



Heat Transfer and Turbulent Nanofluid Flow over a Double Forward-Facing Step

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Abstract: Heat transfer and turbulent nanofluids flow over a double forward-facing step were investigated numerically. The finite volume method was used to solve the continuity, momentum, and energy equations using the $k-\epsilon$ model. Four cases, corresponding to different nanofluids at constant step height, were investigated for Reynolds numbers ranging from 30,000 to 80,000. The bottom of the wall was heated at constant temperature 333K. Whereas the top wall was insulated. The results show that the surface Nusselt number increased with the Reynolds number. The maximum Nusselt number was observed for water/ ZnO nanofluids, with a Reynolds number of 50,000. The behavior of the Nusselt number was similar for all cases at a given Reynolds number and temperature. The results indicate also, that the pressure drop increased with increasing Reynolds number for all cases and the pressure drop in water/ CuO was higher than other at Re 80000. FLUENT 6.3.26 software was employed to run this simulation.

Keywords: Nanofluids; Heat transfer; Double forward-facing step; Nusselt number.

I. Introduction

The purpose of this study is to investigate two-dimensional double forward-facing step flows, and the results of numerical computations for different nanofluids types, and Reynolds numbers at constant bottom wall temperature are presented herein. Numerous studies have been performed on single forward- and backward facing steps; however, the literature on double forward and backward-facing steps is very limited, and the physical basis of flow separation and vortex creation remains unclear. Fluid flow over a backward- or forward-facing step generates recirculation zones and subsequent reattachment regions, due to sudden contraction or expansion in flow passages. Many practical engineering applications, such as the cooling of electronic devices, open channels, power generating equipment, heat exchangers, combustion chambers, and building aerodynamics, involve separating flows [1]. The first attempts to study heat transfer and fluid flow over forward- or backward-facing steps were made in the 1950's. Later, researchers were able to analyze complex flows in three dimensions due to the development of CFD software. Seban et al. [2] and Seban [3] pioneered the study of fluid flow over backward- and forward-facing steps from a heat transfer perspective. The authors discovered that the maximum heat transfer coefficients occur at the reattachment point and decrease toward the outlet. The effect of stream turbulence on the heat transfer rate in the reattachment region on the bottom surface of a backward-facing step was demonstrated by Mabuchi et al. [4]. Improvements in device capabilities have allowed researchers to measure reattachment points and heat transfer characteristics; Mori et al. [5] used a thermal

Tuft probe, Kawamura et al. [6, 7] obtained the temporal and spatial parameters of heat transfer in the reattachment region using a new heat flux probe, and Oyakawa et al. [8, 9] employed jet discharge. The hydrodynamic characteristics of gas flows past a rib and a downward step in feature separation flow regions were studied by Terekhov et al. [10]. An early study on turbulent heat transfer and airflow over a double forward-facing step was reported by Yilmaz and Oztop [11]. The top of the wall and steps were insulated, and the bottom of the wall was heated. The authors used $k-\epsilon$ model and found that the step ratio affected the heat transfer and flow more strongly than the length ratio. Later, Oztop et al. [12] presented a numerical study of heat transfer and turbulent airflow over a double forward-facing step with an obstacle. The bottom of the wall and steps were heated, and the top of the wall was insulated. The results showed that the obstacle aspect ratio (Ar) affected the heat transfer, with the maximum Nusselt number corresponding to $Ar = 1$.

More recently, the majority of studies have been utilizing nanofluid because of its higher thermal conductivity compared to normal fluid [13]. Abu Nada [14] is a pioneer in research on laminar nanofluid flow over a backward-facing step with Cu, Ag, Al₂O₃, CuO, and TiO₂ nanofluid, volume fractions between 0.05 and 0.2 and Reynolds numbers ranging from 200 to 600. An investigation of findings signifies that the Nusselt number increased with the volume fraction and Reynolds number. Later, Kherbeet et al. [15] presented a numerical investigation of heat transfer and laminar nanofluid flow over a micro-scale backward-facing step. The Reynolds numbers ranged from 0.01 to 0.5, nanoparticle types comprised Al₂O₃, CuO, SiO₂, and ZnO, and the expansion ratio was 2. An increasing Reynolds number and volume fraction seemed to lead to an increasing Nusselt number; the highest Nusselt number value was obtained with SiO₂.

