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FORCED CONVECTION HEAT TRANSFER IN CIRCULAR MESO-Tubes

إنتقال الحرارة بالحمل الجبرى فى مجرى أنبوبى دقيق

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خلاصة البحث:

فى هذا البحث تم عمل دراسة معملية لإنتقال الحرارة بالحمل الجبرى لماء يمر داخل مجرى أنبوبى دقيق مسخن. لإتمام هذا الغرض تم تصميم وتنفيذ دائرة إختبار معملية لدراسة إنتقال الحرارة لسريان منتظم وآخر نبضى للماء المار داخل الأنبوب الدقيق. مقطع الإختبار وهو عبارة عن مبادل حرارى من نوع الغلاف والأنبوب مكون من سبعة أنابيب دقيقة محاطة ببخار مشبع من الخارج حتى تحافظ على درجة حرارة السطح الخارجى ثابتة. وقد تم إختبار أربعة أقطار للأنبوب الدقيق وهى ٠,٢٥ ، ٠,٣٥ ، ٠,٥٢، ٠,٧ مم. وقد أجريت التجارب عند قيم مختلفة لرقم رينولدز تتراوح مابين ٢٠٠حتى ٥٠٠٠ لسريان ماء منتظم وآخر نبضى بترددات صغيرة تتراوح مابين ٠,٥ هيرتز حتى ٥ هيرتز. وقد تمت دراسة تأثير تغيير ظروف التشغيل هذه علي كمية الحرارة المنتقلة خلال الأنبوب الدقيق لسريان الماء النبضى المار بداخلها مقارنة بالسريان المنتظم. أيضا تم حساب معامل إنتقال الحرارة المتوسط ومعامل الإحتكاك لسريان الماء المار داخل المجرى الأنبوبى الدقيق.

وقد أظهرت النتائج أن معامل الإحتكاك لسريان الماء فى المجرى الأنبوبى الدقيق يقل مع نقصان قطر الأنبوب ويقل أيضا مع زيادة رقم رينولدز وذلك مثل الأنابيب العادية. كما أظهرت النتائج أن رقم نوسلت يزيد مع زيادة رقم رينولدز لكل من السريان النبضى والمنتظم. كذلك لوحظ نقصان قيمة رقم نوسلت للسريان النبضى مقارنة بالسريان المنتظم وذلك فى مدى ظروف التشغيل للترددات الصغيرة. أيضا تمت المقارنة بالأبحاث السابقة للسريان المنتظم حيث وجد أنه مازال هناك عدم توافق بين النتائج. وقد تم استنتاج صيغة رياضية لرقم نوسلت للسريان النبضى والمنتظم وكذلك تم إستنتاج صيغة رياضية لمعامل الإحتكاك وذلك من النتائج المعملية فى مدى ظروف التشغيل التى تمت دراستها.

Abstract

Convection heat transfer of water flow inside a circular meso-channel was investigated experimentally. To perform this experimental study, an experimental test rig was designed and constructed. Test section was shell and tube heat exchanger. It consists of seven meso-scale circular tubes surrounded by a saturated steam to perform its outer surface at constant temperature. Water flows inside these tubes in steady flow and pulsating flow. The ranges tested for pulsation frequencies are 0.5 Hz to 5 Hz and Reynolds number is of 200 to 5000. Four values of inner diameters was tested in this study ; 0.25, 0.35, 0.52 and 0.7 mm. Experimental measurements for mass flow rate, temperature, pressure, pressure drop and pulsation frequency are taken to perform the required analysis. Therefore, the average value of convection heat transfer coefficient and friction factor for different operating parameters are computed.

The experimental results show that, the friction factor in the studied operating range decreases as Reynolds number increases but its values are deviated from those reported in conventional tubes due to the small dimensions of the meso-tubes. The friction factor decreases with decreasing tube diameter. Also, Nusselt number increases with increasing

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Reynolds number for pulsating flow and steady flow. For pulsating flow, Nusselt number takes lower values than steady flow for the same conditions in the studied operating ranges.

Comparison between the obtained experimental results with the previous data and the classical correlations for conventional tubes was done and the disagreement among the researchers was still. Two empirical correlations for both friction factor and Nusselt number were obtained in the range of the studied operating parameters.

Key words: Convection Heat Transfer, Pulsating flow, Meso-tubes.

1. INTRODUCTION

Compact heat exchangers allow an efficient use of material, volume and energy in the thermal systems. These benefits have driven heat exchanger design toward higher compactness, which can be manufactured using micro-machining and other modern technologies. Ultra-compact heat exchangers also called micro-and meso-channels heat exchangers are distinguished by a very high ratio of surface area to volume, low thermal resistance, small volumes, lower total mass, and low inventory of working fluids. Significant attention has been given to convection heat transfer for liquids flow through micro-and meso-channels due to the rapid grow of technology applications, which required transferring high heat rates in a relatively small space and volume. Forced convection for single-phase of highly purified liquid in micro-and meso-channels is an effective cooling mechanism with a wide range of applications. Micro-and meso-channels heat exchangers to the cooling systems for computer, optical, medical devices and cooling of electronic components are such examples for forced convection flow of fluid through micro-and meso-channels.

Mehendale et al. (2000) reviewed the flow and heat transfer in small channels with hydraulic diameters ranging from 1 µm to 6 mm and adopt the following classifications:

Micro-heat exchanger : $D_h = 1 - 100\mu m$;
Meso-heat exchangers : $D_h = 100\mu m - 1 mm$;
Compact heat exchangers: $D_h = 1 - 6$ mm;
Conventional heat exchangers: $D_h > 6$ mm.

In micro and meso heat exchangers, there are a large number of publications in this field for fluid flow and heat transfer [1, 2, 3,.....etc] but it is still an open problem due to the deviations between the experimental results for micro- and meso-channels than normal size channels (macro-channels). Many investigators have experimentally proved that well established correlations which are used for normal size channels and not apply in the case of micro- and meso-channels. These deviations increase as the channel size decreases.

