Journal of Thermal Engineering
Yildiz Technical University Press, Istanbul, Turkey
Vol. 2, No. 2, pp. 677-682, April, 2016.

http://eds.yildiz.edu.tr/journal-of-thermal-engineering/Articles
Manuscript Received May 28, 2015; Accepted June 06, 2015

This paper was recommended for publication in revised form by Regional Editor Lian-Ping Wang

EXPERIMENTAL FULLY-DEVELOPED THERMAL CONVECTION FOR NON-DARCY WATER FLOW IN METAL FOAM

***Nihad Dukhan**
University of Detroit Mercy
Detroit, Michigan, USA

Özer Bağcı
Istanbul Technical University
Istanbul, Turkey

Mustafa Özdemir
Istanbul Technical University
Istanbul, Turkey

Levent Kavurmacıoğlu
Istanbul Technical University
Istanbul, Turkey

Keywords: experiment; metal foam; convection; Forchheimer; fully developed
* *Corresponding author: Phone:* +313 993 3285
E-mail address: nihad.dukhan@udmercy.edu

ABSTRACT

Experimental heat transfer data for water flow in commercial open-cell aluminum foam cylinder heated at the wall by a constant heat flux, is presented. The foam had 20 pores per inch (ppi) and a porosity of 87%. The measurements included wall temperature along flow direction as well as average inlet and outlet temperatures of the water. Flow speeds were in the non-Darcy regimes (transitional and Forchheimer). Heat fluxes were 14,998 W/m^2 and 26,347 W/m^2. The behavior of the wall temperature clearly shows thermal fully-developed conditions. The experimental Nusselt number is presented as a function of axial distance in flow direction, and showed what seemed to be a periodic development. A correlation for the average Nusselt number as a function of flow Reynolds number is provided. The experimental data can be used for validation of other analytical solutions, numerical models and heat-exchange engineering designs based on metal foam.

INTRODUCTION

Open-cell metal foams are excellent candidates for heat exchange designs. The foams have relatively high conductivities and very large surface area per unit volume. The internal structure of the foam interferes with fluid flow and causes vigorous mixing, which enhances convection between the internal solid surface of the foam and the fluid. Boomsma et al. [1] have shown that compressed open-cell aluminum foam heat exchangers generated thermal resistances that were two to three times lower than the best commercially available heat exchanger, at the same pumping power.

Due to the complexity of heat transfer phenomenon inside metal foam, researchers have solved simplified forms of the governing equations, and relied on numerical simulations. Calmidi and Mahajan [2] numerically studied forced convection of air flow in aluminum foam. Hwang et al. [3] indicated that the local Nusselt number for air flow in metal foam increased with increasing Reynolds number. Angirasa [4] numerically studied convection heat transfer due to water flow in metal foam heat dissipaters. He invoked local thermal equilibrium. The validity of the local thermal equilibrium assumption is questionable due to the difference in the thermal conductivities of the solid and fluid phases.

Lu et al. [5] analyzed forced convection in a tube filled with metal foam subjected to constant wall heat flux. The two-equation model, which relaxes the thermal equilibrium assumption, was solved. They employed the Brinkman-extended Darcy momentum model. A closed-form solution for the solid and the fluid temperatures was presented. They exploited the solution for investigating the effect of various foam parameters in practical heat-exchange designs. Analytical solutions in porous media continue to be sought [6-8] due to their utility, identifying trends of critical variables, parametric studies and for validating numerical models.

Direct comparisons to experimental values of key variables is lacking in many analytical porous media studies concerning heat transfer [6-8], and seem to be non-existent in the literature concerning heat transfer in metal foam. Experimental verifications, when possible, add confidence to analytical solution and validate numerical modeling and simulations. Experimental data also has intrinsic value and they can provide correlations for practical engineering design. In 2006, Lu et al. [5] stated that no experimental data was available for metal-foam-filled pipes. The circular geometry is preferred in many heat exchange applications. In a 2012 comprehensive review, Zhao [9] indicated that there has been a lack of reliable experimental heat transfer data for open-cell metal foam in general. A summary of available experimental data for heat transfer in metal foam will be summarized next.

