THERMAL CHARACTERISTICS OF COMBINED COMPRESSOR – EJECTOR REFRIGERATION / HEAT PUMP SYSTEMS FOR HVAC&R

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Thermal characteristics of combined compressor – ejector refrigeration/heat pump systems applied in heating, ventilation, air conditioning and refrigeration (HVAC&R) of buildings are investigated. An original model for estimation of the thermal characteristics of the combined cycles is developed, to determine the influence of the evaporation, interstage, condensation, and generating temperature conditions on mechanical and thermal COPs of the combined system, and to optimize the thermal parameters of the cycle. Results are presented for different temperature conditions, with R134a as a suitable refrigerant. A comparison between the thermal characteristics of the simple mechanical vapor compression cycle, the simple ejector thermocompression cycle, and the combined compressor – ejector refrigeration / heat pump cycle is given. The benefits of implementation of combined compressor – ejector refrigeration/heat pump cycles in HVAC&R systems are discussed. The temperature lift or temperature difference between condensing temperature and interstage temperature significantly influences the thermal (ejector) coefficient of performance. If temperature lift is between 10 K and 20 K, high values of thermal COPs can be achieved (0.5÷1.0, for generating temperature equal to 80℃; 1.0÷1.8, for generating temperature equal to 120℃); If temperature lift is between 30 K and 40 K, very low values of COP_{th} *can be obtained (0.05÷0.3). High values of mechanical COPs can be achieved (24.8÷6.9), for compressor stage temperature lift 10÷30 K.*

Key words*: energy efficiency, compressor, ejector, thermocompression, HVAC&R, refrigeration, heat pump*

1. Introduction

Driven by technical, economical and ecological reasons connected with the global warming and climate changes, energy efficiency improvement becomes a main topic and subject of many research and development activities. A sustainable development in the energy sector is based on the concept of dispersed multi-energy generation, conversion, transmission, and storage; renewable energy systems (solar energy, wind energy, geothermal energy and hydro energy); combined co-generation, tri-generation and multi-energy generation systems; energy efficient technologies in the industrial sector; energy efficient buildings and energy-efficient systems for HVAC&R; thermal and cooling storage; electrical energy storage (electrical bateries; cryogenic energy storage – (Liquid Air Energy Storage); thermal systems with fuel cells, hydrogen production); thermal cycles with CO_2; decarbonization, carbon capture and storage [1,2]. Application of renewable energy resources (solar energy, wind energy, geothermal energy, hydro energy, etc.), use of low-temperature waste heat, application of natural refrigerants, construction of energy-efficient buildings and energy-efficient systems for HVAC&R of buildings and facilities is a high priority for sustainable energy development [3].

The subject of this paper are combined thermal systems consisting of a compressor refrigeration / heat pump system and an ejector thermocompression system. The ejector system uses thermal energy from renewable sources, or low-temperature waste heat from technological processes, or heat generated in the tri-generation systems as a motive energy for the thermal ejector cycle. The compressor uses mechanical energy for vapor compression cycle. The combination of these two cycles, the thermal ejector cycle and the mechanical compressor cycle, provides an energy-efficient system, appropriate for implementetion in HVAC&R.

Utilization of solar energy as a thermal energy source for the ejector cycle in a hybrid, mechanical compression enhanced ejector cycle is suggested in [4,5]. The use of solar powered ejector cycle as a cooling cycle to reduce the temperature in the condenser of a traditional vapor compressor cooling system is described in [6,7,8]. A review of combined ejector and mechanical vapor-compression refrigeration systems and heat pumps is given in [9]. The ejector cycle has low operating cost because of the use of cheap thermal energy that is available [10,11]. Nevertheless, the ejector cycle has a low coefficient of performance (COP) in comparison with the mechanical compression cycle [12,13]. The combination of both: thermal ejector cycle and mechanical compressor cycle leads to high COP values, competitive with the values of COP achieved with absorption refrigeration machines [14,15].

Ejector systems have no moving parts, relatively low capital cost, simple operation, reliability, and low maintenance cost. Despite the simplicity of construction, the flow phenomena occurring in the ejector flow field are very complex and are subject of numerous scientific research, physical and numerical experiments and CFD analysis [16,17,18,19].

