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EJECTOR POWERPLANT SYSTEM WITH NATURAL WORKING FLUID

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ABSTRACT

This paper presents the potential application of ejector and the efficiency improvement it brings to powerplants that utilize low-temperature renewable and recoverable heat sources. The ejector significantly increases the efficiency of the organic Rankine cycle (ORC) by increasing the turbine temperature drop which is made possible by the expansion, mixing and recompression processes in the ejector. The driving fluid in the ejector of the modified cycle is the high-pressure liquid in the separator that is just circulated back to the evaporator in the ORC. Fundamental thermodynamic analysis of the novel ejector powerplant cycle was undertaken. Ammonia and propane, which are both natural working fluids, were used in the analysis. The analysis was limited by considering that the lowest pressure in the system must be higher than the atmospheric pressure to avoid vacuum leak. The calculation results showed that propane and ammonia can both give maximum efficiencies at different evaporator temperatures and fixed condenser temperatures in the range of 20 – 27%. At evaporator and condenser temperatures of 40°C and 30°C, respectively, ORC gives only 2.5% for the two working fluids while the ejector system gives 21.5% and 27.5% for ammonia and propane, respectively. The difference in efficiency reduces as the evaporator temperature increases.It has been shown that propane can give higher efficiencies and lower velocities in the ejector than ammonia at considerably lower evaporating temperatures, leading to more economical, safer and simpler design and operation. The ejector system with ammonia and propane as working fluids offers environment friendly power generation from low-temperature sources at improved efficiency.

Key Words: *Ejector, powerplant, propane, ammonia, organic Rankine cycle*

1. INTRODUCTION

Ejectors recover the wasted work of an expanding fluid flow [1–2]and have become commonly used in refrigeration systems [3–7]. In this study, utilization of ejector in powerplants, especially those that use low-temperature sustainable heat sources, has been theoretically investigated. It has been theoretically established in this study that the ejector can be used in powerplants that use low-temperature heat sources which are still under-utilized.

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There are vast available natural or waste energy sources that can provide sustainable or recoverable heat for power generation; however, they can usually provide low temperature only. These low-heat grade sources, where the temperature is from 20°C to around 200°C, include solar-thermal panels and ponds, waste heat from commercial and industrial processes, low-temperature geothermal sources, OTEC [8] and combined solar thermal and OTEC technologies [9, 10].

Organic Rankine cycle (ORC) powerplant, with schematic and theoretical thermodynamic cycle diagrams shown in Figure 1 and Figure 2,respectively, can utilize those heat sources, but the efficiency of the resulting powerplant is very low because of the low temperature difference between the heat source and the heat sink. The performance of the proposed novel ejector powerplant cycle in comparison with that of the ORC system for temperature of heat sources from 20°C to 125°C which covers all low-grade heat sources has been theoretically investigated in this study. The working fluids used in the analysis were ammonia and propane, which showed good performance for ORC and OTEC powerplants [8].

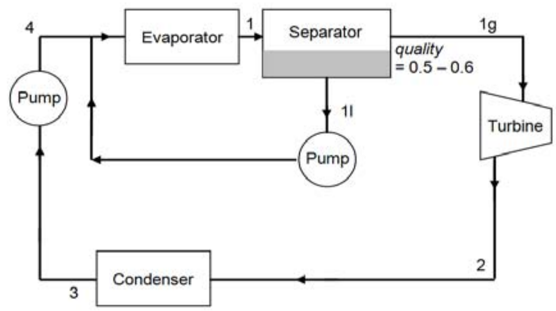

Figure 1. Schematic diagram of the ORC powerplant system.

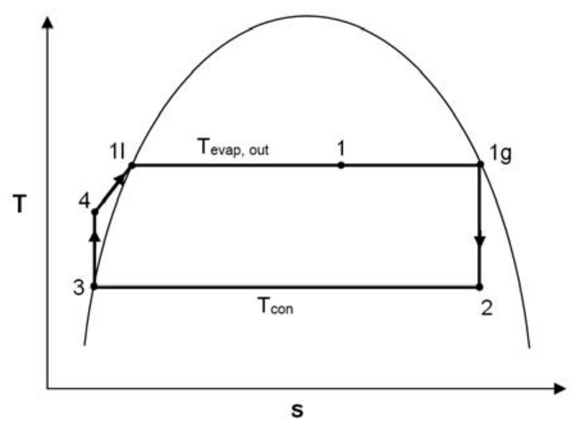

Figure 2. theoretical thermodynamic cycle diagram of the ORC powerplant system.

2. EJECTOR POWERPLANT SYSTEM

The ejector powerplant cycle is described in this section, and the basic governing equations are presented.

