RESEARCH ON REVERSIBLE HEAT PUMP INSTALLATION FOR GREENHOUSE HEATING
/
ИЗСЛЕДВАНЕ НА РЕВЕРСИВНА ТЕРМОПОМПЕНА ИНСТАЛАЦИЯ ЗА ОТОПЛЕНИЕ НА ОРАНЖЕРИИ

Lecturer Ph.D. Eng. Kolev Z. D.[1], Lecturer Ph.D. Eng. Kadirova S. Y.[*2], Student Nenov T. R.[3]
[1] Rousse University "Angel Kunchev", Agrarian – Industrial Faculty / Bulgaria
[2] Rousse University "Angel Kunchev", Faculty of Electrical Engineering, Electronics and Automation / Bulgaria
[3] Rousse University "Angel Kunchev", Faculty of Electrical Engineering, Electronics and Automation / Bulgaria
Tel: 00359 877 089537; E-mail: skadirova@uni-ruse.bg

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ABSTRACT
The aim of the article is to investigate the influence of the airflow velocity on the main parameters of a reversible heat pump installation, as the airflow crosses the external surface of heat exchange apparatus of the "water – air" convector. The main installation parameters, working in "heating" regime at specific values of airflow velocity have been investigated. The investigated parameters are: heat convection coefficients and convector heat transfer coefficient; internal and external installation heat flows; heat pump performance coefficient.

РЕЗЮМЕ
Целта на публикацията е да се изследва влиянието на скоростта на въздушния поток, обтичащ външната повърхност на топлообменния апарат на „водо – въздушен" конвектор, върху основните параметри на реверсивна термопомпена инсталация. Изследвани са основните параметри на инсталацията, работеща в режим отопление, при конкретни стойности на скоростта на въздушния поток. Изследваните параметри са: коефициенти на топлопредаване и коефициент на топлопреминаване на конвектора; вътрешен и външен топлинни потоци на инсталацията; коефициент на трансформация на термопомпата.

INTRODUCTION
Nowadays, greenhouse farming is a growing industry in many countries because of world population increase. That is why greenhouse food production is an additional alternative for meeting increased food demand year around. The way to produce greenhouse crops is very expensive and there are many variables to consider before the farmer decides to take this route. All plant growth factors can be controlled and maintained at optimum level in year around in the greenhouses (*Esen M., Yuksel T., 2013)*.

In developing countries, greenhouses are small-size enterprises, which are generally established by farmers. In general, these are the ones that don't need heating or are heated by farmers through their own methods. The increase in nutrition needs and the rise in the standard of nutrition which is consumed have made greenhousing more important (*Benly H., 2010; Yang S., Rhee J., 2013).*

The energy consumption in agriculture has increased considerably with the introduction of high-yielding varieties and mechanized-crop production practices. A higher heating cost for greenhouses using natural gas or oil has resulted and many growers have preferred to choose alternative energy sources (*Benly H., 2010; Biris S. St. et al, 2009; Constantinescu D. M. et al, 2009; Russo G. et al, 2014).* This problem is very up-to-date, taking into account the significant price increase of natural gas in Europe at the moment.

Underground water source heat pumps are a highly efficient, renewable energy technology for space heating and cooling. These technologies rely on the fact that the underground water sources have a relatively constant temperature (*Benly H., 2010; Kurpaska S., 2011).*

Greenhouse heating is one of the fields requiring high energy consumption during the cold seasons *(Awani S. et al, 2015)*. It's important to optimize the heat pump performance coefficient in the heating installations, which is an indicator of their efficiency. This coefficient depends on the heat flow that the heaters give to the air in the greenhouse. On another hand, the heat flow depends on the settings of the heaters, which determine the coefficients of heat exchange.

In this article, the influence of the fan setting of "water – air" convector in a reversible heat pump installation on the main installation parameters has been investigated. The investigated parameters are: heat convection coefficients and convector heat transfer coefficient; internal and external installation heat flows; heat pump performance coefficient. The researches have been performed under laboratory conditions.

The aim of the article is to obtain results for quantitative change of the investigated parameters, which can be used for greenhouse microclimate maintenance, and optimization of the energy costs for heating.

