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Application of waste low heat as motive heat source for ejection air-conditioning systems for yachts

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ABSTRACT

Heat recovery is a common approach for effective energy management. With utilization of the waste heat the investment and operation costs can be reduced. The possibility of utilisation of the waste heat from flue gases in the maritime industry is presented and discussed. Combustion engine is a main source for electric energy consumed by all electrical devices in ships and yachts. Currently, the classic compression refrigeration systems driven by electricity generated in the generators are used for production of cold water used in AC units. Conversion of fuel energy into mechanical and electrical energy is related with creation of a significant amount of heat which is irretrievably removed. The proposed application of heat driven ejection refrigeration system may be thought as an excellent example of an industrial application with a strong potential for implementation. At the same time it combines all positive aspects of the environmentally-friendly cold production approach using clean technology and meets all standards in the use of ecological working fluids.

1. INTRODUCTION

Medium-speed piston engines are usually used on maritime vessels and yachts to propel the ships and to drive the generators to produce electricity. Piston engines works as a heat engines converting the thermal energy of the fuel to mechanical work and further to electrical energy. Usually, the machinery systems fitted on ships and yachts are designed to work with maximum efficiency and run for long periods. Analyzing data of various piston engine manufacturers it can be concluded that engines efficiency is in the range of 30-45 %. This means that the energy in the fuel cannot be completely converted into mechanical work. The most common and maximum energy loss from the engine is in the form of waste heat. This loss of heat has to be discharged and transferred to ambient by cooling fluids such as central cooling water system to avoid malfunction of the engine or breakdown of the machinery equipment. Utilization of the waste heat is a promising way for improvement the overall energy efficiency of the vessel. Various types of waste heat recovery technologies and the potential for ship-owners to decrease the fuel consumption costs, reduction of the emissions, and the positive effect on the ship EEDI (Energy Efficiency Design Index) were presented in report of Diesel, MAN & Turbo (2011) and paper by Shu et al. (2013). The energy retrieve from the engine depends to a great extent on the size of the main engine of the ship, its load and ambient temperatures. The engine size, operation route, loading condition and environment should be taken into consideration before choosing an appropriate way to waste heat utilization. In any heat recovery situation it is essential to know the amount of the recoverable heat amount and also how it can be applied. Potential benefits of the ship waste heat recovery using a supercritical ORC (Organic Rankine Cycle) were discussed by McCracken and Buckingham (2015). Shu et al (2017) presented the evaluation of the ORC applied for the waste heat recovery based on thermal-economic model. Also, one of the commonly known application is use the waste heat to drive the absorption refrigeration unit. The main problem with absorption systems is that they required more space and they are significantly expansive than conventional vapor-compression systems. This is because absorption systems have more components and as the heat and mass transfer of absorption equipment is poor so that large surface areas are required. Ezgi (2014) presented design and thermodynamic analysis of a water-lithium bromide absorption heat pump as an HVAC system for a naval surface ship application. Despite that the absorption system can be used in naval ships, they are not suitable for yachts due to its machinery room space restrictions. Typical distribution of the waste heat for piston engine is shown in Table 1. It should be noted that not all of the heat sources listed can be recovered separately, rather as combinations, depending on the engine unit. As showed in Table 1 part of heat from high temperature jacket cooling circuits and exhaust gases can be recovered without major technical shortcomings intended for heating on maritime vessels. Also, due to promising temperature level of the waste heat source it is possible to use this heat as a motive source for the ejection refrigeration system operating for air-conditioning purposes, Ezgi and Girgin (2015). As shown by Butrymowicz et al. (2014) and Śmierciew et al. (2017) the ejection refrigeration systems can operate with low-temperature heat source. Moreover, the temperature requirements for air-condition system on the ship are favorable for ejection systems, since design temperature for summer is 24 ~ 28 °C, Yan et al. (2011).

