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Adiabatic Behavior of the Vuilleumier Heat Pump

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Abstract. The functioning of a Vuilleumier heat pump with four variable-volume chambers and six heat exchangers was numerically simulated using a theoretical adiabatic physico-mathematical model. All four variable-volume chambers were assumed as adiabatic and the heat exchangers were considered as isothermal. The model uses the piston movement laws given by the mechanism and not simplified sinusoidal ones. The adiabatic functioning was described through thermodynamic diagrams showing pressure and temperatures cyclical variations inside the machine and inside each adiabatic chamber. The model allows to estimate the heat pump performances, highlighting the energy exchanged by the working agent inside heaters and coolers and the coefficient of performance. The adiabatic model is able to emphasize the energy transfer that appears between the branches of the machine, feature that simpler theoretical models do not have. This analysis of the functioning with the adiabatic model could be extended for a Vuilleumier heat pump that rejects heat at two different temperatures.

1. Introduction

The machine analyzed in the paper was invented by the Swiss-born US American engineer Rudolph Vuilleumier [1] and bears the name of its inventor. This closed cycle gas device functions using three thermal reservoirs and can be used as a heat pump or as a refrigerating / cryogenic machine. Due to some constructive and functional similarities – both are regenerative thermal machines using a succession of heat exchangers comprising a cooler, a regenerator and a heater – the Vuilleumier machine is usually related to the Stirling engine [2]. Theoretical and experimental studies about various aspects of Vuilleumier cycle machines functioning are of topical interest.

Several Vuilleumier heat pumps using natural gas as fuel and helium as working agent and capable of providing various rates of heat flow at output – e.g. 7.5 kW or 20 kW – were developed earlier by professor Carlsen in Denmark [3]. These heat pumps use crank mechanisms for driving the displacers.

The energy and exergy analysis of a Vuilleumier heat pump designed for using industrial waste heat was performed in [4]. The influences of the temperature of the high temperature heater, of the pressure and of the rotation speed over the functioning were studied.

In 2017, a study concerning the influence of the working agent properties over the performances of the Vuilleumier heat pump was published. The heat pump functions using not a single gas as working agent but a mixture of hydrogen, helium and nitrogen [5].

A hybrid Stirling and Vuilleumier system providing both heat pump effect and work was studied by Geue et al. [6]. This machine can be set to work as a Stirling engine, a Vuilleumier heat pump or as a combined unit.

Another topical direction is the Vuilleumier pulse tube cryocooler. In 2017 Wang et al. [7] published experimental results and numerical simulation results for a pulse tube cryocooler fitted with a thermal compressor and a Stirling type precooler, capable of reaching 15.1 K.

Few papers dealing with theoretical physico-mathematical models of the Vuilleumier machine are available. One of the first documents presenting a model of a Vuilleumier machine was prepared for US Air Force Flight Dynamics Laboratory by White [8]. The model uses sinusoidal movements for the two displacer pistons and takes into account nine functional spaces: six heat exchangers and three variable volume spaces. The intermediate temperature spaces were taken into account as a single volume, in accordance with the actual construction of the machine. In all of these spaces only isothermal processes take place, at various temperature levels. Recently (2015) Rogdakis and Dogkas published a paper [9] analyzing the functioning of a Vuilleumier cryocooler. An isothermal model and an adiabatic model were developed and used. The cryocooler was modeled using nine functional spaces. The real movement of the two pistons was simplified, volume variations being considered pure sinusoidal (i.e. the rods are infinite long).

In previous published works, in 2005 [10] and 2006 [11], the authors developed for the Vuilleumier heat pump developed a semi-adiabatic model and an adiabatic (or full-adiabatic) model, respectively. Both models describe the functioning of a Vuilleumier machine comprising ten functional chambers, the intermediate temperature chambers being considered separate and not as a single space. The models also take into account the real movements of the displacers, as given by the drive mechanisms. The semi-adiabatic model has only two adiabatic spaces - the low temperature and the high temperature chambers. The rest of the functional spaces were considered isothermal. For the adiabatic model all four chambers inside cylinders were considered adiabatic spaces. In [11] the differential equations governing the functioning were established, but the system was not solved at that time.

The present paper develops further the previous work, analyzing the functioning of the Vuilleumier heat pump by numerical simulation, using the adiabatic model established in [11].

2. Construction and physical models of the Vuilleumier machine

A typical Vuilleumier machine has two cylinders, each having a double-acting displacer piston that delimit four variable volume chambers inside the cylinders (figure 1, from our previous work [11]).

