EFFICIENCY OF THE HEAT PUMP COOPERATING WITH VARIOUS HEAT SOURCES IN MONOVALENT AND BIVALENT SYSTEMS

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CSBE101186 – Presented at Section IV: Rural Electricity and Alternative Energy Sources conference

ABSTRACT This paper presents the findings of tests carried out on the efficiency of compressor heat pumps cooperating with various types of lower heat sources. Lower heat sources are as follows: horizontal ground heat exchangers, vertical exchangers and sources operating in the bivalent system. The system for receiving energy comprised a traditional heating system and liquid-air exchangers. A strong relationship between the heating efficiency of the analysed systems and temperature inside the structure was noted. Furthermore, it was indicated that for heat requirements of approximately 1 $MJ \cdot m^{-2}$ the applied bivalent system was fully capable of meeting this heat requirement.

Keywords: coefficient of performance, heat pump, bivalent system.

INTRODUCTION The limited amount of natural resources, including fossil fuels, and the capacity of the natural environment to accept pollution without dangerous changes occurring in the functioning of the global ecosystem, constitute the basis for taking measures in order to substitute fossil fuels with renewable sources of energy, including the use of heat pumps. Heat pumps may be operated mono- or bivalently. Under the monovalent system the heat source may be e.g. soil (heat exchangers both in horizontal and vertical systems). In turn, under the bivalent system, heat is transferred by the pump from the tank which stores the liquid which is heated up through the conversion of sunlight in the collectors. The possibility of using heat pumps in the heat systems of every heat reception system depends not only on energy and ecology but also on economic matters. Economic factors concern efficiency which depends on the type and condition of the lower hear source, as well as the manner of receiving the heat from the upper source. Research on the use of heat pumps has been carried out at many research centres. Hamdan et al (1992) analysed the use of renewable sources of energy for heating glasshouses. Analyses were carried out on the operating efficiency of independent sources (heat pumps, solar collectors) and bivalent (hybrid) systems in the form of combined pump and solar collector action. The above researchers noted that under test conditions (a region with a high level of sun exposure) the lowest costs connected with the supply of heat were generated with the use of the bivalent system. Xu et al (2006) analysed the energy effects of heat pumps in which use was made of atmospheric air heated up in flat liquid collectors as the lower heat source. Energy from the upper source was used for heating processing water. Kaygusuz and Ayhan (1999) described and analysed the heat pump cooperating system (atmospheric air was used as the lower source) with the upper source, in which the energy was stored in an accumulator filled with a body subject to phase transition. They defined the Coefficient of Performance (COP) system. Nagano et al. (2006) elaborated an innovative system for visualising and analysing heat pump operating efficiency which makes use of the ground exchanger as a lower source. The heat pump was used for heating purposes in a prototype glasshouse. In the final analysis the researchers presented synthetic heat operating indicators (work efficiency, financial ratios), whilst the energy effects were calculated in terms of limitation of the emission of substances released from the combustion of conventional fuels. Hawlader et al. (2001) researched the energy effects of the system in which the heat pump cooperated with the solar collector. Energy from the upper source was used for heating liquid in the accumulator. The researchers defined the Coefficients of Performance for given system constituents. They also defined the rate of return on financial outlay.

Kaygusuz (1995) performed system simulation tests in which the lower heat source drew energy from the conversion of sunlight in solar air collectors, whilst the heat pump cooperated with the accumulator, filled with a solid subject to phase transition. The tests were carried out under laboratory conditions. Following analyses Kaygusuz defined the energy effects and indicated the legitimacy of conducting a thorough economic analysis for this system, which would be employed in real structures.

