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CONCEPTUAL DESIGN OF A TRANSCRITICAL HEAT PUMP WITH CO_2 REFRIGERANT FOR WASTEWATER HEAT RECOVERY APPLICATION RESEARCH

Adam Miča[(a)], Andrej Kapjor[(b)], Jozef Jandačka[(c)], Michal Holubčík[(d)], Alexander Čaja[(e)]

Žilinská univerzita v Žiline
Žilina, 010 26, Slovakia
[(a)] adam.mica@fstroj.uniza.sk

ABSTRACT

Heat pumps can be more environmentally friendly than heating sources using the combustion of solid fuels. To avoid environmental damage from refrigerant leaks, transcritical heat pumps using CO_2 as a working fluid could be used. To increase the efficiency of the system, a two-phase ejector with an adjustable needle is considered in this work. Considering the type of operation, heat transfer using plate heat exchangers is considered in the paper. The paper deals with the design of an experimental transcritical heat pump concept with CO_2 refrigerant for use in residential wastewater heat recovery applications.

Keywords: Transcritical heat pump, Carbon Dioxide, Heat recovery, Wastewater

1. INTRODUCTION

The refrigeration sector - including air conditioning - consumes about 20% of the overall electricity used worldwide (IIR, 2019). For this reason, it is necessary to reduce the energy dependence of industry and buildings. (European Commission, 2019). One of the possible solutions is wastewater heat recovery.

Domestic hot water (DHW) is usually heated from 50°C to 60°C. (Arnell, et al., 2017) found that water enters sewers from buildings with a temperature of 20 °C. (Cecconet, Daniele, et al., 2020) investigated the wastewater heat recovery system for a multi-functional building, while providing all the heat and cooling needs, the heat pump consumed 59% less energy. Heat pumps are suitable for wastewater heat recovery. Synthetic refrigerants can cause global warming or ozone depletion. An alternative is the use of natural refrigerants.

Carbon dioxide (CO_2, R-744) is a natural refrigerant. Critical point of carbon dioxide is at a pressure of 7.37 MPa and a temperature of 304.1 K, therefore it is necessary to work in transcritical conditions in heating applications. A conventional vapor compression cycle using CO_2 in transcritical operation has a lower COP compared to conventional cycles. This is due to higher operating pressures compared to conventional refrigerants. It causes rather the need for the compressor power consumption. (Kim, Pettersen and Bullard, 2004). This paper considers the energy of the refrigerant stream in the expansion process as the first stage of compression using an ejector.

To increase the COP of the system the variable speed drive (VSD) regulation of the compressor is assumed. The performance of a two-phase ejector is significantly governed by its characteristic dimensions. Therefore, the COP of the system decreases when it is operating under design conditions (Lee, Kim, & Kim, 2014). This paper focuses on, an ejector with an adjustable needle to increase the COP system by variable geometry (Gullo, Kærn, Haida, Smolka, and Elbel, 2020). The paper aims to conceptual design a of a small transcritical heat pump with CO^2 refrigerant using a geometry variable two-phase ejector.

2. EXPERIMENTAL HEAT PUMP

The first step in heat recovery is to identify the energy potential of the heat source. The value of the energy potential of wastewater expresses is flow and temperature over time. The measurement of wastewater flow is very complicated due to the heterogeneity of the mixtures. It is assumed that the amount of water delivered to the residential building is the same as the amount of water that is left to the sewer. Then the flow of the sewage will be the same as the consumption of hot and cold water. The average temperature in sewers in the Czech Republic is 15.2 °(Cecconet, Daniele, et al., 2020). The temperature in the sewers is lower due to the heat losses of the sewer pipe located in the ground. Based on this information, it is possible to predict the thermal performance of wastewater from residential buildings.

The experiment aims to determine the system parameters (pressure, temperature, and mass flow) of the experimental heat pump with CO_2 refrigerant under constant and variable operating conditions. The measurement will take place in laboratory conditions, which will be set to correspond as much as possible to the real conditions of the standard operation of the heat pump.