The objective of this paper is to contribute new data regarding water and nanofluids flow over double forward-facing steps to improve the design of heat exchangers.

II. Model description

A. Physical Model

A schematic diagram of the double forward-facing step and the flow shape employed in this study is presented in Figure 1. The bottoms of the wall and the steps were heated to a given temperature (T_h), while the top of the wall was adiabatic. The first and second step heights (h_1 , h_2) was fixed at 20mm. The entrance width (H) was 100 mm. The Reynolds number was varied from 30,000 to 80,000 and calculated based on entrance width (H), and The temperature of heated wall was 313.

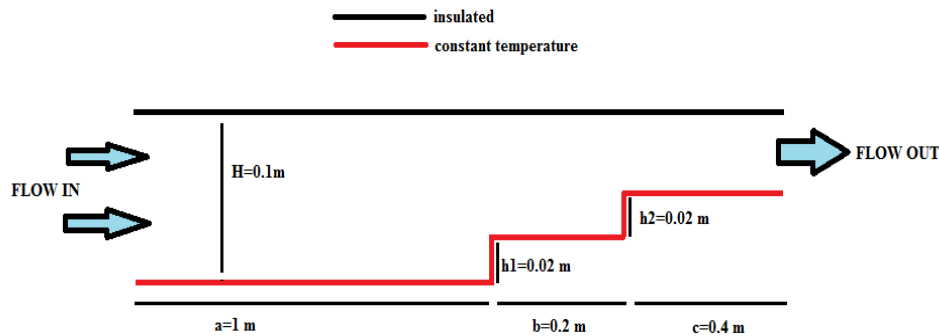


Figure 1: Schematic diagram of physical model.

B. Governing equation

The two-dimensional instantaneous governing equation of mass, momentum and energy equations for study incompressible in fully developed flow can be written in conservation form expressed in Cartesian coordinates as follows [19]:-

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \overline{\rho u'_i u'_j} \right] \quad (2)$$

$$\frac{\partial \rho u_i T}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\frac{k}{c_p} \frac{\partial T}{\partial x_j} \right) \quad (3)$$

The Reynolds stress tensor $-\overline{\rho u'_i u'_j}$ can be determined according to the Boussinesq assumption as

$$-\overline{\rho u'_i u'_j} = \mu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \delta_{ij} \rho k \quad (4)$$

-Where μ_t is the turbulent eddy viscosity and is estimated by the (k- ϵ) two equations turbulent model.

$$\mu_t = \mu \rho k^2 / \epsilon \quad (5)$$

The differential equation of k and ε are given as

$$\frac{\partial(\rho u_j k)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k - \rho \epsilon \quad (6)$$

$$\frac{\partial(\rho u_j \epsilon)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon} \right) \frac{\partial \epsilon}{\partial x_j} \right] + C_{\epsilon 1} G_\epsilon \frac{\epsilon}{k} - C_{\epsilon 2} \rho \frac{\epsilon^2}{k} \quad (7)$$

-Where $G_k = -\rho u'_i u'_j \left(\frac{\partial u_j}{\partial x_i} \right)$ is the turbulent production term.

The remaining coefficients that appeared in the above equation are as quoted by [19] : $C_\mu=0.09$, $C_{\epsilon 1}=1.44$, $C_{\epsilon 2}=1.92$, $\sigma_k=1$ and $\sigma_\epsilon=1.3$

The Reynolds number is computed based on inlet channel height (H).

$$Re = \frac{\rho u H}{\mu} \quad (8)$$

$$Nu_{av} = \frac{1}{L} \int_0^L Nu_L dL \quad (9)$$

Where (L) is the length of the heated wall.

