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The impact of environmentally friendly refrigerants on heat pump efficiency

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Abstract

This paper discusses the issue of heat pump simulation. Lower operating costs can be achieved through good equipment design and appropriate operation. Detailed technical and economic analysis inform the selection of system components with the highest possible efficiency and lower energy demand. The use of systems based on environmentally friendly refrigerants can deliver long product life, which is associated with a shorter payback period for the investment. In addition, smart design processes optimize the performance of equipment. The rapidly increasing share of energy costs in terms of total investment cost is driving innovative solutions and tailor-made solutions to specific operating conditions.

Keywords: refrigerants; heat pump; energy efficiency

1. Introduction

The European Union is encouraging energy-efficient production and more environmentally friendly industrial plants through regulatory architecture. Formerly widely used organic compounds, such as chlorofluorocarbons (HCFCs), which include R12 and R22 in refrigeration technology, were withdrawn from technical application in the EU under the Montreal Protocol (1987). These compounds deplete the ozone layer. Therefore, HCFCs were replaced by hydrofluorocarbons (HFCs), which have a high value of Global Warming Potential (GWP), and mixtures of: zeotropic (R407C), azeotropic (R404a), natural hydrocarbons and/or carbon dioxide. The chemical industry has created a new range of synthetic refrigerants with a low GWP value, belonging to the HFO group. This group of synthetic refrigerants includes R1234yf and R1234ze, which could be used to replace the commonly used R134a. The new refrigerants appear to be environmentally friendly and conducive to high equipment efficiency. The aim is to reduce environmental impact. Appropriate determination of heat and flow parameters (heat transfer coefficient and pressure drop) will inform efficient use of these refrigerants.

Much research is focusing on experimental evaluation of prediction tools with new working fluids. Boiling and condensation involve large rates of heat transfer due to (i) the latent heat of evaporation and (ii) increased turbulence between the liquid and solid surface [1]. Nonetheless, significant gains can be made by increasing the act as a crudeefficiency of energy utilization, i.e. recovering low-grade waste heat [2, 3] and improving process efficiency [4].

There is a move toward devices with alternative working fluids, such as natural refrigerants. This is driven by energy efficiency and environmental goals. Climate change concerns are aiding the search for alternative methods of reducing energy consumption devices, especially in refrigeration and air conditioning. Manufacturers of refrigeration components and working fluids are taking up the challenge of coping with the new demands and the plans the EU has announced for the years ahead. The interest in energy-saving products is growing in developing countries too, for example China, Turkey and Brazil. Rising energy prices provide the most effective incentive to reduce energy use.

Compared to other refrigerants carbon dioxide is relatively safe, because it is non-flammable, non-explosive and neutral to most plastics. It is used in small refrigeration systems, the food industry and air conditioning [5]. The use of carbon dioxide is a huge challenge for designers and producers of these devices. For example, the design of high-efficiency heat exchangers/evaporators which have carbon dioxide as a working fluid need to make an exact calculation of the heat transfer coefficient during flow boiling [6]. A new approach to design was required for compressors to implement supercritical and cascading systems with carbon dioxide. These difficulties were overcome in part, because the cascade systems are mounted. Moreover, there are reports of the use of only carbon dioxide for cooling systems and there are a grow-

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ing number of systems that use natural refrigerants such as R600a, R290, etc.

The air heat pump is a device which absorbs heat from the outside air and delivers it to heating systems. Therefore, heat exchange elements determine the efficiency of the whole system. The temperature difference between the working fluid and the cooling area should be as low as possible. To limit unit costs, the size of the units should be kept as low as possible. Therefore, much research has gone into the intensification of heat transfer, resulting in compact design heat exchangers. Numerous technologies have been used to deliver high heat fluxes: microchannels [7], impinging jets [8, 6], hybrid microjet-microchannels [9] etc.

The research generated a new style of construction of heat exchangers, with enhanced heat transfer (higher heat transfer coefficient). It should be noted that while aiming to achieve the highest possible heat transfer coefficient k [W/m^{2}K], the increase in flow resistance should be kept in check. All gains and losses associated with modification of heat exchangers need to be considered.

Various factors and assumptions are involved in selecting heat exchangers for an installation and they have a significant impact. The most problematic assumptions are: the choice of material appropriate for the working fluid, the characteristics of working fluid pollution, the evaluation and determination of heat transfer coefficients, uniformity of flow, and flow resistance.

Heat exchangers exposed to water (condenser) are exposed to corrosion and contamination in the heat transfer area. Therefore, it is important that the materials used in the condenser should be easy to conserve and resistant to external factors.

To compare the operating cooling systems with different working fluids, an analysis of the impact of various working fluids for efficiencies and energy consumption was carried out and is reported on in this paper. Considerations are based on the selected refrigerants: R290, R600a, R1234yf, R1234ze, R365fmc and the commonly used substances R134a, R32, R22.

2. Modeling

The analysis was carried out for the following working fluids: R290, R600a, R134a, R1234yf, R1234ze, R365fmc, R32 and for withdrawn refrigerants from the chlorofluorocarbons group (HCFCs) R22. To perform the analytical calculations the refrigerant flow through a round tube with inner diameter fo 5 mm was assumed, and the thermal resistance of the wall was omitted. The heat transfer coefficient on the air side was assumed to be constant. Other assumptions: an expanded heat transfer surface area on the air side and the heat transfer coefficient referring to the surface of 500 W/m^{2}K. All assumptions are shown in Table 1.