MATERIAL AND METHODS

Object of research

The principal scheme of the heat pump installation is shown in Fig.1. The heat pump is of "water – water" type, CEAT brand, Aurea 20 model. The convector is of "water – air" type, BUMYANG brand, FVC20MLL2 model. Under "heating" regime, the heat pump heat exchanger in the external circle works as an evaporator, and the internal one works as a condenser. In the evaporator, the refrigerant takes heat energy from the cold water, circulating in the external circle, at the same time in the condenser the refrigerant gives heat energy to the hot water, circulating in the internal circle. The convector gives heat energy to the internal ambient air to ensure the required temperature (*Bobilov V. et al, 2011; Kolev Z. et al, 2015; Kolev Z. et al, 2015b*).

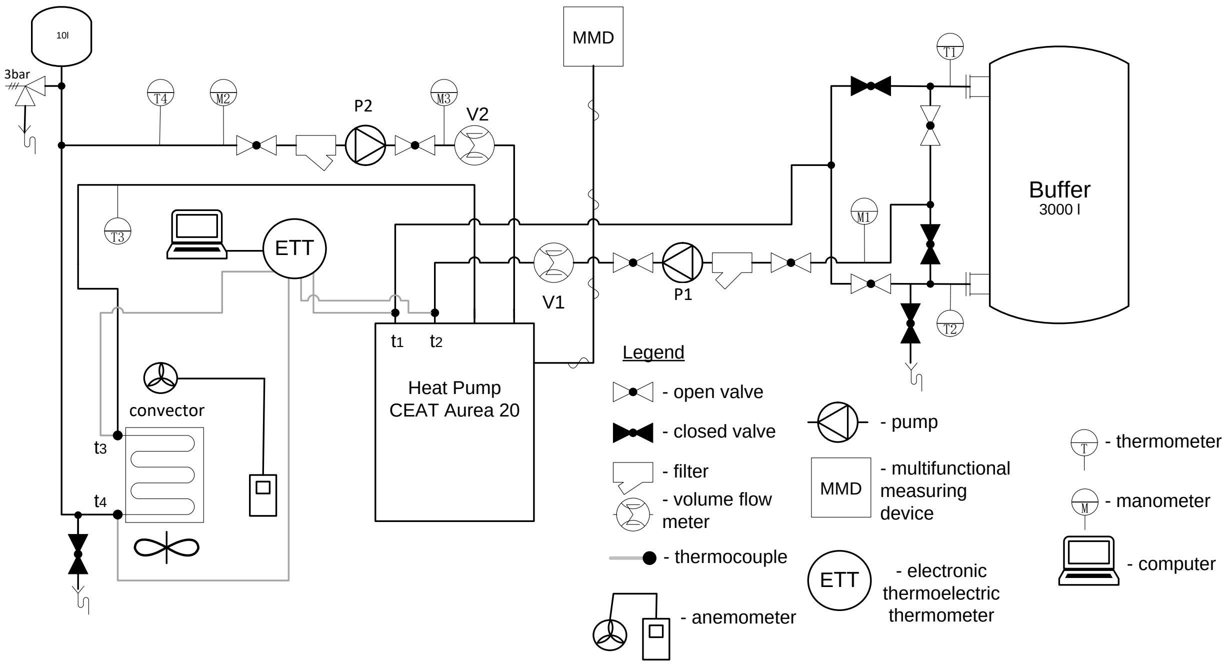
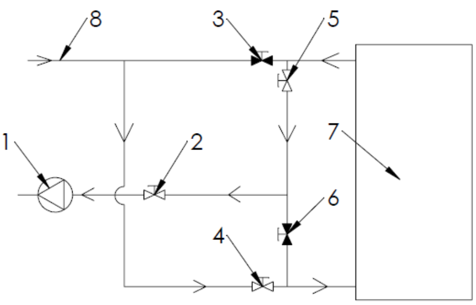


Fig. 1 - Main scheme of the heat pump installation

Under "heating" regime the water in the buffer represents the heat source, from which the installation takes heat energy, ensuring heat pump working. For the installation to work for a long time without reducing considerably the temperature of the buffer water, it is necessary to achieve a stratification of the water in the buffer. The work scheme presenting the "heating" regime is shown in Fig. 2.



