

STRUJNE KARAKTERISTIKE, MODELIRANJE I SIMULACIJA JEDNOFAZNIH I DVOFAZNIH EJEKTORA

FLOW CHARACTERISTICS, MODELING AND SIMULATION OF SINGLE-PHASE AND TWO-PHASE EJECTORS

Vasko ŠAREVSKI,

Faculty of Mechanical Engineering, University “Sv. Kiril i Metodij”, Skopje, R. Macedonia
Karpos II bb 1000 Skopje, e-mail: vasko@mf.edu.mk,

Milan ŠAREVSKI,

Faculty of Mechanical Engineering, University “Sv. Kiril i Metodij”, Skopje, R. Macedonia
Karpos II bb 1000 Skopje, e-mail: milan@mf.edu.mk,

ABSTRACT

A review on implementation of the ejector thermo compression in refrigeration systems and heat pumps is presented in this paper. Thermal, flow and performance characteristics of the refrigeration single-phase and two-phase ejectors are analyzed. A calculating procedure for estimation of the main parameters and performance characteristics of the ejectors is presented. The two main sources of thermodynamic irreversibility: the process of momentum transfer in the mixing section and shock waves, or dispersed shock waves, or pseudo-shock waves in the fluid flow field are discussed. The shock waves are feature for gas and dry vapor fluid flow; dispersed shock waves are feature for dominantly vapor two-phase fluid flow; pseudo-shock waves are feature for dominantly liquid two-phase fluid flow. With appropriate choice of refrigerant and with optimal design of the ejector flow field elements the ejector systems can be successfully applied in various refrigeration / heat pump systems, and in combined (hybrid) and polygeneration thermal systems for utilization of low temperature heat, geothermal energy, solar energy and waste heat.

Key words: *ejector; thermocompression; refrigeration; optimization; flow field*

I. Introduction

The energy efficiency improvement strategy and the concept of cleaner production have led to research and development of new polygeneration thermal systems and new combined (hybrid) thermal systems with utilization of low temperature heat, solar energy, geothermal energy and waste heat. In this context the thermal systems with ejector thermo compression recently have attracted many research activities and successful application in various thermal systems: steam jet and various refrigeration systems working with different refrigerants; two-phase ejectors in compressor refrigeration systems as devices for reduction of throttling losses; two-phase ejectors in compressor refrigeration systems as devices for second step compression; polygeneration thermal systems and new combined (hybrid) thermal systems: combined ejector-compression systems, ejector-absorption systems, ejector-adsorption refrigeration systems.

The base of theory of gas and steam ejectors is given in the fundamental publications (for example: Abramovic, [1]). The theory of ejector steam jet refrigeration systems is also given in the base publications (Cerepnalkovski, [2]). The fluid flow analysis of the ejectors and considerations presented in this paper are based on the fundamental principles and publications (Abramovic, [1]; Loicianski, [3]; White, [4]; etc), as well as on the numerous computational and experimental investigations published in recent years.

Chunnanond and Aphornratana [5] provide a literature review on ejectors and their applications in refrigeration, where background and theory of ejectors and jet refrigeration cycle, 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. A review on solar-driven ejector refrigeration systems, the development history and progress in ejector refrigeration systems are reported and categorized by Abdulateef *et al* [6]. An overview of historical and present developments of ejector refrigeration systems is given by Elbel and Hrnjak [7] and Elbel [8]. State-of-the-art of simple and hybrid jet com-

pression refrigeration systems and the working fluid influence is presented by Bravo Gonzales *et al* [9]. Recent development in ejector refrigeration technologies is given by Chen *et al* [10], where numerous studies are reported and categorized in various topics including, refrigerant selection, mathematical modeling and simulations, geometric optimization, operating conditions optimization and specific ejector refrigeration systems. A review on implementation of two-phase ejectors in compressor refrigeration systems and heat pumps for enhancement of their performance characteristics is presented by Sarkar [11]. Among the application of the vapor ejectors in the ejector refrigeration systems, the ejectors have been successfully applied in thermal industrial concentrating and desalination plants (Šarevski and Šarevski, [12], [13], [14], [15], [16]), in industrial steam-condense systems (Šarevski and Šarevski, [15]), in industrial processing vacuum systems (Šarevski and Šarevski, [16], [17]) etc.