Yumruta and Unsal (2000) elaborated a mathematical model for analysing heat pump cooperation, in which the lower heat source used water collected in the liquid accumulator located in the soil. The findings of water temperature variability simulation were depicted as a function of variable temperature of surrounding soil and total heat loss from the analysed accumulator. Hepbasli et al (2003) defined the energy effects of the system in which the compressor heat pump cooperated with vertical ground heat exchangers. Following analyses the thermal efficiency of ground heat exchanger pipes and the Coefficient of Performance were defined. Esen et al (2006) analysed the efficiency of heat pumps cooperating with ground heat exchangers. A mathematical model of heat exchange between the ground and piping constituting the lower heat source of the exchanger was developed. Research demonstrated a high degree of compliance between measured and calculated ground temperature with the use of a model. Trillat-Berdal et al (2006b) analysed heat pump operation in which the lower heat source was the intake of geothermal water and solar collectors. When using the existing numeric model they defined the operating parameters of the system under consideration and presented the energy and economic effects and the quantity findings of reducing the emission of harmful substances into the atmosphere. Research carried out by Kurpaska (2008) presented the findings of an energy analysis connected with the selection of lower heat source for cooperation with the compressor heat pump.

On the basis of the above presented research findings one may state that they do not comprehensively include the entire analysis on which the taking of a decision to use a heat pump for heating a garden structure depends. For this reason the carrying out of such an analysis is the main purpose of the paper.

MATERIAL AND METHOD The system under consideration comprises solar collectors, a tank for storing heated water, ground heat exchangers (horizontal and vertical), a compressor heat pump with buffer tank and a system of heat reception through a heating device located in the foil tunnel. The total length of the horizontal heat exchanger (with external diameter of 42 mm) was 300m, whilst the vertical exchangers (piping with the same diameter as in the case of the horizontal exchanger) constituted three boreholes, each 20 m deep, distributed in U layout (two sensors) and 2U layout – one ground sensor. In the foil tunnel, with overall surface area of $54m^2$, a standard heating system and two liquid-air exchangers were installed. The heat reception system comprised a system of heating pipes (total surface area of $18.2m^2$) and exchangers with heat exchange surface area equal to $6m^2$ each. Figures 1 and 2 present the gauge position layout. During experimentation the following were measured: surrounding climate parameters, the flow stream medium in: solar collectors, lower and upper heat pump source, feed temperature, return of the circulating medium and electricity consumption by system elements (the heat pump compressor and the circulating pumps).

These levels were gauged continuously with the use of the authors' gauge system which also permitted the archiving of gauge levels.

The Coefficient of Performance was calculated as follows:

$$COP = \frac{Q_{wew}}{P_{PC} \cdot \tau_{PC} + \sum_{i=1}^{2} P_{wym,i} \cdot \tau_{wym,i}}$$

where: Q_{wew}- heat delivered to the interior of the structure, J; P_{Pc}- electricity used by the heat pump, W; τ_{PC}, $\tau_{wym,i}$- respective operating time of the heat pump (τ_{PC}) and of circulating pumps in the heating system ($\tau_{wym,i}$), s; $P_{wym,i}$- electric power of the circulating pumps in the heating system, W.