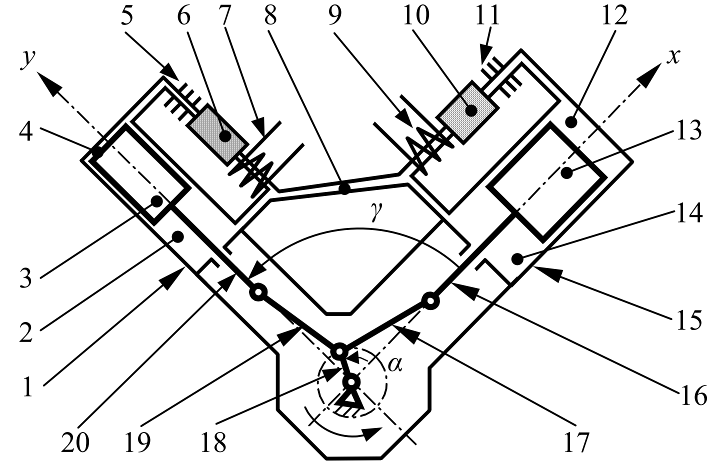

Figure 1. Schematic diagram of the Vuilleumier heat pump (from our work [11, fig. 1]): 1 – cold cylinder; 2 – intermediate temperature space of the cold cylinder; 3 – cold displacer; 4 – low temperature space; 5 – low temperature heater; 6 – low temperature regenerator; 7 – intermediate temperature cooler of the cold cylinder; 8 – connecting pipe; 9 – intermediate temperature cooler of the hot cylinder; 10 – hot temperature regenerator; 11 – high temperature heater; 12 – high temperature space; 13 – hot displacer; 14 - intermediate temperature space of the hot cylinder; 15 – hot cylinder; 16, 20 – piston rods; 17, 19 – rods; 18 – crankshaft; α –angle between positive direction of the hot cylinder axis and the crank; γ – angle between hot cylinder axis and cold cylinder axis.

The variable volume chambers of each cylinder are interconnected through three heat exchangers – a cooler, a regenerator and a heater. A connecting pipe allows the transfer of working agent between the two branches of the machine. The Vuilleumier machine is heat-driven, so the pressure variation inside is obtained through heat exchanges. Pressure rises when more mass of agent is displaced inside the high temperature space 12 and heated during the displacing process by heater 11.

Pressure decreases when more mass is displaced toward intermediate temperature spaces 2 and 14, and this mass is cooled inside intermediate temperature coolers 7 and 9. When pressure decreases some mass of agent flows inside the low temperature space 4, receiving heat from the low temperature heater. In order to obtain the right pressure variation, the movement of the displacers must be correlated through a proper drive. If the diameter of the piston rods 16 and 20 are negligible, the total volume occupied by the working agent inside the device is constant. As heat pump, the useful effect of this device is the heat rejected inside the coolers at intermediate temperature.