1 - circulation pump;

2, 4 and 5 - open valves;

3 and 6 - closed valves;

7 – water buffer;

8 – from heat pump

Fig. 2 - Work scheme presenting the "heating" regime

The convector's heat exchanger has been implemented as a two-pipe staggered ribbed tube sheaf. The tubes and ribs are made from a copper alloy. The main scheme of the convector's heat exchanger is shown in Fig. 3 *(Bobilov V. et al, 2011; Kolev Z. et al, 2015; Kolev Z. et al, 2015b)*.

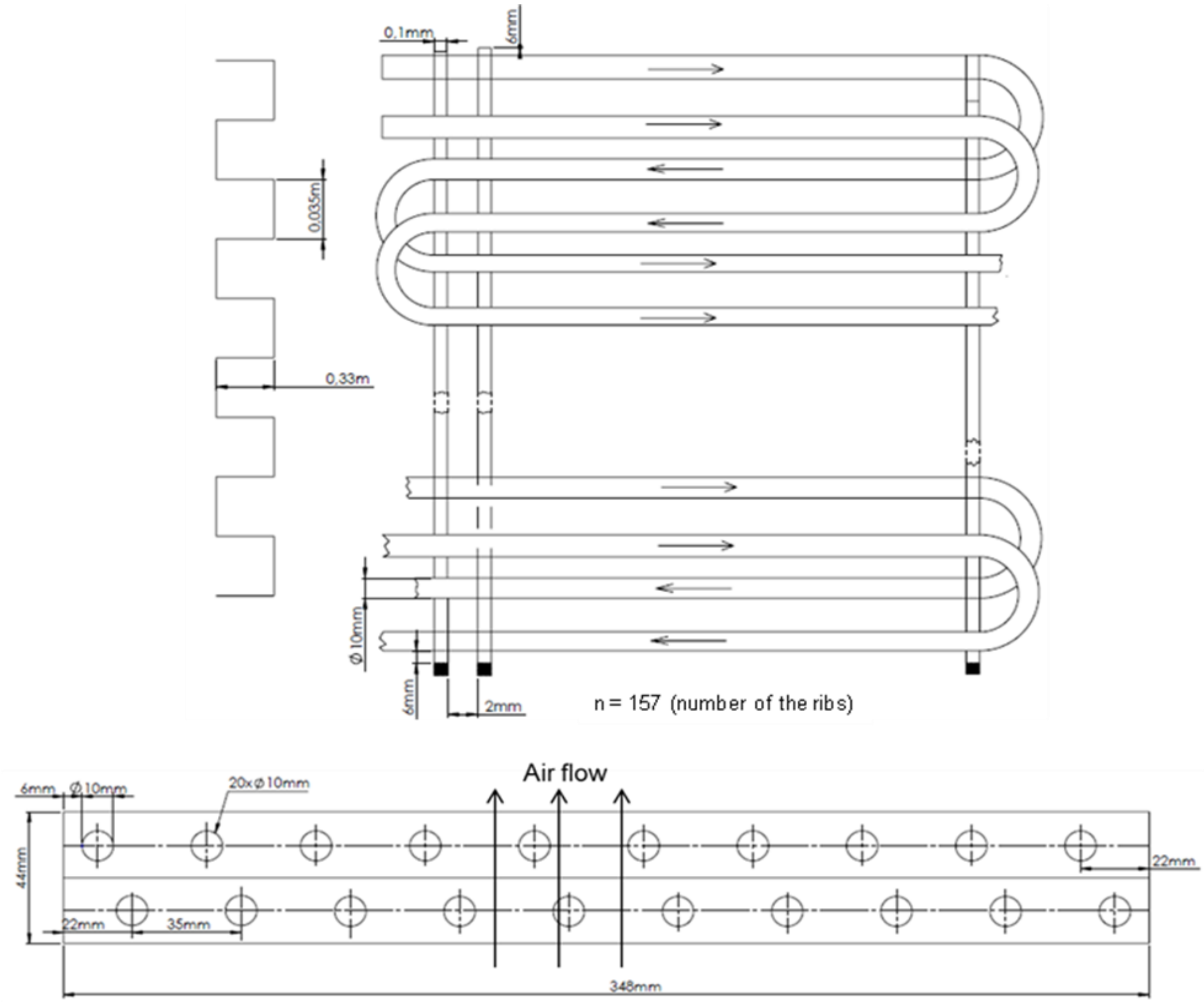


Fig. 3 - Main scheme of the convector's heat exchanger

In the convector's tubes is running water, which is heated or cooled by the heat pump, depending on the installation working regime. The pipe bundle is crossed by airflow, generated by the axial fan.

