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THEORETICAL AND EXPERIMENTAL STUDY OF THERMAL CONTACT RESISTANCE EFFECT ON THERMOELECTRIC GENERATION USING PELTIER MODULES

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Abstract. *To meet the growing global demand for energy, new forms of power generation are of extreme importance, as well as increasing the efficiency of existing methods. The present study intends to model analytically the heat transfer through a thermoelectric generator (TEG) comprised by a Peltier module. The thermoelectric generation from these modules have promising future prospects, as they are light, small and targets of continuous improvement. Was designed the implementation of a TEG in a vehicular exhaust for recovery of thermal energy present in the exhausting gases. Aiming this application, many variables shown to be relevant: flow characteristics, use of heat sinks, their geometry and composition and, mainly, the description of the interface between the surfaces of the module and the heat sinks. In this interface, the contact thermal resistance is a function of mounting pressure, surface roughness and the presence of interstitial material. After modeling the heat transfer, was implemented a numerical code to assess the sensitivity of the output voltage and contact resistance of the module with changes in the TEG operating conditions. Furthermore, to evaluate the simulated results, was developed an experimental bench reproducing TEG operating conditions, under the effect of different temperatures, flow velocities, contact pressures, roughness and interstitial material. With the use of only one module, the theoretical values for the output voltage was in the range of 0.92V and 0.97V for the worst and best case respectively, and the values of experimental measurements under the same operating conditions provided 0.75V and 1.07V. The discrepancy found between the theoretical and experimental approach resulted in a maximum relative error of 22.7%. The best output voltage values were obtained when minimized the thermal contact resistance, indicating that this is a way to increase the efficiency of a TEG and help meet the current demand of energy.*

Keywords: *Peltier module, thermoelectric generation, thermal contact resistance*

1. INTRODUCTION

The demand for electricity has increased over the years (Statistical Yearbook of Energy, 2013) as well as the concern and importance to find more efficient means of power generation and less affecting the environment. Much of the energy generated comes from non-renewable sources, applied on inefficient thermodynamic processes, for example, gas turbines, internal combustion engines and power plants have usual efficiencies ranging from 25% to 60%, that is, there is much wasted potential.

According to the U.S. Energy Information Administration, the average world consumption of energy is 18TW, with the production from fossil fuels representing 86% (15.48 TW). If an average efficiency of 40% is considered, the amount of wasted energy is 9.3 TW considering only this mean of generation.

To illustrate the amount the heat rejected in these processes, the USA energy chain (Lawrence Livermore National Laboratory, 2013) can be used, where it is shown that the rejected energy is approximately 57% of the total generated energy.

The challenge becomes, then, to find a way to use the rejected heat and that does not affect the environment when in operation. Among several alternatives, thermoelectric modules have great potential for waste heat recovery for power generation, as they are compact, lightweight and durable and can be used from power plants to automotive vehicles, offering the potential to increase the overall efficiency of the system.

These modules are based on the Seebeck effect, in which an electromotive force is generated at the junction between two thermoelements of different conductive materials from their heating, where the greater the difference between the temperatures of the hot side, Th, and cold, Tc, the higher the output voltage

$$V = \alpha(T_h - T_c) \tag{1}$$

Where α is the Seebeck differential coefficient between the pair of thermocouples making up the module, measured in V / K. Because of this behavior, these modules are elements very propitious for being inserted into pipes, in which flows transporting heat not used in thermodynamic cycles passes. Fig. 1 shows in detail the constructive features of a generic module

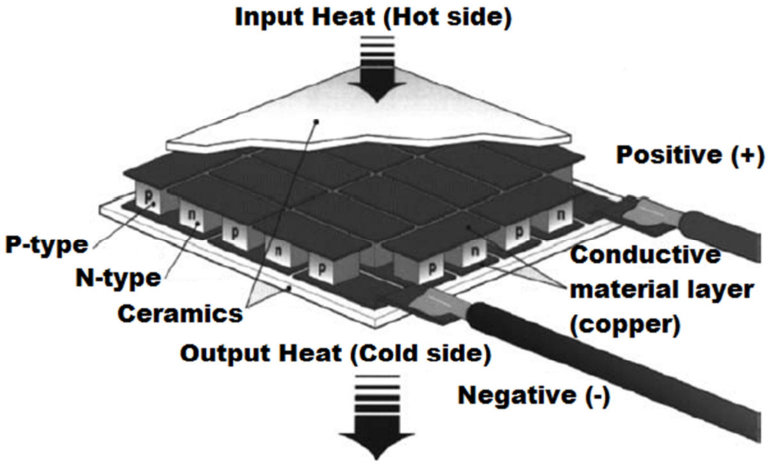

Figure 1. Detailed diagram of the construction of a Peltier module (Rowe, 2006)

