ENERGETSKA EFIKASNOST KOMBINOVANIH KOMPRESORSKO-EJEKTORSKIH RASHLADNIH SISTEMA

ENERGY EFFICIENCY OF COMBINED COMPRESSOR-EJECTOR REFRIGERATION SYSTEMS

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U ovom radu istražuju se radne karakteriskike, performansi i energetska efikasnost kombinovanih kompresorsko-ejektorskih rashladnih sistema. Prikazan je pregled istraživanja ciklusa hlađenja zasnovanog na kombinaciji ejektorskog ciklusa, koji koristi toplotu niske temperature (solarna energija, geotermalna energija, otpadna toplota), sa ciklusom mehaničke kompresije. U ovom kombinovanom ciklusu hlađenja primenjuje se dvostepena kompresija: prva mehanička kompresija i druga ejektorska kompresija. Između stepena instaliran je ekonomajzer.

Predstavljen je proračunski model za procenu radnih karakteristika i performansi kombinovanih kompresorsko-ejektorskih rashladnih sistema. Korišćeni su tradicionalni postupci izračunavanja realnih ciklusa sa mehaničkom kompresijom i dobijene su radne karakteristike i performansi. Karakteristike ejektorskog rashladnog ciklusa mnogo zavise od performansi ejektora. Dat je postupak proračuna i optimizacije ejektorskog rashladnog ciklusa i optimizacije elemenata strujnog prostora ejektora. Složeni procesi transoničnog strujanja u rashladnom R245fa ejektoru analizirani su korišćenjem CFD simulacionog modela. Uprkos jednostavnoj geometriji ejektora, strujni procesi su veoma složeni i trebaju dobiti odgovarajuću pažnju tokom postupka projektovanja.

Temperature kondenzacije i isparivanja ili temperaturni lift $\Delta T=(T_c-T_e)$, imaju snažan uticaj na ciklus hlađenja ejektora COP. Za $\Delta T=(15–20)$ K, COP može dostići vrednosti iznad 1.2; za $\Delta T=(35–40)$ K COP može biti niži od 0.1. Visoki COP (termički COP_{th} i mehanički COP_{mech}) mogu se dobiti kombinovanim kompresorsko-ejektorskim sistemom hlađenja. Ovaj rashladni sistem pogodan je za primenu u klimatizacionim sistemima: voda za hlađenje, temperatura isparavanja $T_e=5$°C; skladiranje hladnoće - proizvodnja leda, temperatura isparavanja $T_e=-5$°C; temperatura kondenzacije $T_c=30–40$°C; uslovi rada toplotne pumpe, temperatura kondenzacije $T_c=45–50$°C. Temperatura generiranja $T_g=70–120$°C, u zavisnosti od niskotemperaturne toplote. Održavanjem preporuke o temperaturnom liftu ejektora od $\Delta T=15$-20-(25) K, visoki COP (termički COP_{th} i mehanički COPmech) mogu se dobiti kombinovanim kompresorsko-ejektorskim sistemom hlađenja: COP_{th} od (0.6–0.8) do (1.2–1.7); COP_{mech} od 8.0 do 21.0. Kombinovani kompresorsko-ejektorski rashladni sistem, kao optimalan rashladni sistem, pogodan je za korišćenje niskotemperaturne toplote i konkurentan je apsorpcionim rashladnim sistemima.

Ključne reči: *kombinovani kompresorsko-ejektorski sistem; hlađenje; COP; CFD; niskotemperaturna toplota*

Performance characteristics and energy efficiency of combined compressor-ejector refrigeration systems are investigated. An overview of the recent investigations of a refrigeration cycle based on the combination of an ejector cycle, which utilizes low temperature heat (solar energy, geothermal energy, waste heat), with a vapor mechanical compression cycle, is presented. Two-stage compression is applied in this combined refrigeration cycle: first mechanical compression and second vapor ejector compression. An economizer is installed between the stages.

A calculating model for estimation of the performance characteristics of combined compressor-ejector refrigeration systems is presented. Traditional calculating procedures for real mechanical compressor cycles have been used and performance characteristics are obtained. The performance characteristics of the ejector refrigeration cycle strongly depend on the performance characteristics

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of the ejector. A calculating and optimizing procedure for estimation of the performance characteristics of the ejector cycle and optimization of the ejector flow field elements is given. The complex transonic flow processes in a refrigeration R245fa ejector are analyzed using CFD simulation model. Despite the ejector simple geometry, the flow processes are very complex, and they ought to get proper attention during the design procedures.

