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A CONCEPT OF PERFORMANCE IMPROVEMENT FOR R744 TWO-PHASE EJECTOR WITH BYPASS DUCT

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ABSTRACT

A concept of the two-phase ejector with a bypass duct for CO_2 applications is proposed in this study. A geometry and bypass positioning, idea of regulation as well as integration with suction nozzle duct was designed and described. Preliminary numerical analysis of the proposed bypass geometry was performed. Computational platform *ejectorPL* integrated with well validated mathematical model of transcritical R744 two-phase flow were used. Gas cooler and evaporator conditions characteristic for large systems such as supermarket refrigeration units were examined. Separation pressure of 32 bar in liquid receivers were simulated for the bypass variants with two duct shapes and six attachment positions. The perspective results were obtained for this low pressure conditions. Namely, the suction mass flow rate increment was of 36.9% for the bypass angle of 19°. Hence, the bypass implementation resulted in the efficiency improvement from 22.2% to 30.4%. Significant influence of the bypass geometry into the overall ejector efficiency was reported. Finally, a possible shape optimisation of bypass duct as well as further analysis focused on the regulation and control strategy were given.

Keywords: R744, CO2, transcritical ejector, bypass implementation, efficiency improvement

1. INTRODUCTION

Global phase-in of environmentally friendly working fluids has had a crucial impact on the refrigeration area. According to the so called F-gas regulation, the vast majority of present refrigeration units working with synthetic refrigerants should be improved or totally replaced before 2022 [1]. According to non-flammability and non-toxicity of carbon dioxide, the highest safety level of exploitation is ensured in such installations [2]. Finally, R744 (carbon dioxide) gives a reference level for a global warming potential (GWP) factor, while it takes the value of 1. No depletion of the ozone layer is another advantage of carbon dioxide, but this advantage also applies to a whole group of natural refrigerants However, the CO_2 cycle results in significant losses during a throttling process that starts from an area close to the critical point [3]. The necessity of operation in this region is a consequence of a relatively low critical point for R744 of approximately 31.06°C [4].

A large possibility for cycle improvement is the recovery of a relatively high potential of work related to the throttling process of carbon dioxide in comparison to synthetic refrigerants [3]. Ejectors are the most likely solution for an indirect work recovery in refrigeration units [5]. The reasons are related to reliability, no moving parts, relatively simple construction in comparison to direct expanders, e.g., a gear expander. Moreover, the ejector functionality allows for additional fluid circulation or operation as a pumping device [6], [7].

Developed geometrical relationships of an ejector construction allowed for the design of high efficiency ejectors under the given pressure conditions. A further development step was focused on ensuring the proper regulation idea according to the variable load of a refrigeration unit. Two different approaches were proposed in the literature. First, a solution based on an adjustable geometry ejector was experimentally examined by Liu et al. [8]. The authors presented satisfactory results for controlling cycle performance based on needle insertion into the ejector throat. The coefficient of performance (COP) improvement up to 60% was reported due to regulation based on the controllable ejector with a needle. Nevertheless, in this study, the air conditioning cycle was used for the experimental tests with a relatively low compressor power of approximately 10 kW.

Evaluation of a controllable ejector performance for various loads characteristic of large refrigeration units such as in supermarkets was presented by Smolka et al. [9]. A full numerical comparison of a controllable and fixed geometric efficiency was based on the same baseline models characterised by high efficiency as in the work of Palacz et al. [10]. The results noted a very sensitive function of the ejector efficiency relative to a needle position. In the case of a needle position that is too deep, the suction flow was totally choked. Nevertheless, proper adjustment of the needle position resulted in increased ejector efficiency of up to 25% in comparison with the fixed ejector geometry (without the needle).

A second solution is characterised by the idea of regulation opposite from the first described controllable ejectors. Namely, instead of one ejector with an adjustable throat area, a solution based on several parallel working ejectors and binary regulation of such a system was proposed by Hafner et al. [11]. Due to discrete regulation possibilities, a linear profile for controlling this device is ensured. Experimental performance mapping of this multi-ejector module was done by Banasiak et al. [12]. The reported overall performance of ejectors contained in the multi-ejector module was on the level of 30%. The power of the laboratory facility used was 70 kW with a temperature of 35°C at a gas cooler outlet where the evaporation temperature was -3°C to mimic supermarket operation in a warm south European climate. Those authors examined a wide range of operating conditions and confirmed applicability of this device to cooperation with a classical high pressure throttling valve.

