OPTIMIZATION MODEL OF SINGLE STAGE VAPOUR COMPRESSION REFRIGERATION SYSTEMS

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În această lucrare este prezentat un model de optimizare a instalațiilor frigorifice cu comprimare mecanică de vapori (IFV) într-o treaptă pe baza termodinamicii proceselor ireversibile. Regimul funcțional vizat este cel de economicitate, corespunzător activității de proiectare în condiții optime a IFV, şi care conduce la un coeficient de performanță frigorifică maxim (COP^{max}) pentru o anumită valoare nominală a puterii frigorifice. Modelul de analiză permite determinarea parametrilor constructivi şi funcționali optimi care, în funcție de tipul agentului frigorific utilizat, conduc la un COP^{max}. S-a efectuat o simulare numerică în raport cu temperatura spațiului răcit pentru R134a. Considerând o IFV proiectată în condiții optime, în funcție de puterea frigorifică, se constată existența unei valori de maxim maximorum a coeficientului de performanță frigorifică corespunzător unei puteri frigorifice optime care este mai mică decât cea nominală.

The paper presents an optimization model of single stage vapour compression refrigeration systems (VCRS) based on the thermodynamics of irreversible processes. The targeted operation regime is the economical one, corresponding to optimum design conditions of VCRS and which leads to a maximum cooling efficiency (COP^{max}) for a certain nominal cooling capacity. The model allows determining the optimum constructive and functional parameters which lead to COP^{max} depending on the refrigerant type. A numerical simulation has been carried out with respect to the cooled space temperature for R134a. Taking into consideration a VCRS system optimumly designed, in function of the cooling capacity, the cooling efficiency has a maximum maximorum value corresponding to an optimum cooling capacity lower than the nominal one.

Keywords: vapour compression refrigeration systems, irreversible processes, economical operation regime, cooling efficiency, refrigerant.

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1. Introduction

Vapour compression refrigeration systems (VCRS) are the most commonly used type of refrigeration systems in current applications (food, chemical, aeronautics, automotive industry, household and air conditioning) [1]. Since the VCRS are energy consuming, their optimization is very important. In general, the VCRS are designed to work in optimum conditions. That is why here, the targeted operation regime is the economical one [2]. This regime, in condition of imposed cooling capacity and finite size constraints, involves finding the optimum constructive (thermal conductance distribution of heat exchangers) and functional parameters (temperature differences between working fluid and heat sources) which lead to a minimum compressor power consumption and respectively, to a maximum cooling efficiency [3]. Furthermore, in real operating conditions, the performances and optimum parameters of VCRS are directly influenced by external (heat transfer at finite temperature difference working fluid - heat sources) and internal (imperfection of the processes which compose the thermodynamic cycle in general and in particular of the compression and expansion processes) irreversibility sources and also by the refrigerant type [4].

The thermodynamic cycle of the single stage VCRS is shown in Fig.1.

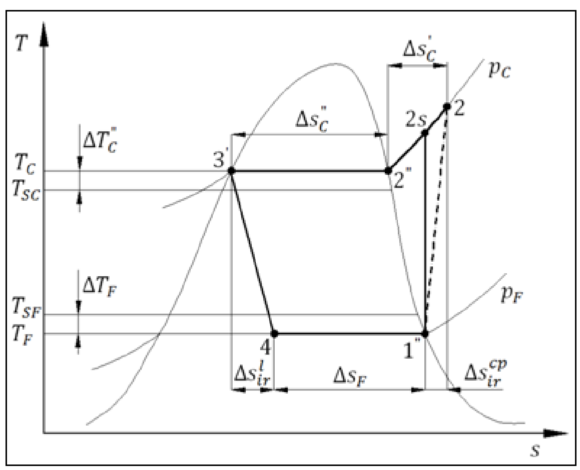

Fig. 1. The thermodynamic cycle of a single stage VCRS in T-s diagram

2. The optimization model

In order to develop the optimization model the following have been considered: constant temperatures of the heat sink and cold heat source; steady state operation regime (constant heat and mass flow rates of the VCRS); the condenser is divided in two zones – one zone designated with (') in which the desuperheating process takes place (the working fluid cools down till the dry saturated vapour state (x=1)) and a second zone designated (") in which the condensing process takes place.