Another general overview of the researches is carried out in single-phase heat transfer and flow in micro-pipes by Celata et al. (2004). Laminar flow and laminar-to-turbulent flow transition are analyzed to clarify the discrepancies among the results obtained by different researchers. In laminar flow regime the friction factor is in good agreement with Hagen-Poiseuille theory for Reynolds number below 600-800. For higher values of Reynolds number, the experimental data deviated from the Hagen-Poiseuille law to the side of higher values of friction factor. The transition from laminar to turbulent regieme occurs for Reynolds number in the range of 1800-2500. Experiments done by Celata et al. (2004) show that, heat transfer correlations in laminar and turbulent regimes developed for conventional tubes are not properly adequate for calculation of heat transfer coefficient in micro-tubes.

Wang and Peng (1994) investigated experimentally the single phase forced convection of water/methanol flowing through micro-channels with rectangular cross-section. The results provide that the fully developed turbulent convection regime was initiated at Re=1000-1500. The transition to turbulent mode and heat transfer characteristics of both transition and laminar flow were highly affected by liquid temperature, velocity and micro-channel size.

Peng and Peterson (1996) studied experimentally the forced convection heat transfer and flow characteristics of water flowing through meso-channel plates with extremely small rectangular channels having hydraulic diameters of 0.133-0.367 mm and different geometric configurations. Their results indicated that the geometric configuration had a significant effect on the convection heat transfer for water and flow characteristics.

Adams et al (1998) performed an experimental investigation of convection heat transfer characteristics of water flowing in circular meso-tubes ranging from 0.76 mm to 1.09 mm. The obtained data show that the values of Nusselt number for the meso-channels are higher than those predicted by traditional large channel correlations.

Owhaib and Palm (2004), investigated experimentally the heat transfer coefficient of single-phase forced convection of R134a, liquid, through single circular meso-channels with 1.7, 1.2 and 0.8 mm inner diameters. Their results were compared to the suggested correlations for meso-channels from the literature and found that the classical correlations were in good agreement with their experimental results in the turbulent region. On the other hand, none of the micro-channels correlations, which published later, were in agreement with the experimentally obtained data.

In general, it is clear from the previous published data that, heat transfer in micro- and meso-channels has a better performance than the conventional flow in macro-channels. Also, the classical thermal and fluid dynamic theories developed for macro-channels are not applicable to fluids in micro-and meso-channels. Therefore, further systematic studies are required to generate a sufficient physical knowledge of the mechanisms that are responsible for the variation of the flow structure and heat transfer in micro-and meso-channels. In this work, an experimental study was done to investigate the effect of different operating parameters on the rate of heat transfer and pressure drop for water flows through circular meso-channels for pulsating flow and compared with steady flow. Also, a comparison with the classical theory and the available correlations for the micro-and meso-channels was done.

2. EXPERIMENTAL TEST RIG

Experimental test rig is designed and constructed to study the convection heat transfer rate and pressure drop for water flow inside a circular meso-channel. Figure (1) shows the schematic diagram for the experimental test rig, which performed to achieve this aim. The experimental test rig consists mainly of a vertical test section, cooling water circuit and heating steam circuit. The main details of the test section are illustrated in Fig. (2.a). The test section was shell and tube ultra compact heat exchanger. The shell was made of clear acrylic, and the tube is made from stainless "used in anesthesia". To reduce the heat losses from the test section, with 50 mm in diameter, 5 mm thickness and 70 mm long. To minimize heat loss a 30-mm thickness of glass wool used to insulate the outer surfaces of the test section. The shell contained seven vertical meso-tubes. All measurements were taken at the center meso-tube, which called tested tube. The outer surface of the meso-tubes was maintained at constant wall temperature by condensing of dry saturated steam on the outside. Four diameters for meso-tubes are tested. These inner diameters are 0.25, 0.35, 0.52 and 0.7 mm. The outer diameters for meso-tubes are 0.51, 0.61, 0.78 and 1 mm, respectively.

Cooling water circuit consists mainly of constant head tank, pulsating generator and its control unit. Cooling water flows inside a vertical meso-tubes as a pulsating flow or a steady flow. The pulsation generator comprises of a solenoid valve, which is equipped by an electronic control unit to vary and control the frequency of pulsating.

The heating steam circuit was provided the test section by the required heating steam. The capacity of the heat exchanger is much lower than 2 kW. Therefore, an electric boiler 2 kW rated power with the basic dimensions of 0.5 m in diameter and 0.9 m height is used to generate the required heating steam. A water trap is installed before the test section directly to insure that the heating steam enters test section at dry saturation condition. The heating steam was flowing on the outer surface of meso-tubes (inside acrylic shell), and then it condensed and collected outside the test section to measure its flow rate.

3. EXPERIMENTAL MEASUREMENTS TECHNIQUE

To start any experiment, the experimental test rig was allowed to equilibrate for approximately one hour until steady state condition had been reached. Mass flow rate for the heating steam inlet to the shell could be controlled to obtain the required heat flux, which applied on the outer surface of the meso-tubes. Also, mass flow rate for cooling water, which flows inside these tubes, was controlled. In case of pulsating flow the pulsation frequency was adjusted to a certain value. Once the desired steady state was reached, the required measurements were taken. These measurements are temperature of cooling water at inlet and outlet of the tested tube, mass flow rate for cooling water and pressure drop through tested tube. The temperature of the outer surface of the tested tube was measured by thermocouple 0.5 mm diameter, as shown in Fig. (2.a). For pulsating flow average values for measurements were taken. Then inner surface temperature can be computed from simple Fourier equation of conduction heat transfer. Pulsation frequency is also measured. Temperature and pressure at the inlet for the heating steam are measured. In the same time condensate temperature and mass flow rate are also measured.

In order to obtain a measure of the reliability of the experimental data an uncertainty analysis was performed for the principle parameters of interest. The root-mean-square random error propagation analysis was carried out in the standard fashion using uncertainties of the basic independent variables (experimentally measured). These variables are included meso-tube dimension, pressure, temperatures and mass flow rates, which used to calculate the uncertainty in computed output results. Temperatures are measured by using copper-constantan thermocouple wires, which connected to a temperature recorder having minimum readable value of ± 0.1 °C. Inlet steam pressure was measured by Bourdon pressure gauge with minimum readable value of ± 0.05 bar. Pressure difference for flow of water inside the meso-tubes was measured in separate experiments by the test section shown in Fig. (2.b). Pressure difference ΔP was measured by using a differential manometer, which using mercury as a measuring fluid. By applying Bernoulli equation between inlet and outlet of the tested tube, we obtain:

$$\Delta P_f = \Delta P + g\,L - \left(\Delta P_{cont} - \Delta P_{enl}\right) \qquad (1)$$

where:

ΔP_f pressure drop due to friction.

