DETERMINATION OF COP MAXIMUM OF COLD WATER LOOP OF HEAT PUMP HEATING SYSTEM BY MEANS OF NUMERICAL-GRAPHICAL OPTIMIZATION PROCEDURE

by

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The paper presents the energy optimization of the cold water loop of the heat pump heating system using analytical-numerical procedure. The aim of the study is obtain the maximum COP of the heating system by optimum of the wall water mass-flow rate and well pump power. The objective function is the heating system's COP. All components of the heating system: evaporator, condenser, compressor, circulation pump, and well pump are described by steady-state, lumped mathematical model. The model's equations are coupled, non-linear, multivariable and algebraic, the solution is feasible using an iterative numerical method. The MATLAB's program with Gauss elimination and Newton-linearization method is applied for solving the model. The obtained numerical data are presented in 3-D graphics. The optimum value of the cold-well water mass-flow rate is obtained from the graphics or by using a selection algorithm. The results of the study are the adequate mathematical model for energy optimization of the heating system, the numerical algorithm for solving the model and the ultimate goal to obtain the optimum of the power of well pump and compressor.

Key words: *energy optimum, heat pump, numerical procedure, well pump, compressor, COP of heating system*

Introduction

The COP is a crucial feature of heat pump heating systems, as it shows the energy efficiency of the system, which, in turn, is a vital indicator from the point of view for environmental protection and economy.

In the leading journals there are numerous studies and articles about the performance of the heat pump heating systems and their COP. The studies focus on various aspects, different mathematical models, methods and measurement systems.

Raiyan *et al*. [1] studied the air-to-air heat pump performance and COP with respect to the effects on the environment regarding the R22, propane-R290 and R22 mixtures of the refrigerants. A mixture of different compositions was prepared and designated with the X6 and X7. It was found that the X6 and X7 blend (R22 + R290) showed better performance than R22 alone in terms of COP, ozone depletion potential, and global warming potential. The problem is that the mixture is flammable due to the presence of propane.

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Sun, *et al*. [2] presented the case study about a method to predict the COP of the heat pump and the COP of ground source heat pump (GSHP) system with limited parameters. The method was based on an artificial neural network model and an adaptive neuro-fuzzy inference system model. The authors detected that the models provided high accuracy and reliability for calculating performance indexes of GSHP system.

Amoabeng *et al.* [3] tested the facility of the new designed heat pump calorimeter to investigate the performance and COP of a water-to-water heat pump using standard test conditions. The analysis of the test showed that the newly-designed calorimeter was able to save about over 75% of the total power consumption compared to the conventional calorimeter.

Tu *et al.* [4] in their work constructed a numerical model in order to evaluate the thermal performance by using single-well circulation (SWC) coupled GSHP systems. Numerical experiments were performed to observe the evolution of outlet temperature, the distribution of subsurface temperature field and the long-term development of outlet temperature. It was found that the thermal effective radius of SWC system is much larger than that of GSHP systems.

Kassai [5, 6] in his works dealt with the energy efficiency of cooling systems and heat pumps, but did not perform the energy optimization of the systems.

The careful and extensive analysis of the relevant literature highlighted the fact that the articles failed to address the heat pump heating system energy optimization. Instead, the papers studied the system performance and COP in the different ways, and with special purposes.

One of the few who has studied this field is Nyers, dealing with the energy optimization of the hot water loop of heat pump heating systems in his articles [7, 8] and dissertation [9].

The present research, however, investigates and determines the maximum COP or maximum energy efficiency of the cold water loop of the heat pump heating system using a numerical-graphical procedure. The optimization procedure involves creating a mathematical model corresponding to the objective function and to the applied mathematical procedure to solve the model.

The mathematical model describes the stationary behavior of the heat pump heating system and consists of the lumped, coupled, non-linear, multi-variables and large numbers of the algebraic equation system. Due to the complexity of the model, solving the equation system of the model was possible only by using an iterative numerical method. The obtained numerical results were arranged in a 3-D matrix. To this end, 3-D and 2-D graphical grids were selected as the most appropriate way to analyze and present the results. In order to solve the model an iterative numerical method and the 3-D graphical representation were found and applied in a program within the MATLAB package. This iterative numerical method is based on Newton-linearization and Gaussian elimination.

