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Calculation of Heat Output of the Combined System with a Solar Collectors and Heat Pump

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Abstract: This study deals with the study and calculation of the thermal conductivity of the tube solar collector with heat pump. The basic equations describing the energy balance of the flat solar collector with the heat pump. This study discusses the calculation of the amount of heat in the heat pump in the combined system of solar heating. A heat balance equation of the heat pump in the integrated system of solar heating method for computing was developed using the techniques above, determine the amount of heat and energy balance equation. The results show that the arrangement of the compressor in the evaporator zone of impact. The evaporator absorbs the heat emitted from the surface of the compressor which is added to the main flow. In view of the fact that the temperature difference between the compressor stack (about 800°C) and the evaporator (about 180°C) is 600°C strand it suggests that the main flow is radioactive heat transfer between the surfaces. As is known in the heat radiation is eliminated between the compressor and the evaporator by setting the screen there between. Accordingly, there has been some decrease in performance. However, even under these conditions the compressor does not overheat because its convective cooling is maintained. Energy conversion efficiency (KPI) are from 2.5-4.5 units.

Key words: Solar collector, combined technology, a heat pump, a compressor, a condenser, an evaporator

INTRODUCTION

Solar energy and renewable energy sources for heat and electric energy without damaging impact on the environment (Zhai *et al*., 2013; Al-Sulaiman *et al*., 2011). Flat plate solar collector is the main component of solar heating and district heating systems. The energy equation does not account for the analysis of the efficiency of solar collectors, so it is not a sufficient criterion for the efficiency of the solar collector (Dutta Gupta and Saha, 1990). It is necessary to optimize the design of the solar system and research (Tyagi *et al*., 2007; Liu *et al*., 1995; Torres *et al*., 2001). The effectiveness of the entropy of solar collectors were considered. Bejan *et al*. (1981) found that solar systems depends on the irreversibility of heat transfer.

As compared with a flat plate at the evacuated tubular collectors from the standpoint of exergy (Suzuki, 1988). Calculate energy and conducted an analysis of solar flat plate (Jafarkazemi and Ahmadifard, 2013). The temperature of the liquid in a flat plate solar collector is an ambient temperature (Luminosua and Fara, 2005). Determine the optimal flow rate and calculated the maximum efficiency of a flat plate collector (Farahat *et al*., 2009). Effective use of alternative energy sources is one of the main objectives of the current situation in the use of energy in buildings. Increasing the share of Renewable Energy Sources (RES) in buildings is one of the possible ways of achieving this goal. Reduction of greenhouse gas emissions on sustainable systems using renewable energy sources can have a positive impact on increasing the reliability of energy supplies. Reduced operating and capital costs can be carried out with the help of the effectiveness of relevant measures in the field of construction and engineering projects using combined systems, consisting of renewable energy sources.

Huang *et al*. (2005) has been studied and found to the long-term efficacy of the combined collector and heat pump solar heating system, where the price of electricity was cheaper than conventional gas heating system.

Hawlader *et al*. (2001) calculated the capacity hot-water heating systems using solar energy with the assistance of heat. They have shown that the system performance is largely dependent on the area of the collector the compressor speed and solar radiation. Thermal storage systems in the studied systems

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use active and passive building. Building heating systems using a heat pump shown that is possible to save 15-28% of the energy consumption (Brandl, 2006).

Heat pumps (in fact on the contrary refrigerators) received the most widespread elements as heaters and air conditioners in the highly developed (usually poor own natural energy resources) countries (Austria, United Kingdom, Sweden, Japan and others) (Brandi, 1998).

According to Siroky *et al.* (2011) are currently installed in Austria heat pumps 200,000. This saves about 252,000 tons of fuel per year (oil equivalent). Therefore, the number of heat pumps in this small country is increasing every year by approximately 5,000 units.

Madani *et al.* (2011) developed a mathematical model with lumped parameters for steady state in the heating system with a heat pump. Kinab *et al.* (2010) has been developed and investigated a mathematical model of the capacitor in the heat pump system. Qiao *et al.* (2013) we established a new mathematical model of convective heat transfer to the refrigerant and the water inside the capacitor plates. Santa (2012) investigated the condensing heat transfer coefficients in the heat pump.

MATERIALS AND METHODS

Modeling the joint operation "solar collector+heat pump" system: The circuit interface elements heat pump operation process with solar collectors is shown in Fig. 1. Where GU1-solar energy, GU2-warm air, GU3-solar collector, GU4-heat pump GU5-heat the battery.