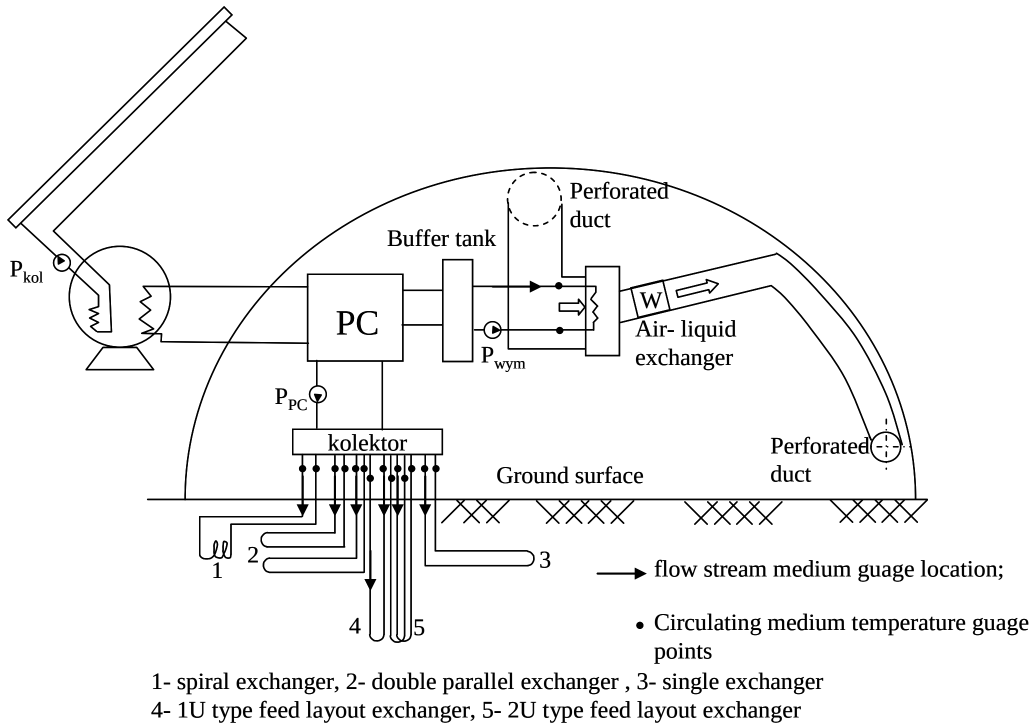


Fig. 1. Diagram depicting the gauge position and heat reception system

In turn, heat supplied to the interior of the structure (Q_{wew}) in $d\tau$ differential time is equal:

a) for liquid-air exchangers:

$$Q_{wew,wym} = \sum_{i=1}^{2} \left(\sum_{i=1}^{n} m_i \cdot c_p \cdot \left(t_{z,i} - t_{p,i} \right) \right) d\tau$$

b) for traditional heating systems:

$$Q_{wew,sg} = m \cdot c_p \cdot \left(t_z - t_p \right) d\tau$$

where: m- flow stream medium, $kg \cdot s^{-1}$; c_p- medium specific heat capacity, $J \cdot kg^{-1} \cdot K^{-1}$; t_z, t_p- feed temperature and return of the heating medium, ^oC.

All calculations were carried out in relation to the unit surface area of solar collectors.

When analysing the composition of the bivalent system (collectors – storage tank – heat pump) sunlight energy reaching the solar collectors was defined (E_s) as well as the heat supplied by the following collectors: liquid ($Q_{kol,\ pl}$) and vacuum ($Q_{kol,\ pr}$). On this basis, within the given time range ($d\tau$) these levels were calculated by means of the following relationship:

$$E_s = \sum_{i=1}^{n} R_z \cdot F_{k,pl(pr)} \cdot d\tau$$

$$Q_{kol,pl(pr)} = m_{pl(pr)} \cdot c_p \cdot \left(t_{z,pl(pr)} - t_{p,pl(pr)}\right) d\tau$$

where: R_z- sunlight intensity, W·m^{-2}; $F_{k,pl(pr)}$- solar collector surface area, m^2; symbols: *pl* and *pr* refer to the appropriate flat plate collector ($_{pl}$) and vacuum collector ($_{pr}$).