The following list shows the types and main characteristics of the components forming the heat pump heating system: the refrigerant R134a, plate evaporator with the nominal surface area 2 m^2 and active surface area 1.6 m^2, well pump nominal power 750 W, compressor nominal power 4000 W.

Since the energy optimum of the cold-water loop of the heat pump heating system was investigated, so during the procedure the effect of the condenser and the circulation pump on the COP was eliminated, *i.e.* the condensation temperature and the performance of the circulation pump were kept constant.

Basically, in this optimization procedure the effect of well pump and compressor power on the COP of heat pump heating system was investigated and determined the maximum COP. The optimization was not directly performed by modifying the power of the well pump and compressor, but rather indirectly, by altering the cold-well water and refrigerant mass-flow rates. During the optimization procedure, the mass-flow rates were modified over a wide range, from 0.2-0.95 kg/s of cold water and 0.06-1.1075 kg/s of refrigerant.

Based on the mass-flow rates the effective and the real power of the well pump and compressor were determined. In creating and applying the equation of the efficiency of the well pump and compressor the authors took into account the energy efficiency of these energy components of system.

Applying the efficiency of the well pump and compressor was crucial as the value and character of the efficiencies greatly influenced the value and location of the maximum COP in the 3-D grid.

In the optimization procedure the objective function was the equation of the COP of the heating system, since it is the quotient of the generated heat and the power used by the drive electric motors of the well pump and compressor.

The maximum COP value was determined by the 3-D grid of COP and the matrix contained the COP values. The 3-D grid and specifically the isohypse diagram made the area of near maximum values of the COP clearly visible and well-plotted, as seen in the yellow surface.

Even though the nominal power of the well pump is approx. five times smaller than a compressor with concave power characteristics, its effect on the COP values is still similar. This can be explained by the fact that the heat transfer coefficient of the water is five or six times higher than that of the evaporative refrigerant. In the case of the compressor with a convex power characteristic, the effect of the compressor power on the COP is slightly more intense. The ratio between real powers is 4000 W: 750 W = 5.3. The results obtained by the proposed numerical-graphical optimization procedure are acceptable since they are consistent with the results obtained by measuring the variables on a similar heat pump heating system in practice.

The near optimum real pump power values of the well pump are within the range of 480-560 W, while the compressor real power values are within the range from 3000-4050 W. The near maximum COP is 3.25 and the maximum is 3.26. The surface of the near optimum range is relatively large, since the effect of the well pump and compressor power on the COP is small. This COP refers to the cold-water loop of the heat pump heating system because the purpose was to investigate the energy efficiency of this loop. The effect of the hot water loop on the COP was not considered in this case.

These good results obtained by the simulation are promising from the point of view of the research and it is the basis for continuation, since the authors intend to continue the energy optimization of the heating system in an analytical procedure.

The advantage of the analytical procedure is that it determines the optimum value of the real pump power directly as a numerical number, thus it becomes unnecessary to search and read the chart so as to search for the maximum COP and the optimum value of the well pump's real power.

The presented research can be considered novelty for the following reasons: the formulation and purpose of the energy optimization problem, the new numerical-graphical energy optimization procedure based on the known mathematical methods and the new results obtained regarding the maximum COP of the heat pump heating system depending on the real power of the well pump and compressor.

Physical system

While the investigated physical system in this case is the entire heat pump heating system as such, this research specifically focuses on determining the energy optimum of the

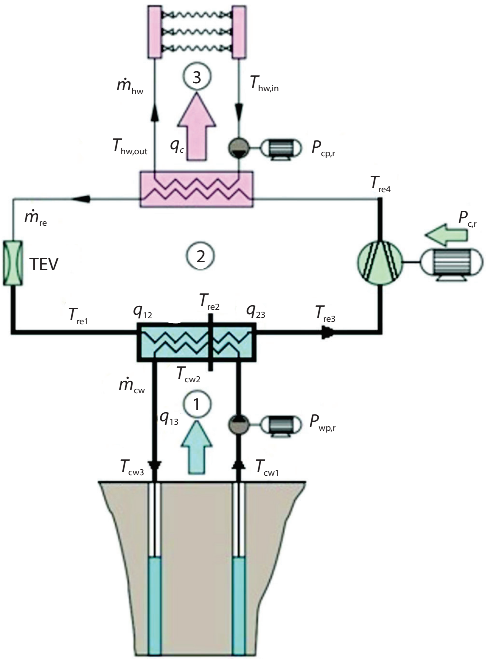

Figure 1. Physical system; *1 – cold water loop, 2 – refrigeration circuit, and 3 – warm water loop*

cold water loop in the heating system. Thus, the active components of the targeted physical system are the evaporator, the compressor and the well pump, whereas the passive components are the condenser, the circulation pump and the thermal expansion valve (TEV), fig. 1.