Performance characteristics of the ejector refrigeration system and of other ejector thermal systems strongly depend on performance characteristics of the ejector. Despite apparent simplicity of the ejector, the phenomena affecting ejector performances are rather complex, and many studies, ranging from one-dimensional models to CFD simulations, have been carried out by many authors. A review of various steady and dynamic models, single-phase flow and two-phase flow models of the ejectors is given by He, *et al* [18]. For the purposes which are focused on global plant performance rather than having the pretension to accurately simulate ejector behaviour and flow features in the ejector flow field elements, simplified 1-D models can be employed (Huang *et al* [19]; Roman and Hernandez [20]; Garcia del Valle *et al* [21], etc). Calculation and analysis of sound velocity in vapor-liquid two-phase flow, as well as theoretical and experimental investigations of transonic flow phenomena in two-phase ejectors are given by Berana *et al* [22]; Berana and Nakagawa [23]; Wang and Zhang [24]; Karwacki *et al* [25]; Banasiak and Hafner [26]. Computational fluid dynamics (CFD) is a valuable tool to analyze and improve ejector performance. Several validated CFD models which use commercial CFD software packages can be found in the literature (Bartosiewicz *et al* [27], Hemidi *et al* [28], Zhu and Li [29], Scott *et al* [30], Ruangtrakoon *et al* [31], Varga *et al* [32], Colarossi *et al* [33], Yazdany *et al* [34]). The good prediction of the entrainment ratio with CFD simulations, even over a wide range of operating conditions, do not necessarily mean a good prediction of the local flow features in ejector flow field. Different results of the flow have been obtained using different CFD (κ - ϵ ; κ - ω -*sst*) turbulence models (Hemidi *et al*, [28]; Zhu and Jiang, [35]).

The purpose of this paper is to give a review on recent investigations of single-phase and two-phase ejectors to describe a modeling method and calculation procedure for estimation of the ejector characteristics and according to systematized results of investigations to present overview and recommendations for optimization of ejector flow field elements.

II. Main Characteristics and Parameters of Vapor and Two-Phase Ejectors

In the ejector primary nozzle (1) motive fluid accelerates and expands (1-2) (Fig. 3.1) 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 usually supersonic, and the nozzle profile is convergent-divergent. Exiting the primary nozzle the fluid additionally expands entering into the mixing chamber, where complex flow phenomena appears 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 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 gets critical flow (choking flow), than these operating conditions of the ejectors are often referred to “double choking” operation.

A supersonic flow from a nozzle discharging in a variable pressure environment such as the case of an ejector, and interaction with the secondary flow is rather complex flow process, characterised by a series of oblique / normal shock waves. This flow phenomenon is directly connected with optimum location of the primary nozzle exit as an important geometrical parameter with great influence on the ejector entrainment ratio and on the ejector performance characteristics. Among this geometrical parameter, the complex flow process of interaction between the primary and secondary flows is the reason for appropriate optimal shaping of the secondary nozzle as an important factor with great influence on the ejector performance characteristics. The combined flows are mixed flowing within the mixing chamber (2-5, 4-5), where appears a complex process of momentum transfer. The compression of the fluid is achieved as the combine stream flows through the mixing chamber and diffuser (4). The kinetic energy of the combine stream flow is transformed to

enthalpy rise in the diffuser, expressed by rise of the pressure. The combined flow at the mixing chamber often is supersonic. If the velocity of the combined flow is supersonic then a normal shock wave occurs (5-5.1). The shock wave is a process where sudden change in the flow space appears, the velocity suddenly falls from supersonic to subsonic and the pressure rises, which is characteristic of single phase (gas and dry vapor) flow. In two-phase flow this complex process is accompanied by mass transfer from one phase to the other, where pseudo-shock waves, feature for dominantly liquid two-phase flow, or dispersed shock waves, feature for dominantly vapor two-phase flow, occur in the flow field. Additional compression (5.1-6) is realized in the subsonic diffuser (4).