The purpose of this paper is to present an original calculating model for thermal characteristics of combined compressor – ejector refrigeration / heat pump cycles applied in HVAC&R systems, to determine the influence of the evaporation and condensation temperature conditions on mechanical and thermal COPs of the combined system, and to optimize the thermal parameters of the refrigeration / heat pump cycle. A comparison between thermal characteristics of the simple mechanical vapor compression cycle, the simple ejector thermocompression cycle, and the combined compressor – ejector refrigeration / heat pump cycle is given, and the benefits of implementation of combined compressor – ejector refrigeration / heat pump cycles in HVAC&R systems are discussed.

2. Description of the combined compressor-ejector system

Scheme of a combined compressor-ejector refrigeration / heat pump system and $p-h$ diagram of the cycle is given in Fig.1. Low-temperature heat (geothermal energy, solar energy, waste heat from industrial processes, heat in tri-generation systems, etc.) is used in thermal ejector cycle. Two-stage compression is applied in the combined compressor-ejector cycle. The first stage is mechanical compression (C). The second stage is vapor ejector compression (EJ_v). Between the stages an economizer (EC) is introduced. Two control-throttling valves are installed: the first one is located between the condenser (Co) and the economizer (EC). The second one is installed between the economizer (EC) and the evaporator (E). Low temperature heat is utilized for production of generating motive vapor in the generator (G).

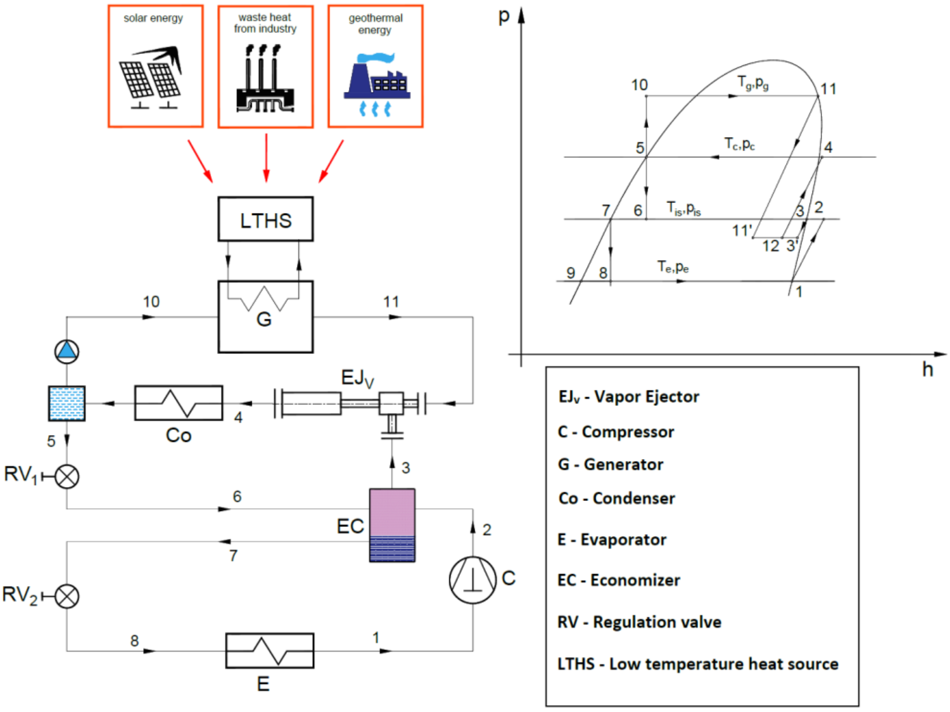

Fig. 1 Combined compressor - ejector refrigeration / heat pump system and p – h diagram of the cycle

In the generator (G) the heat Q_g from the low temperature heat source (LTHS) is transfered to the refrigerant, which evaporates, and with high energy potential (generating pressure p_g and temperature T_g) exits the generator. This potential is used for realization of the refrigeration cycle.

In the primary nozzle of the ejector (EJ_v) generating motive vapor (primary flow) expands (11–11'). The primary flow draws the secondary flow into the ejector mixing chamber. The secondary flow flows through the ejector secondary nozzle and expands (3–3'). Inside the ejector mixing section complex process of momentum transfer occurs (11'–12, 3'–12). The ejector mixing section process is the first main source of thermodynamic irreversibility, hydraulic loss and loss of total pressure. If the entrainment ratio is high (secondary mass flow rate is much larger than primary mass flow rate), then the loss of total pressure in the ejector mixing chamber process is strongly expressed. The pressure ratio of the ejector is limited, which causes limitation of the ejector cycle performance characteristics.