With the formalization of the model followed an analogy with the real object. Input and output signals are designated x_{1i} and y_{1i} where the symbol i shows the number of the element to which the signal and the numeral 1, according to the accepted notation (Fig. 1) membership of the subsystem parameters. Accordingly: q_{1i} is heat flux, t_{1i} is the temperature of the heat flow. Assumptions made quasistationarity links between elements and independence of solar energy and heat of atmospheric air flows from each other which are expressed through a combination of the following Eq. 1:

$$\begin{aligned} & y_{11}(e_{m1}) = x_{12}(e_{m1}) y_{12}(e_{m1}) = x_{13}(e_{m1}); \; y_{12}(t_{m1}) = x_{13}(t_{m1}); \\ & y_{13}(q_{13}; t_{12}) = x_{14}(q_{13}; t_{12}); \; y_{14}(q_{14}; t_{14}) = x_{15}(q_{14}; t_{14}); \\ & y_{14}(q_{14}; t_{11}) = x_{13}(q_{14}; t_{11}); \; y_{15}(q_{15}; t_{13}) = x_{13}(q_{14}; t_{11}) \end{aligned} \quad (1)$$

The model has a special input terminals which receive control signals which are two-the power of the heat pump compressor (p_{mH}) and backup power source (p_{pu}). Feedbacks from GU4 in GU3 from GU5 in GU4 show coolant from the outlet of the circulation evaporator of the heat pump to the input of the solar collector and $y_{14}(q_{14};\, t_{11})$ and exit from the storage tank to the input of the heat pump condenser $y_{15}(q_{15};\, t_{13})$. Variable intensity of solar radiation in the heating months, simulated output from GU1 and can be represented by an Eq. 2:

$$y_{11}(e_{m1}) = e_m = \sum_{n=1}^{n} a_{1n} \cdot m^{n-1} \quad (2)$$

Variable air temperature in the heating months, the simulated output from GU2 and can also be represented by an Eq. 3:

$$y_{12}(t_{m1}) = t_{m1} = \sum_{n=1}^{n} b_{1n} \cdot m^{n-1} \quad (3)$$

Where:
a_{1n}; b_{1n} = Constant polynomials
m = The serial number of the billing month

Using the technique of Beckmann and Duffy solar collector heat balance equation,researching with a heat pump can be expressed as 2 equations describing the conditions of heat exchange with the sun and reverse the shadow side solar collector (Fig. 2 and 3):

$$k_1(t_{m1} - t) \cdot S_{sc} + e_m \cdot \beta \cdot S_{sc} - C_{sc1} \cdot \frac{dt}{d\tau} = q_{sc} + q'_{ma6} \quad (4)$$

$$k_2(t_{m1} - t) \cdot S_{sc} - C_{sc2} \cdot \frac{dt}{d\tau} = q_{sc} + q''_{ma6} \quad (5)$$

Total differential equation would be:

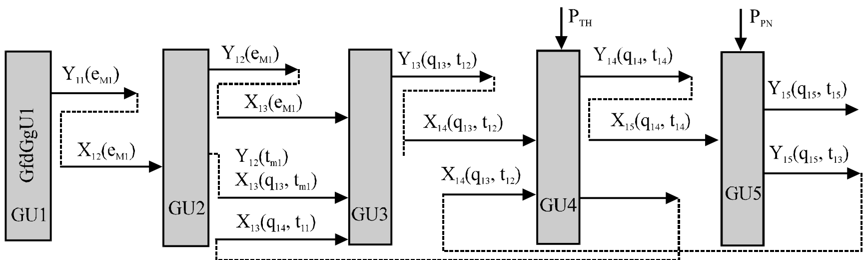

Fig. 1: Model and subsystem interface circuit elements solar power plant

Fig. 2: Tubular solar collector

Fig. 3: General view of the solar

$$2k\left(t_{m1}-t\right)\cdot S_{sc}+e_m\cdot\beta\cdot S_{sc}-C_{sc}\cdot\frac{dt}{d\tau}=q_{sc}+q_{ma} \quad (6)$$

Where:

$q_{cs}+q_{TAB}=q_{x1}$	=	Cooling capacity of the heat pump
t	=	The temperature of the solar collector
β	=	Optical coefficient of the solar collector
C_{2k1}, C_{2k2} and C_{2k}	=	Heat capacity (solar collector) face the back and parts of the total
k, k_1, k_2	=	Overall heat transfer coefficients and two sides solar collector

Making the separation of variables, we obtain:

$$\int_0^{\tau}d\tau=-\frac{C_{\Gamma k}}{2\cdot k\cdot S_{sc}}\cdot\int_{t_{m1}}^{t}\frac{d\left[2\cdot k\cdot S_{sc}\left(t_{m1}-t\right)+e_m\cdot\beta\cdot S_{sc}-q_{x1}\right]}{2\cdot k\cdot S_{sc}\left(t_{m1}-t\right)+e_m\cdot\beta\cdot S_{sc}-qx1}$$

The solution is a function of:

$$t=t_{m1}-\frac{q_{x1}-e_m\cdot\beta\cdot S_{sc}}{2\cdot k\cdot S_{sc}}\left[1-\exp\cdot\left(-\frac{2\tau\cdot S_{sc}\cdot k}{C_{sc}}\right)\right] \quad (7)$$