2.1 Schematic and Thermodynamic Diagrams

The schematic and theoretical thermodynamic diagrams of the ejector powerplant system are shown in Figure 3 and Figure 4. Similar to the ORC system, a separator is used after the evaporator in the ejector system to make sure that only the vapor part of the evaporator discharge will enter the turbine. The mixture in the evaporator does not become superheated because of the low temperature of the fluid coming from the heat source, and the quality at the separator is commonly 50 – 60% only. The high-pressure hot liquid in the separator is only pumped back to the evaporator inlet in ORC systems; therefore, the available high energy of the liquid is not utilized for generating power and increasing the efficiency. On the other hand, an ejector can be connected to the separator to expand through the nozzle the high-pressure and high-temperature liquid into low-pressure, low temperature and high-speed two-phase flow that can entrain the vapor part of the fluid coming from the turbine.

The vapor part of the evaporator outlet can run a turbine similar to the operation of an ORC system. The liquid part of the outlet flow of the turbine can then be pumped directly into the evaporator. Essentially, the turbine outlet pressure and temperature can be decreased without affecting the temperature of the condenser. Given the same evaporator temperature, the work output of the turbine in the ejector system is higher than that in the ORC system. Therefore, the efficiency of the ejector powerplant is higher as well.

The mixture of the motive two-phase flow from the ejector nozzle and the driven vapor from the turbine can be charged by the diffuser into the condenser. The resulting condensate then can be pumped to the evaporator to mix with the liquid from the turbine.

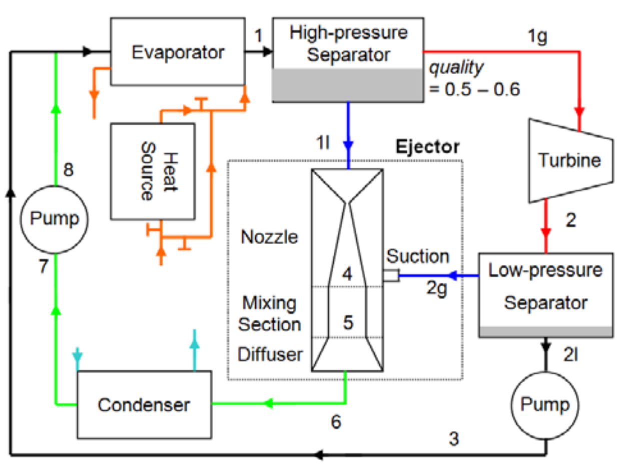

Figure 3. Schematic diagram of the ejector powerplant system.

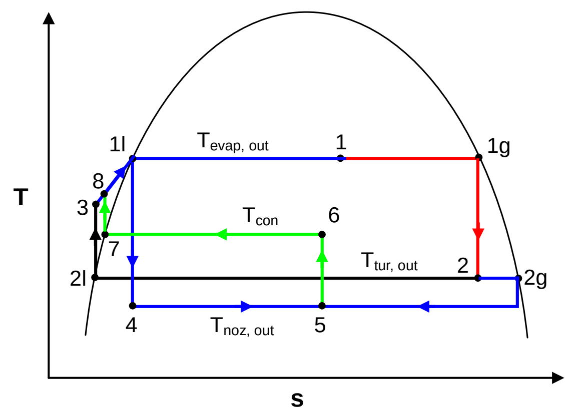

Figure 4. Theoretical thermodynamic cycle diagram of the ejector powerplant system.

2.2 Mathematical Formulation

As shown in Figure 4, constant processes are assumed during heat addition to the evaporator,and heat extraction from the condenser. Isentropic flows in the turbine and pumps are assumed.

The analysis is carried out based on unit mass of evaporator inlet flow. In the high-pressure separator,

$$h_1 = h_{1l} + x_{evap}(h_{1g} - h_{1l}) \qquad (1)$$

The quality of the evaporator discharge was chosen as 0.55 based on common values reported in [8].

The power output of the turbine is given by

$$w_{tur} = x_{evap}(h_{1g} - h_2) \qquad (2)$$

At the turbine outlet, the enthalpy is expressed as

$$h_2 = h_{2l} + x_2(h_{2g} - h_{2l}) \qquad (3)$$

The quality is obtained from isentropic process which leads to equation

$$x_2 = \frac{s_{1g} - s_{2l}}{s_{2g} - s_{2l}}$$

The efficiency of an ejector is defined as the ratio of the change in kinetic energy of the flow through the diffuser to that of the flow through the nozzle.

