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Heat transfer augmentation for the car radiator by using nanofluid

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***Abstract*—The adding solid nanoparticles to liquids are significant topics to enhance heat transfer for many industrial applications in the last ten years. This article included the friction factor and forced convection heat transfer of SiO_2 nanoparticle suspended to water as a base fluid into a car radiator experimentally. Four different concentrations of nanofluids in the range of 1 to 4vol. % have been used. The flow rate changed in the range of 1 to 5 LPM to get Reynolds number with the range of 250 to 2000. The results showed that the friction factor decreases with an increase in flowrate and increase with increasing in volume concentration. Furthermore, the inlet temperature to the radiator has insignificantly affected to the friction factor. Likewise, the heat transfer increases with increasing in flowrate, nanofluid volume concentration and inlet temperature. Meanwhile, application of SiO_2 nanofluid with low concentrations can enhance heat transfer rate up to 30% as a comparison with pure water. The experimental results compared with the investigators experimental data, and there is a good agreement.**

***Index Terms* — Laminar, Nanofluid, Heat transfer, Car radiator, CFD.**

I. INTRODUCTION

The low thermal properties of liquids have led to investigations into additives of small size (less than 100 nm solid particles) to enhance their heat transfer properties and hydrodynamic flow [1]. Base fluids, such as water, ethylene glycol and glycerol, have been used as conventional coolants in automobile radiators for many years; however, these fluids have low thermal conductivities. The low thermal conductivities have thus prompted researchers to search for fluids with higher thermal conductivities than that of conventional coolants. Therefore, nanofluids have been used instead of the commonly used base fluids [2-4]. Experimental studies of friction factor and nanofluids heat transfer enhancement with the flow velocity and nanofluid volume fraction inside heated tube under laminar flow condition has been presented by [5-8]. There are many different applications of thermal-fluid systems, including automotive cooling systems [9]. A numerical study on laminar heat transfer using CuO- and Al_2O_3-ethylene glycol and water inside a flat tube of a car radiator was performed by Vajjha et al. [10]. A CFD model of the mass flow rate of air passing across the tubes of a car radiator was presented by Peyghambarzadeh et al [11]. The air flow was simulated using the commercial software ANSYS 12.1, where the geometry was created in the software SOLID WORKS, followed by creating both the surface mesh and the volume mesh accordingly. The results were compared and verified according to the known physical situation and existing experimental data. The results serve as a good database for future investigations. New correlations for the viscosity and thermal conductivity of nanofluids as a function of volumetric particle concentration and temperature developed from the experiments were used in this paper. The convective heat transfer coefficient and shear stress of the nanofluid showed marked improvement over the base fluid, showing higher magnitudes in the flat regions of the tube. The results showed that increasing the nanofluid volume fraction increased the friction factor and convective heat transfer coefficient; however, there was also an increase in pressure loss as the particle volume fraction increased. A numerical study that analysed mixed convection flows in a U-shaped grooved tube "in a radiator" was conducted by Park and Pak [12]. A modified SIMPLE algorithm for the irregular geometry was developed to determine the flow and temperature field. The results have been used as fundamental data for tube design by suggesting optimal specifications for radiator tubes. Two liquids, water and an ethylene glycol/water mixture, used as the coolant fluid in a meso-channel heat exchanger were studied numerically by Dehghandokht et al, [13]. The predicted results (heat transfer rate, pressure and temperature drops in the coolants) from the numerical simulation were compared with the experimental data for the same geometrical and operating conditions and showed good agreement. Additionally, the results showed the heat exchanger was enhanced, with heat transfer rate approximately 20% higher than that of a straight slab of the same length; the enhanced heat exchanger has a good potential to be used as a car radiator with reasonably enhanced heat transfer characteristics using an ethylene glycol/water mixture as the coolant. The application of a copper-ethylene glycol nanofluid in a car cooling system has been studied by Leong et al. [14]. The friction factor has been evaluated experimentally by [15-19]. An experimental investigation for the determination of heat transfer coefficient with SiO_2 nanofluid under cooling and heating conditions at fluid inlet temperatures of 20, 50 and 70 ^oC for Reynolds number range from 200 to 10000 and 22 nm size diameter particles suspension in water has been conducted by Ferrouillat et al, [18]. The results showed that the overall heat transfer coefficient with nanofluid was more than the base fluid. Escher et al, [20] proposed silica nanofluid to improve the performance of microchannel heat sinks. Hussein et al, [21] studied the thermal properties measurements experimentally. They concluded that the thermal conductivity of SiO_2 nanofluid increased with increasing of the volume fraction. This paper, presents an experimental study of forced convection heat transfer into a car radiator with SiO_2 nanoparticles suspension to water under laminar flow. The test rig is setup to measure temperatures and pressure drop between inlet and outlet of the radiator. The range of Reynolds number is 250 to 2000 and the volume fraction of nanofluid is from 1% to 4%. The aim of this study is to enhance heat transfer in the automotive cooling system. The experimental results are validated with other researchers' data.