The multi-ejector module mentioned in the previous paragraph was evaluated numerically in the work of Bodys et al. [13]. The parallel work of the ejectors examined including motive, suction and outlet collectors was based on the 3-D simulations. The Homogeneous Equilibrium Model of transonic two-phase R744 flow was developed by Smolka et al. [14] and introduced to the computational tool *ejectorPL* (available online: www.itc.netrom.pl) described by Palacz et al. [15]. The operating conditions tested were characteristic of the high ambient conditions in a southern European climate. The numerical evaluation confirmed the possibilities of linear adjustment to the system load. Nevertheless, the overall efficiency of the multi-ejector pack was decreasing with increasing load. The authors stated that these increasing losses are related mainly to the mixing processes in the outlet collector. The benefits offered by full 3-D domain simulation allowed the independent analysis of each ejector in the case of the parallel work mode. This analysis resulted in the stable work of each device with a high efficiency of approximately 35%. Finally, some propositions for further improvement were stated in the optimisation of the outlet collector.

The regulation methods mentioned provide the possibility of the load regulation. Nevertheless, as described in [9], [10], [14], [16]–[19], the high sensitivity of ejectors to operating conditions and designed geometrical parameters forces these devices to work with an optimal efficiency that is close to on-design operating conditions. Moreover, the optimal efficiency could be obtained only with a specified mass entrainment ratio and a corresponding pressure lift. However, according to various systems operations, an ejector is forced to work at a variable pressure lift. Then, the efficiency decreases due to a decreasing entrainment ratio based on unfavourable pressure distribution along the ejector axis and consequently, reduced suction phenomena.

To ensure a suspension of such a situation, an additional duct called bypass could take the role of a suction nozzle substitute. The duct mentioned, located in the ejector diffuser, would provide a bypass flow to the suction nozzle. Simultaneously, the suction stream would be delivered in a more favourable pressure region. The idea of the bypass duct was proposed by the authors of [20] and examined by Chen et al. [21]. The analysis contained three different pressure lifts between the suction and the outlet ports. The results were reported after some geometrical optimisation of the bypass duct in the second of the papers mentioned [21]. Namely, in the lower pressure case examined, the improvement of the mass entrainment ratio was enlarged from 10.7% (baseline design [20]) to 32.8% due to the optimised position of the bypass. Moreover, after some corrections of the bypass shape, the reported improvement was 48.7%. Nevertheless, this large improvement was examined for the narrow range of operating conditions. Next, analysis of the bypass positions and its shape was quite limited, concerning only simple orthogonal duct shapes. In addition, the authors of that study used air as a working fluid, and the ideal gas law was used for the density calculations. Moreover, the pressure at the suction port and the bypass duct was assumed to be constant. Finally, the suction nozzle and the bypass duct were simulated as separate ejector ports; an analysis of the suction nozzle and the bypass integration was not conducted. Nevertheless, this interesting ejector concept certainly deserves further studies.

In this study, the bypass-type ejector is proposed and analysed for CO_2 applications. To the best knowledge of the authors, the bypass investigation in R744 ejectors for refrigeration applications has not been provided to date. The bypass geometry and its positioning, the idea of regulation as well as the integration with the suction nozzle duct were proposed and discussed. Adapting the previously developed [14] and well validated mathematical model of transcritical R744 two-phase flow [14], [15], the analysis of the bypass concept was performed. The series of variant numerical simulations was provided using the computational platform *ejectorPL* [15]. The motive nozzle and suction nozzle inlet conditions reflecting typical gas cooler and evaporator conditions for large systems such as supermarket refrigeration units were examined for three different levels of the ejector outlet pressure (corresponding to pressures in the liquid receiver). Promising results of the mass entrainment ratio were obtained for the lowest pressure conditions leading to the same efficiency as in the case of the ejector operation with higher separation pressures. In addition, the distribution of the sucked stream between the suction nozzle and the bypass duct were analysed in the axisymmetric CFD study. The pressure and Mach number distributions along the ejector axis as well as in the bypass location were presented and discussed. Finally, potential shape optimisation for higher evaporation pressures was also given.