When $\tau=\infty$ we obtain the formula for the steady state:

$$t=t_{m1}-\frac{q_{x1}-e_m\cdot\beta\cdot S_{sc}}{2\cdot k\cdot S_{sc}} \quad (8)$$

The vector equation of energy balance GU3 element modeling solar collector:

$$q_{m\text{н}x1}=p_{m\text{н}}\cdot\tau\cdot\left[\sum_{n=1}^{n}c_n\cdot\left(t_k-t\right)^{n-1}-1\right]$$

$$q_{m\text{н}m1}=p_{m\text{н}}\cdot\tau\cdot\left[\sum_{n=1}^{n}c_n\cdot\left(t_k-t\right)^{n-1}\right] \quad (9)$$

Results quasi natural modeling two solar contour shown in Fig. 4a-f which shows the operation modes of the solar collector at a power of 500, 1000, 1500, 3000, 4000 W. When the solar collector capacity (at the entrance) P_{sc} = 500 W developed the installation output power (q_{m1}) up to 1700 W. In this case, the coolant temperature in the solar collectors (t_2) was lowered to -20°C while the temperature in the heater to +38°C (input) and +32°C (output). The observed dynamics of temperature led to a decrease in heat output from 1.7-1.25 kW.

Similar patterns have been observed in other stages of “solar collector” capacity. For example, when P_{sc} = 1000 W in the solar collector temperature was lowered to -16°C and the inlet and outlet of heating devices 46 and rose to 42°C. Output power was about 1800 and 1400 W beginning at the end of the process.

When P_{sc} = 2000 W showed stabilization t_2 within 6...+12°C indicating that the power balance of the solar collector and the cooling capacity of the heat pump. Conversion factor was 3.5 at the beginning of the process which corresponds to the output power of the heat pump 2450 W.

When P_{sc} 3000 = 4000 W and the coolant temperature in solar collectors in the experiments were not decreased and gradually raised (from 15-18°C and from 15-23°C), despite the fact that the heat pump continuously absorbs heat coolant. The pattern can be explained by the fact that the heat output of the solar collector exceeds coldness performance of the heat pump. Those the heat pump does not have time to “siphon” the incoming energy. The values of conversion rate is at the beginning and end of the process for the installed capacities from 4.2-3.5 and from 4.8-4.0, respectively. Reduction of transform coefficients due to the higher temperature at the outlet of the heat pump (t_3 and t_4) and consequently, deterioration of cooling the coolant in the condenser.

Studies have confirmed the theoretical results. In general the power developed by the heat pump capacity is equal to the sum of “solar collector” and the compressor. In mode 500 and 1000 W exceeded the total amount that can be explained by an additional heat absorption of the evaporator air heat exchanger as the