ΔP_{cont} pressure drop due sudden contraction.

ΔP_{enl} pressure drop due sudden enlargement, Ref.[15] as:

$$\Delta P_{cont} = K_{cont}\left(\frac{u_i^2}{2g}\right)$$

$$K_{cont} = sudden\,contraction\,pressure\,drop\,coeff.$$

$$=0.5\left(1-\left(\frac{D_1}{D_2}\right)^2\right)$$

ΔP_{enl} pressure drop due sudden enlargement,

$$\Delta P_{enl} = K_{enl}\left(\frac{u_i^2-u_2^2}{2g}\right)$$

$$K_{enl} = sudden\,enlargement\,pressure\,drop\,coeff.$$

$$=\left(1-\left(\frac{D_1}{D_2}\right)^2\right)^2$$

D inside meso-tube diameter

L length of the tested tube.

The friction factor f is calculated from:

$$f = \Delta P_f\left(\frac{D}{L}\right)\left(\frac{2}{\rho u_i^2}\right) \qquad (2)$$

Water flow rate was measured by using a calibrated glass tank and stop watch. Also, the amount of condensate was small then it was measured by using a calibrated glass tank and stop watch. The largest calculated uncertainty in the current investigation, were less than 15% for Nusselt number, 8% for Reynolds number and 9.5% for pressure drop.

At steady state condition, the total heat input from the heating steam (Q_t) can be divided into useful heat to the water flow inside the meso-tubes (Q_{us}) and the remaining amount of heat can be transferred to the surroundings as heat loss (Q_{loss}). Then, useful heat can be determined as the difference between input heat and heat loss and calculated from measuring water flow rate and the temperature rise in water as;

$$Q_{us} = Q_t - Q_{loss} = \dot{m}_w Cp_w (T_{wo} - T_{wi}) \quad (3)$$

Where $\dot{m}_w$, Cp_w, $T_{w,i}$ and $T_{w,o}$ are the water flow rate, specific heat of water, inlet and outlet water temperatures, respectively. Water properties are calculated at average temperature, $T_{w,av} = \left(\frac{T_{wi} + T_{wo}}{2}\right)$.

The total input heat can be determined as;

$$Q_t = \dot{m}_{st} (i_g - i_o) \quad (4)$$

Where $\dot{m}_{st}$, i_g and i_o are steam flow rate, specific enthalpy for dry saturated steam at inlet and specific enthalpy for condensate at outlet from the test section respectively.

The heat flux (q") can be calculated from the following equation as;

$$q'' = \frac{Q_{us}}{A_s} \quad (5)$$

Where; A_s is the inner heat transfer surface area, $A_s = \pi DLN$, N is the number of meso-tubes in the shell side "N=7".

The convection heat transfer coefficient (h) can be calculated as;

$$h = \frac{q''}{T_{s,in} - T_{w,av}} \quad (6)$$

Where $T_{s,in}$ is the temperature of the inner surface of the tested tube.

The average values of the dimensionless numbers such as Nusselt number (Nu), Reynolds number (Re), Poiseuille number (Po), Prandtl number (Pr), Strouhal number (St) and Graetz number (Gz) are defined according to the following equations:

$$Nu = \frac{hD}{k}, \quad Re = \frac{\rho u D}{\mu}, P_o = f\,Re$$
$$Pr = \frac{Cp\mu}{k}, \quad St = \frac{FD}{u}, Gz = RePr\frac{D}{L} \quad (7)$$

Where ρ, u, μ, *f* and F are water density, water velocity inside the tested tube, dynamic viscosity of water, friction factor and pulsation frequency, respectively.

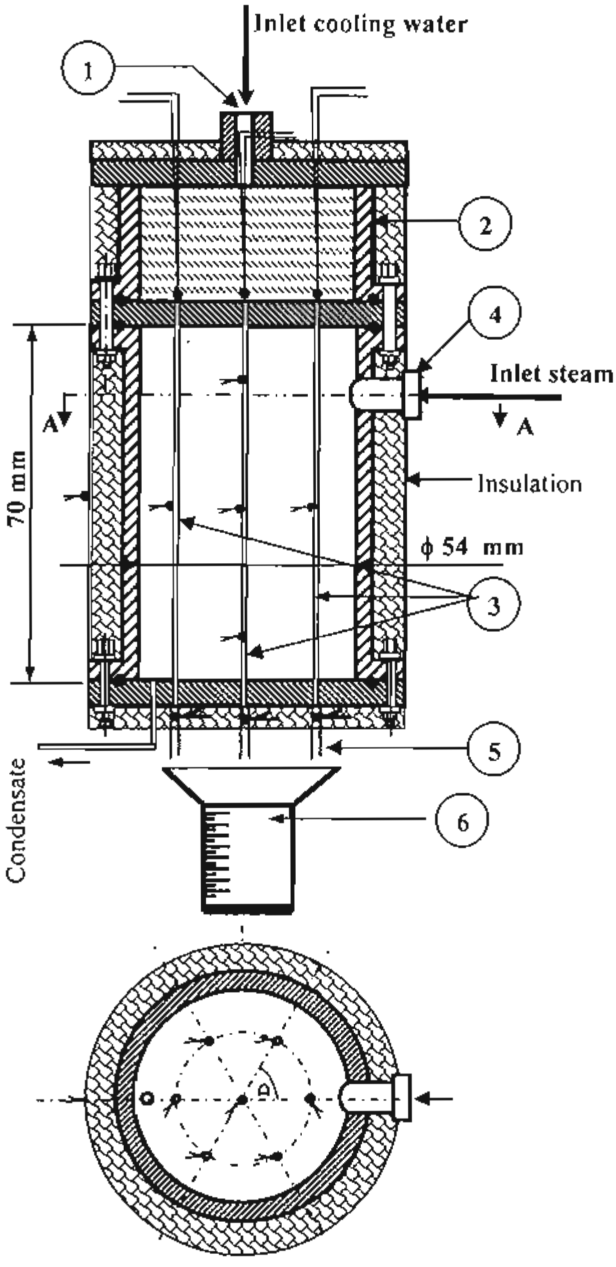
4. RESULTS AND DISCUSSIONS

Study of flow and heat transfer for water flowing inside circular meso-channel was performed. Condensation of dry saturated steam on the outer surface of the meso-tubes kept its surfaces at constant wall temperature. Pressure drop and heat transfer experiments were conducted to compute friction factor and convection heat transfer coefficient for the water side, which flow inside the tested tube.

