing diameter and increasing Reynolds number. Besides, it was seen from this figure that increase of this ratio decelerates after Re>6000 and approach to a constant value for turbulent region.

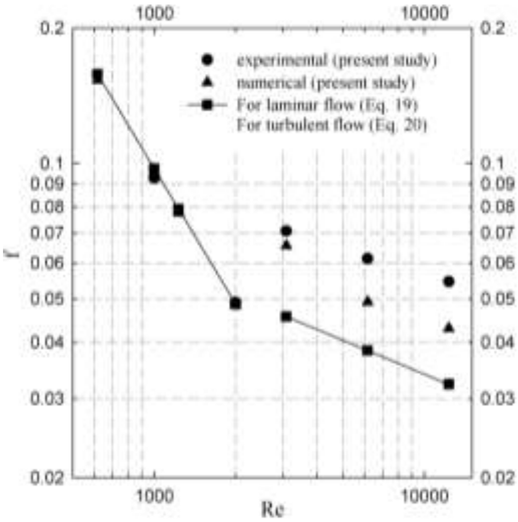



**Fig. 7**. Friction factor versus Reynolds number for $d_h$ = 2 mm

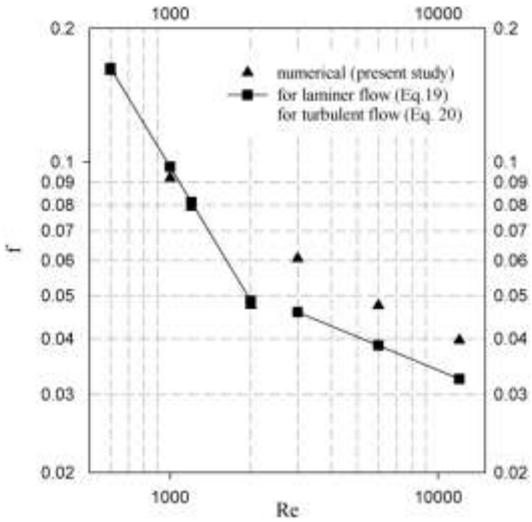



**Fig.8**. Friction factor versus Reynolds number for $d_h$ = 1 mm.

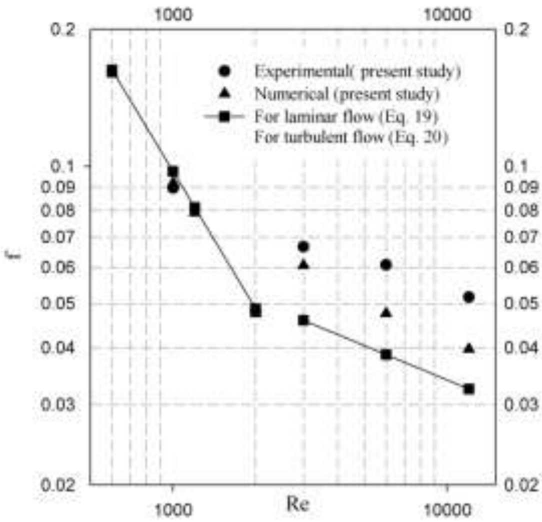



**Fig. 9.** Friction factor versus Reynolds number for $d_h$ = 0.5 mm.

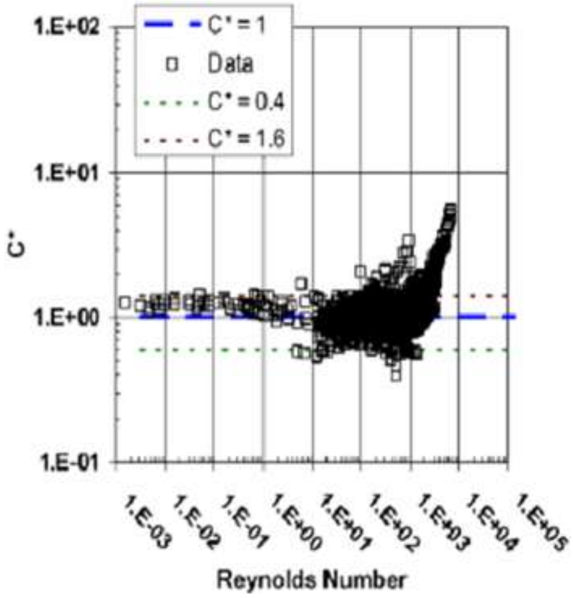



**Fig. 10**. Poiseuille number versus Reynolds number (Steinke and Kandlikar, 2006).

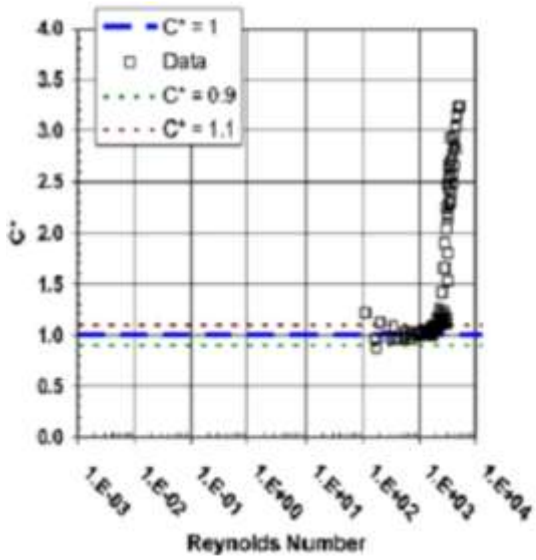



**Fig.11**. Poiseuille number versus Reynolds number (Bucci et al., 2003).

Fig.s 13- 15 show the results of Nusselt number for all the studied diameter. The experimental and numerical results obtained from this study are compared with conventional macro-sized channel results presented in literature. For laminar region, it is clear from these figures that the numerical data of Nusselt number is less than those of experimental data about 6-14%. However; the data obtained from conventional correlations are less about 10-23% than those of experimental data.