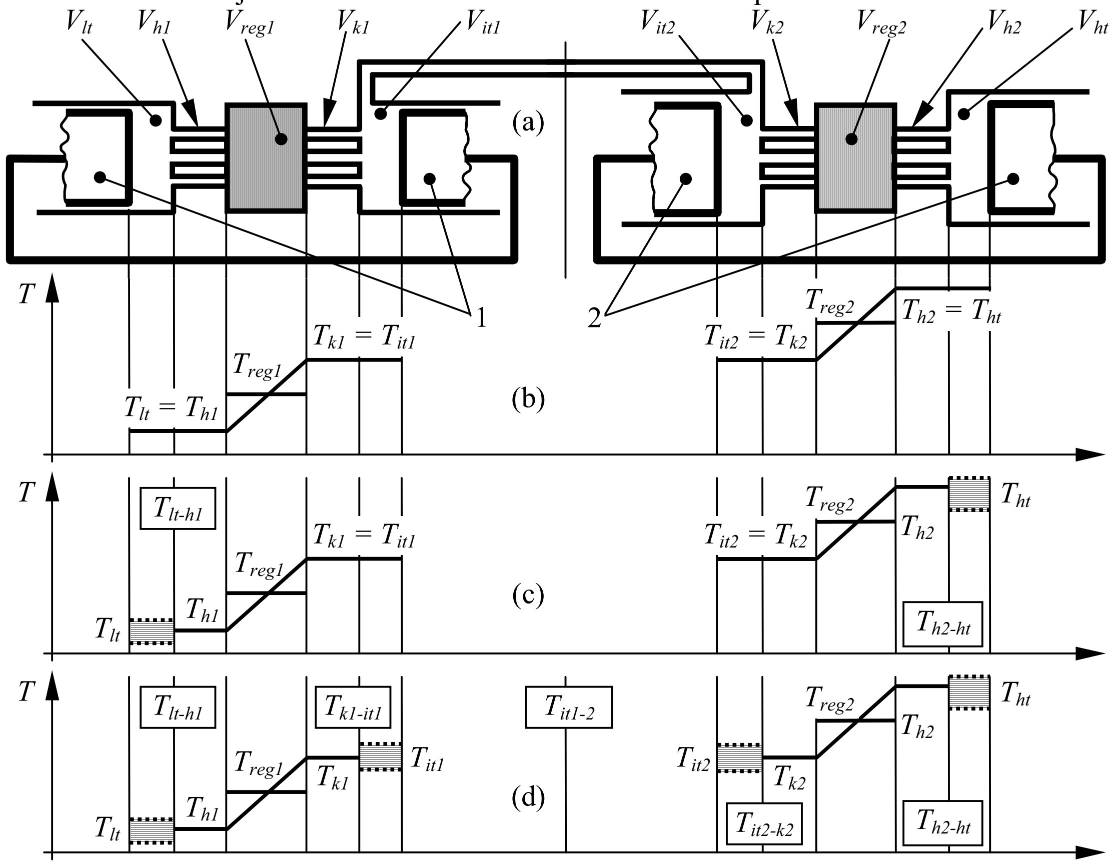

Figure 2. Temperature levels for the theoretical models of the Vuilleumier machine (adapted from our works [10, fig. 2] and [11, fig. 2]): (a) – equivalent scheme; (b) – isothermal; (c) – semi-adiabatic; (d) – adiabatic; 1 – cold displacer; 2 – hot displacer.

The subscripts in figure 2 represent: 1 – cold cylinder; 2 – hot cylinder; lt – low temperature; h – heater; reg – regenerator; k – cooler; it – intermediate temperature; ht – high temperature. Several subscripts were used to indicate the parameters of the separating sections between neighboring chambers. There are nine separating sections, but only five are of interest for the adiabatic model: lt-h1, k1-it1, it1-2, it2-k2 and h2-ht.

As in [10] and [11], displacer pistons were divided in two parts. In this way, the variable volume chambers could be represented as in equivalent scheme in figure 2 (a), where all ten chambers are placed in a single line.

In figure 2 (b), (c) and (d) the hypotheses about temperature levels were presented for three theoretical models of the Vuilleumier machine: isothermal, semi-adiabatic and adiabatic. For all models the heaters, regenerators and coolers are isothermal. The temperature of the working agent inside each regenerator is varying between the temperatures of the neighboring heat exchangers, so the heat regeneration efficiency is 100%. For convenience, the temperature of each regenerator is considered to be constant and equal with logarithmic mean of the temperatures of the neighboring heater and cooler. Variable volume spaces could be isothermal or adiabatic, depending on the chosen physico-mathematical model.

Several other hypotheses were accepted: a constant mass of perfect gas is the working agent, so there are no friction losses. The flow areas are so large that the speed of the agent and the corresponding kinetic energy are negligible in any section (no variation of the section area are allowed inside the machine). As a consequence, pressure inside all ten chambers has the same value at any given moment of time. Besides, the variation of the gravitational potential energy inside the machine is negligible. The device is working with a cyclic stationery regime, after each revolution of the crankshaft the parameters being the same in any point. The variable volumes were calculated using the real movements of the displacers, taking into account piston rods diameters and all clearance volumes. There are no heat transfers between adjacent chambers through their metallic walls.

3. Mathematical model

The adiabatic mathematical model of a Vuilleumier heat pump comprising four adiabatic variable volume chambers and six isothermal heat exchangers has nine differential equations. Supplementary, five conditional temperatures (that depend on the sense of the gas flow inside the machine) must be defined. The differential equations were obtained using the method used by Urieli and Berchowitz [12] for the adiabatic model of the Stirling engine. The differential expression of the mass of working agent inside each heat exchanger is obtained from the equation of state written in differential form. The differential expression of the mass inside each variable volume chamber is obtained from the first law of thermodynamics for open systems, also written in differential form. The total mass m_T of the working agent is constant during a cycle, so $dm_T = 0$. From the differential mass balance, obtained by summing the differential mass in each chamber, the differential expression of the pressure is obtained. Temperature differential inside each variable volume chamber is obtained from the equation of state written in differential form.

The proposed adiabatic model uses nine unknown functions: masses and temperatures inside all four variable volume chambers and pressure p. It must be observed that, for each chamber, both mass and temperature were obtained from the equation of state. So, strictly speaking, these are not independent functions. The number of equations of the system could be reduced to five or even to a single equation. This is not a practical option, because of the sheer length of such equations, which are almost impossible to be handled on a personal computer with a normal wide screen.