The constant values in this study are the surface of the condenser and the condensation temperature. The circulation pump power and the throttle of the thermo expansion valve are also constant values.

Mathematical model

The energy optimum procedure of the heat pump heating system ought to be performed for the stationary operation mode of the heating system, especially given the fact that the heating system operates in 98% stationary mode.

Taking this into account, the mathematical description of the heating system is stationary *i.e.* independent of the time and with lumped variables.

The COP – Objective function

The COP of the heat pump heating system is ratio between heat flow used for heating and the sum power used by the electric motors to drive the pumps and compressor. In the optimization procure the COP is applied as an objective function:

$$COP = [q_{12}(\dot{m}_{\mathrm{cw}}) + q_{23}(\dot{m}_{\mathrm{cw}}) + P_{\mathrm{c,e}}(\dot{m}_{\mathrm{re}})][P_{\mathrm{wp,r}}(\dot{m}_{\mathrm{cw}}) + P_{\mathrm{c,r}}(\dot{m}_{\mathrm{re}}) + P_{\mathrm{cp,r}}]^{-1} \tag{1}$$

Evaporator

Considering its operation, the evaporator consists of the evaporation and superheating sactions.

The stationary operation of the evaporator is described by six algebraic equations.

There are three balance equations of the heat flow in each section.

The governing equation system of the evaporator

Evaporation section

– Heat flow loss of the cold water in the evaporation section

$$q_{12} = \dot{m}_{\mathrm{cw}} Cp_{\mathrm{cw12}}\,(T_{\mathrm{cw1}} - T_{\mathrm{cw2}}) \tag{2}$$

– Heat flow through the wall with arithmetic temperature different

$$q_{12} = k_{12}A_{12}\Delta T_{12} = k_{12}A_{12}0.5(T_{\mathrm{cw1}} - T_{\mathrm{re1}} + T_{\mathrm{cw2}} - T_{\mathrm{re2}}) \tag{3}$$

– Heat flow absorbed by the evaporating refrigerant as the latent heat

$$q_{12} = \dot{m}_{\mathrm{cw}}(1 - x)\Delta h_{\mathrm{re12}} \tag{4}$$

– Temperature of the wall in the evaporation section

$$T_{\text{wall12}} = 0.5(T_{\text{cw1}} + T_{\text{cw2}}) - q_{12}(\dot{m}_{\text{cw}}Cp_{\text{cw12}}) - 1 \tag{5}$$

Superheating section

– Heat flow loss of the cold water in the superheating section

$$q_{23} = \dot{m}_{\text{cw}}Cp_{\text{cw23}}(T_{\text{cw2}} - T_{\text{cw3}}) \tag{6}$$

– Heat flow through the wall with arithmetic temperature different in the superheating section

$$q_{23} = k_{23}(A_{13} - A_{12})\Delta T_{23} = k_{23}(A_{13} - A_{12})(T_{\text{cw2}} - \text{T}_{\text{re2}} + T_{\text{cw3}} - T_{\text{re2}} + \Delta T_{\text{sup}})0.5 \tag{7}$$

– Heat flow for the superheating of the refrigerant vapor

$$q_{23} = \dot{m}_{\text{re}}Cp_{\text{re23}}(T_{\text{re2}} - T_{\text{re3}}) \tag{8}$$

Auxiliary equations of the evaporator

– Active surface area of the evaporator

$$A_{13} = A_{12} + A_{23} \tag{9}$$

– Overall heat flow in the evaporator

$$q_{13} = q_{12} + q_{23} \tag{10}$$

– Arithmetic temperature different in the evaporation section

$$\Delta T_{12} = 0.5(T_{\text{cw1}} - T_{\text{re1}} + T_{\text{cw2}} - T_{\text{re2}}) \tag{11}$$

