

APPLICATION OF LOW-GWP REFRIGERANTS R1233ZDE AND R1234ZEE FOR LARGE SCALE EJECTOR REFRIGERATION SYSTEM OPERATING UNDER REAL INDUSTRIAL CONDITIONS

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ABSTRACT

Application of ejector systems for heat recovery and transform this heat for usable cooling power is one of the most effective and economic way for utilisation of waste heat. This is an exceptionally attractive technological solution for companies with large amount of low-temperature energy sources. The ejector system as one of the few technologies is able to operate with temperature of heat source lower than 80 °C. The paper provides with the results of experimental investigations of the ejector refrigeration system applied in industrial company for recovering waste heat from reciprocating compressors units. The ejector systems have been tested for operation with extremely low GWP and non-flammable R1233zd(E) and R1234zeE refrigerants. Two systems have been studied, one with the capacity of 200 kW and second one with capacity of 600 kW. The system has achieved the entrainment ratio of up to 0.24 which allowed for obtaining up to 45 kW of refrigeration capacity for R1233zdE. In the case of R1234zeE the entrainment ratio up to 0.30 was achieved and cooling capacity of 144 kW was produced. The basics performance lines of the tested refrigeration systems will be presented and discussed in the paper.

Keywords: Refrigeration, low-temperature heat, waste heat utilization, COP,

1. INTRODUCTION

The industrial sector is estimated to account for about one-third of the total energy consumed in the USA (Cavazzini et al., 2019), while in the European Union, this share is approximately 26% (Agathokleous et al., 2019). The EU aims to achieve carbon neutrality by 2050, which can be accomplished by seizing the significant opportunities to enhance energy efficiency and reduce waste heat in industry. Eliminating energy waste is beneficial for industrial companies of all sizes. For instance, electricity represents the largest portion of the total energy cost of compressed air systems, with energy costs comprising nearly 80% of a compressor unit's total cost of ownership over its 10-year life cycle. Over a compressor's lifetime, energy expenses are typically several times higher than the purchase price. Large plants, especially those operating high-capacity compressed air systems 24/7 and requiring process heat, have pioneered heat recovery. These industries include the food and drink, pharmaceutical, and textile sectors, where the scale of operations correlates with greater energy savings. Meanwhile, smaller companies have been transforming or creating spin-offs known as energy service companies (ESCOs), which focus on implementing energy-saving projects, modernizing energy infrastructure, outsourcing energy services, generating and supplying energy, and managing risks. Despite these efforts, the adoption rate of heat recovery systems among small- and medium-sized businesses remains low (Christodoulides et al., 2022), with priorities typically placed on efficiency, reliability, and energy efficiency before heat recovery. Several technical and non-technical barriers exist, with the need for efficient and cost-effective technologies to recover, reuse, upgrade, or transform heat being a significant challenge (European Commission, European Climate 2022). Industries that operate continuously and require process heat benefit the most from heat recovery, which offers substantial advantages across different plant sizes by reducing thermal pollution and greenhouse gas emissions (Soltani et al., 2020). This solution is increasingly popular in various sectors for both space and water heating (Zhang et al., 2021). Recovered heat serves as an additional heat source, decreasing reliance on traditional coal or gas-fired boilers and thus lowering heating fuel costs and carbon dioxide emissions. However, heat recovery

systems need to be tailored to specific needs (Wahlroos et al., 2018). Common technologies for waste heat recovery include various heat pumps (Naldi et al., 2015), ORC systems (Cavazzini et al., 2019), OFC systems (Baccioli et al., 2017), Stirling engines (Laazaar and Boutammachte, 2022), and combined systems (Mohammed et al., 2020). Transcritical CO₂ cycles, including ejector refrigeration cycles, are particularly noteworthy (Yadav et al., 2022). The recently adopted amendment to the Regulation of the European Parliament and EU Council No. 517/2014, known as F-Gas Regulation (EU) 2024/573, encourage industries to phase out HFC refrigerants and develop technology for low-GWP refrigerants (Regulation (EU) 2024).

