# Enhancement Of Ejection Systems Efficiency By Means Of Regenerative Heat Exchager

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# Enhancement of ejection systems efficiency by means of regenerative heat exchanger

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## ABSTRACT

Recently the ejection system became more and more interesting for investigators. The reason for this is, on the one hand, the necessity to apply new low-GWP working fluids, and on the other hand, the necessity to save electric energy required by classic compression refrigeration systems. Ejection system can utilize free renewable or low-grade waste heat as a motive source. Despite all advantages of the ejection systems, such as simple construction, no moving parts, no maintenance operation one of the main disadvantage of these systems is their low efficiency described by coefficient of performance (*COP*). Improvement of the *COP* of the system by simple and low-cost method seems to be the natural direction for the investigations. The internal heat transfer by means of the internal heat exchanger which is located at the ejector discharge line/ liquid line may be thought as the most simple and effective approach. In the internal heat exchanger heat taken from the superheated vapor is transferred to the liquid feeding the vapor generator. The vapor superheating at the ejector outlet may be thought as a waste heat so along with condensation heat is transported to the ambient. As an effect of the heat transfer the temperature of this liquid increases. In effect thermal load required by the vapor generator decreases. In the paper the results of calculation for low-GWP refrigerant are presented. Experimental results obtained from the ejection system operating with low-GWP refrigerant R1234zeE driven by low-grade heat are used for validation. The basic parameters describing heat exchangers, e.g. temperature changes, thermal capacity, pressure losses, heat exchanger effectiveness are presented and discussed. Finally, the improvement of the *COP* is shown. The experiments show that for the analyzed operation parameters the heat exchanger is able to use the superheating and preheat the liquid for about 20 K and as an effect the *COP* increased up to 15%.

## 1. INTRODUCTION

Residential, tertiary and industry are three main sectors which consume energy for heating and cooling. The households buildings representing the highest share accounted for 45% of the final energy for heating and cooling consumption, while industry share is 37% and services of 18%. Energy is used in buildings for heating and cooling, hot water, lighting, and appliances. The majority of this energy come from the burning of fossil fuel. Even if the renewable energy is the world fastest-growing energy source with increase by 2.6% per year, still the fossil fuels continue to supply more than 75% of world energy use in 2040. The building sector therefore plays a significant role in mitigating the impacts of climate change through reducing the demand; i.e. energy conservation, and by maximizing the use of renewable energy (Li et al. 2017). This has increased the need for new energy substitutes and conversion methods to meet an increasing energy demand and pave the way to cost-effective heating and cooling solutions. One of the possibilities of the combine heating and cooling demands is utilization of the district heat for chilled water production. This can be accomplished by application of sorption or ejection systems. The ejection refrigeration system can be applied for central cooling systems, and because of its ability to operation with low-temperature heat source it can be driven by the district heating system. Exemplary schematic diagram of the ejection system is presented in Fig. 1. Refrigerant in vapor state is pressurized by means of the ejector instead of a mechanical compressor. Therefore, the electrical power consumption is reduced. Also, potential operation and maintenance problems related with mechanical compressors such as damages of moving parts and oil circulation are also eliminated. New ecologic low GWP refrigerant R-1234zeE may be proposed as a working fluid for these systems.

