INTERNATIONAL JOURNAL OF HEAT AND TECHNOLOGY

ISSN: 0392-8764
Vol. 35, Special Issue 1, September 2017, pp. S288-S292
DOI: 10.18280/ijht.35Sp0139

A publication of IIETA

IIETA International Information and Engineering Technology Association

Advancing the World of Information and Engineering

http://www.iieta.org/Journals/IJHT

Numerical analysis of hybrid nanofluids as coolants for automotive applications

Gabriela Huminic*, Angel Huminic

Transilvania University of Brasov, Mechanical Enginering Department, 29 Bulevardul Eroilor Street, Brasov 500036, Romania

Email: gabi.p@unitbv.ro

ABSTRACT

The study the laminar convective heat transfer of hybrid nanofluids in a 3D flattened tube under constant heat flux condition was carried out. The effects of Reynolds number and concentration of hybrid nanoparticles on heat transfer characteristics of MWCNT–Fe_3O_4/water hybrid nanofluids were numerically investigated. The simulations were performed for the Reynolds number in the range 50-1000 and for two volume concentrations of hybrid nanoparticles, 0.1% and 0.3% respectively. Results showed that the heat transfer coefficient increases with increasing volume concentration of hybrid nanoparticles and increasing Reynolds number. At low Reynolds numbers, the application of MWCNT–Fe_3O_4/water hybrid nanofluids in flattened tube can enhance heat transfer up to 31% compared with water (base fluid). Numerically results were validated by comparison of simulations with results available in literature.

Keywords: Hybrid Nanofluids, Flat Tube, Heat Transfer.

1. INTRODUCTION

In past decade, extensive research was focused on applications of nanofluids in the automotive industry, especially to automotive radiator. The researches carried out so far suggest that the nanofluids could be an alternative to conventional coolants (water, ethylene glycol, water and ethylene glycol mixture) used in various thermal applications [1-3], as well in automotive radiators [4,5].

Vajjha et al. [6] carried out a numerically study concerning to heat transfer characteristics of the Al_2O_3 and CuO nanoparticle into water-ethylene glycol mixture (40:60). The two nanofluids flow through a flattened tube under laminar flow. The heat transfer coefficient (h=50 W/m^2K) and the temperature (T=303 K) on walls of tube were imposed. The study was performed in following conditions: the Reynolds number,Re=100-2000, the inlet temperature, T_{in} = 90 °C, the volume concentration of nanoparticles, ϕ=0-10% for the Al_2O_3 and ϕ=0-6% for the CuO nanoparticles. Their results indicated that thermal performances of the flattened tube were improved due to using nanofluids. There was an increase in heat transfer coefficient with increasing of Reynolds number and also with increasing of volume concentration of nanoparticles.

Huminic and Huminic [7] numerically investigated the effect of volume concentration of nanoparticles, of Brownian motion and of Reynolds number on thermal performances of the flattened tube using CuO/ethylene glycol nanofluids under laminar flow. The heat transfer coefficient (h=50 W/m^2K) and the temperature (T=303 K) on walls of tube were imposed. The boundary conditions imposed in this study was following: the Reynolds number, Re=10-125, the inlet temperature, T_{in} = 50 °C, the volume concentration of nanoparticles, ϕ=0-4% for the CuO nanoparticles. The numerical results were compared with results obtained in the case circular and elliptic tubes. The authors founded that the thermal performances of flattened tubes were significantly enhanced compared to that of the elliptic and circular tubes using CuO/ethylene glycol nanofluids as coolant.

Delavari et al. [8] numerically investigated heat transfer and flow characteristics of ethylene glycol and water based on Al_2O_3 nanoparticles flowing through a flattened tube under laminar and turbulent flow. Single and two-phase approaches were employed. The study was performed in following conditions: the Reynolds number, Re=1220-2440 (laminar flow) and Re=9350-23000 (turbulent flow), the inlet temperature, T_{in} = 35-60 °C, and the volume concentration of nanoparticles, ϕ=0-1% . They observed that the numerical results using two-phase approach were closer to the experimental data than the results obtained using single-phase method.

