International Communications in Heat and Mass Transfer 39 (2012) 1054–1058

Contents lists available at SciVerse ScienceDirect

International Communications in Heat and Mass Transfer

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

Experimental investigation of thermoelectric power generation versus coolant pumping power in a microchannel heat sink☆

A. Rezania *, L.A. Rosendahl, S.J. Andreasen

Department of Energy Technology, Aalborg University, DK-9220 Aalborg, Denmark

ARTICLE INFO

Available online 20 July 2012

Keywords:
Thermoelectric generators
Pumping power
Microchannel heat sink
Experimental investigation

ABSTRACT

The coolant heat sinks in thermoelectric generators (TEG) play an important role in order to power generation in the energy systems. This paper explores the effective pumping power required for the TEGs cooling at five temperature difference of the hot and cold sides of the TEG. In addition, the temperature distribution and the pressure drop in sample microchannels are considered at four sample coolant flow rates. The heat sink contains twenty plate-fin microchannels with hydraulic diameter equal to 0.93 mm. The experimental results show that there is a unique flow rate that gives maximum net-power in the system at the each temperature difference.

1. Introduction

In the power generation systems, a key factor is the optimization of the systems design, together with its heat source and heat sink. In the case of the thermoelectric generators (TEG), for increasing the convective surface area, in order to provide high density of heat dissipation at the surface of the structure, microscale heat transfer systems can enhance the thermal coupling to the hot and cold reservoirs [1]. Therefore, the challenge is to design an effective heat exchanger within microelectronic dimension restrictions [2]. Microscale single-phase heat transfer has been widely used in industrial and scientific applications [3]. Using microchannel heat sinks provides low weight and compact energy systems, compared to the traditional macroscale heat sinks, and increases modularity. In contrast to macrochannels, a reduced flow rate in the microchannel heat sink is sufficient to maintain the same average temperature difference between the hot and cold sides of the TEGs. Using microchannel heat sinks also increases the Nusselt number in the channels [4].

In microchannels, the convective heat transfer increases at the high relative roughness of the channel walls, so that the critical Reynolds number changes with the wall roughness in a different way compared to the typical macrochannels. There are observations of earlier transition from laminar to turbulent flow regime in microchannels compared to the macrochannels theory [5,6]. The earlier transition happens at lower Reynolds number when the hydraulic diameter of the microchannel decreases [7,8]. The studies indicate that, the geometric configuration of the microchannel heat sinks has a critical effect on the convective heat transfer of the laminar flow in the heat sink [9]. In order to achieve overall heat transfer enhancement a rectangular microchannel is the best shape, and its heat transfer coefficient is the highest amongst trapezoidal and triangular shaped microchannels [10].

A comparison of thermal efficiencies is conducted with triangular, rectangular and trapezoidal microchannels with the same hydraulic diameter by Chen et al. [11]. Their results show that the cross-sectional shape has a significant effect on the temperature distribution of heat sinks, and the lowest pumping power is required for the triangular microchannel heat sink. Kroeker et al. [12] found that, on the basis of equal hydraulic diameter and equal Reynolds number, a rectangular channel has less thermal resistance compared with heat sinks with circular channel. Additionally, the laminar friction factor or flow resistance reaches a minimum value as the channel aspect ratio approaches 0.5 [9]. As reported by Tuckerman and Pease [13], when the channel width and the fin thickness are equal, the heat sink gives the minimum thermal resistance. The inlet/outlet plenum effect gives a non-uniform velocity and flow rate distribution in each channel under a given pressure drop in the heat sink, so that a non-uniform temperature distribution happens in the heat sink [14]. The heat sinks with the vertical coolant supply and collection via inlet and outlet ports of the heat sink provide better uniformities in temperature and velocity, so that this type of heat sink can make better performance due to smaller thermal resistance among the studied heat sinks.

The heat exchanger designs have mostly disconnected to the higher performance TEGs. The development work mostly focused on thermoelectric materials required a significant amount of engineering parametric analysis. Since, the ZT of the thermoelectric materials is not the only factor to improve the output power of the system, the whole parts of the energy conversion system, such as thermal contacts of the hot and cold heat exchangers, needed to be considered. The thermal resistances of the heat exchangers have a strong influence on the

☆ Communicated by W.J. Minkowycz.

* Corresponding author.
E-mail address: alr@et.aau.dk (A. Rezania).