As the combined stream flows through the mixing chamber and the diffuser, compression is achieved. The kinetic energy of the combined vapor stream flow is transformed into rise of enthalpy in the diffuser, and expressed by rise of the pressure. The combined flow at the mixing chamber is supersonic. A normal shock wave occurs (12–12.1) if the velocity of the combined flow is supersonic. In the shock wave the compression is occuring. The shock wave is a thermodynamicaly irreversible process, with entropy increase. This process is the second main source of thermodynamic irreversibility and exergy decrese in the ejectors. When the first main source of thermodynamic irreversibility (process of momentum transfer in the mixing chamber) is weaker, the second one is strongly expressed and vice versa. The thermodynamic irreversibility of these processes cannot be avoided by any design effort because both of them are physics phenomena. Additional compression is realized in the subsonic diffuser (12.1 – 4).

Performance characteristics of the ejector refrigeration systems depend on the performance characteristics of the ejector. The characteristics of the ejector are determined by the refrigeration system operating conditions, and by the degree to which the ejector flow field has been optimally designed. Despite ejector apparent simplicity, the physics phenomena affecting ejector performance are complex and should receive proper attention in the design procedure.

In the condenser (Co) the flow compressed up to condensing pressure (p_c) and corresponding condensing temperature (T_c) condenses (4–5). Via control throttling (regulating) valve (RV_1) (5–6) the secondary flow goes in the intercooler (EC). Via control throttling (regulating) valve (RV_2) (7–8) the refrigerant goes in the evaporator (E), where evaporates (8–1), achieving refrigeration effect (cooling, chilling effect) Q_e at evaporating temperature T_e and corresponding evaporating pressure p_e. The vapor is compressed with mechanical compressor (C) up to the interstage pressure (p_{is}).

With pump (P) the primary flow is pressurized (5–10) up to the generating pressure p_g. In the generator (G) the primary flow is heated and evaporated (10–11) consuming heat Q_g.

Thermal coefficient of performance $COP_{r\,th}$ is the most important indicator for energy efficiency of ejector refrigeration systems, defined as a ratio between the refrigeration effect Q_e and generating heat Q_g.

$$COP_{r\,th} = Q_e/Q_g = M_{sec}\,(h_3 - h_8)\,/\,M_{pr}\,\,(h_1 - h_9) \,=\, \omega\,(h''_e - h'_c)\,/\,\,(h''_g - h'_c) \tag{1}$$

The COP depends on the entrainment ratio $\omega = M_{sec} / M_{pr}$, which is the ratio between the secondary and primary flow rates, and on the enthalpies of saturated liquid (h') and saturated vapor (h'') at evaporating (h_e), condensing (h_c) and generating (h_g) pressures.

For heat pumps the thermal coefficient of performance $COP_{hp\,th}$ is defined as a ratio between the condensation heat Q_c and generating heat Q_g:

$$\begin{aligned} COP_{hp\,th} &= Q_c/Q_g = (Q_e + Q_g)/Q_g = (M_{sec} + M_{pr})(h_6 - h_7)/M_{pr}\,\,(h_1 - h_9) = \\ &= 1 + \,\omega\,(h''_e - h'_c)\,/\,\,(h''_g - h'_c) \end{aligned} \tag{2}$$

Evaporator (cooling) capacity:

$$Q_e = M_{sec}\,(1 - x_6)(h_1 - h_8) \tag{3}$$

Compressor power consumption:

$$P = M_{sec}\,(1 - x_6)(h_2 - h_1) \tag{4}$$

Mechanical $COP_{r\,mech}$ is the most important indicator for energy efficiency of the vapor compression refrigeration systems:

$$COP_{r\,mech} = Q_e/P \tag{5}$$

For heat pumps the mechanical coefficient of performance $COP_{th\,mech}$:

$$COP_{hp\,mech} = Q_c/P \tag{6}$$

3. Model of the thermal characteristics of combined compressor – ejector cycle

The calculation model for estimation of the thermal characteristics of the combined compressor – ejector refrigeration / heat pump cycle is based on the standard calculation procedure for vapor compression cycles and the calculation model for ejector cycles given in details [1].

The model involves the following input parameters:

- evaporation temperature T_e ; condensing temperature T_c ; generating temperature T_g ;
- refrigerant (R134*a*);
- refrigeration cooling capacity Q_e;
- compressor efficiency η_{com}; ($\eta_{com} = 0.7 \div 0.8$).