Also, the heat transfer and flow characteristic of the water based Al_2O_3 and CuO nanoparticle flowing through a flattened tube under laminar flow were numerically investigated by M. Elsebay et al. [9]. The study was performed in following conditions: the Reynolds number, Re=250-1750, the inlet temperature, T_{in}=80 °C, the temperature, T_w=303 K, the heat transfer coefficient, h_w=50 W/m^{2}K and the volume concentration of nanoparticles, ϕ=0-7%. They observed that although the heat transfer coefficients increase with 45 and 38% for Al_2O_3/water and CuO/water nanofluids, the increase of heat transfer coefficient was correlated with a significantly increase in the friction factor and in pressure drop of 271 and 267% for Al_2O_3/water and 266 and 226% for CuO/water nanofluids.

It is known that nanofluids have a high thermal conductivity but also a high viscosity, which leads to limited use of nanofluids in the automotive industry. In last year's, research was focused on the development of new working fluids which combining of two or three solid materials into conventional fluids. These new fluids are called hybrid nanofluids. Thus, the main goal of this work is the study of influence of concentration of hybrid nanoparticles and Reynolds number on flattened tube.

2. SOLUTION METHODOLOGY

2.1. Physical model

In this study, the simulations were performed for a flattened tube of an automobile radiator (Fig. 1). The geometrical dimensions of the flattened tube were: the height, H=2.56 mm, the width, W=16.1544 mm, and the length, L=500 mm.

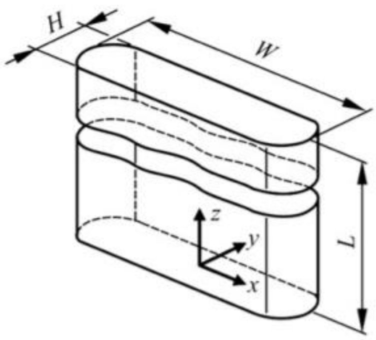

Figure 1. Physical model of flattened tube

2.2. Governing equations

The single phase model to investigate the thermal and fluid dynamic behavior of hybrid nanofluids was employed. The governing equations adopted in this study were based on the following assumptions: steady state process, fluid inside the flattened tube was Newtonian and incompressible, and radiative effect was negligible. Also, governing equations were solved in the Cartesian coordinate system using the ANSYS CFX-14.0 software.

The continuity equation:

$$\frac{\partial}{\partial x_i}\left(\rho U_i\right)=0 \tag{1}$$

The momentum equations:

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

where i, j ∈ {1, 2, 3}.

The energy equation was solved to calculate the temperature distribution:

$$\frac{\partial}{\partial x_i}\left(\rho c_p U_j T\right)=\frac{\partial}{\partial x_i}\left(k\left(\frac{\partial T_i}{\partial x_i}\right)\right) \tag{3}$$

2.3. Grid independence test and validation

In order to grid independence test, three grid sizes were compared in terms of velocity and temperature. A multi-block meshing scheme with hexahedral elements was used for the generation of the structure grids (Fig.2).

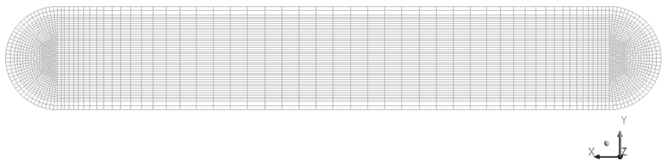

Figure 2. Grid layout used in the present analysis

Table 1 summarizes the grid independence study by comparing the velocity and temperatures profiles for water at Re=100, T=313.15 K and Y=0 m. After this grid independence study, grid II 90 × 40 × 200 was chosen as the optimal grid size.

Table 1. Grid independence study

Grid	Velocity [m/s]	Temperature [K]
60x20x100	0.02420	309.013
90x40x200	0.02421	309.016
120x60x300	0.02422	309.016

In order to validation of the numerical model, the results were compared with the Shah and London correlation [10] and also with the numerical results carried out by Vajjha [6].

