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Effects of SiO_2/Water Nanofluid Flow in a Square Cross-Sectioned Curved Duct

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Abstract

Forced convection SiO2/water nanofluid flow and heat transfer was numerically performed in 180-degree three-dimensional curved duct with square cross section under steady and laminar flow conditions in this investigation. Dean number was changed from 102 to 898. All surface of curved duct was exposed to uniform and constant heat flux of $q'' = 15000$ W/m^2. Nanoparticle volume fractions was ranged 1.0%-4.0%. The average Nusselt number and average Darcy friction factor were determined for each nanoparticle volume fractions. Velocity and temperature profiles and secondary flows were analyzed in detail. In addition, numerical study results are expressed with engineering correlations as changing average Nusselt number and average Darcy friction factor with Dean number and nanoparticle volume fraction.

Key words

Laminar flow, Dean number, SiO_2/water nanofluid, square cross-sectioned curved duct, forced convection,

1. INTRODUCTION

Conventional heat transfer fluids such as water, oil and ethylene glycol have a low thermal conductivity. So, using conventional fluids limit the performance and compactness for example at heat exchangers and electronic equipments. The new heat transfer fluids (nanofluids) having high thermal conductivity are capable to eliminate this effect.

Nanofluids are a new kind of heat transfer fluids containing a small quantity of nano-sized particles (generally less than 100 nm) that are uniformly and stably suspended in a liquid. The dispersion of a small number of solid nanoparticles in conventional fluids remarkably changes their thermal conductivity. Compared to the existing techniques for enhancing heat transfer, the nanofluids show a superior potential for increasing heat transfer rates in a variety of cases. In fact, Maxwell (1873) firstly suggested adding solid particles for increasing of thermal properties [1]. But S. Choi firstly studied about nanofluids at 1995 [2].

Lee et al. [3] have proved that oxide ceramic nanofluids consisting of CuO or Al_2O_3 nanoparticles in water or ethylene glycol exhibit enhanced thermal conductivity. A maximum increase in thermal conductivity of approximately 20% was observed in that study, having 4.0% vol. CuO nanoparticles with mean diameter 35 nm dispersed in ethylene glycol. But larger particles with an average diameter of 40 nm led to an increase of less than 10%.

As nanofluids are rather new, relatively few theoretical and experimental studies have been reported on convective heat transfer coefficients in confined flows. Park and Cho [4] and Xuan and Li [5] obtained experimental results on convective heat transfer for laminar and turbulent flow of a nanofluid inside a tube. They produced the first empirical correlations for the Nusselt number using Cu/water, TiO_2/water and Al_2O_3/water nanofluids. The results indicate a remarkable increase in heat transfer performance over the base fluid for the same Reynolds number.

To increase of heat transfer can apply lots of geometries. When compared the curved ducts with straight ducts, curved ducts have more pressure losses. However, curved ducts were widely used in industrial concerning heat and mass transfer devices. The applications areas of curved ducts are heat exchanger, turbomachines, air condition systems, centrifugal pumps, rocket engines and gas turbines [6].

Flow and heat transfer in curved duct has more complicated system than straight duct. Therefore, analytical study is nearly impossible [7]. Accordingly, experimental and numerically studies on curved duct are the other alternative investigations. In literature reviews, there have been various experimental and numerical study about laminar flow in curved duct at uniform surface temperature and uniform heat flux boundary condition. Facao and Oliveira [7] have used rectangular cross-sectioned curved duct and obtained Nusselt number changed with Dean number. It is found that Nusselt number is higher than at straight duct. Sturgis and Mudauer [8] investigated heat transfer enhancement using of rectangular cross-sectioned curved duct. In this study, duct curvature radiuses were changed. Working fluid was air. It is found that increasing of duct curvature radius, convection heat transfer coefficient also increased. Fang Liu [9] studied effects of geometries on heat transfer enhancement of thermal fluids in curved duct.

