Energy Conversion and Management 94 (2015) 190–197

Contents lists available at ScienceDirect

Energy Conversion and Management

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

Optimum structural design of a heat exchanger for gas-circulation systems

Wenjin Wang, Bo Li, Heng Zhao, Yi Hu, Youqing Wang *

School of Optical and Electronic Information, Huazhong University of Science and Technology, Wuhan 430074, China

ARTICLE INFO

Article history:
Received 6 October 2014
Accepted 23 January 2015

Keywords:
Optimization design
Performance analysis
Computational fluid dynamics
Heat transfer
Pressure loss

ABSTRACT

T-type gas-circulation systems are widely used in gas lasers to remove waste heat from the discharge process. The structure of the heat exchanger is a very important factor that affects the performance of a T-type gas-circulation system. To develop a high-performance heat exchanger for such a gas-circulation application, a computational fluid dynamics approach was adopted for this study. A three-dimensional numerical model was established. A detailed study focused on the influence of the shape of the channel and the location of the finned tubes on the performance of the heat exchanger. Based on the heat-transfer characteristics and the flow structure, a novel geometric structure was proposed to reduce the volume of the heat exchanger. Comprehensive simulations to determine the optimum locations for the finned tubes were also conducted. As a result of this optimization, the heat exchanger for a T-type gas-circulation system could be made more compact and its pressure loss penalty decreased by 11.5% even though its heat-transfer ability remained unchanged. In addition, the results of a theoretical analysis and numerical simulation were found to be in good agreement with the results of the experiment, indicating the validity of the results of the research.

1. Introduction

Fin and tube heat exchangers are widely applied in many industrial systems such as transportation, refrigeration, petrochemicals, and food processing. Improving the performance of these heat exchangers is crucial to satisfying efficiency standards at low cost and with minimal environmental impact. As a result, there is an urgent need for high-efficiency, compact heat exchangers.

Design optimization of these heat exchangers has attracted much attention from researchers over the past decades. During this time, many researchers have made valuable contributions using traditional mathematical methods [1]. However, such methods are tedious and error-prone. In recent years, many novel methods have been applied to the design and optimization of compact heat exchangers. Ghazi et al. [2] demonstrated the successful application of a genetic algorithm to the thermo-economic optimization of heat-exchanger equipment. Using the same principle, Xie et al. [3] succeeded in optimizing the structure and size of a compact heat exchanger. Entransy theory was a new physical concept proposed by Guo et al. [4]. Using this method, the thermal resistance optimization of a heat exchanger was carried out [5]. Yang et al. [6] successfully applied constructal theory to the optimal design of a shell-and-tube heat exchanger conforming to the Tubular Exchanger Manufacturers Association standards. In 2012, the first application of a genetic algorithm in combination with particle swarm optimization was explored as a means of optimizing the design of a plate-fin heat exchanger [7]. Computational fluid dynamics (CFD) is an effective tool for application to the optimization of the design of a heat exchanger in that it enables the study of thermal properties [8]. It can be applied to the optimization of various types of heat exchangers, especially when the structure is complicated. Eiamsa-ard et al. [9] numerically investigated the performance of a three-dimensional channel with triangular wavy baffles using CFD. Al-Waked et al. [10] developed a CFD model that supports the conjugate heat and mass transfer problem of a plate heat exchanger. Tonomura et al. [11] proposed a CFD-based optimization method for the design of plate-fin micro-devices. Wang et al. [12] performed many studies of heat exchanger with longitudinal vortex generators basing on CFD. With the same method, He et al. [13] gave the most suitable values of the parameters of the vortex generators.

However, these earlier investigations focused mainly on heat-transfer enhancement. Within the literature, there are few reports addressing the optimum design of a heat exchanger. To design a heat exchanger that offers both high efficiency and compactness,

* Corresponding author.
E-mail address: yqwang13@163.com (Y. Wang).

http://dx.doi.org/10.1016/j.enconman.2015.01.061
0196-8904/

Nomenclature

D	diameter, mm
C	empirical constant
S_l	longitudinal tube pitch, mm
S_t	spanwise tube pitch, mm
L	fin length, mm
L_c	channel length, mm
W	fin width, mm
H	fin pitch, mm
d	fin thickness, mm
p	pressure, Pa
Δp	pressure drop, Pa
T	temperature, K
E	total energy, J
u	velocity vector, m/s
k	turbulence kinetic energy, m^2/s^2
m	mass flow rate, kg/s
c_p	specific heat of gas, J/kg K
Q	heat transfer rate, W
x, y, z	coordinate direction, m
i, j	Cartesian coordinate