C. Numerical Procedure and grid dependence

Simulations were carried out using FLUENT 6.3.26 The gambit 2.3.16 was used for meshing, and the k-ε standard model in Fluent was used to analyze the water and nanofluids flow and heat transfer over the double forward-facing step in the turbulent region. Grid independence was verified by increasing the grid size step wise, which yielded similar results. Independent verification was performed for Re=30,000 and water used as working fluid the initial grid sizes of (6522, 13765, 22647, 34780, 89228, 129438 and 178972). The difference in the Nusselt number relative to that of the selected grids was less than 0.5% and grid number 6 was adopted as shown Figure 2.

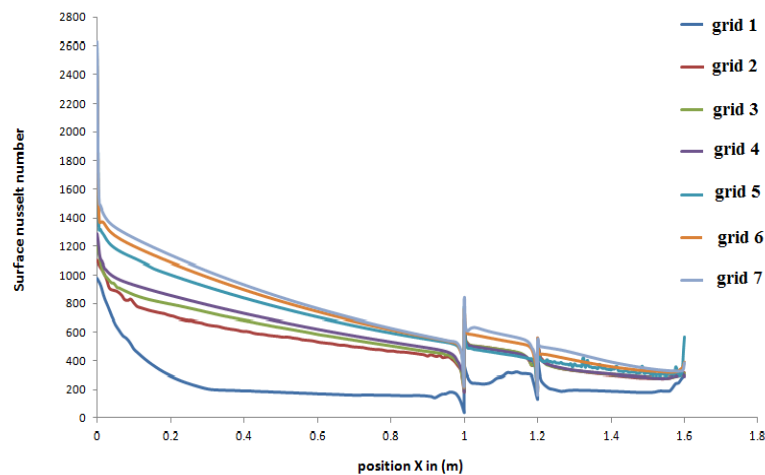


Figure 2: Grid dependence test

III. Thermo-physical properties of nanofluids

Thermophysical properties of the base fluid and the nanoparticles used in this study shown in table.1. The effective properties of nanofluids are defined as follow:

Density:

$$\rho_{nf} = \phi \rho_p + (1 - \phi) \rho_{bf} \quad (10)$$

Heat capacity:

$$c_{p_{nf}} = \frac{\varphi \rho_p c_{p_p} + (1-\varphi) \rho_{bf} c_{p_{bf}}}{\rho_{nf}} \quad \text{.....(11)}$$

The Eqs. (6) and (7) were introduced by [15].

The thermal conductivity:

$$\frac{k_{nf}}{k_{bf}} = \frac{k_p + (n-1)k_{bf} - (n-1)\varphi(k_{bf} - k_p)}{k_p + (n-1)k_{bf} + \varphi(k_{bf} - k_p)} \quad (12)$$

This was introduced by [16].

Where (n) is a shape factor and equal to (3) for spherical nanoparticles.

The viscosity:

The effective viscosity can be obtained by using the following mean empirical correlation [17].

$$\mu_{nf} = \mu_{bf} \frac{1}{(1-34.87(d_p/d_{bf})^{-0.3} \varphi^{1.03})} \quad (13)$$

Where: $d_{bf} = \left[\frac{6M}{N\pi\rho_{bf}} \right]^{1/3}$
..... (14)

Where: M is the molecular weight of base fluid, N is the Avogadro number = $6.022 \times 10^{23} \text{ mol}^{-1}$, ρ_{bf} is the mass density of the base fluid calculated at temperature $T_0=300 \text{ K}$. the table 1. Show the thermo-physical properties of nanoparticles and working fluids.

IV. Results and Discussion

4.1 Effect of the Reynolds number

The effects of the Reynolds number on the local Nusselt number for the turbulent ranges are presented in Figs.3 (A, B and C). With the increase of Reynolds number, the Nusselt number increased in the turbulent ranges for all nanofluids. Results indicate that since higher Reynolds number leads to higher velocity and temperature gradients at heated wall, consequently the local nusselt number is increased by increasing Reynolds number.

