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Theoretical investigation of a novel hybrid refrigeration cycle based on the partial thermal isochoric compression

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ABSTRACT

The objective of this work is the investigation of a novel refrigeration cycle which utilizes both electricity and heat input. This hybrid cycle compresses the refrigerant by using a conventional compressor in the low-pressure stage and a novel isochoric compression in the high-pressure stage by adding heat in the system. Between the compression devices, there is an intercooler which is vital for reducing the heat input demand. The isochoric process is performed in a closed vessel where refrigerant pressure increases by adding heat, according to the gas laws. Different working fluids are studies like CO_2, R134a, R404A, R152a, R290 and R32. According to the final results, the electricity consumption is reduced by the incorporation of the partial thermal compression compared to the conventional system with only mechanical compression. The CO_2 and the R32 seem to be the most promising refrigerants for utilization in the novel suggested cycles. For the case with 75% mechanical compression and 25% thermal compression, the electricity savings with CO_2 are found 15.61% for the case with the evaporating temperature at −25 °C and the heat rejection temperature at 35 °C. More specifically, in this scenario for 100 kW refrigeration production there is a need for 47.29 kW electricity and 19.72 kW heat input, while the system COP is 1.492 and the exergy efficiency 41.01%.

1. Introduction

Refrigeration is one of the most important contributors to worldwide energy consumption [1]. High amounts of electricity are spent in order to cover the needs for refrigeration, as well as for space cooling. This fact makes the refrigeration sector to be one critical sector in energy problems such as global warming and the increased worldwide energy consumption [2]. Moreover, the refrigeration machines utilize harmful refrigerants which create problems in the environment. For example, conventional refrigerants such as R134a and R404A have high GWP values, 1430 and 3922 respectively, which leads to more intense global warming issues [3].

On this direction, there is a need for developing more efficient refrigeration systems which have to be based on natural refrigerants or lower GWP refrigerants [4]. Moreover, the utilization of renewable energy sources is another way of creating sustainable refrigeration systems [5]. Solar energy, geothermal energy and waste heat are energy sources which can be applied for driving or assisting refrigeration systems.

The conventional refrigeration systems are the mechanical compression configurations which consume electricity (or work) for the compressor operation. These are highly efficient systems in terms of coefficient of performance (COP) but they need the consumption of work which is a high-quality energy form. On the other hand, there are refrigeration machines which utilize thermal energy for compressing the working fluid. These devices perform thermal compression with various ways and they try to increase the refrigerant pressure by consuming heating. These refrigeration machines achieve a lower COP compared to the mechanical compression systems because the heat is an energy kind of lower exergetic value compared to the electricity. Moreover, the thermally –driven refrigerators can operate with renewable energy souses (e.g. solar) or waste heat and so they can create sustainable configurations easily.

The most usual thermal compression machine is the absorption chiller. This system is based on the thermal compression through an absorption procedure which is performed by using a working pair. Usually, LiBr-H_2O for cooling applications is used or H_2O-NH_3 for refrigeration applications [6]. The single stage absorption machines need

Abbreviations: COP, Coefficient of Performance; EES, Engineering Equation Solver; GWP, Global Warming Potential; ODP, Ozone Depletion Potential

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Nomenclature

h	Specific enthalpy, kJ/kg
m	Mass flow rate, kg/s
p_l	Low pressure, bar
p_m	Medium pressure, bar
p_h	High pressure, bar
Q_C	Heat rejection rate, kW
Q_E	Refrigeration load, kW
Q_h	Heat input, kW
T	Temperature, °C
v	Specific volume, m^3/kg
W	Work input, kW

Greek symbols

α	Pressure ratio parameter
η	Efficiency

Subscripts and superscripts

ex	Exergy
is	Isentropic
h	Heat input

temperature levels around 100 °C and they operate with a COP close to 0.7, while the multi-effect can operate with COP up to 1.5 with temperature levels up to 180 °C [7]. The next category is the adsorption machines which creates refrigeration by using an adsorbent material (usually silica gel). These systems operate with a non-continuous cycle and they operate with COP close to 0.6 and with driving temperatures around 80 °C [8]. Another category of the sorption machines is the cooling systems with desiccant technology. This is a thermally driven system for regulating air humidity [9].

Another great literature category of thermally-driven refrigeration machines includes ejector devices [10]. The ejector is a mixing device of streams with different pressure and is driven by heat input. The COP of the cycle is generally low in the range of 0.2 to 0.5 but it is depended on the operating conditions. Usually, the ejector devices are also employed in greater refrigeration systems for producing a part of the total pressure increase (gas ejectors). Moreover, there are liquid ejectors which are used for pumping liquid amounts inside the refrigeration cycle and they take the place of a simple circulator in order to reduce the rotating parts of the installation [11].

The previous ideas are interesting and they are developing in order to reduce electricity consumption and to substitute it with thermally-driven systems. However, there are restrictions about the COP values and the working fluids/pairs utilization. A new idea is the utilization of a thermal compression by performing an isochoric compression instead of the isentropic compression (or the adiabatic with an isentropic efficiency) in the conventional compressor. This idea has been recently presented by Bellos et al. [12] in a solar driven system for refrigeration production with R404A. In this work, the electricity saving had been found in the range of 15–25% which is a satisfying range for designing a novel system. Moreover, the idea of a thermal compression device instead of the circulating pump in an organic Rankine cycle has been presented by Yamaguchi et al. [13].

The objective of this work is the improvement of the previously examined idea of Bellos et al. [12]. The cycle of the previous study includes an isochoric compression after the compressor of a refrigeration system in order to reduce the contribution of the compressor and to create a hybrid system with lower electricity consumption, However, the heat input in the system of Ref. [12] was relatively high and thus, this work comes to suggest an improved configuration for reducing this heat input. More specifically, the incorporation of an intercooler between the compressor and the isochoric compression is studied in order to reduce the energy input in the system by thermally compressing a gas of lower temperature level. Furthermore, this work investigates various working fluids, conventional and natural, in order to suggest in which cases the thermal compression through the isochoric process has to be applied. The emphasis is given in the CO_2 which is a promising refrigerant for the future refrigeration systems [14]. The analysis is performed with a developed thermodynamic model in Engineering Equation Solver (EES) [15].

2. Material and methods

2.1. The examined refrigeration thermodynamic cycle

Fig. 1 illustrates the examined refrigeration cycle. This system is such as typical two-stage refrigeration with intercooler but instead of the high-pressure compressor, there is a device for the thermal compression. The process (1 → 2) is a compression in the compressor, the process (2 → 3) is the cooling of the steam by using the ambient source, the process (3 → 4) is the isochoric compression, the process (4 → 5) is the heat rejection to the ambient through the condenser, the process (5 → 6) is the expansion in the throttling valve and the process (6 → 1) is the refrigeration production in the evaporator. Fig. 2 depicts the pressure-enthalpy diagram of the various processes.