In Fig. 1 the following notations have been used: T_{SC} - temperature of the heat sink; T_{SF} - temperature of the cold heat source; T_C - condensing temperature; T_F - evaporating temperature; $\Delta T_C^{''}$ - temperature difference between the working fluid and the heat sink during the condensation process; ΔT_F - temperature difference between the working fluid and the cold heat source during the evaporation process; Δs_F - mass specific entropy variation during the evaporation process (4 - 1"); Δs_{ir}^{cp} - mass specific entropy variation during the compression process (1" - 2); $\Delta s_C^{'}$ - mass specific entropy variation during the desuperheating process (2 - 2"); $\Delta s_C^{''}$ - mass specific entropy variation during the condensation process (2" - 3'); Δs_{ir}^{l} - mass specific entropy variation during the expansion process (3' - 4);

The proposed optimization model is based on the following equations:

Heat transfer and Energy balance equations:

- at the evaporator:

$$\dot{Q}_F = K_F \cdot \Delta T_F \, ; \quad \Delta T_F = T_{SF} - T_F \tag{1}$$

$$\dot{Q}_F = \dot{m} \cdot q_F \, ; \quad q_F = h_1^{''} - h_4 \tag{2}$$

where: $\dot{Q}_F$ is the cooling capacity; K_F is the evaporator thermal conductance; $\dot{m}$ is the refrigerant mass flow rate; $h_1^{''}$ and h_4 are respectively, the refrigerant mass specific enthalpies at the outlet and inlet of the evaporator which give the specific thermal cooling load q_F.

- at the condenser, in the adiabatic case:

$$\left|\dot{Q}_C\right| = \left|\dot{Q}_C^{'}\right| + \left|\dot{Q}_C^{''}\right| \tag{3}$$

where: $\left|\dot{Q}_C\right|$ is the heat flux rejected at the condenser; $\left|\dot{Q}_C^{'}\right|$ is the condenser heat flux rejected in the desuperheating zone and $\left|\dot{Q}_C^{''}\right|$ is the condenser heat flux rejected in the condensing process.

The heat rejected in the desuperheating process can be written as:

$$\left|\dot{Q}_C^{'}\right| = K_C^{'} \cdot \Delta T_C^{'} \, ; \quad \Delta T_C^{'} = T_C^{'} - T_{SC} \tag{4}$$

where: $K_C^{'}$ is the thermal conductance corresponding to the desuperheating zone; $\Delta T_C^{'}$ is the temperature difference between the refrigerant and heat sink and $T_C^{'}$ is the refrigerant mean thermodynamic temperature during the desuperheating process which can be computed as follows:

$$T_C^{'} = \frac{\left|\dot{Q}_C^{'}\right|}{\dot{S}_C^{'}} = \frac{\left|q_C^{'}\right|}{\Delta s_C^{'}} = \frac{h_2 - h_2^{''}}{s_2 - s_2^{''}} \tag{5}$$

In eq. (5) $\left|q_C^{'}\right|$ is the mass specific heat rejected in the desuperheating process; h_2, s_2 and $h_2^{''}$, $s_2^{''}$ are respectively, the inlet and outlet of the desuperheating zone mass specific enthalpies and entropies of the refrigerant.

The heat flux rejected during the condensing process can be written as:

$$\left|\dot{Q}_C^{''}\right| = K_C^{''} \cdot \Delta T_C^{''}; \quad \Delta T_C^{''} = T_C - T_{SC} \tag{6}$$

$$\left|\dot{Q}_C^{''}\right| = \dot{m} \cdot \left|q_C^{''}\right|; \quad \left|q_C^{''}\right| = h_2^{''} - h_3^{'} \tag{7}$$

where: $K_C^{''}$ is the thermal conductance corresponding to condensing zone; $h_2^{''}$ and $h_3^{'}$ are respectively the refrigerant mass specific enthalpies at the beginning and ending of the condensing process, which give the heat rejected during the condensing process $\left|q_C^{''}\right|$.

The cycle energy balance equation is:

$$\left|\dot{Q}_C\right| = \dot{Q}_F + P_{cp} \tag{8}$$

where P_{cp} is the compressor power consumption which can be computed as follows:

$$P_{cp} = \dot{m} \cdot \left|l_{cp}\right|; \left|l_{cp}\right| = h_2 - h_1^{''} \tag{9}$$

where $\left|l_{cp}\right|$ is the real specific mechanical work consumption during the compression process.