4.1 Friction Losses

To compute the friction factor and in turn Poiseuille number for different values of Reynolds number, it is important to measure pressure drop for different flow velocities inside the tested tube. Figure (3) shows the variation of the measured pressure drop with respect to Reynolds number for different values of the tested tube diameters. It is observed that pressure drop increases with decreasing micro-tube diameter, for the studied operating range. Also, pressure drop increases with increasing Reynolds number.

Then, the friction factor can be computed from the measured pressure drop for the tested tube and plotted against Reynolds number as shown in Fig. (4). A comparison between the obtained experimental results and the conventional correlation for laminar flow inside smooth circular tube (Hagen Poiseuille law $f = 64/Re$) was done up to Re=2000. It is noticed that, friction factor for



Sectional Plan at A-A

1.Inlet water, 2. Header, 3. Meso tube,
4. Inlet steam, 5.Exit water guide,
6. Exit flow collector
Thermocouples $\theta = 60°$ angle

Fig. 2.a Test Section

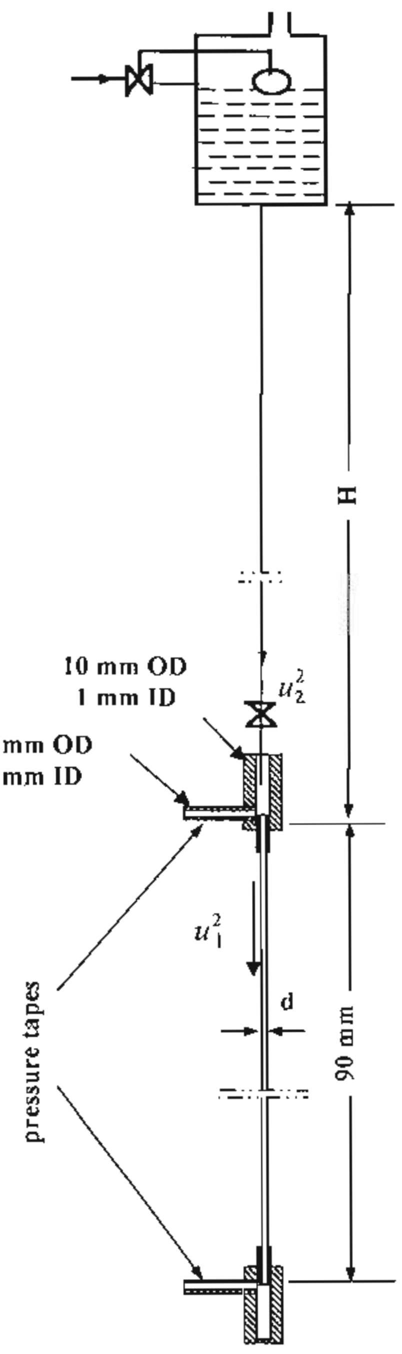


Fig.2-b Test section for friction factor.
d= 250, 350, 600, 700 μm
H = 18 m

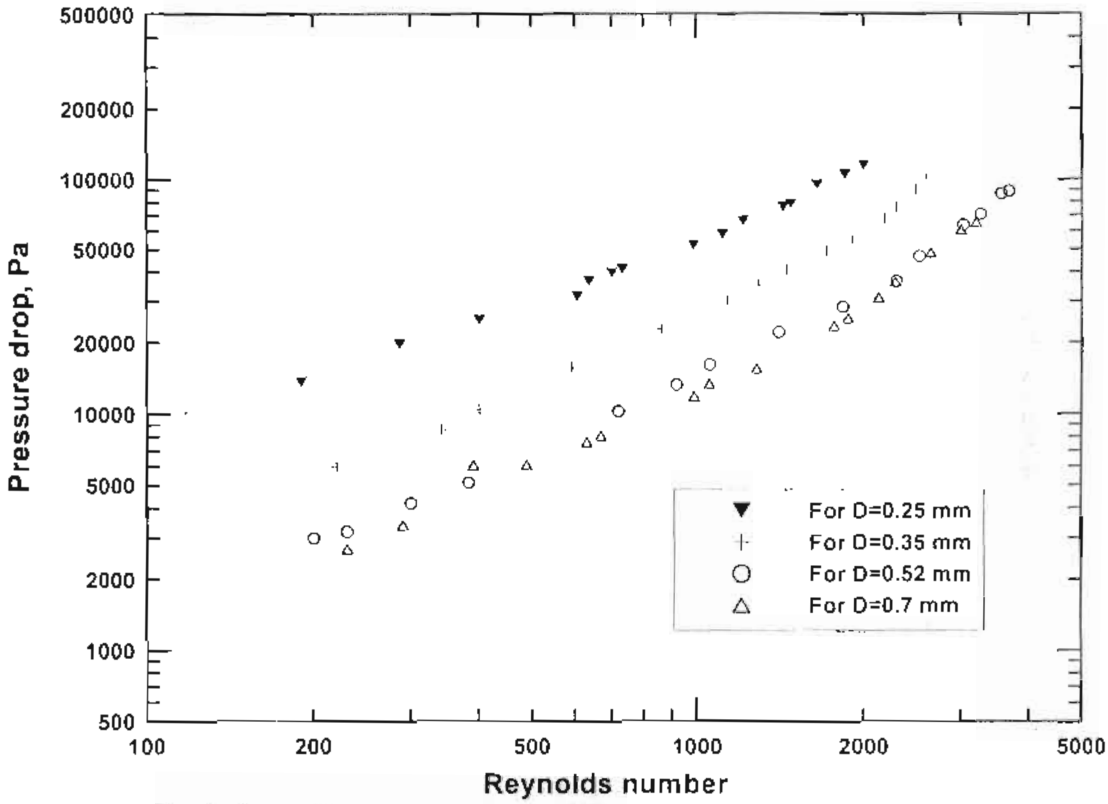


Fig.(3) Variation of the measured pressure drop versus Reynolds number for different values of the tested tube diameters.

