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Waste Heat Recovery of Industrial Exhaust Gas: Study of Variable Power Ejector Cooling Systems

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Abstract—Large quantities of energy are wasted in the form of heat. For a given waste energy source, when its temperature is high enough, a desirable recovery strategy is a first Organic Rankine Cycle (ORC) stage for electricity production, followed by using the remaining available energy, at lower temperature, for either building heating in winter or building cooling in summer. Thermal cooling can be achieved with an ejector cycle. However, a single ejector having a fixed geometry does not have the flexibility to follow the variation of the building cooling load, especially if the outside temperature, and thus the condensing pressure, fluctuates. The proposed solution is to use multi-ejector blocks, designed to handle the changing conditions. This theoretical study used a thermodynamic model built with Python that allowed to model the different multi-ejector design. The first multi-ejector block is one of scaled ejectors designed to provide cooling at the same condensing pressure. The second is the use of different geometry ejectors designed to handle variable condensing pressures. This study explores a range of operating conditions that could be obtained by using waste heat or heat of renewable sources. The simulations were done for a cycle with R-600a, a natural refrigerant showing the potential to provide high system flexibility. The results show that for fixed operating pressures with variable cooling load requirements, the simultaneous use of scaled ejectors would be preferred over the use of different geometry ejectors.

Keywords: Refrigeration System; Parallel Ejector; Multi-ejector; Variable Geometry Ejector; Waste Heat Recovery.

I. INTRODUCTION

The climate change challenge requires us to find alternatives to fossil fuels, and to use energy more efficiently. Waste heat from various industrial processes, such as exhaust gases [1], can be used to power an ORC that will produce electricity and reject low quality heat [2], [3]. However, this low-quality heat can still be used to supply building heating in winter, and building cooling in summer, through a thermal activated technology assuming that buildings are in the vicinity.

The idea of using waste heat to produce cooling is not a new concept [4]. One technology that has been extensively studied for the production of cooling from low quality waste heat is the use of an Ejector Refrigeration System (ERS). As an example, an ERS with refrigerant R-113, powered by the waste heat from a 2000 cc car engine, could produce 8.23 kW of cooling for 0.21 kW of electricity [5]. The interested reader may refer to [6] for a recent review on ejector-based refrigeration technology.

The most well-known ejector model found in the literature is that of Keenan et al. [7]. Utilizing the Venturi effect, a primary motive flow ($\dot{m}_1$) entrains a secondary suction flow ($\dot{m}_2$) for which the resulting fraction of entrained flow is called the entrainment ratio ($\omega = \dot{m}_2/\dot{m}_1$). The main working principles of the ejector are the mixing of both the primary and secondary fluid streams in a constant section area, the shock train formation, and the fluid recompression in a diffuser, where the kinetic energy is transformed into pressure. The compression ratio ($C_R = P_{out}/P_2$) is defined as the outlet pressure (P_{out}) divided by the secondary pressure (P_2). Both the entrainment and compression ratios are used to measure the ejector performance. The entrainment ratio is constant for a given set of outlet pressures. For this interval, the ejector performance is said to be "on-design" as both primary and secondary flows are choked. Beyond this critical outlet pressure point, the ejector performance is "off-design" or in the single choke range and a further increase of the outlet pressure may lead to back-flow or negative entrainment ratio [8]. This performance curve is presented in Figure 1.

Both the entrainment and compression ratios are mainly dictated by the operating conditions [9]. Therefore, the ERS is designed to work for a specific set of operating conditions. To overcome the design limitations of a fixed geometry ejector, different solutions are explored. The first option is the use of a Variable Geometry Ejector (VGE). Varying the throat area, using a needle or a spindle, modulates the primary mass flow rate [10], [11]. As expected, variable geometry ejectors offer greater flexibility by increasing or decreasing the entrainment ratio. However, Rand et al. [11] found that this method led to an exact opposite reaction with regards to the critical compression ratio, which is detrimental to the application. The alternate option for single phase ejectors is the use of multiple ejectors in a parallel configuration. This option has mainly been used to increase the performance of R-744 vapor compression refrigeration systems to counter the high expansion losses with this refrigerant [12] through the use of a multi-ejector block to replace the expansion valve. The technology has since been applied in large and medium scale systems. In the case of single-phase vapor ejectors, several works proposed the use of a multiple ejector setup where each ejector is designed to operate at a specific outlet pressure, allowing the ejector setup to operate over a wide range of outlet pressure by switching from one ejector to the other [13-18]. Thus, in all these works, there was always only one ejector in operation at the same time. Recently, Rand et al. [19] addressed the situation where the system capacity is changing over the period of operation. They performed tests on two parallel scaled ejectors working simultaneously. While performing tests in an open loop test bench and using two ejectors that are designed to provide scaled cooling loads at the same critical outlet pressure, they found that both ejectors led to the same performances whether they were operated separately or simultaneously.