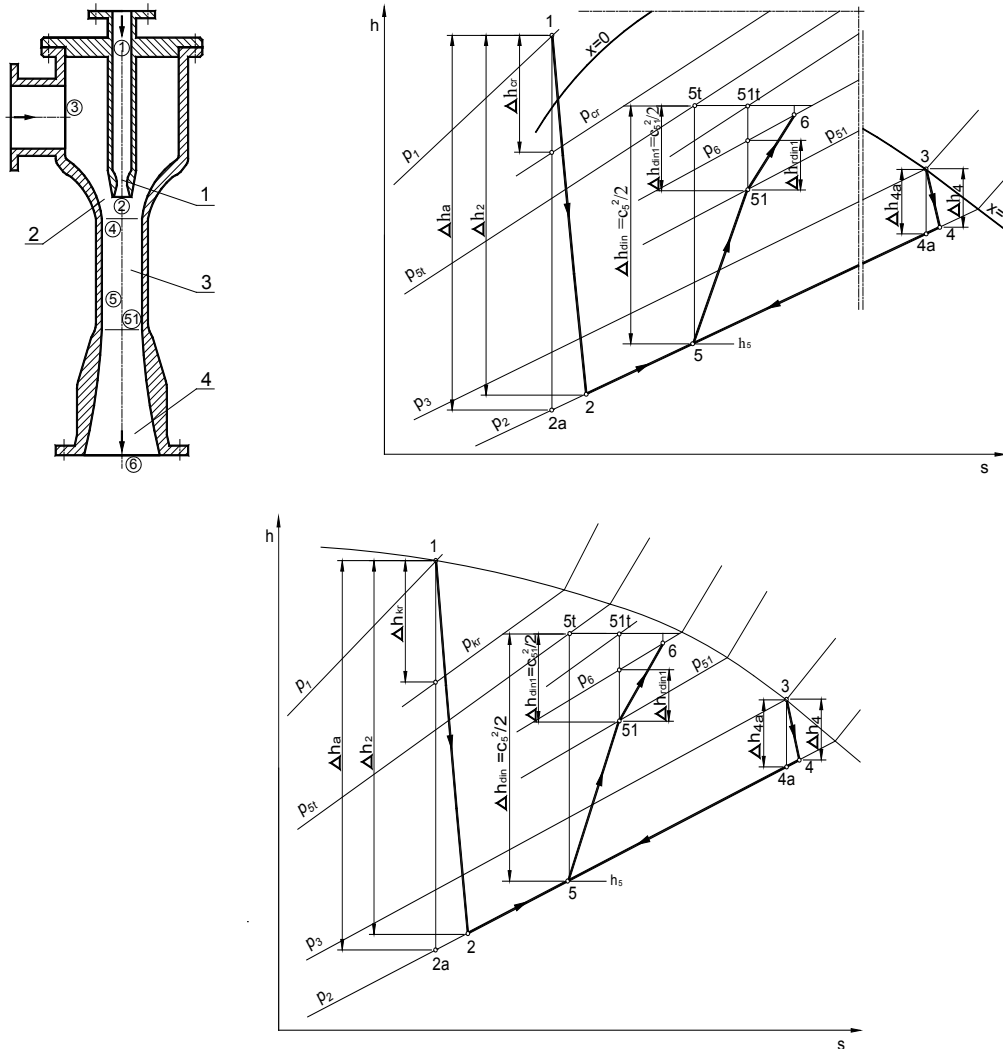


Figure 1 Scheme of an ejector and $h-s$ diagram for two-phase expansion and vapor expansion

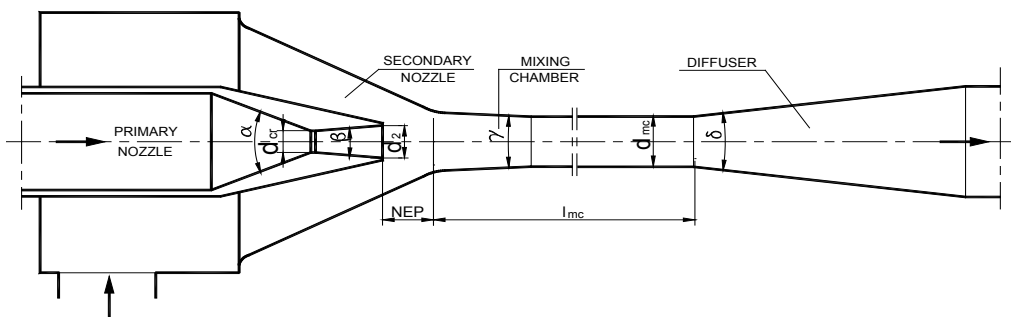


Figure 2 Geometrical parameters of the ejector flow field

Two general design models of ejectors: ejector with constant area mixing chamber and ejector with variable area mixing chamber (ejector with “constant pressure mixing chamber”) can be distinguished (Chunnanond and Aphornratana, 2004). Various design of the ejector flow field elements exist with design characteristics between these two general models. The following key geometrical parameters of the ejector

flow field (Fig. 2) have strong influence on the ejector performance characteristics: primary nozzle converging angle (α); primary nozzle diverging angle (β); profile of the secondary nozzle; primary nozzle exit position (NEP); mixing chamber converging angle (γ); mixing section length and diameter ratio (l_{mc}/d_{mc}); mixing section cross-section and primary nozzle exit cross-section ratio; diffuser diverging angle (δ).

The entrainment ratio of an ejector ($\omega = M_{sec} / M_{pr}$) is defined as ratio between the ejector secondary flow rate (M_{sec}) and primary flow rate (M_{pr}). The entrainment ratio is the most important parameter of an ejector, which depends on the operating conditions of the thermal system: generating pressure and temperature; condensing pressure and temperature; evaporating pressure and temperature; and pressure or temperature lift. The coefficient of performance COP of these systems directly depends on the ejector entrainment ratio.

