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Chapter 7. Innovative technology of conversion of low-grade heat resources for cooling capacity of air-conditioning systems

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Streszczenie. Zrównoważony kierunek rozwoju technologii chłodniczej obejmuje łączenie agregatów strumieniowych zasilanych ciepłem z układami sprężarkowymi napędzanymi elektrycznie w celu efektywniejszego wykorzystania niskotemperaturowych źródeł ciepła. W pracy zostało przeanalizowane zastosowanie bezpiecznych dla środowiska czynników roboczych dla tych układów. Pomimo wyraźnego potencjału takich systemów ich praktyczne wdrażanie jest wciąż stosunkowo rzadkie. W niniejszym rozdziale przedstawiono eksperymentalną weryfikację innowacyjnej hybrydowej instalacji chłodniczej ze strumienicą parową i sprężarką o mocy 23 kW, pracującej w trybie hybrydowym, przeznaczonej do dostarczania wody lodowej o temperaturze 6/12°C do zastosowań klimatyzacyjnych. W zależności od warunków pracy instalacja może pracować w trybie zdominowanym przez podukład strumienicowy lub sprężarkowy, łagodząc znane ograniczenia strumienic przy podwyższonym ciśnieniu skraplania. Testy potwierdziły, że system stale zapewnia chłodzenie, osiągając współczynnik wydajności (COP) na poziomie 0.30 w trybie strumienicowym i średni COP 4,70 w trybie sprężarkowym. Przełączenie robocze – zdefiniowane jako punkt biwalentny – następuje przy temperaturze skraplania wynoszącej około 25°C. Przeprowadzona ocena energetyczna wskazuje na znaczne oszczędności energii podczas pracy w trybie strumienicowym, co dowodzi jego wysokiej efektywności w odpowiednich warunkach. Niniejsze badanie wypełnia lukę między koncepcjami teoretycznymi a praktycznymi rozwiązaniami, zapewniając eksperymentalną ocenę systemu hybrydowego. Uproszczona analiza ekonomiczna dodatkowo potwierdza praktyczność tego rozwiązania, gdyż oszacowano, że ciągła praca wymagałaby około 42 836 kWh energii elektrycznej rocznie. Podsumowując, wyniki dostarczają istotnych wskazówek inżynierom projektującym hybrydowe systemy chłodnicze, których celem jest zmniejszenie wpływu na środowisko przy jednoczesnym zapewnieniu szerokiego zakresu eksploatacyjnego.

Abstract. A sustainable direction in refrigeration technology involves coupling heat-powered ejector units with electrically driven compressors to exploit low-temperature heat sources more effectively. The aspects of environmentally safe working fluid selection have been analysed. Despite the clear potential of such systems, their practical deployment remains relatively rare. This paper reports the experimental verification of a 23 kW dual-mode hybrid vapour ejector–compressor refrigeration installation intended to supply chilled water at 6/12°C for HVAC applications. Depending on operating conditions, the setup can function either in ejector-dominated or compressor-dominated mode, mitigating the well-known constraints of ejectors at elevated condensation pressures. Tests confirmed that the system consistently delivers cooling, reaching a coefficient of performance (COP) of 0.30 in ejector operation and an average COP of 4.70 during compressor operation. The operational switch – defined as the bivalent point – appears at a condensation temperature of roughly 25°C. The performed energy assessment indicates notable energy savings when operating in ejector mode, demonstrating its high effectiveness under suitable conditions. This study narrows the gap between theoretical concepts and practical realization by providing an experimental assessment of the hybrid system. A simplified economic analysis further confirms the practicality of the solution, estimating that continuous operation would require about 42,836 kWh of electricity per year. Overall, the findings deliver essential guidance for engineers designing hybrid refrigeration systems aimed at lowering environmental impact while ensuring a wide operational range.

Słowa kluczowe: strumienica, czynniki o niskim wskaźniku GWP, hybrydowe układy chłodnicze, COP, efektywność energetyczna, praca biwalentna

Keywords: ejector, low-GWP refrigerant, hybrid refrigeration system, COP, energy efficiency, bivalent operation

Introduction

The worldwide shift toward more energy-conscious and environmentally responsible technologies has accelerated developments in refrigeration science. Conventional vapour compression units – although reliable and highly developed – rely on mechanical compression, making them electricity-intensive and environmentally burdensome [1]. These systems compress the refrigerant mechanically and remove heat through condensation, but this process comes at the cost of substantial electrical input and related greenhouse-gas consequences [2].

Ejector-based refrigeration, on the other hand, uses a high-pressure stream to entrain and compress a secondary low-pressure fluid. As they contain no moving components, ejectors represent durable and mechanically simple devices [3]. Their advantage becomes clear when low-temperature heat sources – such as waste heat – are available. Studies such as [4] discussed the integration of ejectors with other thermodynamic cycles in maritime energy systems, illustrating their compactness and efficiency. Shipboard air-conditioning applications were investigated in [5], showing meaningful reductions in operating costs and environmental impact. Solar-assisted ejector systems have also been demonstrated as effective [6]. As industrial processes often reject large amounts of thermal energy, they represent natural application areas for ejector-driven recovery solutions [7]. Numerical investigations of combined ejector-compressor setups have revealed their promise for improving system efficiency. Performance enhancements via compression-supported cycles were examined in [8], whereas in [9] solar-coupled configurations for urban cooling were explored. However, these contributions remain primarily numerical and typically do not verify their results through experiments. This work addresses this gap by experimentally analyzing separate ejector and compressor operation within a hybrid configuration to determine real-world working boundaries and evaluate achievable energy savings.