The compressor outlet state (2 in Fig. 1) can be established by defining the isentropic efficiency, which can be approximated with $\eta_{is}^{cp} \approx T_F / T_C$ [1]:

$$\eta_{is}^{cp} = \frac{\left|l_{cp}^{is}\right|}{\left|l_{cp}\right|} = \frac{h_{2s} - h_1^{''}}{h_2 - h_1^{''}} \Rightarrow h_2 = h_1^{''} + \frac{h_{2s} - h_1^{''}}{\eta_{is}^{cp}} \tag{10}$$

where $\left|l_{cp}^{is}\right|$ is the isentropic specific mechanical work consumption during the compression process .

The cycle entropy balance equation is:

$$\frac{\dot{Q}_F}{T_F}-\left(\frac{\left|\dot{Q}_C^{'}\right|}{T_C^{'}}+\frac{\left|\dot{Q}_C^{''}\right|}{T_C}\right)+\dot{S}_{ir}=0 \tag{11}$$

In eq. (11) $\dot{S}_{ir}$ is the entropy flux generated by the imperfection of the thermodynamic cycle, in general and in particular by the imperfection of the compression and expansion processes. It can be expressed as:

$$\dot{S}_{ir}=\dot{S}_{ir}^{l}+\dot{S}_{ir}^{cp}+\dot{S}_{ir}^{int} \tag{12}$$

where: $\dot{S}_{ir}^{l}$ is the entropy flux generated during the expansion process (3'-4, in Fig. 1); $\dot{S}_{ir}^{cp}$ is the entropy flux generated during the compression process (1-2, in Fig. 1) and $\dot{S}_{ir}^{int}$ is the entropy flux generated by the other internal irreversibility sources of the thermodynamic cycle.

Using eq. (5), the equation (11) becomes:

$$\frac{\dot{Q}_F}{T_F}-\frac{\left|\dot{Q}_C^{''}\right|}{T_C}-\dot{S}_C^{'}+\dot{S}_{ir}=0 \tag{13}$$

The following notation can be used:

$$\dot{S}=-\dot{S}_C^{'}+\dot{S}_{ir} \tag{14}$$

Based on eq. (14), eq. (13) can be further written as follows:

$$\frac{\dot{Q}_F}{T_F}-\frac{\left|\dot{Q}_C^{''}\right|}{T_C}+\dot{S}=0 \tag{15}$$

Cooling efficiency of the VCRS is:

$$COP=\frac{\dot{Q}_F}{P_{cp}} \tag{16}$$

The objective of the optimization model is the *COP* maximization in conditions of imposed cooling capacity and finite size constraint (total thermal conductance imposed).

The partial finite size constraint can be expressed as:

$$K_T=K_F+K_C^{''} \tag{17}$$

As it can be seen from eq. (16), in conditions of imposed cooling capacity ($\dot{Q}_F$), the maximization of the COP is achieved when the compressor power consumption (P_{cp}) is minimum.

Combining eq. (3) and (8), the compressor power consumption can be written:

$$P_{cp} = \left|\dot{Q}_C^{'}\right| + \left|\dot{Q}_C^{''}\right| - \dot{Q}_F \tag{18}$$

In order to simplify the optimization model the heat flux rejected in the desuperheating zone can be established as a part (*pc*) from the heat flux rejected during the condensing process, as follows:

$$\left|\dot{Q}_C^{'}\right| = pc \cdot \left|\dot{Q}_C^{''}\right|; \quad pc = \frac{h_2 - h_2^{''}}{h_2^{''} - h_3^{'}} < 1 \tag{19}$$

So, using eq. (19), the expression (18) becomes:

$$P_{cp} = (pc+1) \cdot \left|\dot{Q}_C^{''}\right| - \dot{Q}_F \tag{20}$$

Next, the expression (20) will be processed. Starting from eq. (1) the evaporating temperature can be written as:

$$T_F = T_{SF} - \frac{\dot{Q}_F}{K_F} \tag{21}$$

Substituting eq. (21) in eq. (15), it yields:

$$\frac{\dot{Q}_F}{T_{SF} - \dot{Q}_F / K_F} - \frac{\left|\dot{Q}_C^{''}\right|}{T_C} + \dot{S} = 0 \tag{22}$$

Using eq. (6), the expression (22) becomes:

$$-\frac{K_C^{''} \cdot (T_C - T_{SC})}{T_C} + A + \dot{S} = 0 \tag{23}$$

where, the following notation was used:

$$A = \frac{\dot{Q}_F}{T_{SF} - \dot{Q}_F / K_F} = f(K_F) \tag{24}$$

From eq. (23) it results that:

$$\frac{T_{SC}}{T_C} = 1 - \frac{A + \dot{S}}{K_C^{''}} \tag{25}$$

Using the eqns. (6) and (20) the expression of compressor power consumption becomes:

$$P_{cp} = (pc+1) \cdot K_C^{''} \cdot T_{SC} \cdot \left(\frac{T_C}{T_{SC}} - 1\right) - \dot{Q}_F \tag{26}$$