The condenser and evaporator temperatures or temperature lift $\Delta T=(T_c–T_e)$, have a strong influence on ejector refrigeration cycle COP. For $\Delta T=(15–20)$ K the COP can reach values above 1.2; for $\Delta T=(35–40)$ K the COP can be lower than 0.1. High COP (thermal COP_{th} and mechanical COP_{mech}) can be obtained with combined compressor-ejector refrigeration system. The combined compressor–ejector refrigeration system is suitable for air conditioning application: chilling water, evaporating temperature T_e=5 °C; cold storage – ice production, evaporating temperature T_e=–5 °C; condensing temperature T_c=30–40 °C; heat pump operating conditions, condensing temperature T_c=45–50 °C. The generating temperature Tg=70–120 °C, depending on low temperature heat. Maintaining the recommendation for ejector temperature lift of ΔT=15-20-(25) K, high COP (thermal COP_{th} and mechanical COP_{mech}) can be obtained with combined compressor-ejector refrigeration system: COP_{th} from (0.6–0.8) up to (1.2–1.7); COP_{mech} from 8.0 up to 21.0. The combined compressor-ejector refrigeration system, as an optimal refrigeration system, is suitable for utilization of low temperature heat and competitive with absorption refrigeration systems.

***Key words:** combined compressor-ejector system; refrigeration; COP; CFD; low temperature heat*

1 Introduction

The subject of this paper are the thermal and performance characteristics of combined compressor-ejector refrigeration systems. The refrigeration cycle in these systems is based on an ejector cycle, which utilizes low temperature heat (solar energy, geothermal energy, waste heat), enhanced with a vapor mechanical compression cycle. Utilization of sources of low grade or waste heat is beneficial from environmental and economic points of view [1].

Performance characteristics of simple and combined ejector refrigeration systems strongly depend on performance characteristics of the ejectors. A wide range of applications of single-phase and two-phase ejectors is systematized in Chapter 3 in [2], where numerous recently published investigations are listed. The development history and progress in ejector refrigeration technologies are presented in the review articles (Chunnanond and Aphornratana, 2004 [3]; Elbel and Hrnjak, 2008a [4]; Abdulateef et al., 2009 [5]; Sumery et al., 2012 [6]; Bravo Gonzales et al., 2012 [7]; Sarkar, 2012 [8]; Chen et al., 2013 [9]). Chunnanond and Aphornratana (2004) provide a literature review of ejectors and their applications in refrigeration, where background and theory of ejectors and jet refrigeration cycles, performance characteristics, working fluids and improvement of jet refrigerators, as well as other applications of the ejectors in hybrid ejector-compressor and ejector-absorption refrigeration systems are given. Application of refrigeration ejectors, design procedure and CFD modeling is presented by Grazzini et al. (2017) [10].

A review of solar-driven ejector refrigeration systems and the developmental history and progress in ejector refrigeration systems are reported and categorized by Abdulateef et al. (2009) [5]. An overview of historical and present developments of ejector refrigeration systems is given by Elbel and Hrnjak (2008a) [4] and Elbel (2011) [11]. The state-of-the-art of simple and hybrid jet compression refrigeration systems and working fluid influence is presented by Bravo Gonzales et al. (2012) [7]. Recent developments in ejector refrigeration technologies are given by Chen et al. [9].

2 Performance characteristics of combined compressor – ejector refrigeration systems

A compression enhanced ejector system with utilization of low-temperature heat or waste heat is suggested by Sokolov and Hershgal [1] as a mechanically efficient way for improvement of the simple vapor ejector refrigeration cycle.

2.1 Simple vapor ejector refrigeration cycle

A scheme of a simple ejector refrigeration system and *p–h* diagrams of the cycle processes for wet and dry expansion, depending on refrigerant thermodynamic properties, are given in Fig. 1. Low-temperature heat (solar energy, geothermal energy, waste heat) can be used in the ejector refrigeration cycle, which is an advantage of ejector systems. In the generator (G) the heat Q_g of the low-temperature heat source (LTHS) is transformed to the refrigerant, which evaporates and with high energy potential (generating pressure pg and temperature T_g) exit the generator. This potential is used for realization of the refrigeration cycle. Thermal storage of generating heat (TC) and cooling storage (CS) of the refrigeration effect are useful options, especially in solar energy systems, to provide steady-state design operating conditions of the ejector and of the refrigeration system.