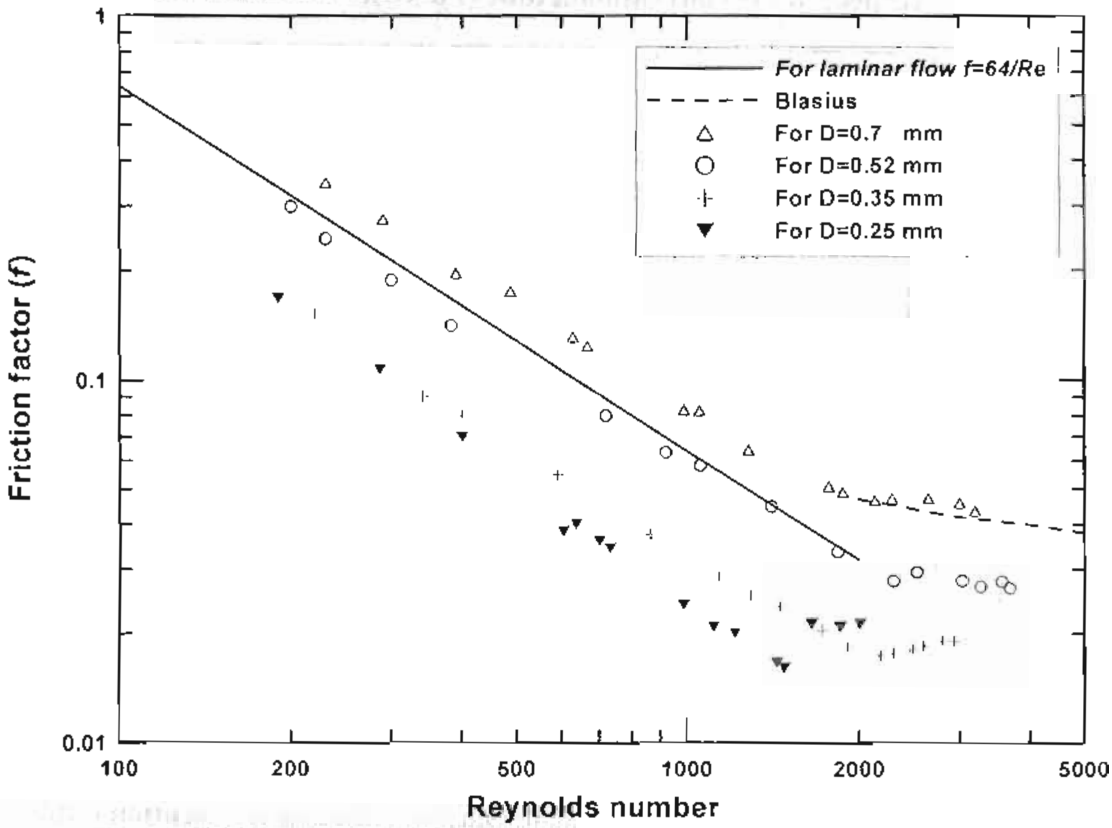


Fig.(4) Friction factor against Reynolds number for different values of the tested tube diameters.

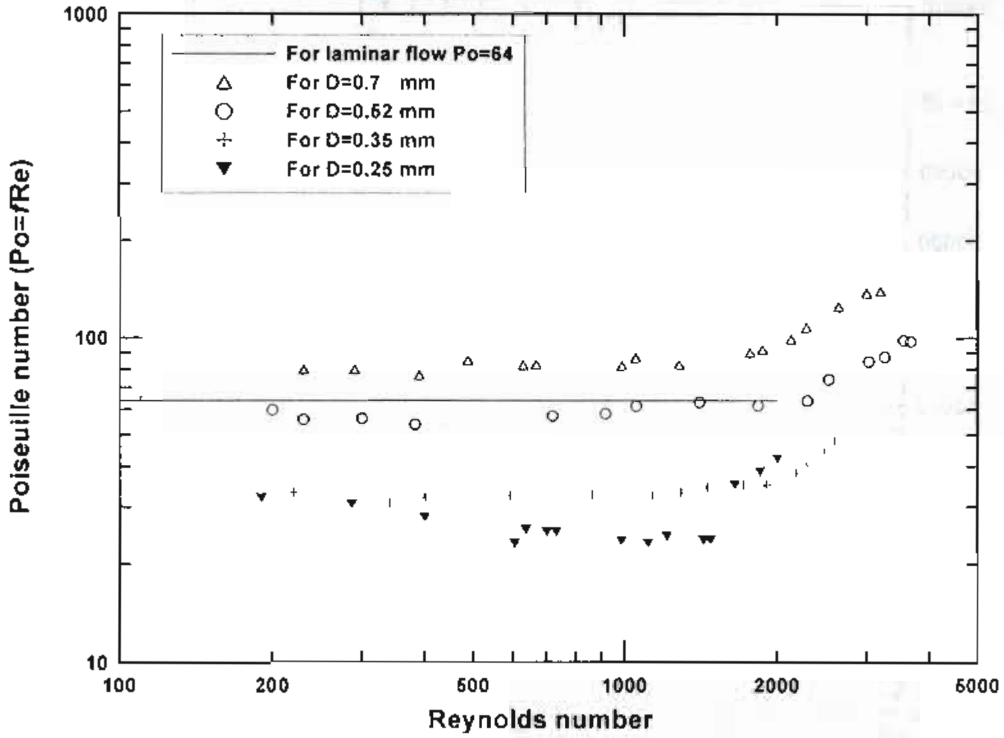


Fig.(5) Poiseuille number versus Reynolds number for the present work with respect to the conventional tube (Po=64).