2. RESEARCH DESCRIPTION

Ejector refrigeration devices have been extensively studied over the past 20 years. Most research focuses on theoretical analyses of ejector cycles rather than experimental studies, especially concerning the use of low GWP refrigerants. The majority of studies indicate that solar radiation or waste heat is typically used as the motive source, with temperatures often at 90°C or higher. However, many industrial waste heat sources have lower temperatures, which are insufficient to directly feed into district heating networks. Therefore, it is crucial to utilize low-grade heat at temperatures below 80°C or even 70°C. Previous studies (J. Gagan et al., 2018) have shown that the ejector refrigeration cycle can be effectively driven by heat sources below 70°C when using refrigerants R600a and R1234ze(E). This paper tests a new refrigerant, R1233zd(E), which has a GWP of 1 and is non-explosive and non-flammable, unlike R600a and R1234ze(E). Refrigerant R1233zd(E) is classified as fully safe (ASHRAE Class A1), whereas R1234ze(E) is classified as A2L (low-flammable), and R600a is classified as A3 (flammable and explosive). Despite the widespread use of R1234ze(E) and R600a, customers might prefer A1 refrigerants, particularly in large thermal capacity systems with substantial refrigerant charges. To the authors' knowledge, only one paper has provided experimental investigation results of an ejector system operating with R1233zd(E) (Mahmoudian et al., 2021), using heat sources at 95°C – 105°C. This study applied the system outside of a laboratory setting, using low and ultra-low grade heat within a temperature range of 53 - 70°C (with a saturation temperature in the vapor generator of 48 – 60°C). Utilizing low-grade waste heat is challenging not only for thermally driven systems but also for other available technologies.

2.1. Testing bench

The experiments were conducted using the system illustrated in Figure 1.

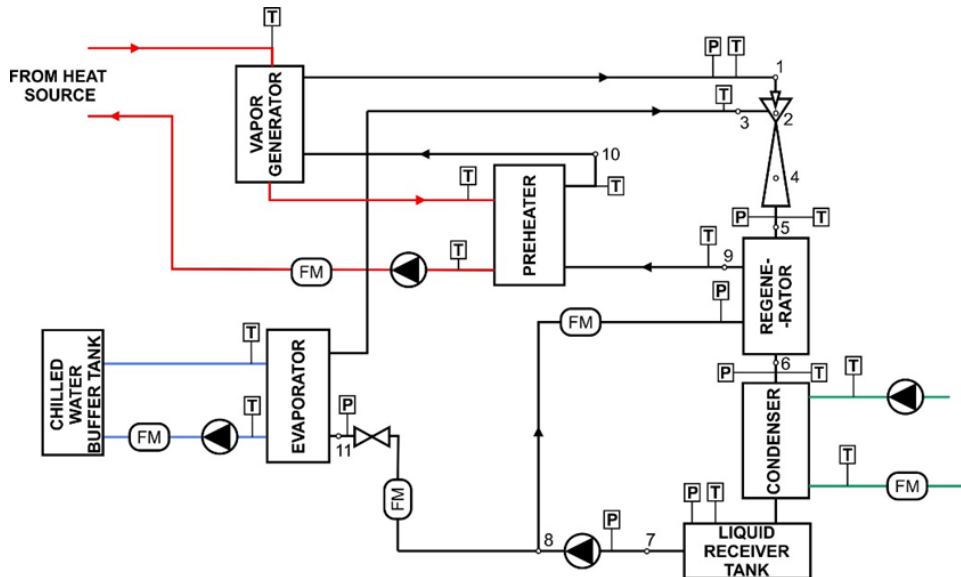


Figure 1: Schematic of tested ejector refrigeration system driven by industrial waste heat

Two independent testing stands were built, one with thermal motive power of 200 kW and the second one with 600 kW of motive heat capacity. These systems were designed and constructed by researchers from Bialystok University of Technology in collaboration with engineers from Marani Ltd. in Zabrze, Poland. The 200 kW testing

bench was implemented at Timken Poland Ltd. in Sosnowiec, Poland, while the 600 kW was investigated at the University. Key geometric parameters of the ejector used in the 200 kW system are: $d_t = 21.6$ mm, $d_m = 76.3$ mm, and $d_d = 156$ mm. Three identical ejectors are combined in one 600 kW capacity ejector system, as shown in Fig.2.