For turbulent regime, it was seen from these figures that experimental data deviate about 20% from the numerical results, especially for the cases of $d_h$ < 2 mm and Re>6000. Besides, the error between the experimental and conventional results is about 15-26%. In Fig.s 14 and 15, for the cases of $d_h$ =1 and 0.5 mm, the experimental results of Adams et al. (1998) presented in Eq.(30) for micro channels are also plotted.

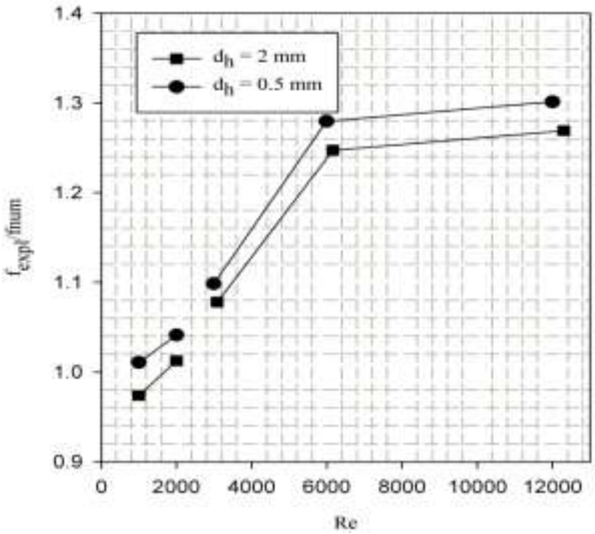



**Fig.12.** The ratio of experimental to numerical results for friction factor versus Reynolds number.

According to these figures, it can be concluded that the tendency of heat transfer is close to results of Adams et al. (1998). They experimentally investigated the forced convectional heat transfer in single-phase turbulent flow through circular microchannels with 0.76 and 1.09 mm diameter. They said that the Nusselt number rises with increasing Reynolds number and decreasing channel diameter and found that the Nusselt number values are higher than the values predicted by Gnielinski correlation (1976) and proposed a modified Gnielinski equation for micro channels as follows

$$Nu = Nu_{GN}(1 + cRe(1 - (d/d_0)^2) \qquad (30)$$

where $Nu_{GN}$ is given by Gnielinski (1976)

$$Nu_{GN} = \frac{(f/8)(Re-1000)Pr}{1+12.7(f/8)^{1/2}(Pr^{2/3}-1)} \qquad (31)$$

Although Adams et al. (1998) concludes the opposite, there also are many recent literatures stating that the microchannels act as macro channels (Obwaib and Palm, 2004; Lelea et al., 2004). The results of this study support the estimation of Adams et al. (1997).