BYPASS EJECTOR IDEA

Ejector operation with pressure lift decreased beyond the rated (design) conditions typically results in lower overall efficiency of the ejector. Physical reasons are based on the unfavourable pressure distribution in a mixing zone due to chocked flow conditions in this area. To overcome geometrical constraints and increase the suction flow rate, an additional duct introduced after the blocked flow region might be considered. This duct plays the role of the suction nozzle bypass, and this nomenclature will be used in this study. The proposed solution for carbon dioxide cycles could have a great impact on the overall COP of the system. Moreover, according to the proposed regulation idea presented in Fig. 1, no additional connector will be required. Implementation of bypass is based on a classic ejector geometry. Hence, basic ejector sections such as converging-diverging motive nozzle, suction nozzle, mixer and diffuser are indicated in Fig. 1, where one half of an ejector geometry with respect to the device axis is schematically presented. The bypass duct volume was marked by a blue area located in the suction duct before the suction nozzle. The shape and position of the suction nozzle duct are crucial for effective bypass implementation. Simultaneous connection of the suction nozzle and bypass with only one inlet suction port (see Fig. 1) gives more reliability and allows for avoidance of an additional valve. In this paper, the concept of the bypass opening is based on two separate parts of the ejector. The two parts mentioned are obtained as a result of the precise cutting of the standard ejector. Therefore, part A and part B will be created. Part A is stationary. The volume of the bypass duct (blue in Fig. 1) is obtained after offset of the moving part B in the direction of the ejector outlet. A proper location of the suction port allows for supplying both suction nozzle and bypass. Dependent on displacement of part B, proper bypass width is obtained. In the case of the zero offset, part A and B are connected, and the standard fixed ejector geometry is utilised.
The proposed solution does not require any additional pipeline systems for the bypass activation, apart from a system enabling retraction of part B from the rest of the ejector geometry.

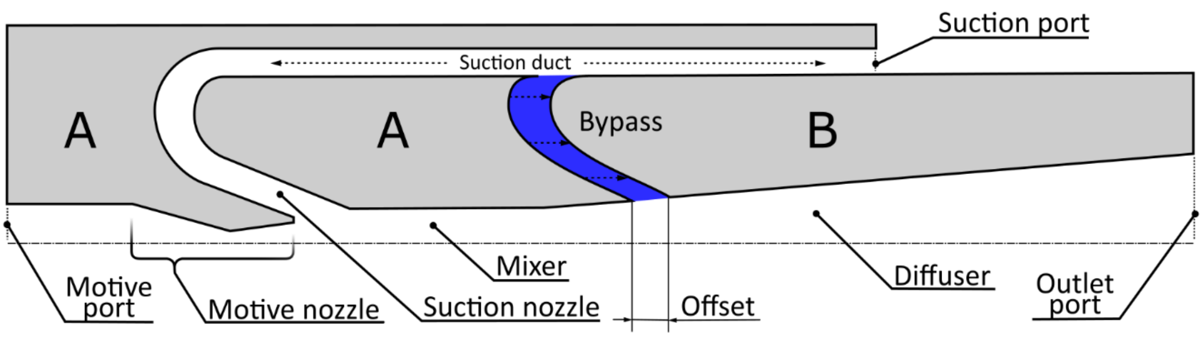

Figure 1. Idea of bypass implementation to the fixed-geometry ejector.

In this study, the proposed bypass idea was preliminarily investigated using the CFD methods. Hence, the main efforts were focused on the flow analysis and potential of overall improvement in the selected operating conditions. The design development of the bypass ejector should be considered as the other study involving additional factors, i.e., the manufacturing of a regulation mechanism and its sealing, as well as the connection between the outer and inner part A. Such a study should be preceded by predictions of the available potential of the bypass solution. The data mentioned are included in this paper.