For the purpose of these analyses the well-known and experimentally validated Friedel correlation was used to determine the pressure drop [10], which is described by the relation 1:

Table 1: Assumptions used in the analysis

Parameter	Unit	Value
T_w—wall temperature	°C	-9
T_e—evaporation temperature	°C	-15
x—quality	-	0.5
G—mass flux	kg/m^{2}s	0–3000
q—heat flux	W/m^2	500
L—length of channel	m	1
d—hydraulic diameter	mm	5

$$\Delta P_{frict} = \Delta P_l \phi_{Fr}^2 \tag{1}$$

Where liquid pressure drop is calculated by equation 2, and the two-phase multiplier is calculated by equation 3.

$$\Delta P_l = 4 \cdot f_l \cdot (\frac{L}{d}) \cdot G^2 \cdot (1 - x^2) \cdot (\frac{1}{2\rho_l}) \tag{2}$$

$$\Phi_{Fr}^2 = E + \frac{3.24 \cdot F \cdot H}{Fr^{0.045} \cdot We^{0.035}} \tag{3}$$

The two-phase multiplier is defined by equation 4 respectively, for liquid and vapor (gas) phase.

$$\begin{aligned} f_l &= \begin{cases} \frac{16}{Re_l} & \text{for} \quad Re_l < 2300 \\ \frac{0.079}{Re_l^{0.25}} & \text{for} \quad Re_l \geqslant 2300 \end{cases} \\ f_g &= \begin{cases} \frac{16}{Re_g} & \text{for} \quad Re_g < 2300 \\ \frac{0.079}{Re_g^{0.25}} & \text{for} \quad Re_g \geqslant 2300 \end{cases} \end{aligned} \tag{4}$$

Parameters E, F, and H, which occur in equation 3 are defined by the following equations:

$$\begin{aligned} E &= (1-x)^2 + x^2(\frac{\rho_l f_g}{\rho_g f_l}) \\ F &= x^{0.78}(1-x)^{0.224} \\ H &= (\frac{\rho_l}{\rho_g})^{0.91}(\frac{\mu_g}{\mu_l})^{0.19}(1 - \frac{\mu_g}{\mu_l})^{0.7} \end{aligned} \tag{5}$$

Shah [11] correlation was used for the heat transfer coefficient working fluid 6 during this analysis.

$$\alpha_{TPB} = \alpha_C \cdot \psi_S \tag{6}$$

where amplification factor is defined as function boiling number Bo describing the nucleate boiling and convective number Co describing convective boiling. In order to determine the amplification factor an additional variable Ns is introduced, which is a function of the Froude number, Fr (equation 7).

$$\begin{aligned} N_S &= C_0 \quad for \quad Fr > 0.04 \\ N_S &= 0,38 \cdot Fr^{-0.3} \cdot C_0 \quad for \quad Fr > 0.04 \end{aligned} \tag{7}$$

It is also necessary to designate a parameter using equation 8.

$$\psi_{CB} = \frac{\alpha_{TPB}}{\alpha_C} = 1.8 \cdot N \tag{8}$$

In addition, the following should be estimated: the value of the function and the function previously designated parameter N_s (9,10).

When $N_s > 1.0$

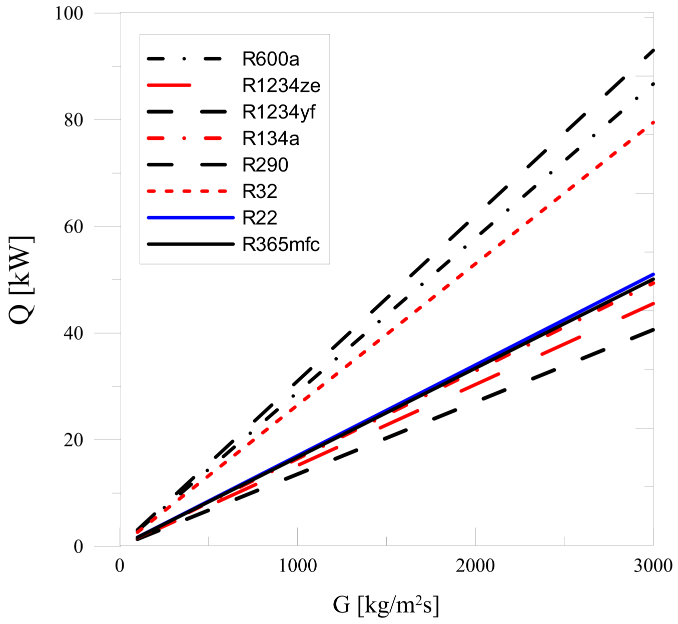

Figure 1: Model System

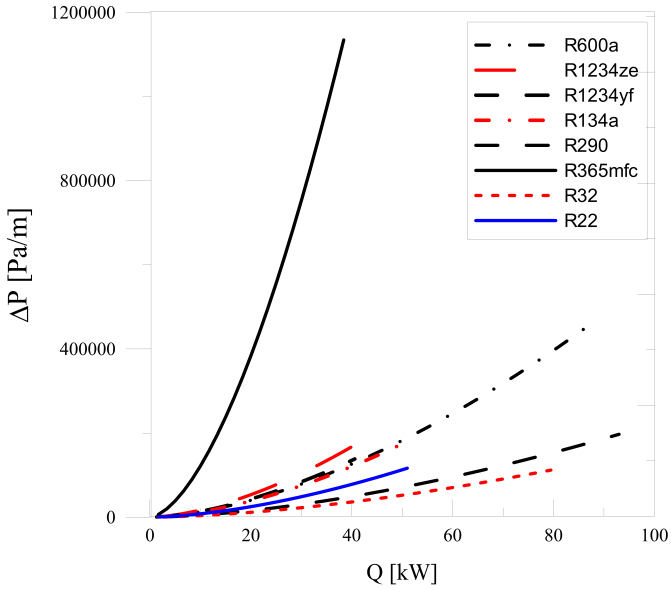

Figure 2: Evaporator pressure drop as a function of transferred heat

$$\psi_{NB} = 230 \cdot Bo^{0.5} \text{ for } Bo > 0.3 \cdot 10^{-4}$$
$$\psi_{NB} = 1 + 46 \cdot Bo^{0.5} \text{ for } Bo < 0.3 \cdot 10^{-4} \quad (9)$$