The performance of ejector-only systems is inherently sensitive to outdoor conditions, especially ambient temperature. As the condensation pressure increases, the cooling capacity drops significantly. Ejectors have therefore been studied extensively over the past decades [10]. Their geometry, thermodynamic behaviour, operating constraints, and refrigerant choices have all been explored through theory, simulations, and experiments. In [11], for example, a compound ejector cycle powered by both waste heat and solar energy was proposed, achieving notable COP improvements with low-GWP refrigerants such as R450A. In [12], ejector-compression concepts were adapted for vehicle cabin cooling using engine waste heat and low-GWP mediums like R1234yf, reporting substantial increases in COP and capacity. Similar efforts across many industrial branches confirm the potential of ejector technologies for reducing power consumption and operating expenses [13]. It was shown that solar-supported ejector units can enhance heat pump drying [14]. Adjustable-geometry ejectors [15] have demonstrated improved COP relative to fixed designs. In [16], solar installations with additional ejectors for expansion work recovery were assessed. In marine applications, a dual-loop ejector configuration for engine waste heat recovery was developed [17], improving fuel use and lowering CO_2 emissions. Flue-gas heat-powered air-conditioning systems for ships were assessed in [18]. In [19], a combined ejector–compressor solution for refrigerated ship compartments was introduced. Other researchers used ejectors to enhance hybrid or auxiliary systems: solar-powered ejector-desalination units were evaluated in [20]; ejectors were applied in [21] for battery temperature control with reduced compressor power; expansion-energy recovery ejectors in solar cycles were studied [16]; in [22] it was showed that multi-ejector systems in commercial refrigeration outperform parallel compression; in [23] ORC-supported refrigeration was improved with a multi-ejector control strategy; in [24] solar and air heat sources were combined in a dual-source heat pump with ejector enhancement.

The term "hybrid" encompasses a wide spectrum of system architectures, from CO_2 ejector-assisted booster arrangements to two-stage and expansion-energy-recovery configurations. Strong gains when merging a transcritical CO_2 cycle with an ejector were identified [25], especially with R32 as the refrigerant. In [26], refrigeration and power generation in a CO_2 ejector cycle were combined, improving energy efficiency through optimized back-pressure control. Two-stage ejector cycles were studied in [27]. Two-phase ejectors were reviewed in [28], confirming their useful role in cooling and heat pump systems. More complex cycles – such as cogeneration or cascade

– have also adopted ejectors. Efficiency improvements in cogeneration systems through ejector were demonstrated in [29]. ORC-supported ejector refrigeration in partial thermal storage conditions was evaluated in [30]. Solar ejector systems using R290 were optimized in [31]. Ejectors have also been explored directly as expansion-valve substitutes, improving overall cycle COP [22].

The innovative aspect of the system studied in this work lies in its ability to alternate between ejector-based and compressor-based operation depending on real-time condensation pressure and ambient conditions. The analogy to bivalent heat pumps is intentional: just as heat pumps rely on auxiliary heat sources under unfavourable weather, this hybrid unit supplements a heat-driven ejector with a mechanical compressor whenever condensation pressure becomes excessive. At low condensation temperatures, the ejector performs the vapour compression using only low-grade heat, reducing the load on the electrical compressor and improving overall efficiency. When outdoor temperatures rise and condensation pressure exceeds allowable limits, the compressor assumes full responsibility for maintaining cooling performance. This operational flexibility is particularly useful in environments with large temperature swings, where ejectors alone cannot manage peak loads.

The bivalent point – where system operation must switch from ejector to compressor – plays a central role in defining this strategy. In heat pumps, the bivalent point is the outdoor temperature at which auxiliary heating becomes necessary. Analogously, in the hybrid cycle studied here, the bivalent point corresponds to a condensation pressure beyond which the ejector can no longer deliver the required cooling. Past this threshold, compressor operation ensures uninterrupted air-conditioning performance.

From a fluid-dynamic perspective, the ejector may function in double-choked (on-design) or single-choked (off-design) regimes. Under double choking, the entrainment ratio remains constant despite variations in back-pressure. Once back-pressure surpasses a specific transition value, entrainment decreases. This transition pressure corresponds directly to the bivalent point. Designing an ejector with a higher entrainment ratio shifts the transition point to lower pressures, while lower entrainment ratios result in higher allowable back-pressures. The designer can select the optimal trade-off depending on local climate conditions. In the hybrid system, the ejector effectively reduces compressor energy consumption when conditions are favourable, while the compressor provides robustness when ejector operation becomes infeasible. To the authors' knowledge, such an approach has not been widely documented previously.

This research provides experimental results of the evaluation of a hybrid ejector–compressor refrigeration device capable of switching between two fully independent operating regimes. The ejector is intended for periods of low condensation pressure, whereas the compressor maintains operation at high condensation pressure. This capability allows for continuous air-conditioning performance without sacrificing efficiency. Prior publications have primarily focused on numerical simulations of multi-stage, cascade, or advanced absorption–ejector cycles. While useful, such analyses do not account for real-world limitations. In contrast, the present work confronts these practical challenges, including flow stability, mode switching, and the identification of the actual bivalent point. Despite overlapping terminology with previous studies, the experimental method, emphasis on practical operation, and dual-mode functionality distinguish this system sharply from earlier concepts. The results confirm the feasibility of integrating ejector and compressor functionalities into a single device and open pathways for future developments, such as advanced control strategies for dynamic mode switching.

Other thermally driven technologies (absorption, adsorption) cannot form bivalent systems of the type examined here. An ejector system uniquely allows for a seamlessly embedded compressor that can be activated as needed through a suitable control algorithm. This architecture ensures uninterrupted cooling delivery regardless of the state of the motive heat source, the district heating network, or off-design conditions in the heating loop. Under unfavourable ejector performance, the compressor cover ensures stable operation.

The hybrid configuration can be supplied by various heat sources depending on application. Three potential heat sources for the presented system include: (a) district heating networks, (b) solar thermal collectors, and (c) waste heat streams. Ejector refrigeration cycles involve three characteristic temperature levels. For typical use cases, the following temperature ranges are relevant: (a) motive heat: 55 to 120°C, (b) evaporation: 0 to16°C, (c) condensation: 20 to 40°C. A key advantage of ejector technology is its ability to operate on low-grade heat that cannot drive absorption or adsorption units. However, ejector performance is highly sensitive to condensation temperature, particularly at the lower end of motive-heat availability.