There are a few studies about heat transfer enhancement using of nanofluids at curved ducts. Akbarinia et al. [10] carried out numerical study of laminar mixed convection of a nanofluid in horizontal curved tubes. It is found that increasing the buoyancy forces causes to reduce the skin friction and free convection has a negative effect on the heat transfer enhancement in the presence of centrifugal force. Heat transfer coefficients are determined with SiO_2 nanoparticle in base liquid water at various concentrations. Ghaffari et al. [11] investigated numerically turbulent mixed convection heat transfer flow of Al_2O_3/water nanofluid in a horizontal curved tube. They performed the study using two phase mixture model. They analysed the effects of nanoparticle volume fractions on heat transfer. It was obtained from the numerical results that the nanoparticle volume fraction improves the Nusselt number.

The objective of this paper is to study effect of SiO_2/water nanofluid on flow and heat transfer characteristics for different Dean numbers and different nanoparticle volume fractions in 180-degree square cross-sectioned curved duct. The average Nusselt number and Darcy friction factor are presented and discussed. Also, velocity and temperature distributions are presented for varying nanoparticle volume fraction and the Reynolds number.

2. PROBLEM DESCRIPTION AND GOVERNING EQUATIONS

Nanofluid flow in 180-degree square cross-sectioned 3D curved duct was modelled in this study. SiO_2/water nanofluid was used for working fluid. Particle size was 20 nm. ANSYS 15.0 was used to obtain numerical results. Geometry of curved duct is shown in Figure 1. Curvature radius was R=60 mm and length of the one edge of square duct was 10 mm. All the duct's walls were heated by 15000 W/m^2. Inlet temperature of fluid flow was 300 K. The following assumptions were adopted for this numerical study: (i) Both heat transfer and fluid flow in duct are in three-dimensional and steady-state; (ii) fluid flow is incompressible and laminar flow; (iii) the physical properties of nanofluid, such as density, specific heat, thermal conductivity and viscosity are taken as temperature independent; (iv) negligible buoyancy effect, viscous dissipation and radiation heat transfer; and (v) the base fluid and the nanoparticles are in thermal equilibrium.

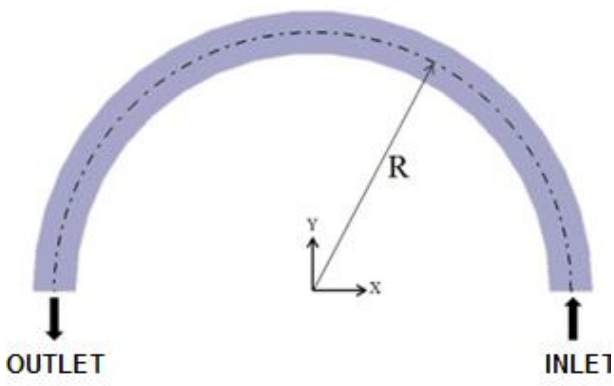

Figure 1. Schematic diagram of horizontal curved duct

Continuity, momentum and energy equations are given below for steady-state, incompressible, laminar flow condition in Eq (1) – (3), respectively.

$$\vec{\nabla}.\vec{V}=0 \tag{1}$$

$$\rho\frac{D\vec{V}}{Dt}=-\Delta p+\mu\nabla^2\vec{V} \tag{2}$$

$$\rho Cp\frac{DT}{Dt}=k\nabla^2 T \tag{3}$$

The properties of density (ρ), specific heat (C), viscosity (μ) and thermal conductivity (k) of nanofluids have been calculated as given below, respectively [12]. Also, φ represents the volume fraction of nanoparticle.

$$\rho_{nf}=(1-\varphi)\rho_w+\varphi\rho_p \tag{4}$$

$$C_{nf}=\frac{(1-\varphi)(\rho C)_w+\varphi(\rho C)_p}{(1-\varphi)\rho_w+\varphi\rho_p} \tag{5}$$

$$\mu_{nf}=\mu_w\left(1+\frac{\phi}{100}\right)^{11.3}\left(1+\frac{T_{nf}}{70}\right)^{-0.038}\left(1+\frac{d_p}{170}\right)^{-0.061} \tag{6}$$

$$k_{nf}=k_w 0.8938\left(1+\frac{\phi}{100}\right)^{1.37}\left(1+\frac{T_{nf}}{70}\right)^{0.2777}\left(1+\frac{d_p}{150}\right)^{-0.0336}\left(\frac{\alpha_p}{\alpha_w}\right)^{0.01737} \tag{7}$$

where the subscript of w, p and nf represents water, nanoparticle, and nanofluid, respectively.