when $N_s < 1.0$

$$\psi_{BS} = F_S Bo^{0.5} \exp(2.47 \cdot N_S^{-0.1}) \text{ for } 0.1 < N_S$$
$$\psi_{BS} = F_S Bo^{0.5} \exp(2.74 \cdot N_S^{-0.15}) \text{ for } N_S < 1.0 \quad (10)$$

In equation 11 the parameter F_s is a constant depending on boiling number, Bo.

$$F_S = 14.7 \quad \text{for} \quad Bo > 11 \cdot 10^{-4}$$
$$F_S = 15.4 \quad \text{for} \quad Bo < 11 \cdot 10^{-4} \quad (11)$$

Thus, the amplification factor value is determined as a larger value of the parameters, where the first parameter is designated as 8 and the second 9 or 10:

$$\psi_S = max(\psi_{NB}, \psi_{CB})$$
$$\psi_S = max(\psi_{BS}, \psi_{CB}) \quad (12)$$

All the relationships are described above by equations (1–7), where indices designated as g and l mean respectively the flow of vapor phase or flow of the liquid phase. Based on this analysis graphs were drawn to show the impact of the heat flux received by the evaporator on the other parameters of the heat exchanger. The impact of working fluid properties on the size of the heat exchanger is shown in Figures 1–4.

As can be seen from the theoretical analysis, there is comparable heat flux received from the refrigerant in the evaporator of the heat pump based on R290, R134a, and R600a. It is similar in the case of other working fluids considered in this analysis. Comparable heat flux can be achieved with working fluids such as R1234yf, R1234ze, R365mfc and R32. It should be noted that the heat flux in these cases is similar to the heat flux obtained with R22, which is currently being withdrawn from use. Working fluid R32 achieves the highest value of heat transfer coefficient, while it has the lowest pressure drop. The heat pump is a device designed to collect heat from the outside air and to deliver it to the heating system of buildings. The principle of this device is as follows: heat is absorbed from a low temperature heat source to provide energy for heating and evaporation of a working fluid with a low boiling point. Next, in the compressor the vapor pressure is increased. The condenser receives energy from the refrigerant at high pressure, which is used for example to heat water in the central heating system. Then, the fluid exiting the expansion valve is delivered back to the evaporator. The efficiency of this cycle is measured as the ratio of heat transferred in the condenser to the energy of the compressor (13):

$$COP_h = \frac{\dot{Q}_{cond}}{N_{comp}} \quad (13)$$

Where the system is simple, such as the cycle shown in Fig. 8, the mass flux of refrigerant is constant in each element of the heat pump. Thus, heat flux and power of the compressor can be simplified to the difference enthalpy of the inlet and outlet of refrigerant. Then it can be described by the following equation (14):

$$COP_h = \frac{h_{cond\ in} - h_{cond\ out}}{h_{comp\ out} - h_{comp\ in}} \quad (14)$$

The additional pressure drops result from the flow of working fluid through the channel and the phase change in the evaporator and condenser should take into account the length of the channel and the speed of the working fluid. Assuming flow through channels with a diameter of 5 mm, the length can be described by equation (15):