The interstage temperature (T_{is}) is taken as a variable parameter:

- interstage temperature T_{is} ; ($T_c > T_{is} > T_e$)

The calculation model is implemented in MATLAB/REFPROP for both cycles of the compressor and the ejector stage to estimate the following parameters:

- pressures, temperatures, entalpies, entropies, densities;

For compressor stage the following parameters are calculated:

- specific cooling capacities; compressor power consumption; heating capacities; flow rates;

The calculation model for the ejector stage is given below.

The ejector main elements are: primary nozzle, secondary nozzle, mixing chamber and diffuser (Fig. 3). Thermal *h* - *s* diagram of the processes in the ejector is given in Fig. 4.

In the ejector primary nozzle (1) motive vapor expands (11-11') (Fig. 4) from the high pressure p_1 to the pressure p_2 which is lower than secondary flow suction pressure. The flow at the outlet of the primary nozzle is supersonic. The nozzle profile is convergent-divergent.

In the narrowest (critical) flow cross-section of the nozzle the pressure is equal to the critical pressure $p_{cr}= p_1 (2/(\kappa+1))^{(\kappa/(\kappa-1))}$, and the speed is equal to the local speed of sound $a = \sqrt{\partial p/\partial \rho}$. The velocity equal to the local speed of sound is:

$$c_{cr}=(2\Delta h_{cr})^{1/2}=(\kappa R T_1)^{1/2}(2/(\kappa+1))^{1/2} \tag{7}$$

Using isentropic expansion from p_1 to p_{cr} the specific volume v_{cr} and the density $\rho_{cr} = 1 / v_{cr}$ can be calculated. The primary mass flow is, $M_{pr} = A_{cr} c_{cr} \rho_{cr} \zeta_{cr}$.

For well-designed nozzles with smooth surface treatment the flow coefficient is ζ_{cr}=0.97–0.98.

The speed of the flow exiting the primary nozzle is:

$$c_{11'}=\Psi_{pr} c_{11's}=[2(h_{11}-h_{11'})]^{1/2}=(2\Delta h_a \eta_{pr})^{1/2}=(2\Delta h_s \eta_{pr})^{1/2} \tag{8}$$