Figure 1 shows the diagram of the experimental device. The experimental setup consists of a heat pump, three pumps (P1, P2, and P3), a heat exchanger (VT), a heat source (ZT), and a storage tank (AN).

Refrigerant vapors are compressed at the compressor into the transcritical region. The refrigerant flows into the oil separator, where the oil is separated from the refrigerant. Next, the refrigerant flows into the gas cooler, where heat transfer occurs between the refrigerant and the secondary fluid. The refrigerant at a lower temperature from the gas cooler flows into the ejector, where the expansion effect occurs due to the flow through the nozzle. At the same time, the refrigerant vapors flow from the evaporator into the suction nozzle of the ejector. In the ejector, these two flows are mixed and a multi-phase flow occurs . Furthermore, the system includes a liquid separator (S) to separate the liquid fractions of the refrigerant from the vapor. In case the liquid separator cannot completely separate the liquid refrigerant, a suction accumulator (AS) is installed before the compressor. The vapors are drawn into the compressor and the cycle repeats. The liquid refrigerant is separated in the separator (S) and flows through the expansion valve to the evaporator . In the expansion valve, the liquid refrigerant expands to the conditions of the evaporator, where heat transfer occurs between the refrigerant and the secondary fluid.

The heat source for the experiment will be cold water. We will heat the cold water with a heat source and adjust the flow using a pump with VSD regulation (P1) so that the temperature and flow of cold water correspond to the temperature of wastewater in residential buildings. The heat exchanger (VT), in which the sewage will transfer energy, must be cleaned regularly to avoid fouling due to the heterogeneous of sewage water. To avoid service interventions in the heat pump, a circuit with a 30% mixture of ethylene glycol and water is placed in the system. This is used for heat transfer between sewage water and CO_2. The circulation of the mixture is ensured by the circulation pump (P2).

The cold water is heated in the gas cooler of the heat pump (GC) and flows into the storage tank. The flow of heated water will be adjusted as needed by the pump with VSD regulation (P3).

During the measurements, the frequency of the compressor will vary from 30Hz to 60Hz. Depending on the frequency of the compressor, the operating parameters of the system will change. The geometry of the ejector will adapt to changes in operating parameters. It will be changed by inserting or extending the needle. During this process, temperature, pressure, and refrigerant flow will be continuously measured. Pressure, temperature, and flow measurements will be performed at the measurement points. The temperature sensors will be placed on the outside of the pipe and sufficiently isolated.

The main goal of the heat pump experimental assembly is to investigate a wide range of operating parameters and dependencies between individual components with regard to their safety. The next step in the research will be to compare different types of components and apply the knowledge to the selection and precise design of components to achieve the best COP.