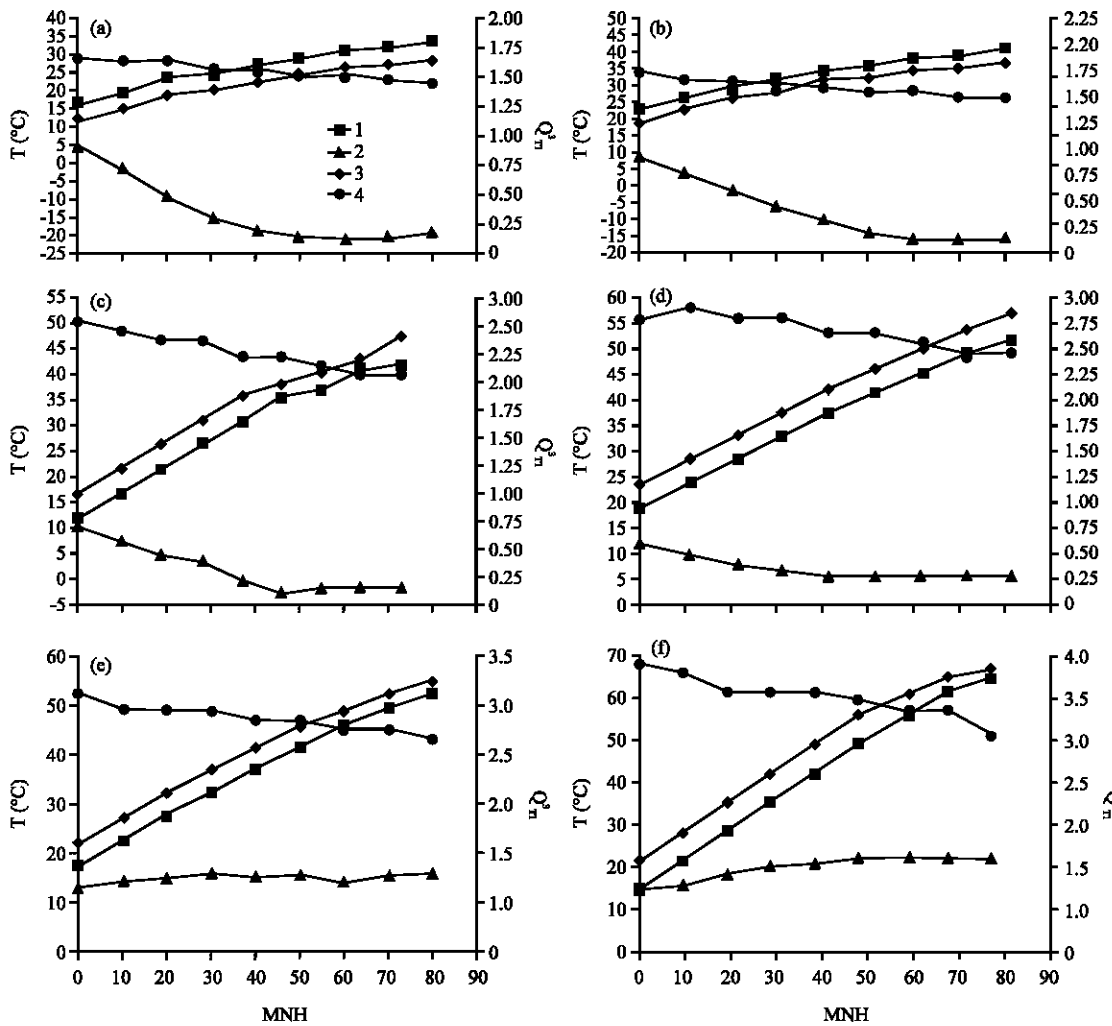

Fig. 4: Results of research 2 contour modeling work solar collector with heat pump; 1, 3-t_3, t_4-coolant temperature at the inlet and outlet of the condenser; 2-t_2-temperature of the coolant in the solar collectors; 4-qtl-h heat output settings: a) P_{rc} = 500 W; b) P_{rc} = 1000 W; c) P_{rc} = 1500 W; d) P_{rc} = 2000 W; e) P_{rc} = 3000 W and f) P_{rc} = 4000 W

surface temperature is below ambient temperature. Increasing the temperature of the condenser (t_3, t_4) and reduction of evaporator temperature (t_2) lead to reduced thermal output of the heat pump which is also consistent with the theoretical results.

State of the problem: Consider the equation of heat-mass transfer, currently adopted for the calculation of thermal fields in soil foundations (and not only). According to modern ideas about the nature of heat propagation there are three basic types of heat transfer in continuous media (including the ground):

- Conductive
- Convection
- Radiation (i.e., radiation heat transfer)

To describe the conductive heat transfer using Fourier's law which has the form (Bejan *et al*., 1981):

$$dQ = \lambda \times \frac{dt}{dn} \times dF \times dt \quad (10)$$

Where:

dQ	=	The amount of heat that has passed in time
dt	=	Through the element
dF	=	surface normal thereto
dT/dn = grad (T)	=	Temperature gradient
T	=	Temperature
λ	=	Coefficient of heat transfer
t	=	Time

Description of the technology and systems: With this in mind, we proposed a new structural and technological

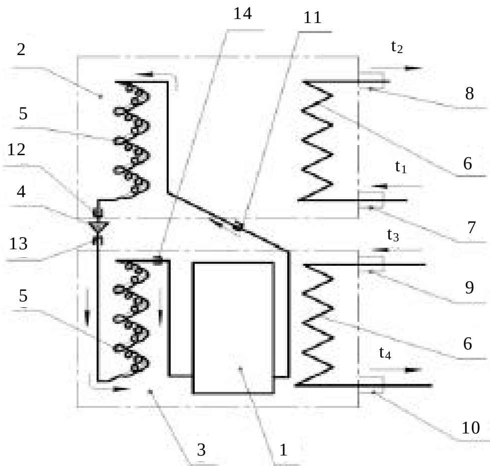

Fig. 5: Scheme of structural unit of the new HP

Fig. 6: Detail of experimental studies HP

solution that takes into account the shortcomings of the known analogues. Driving the new solution is shown in (Fig. 5). The heat pump includes: a compressor 1, a condenser 2 an evaporator 3, a throttle device 4 which made structurally a like as coil heat exchangers of the condenser and the evaporator "pipe in pipe", comprise an inner pipe 5 through which circulates the coolant and the external conduit 6 which namely the annulus, circulates coolant bathing the conduit 5. In this case the inner pipe 5 is arranged in a spiral pattern to create a helical channel for circulating the coolant in the annulus. External piping equipped with condenser and evaporator entry spigots 7, 8, 9 and the output 10 of coolant with the proviso that the nozzles are arranged to enter the refrigerant outlet side and output from the refrigerant inlet. Structurally, the condenser and evaporator piping heat exchanger coils stacked one above the other, along a helical path with the same average diameter and pitch of turns, forming a cylindrical shape with the placement of the condenser above the evaporator and the compressor within the evaporator. Sequential connection compressor to the condenser, a throttling device with condenser, throttling device and an evaporator with an evaporator to a compressor formed respectively by means of choke connectors 11, 12, 13 and 14.

