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EXPERIMENTAL INVESTIGATION OF HYBRID VAPOUR COMPRESSION-EJECTION REFRIGERATION SYSTEM

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ABSTRACT

Application of ejector systems for utilisation of low temperature heat and transform this heat for usable cooling power is one of the most effective and economic way for energy management. The basic concept of the hybrid system is to use a gas ejector in a vapor mechanical compression refrigeration system. The hybrid system is used for chilled water production required by air cooler in AC unit. The proposed system uses a gas ejector as the main device for compression of refrigerant during the period with relatively low condensing temperature, while for the period with high condensing temperature the compression of the refrigerant is provided by mechanical compressor. In effect the cooling power is produced with maximum use of low-temperature heat, such as a district heating system, solar power or waste heat. In the paper the operation of hybrid system in ejector mode and compression mode will be shown. The critical temperature that swich the modes in the hybrid system will be determined. The performance lines will be presented and discussed.
Keywords: ejector, low-GWP refrigerant, hybrid refrigeration, COP, energy efficiency

1. INTRODUCTION

The global push for energy efficiency and sustainability has driven significant innovation in the field of refrigeration technology. Traditional vapor compression refrigeration systems, while effective, are highly energy-intensive and rely heavily on mechanical compressors, leading to significant electricity consumption and environmental impact (Dincer and Kanoglu, 2010). To address these challenges, hybrid vapor compression-ejection refrigeration systems have been developed, offering a promising solution by utilizing low-grade heat sources to enhance energy efficiency (Lawrence and Elbel, 2013). Vapor compression refrigeration systems have been the cornerstone of cooling technology for decades. These systems operate by compressing a refrigerant, which then releases heat as it condenses. The refrigerant is expanded and evaporates to produce cooling (Palacz, and Smolka, 2016). Despite their widespread use and reliability, the mechanical compressors in these systems consume a substantial amount of electricity, leading to high operational costs and significant environmental impacts due to greenhouse gas emissions. Ejector refrigeration systems present an alternative approach by using a high-pressure fluid to entrain and compress a secondary low-pressure fluid (Cao and Yu, 2011). The ejector functions as a compressor without any moving parts, offering simplicity and durability. These systems are particularly effective for harnessing low-grade heat sources such as waste heat, solar thermal energy, or district heating systems. However, the efficiency of standalone ejector systems can be limited, especially under varying operating conditions (Zheng, et al., 2012). The hybrid vapor compression-ejection refrigeration system combines the advantages of both traditional vapor compression and ejector systems. In this hybrid configuration, the ejector is used to compress the refrigerant during periods of low condensing temperatures, utilizing low-temperature heat sources and reducing the load on the mechanical compressor. This approach enhances overall system efficiency and decreases energy consumption. During periods of high condensing temperatures, the mechanical compressor takes over to ensure stable cooling performance. This dual-mode operation optimizes energy use and improves the system's adaptability to different environmental conditions. Hybrid systems are particularly advantageous in applications requiring chilled water production, such as air conditioning units (Guan and Yu, 2018). By maximizing the use of low-temperature heat sources, these systems can significantly reduce energy consumption and operational costs. Furthermore, hybrid systems contribute to lower environmental impacts by minimizing reliance on fossil fuels and reducing greenhouse gas emissions. Recent research has focused on optimizing the performance of hybrid vapor compression-ejection refrigeration systems (Pian and BoroumandJazi, 2019). Experimental investigations have shown that these systems can maintain stable cooling capacity and a high coefficient of performance (COP) even under varying thermal loads and environmental conditions. The ability of hybrid systems to effectively utilize low-temperature heat sources has been validated through various experimental setups, providing valuable insights into their practical applications and performance metrics such as mass entrainment ratio, cooling capacity, and COP. The primary benefits of hybrid vapor compression-ejection refrigeration systems include enhanced energy efficiency, reduced operational costs, and lower environmental impact. By leveraging low-grade heat sources, these systems can achieve significant reductions in electricity consumption. However, challenges remain in optimizing system design for specific applications, ensuring consistent performance across different operating regimes, and developing effective control strategies to manage the transition between ejector and mechanical compressor modes. The hybrid vapor compression-ejection refrigeration system represents a significant advancement in sustainable cooling technology. By effectively harnessing low-temperature heat sources, these systems provide a viable solution for reducing energy consumption and mitigating the environmental impact of cooling processes. Continued research and development are essential to address existing challenges and further improve the efficiency and applicability of these innovative systems. As the demand for energy-efficient cooling solutions grows, hybrid refrigeration systems are poised to play a crucial role in advancing global sustainability efforts.