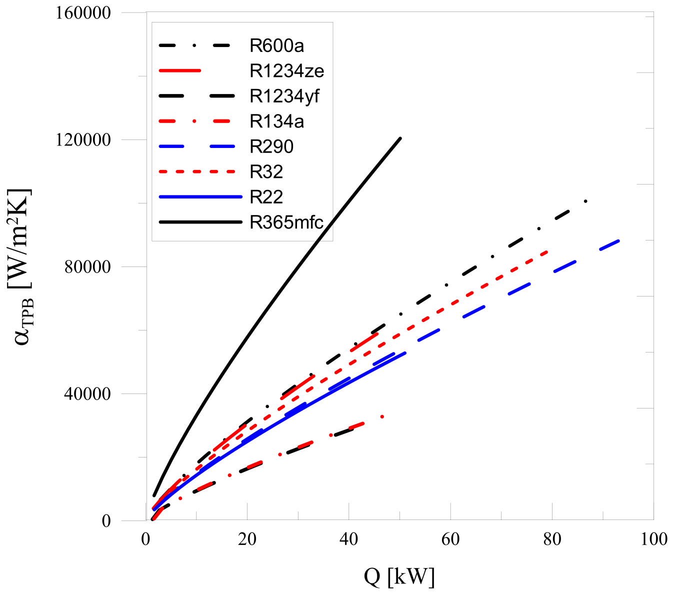

Figure 3: Heat transfer coefficient as a function of transferred heat

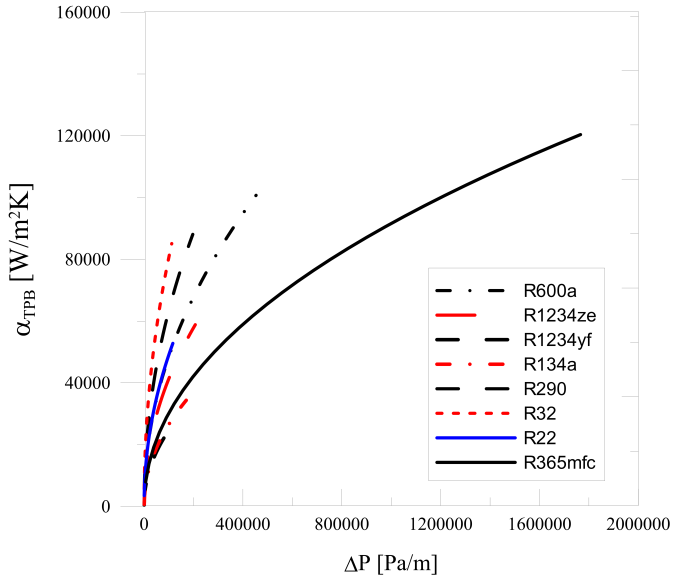

Figure 4: Heat transfer coefficient as a function of evaporator pressure drop

Table 2: Design as assumptions

Parameter	Unit	Value
Evaporating temperature	°C	0
Condensing temperature	°C	50 60 70 80 90
Subcooling	K	0 5 10 15 20
The compressor efficiency	%	70

$$\dot{Q}_{cond} = \alpha \cdot A \cdot \Delta T \qquad (15)$$

Both the heat transfer coefficient and the pressure drop depend on the mass flux of the working fluid. Therefore, it is necessary to iterate the solution of the problem. This analysis assumes that the heat pump system will be a system that is supporting the primary source of heat. In addition, it is assumed that it will work with the following constant conditions: outside air temperature of 10°C, temperature of the refrigerant in the evaporator of 0°C. The assumptions used to carry out this analysis are presented in Table 2. As before, the analysis of the heat pump was carried out for the selected working fluids: R290, R600a, R134a, R1234yf, R1234ze, R365mfc, and R32.

3. Results

The impact of condensing temperature and subcooling on the efficiency of the heat pump is shown in Figures 5–12.

The impact of subcooling on the COPh is non-linear and highly dependent on the saturation temperature of the working fluids. These relationships are shown in Figures 13–20.

4. Summary

The cost of refrigeration and air conditioning installations depends on the size of the device, which includes: compressors, circulation pumps and heat exchangers. The operating cost can be reduced by good design and operation. In addition, economic and technical analysis can inform the selection of components with the highest efficiency and lowest energy demand. Systems based on next-generation environmentally friendly refrigerants have the potential to benefit from a longer working life and quicker return on the cost of investment. Moreover, good design can optimize the work of the whole installation. This paper presents the modeling of the heat pump and impact of pressure drop during flow inside heat exchangers. Evidently, the temperature of the working fluid inside the condenser and evaporator have a significant impact on the amount of heat which can be transferred and therefore on the efficiency of the heat pump. The physical and chemical properties of working fluids should be taken into account during analysis of these devices. It is stressed that the calculated value can vary from the real value due to the presence of pollution in the working fluid. The operating conditions of these devices also have a significant impact. Systematic reviews and appropriate service are fundamental. The system should be well-positioned with easy access to heat exchangers (quick and easy clean). The aim is to optimize system efficiency while keeping the investment and service costs as low as possible.