III. Flow Characteristics, Modeling and Optimization

Performance characteristics of the ejector strongly depend on the thermal system operating conditions, as well as on the degree to which the ejector flow field has been optimally designed. Despite ejector apparent simplicity, the phenomena affecting ejector performances are rather complex, and many studies, ranging from one-dimensional models to CFD simulations, have been carried out by many authors. Computational fluid dynamics (CFD), supported by results of experimental investigations, is a valuable tool to analyze and improve ejector performance. Simulation of the complex flow processes in ejector flow field, connected with supersonic, transonic and subsonic fluid flows, interaction between the primary and secondary flows in conditions of large velocity gradient, shock waves etc, especially if that flow phenomena are connected with two-phase fluid flow, is an extremely complex task.

A review of various steady and dynamic models, single-phase flow and two-phase flow models of the ejectors is given by He, [18], and additionally the simplified empirical and semi-empirical models based on measured data are discussed. For the purposes which are focused on global plant performance rather than having the pretension to accurately simulate ejector behaviour, simplified 1-D models can be employed (Huang *et al* [19], Roman and Hernandez [20], Garcia del Valle *et al* [21], etc).

Numerical and experimental investigations on supersonic ejectors are presented by Bartosiewicz *et al* [27]. The papers given by Hemidi *et al* [28], deal with comparisons between CFD and experiments for a supersonic ejector. The influence of key geometry parameters on the performance of the ejectors with different refrigerants using CFD models calibrated by experimental results is presented by Zhu and Li [29], Ruangtrakoon *et al* [31], Varga *et al* [32], Scott *et al* [30].

The phase change process in two-phase condensing ejectors, which utilize the beneficial thermodynamics of condensation to produce an exit static pressure that can be in excess of either entering static pressure, is driven by both turbulent mixing and interphase heat transfer (Colarossi *et al*, [33]. The semi-empirical model used in conjunction with CFD presented by Colarossi *et al* [33] describes the construction of a multidimensional simulation capability built around an Eulerian pseudo-fluid approach. The numerical model of two-phase supersonic ejectors for work-recovery applications developed by Yazdany *et al* [34] integrates models for real-fluid properties, local mass and energy transfer between the phases, and two-phase sonic velocity in the presence of phase change. A one-dimensional model of the R744 two-phase ejector for expansion work recovery is presented by Banasiak and Hafner [26].

The flow at the exit of the primary nozzle is supersonic, and the nozzle profile is convergent-divergent. Depending on refrigerant thermodynamic properties the vapor expansion can be wet (R718, R134a, etc) or dry (R245fa, R600, etc). The presence of high flow speed in the vapor ejector suggests the existence of metastable conditions. The profile of the primary nozzle is one of the key geometrical parameters with strong influence on the ejector efficiency and performance characteristics. According to Abramovic (1969), values of 30° to 60° for the angle of the converging section (α) and values of 10° to 16° for the angle of the diverging section (β) are most common for gas ejectors. According to ASHRAE (1983), values of 10° to 12° for the angle of the diverging section are most common for steam jet refrigeration, but can range from 8° to 15° (Elbel *et al* [8]). The angle of converging section is 30° and the angle of diverging section is 2° for two phase ejector primary nozzle in the work presented by Banasiak and Hafner [26]. In the work presented by Karwacki *et al*, [25], the nozzle converging section is profiled, and the angle of diverging section is 8° . A 2.3° motive nozzle diverging angle is tested by Lawrence and Elbel [36], and a 4.0° motive nozzle diverging angle resulted in reduced performance compared to the 2.3° diverging angle.

Shock waves in supersonic two-phase flow of CO_2 in converging-diverging nozzles are investigated by Berana *et al* [22]; Berana and Nakagawa [23]. The divergence angles ($\beta/2$) with significant variation of decompression are 0.076° , 0.153° , 0.306° and 0.612° . Pseudo-shock waves (feature for dominantly liquid

two-phase flow) and dispersed shock waves (feature for dominantly vapor two-phase flow) are obtained from their experiment. Both are weaker than equilibrium shock waves and indicated relaxation phenomena.

According to analyses of the previously mentioned publications and other publications given in references, in this work is suggested that motive nozzle converging angle should be between 30° - 40° and motive nozzle diverging angle between 10° - 12° for vapor ejectors, and lower than 2° for two phase ejectors.

Exiting the primary nozzle the fluid additionally expands entering into the mixing chamber, where complex flow phenomena appears between the primary and secondary flow. A supersonic ejector primary flow can be reached even with converging nozzle only (Abramovic, [1]), by additional expansion and interaction with secondary flow. The primary flow draws and entrains the secondary flow into the mixing chamber. 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 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. The secondary fluid gets critical flow (choking flow). These operating conditions of the ejectors are often referred to “double choking” operation. The experiments conducted by Eames *et al* [37] showed that choking of the secondary flow in the mixing chamber of the ejector plays an important role in the ejector performance. Maximum COP ($\text{COP} = Q_e / Q_g$) and maximum entrainment ratio ω ($\omega = m_{sec} / m_{pr}$) is obtained when the ejector operates at its critical flow condition.