Methodology for determining the parameters

The actual velocity of the airflow in the air gap between heat exchanger tubes and ribs has been calculated by the equation (*Kolev Z. et al, 2015; Kolev Z. et al, 2015b*):

$$w_2 = w_{2-measured} \cdot \frac{S_2}{S_{2-air\ gap}} \ \text{[m/s]} \tag{1}$$

where: $w_{2-measured}$ is the measured airflow velocity at the convector's output, m/s;

S_2 - cross sectional area of the heat exchanger, crossed by the airflow, m²;

$S_{2-air\ gap}$ - cross sectional area of the air gap between the tubes and ribs, m².

The heat convection coefficient between the water in the convector tubes and their internal surface has been determined by the equation (*Kolev Z. et al, 2015; Kolev Z. et al, 2015b*):

$$\alpha_1 = \frac{Nu_1 . \lambda_1}{d_{tube1}} \ [\mathrm{W/(m^2K)}] \tag{2}$$

where: the criterion of Nuselt has been calculated by the equation $Nu_1 = 0{,}021.Re_1^{0,8}.Pr_1^{0,43}$;

λ_1 - thermal conductivity coefficient of the water in tubes, W/(m.K);

d_{tube1} - internal diameter of the tubes, m.

The heat convection coefficient between the external surface of the convector and the airflow has been determined by the equation (*Kolev Z. et al, 2015; Kolev Z. et al, 2015b*):

$$\alpha_2 = \frac{Nu_2.\lambda_2}{d_{tube\,2}} \quad [\mathrm{W/(m^2K)}] \tag{3}$$

where: the criterion of Nuselt has been calculated by the equation $Nu_2 = 0{,}25.\left(\frac{d}{t}\right)^{-0{,}54}.\left(\frac{D-d}{2.t}\right)^{-0{,}14}.Re_2^{0{,}65}.Pr_2^{0{,}4}$;

λ_2 - thermal conductivity coefficient of the air crossing heat exchanger external surface, W/(m.K);

$d_{tube\,2}$ - tubes external diameter, m.

The heat transfer coefficient between the water in the convector tubes and the airflow crossing the convector external surface has been determined by the equation for single layer flat wall, ignoring the thermal resistance of the tubes and ribs (*Kolev Z. et al, 2015; Kolev Z. et al, 2015b*):

$$U = \frac{1}{\frac{1}{\alpha_1} + \frac{1}{\alpha_2}} [\mathrm{W/(m^2K)}] \tag{4}$$

The heat flows $\dot{Q}_{con}$ and $\dot{Q}_{eva}$ have been determined by the basic calorimetric equation, (*Bobilov V. et al, 2011; Kolev Z. et al, 2015; Kolev Z. et al, 2015b*):

$$\dot{Q} = \dot{m}.c_{pm}.\Delta t \quad [\mathrm{W}] \tag{5}$$

where: $\dot{m}$ is the water mass flow rate, kg/s;

c_{pm} - water mass heat capacity, J/(kg.K);

Δt - water temperature difference, K.

$\dot{Q}_{con}$ is the heat flow in the internal installation circle which has been exchanged in the heat pump condenser. On the other hand, this is the heat flow between the convector and the ambient air.

$\dot{Q}_{eva}$ is the heat flow in the external installation circle which has been exchanged in the heat pump evaporator.

The heat pump performance coefficient under "heating" working regime (winter regime) has been determined by the equation (*Bobilov V. et al, 2011*):

$$COP|_W = \left.\frac{\dot{Q}_{con}}{W_{comp}}\right|_W \tag{6}$$

where W_{comp} is the electric power, consumed by the heat pump, W.

The installation input parameters for the research are presented in Table 1.