According to our previous work, the system of differential equations of the adiabatic model of the Vuilleumier machine [11, equations (8)-(12) and (16)-(19)] is:

$$dm_{lt} = \frac{1}{R\,T_{lt-h1}}\left[p\,dV_{lt} + \frac{V_{lt}}{k}dp\right]; \tag{1}$$

$$dm_{ht} = \frac{1}{R\,T_{h2-ht}}\left[p\,dV_{ht} + \frac{V_{ht}}{k}dp\right]; \tag{2}$$

$$\begin{aligned} dm_{it1} = {} & \frac{dp}{R}\left[\frac{V_{it1}}{k\,T_{it1-2}} + \left(\frac{T_{k1-it1}}{T_{it1-2}} - 1\right)\left(\frac{V_{lt}}{k\,T_{lt-h1}} + \frac{V_{h1}}{T_{h1}} + \frac{V_{reg1}}{T_{reg1}} + \frac{V_{k1}}{T_{k1}}\right)\right] + \\ & + \frac{p}{R\,T_{it1-2}}dV_{it1} + \left(\frac{T_{k1-it1}}{T_{it1-2}} - 1\right)\frac{p}{R\,T_{lt-h1}}dV_{lt}\,; \end{aligned} \tag{3}$$

$$dm_{it2} = \frac{dp}{R}\left[\frac{V_{it2}}{k\,T_{it1-2}} + \left(\frac{T_{it2-k2}}{T_{it1-2}} - 1\right)\left(\frac{V_{ht}}{k\,T_{h2-ht}} + \frac{V_{h2}}{T_{h2}} + \frac{V_{reg2}}{T_{reg2}} + \frac{V_{k2}}{T_{k2}}\right)\right] + \\ + \frac{p}{R\,T_{it1-2}} dV_{it2} + \left(\frac{T_{it2-k2}}{T_{it1-2}} - 1\right)\frac{p}{R\,T_{h2-ht}} dV_{ht}\;; \tag{4}$$

$$dp = A / B\;; \tag{5}$$

$$A = -k\,p\left(dV_{it1} + dV_{it2} + \frac{T_{k1-it1}}{T_{lt-h1}} dV_{lt} + \frac{T_{it2-k2}}{T_{h2-ht}} dV_{ht}\right); \tag{6}$$

$$B = V_{it1} + V_{it2} + \frac{T_{k1-it1}}{T_{lt-h1}} V_{lt} + \frac{T_{it2-k2}}{T_{h2-ht}} V_{ht} + k\,T_{k1-it1}\left(\frac{V_{h1}}{T_{h1}} + \frac{V_{reg1}}{T_{reg1}} + \frac{V_{k1}}{T_{k1}}\right) + \\ + k\,T_{it2-k2}\left(\frac{V_{h2}}{T_{h2}} + \frac{V_{reg2}}{T_{reg2}} + \frac{V_{k2}}{T_{k2}}\right); \tag{7}$$

$$dT_{lt} = T_{lt}\left(\frac{dp}{p} + \frac{dV_{lt}}{V_{lt}} - \frac{dm_{lt}}{m_{lt}}\right); \tag{8}$$

$$dT_{it1} = T_{it1}\left(\frac{dp}{p} + \frac{dV_{it1}}{V_{it1}} - \frac{dm_{it1}}{m_{it1}}\right); \tag{9}$$

$$dT_{it2} = T_{it2}\left(\frac{dp}{p} + \frac{dV_{it2}}{V_{it2}} - \frac{dm_{it2}}{m_{it2}}\right); \tag{10}$$

$$dT_{ht} = T_{ht}\left(\frac{dp}{p} + \frac{dV_{ht}}{V_{ht}} - \frac{dm_{ht}}{m_{ht}}\right); \tag{11}$$

$$\begin{array}{lcl}
\begin{cases} T_{lt-h1} = T_{lt} \\ T_{lt-h1} = T_{h1} \end{cases} & if & \begin{array}{l} dm_{lt} \le 0 \\ dm_{lt} > 0 \end{array} \\
\begin{cases} T_{k1-it1} = T_{k1} \\ T_{k1-it1} = T_{it1} \end{cases} & if & \begin{array}{l} d(m_{lt} + m_{h1} + m_{reg1} + m_{k1}) \le 0 \\ d(m_{lt} + m_{h1} + m_{reg1} + m_{k1}) > 0 \end{array} \\
\begin{cases} T_{it1-2} = T_{it1} \\ T_{it1-2} = T_{it2} \end{cases} & if & \begin{array}{l} d(m_{lt} + m_{h1} + m_{reg1} + m_{k1} + m_{it1}) \le 0 \\ d(m_{lt} + m_{h1} + m_{reg1} + m_{k1} + m_{it1}) > 0 \end{array} \\
\begin{cases} T_{it2-k2} = T_{it2} \\ T_{it2-k2} = T_{k2} \end{cases} & if & \begin{array}{l} d(m_{k2} + m_{reg2} + m_{h2} + m_{ht}) \ge 0 \\ d(m_{k2} + m_{reg2} + m_{h2} + m_{ht}) < 0 \end{array} \\
\begin{cases} T_{h2-ht} = T_{h2} \\ T_{h2-ht} = T_{ht} \end{cases} & if & \begin{array}{l} dm_{ht} \ge 0 \\ dm_{ht} < 0\,. \end{array}
\end{array} \tag{12}$$

This is an initial value problem, and the system could be solved using a Runge-Kutta method.