$$\eta_{eje} = \frac{\Delta KE_{diff}}{\Delta KE_{noz}} = 1 \qquad (5)$$

The ideal efficiency of unity for the ejector is set in this study.
The change in kinetic energy through the nozzle is

$$\Delta KE_{noz} = (1 - x_{evap})(h_{1l} - h_4) \qquad (6)$$

The velocity at the nozzle outlet is

$$u_4 = \sqrt{2(h_{1l} - h_4)} \tag{7}$$

Where, the enthalpy is

$$h_4 = h_{4l} + x_4(h_{4g} - h_{4l}) \tag{8}$$

and the quality is

$$x_4 = \frac{s_4 - s_{4l}}{s_{4g} - s_{4l}} \tag{9}$$

Isentropic expansion through the nozzle is assumed as such that

$$s_4 = s_{1l} \tag{10}$$

Energy balance in the ejector is also assumed leading to

$$h_6 = \frac{(1 - x_{evap})h_{1l} + x_{evap}x_2h_{2g}}{1 - x_{evap} + x_{evap}x_2} \tag{11}$$

Where,

$$x_6 = \frac{h_6 - h_{6l}}{h_{6g} - h_{6l}} \tag{12}$$

and

$$s_6 = s_{6l} + x_6(s_{6g} - s_{6l}) \tag{13}$$

The change in kinetic energy through the diffuser is $\Delta KE_{diff} = (1 - x_{evap} + x_{evap}x_2)(h_6 - h_5)$

Equations (5), (6) and (14) can be used to derive the equation

$$h_5 = \frac{(1 - x_{evap} + x_{evap}x_2)h_6 - x_{evap}x_2(h_1 - h_4)}{1 - x_{evap} + x_{evap}x_2}$$

Where,

$$x_5 = \frac{h_5 - h_{5l}}{h_{5g} - h_{5l}} \tag{16}$$

The velocity at the inlet of the diffuser can be obtained as

$$u_5 = \sqrt{2(h_6 - h_5)} \tag{17}$$

The compression process through the diffuser is assumed as isentropic such that

$$s_5 = s_6 \tag{18}$$

The work done by the liquid pump can be expressed as

$$w_{lp} = x_{evap}(1 - x_2)(h_3 - h_{2l}) \tag{19}$$

Isentropic process is assumed for the liquid pump.

$$s_3 = s_{2l} \tag{20}$$

Likewise, the work done by the condensate pump can be expressed as

$$w_{cp} = (1 - x_{evap} + x_{evap} x_2)(h_8 - h_7) \qquad (21)$$

Isentropic process is also assumed for the condensate pump.

$$s_3 = s_{2l} \qquad (22)$$

The net work of the powerplant is

$$w_{net} = w_{tur} - w_{lp} - w_{cp} \qquad (23)$$

The heat input to the powerplant is given by

$$q = h_{1l} - x_{evap}(1 - x_2)h_3 - (1 - x_{evap} + x_{evap} x_2)h_8$$

The corresponding efficiency of the powerplant is

$$\eta = \frac{w_{net}}{q} \qquad (25)$$

3. RESULTS AND DISCUSSIONS

The calculated theoretical efficiencies of the ejector and ORC systems using ammonia and propane are shown in Figure 5 and Figure 6. REFPROP [11] was used in the calculation. The higher limits of evaporator temperatures are selected based on the critical points of ammonia and propane such that liquid andvapor mixture is present in the evaporator. The turbine and nozzle outlet temperatures are also selected such that the minimum pressure in the system is slightly higher than that of atmospheric pressure to avoid leakage into the system. This condition corresponds to the maximum efficiency that the ejector system can provide at an evaporator temperature without danger of leakage into the system.

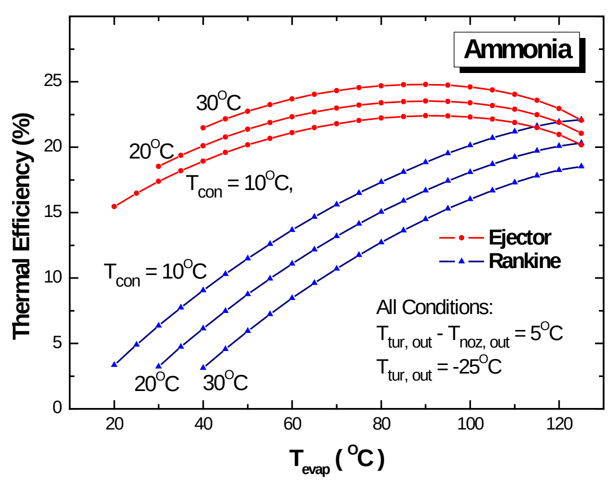

Figure 5. Thermal efficiencies of ejector and Rankine powerplant using ammonia.