Studies with Air

Calmidi and Mahajan [2], measured the wall temperature of aluminum foam sample bounded by substrates and heated. For comparing to their numerical results, they used the measured wall temperature to obtain the average surface heat transfer coefficient. Hwang et al. [3] obtained wall and exit temperature measurements for air flow through metal foam having dimensions 60 × 25.4 × 60 mm^3. The data was used to investigate heat transfer in terms of Nusselt number. Bhattacharya et al. [10] used the same kind of temperature measurements to obtain the effective conductivity of metal foam, while Bhattacharya and Mahajan [11] used similar techniques to assess the thermal performance of a finned-metal-foam heat sink. Zhao et al. [12] tested convection due to air flow in several foam samples 127 × 127 × 12 mm^3 each. Kurbas and Celik [13] published the results of an experimental study of forced and mixed convection heat transfer in a foam-filled rectangular channel for air as the working fluid. Similar measurements were conducted. Mancin et al. [14] presented experimental data for forced convection due to airflow in several samples of aluminum and copper foam. Overall and interstitial heat transfer coefficients were obtained; foam finned surface efficiency were investigated. Each foam sample was 100 mm long and had a cross section of 20 mm × 20 mm. Dukhan et al. [15] measured actual air temperature inside a cylinder of metal foam heated with constant heat flux and cooled by air. The tube had a length of 152.4 mm in the flow direction and an inside diameter of 255.6 mm. Wall temperature measurements were limited, and the Nusselt number was not discussed.

Studies with Air

Experimental studies concerning heat transfer in heated metal foam with water as the cooling fluid are few indeed. Water flow provide much higher heat transfer rates due to it higher thermal conductivity (relative to that of air) and also due to dispersion which is negligible for air flow in porous media.

Boomsma et al. [1] experimentally investigated compressed aluminum foam performance as a compact heat exchanger using water as a coolant. The foam size was 40 mm × 40 mm × 2 mm. Hetsroni et al. [16] investigated the cooling of 40-ppi metal-foam heat sink for transmission window. The foam dimensions were 10 mm × 45 mm × 2 mm. Kim et al. [17] studies convection in a sample of metal foam 9 × 90 × 30 mm^3. One common fact about these studies is that they employed small foam sample sizes (small length at least in one direction), which makes there results specific to the samples tested.

In the current study, direct measurements of wall and inlet and outlet temperatures for water flow inside heated commercial aluminum foam are presented. The length of the foam cylinder tested is long to ensure full thermal development and minimize size effects. Flow velocities encountered Darcy and non-Darcy regimes. Such measurements have an intrinsic value. To the knowledge of the present authors, the experimental data are novel, and can be used for validation of analytical and numerical models of heat transfer in metal foam and for assessing the performance of heat-exchange engineering designs based on such media.

EXPERIMENT

The experimental heat transfer model was principally a cylindrical aluminum tube brazed to an aluminum foam core, Fig. 1(a). Brazing of the two similar materials is known to minimize thermal contact resistance, and hence facilitate heat transfer from the wall to the foam. The tube had a length of 30.50 cm in the flow direction, an inside diameter of 5.08 cm and a wall thickness of 0.63 cm. The foam was obtained commercially (ERG Materials and Aerospace [18]), and was made from aluminum alloy 6101-T6 and had 20 pores per linear inch (ppi). The porosity of the foam was 87.6 % (calculated from measurements of its volume and weight). Other geometric parameters can be found in [18].

For measuring the wall temperature, 17 holes were drilled. The diameter of each hole was 1 mm and the depth was 4 mm. The location of the first hole was 150 mm from the inlet of the foam tube; the rest of the holes were at 10 mm from each other along the length of the tube (the flow direction). This caused the last hole to be at 1 mm from the outlet of the foam tube. In each hole, a type-K thermocouple was inserted, Fig. 1(b). Thermal epoxy then filled the remaining volume of each hole, while ensuring that the bead of the thermocouple was touching the bottom of the hole. The epoxy guaranteed that there were no air pockets trapped in the holes.

A surface band heater was wrapped around the outer surface of the aluminum cylinder. The heater had a sandwich-like inner structure consisting of a mica plate wrapped by resistance heater ribbons of 1.66 Ohm/m; two more mica layers on both sides for electrical insulation and steel sheet layers to cover the inner elements and to give structure to the heater. The heater had an electrical power of 1,780 Watts at 60 Volts; it was powered by two 40-VDC power supplies connected in series.