The ratio of experimental to numerical values of Nusselt versus Reynolds number is shown in Fig.16. This ratio for laminar and turbulent regions are varied between 1.05-1.17 and 0.97 - 1.24, respectively. It is seen from this figure that this ratio is almost constant for $d_h$ = 2 mm and Re >10000 as it increases rapidly for $d_h$=0.5 mm and Re>10000. This character shows that, especially for $d_h$<1 mm, the results of numerical calculation (obtained by Fluent in this study) based on governing equation used for macro-channels is less than the experimental results with increasing discrepancy versus decreasing diameter.

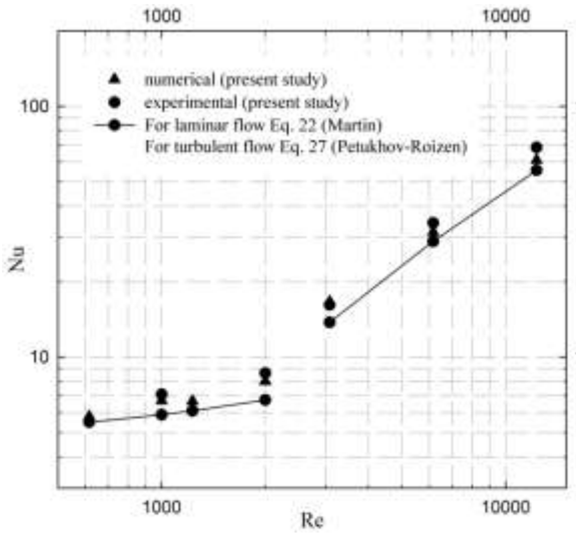



**Fig. 13.** Nusselt number versus Reynolds number for $d_h$=2 mm.

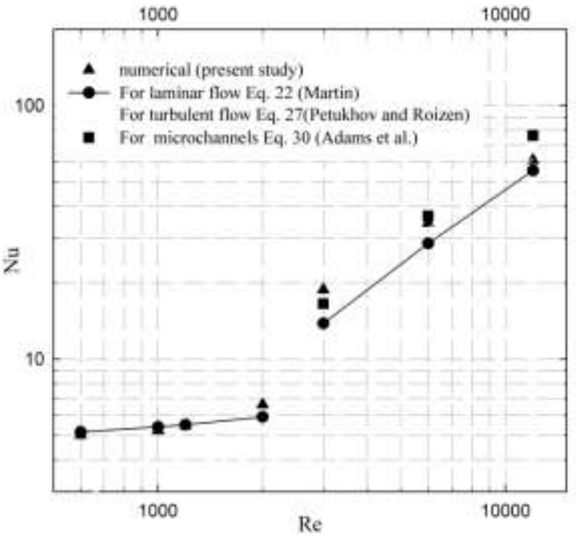



**Fig. 14.** Nusselt number versus Reynolds number for $d_h$=1 mm.

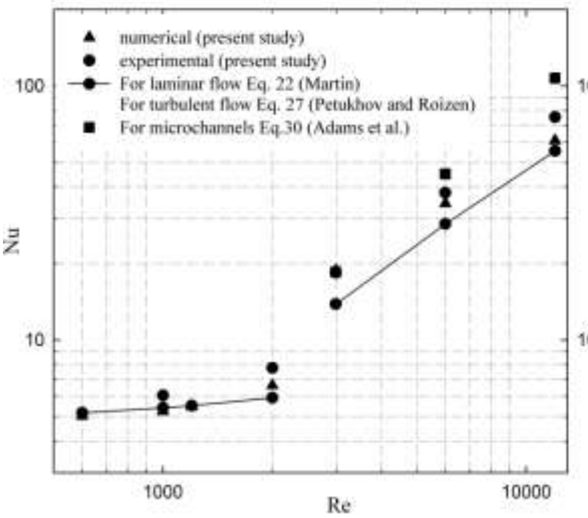



**Fig. 15.** Nusselt number versus Reynolds number for $d_h$=0.5 mm.

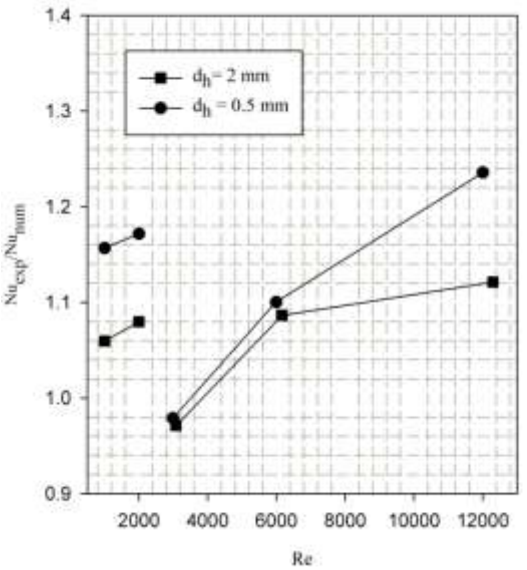