Figure 6 schematically displayed the collection, transmission, processing and storage of data in research. The basis of the information system constitute the central unit (server), database, personal computers, software and data acquisition modules with heat pump coolers of milk and buffer tank.

Heat pump module collects data from these sensors 12, temperature sensors (DS18B20 encapsulated) two fluid flow sensors (G1WFM) and two pressure sensors (Wika-R1). Milk cooler module is connected to the 4th temperature sensor (DS18B20) and one flow rate sensor (G1WFM). battery module collects data from 15 temperature sensors (DS18B20), successively arranged along the vertical direction.

The process of collecting information triggers the central unit by sending a request to each module separately. After receiving the request, each module begins sensors survey and collect data in a single package which is subsequently sent back to the central unit. Server receives packets with the "raw" data processes them according to the relevant algorithms for easy storage. The server then sends the data to the database for storage. You can view the current values, using special software from the user PC. Figure 6 shows a fragment of the experimental studies.

Experimental studies conducted HP for 3 variants: 1st variant-the compressor is located in the center of the evaporator. About 2nd option-the compressor is off-center, close to the wall of the evaporator. About 3rd option-the compressor is located in the center of the evaporator but ruled out heat exchange by radiation from the walls of the evaporator. Each experiment is carried out at temperatures cocircuit arrangement of the temperature sensors and the pressure is shown in Fig. 7. Research evaporator is characterized by four other sensors 1 and 2 and sensors at the inlet and outlet of the evaporator coil (Fig. 7). They also show the temperature of milk in the milk cooler.

Research capacitor characterize four sensors 3 and 4 and sensors at the inlet and outlet of the condenser exchanger (Fig. 7). They also show the water temperature in the accumulating tank.

The temperature of the compressor show mode sensor 6 attached to the side wall and the sensor 7 is

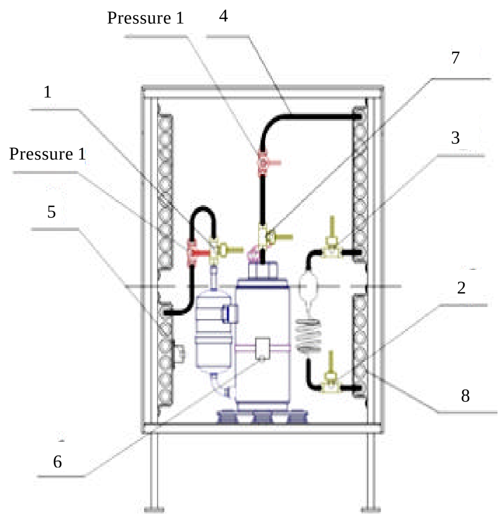

Fig. 7: Arrangement of the sensors in HP

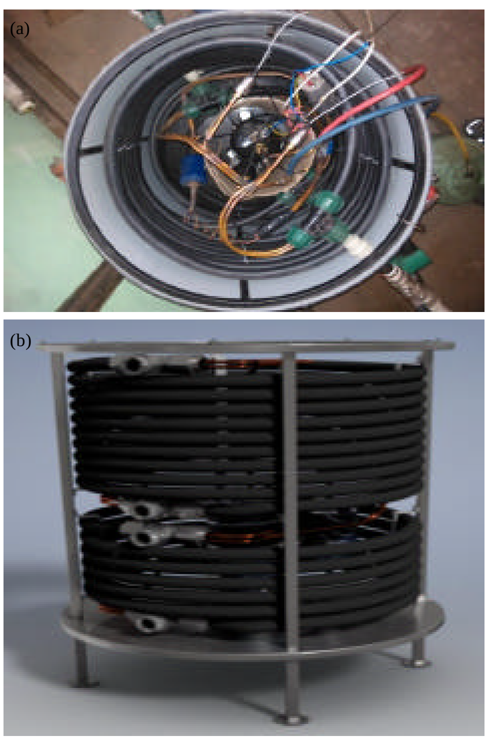

Fig. 8: a, b) Heat pump

attached to the compressor cover. Sensor 6 indirectly indicates the temperature state of the motor which is located inside the compressor 6. opposite point temperature sensor 7 characterized by a cap which is directly compressing refrigerant. The temperature condition in the space between the compressor and the evaporator show sensors 5 and 8 (Fig. 8a-b).