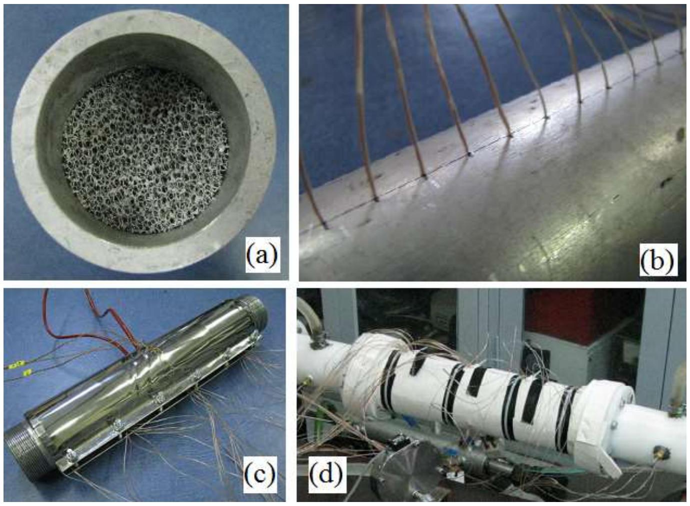

Figure 1 Experimental model construction: (a) aluminum tube with brazed foam, (b) Thermocouple inserted into holes, (c) Heater wrapped around the pipe and (d) Heat transfer model insulated

Thermal grease was placed between the heater surface and the pipe surface to minimize contact resistance. The heater covered the outside area, Fig. 1(c). The whole assembly was then covered with five layers of ceramic fiber paper (4-mm thick) with a thermal conductivity of 0.058 W/mK in order to insulate the test section from ambient air, Fig. 1(d).

Experiments were performed in an open-loop water tunnel as shown in Fig. 2. The heat transfer test section was connected to two 51.4-mm-diameter 200-mm-long Polyethylene tubes at its two ends using specially-designed flanges, Fig. 1(d). Temperature probes were inserted in these tubes to measure the inlet and outlet temperature of the water. Each probe has five thermocouples spanning the cross section in order to obtain a good average value for the inlet and outlet temperatures. The outlets of the Polyethylene tubes were connected to stainless steel pipes 32 mm in diameter and 110 cm in length. A hose and a valve were used for connecting the outlet of one steel pipe to a 50-liter tank for collecting water at the outlet over a known length of time for measuring mass flow rates.

An elevated (3.5 m above ground) plastic tank (diameter 41 cm, height 44 cm) with a network of hoses and valves, that guaranteed a constant water height (33.2 cm) in the tank at all times, supplied water to the test section. Heavily-filtered tap water was supplied to the tank using a 1.27-cm hose. Four 1.90-cm outlet houses were attached to the tank at a height of 36.3 cm from the bottom of the tank. The supply line was at 41 cm from the bottom of the tank (3.7 cm higher than the four outlet hoses).

To supply constant-pressure flow to the test section, one end of another 1.90-cm hose was connected at 3.1 cm from the bottom of the tank, while the other end was connected to the test section. As such, a constant water height of 33.2 cm in the tank during each experimental run was maintained. The flow rate provided by the tank was practically constant (less than 4 % variation). The experimental rig was able to produce and hold very low water speeds (starting at 7.6×10^{-5} m/s).

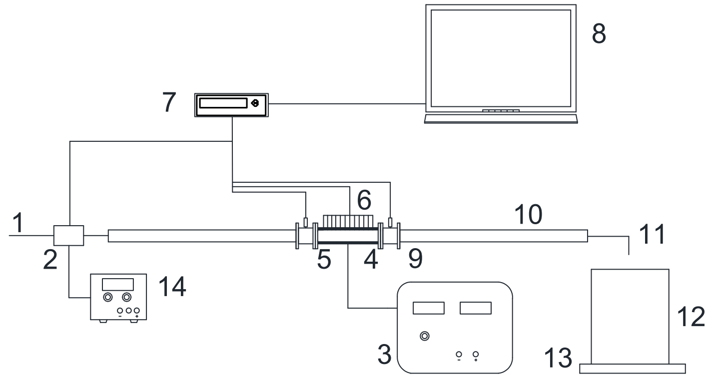

Figure 2 Schematic of the experimental set-up: 1. Filtered water inlet, 2. Magnetic flow meter, 3. 40-VDC power supply, 4. Heater, 5. Test section, 6. Thermocouple wires, 7. Data logger, 8. PC, 9. Polyethylene tubes, 10. Stainless steel tubes, 11. Water outlet, 12. 50-liter collecting tank, 13. Mass scale, 14. Second DC power supply

