JOURNAL OF APPLIED SCIENCES RESEARCH

JOURNAL home page: http://www.aensiweb.com/jasr.html 2014 March; 10(3): pages 165-170.

Published Online: 15 January 2014. **Research Article**

# Thermo-Economic Analysis of Fouling Effect in Circular Duct with Constant Wall Temperature Using Nanofluid

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**Received:** 25 January 2014; **Revised:** 8 April 2014; **Accepted:** 20 April 2014; **Available online:** 15 May 2014.



ABSTRACT

In this paper, the second law based thermo-economic analysis of turbulent Cu-water nanofluid flow is studied analytically inside isothermal circular duct. The effects of fouling layer on heat transfer and pressure drop are investigated. The fouling thickness, nanoparticles volume fraction and dimensionless temperature are considered to be the main parameters in this study. Variation of the total cost of irreversibility per unit exergy of heat transfer under fouling thickness is presented and discussed.

*Keywords:* Irreversibility, Turbulent forced convection, Nanofluid, Fouling.

## Nomenclature

| | |
|---|---|
| $c_P$ | specific heat, $kJ/kg\,K$ |
| $D_i$ | Inner diameter, $m$ |
| $D_o$ | outer diameter, $m$ |
| $f$ | friction factor |
| $h$ | heat transfer coefficient, $W/m^2\,K$ |
| $k$ | thermal conductivity of the fluid, $W/m\,K$ |
| $L$ | length of duct, $m$ |
| $\dot{m}$ | mass flow rate, $kg/s$ |
| $Nu$ | Nusselt number |
| $P$ | pressure, $Pa$ |
| $\Pr$ | Prandtl number |
| $\Delta P$ | pressure drop, $Pa$ |
| $\mathrm{Re}$ | Reynolds number |
| $S$ | specific entropy, $kJ/kg\,K$ |
| $T$ | temperature, $K$ |
| $TCI$ | total cost of irreversibility, $\$/h$ |
| $t_f$ | thickness of fouling, $m$ |
| $u$ | velocity, $m/s$ |
| $x$ | local position along the flow direction, $m$ |
| $\dot{X}$ | rate of exergy, $W$ |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | cost ratio |
| $\mu$ | viscosity of the fluid, $Pa.s$ |
| $\rho$ | density, $kg/m^3$ |
| $\theta$ | dimensionless temperature, $(=\frac{T_s-T_1}{T_s})$ |
| $\chi$ | nanoparticles volume fraction |
| $\lambda_T$ | Unit cost of heat exergy, $\$/J$ |
| $\lambda_P$ | Unit cost of pressure exergy, $\$/J$ |
| $\eta$ | total cost of irreversibility per unit exergy of heat transfer, $\$/J$ |

*Subscripts*

| | |
|---|---|
| 1 | inlet of duct |
| 2 | outlet of duct |
| $bf$ | base fluid |
| $f$ | fouling |
| $i$ | segment inlet |
| e | segment outlet |
| $m$ | balk temperature |
| w | fouling surface |
| s | Duct surface |
| $nf$ | nanofluid |
| $o$ | surrounding |
| $p$ | particles |

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## INTRODUCTION

Enhancement of convective heat transfer is very important for many industrial heating or cooling systems. The heat convection can passively be enhanced by fluid thermophysical properties. One way of improving the thermal conductivities of fluids is to suspend small solid particles in the fluid. Kulkarni et al. [1], that proposed $AL_2O_3$ nanofluids as jacket water coolant in a diesel engine for electric generation, their results show that the efficiency of waste heat recovery heat exchanger increased by nanofluid. Li and Xuan [2] and Xuan and Li [3] investigated experimentally the convective heat transfer and flow characteristics for Cu–water nanofluid flowing through a straight tube with a constant heat flux under laminar and turbulent flow conditions. Das et al. [4] studied the increase of thermal conductivity with temperature for Water-$Al_2O_3$ and water-Cuo nanofluids as the suspension material using the temperature oscillation technique. The advantages of nanofluids with respect to heat transfer

were discussed in [5]. Palm et al. [6] studied Laminar forced convection flow of nanofluids between two coaxial and parallel disks with central axial injection using temperature dependent nanofluid properties.

Fouling of heat transfer surfaces is one of the most important problems in heat transfer equipment. The fouling layer increasing heat transfer resistance because this has a low thermal conductivity and the cross sectional area of duct is reduced, which causes an increase in pressure drop across the duct [7-13].