Substituting eq. (25) in eq. (26) and after several mathematical operations yields:

$$P_{cp} = (pc+1)\cdot T_{SC}\cdot\left(\frac{A+\dot{S}}{1-(A+\dot{S})/K_C''}\right)-\dot{Q}_F \tag{27}$$

Based on eq. (27) the expression (16) becomes:

$$COP = \left[(pc+1)\cdot\frac{T_{SC}}{\dot{Q}_F}\cdot\left(\frac{A+\dot{S}}{1-(A+\dot{S})/K_C''}\right)-1\right]^{-1} \tag{28}$$

As it can be seen from eq. (28) the COP depends on the following parameters: $\dot{Q}_F$, pc, T_{SF}, T_{SC}, $\dot{S}$ and on the variables K_F and K_C''. If the values corresponding to $\dot{Q}_F$, pc, T_{SF}, T_{SC}, and $\dot{S}$ are known, then the minimum compressor power consumption which leads to COP^{max}, can be achieved only if the expression *E* is minimum, where:

$$E = \left(\frac{A+\dot{S}}{1-(A+\dot{S})/K_C''}\right) \tag{29}$$

From eqns. (24) and (29) it results that the expression $E = f\left(K_F, K_C''\right)$.

Taking the derivative of *E* with respect to K_F and setting it equal to zero ($\partial E/\partial K_F = 0$), it gives the optimum thermal conductance distribution between evaporator and condenser (K_F^{opt}, $K_C''^{opt}$), corresponding to the minimum value for expression E which leads to minimum power consumption and COP^{max}.

After several mathematical computations the expression $\partial E/\partial K_F = 0$ becomes:

$$-\frac{A^2}{K_F^2}\cdot\left(1-\frac{A+\dot{S}}{K_C''}\right)-\left(A+\dot{S}\right)\cdot\frac{A^2/K_F^2\cdot K_C''-\left(A+\dot{S}\right)}{\left(K_C''\right)^2}=0 \tag{30}$$

The expression (30) leads to:

$$\frac{A^2}{K_F^2}=\frac{\left(A+\dot{S}\right)^2}{\left(K_C''\right)^2} \tag{31}$$

From eq. (31), the thermal conductance of the condenser corresponding to the condensing zone can be obtained:

$$K_C''=\frac{K_F\cdot\left(A+\dot{S}\right)}{A} \tag{32}$$

Using eqns. (24) and (32), K_C'' can be written as follows:

$$K_C^{"} = K_F + \frac{\dot{S}\cdot\left(T_{SF}\cdot K_F - \dot{Q}_F\right)}{\dot{Q}_F} \tag{33}$$

Based on the finite-size constraint from eq. (17) and eq. (33) the optimum thermal conductance of the evaporator K_F^{opt} can be computed:

$$K_F^{opt} = \frac{K_T + \dot{S}}{2 + \dot{S}\cdot T_{SF}/\dot{Q}_F} \tag{34}$$

Substituting eq. (34) in eq. (33), at yields the optimum value for the thermal conductance at the condenser in the condensing zone $K_C^{"opt}$:

$$K_C^{"opt} = K_F^{opt} + \frac{\dot{S}\cdot\left(T_{SF}\cdot K_F^{opt} - \dot{Q}_F\right)}{\dot{Q}_F} \tag{35}$$

Using the eq. (17) and if in eq. (34) the term $\dot{S}$ is considered to be zero, the thermodynamic cycle is endoreversible and there is no desuperheating zone in the condenser; then it results the well known equipartition principle [5, 6]:

$$K_F^{opt} = K_C^{"opt} = \frac{K_T}{2} \tag{36}$$

which is a validation of the proposed optimization model in the case.

Substituting eqns. (34) and (35) in eqns. (27) and (28), the minimum compressor power consumption and maximum *COP* can be written:

$$P_{cp}^{\min} = (pc+1)\cdot T_{SC}\frac{A^{opt}+\dot{S}}{1-(A^{opt}+\dot{S})/K_C^{"opt}} - \dot{Q}_F \tag{37}$$

$$COP^{max} = \left[(pc+1)\cdot\frac{T_{SC}}{\dot{Q}_F}\cdot\left(\frac{A^{opt}+\dot{S}}{1-\dfrac{A^{opt}+\dot{S}}{K_C^{"opt}}}\right)-1\right]^{-1} \tag{38}$$

where,

$$A^{opt} = \frac{\dot{Q}_F}{T_{SF} - \dot{Q}_F/K_F^{opt}} \tag{39}$$

Furthermore, the optimum temperature differences between the working fluid and the heat sources can be determined.