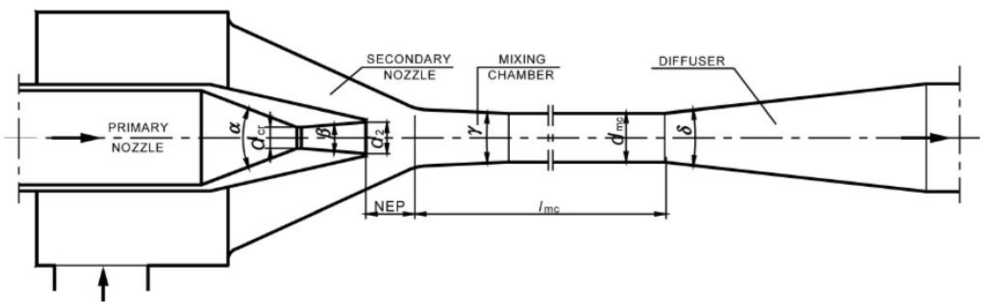

Fig. 2 Geometrical parameters of the ejector flow field

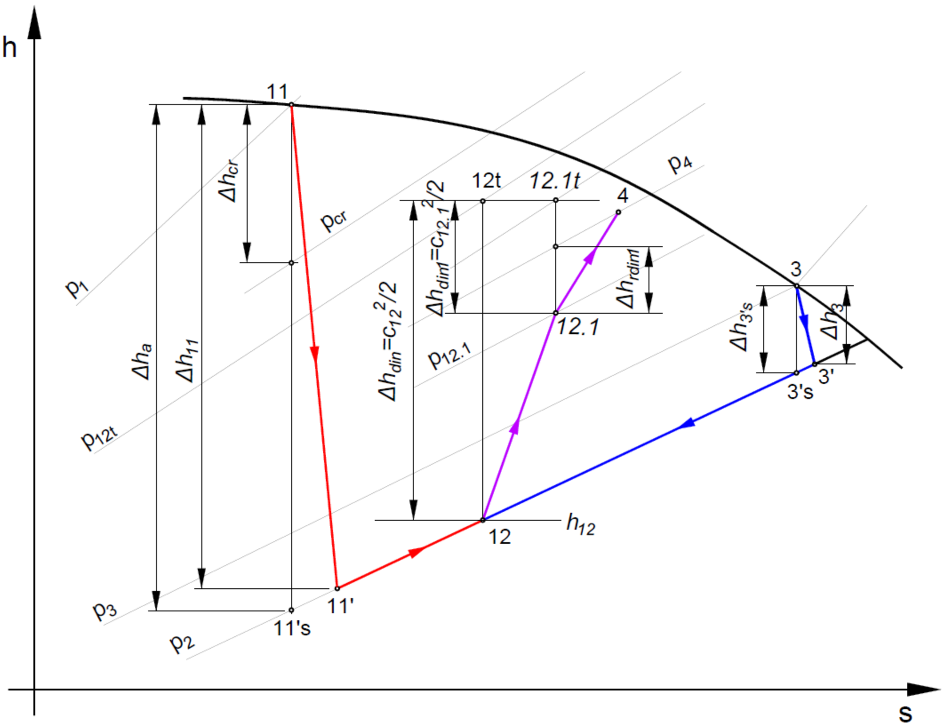

Fig. 3 h - s diagram of the processes in the ejector (refers to refrigerant with wet expansion)

Exiting the primary nozzle, the fluid additionally expands, entering into the mixing chamber, where complex flow phenomena appear between the primary and secondary flow. The primary flow draws the secondary flow into the mixing chamber. The secondary flow comes through the secondary nozzle where it expands (3-3'). The shear layer between the primary and secondary fluids flowing with a large velocity difference leads to the acceleration of the secondary flow. If the secondary fluid hits critical flow (choking flow), then these operating conditions of the ejectors are often referred to as a "double choking" operation.

The speed of the secondary flow is:

$$c_{3'} = \Psi_{sec} \cdot c_{3's} = [2 \cdot (\square_3 - \square_{3'})]^{1/2} = (2 \cdot \Delta\square_{3's} \cdot \eta_{sec})^{1/2} \tag{9}$$

The calculation of all thermodynamic quantities (temperature, enthalpy, entropy, density) for states 11' and 3' can be performed using the equations for polytropic expansion (11–11') and (3–3'), data on the thermodynamic properties of refrigerant (REFPROP).

According to the analysis of many publications, the expected values of the velocity coefficient for the nozzles ($\Psi_{pr};\Psi_{sec}$) are 0.95÷0.98, and the corresponding utilization coefficient (η_{pr} ; η_{sec}) is 0.92÷0.96.

Using the momentum conservation law and the momentum equation for the mixing chamber, assuming that the process takes place at the constant cross section mixing chamber, if the flow friction forces are covered by the mechanical efficiency coefficient (η_{mc}=0.95÷0.98), the equation for the velocity of the combined flow is obtained:

$$c_{12}=\eta_{mc}\left(c_{11'}m_{pr}+c_{3'}m_{sec}\right); \quad m_{pr}=\frac{M_{pr}}{M_{pr}+M_{sec}}; \quad m_{sec}=\frac{M_{sec}}{M_{pr}+M_{sec}} \tag{10}$$

The kinetic energy losses, i.e. the total pressure losses in the mixing chamber in the process of momentum transfer are:

$$\delta_e = \frac{\Delta E}{E_{11'} + E_{3'}} = \frac{m_{pr} c_{11'}^2 + m_{sec} c_{3'}^2 - c_{12}^2}{m_{pr} c_{11'}^2 + m_{sec} c_{3'}^2} \tag{11}$$

Using the energy conservation law for the mixing chamber, the enthalpy of the combined flow can be determined:

$$h_{12} = m_{pr} h_{11'} + m_{sec} h_{3'} + m_{pr} \frac{c_{11'}^2}{2} + m_{sec} \frac{c_{3'}^2}{2} - \frac{c_{12}^2}{2} \tag{12}$$

With the pressure $p_{11'}=p_{3'}=p_{12}$ and with the enthalpy h_{12} (state 12 is defined), the values of the other thermodynamic quantities can be determined. The dynamic component $\Delta h_{din} = c_{12}^2$ and the total pressure are defined by the velocity c_{12}.

Vapor compression occurs when the combined flow flows through the mixing chamber and the diffuser. The kinetic energy $\Delta h_{din}=c_{12}^2/2$ is transformed into an increase in enthalpy, expressed by increased pressure.

