PERFORMANCE ANALYSIS OF SYSTEM HEAT PUMP – HEAT RECUPERATOR USED FOR AIR TREATMENT IN PROCESS INDUSTRY

by

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A detailed parametric analysis and performance optimization of the system heat pump – heat recuperator is given in this paper. Mathematical model used for analysis is formed according to the physical model of the system and practical experience. Different types of compressors, and various temperature ranges are treated. Special attention is paid to influence of condensing and evaporating temperatures on system performances, so as to different features of heat recuperator. It is found out that in accordance to the working regimes and the compressor type, it is possible to determine optimal conditions in which the system will consume the minimum of electricity.

Key words: *heat pump, heat recuperator, air de-humidification, drying, electricity consumption*

Introduction

Drying processes are accompanied with high consumption of heat per unit of removed water and high volume air flow rates. With a well-known fact of rising demands for fuels and electricity, classical air treatment in conventional driers does not have a chance for commercial use. However, the significant energy efficiency improvements of drying processes (decreasing of heat duty consumption and increasing of drying potential by reduction of entering air humidity) can be obtained with system *heat pump – heat recuperator* (HP-HR). In air-conditioning and drying systems, recuperators (as the heat recovery devices) are commonly used to re-use waste heat from exhaust air normally expelled to atmosphere. Devices typically comprise a series of parallel plates of aluminum, plastic, stainless steel, or synthetic fiber, alternate pairs of which are enclosed on two sides to form twin sets of ducts at right angles to each other, and which contain the supply and extract air streams. Heat from the exhaust air stream is transferred through the separating plates into the supply air stream. Manufacturers claim gross efficiencies of up to 80% depending upon the specification of the unit.

Using refrigeration machine for simultaneous cooling (de-humidification) and heating purposes gives numerous benefits. Environmental – reduce CO_2 emissions and facilitate transition process from finite fossil fuel to sustainable, renewable energy sources. Also it allows significant money savings, and energy security improvement of country.

Air treatment has a very important role in chemical and food industry as well as in air-condition operations. In these sectors air heating, cooling, and de-humidifying are key processes

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for preparing and preserving of products. There is a variety of papers dealing with these topics, but neither about which type of compressor (as the main component of any refrigeration system) to use. In [1] authors deal with a problem of air conditioning for supermarket applications with air handling units and heat pump de-humidifier. The effectiveness of heat pump de-humidifier, solar-assisted systems, and natural drying processes were tested in the process of tomato de-humidification, and various relevant parameters were given in [2]. In [3] an experimental investigation was performed for determining the effects of surface load, drying air temperature, evaporator bypass air ratio, air velocity, *etc.,* on the drying characteristics of mackerel. Authors in [4-6] investigated different technologies of drying processes. According to [3, 6] drying process consumes 10-15% of total industrial energy demand. In [7] it is stated that conventional dryer techniques require large energy amounts for heating and removing water with a relatively high temperature moist air leaving the open circuit dryers. It is considered that warm and moist air is not only a waste but also it is detrimental to the environment [3]. Also, in [8] is briefly described the usage of heat pump de-humidifier. Authors in [9] have analyzed air treatment methods for the large scale sugar silos and potentials of energy savings.

Model

Exhaust air from the drying process (condition 1 on fig. 1) flows through HR where it changes until 2; at the same time the re-circulated air is preheated in same HR from 3 to 4. Precooled air (condition 2) flows through evaporator wherein it is further cooled and dried to the desired water content (condition 3). Evaporation temperature and consequently the mean temperature of the evaporator surface have to be properly defined to achieve the demanded air parameters. Heating of dried air (conditions 5 and 6) is conducted in two phases. First one is achieved by the heat rejected from the condenser of HP until the maximum temperature which is allowed by condensing temperature of the refrigeration machine. The second one is