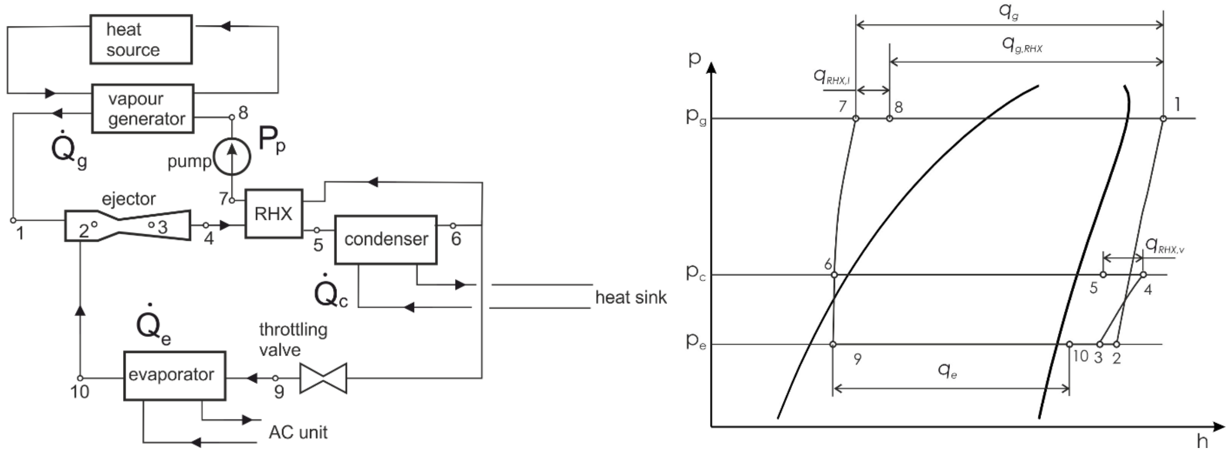


**Figure 1**: Schematic of the ejector refrigeration system with internal heat exchanger (RHX)

Ejectors are very simple devices, which contain no moving parts so that their maintenance is not required. The ejection refrigeration system can be also very compact. The ejectors have many advantages over other heat driven devices, e.g. sorption systems. However, despite that, the ejection refrigeration systems, similarly like absorption systems generally may be thought as low efficient devices. Rattner and Garimella (2016a, 2016b) presented the operation of the absorption systems with heat source temperatures up to 130°C. The *COP* of the discussed systems used for AC purposes was 0.14 for chilled fluid temperature 12/8°C and *COP* = 0.17 for temperatures 18/14°C. The authors have pointed out that using of different pairs of sorbent-refrigerant can lead to lowering of generator temperature. Theoretical analysis of the performance of the ejection systems depending on working fluid was presented by Besagni *et al.* (2015). By means of lumped parameters model the authors have found that for motive temperatures lower than 100°C the refrigerant R-134a can be effectively used, hydrocarbons R-600 are suitable for generator temperatures up to 130°C, and pentane (R-601) can be recommended for generator temperatures above 130°C. In paper presented by Chen et al. (2017) analysis were made for several refrigerants, i.e. R-134a, R-123, R-124, R-290, R-600a, and R-141b. The authors have assessed *COP* of the system for given various operation conditions. Refrigerant R-1234yf was used as a substitute of R-134a in AC system and heat pump in paper published by Wang (2014). The author found that direct replacement of the refrigerants can lead to deterioration of the heat capacity and *COP* of the R134a system up to 27%.

There is a clear need for further development of simple and low-cost method for improving of *COP*. The internal heat transfer by means of the internal heat exchanger which is located at the ejector discharge line/ liquid line may be thought as the most simple and effective approach. As it was shown in Fig. 1 the proposed ejection system is equipped with the internal heat exchanger (RHX) in other to improve *COP* of the system. The rationality of this solution results among others from the utilization of superheating of the vapor discharged from the ejector (Butrymowicz et al., 2014). Thermodynamic cycle in pressure-enthalpy plot is shown in right-hand side of the Fig. 1. In the figure the regenerated heat streams are shown as an enthalpy differences (described as $q_{RHX}$). The vapor superheating at the ejector outlet $q_{RHX,v}$, may be thought as a waste heat so along with condensation heat is transported to the ambient. In the discussed internal heat exchanger heat taken from the superheated vapor is transferred to the liquid feeding the vapor generator, $q_{RHX,l}$. As an effect the temperature of this liquid increases. In effect thermal load required by the vapor generator decreases, $q_{g,RHX} < q_g$. Moreover, the cost of application of the additional heat exchanger in the system may be thought as relatively small. Additional advantage of the utilization of the vapor superheating is related with decrease of the thermal load of the condenser. As an effect the required quantity of the condenser cooling fluid is smaller. Coefficient of performance of the ejection system in general form is given by:

$$COP = \frac{\dot{Q}_e}{\dot{Q}_g + P_p} \tag{1}$$

where $\dot{Q}_e$ is cooling capacity, $\dot{Q}_g$ is motive heat flux, $P_p$ is power consumed by mechanical pump. The cooling capacity $\dot{Q}_e$, of the system was calculated as:

$$\dot{Q}_e = \dot{m}_e \cdot \Delta h_e \,, \tag{2}$$

where $\dot{m}_e$ is mass flow rate of refrigerant flows through the evaporator, and $\Delta h_e$ is enthalpy difference at the evaporator outlet and inlet. The motive heat flux $\dot{Q}_g$ was calculated as follows:

$$\dot{Q}_g = \dot{m}_g \cdot \Delta h_g \,, \tag{3}$$

where $\dot{m}_g$ is motive vapor mass flow rate, and $\Delta h_g$ is enthalpy difference between the vapor generator outlet and inlet. On the basis of the measurement of temperature and pressure of vapor at the heat exchanger inlet and outlet and mass flow rate of vapor, the thermal capacity of the heat exchanger can be found as:

$$\dot{Q}_{RHX} = (\dot{m}_g + \dot{m}_e) \cdot (h_{in} - h_{out}) \,, \tag{4}$$

where the specific enthalpies are calculated from the equation of state $h = h(t,p)$; the inlet parameters correspond to the ejector discharge, and outlet parameters correspond to the condenser inlet.

In order to evaluate the improvement of the *COP* of the system the *COP* of the classic system must be known. It was assumed that the thermal capacity of the internal heat exchanger $\dot{Q}_{RHX}$ must be added as a motive heat to the system operating without the heat exchanger since liquid that feeds the generator is not preheated. In that case $COP_{\mathrm{st}}$ of the standard system without the internal heat exchanger is:

$$COP_{st} = \frac{\dot{Q}_e}{\dot{Q}_g + \dot{Q}_{RHX} + P_p} \,. \tag{5}$$

Finally, improvement of *COP* is assessed as follows:

$$\Delta COP_{\%} = \frac{COP - COP_{st}}{COP_{st}} \cdot 100\% \,. \tag{6}$$

Overall heat transfer coefficient for the discussed internal heat exchanger is calculated as:

$$K = \frac{\dot{Q}_{\mathrm{RHX}}}{\Delta T_{\log} \cdot A_{\mathrm{RHX}}} = \frac{(\dot{m}_g + \dot{m}_e) \cdot (h_{in} - h_{out})}{\Delta T_{\log} \cdot A_{\mathrm{RHX}}} \,, \tag{7}$$

where $\Delta T_{\log}$ is mean temperature difference between vapor and liquid, and $A_{\mathrm{RHX}}$ is heat transfer surface area. Since heat transfer conditions may be assumed as uniform in the discussed heat exchanger then logarithmic mean temperature difference may be applied:

$$\Delta T_{\log} = \frac{\Delta T_{liq} - \Delta T_{vap}}{\ln\left(\frac{\Delta T_{liq}}{\Delta T_{vap}}\right)} \,. \tag{8}$$

The heat exchanger effectiveness, $\eta_X$, is defined as the ratio of actual rate of heat transfer from the hot (vapor) to cold (liquid) fluid to the maximum possible rate of heat transfer. The effectiveness of the discussed heat exchanger is calculated as follows:

$$\varepsilon = \frac{\dot{Q}_{\mathrm{RHX}}}{\dot{Q}_{\max}} \,, \tag{9}$$

where $\dot{Q}_{\max}$ is lower value of heat flux exchanged on the vapor or liquid side

$$\dot{Q}_{\max} = \min\left[\dot{Q}_{vap};\dot{Q}_{liq}\right] \, . \tag{10}$$

Another important issue that should be discussed is the problem concerning the effect produced by the additional pressure loss due to the application of the internal heat exchanger. The pressure loss at the vapor side requires higher compression ratio for the ejector which can deteriorate the ejector capacity in terms of the entrainment ratio. The additional pressure loss at the liquid side of the heat exchanger requires higher compression ratio for the liquid pump which increases power consumption by this pump. Both effects can deteriorate *COP* of the system. The discussed pressure loss at the vapor side in the heat exchanger were measured directly during the experiments. Pressure transducers were located at the ejector outlet (at the diffuser outlet) and at the inlet to the condenser, see Fig. 2. The internal heat exchanger was located between these pressure transducers so that the pressure loss produced by this discussed heat exchanger can be calculated as:

$$\Delta p_{RHX} = p_d - p_{c,sat} \, . \tag{11}$$

## 2. TESTING STAND

The schematic of the experimental testing stand is presented in Fig. 2. The main elements of the stand are listed in Fig. 2 caption. Location of the temperature sensors (RTD) and pressure transducers (P) at the testing stand is presented in Fig. 2. Accuracy of the sensors were as follows: 0.25% for the pressure transducers, 0.20% for the RTD temperature sensors, and 0.15% for the Coriolis mass flow meters. Measurement error was calculated by means of the total derivative approach. All of the measurement sensors and the measurement circuits were calibrated before each of the experimental run.

Three auxiliary loops were applied in the test stand that serve for the thermal load of the evaporator, for the condenser cooling, and as the heat source for the vapor generator. The Coriolis mass flow meters and temperature sensors at the heat exchangers inlets and outlets were installed at these systems. The auxiliary systems allowed for adjusting of the operation parameters in a wide range of thermal capacity. Water was used as a working fluid in the vapor generator heat source system and ethylene glycol was used as a heat transfer fluid in the evaporator heat load system and in the condenser cooling system. Dry cooler is used in the condenser cooling system.

The presented system is proposed for operation with low-grade motive source. During experiments operation of the motive source was simulated by the electric heater system of the thermal capacity approximately 100 kW. During experiments the electric heater operated with full capacity. Mass flow rate of water circulated in the heating loop was set to maintain its temperature at the vapor generator inlet below 70°C. The pressure and the temperature of the motive vapor at the ejector inlet were controlled by means of the control valves. Regenerative heat exchanger was applied for improvement of the effectiveness of the ejection device.
The refrigerant is circulated by the diaphragm pump. The pump is driven by the motor of power 3 kW achieved at 700 RPM. In general, the pump is used for circulation of a part of the refrigerant, i.e. it feeds the vapor generator only. However, as it seen in Fig. 1 and Fig. 2 in presented system the pump is located behind the condenser before the fluid is split into the motive and the entrained streams. Therefore, the pump is pumping all amount of refrigerant. As an effect the pump consumes more energy. However, location of the pump in presented system enables more precise and easier regulation and control of the system operation. In refrigeration system operating with commonly used refrigerants, e.g. R-404A or R-134a the throttling valve that feeds the evaporator operates at pressure difference between the condenser and the evaporator 500 kPa and more. For low-pressure refrigerants this pressure difference it 200-300 kPa. Without initial pressure increase operation of the throttling valve will be less stable and predictable. For this reason, we have decided for such location of the pump. However, since the liquid pump consumes almost negligible shear in the total motive power consumption of the system, then the discussed system configuration does not lead to noticeable deterioration of *COP* and provides easier control of the operation.
The measurement system was based on two National Instrument systems. The computer used dedicated software which was capable to receive on-line data from refrigeration properties software. The frequency of the measurement reading was 1 Hz. Time of measurement was between 90-120 seconds. All readings were averaged to make one experimental point. The operating conditions were taken as stable when pressure and temperatures profiles not shown significant changes.