District heating typically offers the least favourable heat source conditions, especially during summer, with supply temperatures usually below 65°C due to recent EU regulations [32]. While some networks still operate at higher temperatures, ongoing modernization efforts aim at lowering them progressively. Although no strict physical constraints prevent using lower motive temperatures, energy efficiency inevitably decreases. Waste heat can vary widely but often exceeds 70°C. Solar-collector systems can reach temperatures above 120°C, though exceeding the refrigerant's critical temperature pushes the cycle into supercritical operation.

Standards such as [33, 34], and [35] establish expected chilled-water temperatures (6/12°C) for typical HVAC systems, implying evaporating temperatures of 0–3°C. For high-temperature cooling (14/17°C or 14/19°C), the evaporating temperature may be 8–10°C. Condensing temperatures are generally expected to fall between 20 and 40°C.

Although district heating represents the most challenging heat source, using its surplus capacity for summertime cooling is increasingly important in modern energy systems. For that reason, the present study focuses on operating conditions representative of district-heating temperatures. The experimental results presented later include performance curves enabling estimation of COP, cooling capacity, motive-heat consumption, and feasible working ranges.

To verify assumptions about the hybrid system's bivalent nature and performance boundaries, experiments were conducted on the developed test stand. Details of the research procedure are provided.

Working fluid selection for ejector refrigeration systems

The efficiency of an ejector depends on numerous factors, including inlet and outlet pressures and temperatures, thermo-physical properties of the working fluid, and the ejector geometry. A considerable body of numerical and experimental work has explored ejector performance under various working fluids and boundary conditions. Among these factors, fluid selection remains critical due to its strong influence on cycle efficiency. Equally important are the operating parameters of the cycle. Since ejector systems use thermal energy as the driving force, they are often compared with absorption units. In absorption cycles, the generator temperature $t_{(g)}$ typically exceeds 100°C. When the generator temperature decreases, cycle performance rapidly deteriorates due to diminishing concentration differences between the strong and weak solutions. As a result, vapour generation temperature $t_g < 80°C$ is generally considered the lower practical limit for absorption operation.

In contrast, most working fluids used in ejector systems do not require generator temperatures near 100°C, and there is no inherent physical barrier preventing ejector operation at much lower temperatures. Therefore, ejector cycles operating below $t_g < 80°C$ become especially attractive and, in this range, may compete effectively with absorption units [36]. For low-temperature heat sources, organic fluids offer clear advantages over water due to the significantly lower specific volume of their vapour, which becomes important in compact system designs.

Another aspect of fluid selection is the suitability of a refrigerant for a given ejector cycle configuration, which depends on the shape of its saturation dome and saturation pressure level. Most low-boiling fluids exhibit a "bell-shaped" saturation curve in the T–s diagram, with substances such as water or ammonia forming nearly symmetric liquid and vapour lines due to their high latent heat. However, several low-boiling synthetic refrigerants display highly asymmetric vapour saturation curves.

Based on the slope of the saturated vapour curve, refrigerants can be categorised as wet fluids (with a negatively sloped vapour saturation line); isentropic fluids (whose vapour saturation line is nearly vertical); and dry fluids, characterised by a positively sloped vapour saturation line. The most preferred selection for the ejector refrigeration systems are dry fluids.

The supersonic ejector performance with various working fluids has been widely investigated [37, 38]. Numerous theoretical and experimental studies have analysed the behaviour of ejectors operating with different refrigerants [39, 40, 41]. Literature findings consistently indicate that isobutane (R600a) is one of the most advantageous choices for ejector applications [37, 42]. Isobutane is inexpensive, well-characterised, and already used in many practical systems, with the primary drawback being its flammability and potential explosiveness. Therefore, safety considerations must be taken into account when designing ejector refrigeration systems using isobutane. The most critical precaution is minimising the total refrigerant charge, which substantially enhances safety without altering the fundamental operating characteristics. Proper system tightness and the use of safety valves compliant with EN 378 are also essential.

Current regulation concerning fluorinated refrigerants [43] effectively bans the use of new refrigerants with a GWP above 150 in most refrigeration applications. This significantly narrows the range of workable candidates for AC-R and heat-pump systems. Consequently, increasing attention is given to the new class of synthetic HFO refrigerants, particularly R1234, which has become available on the European market [44]. Since these fluids are still relatively unfamiliar, their key thermodynamic properties are compared here with well-known refrigerants of similar characteristics: isobutane (R600a), the discontinued HCFC R123, and the widely used R134a (see Fig. 1–3).

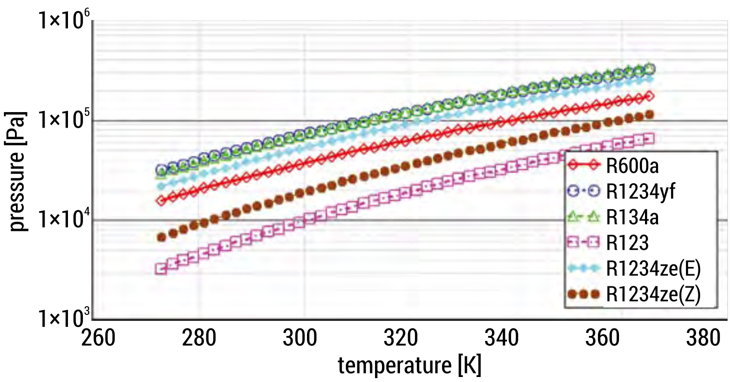

FIGURE 1. Saturation pressure versus temperature for selected environmentally friendly working fluids

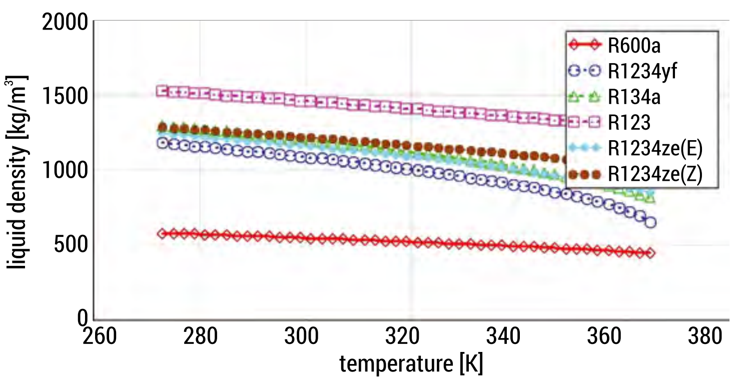

FIGURE 2. Liquid density versus temperature for selected environmentally friendly working fluids