– Arithmetic temperature different in the superheating section

$$\Delta T_{23} = 0.5(T_{\text{cw2}} - T_{\text{re2}} + T_{\text{cw3}} - T_{\text{re2}} + \Delta T_{\text{sup}}) \tag{12}$$

– Overall heat transfer coefficient in the evaporation section

$$k_{12} = \alpha_{\text{cw12}}\alpha_{\text{re12}}(\alpha_{\text{cw12}} + \alpha_{\text{re12}})^{-1} \tag{13}$$

– Overall heat transfer coefficient in the superheating section

$$k_{23} = \alpha_{\text{cw23}}\alpha_{\text{re23}}(\alpha_{\text{cw23}} + \alpha_{\text{re23}})^{-1} \tag{14}$$

– Heat transfer coefficient of the cold water [10]

$$\alpha_{cw12} = \alpha_{\text{cw23}} = 0.089\,(\text{Re}_{\text{cw23}})^{0.79} \cdot (\text{Pr}_{\text{cw23}})^{0.4}\,\lambda_{\text{cw23}}(D_{\text{eqv}})^{-1} \tag{15}$$

– Heat transfer coefficient of the refrigerator in the evaporation section [12]

$$\begin{aligned}\alpha_{\text{re12}} = \alpha_{\text{re12}x} &= (\lambda_{\text{re}}'/D_{\text{eqv}})(0.603)(\,\text{Re}_{\text{re}})^{0.5}(\,\text{Pr}_{\text{re}})0.1(x_{\text{atlag12}x})^{-2}\cdot\\ &\cdot[(G_{\text{re13}})2/((\rho'_{\text{re}})^2Cp'_{\text{re}}\,\Delta T_{12})]^{-0.1})((\rho'_{\text{re}})2\Delta h_{\text{re12}}/(G_{\text{re13}})^2)^{0.5}\cdot\\ &\cdot(\rho'_{\text{re}}\,\sigma'_{\text{re}}/(\eta'_{\text{re}}G_{\text{re13}}))^{1.1}\,(\rho'_{\text{re}}/(\rho_{\text{re}}' - \rho_{\text{re}}''))^2\end{aligned} \tag{16}$$

– Heat transfer coefficient of the refrigerator in the superheating section [9]

$$\alpha_{\text{re23}} = \frac{\lambda_{\text{re23}}}{D_{\text{eqv}}}C_{\text{re23}}\,\text{Re}_{\text{re23}}^{n_{\text{re23}}}\,\text{Pr}_{\text{re23}}^{m_{\text{re23}}} \tag{17}$$

Compressor

Real compressor power demand was determined using the enthalpy increment of the refrigerant vapor. When introducing the compressor efficiency, the energy losses during the operation of the compressor can also be considered.

The power equation of the compressor describes the stationary operation mode.

– Real power of the compressor

$$P_{\text{c,r}} = \dot{m}_{\text{re}}[h_{\text{re4}} - (h_{\text{re2}} + h_{\text{re23}})](\eta_{\text{c}})^{-1} \tag{18}$$

– Effective power of the compressor

$$P_{c,e} = P_{c,r}\,\eta_c \tag{19}$$

– Based on the characteristic of the real compressor was created the polynomial of the efficiency of the compressor [11]

