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Abstract

In this research, an experimental investigation was conducted on a dual-evaporator ejector system (DEES). The experiments were conducted under two distinct configurations, known as condenser outlet split (COS) and diffuser outlet split (DOS), across varying compressor inlet pres-sures. The system was initially operated in accordance with the COS configuration followed by operation under the DOS configuration. The comparison revealed a 9% reduction in the compressor work within the DOS configuration relative to the COS configuration. Evaporator#2 cooling capacity was 14% higher in the DOS compared to the COS. Moreover, the total cooling capacity achieved in the COS mode exhibited a 16% increase in comparison to the DOS mode. Furthermore, research findings indicate that by operating the DEES in the DOS configuration, full refrigerant separation can be achieved, leading to enhanced operational efficiency.

Keywords: Ejector, diffuser, dual evaporator, refrigeration system

INTRODUCTION

In recent years, research on the utilization of ejectors in refrigeration systems has seen a surge in interest due to their straightforward design, cost-effectiveness, and durability. By integrating ejectors, the efficiency of vapor compression refrigeration systems can be enhanced (1). Since the liquid-vapor separator, used in standard ejector refrigeration systems, has a maximum efficiency of 85%, it reduces the efficiency of the system. To mitigate these losses, researchers suggest replacing the separator with a second evaporator in vapor compression refrigeration systems employing ejectors (2) This modification not only enhances cooling capacity but also minimizes efficiency losses. Determining the separation of refrigerant in refrigeration systems with dual evaporators is crucial (3). Literature on systems with dual evaporators and an ejector typically recommends splitting the refrigerant flow at the condenser outlet to optimize performance (4). In dual evaporators (dual temperatures) refrigeration systems, it is observed that the ejector provides a beneficial impact on performance (5). Recent studies, such as Fan et al. (6), have investigated the performance of solar-assisted ejector compression heat pump cycles using R290/R600a refrigerants for water heating. These studies have demonstrated significant enhancements in coefficient of performance (COP) and heating capacity - up to 33% and 47%, respectively - compared to traditional vapor compression heat pump cycles. Śmierciew et al. (7) experimentally evaluated an ejector-based refrigeration system using isobutene. Their findings indicated that a higher quality at the nozzle inlet led to improved agreement between the proposed model's mass flow rates and the experimental data. Similarly, Tahir Erdinç et al., (8) evaluated the performance of an ejector heat pump system. They found a 22.6% increase in COP with ejector utilization. Çalışkan and Ersoy (9) found that incorporating an ejector in a dual evaporator CO2 system increases the COP value by up to 47%. In the investigation conducted by Işkan et al. (10), the comparative analysis of the performance of R134a and the alternative refrigerant R516A within a DEES was examined. The experimental results revealed that R134a exhibited superior cooling capacity compared to R516A, leading to an elevation in the COP ranging between 1% and 5% when R134a was adopted in the system. Furthermore, it was deduced that there existed a positive correlation between the performance parameters and increasing air velocities. However, air velocities exceeding 2.2 m/s exhibited a diminishing effect on system performance. Vaibhav Jain et al. (4) conducted an analysis on a DEES operating under the COS configuration whereby varying condenser water temperatures were employed. Their study revealed an escalating trend in the entrainment ratio, rising from 0.396 to 0.701 as the condenser water temperature decreased. In a separate study, Fingas et al. (11) performed an experimental assessment on the efficiency of an ejector within a bi-evaporator ejector heat pump system utilizing R290 as the refrigerant, with experiments being conducted according to the COS mode. The results of their investigations indicated a notable enhancement in the heating COP of the system by up to 38% when compared to a direct expansion system. Ünal et al. (12) examined the performance of R1234yf, R1234ze(E), and R600a refrigerants in a DEES refrigeration system. It was observed that the type of refrigerant had a significant effect on the size of the ejector. In the COS configuration, the refrigerant undergoes division prior to its arrival at the ejector. This partitioning serves to diminish the quantity of refrigerant that enters the primary inlet of the ejector. With a reduction in mass flow rate at the primary inlet of the ejector, there is a corresponding decline in the velocity increase. Lower refrigerant velocity leads to a reduction in mass flow rate at the ejector outlet. An alternative approach to address this issue involves separating the refrigerant after the diffuser outlet, as proposed by Lawrance and Elbel (13). By implementing this separation, all refrigerant leaving the condenser is directed to the primary inlet of the ejector, with the objective of elevating the rate of velocity enhancement resulting from the pressure drop. Consequently, the strategy entails the operation of a system featuring dual evaporators, wherein the refrigerant is divided into two streams subsequent to the diffuser outlet and distributed to the first and second evaporators. This approach is designed to optimize the efficiency of the ejector system.