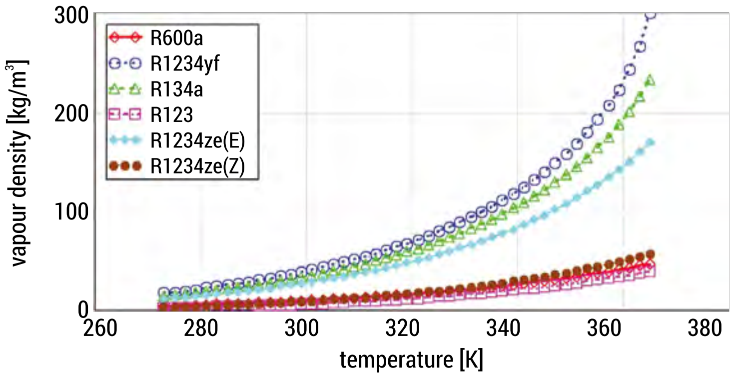

FIGURE 3. Vapour density vs temperature for selected environmentally friendly working fluids

Analysis of the presented diagrams leads to the following observations:

- All examined fluids show similar saturation-curve shapes, particularly on the vapour side, meaning they all behave as "dry" refrigerants in ejector applications.
- Saturation pressures of R1234ze(Z) and R123 are lower than those of R600a; in turn, R600a exhibits lower saturation pressures than R1234ze(E), R1234yf, and R134a.
- Liquid density is lowest for isobutane and highest for R123; refrigerants from the R1234 family display liquid densities comparable to each other and close to R134a.
- Vapour densities rise with temperature for all fluids, but the magnitude of this change separates them into two groups: R1234yf and R134a show the largest increase, while R123, R1234ze(Z), and R600a display much smaller variations. R1234ze(E) falls between these two extremes.

The outcomes of the numerical analysis for the selected refrigerants are presented in the figures that follow. The examined set includes both synthetic fluids (R-123, R-134a, and the R-1234 family) and a natural refrigerant (isobutane). It also spans representatives of "dry" fluids (isobutane, R-123, R-1234 types) as well as one "wet" refrigerant (R-134a). It should be emphasized that R-123 is an HCFC, and its use in newly installed systems is prohibited in the United States, the European Union, and numerous other countries. For the remaining fluids, no dedicated restrictions exist beyond the general limitations on HFC emissions defined in the European F-Gas Regulation [43].

Analysis of the results indicates that, across a broad spectrum of operating conditions, all investigated refrigerants provide comparable cycle efficiencies. Each of them can therefore be regarded as an attractive candidate relative to other fluids commonly referenced in the literature. It is worth noting that ejector-cycle performance data for R-1234 refrigerants have not yet been reported in published studies. Based on the characteristics illustrated in Fig. 1–3 and Fig. 4 and Fig. 5, refrigerant R-123 would appear to be the most favourable option, as it ensures similar cycle performance while operating at reduced pressures and lower vapour densities. Nonetheless, R-123 cannot be treated as a viable choice because of its non-zero ODP (ODP > 0), which led to its phase-out.

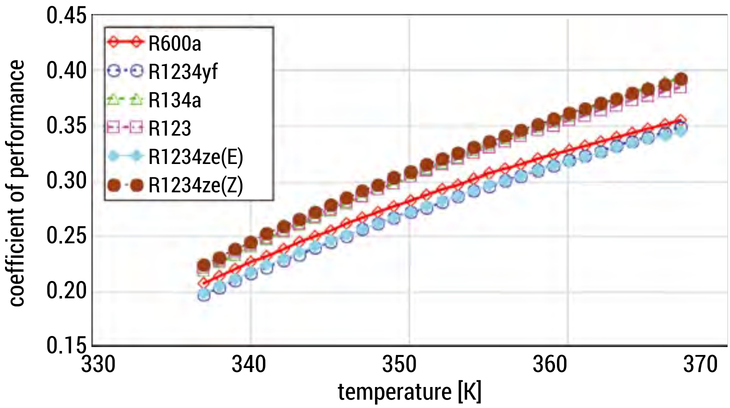

FIGURE 4. COP versus vapour generation temperature for evaporation temperature T_e = 278 K and condensation temperature T_c = 301 K

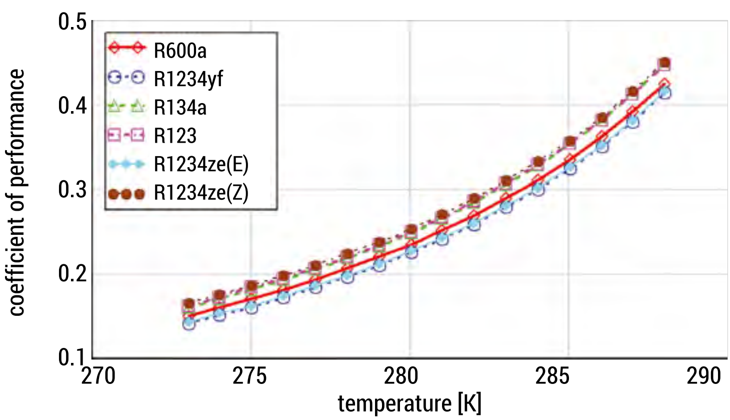

FIGURE 5. COP versus evaporation temperature for vapour generation temperature T_g = 333 K and condensation temperature T_c = 301 K

Refrigerant R-134a is also being discontinued due to its high GWP of 1400. The closest drop-in successor is R-1234yf, already implemented in automotive air-conditioning systems, although it is mildly flammable. Previous work by the authors demonstrated that isobutane can operate effectively in ejector refrigeration systems. Its primary advantage is that it is a natural refrigerant with negligible environmental impact, and for systems with very small refrigerant charges – similar to household refrigerators widely used across Europe – its flammability presents no significant hazard. In contrast, for larger-capacity installations, the explosive nature of isobutane may raise safety concerns.