The thermal compression is an isochoric process which has to be performed in a closes vessel in order to keep the volume constant (process 3 → 4). This process is a special process which can be achieved with a non-continuous way by using more than one vessel. In Ref. [12], an analytical strategy for performing this process with three different vessels is described. Briefly, it can be said that there are at least 3 vessels in a process as this, and when the first vessel is heated, the other vessel feeds the condenser with high-pressure gas, while the third vessel takes the produced gas from the compressor. Practically, the use of three (or more) vessels makes the non-continuous thermal compression to be a continuous process.

The pressure level (p_m) is an intermediate pressure level which is

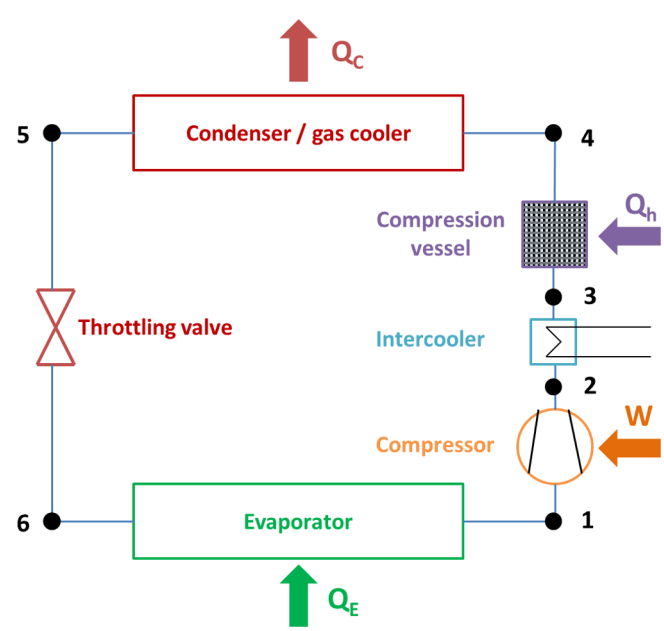

Fig. 1. The examined novel refrigeration system with mechanical and thermal compression which consumes work (W) and heat (Q_h).

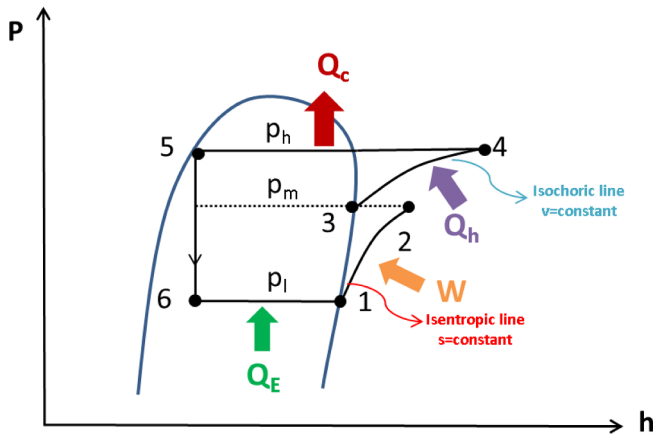

Fig. 2. The pressure-enthalpy depiction of the refrigeration cycle for subcritical operation.

between the low-pressure level (p_l) and the high-pressure level (p_h). The low-pressure level is the saturation pressure of the evaporator temperature, while the high pressure is the saturation pressure of the condenser temperature. In the cases with the transcritical operation, only for some cases with CO_2 as the refrigerant, the high pressure is determined by performing a simple sensitivity analysis for finding its optimum value. During the transcritical operation, the condenser is practically a gas cooler which is used for the heat rejection to the ambient. In any case, the medium pressure (p_m) is a critical parameter of this work and it is studied parametrically by using the following dimensionless parameter (α):

$$\alpha = \frac{p_m - p_l}{p_h - p_l} \tag{1}$$

High values of the parameter (α) mean that the mechanical compressor has a higher contribution to the total gas compression. In this work, the parameter (α) studied from 0.5 up to 1.0. The value of 0.75 is assumed to be a typical one for performing comparison and parametric studies. This value indicates that the thermal compression is 25% of the total compression in terms of pressure increase.

2.2. Mathematical formulation

The main mathematical equations of the developed model are given in this section. The refrigeration production (Q_E) is calculated as below:

$$Q_E = m\cdot(h_1 - h_6) \tag{2}$$

The heat rejection to the ambient (Q_C) is calculated as below:

$$Q_C = m\cdot(h_4 - h_5) \tag{3}$$

The work input (W) in the compressor can be written as:

$$W = m\cdot(h_2 - h_1) \tag{4}$$

The heat input in the thermal compression (Q_h) can be written as:

$$Q_h = m\cdot(h_4 - h_3) \tag{5}$$

During the compression, the entropy is assumed to increase and the isentropic efficiency (η_{is}) can be written as:

$$\eta_{is} = \frac{h_{2,is} - h_1}{h_2 - h_1} \tag{6}$$

The state point (2, is) is the point with pressure (p_m) and entropy equal to the state point (1):

$$s_{2,is} = s_1 \tag{7}$$

The pressure in the state point (3) is the medium one and in the state point (4) is the high one. The state point (4) has the same specific volume as the state point (3):

$$v_4 = v_3 \tag{8}$$

During the intercooling, the temperature in the state point (3) is assumed to be the maximum between the (T_c) and the saturation temperature at the medium pressure.

$$T_3 = \max[T_C\ ,\ \ T_{sat}(p_m)] \tag{9}$$

The process in the throttling valve is assumed to be adiabatic and consequently, the enthalpy does not change:

$$h_5 = h_6 \tag{10}$$

The system can be evaluated using the coefficient of performance (COP) which is an energetic index and it is defined as below:

$$COP = \frac{Q_E}{W + Q_h} \tag{11}$$

However, the COP is not a so proper index because it evaluates the heating input and the electricity in the same way. Thus, the exergy efficiency (η_{ex}) is also used. This index evaluates properly energy energetic quantity by converting it to the respective work quantity:

$$\eta_{ex} = \frac{Q_E\cdot\left(\frac{T_0}{T_E} - 1\right)}{W + Q_h\cdot\left(1 - \frac{T_0}{T_h}\right)} \tag{12}$$

In the previous expression, all the used temperature levels have to be in Kelvin units. The reference temperature (T_0) is assumed to be 298.15 K. Moreover, it has to be said that the heat source temperature (T_h) is assumed to be equal to the refrigerant temperature at the state point (4) in order to perform properly the heat input during the isochoric compression.