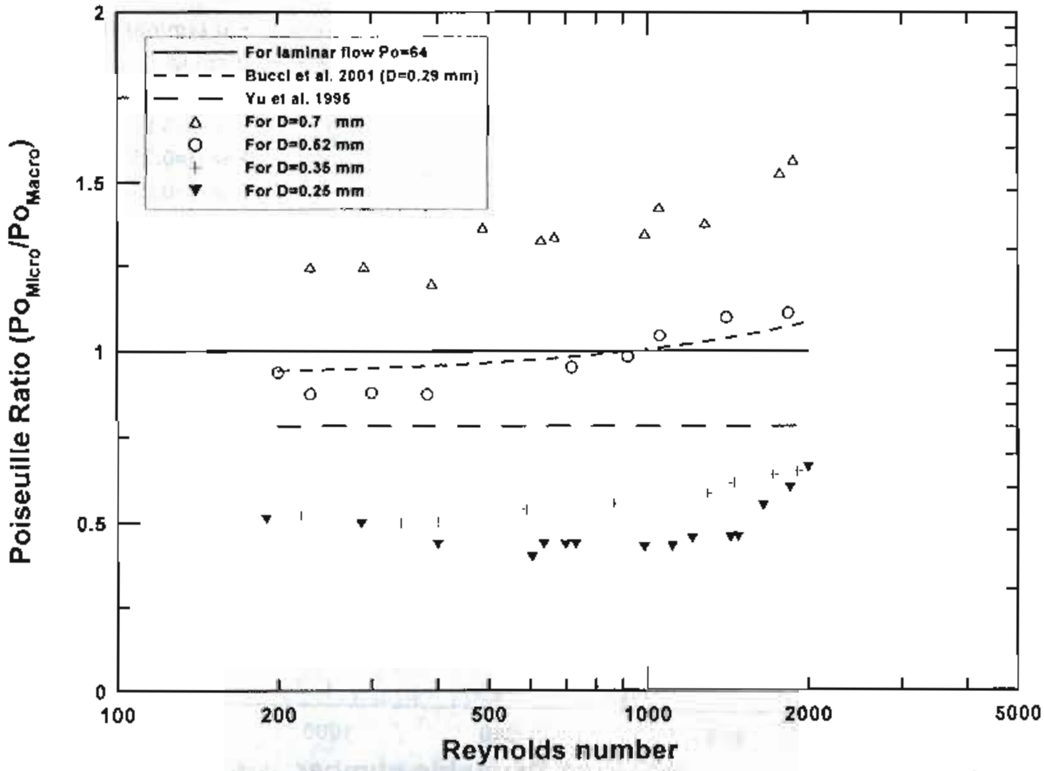


Fig.(6) Ratio of poiseuille number for meso-tube to the conventional tube versus Reynolds number for the present work and published data.

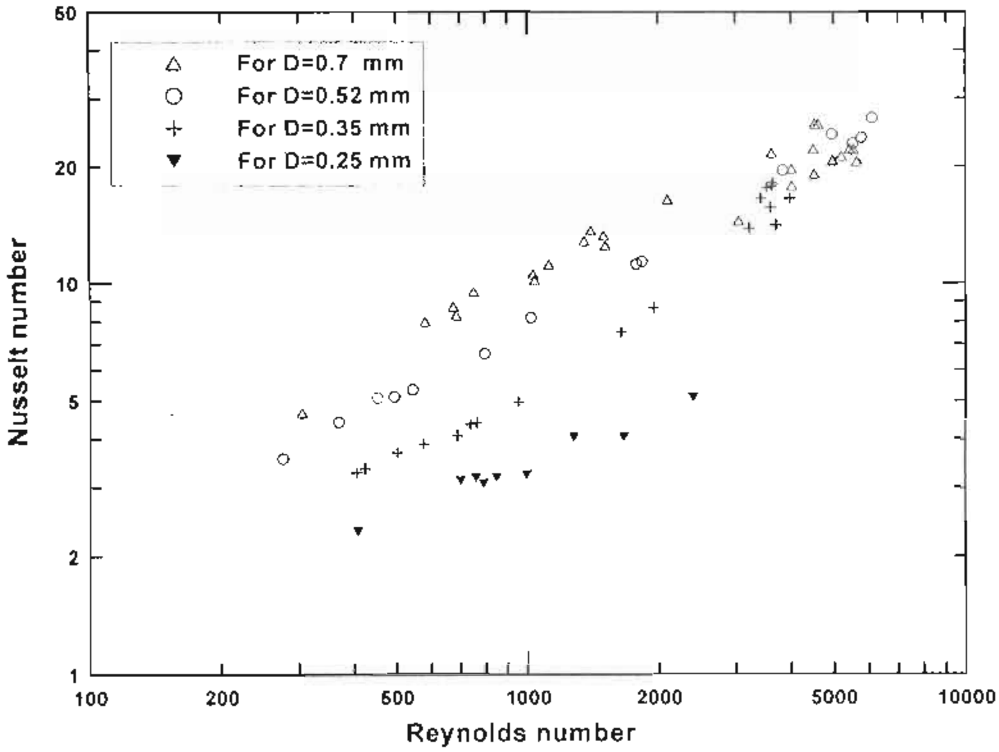


Fig.(7) Nusselt number versus Reynolds number for different values of tested tube diameters.

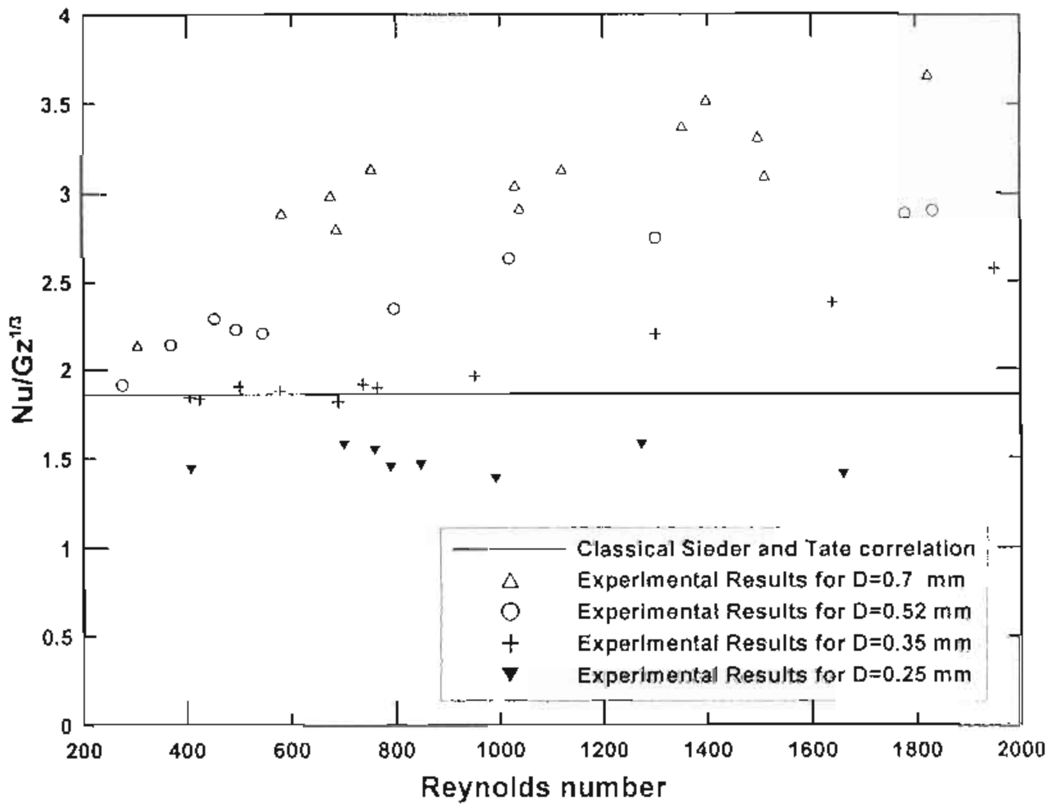


Fig.(8) Comparison between the present experimental results with the conventional tube.