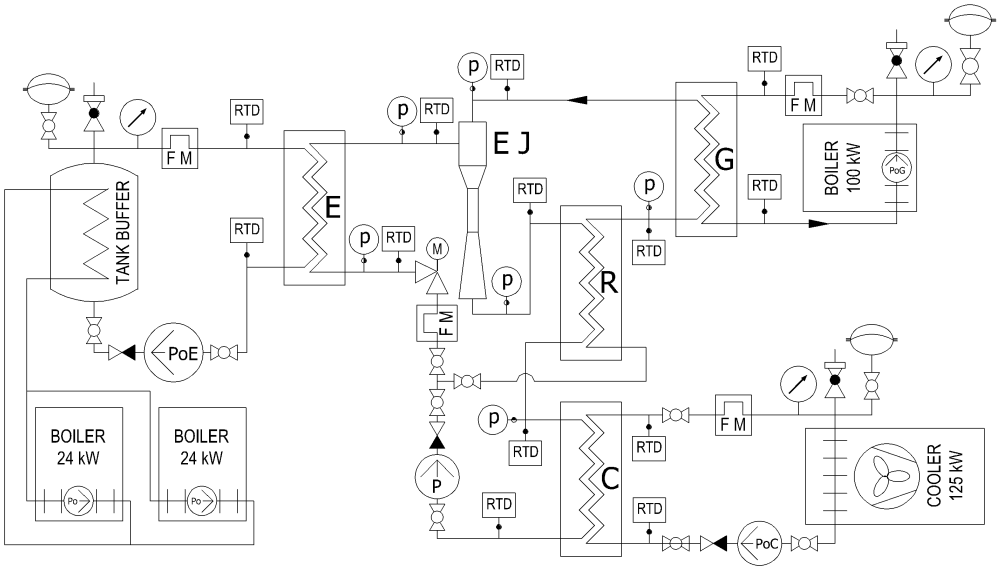


**Figure 2**: Schematic of the testing stand: vapor generator (G); condenser (C); evaporator (E); ejector (EJ); refrigerant pump (P); internal heat exchanger (R); M - throttling valve, RTD – temperature sensor, p – pressure transducer, FM – flow meter

Two experimental runs were conducted with operating conditions given in Table 1.

**Table 1:** Experimental operating conditions

| | **Run No. 1** | **Run No.2** |
|---|---|---|
| motive temperature (saturation) | $t_g = 58$ ºC | $t_g = 58$ ºC |
| motive superheating | $\Delta T_g = 5$ K | $\Delta T_g = 5$ K |
| evaporation temperature (saturation) | $t_{e1} = 0$ ºC | $t_{e2} = 6$ ºC |
| evaporation superheating | 5 K | 6 - 8 K |
| condensation temperature | variable | variable |

## 3. RESULTS

The main goal of application of the internal heat exchanger is to preheat liquid refrigerant feeding the vapor generator. Thermal energy required to preheat of liquid refrigerant is taken from refrigerant vapor at the ejector discharge which is at superheated vapor state. This superheat can be thought as a waste heat because it is rejected at the condenser if the internal heat exchanger is not applied.

Performance lines in terms of the coefficient of performance *COP* versus condensation temperature for both test runs are shown in Fig. 3.

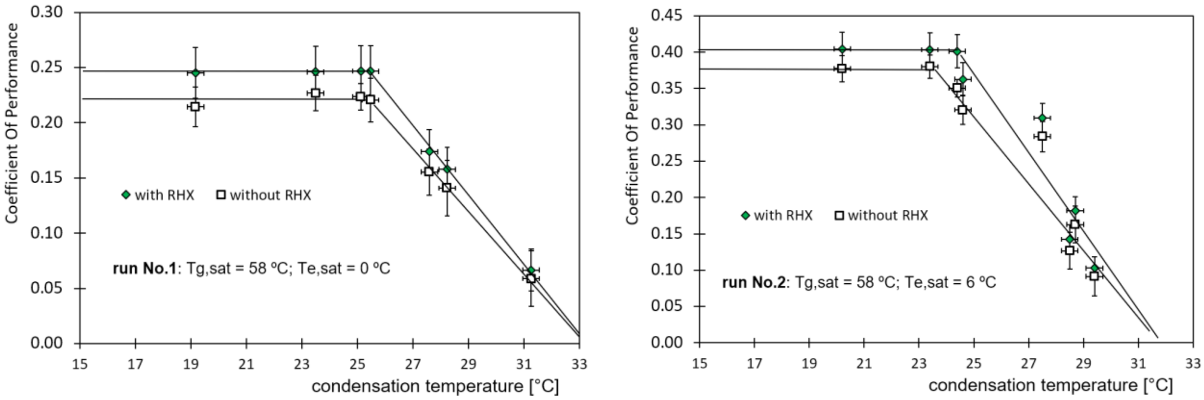


**Figure 3**: *COP* of the system versus condensation temperature; run No. 1 (left), run No. 2 (right)

Heat flux transferred in the internal heat exchanger was calculated by eq. (4). Calculated $\dot{Q}_{RHX}$ was added to the motive heat source according to eq. (5) in order to calculate *COP* of the standard system operating without the internal heat exchange. This allows for the assessment of *COP* improvement due to application of the internal heat exchanger according to eq. (6). Results are shown in Fig 3. It is seen that without regenerative heat exchanger the system is less efficient. For run No.1 $COP_{st}$ = 0.22 while after heat regeneration the *COP* increases to *COP* = 0.25. For run No.2 $COP_{st}$ = 0.37 while after heat regeneration the *COP* increases to *COP* = 0.41.