2.3. Followed methodology

In this work, the novel refrigeration system is studied with a developed model in Engineering Equation Solver (EES) [15]. The equations of Section 2.2 have been used and the system is studied in steady-state conditions. In the first part of this work, six different working fluids are tested for the case of refrigeration production at $T_e = -25\,^{\circ}C$ and heat rejection at $T_c = 35\,^{\circ}C$. More specifically, the examined working fluids are the following: CO_2, R134a, R404A, R152a, R290 and R32. It is important to state that the symbol (T_c) represents the heat rejection temperature which is the condensation temperature for subcritical cases and the gas cooler outlet temperature for the transcritical operation. The CO_2 is the only working fluid which is in the transcritical region for heat rejection at 35 °C because its critical temperature is at 31 °C [16]. The high pressure of CO_2 has the nominal value of 95 bar which is reasonable value and higher than its critical pressure at 73.8 bar [17]. In this work, the heat rejection is studied parametrically from 30 °C up to 50 °C and in other cases; the maximum pressure is optimized in order to minimize the work consumption.

During this initial working fluid evaluation, the isentropic efficiency is selected at 100%, the superheating and the subcooling are zero, as well as there are no pressure losses in the devices. The previous assumptions have been done in order to study the system theoretically and to check the newly suggested thermodynamic process. In the next part of this work, the isentropic efficiency is studied parametrically in order to check the system performance for realistic operating conditions. Moreover, extra parametric studies are conducted for testing the new examined system. Lastly, it has to be said that the refrigeration capacity is selected at 100 kW in all the cases.

3. Results

3.1. Working fluid investigation

The first part of the results section is devoted to presenting the system performance for different refrigerants and different pressure ratio parameter (α) values. The examined working fluids are CO_2, R134a, R404A, R152a, R290 and R32. The pressure ratio parameter indicates the values of the medium pressure (p_m) and it is a critical parameter of this work. Moreover, it has to be said that the results of this section regard the operating scenario with refrigeration production at −25 °C and heat rejection temperature at 35 °C, while the refrigeration load is equal to 100 kW in all the cases.

Fig. 3 shows the work and the heat demand for the examined refrigerants and for the various values pressure ratio parameter from 0.5 up to 1.0. It is obvious that higher values of the pressure ratio parameters lead to higher work consumption and to lower heat input demand. This is a reasonable result because higher values of the pressure ratio parameter indicate that the mechanical compression has a greater percentage in the total compression. CO_2 is the refrigerant with the highest work demand because it operates in a transcritical mode in the cases with (α) greater than 0.7. However, the heat input in the CO_2 case is relatively low compared to the other refrigerants; a very important result which indicates the use of the novel idea in CO_2 refrigeration systems. Moreover, R32 seems to be another interesting refrigerant for the application in the novel system. On the other hand, the refrigerants R404A and R134a seem not to be ideal for the examined hybrid refrigeration system because they need high amounts of useful heat input. The exact values for the heat input are summarized in Table 1 and for the work input in Table 2.

Fig. 4 shows the heat input demand reduction for the examined cases compared to the system without intercooling, as in Ref. [5]. It is clear that the intercooling is extremely beneficial for the refrigerants CO_2, R32 and R152a, while it is not so effective for the others. This is the reason that indicates the previously referred working fluids and especially CO_2 and R32 as the most suitable for the novel refrigeration system. It has to be said that the maximum heat input reduction is 81.33% for the CO_2 case which is a very high value. About the refrigerants R134a, R404A and R290, the intercooling is not a so effective method and thus their curves are close to the horizontal-zero axis. Practically, for these fluids, the compressor outlet temperature is

Table 1
Heat input demand for the various working fluids (T_e = −25 °C and T_c = 35 °C).

	Heat input – Q_h (kW)					
α	R134a	R404a	R152a	R290	R32	CO_2
0.50	142.30	151.60	116.90	136.60	72.83	81.72
0.55	115.60	123.70	94.64	111.60	58.43	64.31
0.60	93.47	100.40	75.26	90.70	46.68	49.97
0.65	74.87	80.65	59.32	72.97	37.00	38.09
0.70	59.09	63.81	46.09	57.82	28.92	28.16
0.75	45.43	49.32	35.00	44.76	22.11	19.72
0.80	33.15	36.75	25.65	33.00	16.32	11.97
0.85	22.78	25.16	17.71	22.69	11.35	5.29
0.90	13.97	15.30	10.91	13.92	7.05	2.90
0.95	6.44	6.99	5.06	6.43	3.30	1.30
1.00	0.00	0.00	0.00	0.00	0.00	0.00

Table 2
Work input demand for the various working fluids (T_e = −25 °C and T_c = 35 °C).

	Work input – W (kW)					
α	R134a	R404a	R152a	R290	R32	CO_2
0.50	142.30	151.60	116.90	136.60	72.83	81.72
0.55	24.72	27.57	23.11	24.25	23.69	38.93
0.60	25.85	28.89	24.21	25.49	24.96	41.16
0.65	26.90	30.14	25.23	26.65	26.16	43.29
0.70	27.90	31.30	26.20	27.75	27.31	45.33
0.75	28.83	32.40	27.12	28.79	28.40	47.29
0.80	29.72	33.44	27.99	29.77	29.44	49.17
0.85	30.56	34.43	28.82	30.71	30.43	50.97
0.90	31.36	35.37	29.61	31.61	31.39	52.72
0.95	32.12	36.26	30.36	32.46	32.31	54.41
1.00	32.85	37.11	31.08	33.28	33.19	56.04