Table 1: The typical distribution of waste heat for piston engine

Energy source	Temperature	Portion of fuel energy
Exhaust gas	~ 400-500 °C	~ 30%
Jacket water	~ 85 °C	~ 6.5%
High-temperature air charger	~ 90 °C	~ 9%
Lubricating oil	~ 70 °C	~ 5.5 %
Low-temperature air charger	~ 40 °C	~ 4 %
Generator cooling (on gen-set)	~ 35 °C	~ 1.3%
Engine radiation	~ 35 °C	~ 1.5%

This paper presents the Phase 1 of the project dealing with developing of the ejection air-conditioning system driven by waste heat. The potential application of the ejection refrigeration system operating for the air-conditioning purposes and driven by waste heat collected from the vessel piston engine has been analysed. Preliminary calculation of the proposed system operating with environmentally friendly new HFO (*hydro-fluoro-olefins*) group refrigerant R-1234zeE are presented and discussed. Proposed fluid fulfils the requirements of Regulation of the European Parliament and the EU Council No. 517/2014 enacted on April 16th, 2014, Regulation (EU) (2014). The geometry of the ejector designed for the specific case and performance operation line are analyzed in the paper.

2. EJECTION REFRIGERATION SYSTEM

The ejection refrigeration system (Fig. 1) is a modification of a well-known vapor compression cycle. Instead of pressurizing the refrigerant by a mechanical compressor, an ejector compresses refrigerant vapor flowing from the evaporator and discharges it to the condenser. The motive vapor is generated in the vapor generator which is heated by heat recovered system from the piston engine. Recovery system consists of the high-temperature heat-exchanger powered by exhaust gas and the low-temperature heat-exchanger powered by rest of thermal energy source collected from the piston engine.

The main difference between the ejection cycle and the conventional compression refrigeration cycle, besides elimination of a compressor, is that the ejection cycle requires three heat sources at different temperatures, namely the vapor generator level, which is the temperature of the waste heat source, a condensation level, which is the ambient temperature, and the evaporation temperature required for desirable cooling effect. The performance of the ejector depends on several quantities such as: operation pressures and temperatures at the ejector inlets and outlet, working fluid properties and the ejector geometry. The basic parameters describing the ejection cycle performance are mass entrainment ratio:

$$U = \frac{\dot{m}_e}{\dot{m}_g}, \tag{1}$$

where: $\dot{m}_g$ and $\dot{m}_e$ are the primary (motive) and the secondary fluid mass flow rates, respectively, and compression ratio:

$$\Pi = \frac{p_c - p_e}{p_g - p_e}, \tag{2}$$

where: p_c is condensation pressure, p_e is evaporation pressure, p_g is saturation pressure in the vapor generator. The coefficient of performance of the system is defined as:

$$COP = \frac{\dot{Q}_e}{\dot{Q}_g}, \tag{3}$$

where $\dot{Q}_g$ and $\dot{Q}_e$ are heat source capacity and cooling capacity, respectively. Thermal capacities are calculated as follows:

$$\dot{Q}_g = \dot{m}_g \Delta h_g, \tag{4}$$

$$\dot{Q}_e = \dot{m}_e \Delta h_e. \tag{5}$$

where Δh_g and Δh_e are enthalpy differences at outlet and inlet of the vapor generator and the evaporator, respectively.