Performance enhancement of a transcritical CO_2 air conditioner with a controllable ejector at variable operating conditions and variable compressor frequencies is obtained by Lui *et al* (2012) with experimental investigations of ejectors with various geometries. The COP reached a maximum when the distance between motive nozzle exit and mixing section entrance is three times the mixing section diameter.

According to the experimental and numerical investigation of the optimum two-phase ejector geometry for a small-capacity R744 heat pump performed by Banasiak *et al* [26], the optimum ratio between mixing section length and diameter is between 7 and 10. The optimum mixing section length of a two phase ejector (Lui *et al*, [38]) is nine times the mixing section diameter.

According to analyses of the previously mentioned publications and other publications given in references, in this work is suggested that the optimum distance between motive nozzle exit and mixing section entrance is $(1-3) d_{mc}$. Higher values correspond to higher primary flow stagnation pressures. Higher values of the mixing section converging angle are required to maximize the ejector performance when the primary flow pressure rises. The optimum ratio between mixing section length and diameter is between 7 and 11. Lower values correspond to steam and vapor ejectors. Higher values correspond to two phase ejectors.

The process of the momentum transfer in the mixing chamber is the first main source of thermodynamic irreversibility and exergy decrement in the ejectors.

The combined flow at the mixing chamber outlet often is supersonic. If the velocity of the combined flow is supersonic then a normal shock wave occurs. The shock wave is a process where sudden change in the flow space appears, which is characteristic of single phase (gas and dry vapor) flow. In two-phase flow this complex process is accompanied by mass transfer from one phase to the other, where pseudo-shock waves, feature for dominantly liquid two-phase flow, or dispersed shock waves, feature for dominantly vapor two-phase flow, occur in the flow

In the shock wave partially the compression is realized. However, the shock wave is thermodynamic irreversible process, with entropy rise, and it is the second main source of thermodynamic irreversibility and exergy decrement in the ejectors. The existent of pseudo-shocks in two phase flow is obtained and proved by experiments (Elbel and Hrnjak [7]; Elbel, [8]; Banasiak and Hafner [26], etc) with measurement of the pressure along the mixing chamber wall.

Additional compression is realized in the subsonic diffuser. According to wide range of publications about subsonic diffuser and according to the recent experimental investigations of two-phase ejectors the diffuser angle of divergence is 3° - 5° . The optimal diffuser angle of divergence for and vapor ejectors suggested in this work is 5° - 7° , and for two phase ejectors is 3° - 5° . When the amount of liquid in the mixture is much larger, then expected optimal diffuser angle is lower.

IV. Calculation Procedure of Vapor and Two-Phase Ejectors

The calculated procedure for estimation of the main geometrical parameters, flow characteristics and performance characteristics of vapor and two-phase ejectors is based on the principle laws of the mechanics and physics for steady-state and steady-flow conditions: energy equation for an adiabatic process, momentum equation and continuity equation.

For vapor ejectors with dry expansion (R245fa, R600, for example) the calculations can be conducted using data for thermodynamic properties (equations, tables, diagrams), or applicative software of the superheated refrigerant. For steam and vapor ejectors with wet expansion (R718, R134a, etc) the calculations and the nozzle profiling procedure can be conducted according to thermodynamic properties of wet vapor. Approximately, if the liquid phase can be neglected, the calculations can be conducted with saturated vapor isentropic expansion using data for thermodynamic properties (equations, tables, diagrams), or applicative software of the saturated refrigerant. The method of conditional mean isentropic exponent can be used.

For the two-phase ejectors the flow analyses is based on the assumption that saturated vapor-liquid mixture is in thermodynamic equilibrium state at any cross-section of the ejector, and that liquid and vapor are uniformly mixed and flow at the same velocity without inter-phase slip. The calculations and the nozzle profiling procedure can be conducted according to thermodynamic properties of wet vapor using data for thermodynamic properties (equations, tables, diagrams), or applicative software of the wet refrigerant.

In the primary nozzle motive fluid accelerates and expands (1-2) (Fig. 3.1), 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 usually supersonic, and the nozzle profile is convergent-divergent. The primary nozzle exit velocity

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

The primary nozzle exit cross-section area and diameter

$$A_2 = V_2 / c_2 ; \quad V_2 = M_{pr} v_2 ; \quad d_2 = (4 V_2 / (\pi c_2))^{1/2} \quad (2)$$

At primary nozzle throat cross-section the pressure is equal to the critical and the velocity is equal to the sound velocity

$$a = \sqrt{\partial p / \partial \rho} \quad (3)$$

For vapor ejectors (dry and/or wet expansion) the nozzle throat (critical) cross-section diameter and critical velocity, equal to sound velocity, are

$$d_{cr} = (4 V_{cr} / (\pi c_{cr}))^{1/2} ; \quad c_{cr} = (2 \Delta h_{cr})^{1/2} = (\kappa R T_1)^{1/2} (2 / (\kappa + 1))^{1/2} \quad (4)$$