Greek symbols

λ	thermal conductivity, W/m K
ρ	density, kg/m^3
β	thermal expansion coefficients
μ	fluid viscosity, kg/m s
ε	turbulent energy dissipation rate, m^2/s^3
σ	inverse effective prandtl numbers

Subscripts

ex	experimental
in	inlet
out	outlet
eff	efficiency
w	water

this study addressed the influence of the flow channel geometry and location of the finned tubes on the performance of the heat exchanger. In comparison with a conventional design, the new heat exchanger is very suitable for gas-circulation systems [14]. It is compact and easy to install.

2. Experimental setup

Flow field experiments were performed on the fast axial flow (FAF) CO_2 laser shown in Fig. 1. Its gas-circulation system has a typical T-type structure. The experimental system consists of a turbo blower, discharge tubes, heat exchanger, and air-current channel. The working gas is driven by the turbo blower and flows through the discharge tube. The heat generated by the electrical discharge is absorbed by the working gas and transported across to the main heat exchanger by means of convection. The main heat exchanger was the subject of the test. To measure the pressure loss across the test section, nine pressure taps at the inlets and outlets were connected to pressure sensors with an accuracy of 0.5 Pa. The temperatures of the inlet and outlet gas were measured by using temperature transducers. The information thus obtained was displayed on the control panel (CP3000 developed by Convergent Laser Technologies Co., Ltd.). During the experiment, the laser was operated at its rated power.

3. Model description and methods

Fig. 2(a) shows the geometry of the heat exchanger. The main heat exchanger consists of two symmetrical heat-exchanger devices. Each has four rows of tubes along the direction of flow. They are joined together by a long gas passage. Table 1 lists the geometric values.

Fig. 2(b) and (c) shows front and side views of the computational domain, respectively. Due to the symmetry, the area in Fig. 2(a) is selected as the computational domain, and the centric surfaces of the two neighboring fins are taken as the upper and lower boundaries of the computational domain. The actual computational domain is extended by an amount equal to 15 times the fin spacing from the inlet to ensure a uniform velocity distribution. At the outlet, the domain is extended by an amount equal to 45 times the fin spacing to avoid recirculation. The extended regions are indicated by a red[1] line.

3.1. Governing equations and boundary conditions

The active medium of the FAF CO_2 laser is composed of CO_2, N_2, and He, with a molar ratio of 5:29:66. The pressure of the working gas is about 12 kPa. The fluid is considered incompressible with constant physical properties. Due to the high speed of the gas, the gas fluid in the flow channel was assumed to be turbulent. Therefore, the RNG $k-\varepsilon$ turbulence model was used. In addition, the fin thickness and heat conduction in the tubes and fins were also considered. The temperature of the fins was determined by solving the conjugate heat transfer problem in the fin surface region.

The governing equations can be expressed as follows:

$$\text{Ideal gas equation}: \frac{1}{p}\frac{\partial p}{\partial x} - \frac{1}{T}\frac{\partial T}{\partial x} - \frac{1}{\rho}\frac{\partial \rho}{\partial x} = 0 \tag{1}$$

$$\text{Continuity equation}: \frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{2}$$

$$\text{Energy equation}: \frac{\partial}{\partial x_j}[u_i(\rho E + P)] = \frac{\partial}{\partial x_j}\left(\lambda_{eff}\frac{\partial T}{\partial x_i}\right) \tag{3}$$

$$\text{Momentum equation}: \frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu\frac{\partial u_i}{\partial x_j} - \rho u_i u_j\right) \tag{4}$$

In the RNG $k-\varepsilon$ turbulence model:

$$\text{Kinetic energy}: \frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\sigma_k \mu_{eff}\frac{\partial k}{\partial x_j}\right] + G_k + \rho\varepsilon \tag{5}$$

[1] For interpretation of color in Figs. 2 and 11, the reader is referred to the web version of this article.