4.2 Effect of different types of nanoparticles

Different types of nanoparticles such as Al₂O₃, CuO and ZnO and pure water as a base fluid are used. The Nusselt number for different nanofluids and different values of Reynolds number are shown in Fig.3 (A, B, C and D). It can be clearly seen that ZnO nanofluid has the highest average Nusselt number, followed by CuO, Al₂O₃ respectively. This is because ZnO has the lowest thermal conductivity than other nanofluids, but higher than water also, the heat capacity for ZnO is small compared with other nanofluids that means the heat transfer in particles move quickly and that let to get high temperature in it.

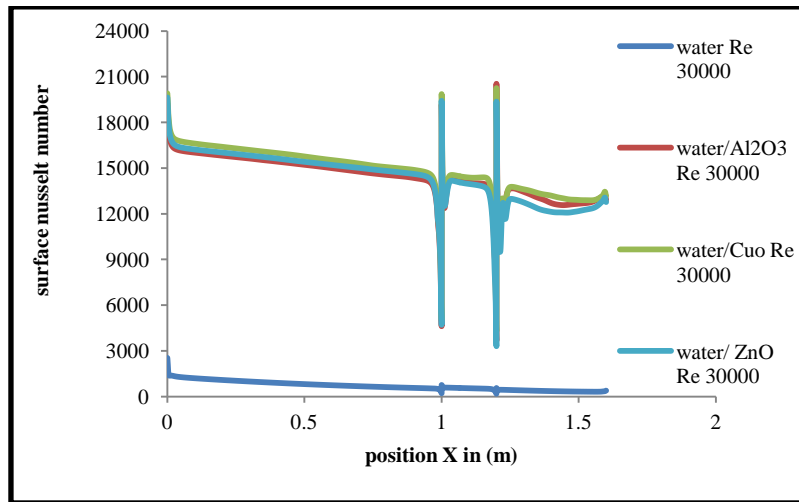
4.3 Pressure drop

The pressure drop variation with axial distance for water/ Al₂O₃ at different Reynolds numbers and different nanofluids at Re 80000 is presented in Fig.4 and Fig.5. According to the results, the pressure drop intensified as the Reynolds number increased for water flow and also, the water/ CuO nanofluids has higher drop than other nanofluids at constant volume fraction. Generally, the highest pressure drop occurred at the downstream inlet region due to recirculation flow which caused the improvement of heat transfer.

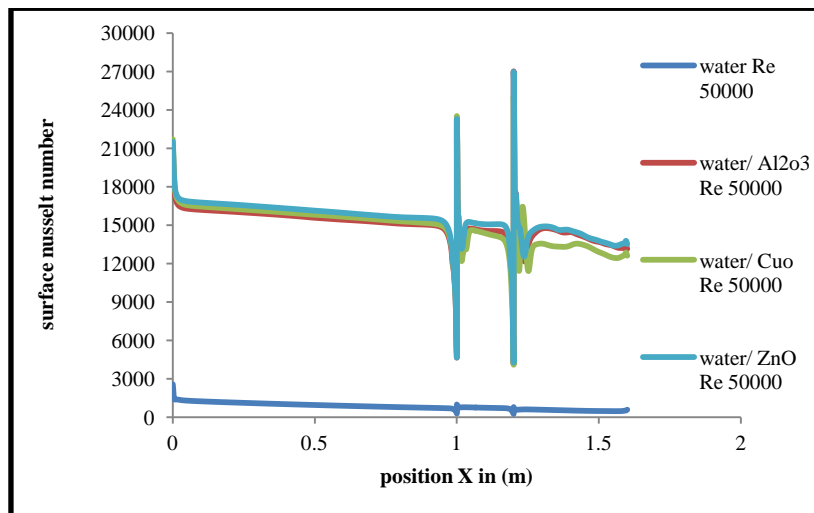
V. Conclusion

Turbulent nanofluids forced convection and heat transfer over a double forward-facing step was studied. Four cases, corresponding to three different nanofluids and base fluid, were investigated at different Reynolds numbers