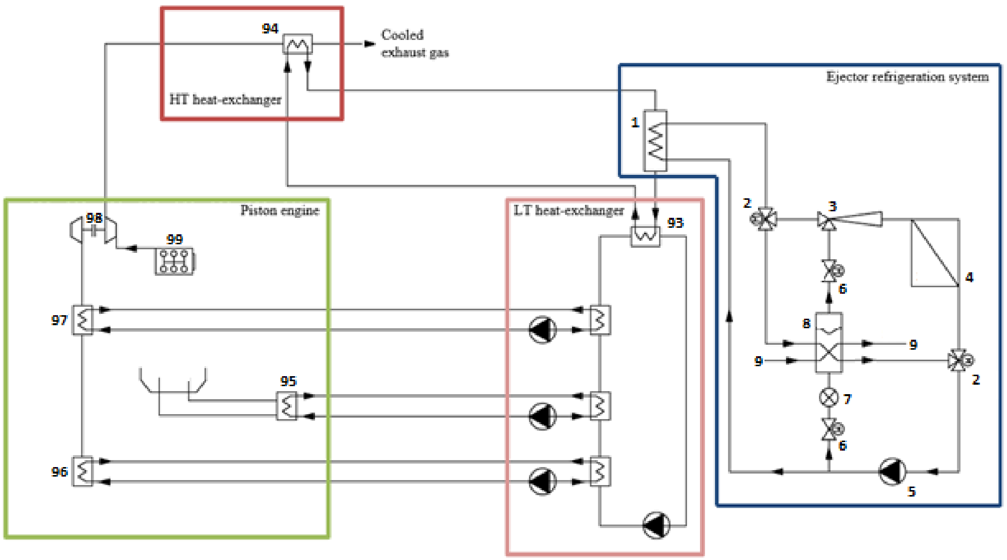

Figure 1: Schematic diagram of the waste heat recovery system that drives the ejection refrigeration system 1– vapor generator; 2-three-way valve; 3-ejector; 4-condenser; 5-pump; 6-shut-off valve; 7-expansion valve; 8-evaporator; 9- cooling/hot water piping system; 99-piston/jacket water cooling; 98-turbine; 97-HT charge air; 98-LT charge air 95-lubricating oil; 94-exhaust gas heat-exchanger; 93-low-temperature heat-exchanger.

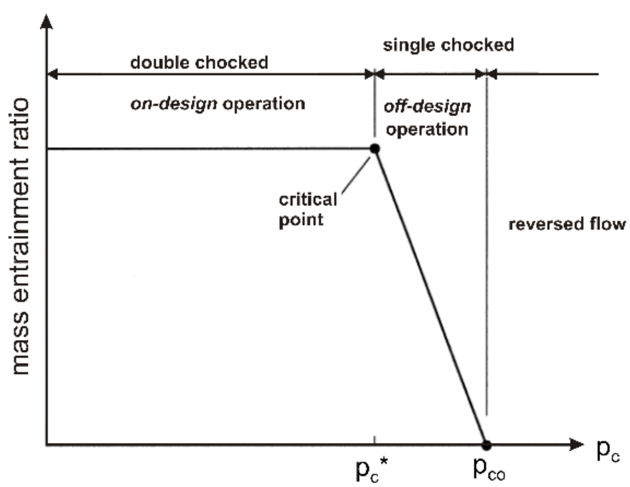

Figure 2: The performance curve and operational modes of the vapor ejector

A typical performance curve of the vapor ejector for primary and secondary pressures (motive vapor pressure and evaporator pressure, respectively) is shown in Fig. 2. Three operation modes are distinguished. The most desirable operation mode is a critical mode, also called "on-design" operation mode. Under the critical mode the ejector operates with maximum mass entrainment ratio. Independently of the type of operation mode, the primary fluid reaches the supersonic velocity during expansion in the motive converging-diverging nozzle. The flow is also chocked. In the critical operation mode secondary flow reaches the speed of sound because of favorable conditions related with momentum transfer and pressure difference between both streams. When the secondary fluid achieves the speed of sound the flow is chocked. Now both fluids are chocked and remain in this state as long as the back-pressure is lower than the critical pressure. In the critical mode, the shock wave which creates the compression effect is expected downstream the flow, either at the constant area mixing chamber or at the diffuser. However, if the back pressure increases the shock wave moves upstream the flow. For the back-pressure lower than the critical pressure the shock wave influences neither primary nor secondary mass flow rate. Therefore, the entrainment ratio is constant for the back-pressures lower than critical pressure. When the back-pressure is equal to critical pressure the shock wave is located exactly where the secondary flow achieves the speed of sound and where it was chocked. With a further increase in the back-pressure the shock wave moves upstream into the suction chamber. As a result, the pressure of the primary and the secondary fluid increases and velocity of both fluids subsequently decreases. Secondary flow cannot reach the speed of sound and the mass flow of the fluid is lower than in the critical operation mode. This mode is also called "off-design". In this mode the mass flow rate of the secondary flow is described by the equation continuity equation. The increase in the back-pressure leads to decrease in the velocity of the secondary fluid. Consequently, the mass flow rate decreases. Primary flow in the off-design operation mode is still chocked. This is the reason why the mass entrainment ratio decreases with the increase in the back-pressure in the subcritical or off-design operation mode. Under particular conditions, i.e. with the back pressure higher than breakdown pressure, the pressure in the suction chamber exceeds evaporation pressure and thus the reverse flow appears, whereas the ejector stop its operation.