Prior research has primarily focused on the COS configuration, while theoretical investigations on the DOS layout have been limited. To address this research gap, the present study examines the performance of DEES operating in both COS and DOS configurations. Experimental tests were conducted using R134a as the refrigerant, with results compared through graphical analysis. The experiments were repeated at varying compressor inlet pressures to evaluate the influence of compressor inlet pressure on system performance.

EXPERIMENTAL SETUP

The experimental setup has been implemented in two distinct configurations, namely the condenser outlet split (COS) and diffuser outlet split (DOS). The evaporators within the system have been positioned according to the DOS and COS configurations. The system incorporates two plate heat exchangers identified as Evaporator#1 and Evaporator#2, in addition to a tubular welded heat exchanger denoted as Evaporator#3. Moreover, there is integration of a hermetic type compressor, condenser, ejector, and various auxiliary components. As illustrated in Figure 1, the experimental setup comprises a single refrigeration cycle along with two water cycles. The water utilized in Evaporator#1 and Evaporator#2 functions to initiate the evaporation of the refrigerant. Within the water cycles, pumps have been installed to elevate the water pressure within the tanks. In the DOS configuration, the separator component positioned downstream of the ejector has been employed to segregate the refrigerant into dual streams. The connection between all elements within the system has been established through insulated copper piping. The COS configuration encompasses evaporators, with the schematic representation of the refrigerant flow paths elucidated in Figure 1a. Similarly, the DOS configuration entails evaporators, with a visual depiction of the refrigerant pathways demonstrated in Figure 1b. Subsequent to this, Figure 2 indicates the complete experimental setup, whereas Figure 3a illustrates the configuration of the ejector and the subsequent separation process. Additionally, Figure 3b showcases the position of the evaporators in the DOS mode. The designated points in Fig. 1 and Fig. 2 are introduced in Table 1.

Table 1 The designated points in Fig. 1 and Fig. 2

No	COS / DOS configurations
1	Inlet of compressor
2	Outlet of compressor
3	Outlet of condenser
4	Inlet of ejector/Outlet of ejector's nozzle
5	Second inlet of ejector/ Suction chambers of ejector
6	Mixing chamber
7	Inlet of evaporator#3/ Inlet of evaporator#1
8	Outlet of evaporator#3/ Outlet of evaporator#1
9	Outlet of diffuser

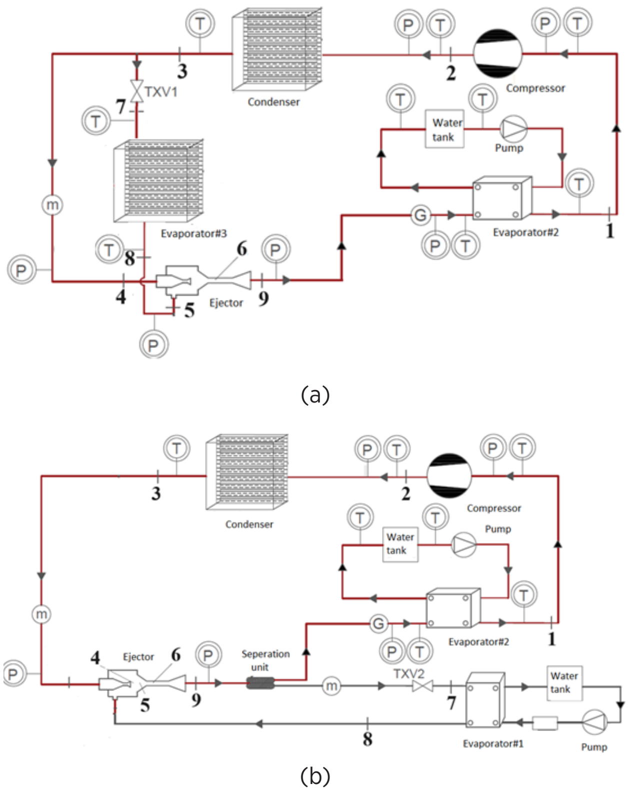
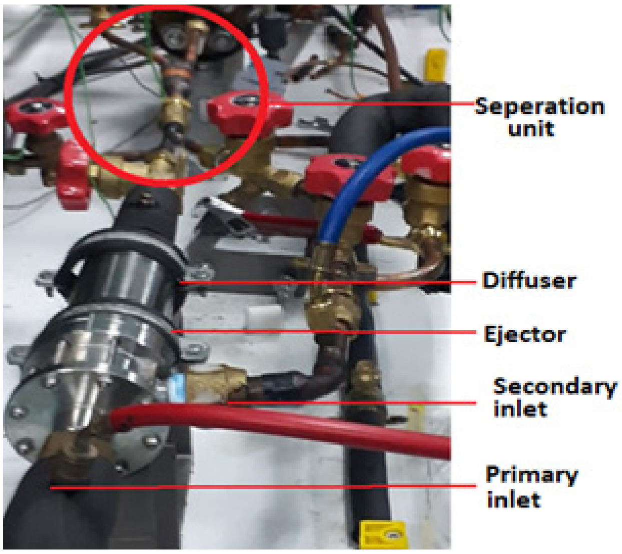