For a given run, control valves were adjusted and water was allowed to flow into the foam. A valve at the inlet provided fine control over the mass flow rate. Heat was supplied by the heater to produce and maintain a heat flux of 14,998 W/m^2 for only the lowest flow rate (15.02 g/s) and 26,347 W/m^2 for the higher flow rates. It should be noted that these fluxes are based on the inside diameter of the pipe, and they exclude an estimated heat loss of about 5%. For any given run, it took about 30 minutes for steady state conditions to be reached. At steady state and for a fixed valve opening (flow rate), water exiting the test section was captured in the collecting tank over a known period of time. Knowing the time and the mass of the collected water, the mass flow rate and the average flow velocity were determined. The free leads of all thermocouples were connected to the data logger, which sent the steady-state temperature readings to a computer.

Uncertainty Analysis

Uncertainty in the directly-measured quantities: length, mass, time and temperature was based in errors associated with measuring devices provided by manufacturers. The propagation of error in derived quantities, i.e., average flow velocity, Reynolds number and Nusselt number, was assessed via uncertainty analysis as described by Figliola and Beasley [19]. The uncertainties in length and diameter of the foam were 0.18% and 1.0%, respectively. As for the temperature, the error in the reading of a thermocouple was 0.4%, as given by the manufacture.

As an example of uncertainty in derived quantities for Darcy-regime data, the average flow velocity, U was obtained by dividing the mass flow rate, $\dot{m}$ by density, ρ and the cross-sectional area of the test section, A. Hence the percent uncertainty in average velocity δ_U/U is given by Figliola and Beasley [19]:

$$\frac{\delta_U}{U} = \pm\sqrt{\left(\frac{\delta_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{\delta_\rho}{\rho}\right)^2 + \left(\frac{\delta_A}{A}\right)^2} \qquad (1)$$

which results in 0.37%. The uncertainty in the mass flow rate had been obtained in the same manner, which resulted in identical values as those obtained for the mass flow rate due to the very small uncertainties in the density and the cross-sectional area. For non-Darcy flow data, the flow rate was measured directly by a flow meter with an uncertainty of 2%, reported by the manufacture.

The uncertainty in μ was estimated as 1×10^{-7} N.s/m^2, taken as the accuracy of the reported values in property tables [20]. This value is small enough to cause negligible impact on the overall uncertainty in Reynolds number, therefore it was ignored. Using similar calculation as shown by Eq. (1), the uncertainty in Reynolds number was obtained as 2.69%.

The effective thermal conductivity of the solid aluminum ligaments of the foam was obtained from an analytical one-dimensional model given by Calmidi and Mahajan [21]. These researchers indicated that this model was excellent in matching their measured values of the effective conductivity. The fluid phase effective thermal conductivity was obtained as 0.58 W/m.K. Similarly, the uncertainty in the effective conductivity was conservatively assumed to be 10%. The uncertainty in the heat flux was assumed to be 10%. The uncertainty in Nusselt number was obtained as ±14.12%.

RESULTS AND DISCUSSION

The data runs covered transitional and Forchheimer regimes. These regimes have been identified previously by flow experiments using the same set-up [22].

According to Nield et al. [23] who investigated thermal entry length for the case of a circular-tube porous media subjected to constant heat flux, that the thermal entry length z_t is given by

$$\left(\frac{z_t}{4r_0Pe'}\right)^{1/2} \cong 0.1 \tag{2}$$

in which the current nomenclature was employed, and *Pe'* is the Péclet number defined by

$$Pe' = \frac{\rho c_p r_0 U}{k_e} \tag{3}$$

According to Eq. (3), the longest thermal entry region would result for the highest velocity. This resulted in a maximum entry length of 2.34 cm for the current study. Hence the end of the entry region is far away from where the wall temperature of the current study was obtained, i.e., data was collected in the fully developed region, as will be shown below.