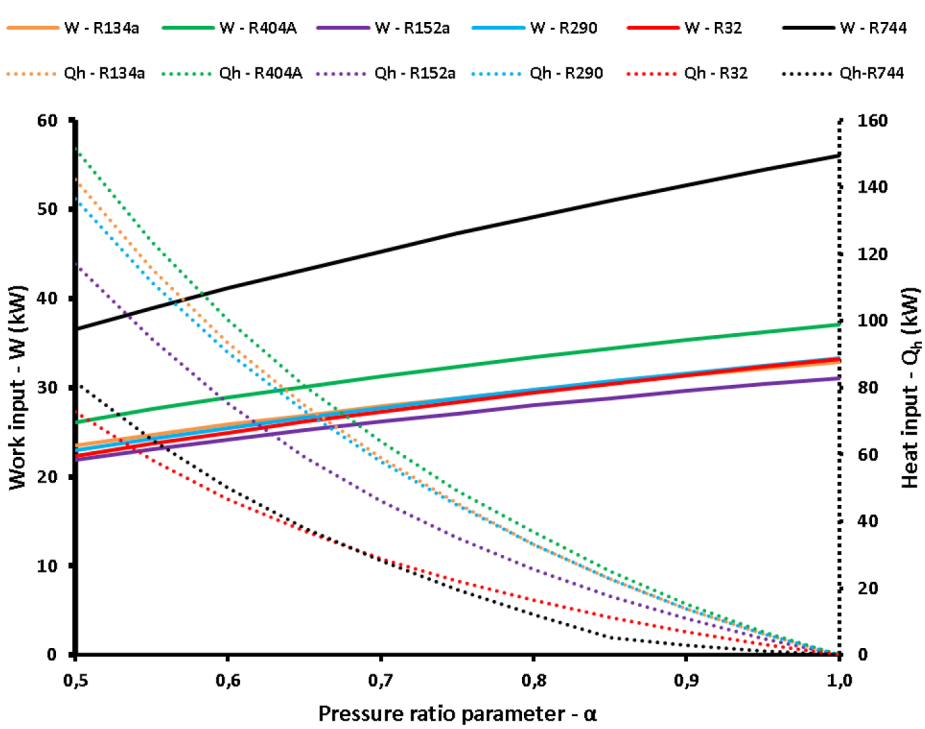

Fig. 3. Work and heat input for the different values of the pressure ratio parameter (α) for the examined refrigerants (T_e = −25 °C and T_c = 35 °C).

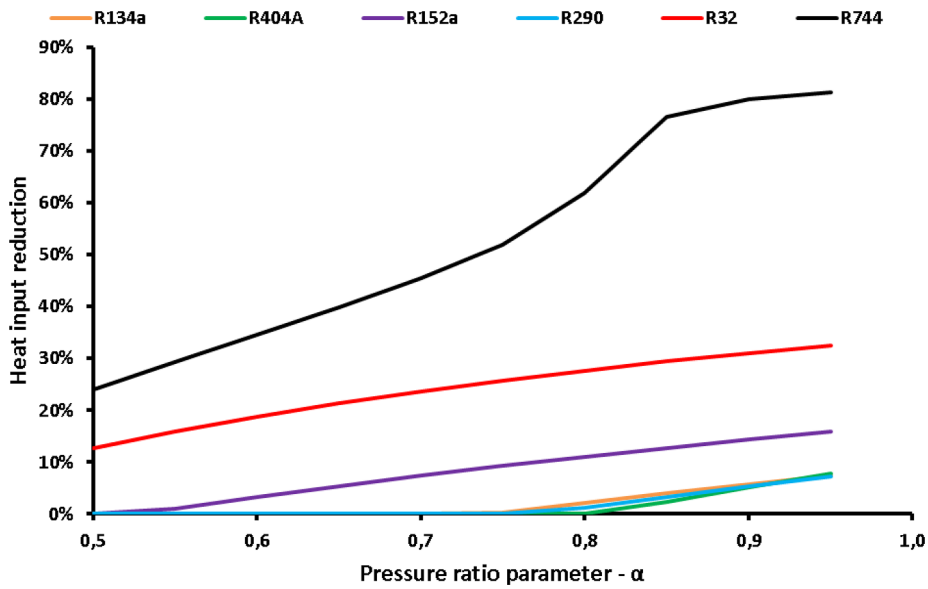

Fig. 4. Heat input reduction compared to the system without intercooler for the different values of the pressure ratio parameter (α) for the examined refrigerants ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$).

not high and thus the intercooling is not something important and effective for them.

Fig. 5 shows the system COP for the examined cases. This is an energetic index which evaluates the heat and the work input with the same way. It is obvious that the COP takes values in the range of 1 up to 3.5. The R32 and the R152a are the most efficient choices due to the lower heat input demand. The CO_2 seems to be a good choice in the cases with low values of the parameter (α), while in a higher value of this parameter, the COP of the CO_2 has a strange behavior. Practically, after the limit of α = 0.8, the COP is approximately constant (it has a very rough increase) with the (α) parameter values. This fact is explained by the compressor operation in the transcritical mode because for higher (α) values the medium pressure is over the critical point. This fact makes the work input to be high and so the COP has this different behavior compared to the other subcritical fluids.

Fig. 6 illustrates the exergy efficiency of the refrigeration system for various cases. The exergy efficiency evaluates suitable the heat input and it converts it to the equivalent work, so it is the most suitable index for the system evaluation. R152a and R32 are the most efficient refrigerant exergetically, while the CO_2 is the less efficient for high values of the parameter (α). So, it has to be said that the CO_2 has to be used in systems with the pressure ratio up to 75% because, in higher values, the CO_2 stops to be enhanced with the use of the thermal compression. Moreover, Fig. 6 indicates that there is an optimum (α) value for all the refrigerants which maximize the exergy efficiency of the system. The optimum values are in the range of 0.70 up to 0.95. Table 3 includes the values of the COP, exergy efficiency, heat input and work input for the optimum exergetic scenarios.

At the end of this section, Fig. 7 depicts the temperature level after thermal compression (T_4). This temperature level is practically the heat input temperature level (T_h) and it is crucial in order to evaluate the heat input quality (or its exergy value). In other words, in the cases with

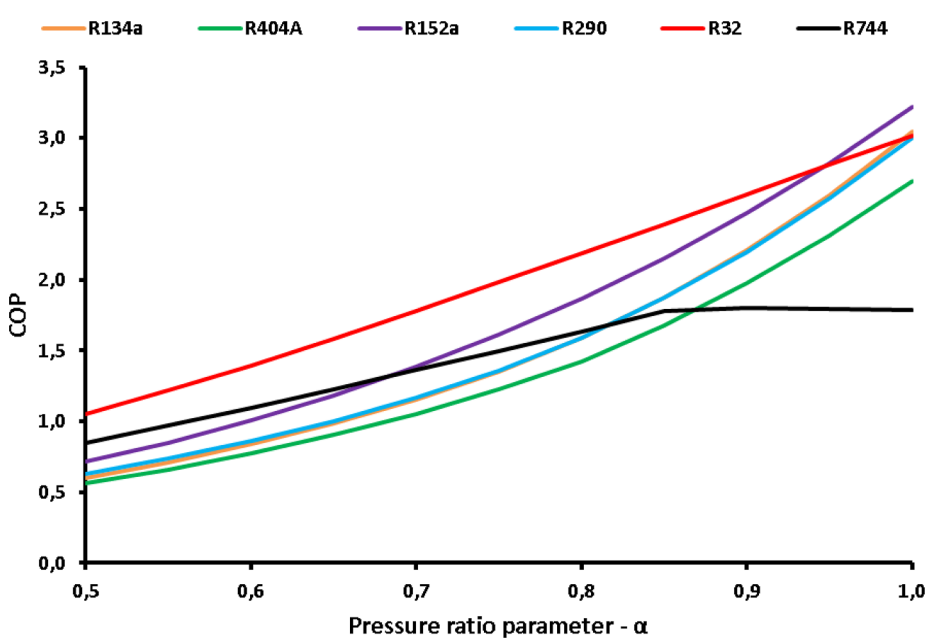

Fig. 5. COP for the different values of the pressure ratio parameter (α) for the examined refrigerants ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$).