RESULTS AND DISCUSSION

During operation of the compressor, performing compression research of the refrigerant is heated to a temperature of the evaporator t_k and cooled to a temperature t_{uc}. Accordingly, radiative heat transfer is formed between them, heat transfer and convection. Thus, the surface receives a constant evaporator temperature the surface temperature of the compressor will increase to a steady state value. You need to set a pattern of change in the formed sheet. The differential heat equation will be (Farahat *et al*., 2009):

$$k \cdot W \cdot \tau - Э_{rad} - Э_{con} = C_c \cdot \frac{dT_{com}}{d\tau} \quad (11)$$

Where:

$Э_{rad} = T^4_{con}$- $T^4_{eva}/\Delta d^2$	=	Heat transfer from the compressor to the evaporator radiation
$Э_{con}$ = F. α. (t_k-t_u)	=	Heat transfer from the compressor to the evaporator convection
C_{com}	=	The heat capacity of the compressor
T_{com} T_{eva}	=	The surface temperature of the evaporator and the compressor
Δd	=	Width of the gap between the inner cylinder and the inner surface of the outer hollow cylinder (distance between the surfaces of the compressor to evaporator)
F	=	Cylindrical surface area of the compressor through which the heat exchange process with the inner cylindrical surface of the evaporator
K. W.τ	=	The heat generated by the compressor during operation
3Aecb: k	=	Ratio showing the proportion of electric power consumed by the compressor which is released in the form of heat through the surface of the compressor
W	=	Compressor motor power
τ	=	Time

$$Э_{eva} = \frac{T^4_{eva} - T^4_{com}}{\Delta d^2} \quad (12)$$

$$k \cdot w \cdot \tau - Э_{rad} - Э_{con} = \frac{C_{com} \cdot \partial T_{com}}{\partial \tau} \quad (13)$$

$$Э_{con} = F \cdot \alpha \cdot \left(t_{eva} - t_{comc}\right) \quad (14)$$

$$\frac{k \cdot w \cdot \tau - T^4_{eva} - T^4_{com}}{\Delta d^2 - F \cdot \alpha \cdot \left(t_{eva} - t_{comc}\right)} = \frac{C_{com} \cdot \partial T_{com}}{\partial \tau} \quad (15)$$

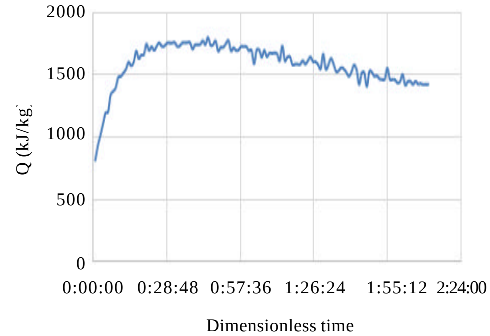

Fig. 9: Dependence of the dimensionless heat from the Q dimensionless time

$$Q = k \cdot w \cdot \tau - T^4_{eva} - T^4_{com} - F \cdot \alpha \cdot \left(t_{eva} - t_{com}\right) - C_{com} \cdot \partial T_{com}/\partial\tau \tag{16}$$

Differentials τ:

$$Q - k \cdot w \cdot \tau - T^4_{com} - T^4_{eva}/\Delta d^2 = -F \cdot \alpha \cdot \left(t_{eva} - t_{com}\right) - C_{com} \cdot \partial T_{com}/\partial\tau \tag{17}$$

$$Q - k \cdot w \cdot \tau - T^4_{com} - T^4_{eva}/\Delta d^2 + F \cdot \alpha \cdot \left(t_{eva} - t_{com}\right) = -C_{com} \cdot \partial T_{com}/\partial\tau \tag{18}$$

$$Q - k \cdot w \cdot \tau - T^4_{com} - T_{eva}/\Delta d^2 + F \cdot \alpha \cdot \left(t_{eva} - t_{com}\right)/C_{com} = \partial T_{com}/\partial T \tag{19}$$

The resulting equation integrate with respect to τ and find τ:

$$1/F \cdot \alpha \int -d\left(t_{eva} - t_{com}\right) F \cdot \alpha + Q - k \cdot w \cdot \tau - T^4_{Com} - T^4_{eva}/\Delta d^2/Q - k \cdot w \cdot \tau - T^4_{com} - T^4_{eva}/\Delta d^2 + F.\alpha(t_{com} - t_{eva}) \tag{20}$$

Next, we find:

$$Q = \frac{kw\tau - \dfrac{T_x^4 - T_{eva}^4}{\Delta d^2} - F \cdot \alpha \cdot t_{con} - e^{-\tau F\alpha/C_{com}}}{\dfrac{T_{con}^4 - T_{eva}^4}{\Delta d^2}} + t_{evac} \tag{21}$$

$$Q = \frac{kw\tau - \dfrac{T_k^4 - T_{eva}^4}{\Delta d^2} - F \cdot \alpha \cdot t_{eva} - e^{-\tau F\alpha/C_{com}}}{\dfrac{T_k^4 - T_{eva}^4}{\Delta d^2}} + t_{con} \tag{22}$$

Method for computing was developed using the techniques above, determine the amount of heat and energy balance equation (Fig. 9 and 10).