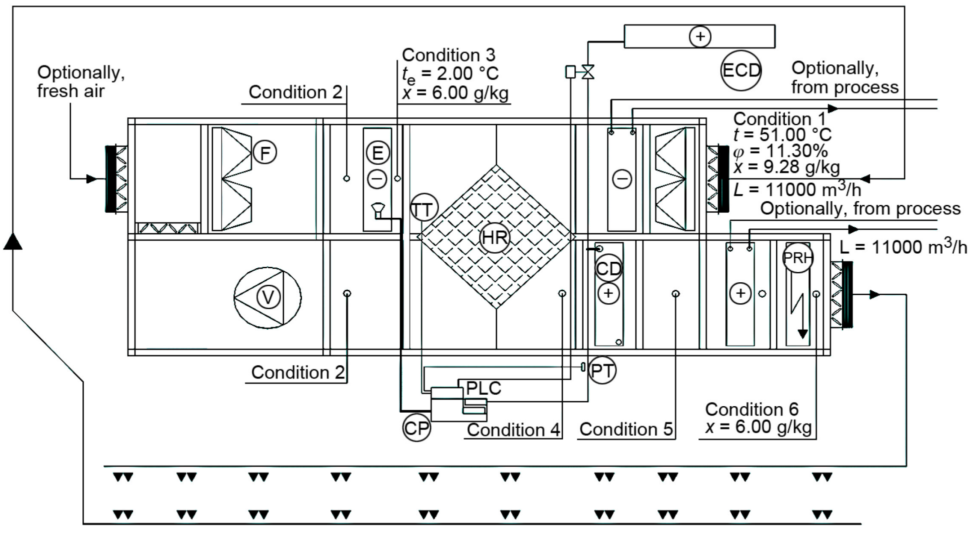

Figure 1. Schematic view of the system; *F – filters, E – evaporator, CP – compressor, CD – condenser, ECD – external condenser, PRH – preheater, HR – air heat recuperator, V – fan, PT – pressure measurement, PLC – power line communication, TT – temperature measurement, t – air temperature, x – absolute air moisture, φ – relative air humidity, L – air flow rate,* t_e *– temperature*

achieved through the auxiliary heater by electricity. It should be noted that some other heat sources should be used for additional air heating, such as vapor or exhaust hot gas (fig. 1). In practice, because of its simplicity and small investment costs the electrical heater is often acceptable. Using electricity for the heating purposes is not thermodynamically reasonable, but under certain circumstances it could be generally accepted solution (considering the low price of electricity in Serbia).

Based on the refrigerant de-superheating in heat pump process it is possible to reach slightly higher temperatures of air comparing to condensing temperature. However it is well known that the amount of de-superheating heat is relatively poor in relation to the heat of condensation, so in this analysis it is assumed that air temperature on the outlet of condenser is equal to the condensing temperature of refrigerant. It should be noted that rejection heat is strongly influenced by evaporating and condensing temperatures, so it is not possible with high condenser-evaporator temperature differences to reach the desired air temperature because of the drastically reduced refrigeration capacity of compressor.

The proper choice of evaporation temperature is one of the key factors which determine desired final parameters of prepared air entering a drying chamber. Requirements for reduced air flows leads to as low evaporation temperatures as possible (for example, 2-3 °C), where evaporation temperature below 2 °C is unacceptable because of the frosting on the evaporator surface. The selection of evaporation temperature also can be the subject of optimization, but considering quite small values of the required air humidity ratios (in industry drying applications it usually does not exceed 6 g_{vapor}/kg_{dryair}), in this analysis the evaporation temperature is adopted as constant.

Total consumption of electricity P_{ef}, [kW] includes absorbed power needed for compressors $P_{c,ef}$, electric power input of fans $P_{V,ef}$, and electrical power used in additional heater $P_{PRH,ef}$. Discharge temperature for defined working conditions is dependent on the type of compressor (reciprocating, screw…), the compressor construction, so as the used refrigerant. For air temperatures for typical drying processes in range of 60-80 °C, the refrigerant choice is relatively restricted. For further analysis R134a is chosen, despite the fact that its global warming potential (GWP) is quite high, GWP = 1430 [10], but it is de-chlorinated so its ozone depletion potential is equal to zero. Further, R134a is also convenient for use because of its low working pressures (alike ammonia R717). Ammonia is not suitable for these purposes, because of the high value of the polytrophic (isentropic) exponent, which leads to high discharge temperatures.