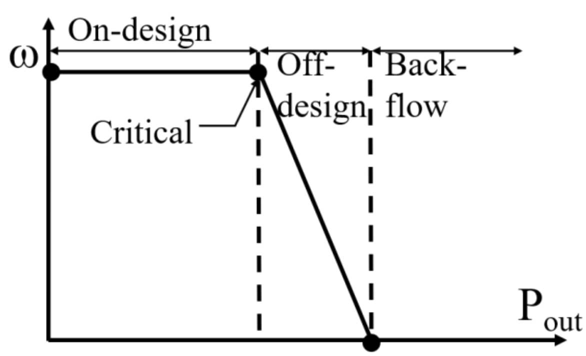

Figure 1. Schematic representation of a typical ejector performance curve.

This paper investigates the use of the low-quality heat rejected by an ORC system to supply building cooling over wide ranges of capacity and outlet pressure. The main objective of the present work is to compare variable throat diameter ejectors to that of scaled parallel ejectors working at the same time. A thermodynamic model of the ejector is developed and used to analyze the proposed system by fixing a primary pressure interval. The evaporator pressure is fixed throughout all the tests. The focus of this study surrounds the variation of the primary nozzle diameter on the ejector performances.

Moreover, considering the high global warming potential (GWP) of current hydrofluorocarbon (HFC) refrigerants, for which a phase-out is currently happening, and the risks associated with the use of the new generation of hydrofluoroolefins and hydrochlolofluoroolefins (HFO/HCFO) refrigerants, as they produce the "forever chemicals" Perfluoroalkyl and Polyfluoroalkyl Substances (PFAS) when released into the atmosphere, the simulations are done using the natural refrigerant R-600a, as proposed by Ciconkov [20].

II. Thermodynamic Modelling

The studied cycle relies on recovering high-temperature waste heat from the exhaust gas of an industrial process. The waste heat then supplies a combined ORC and ERS cycle. The ejector heat driven refrigeration cycle contains parallel ejectors in the form of a multi-ejector block. The entire cycle is viewed in Figure 2.

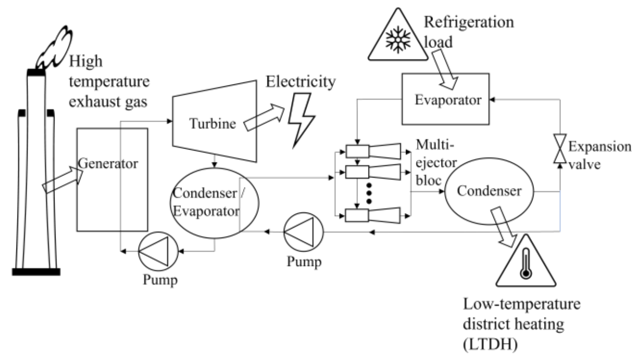

Figure 2. Schematic representation of a proposed combined ORC and ERS cycle to recover waste heat from the exhaust gas of a biomass co-generation plant.

The ejector model is based on the formulation of Metsue et al. [21] and has been extensively validated for various types of refrigerants and ejector geometries. For the ejector model, the overheat modules have a value of 5°C, which means that T_1 and T_2 are 5°C above their saturated values. The refrigerant used in this theoretical study is isobutane (R-600a). Once the ejector properties are calculated, it provides the primary and secondary mass flow rates as well as the critical outlet conditions.

The thermodynamic model predicts the performance and provides outlet data of the ejector based on specified input conditions and assumptions. The input data for the ejector are:

- Pressure and temperature at the primary and secondary inlets: P_1, T_1, P_2 and T_2.
- Primary nozzle throat and exit diameters D_t and D_e and constant area diameter D_y.
- Loss coefficients for the primary nozzle η_p (section 0-e), primary flow within the mixing chamber $\eta_{p,y}$ (section e-y), secondary flow within the mixing chamber η_s (section 0-y), mixing between the primary and secondary streams η_m (section y-m) and diffuser η_d (section 2-d).

