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Experimental and computational study on thermoelectric generators using thermosyphons with phase change as heat exchangers

M. Araiz, A. Martínez *, D. Astrain, P. Aranguren

Mechanical, Energy and Materials Engineering Department, Public University of Navarre, 31006 Pamplona, Spain
Smart Cities Institute, 31006 Pamplona, Spain

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ABSTRACT

An important issue in thermoelectric generators is the thermal design of the heat exchangers since it can improve their performance by increasing the heat absorbed or dissipated by the thermoelectric modules. Due to its several advantages, compared to conventional dissipation systems, a thermosyphon heat exchanger with phase change is proposed to be placed on the cold side of thermoelectric generators. Some of these advantages are: high heat-transfer rates; absence of moving parts and lack of auxiliary consumption (because fans or pumps are not required); and the fact that these systems are wickless. A computational model is developed to design and predict the behaviour of this heat exchangers. Furthermore, a prototype has been built and tested in order to demonstrate its performance and validate the computational model. The model predicts the thermal resistance of the heat exchanger with a relative error in the interval $[-8.09; 7.83]$ in the 95% of the cases. Finally, the use of thermosyphons with phase change in thermoelectric generators has been studied in a waste-heat recovery application, stating that including them on the cold side of the generators improves the net thermoelectric production by 36% compared to that obtained with finned dissipators under forced convection.

1. Introduction

The current energy situation, characterised by an increase in the energy consumption and the dependency on fossil fuels, has led several researches in order to improve the efficiency of the processes or to develop different ways for energy production, using, for instance, renewable sources. In this sense, thermoelectric generators (TEGs) can be used to produce electric energy from waste heat that, in other case, would be released to the ambient. This would increase the efficiency while using a free source of energy.

TEGs are made up of thermoelectric modules (TEM) based on the Seebeck effect, which are in charge of the transformation of heat into electricity; and heat exchangers, whose aim is to improve the efficiency of these devices. These exchangers have as purpose the reduction of the thermal resistances between the heat source and the hot side of the thermoelectric modules as well as that between the cold side of the modules and the ambient. In this way, the temperature difference between the hot and the cold side of the thermoelectric modules gets close to the maximum temperature gradient possible, which is the temperature difference between the heat source and the ambient, increasing the efficiency of the generator. The need to reduce these thermal resistances and to optimize the design of heat exchangers has been already proven [1,2].

A wide range of heat exchangers can be placed at the cold side of the modules in order to reduce the thermal resistance between this cold side and the ambient. Finned heat sinks are extensively used due to their simplicity as well as their relative low cost compared to other kind of heat exchangers. Working as active cooling systems, these dissipators can reach high cooling power rates [3–5]. Other kind of heat exchangers are the ones based on a liquid, such as water [6,7], which increase the performance of the system as they have higher convective coefficients. In these cases, there is an auxiliary consumption due to the electric power required to feed both the fans that make the air pass through the fins or the pumps that move the liquid inside the system [8].

Nowadays, there is a deep research in heat exchangers with phase change that improve heat transfer even more, with a small temperature drop, due to the use of the latent heat of an internal fluid [9]. Heat pipes are the most common dissipators used of this range. They have an evaporator in contact with the cold side of the TEMs to absorb the heat that needs to be dissipated. This absorbed heat evaporates the fluid inside, which flows up to the condenser. Once up there, it condensates, as it releases the heat to the ambi-

* Corresponding author at: Mechanical, Energy and Materials Engineering Department, Public University of Navarre, 31006 Pamplona, Spain.
E-mail address: alvaro.martinez@unavarra.es (A. Martínez).