II. EXPERIMENTAL WORK

A. Experimental setup and procedure

The test rig shown in Fig. 1 has been used to measure heat transfer coefficient and friction factor in the automotive engine radiator.

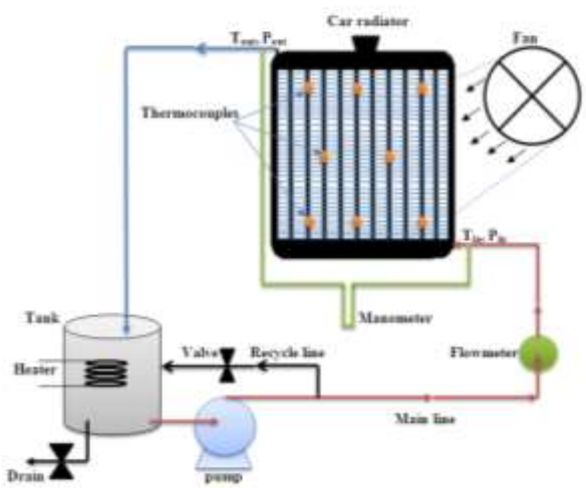


Fig. 1. Schematic diagram of test rig

This experimental setup includes a reservoir plastic tank, electrical heater, a centrifugal pump, a flow meter, tubes, valves, a fan, a D.C power supply; ten thermocouples type T for temperature measurement, manometer tube with mercury and heat exchanger (automobile radiator). An electrical heater (1500W) inside a plastic storage tank (40 cm height and 30 cm diameter) put to represent the engine and to heat the fluid. A voltage regular (0-220 V) provided the power to keep the inlet temperature to the radiator from (60-80 ^oC). A flow meter (0-70 LPM) and two valves used to measure and control the flowrate. The fluid flows through plastic tubes (0.5inch) by a centrifugal pump (0.5 hp and 3 m head) from the tank to the radiator at the flowrate range (1-5) LPM. The total volume of the circulating fluid is (3 liters) and constant in all the experimental steps. Two thermocouples (copper – constantan) types T have been fixed on the flow line for recording the inlet and outlet fluid temperatures. Eight thermocouples type T have been fixed to the radiator surface to ensure more of surface area measurement. Two thermocouples type T also fixed in front of the fan and another side of radiator to measure air temperatures. Very small thickness and high thermal conductivity of the copper flat tubes caused to make the inside temperature of the tube with the outside one are equated. A handheld (-40^oC to 1000^oC) digital thermometer with the accuracy of ± (0.1^oC) used to read all the temperatures from thermocouples. The calibration of thermocouples and thermometers carried out by using a constant temperature water bath and their accuracy estimated to be 0.15 ^oC [19]. Two small plastic tubes (0.25 inch) diameter connected at inlet and outlet radiator and joined to U-tube mercury manometer with the accurate scale 0.5 mm Hg to measure the pressure drop on the inlet and outlet of it. The car radiator has louvered fin and 32 flat vertical copper tubes with flat cross sectional area. The distances among the tube rows filled with thin perpendicular copper fins. For the air side, an axial force fan (1500 rpm) installed close on axis line of the radiator. The D.C power supply (type Teletron 10-12V) used to turn the axial fan instead of a car battery. The thermal properties of nanofluids have been measured by [22] and used in this study. The pH values of nanofluid have been measured using OAKTON device shown in for the nanofluid volume concentration of (1-4%). The pH values before and after experimental tests referring to nanofluids stability and changing in thermal properties. If the pH values of the suspension decrease, the force among particles will increase which the moving of the nanoparticles suspension lead to enhance the heat transfer process. So, to augment heat transfer for many applications should keep low values of the nanofluid pH [15]. The pH meter has been calibrated using a single point calibration technique, with a Hatch standard buffer solution of pH 7.00 ± 0.02. According to [22] to find out the pH values, samples with pH of 8.2, 8.6, 9.3 and 10.2 were prepared. Samples checked after finishing each test but found no visible sedimentation. SiO_2 nanoparticles with volume concentration (1, 2, 3 and 4%) and 30 nm size diameter added to base fluids (deionized water). The thermal properties of SiO_2 nanoparticle and base fluids showed in Table 1.