**Fig. 16.** The ratio of experimental to numerical results for Nusselt number versus Reynolds number.

Fig.s 17- 19 show the local Nusselt number along z- axis for all the studied diameters. For laminar region, for 600<Re<2000, it is clear from these figures that the numerical Nusselt data is close to each other. For turbulent region, for Re>2000, the increase ratio of Nusselt number is much more than that of laminar region.

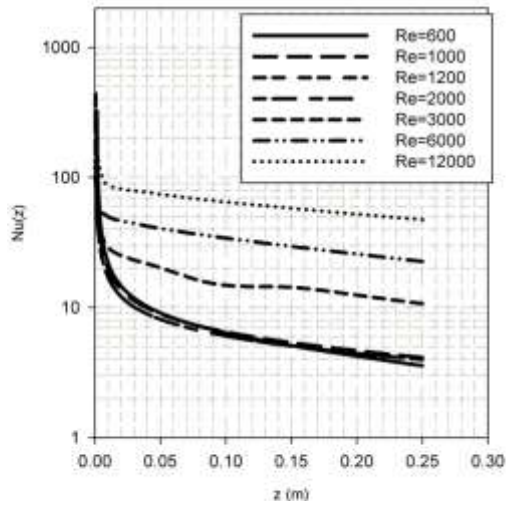



**Fig. 17.** Local Nusselt number along z-axial direction for $d_h$=2 mm.

The static temperature and velocity magnitude fields at the outlet for $d_h$= 0.5, 1 and 2 mm (Re=12000) are given in Fig. 20. As is seen from this figure, decreasing of the gap size from 1 mm to 250 μm causes to decrease of temperature values about 5 K and increase of velocity values about two times at the outlet of the annular duct.

Some of the earlier investigators reported that the entrance and exit losses, and the developing region effects cause to large discrepancy from the theoretical values (Kandlikar et al., 2006). As is seen from Fig.s 17-19, the flow is not yet fully thermally developed. The increasing of the developing region length cause to diverge from conventional values.