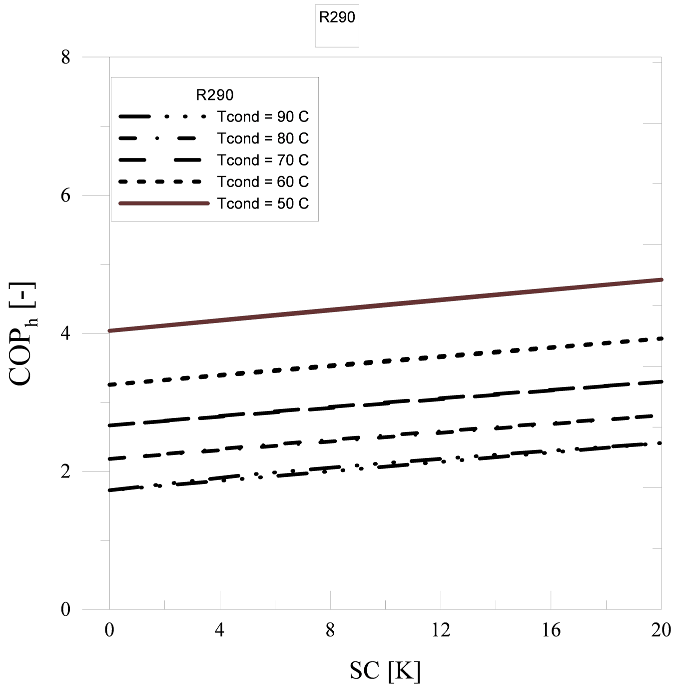

Figure 5: COPh for R290, with variable subcooling

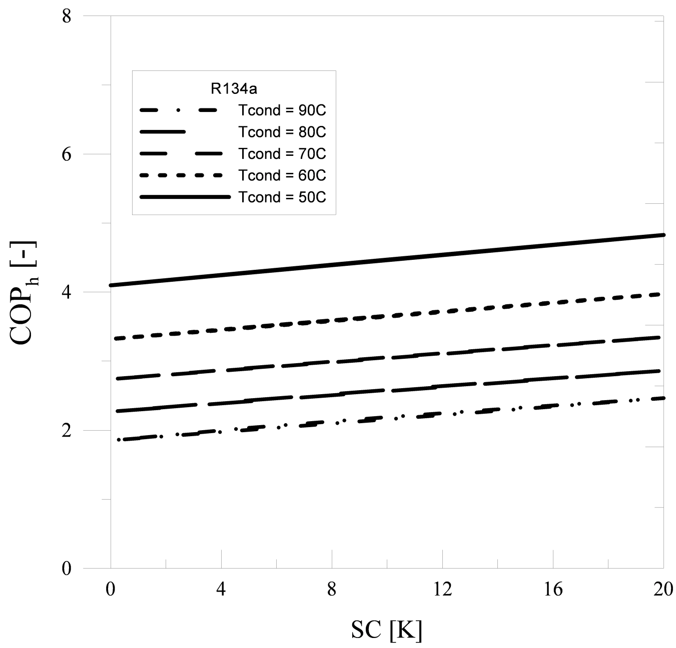

Figure 6: COPh for R134a, with variable subcooling

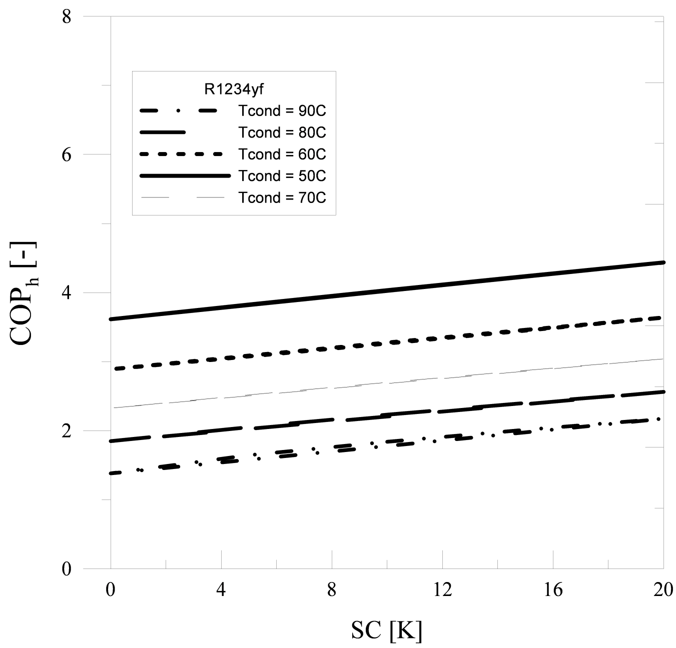

Figure 7: COPh for R1234yf, with variable subcooling

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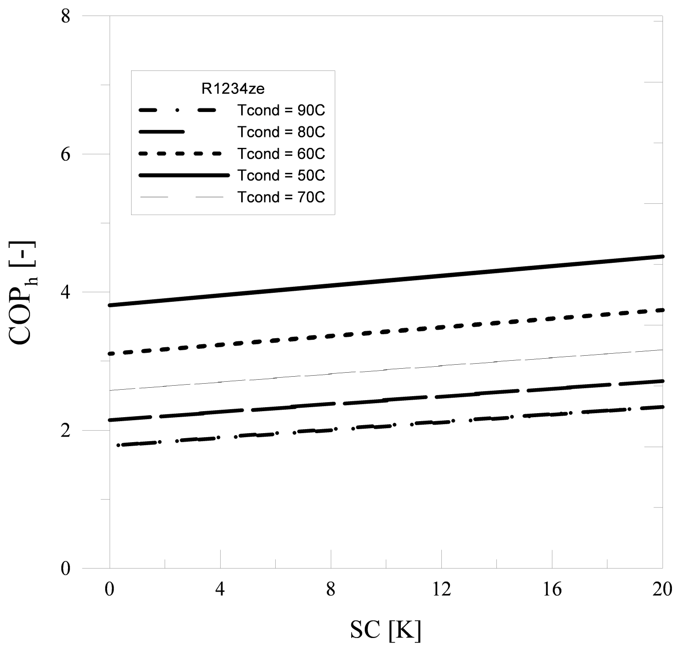