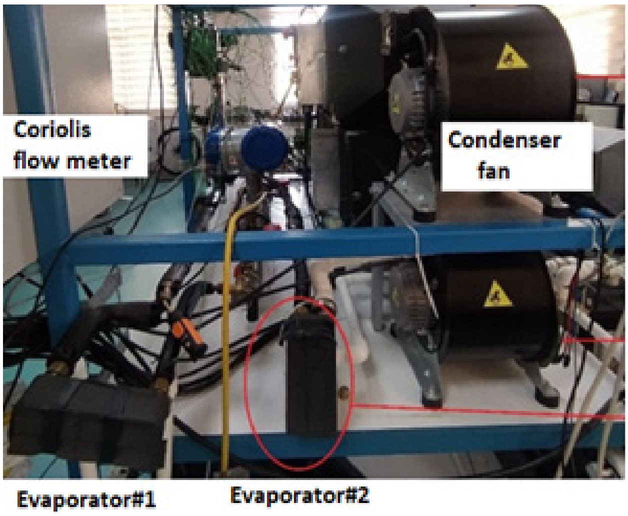
Figure 1 a) COS and b) DOS configurations flow diagram

In the DOS configuration, the refrigerant from the condenser is directed to the primary inlet of the ejector. Upon entry at the first inlet of the ejector, the fluid undergoes an increase in velocity as a result of nozzle constriction, leading to a pressure reduction and subsequent intake into the suction chamber of the ejector. Subsequently, the refrigerant passing through the TXV undergoes isenthalpic throttling at the ejector outlet. This process culminates in a pressure reduction prior to the entry into evaporator#1, initiating the initial stage of evaporation within evaporator#1. The refrigerant is then sucked into the secondary inlet of the ejector under a vacuum, where it commingles with the refrigerant from the primary inlet. Subsequently, the blended streams within the suction chamber are propelled into the mixing chamber under consistent pressure. As the refrigerant, existing in a liquid-gas phase with elevated velocity, decelerates within the divergent diffuser, the pressure is boosted owing to the diverging geometry of the diffuser. The refrigerant, present in the liquid-gas phase upon exiting the diffuser, fully transitions into the gaseous phase within evaporator#2. It subsequently proceeds into the compressor for pressurization, before being routed to the condenser, thereby concluding the operational cycle.

Figure 2 Experimental setup and measurement devices



(a)



(b)

Figure 3 a) Diffusor and separation process b) The position of the evaporators in the DOS mode

Thermodynamic Equations and Diagrams

Figure 4 illustrates the pressure-enthalpy (P-h) diagrams for the DOS and COS configurations. Equations have been derived based on designated points and are provided in Table 2.