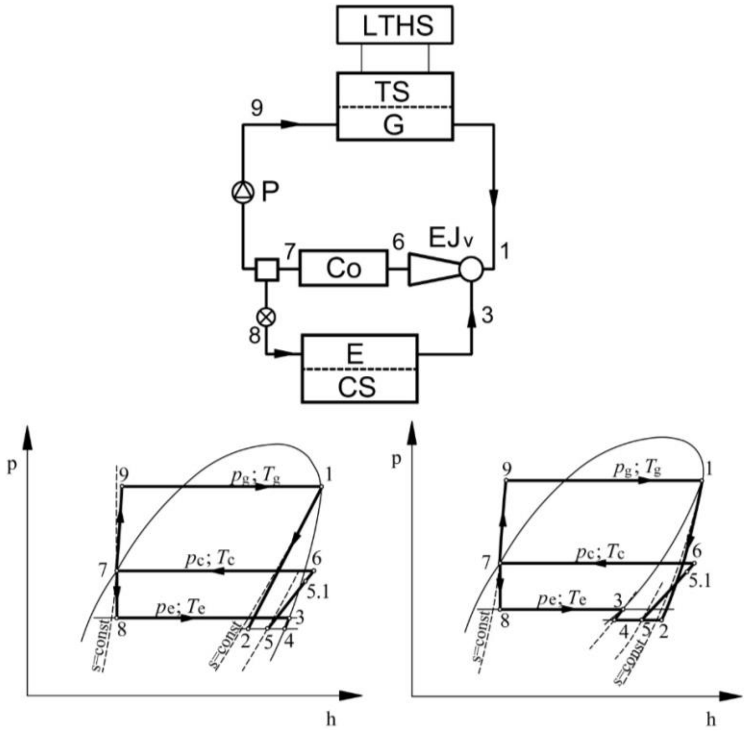


Figure 1 Simple ejector refrigeration machine and p-h diagram of the cycle for wet and dry expansion

2.2 Combined compressor – ejector refrigeration systems

It is demonstrated that a combination of mechanical and thermal energies may provide a wide range of design alternatives, which should yield a competitive refrigeration system. A refrigeration cycle based on the combination of an ejector cycle with a vapor compression cycle is described by Sun (1998) [12]. This integration maximizes the performance of the conventional ejector cycles and provides high COP for refrigeration. The analyses show that the cycle proposed by the author has a significant increase in system performance over the conventional systems and its COP values are competitive with the absorption machines. Suitability of coupling a solar-powered ejector cycle with a vapor compression refrigerating machine is analyzed by Chesi et al. (2012, 2013) [13,14]. The solar powered ejector machine is used to increase the efficiency of a traditional vapor compressor refrigeration machine. A study of an ejector-vapor compression hybrid air-conditioning system using solar energy is presented by Dang et al. (2012) [15]. A review of ejector enhanced vapor compression refrigeration and heat pump systems is given by Sarkar (2012) [8].

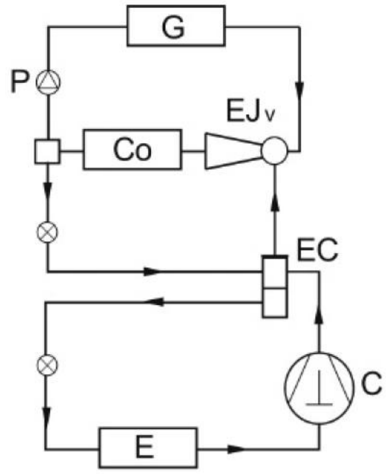


Figure 2 Combined compressor – ejector refrigeration system

A scheme of a combined compressor – ejector refrigeration system is given in Figure 2.

An overview of the recent investigations of a refrigeration cycle based on the combination of an ejector cycle which utilizes low temperature heat (solar energy, geothermal energy and waste heat) with a vapor mechanical compression cycle is presented in Section 6.1 in [2]. Two-stage compression is applied in this combined refrigeration cycle: the first one mechanical compression (C) and the second one vapor ejector compression (EJ_v). An economizer (EC) is installed between the stages. Two throttling valves are installed: the first one between the condenser (Co) and economizer (EC) and the second one between the economizer (EC) and evaporator (E). It is suitable for COP increase. Low temperature heat is utilized for production of generating motive vapor in the generator (G).