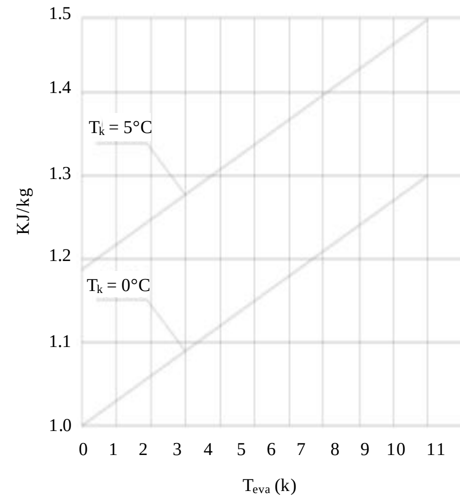

Fig. 10: The dependence of the compressor from the temperature difference in the condenser and evaporator

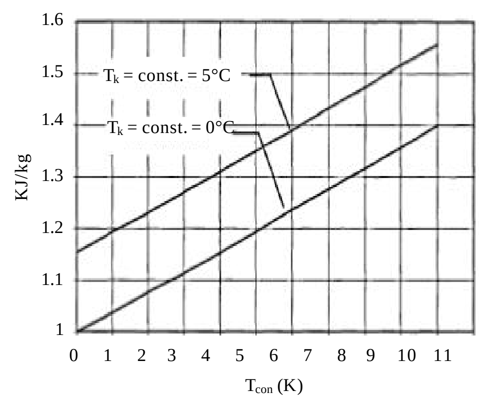

Fig. 11: Dependence of the compressor from the temperature difference in the evaporator

The losses in the capacitor: In the analysis of external losses should also take into account the loss in the evaporator. Temperature in than the temperature of researching fluid in the evaporator (Fig. 11). Consequently, this loss is consistent with the loss of external irreversibility due to the presence of a finite temperature difference between the condensing agent and the coolant in the condenser and is caused by the difference between the temperature and the boiling point working fluid in the evaporator. At equality of the irreversibility of the temperature in the evaporator stronger impact on energy cycle costs than in the condenser irreversibility. Increased T and increases the compressor and reduces the conversion factor heat pump bution of heat flows in heat exchangers (Fig. 11). Located in the gaseous researching agent in the condenser is first cooled to the saturation temperature (overheating

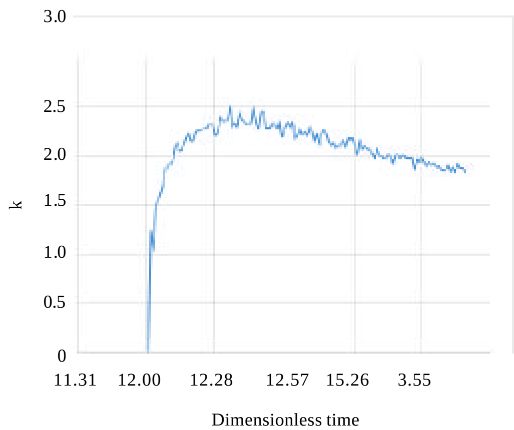

Fig. 12: The dependence of the energy conversion efficiency of the heat pump on the dimensionless time

weaning process) and then condensed. The process of heat exchange between the working agent and heated medium passes under a large temperature difference (Fig. 12) which leads to significant losses. The temperature difference between the condensing coolant and working agent is denoted by Tcon in Fig. 12, tcon-temperature pressure in the condenser. Logically, from the energy point of view strive to decrease the limiting temperature difference tcon and on the other hand this could lead to a substantial increase of the condenser surface and consequently, a considerable increase in installation costs. Agents for various operating values of heat transfer coefficients in the condensation are different therefore the heat transfer coefficients and the values for them are too dissimilar. Therefore with the same surface of the capacitor for different operating agents necessary to maintain unequal difference Tcon temperature. Thus, a worker agent with a higher heat transfer coefficient, ceteris paribus will operate at a lower temperature difference and it will lead to lower energy consumption. Temperature pressure determines the degree of perfection of the heat exchanger, in this case from the perfection of the capacitor depends on the value of tcon. Increasing Tcon increases the compression work of the compressor heat pump. Analysis of the data shows that the temperature difference T and significantly affects the operation of the compressor (the change of tcon 1 K causes a change in the compressor 3%) and efficiency heat pumps (with increasing tcon and 1 K conversion coefficient is reduced by 4-5% (Fig. 11 and 12).

The losses in the evaporator: Temperature Tver (VER) is higher than the temperature of the working agent t eva in the evaporator. Consequently, this loss is consistent with the loss of external irreversibility due to the presence of loss in the evaporator. Temperature tver (RES) (Fig. 11) than the temperature of the researching agent Teva in the evaporator. Consequently, this loss is consistent with the loss of external irreversibility due to the presence of a finite temperature difference between the condensing agent and the coolant in the condenser and is caused by the difference between the RES and the temperature of working fluid in the evaporator boiling. At equality tcon and teva irreversibility evaporator stronger effect on the energy consumption in the cycle than in the condenser irreversibility. The results of the settlement analysis of the influence of the temperature difference in the evaporator to the compressor ty and conversion coefficient μ shown in Fig. 11. As follows from the data presented in Fig. 12 and 2.9. The impact Teva is the same as Tcon. Increased T and increases the compressor and reduces the conversion factor heat pump temperature difference between the condensing agent and the coolant in the condenser and is caused by the difference between the VER temperature and the boiling working fluid in the evaporator.