The analyzed system devices comprise of two seemingly independent circuits, which are connected and synergistically integrated into unique process. In the steady-state condition, drying plant operation can be considered as *static*, which means that the condition of the exhaust air from drying process is in a small range of temperatures and humidity ratios. Thus it is reasonable to assume the constant parameters of entering air in heat recovery unit (condition 1 of exhaust air). Moisture condensation can occur during cooling of exhaust air in HR. In order to simplify the analysis, and to make the easier control of the requested humidity ratio of air entering the evaporator of HP, it is provided dry heat transfer on HR by proper choice of operating system parameters. It is possible to determine the geometric features of a certain type of HR which provides the required operating parameters.

The required refrigeration capacity of compressor is determined by heat required for drying and cooling air (conditions 2 and 3), whereafter cooling humidity ratio of air reaches the required value (in this analyses 6 g_{vapor}/kg_{dryair}). Parameters of air are marked with subscripts 1 to 6, and refrigerant with 1R to 9R.

Heat duty of heat exchangers (evaporator E and condenser CD in fig. 1) for cooling and heating of treated air are, respectively:

$$\Phi_{\mathrm{kd}} = m(h_5 - h_4) \tag{1}$$

$$\Phi_{\mathrm{e}} = m(h_2 - h_3) \tag{2}$$

Heat pump consists of compressor, condenser, evaporator, expansion (throttling) device, and related elements of automation. A refrigeration process calculation was carried out by assuming real working conditions of installation. It means that refrigerant vapor is superheating on the outlet of evaporator for 7 °C (condition 8R in fig. 2) which is resulted with the throttling device, and additionally 3 °C in the suction lines of compressors. It has already accepted that the air temperature at the outlet of condenser is considered as high as the condensing temperature. Practically the air temperature is slightly higher than condensing temperature, but it does not have a significant impact on the accuracy of the prediction of system operation. Refrigerant R134a is a single component substance without temperature glide in two phase region. Sub-cooling is a result of heat exchange in condenser and it is calculated with 3 °C. No further sub-cooling is taken into account.

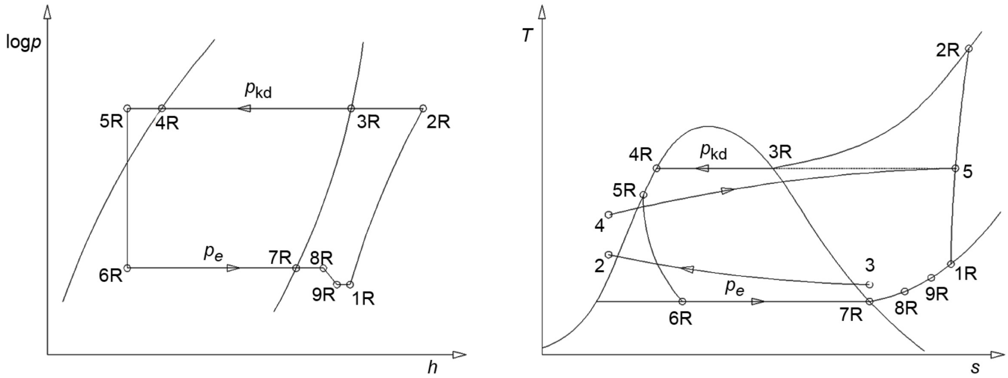

Figure 2. Log *p-h* and *T-s* diagrams of heat pump cycle

Refrigeration capacity and absorbed power of given compressor are strongly influenced with the temperature of the cooled air. For analyzed types of HR, air temperatures entering the evaporator were in range 22-32 °C (condition 2), and air outlet temperatures determined by humidity ratio of 6 $\mathrm{g_{vapor}/kg_{dryair}}$ were in range of 8-9.3 °C (condition 3). Changes of evaporator capacity for defined compressor occur as a result of evaporation temperature variation. As the evaporation temperature is within small range of operating conditions, it is assumed $t_{\mathrm{e}} = 2$ °C.