Geometry of the ejector with the bypass

The shape of the bypass was obtained on the basis of the suction nozzle modification and generation of a proper turn, selecting two radii (r_1, r_2) and the dimension L_1 as presented in Fig. 2. Next, the bypass entered the diffuser volume with the angle β between the ejector axis and walls of the bypass. Moreover, the dimension d_1 was a width of the bypass. In the simulations performed, the assumption of a bypass width equal to half of the mixer diameter was used.

The parameters used for definition of the bypass positioning are L_{MIX} and L_{BPS}. The dimension L_{MIX} describes the length of the ejector mixer. The second dimension, i.e., introduced as L_{BPS}, gives the position of the connection of the bypass and the diffuser wall.

For a complete description of the bypass geometry, dimension L_{BSC} was also introduced. This dimension denotes the length of the bypass suction chamber and varies according to the angle β and the bypass position. This dimension will also be used to discuss the results obtained. The beginning of the bypass suction chamber is defined by the point located closer to the mixing chamber (i.e., L_{BPS} - L_{BSC}), while the end of the bypass suction chamber is defined as the point closer to the ejector outlet (i.e., L_{BPS}).

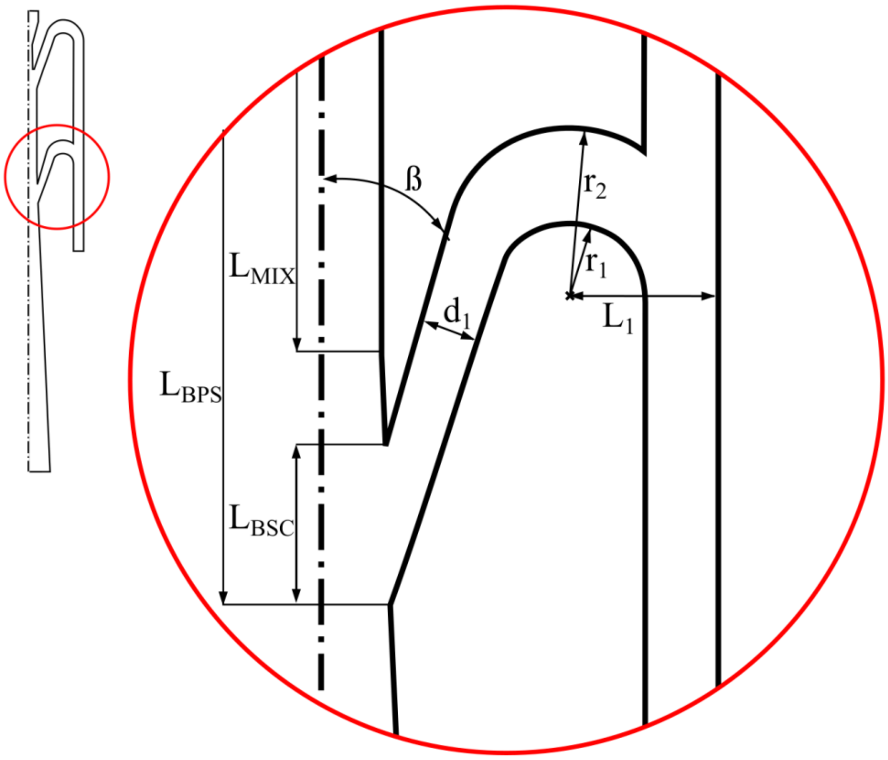

Figure 2. The proposed bypass geometry.

Finally, the bypass position is presented in the form of dimensionless ratio between L_{BPS} and L_{MIX}:

$$Position = \frac{L_{BPS}}{L_{MIX}} \qquad (1)$$

This parameter shows how far the bypass is located inside the diffuser. The bypass idea was tested in its several positions. Moreover, two angles β were examined during the computational procedure, i.e., 19° and 38°. The angle of 19° was assumed as a direct translation of the suction nozzle angle, while the second angle was a factor of two larger than the first angle. Because of some geometrical restrictions on the bypass suction chamber, the positions examined were not the same for each angle.