Table 1

Installation input parameters

Set of the convector's fan	Set of the heat pump	Internal water volume flow rate	External water volume flow rate	Temperature of the ambient air	Actual airflow velocity
"LOW" Degree	The switch ON temperature of the heat pump is *t₄ = 42 ºC* The switch OFF temperature of the heat pump is *t₄ = 45 ºC*	$\dot{V} = 0.000148\, m^3/s$	$\dot{V} = 0.000411\, m^3/s$	$t_{ambient\ air} = 14.2\ ^oC$	$w_2 = 0.92\, m/s$
"MID" Degree				$t_{ambient\ air} = 15.2\ ^oC$	$w_2 = 2.51\, m/s$
"HIGH" Degree				$t_{ambient\ air} = 15.2\ ^oC$	$w_2 = 5.06\, m/s$

The ambient temperature $t_{ambient\ air}$ has constant value during the experiment.

The average temperature of the water in the buffer remains approximately equal to $t_{ambient\ air}$.

RESULTS

- **Research on the influence of airflow velocity w_2 on the heat convection coefficient α_1**

The change of the coefficient α_1 depending on the velocity w_2 of the airflow in the air gap between the heat exchanger tubes and ribs is presented in Fig. 4.

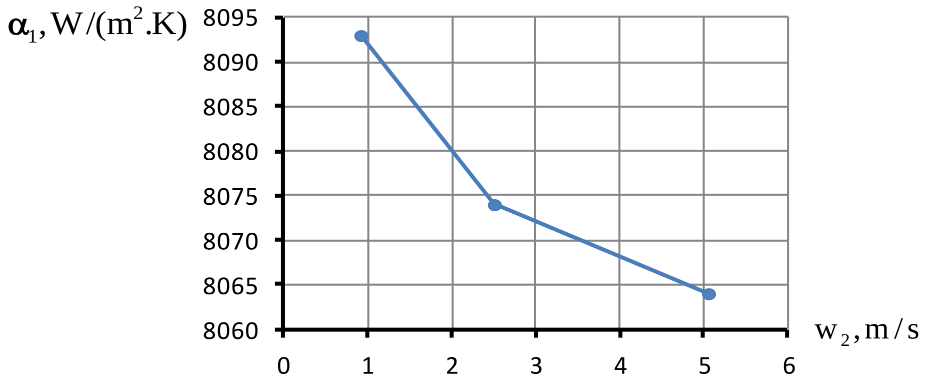


Fig.4 - Graphical dependence between α_1 and w_2

The reason for the decrease of the heat convection coefficient α_1 by increasing the airflow velocity is the decrease of the average water temperature in the internal circle. The increase of the water temperature leads to the increase of water kinematic viscosity, which in turn leads to the decrease of the Reynolds criterion.

- **Research on the influence of the airflow velocity w_2 on the heat convection coefficient α_2**

The change of the coefficient α_2 depending on the velocity w_2 is presented in Fig. 5.

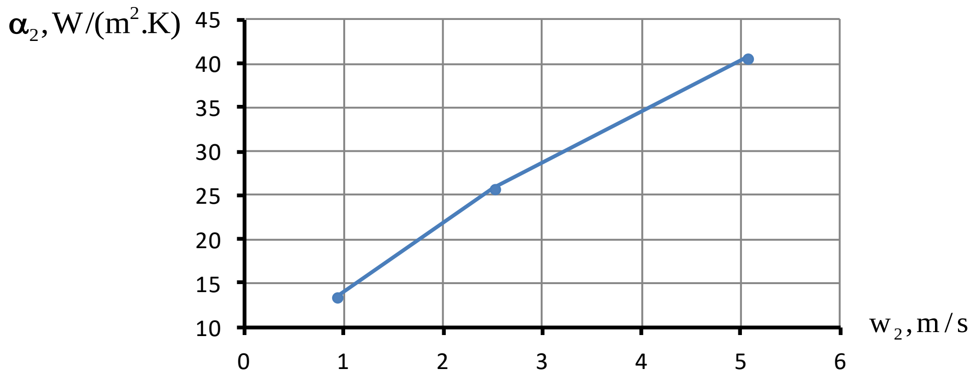


Fig. 5 - Graphical dependence between α_2 and w_2

The increase of the heat convection coefficient α_2 results from the significant increase of Reynolds criterion by the increase of the airflow velocity. On the other hand, the temperature of the air crossing the convector is decreased and leads to a decrease of the water kinematic viscosity.