In general the hybrid ejector-compressor syetm can be used with differend heat source depending on specific requirements. So far, three potential heat source variants are considered: (a) district heating network; (b) solar collector systems, (c) waste heat sources. The ejector refrigeration device is defined by three temperature levels. Considering the potential heat sources and application range, the following can be defined: (a) motive temperature range: tgs = 55…120°C; (b) evaporation temperature range: tes = 0…10 (12)°C and (c) condensation temperature range: tks = 20…35°C

The district heating network is characterized by the lowest available heat carrier temperature, typically not exceeding 65°C. To effectively use this source for powering the ejector system, it is necessary to reduce the heat carrier temperature to around 40°C. This requirement comes from the district heating provider, which mandates that the temperature of the returning heat carrier to the cogeneration plant or heating plant must be such. Lowering the heat carrier temperature poses a technological challenge due to the constant temperature process of the refrigerant's evaporation. Thus, the heat carrier cannot reduce the temperature below the refrigerant's boiling point. It should be noted that there are no physical restrictions on the drive source temperature, but the lower it is, the lower the energy efficiency. The heat carrier temperature from waste heat sources (various origins) ranges widely, starting from 80°C and upwards. From solar collectors, depending on their construction, type, and external conditions, the heat carrier temperature can reach up to 120°C or more. Temperatures above 120°C for many working fluids exceed the critical point temperature, necessitating a supercritical thermodynamic cycle. Depending on the system's purpose—standard cooling or high-temperature cooling—it can be defined that producing cooling for air conditioning purposes with standard parameters, i.e., chilled water temperatures of 6/12°C, requires an evaporating temperature of the working fluid in the range of 0-3°C. For high-temperature cooling, i.e., when supplying chilled water at 14/17 (19)°C to an air handling unit, the expected saturation temperature in the evaporator is within 8-10°C. Regarding expected condensation temperature values, it can be anticipated that regardless of the drive source conditions and evaporation temperature within the specified range, it will fall within the range of +20…+40°C.

Since the district heating network offers the worst drive parameters from a thermal perspective, the study has focused most attention on such operating conditions. The results of these studies are presented in the paper as operational characteristics, which can estimate energy efficiency, the amount of cooling produced, the amount of heat energy consumed, and the temperature range within which the system can operate with satisfactory energy efficiency.

2. RESEARCH DESCRIPTION

2.1. Testing bench

The experiments were conducted using the system illustrated in Figure 1. The refrigerant R12364ze(E) was ued as a working fluid. The main components of the research setup are: steam generator (G), condenser (C), evaporator (E), steam header, compressor (komp), refrigerant pump, regenerative heat exchanger (R), expansion valve, regulating and shut-off valves. The fundamental premise for proposing the above configuration of the system is to achieve stable operation of the refrigeration system across the entire range of operating parameters, regardless of drive parameters and environmental conditions. It is worth referring here to a similar situation that occurs in heat pump technology. In Central European climatic conditions, it is essentially standard to design bivalent systems in which the heat pump provides heating power for ambient conditions above a certain established temperature (referred to as the bivalence temperature), while below this temperature, the heating power of the heat pump is insufficient, and therefore supplemented by a peak heat source. A similar situation occurs in the case of a thermally driven refrigeration system. For such a system, it is possible to propose a configuration in which, at the expense of energy efficiency, its operation is ensured for the widest possible range of operating parameters, and above all - for the highest environmental temperatures. The thermally driven refrigeration system operates up to a certain threshold value of ambient temperature, and when it is exceeded, the peak system is activated. This solution is able to ensure the operation of the system with the highest efficiency for the main part of the summer season, and only in the most unfavorable conditions - this system would be supplemented by a peak compression refrigeration system. The proposed refrigeration system has the unique advantage that the installation already includes the fundamental part of the refrigeration system (evaporator with feed valve and supply automation, condenser), so a compressor can be built into this system to achieve the effect of a bivalent system. It is worth emphasizing that other cooling production technologies (absorption systems) do not allow the construction of bivalent systems in this form. In the case of stream systems, the matter practically boils down to installing only a compressor in the system and applying an appropriate control algorithm for switching to compressor operation mode. The proposed solution ensures a significant increase in the operational safety of the system by ensuring the delivery of cooling to the recipient regardless of circumstances. Thus, the compressor system will support the operation of the stream system in conditions of occurrence of non-design parameters, but also in the event of a district heating network failure or a decrease in network heat parameters.