Heat transfer is a fundamental source of thermodynamic irreversibility in all real engineering devices. The second law analysis is the technique for optimization in thermal equipments and systems, which makes good engineering sense to focus on the irreversibilities of fluid flow and heat transfer processes, based on the Second Law analysis of thermodynamics in tube was proposed by Sahin [14, 15]. In this method, both heat transfer and fluid friction generate entropy. Since entropy generation is proportional to the destruction of exergy, an optimum design condition of flow generates minimum irreversible losses.

Many researchers have investigated the problem of entropy generation minimization in fluid flow with heat transfer. Bejan [16, 17] found out the method for analysis the entropy generation in fluid flow with heat transfer. Sahin [18, 19] also investigated the geometries of duct like circular, square, rectangular, equilateral triangle, and sinusoidal on the entropy generation, for duct with uniform wall temperature and uniform heat flux boundary conditions.

Oztop [20] made a study on entropy generation in semicircular ducts. Jarungthammachote [21] investigated entropy generation for hexagonal duct subjected to constant heat flux. Rafee [22] presented a study on entropy generation in annuli with nanofluid. He found that For very large length ratios, applying the nanoparticles is not advised. Evans et al. [23] have conducted Second law based thermo-economic studies in a series of works related to feed water heaters. Zubair et al. [24] developed detailed sizing rules for two phase heat exchangers with respect to thermo-economic optimization.

The present study aims to provide a thermo-economic analysis for convection Cu–water nanofluid in circular duct with constant wall temperature. The effects of Reynolds number, nanoparticles volume fraction, fouling thickness on irreversibility are investigated.

*2 Methodology:*

*2.1 Physical model and thermophysical properties of nanofluid:*

The geometries of the present problem have been shown in Fig. 1. The geometries consist of circular duct with constant surface temperature, thickness of fouling is $t_f$. It can be assumed that the flow is steady, fully developed, incompressible, and turbulent.

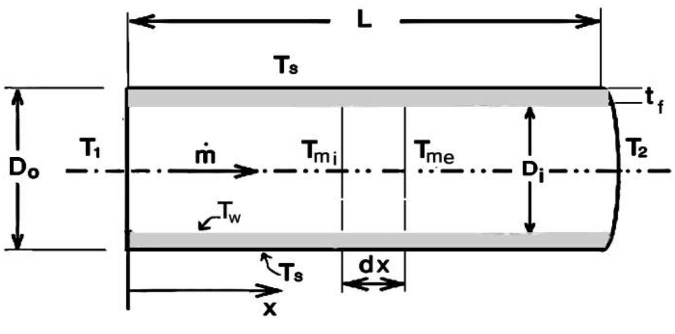


**Fig. 1:** Schematic diagram of the circular duct.

The nanofluid in the channel is Newtonian and assumed that the fluid phase and nanoparticles are in the thermal equilibrium state and they flow with the same velocity.

The thermo-physical properties (Table. 1) of the nanofluid are obtained from the following relations [25]:

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

$$(\rho c_p)_{nf} = (1-\chi)(\rho c_p)_{bf} + \chi(\rho c_p)_p \quad (2)$$

$$\frac{k_{nf}}{k_{bf}} = \frac{k_p + 2k_{bf} - 2\chi(k_{bf}-k_p)}{k_p + 2k_{bf} + \chi(k_{bf}-k_p)} \quad (3)$$

$$\mu_{nf} = \frac{\mu_{bf}}{(1-\chi)^{2.5}} \quad (4)$$

**Table 1:** Thermophysical properties of pure fluid and nanoparticles.

| Property | Fluid phase(water) | Solid phase (Cu) |
|---|---|---|
| $C_P\ (J/kg\,K)$ | 4179 | 385 |
| $\rho\ (kg/m^3)$ | 997.1 | 8933 |
| $k\ (W/m\,K)$ | 0.613 | 400 |

*2.2 Thermo-economic evaluation criteria:*

The energy balance for a control volume with length *dx* is

$$\dot{m}c_P T_{mi} = h\pi D_i\,dx(T_W - T_m) + \dot{m}c_P T_{me} \quad (5)$$

where $\dot{m}$ is the mass flow rate given by

$$\dot{m} = \rho u \frac{\pi D_i^2}{4} \quad (6)$$

Conduction heat transfer from the fouling layer it is

$$\frac{T_S - T_W}{\ln(D_o/D_i)/2\pi k_f\,dx} = h\pi D_i\,dx(T_W - T_m) \quad (7)$$

where

$$D_i = D_o - 2t_f \quad (8)$$

According to above equations, $T_W$ fouling surface temperature and $T_{me}$ outside temperature for each segment respectively are obtained

$$T_w = \frac{T_S + \left(\frac{h\ln(D_o/D_i)T_m}{2\pi k_f\,dx}\right)}{1 + \left(\frac{h\ln(D_o/D_i)}{2\pi k_f\,dx}\right)} \quad (9)$$

$$T_{me} = T_{mi} - (h\pi D_i\,dx/\dot{m}c_p)(T_W - T_{mi}) \quad (10)$$

In a steady-state and steady-flow process, if the exergy rate at the inlet is smaller than that at the outlet, the rate of exergy of the flow lost in the duct when the changes of kinetic energy and potential energy were neglected, can be expressed as follows [16]:

$$\Delta\dot{X} = \dot{X}_2 - \dot{X}_1 = \dot{m}(h_2 - h_1) - T_o(S_2 - S_1) \quad (11)$$