The combined compressor – ejector refrigeration system is suitable for air conditioning application: chilling water, evaporating temperature T_e = 5 °C; cold storage – ice production, evaporating temperature T_e = –5 °C; condensing temperature T_c = 30 – 40 °C; heat pump operating conditions, condensing temperature T_c = 45 – 50 °C. The generating temperature T_g = 70 – 120 °C depending on low temperature heat source. The optimization of the inter stage pressure is crucial to achieve high thermal $COP_{th} = Q_e/Q_g$ and high mechanical $COP_{mech} = Q_e/P_{mech}$. Compressor power consumption depends on inter stage pressure and on performance characteristics (efficiency of the compressor). The optimization of the inter stage pressure can be conducted using thermo-economics criteria according to the capital investments and energy costs (electrical energy costs and low temperature energy costs). Numerical experiments have been conducted for various temperature conditions (given previously), with R245fa as a suitable refrigerant for air conditioning operating conditions, using calculating procedure for the ejector flow field optimization and performance characteristics presented in Chapter 3 [2] and traditional calculating procedures for real mechanical compressor refrigerating cycles. Maintaining the recommendations for ejector temperature lift of ΔT = 15–20–(25) K, high COP (thermal $COP_{th} = Q_e/Q_g$ and mechanical $COP_{mech} = Q_e/P_{mech}$) can be obtained with combined compressor-ejector refrigeration system. For the previously given operating temperature operating conditions (T_e, T_c, T_g) thermal COP_{th} from (0.6 – 0.8) up to (1.2 – 1.7) and mechanical COP_{mech} from 8.0 up to 21 can be obtained. Lower values of COPs correspond for lower evaporating temperatures (T_e = –5 °C) and higher condensing temperatures (T_c = 45 – 50 °C). Higher values of COPs correspond to higher evaporating temperatures (T_e = 5 °C) and lower condensing temperatures (T_c = 30 – 35 °C). Lower values of COP_{th} correspond for higher values of recommended ejector temperature lift of (ΔT = 20–25 K) and lower generating temperature (T_g = 70 °C). Higher values of COP_{th} correspond for lower values of recommended ejector temperature lift of (ΔT = 15 K) and higher generating temperature (T_g = 120 °C).

The combined compressor-ejector refrigeration system, as an optimal refrigeration system, is suitable for utilization of low temperature heat and competitive with absorption refrigeration systems.

3 Main parts of the ejector, operating characteristic, optimal geometric parameters, and performance characteristics

The main parts of the ejector are the primary nozzle, the secondary nozzle, the mixing chamber and the diffuser. A diagram of the flow field of an ejector is given in Fig. 3. Thermal h - s diagram of the processes in the ejector is given in Fig. 4. The diagram refers to refrigerant with wet expansion.

In the ejector primary nozzle (1) motive fluid accelerates and expands (1-2) (Fig. 4) from the high pressure p1 to the pressure p_2 which is lower than secondary flow suction pressure. The flow at the outlet of the primary nozzle is usually supersonic, and the nozzle profile is convergent-divergent.

In the narrowest flow section of the nozzle $A_{cr}=d^2_{cr}\pi / 4$ 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 narrowest flow cross-section of the nozzle is called the critical cross-section.

For steam ejectors (with dry or wet expansion) the critical cross-sectional velocity equal to the local speed of sound is:

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

Using isotropic expansion from p_1 to p_{cr} the specific volume vcr 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:

$$c_2=\Psi_{pr} c_{2s}=[2(h_1-h_2)]^{1/2}=(2\Delta h_a \eta_{pr})^{1/2}=(2\Delta h_s \eta_{pr})^{1/2}$$

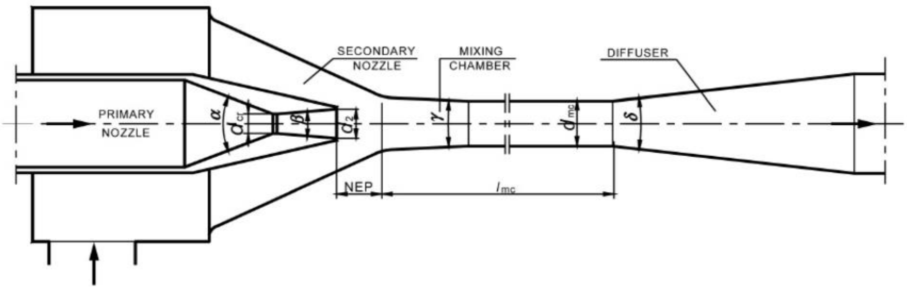


Figure 3 Geometrical parameters of the ejector flow field

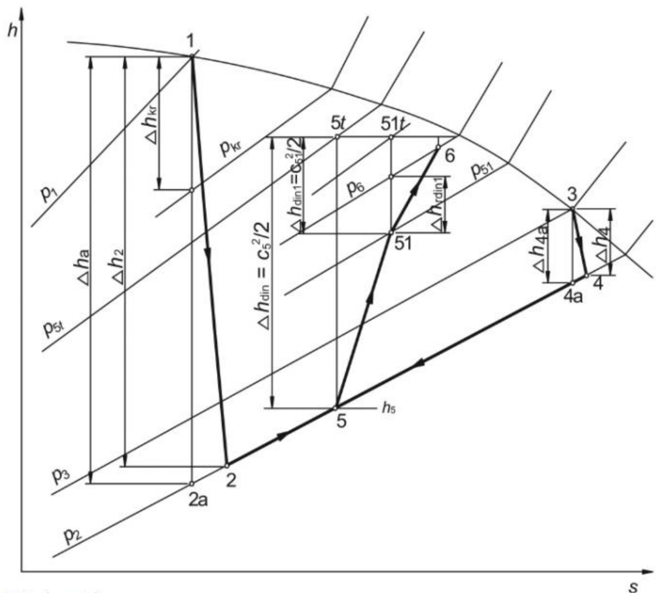


Figure 4 h - s diagram of the processes in the ejector (refers to refrigerant with wet expansion)

