Applied Thermal Engineering 61 (2013) 268–277

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Heat transfer improvement due to the imposition of non-uniform wall heating for in-tube laminar forced convection

M.R. Hajmohammadi [a,*], S. Poozesh [b], M. Rahmani [c], A. Campo [d]

[a] Department of Mechanical Engineering, Amirkabir University of Technology, Hafez Ave., Tehran 158754413, Iran
[b] Department of Mechanical Engineering, University of Kentucky, Lexington, KY 40508, USA
[c] Department of Mechanical Engineering, Arak University of Technology, Arak 41167-38181, Iran
[d] Department of Mechanical Engineering, The University of Texas at San Antonio, San Antonio, TX 78249, USA

HIGHLIGHTS

- Considered the bearing of non-uniform distribution of heat flux on the hot spots.
- Determined the optimal distribution of heat flux that minimizes the hot spots.
- Results are compared with those obtained by EGM method.

ARTICLE INFO

Article history:
Received 24 June 2013
Accepted 8 August 2013
Available online 20 August 2013

Keywords:
Heat transfer enhancement
Circular pipe
Optimized heat transfer
Constructal
Entropy generation minimization

ABSTRACT

This paper explores the bearing that a non-uniform distribution of heat flux used as a wall boundary condition exerts on the heat transfer improvement in a round pipe. Because the overall heat load is considered fixed, the heat transfer improvement is viewed through a reduction in the maximum temperature ('hot spot') by imposing optimal distribution of heat flux. Two cases are studied in detail 1) fully developed and 2) developing flow. Peak temperatures in the heated pipe wall are calculated via an analytical approach for the fully developed case, while a numerical simulation based on CFD is employed for the developing case. By relaxing the heat flux distribution on the pipe wall, the numerical results imply that the optimum distribution of heat flux, which minimizes the peak temperatures corresponds with the 'descending' distribution. Given that the foregoing approach is quite different from the 'ascending' heat flux distribution recommended in the literature by means of the entropy generation minimization (EGM) method, it is inferred that the optimization of heat transfer and fluid flow, in comparison with the thermodynamic optimization, may bring forth quite different guidelines for the designs of thermal systems under the same constraints and circumstances.

1. Introduction

The dual optimization of heat transfer and fluid flow constitutes subjects of paramount interest in heat exchange equipment. In the last decades, the analysis of heat transfer and fluid flow in multiscale flows and the choice of configurations for maximum heat transfer density have been widely studied by many researchers. Some of these studies are carried out based on the Constructal law proposed by Bejan [1]. They are normally carried out to endow the flow configuration with some degree of freedom to improve the access to the currents that flow through it [1–3]. During the past decades, Constructal theory has been applied by many researchers so as to improve the performance of heat transfer systems [4–13]. More recently, 'Entransy' [14] which is a new physical quantity reflecting heat transfer ability of an object has been introduced and 'Constructal Entransy dissipation rate minimization' has been applied for optimizing heat transfer systems [14–17].

In heating devices, such as heat exchangers, the magnitude of heat flux and the total heat transfer area in a pipe, $A_h = \pi DL$ (D is the pipe diameter and L is the pipe length) are specified for the analysis of heat transfer and fluid flow inside the pipes. In such cases, the amount of total heat transfer, $Q = A_h q''$, to the moving fluid is known. It is also natural to consider the total heat transfer area, A_h, as a constraint. Thus, the thermal objective is to control the heated wall temperature, such that the temperature peaks (the so called "hot spots") do not exceed a pre-set allowable value. To accomplish this goal, the flow configuration may be manipulated utilizing several techniques of heat transfer augmentation. For example, Bejan and Sciubba [18]

* Corresponding author. Tel.: +98 9125486617; fax: +98 2166419736.
E-mail address: mh.hajmohammadi@yahoo.com (M.R. Hajmohammadi).

1359-4311/$ – see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.applthermaleng.2013.08.009

proposed that by optimizing the space between heat sources in an array cooled by forced convection, minimum peak temperatures can be achieved. Another technique presented by Bejan and Fautrelle [19] included the insertion of additional heat sources in the entrance region of the flow between the current heat sources. Hajmohammadi et al. [20] proposed to place a thick conducting plate above the heat sources cooled by forced convection boundary layer flow, to conduct the heat current in optimal manner. Another related scheme was reported in Refs. [21–26] delineating optimal distributions and arrangements of the heat sources to reach the minimum "hot spot" temperature. For instance, Balaji [21] exploited micro genetic algorithm (MGA) to optimize the location of three discrete heat sources which could be placed anywhere inside a ventilated cavity and cooled by forced convection. Using artificial neural networks (ANN), Sudhakar et al. [22] sought the optimum configuration for five discrete heat sources mounted on a wall of a three-dimensional vertical duct under mixed convection heat transfer. Muftuoglu and Bilgen [23] performed a numerical simulation to determine the optimum location of discrete heaters placed in an open square cavity. Further, analytical calculations were carried out independently by da Silva et al. [24] and Hajmohammadi et al. [25] to determine the optimal distribution of heat sources over a plate cooled by forced convection. Recently, Hajmohammadi et al. [26] suggested the optimally placement of insulated segments between the heated segments along the heated pipe wall. Applying this technique, these authors indicated that the effective length of the thermal entrance region is enlarged and as a result, the maximum temperature of the heated pipe wall is eventually reduced.