The proposed model is based on the following assumptions:

- Flow is 1D, steady-state, adiabatic and its variables are uniform at each cross-section. The primary and secondary inlets as well as the outlet velocities are considered negligible.
- Isentropic compression and expansion coefficients are used to represent losses from friction losses along the walls and within the shear layer between the primary and secondary streams.

- Both primary and secondary flows start mixing up until a certain labelled mixing section within the constant area section. In this section, both pressures are assumed to be equal. This section is compound-choked for the critical operating regime.
- Real gas properties are obtained from the tabulated database COOLPROP available in the Python library.

Throughout the model, for each section, conservation of mass, energy and momentum is applied regarding combining inlet flows and corresponding outlet flows. The corresponding outlet data are:

- Primary and secondary mass flow rates, $\dot{m}_1$ and $\dot{m}_2$ respectively.
- Thermodynamic properties at each cross section.
- Critical outlet pressure, P_{crit}.

For more details, the reader can refer to [21]. The corresponding geometrical values referenced in the thermodynamic model are seen in Figure 3.

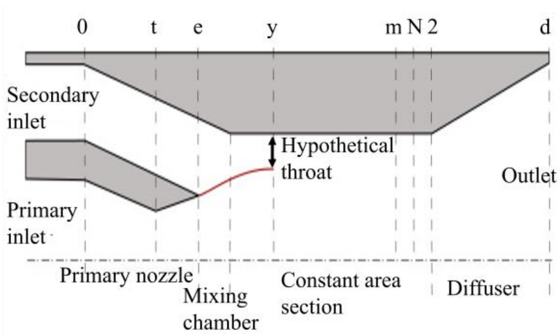

Figure 3: Thermodynamic model ejector geometry.

TABLE I. PARALLEL EJECTOR GEOMETRICAL PARAMETERS (DESIGN VALUES).

	Parameter	*Small ejector*		*Large ejector*	
D_t	Primary nozzle throat diameter (mm)	3.20	4.20	4.60	4.20
D_e	Primary nozzle exit diameter (mm)	4.48	4.48	6.44	6.44
D_y	Diameter of the constant area section (mm)	7.64	7.64	11	11
η_p	Loss coefficient of the primary nozzle (section 0-e)	0.9	0.9	0.9	0.9
$\eta_{p,y}$	Loss coefficient of the primary flow within the mixing chamber (section e-y)	1	1	1	1
η_s	Loss coefficient of the secondary flow within the mixing chamber (section 0-y)	1	1	1	1
η_m	Loss coefficient of the mixing between the primary and secondary streams (section y-m)	0.9	0.9	0.9	0.9
η_d	Loss coefficient of the diffuser (section 2-d)	0.9	0.9	0.9	0.9
AR	Area ratio (A_y /A_t)	5.7	3.3	5.7	6.9
NR	Nozzle ratio (A_e /A_t)	1.96	1.13	1.96	2.35

III. RESULTS AND DISCUSSION

The current study looks at the individual performance of both a small and large ejector. The entrainment ratio as a function of the outlet pressure is first used to characterize the behavior of the individual ejectors as predicted by the thermodynamic model.

The entrainment ratio as a function of the outlet pressure is presented in Figures 4 and 5. The curves are done for all the combinations of the following inlet conditions: P_1=869.2 kPa and P_1=1986.5 kPa, P_2=157.0 kPa and P_2=206.5 kPa for the ejector described in Table 1. Additional tests are done at P_1=869.2 kPa and P_2=157.0 kPa for both ejectors, but with a different throat diameter value of D_t=4.2 mm. Figure 4 displays the entrainment ratio as a function of the outlet pressure at P_1=869.2 kPa for the small and large ejectors at (a) P_2=157.0 kPa and (b) P_2=206.5 kPa. Figure 5 shows the entrainment ratio as a function of the outlet pressure for the same values of P_2 but at P_1=1986.5 kPa.