$$\eta_c = a + bP_{c,r} + c(P_{c,r})^2 \tag{20}$$

– Coefficients of the concave characteristic of compressor [11], fig. 2,

$$a = 9.00836363636 \cdot 10^{-0.01}$$

$$b = -4.28439393939 \cdot 10^{-0.05}$$

$$c = 2.52575757576 \cdot 10^{-0.09}$$

– Coefficients of the convex characteristics of compressor [11], fig. 3,

$$a = 8.73860479381 \cdot 10^{-0.01}$$

$$b = 2.73912131120 \cdot 10^{-0.06}$$

$$c = -2.27742786917 \cdot 10^{-0.09}$$

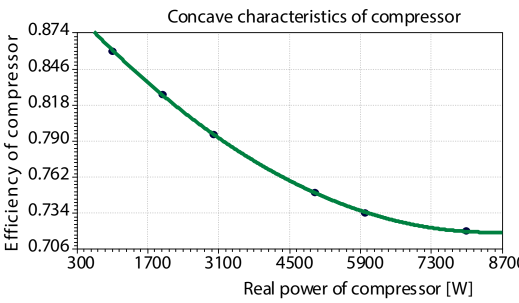

Figure 2. Concave efficiency-real power characteristics of the compressor

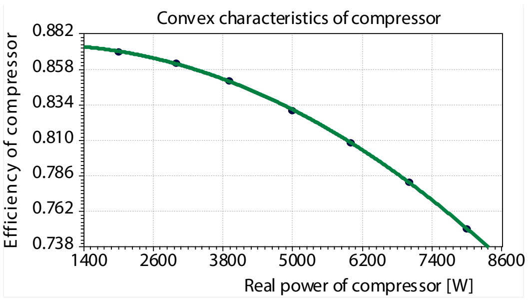

Figure 3. Convex efficiency-real power characteristics of the compressor

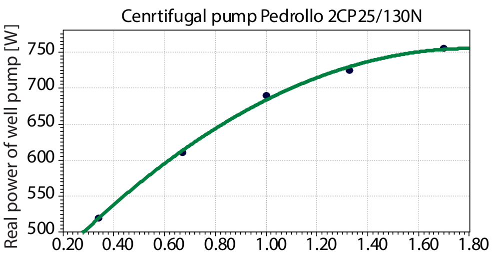

Figure 4. The efficiency-mass-flow rate characteristics of well pump

Well pump

The stationary real power equations of the well pump set up are based on the characteristic curve of a real centrifugal pump manufactured by an Italian company. The pump curve represents the correlation between the real power of the pump and the water mass-flow rate [12, 13].

Based on these two sets of data the equation of the well pump real power was constructed as a function of the cold-water mass-flow rate using the EXPERT CURVE software package.

The equation below refers to the real power of the well pump as a function of the mass-flow rate of cold-well water [11], fig. 4:

$$P_{wp,r} = a + b\dot{m}_{cw} + c(\dot{m}_{cw})^2 \tag{21}$$

Coefficient data of the Pedrollo 2CP25/130N centrifugal pump [11]

$$a = 3.97667350155 \cdot 10^{0.02}$$

$$b = 3.95171096684 \cdot 10^{0.02}$$
$$c = -1.09264883836 \cdot 10^{0.02} \tag{22}$$

Numerical mathematical procedure

The mathematical model of the heat pump heating system describing the stationary operation of the heating system is the system of the time independent, lumped, coupled, multi-variable, non-linear and large number algebraic equations.

The equation system of the model cannot be solved analytically, thus an iterative numerical procedure was applied, fig. 5.

The numerical program developed for this task was found in the MATLAB program package.

The numerical iterative procedure of the MATLAB is based on Newton-linearization and Gauss elimination methods.

Given that the goal was the energy optimization of the real well pump power with the appropriate real compressor power, the numerical optimization procedure has been used different mass-flow rates of cold water and refrigerant. The range of the variation of mass-flow rate was 0.2-0.95 kg/s for the cold water and 0.06-0.095 kg/s for the refrigerant.

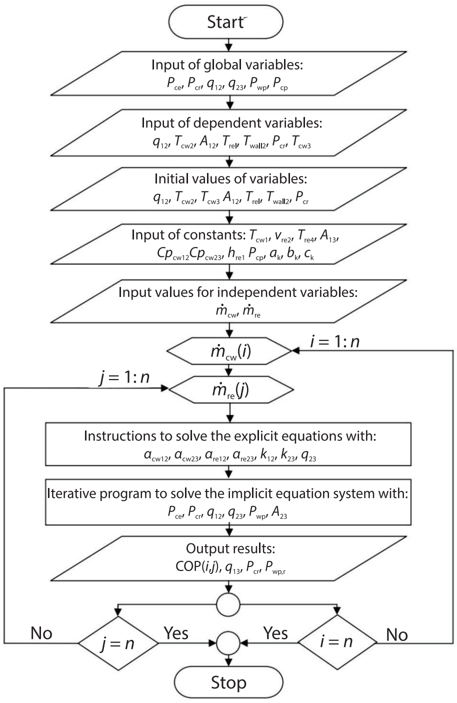

Figure 5. Algorithm of the numerical-graphical procedure

During the simulation, the increment of the cold-water flow rate per iteration was 0.025 kg/s, while the refrigerant mass-flow rate increased by 0.0025 kg/s. The number of iterations was 20.