Currently, the efficiency of a common Peltier module ranges from 6% to 11% (Rowe, 2006), so, in practice, are used several modules in series that, adding up the output power generated, results in more significant amounts.

2. THERMOELECTRIC GENERATOR

Given that the power generated depends on the temperature difference between the hot and cold side of the module, this study was proposed to analyze the active heat transfer in the same and, therefore, was designed a thermoelectric generator (TEG). It consists of a Peltier module and a pair of aluminum fins that aim to increase the heat transfer (Çengel, 2006), shown in Fig. 2, where N and P elements represent the two thermocouples of which the module consists

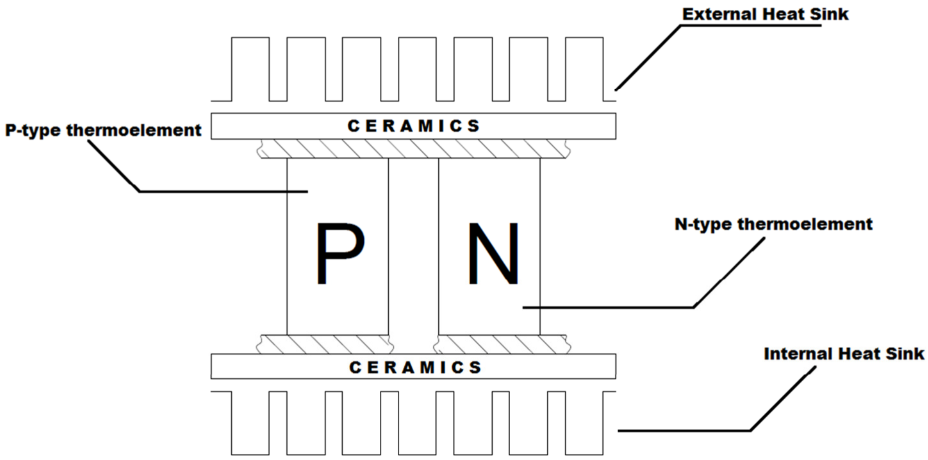

Figure 2. Schematic drawing of a pair of rectangular heat sinks installed on each side of a module, thereby forming a TEG

Three heat transfer modes operate in the TEG, conduction, convection and radiation; however, in situations considered in this study, the heat transferred by radiation is of inferior order than the other two and, therefore, was not included. The heat flow, by conduction, convection and the two combined are defined in Eqs. 2, 3 and 4, given in W/m^2

$$\dot{Q}_{cond} = -k\frac{dT}{l} \tag{2}$$

$$\dot{Q}_{conv} = h(T_w - T_\infty) \tag{3}$$

$$\dot{Q}_{total} = \dot{Q}_{cond} + \dot{Q}_{conv} \tag{4}$$

Where k is the thermal conductivity of the material, dT is the temperature gradient in the general direction, l is the thickness of the material in which there is a conductive heat flux, Tw is the temperature at the surface and T∞ is the temperature of the medium.

Due to flat geometry of the module, the TEG was installed in a duct of rectangular section and, for this configuration, Nusselt numbers to determine the heat transfer coefficient by convection of the outer and inner flows were defined based in approaches of Churchill and Ozoe (1973) and Jensen and Vlakancic (1997), respectively.

So, for the external flow,

$$Re_{ext} = \frac{\rho U_{ext} X}{\mu_{ext}} \tag{5}$$

$$Nu_{ext} = \frac{\left[0{,}3387 Pr^{1/3} {Re_{ext}}^{1/2}\right]}{\left[1 + \left(\frac{0{,}0468}{Pr}\right)^{2/3}\right]^{1/4}} \tag{6}$$

$$h_{ext} = \frac{Nu_{ext} k}{X} \tag{7}$$

Where Re is the Reynolds number, ρ is the specific mass of the fluid (kg/m3), U is the flow velocity (m/s), μ is the dynamic viscosity (Pa.s), Pr is the Prandtl number, Nu is the Nusselt number h is the convection coefficient (W/K.m2).