3. RESULTS

The analysis of the performance of the ejection system in terms of mass entrainment ratio and compression ratio was performed for refrigerant R1234ze(E). Performance of the system operating with the ejectors of two different geometries were analyzed. The first ejector geometry was developed based on model proposed by Huang et al. (1999) and Kumar et al. (2014) for assumed operation conditions: motive source heat flux Q_g = 100 kW, motive temperature and pressure: t_g = 130 ºC, p_g = 4 MPa (supercritical conditions), expected cooling capacity Q_e = 25 kW, evaporation temperature t_e = 5 ºC, superheating in evaporator ΔT_e = 5 K. Thermodynamic cycle in the pressure-enthalpy diagram is shown in Fig. 3. It was assumed that the condenser will be cooled by sea water of temperature 35 ºC as required by the standard ISO 7547:2002(E). It was also assumed that temperature increase in circulation pump is negligible. The second geometry was obtained by slight modification of the mixing chamber and the diffuser dimensions.

The performance lines for both geometries were found. Numerical model based on Chen et al. (2013) was built. Real gas properties were applied, Akasaka (2010). Two cases for the evaporator temperature t_{e1} = 0 ºC and t_{e2} = 5ºC were analyzed. The motive temperature is assumed as constant t_g = 130°C for both cases and condensation temperature is varied. The results of obtained for the basic geometrical diameters of the two analyzed ejectors are shown in Table 2.

Table 2. Geometrical parameters of ejector

	Geometry No.1	Geometry No.2
Nozzle throat diameter	5.7 mm	5.7 mm
Diameter of the nozzle outlet	11.8 mm	11.8 mm
Diameter of mixing chamber	20 mm	18 mm
Diameter of diffuser	60 mm	60 mm

Since motive parameters are constant then the diameters of the nozzle throat and nozzle outlet are the same for both geometries. As it was stated previously the mixing chamber diameter was modified for the second geometry.

The effect of the condensation temperature on mass entrainment ratio are shown in Fig. 4 and Fig. 5. In Fig. 4 the performance lines for evaporation temperature t_{e1} = 0 ºC.

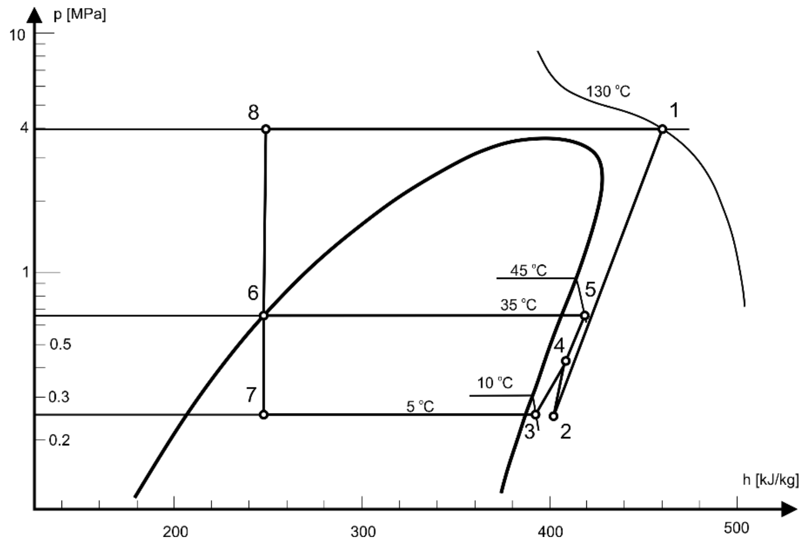

Figure 3: The ejection refrigeration cycle for R1234zeE refrigerant and assumed operation parameters