The calculations of average Nusselt number (Nu), average Darcy friction factor (f) and Dean number (De) are given below:

$$Nu=\frac{hD_h}{k} \tag{8}$$

$$f=\frac{(-dp/dx)D_h}{\rho U_m^{\ 2}/2} \tag{9}$$

$$De=Re\sqrt{\frac{D_h}{R}} \tag{10}$$

The average heat transfer coefficient was calculated with Eq. (11).

$$h=\rho U_m A_c c_p\,(T_{mo}-T_{mi})/A_s(T_w-T_m) \tag{11}$$

3. NUMERICAL PROCEDURES

In the computations, the finite-volume method based commercial CFD software Ansys Fluent 15.0 was used to perform the numerical calculations by solving the governing equations along with the boundary conditions. The convection terms in mass, momentum and energy equations were discretized using a second order upwind scheme. The standard scheme was employed for discretization of pressure and the SIMPLE algorithm. The Green-Gauss cell based method was applied for discretization of the momentum and energy equations. To obtain convergence, each equation for mass, momentum, and energy were iterated until the residual falls below $1x10^{-6}$. No convergence problems were observed during the calculations. The hexahedral mesh distribution was used for the curved duct having square cross-section. The numbers of mesh points or control volumes were increased close to wall of the duct to enhance the resolution and accuracy as given in Figure 2.

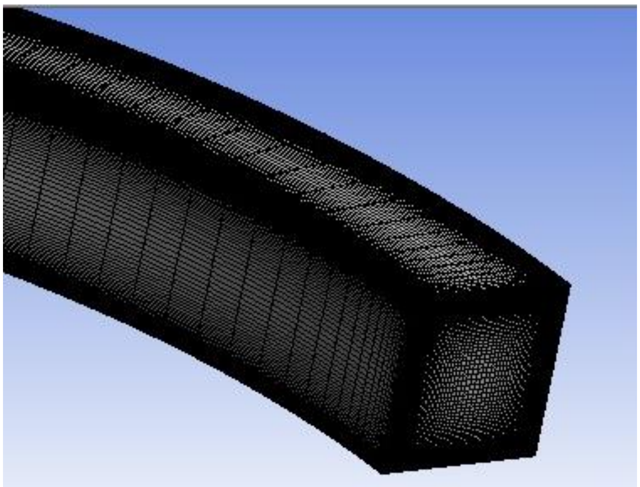

Figure 2. Mesh distribution of the curved duct

The mesh independence study was performed for the curved duct by refining the mesh number until the variation in both average Nusselt number and average Darcy friction factor are less than 0.1%. To obtain the optimum mesh number, a grid independence study was conducted using thirteen different mesh numbers changing from $5x10^2$ to 1.255 $x10^6$ for De=898. Changing of average Nusselt number and average Darcy friction factor values with mesh number for pure water flow is given in Figure 3 as an illustration. It was observed that a further refinement of mesh number from $4x10^5$ to $1.25x10^6$, the changing of average Nusselt number and average Darcy friction factor is negligible. If Fig. 3 is observed, optimum mesh number with minimum computational time and maximum accuracy approximately can be seen at $4x10^5$.

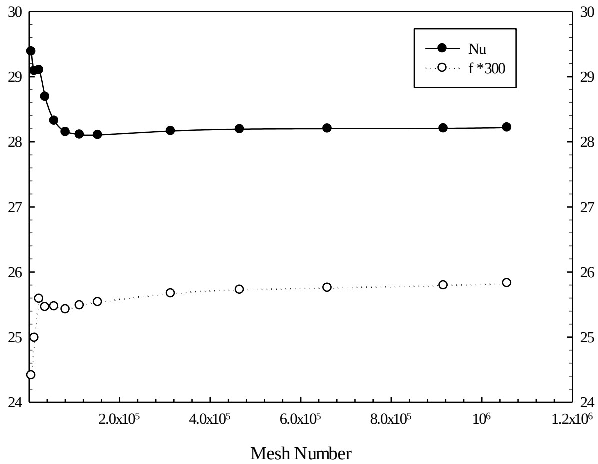

Figure 3. Changing of average Nusselt number and average Darcy friction factor with mesh number for pure water at De=898

To understand the accuracy of the numerical simulation for pure water, values of average Nusselt number were compared with experimental data studied by Dravid et al.[13] and Kalb and Sieder [14] in Figure 4. The results obtained by present study are good agreement with the the correlation proposed studies.