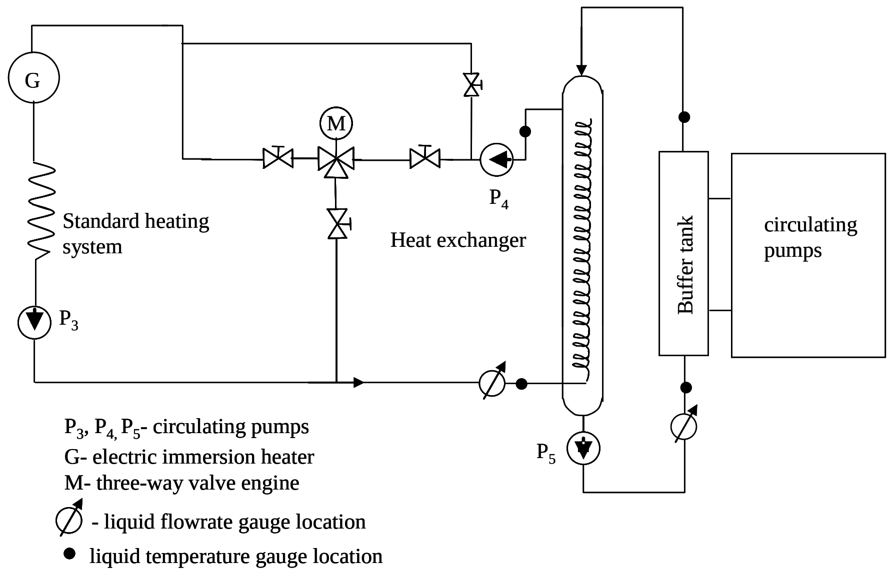


Fig. 2. Diagram depicting the hydraulic system feeding the standard heating system

When defining the heat pump operating efficiency in the bivalent system the following approach was employed:

$$\eta = \frac{Q_{wew,wym(sg)}}{\sum_{i=1}^{2} P_{kol,i} \cdot \tau_{kol,i} + P_{PC} \cdot \tau_{PC} + \sum_{i=1}^{2} P_{wym,i} \cdot \tau_{wym,i}}$$

Where, $P_{kol,\ i}$, electric power of circulating pumps in the collectors, W; $\tau_{kol,i}$ pump operating time in the collector system, s.

Below are presented the findings of experiments carried out on: horizontal exchangers, vertical exchangers and the hybrid system.

RESULTS AND DISCUSSION Research was carried out during the autumn and spring of 2008/2009. Figure 3 presents the thermal efficiency of the analysed heating systems installed in the structure. One may note their linear dependence between efficiency and temperature outside the structure. The reduced efficiency of heating systems at higher internal temperature and the absence of additivity in the thermal efficiency of individual heating systems (the total efficiency of individual heating systems is higher than the total efficiency of both exchangers) may be explained by the differentiated temperatures in feeding the circulating medium of both systems and the dependence of heat transfer intensity by the heating system on temperature inside the structure.

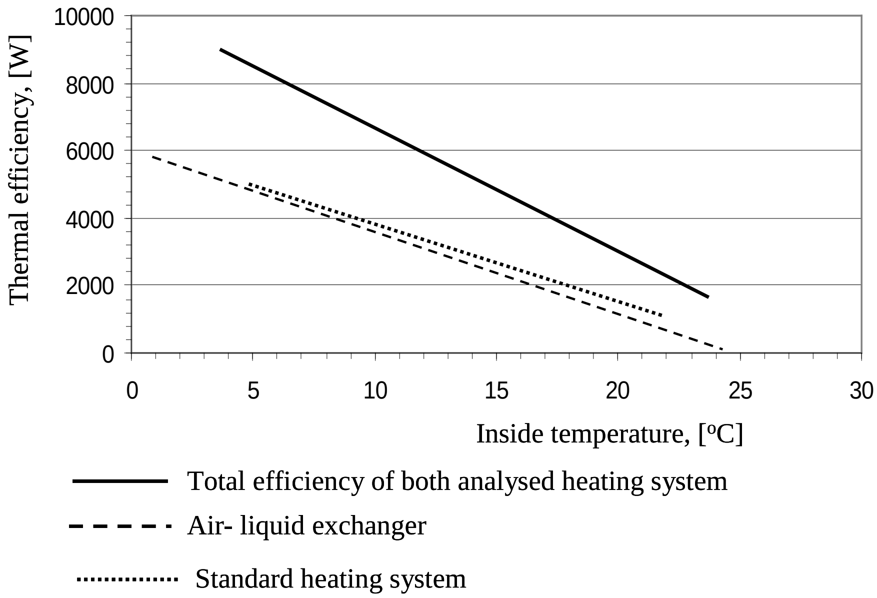


Fig. 3. Thermal efficiency of analysed heating systems

Figures 4 and 5 present COP for lower heat source in the form of horizontal heat exchangers (fig. 4) and vertical exchangers (fig. 5). As can be seen, average COP value depending on the type of exchanger is COP= 1.56 (horizontal ground heat exchangers) and 1.49 (vertical exchangers).