B. Experimental data analysis.

According to Newton's cooling law the following procedure followed to obtain heat transfer coefficient and corresponding Nusselt number as:

$$Q_c = hA\Delta T = hA_s(T_b - T_s) \tag{1}$$

As: is surface area of tube, T_b: is the bulk temperature

$$T_b = \frac{T_{in} + T_{out}}{2} \tag{2}$$

(T_{in}, T_{out}) are inlet and outlet temperatures and T_s is the tube wall temperature which is the mean value by two surface thermocouples as:

$$T_s = \frac{1}{a}\sum_{i=1}^{a} T_i \tag{3}$$

And heat transfer rate calculated by:

$$Q_c = \dot{m}C\Delta T = \dot{m}C(T_{in} - T_{out}) \tag{4}$$

$\dot{m}$: is mass flowrate which determined as:

$$\dot{m} = \rho \times \dot{V} \tag{5}$$

The heat transfer coefficient can be evaluated by collecting Eq. (1) and (4):

$$h = \frac{\dot{m}C(T_{in} - T_{out})}{nA_s(T_b - T_s)} \tag{6}$$

And Nusselt number can be calculated as:

$$Nu = \frac{h_{exp} \times D_h}{k} \tag{7}$$

D_h: is the hydraulic diameter of the tube which estimated by description the problem undertaken as cylindrical geometry coordinates. The dimensions of the flat tube are major and minor diameter (D=9mm, d=3mm), the length (L) and hydraulic diameter (D_h) of the flat tube are 345 mm and 4.68 mm as shown in Fig. 2.

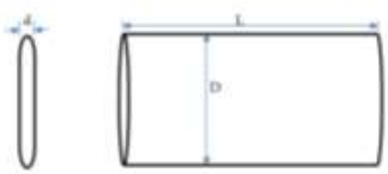


Fig. 2. Flat tube configuration

TABLE I: NANOPARTICLE THERMAL PROPERTY

Materials	ρ (Kg/m^3)	C (J/Kg. °C)	K (W/m. °C)	μ (Pa. s)
SiO_2	2220	745	1.4	-
Pure water	998	4180	0.6067	0.0014

Reynolds number calculated regarded hydraulic diameter (D_h) as:

$$D_h = \frac{4 \times Area}{Perimeter}$$

$$D_h = \frac{4 \times \left[\frac{\pi}{4}d^2 + (D-d) \times d\right]}{\pi \times d + 2 \times (D-d)} \tag{8}$$

The pressure drop can be calculated as:

$$\Delta P = S \times g \times H \tag{9}$$

The friction factor can be evaluated as:

$$f = \frac{2 \times \Delta P}{\frac{L}{D} \times \rho \times u^2} \tag{10}$$

Reynolds number (Re) is determined as:

$$Re = \frac{4\dot{m}}{\pi D_h \mu} \tag{11}$$

C. Uncertainty analysis

The uncertainty analysis performed by calculating the measurements error. The Reynolds number uncertainty range may be come from the errors in the measurement of volume flowrate Eq (5) and hydraulic diameter of the tubes Eq (8). The uncertainty of Nusselt number refers to the errors in the measurements of volume flowrate Eq (5), hydraulic diameter Eq (8), and all the temperatures Eq (2-3). According to uncertainty analysis described by [23], the measurement errors of the main parameters are summarized in Table (2).