Refrigeration capacity of installation is defined by heat balance equation:

$$\Phi_{\mathrm{e}} = m_{\mathrm{R}}(h_{8\mathrm{R}} - h_{6\mathrm{R}}) = m(h_2 - h_3) \tag{3}$$

Swept volume of compressor $V_{\mathrm{S,}}$ [$\mathrm{m^3s^{-1}}$] is obtained by:

$$V_{\mathrm{s}} = \frac{\Phi_{\mathrm{e}}}{\lambda\, q_{\mathrm{v}}} \tag{4}$$

where volumetric refrigeration effect q_{v}, [$\mathrm{kJm^{-3}}$] is defined by:

$$q_{\mathrm{v}} = \frac{h_{8\mathrm{R}} - h_{6\mathrm{R}}}{v_{1\mathrm{R}}} \tag{5}$$

Volumetric efficiency for reciprocating compressors λ is obtained according to well known procedure [11, 12]. For screw compressor it is used calculated data from manufacturers. Isentropic power rate of compressor is:

$$P_{\mathrm{c},t} = m_{\mathrm{R}}(h_{2\mathrm{R}} - h_{1\mathrm{R}}) \tag{6}$$

and absorbed power is:

$$P_{\mathrm{c,ef}} = \frac{P_{\mathrm{c},t}}{\eta_{\mathrm{is}}} \tag{7}$$

Isentropic efficiency η_{is} is:

$$\eta_{\mathrm{is}} = \frac{\Delta h_{\mathrm{c,is}}}{h_{2\mathrm{R}} - h_{1\mathrm{R}}} \tag{8}$$

In eq. (8) $\Delta h_{\mathrm{c,is}}$ is the isentropic work of compression, and $h_{2\mathrm{R}} - h_{1\mathrm{R}}$ is the actual work of compression. The isentropic efficiency is mainly a function of the pressure ratio and type of a compressor. To define actual work of compressor in this research, realistic data are obtained from certain respectable compressor manufacturers.

Total condensing heat calculated for refrigeration cycle is defined by:

$$\Phi_{\mathrm{tot}} = m_{\mathrm{R}}(h_{5\mathrm{R}} - h_{2\mathrm{R}}) \tag{9}$$

External condensing heat is given by:

$$\Phi_{\mathrm{ecd}} = \Phi_{\mathrm{tot}} - \Phi_{\mathrm{kd}} \tag{10}$$

Throttling valve should provide refrigerant mass flow rate m_{R}, [kgs^{-1}] for defined pressure difference of condensation and evaporation Δp and its capacity is:

$$m_{\mathrm{R}} = \mu A_{\mathrm{V}} \sqrt{2\Delta p \rho} \tag{11}$$

where μ is the discharge coefficient given by valve manufacturer, A_{V}, [m^2] – the valve flow cross-section area, and ρ [kgm^{-3}] refrigerant density at the valve inlet.

Numerical calculations conducted in this paper cover a wide range of operating parameters for various compressor types. For example, conventional screws with R134a as refrigerant cannot operate in high condensing temperatures (over 65 °C). As the final temperatures of prepared air that is entering drying chamber are in the range of 60-80 °C (condition 6), it is not always possible to reach the requested temperature level only by HP, so additional heating is needed. Furthermore it is not possible within certain operating conditions to transfer all available condensing heat to the air, so an external condenser is essential (ECD in fig. 1). In some regimes when the HP operates with high condensing temperatures, available heating capacity of condenser is insufficient to reach needed air temperature level which leads to higher power consumption on additional heater.

Results and discussions

By assuming that system operates with re-circulation without any fresh air, mass flow rate was m = 3.53 kg/s (the volumetric flow rate at the outlet of drying process V = 11000 m^3/h, with parameters of air in condition 1 of p = 1 bar, t_1 = 51 °C, φ_1 = 11.3%), demanded humidity ratio of prepared air is x_6 = 6 g/kg, and evaporation temperature is t_e = 2 °C. Varied process parameters are temperature of prepared air for drying (in the range of t_6 = 60-80 °C), condensing temperature (t_{kd} = 50-80 °C), and geometrical parameters of HR expressed with effectiveness. For all defined relevant ranges of parameters, the performances of described drying system including different types of compressors: (1) screw in range of t_{kd} = 50-65 °C, (2) reciprocating in range of t_{kd} = 50-65 °C, and (3) reciprocating in range of t_{kd} = 65-80 °C, were tested. Based on predictions, graphical presentation of results relating to comparison of electricity consumption, depending on the operating conditions is given (figs. 3-10).