Temperature of refrigerant vapor at the ejector discharge in comparison to saturation conditions versus condensation temperature is shown in Fig. 4 for test run No. 1, and in Fig. 5 for test run No. 2.

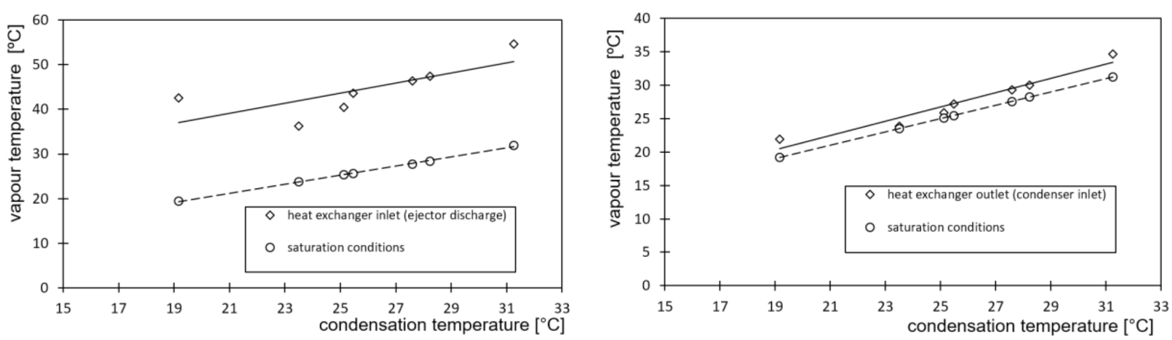


**Figure 4**: Vapor temperature at the RHX inlet (left) and outlet (right) for $t_e$ = 0 ºC (run No. 1)

It is seen on left-hand side of Fig. 4 and Fig. 5 that vapor temperature at heat exchanger inlet vary between 32 ºC and 50 ºC, therefore, the superheating is in range 10 K to 20 K. Outlet temperature of vapor in shown in right-hand side of Fig 4 and Fig. 5. The temperature was measured with accuracy of ± 0.2 K and this value is too small to be clearly present in the figures. These figures show that vapor leaves the internal heat exchanger at nearly saturation conditions. That means the entire or almost entire superheating was utilized for preheating of the liquid before it enters the vapor generator. The changes of the temperature of refrigerant on both sides of RHX are presented in Fig. 6. The temperature changes were calculated as:

$$\Delta T_{vap} = t_{vap,in} - t_{vap,out}\,, \qquad \Delta T_{liq} = t_{liq,out} - t_{liq,in}\,. \tag{11}$$