For the internal flow, as the section of the duct has non-cylindrical geometry, the use of a hydraulic diameter, D_h, is necessary. Being the TEG's cross section rectangular, of X width and Y height, the nominal area, A_n, is

$$A_n = XY \tag{8}$$

Thus,

$$D_h = \frac{4A_n}{perímetro} = \frac{4XY}{2(X+Y)} = \frac{2XY}{(X+Y)} \tag{9}$$

$$Re_{int} = \frac{\rho U_{int} D_h}{\mu_{int}} \tag{10}$$

$$A_{real} = A_n - NLt \tag{11}$$

Where N, L and t Where C, G and T are the number of fins, the height and width of the fins, respectively.

According to Jensen and Vlakancic (1997), this approach can be used only if the dimensionless height of the fin, J, is greater than 0.06

$$J = \frac{2t}{Y} \geq 0{,}06 \tag{12}$$

The internal friction coefficient is

$$f_{geométrico} = \left(\frac{AS_{real}}{AS_n}\right)^{0{,}29} \tag{13}$$

Where real AS_{real} and AS_n represent the real and nominal areas of the inner surface of the duct, respectively,

$$AS_{real} = A_{aleta} + A_{na} \tag{14}$$

A_{aleta} is the total surface area of the fins and A_{na} is the total area of the spaces between the fins.

$$A_{na} = X^2 - Nxt \tag{15}$$

$$A_{aleta} = 2NXL \tag{16}$$

$$AS_n = 2XY + 2X^2 \tag{17}$$

And, finally,

$$\frac{Nu_{int}}{Nu_{st}} = \left(\frac{X}{Y}\right)^{-1/2} \left(\frac{A_n}{A_{real}}\right)^{0{,}8} f_{geometrico} \tag{18}$$

Where Nu_{st} is the Nusselt number for the duct without fins, defined by Gnielinski (1976),

$$Nu_{st} = 0{,}012 Pr^n [Re_{int}^{0{,}87} - 280] \tag{19}$$

Where $n = 0{,}4$, because the air is warming, as defined by Dittus-Boelter equation (Çengel, 2006). Found the values for Nu_{int} and D_h, is defined the internal convection coefficient as

$$h_{int} = \frac{Nu_{int} k}{D_h} \tag{20}$$

In addition to quantifying the heat flows, a detailed study of heat transfer that occurs in the TEG is essential to increase its efficiency. This may be achieved by studying the thermal resistance circuit shown in Fig. 3

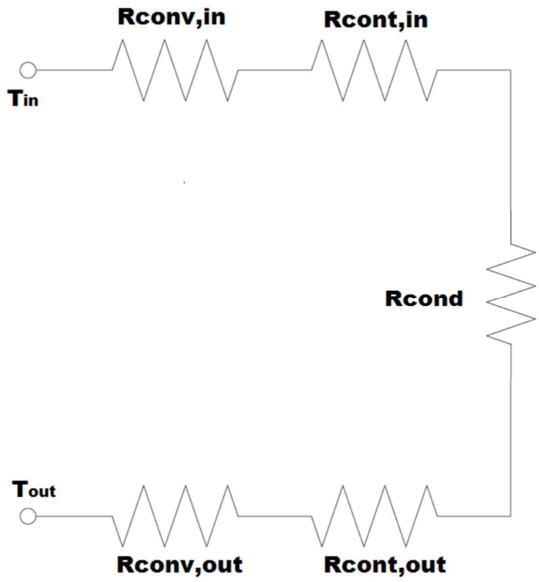

Figure 3. Thermal resistance circuit acting on TEG.

The total thermal resistance of the TEG is given by Eq. 21 and has the unit K/W

$$R_{eq} = R_{conv,int} + R_{cont,int} + R_{cond} + R_{cont,ext} + R_{conv,ext} \tag{21}$$

The conduction and convection resistances are, respectively

$$R_{conv,int} = \frac{1}{h_{int} A S_{real}} \tag{22}$$

$$R_{conv,ext} = \frac{1}{h_{ext} A S_{real}} \tag{23}$$

$$R_{cond} = \frac{H}{k_{modulo} A_{modulo}} \tag{24}$$

Where H is the width of de module.