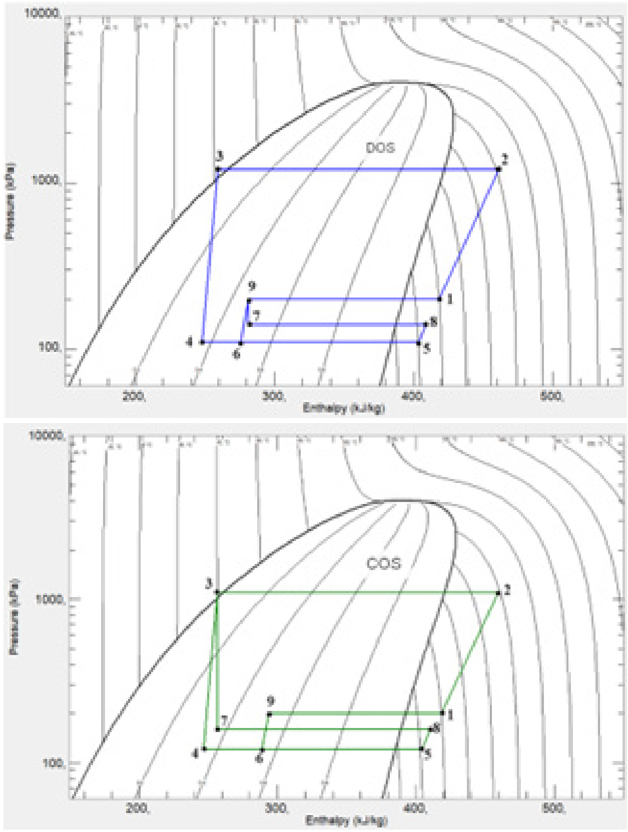


Figure 4 P-h diagrams of DOS and COS configurations

Table 2 Thermodynamic Equations

	COS	DOS
Compressor power	$\dot{W}_{comp} = (h_2 - h_1).\dot{m}_1$	$\dot{W}_{comp} = (h_2 - h_1).\dot{m}_{total}$
Condenser capacity	$\dot{Q}_{cond} = (h_2 - h_3).\dot{m}_1$	$\dot{Q}_{cond} = (h_2 - h_3).\dot{m}_{total}$
Evaporator#1 cooling capacity	$\dot{Q}_{evap\#1} = (h_2 - h_3).\dot{m}_1$	-
Mass flow rate	$\dot{m}_{total} = (\dot{m}_1 + \dot{m}_2)$	$\dot{m}_{total} = (\dot{m}_1 + \dot{m}_2)$
Evaporator#2 cooling capacity (water)	$\dot{Q}_{evap\#2(water)} = \dot{m}_{water}.c_{p(water)}.\Delta T$	$\dot{Q}_{evap\#2(water)} = \dot{m}_{water}.c_{p(water)}.\Delta T$
Evaporator#3 cooling capacity	-	$\dot{Q}_{evap\#3} = (h_8 - h_7).\dot{m}_2$
H evap1 inlet	$h_9 = h_7$	-
H evap2 inlet	$h_9 = h_1 - (\dot{Q}_{evap\#2(water)} / \dot{m}_1)$	$h_9 = h_1 - (\dot{Q}_{evap\#2(water)} / \dot{m}_1)$
H evap3 inlet	-	$h_3 = h_7$
Total cooling capacity	$\dot{Q}_{total} = \dot{Q}_{evap\#1} + \dot{Q}_{evap\#2}$	$\dot{Q}_{total} = \dot{Q}_{evap\#1} + \dot{Q}_{evap\#2}$

The setup utilizes two distinct types of flow meters: a turbine-type flow meter installed at the outlet of the condenser, and a Coriolis-type flow meter installed between evaporator#1 and the separation unit at the diffuser outlet. In conventional ejector refrigeration cycles, a distinct separator is employed for the separation of liquid and vapor phases. However, in this particular system, the task of liquid-vapor separation is undertaken by the second evaporator. As depicted in Figure 1b, the refrigerant pathways and system components are delineated during the operation of the ejector refrigeration system in the DOS configuration. Furthermore, Table 3 details the specifications of the experimental equipment, while Table 4 outlines the specifications of the measurement devices utilized.

Table 3 The component characteristics of experimental setup

Component	Type	Characteristics
Evaporator#1	Air	Heat Transfer Coefficient: 52 Wm^2K^{-1}, Sensible Heat Rate: 1
Evaporator#2	Water	Heat transfer surface: 0,5m^2 # of plate: 24
Evaporator#3	Water	Heat transfer surface: 0,216m^2 , # of plate 50
Compressor	Hermetic, variable speed drive	Danfoss MTZ 022-4 b 2.9 kW, 380-400 V, 50 Hz, 2900 rpm
TXV 1-2	Externally equalized	