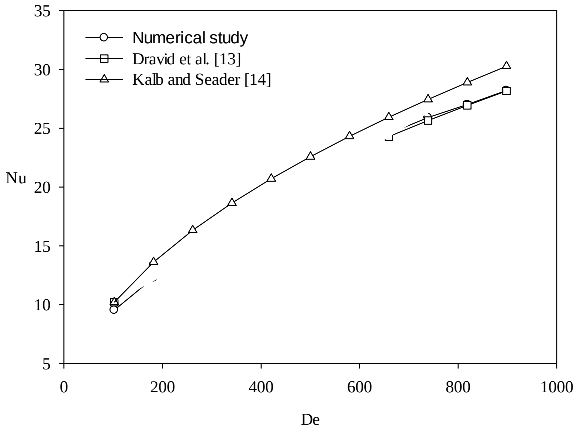

Figure 4. Comparison of average Nusselt number for pure water of the present study with experimental data given in the literature

4. RESULTS AND DISCUSSION

Laminar forced convection flow in a curved duct having square cross-sectioned is numerically investigated. Figure 5 is presented to understand the effect of nanoparticles volume fractions on average Nusselt number. As seen Figure 5, average Nusselt number increases with increasing nanoparticle volume fractions. Average Nusselt number also increases with increasing Dean number. The highest average Nusselt number was obtained at 4.0% nanoparticle volume fraction and the highest Dean number. This result is attributed that adding nanoparticles increases viscosity of nanofluid which enhances the random and irregular movement. Pure water has the lowest average Nusselt number.

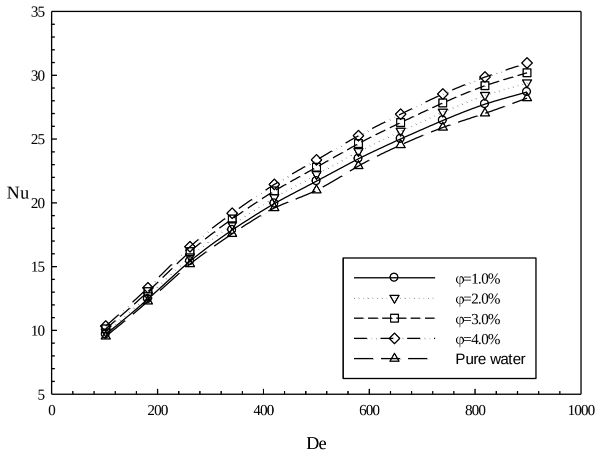

Figure 5. Changing of average Nusselt number with Dean number for different nanoparticle volume fractions of SiO_2/water nanofluid

Figure 6 shows changing of average Darcy friction factor with nanoparticle volume fractions and Dean numbers. It can be noticed that average Darcy friction factor is not affected of changing of nanoparticle volume fractions. This is advantage of SiO_2/water nanofluid flow for reducing pumping power. But, average Darcy friction factor decreases with the increasing Dean number.

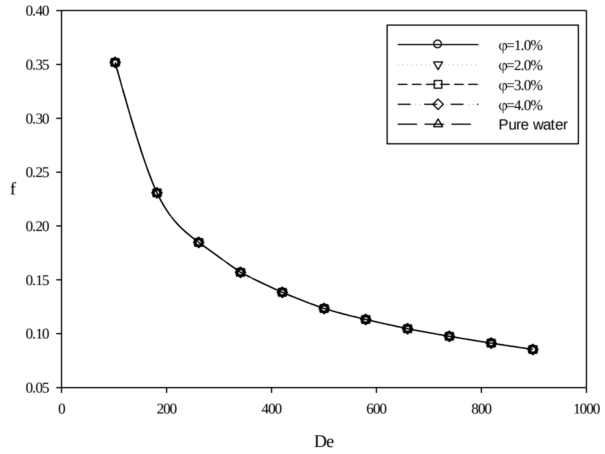

Figure 6. Changing of average Darcy friction factor with Dean number for different nanoparticle volume fractions of SiO_2/water nanofluid

Figure 7 shows that temperature distributions of duct outlet at De=500 for different nanoparticle volume fractions. It can be revealed that temperature distributions are symmetric. The outer wall of curved duct has lower temperature compared to the inner wall because of highest flow velocity at the outer wall. Flow velocity of the outer wall is high according to flow velocity of the inner wall. Also, increasing of nanoparticle volume fractions slightly effects the temperature distributions of the inner wall.