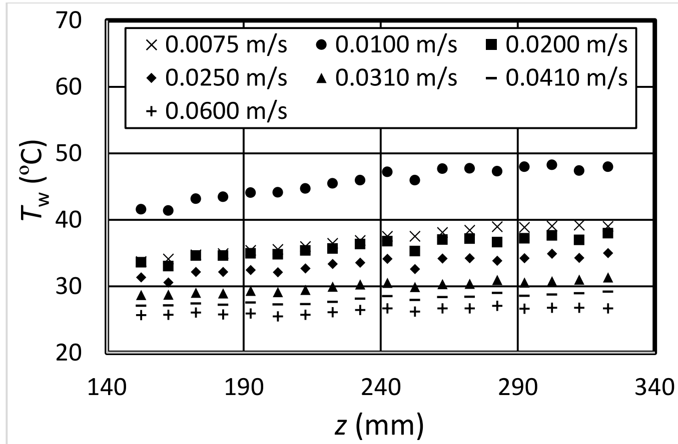

Figure 3 Wall Temperature as a Function of Axial Distance for Non-Darcy Regimes

Wall Temperature

Figure 3 is a plot of the wall temperature as a function of axial distance from the entrance for various Non-Darcy velocity cases. This temperature increases in the flow direction as expected. This trend in the wall temperature of heated metal foam is very similar to what has been presented in Calmidi and Mahajan [2], Kurtbas and Celik [13] and Dukhan et al. [15]. The wall temperature is lower for higher flow velocities indicating better rates of heat transfer for higher velocities, as expected in forced convection. It should be noted the case for flow velocity of 0.0075 m/s, was subjected to a lower heat flux (14,998 W/m^2) compared to the rest of the cases in Fig. 3(b) which were subjected to 26,347 W/m^2.

Estimation of Bulk Temperature

The bulk fluid temperature is defined by

$$T_\mathrm{b} = \frac{1}{UA}\int_A uT\mathrm{d}A \tag{4}$$

where the flow velocity *U* is averaged over the cross-sectional area. The actual bulk temperature of water along the tube is extremely difficult to obtain experimentally. It would require extensive measurements of fluid temperature over the cross section at multiple locations along flow direction. Similar measurements for the velocity distribution would also be required. Subsequently, the integration given in Eq. (4) would have to be evaluated. The measurements needed would be very intrusive and would likely alter the internal structure of the foam, and change flow and heat transfer phenomena inside it.

A reasonable estimation of the bulk temperature can be obtained as follows. It is well established from analysis, e.g. [7], that for thermal fully-developed conditions that

$$\frac{\mathrm{d}T_b}{\mathrm{d}z} = \frac{\mathrm{d}T_w}{\mathrm{d}z} = \mathrm{constant} \tag{6}$$

In other words, the slope of the bulk temperature is the same as that of the wall temperature for each flow velocity, which is readily available form plots of the wall temperature (Fig. 3). Having this slope and the average outlet temperature (approximately equal to the bulk temperature at the exit) is sufficient to obtain a straight line representing the variation of bulk temperature in the fully-developed region. This carries a little error due to the existence of the exit region. Nonetheless, the use of bulk temperature estimated in this manner is better for estimating *Nu* as oppose to using a fixed temperature, i.e., the inlet temperature, which is constant and does not capture any thermal physics inside the foam.

Local Nusselt Number

Following [23], the local Nusselt number is defined as

$$\mathrm{Nu} = \frac{Dq"}{k_{fe}(T_w - T_b)} \tag{7}$$

where the effective thermal conductivity of the fluid, k_{fe} was obtained from a model given in Calmidi and Mahajan [21]. The behavior of Nusselt number is shown in Fig. 4 for the non-Darcy regimes.

Higher velocities produce significantly higher Nusselt numbers, as shown in Fig. 4(b) for non-Darcy regimes. In terms of trends, the situation for these regimes is a little different. For the transitional velocities of 0.0075 and 0.010 m/s, Nu is seen to have constant values, albeit with some scatter. For the higher velocities 0.020 and 0.025 m/s, Nu is significantly higher, which may be due to regime change from transitional to Forchheimer. Also, there seems to be a periodic behavior, signifying possible periodic development where the heat transfer alternate between developing and almost developed states. Similar behavior was present in the work of Kurbas and Celik [13] air flow in a foam-filled rectangular channel. There the behavior was explained by considering buoyancy-driven secondary flows and a change in cell velocity. The current authors propose that the behavior in Nu may be related to flow phenomenon inside the cells of the foam, e.g., inertial cores.