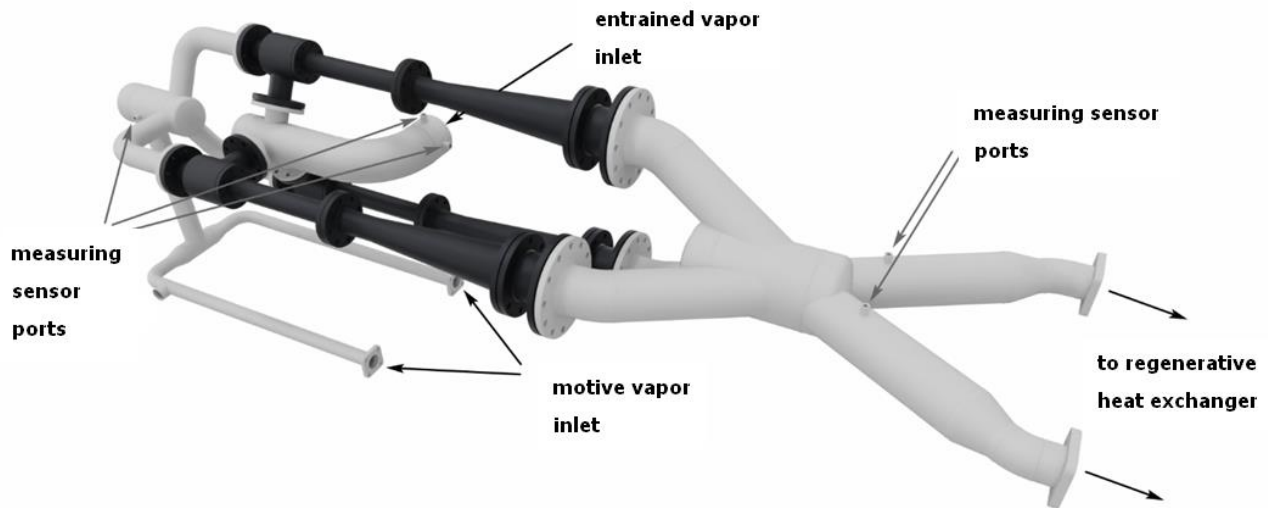


Figure 2: Schematic of tested ejector refrigeration system driven by industrial waste heat

The necessary motive heat and heat sink for the evaporator and condenser are supplied by independently controlled thermal loops, using water in the high-temperature loop and glycol in the low-temperature loop. Parameters of the heat transfer fluid are monitored by mass flow meters and temperature sensors at the inlets and outlets of the heat exchangers. For the larger system, the main source during investigation was oiled-fired boiler operating in such way, that approximately 600 kW is transferred to the ejector system. For smaller system, the motive heat was gathered from three large compressors, each with a 200 kW motive power. These compressors provide constant compressed air for the company's daily operations. Heat generated by these compressors is collected by plate heat exchangers using circulating oil. Under normal conditions, some of this heat is used to heat tap water twice daily, while the rest is rejected to the ambient. To utilize this waste heat, the ejector refrigeration system's motive thermal loop is connected to the compressors' cooling system. The cold produced by the ejector system is stored in a buffer tank and used by the air conditioning system as needed. Due to the variable thermal load required by the air conditioning system, an additional electrical heater was used to stabilize the evaporator's operation during the experiments by providing a controlled artificial heat load. In the condenser cooling loop, water is used as the working fluid, connected to a water system with a cooling tower for all cooling processes. The refrigeration system and ejector were designed to operate with 200 kW of motive heat at an inlet temperature of 70°C. The actual industrial operating conditions, including motive heat power and temperature and condenser cooling conditions, depend on the company's specific and current demands, resulting in varying conditions for the ejector system. Refrigerant circulation is driven by a SEMA liquid pump, powered by a 5.5 kW/1450 rpm electric motor. This multi-stage, magnetically coupled pump combines the functionality of a low net positive suction head first stage impeller with the performance of a side channel design. The low-pressure working fluid and the low pressure difference between the condenser and evaporator can affect the throttling valve's stable operation. To ensure stability and precise control, the circulating pump is placed at the condenser outlet, pressurizing the refrigerant before it is split and delivered to the vapor generator and evaporator through the throttling valve. This configuration ensures the throttling valve operates at a high pressure difference. The system's operation was considered steady-state when no changes in the pressure profile were observed. The plate type heat exchangers were applied in the tested refrigeration system. This includes vapor generator, regenerative heat exchanger, evaporator, and condenser. The regenerative heat exchanger was used to preheat the liquid refrigerant feeding the vapor generator and to precool the superheated refrigerant flowing to the condenser. Such configuration has two positive effects: 1) the thermal load of the condenser is reduced, and 2) motive heat required by the system is also reduced, thus increasing COP (i.e. in this case with the same thermal capacity of waste heat source it is increased refrigeration capacity of the system). The system was equipped with the temperature sensors of accuracy ± 0.25 K and pressure transducers of accuracy 0.25 % of the full measurement range. These sensors are marked in Figure 1 as (T) and (P),

respectively. The mass flow of the refrigerant was measured with accuracy of 0.15% of the measured value by means of Coriolis mass flow meters. The measurements were collected by data acquisition system and stored on the PC computer. Thermodynamic properties of the refrigerant was calculated using NIST REFPROP database.