Shah and London correlations [10] for the constant wall heat flux in a circular tube under laminar flow are given below:

$$Nu=1.953\left(\mathrm{Re}\,\mathrm{Pr}\frac{d_h}{L}\right)^{1/3}$$

$$\text{for}\left(\mathrm{Re}\,\mathrm{Pr}\frac{d_h}{L}\right)\geq 33.33 \tag{4}$$

$$Nu=4.364+0.0722\left(\mathrm{Re}\,\mathrm{Pr}\frac{d_h}{L}\right)$$

$$\text{for}\left(\mathrm{Re}\,\mathrm{Pr}\frac{d_h}{L}\right)<33.33 \tag{5}$$

In Eqs. (4) and (5), the hydraulic diameter of the flattened tube was d_h=4.536·10^{-3} m. The Reynolds and Prandtl numbers were defined as:

$$\mathrm{Re}=\frac{U\ d_h}{\upsilon}=\frac{\rho U\ d_h}{\mu}\ \text{and}\ \mathrm{Pr}=\frac{\mu\ c_p}{k} \tag{6}$$

As seen in Table 2 there a good agreement between the calculated results and reported theoretical and numerical results.

Table 2. The validation of numerical model

Working fluid	*Nusselt number, Nu*	
Water	*Shah-London correlation [10]*	*Numerical results*
	Re=75	
	4.575	4.695
	Re=100	
	4.646	5.566
Ethylene glycol	*Vajjha et al.[6]*	*Numerical results*
	Re=100	
	6.49	6.68

2.3. Boundary conditions

Numerical simulations were performed in a laminar flow with the Reynolds numbers in the range of 50-1000 for hybrid nanofluids based on water with volume concentrations of 0.1% and 0.3% for MWNCT+Fe_3O_4 hybrid nanoparticles.

In this study, the boundary conditions imposed were the following:

- Inlet: velocity (Re=50-1000) and temperature (T=313 K);
- Outlet: pressure (P=0);
- Wall: heat transfer coefficient (h=50 W/m^{2}K) and temperature (T=293 K), non-slip condition;
- Interference surfaces of the domains: conservative interface flux.

2.4. Data reduction

Heat transfer rate can be expressed as

$$Q = \dot{m}c_p(T_{in} - T_{out}) \tag{7}$$

Heat transfer coefficient can be written

$$h = \frac{Q}{A_p(T_b - T_w)} = \frac{\dot{m}c_p(T_{in} - T_{out})}{A_p(T_b - T_w)} \tag{8}$$

Peripheral area is

$$A_p = 2(LH + LW) \tag{9}$$

Hydraulic diameter of tubes is given by

$$d_h = 4\frac{A}{P} \tag{10}$$

3. THERMO-PHYSICAL PROPERTIES OF HYBRID NANOFLUIDS

The thermo-physical properties of hybrid nanofluids used in this study were chosen from the study of Sundar et al. [11]. In Table 3 were presented the thermo-physical properties of water based on MWCNT-Fe_3O_4 (26%:74%) hybrid nanoparticles.

Table 3. Thermo-physical properties of water [12] and MWCNT+Fe_3O_4 /water at 313 K [11]

Working fluid		*MWCNT-Fe_3O_4/water*	
Thermo-physical properties	*Water*	*ϕ=0.1%*	*ϕ=0.3%*
Thermal conductivity [W/mK]	0.633	0.720	0.7656
Viscosity [Pa s]	$0.658 \cdot 10^{-3}$	$0.610 \cdot 10^{-3}$	$0.760 \cdot 10^{-3}$
Density [kg/m^3]	992.20	995.85	1003.56
Specific heat [J/kg K]	4175	4179.66	4180.99