The work exchanged inside each variable-volume chamber was calculated using the defining relation:

$$W = \oint p\,dV\,. \tag{13}$$

For any chamber of the machine the heat exchanged could be calculated using the first law of Thermodynamics written for an open system. For a control volume and for a period of time, the first law takes the following form [9]:

$$Q = \Delta U + W + m_o\,h_o - m_i\,h_i\,, \tag{14}$$

where: Q – heat, U – internal energy, W – work, m – mass of agent exchanged by the control volume, h – mean specific enthalpy. Subscripts "i" and "o" stand for inlet and outlet. The kinetic energy and gravitational potential energy were neglected.

For the heat exchangers $W = 0$, since there is no volume variation. For a complete cycle $\Delta U = 0$ for any chamber. Now equation (14) becomes:

$$Q = \oint c_p\,T_o\,dm_o - \oint c_p\,T_i\,dm_i\,. \tag{15}$$

During a complete cycle, the working agent flows through any section of the machine in both directions. For a complete cycle the mass flowing to the right is equal with the mass flowing to the left. Two sections border each heat exchanger, one on the left and one on the right. Due to the functioning particularities of the Vuilleumier machine, for expressing mass that enters a heat exchanger both bordering sections must be taken into account. The same is for the mass that leaves the heat exchanger. It is difficult to express separately the mass that enters the heat exchanger, because is divided between the two bordering sections. It is much easier to use the masses that flow through each bordering section instead of masses flowing in or out.

For any section the mass that flows to its right is equal to the variation of the mass of working agent from all the chambers placed at the left of the analyzed section, taken with opposite sign. This statement expresses the law of mass conservation. In consequence, the masses appearing in equation (15) could be replaced by the variations of the masses found at the left of the sections bordering the heat exchanger, taken with opposite sign. Equation (15) becomes:

$$Q = \oint c_p\,T_{rs}\,dm_{l\,rs} - \oint c_p\,T_{ls}\,dm_{l\,ls}\,, \tag{16}$$

where subscripts "rs" and "ls" stand for right and left sections of the heat exchanger and "l rs" and "l ls" stand for the total mass of working agent found at the left of the right section respectively to the left of the left section of the same heat exchanger. At cycle level, only the section with conditional temperature has meaning. So, when applying equation (16) to each of the four heat exchangers, only one term counts.

The coefficient of performance of the Vuilleumier heat pump is:

$$\varepsilon_{Vhp} = \frac{\left|Q_{it1} + Q_{it2}\right|}{Q_{ht}}\,, \tag{17}$$

since the useful effect is the heat removed from the cycle at intermediate temperature and the energy used for obtaining the useful effect is the heat received by the high temperature heater.

4. Results of the numerical simulation of the Vuilleumier heat pump and discussion

The analyzed Vuilleumier heat pump features the same dimensions as the one studied in previous work [10]: diameter of the cold displacer (equal to the cold cylinder bore) $D_1 = 0.1$ m; diameter of the piston rods $d_1 = d_2 = 0.02$ m; diameter of the hot displacer $D_2 = 0.12$ m; $\gamma = 90^{\circ}$; crank radius $r = 0.05$ m; rods length $l_1 = l_2 = 0.2$ m; distances between dead centers and nearby cylinder heads

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$f_{TDC1} = f_{BDC1} = f_{TDC2} = f_{BDC2} = 0.001$ m; heaters and coolers volume $V_{h1} = V_{h2} = V_{k1} = V_{k2} = 0.05\ V_{Sd2}$; regenerators volume $V_{reg1} = V_{reg2} = 1.2\ V_{Sd2}$, where $V_{Sd2} = 1.142\ 10^{-3}$ m^3 is the volume swept by the high temperature displacer.

The chosen working agent was hydrogen with specific gas constant $R_{H2} = 4121$ J kg^{-1} K^{-1}. At ambient temperature of 15 ºC the pressure inside the heat pump was set at 50 bar, which corresponds to a hydrogen total mass of $m = 0.0207$ kg. Temperature levels were chosen as follow: $T_{h1} = 923$ K, $T_{k1} = T_{k2} = 343$ K and $T_{h2} = 278$ K.