The mixed flow, after the process of momentum transfer in the mixing chamber, is supersonic. Transition from supersonic flow to subsonic occurs in a normal shock wave where the speed drops sharply from supersonic to subsonic, and the pressure rises sharply (12-12.1). Mach number of the supersonic flow, upstream of the shock wave is, $\lambda_1 = c_{12} / a_{cr} > 1$, and downstream of the shock wave is, $\lambda_2 = c_{12.1} / a_{cr} < 1$. Across the shock wave $\lambda_1 \lambda_2 = 1$. The pressure rise in the shock wave is:

$$\frac{p_{12.1}}{p_{12}} = \frac{\lambda_1^2 - (\kappa - 1)/(\kappa + 1)}{1 - (\kappa - 1)\lambda_1^2/(\kappa + 1)} \tag{13}$$

This equation defines the pressure $p_{12.1}$ and the velocity after the shock wave $c_{12.1}$. The dynamic component $\Delta h_{din1} = c_{12.1}^2 / 2$ and the total enthalpy h_{din12t} for state 12.1 are defined by the velocity $c_{12.1}$. According to the entropy $s_{12.1}$, the increase in entropy through the shock wave can be calculated and the thermodynamic irreversibility can be estimated. In the shock wave the compression is realized. However, the shock wave is a thermodynamically irreversible process, with entropy rise. It is the second main source of thermodynamic irreversibility and exergy decrement in the ejectors.

The efficiency of the subsonic diffuser defined as a ratio between isentropic compression work Δh_{rdin1} and dynamic pressure at the subsonic diffuser inlet $\Delta h_{din1}=c_{12.1}^2/2$:

$$\eta_d = \Delta h_{rdin1} / \Delta h_{din1} = \Delta h_{rdin1} / (c_{12.1}^2 / 2) \tag{14}$$

The values of diffuser efficiency η_d are from 0.60 up to 0.80, depending on shape and operating conditions. Using the previous equation, the pressure $p_4=p_e$ can be determined and the thermodynamic values for state 4 at the ejector outlet can be obtained.

The efficiency coefficients (efficiency) η_{pr}, η_{sec}, η_{mc} и η_d, which define the efficiency of the ejector depend on the design characteristics and the shape of the elements of the flow field of the ejector, the thermodynamic properties of the refrigerant, as well as the fluid flow conditions.

The calculation model for ejector cycles is verified by comparing the results for thermal COP with experimental results obtained in an ejector concentrator [1]. Calculation model results are in the range of measurement uncertainty of experimental results. Standard calculation procedure for vapor compression cycles is applied.

4. Results of the model for thermal characteristics of combined compressor – ejector system

Analyses of the thermal characteristics of the system are performed for cooling (summer air conditioning mode) and for heating – heat pump (winter air conditioning mode), for different temperature conditions, i.e., evaporation, intersatge, condensation, and generation temperatures.

Cooling operating conditions (summer mode):

- Recirculation water for cooling (chilling water)
  - temperature 7 / 12 °C - evaporation temperature $T_e = 2 \div 5$ °C
- Cooling of products or cooling storage
  - temperature of $0 \div 7$ °C - evaporation temperature $T_e = -5 \div +3$ °C
- Condensation temperature $T_c = 30 \div 50$ °C.

Heat pump operating conditions for heating (winter mode) and for preparing sanitary hot water:

- Heat source: water
  - temperature mode in the evaporator $T_{w1}/T_{w2} = (12 \div 15) / (5 \div 10)$ °C
    evaporation temperature $T_e = 2 \div 7$ °C;
  - temperature mode in the evaporator $T_{w1}/T_{w2} = 20 / 15$ °C
    evaporation temperature $\mathrm{T_e} = 15 \div 10$ °C;
- Heat source: air
  - temperature mode in the evaporator $T_{v1}/T_{v2} = 10 \div (-5) / 7 \div (-8)$ °C
    evaporation temperature $\mathrm{T_e} = 5 \div (-10)$ °C.
- Condensation temperature
  - $T_c = 55$ °C - temperature mode of water in the heating system
    $T_{w1}/T_{w2} = 50/45$ °C or 50/40 °C
  - $T_c = 60$ °C - temperature mode of water in the heating system
    $T_{w1}/T_{w2} = 55/50$ °C or 55/45 °C.

Generator working conditions:

- $T_g = 70 \div 130$ °C - depends on the characteristics of the low-temperature generator heat source (solar energy, geothermal energy, waste heat).

Optimisation of the interstage pressure is essential for achieving high thermal and mechanical coefficients of thermotransformation ($COP_{th}=Q_e/Q_g$ and $COP_{mech}=Q_e/P_{mech}$ in the cooling mode – refrigeration system, respectively $COP_{th}=Q_c/Q_g$ and $COP_{mech}=Q_c/P_{mech}$ in heating mode – heat pump), where Q_e is the cooling capacity / heat source capacity, Q_c is the heat capacity of the condenser / heat consumption, Q_g is the heat capacity of the low-temperature heat generator, and P_{mech} is the power consumption of the compressor unit. Compressor power consumption of a compressor chiller / heat pump depends on interstage pressure and compressor performance and efficiency. Generator heat consumption depends on interstage pressure and ejector performance and efficiency.