The above discussed limitations motivated the search for another low-GWP refrigerant appropriate for ejector-cycle applications. Based on the comparative results presented, R-1234ze(E) emerges as a promising synthetic alternative. It offers operating characteristics and achievable cooling capacities that are broadly comparable to those obtained with isobutane, making it a strong candidate for further development in ejector refrigeration systems.

Testing bench

The experimental campaign was performed on the setup presented in Fig. 6. Photographs of the hybrid refrigeration unit and the wind tunnel used during the tests are shown in Fig. 7. The working fluid throughout the tests was R1234ze(E), a refrigerant selected in the previous section of this paper and characterised by extremely low GWP and widely regarded in the European Union as an environmentally acceptable option [44]. The main elements of the installation included a vapour generator, condenser, evaporator, compressor, refrigerant pump, regenerative heat exchanger as well as throttling, regulating, and shut-off valves. The system layout was selected so that stable refrigeration operation could be maintained across the full spectrum of operating conditions, irrespective of variations in motive heat supply or ambient temperatures.

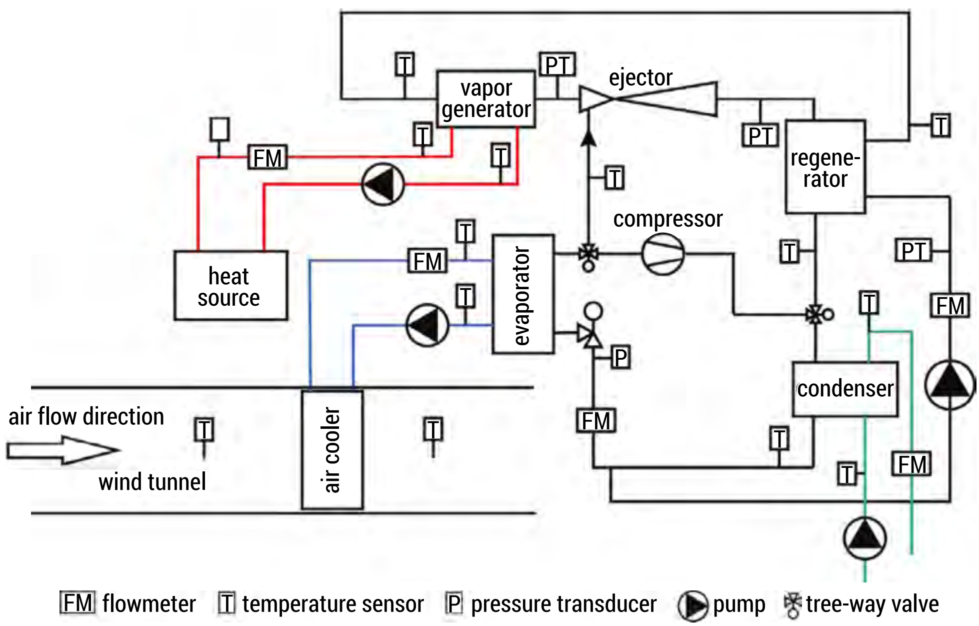

FIGURE 6. Schematic of the tested hybrid ejector-compressor refrigeration system

FIGURE 7. **Photo of the tested hybrid refrigeration unit**

When the unit operated in ejector mode, the thermal energy delivered by the heat source (e.g., district heating network) was supplied to the vapour generator, where the working fluid evaporated under elevated pressure. The resulting vapour – often superheated – was directed into the ejector, expanding through the motive nozzle to very low pressure and high velocity, typically reaching supersonic values. The pressure at the motive nozzle outlet remained below that in the evaporator, and the combined effect of this pressure difference and the high momentum of the motive stream caused vapour suction from the evaporator. Mixing and subsequent compression of the motive and entrained streams took place within the ejector body. At the ejector outlet, the medium was still a superheated vapour, enabling efficient heat transfer in the regenerator, where this excess heat was passed to the liquid entering the vapour generator. After the regenerator, the vapour entered the condenser and was liquefied. The condensed refrigerant was then divided: one fraction was routed through a circulation pump to the regenerator and subsequently to the vapour generator, while the remainder passed through the expansion valve to the evaporator. In compressor mode, the heat-supply loop and the ejector were disabled, and the installation worked as a conventional vapour-compression system.

The supersonic ejector was instrumented with paired temperature and pressure sensors located in the suction chamber, mixing chamber, and diffuser. The placement of RTD temperature sensors and pressure transducers is indicated in Fig. 6. The measurement accuracies were as follows: 0.25% for pressure transducers, 0.20% for RTD sensors, and 0.15% for Coriolis mass-flow meters. These values relate to the instrument readings and ensured precise monitoring of the thermal and flow parameters during operation. Measurement uncertainties were determined using the total derivative method. All sensors and measurement lines were calibrated prior to each test run. The data acquisition system continuously recorded all measurements, which were then stored on a PC. Only one flow meter was installed downstream of the ejector; the mass flow rate through the motive nozzle was evaluated using gas-dynamic relations, since under critical (choked) conditions the flow becomes dependent solely on upstream pressure and temperature. This method offers an accuracy of approximately ±1% [45] while also minimizing additional flow resistance. The accuracy parameters for the measurement equipment are summarized in Table 1.

TABLE 1. Accuracy of experimental equipment

Measurement Parameter	Sensor Type	Accuracy	Notes
Pressure	Pressure transducers	±0.25% of reading	Located in suction, mixing, and diffuser
Temperature	RTD sensors	±0.20% of reading	Calibrated before each run
Mass flow rate	Coriolis mass flow meters	±0.15% of reading	Single flow meter downstream of ejector
Electric power	Three-phase, Three-wire Active Power Transducer	±0.5% of reading	Compliant with EN 60688 and EN 61010-1 standards