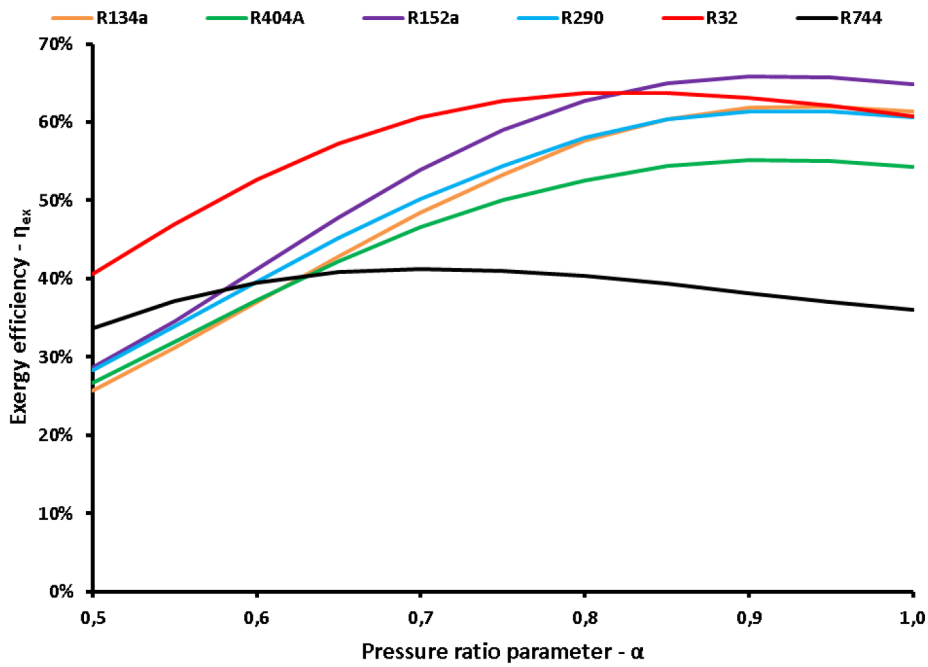

Fig. 6. Exergy efficiency for the different values of the pressure ratio parameter (α) for the examined refrigerants ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$).

Table 3
Optimum cases exergetically for the examined refrigerants ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$).

Refrigerants	A	COP	η_{ex}	Q_h (kW)	W (kW)
R134a	0.95	2.593	62.03%	6.44	32.12
R404a	0.90	1.974	55.16%	15.30	35.37
R152a	0.90	2.468	65.86%	10.91	29.61
R290	0.90	2.196	61.43%	13.92	31.61
R32	0.85	2.394	63.77%	11.35	30.43
CO_2	0.70	1.361	41.26%	28.16	45.33

lower temperature (T_4) there is the possibility for using low-grade heat sources such as waste heat, solar energy and geothermal energy. Fig. 7 proves that this temperature level reaches up to 200 °C. In the cases with the need for high temperatures (> 100 °C), the heat input has to be given by a special heat source such as fuel consumption. In other cases, flat plate collectors or geothermal energy can be used, as for example two classical renewable energy sources.

CO_2 is the refrigerant with the lower values of the temperature (T_4) because the intercooling is extremely beneficial for this refrigerant. For the case of (α = 0.75), the temperature (T_4) is equal to 56.31 °C which is a low value. So, it can be said that the use of solar energy or geothermal energy can be incorporated in CO_2 refrigeration systems for work demand reduction.

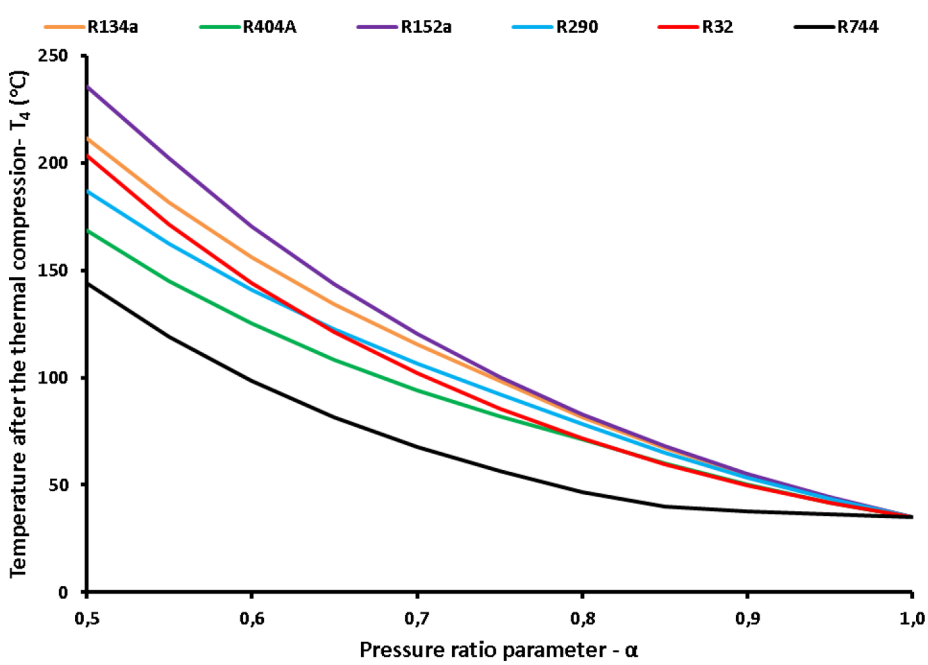

Fig. 7. Temperature after the thermal compression for the different values of the pressure ratio parameter (α) for the examined refrigerants ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$).

3.2. Parametric analysis

The previous section proved that the CO_2 and the R32 are the most interesting refrigerants for utilization in the suggested refrigeration system. Thus, these refrigerants are studies deeper in Section 3.2 by performing some parametric studies. Moreover, it is important to state that the typical case with ($\alpha = 0.75$) is selected. In this case, the mechanical compression is responsible for the ¾ of the total pressure increase, while the thermal compression for the ¼.