Table 4 Specifications of the measurement devices utilized in experimental setup

Parameters	Component	Accuracy	Measurement Range
Temperature	K-type thermocouple	± % 0.8	-100 – 1370 ºC
Pressure	Electromagnetic manifold	± % 0.5	-1 – 60 bar
Pressure	Pressure transmitter	± % 0.5	4 – 20 mA
Pressure	Bourdon manometer	± % 0.5	-1 – 55 bar
Air Velocity	Anemometer	± % 2	0 – 30 m s^{-1}
Water volumetric flow rate	Electromagnetic flowmeter	± % 0.3	0 – 1 m^3s^{-1}
Refrigerant mass flow rate	Coriolis mass flow rate	± % 0.1	0 – 5 kg m s^{-1}
Compressor Frequency	Frequency inverter	± % 0.2	10-50 Hz

Throughout the experiment, certain parameters were held constant, including the compressor inlet pressure, water inlet temperature, condenser air velocity, and compressor frequency. The specific values of these parameters are delineated in Table 5.

Table 5 Constant values of test while varying the compressor inlet pressures

Compressor inlet pressure (kPa)	170 – 230 (by 10 kPa steps)
Evaporator#2 – water inlet temperature (°C)	20-21
Evaporator#1 -air inlet temperature(°C)	25-26
Evaporator#2 water mass flow rate (kg/s)	0,2
Compressor frequency (Hz)	50

RESULTS AND DISCUSSIONS

This section examines the results achieved during the operation of the system in both COS and DOS configurations, while varying the compressor inlet pressures. Pressure and temperature values obtained from the experiments are presented in Table 6. An examination of Table 6 reveals higher compressor outlet pressure and condenser outlet temperatures in the DOS configuration compared to the COS configuration.

Figure 5 illustrates the changes in compressor compression ratios while varying compressor inlet pressures. The analysis of Figure 5 reveals a negative correlation between compression ratios and increasing compressor inlet pressures. Through experimental observations, it was noted that the compression ratios for both configurations exhibited a decrease of approximately 20% as the compressor inlet pressures increased from 170 kPa to 230 kPa. This trend can be attributed to the maintenance of consistent condensing temperatures as the compressor inlet pressures elevate. Consequently, the compression ratio exhibited an inversely proportional relationship with the compressor inlet pressure due to the constant nature of condenser water temperature and condensing pressure.

Furthermore, the compression ratio was found to be higher in the DOS configuration compared to the COS configuration. Specifically, at a compressor inlet pressure of 190 kPa, the compression ratio was calculated to be 4.13 for COS and 5.03 for DOS configuration, indicating a 22% increase. This disparity was linked to varying refrigerant mass flow rates in the two configurations. The DOS configuration, characterized by lower mass flow rates, yielded a higher compression ratio compared to the COS configuration. Even the total refrigerant volume remained constant in both configurations, a greater mass flow rate of refrigerant passed through the compressor in the COS configuration. This discrepancy stemmed from the separation of refrigerant leaving the ejector in DOS configuration, where some refrigerant was directed to evaporator #1.

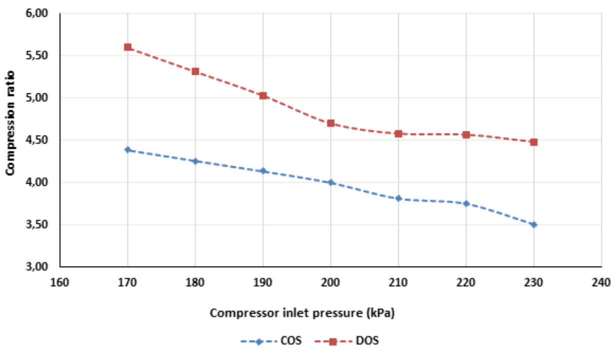


Figure 5 The change of compression ratio depending on compressor inlet pressure

yielded lower compressor power values compared to the COS configuration. For instance, at a compressor inlet pressure of 180 kPa, the calculated compressor power was 0.57 kW for COS and 0.52 kW for DOS, marking a 9% reduction. Upon an assessment of Figures 5 and 6, it is apparent that decreasing compressor inlet pressures resulted in higher