- **Research on the influence of the airflow velocity w_2 on the heat transfer coefficient U**

The change of the coefficient U depending on the velocity w_2 is presented in Fig. 6.

The reason for the increase of the heat transfer coefficient U by increasing the airflow velocity w_2 is the increase of the external heat convection coefficient α_2, which has a more significant influence on U than the decrease of the internal heat transfer coefficient α_1. It can be concluded that the values of U have insignificant deviation comparing to those of α_2 (the differences are of the orders of tenths of 1 W/(m^2.K)).

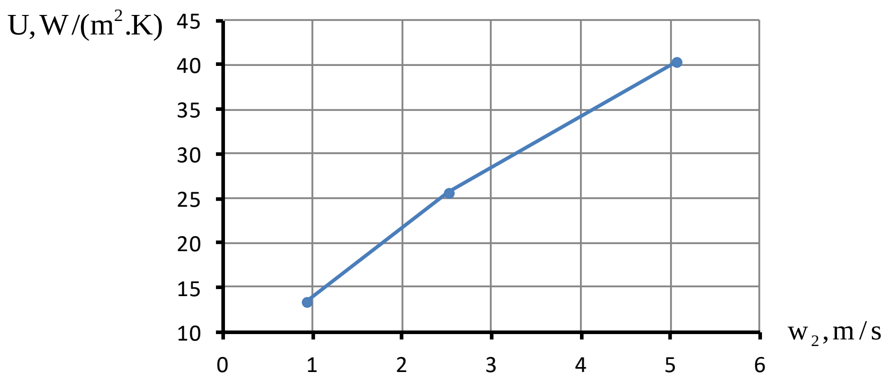


Fig. 6 - Graphical dependence between U and w_2

- **Research on the influence of the airflow velocity w_2 on the heat flow $\dot{Q}_{con}$**

The change of the heat flow $\dot{Q}_{con}$ depending on the velocity w_2 is presented in Fig. 7.

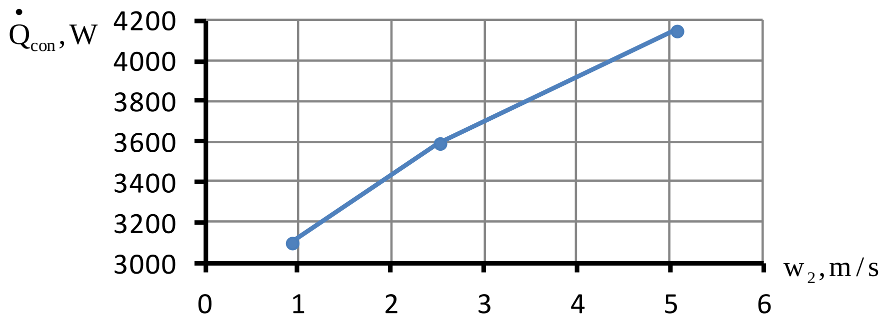


Fig. 7 - Graphical dependence between $\dot{Q}_{con}$ and w_2

The heat flow $\dot{Q}_{con}$ can be determined also based on the calculated heat convection coefficients and heat transfer coefficient.

Based on the basic calorimetric equation, the heat flow $\dot{Q}_{con}$ in the convector has been increased by increasing the airflow velocity, because the temperature difference "t_3(convector input) - t_4(convector output)" has increased. On the other hand, the water temperature in the internal circle has decreased, which leads to a decrease of the water density.

In terms of the "Newton – Rickman" law for the process of heat convection between the water and the internal surface of the tubes, the increase of the temperature difference "(average water temperature in the central section of tubes) - (average temperature of the inner tubes surface)" has a higher influence on the heat flow comparing to the decrease of the internal heat convection coefficient α_1.

From the perspective of the "Newton – Rickman" law for the process of heat convection between the external surface of heat exchange apparatus and the crossing airflow, the external heat convection coefficient α_2 has a significant influence on the heat flow.

Regarding the heat transfer process in the convector, the significant increase of the heat transfer coefficient U influences the heat flow.

- **Research on the influence of the airflow velocity w_2 on the heat flow $\dot{Q}_{eva}$**

The change of the heat flow $\dot{Q}_{eva}$ depending on the velocity w_2 is presented in Fig. 8.