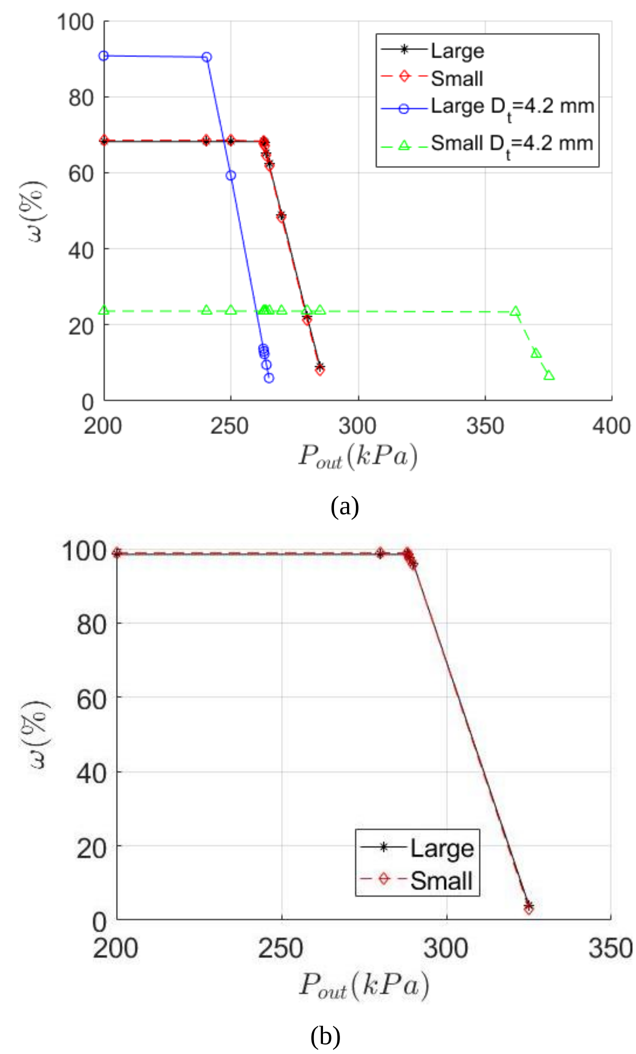

Figure 4. Entrainment ratio as a function of the outlet pressure at P_1=869.2 kPa for the small and large ejectors at: (a) P_2=157.0 kPa, (b) P_2=206.5 kPa.

Looking at the performance at P_1=869.2 kPa and P_2=157.0 kPa (Fig. 4a), the small ejector has an on-design entrainment ratio of ω=68.5% and a critical outlet pressure of P_{out}=262.8 kPa. The large ejector has an on-design entrainment ratio of ω=68.2% and a critical outlet pressure of P_{out}=263.2 kPa. Regarding the results at a throat diameter value of D_t=4.2 mm, the small ejector has an on-design entrainment ratio of ω=23.6% and a critical outlet pressure of 361.9 kPa, while the large ejector has an entrainment ratio of 90.7% and a critical outlet pressure of 240.5 kPa. For the same P_1 and increasing P_2 to 206.5 kPa (Fig. 4b), the small ejector has an on-design entrainment ratio of ω=98.9% and a critical outlet pressure of P_{out}=288.2 kPa. The large ejector has an on-design entrainment ratio of ω=98.5% and a critical outlet pressure of P_{out}=288.5 kPa. As found in Figure 4 and the performance wrote down above, the scaled ejectors led to almost identical performances, as expected.

When increasing P_1 to 1986.5 kPa (Fig. 5), both ejectors have on-design entrainment ratio of ω=14.5% and critical outlet pressure of 514 kPa with P_2=157.0 kPa, and ω=26.2% and critical outlet pressure of 532 kPa with P_2=206.5 kPa. As for the simulation at P_1=869.2 kPa, the scaled ejectors led to almost identical performances when operated at the same P_1 and P_2. Then, the increase of P_2 led to both an increase of the entrainment ratio and the critical outlet pressure. Finally, for the same secondary pressure, the comparison of the results in Figures 4 and 5 show that operating at a higher primary pressure leads to a lower entrainment ratio but to a higher critical outlet pressure. This ability to modulate the primary pressure of a given ejector, and the associated primary mass flow rate and waste heat requirement, could be an appropriate solution when there are changes to the availability of the waste heat source.