Three auxiliary loops were integrated into the experimental rig: one for applying the thermal load to the evaporator, one for condenser cooling, and one delivering heat to the vapour generator. Each loop contained a PROMASS Coriolis meter

and temperature sensors installed at the inlet and outlet of the respective heat exchangers, enabling wide-ranging control of refrigerant flow rate and operating conditions. The evaporator load system used glycol as the heat-transfer medium, whereas water served as the working fluid in both the condenser cooling loop and the generator heating loop. The cooling loop was connected to both the evaporator and an air cooler positioned inside the wind tunnel. An automatically regulated dry cooler ensured stable condenser cooling, while the motive-heat loop incorporated a 100 kW electrically powered heater with automatic control. Control valves from CAREL and DANFOSS allowed fine adjustment of system parameters. A Hydra-Cell diaphragm pump driven by a 3 kW/700 RPM motor circulated the liquid refrigerant. The hybrid system used a Frascold Q7-28.1-AXH compressor. The generator, condenser, and evaporator were built using plate heat exchangers from SWEP and SECESPOL, and a regenerative plate heat exchanger was installed between the ejector outlet and the condenser. The wind tunnel was equipped with components enabling simulation of an air-handling unit; however, as most were not used for the purposes of this study, the diagram in Fig. 6 shows only the functional elements. Airflow rate and air temperatures (fresh and processed air) were also measured.

Results of Experimental Investigations

Two full experimental campaigns were executed, each consisting of a sequence of operating points. For every point, measurements were taken only after the system reached a steady thermal and flow regime – defined as fluctuations limited to the magnitude of inherent measurement noise rather than changes caused by the system approaching thermodynamic equilibrium. Once stability was confirmed, over one hundred data samples were recorded at 5-second intervals. These samples were subsequently averaged into a single representative data point. After adjusting the operating conditions, particularly by increasing the condensation pressure, the same procedure was repeated to obtain the next averaged point. This ensured a sufficiently robust dataset for evaluating the system performance. In ejector mode, each of the two measurement series contained six averaged points. The aggregated results are provided in Tables 2–4. An analogous methodology was applied in the compressor-mode investigations. These tests began at elevated condensation temperatures, where the ejector was no longer capable of delivering sufficient cooling performance. In this mode, run Series 1 yielded 10 averaged points, while Series 2 produced 8 points.

For run No. 1, the typical operating conditions were: $t_{g,sat}$ = 59.0°C, superheat Δt_g = 3–5 K, and $t_{e,sat}$ = –1.0°C. For run No. 2, the averaged values were: $t_{g,sat}$ = 57.5°C, superheat Δt_g = 7–9 K, and $t_{e,sat}$ = 3.0°C. In both campaigns, condensation temperature was treated as the primary variable. Temperatures of the heat-transfer fluids entering and exiting the heat exchangers are listed in Table 2), where HW and CW correspond to hot-water and chilled-water flows. These chilled-water values reflect the actual stable measurements obtained during controlled operation.

TABLE 2. Operation parameters for the tested ejector system

Heat carrier fluid (hot water)		Chilled water		Heat carrier fluid (hot water)		Chilled water	
t_{HWin}	t_{HWout}	t_{CWin}	t_{CWout}	t_{HWin}	t_{HWout}	t_{CWin}	t_{CWout}
°C	°C	°C	°C	°C	°C	°C	°C
run No.1				run No.2			
64.7	59.9	13.9	5.5	66.5	61.6	13.7	4.4
63.8	59.1	14.0	5.5	66.3	61.4	13.5	4.4
63.4	58.8	14.2	5.9	65.9	61.1	13.5	4.0
63.0	58.3	14.5	8.0	65.3	60.5	13.5	3.8
62.9	58.3	13.4	7.0	64.3	59.7	13.2	3.7
63.0	58.4	13.7	7.8	63.3	58.8	13.0	5.6

The necessary thermal and flow variables – including temperatures, pressures, and mass flow rates – were recorded and subsequently used to determine specific enthalpies and specific volume. These quantities served as inputs to the heat-flux calculations. The results are shown in Table 3 and Table 4. The symbol definitions for quantities used in the above tables are as follow: p_c – condensation pressure; Q_g – thermal heating load; Q_e – cooling capacity; W – power consumed by liquid pump; m_g – mass flow rate of primary (motive) fluid; m_e – mass flow rate of secondary (entrained) fluid; *U* – mass entrainment ratio; *COP* – Coefficient of Performance.

TABLE 3. Mean operating parameters for run No. 1.

p_c	Q_g	Q_e	W	m_g	m_e	U	COP	p_c	Q_e	W	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	–	–	–	–
465.9	92.9	24.4	0.43	0.466	0.112	0.241	0.262	–	–	–	–
496.6	91.1	23.8	0.41	0.463	0.111	0.241	0.260	500.3	24.4	5.2	4.70
505.9	90.9	12.8	0.38	0.461	0.079	0.171	0.141	527.7	24.7	5.2	4.78
510.4	90.5	10.3	0.36	0.461	0.057	0.124	0.113	569.7	24.0	6.1	3.90
515.9	90.1	7.8	0.35	0.461	0.050	0.108	0.086	601.0	23.6	6.0	3.94
–	–	–	–	–	–	–	–	614.9	22.6	7.6	2.97
–	–	–	–	–	–	–	–	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

TABLE 4. Mean operating parameters for run No. 2.

p_c	Q_g	Q_e	W	m_g	m_e	U	COP	p_c	Q_e	W	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	–	–	–	–
473.5	95.6	25.4	0.40	0.441	0.146	0.330	0.265	–	–	–	–
484.4	94.4	25.1	0.39	0.445	0.136	0.307	0.265	–	–	–	–
496.8	93.0	22.6	0.38	0.448	0.113	0.252	0.242	–	–	–	–
502.6	91.1	17.9	0.37	0.450	0.096	0.214	0.196	503,9	24.4	2.3	10.53
518.6	89.3	10.8	0.36	0.459	0.067	0.146	0.121	555.8	23.7	4.3	5.57
–	–	–	–	–	–	–	–	578.3	23.8	2.8	8.50
–	–	–	–	–	–	–	–	616.3	23.3	2.5	9.36
–	–	–	–	–	–	–	–	667.2	22.5	2.7	8.31
–	–	–	–	–	–	–	–	729.8	25.8	2.4	9.64
–	–	–	–	–	–	–	–	871.3	21.9	3.4	6.38
–	–	–	–	–	–	–	–	1012.8	21.2	3.0	7.07

The experimental results demonstrate that the hybrid refrigeration system successfully utilized motive heat at temperatures slightly above 60°C to generate cooling. In ejector mode, approximately 25 kW of cooling output was produced using about 100 kW of heating power. Differences between heat-flux values on the hot and cold sides of the heat exchangers were within 9% for the generator, 13% for the evaporator, and 12% for the condenser. As expected, pump power consumption remained below 1% of the supplied heating capacity, confirming that the pumping power can legitimately be neglected when assessing the COP of the ejector cycle. Evaporator saturation temperatures of 0–2°C correspond to typical chilled-water air-conditioning conditions. All COP and cooling-capacity values reported in this study are based directly on controlled experimental measurements, ensuring their reliability.