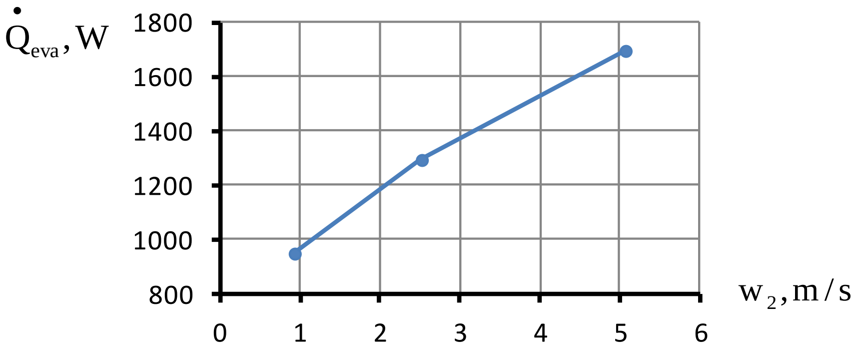


Fig. 8 - Graphical dependence between $\dot{Q}_{eva}$ and w_2

The heat flow $\dot{Q}_{eva}$ from the buffer water to the evaporating refrigerant in the evaporator increases with increasing airflow velocity through the convector, due to increased temperature difference "t_2 (buffer output) - t_1(buffer input)".

- **Research on the influence of the airflow velocity w_2 on the heat pump performance coefficient $COP|_W$**

The change of the performance coefficient $COP|_W$ depending on the velocity w_2 is presented in Fig. 9.

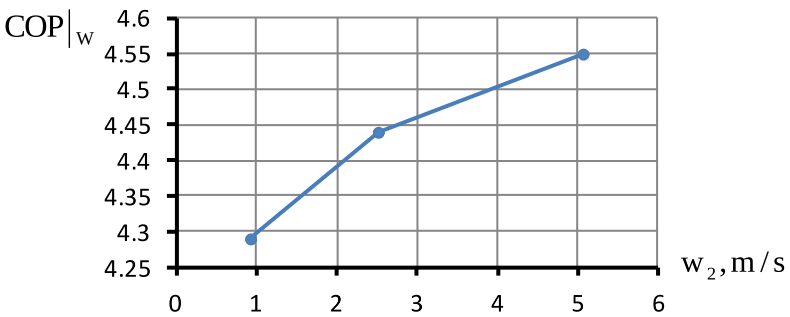


Fig. 9 - Graphical dependence between $COP|_W$ and w_2

The performance coefficient increases when increasing the airflow velocity. This results from the higher influence of the increase of the heat flow through the convector than the change of the average electrical power, consumed by the heat pump.

From the results observed, presented in Fig. 9 we can conclude that there is relatively low variation of the coefficient $COP|_W$ in terms of airflow velocity variation.

At a lower airflow velocity, i.e. less power consumption of the convector's fan, the operation efficiency of the heat pump heating installation will be obtained with a value close to that at a higher airflow velocity.

CONCLUSIONS

The main parameters of the installation working under "heating" regime at specific values of airflow velocity have been investigated. Based on them, the microclimate in greenhouses as well as the energy costs for heating can be optimized by correct control of the heat pump heating installation.

The change of the airflow velocity w_2 leads to a higher degree of change of the external heat convection coefficient α_2, respectively of the heat transfer coefficient U, comparing to the change degree of the internal heat convection coefficient α_1.

On the other hand the change of the airflow velocity w_2 leads to a higher change degree of the heat flow $\dot{Q}_{eva}$ exchanged by the heat pump evaporator, than the heat flow $\dot{Q}_{con}$ exchanged by the heat pump condenser.

At a lower airflow velocity, i.e. less power consumption of the convector fan, the operation efficiency of the heat pump heating installation will be obtained with a value close to that at a higher airflow velocity.

Last but not least, the used laboratory installation is suitable for additional researches about heat pump heating installations for greenhouses. The additional parameters of the installations which can be investigated are: the temperature of the ambient air, the temperature of the water energy source, the water volume flow rate in the internal installation circle, the settings of the heat pump (switch ON temperature, switch OFF temperature, as well as the temperature range between them).

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