Viewing from the existing literatures there is a lack of sound velocity data in two-phase flow (Wang and Zhang, [24]). The calculation of the sound velocity and the flow analysis of the two-phase ejectors are complex tasks. The evaluation of the sound velocity and fluid flow analyses can be carried out numerically

$$a = \sqrt{(\Delta p / \Delta \rho)_{s=const}} \quad (5)$$

The two-phase ejector primary nozzle profile can be obtained applying the energy and continuity equations according to following numerical algorithm

$$p=p-\Delta p \Rightarrow t \Rightarrow \Delta h \Rightarrow c \Rightarrow x \Rightarrow v \Rightarrow \rho \Rightarrow \Delta \rho \Rightarrow a \Rightarrow f \Rightarrow A$$

The primary nozzle throat cross-section $A=A_{min}$, ($f=f_{min}$), is critical cross-section, the pressure is critical pressure $p=p_{cr}$, and the velocity is equal to the sound velocity $c=a$.

Exiting the primary nozzle the fluid additionally expands entering into the mixing chamber, where complex flow phenomena appears between the primary and secondary flow. The primary flow draws and entrains the secondary flow into the mixing chamber. The secondary flow comes through the secondary nozzle where it expands (3-4). The velocity of the secondary flow c_4 is

$$c_4 = \Psi_{sec} c_{4s} = [2(h_3 - h_4)]^{1/2} = (2\Delta h_{4s} \eta_{sec})^{1/2} \quad (6)$$

The shear layer between the primary and secondary fluids flowing with 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 design conditions are “double choking”

operation conditions, the secondary fluid gets critical flow (choking flow). A similar calculating procedure can be conducted for the secondary nozzle like that of the primary nozzle.

According to the analysis of the publications cited in references, the expected values of velocity coefficients (Ψ_{pr} ; Ψ_{sec}) are 0.92 – 0.98, and corresponding nozzle efficiencies (η_{pr} ; η_{sec}) are 0.85 – 0.96. Lower values correspond to two-phase nozzles, and higher values correspond to dry expansion nozzles.

The combined flows are mixed flowing within the mixing chamber (2-5, 4-5), where appears a complex process of momentum transfer. By using the momentum equation for the mixing chamber and assuming that constant-pressure mixing occurs inside the constant-area mixing section of the ejector, the cross section areas are $A_2+A_4=A_5$, pressures $p_2=p_4=p_5$ and if the flow friction forces P_{fr} are comprised with mixing chamber efficiency coefficient $\eta_{mc}=0.95-0.98$, the velocity of the combined flow is

$$c_5 = \eta_{mc}(c_2 m_{pr} + c_4 m_{sec}); m_{pr} = M_{pr} / (M_{pr} + M_{sec}); m_{sec} = M_{sec} / (M_{pr} + M_{sec}); \quad (7)$$

The main losses in the ejector occur in the mixing chamber in the process of momentum transfer. The loss of the kinetic energy or loss of the total pressure is

$$\delta_e = \Delta E / E_1 = m_{sec}(c_2^2 - c_4^2) / c_2^2 = (1 - m_{pr})(c_2^2 - c_4^2) / c_2^2 \quad (8)$$

The process in the mixing chamber is the first main source of thermodynamic irreversibility and exergy decrement in the ejectors. If the primary mass flow rate is much larger than secondary mass flow rate ($m_{pr} \gg m_{sec}$), then the loss of total pressure in the mixing process is negligible.

The compression of the fluid is achieved as the combine stream flows through the mixing chamber and diffuser. The kinetic energy $\Delta h_{din} = c_5^2 / 2$ in the diffuser is transformed to enthalpy rise, expressed by rise of the pressure. The combined flow at the mixing chamber outlet often is supersonic. If the velocity of the combined flow is supersonic then a normal shock wave occurs. The shock wave is a process where sudden change in the flow space appears, the velocity suddenly falls from supersonic to subsonic and the pressure rises, which is characteristic of single phase (gas and dry vapor) flow. In two phase flow this complex process is accompanied by mass transfer from one phase to the other, where pseudo-shock waves, feature for dominantly liquid two-phase flow, or dispersed shock waves, feature for dominantly vapor two-phase flow, occur in the flow field (Berana *et al* [22]; Berana and Nakagawa [23], Banasiak and Hafner [26], Zhu *et al* [29]). 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 $\lambda_2 = c_{s1} / a_{cr} < 1$. Across the shock wave $\lambda_1 \lambda_2 = 1$. Approximately, the sound velocity a_{cr} and can be estimated by numerical method ($a = \sqrt{(\Delta p / \Delta \rho)_{s=const}}$). Using the conditional pseudo isentropic exponent (Karwacki *et al* 2011) and/or numerical estimation of the conditional pseudo isentropic exponent ($\kappa = (\ln(p + \Delta p / p) / \ln(\rho + \Delta \rho / \rho))_{s=const}$) and according to the gas dynamic theory the pressure rise across the shock wave (pseudo-shock waves; dispersed shock waves) can be approximately estimated

$$\frac{p_{s1}}{p_5} = \frac{\lambda_1^2 - (\kappa - 1) / (\kappa + 1)}{1 - (\kappa - 1) \lambda_1^2 / (\kappa + 1)} \quad (9)$$