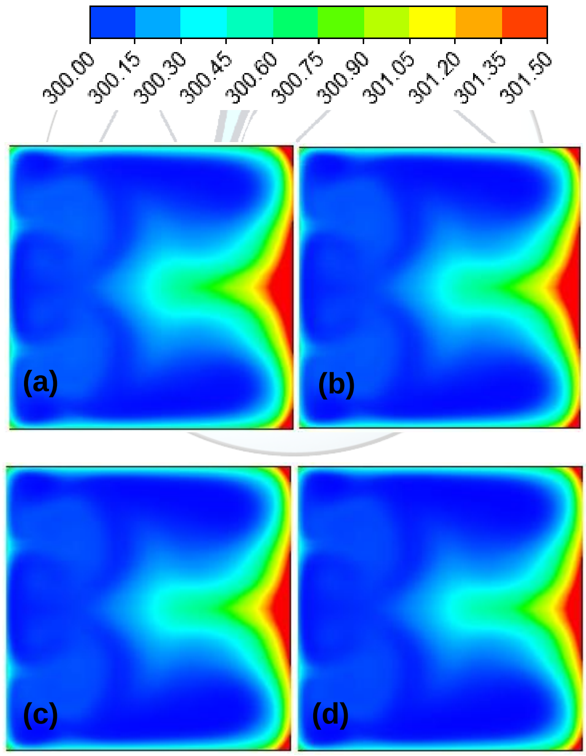

Figure 7. Temperature distribution of duct outlet for different nanoparticle volume fractions of SiO_2/water nanofluid at De=500 (a) 1.0%, (b) 2.0%, (c) 3.0%, (d) 4.0%

Figure 8 shows that the velocity streamlines of duct outlet for different nanoparticle volume fractions of SiO_2/water nanofluid at De=500. It was obtained that velocity distributions are symmetric and secondary flows occurs. As nanoparticle volume fraction increases, the velocity at the outer wall increases.

Figure 9 shows that temperature distribution of duct outlet for different Dean numbers for 4.0% nanoparticle volume fraction of SiO_2/water nanofluid. It was revealed that Dean number significantly effects the temperature distribution in the duct. Temperature distribution decreases with increasing Dean number.

Figure 10 shows that velocity streamlines of duct outlet for different Dean number at 4.0% nanoparticle volume fraction of SiO_2/water nanofluid. It was obtained that Dean number significantly effects velocity distributions, too. At De=102, there is a secondary flow which has one center. However, at De=500, there are a lot of secondary flow. On the other hand, at De=898, there is no secondary flow. Also, velocity distribution increases with increasing Dean number.

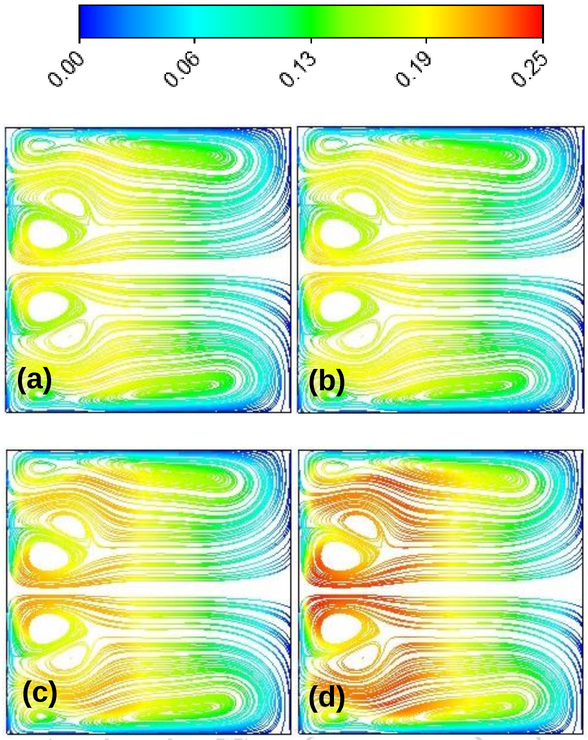

Figure 8. Velocity streamlines of duct outlet for different nanoparticle volume fractions of SiO_2/water nanofluid at De=500 (a) 1.0%, (b) 2.0%, (c) 3.0%, (d) 4.0%