4. RESULTS AND DISCUSSIONS

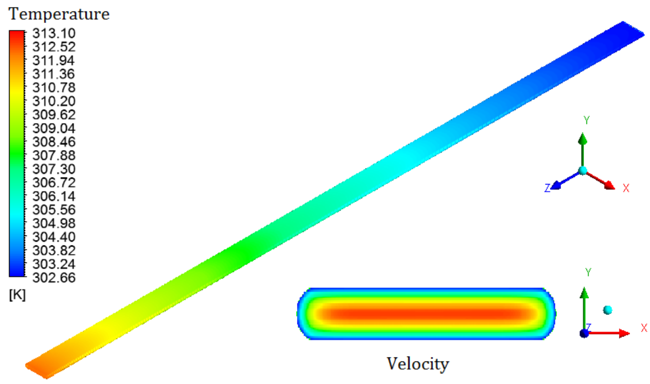

Figure 3 Temperature distribution and velocity profile for water at Re=50

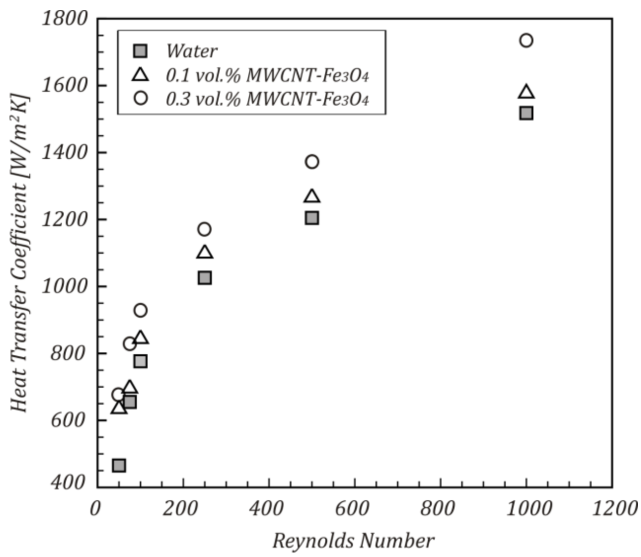

Figure 4. The variation of the heat transfer coefficient for different Reynolds number

In the present study, the heat transfer performances of flattened tube using hybrids nanofluid with volume concentrations of 0.1% and 0.3% of MWCNT-Fe_3O_4 hybrid nanoparticles in water were studied at different Reynolds numbers (Re=50-1000) and at the temperature of T=313.15 K.

The temperature and velocity profiles for water of the flattened tube at Re=50 were given in Fig. 3.

Fig. 4 shows the results of the average heat transfer coefficient for the Reynolds number within the range Re=50-1000, and two concentrations of MWCNT-Fe_3O_4 hybrid nanoparticles. As shown, heat transfer coefficient increases with increasing Reynolds number and increasing concentration of nanoparticles. Also, at low Reynolds numbers, the heat transfer coefficient increase was significantly higher than at high Reynolds numbers. Moreover, it is observed that heat transfer coefficient of hybrid nanofluids was considerably higher than that of water (base fluid).

Fig. 5 shows the effect of the Reynolds number on the ratio of convective heat transfer coefficient (h_{nf}/h_{bf}) defined as the ratio between the heat transfer coefficient of hybrid nanofluid (h_{nf}) and the heat transfer coefficient of base fluid (h_{bf}). From the numerical results, higher heat transfer coefficient ratio can be observed with increasing of the concentration of hybrid nanoparticles in water. The maximum heat transfer coefficient ratio was 1.45 at 0.3 vol.% MWCNT-Fe_3O_4 hybrid nanoparticles and Re=50. At same concentration of hybrid nanoparticles and Re=1000, heat transfer coefficient ratio was 1.143. Also, heat transfer coefficient ratio decreases with increasing Reynolds number for all studied concentrations of nanoparticles.

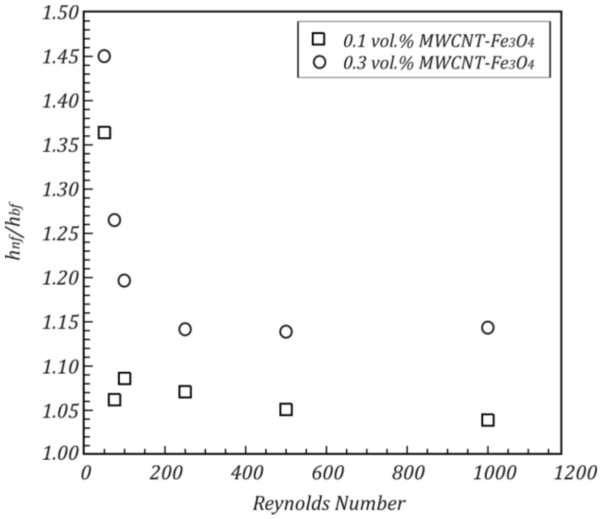

Figure 5. The heat transfer coefficient ratio versus Reynolds number

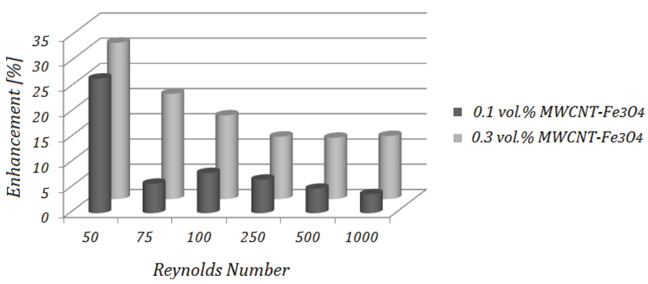

Figure 6. The heat transfer enhancement for different Reynolds number

The heat transfer enhancement for different Reynolds number and concentrations of hybrid nanoparticles is shown in Fig. 6. As shown in Fig. 6, the heat transfer enhancement increases with increasing concentration of hybrid nanoparticles and decreases with increasing Reynolds number.