Using eqns. (1) and (34) the optimum temperature difference at the evaporator can be expressed as:

$$\Delta T_F^{opt} = \frac{\dot{Q}_F}{K_F^{opt}} \tag{40}$$

Using eqns. (20) and (37), the minimum heat flux rejected at the condenser in the condensing zone can be written as:

$$\left|\dot{Q}_C^{''min}\right| = \frac{P_{cp}^{min} + \dot{Q}_F}{(pc + 1)} \tag{41}$$

Based on eqns. (6), (35) and (41), the optimum temperature difference at the condenser in the condensing zone can be computed:

$$\Delta T_C^{''opt} = \frac{\left|\dot{Q}_C^{''min}\right|}{K_C^{''opt}} \tag{42}$$

Based on eqns. (6) and (42), the optimum temperature difference at the condenser in the condensing zone leads to the optimum condensing temperature as follows:

$$T_C^{opt} = T_{SC} + \Delta T_C^{''opt} \tag{43}$$

The optimum condensing temperature determines the compressor outlet state (2 in Fig. 1) and inlet and outlet states at the condenser in the condensing zone (2'' and 3' in Fig. 1). In these conditions, the heat flux rejected in the desuperheating zone is minimum ($\left|\dot{Q}_C^{'min}\right|$) and the percent ($pc$) which determines this heat flux and the mean thermodynamic temperature of the working fluid (eq. 5) are optimum, involving pc^{opt} and $T_C^{'opt}$, respectively.

Furthermore, the optimum temperature difference between working fluid and heat sink in the desuperheating zone is:

$$\Delta T_C^{'opt} = T_C^{'opt} - T_{SC} \tag{44}$$

Based on eqns. (4), (19) and (44), the optimum thermal conductance of the condenser in the desuperheating zone can be determined:

$$K_C^{'opt} = \frac{\left|\dot{Q}_C^{'min}\right|}{\Delta T_C^{'opt}} = \frac{pc^{opt} \cdot \left|\dot{Q}_C^{''min}\right|}{\Delta T_C^{'opt}} \tag{45}$$

Using eqns. (3), (4) and (6), the overall thermal conductance of the condenser can be determined, which in optimum conditions can be expressed as:

$$K_C^{opt} = K_C^{''opt} + K_C^{'opt} \cdot \frac{\Delta T_C^{'opt}}{\Delta T_C^{''opt}} \tag{46}$$

Finally, the optimum variable values which lead to the economical functional regime, characterized by COP^{max}, can be synthetically expressed as:

$$parameters \begin{Bmatrix} \dot{Q}_F \\ T_{SF} \\ T_{SC} \\ K_T \\ pc \\ \dot{S} \end{Bmatrix} => optimized\ variables \begin{Bmatrix} K_F^{opt} \\ K_C^{''opt} \\ K_C^{opt} \\ \Delta T_F^{opt} \\ \Delta T_C^{''opt} \end{Bmatrix}$$

The economical functional regime is the functional regime targeted during the design activity of single stage VCRS.

3. Numerical simulation

As it can be seen from eqns. (34), (35), (38) and (39) the values of K_F^{opt}, $K_C^{''opt}$ and COP^{max} depend on the following parameters: $\dot{Q}_F$, pc, T_{SF}, T_{SC}, K_T and $\dot{S}$. It is difficult to establish proper values for pc, K_T and $\dot{S}$ because they depend on the type of refrigerant being used. In order to outcome this difficulty, a program has been developed in Engineering Equation Solver (EES) [7]. For a certain initial set of parameters $\dot{Q}_F$; T_{SC}; T_{SF}; ΔT_F; $\Delta T_C^{''}$ and type of refrigerant, the program allows to: determine the thermodynamic state parameters (pressure, temperature, quality and mass specific properties: enthalpy, entropy, volume) in all points of the single stage VCRS thermodynamic cycle (Fig. 1); $\left|\dot{Q}_C^{'}\right|$, $\left|\dot{Q}_C^{''}\right|$ and thus $\left|\dot{Q}_C\right|$; K_F, $K_C^{'}$, $K_C^{''}$ and K_T; $\dot{S}$ and pc parameter values.