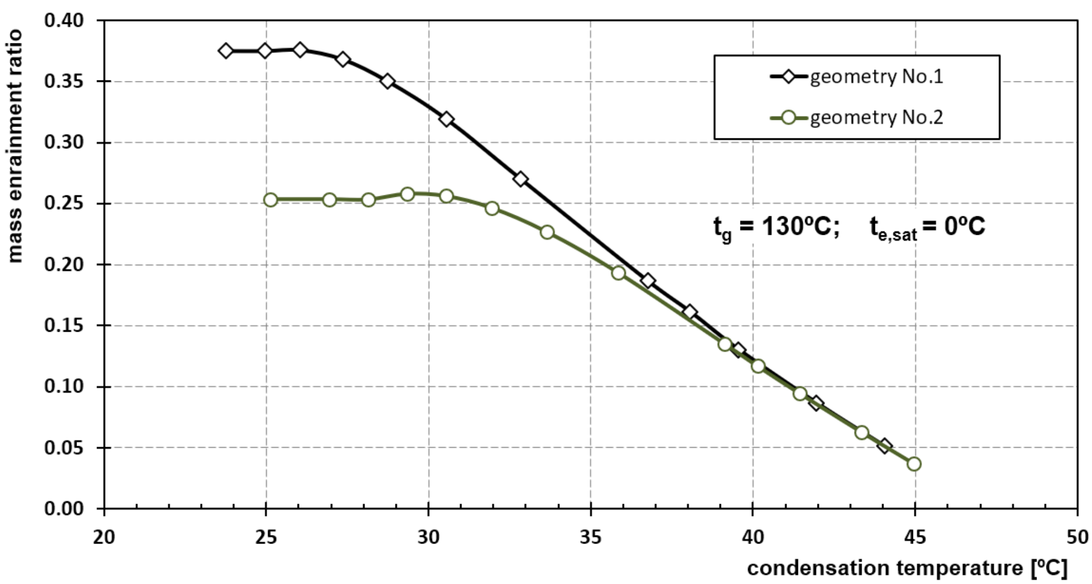

Figure 4: The effect of condensation temperature on mass entrainment ratio for two analyzed ejector geometries

The results show that for both tested geometries the performance line is typical for the gas ejector. It is seen from this figure that for all investigated cases the ejector operates both under on-design and off-design conditions. However, for the case with geometry No. 1 the ejector operates with higher mass entrainment ratio. The maximum mass entrainment ratio predicted by the model is U_1 = 0.37. Simultaneously, the critical condensation temperature is t_{c1*} = 26 °C. With this critical condensation temperature, the ejector driven by t_g = 130 °C starts to operate under off-design conditions. With a further increase of the condensation temperature the mass entrainment ratio consequently decreases. The slope of the performance line indicates that the ejector of the geometry No. 1 will not operate for temperatures of condensation above 45 °C. It must be noted that ejector is dedicated for A/C system for yachts and therefore high condensation temperature can be expected especially in hot climate. Because of this the critical condensation temperature at level t_{c1*} = 26 °C can be though as moderate in proposed system. On the other hand, the mass entrainment ratio at level U_1 = 0.37 is relatively high compared to initially assumed value of $U \approx 0.25$.

Therefore, the geometry was slightly modified in order to reduce mass entrainment ratio and extend the on-design operation of the ejector. The diameter of the mixing chamber was modified using trial-and-error method. The performance line for second geometry is also shown in Fig. 4. It is seen that decreasing of mixing chamber diameter reduces the mass entrainment ratio and extends the on-design operating regime, as expected. The maximum reported mass entrainment ratio was U_2 = 0.25. For geometry No.2 the critical condensation temperature is t_{c2*} = 31 °C. Analogically as for geometry No. 1 the slope of the performance line predicted by the model indicates that the ejector with geometry No. 2 will not operate for temperatures of condensation above 45 °C.