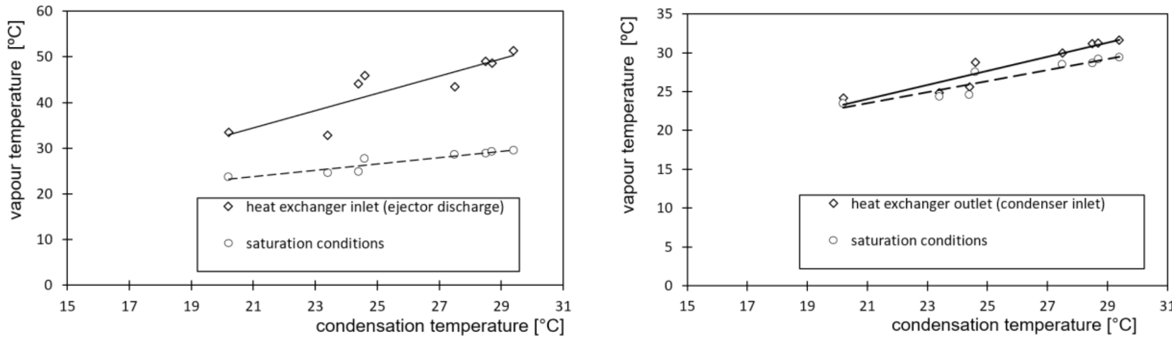


**Figure 5**: Vapor temperature at the RHX inlet (left) and outlet (right) for $t_e$ = 6 ºC (run No. 2)

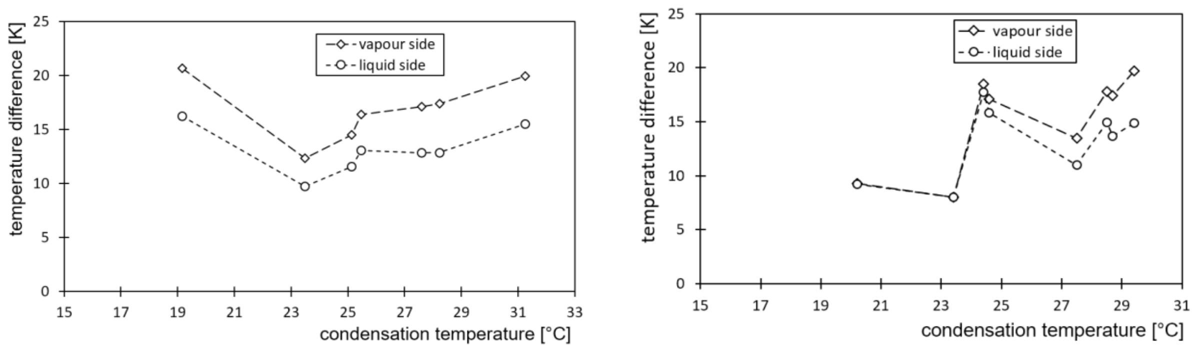


**Figure 6**: Temperature change in the internal heat exchanger: run No. 1 (left), run No. 2 (right)

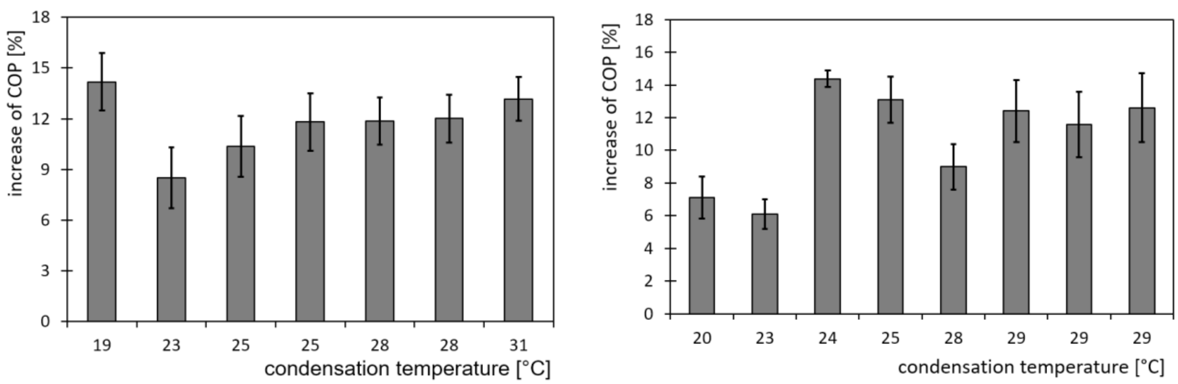


**Figure 7**: Improvement of *COP*: run No. 1 (left), run No. 2 (right)

Using eq. (6) the improvement of the *COP* was evaluated. Results are given in Fig. 7. The results clearly show that application of internal heat exchanger improves *COP* of the system. It is seen in Fig. 7 that when system operates at the evaporation temperature $t_e$ = 0ºC the improvement of *COP* is more stable in comparison with the second test run at which $t_e$ = 6ºC. For both test runs the average value of improvement of the *COP* is in range 10 - 12%. The achieved improvement can be thought as a very attractive. Also, it can be seen that *COP* improvement does not depend on condensation or evaporation temperatures.

