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Mechanical Engineering

Elixir Mech. Engg. 42 (2012) 6135-6138

ISSN: 2229-712X

Effective parameters on second law analysis for circular segment ducts in fully developed laminar flow under constant wall heat flux

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ARTICLE INFO

Article history:
Received: 16 November 2011;
Received in revised form:
16 December 2011;
Accepted: 27 December 2011;

Keywords
Entropy generation,
Circular segment duct,
Laminar flow,
Pumping power,
Constant wall heat flux.

ABSTRACT

In this study, the entropy generation of a fully developed laminar flow in circular segment ducts with constant wall heat flux is investigated. Entropy generation is obtained for various segment angels (2φ), various wall heat flux and various Reynolds number. It is concluded that segment angel and wall heat flux have considerable effect on entropy generation. For the increasing value of these parameters, both entropy generation and pumping power ratio are increased at fixed Reynolds number.

Introduction

In a thermodynamic process the loss of exergy is primarily due to the associated irreversibilities which generate entropy. Most convective heat transfer processes are characterized by two types of exergy losses, e.g. losses due to fluid friction and those due to heat transfer across a finite temperature difference. The above two interrelated phenomena are manifestations of thermodynamic irreversibility and investigation of a process from this standpoint is known as second law analysis. However, there exists a direct proportionality between the wasted power (the rate of available work lost) and the entropy generation rate. [1,2]. For efficient optimal thermodynamic design entropy generation must be reduced. In this context, geometry of duct (cross-sectional area) is an important parameter on entropy generation. Various cross-sectional ducts are used in heat transfer devices due to the size and volume constraints to enhance heat transfer with passive method. Pressure drop and heat transfer analysis in various shaped ducts were summarized by Shah and London [3].

Sahin [4] presented the second law analysis for different shaped duct such as triangular, sinusoidal etc, in laminar flow and constant wall temperature boundary conditions. He made a comparison between these ducts to find an optimum shape. He found that the circular duct geometry is the favorable one among them. He made another study in order to investigate the constant heat flux effects on these cross-sectional ducts without taking into account the viscosity variation in the analysis [5]. Viscosity variation was considered by Sahin [6] for turbulent flow condition for circular ducts [7].

Recently, Oztop [8] made a study on entropy generation in semicircular ducts. Dagtekin et al.[9]investigated the problem for circular duct with different shaped longitudinal fins for laminar flow using the similar methods of Sahin [5]. Oztop et al. [10] made a study on entropy generation in rectangular ducts with semicircular ends ducts with two boundary conditions: constant wall temperature and constant wall heat flux. Jarungthammachote [11] investigated entropy generation for hexagonal duct subjected to constant heat flux and Falahat [12] made a study entropy generation in parallel plate ducts with span wise periodic triangular corrugations at one wall at constant heat flux.

The main aim of this paper is to investigate entropy generation through circular segment ducts. To the best of the author's knowledge the entropy generation in circular segment ducts with constant wall heat flux has not yet been investigated. The present paper reports an analytical study of entropy generation in laminar flow. The effect of Reynolds number, wall heat flux and segment angel on entropy generation and pumping power are analyzed.

PHYSICAL MODEL OF PROBLEM

The physical model of annular sector duct is depicted in Fig. 1. The hydraulic diameter of any duct given by

$$D_h = \frac{4A_c}{p} \tag{1}$$

Where A_c is cross-sectional area and P is perimeter. The hydraulic diameter for annular sector cross-sectional area can be written as

$$D_h = \frac{2\sqrt{\varphi - \sin(\varphi)\times\cos(\varphi)}}{\sin(\varphi)+\varphi}\sqrt{A_c} \tag{2}$$