Based on this program, a numerical simulation has been carried out. In order to verify the correctness of the optimization model, the input data has been chosen according to [8], as follows: $\dot{Q}_F = 30\ kW$; $T_{SC} = 303\ K$; $\Delta T_F = 6\,K$; $\Delta T_C^{''} = 8\,K$, for the refrigerant R134a. Also, the temperature of the cold heat source has been considered within the range $T_{SF} = 253 \div 283\,K$. The results are presented below.

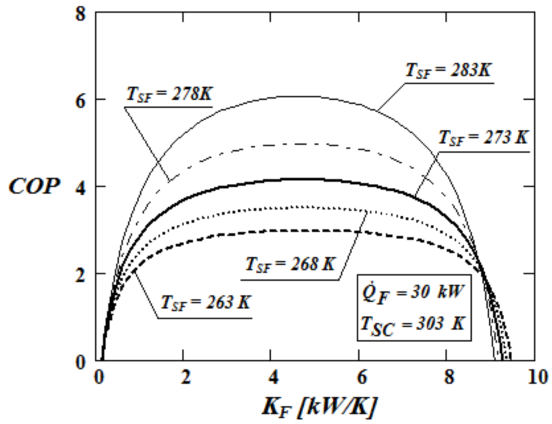

Fig. 2. COP variation in function of the evaporator thermal conductance for various temperatures of the cold heat source

Fig. 2 presents the COP as a function of the evaporator thermal conductance for various temperatures of the cold heat source. As it can be seen, the *COP* presents a maximum value (COP^{max}) for certain optimum values of K_F^{opt}. In condition of partial finite size constraint ($K_T = ct.$), the values K_F^{opt} lead to the optimum values for the thermal conductance of the condenser in the condensing zone ($K_C^{''opt}$). For K_F^{opt} and $K_C^{''opt}$ the optimization model points out their analytical expressions (eqns. 34 and 35).

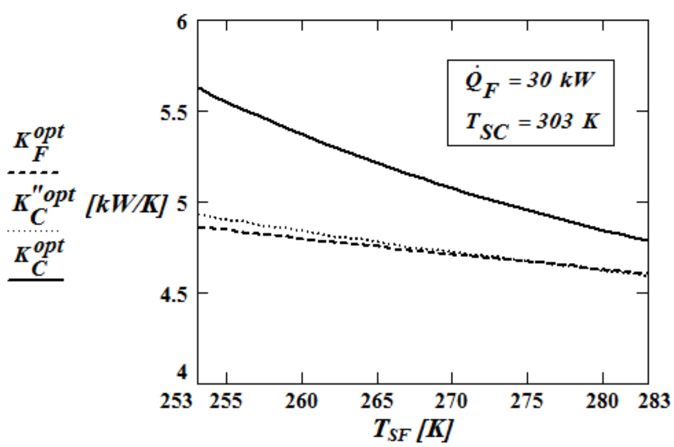

Fig. 3. Variation of optimum thermal conductances distribution in function of the temperature of the cold heat source

Fig. 3 presents the variation of the optimum thermal conductance distribution K_F^{opt}, $K_C^{''opt}$ and K_C^{opt}, respectively, as a function of the cold heat source temperature (T_{SF}), for refrigerant R134a. The increase of T_{SF} leads to the decrease of the optimum thermal conductances. For $T_{SF} < 271$ *K*, between the optimum thermal conductances exists the following relation: $K_C^{opt} > K_C^{''opt} > K_F^{opt}$. For $T_{SF} = 271$ *K*, $K_C^{''opt} = K_F^{opt}$ and for $T_{SF} > 271$ *K*, the values for $K_C^{''opt}$ are very close to those of K_F^{opt}. Also, Fig. 3 points out that the desuperheating process should not be neglected in the design of VCRS because it has a strong influence on the optimum overall conductance of the condenser. Similar values for the optimum thermal conductances have been also reported in paper [8]

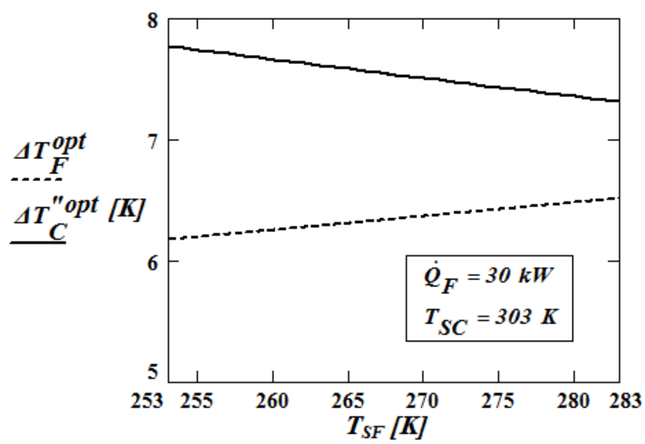

Fig. 4. Variation of optimum temperature difference in function of the temperature of the cold heat source