It is noted that the air pressure drop on the evaporator and condenser for defined flow rate depends on the geometric parameters of plate fin heat exchangers as well as on the de-humidification rate on evaporator. Exhaust air (condition 1 in fig. 1) flows through filter section and for clean filter air pressure drop is around 20 Pa, but during the time it increases. Total pressure drop on exchanger surfaces (without HR) and filter is changeable over operation time. In spite of this fact the energy consumption needed for overcoming these flow resistances is not considered because the identical pressure drop appears in every operating regime, and practically is independent of temperatures. This is the value added as a constant in all regimes. Furthermore, this increment in electricity consumption is not significant. For example, according to [13] for pressure drop of 150 Pa and defined flow rate, fan power is approximately increased by 650 W.

In figs. 3 to 8 the electricity consumption for various systems (with different type of compressor) in function of condensation temperature and temperature of drying air (condition 6 in fig. 1) are shown. The final temperatures of drying air (in each of figs. 3-8, these temperatures are labeled by associated tags) are used as a parameter. Furthermore, high temperature (HT) and low temperature (LT) next to numbers denoting drying air temperatures define the reciprocating compressors are used. Total energy consumption is calculated as sum of absorbed power of compressor, duty of heater, and fan power (depending on geometry and surface of HR).

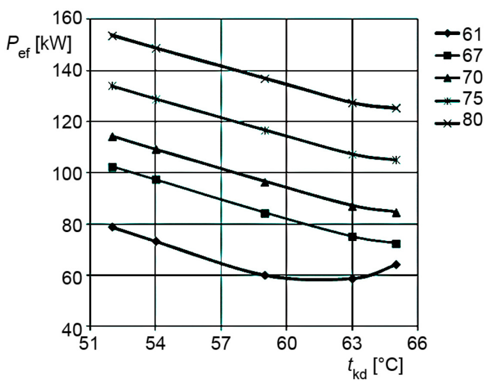

Figure 3. System with screw compressor

Total electricity consumption of system with screw compressors as a function of condensing and drying air temperatures is shown in fig. 3. Trend of reducing electricity consumption is registered with increasing t_{kd} except for curve which refers to t_6 = 61 °C (lowest one on graph). In that case, in range of higher condensing temperatures it is possible to reach the requested temperature of drying air by condenser heat only (that means, HP condenser capacity can meet heating needs), and consequently there is no need for additional heating. Clearly the minimum of function can be easily defined (t_{kdmin} = 61 °C). For other considered operating conditions of the system it is concluded that the higher temperature of condensation leads to lower electricity consumption.

Likewise, the system with screw compressor HP, for temperature of drying air t_6 = 61 °C there is no need for additional heater, because needed parameters of air can be achieved by condensing heat. That is the reason of appearing the local minimum on the lowest curve in fig. 4 which is defined for system with reciprocating compressors in LT range. For higher values of treated air (higher than 65 °C) this system consumes less electricity with higher values of t_{kd}, because smaller amount of heat duty needed on additional heater.