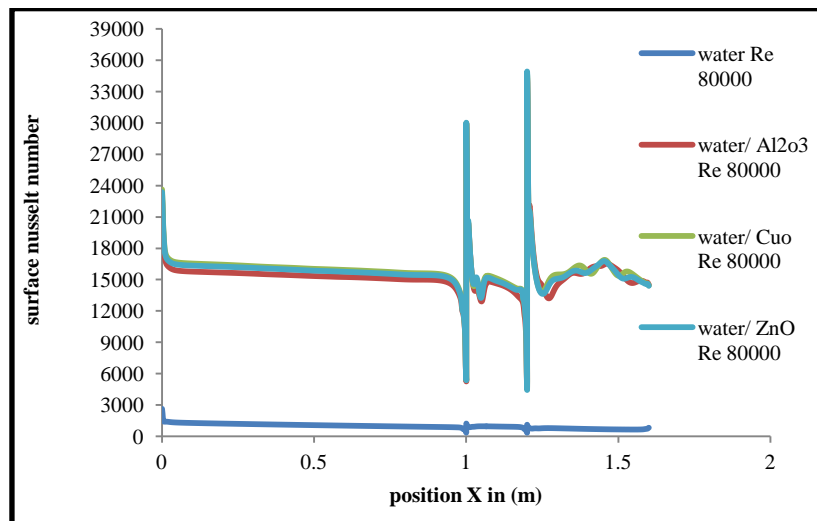
and constant temperature. Recirculation zone increased the separation length at the same Reynolds number and temperature, which increases the Nusselt number. The results show that increasing the Reynolds number and temperature increased the Nusselt number for all cases. The enhancement of the Nusselt number occurred at the water/ ZnO nanofluid at 50000. Reynolds number compared with other nanofluids. In addition, the obtained results indicate an increase in the pressure drop with increasing Reynolds number for all cases.



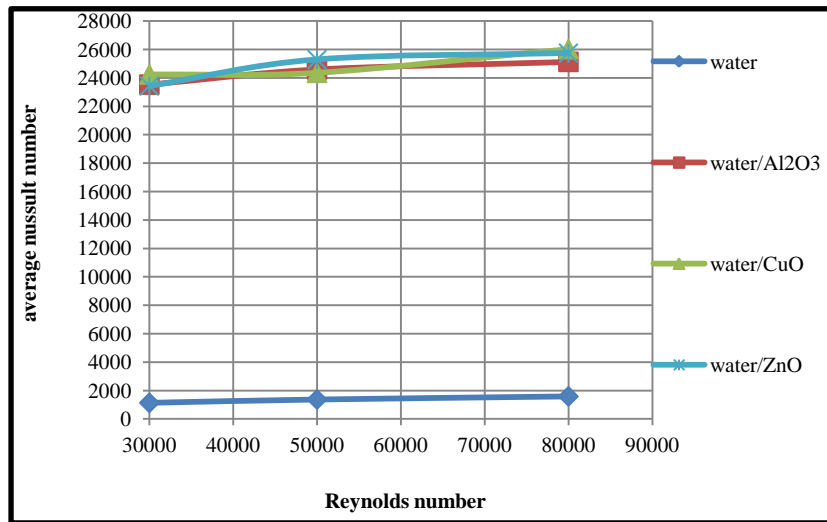
(A)



(B)



(C)



(D)
 Figure 3. (A) surface nusselt number at Re 30000, (B) surface nusselt number at Re 50000, (C) surface nusselt number at Re 80000 and (D) Average nusselt number at different Reynolds number.