Fig. 4 shows the variation of the optimum temperature differences at the evaporator (ΔT_F^{opt}) and condenser in the condensing zone ($\Delta T_C^{''opt}$) as a function of T_{SF} for refrigerant R134a. As it can be seen, the increase of T_{SF} leads to the increase of ΔT_F^{opt}, and to the decrease of $\Delta T_C^{''opt}$. For a given T_{SF} value between the optimum temperature differences the following relation exists: $\Delta T_F^{opt} < \Delta T_C^{''opt}$. In the design of VCRS, assuming $\Delta T_C^{''opt}$ as the design temperature difference between the working fluid and heat sink will lead to the correct K_C^{opt} only if the desuperheating process is taken into consideration. The values of the optimum temperature differences are similar to those obtained in the paper [8].

Fig. 5 presents the variation of the maximum cooling efficiency (COP^{max}) as a function of T_{SF} for refrigerant R134a. The increase of T_{SF} leads to the increase of COP^{max}. For T_{SF} =263 K, the value obtained for COP^{max} =2.97 is close to the one obtained in paper [8].

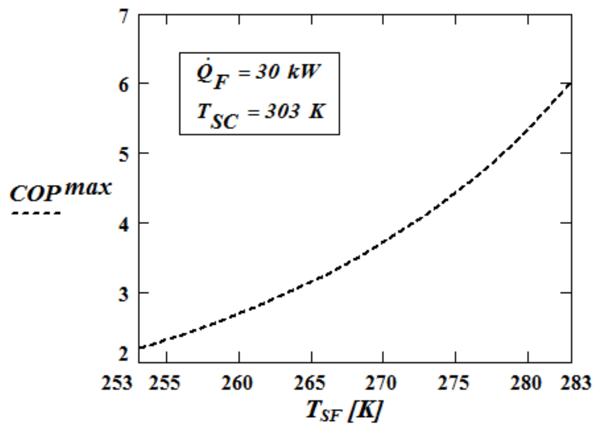

Fig. 5. Variation of COP^{max} in function of the temperature of the cold heat source

Using Fig. 3 to Fig. 5, for given $\dot{Q}_F$, T_{SF}, T_{SC} and a certain type of refrigerant (in this case R134a) the optimum variables which lead to COP^{max} can be chosen, thus serving in the optimum design of IFV.

4. Study on the behavior of VCRS at non-nominal cooling capacity

If we choose a nominal cooling capacity $\dot{Q}_F = 30\ kW$, $T_{SF} = 263\ K$, $T_{SC} = 303\ K$ based on the program developed in EES the following parameters which are present in the expression (38) of COP^{max} can be determined: $pc = 0.09782$; $K_F^{opt} = 4.768\ kW/K$; $K_C^{''opt} = 4.80\ kW/K$; $\dot{S} = 0.0007567\ kW/K$. Assuming that a single stage VCRS has been designed according to this optimum data, the influence of cooling capacity on the COP^{max} can be pointed out. This influence is presented in Fig. 6, where the cooling capacity takes values within the interval $\dot{Q}_F = 1 \div 200$ kW.

In Fig. 6, it must be noted that all the values corresponding to the COP^{max} have been determined using the analytical expression (eq. 38). Thus, for a given set of parameters, COP^{max} presents a maximum maximorum value corresponding to an optimum cooling capacity. This observation has also been made in [9].