The main results obtained after the numerically solving of the system of differential equations of the adiabatic model are presented in the following figures.

The pressure-volume diagrams (indicator diagrams) in figure 3 show that work is produced inside low and high temperature chambers. Each intermediate temperature chamber receives work through the piston shared with the nearby variable volume chamber. Due to the piston rods diameter, the theoretical Vuilleumier heat pump is yielding a small cyclical work, as represented in the Sankey diagram in figure 6.

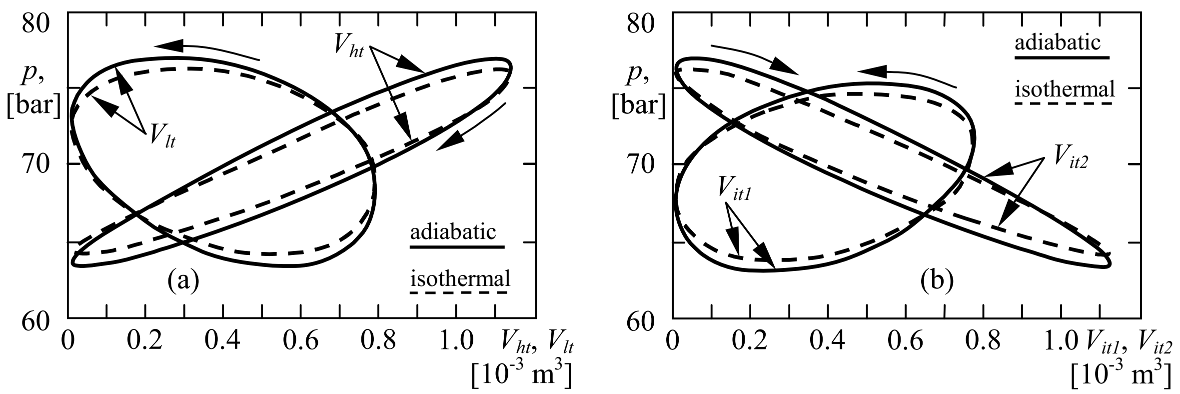

Figure 3. Pressure-volume diagrams for all variable-volume chambers of the Vuilleumier machine:
(a) – for low temperature and high temperature chambers;
(b) – for both intermediate temperature chambers.

For comparison, figure 3 also presents the pressure-volume diagrams of the same Vuilleumier heat pump calculated with an isothermal model that takes into account the real movement of the pistons. It can be seen that the pressure variation interval is larger for the adiabatic model than for the isothermal model. This conclusion is strengthened by the variation of the pressure function of rotation angle, as represented in figure 5. The adiabatic model clearly yields larger work than the isothermal model, indicating a lesser coefficient of performance of the adiabatic heat pump.

For the low temperature chamber the work produced during a complete cycle is equal to the heat received cyclically by the low temperature heater. The explanation is simple: when analyzing the low temperature chamber and the low temperature heater as a whole, for a complete cycle $\Delta U = 0$, temperature of the section h1-reg1 is constant and consequently $Q_{h1} = W_{lt}$. Similarly, the work produced by the high temperature chamber is equal to the heat received by the working agent inside the high temperature heater, $Q_{h2} = W_{ht}$. So, the p-V diagrams for low temperature and high temperature chambers in figure 3 also represent the heats received during a cycle inside low and high temperature heaters h1 and h2.

The results about temperature variation inside the four adiabatic chambers are presented in figure 4 (a) as function of rotation angle and (b) as function of chamber volume. Temperatures T_{h1} and T_{h2} of the low and high temperature heaters, temperature T_k of the intermediate temperature coolers and mean temperatures of the adiabatic chambers were also represented. It can be seen that for any adiabatic chamber there is a part of the cycle for which the temperature of the working agent is higher than the temperature of the nearby heat exchanger. For the rest of the cycle the adiabatic chamber temperature is lesser than the temperature of the nearby heat exchanger.