The primary nozzle should be profiled to a pressure close to p_2. The primary flow further expands in an environment with variable pressure, interacting with the secondary flow, and reaches velocity c_2. The cross section of the primary flow having pressure p_2 is: $A_2 = M_{pr}\, v_2 / c_2$

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 and entrains the secondary flow into the mixing chamber (3). The secondary flow comes through the secondary nozzle (2) where it expands (3-4). The secondary nozzle is formed by the outside profile of the primary nozzle and inside profile of the secondary nozzle, as well as by interaction with additionally expanded primary flow. The shear layer between the primary and secondary fluids flowing with a large velocity difference leads to the acceleration of the secondary flow. The mixing process after the primary nozzle exit plane is rather complex due to the interaction between the two fluid streams. 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 hypothetical critical (narrowest) cross-section of the secondary nozzle (A_{sec}) is equal to the difference between the cross-section of the mixing chamber (A_{5mch} = A5) and the cross-section of the primary flow, $A_{sec} = A_4 = A_5 – A_2$

The speed of the secondary flow is:

$$c_4 = \Psi_{sec} \cdot c_{4s} = [2 \cdot (h_3 - h_4)]^{1/2} = (2 \cdot \Delta h_{4s} \cdot \eta_{sec})^{1/2}$$

The secondary mass flow is, $M_{sec} = A_4 c_4\, \rho_4$

The calculation of all thermodynamic quantities (temperature, enthalpy, entropy, density) for states 2 and 4 can be performed using the equations for polytropic expansion (1–2) and (3–4), data on the thermodynamic properties of refrigerant (equations, tables, diagrams, application software), or using the conditional mean isentropic exponent (κ) method.

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

In the mixing chamber of the ejector, a complex process of momentum transfer and quantity of movement between the primary and secondary flow (2–5; 4–5) occurs. Using the law of conservation of momentum and the momentum equation for the mixing chamber, assuming that the process takes place at a constant pressure in a mixing chamber with a constant cross section, $A_2+A_4=A_5$, $p_2=p_4=p_5$, and if the flow friction forces are covered by the mechanical efficiency coefficient of the mixing chamber (η_{mc}=0.95–0.98), the equation for the velocity of the combined flow is obtained:

$$c_5=\eta_{mc}\left(c_2 m_{pr}+c_4 m_{sec}\right); \quad m_{pr}=\frac{M_{pr}}{M_{pr}+M_{sec}}; \quad m_{sec}=\frac{M_{sec}}{M_{pr}+M_{sec}}$$

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

$$\delta_e=\frac{\Delta E}{E_1}=m_{sec}\frac{\left(c_2^2-c_4^2\right)}{c_2^2}=m_{sec}\frac{\left(c_2^2-c_4^2\right)}{c_2^2}$$

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

$$h_5 = m_{pr}h_2 + m_{sec}h_4 + m_{pr}\frac{c_2^2}{2} + m_{sec}\frac{c_4^2}{2} - \frac{c_5^2}{2}$$

With the pressure $p_2=p_4=p_5$ and with the enthalpy h_5, state 5 is defined, the values of the other thermodynamic quantities can be determined. The dynamic component $\Delta h_{din} = c_5^2$ and the total pressure are defined by the velocity c_5.

Fluid compression occurs when the combined flow flows through the mixing chamber and the diffuser. The kinetic energy $\Delta h_{din}=c_5^2/2$ is transformed into an increase in enthalpy, expressed by an increase in pressure.

The combined flow, after the process of momentum transfer and transfer of quantity of motion in the mixing chamber, is usually supersonic. Decrease of speed of the supersonic flow, i.e. transition from supersonic flow to subsonic occurs in a direct shock wave. Shock wave is a process in which the speed drops sharply from supersonic to subsonic, and the pressure rises sharply (5 - 5.1).

Mach number of the supersonic flow, upstream of the shock wave is, $\lambda_1 = c_5 / a_{cr} > 1$. Mach number downstream of the shock wave is, $\lambda_2 = c_{51} / a_{cr} < 1$. Across the shock wave is, $\lambda_1 \lambda_2 = 1$.This physical law shows that when decelerating a supersonic flow through a shock wave, the higher the Mach number of the supersonic flow before the shock wave, the lower the Mach number of the supersonic flow after the shock wave, and the stronger the shock wave.