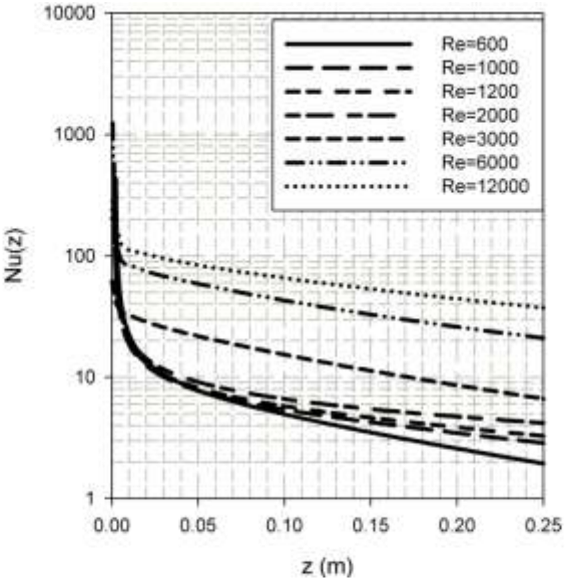



**Fig. 18.** Local Nusselt number along z-axial direction for $d_h$=1 mm.

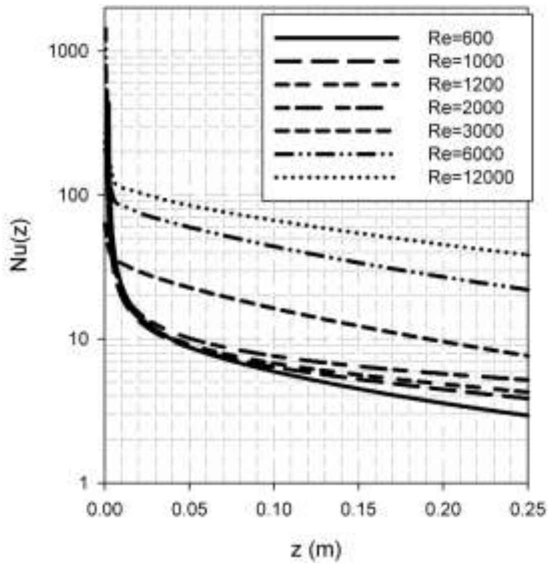



**Fig. 19.** Local Nusselt number along z-axial direction for $d_h$=0.5 mm.

On the other hand; Steinke and Kandlikar (2005, 2006) point out that simultaneously developing flows are the most complex and much more accurate data is needed in this regime. One of the reasons caused discrepancy may be this case too. Also in this study, the flow and temperature field develope as simultaneously. Another reasons of discrepancy may be due to the entry effects or experimental equipment deteriorations (Reynaud et al., 2005).

The experimental/numerical heat transfer and friction factor results of turbulent region obtained from this study are not good agreement with conventional macro-sized annular duct correlation results. The reasons of this discrepancy may be due to experimental uncertainties and control complications of uniform temperature boundary condition at the tube wall accepted in this study. As a matter of fact, Morini (2004) pointed out based on the results presented by several researchers related to the mini/micro duct flow that in many cases the experimental data of friction factor and Nusselt number disagree with both the conventional theory and also each other.

Nevertheless; the numerical analysis programmes use the governing equations based on conventional Navier-Stokes equations. Thus, the applicability of this programmes to mini/micro channels can be questionable. The difference between experimental and numerical results may be occured due to this reason. Besides, the correlations in literature from Eq. 19 to Eq. 28 are already used for macro-sized annular ducts. Thus, it can be expected a discrepancy between literature and experimental results. The diverge between the values of experimental and literature for friction factor results were supported by Steinke and Kandlikar (2006) and Bucci et al.(2003) (Fig.s 10-11). For turbulent flows, in many experiments (Whu and Little, 1984; Choi et al., 1991; Yu et al., 1995; Adams et al., 1998), it is found that the heat transfer rate is higher than conventional results.