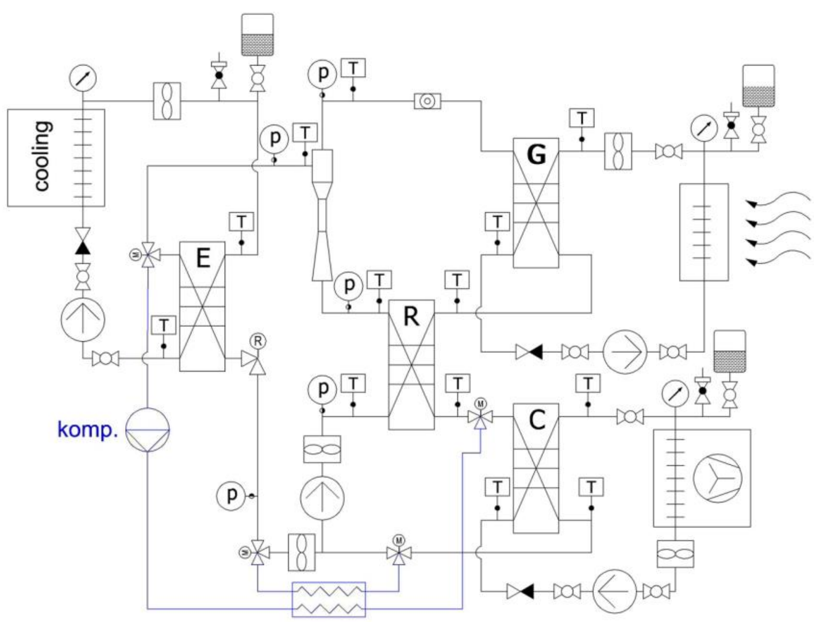


Figure 1: Schematic of tested ejector refrigeration system driven by industrial waste heat

The ejector was equipped with pairs of temperature and pressure (T,p) sensors located in the suction chamber, the mixing chamber, and in the diffuser. Location of the temperature sensors (RTD) and pressure transducers (P) at the testing stand is presented in Fig. 1. Accuracy of the sensors were as follows: 0.25% for the pressure transducers, 0.20% for the RTD temperature sensors, and 0.15% for the Coriolis mass flow meters. Measurement error was calculated by means of the total derivative approach. All of the measurement sensors and the measurement circuits were calibrated before each of the experimental run. The measurements were collected by data acquisition system and stored on the PC computer. The test rig included three auxiliary loops: one for the evaporator's thermal load, another for condenser cooling, and a third for the generator's heat source. Each system had mass flow meters and temperature sensors at the inlets and outlets of the heat exchangers, allowing for the adjustment of refrigerant flow rates and operational parameters over a broad range. Glycol was used as the heat transfer fluid in the evaporator heat load system, while water was used in both the condenser cooling system and the generator heat source system. An automatically controlled dry cooler was part of the condenser cooling system, and the motive heat load system featured an automatically controlled 100 kW electrical heater. Control valves enabled the adjustment of operating parameters for the motive vapor entering the motive nozzle of the ejector. A HYDRA-CELL diaphragm pump with a 3 kW/700 RPM motor acted as the circulation pump for the refrigerant. Plate heat exchangers were utilized for the generator, condenser, and evaporator, and the test stand was equipped with a regenerative plate heat exchanger located between the ejector outlet and the condenser.