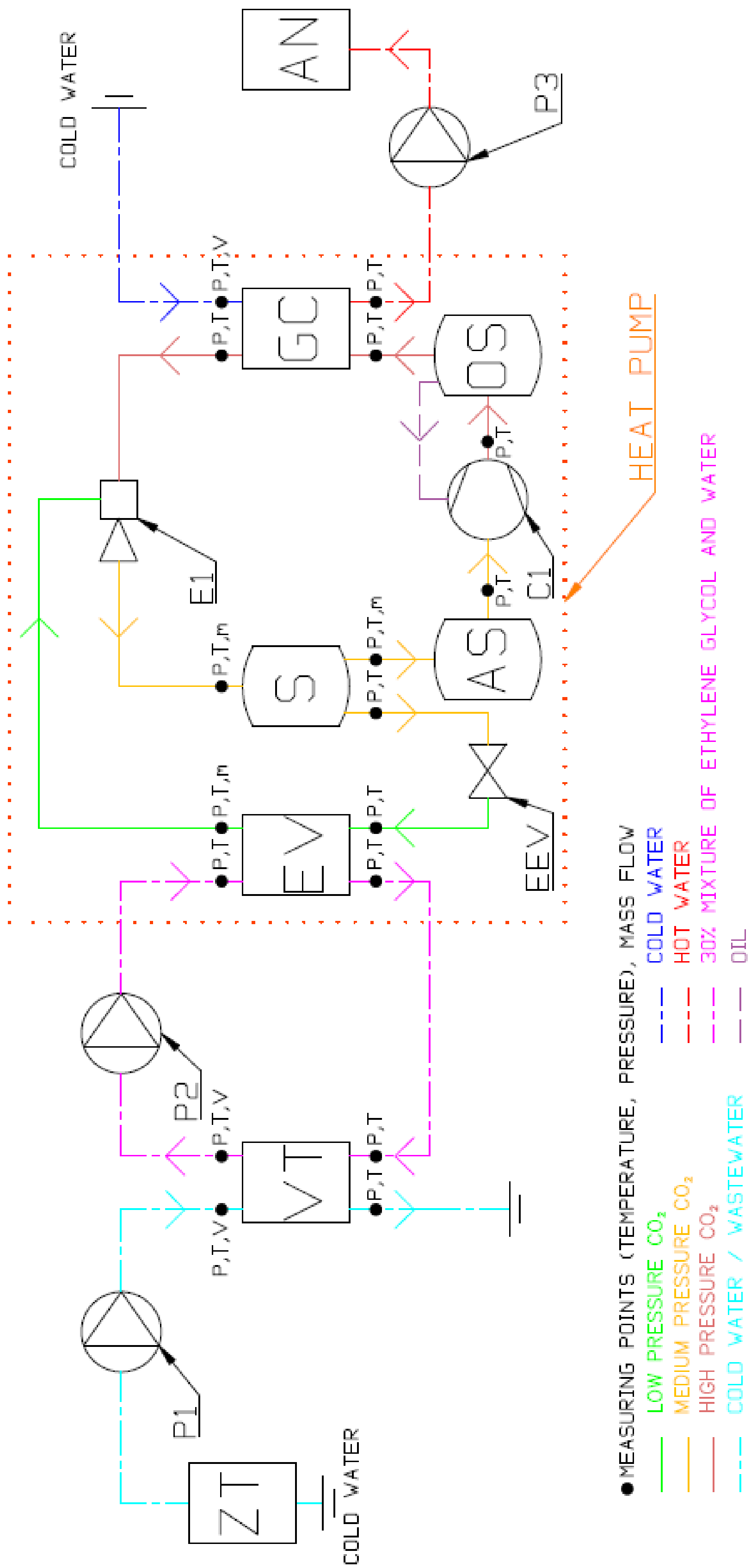

Figure 1: Diagram of experimental heat pump

2.1. Main components

The gas cooler is designed as a plate heat exchanger with 20 heat exchange plates with a total heat exchange area of 0.738 m^2. The ejector is designed for an inlet temperature of 35°C. The gas cooler must cool the CO_2 from 94°C (the compressor outlet) to 35°C (the ejector inlet) at optimal pressure. The heat performance of the gas cooler at nominal operation is 6kW.

The evaporator is designed as a plate heat exchanger with 10 heat exchange plates with a total heat exchange area of 0.272 m^2. The design is covered to overheat the vapor by 5°C. The inlet temperature of CO_2 to the evaporator is -10°C at a pressure of 26.4 bar. The heat performance of the evaporator at nominal operation is 3.8 kW. The mixture of ethylene glycol and water is cooled from 0°C to -6°C.

The heat exchanger (VT) is designed to transfer heat from water with a temperature corresponding to sewage water and ethylene glycol. It is designed as a plate heat exchanger with 10 heat exchange plates with a total heat exchange area of 0.09m^2. The heat performance of the heat exchanger at nominal operation is 3.8kW.

The system used a semi-hermetic reciprocating compressor with two pistons with an input power of 2.2kW. The compressor is equipped with a frequency converter that ensures variable operating conditions. The nominal operation of the compressor is at 50Hz. The operating range is from 30Hz to 60Hz.

The Components of the system are described by mathematical models taking into account operating parameters. Mathematical models of heat exchangers and compressors are well-known in the literature, due to the mathematical model of the two-phase ejector will be described in the next section.