Figure 8: COPh for R1234ze, with variable subcooling

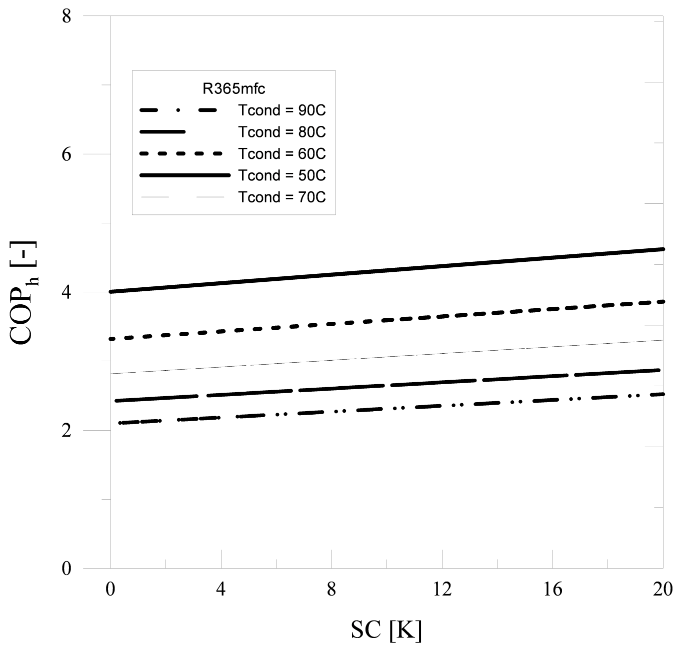

Figure 10: COPh for R365mfc, with variable subcooling

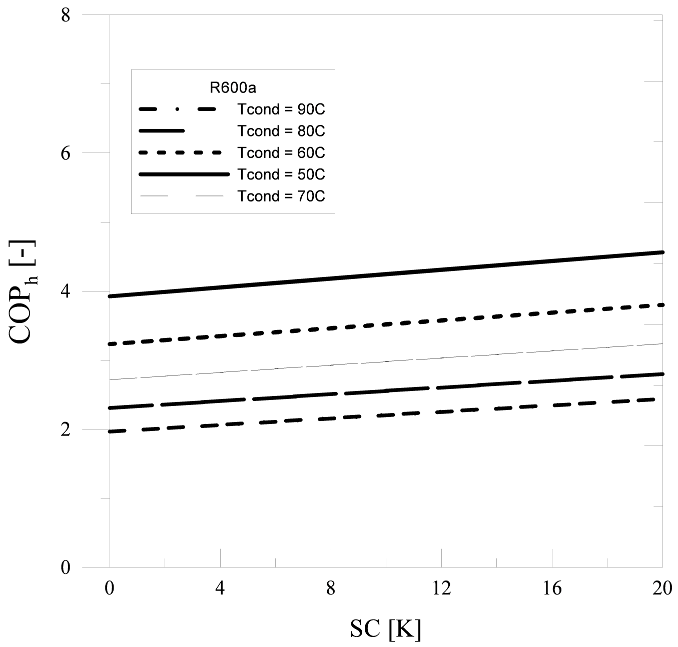

Figure 9: COPh for R600a, with variable subcooling

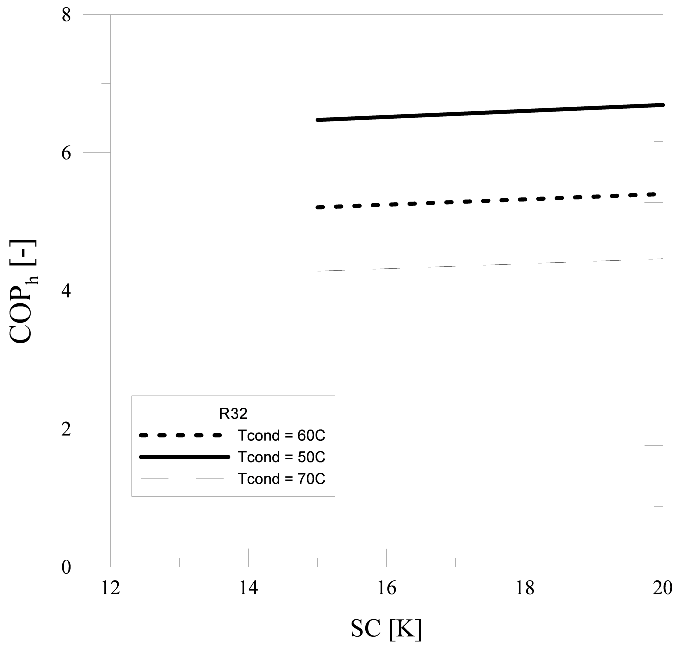

Figure 11: COPh for R32, with variable subcooling

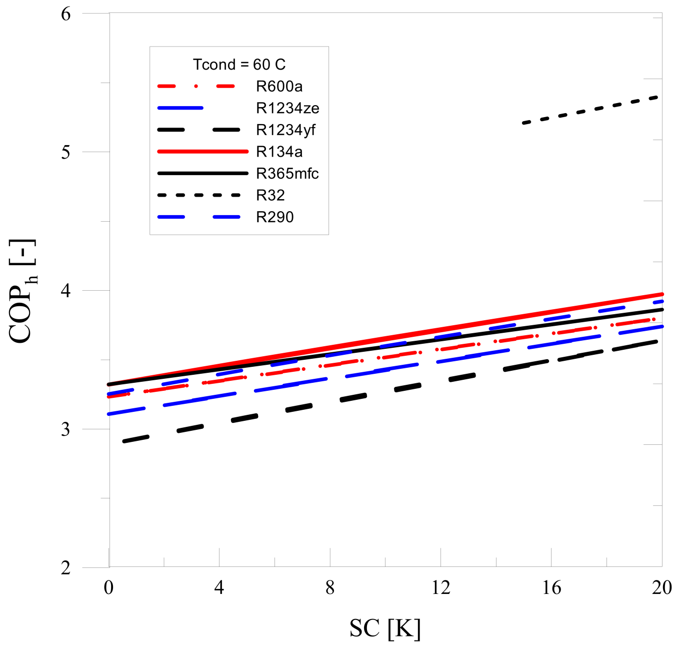

Figure 12: Comparison COPh for various refrigerants with variable subcooling at T_{cond} = 60°C