Table 6 Experimental results for DOS and COS configurations

Compressor inlet pressure (kPa)	Configuration	Compressor outlet pressure (kPa)	Condenser outlet temperature (°C)	Evaporator#2 outlet temperature (°C)
170	DOS	985	27.9	25.1
	COS	779	26.4	22
180	DOS	967	29.4	25.1
	COS	786	26.6	22.2
190	DOS	986	31.1	25
	COS	793	26.7	22
200	DOS	940	33.2	24
	COS	800	26.7	22.1
210	DOS	943	34.3	26
	COS	812	26.2	22.1
220	DOS	1001	36.5	24.9
	COS	829	26.6	22
230	DOS	1008	36.3	24.6
	COS	800	29	22.2

compression ratios but lower compressor power values in the DOS configuration. This phenomenon indicates that the DOS configuration achieved elevated condenser temperatures with reduced compressor power values. This feature can be leveraged for greater heating capacity when the system operates as a heat pump, as the DOS system functions at elevated condenser water temperatures while consuming lower compressor power values.

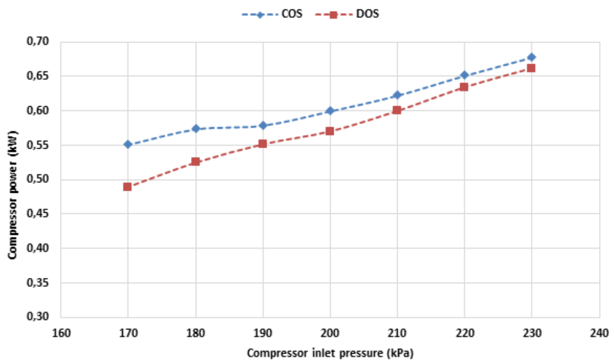


Figure 6 The change of compressor work depending on compressor inlet pressure

Figure 7 illustrates the variations in cooling capacity of evaporator#2 in relation to the compressor inlet pressure. The data presented in Figure 7 indicates that the DOS configuration exhibited higher cooling capacity for evaporator#2 compared to the COS configuration. For instance, at a compressor inlet pressure of 180 kPa, the cooling capacity for evaporator#2 was measured at 0.59 kW for COS and 0.67 kW for DOS, showcasing a 14% increase.

The enhanced cooling capacity in the DOS configuration can be attributed to the lower quality of the refrigerant leaving the ejector. In the DOS configuration, due to the ejector (diffuser) outlet split, the refrigerant enters both evaporators with an equivalent quality, resulting in a notable boost in the cooling capacity of evaporator#2. Conversely, in the COS configuration, the degree of quality at the inlet of evaporator#2 is primarily influenced by the separation rates of the refrigerant following condenser (14).

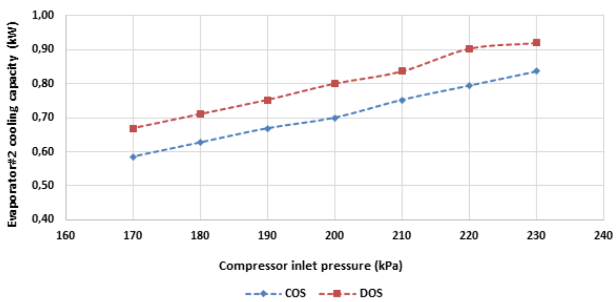


Figure 7 The change of evaporator#2 cooling capacity depending on compressor inlet pressure

Figure 8 illustrates the variations in cooling capacity of evaporator #1 (Qevap#1) in relation to compressor inlet pressures. In COS configuration, the refrigerant fluid underwent splitting at the condenser outlet, resulting in higher measured mass flow rates in comparison to that observed in DOS configuration. Consequently, the Qevap#1 value exhibited an elevation in COS configuration owing to the increased mass flow rates. Notably, at a compressor inlet pressure of 220 kPa, Qevap#1 was recorded as 0.49 kW and 0.19 kW in COS and DOS modes, respectively. It was observed that mass flow rates were lower in DOS mode, primarily attributed to the reduced pressure of mixture within suction chamber of the ejector, consequently resulting in diminished quantities of entrained refrigerant.