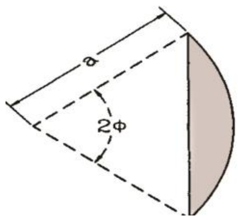

Figure 1. Cross section of circular segment duct

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ENTROPY GENERATION ANALYSIS

The total entropy generation within a control volume of thickness dx , shown in Fig. 2, along the duct can be written as follows

$$d\dot{S}_{gen} = \dot{m}\, ds - \frac{\delta \dot{Q}}{T_w} \tag{3}$$

Where $\delta \dot{Q} = qP dx$ is heat transfer rate to the fluid flowing in this system for an incompressible fluid we have

$$T\, ds = C_P\, dT - v\, dP \tag{4}$$

Substituting ds from Eq. 4 into Eq. 3, $d\dot{S}_{gen}$ can be written as

$$d\dot{S}_{gen} = \dot{m} C_P \left(\frac{T_w - T}{T T_w} dT - \frac{1}{\rho C_P T} dP \right) \tag{5}$$

Pressure drop in Eq. 5 is given in the following equation.

$$dP = -\frac{f \rho U^2}{2 D_h} dx \tag{6}$$

The bulk temperature variation of fluid along a duct is given in the following equation.

$$T = T_o + (4q / \rho U D_h C_P)\, x \tag{7}$$

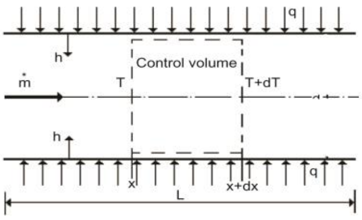

Figure 2.control volume for entropy generation

And then, for the constant wall heat flux boundary conditions, the total entropy generation is obtained by integration of Eq. (5) using Eqs. (2) and (7). The dimensionless total entropy generation based on the flow stream heat capacity rate $(\dot{m}/C_P)$ is defined as

$$Ns = \ln\left[\frac{(\mathrm{Re} + \tau\eta_1)(1+\tau)}{(\mathrm{Re} + \tau\,\mathrm{Re} + \tau\eta_1)}\right] + \eta_2\, \mathrm{Re}^2 \ln\left[\frac{\mathrm{Re} + \tau\eta_1}{\mathrm{Re}}\right] \tag{8}$$

In this equation η_1 and η_2 are,

$$\eta_1 = 4Nu\,\lambda / \mathrm{Pr} \tag{9}$$

$$\eta_2 = \mu^3 f\, \mathrm{Re} / 8\rho^2 D_h^{\,3} q \tag{10}$$

In these equations some parameters can be made dimensionless as follows

$$St = \frac{h}{\rho U C_P} = \frac{Nu}{\mathrm{RePr}} \tag{11}$$

$$\tau = \frac{T_W - T}{T_o} \tag{12}$$

$$\lambda = \frac{L}{D_h} \tag{13}$$

The fully developed flow and heat transfer characteristics obtained by Sparrow and Haji-Sheikh [13] are given in Table 1.

PUMPING POWER TO HEAT TRANSFER RATIO

The power required to overcome the fluid friction in the duct in dimensionless form is

$$PPR = \frac{A_c\, \Delta P U}{\dot{Q}} \tag{14}$$

For constant wall heat flux boundary conditions, the pumping power to heat transfer ratio for fully developed laminar flow becomes

$$PPR = \eta_2\, \mathrm{Re}^2 \tag{15}$$

RESULT AND DISCUSSION

A second law analysis is conducted for circular segment duct in laminar flow regime. Water has been used as working fluid. The thermophysical properties used are shown in Table 2. Fig. 3 shows dimensionless entropy generation for different segment angel (2φ) of circular segment duct at different Reynolds number values. In this figure, total entropy generation decreases considerably while the Reynolds number is increased. As the value of segment angel (2φ) is increased, total entropy generation increased for fixed Reynolds number. As can be shown, the circular segment geometry, gives the lowest dimensionless entropy generation and circle geometry $(2\varphi = 360^0)$ has a largest dimensionless entropy generation.

Fig. 4 shows variation of pumping power for different segment angel (2φ) and Reynolds numbers. As the segment angel (2φ) is increased pumping power to heat transfer ratio increases. With increasing of Reynolds number pumping power to heat transfer ratio values also increase. For a fixed Reynolds number as segment angel (2φ) values are increased pumping power to heat transfer ratio values are increased, especially for higher values of Reynolds number. These results indicate that for larger aspect ratios friction losses are higher than that of lower aspect ratios as expected.