TABLE II: UNCERTAINTIES OF MEASURED PARAMETERS

No.	Parameter	Values	Uncertainties
1	D_h	4.68 mm	0.58%
2	Re	250 - 2000	2.64%
3	Nu	8 - 22	0.35%

Furthermore, to check the reproducibility of the experiments, some runs were repeated later which proved to be excellent. The results of experimental are compared with the Darcy Eq. (12) for friction factor and Shah-London Eq. (13) for Nusselt number as:

$$f = \frac{64}{Re} \tag{12}$$

$$\begin{aligned} Nu_{av} &= 1.953\left(Re_{D_h} Pr \frac{D_h}{L}\right)^{\frac{1}{3}} \\ &\text{For } \left(Re_{D_h} Pr \frac{D_h}{L}\right) >= 33.33 \\ Nu_{av} &= 4.364 + 0.0722\left(Re_{D_h} Pr \frac{D_h}{L}\right) \\ &\text{For } \left(Re_{D_h} Pr \frac{D_h}{L}\right) < 33.33 \end{aligned} \tag{13}$$

III. RESULTS AND DISCUSSION

The friction factor at different velocity and nanofluid volume concentration is showed in Fig. 3. It appears that the friction factor decreases with increasing in Reynolds number and nanofluid volume concentration. Fig. 4 shows the friction factor at different velocity and nanofluid volume fraction. It can be seen that the friction factor decreases with the increase of the velocity but increases with the increase of the nanofluid volume fraction. It can be noted that the effect of the nanofluid volume fraction is significant on the friction factor. Fig. 5 illustrates the friction factor at different velocity and the inlet temperature. It seems that the friction factor decreases with the increase of the velocity and decreases slightly with the increase of the inlet temperature. It can be seen that the effect of the inlet temperature is insignificant on the friction factor.

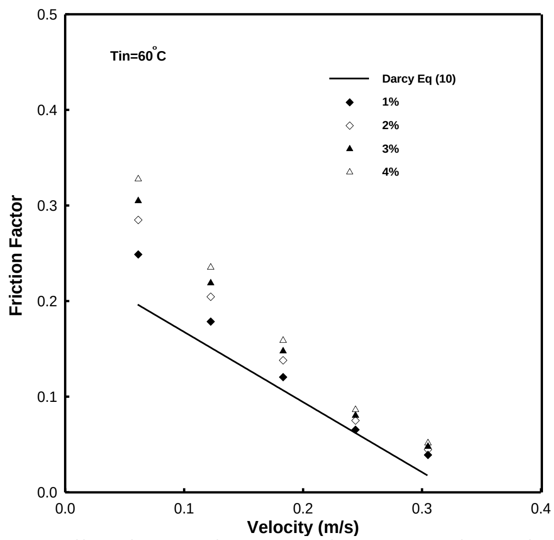


Fig. 3. The effect of the nanofluid volume fraction on the friction factor at different velocity.

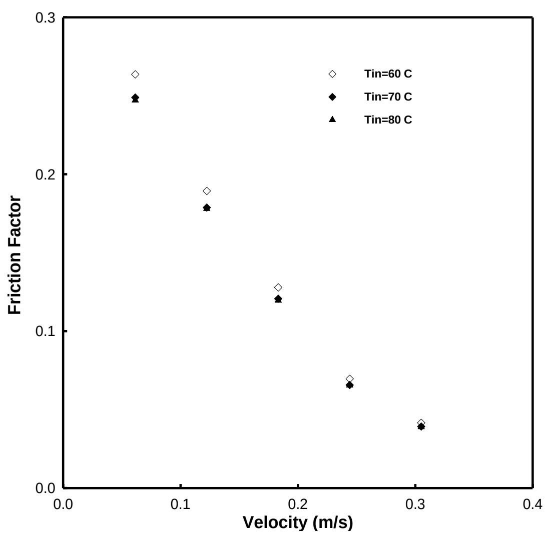


Fig. 4. The effect of the inlet temperature on the friction factor at different velocity.

Fig. 5 shows the heat transfer coefficient at different velocity and nanofluid volume concentration. The heat transfer coefficient increases with the increase of the velocity and the nanofluid volume concentration. The deviation is approximately 40% when adding the nanoparticles on base fluid (pure water). Likewise, Fig. 6 demonstrates the effect of the inlet radiator temperature on the heat transfer coefficient. The highest values of the heat transfer coefficient found at inlet temperature 80 ^oC followed by at 70 ^oC and finally at 60 ^oC. The maximum values of the heat transfer coefficient are (5900, 5400 and 4900W/m^2.^oC) at (60, 70 and 80^oC) respectively. This refers to high heat transfer from the radiator when high inlet temperature would apply.