Optimization of the interstage pressure can be performed according to techno-economic criteria, including capital investment costs and energy costs (electricity costs and low-temperature heat energy costs). Calculations have been performed for different temperature conditions, with R134*a* as a suitable refrigerant for compressor-ejector refrigeration systems for air conditioning applications, using the model (section 3). The calculations are performed for the efficiency coefficient of the compressor $\eta_c=0.75 \div 0.85$, and for mean values of efficiency coeficients of the ejector elements.

Results of the model for thermal characteristics of combined compressor – ejector system for different evaporation temperature (T_e), condensing temperature (T_c), interstage temperature (T_{is}) and generating temperature (T_g) are given in Fig. 4. The temperature conditions (T_e, T_c, T_{is}, T_g) have strong influence on the combined compressor – ejector system performance (efficiency) coefficients:

mechanical – cooling mode ($COP_{r\ mech}$), mechanical – heating mode ($COP_{hp\ mech}$), thermal – cooling mode ($COP_{r\ th}$), thermal – heating mode ($COP_{hp\ th}$).

The temperature lift or temperature difference between condensing temperature T_c and interstage temperature T_{is} ($\Delta T_{Ej} = T_c - T_{is}$) significantly influences the thermal (ejector) coefficient of performance ($COP_{r\ th}$ and $COP_{hp\ th}$). If ΔT_{Ej} is between 10 K and 20 K, high values of $COP_{r\ th}$ can be achieved (0.5 ÷ 1.0, for T_g = 80 °C; 1.0 ÷ 1.8, for T_g = 120 °C); respectively $COP_{hp\ th}$ is (1.6 ÷ 2.1, for T_g = 80 °C; 2.1 ÷ 2.9, for T_g = 120 °C). If ΔT_{Ej} is between 30 K and 40 K, low (very low) values of $COP_{r\ th}$ can be achieved (0.05 ÷ 0.3, for T_g = 80 °C; 0.3 ÷ 0.6, for T_g = 120 °C); respectively $COP_{hp\ th}$ is (1.15 ÷ 1.4, for T_g = 80 °C; 1.4 ÷ 1.7, for T_g = 120 °C).

The temperature lift or temperature difference between interstage temperature T_{is} and evaporating temperature T_e and ($\Delta T_{COM} = T_{is} - T_e$) influences the mechanical (compressor) coefficient of performance ($COP_{r\ mech}$ and $COP_{hp\ mech}$). High values of $COP_{r\ mech}$ can be achieved (24.8 ÷ 6.9); respectively $COP_{hp\ mech}$ is (26.5 ÷ 8.6), for ΔT_{COM} = 10 ÷ 30 K.

Optimization of the interstage temperature T_{is} can be performed using technical (thermal) and economical criteria, which will comprise investments costs and opteration costs (energy efficiency, price of electrical energy, price of heat – waste heat, solar energy, and geothermal energy).