The maximum enhancement ($100(h_{nf}-h_{bf})/h_{bf}$) was 31% for a concentration of 0.3 vol.% nanoparticles and a Reynolds number of Re=50. At same Reynolds number, the enhancement of heat transfer coefficient for a concentration of the nanoparticles of 0.1% was 26.68%.

The pumping power required to circulate the working fluid can be expressed as:

$$W = AU(\Delta P) \tag{11}$$

The table 4 shows the pressure loss and the pumping power at different Reynolds numbers. As seen, the pressure loss increases with increasing Reynolds number in all cases. When using 0.1% MWCNT-Fe_3O_4 hybrid nanoparticles in flattened tube, the pressure loss increases up Re=250, after which it can be observed a decrease of it compared with the water (base fluid). Also, at a concentration of 0.3% MWCNT-Fe_3O_4 hybrid nanoparticles, the pressure loss increases compared with water and 0.1% MWCNT-Fe_3O_4 hybrid nanofluid. Furthermore, it can be seen a decrease of the pumping power at a concentration of 0.1% nanoparticles and Reynolds numbers in the range 75-1000 compared cu water, while at a concentration of 0.3% can be observed a significantly increase of the pumping power.

Table 4. Pressure loss and the pumping power in a flattened tube using hybrid nanofluids

Reynolds number	50	75	100	250	500	1000
Water						
Pressure loss [Pa]	18.21	23.69	28.55	50.07	85.52	157.92
Power [W]	$5.28\cdot10^{-6}$	$1.04\cdot10^{-5}$	$1.71\cdot10^{-5}$	$7.20\cdot10^{-5}$	$2.48\cdot10^{-4}$	$9.15\cdot10^{-4}$
0.1% MWCNT-Fe_3O_4						
Pressure loss [Pa]	20.33	25.02	29.87	50.57	84.09	151.99
Power [W]	$5.48\cdot10^{-6}$	$1.0\cdot10^{-5}$	$1.67\cdot10^{-5}$	$6.87\cdot10^{-5}$	$2.28\cdot10^{-4}$	$8.20\cdot10^{-4}$
% Power reduction	3.92	-3.97	-2.35	-4.62	-7.81	-10.4
0.3% MWCNT-Fe_3O_4						
Pressure loss [Pa]	27.67	34.51	39.61	67.81	122.67	207.98
Power [W]	$9.23\cdot10^{-6}$	$1.79\cdot10^{-5}$	$2.69\cdot10^{-5}$	$1.14\cdot10^{-4}$	$4.07\cdot10^{-4}$	$1.39\cdot10^{-3}$
% Power reduction	74.85	72.18	57.21	58.01	64.16	51.68

5. CONCLUSIONS

The heat transfer performances of a flattened tube were numerically investigated using water based MWCNT-Fe_3O_4 hybrid nanofluids in laminar flow. The influence of the Reynolds number and volume concentration of hybrid nanoparticles on the cooling performances of hybrid nanofluids and water were studied. The numerically results showed that the heat transfer coefficients of the MWCNT-Fe_3O_4/water hybrid nanofluids were significantly higher than those of the base fluid (water). The increase in heat transfer coefficients was higher at low Reynolds numbers. The heat transfer enhancement of hybrid nanofluids depends of the volume concentration of hybrid nanoparticles, increasing concentration of hybrid nanoparticles of 0.3 % showing a maximum heat transfer enhancement of 31%.