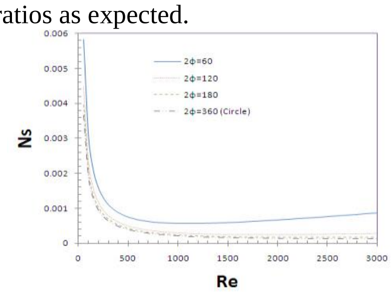

Figure 3. Variation of dimensionless entropy generation for different 2φ values and Reynolds number

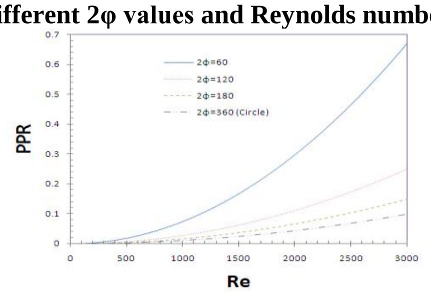

Figure 4. Variation of pumping power for different 2φ values and Reynolds number

Fig. 5 shows the effects of heat flux value on entropy generation at different Reynolds numbers for a fixed segment angel $(2\varphi = 120^o)$, namely, for a fixed cross sectional area. Entropy generation is influenced by the wall heat flux. As Reynolds number is increased, the dimensionless entropy generation decreases, especially for higher q values. Thus, lower entropy generation is obtained for higher value of q (e.g. $q = 3000W/m^2$). The decrease of total entropy generation depends on the increase of wall heat flux as expected. Fig. 6 shows variation of pumping power to heat transfer ratio for various wall heat flux and Reynolds numbers. Increasing Reynolds number yields higher pumping power to heat transfer ratio values for fixed q values. As the wall heat flux is increased pumping power to heat transfer ratio values decreases for fixed Reynolds number, especially for higher values of Reynolds number. These results indicate that for lower wall heat flux friction losses are higher than that of higher wall heat flux as expected.

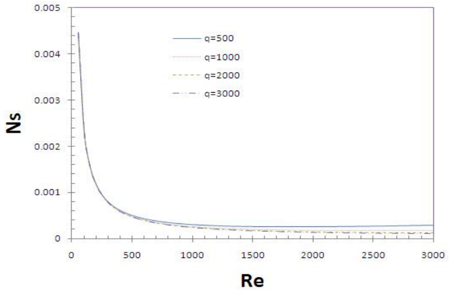

Figure 5. Variation of dimensionless entropy generation for various wall heat flux values and Reynolds number

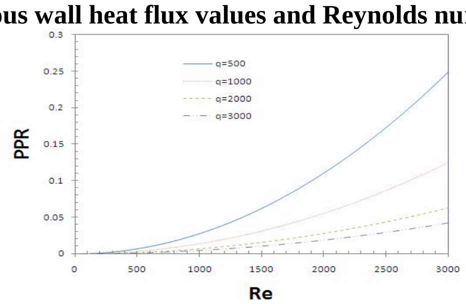

Figure 8. Variation of pumping power for various wall heat flux values and Reynolds number

CONCLUSION

A analytical investigation of entropy generation in ducts of circular segment cross section has been performed and the results compared to circular tube. It was found that the circular segment duct performs better than the circular duct. The effect of segment angel and the value of heat flux were also investigated. Increasing segment angel increased entropy generation. The same was found to be true for increased heat flux. In some cases there seemed to exist minimum value of entropy generation at a certain Re.

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Table 1. The fRe and Nu for Fully Developed Laminar Flow in Circular Segment Ducts [13]

2ϕ	fRe	Nu_{H1}	2ϕ	fRe	Nu_{H1}
0	15.555	3.580	120	15.690	3.894
10	15.558	3.608	180	15.767	4.089
20	15.560	3.616	240	15.840	4.228
40	15.575	3.648	300	15.915	4.328
60	15.598	3.696	360	16.000	4.364
80	15.627	3.756			

Table 2. thermophysial properties of water

SYMBOL	QUANTITY
$C_P\,(J/kgK)$	4182
Pr	7
$T_w\,(K)$	293
$\mu(Ns/m^2)$	9.93×10^{-4}
$\rho(kg/m^3)$	998.2