Figs. 8 and 9 show the impact of the isentropic efficiency in the system performance for the case ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$). More specifically, Fig. 8 depicts the impact of the isentropic efficiency of the system COP and on the exergy efficiency. It is obvious that all the indexes are enhanced with the increase of isentropic efficiency. Moreover, it has to be said that the R32 is more efficient than the CO_2 according to COP and the exergy efficiency. The COP for the CO_2 is ranged from 0.875 up to 1.492, while for the R32 from 1.267 up to 1.900. The exergy efficiency for the CO_2 is ranged from 20.90% up to 41.01%, while for the R32 from 33.32% up to 62.77%.

Fig. 9 depicts the work and the heat input for the different isentropic efficiencies. It is obvious that higher isentropic efficiency leads to lower work demand but it does not effect on the thermal input. The reason for the constant heat input is explained by the existence of the intercooler which leads to the same temperature level at state point (3) in all the cases. Moreover, it is interesting to state that the CO_2 has higher work demand but lower heat demand compared to the R32.

Figs. 10 and 11 show the impact of the evaporator temperature in the system performance for heat rejection temperature at 35 °C and 100% isentropic efficiency. Fig. 10 exhibits the impact of the evaporator temperature on the COP and the exergy efficiency. It can be said that higher evaporator temperature leads to a significant increase on the COP, while the exergy efficiency is kept in approximately constant values. The COP for the CO_2 is ranged from 0.993 up to 2.444, while for the R32 from 1.366 up to 3.162. The exergy efficiency for the CO_2 is ranged from 38.96% up to 41.07%, while for the R32 from 59.39% up to 63.61%.

Fig. 11 indicates that heat and work input are both reduced with the increase of the evaporator temperature. The reason for the small variation of the exergy efficiency is based on the reduction of the exergy factor of the refrigeration in higher evaporator temperatures. So, there is a reduction in the exergy inputs (heat and work) and a simultaneous reduction of the refrigeration exergy production; two contrary factors which make the exergy efficiency to be ranged in similar levels for all the examined evaporating temperatures.

Figs. 12 and 13 show the impact of the heat rejection temperature in the system performance for evaporator temperature at −25 °C and 100% isentropic efficiency. Fig. 12 illustrates the COP and the exergy efficiency for the examined heat rejection temperatures. It can be said that higher heat rejection temperatures lead to lower COP and exergy efficiency, a result which is reasonable according to the thermodynamic theory (see Carnot refrigeration cycle). The COP for the CO_2 is ranged from 1.080 up to 1.636, while for the R32 from 1.476 up to 2.194. The exergy efficiency for the CO_2 is ranged from 26.18% up to 49.27%, while for the R32 from 43.56% up to 71.54%.

Moreover, Fig. 13 shows that higher heat rejection temperature leads to higher work input for both refrigerants. The heat input increases at higher heat rejection temperatures for the R32, while this parameter has a non-linear behavior for the CO_2. This strange behavior can be explained by the operation in the transcritical mode which has a direct impact on the CO_2 performance. In this analysis, the pressure ratio parameter (α) is kept at the value of 0.75 and the high pressure is optimized exergetically in every case. So, the pressure levels are different for every heat rejection temperature, while the medium pressure is not in all the cases in the "same distance from the critical pressure" for all the cases. So, the behavior of the CO_2 is not linear and predictable as the R32. The results of Figs. 12 and 13 show clearly its performance for the different heat rejection temperatures.

3.3. Discussion and future steps

In this work, a novel refrigeration system is investigated for different refrigerants and operating conditions. Table 4 summarizes the performance of the six examined refrigerants. The results are given for the typical case of ($\alpha = 0.75$) which indicates that the thermal compression is responsible for the 25% of the total pressure increase. Also, this table includes the values of the work demand for the conventional system without thermal compression ($\alpha = 1.00$). The work demand reduction in this table shows that the CO_2 presents the maximum reduction which is 15.61%, while the next one is 14.43% for R32. The other refrigerants have lower reductions but without great variation. The next important point in this table is the amount of the heat input which is minimized for CO_2 with 19.72 kW. The R32 has also a relatively low heat input demand at 22.11 kW, while the other refrigerants have higher needs. For example, the R134a needs 45.43 kW and the R290 44.76 kW. These results show that thermal compression can be easily applied in working fluids such as CO_2 and R32.

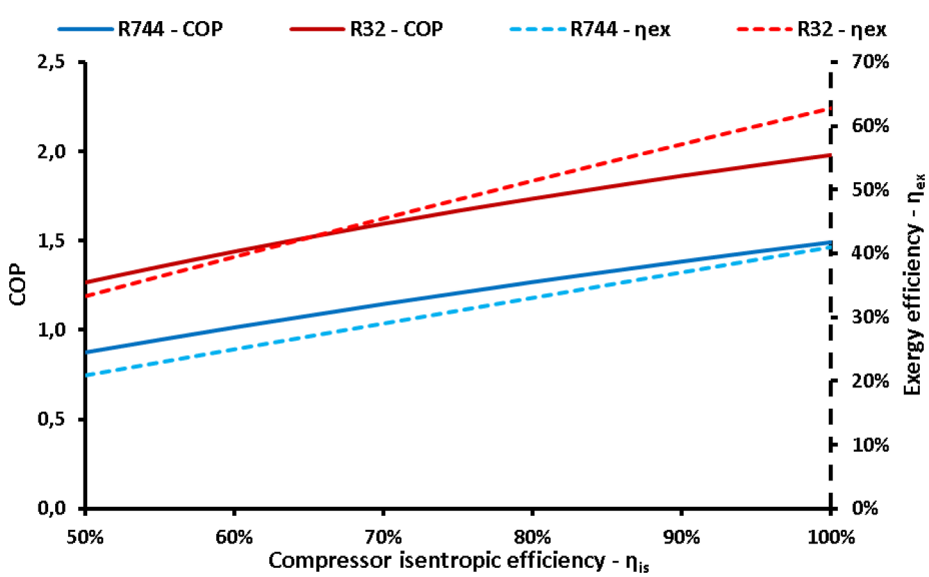

Fig. 8. The impact of the isentropic compressor efficiency on the COP and the exergy efficiency for CO_2 and R32 ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$).