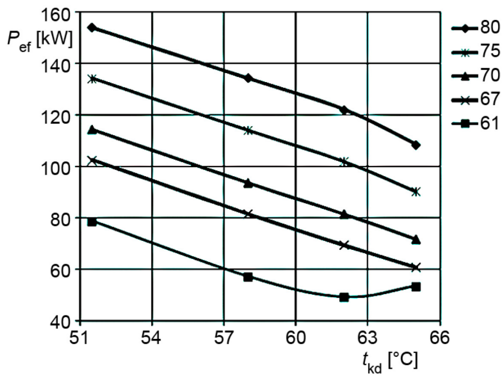

Figure 4. System with reciprocating compressor for lower t_{kd}

Unlike the previous two cases each curve in fig. 5 have minimum. For t_6 = 70 °C there is no need for additional air heater, since the required temperature can be reached by condenser rejected heat, whereby: the minimum is defined by t_{kd} = 70 °C. For drying air temperatures of 75 °C and 80 °C, even though these compressors enables HP to run at high condensing temperatures (which are equal to or in some cases even higher than the required temperature t_6), it is impossible to reach the required parameters of drying air with HP condenser due to insufficient heat output that occurs as a result of reduction in cooling capacity of compressor, so there is a need for additional heater. In this case the temperature of air at outlet of condensing unit is even lower than condensation temperature. For these reasons minimum of P_{eff} for both curves is defined for t_{kd} close to 75 °C.

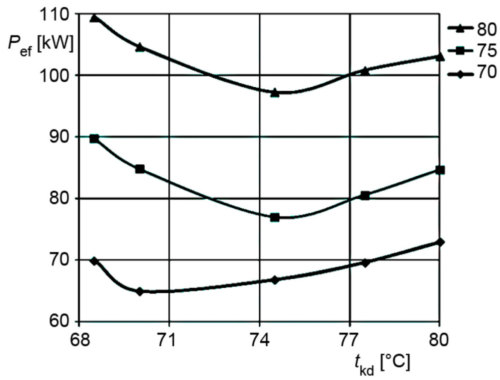

Figure 5. System with reciprocating compressor for higher t_{kd}

Comparison of systems including screw and reciprocating LT compressors is given in fig. 6. For condensing temperature between 51-52 °C these systems are comparable concerning total electric power consumption. If there is no need for HT of drying air, greater efficiency can be achieved using screw compressors (temperature range which is not analyzed in this paper, t_{kd} < 51 °C). For HT of drying air from fig. 6 can be seen that the system with reciprocating compressor indicates an advantage. For the case when the t_6 = 61 °C, the minimum of electricity consumption is achieved for condensing temperature of 61 °C (no need for additional air heater). For HT of drying air (t_6 = 67 °C and t_6 = 70 °C), the system achieves greater efficiency (less consump-

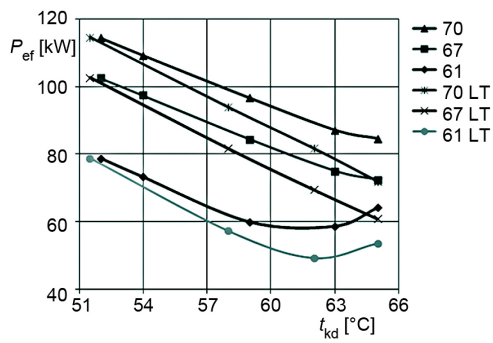

Figure 6. Consumption of systems with screw compressor and reciprocating for lower t_{kd}

tion) when operates at HT of condensation for both systems, whereby in every range of application reciprocating compressors gives a certain advantage over the system with screws.

An interesting comparison of the electricity consumption of the systems with screw and reciprocating HT compressors is given in fig. 7. It is clear that for all temperature ranges of drying air reciprocating compressors offer a certain advantage. The same conclusion can be drawn based on the analysis of reciprocating HT and LT compressors, which is shown in fig. 8.