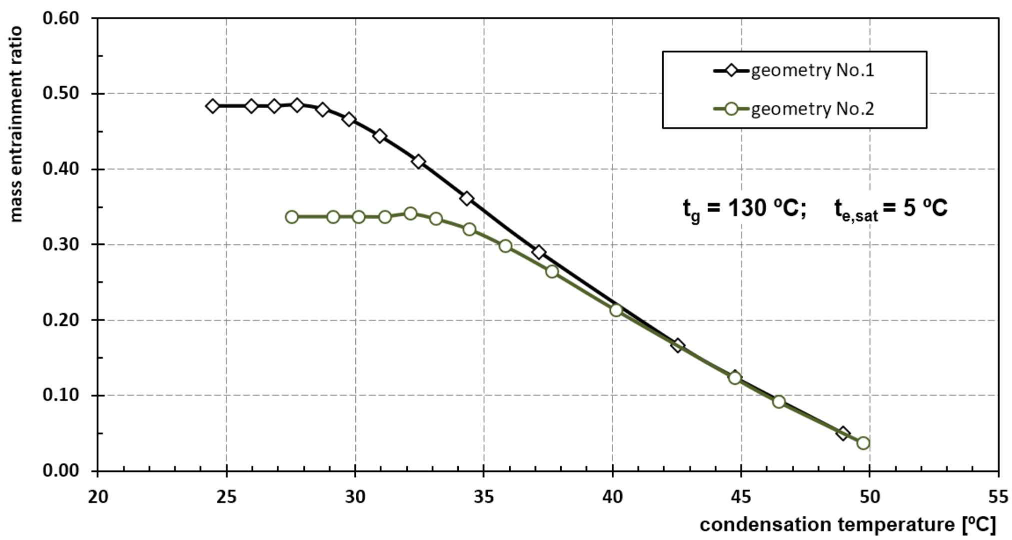

Figure 5:. The effect of condensation temperature on mass entrainment ratio for two analyzed ejector geometries

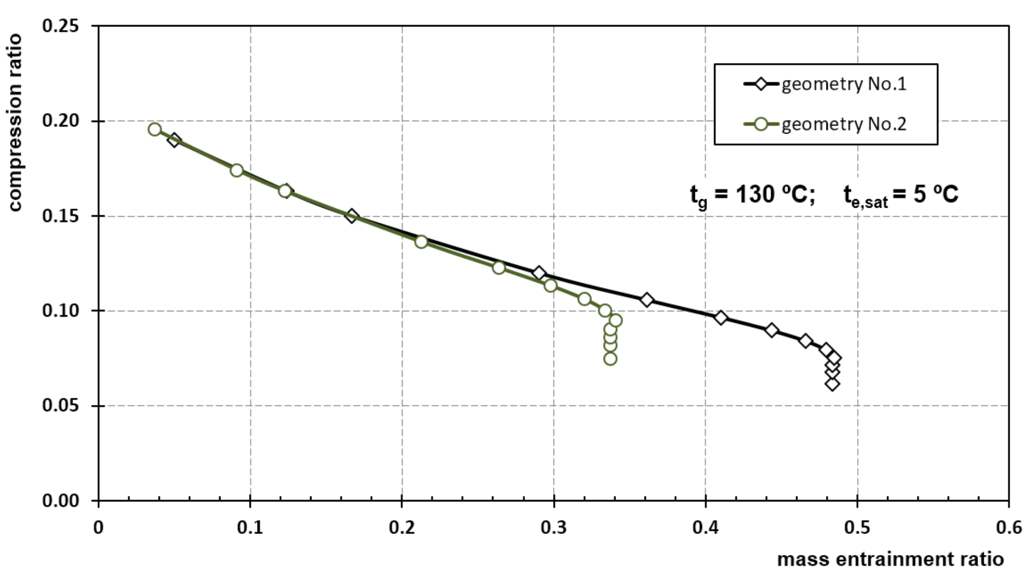

Figure 6: Compression ratio versus mass entrainment ratio for two analyzed ejector geometries

Taking into consideration the requirements for air temperature in ship cabins during summer it can be postulate that the evaporation of refrigerant at temperature 0 ºC is not necessary, because the temperature of fresh supply air should be approximately 19 ºC. Cooling the air up to temperatures slightly above 0 ºC is not an effective solution, because in the next step the air should be heated up to provide proper supply air temperature. The second reason why one should avoid the excessive cooling of the air is to reduce the potential risk of condensation of water vapor on a cold air-cooler surface. Operation performance of ejector of both geometries was tested for evaporation temperature t_{e2} = 5 ºC. Again, the motive temperature was t_g = 130 °C. Results are presented in Fig. 5.