3. MATEMATICAL MODEL OF TWO-PHASE EJECTOR

Figure 2 show the diagram of the ejector and the course of the expansion process through the ejector in the p-h diagram. The thermodynamic process of the ejector in the transcritical CO_2 cycle is based on the flow of compressed CO_2 through the motive nozzle, where the expansion process and two-phase flow occur. CO_2 expands from the pressure in the gas cooler (p_m) to the pressure in the receiving chamber (p_{m1}). At the same time, the enthalpy changes from h_m to h_{m1} and the refrigerant velocity from v_m to v_{m1}. The flow of refrigerant through the ejector creates a negative pressure at the intake inlet of the ejector, where CO_2 flows in as vapor from the evaporator. The refrigerant expands from the pressure in the evaporator (p_s) to the pressure in the receiving chamber (p_{s1}). At the same time, the enthalpy changes from h_s to h_{s1} and the speed to v_{s1}. In the mixing section, the flow from the evaporator and the gas cooler are mixed and an equilibrium state is reached, i.e. CO_2 has a pressure p_{mix}, an enthalpy h_{mix}, and a velocity v_{mix}. Next, CO_2 flows into the diffuser, where kinetic energy is converted into internal energy, which increases the pressure from p_{mix} to p_3 (Zhang & Yamaguchi, 2021).

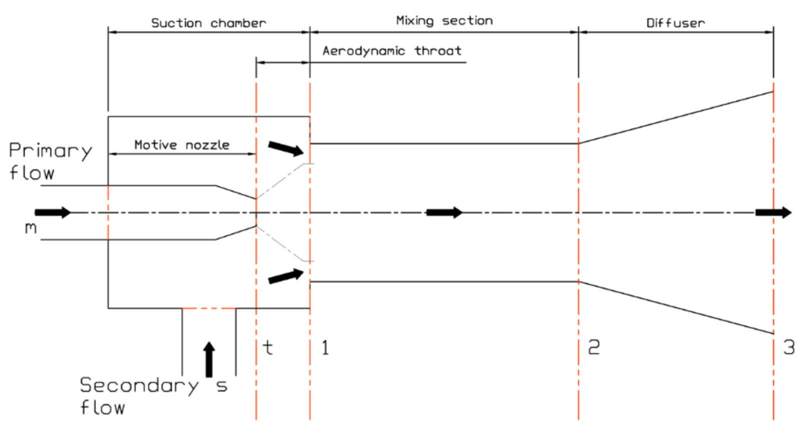

Figure 2: Schematic diagram of typical vapor ejector

In the following sections, the one-dimensional mathematical model of the two-phase ejector is described. The ejector can be divided into four submodels: the motive nozzle model, the suction nozzle model, the mixing section model, and the diffuser model. Known variables that enter the mathematical model are the area and efficiency of all submodels.

3.1. Motive nozzle model

The isentropic efficiency of the motive nozzle η_m, the inlet pressure p_m, and the temperature at the inlet T_m, are known, then the flow velocity at the outlet of the motive nozzle v_t can be derived by estimating the pressure p_t using an iterative method.

$$h_{\mathrm{t}} = h_{\mathrm{m}} - \eta_{\mathrm{m}}\left(h_{\mathrm{m}} - h_{\mathrm{t,is}}\right) \qquad \text{Eq. (1)}$$

where h_t, and h_m are the enthalpies at the nozzle outlet, inlet respectively. $h_{t,is}$ is the enthalpy at the nozzle throat assuming an isentropic expansion. The flow velocity at the motive nozzle can be derived from the equation of conservation of energy between the input and output of the motive nozzle:

$$h_{\mathrm{m}} = h_{\mathrm{t}} + \frac{\mathrm{v}_{\mathrm{t}}^2}{2} \qquad \text{Eq. (2)}$$

the flow velocity v_t must equal the speed of sound under the given conditions. The speed of sound was derived according to (Lund H, & Flatten T. 2010).