In the shock wave partially the compression is realized. However, the shock wave is thermodynamic 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. The both of them are physics phenomena and cannot be avoided with any design effort. The existence of pseudo-shocks in two phase flow is obtained and proved by experiments (Elbel and Hrnjak [7]; Elbel, [8]; Banasiak and Hafner [26], etc) with measurement of the pressure along the mixing chamber wall.

Additional compression is realized in the subsonic diffuser. According to wide range of publications about subsonic diffuser hydraulic losses, the values of diffuser efficiency η_d are from 0.60 up to 0.80.

V. Performance Characteristics of the Ejectors in Variable Operating Conditions

The thermal system operating conditions determine the design conditions of the ejector. The ejector flow field can be optimally designed and only one optimal ejector geometry corresponded for given thermal system operating conditions. The performance characteristics of the ejector thermal system strongly depend

on the performance characteristics of the ejector. Change of the thermal system operating conditions has strong influence on the ejector performance characteristics.

Maximum coefficient of performance and maximum entrainment ratio of an ejector can be obtained in conditions of “double choking” operation (Fig. 3). If the ejector flow field is designed at its critical flow conditions (design point in Fig. 3), when the secondary flow attains critical (sonic) velocity, then the maximum COP and the maximum entrainment ratio can be obtained, for the given thermal system operating conditions (given exit pressure). For off-design operating conditions, if the exit pressure is lower than design exit pressure then ejector “double choking” operation exists: choking flow at the primary nozzle throat cross-section area and choking flow at the secondary nozzle hypothetical throat area. There is no increase of the secondary flow rate (Fig. 3). If the exit pressure is higher than design exit pressure then the ejector works in “single choking” operation: primary choking flow and subcritical secondary flow. The secondary flow rate is lower for higher exiting pressure. With increment of the exit pressure unsteady flow phenomena with surge in the ejector flow field can be appeared (dot line in Fig. 3). When the exit pressure becomes higher than maximum exit pressure (p_{bp} - back pressure) then reversed flow occurs.

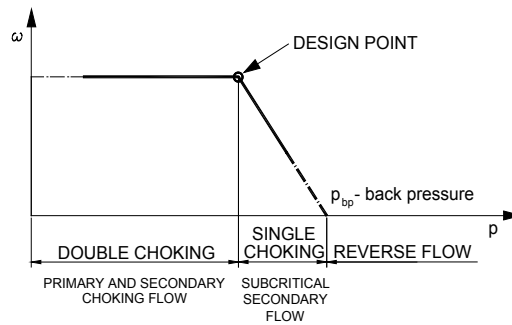


Figure 3 Performance characteristics of an ejector

Numerical experiments have been conducted to obtain performance characteristics of two refrigeration R245fa ejectors. The design conditions for both ejectors are: evaporating temperature $T_e = 15^\circ\text{C}$; condensing temperature $T_c = 38^\circ\text{C}$; generating temperature for the first ejector E1-1 $T_g = 90^\circ\text{C}$; for the second ejector E1-2 $T_g = 120^\circ\text{C}$. Using the adapted calculating procedure to solve the inverse task (estimate performance for given geometry of ejector flow field) the performance characteristics of the ejectors are estimated and given in Fig. 4. From the results given in the figures it can be exposed that the ejector performances are very sensitive on variable operating conditions.

The entrainment ratio dramatically falls in operating conditions with lower evaporating temperatures (Fig. 4a and Fig. 4b). For higher evaporating temperatures entrainment ratio is higher but temperature lift is lower. Increment of the condensing temperature causes decrease of the entrainment ratio. For condensing temperatures lower than design condensing temperature the entrainment ratio remains constant value. Increment of the generating temperature causes dramatic decrease of the entrainment ratio, although the possible temperature lift is higher (Fig. 4c). For generating temperatures lower than design generating temperature (Fig. 4d) the ejector is unable to reach the needed condensing temperature, although high value of the entrainment ratio can be obtained for low temperature lift.

One way to overcome the problems of the ejectors in variable operating conditions is by application of ejector variable technology. Using the primary nozzle with variable area ratio, achieved by applying a movable spindle, the primary flow rate can be regulated to achieve appropriate operating parameters.