Figure 5 shows that increasing of evaporation temperature lead to increase of the mass entrainment ratio. For the ejector of the geometry No. 1 the mass entrainment ratio is U_1 = 0.49 and the critical condensation temperature is approximately t_{c1*} = 29 °C. The slope of the performance line in the off-design regime indicates that ejector should operate for condensation temperature up to 50 °C. For the case of ejector of the geometry No. 2 the maximum entrainment ratio is U_2 = 0.33 and the critical condensation temperature is approximately t_{c2*} = 34 °C. The temperature above which the ejector will not operate is predicted as 50 °C. It must be pointed out that performance of the both ejectors are very promising. The model predicts that on the off-design operating regime for temperature t_{c2} = 40 °C both ejector will have mass entrainment ratio at level of 0.20.

The performance lines in terms on compression ratio versus mass entrainment ratio for t_{e2} = 5 ºC are shown in Fig. 6. The figures show that for the evaporation temperature t_{e1} = 5 ºC the compression ratios corresponding to critical condensation temperatures are Π_{1*} = 0.08 and Π_{2*} = 0.10. The verticals line represents the on-design operating regime of ejector and the slope lines represents the off-design operating regime.

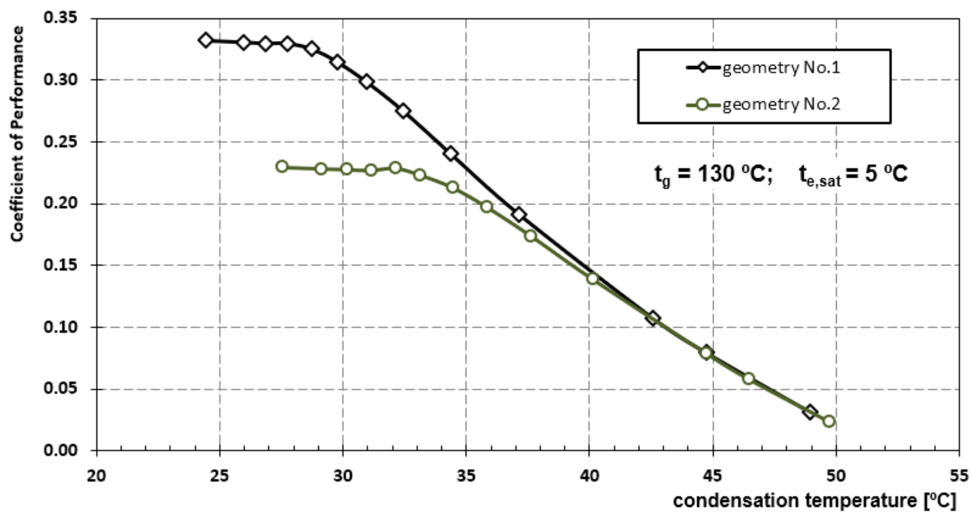

Figure 7: Coefficient of performance versus condensation temperature for two analyzed ejector geometries

The influence of the condensation temperature on the coefficient of performance *COP* of the ejection system is presented in Fig. 7. The coefficient of performance is directly related with mass entrainment ratio, therefore like in the case of mass entrainment ratio, the maximum level of *COP* = 0.33 was obtained for the geometry No. 1 and condensation temperatures below t_{c1*} = 29 °C (i.e. the on-design operation regime). The lowest reported *COP* = 0.23 at the on-design operation was obtained for the geometry No. 2. With mass entrainment ratio above the critical condensation temperature t_{c*} the *COP* of the tested ejector consequently decreases with subsequent increase in the condensation temperature.

4. CONCLUSIONS

The paper presents the results of the numerical prediction of the operation of the ejection refrigeration system for the air-conditioning purposes. Refrigerant R1234zeE was selected as a working fluid. Based on the presented results the following conclusion can be drawn:

- The proposed ejection refrigeration system can utilize the waste heat collected from the piston engine as the motive source.
- Wide range of the disposable temperature of the source allows to use the supercritical state of refrigerant to drive the system.
- Reduction of the mixing chamber diameter leads to extension of the on-design operation regime of the ejector.

- The achievable coefficient of performance can be achieved for analyzed operation conditions
- Temperature t_{c1*} = 29 °C and t_{c2*} = 34 °C were predicted as the critical condensation temperatures for operation temperatures t_e = 5 ºC and t_g = 130 ºC.

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