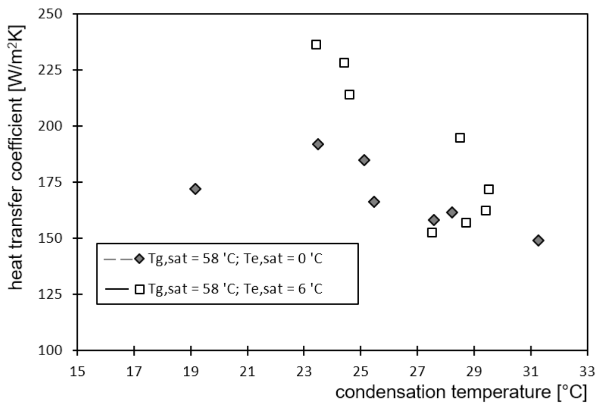


**Figure 8**: Overall Heat transfer coefficient

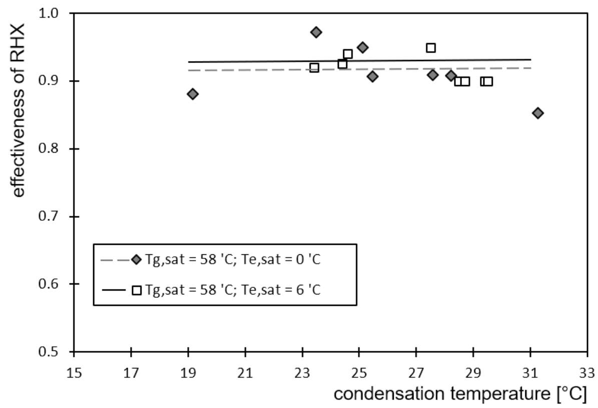


**Figure 9**: Effectiveness of the internal heat exchanger

Overall heat transfer coefficient of the internal heat exchanger is in range 150-225 $W/(m^2 \cdot K)$. It should be noted that information about heat transfer coefficient for R1234zeE are very limited. Based on results given in Fig. 9 it can be stated that RHX has effectiveness close to unity. Therefore, application of this kind of *COP* improvement solution can be very attractive.

Experimental investigations also allow for evaluation of the pressure loss produced by the discussed heat exchanger. For analyzed operating conditions the pressure drop calculated from eq. (11) was in the range of 2-4 kPa. It must be pointed out that the ejector operates between evaporator and condenser so that the static pressure difference produced by the ejector was in the range of 200-300 kPa. Therefore, the additional pressure drop produced by the internal heat exchanger may be assessed as negligible for the analyzed cases.

## 4. CONCLUSIONS

Based on the presented results following conclusions can be drawn:

- Superheating of vapor at the ejector outlet is in range 10 K – 20 K for analyzed cases, which can be utilize to preheat the liquid.
- Experiments proved that regenerative heat exchanger utilize all superheating and vapor is at saturation at the RHX outlet.
- Pressure drop produced by RHX are negligible, and the effectiveness of RHX is close to unity.
- The applied RHX improves the *COP* of the analyzed system for about 10-12%.
- Application of internal heat exchanger is a simple and inexpensive solution for improvement of *COP* of the ejector refrigeration system.

## NOMENCLATURE

| | | |
|---|---|---|
| AC | air-conditioning | (-) |
| *COP* | coefficient of performance | (-) |
| GWP | Global Warming Potential | (-) |
| $h$ | specific enthalpy, | (kJ/kg) |
| $\dot{m}$ | mass flow rate, | (kg/s) |
| $p$ | pressure, | (MPa) |
| $T$ | temperature, | (K) |
| $t$ | temperature, | (°C ) |
| $Q$ | thermal capacity, | (kW) |

**Subscript**

| | |
|---|---|
| c | condenser |
| crit | critical |

| | |
|---|---|
| e | evaporator, secondary fluid |
| d | discharge |
| g | generator, primary fluid |
| l | liquid |
| max | maximum |
| p | pump |
| R | refrigerant |
| RHX | regenerative heat exchanger |
| sat | saturation condition |
| st | standard |
| v | vapor |

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