In the heat conduction analysis on the interface between two solid layers, there is a resistance generated by the roughness of the surfaces, because although the same appear flat and smooth to the naked eye, they appear quite rough if analyzed under a microscope, with numerous peaks and valleys. When these surfaces are pressed against one another, the peaks form good material contact, but the valleys form voids filled with air and, these gaps, due to the low thermal conductivity of air, act as a thermal insulator. Thus, the interface provides some resistance to heat flow, and this resistance per area is the thermal contact resistance, R_{cont}

$$R_{cont} = \frac{1}{h_c} = \frac{\Delta T_{interface}}{\dot{Q}/A} \tag{25}$$

As can be noted, the conductive and convective resistances depend only on the characteristics of the flows and physical properties of the components. Thus, the resistance that can be optimized is the contact thermal resistance, which is a function of contact pressure, interstitial material, and surface roughness, but there is not a governing equation to define it (Çengel, 2006), requiring an empirical relationship to be is established.

There is great uncertainty of the thermal conductance values, h_c, reported in literature, so, for this study, the work of Barzelay et. al (1955) was used as reference, in which there are curves for contact conductance interfaces between aluminum plates with different contact pressure and surface roughness.

From three graphs h_c x P, each for a different value of surface roughness, were read h_c values for three pressures, generating three equations of type

$$h_c = aP + b \tag{26}$$

To include the influence of the roughness, the Ai and Bi coefficients of the line equations, where i = 1, 2, 3, were graphed with the roughness values on the ordinate axis and, again, equations were derived for the curves. These equations were then inserted in Eq. 27, that is the equation of the thermal contact conductance used as a reference

$$h_{c,calculado} = [1{,}307r^2 - 5{,}567r + 15{,}33]P + [409{,}94r^2 - 1842r + 3358{,}3] \tag{27}$$

3. EXPERIMENTAL BENCH

In order to compare with the numerical results provided by the numerical implementation of the proposed mathematical model, an experimental bench was designed and built in order to allow the variation of TEG settings, and alter the value of the thermal contact resistance aiming to evaluate the sensitivity of the heat flow has therewith.

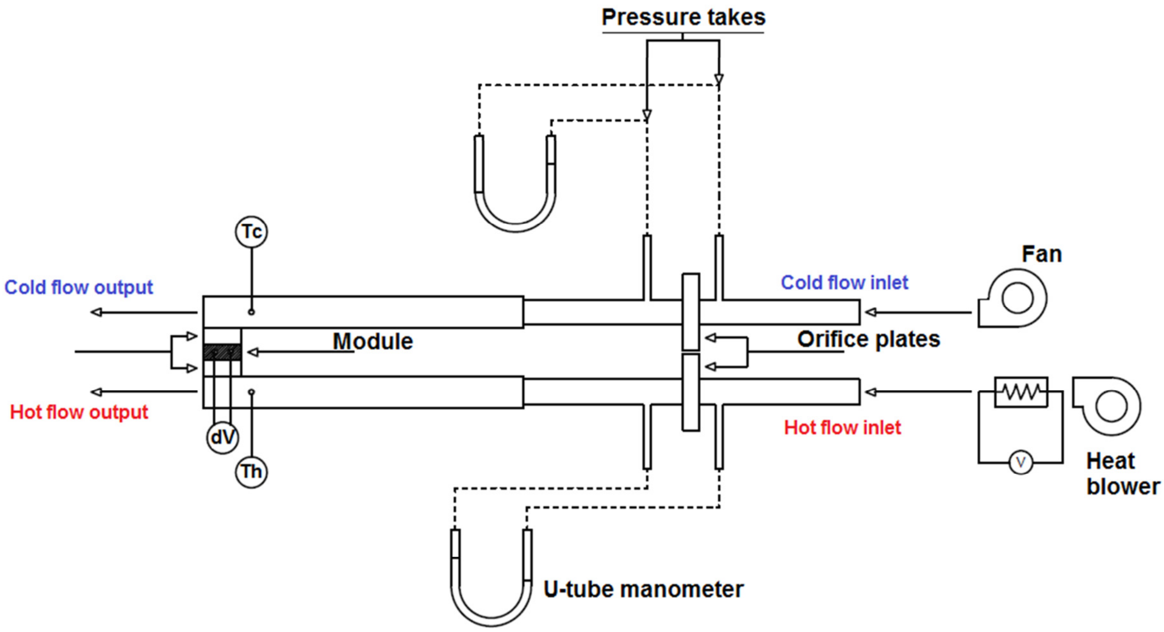

Figure 4. Scheme of the experimental apparatus

The test rig consists of two cylindrical ducts leading the cold temperature and hot temperature flows, in which orifice plates are installed so that is possible to know the flows speeds. Subsequently, the section changes from cylindrical to rectangular, in which are installed the fins and the Peltier module (TEG).