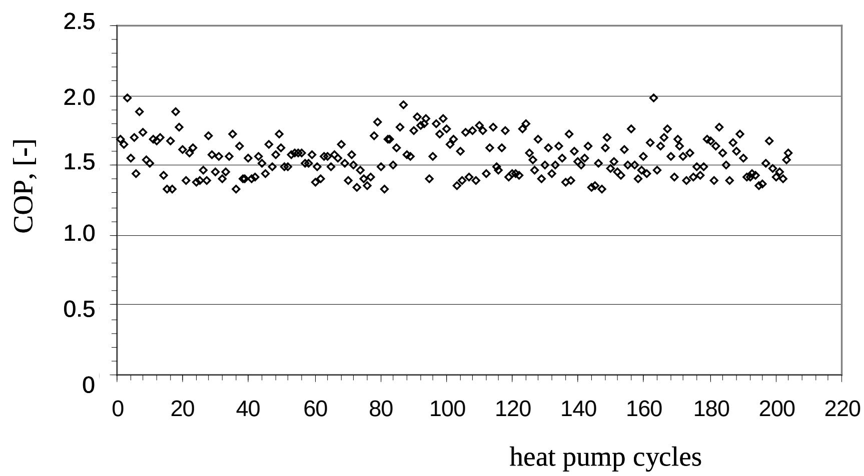


Fig. 4. Pump coefficient of performance during experiments with simultaneous heat distribution to the interior with the use of exchangers and of the traditional heating system (horizontal exchangers)

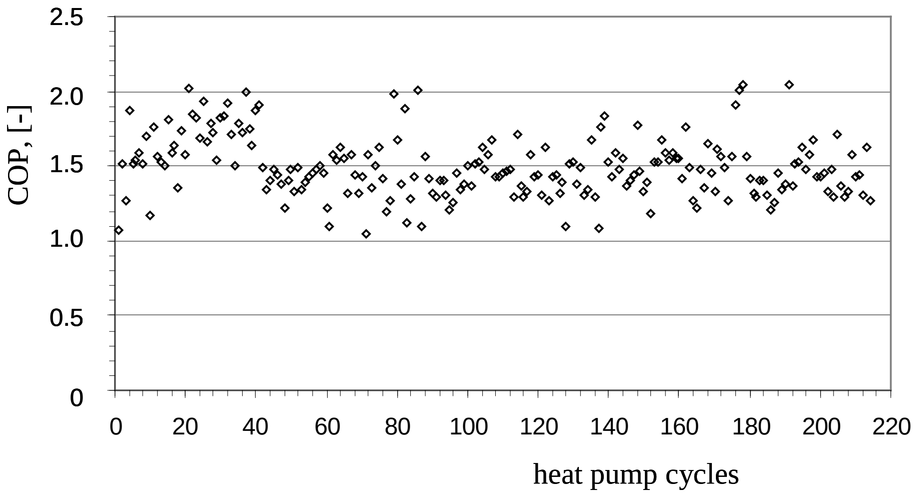


Fig. 5. Pump coefficient of performance during experiments with simultaneous heat distribution to the interior with the use of exchangers and of the traditional heating system (vertical exchangers)

Table 1 presents a synthetic calculation of differences depending on the types of lower heat source exchanges and the reception system.

Table 1. Relative differences (in relation to the traditional heat reception system) in the COP for the analysed experimental conditions

Type of lower heat source	Percentage difference in COP, %	
	Liquid-air exchangers	Exchangers + traditional heat distribution system
horizontal	18.7	21.2
vertical	17.3	23.1

An analysis of the presented calculation clearly shows that, irrespective of the type of lower heat source (vertical or horizontal ground heat exchangers), it is more advantageous to use additional liquid-air exchangers in the structure, apart from the traditional heat distribution system. It stems from data appearing in the table, as a result of applying the bivalent heat reception system, that growth in the COP of more than 21% is observed.