2.2. Methodology

Ejector system described in this paper was applied for utilisation of low temperature motive heat produced by domestic heating network. Heat flux transferred by refrigerant is calculated as

$$\dot{Q} = \dot{m} \cdot \Delta h \,. \qquad \text{Eq. (1)}$$

Specific enthalpy difference between the heat exchanger inlet and outlet was calculated on the basis of temperature and pressure measurements from the equation of state h = f(p,T) using NIST database. For water or glycol side the heat flux was calculated as:

$$\dot{Q} = \dot{m} \cdot c_p \cdot \Delta T \,, \qquad \text{Eq. (2)}$$

where temperature difference of water or glycol was taken directly from measurement, and specific heat capacity for water cp = 4.186 kJ kg^{-1} K^{-1} and for glycol cp = 3.58 kJ kg^{-1} K^{-1} were taken. Power consumed by the pump was calculated as:

$$P_p = \frac{\dot{m}}{v} \cdot \frac{\Delta p}{\eta} \,. \qquad \text{Eq. (3)}$$

and the pump efficiency was assumed η = 0.80. The specific volume of liquid refrigerant was found as v = f(p,T) In the case of compression mode, the power consumption by mechanical compressor was directly measured. Mass entrainment ratio is defined as the ratio between the mass flow rate of refrigerant that flows through the evaporator and the mass flow rate of the motive vapour :

$$U = \frac{\dot{m}_e}{\dot{m}_g} \,. \qquad \text{Eq. (4)}$$

The coefficient of performance of the system (*COP*) in general is defined as ratio of the refrigeration capacity divided by the motive power. For both operating mode, the COP was calculated according to the formulas

$$COP_{\text{ejector-mode}} = \frac{\dot{Q}_e}{\dot{Q}_g + P_p}. \quad \text{Eq. (5a)}$$

$$COP_{\text{compressor-mode}} = \frac{\dot{Q}_e}{P_{el.comp}}. \quad \text{Eq. (5b)}$$

2.3. Results

Two experimental test runs were carried out. For run No.1 the average operating parameters were as follows: $t_{g,sat}$ = 59.0 °C, superheating Δtg = 3-5 K, $t_{e,sat}$ = − 1.0 °C. For run No.2 the average operating parameters were as follows: $t_{g,sat}$ = 57.5 °C, superheating Δtg = 7-9 K, $t_{e,sat}$ = 2.0 °C. For both runs the condenstation temeprature was variable parameter. The necessary thermal and flow parameters such mass flow rates, temperature and pressure were measured, next, the specific entalphies and specific volume were determined and used for calculation of heat fluxes. Results are presented in table 1 and table 2.

Table1. Average operating parameters for run No.1

p_c	Q_g	Q_e	P_{pump}	m_g	m_e	U	COP	p_c	Q_e	P_{comp}	COP
kPa	kW	kW	kW	kg/s	kg/s			kPa	kW	kW	
Ejector mode								Compressor mode			
453.6	95.0	24.3	0.45	0.469	0.110	0.235	0.254	500.3	24.4	5.2	4.70
465.9	92.9	24.4	0.43	0.466	0.112	0.241	0.262	527.7	24.7	5.2	4.78
496.6	91.1	23.8	0.41	0.463	0.111	0.241	0.260	569.7	24.0	6.1	3.90
505.9	90.9	12.8	0.38	0.461	0.079	0.171	0.141	601.0	23.6	6.0	3.94
510.4	90.5	10.3	0.36	0.461	0.057	0.124	0.113	614.9	22.6	7.6	2.97
515.9	90.1	7.8	0.35	0.461	0.050	0.108	0.086	659.1	23.6	4.5	5.21
								713.2	22.8	5.4	4.23
								757.1	22.8	4.2	5.46
								872.2	23.0	2.8	8.16
								1005.6	21.2	5.7	3.75

Table2. Average operating parameters for run No.2

p_c	Q_g	Q_e	P_{pump}	m_g	m_e	U	COP	p_c	Q_e	P_{comp}	COP
kPa	kW	kW	kW	kg/s	kg/s			kPa	kW	kW	
Ejector mode								Compressor mode			
456.7	96.1	26.3	0.41	0.443	0.143	0.324	0.272	503.9	24.4	2.3	10.53
473.5	95.6	25.4	0.40	0.441	0.146	0.330	0.265	555.8	23.7	4.3	5.57
484.4	94.4	25.1	0.39	0.445	0.136	0.307	0.265	578.3	23.8	2.8	8.50
496.8	93.0	22.6	0.38	0.448	0.113	0.252	0.242	616.3	23.3	2.5	9.36
502.6	91.1	17.9	0.37	0.450	0.096	0.214	0.196	667.2	22.5	2.7	8.31
518.6	89.3	10.8	0.36	0.459	0.067	0.146	0.121	729.8	23.4	1.1	21.80
								871.3	21.9	3.4	6.38
								1012.8	21.2	3.0	7.07