Using the average value of the isentropic coefficient (κ) and using the theory of gas dynamics the pressure rise in the shock wave can be calculated:

$$\frac{p_{51}}{p_5} = \frac{\lambda_1^2 - (\kappa - 1)/(\kappa + 1)}{1 - (\kappa - 1)\lambda_1^2/(\kappa + 1)}$$

The previous equations define the pressure p_{51} and the velocity after the shock wave c_{51}. The dynamic component $\Delta h_{din1} = c_{51}^2 / 2$ and the total enthalpy h_{din5t} for state 51 are defined by the velocity c_{51}. According to the entropy s_{51}, 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 partially realized. However, the shock wave is a thermodynamically irreversible process, with entropy rise, and it is the second main source of thermodynamic irreversibility and exergy decrement 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. Both of them are physics phenomena and cannot be avoided by any design effort. Additional compression is realized in the subsonic diffuser.

The efficiency of the subsonic diffuser defined as a ratio between isentropic compression work Δh_{rdin1} from point 51 (inlet state at the subsonic diffuser) up to ejector exit pressure pe (work resulting in pressure rise) and dynamic pressure at the subsonic diffuser inlet $\Delta h_{din1} = c_{51}^2/2$.

$$\eta_d = \Delta h_{rdin1} / \Delta h_{din1} = \Delta h_{rdin1} / (c_{51}^2 / 2)$$

According to a wide range of publications about subsonic diffuser hydrauliclosses, 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 $p_6 = p_e$ can be determined and the thermodynamic values for state 6 at the ejector outlet can be obtained.

According to the analysis given in Section 3.3 [2] it is suggested that the optimal diffuser angle of divergence for steam and vapor ejectors is 5° to 7°.

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. Lower entrainment ratios ω and COP (about 15 - 20% depending on operating conditions) are obtained for the lowest values of the coefficients previously given. With further design research and improvement of production quality, using the results of theoretical (CFD - simulations) and experimental research, the efficiency of the ejectors can be improved, achieving higher values of the efficiency coefficients of the ejector elements. However, the dissipation of ejector efficiency caused by the two main sources of hydraulic losses, thermodynamic irreversibility and exergetic losses (the process of momentum transfer and transfer of quantity of motion), which depend on the thermodynamic properties of the refrigerant and the flow conditions, cannot be improved with any design or research activities.

4 CFD analysis of the ejector flow field using R245fa as a refrigerant

4.1 Dimensions of the ejector flow field

The dimensions of the ejector are calculated using the calculation procedure (model) for optimization and performance of the ejectors given in section 3 for the following design conditions: cooling capacity: 10 kW; refrigerant: R245fa; evaporation temperature: - 3 ℃; evaporation pressure: 0.45 bar; condensation temperature: 18.5 ℃; condensation pressure: 1.15 bar; generation temperature: 90 ℃; generating pressure: 10.016 bar.

Using the calculation procedure, the following geometric parameters are obtained:

Primary nozzle: inlet diameter 20 mm; critical diameter 4.5 mm; outlet diameter 9.0 mm; convergent angle 45 ° / 30 °; divergent angle 12 °; length 45 mm.

Secondary nozzle and mixing chamber: convergent angle 30 ° / 12 °; diameter of the mixing chamber 22 mm; length of the mixing chamber 210 mm.

Diffuser: inlet diameter 22 mm; outlet diameter 36 mm; diffuser angle 4.5 °; length 180 mm.

Connections: Primary flow inlet DN 20 PN 16; Secondary flow inlet DN 32 PN 16; Ejector outlet DN 32 PN 16.

Ejector performance: Condensation temperature <18.5 ℃ M_{sec} / M_{pr} = 1.0, critical operating conditions; Condensation temperature> 18.5 ° C, M_{sec} / M_{pr} <1.0, subcritical operating conditions; Condensation temperature around 38 ° C, M_{sec} / M_{pr} = 0, reverse flow

4.2 Computer design of the ejector

The ejector is designed using the SolidWorks computer modeling software package. Figure 5 and Figure 6 show the spatial appearance and the cross section view of the ejector.