3.1 Experimental procedure

Initially, the fan and heat blower are turned on, and only after the change in temperature of the hot flow reaches steady state, measurements starts.

Data acquisition was made for a range of different configurations of the bench, on the change of parameters that should cause variations in performance of the thermoelectric generator. These settings relate to the following factors:

Relative to the surface treatment of the base of the heat sinks:

- A pair of heat sinks unmodified, used with the same surface finish which were acquired, without any treatment;
- A pair of heat sinks with sanded surface.

Relative to the presence of interstitial material between the base of the heat sink and the thermoelectric module:

- Use of Thermal Grease 1, which have thermal conductivity of 0.4 applied to the interfaces of the base of the fins and the module
- Use of Thermal Grease 2, which have thermal conductivity of 1.2 applied to the interfaces of the base of the fins and the module

Relative to the application of additional pressure on the thermoelectric generator:

- Without applying additional pressure on the generator, i.e., the only pressure acting is the own weight of the structure of the experimental apparatus. This measure will be taken as a measure of zero relative pressure, serving as a parameter for cases with additional pressure;

- Application of additional 62052 Pa (9 psi) on the thermoelectric generator;
- Application of additional 124106 Pa (18 psi) on the thermoelectric generator;

4. RESULTS

The voltage values achieved in the presence of different interstitial materials, with different contact pressures and with different surface roughness are arranged in graphs shown in Figs. 5, 6 and 7. The rise in output voltage with application of thermal grease is consistent with the anticipated by the theory, the inserted material fills the air gaps, increasing the thermal conductance of the contact (Fried, 1962), whereas these greases have a higher thermal conductivity than air.