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Nomenclature

A_b	base area of the evaporator (m^2)
A_{im}	base area of the interface material (m^2)
A_m	base area of one thermoelectric module (m^2)
Bi	Biot number
b_i	systematic uncertainty for the experimental variable i
c_{pl}	liquid specific heat capacity (J/kg K)
d_e	tube external diameter (m)
d_i	tube inside diameter (m)
D	equivalent fin diameter (m)
e	Wall's evaporator's base thickness (m)
e_{im}	interface material thickness (m)
g	acceleration due to gravity (m/s^2)
h_b	boiling heat transfer coefficient (W/m^2 K)
h_{cI}	condensation heat transfer coefficient with turbulent flow (W/m^2 K)
h_{cII}	condensation heat transfer coefficient with laminar flow (W/m^2 K)
H	fin height (m)
I_{ps}	electric current supplied to the electric resistances (A)
i_{lg}	latent heat of vaporization (J/kg)
k_{Al}	thermal conductivity of the aluminium (W/m K)
k_{im}	thermal conductivity of the interface material (W/m K)
k_l	liquid thermal conductivity (W/m K)
L	condensation tube's length (m)
N	number of thermoelectric modules
Nu	Nusselt number
P_{max}	maximum electric power generated with TEG (W)
p_r	reduced pressure (P/P_c)
Pr_l	Prandtl number
$\dot{Q}_c$	heat flux dissipated by the heat exchanger (W)
R_b	boiling thermal resistance (K/W)
R_c	condensation thermal resistance (K/W)
R_{cold}	thermal resistance of the cold side of the TEG (K/W)
R_{cond_b}	conduction thermal resistance through the evaporator's base (K/W)
R_{cond_t}	conduction thermal resistance through the condensation tubes' wall (K/W)
R_{const}	constriction thermal resistance (K/W)
R_{cont}	contact thermal resistance (K/W)
R_{conv}	natural convection thermal resistance (K/W)
R_{HE}	thermal resistance of heat exchanger per module (K/W)
R_{hot}	thermal resistance of the hot side of the TEG (K/W)
Ra	Rayleigh number
Re_{LT}	Reynolds number assuming total mass flowing as liquid
Re_{LS}	Reynolds number assuming liquid phase flowing alone
T_{amb}	ambient temperature (K)
T_c	temperature on the cold side of the thermoelectric modules (K)
$T_{out}^{wall}, T_{in}^{wall}$	outside and inside evaporator's wall temperature (K)
$T_{out}^{tube}, T_{in}^{tube}$	outside and inside tube's wall temperature (K)
T_{sat}	saturation temperature (K)
V_{ps}	voltage supplied to the electric resistances (V)
x	vapour quality

Greek symbols

α	thermal diffusivity (m^2/s)
ΔP_{sat}	difference in saturation pressure corresponding to ΔT_{sat} (Pa)
ΔT_{sat}	difference between wall and saturation temperature (K)
β	coefficient of thermal expansion (K^{-1})
γ	parameter given by Eq. (13)
δ	occupancy ratio
ε	dimensionless contact radius, $\sqrt{\delta}$
λ_c	empirical parameter given by Eq. (5)
μ_g	gas viscosity (N s/m^2)
μ_l	liquid viscosity (N s/m^2)
ν	kinematic viscosity (m^2/s)
ρ_g	gas density (kg/m^3)
ρ_l	liquid density (kg/m^3)
σ	surface tension (N/m)
τ	dimensionless wall thickness, $e/\sqrt{A_b \pi}$
Ψ	dimensionless constriction thermal resistance

ent, and it returns back to the evaporator. Some TEGs applications use heat pipes with a fan to help the heat to be transferred from the condenser to the ambient [10] whereas others take advantage of the free convection to remove the fan auxiliary consumption [11]. In both cases, fluid flow is achieved without pumps just by capillary effect of the wick which is inside the evaporator. This allows a heat pipe to work in any orientation but introduces two drawbacks: the capillary pressure caused by the wick may not be enough to pump the liquid back to the evaporator and an extra thermal resistance must be taken into account due to the conduction through the wick [9].

In order to avoid these inconveniences, this work presents the study of a wickless heat pipe applied to a TEG. Instead of using capillary effect, this device uses the thermosyphon effect, which takes advantage of density differences and gravity force to make the fluid flow inside the exchanger. It does not require any fan or pump to work; it just needs the evaporator to be below the condenser to allow the fluid drain back once it has been condensed. A thermosyphon with phase change (TSP) has-like a heat pipe-an evaporator, a vapour line, a condenser (composed by several finned tubes for the natural convection) and a liquid return line.

The main goal of this work is to design thermosyphons with phase change and study their performance as cold-side heat exchangers in TEGs, which, as they do not need any auxiliary consumption, could improve the net generation of electric energy.

For that, Section 2 describes a computational model that has been developed to predict the performance of thermosyphons with phase change. Section 3 presents a prototype designed, built and tested to prove the performance of these systems and to validate the model. Section 4 shows the experimental and simulated results obtained from the prototype and the computational model respectively. Section 5 presents a parametric study conducted for design and optimization of this kind of heat exchangers, and also a study of the improvement in the production of electricity using TEGs for waste heat harvesting from a chimney. A comparison has been made between the use of finned dissipaters and thermosyphons as cold-side heat exchangers. Finally, Section 6 collects the main conclusions of this work.

2. Computational model

To study the use of thermosyphons with phase change in thermoelectric generators is necessary to, firstly, develop a computational model able to predict the behaviour of these heat exchangers. The model must simulate their performance with enough accuracy and convergence rate, allowing the modification of dozens of parameters. This cannot be achieved using a CFD software [6], due to its high computational cost, even when modifying one single parameter. Because of that, the finite-differences implicit method has been employed, which has been proven to be useful