2.2. Methodology

Ejector system described in this paper was applied for utilisation of waste heat produced by large mechanical air compression unit at the Timken company. The system operates under industrial conditions which depend on the current needs and operating conditions of the company. The analysis of the ejector system operation described in this paper cover presentation of operation parameters and performance lines for each runs. The achieved temperatures and thermal capacity as well as ejector performance in terms of entrainment ratio as well as coefficient of performance are presented. Heat flux transferred by refrigerant is calculated as

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

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 [41]. For water or glycol side the heat flux was calculated as:

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

where temperature difference of water or glycol was taken directly from measurement, and specific heat capacity for water $c_p = 4.186 \text{ kJ kg}^{-1} \text{ K}^{-1}$ and for glycol $c_p = 3.58 \text{ kJ kg}^{-1} \text{ K}^{-1}$ were taken. Average values of these two calculated heat fluxes for each heat exchanger are presented in tables in the next section. The subcooled liquid from refrigerant receiver tank is pressurized by the refrigerant pump. The discharge port of the pump is connected to the regenerator inlet. Both, the pressure and the temperature were measured at the inlet to the pump, therefore the specific volume of liquid refrigerant was found as $v = f(p,T)$. Since only one pressure transducer is mounted at this place, then temperature of liquid at the pump outlet and regenerator inlet is calculated taking the pump efficiency to be $\eta = 0.80$. Power consumed by the pump was calculated as:

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

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 :

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

The coefficient of performance of the system (COP) in general is defined as ratio of the refrigeration capacity divided by the motive power which is the sum of the motive heat and electrical power consumed by the liquid refrigerant pump according to the formula

$$COP = \frac{\dot{Q}_e}{\dot{Q}_g + P_p}. \quad \text{Eq. (5)}$$

2.3. Results

During experimental campaign the 200 kW system was driven by the source operating under approximately 57-59 °C saturation temperature, and the two levels of evaporation temperature were tested, namely standard cooling parameters and mid-/high-temperature cooling. The standard cooling parameters means the evaporation at

approximately 0°C and provides the chilled water with 6/12 °C. Usually this chilled water is used by air collers in AC handling units. The mid-/high-temperature cooling refer to the evaporation at approximately 10°C and provides the chilled water with 16/19 °C. This water can be used in AC systems equipped with active chilled beam. As it was mentioned, the ejector system was applied for utilisation of waste heat generated by large air compression system. The ejector cycle is used for production of refrigeration capacity consumed by the company which temperature of cooling fluid is in the range of 16 – 19 °C. Operation temperatures of the ejector refrigeration system are presented in Table 1. The measurements were performed for various condensation temperatures.

Table1. Average operating parameters for the system with motive heat capacity of 200 kW

Standard cooling, average $t_{e,sat} = -0.2\text{ °C}$						Mid-/high-temp. cooling, average $t_{e,sat} = 11.0\text{ °C}$					
p_g	$t_{g,sat}$	p_e	$t_{e,sat}$	p_c	$t_{c,sat}$	p_g	$t_{g,sat}$	p_e	$t_{e,sat}$	p_c	$t_{c,sat}$
kPa	°C	kPa	°C	kPa	°C	kPa	°C	kPa	°C	kPa	°C
362.8	57.3	48.87	0.4	133.6	25.8	387.5	59.7	77.13	11.2	172.3	33.2
370.0	58.0	48.14	0.0	136.3	26.4	386.2	59.6	76.30	11.0	168.6	32.5
361.4	57.2	47.30	-0.4	137.7	26.7	381.3	59.1	76.08	10.9	163.4	31.6
362.0	57.3	47.77	-0.2	141.6	27.4	379.9	59.0	75.83	10.8	159.2	30.8
367.9	57.8	46.45	-0.8	144.8	28.1	374.1	58.4	76.24	10.9	154.3	29.9
383.1	59.3	48.65	0.2	150.0	29.1	363.4	57.4	75.73	10.8	149.1	28.9
383.8	59.3	47.71	-0.2	151.6	29.4	365.0	57.5	77.56	11.4	146.5	28.4
383.2	59.3	46.47	-0.8	153.6	29.8	366.6	57.7	77.33	11.3	144.3	28.0
						362.9	57.3	76.77	11.1	141.5	27.4
						361.2	57.18	76.38	11.00	139.7	27.07