Performance curves for both ejector and compressor modes are illustrated in Fig. 8 and Fig. 9. The initial objective was to determine the point at which the ejector transitions into off-design operation. The slope of this off-design segment is particularly important: a steep slope suggests a narrow stable operating window, whereas a gentler slope indicates a more forgiving ejector performance. This behaviour determines when the system should switch to compressor mode. The slope characteristics depend on working-fluid thermo-physical properties as well as the ejector geometry.

In both test series, the ejector exhibited strong sensitivity to variations in condensation pressure under off-design conditions, resulting in steep performance curves. Consequently, the compressor mode should be activated as soon as the entrainment ratio shows a declining trend. The observed relationships between condensation pressure and cooling capacity align with core thermodynamic principles, confirming the robustness of the test results. Compression-mode tests began when condenser pressure was slightly below the transition point, and the measurements were collected once this threshold was reached. The resulting curves clearly depict the operating domains of both ejector and compressor modes.

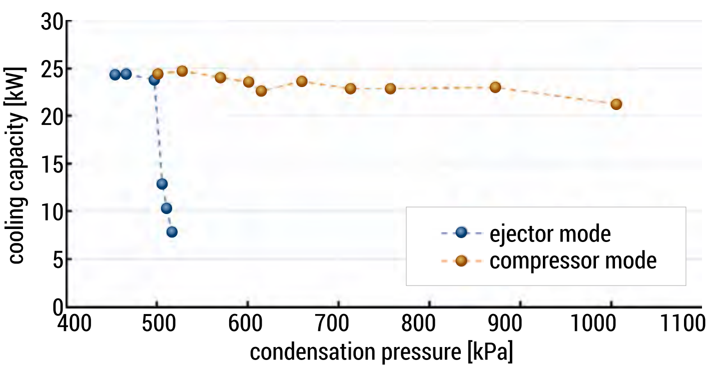

FIGURE 8. The tested hybrid ejector-compressor refrigeration system performance lines for ejector and compressor modes, run No. 1

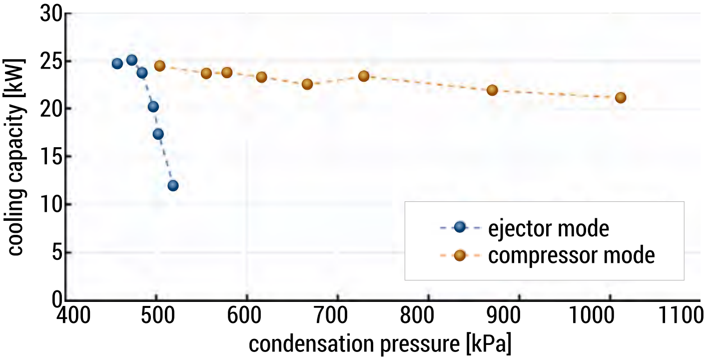

FIGURE 9. The tested hybrid ejector-compressor refrigeration system performance lines for ejector and compressor modes, run No. 2

The results highlight the contrasting characteristics of the two modes. For the ejector, cooling capacity rapidly declines once condensation pressure surpasses ~500 kPa, reflecting high sensitivity to increasing condensation temperature. Conversely, the compressor mode remains stable across a much broader pressure range, reliably delivering cooling even when the ejector becomes ineffective. This demonstrates that the hybrid concept effectively leverages the strengths of each mode, enabling efficient adaptation to variable thermal conditions.

The ejector operated in on-design mode until pc ≈ 500 kPa, beyond which the system entered off-design behaviour characterized by a noticeable reduction in cooling capacity. At pc = 500 kPa, both the ejector and compressor produced comparable cooling output, which is essential for seamless switching between modes without affecting user comfort. No additional system adjustments were required during mode transitions. Therefore, for the tested prototype, pc ≈ 500 kPa (corresponding to ~25°C condensation temperature) can be identified as the bivalence point under these conditions.

The ability to operate with low-temperature heat sources – such as district heating water – demonstrates strong applicability in moderate climates. The bivalence pressure serves as a guideline for designing hybrid systems and selecting optimal control strategies. Substantial energy-saving potential exists in regions where outdoor temperature remains below 25°C for most of the year. In Central Europe, annual mean temperatures of 7–10°C are typical, with only 20–50 days above 25°C. As a result, air-conditioning systems may operate in ejector mode for the vast majority of the year, which is particularly advantageous in facilities requiring 24/7 cooling (data centres, pharmaceutical production, electronics manufacturing, hospitals). It is important to emphasize that the presented validation tests were performed under the least favourable expected conditions: lowest motive-heat temperature and standard cooling loads with evaporator saturation of 0–2°C. Higher motive-heat temperatures or medium-temperature cooling applications would yield even higher efficiencies.

The additional test covered investigations of the system performance connected to a wind-tunnel test facility, enabling variation of the thermal load imposed on the air cooler by adjusting airflow rate. The results are summarized in Table 5.