0735-1933/$ – see front matter © 2012 Elsevier Ltd. All rights reserved.
doi:10.1016/j.icheatmasstransfer.2012.07.010

Nomenclature

A	TEG leg area, m^2
C_p	specific heat of water, 4178 J/kg.K
D	diameter, m
D_h	hydraulic diameter of the channel, m
H	height, m
I	current, A
L	length, m
$\dot{m}$	mass flow rate, kg/s
P	power, W
p	pressure, Pa
Pr	Prandtl number
Re	average Reynolds number
Q	heat absorbed by heat sink, W
T	temperature, K
t	heat sink wall thickness, m
u	coolant flow velocity in the microchannel, m/s
V	voltage, V
$\dot{V}$	volumetric flow rate, m^3/s
w	width, m
x_h	thermal entry length, m

Greek symbols

α	Seebeck coefficients, V/K
Δ	value difference
μ	water dynamic viscosity, N.s/m^2
ρ	water density, kg/m^3

Subscripts

av	average
ch	microchannel
hs	heat sink
i	inlet
n	net
o	outlet
p	pump
pl	plenum
teg	thermoelectric generator

electric power produced by the TEG. Martinez et al. [15] optimized the heat exchangers of a TEG in order to maximize the electric power generated by the TEG. In addition, in order to decrease the coolant pumping power in the TEG systems, an effective design of the microchannel heat sinks is proposed and implemented in a three-dimensional TEG model [16]. The small thermal conductivity of the thermoelectric materials causes the temperature difference between the cold and the hot surfaces of TEG not to change remarkably when the coolant flow rate varies in the heat sink. Due to the Seebeck effect in the TEG, which transfers a certain amount of energy from high-temperature fluid to electricity, the linear variations in temperature of the fluids in the TEG system are different from the logarithmic variation case in the ordinary parallel-plate heat exchanger [17]. Therefore, the heat flow from the hot source is not equal to the heat flow to the low temperature fluid. Further analysis of the influence of coolant flow rate, heat exchanger geometry, and fluid inlet temperature on the power generation by the TEGs is done by Esartea et al. [18].

Nonetheless, when it comes to real-world design of thermoelectric systems for direct thermal to electricity conversion, a narrow discussion of the TEGs integrated to heat exchangers in micro scale is still lacking. Wojtas et al. [19] reported a complete integration of the microfluidic heat transfer system with a micro-TEG, where the thermoelectric Seebeck voltage is measured as a function of the temperature gradient and applied fluid flow rate. Their demonstration enables us to increase the output power performance of the micro-TEGs. The thermoelectric Seebeck voltage is measured as a function of the applied flow rate and temperature gradient. In this work, a micro plate-fin heat sink that is made of aluminum is applied to a TEG. The purpose is considering the power generation versus the pumping power, which is produced by the pressure loss of the coolant fluid in the system. The particular focus of this experimental study is, exploring the optimum coolant flow rate that results the maximum net-power in the system, at a wide average temperature difference range of the hot and cold sides of the TEG ($\Delta T_{teg,av}$). By considering the average temperature of the hot surfaces of the TEG legs, these optimum points are suggested as the optimum applied pressure drop in the microchannel heat sink by Rezania and Rosendahl [20]. In addition, the temperature distribution and pressure loss in some microchannels with variation of the coolant flow rate and heat flux are reported. Fig. 1 illustrates the schematic, thermocouple and pressure gages location, and the setup of the experiments.

2. Theoretical background

The heat transfer rate removed by the coolant flow in the heat sink is [21]:

$$Q = \dot{m}C_p(T_o - T_i). \tag{1}$$

The pumping power to circulate the coolant flow in the heat sink is related to the pressure drop and the volumetric flow rate, and can be calculated as follows:

$$P_{pump} = \dot{V}\Delta p = \dot{V}(p_o - p_i). \tag{2}$$

The thermal entry length of the internal flow at the laminar regime is approximated by [22]:

$$x_h \approx 0.05 D_h \mathrm{Re}\mathrm{Pr}. \tag{3}$$

where D_h is the hydraulic diameter of a rectangular cross section shape channel defined as follows:

$$D_h = \frac{2wH}{w+H}. \tag{4}$$

The Reynolds number in the channels is [23]:

$$\mathrm{Re} = \frac{\rho u D_h}{\mu}. \tag{5}$$

In this study, the effect of the temperature variation on the thermo-fluid properties of the coolant flow is taken into account. In the TEGs, if the hot and cold junctions are maintained at different temperatures, an open-circuit electromotive force develops between them, and is given by $V = \propto \Delta T$, which defines the differential Seebeck coefficient between the two sides of the elements. Therefore, the power generation is as follows [24]:

$$P_{teg} = IV = \propto^2 \sigma \times \Delta T_{teg}{}^2 \times \frac{A}{L_{teg}}, \tag{6}$$

where $\propto^2\sigma$ is the power factor of the thermoelectric materials. We define the net-power in the system as follows:

$$P_n = P_{teg} - P_p. \tag{6}$$