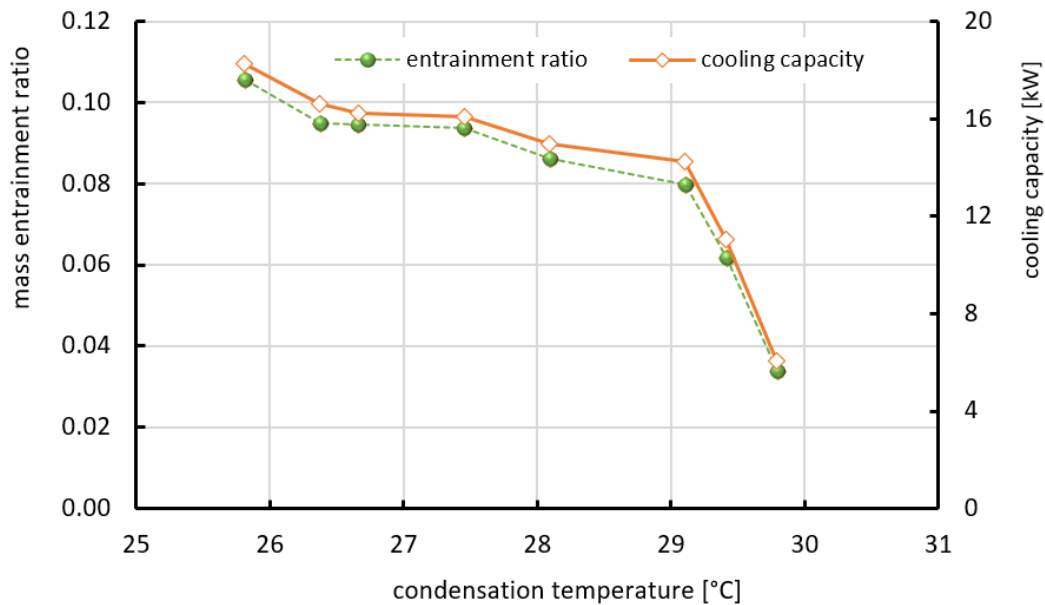


Figure 3: The performance lines for standard cooling

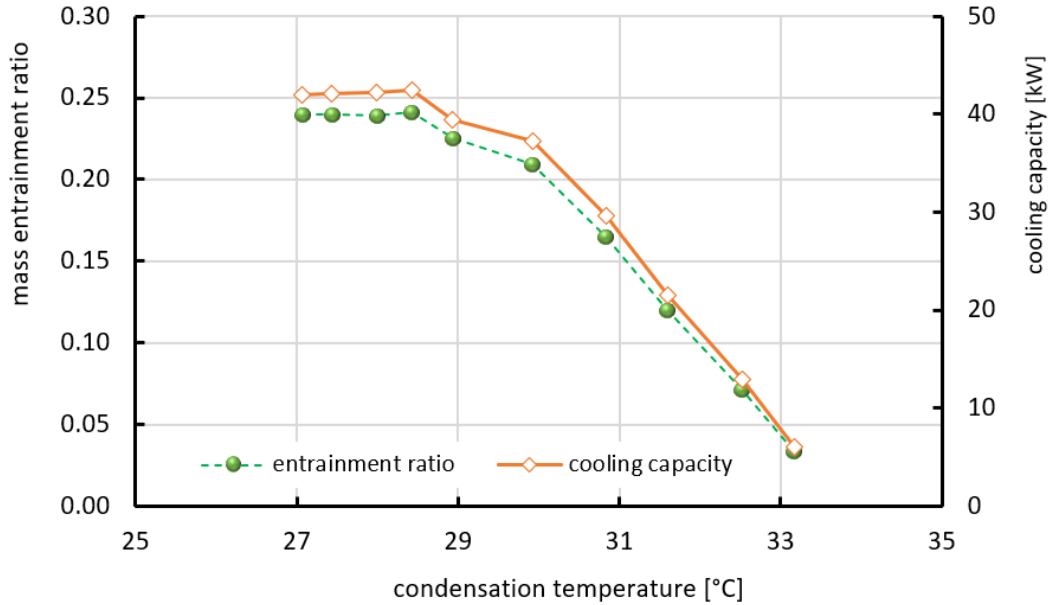


Figure 4: The performance lines for mid-temperature cooling

The ejector system with thermal motive heat capacity 600 kW have been investigated using refrigerant R1234zeE. The saturation temperature in vapor generator was kept at level 50-56 °C and evaporation temperature varied between 3-11°C. The minimum condensation temperature at level 22.5 °C was reported while maximum value was slightly below 30 °C. The variation in temperature in this range allow to investigate the system under on-design and off-design operation regime. Table 2 shows the average operating parameters on the system reported during experiments.