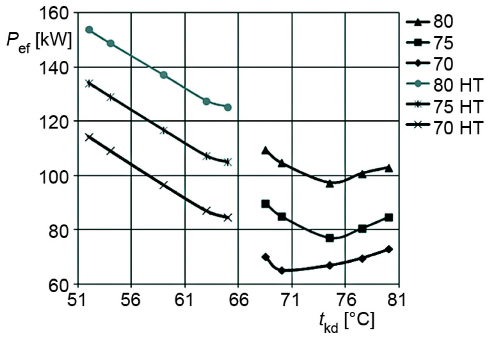

Figure 7. Comparison of systems with screw and reciprocating HT compressors

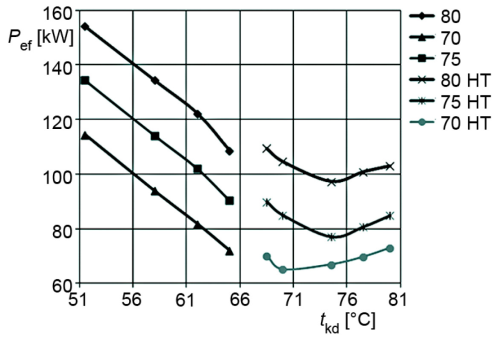

Figure 8. Comparison of systems with reciprocating HT and LT compressors

In figs. 9 and 10 the distribution of electricity consumption were analyzed in two characteristic cases. The electricity consumption of fan, additional heater and reciprocating HT compressors, as well as total consumption of system is shown in fig. 9. Increase of consumption in fans is proportional to condensing temperature rise. The power consumption for additional heater is marked as PRH. Based on the analysis it was found that heater power is significantly reduced to a local minimum which is defined for t_{kd} = 75 °C.

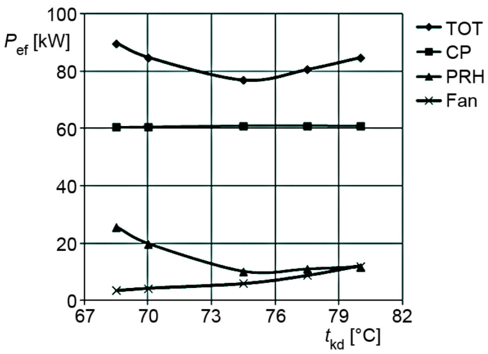

Figure 9. System with reciprocating compressor for t_6 = 75 °C

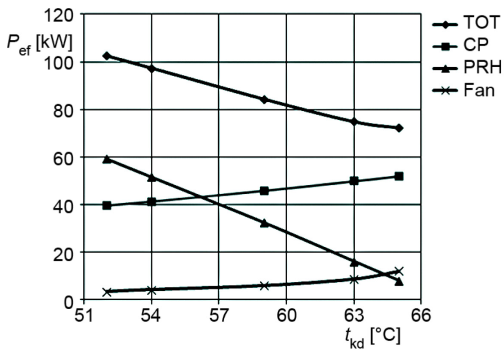

Figure 10. System with screw compressor for t_6 = 67 °C

With higher condensing temperature consumption of PHR begins to increase. In this application, condenser heat duty is not enough to achieve the required drying air parameters. This can be explained by the reduction of the cooling capacity of compressor, and therefore by reduction of heat duty of condenser at higher t_{kd}. Almost linear change of absorbed power of the compressor (CP) shows a slight increase in the whole range of t_{kd}. Total electricity consumption (TOT) curve is the identical one as in fig. 5 defined for t_6 = 75 °C.

Figure 10 shows the electricity consumption for LT of drying air (t_6 = 67 °C). As in the previous case the fan curve has the same trend. It is evident that power consumption of additional heater (PRH) significantly decreases across the range of considered t_{kd} with clear rise of screw compressor absorbed power. The TOT curve shows a lowering trend which is limited only with considered range of condensing temperature.

Conclusions

Based on the analysis the following facts are indicated.

- When the drying air temperature does not exceed 65 °C, the TOT decreases with rising condensing temperature. The same conclusion is valid for all compressor types. It should be noted that according to available data, the authors failed to find a conventional rotary screw compressors which are capable of operating at evaporation temperature of 2 °C and condensing temperature higher than 65 °C. Under these circumstances, it is always rational that the condensing temperature should be as high as possible.
- Ammonia as working fluid is very suitable for HT-HP heat pumps but only if evaporating temperature is not below 25 °C. However with these operating parameters cannot be achieved the required de-humidification rate of drying air.
- Within the range 60-80 °C of drying air temperature, the use of reciprocating compressors with R134a as a refrigerant is strongly recommended since an apparent minimum of TOT is registered. Proper selection of the reciprocating compressor according to the application range is critical.

Nomenclature

h – specific enthalpy, [kJkg^{-1}]
P – power rate, [kW]
v – volumetric flow rate, [m^{3}s^{-1}]

Greek symbol

Φ – heat duty, [kW]

Subscripts

c – compressor
e – evaporator
kd – condenser
R – refrigerant

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