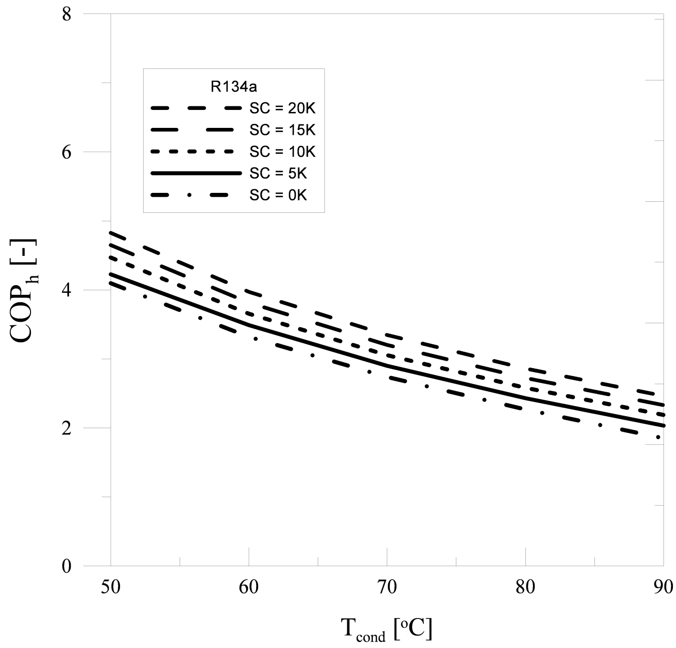

Figure 14: COPh for 134a at variable condensing temperature

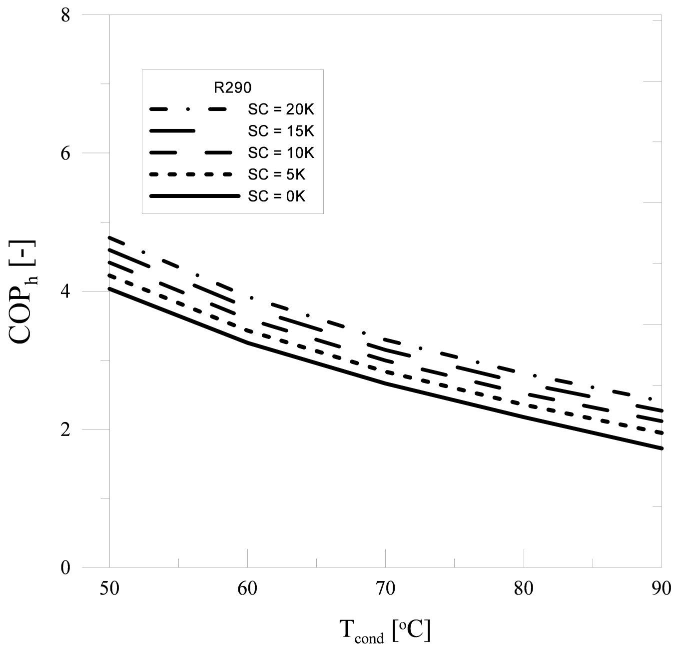

Figure 13: COPh for R290 at variable condensing temperature

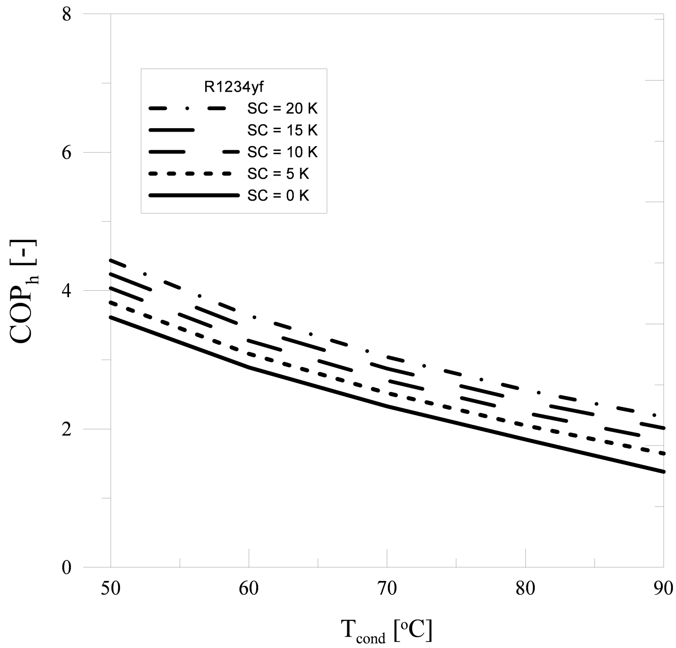

Figure 15: COPh for R1234yf at variable condensing temperature

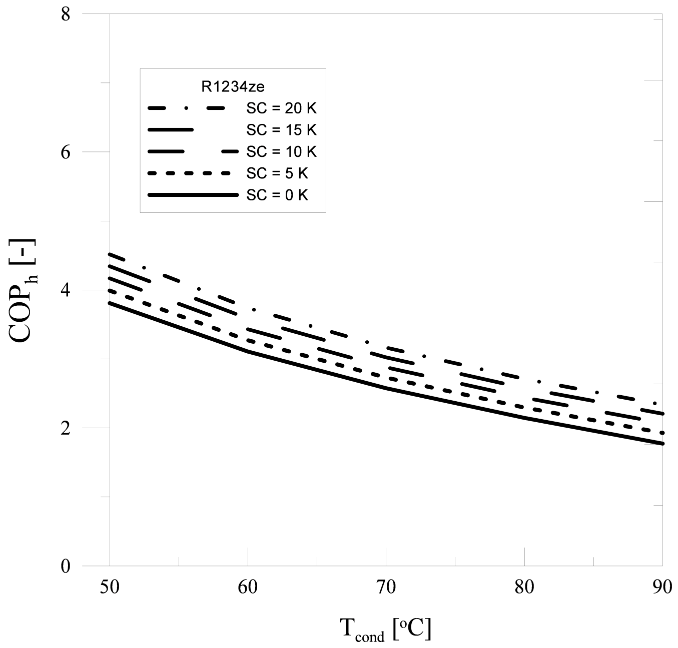

Figure 16: COPh for R1234ze at variable condensing temperature