TABLE 5. **Results of operation of the bivalent hybrid system applied to the wind tunnel**

	V_{Air}	t_{HWin}	t_{HWout}	t_{CWin}	t_{CWout}	$t_{g.sat}$	$t_{e.sat}$	$t_{c.sat}$	$t_{Air.in}$	$t_{Air.out}$	Q_e	Q_g	U	COP
	m^3/h	°C	°C	°C	°C	°C	°C	°C	°C	°C	kW	kW		
E	2000	71.9	67.4	12.9	8.0	59.1	6.7	26.5	34.8	15.7	28.6	91.3	0.39	0.31
E	3000	71.5	67.2	12.5	8.1	59.6	6.5	26.8	33.8	16.4	27.9	84.8	0.36	0.33
C	3000	–	–	13.0	8.1	–	3.3	24.4	34.2	16.5	22.6	–	–=	7.00
E	3500	68.4	64.4	11.8	7.3	60.0	5.6	26.8	33.3	16.3	26.0	81.9	0.34	0.32

The airflow rate was regulated by adjusting the fan rotational speed, with tests performed at 2000, 3000, and 3500 m^3/h. For nearly all conditions, the system operated in ejector mode (denoted as E in Table 5), except for the 3000 m^3/h case, where compressor mode (C) was deliberately activated. The goal was to achieve conventional air-conditioning conditions, namely chilled-water temperatures of 6/12°C. Fresh-air temperature entering the tunnel ranged from 33–35°C. The system maintained evaporator saturation temperatures near 6°C. Chilled-water inlet temperature remained 12–13°C, with a typical 5 K temperature drop resulting in outlet temperatures near 8°C. Maintaining these conditions required generator saturation temperatures around 60°C and heating-water supply temperatures of approximately 70 ± 2°C. The outlet air temperature from the tunnel was around 16°C, which matches typical supply-air temperatures used in comfort-cooling systems. For indoor comfort levels of 22–23°C, supply air of 15–18°C is required. During these tests, condensation temperature was close to 26.5°C, corresponding to a saturation pressure of ~520 kPa.

In the ejector mode, cooling capacity ranged from 26–29 kW, with motive-heat input of 82–91 kW, yielding a COP of approximately 0.30. The recorded mass-entrainment ratios (0.34–0.39) are very favourable, indicating efficient ejector operation. Since generator saturation temperature matched earlier tests and evaporator saturation temperature was slightly higher – beneficial for ejector stability – the ejector likely operated close to its on-design regime.

In the compressor mode (C), stable operation was observed once evaporator temperature decreased from 6°C to 3°C, causing a roughly 4 kW reduction in cooling capacity. No noticeable changes in chilled-water or outlet-air temperatures were recorded.

Compared to earlier theoretical studies on hybrid systems, the present investigations provide with experimental validation under real operating conditions. While simulations have suggested strong potential for ejector-compressor systems, these results provide tangible evidence and highlight practical constraints – particularly the narrow operating range of ejectors at elevated condensation pressures.

Assessment of Energy Savings for Hybrid Operation

The use of the proposed hybrid system may offer meaningful reductions in energy consumption. A detailed analysis has been provided in [46]. This analysis is based solely on the experimental results and can be extended to wider operating ranges. The approach assumes continuous operation and considers only the electrical power demand of the compressor; auxiliary electrical demands (control systems, displays, fans, etc.) are neglected. It is assumed that the cooling capacity is 23 kW and the average COP in compressor mode is 4.70. If the compressor is needed only when outdoor temperature exceeds 25°C, which is assumed as 45 days per year (representative for Central Europe), then annual energy consumption is 5 281 kWh/a, and energy savings are 37 555 kWh/a. On this basis there could be assessed percentage savings of 87.8%.

These savings consider only the reduced electricity use of the compressor. Regarding investment costs, the hybrid system differs from a conventional vapour-compression system only by the addition of a vapour generator, an ejector, and a circulation pump. A modification using a two-phase ejector instead of a mechanical pump is also under development [47], although still at laboratory scale. Additional benefits stem from the ability to utilize waste heat or other low-grade heat sources.

The energy-savings analysis follows a continuous-operation scenario typical for applications such as data centres, where uninterrupted cooling is essential. Systems in the 25 kW cooling-capacity range are common in medium-scale facilities, including hospital operating rooms, pharmaceutical storage and manufacturing areas, food-processing facilities, and various industrial environments requiring stable thermal control.

This scenario illustrates the upper bound of annual savings while emphasizing the strong energy-efficiency potential of the hybrid concept. Although additional operational scenarios could be assessed, the continuous-load case demonstrates system performance under the most demanding conditions. The presented configuration is particularly well suited for applications utilizing low-grade heat sources (district heating, industrial waste heat). Although reversible heat pumps combined with PV systems are an alternative, their performance strongly depends on solar conditions. In contrast, ejector-driven systems deliver stable cooling when integrated with consistent thermal sources. The results clearly show the strong potential of the hybrid system in terms of energy efficiency and practical applicability across diverse climatic conditions. The main conclusions and future research directions follow below.

Conclusions

This study presents the first experimental confirmation of the operating performance of a bivalent hybrid ejector–compressor refrigeration system. The measurements show that the system achieved a COP of 0.30 in ejector mode and 4.70 in compressor mode. Under the conditions examined, the hybrid concept offers the potential for energy savings approaching 80%, making it an attractive option for continuous-duty applications such as industrial cooling, hospital environments, and data centres. From the experimental campaign, the following key conclusions can be formulated:

- Refrigerant R1234ze(E) has been selected as the most suitable for application in ejector and hybrid ejector-compressor refrigeration systems utilising low grade heat resources.
- A comprehensive series of laboratory experiments was completed, assessing the performance of a prototype bivalent hybrid ejector–compressor refrigeration system operating under temperature levels representative for typical air-conditioning systems.
- Despite the relatively low temperature of the motive heat source, the obtained results should be regarded as highly encouraging.
- For the first time, operating characteristics of this type of hybrid configuration have been experimentally established.
- The bivalent point for refrigerant R1234ze(E) was identified at approximately 500 kPa, corresponding to a condensation temperature of 25°C and motive-heat temperature near 65°C.

The findings presented in this paper offer a representative dataset describing the behaviour of a bivalent hybrid ejector–compressor system driven by low-grade heat sources relevant to air-conditioning applications, such as district heating networks. Although the experiments confirm key performance trends and highlight the system energy-saving potential, additional research is planned. Future work will extend the analysis to a broader spectrum of operating conditions, including other heat sources and alternative working parameters. These efforts aim to expand the current understanding of the hybrid system capabilities and strengthen the basis for its practical implementation.

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