$$a^{-2} = a_w^{-2} + \frac{\rho}{T}\frac{C_{p,v}C_{p,l}(\zeta_l - \zeta_v)^2}{C_{p,v} + C_{p,l}} \qquad \text{Eq. (3)}$$

$$a_w^{-2} = \rho\left(\frac{\epsilon_v}{\rho_v a_v^2} + \frac{\epsilon_l}{\rho_l a_l^2}\right) \qquad \text{Eq. (4)}$$

$$\zeta_k = \left(\frac{\partial T}{\partial P}\right)_{s_k} = \frac{T\beta_k v_k}{c_{p,k}} \qquad \text{Eq. (5)}$$

$$C_{p,k} = \rho_k \epsilon_k c_{p,k} \qquad \text{Eq. (6)}$$

where a_v and a_l are the values for the single phase speed of sound of a saturated vapor and liquid, respectively, whilst ϵ_v and ϵ_l are the void fraction of the vapor and liquid phases, respectively. a_w is the speed of sound considering only the pressure to be at equilibrium, and $k \in \{v, l\}, \zeta_k$ and $C_{p,k}$ are defined as follows:

$$\zeta_k = \left(\frac{\partial T}{\partial P}\right)_{s_k} = \frac{T\beta_k v_k}{c_{p,k}} \qquad \text{Eq. (7)}$$

$$C_{p,k} = \rho_k \epsilon_k c_{p,k} \qquad \text{Eq. (8)}$$

where β is the thermal expansion coefficient, c_p is the constant pressure specific heat and ϵ_k is the void fraction, v is specific volume, T is temperature (José M. Cardemil, & Sergio Colle 2012). The mass flow of the motive nozzle $\dot{m}_m$ is defined as follows:

$$\dot{m}_m = \rho_{\mathrm{t}} A_{\mathrm{t}} \mathrm{v}_{\mathrm{t}} \qquad \text{Eq. (9)}$$

where A_t is cross-sectional of nozzle, desinty at nozzle ρ_t is defined as follows:

$$\rho_\mathrm{t} = \frac{1}{\frac{x_\mathrm{t}}{\rho_\mathrm{g,t}} + \frac{1 - x_\mathrm{t}}{\rho_\mathrm{l,t}}} \qquad \text{Eq. (10)}$$

where x_t is the quality, ρ_g, and ρ_l are the densities of the gas and liquid fraction respectively.

3.2. Aerodynamic throat and suction nozzle

Due to the high speed of fluid flow through the ejector between the motive nozzle and the mixing part, further expansion occurs. The following part describes a mathematical model of aerodynamic throat and section nozzle developed by José M. Cardemil, & Sergio Colle.

To establish this pressure, firstly let us consider an isentropic expansion of the primary flow to a hypothetical pressure, equal to the secondary pressure inlet (p_{h1}=p_s). Thus, it is appropriate to apply the energy equation as follows:

$$h_t + \frac{\mathrm{v}_t^2}{2} = h_{h1} + \frac{\mathrm{v}_{h1}^2}{2} \qquad \text{Eq. (11)}$$

where the specific enthalpy is h_{h1} is funcion of p_{h1} and entropy of at the hypothetical pressure. v_{h1} is axial velocity of the primary flow at the hypothetical pressure, respectively. The actual area occupied by the primary flow is obtained by correcting A_{1h} according to an experimental parameter ψ (José M. Cardemil, & Sergio Colle 2012),

$$A_{m1} = \frac{A_{1h}}{\psi^2} \qquad \text{Eq. (12)}$$

the primary flow undergoes a series of oblique shocks as it expands in the suction chamber and thus this expansion process is considered using the constant isentropic efficiency η_m assumption, as follows:

$$\eta_m = \frac{h_t - h_{m1}}{h_t - h_{m1s}} \qquad \text{Eq. (13)}$$

the mass conservation equation for the entrained flow gives:

$$\dot{m}_s = \rho_{s1} A_{s1} \mathrm{v}_{s1} \qquad \text{Eq. (14)}$$

$$A_2 = A_{h1} + A_{s1} \qquad \text{Eq. (15)}$$

where A_{s1} is the effective area occupied by the secondary flow at Section 1. A_2 is the cross-sectional area of the constant area section. The expansion of the entrained flow can be considered as isentropic. Therefore, the density ρ_{s1} is estimated through the equation of state:

$$\rho_{s1} = \rho(p_{s1}, s_s) \qquad \text{Eq. (16)}$$

Is assumed to have the speed of sound at point s_1. Then v_{s1} is calculated according to the equations (Eqs. 11-12). Based on Keenan's theory, the pressure remains constant during the mixing process and thus we impose $p_{s1} = p_{m1} = p_2$ (José M. Cardemil, & Sergio Colle 2012).

3.3. Mixing section

During the fluid flow in the mixing section of the ejector, the pressure and cross-sectional area are constant (p_1=p_2, A_1=A_2). By law, conservation of energy applies:

$$\dot{m}_m\left(h_{m1}+\frac{\mathrm{v}_{m1}^2}{2}\right)+\dot{m}_s\left(h_{s1}+\frac{\mathrm{v}_{s1}^2}{2}\right)=\dot{m}_{mix}\left(h_2+\frac{\mathrm{v}_2^2}{2}\right) \quad \text{Eq. (17)}$$

$$\dot{m}_{mix}=\dot{m}_m+\dot{m}_s \quad \text{Eq. (18)}$$

$$x_{mix}=\frac{(\dot{m}_m x_{m1}+\dot{m}_s x_{s1})}{\dot{m}_{mix}} \quad \text{Eq. (19)}$$

enthalpy at the outlet of the mixing section is a function of pressure and quality h_2=h(p_2, x_{mix}), x_{mix} is the quality of the stream after mixing.

3.4. Diffuser

Considering that the following diffuser decelerates the flow until the stagnation condition is reached, the energy equation for the control volume between Sections 2 and 3 holds

$$h_2+\frac{\mathrm{v}_2^2}{2}=h_3 \quad \text{Eq. (20)}$$

$$\eta_d=\frac{h_3-h_2}{h_{3s}-h_2} \quad \text{Eq. (21)}$$

where h_3 is the discharge enthalpy of the ejector and the isentropic efficiency for a compression process is η_d. h_{3s} is the discharge enthalpy assuming an isentropic compression. The isentropic efficiency of the diffuser varies between 0.9 and 0.96. From Eqs. (20) to (21) it is possible to calculate the discharge pressure P_3 which must be higher than the condenser pressure, otherwise the ejector is not operating in the critical mode (José M. Cardemil, & Sergio Colle 2012).

3.5. Electric expansion valve model

An electric expansion valve is used in the experimental heat pump. The expansion process in the expansion valve will be considered as an adiabatic process. This can be defined as:

$$h_{EEV,in}=h_{EEV,out} \quad \text{Eq. (22)}$$

where $h_{EEV,in}$ and $h_{EEV,out}$ are enthalpies at the inlet and outlet of the electric expansion valve, respectively. By combining the above-described mathematical model of the ejector, the expansion valve, and the conventional mathematical models of the heat exchangers and the compressor, it is possible to create a comprehensive model of the transcritical heat pump.

4. CONCLUSIONS

To reduce the energy dependence of buildings, the most efficient approach to the use of energy is necessary. Wastewater discharged into the sewer still has a lot of energy that we can use. To use the low potential energy contained in the wastewater, it is advisable to use a heat pump. This paper presents the conceptual design of a small scale heat pump unit for wastewater heat recovery. The paper shows a diagram of a transcritical heat pump using a two-phase ejector as an expansion device. The ejector reacts to a change in compressor speed and operating conditions by changing its geometry using an adjustable needle. The next part presents a mathematical model of the two-phase ejector, which can be used to describe the operation and calculate the operating parameters of the system. Together with other mathematical models of the components, it is possible to create a comprehensive model of the heat pump. Such a model can be used for further investigation of other possible applications.

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