If the flow is an incompressible fluid, we can write Equation (11) as

$$\Delta\dot{X} = \dot{m}c_P T_o\left[\frac{T_2 - T_1}{T_o} - \ln\left(\frac{T_2}{T_1}\right) + \left(\frac{P_2 - P_1}{\rho c_P T_o}\right)\right] \quad (12)$$

In this paper, above equation was expressed in a general form as follows

$$\Delta\dot{X} = \dot{m}c_P T_o\left[\frac{T_2 - T_1}{T_o} - \ln\left(\frac{T_2}{T_1}\right)\right] + \dot{m}c_P\left(\frac{\Delta P}{\rho c_P T_o}\right) = \Delta\dot{X}_T + \Delta\dot{X}_P \quad (13)$$

The pressure drop $(\Delta P = P_2 - P_1)$ for incompressible flow in circular cross section duct given by

$$\Delta P = \rho f \frac{L}{D_i}\frac{u^2}{2} \quad (14)$$

In Thermo-economics, this includes total cost of irreversibility. Thus the total cost of irreversibility (TCI) can be expressed as

$$TCI = \lambda_T(\Delta\dot{X}_T + \alpha\Delta\dot{X}_P) \quad (15)$$

where $\lambda_T$ unit cost of heat exergy and $\alpha$ is the cost ratio $(\frac{\lambda_P}{\lambda_T})$.

The exergy used in the heat transfer process can be written in the rate form as [26]

$$\dot{X}_q = \dot{m}c_P(T_2 - T_1)(1 - \frac{T_o}{T_{ave}}) \quad (16)$$

$T_{ave}$ is the average temperature of the fluid inside the duct, estimated as [27]

$$T_{ave} = \frac{T_2 - T_1}{\ln(T_2/T_1)} \quad (17)$$

In this work, a economic criterion evaluating the effect of the fouling on the total cost of irreversibility per unit exergy of heat transfer in forced convective heat transfer in duct, can be written

$$\eta = \frac{TCI}{\dot{X}_q} \tag{18}$$

**Results and discussion**

In this paper, investigated the effect of the fouling on the total cost of irreversibility per unit exergy of heat transfer in forced convective heat transfer in circular duct, Cu-water nanofluid is selected as the working fluid. The surface temperature of duct is 320K and the surrounding temperature is 298 K. The length and diameter of duct are 10 and 10 mm, respectively. For thermally and hydrodynamic fully developed turbulent flow in smooth duct [28],

$$Nu = 0.024 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.4} \tag{19}$$

$$f = (0.79 \ln \mathrm{Re} - 1.64)^{-2} \tag{20}$$

Fig. 2 shows the variation of the total cost of irreversibility per unit exergy of heat transfer $(\eta)$ versus Reynolds number in different thickness of fouling for pure water. The figure shows that η increases by increasing the Reynolds number in all fouling thickness. As the fouling thickness is increased, η increases. For a fixed Reynolds number as fouling thickness values are increased η values are increased, especially for higher values of Reynolds number. These results indicate that for larger fouling thickness, friction irreversibility is higher than that of lower fouling thickness as expected and heat irreversibility decreases by increasing fouling thickness.

Fig. 3 shows the variation of total cost of irreversibility per unit exergy of heat transfer (η) versus fouling thickness in different volume concentration of nanoparticles and fixed Reynolds number. As the fouling thickness is increased, η increases for fixed volume concentration of nanoparticles. For a fixed fouling thickness as volume concentrations of nanoparticles are increased η is increased, especially for higher values of fouling thickness because in higher values of fouling thickness, friction irreversibility increases.