The results presented above indicate the effective utilization of thermal energy at a temperature slightly above 60°C for cooling production by a hybrid refrigeration device. The source of driving heat at such a low temperature can be a district heating network, which is typically supplied with heating water at a temperature not exceeding 65°C. These parameters allow for achieving saturation parameters in the generator at the level of 57-59°C, as shown in the graphs. Approximately 25 kW of cooling power was obtained while consuming just under 100 kW of heating power. Evaporation conditions at the level of 0-2°C correspond to standard cooling conditions. The pressure distribution along the ejector indicates that the device operated in both on-design and off-design modes. Similarly, the plotted characteristics indicate both modes of operation of the ejector system.

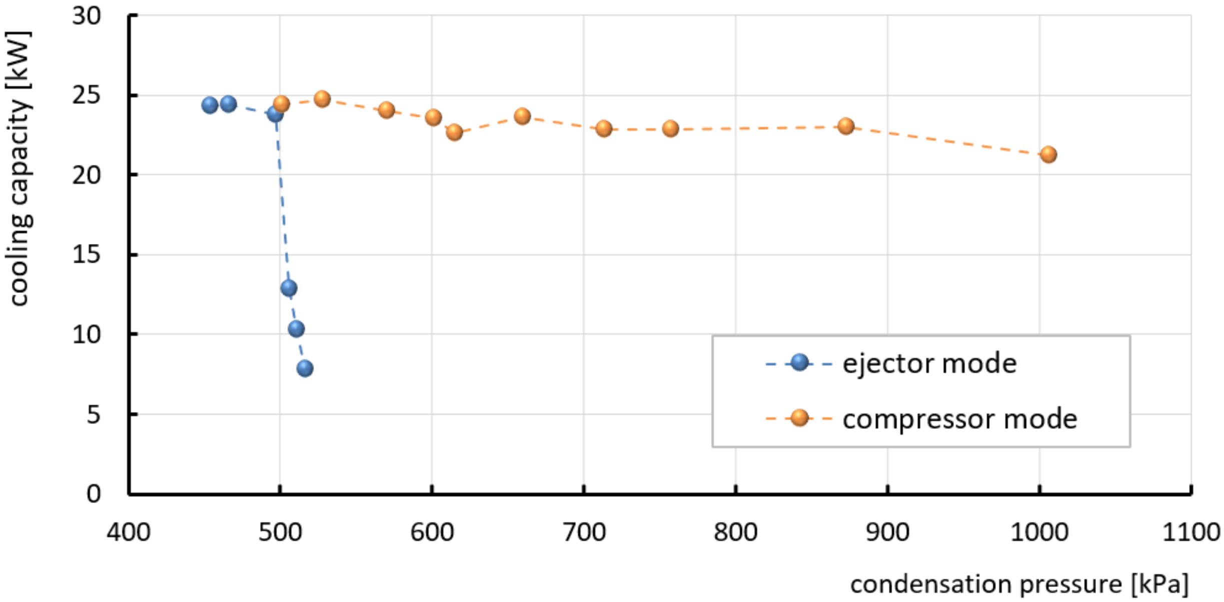


Figure 2: The performance lines for ejector-compressor mode, run No.1

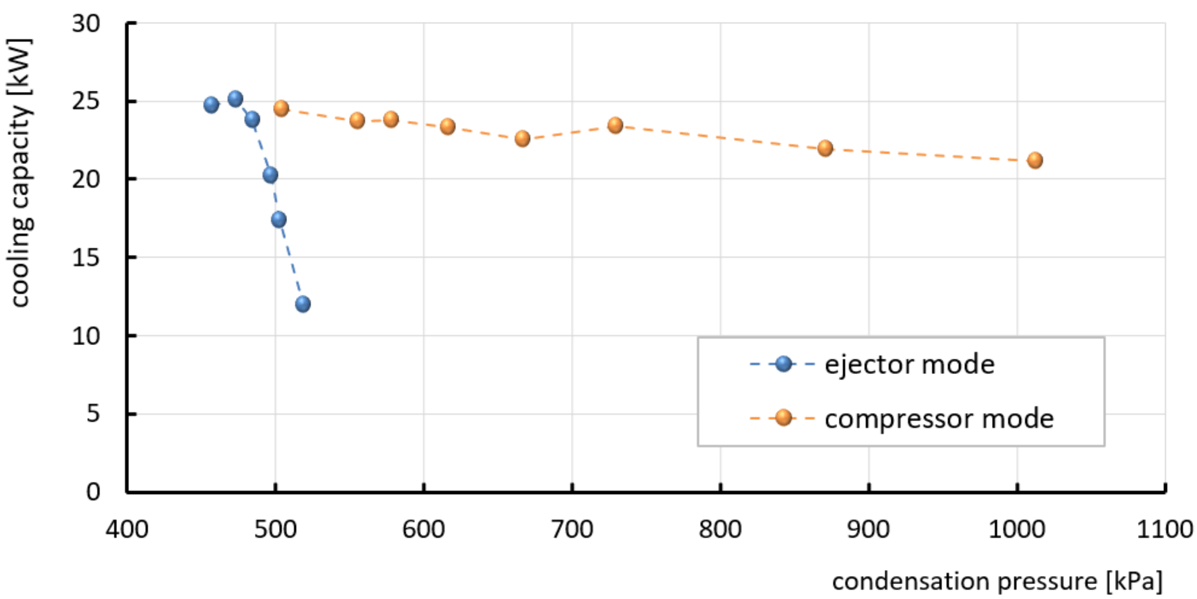
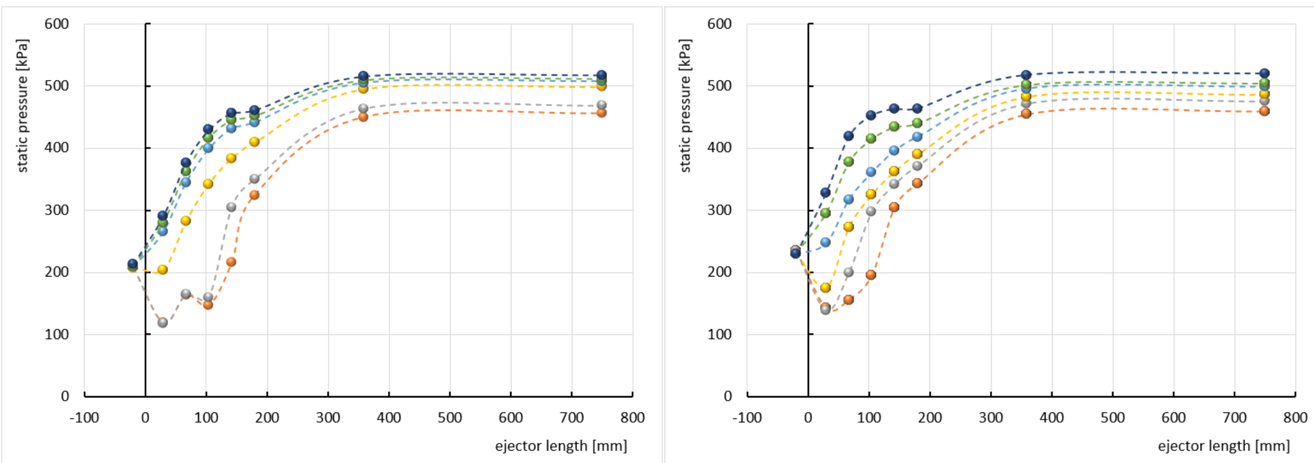


Figure 3: The performance lines for ejector-compressor mode, run No.2

The on-design mode was maintained until the condensation pressure reached 500 kPa, after which the ejector device switched to off-design (non-project) mode, where a clear decrease in the amount of generated cooling was observed as the condensation temperature increased. The condensation pressure pc =500 kPa was the initial temperature for the hybrid system operating in compressor mode. It is worth noting that the compressor produced the same amount of cooling as the ejector operating at pc = 500 kPa. This is a very key conclusion because it indicates that the hybrid system is capable of smoothly switching from ejector mode to compressor mode. Therefore, it can be stated that for the pre-prototype system, the pressure pc = 500 kPa is the bivalence point.It should be emphasized that the validation tests were conducted under the least favorable operating conditions envisaged in the project, i.e., the hybrid system was powered by the source with the lowest temperature while operating under standard cooling conditions, i.e., with evaporation temperatures of the refrigerant in the evaporator at the level of 0-2°C. It is expected that an increase in the source temperature and a switch to medium- or high-temperature cooling will improve the energy efficiency of the system. The pressure distribution along the ejector wall is shown in Fig. 4. It is seen that for the first two operating conditions marked as orange and gray dashed lines and dots the pressure decrese in the initial part of the mixing chamber, and then a sharp increase in pressure is observed. This corresponds to on-design operating regime. The yellow line in the graphs represents the transition operating condition and the green and light blue and dark blue falls within the off-design operating regime.