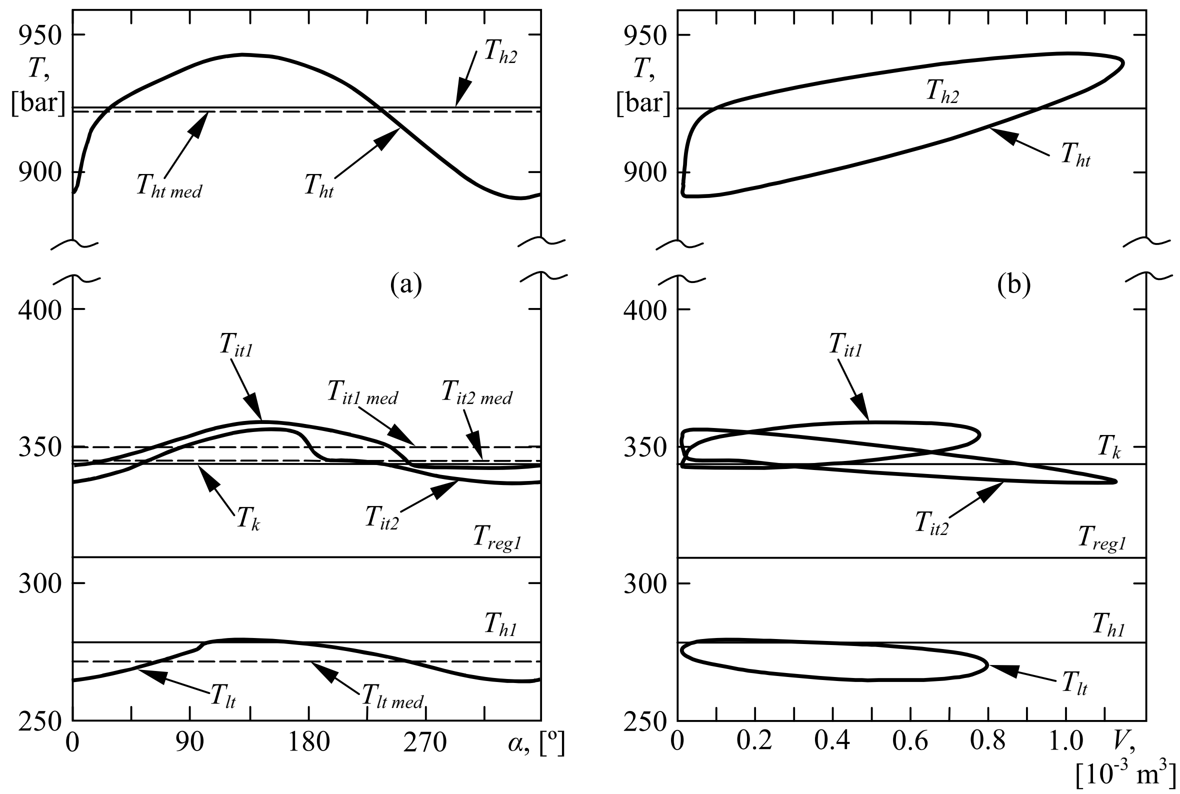

Figure 4. Temperature variations for all variable-volume chambers of the Vuilleumier machine: (a) – function of rotation angle; (b) – function of chamber volume.

As expected, T_{it1} and T_{it2} have different variation laws. Between the mean temperatures of the adiabatic chambers and the temperatures of the heat exchangers there are the following relations: $T_{ht\ med} < T_{h2}$, $T_{it1\ med} > T_{it2\ med} > T_k$ and $T_{lt\ med} < T_{h1}$.

Table 1 presents the heats cyclically exchanged by the Vuilleumier heat pump, calculated with the adiabatic model as well as with semi-adiabatic and isothermal models [10]. The adiabatic model stressed the role of the heaters and coolers as heat exchangers. For the semi-adiabatic model, zero heat exchanged per cycle characterizes the two coolers. In the case of the isothermal model, the heats exchanged cyclically by each heat exchanger are all zero. Since there is no heat exchange for an adiabatic variable-volume chamber, the heat must be exchanged elsewhere – inside a heat exchanger. Because an isothermal variable-volume chamber is characterized by $Q = W$ for a cycle, there is no amount of energy cyclically transferred through its separating sections. So there is no heat exchanged per cycle inside the nearby heater of cooler. As expected, the coefficient of performance predicted by the adiabatic model is smaller than the ones given by the other two models.

Table 1. Heats exchanged inside machine chambers (per cycle).

Model	Q_{lt}	Q_{h1}	Q_{k1}	Q_{it1}	Q_{it2}	Q_{k2}	Q_{h2}	Q_{ht}	ε_{hp}
	[J/cycle]								-
isothermal	718.0	0	0	-701.9	-295.6	0	0	300.4	3.32
semi-adiabatic	0	749.4	0	-732.4	-344.7	0	350.3	0	3.07
adiabatic	0	786.5	-945.2	0	0	-229.4	412.5	0	2.85

The balance of heat of the Vuilleumier heat pump with four adiabatic chambers is synthesized in the Sankey diagram in figure 6. The heat received in the high temperature heater, together with the heat received inside the low temperature heater are transferred to the intermediate temperature coolers, except for a small amount that was transformed in work (due to the presence of piston rods). The heat received by the working agent inside the high temperature heater was taken as reference (100 %) and all values were expressed as percentage from Q_{h2}.