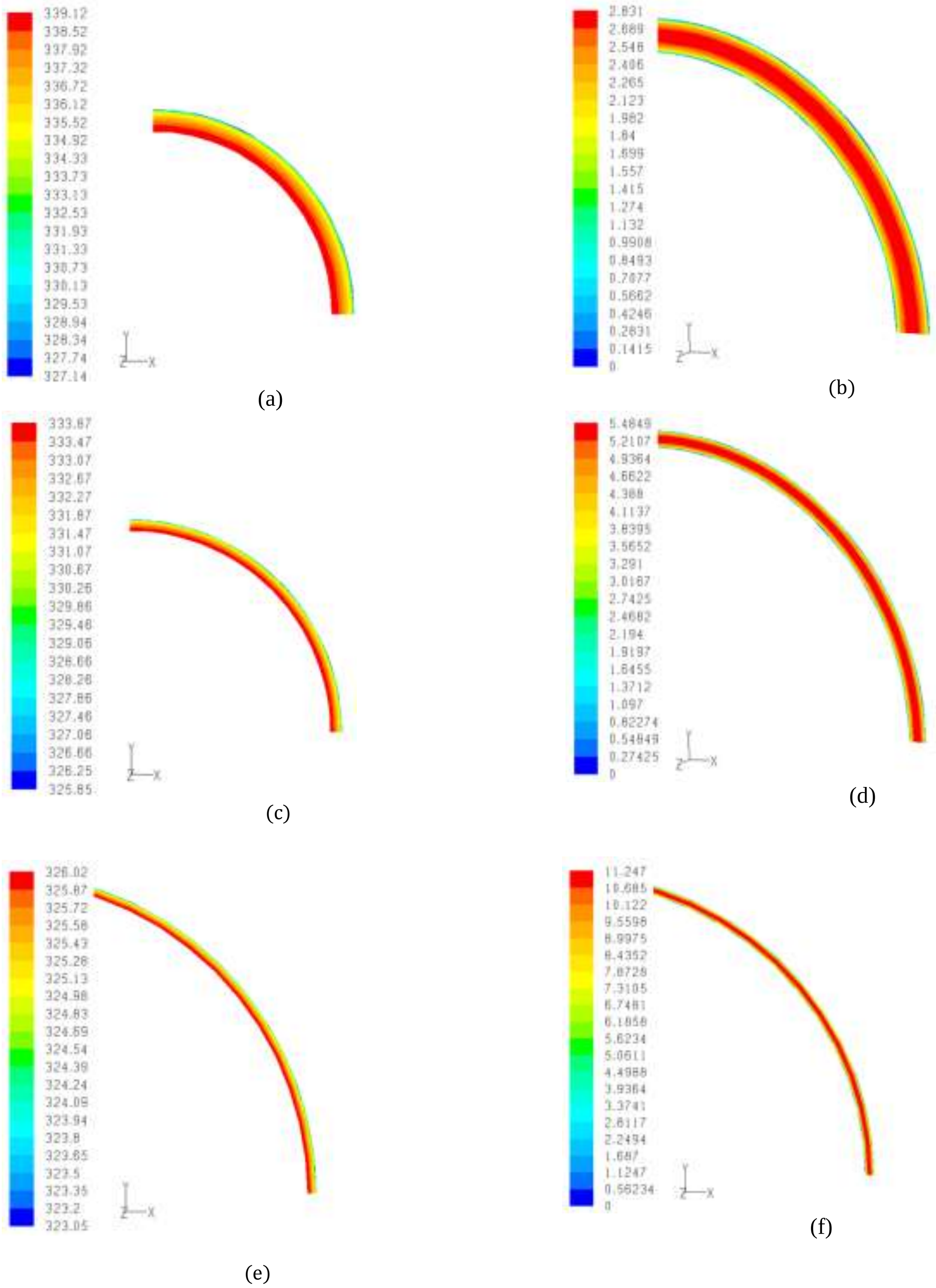



**Fig. 20.** Contours of static temperature (K) and velocity magnitude (m/s) at the outlet for $d_h$=2 mm (a,b), $d_h$=1 mm (c,d) and $d_h$=0.5 mm (e,f), respectively (for Re=12000).

## CONCLUDING REMARKS

In this study, heat transfer and single-phase water flow are experimentally and numerically investigated in the horizontal narrow annular ducts having mini/micro gaps from 250 µm to 1 mm. The study was conducted for the Reynolds number which ranges about from 600 to 12000. The effects of the Reynolds number and gap size on the convective heat transfer and flow have been examined in detail. The key findings of this study are as follows:

1. For laminar and turbulent region, the ratio of experimental to numerical values of friction factor is about 0.97 -1.01 and 1.08 - 1.3, respectively.
2. For laminar and turbulent region, the ratio of experimental to numerical values of Nusselt number is varied between 1.05-1.17 and 0.97 - 1.24, respectively.

3. For turbulent region, the numerical and experimental Nusselt number are found bigger about 9-36% and 18-36% than conventional results presented in literature for macro- sized annular duct, respectively.

4.For turbulent region, the numerical and experimental friction factor results are greater about 23-43% and 45-69% than the conventional results.

5. For turbulent region, the diverge between the experimental and conventional values of friction factor is bigger than that of Nusselt number.

6. The heat transfer and friction results of laminar and turbulent regions obtained from this study are in good agreement with previous results presented by Steinke and Kandlikar (2006), Bucci et al(2003) and Adams et al.(1998).

7. In future, the completion of the experiments in cases of smaller hydraulic diameter and larger Reynols numbers remained in outside the scope of this study corroborates the results and conclusions reached in this study.

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