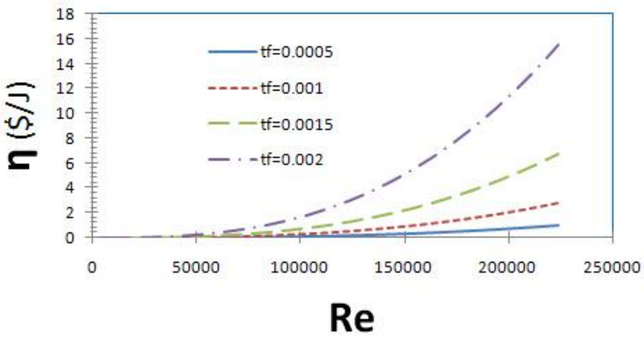


**Fig. 2:** Variation of η with Reynolds number for different fouling thickness $(\chi = 0\ ,\ \theta = 0.05)$.

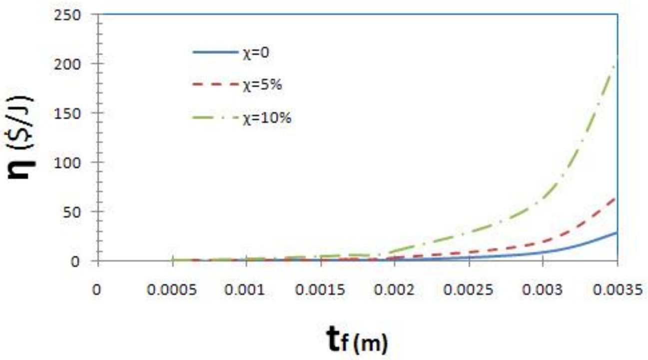


**Fig. 3:** Variation of η with fouling thickness for different nanoparticles volume fraction $(\mathrm{Re} = 94000\ ,\ \theta = 0.05)$

Fig. 4 illustrates the total cost of irreversibility per unit exergy of heat transfer (η) versus fouling thickness in different cost ratios for pure water and 5% nanoparticle volume concentration in fixed Reynolds number. It can be seen that η increases with the increase of fouling thickness for pure water and 5% nanoparticle volume concentration. For a fixed fouling thickness, by increasing cost ratio, η is increased especially for higher values of fouling thickness, because the cost ratio is coefficient of friction irreversibility.

Fig. 5 illustrates the cost of irreversibility per unit exergy of heat transfer (η) versus dimensionless temperature in different fouling thickness for pure water and fixed Reynolds number. As dimensionless temperature is increased, the η decreases, especially for higher values of dimensionless temperature. The decrease of η depends on the decrease of dimensionless temperature as expected.

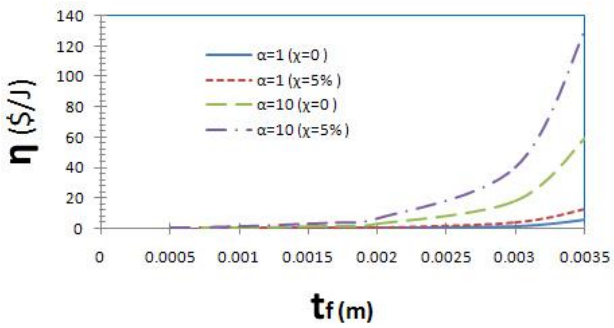


**Fig. 4:** Variation of η with fouling thickness for different cost ratio $(\mathrm{Re}=94000\ ,\ \theta=0.05)$

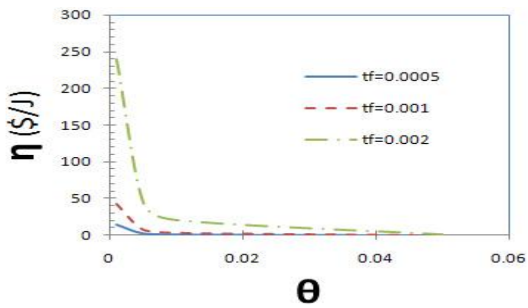


**Fig. 5:** Variation of η with dimensionless temperature for different fouling thickness $(\mathrm{Re}=94000\ ,\ \theta=0.05)$

*2.4 Conclusion:*

The present paper investigated the effect of fouling thickness on the total cost of irreversibility per unit exergy of heat transfer (η) of Cu-water nanofluid in circular duct subjected to constant wall temperature. The Reynolds number and nanoparticles volume fractions are in the ranges of 4000 to 224000 and 0 to 0.1 respectively. The following results can be de derived from this study:

The total cost of irreversibility per unit exergy of heat transfer (η) increases by increasing the Reynolds number in all fouling thickness.

The total cost of irreversibility per unit exergy of heat transfer (η) increases by increasing the fouling thickness in fixed Reynolds number.

As volume concentrations of nanoparticles is increased, η is increasing.

For a fixed fouling thickness, by increasing cost ratio, η is increased in all nanoparticles volume fractions.

As dimensionless temperature is increased, the η decreases in all fouling thickness.

**Acknowledgements**

Financial support for this research was provided by the Mahshahr Branch, Islamic Azad University, Mahshahr, Iran with title: "Economic evaluation of irreversibility and pumping power of nanofluid flow in tube with constant wall temperature".

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