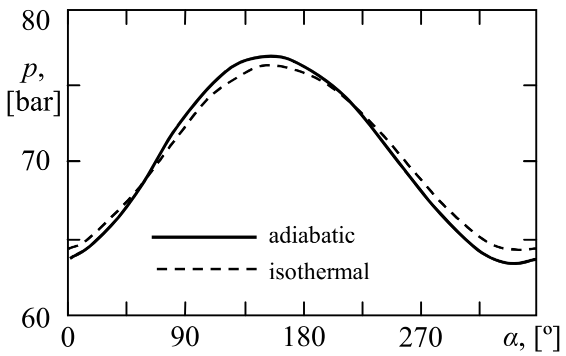

Figure 5. Pressure variation inside Vuilleumier heat pump.

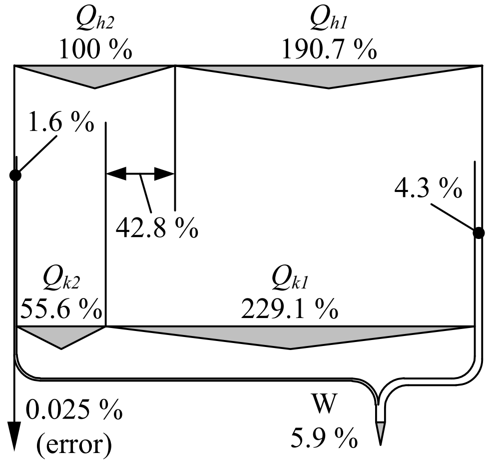

Figure 6. Sankey diagram.

From the heat Q_{h2} received inside the high temperature heater 55.6 % leaves the heat pump through the intermediate temperature cooler of the hot cylinder (as Q_{k2}) and a portion of 42.6 % is transferred to the intermediate temperature cooler of the cold cylinder. The rest of 1.6 % from Q_{h2} is transformed in work in the hot branch of the machine.

The heat Q_{h1} received inside the low temperature heater represents 190.7 % from Q_{h2}. A small share of 4.3 % is transformed in work in the cold branch of the heat pump. The rest from Q_{h1}, namely 186.4 % from Q_{h1}, and the 42.8 % share that comes from the hot branch are rejected at the intermediate temperature cooler of the cold cylinder. This is a notable characteristic of the model with four adiabatic variable-volume chambers – it emphasizes the energy exchange between the two branches of the machine. The semi-adiabatic and the isothermal models of the Vuilleumier heat pump are not able to point out this fact. The Sankey diagram also displays the error made by numerically solving the mathematical model: 0.025 %.

5. Conclusions

The results obtained through the numerical simulation of the Vuilleumier heat pump by using the adiabatic physico-mathematical model provide a clear picture of the functioning and of the theoretical performances of the machine.

The adiabatic model emphasize the real role of the heat exchangers. Besides, it allows for estimating the energy received by the working agent inside the high and low temperature heaters and also the energy rejected inside the intermediate temperature coolers. These estimations could be useful for the preliminary dimensioning of the heat exchangers. The hypothesis of the adiabatic chambers is stronger than the hypothesis of isothermal variable-volume chambers. It is closer to reality to expect that the heat exchanges will take place inside the heaters and coolers and not through the walls of the cylinders that comprise the variable-volume chambers of the machine.

Due to the use of two distinct adiabatic intermediate temperature chambers with a separating section with conditional temperature between them, the model is able to emphasize also the energy transfer from the hot to the cold branch of the machine during a cycle. This energy transfer between branches is related to the enthalpy transferred with the mass that passes from one branch to the other at different temperatures. Isothermal and semi-adiabatic models, that do not have a separating section with conditional temperature between the two intermediate temperature variable-volume chambers,

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are calculating a null net transfer of energy between the branches. So, the adiabatic model is more realistic in its estimation than the other theoretical models.

A comparison of the results obtained through numerical simulation for a Vuilleumier heat pump having the same geometry and similar functioning conditions with the results obtained with other theoretical models (semi-adiabatic and isothermal) shows that the adiabatic model predicts a larger variation of the pressure during the thermodynamic cycle. In spite of larger energy exchanged by the adiabatic model, the coefficient of performance is lower than the isothermal or semi-adiabatic ones. All theoretical models could provide reference performances. Each of them could be considered as the maximum possible performances that could be obtained by a Vuilleumier heat pump working in various ideal conditions. The performances of a real machine will be always below the reference performances. It seems to be logical to use as reference performances the results of the adiabatic model because they are closest to the real functioning (between the theoretical models).

The adiabatic theoretical physico-matematical model comprising four variable-volume adiabatic chambers and five separating sections with variable temperatures is a useful development of the semi-adiabatic model, since yields a more realistic knowledge about the functioning of the Vuilleumier heat pump.

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