CONCLUSION

During operation of the compressor, performing compression research of the refrigerant is heated to a temperature of the evaporator is cooled up to temperature. Accordingly, the heat radiation is formed between them, heat transfer and convection. Thus, the surface receives a constant evaporator temperature the surface temperature of the compressor will increase to a steady state value. The regularities of its change in the course of the resulting balance.

In the analysis of external losses should also take into account the loss in the evaporator. Ambient temperature is higher than the temperature of researching fluid in the evaporator. Consequently, this loss is consistent with the loss of external irreversibility due to the presence of a finite temperature difference between the condensing agent and the coolant in the condenser and is caused by the difference between the temperature and the boiling point researching fluid in the evaporator.

Ambient temperature is higher than the temperature of working fluid in the evaporator. Consequently, this loss is consistent with the loss of external irreversibility due to the presence in the evaporator loss.

REFERENCES

Al-Sulaiman, F.A., I. Dincer and F. Hamdullahpur, 2011. Exergy modeling of a new solar driven trigeneration system. Solar Energy, 85: 2228-2243.

Bejan, A., D.W. Keary and F. Kreith, 1981. Second law analysis and synthesis of solar collector systems. Energy Eng., 103: 23-28.

Brandi, H., 1998. Energy piles and diaphragm walls for heat transfer from and into the ground. Proceeding of the 3th International Geotechnical Seminar on Deep Foundations on Bored and Auger Piles, October 19-21, 1998, Technical University Of Kenya, Ghent, Belgium, ISBN:90-5809-022-1, pp: 38-60.

Brandl, H., 2006. Energy foundations and other thermo-active ground structures. Geotech., 56: 81-122.

Dutta Gupta, K.K. and S.K. Saha, 1990. Energy analysis of solar thermal collectors. Renewable Energy Environ., 1990: 283-287.

Farahat, S., F. Sarhaddi and H. Ajam, 2009. Exergetic optimization of flat plat solar collectors. Renewable Energy, 34: 1169-1174.

Hawlader, M.N.A., S.K. Chou and M.Z. Ullah, 2001. The performance of a solar assisted heat pump water heating system. Appl. Therm. Eng., 21: 1049-1065.

Huang, B.J., J.P. Lee and J.P. Chyng, 2005. Heat-pipe enhanced solar-assisted heat pump water heater. Solar Energy, 78: 375-381.

Jafarkazemi, F. and E. Ahmadifard, 2013. Energetic and exergetic evaluation of flat plate solar collectors. Renewable Energy, 56: 55-63.

Kinab, E., D. Marchio, P. Riviere and A. Zoughaib, 2010. Reversible heat pump model for seasonal performance optimization. Energy Build., 42: 2269-2269.

Liu, G., Y.A. Cengel and R.H. Turner, 1995. Eclmicail Briefs. J. Solar Energy Eng., 117: 249-251.

Luminosu, I. and L. Fara, 2005. Determination of the optimal operation mode of flat solar collector by exergetic analysis and numerical simulation. Energy, 30: 731-747.

Madani, H., J. Claesson and P. Lundqvist, 2011. Capacity control in ground source heat pump systems: Part I; Modeling and simulation. Intl. J. Refrig., 34: 1338-1347.

Qiao, H., V. Aute, H. Lee, K. Saleh and R. Radermacher, 2013. A new model for plate heat exchangers with generalized flow configurations and phase change. Intl. J. Refrig., 36: 622-632.

Santa, R., 2012. The analysis of two-phase condensation heat transfer models based on the comparison of the boundary condition. Acta Polytech. Hungarica, 9: 167-180.

Siroky, J., F. Oldewurtel, J. Cigler and S. Privara, 2011. Experimental analysis of model predictive control for an energy efficient building heating system. Appl. Energy, 88: 3079-3087.

Suzuki, A., 1988. A fundamental equation for exergy balance on solar collectors. ASME. J. Sol. Energy Eng., 110: 102-106.

Torres, R.E., J.G.C.D. Gortari, B.A.I. Salazar and M.P. Nunez, 2001. A design method of flat-plate solar collectors based on minimum entropy generation. Exergy Intl. J., 1: 46-52.

Tyagi, S.K., S. Wang, M.K. Singhal, S.C. Kaushik and S.R. Park, 2007. Exergy analysis and parametric study of concentrating type solar collectors. Int. J. Therm. Sci., 46: 1304-1310.

Zhai, R., Y. Zhu, Y. Yang, K. Tan and E. Hu, 2013. Exergetic and parametric study of a solar aided coal-fired power plant. Entropy, 15: 1014-1034.