Table2. Average operating parameters for the system with motive heat capacity of 600 kW

p_g	$t_{g,sat}$	t_g	p_e	$t_{e,sat}$	p_c	$t_{c,sat}$	U	Π	COP	Qg	Qe
MPa	°C	°C	MPa	°C	Pa	°C				kW	kW
On-design operation											
1.082	53.3	69.0	0.256	4.7	0.508	25.7	0.155	0.305	0.154	616.8	95.5
1.100	53.9	66.8	0.248	3.8	0.503	25.4	0.212	0.299	0.192	615.8	119.0
1.107	54.2	67.3	0.249	3.9	0.504	25.4	0.221	0.297	0.207	613.8	127.7
1.010	50.5	57.7	0.240	2.9	0.462	22.5	0.254	0.288	0.243	570.5	139.1
Off-design operation											
1.146	55.6	61.0	0.269	6.1	0.555	28.6	0.074	0.326	0.070	616.5	36.9
1.149	55.7	62.8	0.305	9.7	0.572	29.7	0.106	0.316	0.138	620.8	53.2
1.138	55.3	70.1	0.317	10.8	0.574	29.8	0.153	0.313	0.126	627.9	75.8
1.130	55.0	55.2	0.258	4.9	0.540	27.7	0.091	0.324	0.106	612.8	49.1
1.165	56.3	76.6	0.250	4.0	0.543	27.9	0.123	0.320	0.099	645.8	58.8

The results indicate that the tested ejector, applied in a 200 kW refrigeration system, operates at both on-design and off-design conditions. Approximately 29°C is identified as the critical temperature at which the operation switches from double-choked to single-choked mode. For standard cooling, the system is able to produce 16-18 kW of cold, and for mid-temperature cooling, the system produces slightly above 40 kW of cold. As expected, the ejector can operate under higher condensation temperatures for higher evaporator temperatures. In the case of the 600 kW installation, the results indicate that the system can produce up to 140 kW of cold at an evaporation temperature of 3°C and a condensation temperature of 22.5°C, consuming 570 kW of motive heat, which corresponds to a COP of 0.24. An increase in condensation temperature results in an increase in evaporation temperature. Simultaneously, the superheating in the generator varies, affecting the cooling capacity and COP. Considering that the saturation temperature is relatively low compared to typical operating values reported in the literature, which are around 90°C, the obtained cooling capacity and COP can be considered promising.

3. CONCLUSIONS

Based on the presented results following conclusions can be drawn:

- The system with 200 kW of motive heat capacity was driven by recovered waste heat and tested under real operating conditions, while the 600 kW system was tested outside the laboratory. However, the waste motive heat for this system was simulated and controlled by an oil-fueled boiler.
- The refrigerant R1233zd(E) was used in the 200 kW ejector refrigeration system and refrigerant R1234zeE was used in 600 kW ejector refrigeration system. To stabilize the thermal load of the evaporator, additional electric heaters were used to demonstrate achievable refrigeration capacity under the tested conditions.
- The obtained results, in terms of achievable mass entrainment ratio, cooling capacity, and COP of the tested system, indicate that for such a low-temperature heat source, the ejector refrigeration system can effectively operate under full, i.e., on- and off-design operating regimes.

ACKNOWLEDGEMENTS

The research results presented in the paper were partially financed by the National Centre of Research and Development, Contract No. POIR.01.01.01–00–0301/18 and partially by the Project No. WZ/W_WM-IIM2_2023 supported by the Ministry of Science and Higher Education, Poland

NOMENCLATURE

<i>c</i>	specific heat capacity ($J \times kg^{-1} \times K^{-1}$)	<i>Q</i>	Heat flux (W)
COP	Coefficient of Performance (-)	<i>R</i>	molar gas constant ($8.314472 J \times mol^{-1} \times K^{-1}$)
<i>h</i>	Specific enthalpy ($J \times kg^{-1}$)	<i>T</i>	temperature (K)
<i>m</i>	mass flow rate ($kg \times s^{-1}$)	<i>U</i>	Mass entrainment ratio (-)
<i>p</i>	pressure (kPa)	<i>v</i>	Specific volume ($m^3 \times kg^{-1}$)
<i>P</i>	Power (W)	η	Isentropic efficiency (-)

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