VI. Conclusions

The flow characteristics of the single-phase and two-phase ejectors are analyzed, and calculating procedure for estimation of ejector geometrical parameters and performance characteristics is presented. Recommendations for optimal geometric parameters of vapor ejectors (with wet or dry expansion) and of two-phase ejectors are given. Among the efficiency of the ejector flow elements, which depend on ejector geometry and fluid flow conditions, two main sources of thermodynamic irreversibility: the process of momentum transfer in the mixing section (the first) and shock waves, or dispersed shock waves, or pseudo-shock waves in the fluid flow field (the second), determine the efficiency and performance characteristics of the ejectors. The shock waves are feature for gas and dry vapor fluid flow; dispersed shock waves are feature for dominantly vapor two-phase fluid flow; pseudo-shock waves are feature for dominantly liquid two-phase fluid flow. When the first main source of thermodynamic irreversibility (process of momentum transfer in the mixing chamber) is weaker (when the primary flow rate is large then secondary flow rate) the second one

is strongly expressed and vice versa. The both of them are physics phenomena and cannot be avoided with any design effort.

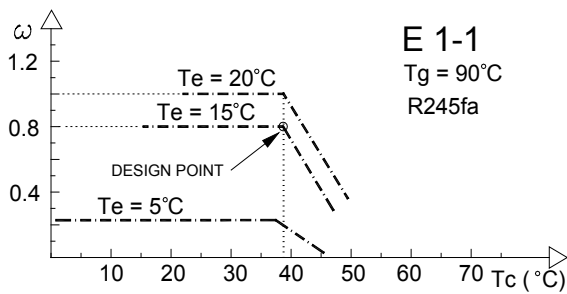


Fig. 4a

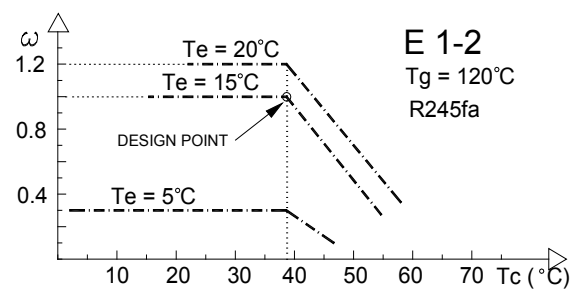


Fig. 4b

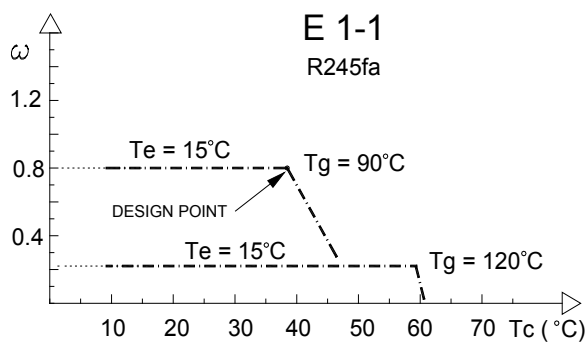


Fig. 4c

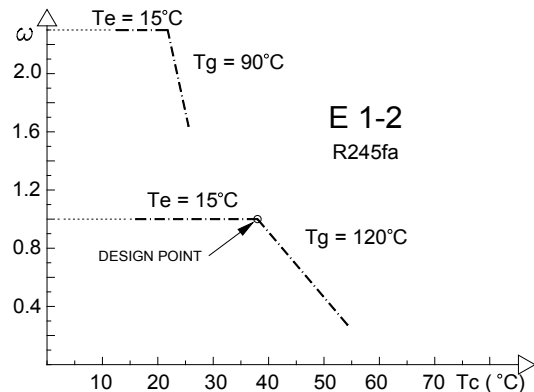


Fig. 4d

Figure 3.4 Performance characteristics of refrigeration R245fa ejectors

For a given operating conditions and for an optimally chosen refrigerant the optimal design of the ejector flow field elements can be obtained, and only one optimal ejector geometry corresponded. Maximum coefficient of performance and maximum entrainment ratio of an ejector can be obtained in conditions of “double choking” operation, and it is recommended design point. The ejectors and systems with ejectors are very sensitive on variable (off design) operating conditions. If the exit pressure is lower than design exit pressure then ejector “double choking” operation exists. There is no increase of the ejector entrainment ratio. If the exit pressure is higher than design exit pressure then the ejector works in “single choking” operation. With increment of the exit pressure unsteady flow phenomena with surge in the ejector flow field can be appeared. When the exit pressure becomes higher than maximum exit pressure then reversed flow occurs. The entrainment ratio dramatically falls in operating conditions with lower evaporating temperatures but temperature lift is higher. For higher evaporating temperatures entrainment ratio is higher but temperature lift is lower. Increment of the generating temperature causes dramatic decrease of the entrainment ratio. For generating temperatures lower then design generating temperature the ejector is unable to reach the needed condensing temperature, although high value the entrainment ratio can be obtained for low temperature lift.

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