Figures 5 Spatial appearance of the ejector

Figures 6 Cross section view of the ejector flow field

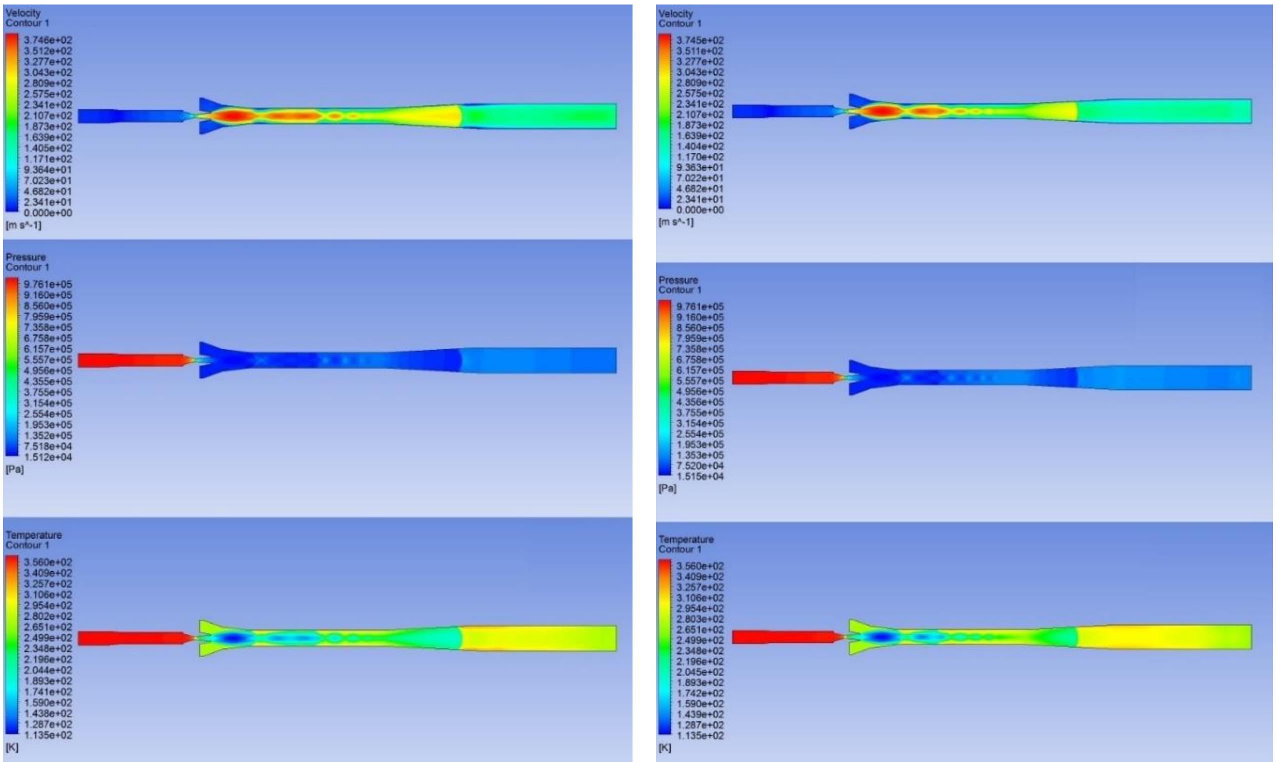
4.3 CFD analysis of the ejector flow field

Numerical experiments are performed using the CFD flow model in the ejectors for different operating modes, with R245fa as the working fluid (refrigerant). Results are obtained for the speed, pressure and temperature in the ejector flow field (Figures 7,8).

Figure 7a) shows the results of the CFD model for calculating the R245fa refrigeration ejector flow field for the design conditions (design mode): generating pressure 10.061 bar / generating temperature 90 °C, evaporation pressure 0.45 bar / evaporation temperature -3 °C, condensation pressure 1.15 bar / temperature at condensation 18.5 °C.