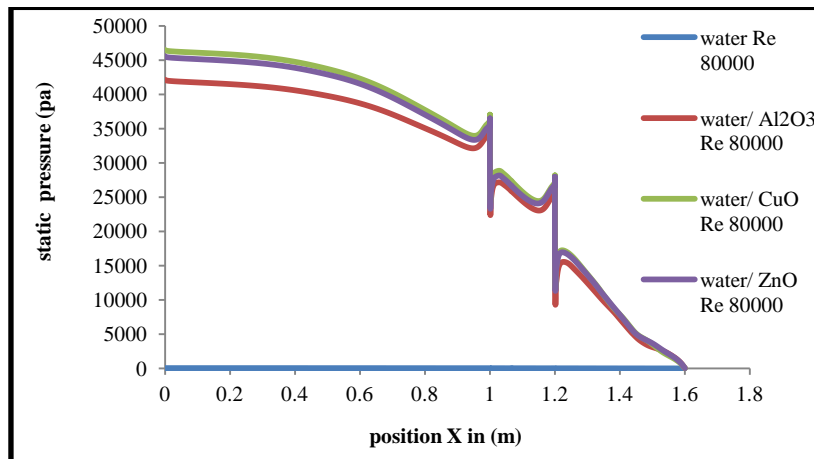


Figure 4: Pressure drop with different Nanofluids at Re 80000.

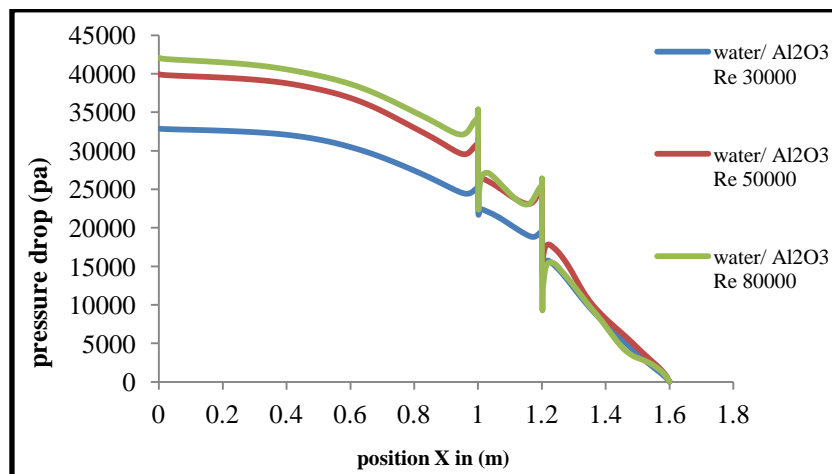


Figure 5: Pressure drop with different Reynolds number for (Al2O3-water)

Table 1. Thermophysical properties of the base fluid and the nanoparticles used in this study [16, 17, 18]

property	water	Al ₂ O ₃	CuO	ZnO
ρ (kg.m ⁻³)	998.2	3970	6500	5600
C_p (J/kg.k)	4182	765	535.6	495.2
K (w/m.k)	0.6	40	20	13
μ (N.s/m ²)	0.001003	-	-	-

VI. Nomenclature

C_p	specific heat capacity (J kg ⁻¹ K ⁻¹)
Nu_L	surface Nusselt number
Nu_{av}	average Nusselt number
P	Pressure (Pa)
Pr	Prandtl number
Re	Reynolds number
T_w	Heated Wall temperature (K)
T	Temperature (K)
u	velocity component (m s ⁻¹)
x, y	spatial coordination (m)
L	Length of the heated downstream wall (m)
h_1	First Step height (m)
h_2	Second Step height (m)
H	Height of inlet channel (m)
a	Length of bottom wall before the first step (m)
b	Length of bottom wall after the first step (m)
c	Length of bottom wall after the second step (m)
M	molecular weight of basefluid
d_p	Particles diameter (nm)
d_{bf}	Basefluid diameter (nm)
N	Avogadro number

Greek symbols

K	thermal conductivity (W m ⁻¹ K ⁻¹)
μ	dynamic viscosity (Pa s)
ρ	density (kg m ⁻³)
ρ_{bf}	Density of basefluid (kg m ⁻³)
φ	Volume friction (%)

Subscripts

nf	nanofluid
p	Nano particles
bf	basefluid

VII. References

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