RESULTS AND DISCUSSION

Global results of the performed simulations for motive pressure 84.5 bar (32 °C) and suction pressure 28.0 bar (1.0 °C) and pressure lift of 4 bar are presented in Table 1. The last column presents value of the relative increment in MER (ΔMER) defined as:

$$\Delta MER = \frac{\chi_{bypass} - \chi_{baseline}}{\chi_{baseline}} \cdot 100\% \qquad (2)$$

First of all, every examined bypass position resulted in a significant increment of the ΔMER value. The observed phenomenon of the MER increment is based on the increment of the suction stream with the constant motive stream. In comparison to the baseline case, the smallest ΔMER that was recorded took a value of 15.5%. Position 1.4 resulted in the maximum improvement of the sucked stream mass flow rate for both angles. The bypass angle of 38° resulted in the ΔMER of 32.0% and for the angle of 19°, this flow parameter increased by 36.9%. As a result, the ejector efficiency was lifted to the level of 30.4% starting from the baseline value of 22%. Hence, the bypass ejector working at lower pressure conditions was as efficient as a baseline ejector operating in high pressure conditions. Another positive statement is related to the character of the efficiency changes. Namely, a similar level of the efficiency was obtained for a given angle within three positions located in the diffuser. If the bypass duct is located in the diffuser, ejector efficiency is dependent on the bypass shape (the angle), while the influence of its precise position becomes less important. Moreover, this also means that the regulation area for the proper angle used is not affected by rapid changes in the ejector performance.

Table 1. Results of the simulations.

Angle -	Position -	Motive port kg/s	Suction port kg/s	MER -	Efficiency %	ΔMER %
Baseline		0.074	0.037	0.504	22.2	-
38°	1	0.074	0.0431	0.582	25.7	15.5
	1.2	0.074	0.0472	0.637	28.1	26.3
	1.4	0.074	0.0493	0.665	29.3	32.0
	1.6	0.074	0.0470	0.635	28.0	26.0
19°	1.0	0.074	0.0432	0.583	25.7	15.6
	1.3	0.074	0.0504	0.680	30.0	35.0
	1.4	0.074	0.0511	0.690	30.4	36.9
	1.5	0.074	0.0507	0.684	30.2	35.8

An analysis of the suction stream distribution is presented in Fig. 3, where the flows through the suction nozzle and the bypass are separately given. The sum of these streams is equal to the value from Table 1, i.e., the mass flow rate of the suction port. Moreover, black symbols present the total stream depending on the bypass position, blue symbols present the suction nozzle stream and red symbols indicate the bypass stream. Similar to Fig. 5, the crosses and circles are used for the bypass angle of 19° and 38°, respectively. The mass flow rate in the suction nozzle in the baseline case was represented by the green dashed line.

According to the total stream results, the optimum bypass position in the case of the angle 19° is barely visible due to the small incremental differences between Positions 1.3, 1.4 and 1.5. A wide range of similar high improvement values allows for higher tolerance in the manufacturing and regulation process. In the case of the angle of 38°, a character of the total stream changes is different and has a visible maximum. The suction nozzle mass flow rate changes are almost linear for both angles, while the values related to the smaller angle are slightly smaller than those for the angle of 38°. Moreover, these streams are growing constantly through the whole range examined. Hence, the maximum points are located at the highest bypass positions. However, the mass flow rate of the suction nozzle is lower than the baseline case for each of the simulated bypasses. In the case of Position 1.5 – after the position of the maximum total stream - this value approaches the baseline. Nevertheless, a character of the bypass results differs between the angles. Moreover, it is not uniform like the changes obtained for the mass flow rate through the suction nozzle. The character of these changes was a source of the same trend of the total stream. The maximum bypass stream is related to Position 1.2 for the angle of 38° and 1.3 for the angle of 19°, which is before the maximum of the total stream. Due to the uniform growth of the flow through the suction nozzle, the highest value of the total stream is slightly farther. Finally, a small difference between the examined angles is visible. Namely, in the case of the smaller angle, the suction stream is higher by almost 8% based on the maximum values The performance of the suction nozzle is not significantly affected by introducing the considered bypass, and a similar growth is reported in both analysed angles, leading to a statement that the re-design process of the suction nozzle is not necessary in the case of the bypass ejector type.