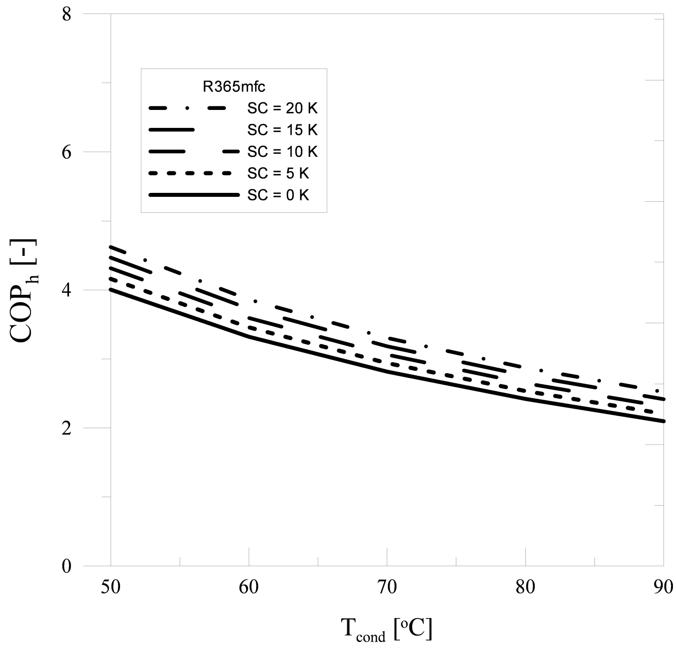

Figure 18: COPh for R365mfc at variable condensing temperature

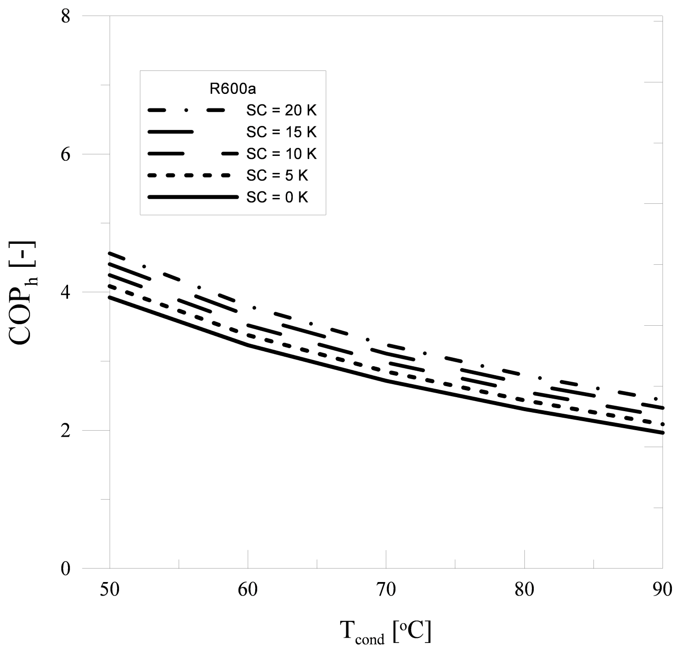

Figure 17: COPh for R600a at variable condensing temperature

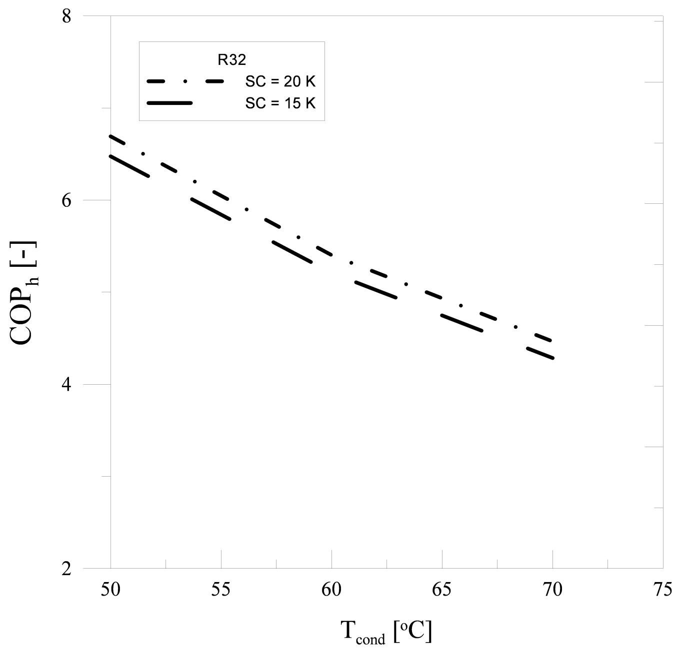

Figure 19: COPh for R365mfc at variable condensing temperature

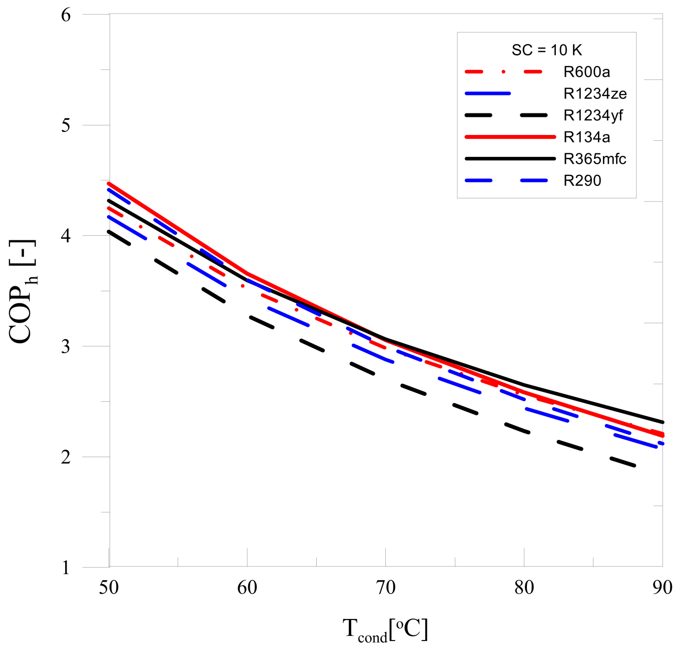

Figure 20: COPh for R365mfc at variable condensing temperature