Concerning the bivalent system fig. 6 presents an example of average changes in liquid temperature in the tank together with designated heat pump operating cycles. During the experiment the accumulation tank contained 2.5m^3 of water. As a result of heat pump operations (total time over 11 cycles was 2.76 hours) average water temperature dropped (as a result of pump operations and heat loss into the environment) from approx. 35 ^{o}C to approx. 22 ^{o}C. In the presented time range (Fig. 6) the pump supplied almost 113 MJ of heat into the interior of the structure over 11 cycles. Scope of change of the amount of collected heat is between 400MJ and 1900MJ; each subsequent heat pump operating cycle collected less energy as a result of worse heat exchange of the lower source.

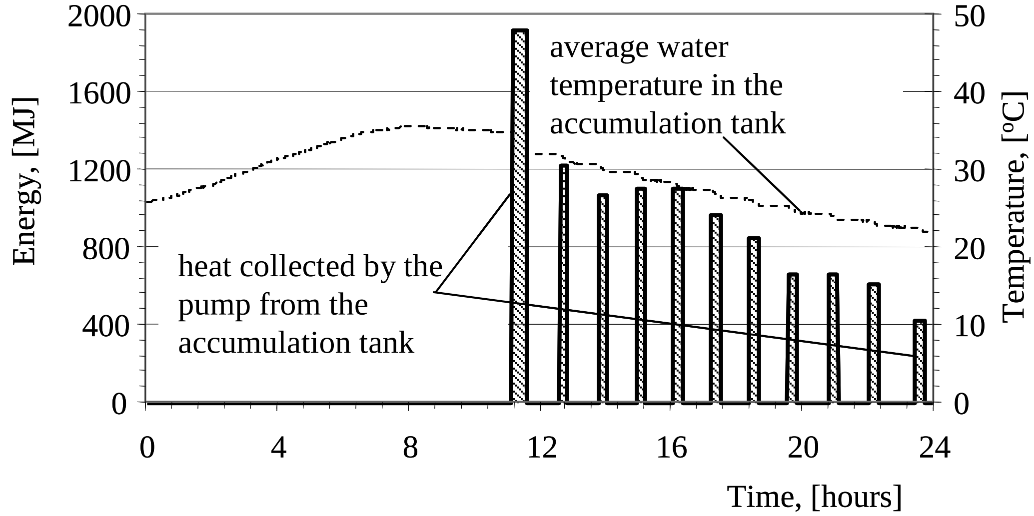
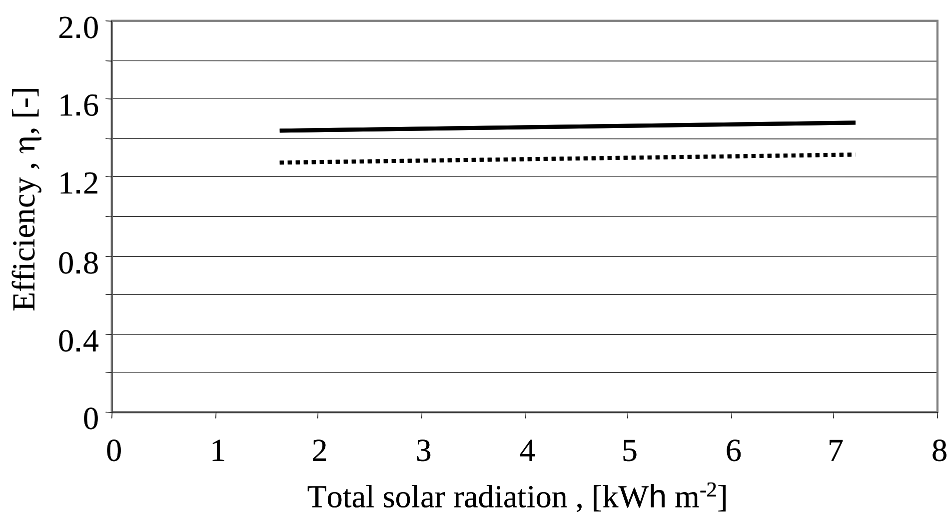