The highest three velocities 0.031, 0.041 and 0.060 m/s are not included in the Fig. 4(b) because the wall temperature for these cases showed diversion from being a straight line. Line fits produce low correlation factors. In addition, for these high velocities, the heat transfer was enhanced and the overall change in wall temperature along the foam was relatively low, which added to the relative uncertainty. As such a Nusselt number could not be obtained for these cases.

Correlation

Figure 5 is a plot of Nu as a function of Re. The flow Reynolds number is defined as

$$Re = \frac{\rho U D}{\mu} \tag{10}$$

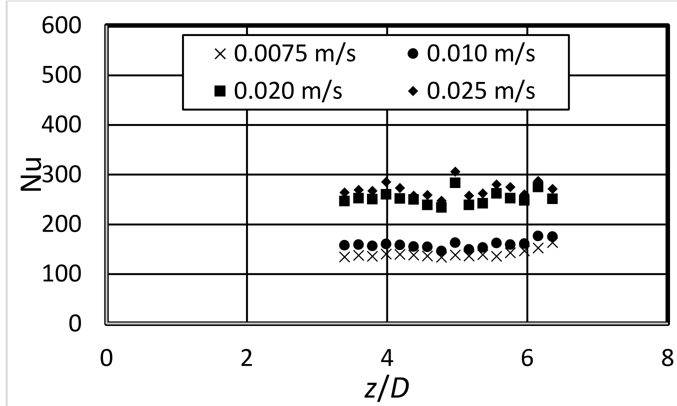

Figure 4 Nusselt Number as a Function of Axial Distance for Non-Darcy Regimes

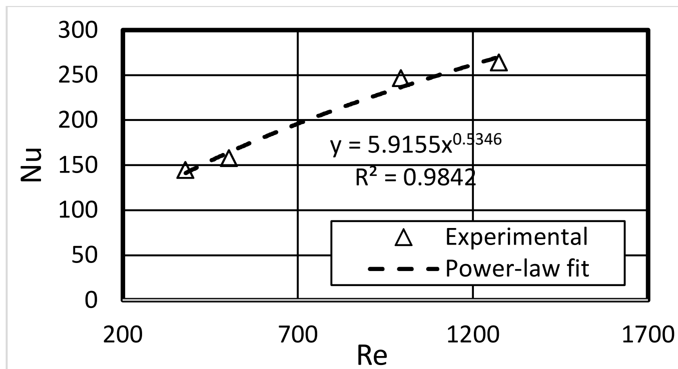

Figure 5 Nusselt Number as a Function of Reynolds Number

The Nusselt number is seen to be a strong function of Re (or flow velocity). The non-Darcy flow Nu correlates well with Re according to a power law:

$$Nu = 5.91 Re^{0.53} \tag{11}$$

The power-law relation between Nu and Re non-Darcy regime data is common [2, 12, 16]

CONCLUSION

Direct measurement of the wall temperature of a heated cylinder filled with metal foam subjected to constant heat flux and cooled by water flow, was conducted. Average inlet and exit water temperatures were also measured.

The data points were in the transitional and Forchheimer regimes and in the thermally fully-developed region. The bulk fluid temperature was estimated and the Nusselt number was calculated. Previously unknown heat transfer phenomenon was observed constituting a seemingly periodic behavior of Nusselt number along the foam. In the non-Darcy regimes, Nusselt number correlated well with the Reynolds number in a power law.

ACKNOWLEDGMENTS

This work was supported by the Scientific & Technological Research Council of Turkey (TUBİTAK) under program 2221: 1059B211301074, for which the authors are very thankful.

NOMENCLATURE

A	cross-sectional area (m^2)
k	thermal conductivity ($W.m^{-1}.K^{-1}$)
Nu	Nusselt number
q''	heat flux ($W.m^{-2}$)
T	temperature (^oC)
u	flow velocity ($m.s^{-1}$)
U	mean flow velocity ($m.s^{-1}$)
r_o	radius of foam cylinder (m)
Re	Reynolds number
z	axial coordinate along flow direction (m)

Greek

μ	viscosity (Pa.s)
ρ	density of fluid ($kg.m^{-3}$)

Subscripts

f	fluid
b	bulk (mean) value
e	effective
TE	thermal equilibrium
w	wall

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