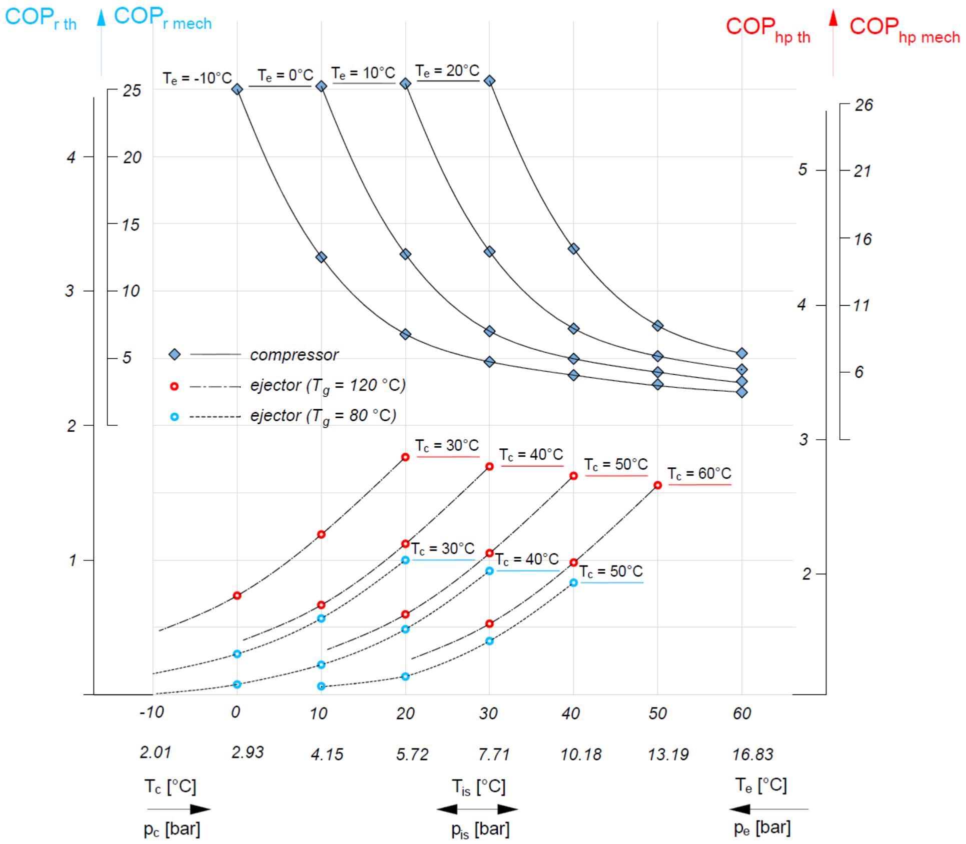

Fig. 4 COPs of the combined compressor - ejector system for different temperature conditions

Thermo-economic analysis and optimization is not subject of this paper. However, the possibility to achieve high values of compressor - ejector system thermal and mechanical COPs provides energy benefits (lower energy consumption), economic benefits (lower energy costs, lower investment costs in comparison with other thermal refrigeration / heat pump systems, e.g. absorption systems) and environmental benefits (lower CO_2 footprint).

5. Conclusions

Thermal characteristics of combined compressor-ejector refrigeration/heat pump systems applied in HVAC&R of buildings are presented. An original model and MATLAB/REFPROP program of combined compressor – ejector cycles are developed. The influence of the evaporation, interstage, condensation, and generating temperature conditions on mechanical and thermal COPs is determined. Results are presented for different temperature conditions, with R134a as a suitable refrigerant. The benefits of implementation of combined compressor – ejector refrigeration / heat pump systems in HVAC&R are discussed.

The temperature lift or temperature difference between condensing temperature T_c and interstage temperature T_{is} ($\Delta T_{Ej}=T_c-T_{is}$) significantly influences the thermal (ejector) coefficient of performance. If temperature lift is between 10 K and 20 K, high values of thermal COPs can be achieved (0.5 ÷ 1.0, for generating T_g = 80 ℃; 1.0 ÷ 1.8, for T_g = 120 ℃). If temperature lift is between 30 K and 40 K, very low values of thermal COPs can be obtained (0.05 ÷ 0.3). The temperature lift or temperature difference between interstage temperature T_{is} and evaporating temperature T_e and (ΔT_{COM} = $T_{is}-T_e$) influences the mechanical (compressor) coefficient of performance. High values of mechanical COPs can be achieved (24.8 ÷ 6.9) for ΔT_{COM} = 10 ÷ 30 K. Optimization of the interstage temperature T_{is} can be performed using technical (thermal) and economical criteria.

Nomenclature

a	local speed of sound
COP	coefficient of performance
c	velocity (m s^{-1})
E	energy (J kg^{-1})
h	enthalpy (J kg^{-1})
M	mass flow rate (kg s^{-1})
m	specific mass flow rate
P	compressor power consumption (W)
p	pressure (bar; Pa)
Q	heat capacity; cooling capacity (W)
s	entropy (J kg^{-1} K^{-1})
T	temperature (K, ℃)
x	vapor quality

Greek letters

ΔT	temperature difference (K)
δ	kinetic energy losses
λ	Mach number
η	efficiency
ω	entrainment ratio

Subscripts

c	condensation
com	compressor
cr	critical
d	diffuser
din	dynamic
e	evaporation
g	generation
is	interstage
mc	mixing chamber
mech	mechanical
pr	primary
sec	secondary
th	thermal
v	air
w	water

Abbrevations

C	compressor
Co	condenser
COP	coefficient of performance
E	evaporator
E_c	economizer
E_{jv}	vapor ejector
G	mechanical
LTHS	low temperature heat source
RV	regulation valve

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