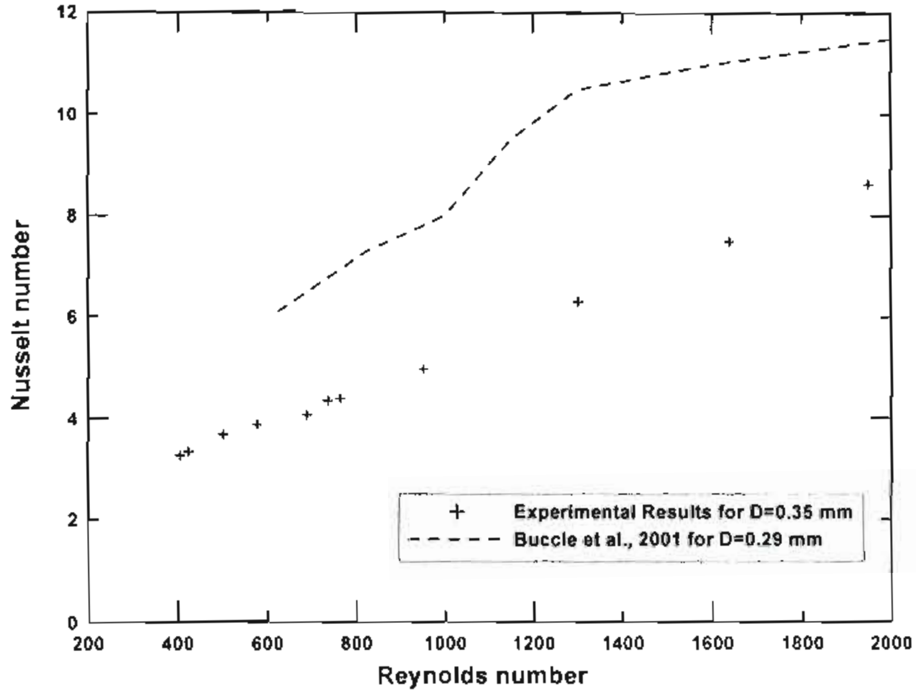


Fig.(9) Comparison between the present experimental results and the experimental results of Bucci et al.,(2001).

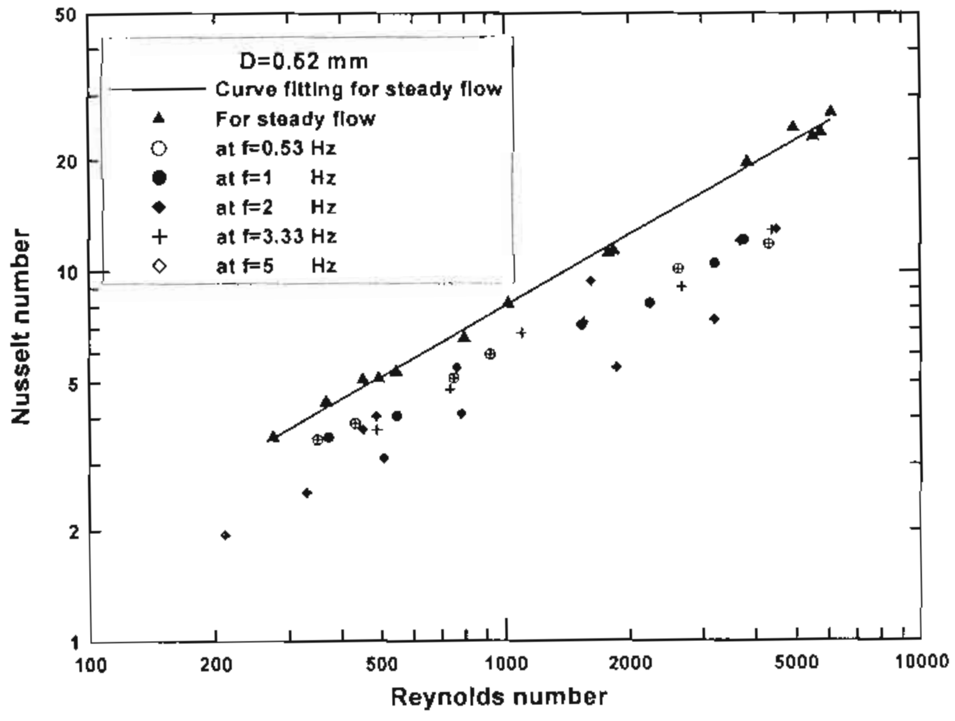


Fig.(10) Nusselt number versus Reynolds number for pulsating flow compared with steady flow.