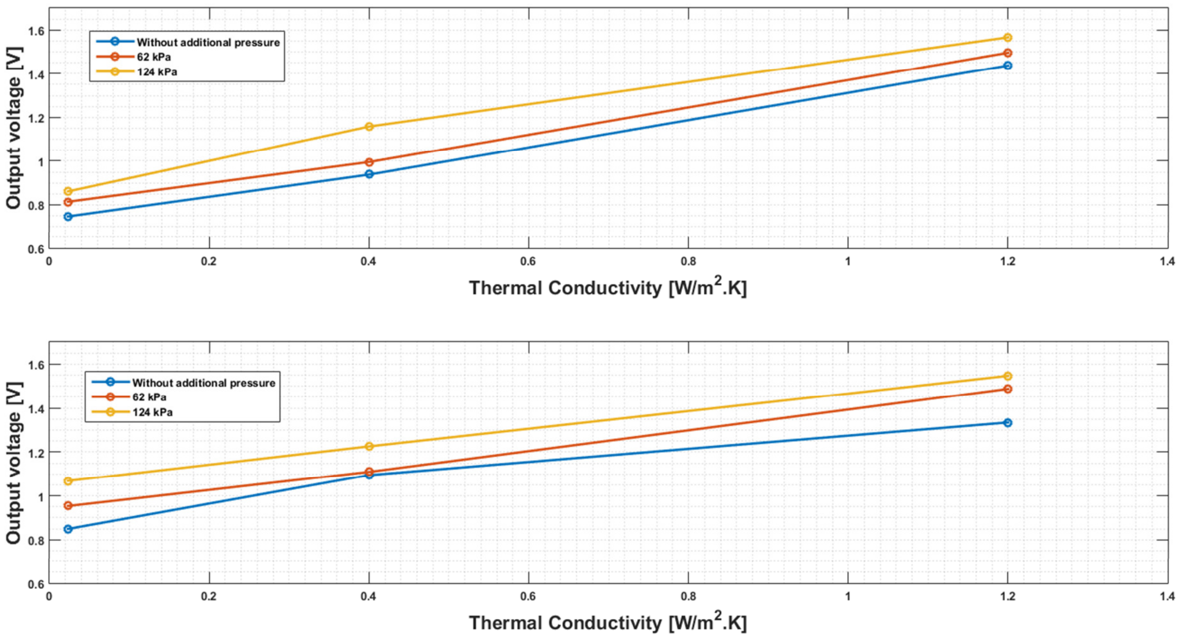

Figure 5. Output voltage values for Pair 1 (top) and pair 2 (bottom) of sinks in the case where there is no interstitial material, with Thermal Grease 1, and 2 Thermal Grease

The variation of contact pressure causes much more subtle increases in output voltage, which indicates a trend that, for a very high pressure, there will be no difference in the reduction of the thermal contact resistance (Barzelay et al., 1955).

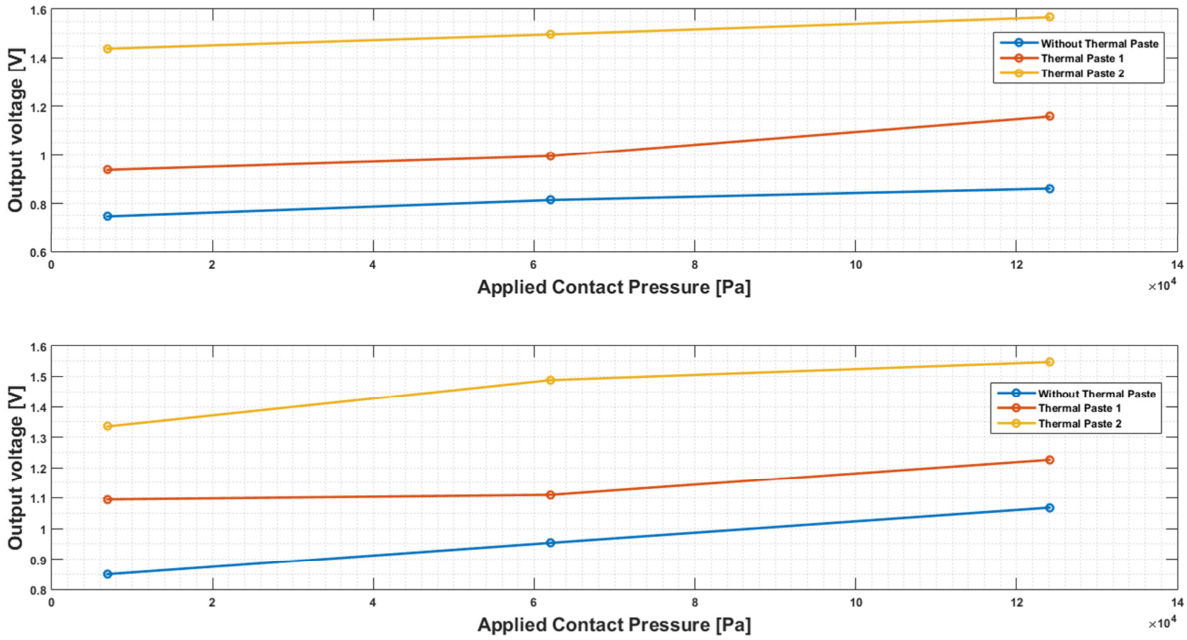

Figure 6. Output voltage values for Pair 1 (top) and Pair 2 (bottom) in cases where there is no additional contact pressure, applying 62 kPa (9 psi), and applying 124 kPa (18 psi)

Data from the literature suggest that a reduction in surface roughness leads to reduction in the contact thermal resistance (Çengel, 2006), increasing the heat flux and hence the output voltage. However, the experimental results are not in agreement with the theory, this can be explained by the fact that the surface roughness has microscopic characteristic, i.e., there is no way to ensure optimal fit between two surfaces, and slight variations in position can generate large variations in resistance. Another important factor is the hardness of the material of which the surfaces are made, i. e., for each material from which the pairs of plates are made of, the same or different from each other, there should be an empirical relationship for thermal conductance.

To compare the output voltages provided by the analytical model with the experimentally measured, only values for the configuration without the presence of interstitial material are shown due to lack of data or equations that preview the variation of the voltage in the presence of the same. Table 1 shows the theoretical and experimental values for each pair of sinks, for all three contact pressure cases considered: no additional pressure, applying 62 kPa and 124 kPa.