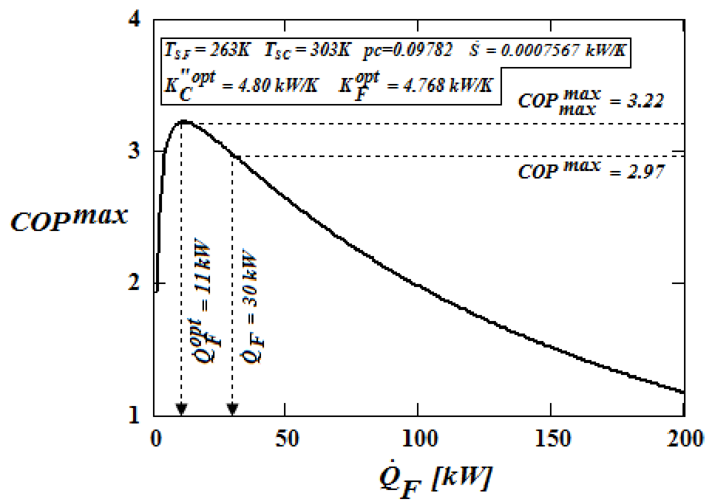

Fig. 6. Variation of COP^{max} in function of the cooling capacity

For the present set of parameters obtained in the nominal regime, the maximum maximorum value is $COP_{max}^{max} = 3.22$, corresponding to an optimum value of the cooling capacity $\dot{Q}_F^{opt} = 11\ kW$. One can observe that COP_{max}^{max} is *7.8 %* higher than in nominal regime ($COP^{max} = 2.97$), while $\dot{Q}_F^{opt}$ is *63%* lower than the nominal one ($\dot{Q}_F = 30\ kW$).

This behavior of VCRS already designed (in optimum constructive and functional condition to obtain COP^{max}) at non-nominal operating regime is justified because the refrigeration system, having larger heat exchangers, is used at a lower cooling capacity involving lower temperature differences and consequently, lower compressor power consumption. Its variation of COP^{max} as a function of the cooling capacity is unique. Thus, for this considered VCRS, the maximum maximorum value can be obtained only for the optimum cooling capacity.

Based on this observation, the idea of designing a single stage VCRS according to optimum conditions for a given nominal cooling capacity and than using it at lower cooling capacity leads to a higher COP than the nominal one. Moreover, single stage VCRS are usually designed by increasing the cooling demand, obtained by summing all the heat fluxes extracted from the cold space, with 5 ÷ 20% [1]. This is done in order to cover inappropriate exploitation situations and, at the same time, according with the previous idea, higher COP than the nominal one will be assured. Taking into consideration that larger cooling capacities involve larger investment costs, a maximum increase of the cooling demand with 20% could be accepted.

Pointing out the COP_{max}^{max} value, corresponding to the $\dot{Q}_F^{opt}$, as a function of the refrigerant type (R134a in this case), is a very important advantage of the present optimization model.

5. Conclusions

The paper presents an optimization model of single stage vapour compression refrigeration systems. The aim is to find the optimum constructive (thermal conductances distribution) and functional parameters (temperature differences working fluid - heat sources) which lead to a maximum cooling efficiency in function of the refrigerant type. The optimization model is developed in conditions of imposed cooling capacity and finite size constraint (known total thermal conductance). Also, external and internal irreversibility sources have been taken into consideration, caused by the heat transfer at finite temperature differences working fluid heat-sources and the imperfection of the processes which compose the thermodynamic cycle in general and in particular of the compression and expansion processes, respectively.

In order to find the optimum thermal conductance distribution between evaporator and condenser, the optimum temperature differences between working fluid - heat sink and working fluid - cold heat source, which lead to a maximum cooling efficiency in relation with the refrigerant type a program, has been developed in Engineering Equation Solver. In a first step, based on this program a numerical simulation has been carried out in function of the temperature of the cold heat source. The relations between optimum values of the thermal conductances and temperature differences have been established. A correct value for the optimum overall thermal conductance of the condenser can be obtained only if the desuperheating process is taken into consideration. Also the influence of the cold heat source temperature on the maximum cooling efficiency has been pointed out. The results obtained are in good correlation with those obtained in similar papers.

In a second step, assuming that a single stage vapour compression refrigeration system has been designed in optimum conditions, the influence of the cooling capacity on the maximum cooling efficiency has been pointed out. The results show that the maximum cooling efficiency presents a maximum maximorum value which corresponds to an optimum cooling capacity much lower than the one for which the refrigeration system has been designed. In the usual single stage VCRS design method the cooling demand is increased with 5 ÷ 20%. This covers the inappropriate exploitation situations and leads to higher COP than the nominal one. Because of investment costs, a maximum increase of the cooling demand with 20% could be accepted.

Moreover, a very important advantage of this optimization model is that it allows establishing the maximum maximorum cooling efficiency and the corresponding optimum cooling capacity in function of the refrigerant type.

Future development of the optimization model could involve a comparison between different refrigerants and also mathematical computations in order to find the analytical expression of the maximum maximorum *COP* and the corresponding optimum cooling capacity.

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