Following the determination of the optimum mass-flow rate of the cold water, the optimum real power of the well pump, the real power of the compressor and the maximum COP of the heating system can be calculated.

Results and discussion

The results of the iterative numerical procedure are organized in the 3-D matrix, however, since the reviewing, interpreting and evaluating of results are challenging for researchers, the results were represented in 3-D graphics.

Both 2-D and 3-D grid applications are available in MATLAB, so the numerical results obtained in the 3-D matrix were edited and plotted by using the grid application.

Figures 6 and 7 show the 3-D space grid of the COP and COP_{max} values of the cold-water loop of the heating system as a function of the well pump and compressor real power.

In addition the main objective, the paper also investigated the effect of the compressor efficiency character on the COP. The research showed a significant difference in the value and position of the maximum COP of the concave and convex characters, as seen in figs. 6 and 7. The concave character exhibits an explicit optimum, as portrayed in fig. 6, while the convex

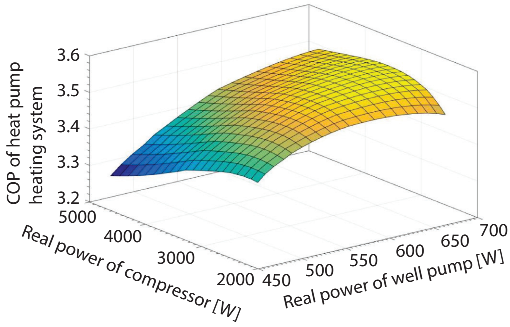

Figure 6. The COP and COP_{max} of the cold water loop of the heat pump heating system with the convex efficiency characteristics of compressor

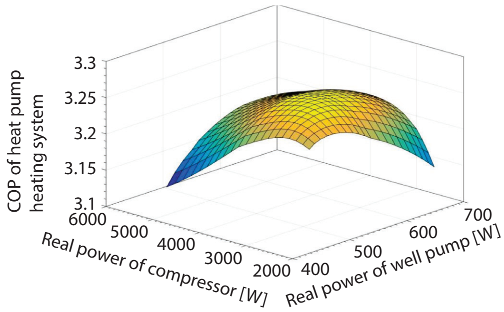

Figure 7. The COP and COP_{max} of the cold water loop of the heat pump heating system with the concave efficiency characteristics of compressor

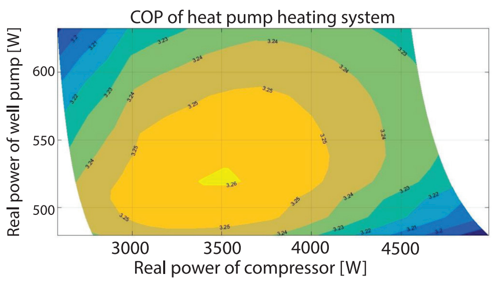

Figure 8. Isohypse of the COP depend on the compressor and well pump real power

character does not, presented in fig. 7. Hence, for a convex character, the compressor power has an optimum value, whereas the well pump power is moving toward its maximum.

Since the heat pump heating system with the convex character compressor regarding the well pump power has no energy optimum, the research focuses only on the analysis of the heating systems with the concave character compressor.

Figure 8 presents the 3-D grid which shows that the variation in COP value is similar depending on the change in well pump and compressor power, even though the compressor has approximately five times larger nominal power. This can be explained by the fact that the heat transfer coefficient of water is an estimated eight times greater than the refrigerant heat transfer coefficient.

In the case of a concave compressor characteristics, presented in fig. 6, the change of the curve of the powers is similar, whereas with convex characterisstics, fig. 7, it is different.

The range of compressor power changes from 2525-5587 W, while the well pump power changes range from 480-633 W, resulting in changes of COP from 3.11-3.26.

The change of the efficiency of the well pump and the compressor corresponds to the quadratic polynomial. The maximum COP is located on the surface marked yellow in the isohypse diagram, fig. 8, and its value is $COP_{max} = 3.26$.