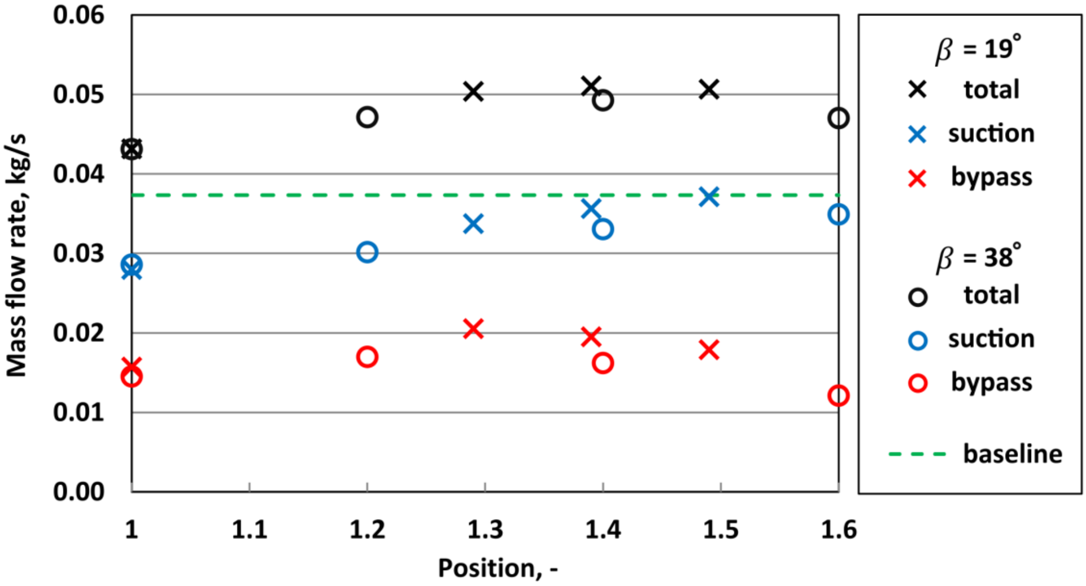

Figure 3. Distribution of total suction mass flow rate for suction nozzle and bypass.

The pressure distribution analysis is presented referring to Fig. 4. In this figure, the absolute pressure field in the whole baseline and bypass ejectors (left) and in the area (dotted red frame) of the bypass suction chamber (right) is presented. Both fields in Fig. 4 contain two symmetrical halves where the left one was obtained from the baseline unit, and the right one was obtained from the bypass ejector simulations.
The absolute pressure field analysis shows some fundamental phenomena. Starting from the motive nozzle, 8.45 MPa pressure is converted to high speed flow. The difference between the baseline and the bypass ejector can be observed in the characteristic shock trains located in both mixing sections. Namely, shift of pressure patterns between analysed cases is clearly visible.
As it was mentioned, the area of the bypass suction chamber was presented in the different pressure range on the right-hand side of Fig. 4 for a better illustration. In addition to the shock train shift, the pressure values in the mixing section are higher for the bypass case than for the baseline case, not only in the ejector axis but also in the wall vicinity. The mixing area of the bypass ejector (right) indicates 2.7 MPa, while the baseline (left) ejector mixer is described by approximately 2.2 MPa, resulting in a difference of approximately 0.5 MPa. These increments could be related to the lower mass flow rate in the mixer, as it was stated on the basis of the mass flow rate distribution presented in Fig. 3. Analysis based on the absolute pressure distribution along the ejector axis (will be presented during conference) resulted in the similar difference of approximately 0.47 MPa. The pressure field in the cross-section of the bypass suction chamber is uniform on the level of approximately 2.8 MPa. Right after the bypass suction chamber, the pressure level is lowered again to approximately 2.5 MPa in the whole diffuser cross-section related to the additional mass flow introduced to the diffuser volume trough the bypass. To avoid this additional pressure drop, adjustment of the bypass width or the diffuser width starting from the end of the bypass suction chamber could bring additional improvements. Nevertheless, the bypass width of the mixer half diameter is found to be quite a good choice for the preliminary analysis of the bypass solution.