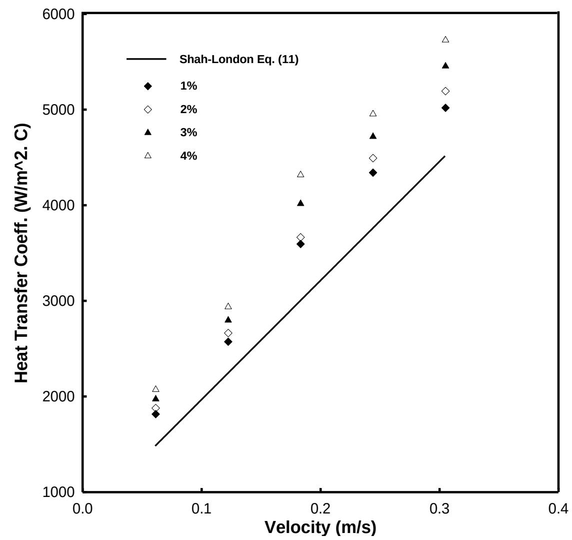


Fig. 5. The effect of nanofluid volume fraction on the heat transfer.

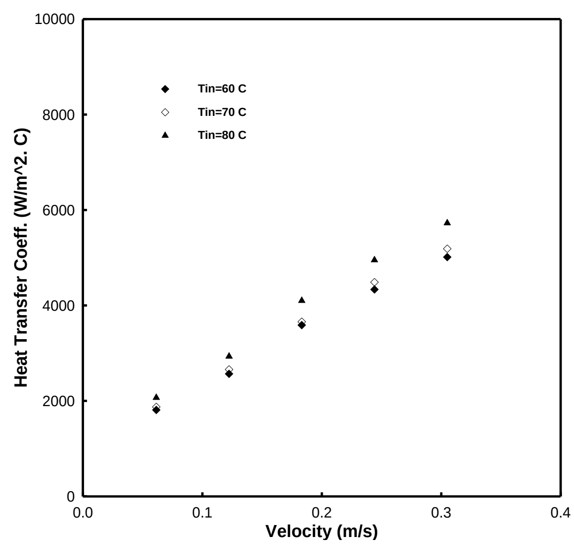


Fig. 6. The effect of the inlet temperature on the heat transfer.

Fig. 7 shows the outlet temperature of the radiator (T_{out}) with different volume flow rate circulating and nanofluid volume concentrations. It seems that the outlet temperature increases with increasing in the volume flow rate and decreasing in nanofluid volume concentration. Fig. 7 indicates the percentage of the heat transfer enhancement depending on the nanofluid volume concentration and the inlet temperature to the radiator. The percentage enhancement can be evaluated by:

$$E\% = \frac{h_{nf} - h_f}{h_{nf}} \times 100 \qquad (12)$$

It can be seen that the heat transfer enhancement increases with increasing of nanofluid volume concentration and inlet temperature respectively. The values of heat transfer enhancement are from 31 to 46% for nanofluid volume concentration from 1% to 4% while the values of heat transfer enhancement are from 26 to 35% for inlet temperature from 60 to 80^{o}C. This refers to nanofluid volume concentration affected more than inlet temperature on heat transfer enhancement, but significantly of using all of them. Fig. 8 demonstrates that the experimental data agreed with other investigator data to be validated. Fig. 8a showed the friction factor results from the experimental work and experimental data of [16-17]. The present work agreed with data of [16-17] with deviation, not more than 9% due to the different of the size diameter of the nanoparticles. On the other hand, Fig.8b shows the validation of Nusselt number from experimental work of [17-18].