On the other hand, a variety of the near optimal real powers of the well pumps and compressors can be achieved the approximate the maximum COP. This is presented as a yellow surface in the isohypse diagram, fig. 8.

The near COP_{max} = 3.25 can be achieved with numerous combinations of the two real powers. The range of values is very wide, for the well pump real power it is from 490-580 W, while the compressor real power varies from 3000-4050 W.

The reason why the near optimum range surface is so large is that the COP of the cold loop of heat pump heating system is not too sensitive to the changes of the well pump and compressor power.

It must be noted that the values presented in this article correspond to the constant condensation temperature (50 °C) and the constant circulating pump real power, P_{wp} = 250 W. Thus, in this study the effect of the warm water loop on COP was not deliberately disregarded, instead, the authors prioritized the internal correlations and effects of the cold water loop on the COP.

Conclusions

- The goal of the study was to determine the energy optimum of the cold water of the heat pump heating system using a numerical-graphical procedure.
- This goal oriented optimization procedure was implemented using the well-known mathematical methods.
- The optimization procedure proved to be a good one, since the values of the COP_{max} and the optimal real powers obtained correspond to the values measured in the operation heating system.
- The numerical-graphical optimization procedure is relatively easy to implement, but it is important to set up a proper, accurate physical and mathematical model.
- For the mathematical model, it is important to select and apply equations of the heat transfer coefficients which proved precise, accurate values for the flowing cold water and evaporating refrigerant
- Namely, the accuracy of the mentioned two heat transfer coefficients greatly determines the accuracy of the COP_{max} and optimum power of the well pump and compressor.
- The other equations in the model are balance equations whose accuracy has been proven mathematically and practically.
- In the case of energy optimization of the heat pump heating system, COP the best characterize the energy efficiency of the heating system.
- Obviously, in the energy optimization procedure the COP is the objective function.
- The energy optimization of the heating system is worth performing for the stationary operation mode, since in the practice the near stationary operation mode is happened approx. 95-98% in the time.
- Therefore, the equations describing the stationary operation of the heating system are independent of time.
- In the study the ultimate goal is energy optimization of the heat pump heating system by using the analytical optimization procedure, so the equations in the mathematical model are not only time-dependent but also space-dependent, i.e. they could be lumped. The analytical energy optimization procedure can only be performed with the lumped equations.
- The results of the optimization procedure provide important information and guidance on the behavior of the heat pump heating system, from the energy point of view.
- It has been found that combining the real power values of the several different well pumps and compressors results a near COP_{max} values.
- These real power values are in fact close to optimal values for the well pump and the compressor.

- Important conclusion is the near-optimal values of the well pump power over the fairly wide range, which is indicated by a yellow surface in the isohypse diagram.
- This yellow surface contains not only near-optimal values but also maximum COP as well.
- The characteristics of the compressor efficiency greatly influences the values of COP and the location and values of COP_{max}, as well as the optimal real power of the well pump and compressor.

Nomenclature

A – surface area, [m^2]
a, b, c, C – coefficients
Cp – specific heat, [Jkg^{-1}K^{-1}]
COP – coefficient of performance, [–]
D – pipe diameter, [m]
G – mass-flow density, [kgs^{-1}m^{-2}]
Δh – latent heat, [Jkg^{-1}]
h – specific enthalpy, [Jkg^{-1}]
k – overall heat transfer coefficient, [Wm^{-2}K]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
P – power, [W]
Pr – Prandtl number, [–]
q – heat flow, [W]
Re – Reynolds number, [–]
T – temperature, [K]
ΔT – temperature difference, [K]
x – vapor quality, [–]

Greek symbols

α – convective heat transfer coefficient, [Wm^{-2}K^{-1}]
η – efficiency, [–]
λ – conductive heat transfer coefficient, [Wm^{-1}K^{-1}]
ρ – density, [kgm^{-3}]
σ – surface tension of the liquid, [kgs^{-2}]

Subscripts

c – compressor
cp – circulation pump
cw – cold water
e – effective
eqv – equivalent
hw – hot water
in – inlet
k – condenser
out – outlet
r – real
re – refrigerant
wp – well pump

Superscripts

m, n – coefficients

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