On the other hand, aside from maximum temperature minimization, 'minimizing the irreversibility' or 'entropy generation minimization' method [27–29], as the alternative strategy for achievement of an optimized heat and fluid flow configuration, has experienced tremendous growth during the 1980s and 1990s. This method has been successively used by Bejan [27] to thermodynamically optimize real devices such as power plants and refrigeration systems [27], single-phase liquid cooling devices [30], fan-supplied tube-fin condensers [31] and heat exchangers [32]. Recently, Esfahani and Shahabi [33] investigated seven heat flux distributions for a thermal boundary condition on the pipe wall in connection to laminar developing pipe flow of a high Prandtl number fluid. When the heat load is considered fixed in all cases, they indicated that the 'ascending-shaped' distribution of heat flux minimizes the entropy generation.

The problem to be addressed in this study deals with investigating the effect that various distribution of heat flux as a thermal boundary condition on the heated pipe wall, exerts on the maximum temperature of the heated wall, when the total heat load, mass flow rate, geometry (length and diameter) of the pipe and as a consequence, the pressure drop are fixed. The primary objective is to find an optimal heat flux distribution which minimizes the maximum temperature in the heated pipe wall, the so-called 'hot spots'. Upon considering the velocity as fully developed, two cases are scrutinized: 1) thermally developed flow and 2) thermally developing flow. From a historical perspective, the problem with arbitrary variations in axial wall heat flux can be handled by the superposition method using Duhamel's theorem [34,35]. Fortunately, manipulating these relations paves the way to the determination of an exact analytical solution for fully-developed flow regime without suffering from the complexities associated with numerical computations or experimental measurements. However, regarding to the complexities related to thermally developing flow, a numerical solution is employed to compute the peak temperatures in the heated pipe wall. After performing a detailed optimization procedure, it is indicated that contrary to the 'ascending-shaped' of heat flux distribution recommended by Esfahani and Shahabi [33] using the entropy generation minimization method, the 'descending-shaped' distribution delivers the optimized heat transfer. Owing that the mass flow rate and pressure drop are taken as fixed parameters in all cases, the optimum 'descending-shaped' distribution obtained in this work by way of minimization of the peak temperature, ultimately supplies the optimized heat transfer and fluid flow. In synthesis, this discovery signifies that the entropy generation minimization method does not always guarantee optimized heat transfer and fluid flow conditions.

2. Physical system

The configuration sketched in Fig. 1 consists of a pipe with n heating segments of length l_i (i = 1, 2, ..., n) attached at the outer surface. Each heater generates heat with uniform heat flux, q''_i. The heating takes place with non-uniform heat flux, $q''(x)$. The magnitude of $q''(x)$ can be expressed as follows

$$q''(x) = q''_1 \widehat{q}_i \tag{1}$$

where

$$\widehat{q}_i = \begin{cases} 1 & 0 \le x < x_1; \ \ (i = 1) \\ \frac{q''_i}{q''_1} & x_{i-1} \le x < x_i; \ \ (i = 2, 3, \ldots, n-1) \\ \frac{q''_n}{q''_1} & x_{i-1} \le x \le L; \ \ (i = n) \end{cases} \tag{2}$$

Here, $\widehat{q}_i$ represents the ratio of the heat generation rate of the i-th heater to that of the first heater, L is the length of the pipe and x_i can be expressed by the following relation

$$x_i = \sum_{k=1}^{i} l_k; \quad i = 1, 2, 3 \ldots, n \tag{3}$$

Additionally, it is assumed that under the prevalent circumstances of variable heat flux distributions, the total heat removal rate from the pipe is fixed, and can be adequately expressed as

$$\sum_{i=1}^{n} l_i q''_i = q''_{ave} L = \text{Const.} \tag{4}$$

where q''_{ave} is the mean value of heat generation rates, q''_i. The constraint of fixed pipe length

$$\sum_{i=1}^{n} l_i = L = \text{Const.} \tag{5}$$

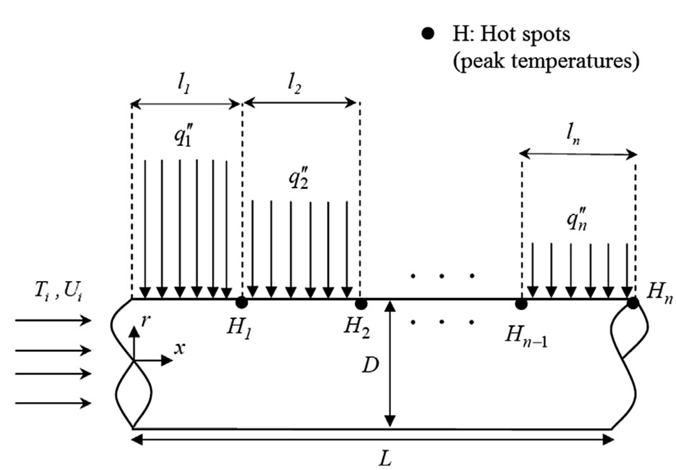

Fig. 1. A schematic view of the physical system.