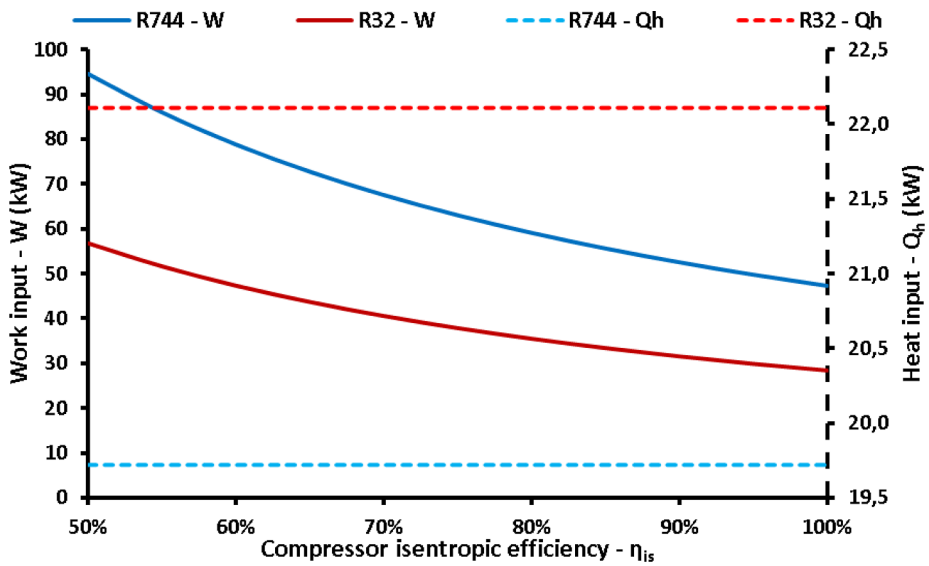

Fig. 9. The impact of the isentropic compressor efficiency on the work and heat input for CO_2 and R32 ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$).

The last examined parameter is the heat input temperature level which is expressed by the (T_4). The CO_2 needs to be heated at 56.31 °C, while the next choice is R404a at 81.81 °C and R32 at 85.40 °C. So, it is obvious that the thermal compression can be done in CO_2 with a low-temperature level which is a very interesting result. More specifically, the temperature of 56.3 °C can be easily achieved by using flat plate solar collectors or geothermal energy. So, it can be said that the thermal compression with CO_2 is an overall optimum case because this CO_2 presents the maximum work input reduction, the minimum heat input demand and the minimum heat temperature level. Furthermore, CO_2 is the environmentally friendly fluid because it has GWP = 1, ODP = 0, it is no toxic, no flammable and it is a cheap working fluid. The only drawback of CO_2 is the reduced efficiency due to the transcritical operation which something that is faced with the use of the partial thermal compression by using a renewable energy source.

At the end of the discussion section, it has to be stated that the use of the vessel before the compressor is another idea which had been studied but it is not presented in this work because it is found that it is not an effective way to reduce the work input. More specifically, the isochoric procedure demands high-temperature levels order to compress thermally the vapor of low temperature, as it is found by preliminary calculations, thus this idea has no interest for further investigation. On the other hand, it is very interesting to investigate the possibility of utilizing partially the rejected heat from the intercooler for feeding with heat input the compression vessel. This is something that will be studied in future studies.

4. Conclusions

The objective of this work is the investigation of a novel hybrid refrigeration cycle which needs both work and heat input. This cycle includes mechanical compression after the evaporator, an intercooler device and a thermal compression vessel. The thermal compression is performed using heat input in a vessel and it is an isochoric process.

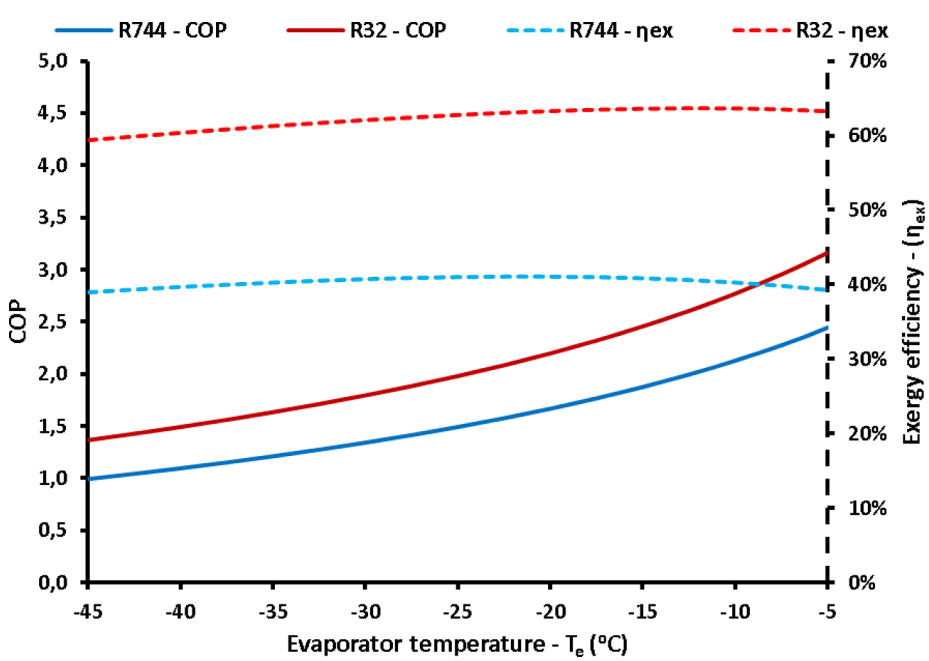

Fig. 10. The impact of the evaporator temperature on the COP and the exergy efficiency for CO_2 and R32 ($T_c = 35\,^{\circ}C$).

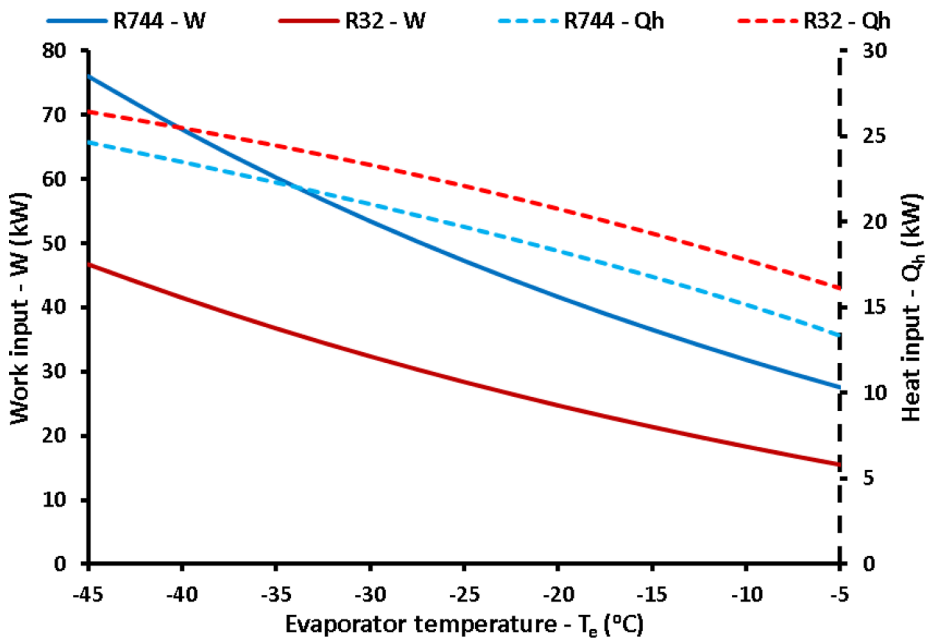

Fig. 11. The impact of the evaporator temperature on the work and heat input for CO_2 and R32 (T_c = 35 °C).