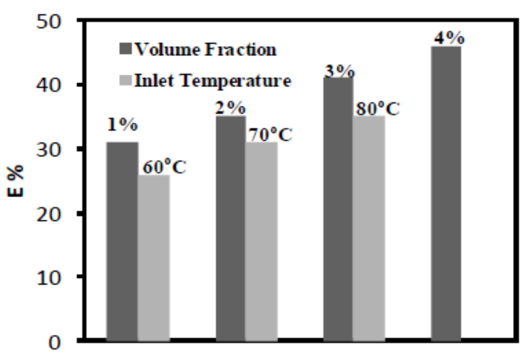
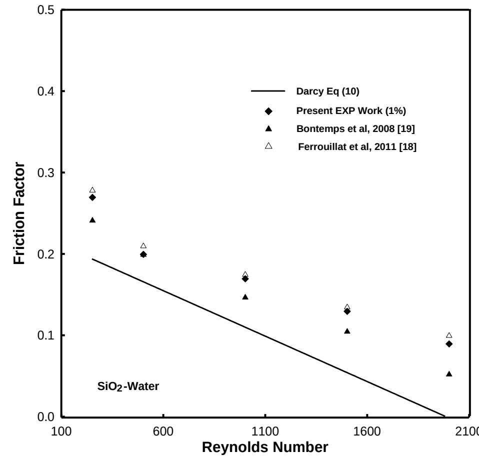
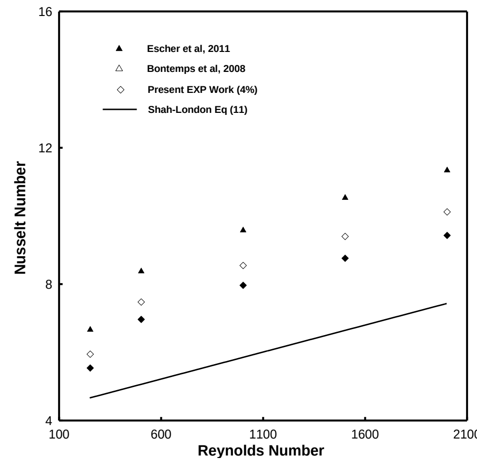


Fig. 7. The effect of the nanofluid volume fraction and the inlet temperature on the heat transfer enhancement.

It seems that there is a good agreement with the maximum deviation 7% may be related to size diameter or the preparation method of the type of nanofluids.



a. Friction factor



b. Nusselt number

Fig. 8. Validation of: a. friction factor and b. Nusselt number

IV. CONCLUSIONS

The experimental study of friction factor and forced convection heat transfer enhancement of SiO_2 suspended to water has been conducted. Significant increases of the friction factor and heat transfer enhancement observed with the volume fraction of nanoparticles addition. The maximum values of friction factor increased to 22% for SiO_2 nanoparticles dispersed to water with 4% volume concentration. Highest values of the heat transfer coefficient enhance up to 40% obtained for SiO_2 nanoparticles in water. The experimental results showed that the friction factor and Nusselt number action of the nanofluids were highly depended on the volume concentration and Reynolds number. The friction factor decreases with the increasing of volume flowrate and the inlet temperature. The friction factor at Reynolds number less than 1000 has a maximum deviation 82% and after that the maximum deviation becomes 40%. A highest Nusselt number enhancement is 45% obtained for SiO_2

nanoparticles in water as a base fluid. The experimental results showed that the Nusselt number behavior of the nanofluids was highly depended on the volume concentration and the volume flowrate. The Nusselt number of 1% SiO_2 nanofluid at 80 ^oC has 36% deviation than pure water but 22% at 60 ^oC. There is a good agreement among present data of friction factor with data of [16-17] and Nusselt number data with data of [17-18]. The results proved that SiO_2 nanofluid have high potential for hydrodynamic flow and heat transfer enhancement and are highly appropriate to industrial and practical applications. When adding nanoparticles to the base fluid the possible enhancement of car engine cooling rates will improve otherwise the engine heat remove or reduce size cooling system. The small cooling systems have led to benefit almost every aspect of car performance and increased in fuel economy.

NOMENCLATURES

C - Specific heat [W/ kg.oC]

D - Diameter [m]

E - Enhancement

f - Friction factor

h - heat transfer coefficient [$W/m^2.^oC$]

k - Thermal conductivity [$W/m.^oC$]

Nu - Nusselt Number

P - Pressure [N/m^2]

Pr - Prandtle Number [C.μ/Knf]

Re - Renolds Number [ρ_{nf} D_h u / K_{nf}]

u - Velocity [m/s]

μ -Viscosity [N.s /m2]

ρ - Density [kg/m3]

τ - Shear stress [N/m2]

ϕ - Volume concentration

Subscripts

f	liquid phases
p	solid particle
nf	nanofluid
h	hydraulic

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