Figure 4: The pressure distribution along the ejector wall, run No.1 (left), run No.2 (right)



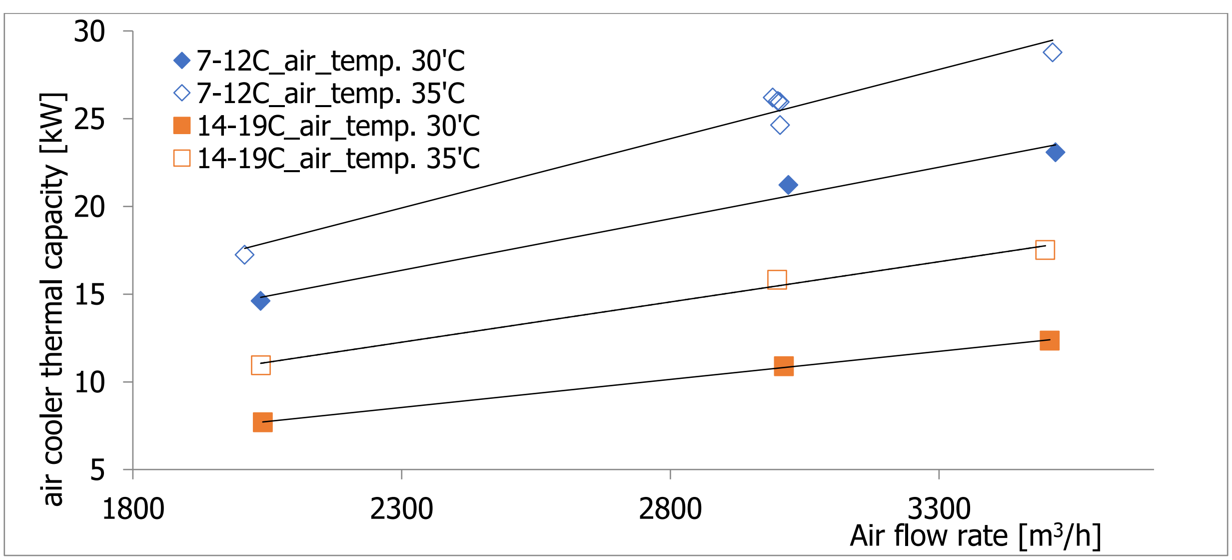


Figure 5: The cooling capacity of air cooler vs air flow rate

Figure 5 presents the exemplary results of investigations performed in the next step, once the hybrid system was applied in AC handlig units and connected to the test tunnel. The suply air temperature was 30°C and 35°C and chille water temperature was set as 7/12°C which corresponds to standards cooling and 14/19°C which is suitable for mid-temperature cooling. As it seen the cooling capacity increases as the air flow increase. The obtained values of thermal capacity of air cooler correspond to those obtained during investigation outside the air tunnel.

3. CONCLUSIONS

Based on the presented results following conclusions can be drawn:

- a series of experimental tests were carried out and the operation of a prototype hybrid refrigeration system was evaluated, for standard cooling operating temperatures
- taking into account the very low driving parameters of the heat source, below 60°C, the results obtained can be considered very promising,
- for the first time, the operating characteristics of a hybrid ejector-compressor system were obtained
- the bivalent operating point was established at tpressure p = 500 kPa
- testing of the hybrid system was carried out in the application to an air conditioning unit

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NOMENCLATURE

c	specific heat capacity (J×kg–1×K–1)	*Q*	heat flux (W)
COP	Coefficient of Performance (-)	*T*	temperature (K)
h	specific entalphy (J×kg–1)	*U*	mass entrainment ratio (-)
m	mass flow rate (kg×s–1)	*v*	specific volume (m3×kg–1)
p	pressure (kPa)	*η*	isentropic efficiency (-)
P	power (W)		

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