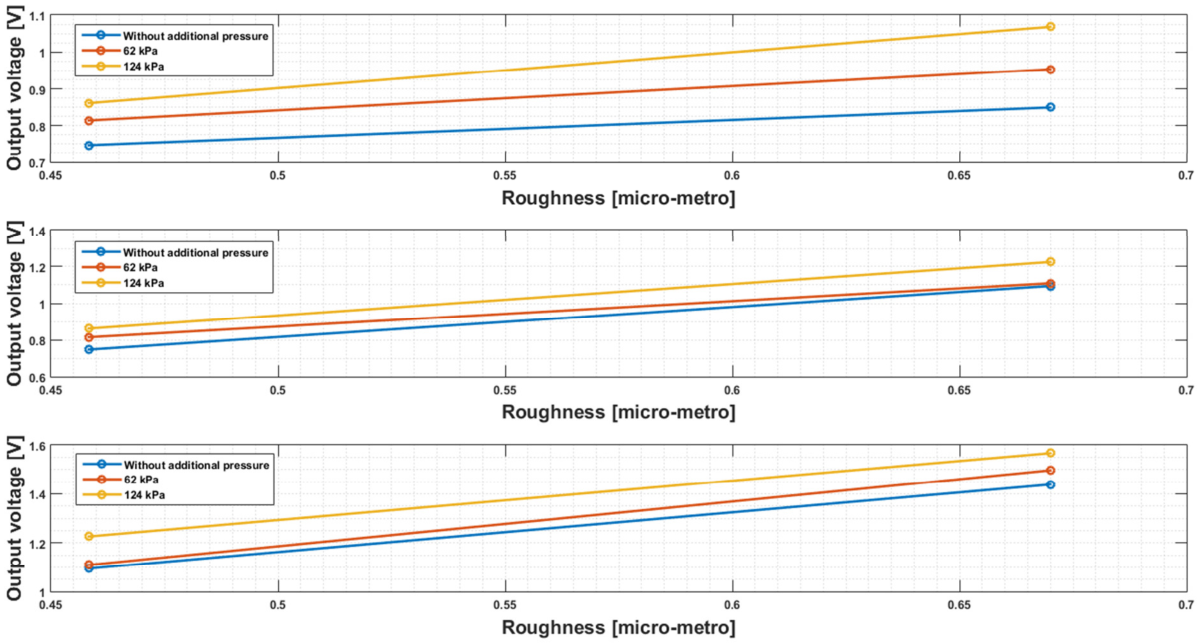

Figure 7. Output voltage values for the cases without interstitial material (first), applying Thermal Grease 1 (second) and applying Thermal Grease 2 (third)

Table 1. Theoretical and experimental output voltage values for the two pairs of heat sinks, without application of interstitial material, and for three contact pressure values

Pair of Sinks 1	**Theoretical Voltage [V]**	**Experimental Voltage [V]**	**|Error| [%]**
Without additional pressure	0.92	0.74	23.96
62052 Pa	0.93	0.81	14.49
124106 Pa	0.94	0.86	9.31
Pair of Sinks 2	**Theoretical Voltage [V]**	**Experimental Voltage [V]**	**|Error| [%]**
Without additional pressure	0.93	0.84	10.38
62052 Pa	0.96	0.95	1.23
124106 Pa	0.96	1.06	10.66

5. CONCLUSION

Set the contact thermal resistance proved to be a hard task because, despite being proven that it depends on the roughness, contact pressure and interstitial material, a theoretical governing equation could not be defined as there is dependence on other factors that could not be assessed in this study, such as variation of the mounting position, which causes a change in fitting the two surfaces, and the material the interfaces between the sinks and the module are composed.

The error sources in the experimental voltage values are due mainly to the rustic character of the experimental bench, because as well as designed, it was built using simple materials, but trying to achieve the best possible result.

To achieve the common goal of finding alternative forms of power generation and increase the efficiency of thermoelectric generators, the study of heat transfer acting in them is essential. Along with improvement of new materials, geometry and operating states, a better understanding of the thermal contact resistance can trigger a significant gain in power generation and consequent use of good quality thermal energy that, otherwise, would be wasted.

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