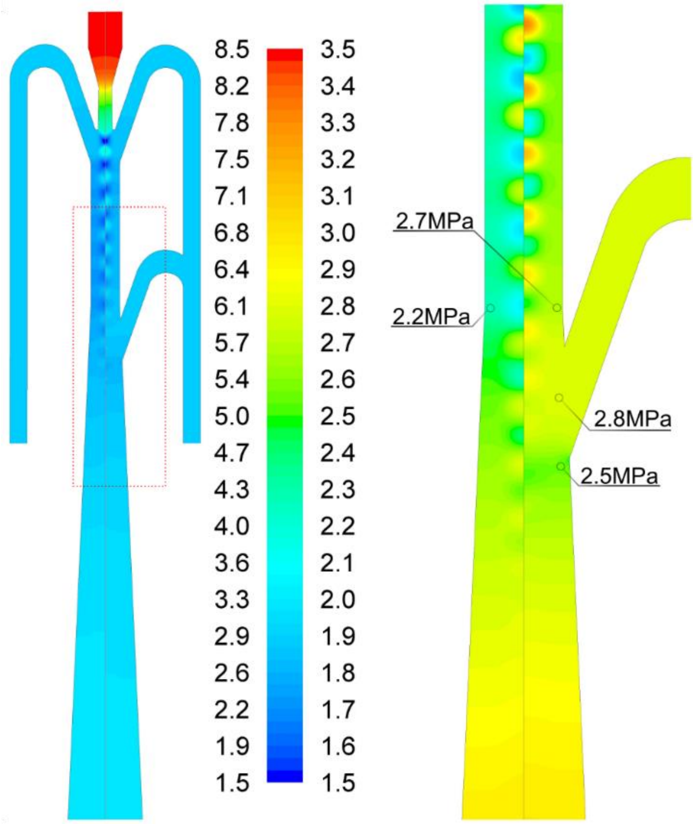

Figure 4. Absolute pressure field (MPa) of the baseline (left) and the bypass ejector (right) in the whole and zoomed views.

CONCLUSIONS

The idea of a bypass implementation to the transcritical carbon dioxide ejector was proposed. A control approach of the bypass was also proposed. The bypass idea presented was numerically examined based on the operating conditions characterised by high accuracy for the HEM approach employed. The numerical simulations were executed using the well-validated computational platform *ejectorPL* [15]. For the operating conditions with the pressure lift of 4 bar, very promising results were obtained. The ΔMER was 32.0% for the bypass angle 38° and 36.9% for the bypass angle 19°.

The translation of the shock train along the ejector axis as well as the higher pressure in the ejector mixer was reported because of the bypass implementation. According to the pressure field distribution in the bypass suction chamber, the assumption of the bypass width of a half of the mixer diameter was evaluated as sufficient in the preliminary analysis. Moreover, in the bypass suction chamber, the uniform pressure distribution in the duct cross-section, as well as the constant linear pressure drop, was one of the features characteristic of the highest improvement of the ejector operation.

Small differences between the MER improvements were obtained for the optimum position and the two neighbouring positions considered. We could conclude that very high accuracy of the bypass positioning might be avoided in the operation with the low pressure lift. In such an operation, the bypass angle becomes a more significant factor, leading to the statement that optimisation of the bypass duct profile should bring more benefits than the detailed analysis of the bypass positioning. Finally, from the point of view of ΔMER, the most crucial parameter is the pressure lift of the operation, and then the bypass shape. Finally, based on the first results presented, the bypass position should be located approximately 40% of the mixer length after the diffuser beginning, regarding a properly designed fixed geometry ejector.

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NOMENCLATURE

Acronyms and abbreviations		**Roman Letters**	
R744	Carbon dioxide	*L*	length, m
GWP	Global Warming Potential	*r*	radius, m
COP	Coefficient of Performance	*d*	width, m
CFD	Computational Fluid Dynamics	**Subscripts**	
MER	Mass Entrainment Ratio	*MIX*	mixer
Greek Letters		*BPS*	bypass
χ	Mass Entrainment Ratio, -	*BSC*	bypass suction chamber
β	angle, °		

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