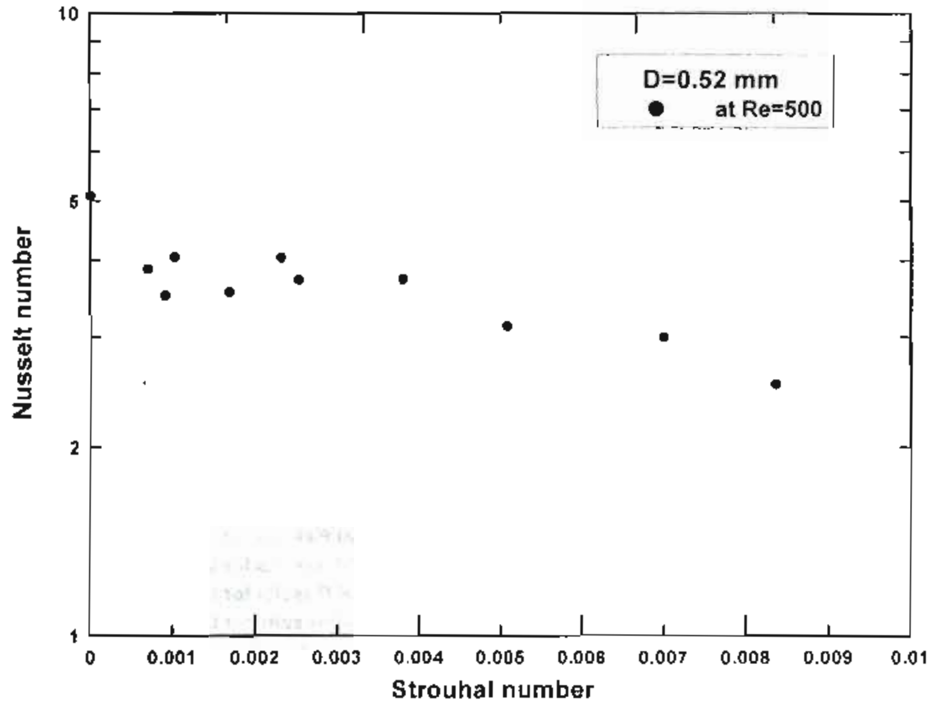


Fig.(11) Nusselt number against Strouhal number for tested tube (D=0.52 mm).

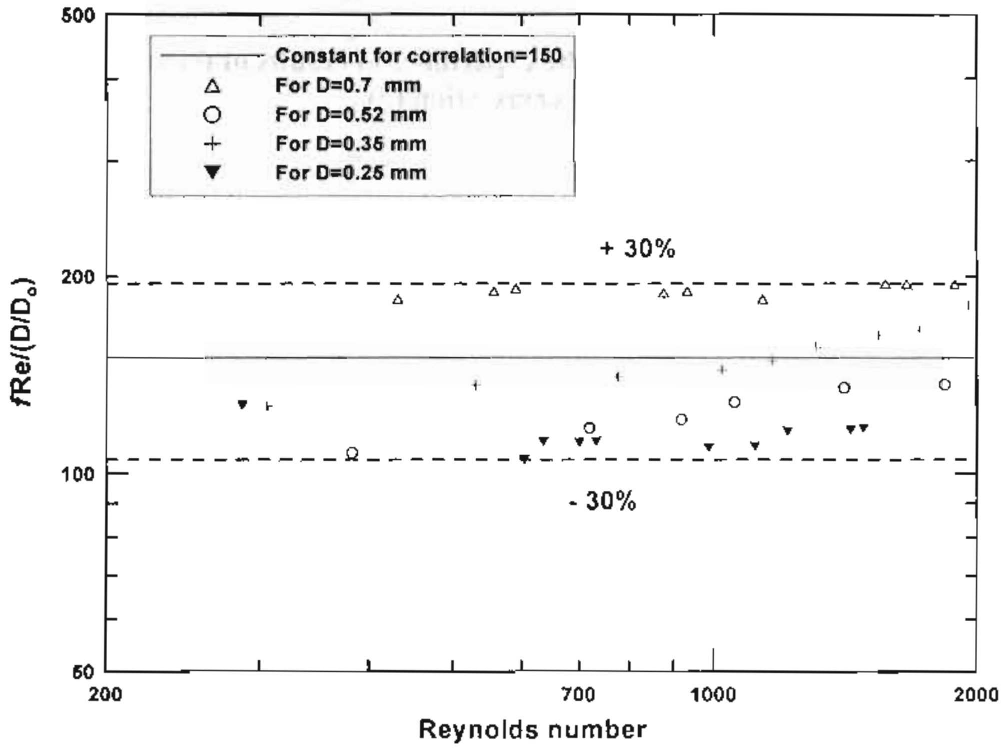


Fig. (12) The deviation for the experimental results of Nusselt number from the proposed correlation (6).

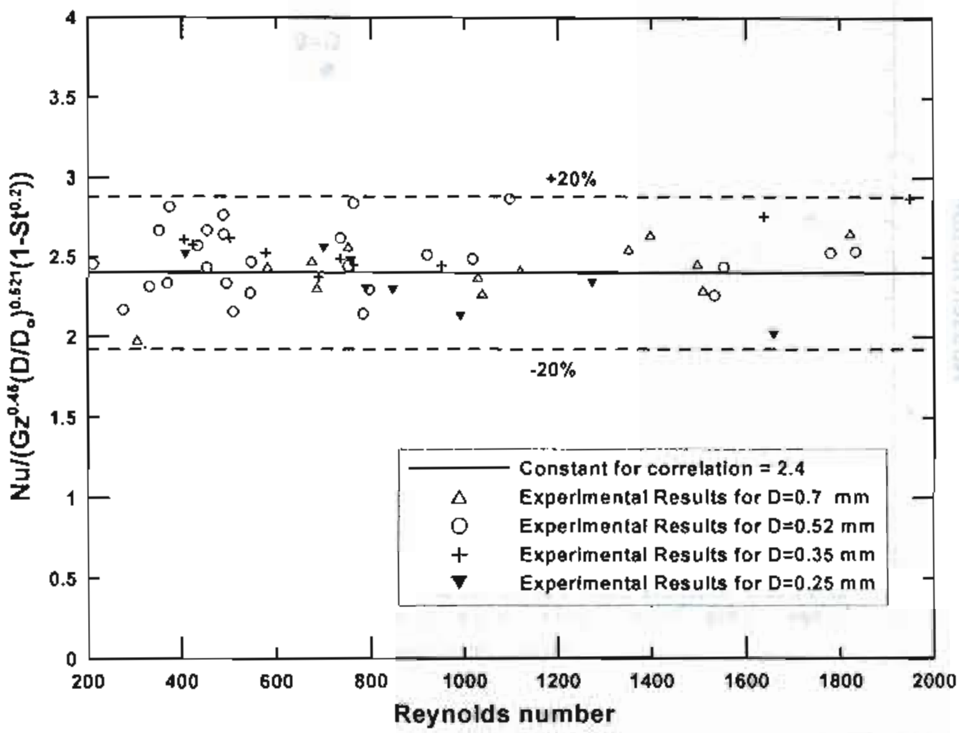


Fig. (13) The deviation for the experimental results of friction factor from the proposed correlation (7).