Fig. 6. The timing of water temperature changes in the tank and the amount of heat provided to the interior of the structure during the experiment in terms

When calculating the work efficiency of the bivalent system (Fig. 7) an analysis was carried out on all work cycles for differentiated capacities of the storage tank (between $2m^3$ and $5m^3$ of water). One notes the lack of dependency between the presented amounts and the sum of sunlight. It is true that solar energy as a result of conversion led to higher water temperature in the accumulation tank, but for the approved system operation cycle period there was no direct connection between these parameters (the sum of sunlight energy) and the average temperature of water in the lower source. Average efficiency value was 1.45 (heat pump) and 1.35 (for the entire hybrid



system).

Fig. 7. Efficiency of the bivalent system and of the heat pump in total sunlight function.

In turn, fig. 8 depicts the quantity of energy in the lower and upper heat pump heat source in reference to the structure's requirement for heat (the requirement for heat was calculated during liquid-air heat exchanger operations at 15°C, whilst energy was calculated for the system operating cycle period).

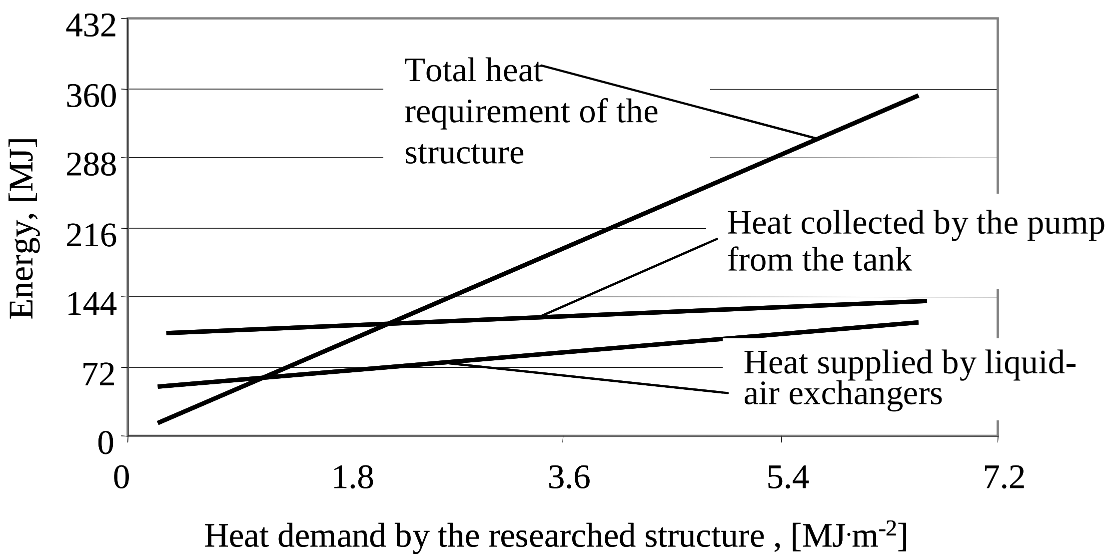


Fig. 8. Energy collected by the heat pump and energy supplied to the inside of the structure in terms of structure heat demand

When the structure requires a large amount of heat the energy supplied to the interior is less than the energy collected by the heat pump from the lower source. In turn, when the structure requires a small amount of heat (up to approx. 0.9 MJ·m^{-2}) the amount of heat supplied by the exchangers exceeded demand. Heat, which was collected by the heat pump from the accumulation tank could theoretically be enough to satisfy the heat demand of the structure at a level of approx. 1 MJ·m^{-2}.

CONCLUSION 1. The average COP of the pump cooperating with the ground exchangers, depending on their type, was between 1.49 (vertical exchangers) and 1.56 (horizontal ground heat exchangers).

2. Irrespective of the type of lower heat source (vertical or horizontal ground heat exchangers), it is more advantageous to use additional liquid-air exchangers in the structure, apart from the traditional heat distribution system.

3. Average efficiency value for each system was 1.45 (heat pump) and 1.35 (for the entire bivalent system).

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