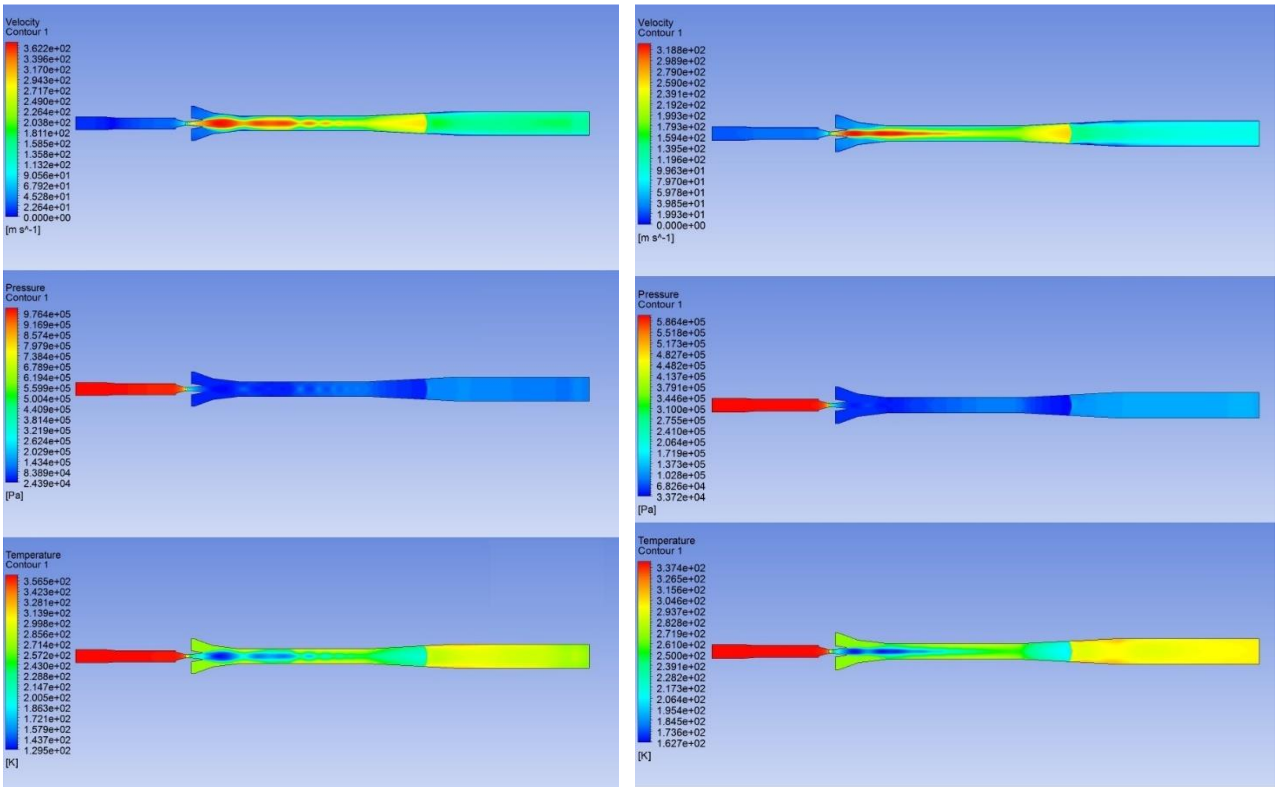
Critical flow parameters (critical speed, critical pressure, critical temperature) are obtained in the narrowest cross section of the primary nozzle. The critical speed is equal to the local speed of sound (M = 1). The value of the critical speed in the critical (narrowest) section of the primary nozzle is about 130 m / s and is significantly lower than the critical air speed (about 300 m / s). In the divergent part of the nozzle, additional expansion occurs during which a strong supersonic flow is achieved (M> 1). The zone of momentum transfer from the primary to the secondary gas flow is characterized by a series of oblique shock waves, especially clearly expressed in the images of the flow results. Strong supersonic flow is achieved in the mixing chamber. The transition from supersonic to subsonic flow takes place through a strong normal shock wave, especially clearly expressed in the images. When speed drops from supersonic to subsonic gas pressure and temperature rise sharply. Compression takes place mainly in the normal shock wave. An additional slight increase in pressure occurs with further deceleration of the gas flow in the subsonic (divergent) ejector.

In Figure 7b) the results for the flow calculation at elevated condensation pressure of 1.22 bar and correspondingly elevated condensation temperature of 20 °C are given. There is a slight decrease in the speed of the secondary flow. In Figure 8a) the results for the flow calculation for elevated evaporation pressure 0.53 bar / evaporation temperature 0 °C and under the same other operating conditions as in the design mode (Figure 7a)) are given. In Figure 8b) the results for the flow calculation for lowered generating pressure 0.61 bar / generating temperature 70 °C and under the same other operating conditions as in the design mode are given.



a) b)

Figure 7 R245fa ejector flow field for design mode (a) and for elevated condensation pressure (b)



a) b)

Figure 8 R245fa ejector flow field for elevated evaporation pressure a) and for lowered generating pressure b)

5 Conclusions

Combined compressor-ejector refrigeration systems are presented and their thermal and performance characteristics are estimated. The simple vapor ejector refrigeration cycle utilizes low temperature heat (solar energy, geothermal energy, waste heat) and in the combined systems it is enhanced with a vapor mechanical compression cycle. Utilization of sources of low grade or waste heat is beneficial from environmental and economic points of view.

The performance characteristics of the ejector refrigeration cycle strongly depend on the performance characteristics of the ejector. A calculating and optimizing procedure for estimation of the performance characteristics of the ejector cycle and optimization of the ejector flow field elements is given. The complex transonic flow processes in a refrigeration R245fa ejector are analyzed using CFD simulation model. Despite the ejector simple geometry, the flow processes are very complex, and they ought to get proper attention during the design procedures.

The combined compressor-ejector refrigeration systems are technically and economically viable and competitive in comparison with conventional refrigeration systems.

6 References

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