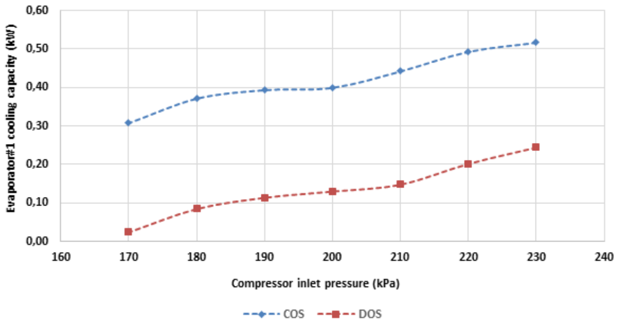


Figure 8 The change of evaporator#1 cooling capacity depending on compressor inlet pressure

Figure 9 depicts the variations in evaporator#2 superheat degrees with respect to compressor inlet pressures. The data reveals a downward trend in evaporator#2 superheat values for both configurations as compressor inlet pressures rise. Additionally, the superheat degrees recorded in DOS configuration were consistently lower than those in the COS configuration. This observation aligns with the previously established higher cooling capacity of evaporator#2 (Figure 7) in the DOS configuration compared to COS. The lower superheat degrees affirm that the refrigerant entering evaporator#2 in the DOS configuration maintained a lower quality, highlighting the performance advantages associated with this system configuration. Furthermore, examination of the figure revealed that the superheat degree of evaporator #2 in the COS configuration ranged between 29°C and 35°C. These values are consistent with those reported in the study by İşkan and Direk (15).

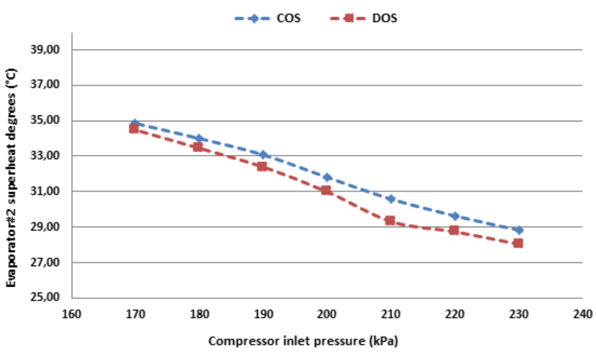


Figure 9 The change of evaporator#2 superheat value depending on compressor inlet pressure

Figure 10 illustrates the variations in total cooling capacity relative to compressor inlet pressures. Upon analysis, it becomes apparent that the COS configuration achieved a higher total cooling capacity at the specified pressures compared to the DOS configuration. This disparity can be attributed to the lower energy difference per unit mass of refrigerant at the evaporators in the COS configuration as opposed to the DOS configuration. For instance, at a pressure of 230 kPa, the DOS configuration provided a total cooling capacity of 1.16 kW, while the COS configuration delivered a total cooling capacity of 1.35 kW, presenting a 16% increase. Similar patterns are observed at other inlet pressures, reinforcing the trend of higher total cooling capacity in the COS mode.

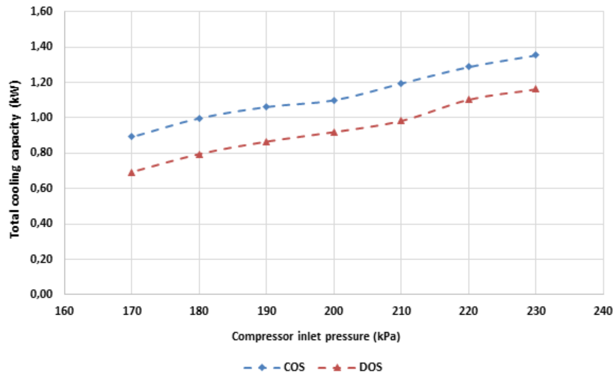


Figure 10 The change of total cooling capacity depending on compressor inlet pressure

CONCLUSION

This study presents an experimental comparison of the performance of an ejector refrigeration system operating in COS and DOS configurations under varying compressor inlet pressures. The experimental investigation yielded significant findings, as outlined below:

-The DOS configuration demonstrated a higher compression ratio than the COS configuration when operating at lower mass flow rates.

-Compressor power was observed to be 9% lower in the DOS configuration compared to the COS configuration.

-Evaporator#2 cooling capacity was approximately 14% higher in the DOS configuration compared to the COS configuration across different compressor inlet pressures.

-Notably, the COS configuration exhibited approximately 16% higher total cooling capacity than the DOS configuration at varying compressor inlet pressures.

These results highlighted the operational efficiency and refrigerant separation capabilities of the dual evaporator ejector system when operated in the DOS configuration. A key recommendation for future research involves the implementation of a constant area ejector model in the DOS configuration, as this adjustment has the potential to alleviate issues related to inadequate cooling due to low ER at decreased compressor inlet pressures.

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