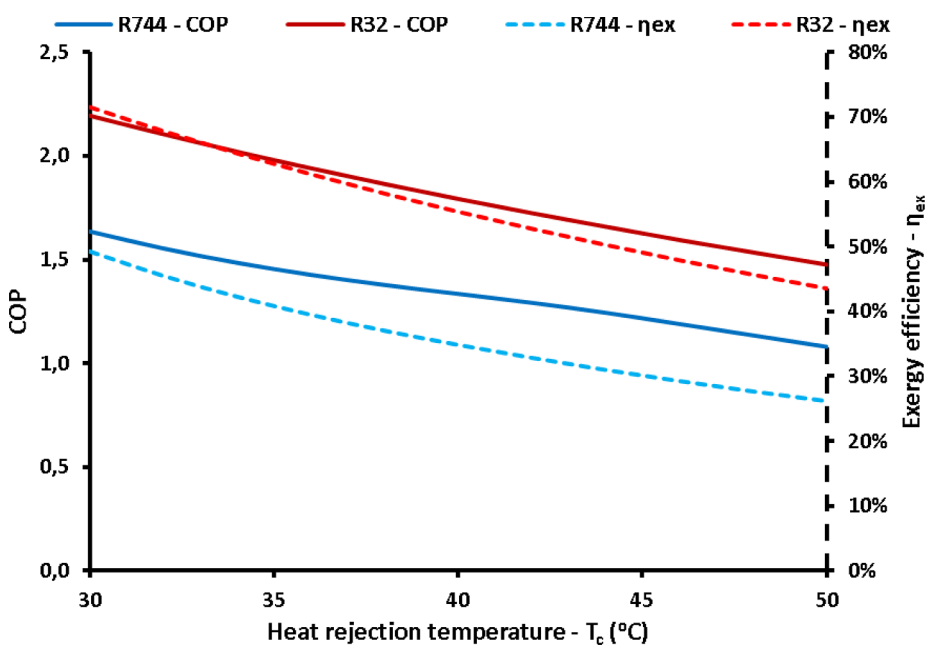

Fig. 12. The impact of the heat rejection temperature on the COP and the exergy efficiency for CO_2 and R32 (T_e = −25 °C and high pressure optimized for CO_2 case).

Various working fluids are studies and the most important conclusions from this work are summarized below:

- CO_2 and R32 are the working fluids with the lowest heat input demand in the thermal compression process among the examined. Moreover, these fluids are extremely enhanced with the use of the intercooling process.
- The work reduction compared to conventional refrigeration cycle with only mechanical compression is in the range of 12.24% up to 15.61% for 25% thermal compression (α = 0.75). The CO_2 is the refrigerant which presents the maximum work input reduction which is 15.61%.
- Another advantage of CO_2 is the low temperature of the heat input, something that makes possible the use of low-grade renewable energy sources for driving the isochoric thermal compression.
- Moreover, it has been found that the exergy efficiency is maximized for different values of the pressure ratio parameter. The optimum values are ranged from 0.70 to 0.95 for the different working fluids.
- Moreover, it has to be said that the higher improvement of the CO_2 with the thermal compression compared to the other refrigerants is explained by the operation in the transcritical mode which gives higher enhancement margins.

Conflict of interests

There is no conflict of interest.

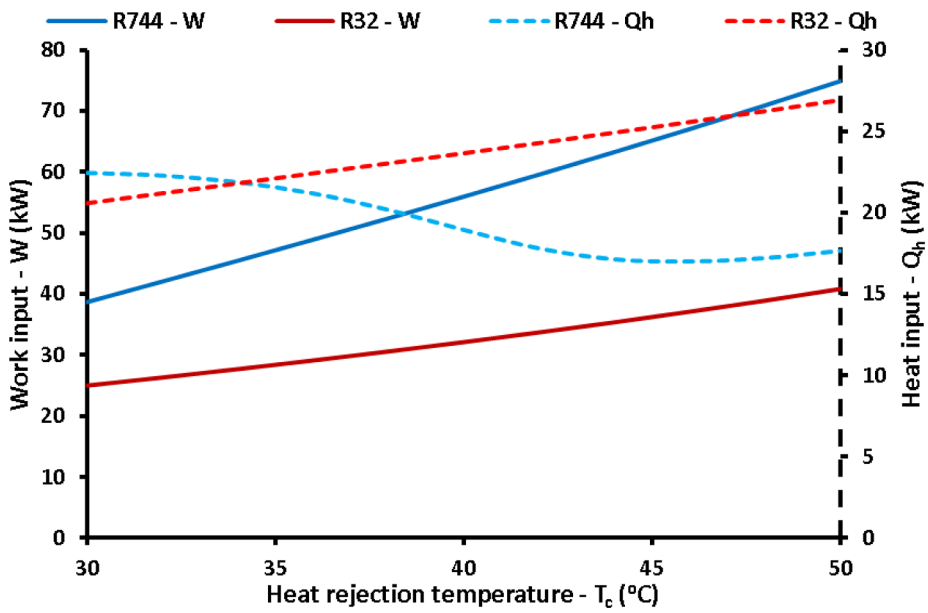

Fig. 13. The impact of the heat rejection temperature on the work and heat input for CO_2 and R32 ($T_e = -25\,^{\circ}C$ and high pressure optimized for CO_2 case).

Table 4
Evaluation of the hybrid system ($\alpha = 0.75$) compared to the conventional one ($\alpha = 1.00$) for ($T_e = -25\,^{\circ}C$ and $T_c = 35\,^{\circ}C$).

Refrigerants	Q_h (kW) ($\alpha = 0.75$)	T_4 (°C) ($\alpha = 0.75$)	W (kW) ($\alpha = 0.75$)	W (kW) ($\alpha = 1.00$)	Work reduction
R134a	45.43	98.51	28.83	32.85	12.24%
R404a	49.32	81.81	32.40	37.11	12.69%
R152a	35.00	100.20	27.12	31.08	12.74%
R290	44.76	92.35	28.79	33.28	13.49%
R32	22.11	85.40	28.40	33.19	14.43%
CO_2	19.72	56.31	47.29	56.04	15.61%

Acknowledgments

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