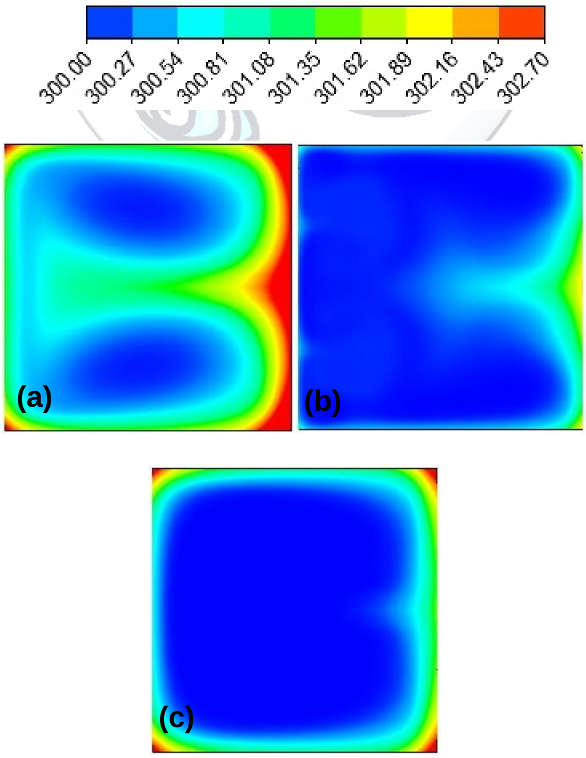

Figure 9. Temperature distribution of duct outlet for different Dean numbers at 4.0% nanoparticle volume fraction of SiO_2/water nanofluid (a) De=102, (b) De=500, (c) De=898

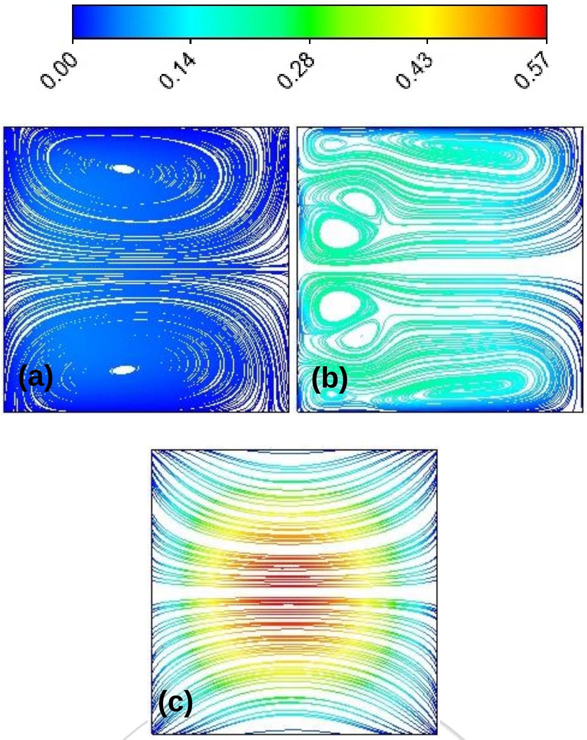

Figure 10. Velocity streamlines of duct outlet for different Dean numbers at 4.0% nanoparticle volume fraction of SiO_2/water nanofluid (a) De=102, (b) De=500, (c) De=898

5. CONCLUSIONS

Numerical simulation of steady state laminar forced convection flow and heat transfer in a 3D square cross-sectioned curved duct using SiO_2/water nanofluid is presented. Effect of different Dean numbers and the nanoparticle volume fractions were investigated. It was noticed from the numerical simulation results that SiO_2/water nanofluid enhances heat transfer. The average Nusselt number increases with increasing the Dean number and nanoparticle volume fractions. Also, Darcy friction factor decreases with increasing Reynolds number. There is no effect of changing of nanoparticle volume fractions on average Darcy friction factor.

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