REFERENCES

[1] Huminic G., Huminic A., Fleaca C., Dumitrache F. (2016). Heat transfer characteristics of a two-phase closed thermosyphons using nanofluids based on SiC nanoparticles, *International Journal of Heat and Technology*, Vol. 34, Sp. 2, pp. S199-S204. DOI: https://doi.org/10.18280/ijht.34S202

[2] Wang X.H., Jiao Y.L., Niu Y.C., Yang J. (2015). Study on enhanced heat transfer features of nano-magnetic fluid heat pipe under magnetic field, *International Journal of Heat and Technology*, Vol. 33, pp. 137-143. DOI: 10.18280/ijht.330119

[3] Sivakumar A., Alagumurthi N., Senthilvelan T. (2015). Experimental and numerical investigation of forced convective heat transfer coefficient in nanofluids of Al_2O_3/water and CuO/EG in a serpentine shaped microchannel heat sink, *International Journal of Heat and Technology*, Vol. 33, pp. 155-160. DOI: 10.18280/ijht.330121

[4] Bigdeli M.B., Fasano M., Cardellini A., Chiavazzo E., Asinari P. (2016). A review on the heat and mass transfer phenomena in nanofluid coolants with special focus on automotive applications, *Renewable and Sustainable Energy Reviews*, Vol. 60, pp.1615-1633. DOI: 10.1016/j.rser.2016.03.027

[5] Zhao N., Li S., Yang J. (2016). A review on nanofluids: Datadriven modeling of thermalphysical properties and the application in automotive radiator, *Renewable and Sustainable Energy Reviews*, Vol. 66, pp. 596-616. DOI: 10.1016/j.rser.2016.08.029

[6] Vajjha R.S., Das D.K., Namburu P.K. (2010). Numerical study of fluid dynamic and heat transfer performance of Al_2O_3 and CuO nanofluids in the flat tubes of a radiator, *International Journal of Heat and Fluid Flow*, Vol. 31 pp. 613-621. DOI: 10.1016/j.ijheatfluidflow.2010.02.016

[7] Huminic G., Huminic A. (2013). Numerical analysis of laminar flow heat transfer of nanofluids in a flattened tube, *International Communications in Heat and Mass Transfer*, Vol. 44, pp. 52-57. DOI: 10.1016/j.icheatmasstransfer.2013.03.003

[8] Delavari V., Hashemabadi S.H. (2014). CFD simulation of heat transfer enhancement of Al_2O_3/water and Al_2O_3/ethylene glycol nanofluids in a car radiator, *Applied Thermal Engineering*, Vol. 73, pp. 380-390. DOI: 10.1016/j.applthermaleng.2014.07.061

[9] Elsebay M., Elbadawy I., Shedid M.H., Fatouh M. (2016). Numerical resizing study of Al_2O_3 and CuO nanofluids in the flat tubes of a radiator, *Applied Mathematical Modelling*, Vol. 40, pp. 6437-6450. DOI: 10.1016/j.apm.2016.01.039

[10] Shah R.K., London A.L. (1978). Laminar flow forced convection in ducts, *Journal of Fluids Engineering*, Vol. 102, No. 2, pp. 431-455.

[11] Sundar L.S., Singh M.K., Sousa A.C.M. (2014). Enhanced heat transfer and friction factor of MWCNT-Fe_3O_4/water hybrid nanofluids, *International Communications in Heat and Mass Transfer*, Vol. 52, pp. 73-83. DOI: 10.1016/j.icheatmasstransfer.2014.01.012

[12] Raznjevic K. (1975). Thermodynamic Diagrams and Tables, Bucharest, Romania: Technical Press, pp. 112

NOMENCLATURE

A	tube surface area, m^2
A_p	peripheral area, m^2
c_p	specific heat, J. Kg^{-1}. K^{-1}
dh	hydraulic diameter of the tube, m
h	heat transfer coefficient, $W.m^{-2}.K^{-1}$
H	height, m
k	thermal conductivity, $W.m^{-1}.K^{-1}$
L	length, m
$\dot{m}$	mass flow rate, $kg.s^{-1}$
Nu	Nusselt number
P	pressure, Pa
P	perimeter, m
Pr	Prandtl number
Q	heat transfer rate, W
Re	Reynolds number
T	temperature, K
U	velocity, $m.s^{-1}$
W	width, m
W	power, W (Eq. 10)
x	cartesian coordinates, m

Greek symbols

ρ	density, $kg.m^{-3}$
υ	kinematic viscosity, $m2.s^{-1}$
μ	dinamic viscosity, Pa s

Subscripts

b	bulk
in	inlet
out	outlet
w	wall