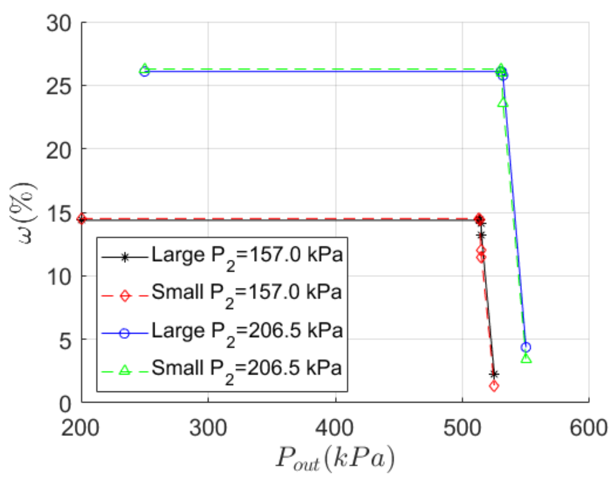

Figure 5. Entrainment ratio as a function of the outlet pressure at P_1=1986.5 kPa for the small and large ejectors at P_2=157.0 kPa and P_2=206.5 kPa.

Considering the results obtained for variable geometry ejectors, their performance is quite different at given operating conditions. Changing the throat diameter of the small and the large ejector to a value of 4.2 mm changes the area ratio (AR) between the constant area section and the ejector throat. The AR is a design parameter with one of the greatest impacts on ejector performance. For the scaled ejectors, AR=5.7. When the throat diameter is changed for 4.2 mm, AR=3.3 for the small ejector and AR=6.9 for the large ejector. Figure 4a shows that the increase of AR leads to a higher entrainment ratio and a lower critical outlet pressure. The work of Rand et al. [11] has led to the same conclusion. These results indicate that if there is a need to change the condensing pressure for fixed inlet conditions, the use of multiple variable geometry ejectors could allow for an easy switch over in an actual system. However, if there is a need for a fixed condensing pressure, the use of variable geometry ejectors is not appropriate since it changes the critical pressure value, and consequently, the simultaneous operation of two or more such ejectors at the same outlet pressure would mean that at least one ejector will operate either above or below its critical outlet pressure. Thus, for fixed condensing pressure applications, scaled ejectors should be preferred.

As discussed above, for applications where the operating pressures remains the same, but the cooling load varies, the simultaneous use of scaled ejectors would be preferred. Being designed to work at the same critical outlet conditions, the use of two or more scaled ejectors in parallel would permit the same condensing pressure. However, they would increase the condensing load. Consequently, it would be necessary to adequately size the condenser to be able to handle partial or maximum loads.

To summarize, the above results suggest that the use of either scaled or variable geometry ejectors in parallel would offer different solutions in terms of cooling. As proposed by Beyrami and Hakkaki-Fard [15], modulating between different capacity ejectors allows to provide renewable cooling during year round operation of a solar thermal powered ejector refrigeration system.

IV. Conclusion

The current study has explored the use of multiple scaled or variable geometry ejectors. The results show that both types of parallel ejector systems offer advantages given the type of application that is required. Indeed, there is an important distinction to be made between the operating conditions and the cooling load needed as it impacts the system design. It is important to know if the waste heat temperature supplying the primary inlet of the ejectors is to remain fixed during yearlong operation. In the case of uncertainty, the installation of two multi-ejector blocks designed for different variable power scenarios offers the greatest flexibility.

Most fixed geometry ejector systems working with a single ejector have focused on designing an ejector to fit a specific limited range of operating conditions. This study proposes adding multi-ejector blocks to expand the operating range without having to go through lengthy re-designs at each stage. With two multi-ejector blocks, one equipped with variable geometry ejectors and the other equipped with scaled parallel ejectors, the system can handle an extremely wide set of operating conditions that could be imagined when trying to utilize a waste heat stream for multiple uses such as in the combined ORC and ERS cycle.

When considering overall performance of the combined ORC and ERS cycle, particular attention should be given to the control of each subsystem. The capacity to modulate the production of heating, cooling and electricity allows a greater opportunity to implement such combined systems elsewhere. More specifically looking at the cooling load, there is a concern on the operating stability when activating and deactivating ejectors during operation of real system. Proper testing during installation in a system should be performed for the use of individual ejector and the simultaneous use of multiple ejectors. Being able to control the activation of an entire multi-ejector block or only certain ejectors within the block needs to be considered in later studies as it is an important factor to ensure maximum performance in an actual system.

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