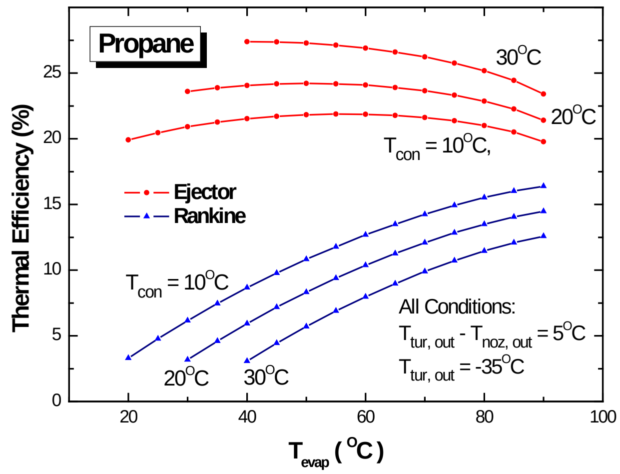

Figure 6. Thermal efficiencies of ejector and Rankine powerplant using propane.

For both working fluids, it is shown that the ejector system is significantly more efficient than the ORC system for the lower subranges of the investigated ranges of evaporator temperature. For a given condenser temperature, there is an optimum evaporator condition that provides the maximum efficiency. Maximum efficiencies for both working fluids ranged from 20 – 27% for the ejector system. The ejector system notably gives higher efficiency than ORC at lower temperature difference between the evaporator and the condenser, which is the potential of the ORC turbine to extract work from the expanding fluid. At evaporator and condenser temperatures of 40°C and 30°C, respectively, ORC provides only 2.5% for the two working fluids while the ejector system provides 21.5% and 27.5% for ammonia and propane, respectively. The difference in efficiency reduces as the evaporator temperature increases. Propane is a good alternative for ammonia, which has been commonly used for refrigeration and ORC systems, because propane can provide higher efficiencies than ammonia at significantly lower evaporating temperatures. Therefore, the required evaporator pressures for propane at such temperatures are lower than those of ammonia.

Computed velocities at the nozzle outlet and diffuser inlet are shown in Figure 7 and Figure 8 for ammonia and propane, respectively. The velocity profiles are linearly increasing as they are directly proportional to change in pressure. The difference between nozzle outlet velocity and diffuser inlet velocity increases because the kinetic energy lost in the mixing section also increases. The figures show that lower velocities are necessary for propane compared with ammonia at the range of maximum efficiencies mentioned above. This consideration makes the nozzle for propane more practical than for ammonia to be designed. The nozzle for propane will also be cheaper, safer and easier to operate.

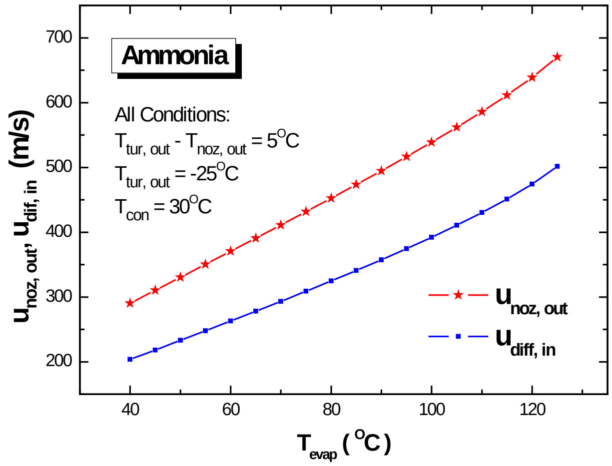

Figure 7. Flow velocities at nozzle outlet and diffuser inlet for ammonia.

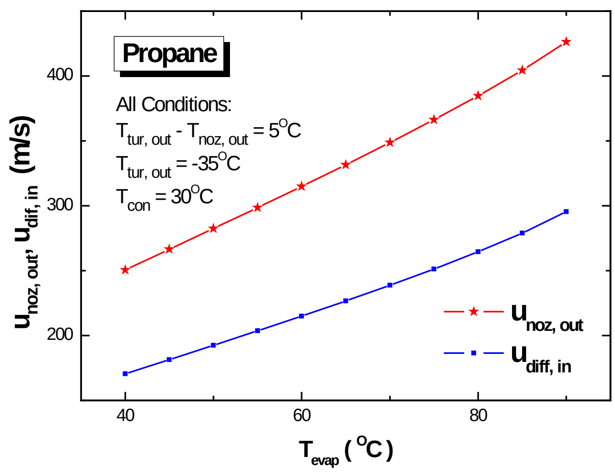

Figure 8. Flow velocities at nozzle outlet and diffuser inlet for propane

Geometric parameters of the ejector that will provide the required operating velocities and pressures need to be designed in further studies.

4. CONCLUSION

Ejector powerplant systems lead to significant improvement in efficiencies of ORC powerplant systems. Ammonia and propane are suitable working fluids for ejector powerplant systems, leading to a range of maximum efficiency from 20 to 27% for the ejector system. However, propane can provide higher efficiencies and lower velocities in the ejector than ammonia at lower operating temperatures and pressures. The ejector system significantly provides higher efficiency than the ORC system at lower temperature difference between the evaporator and the condenser, which is the potential of the ORC turbine to provide power. At evaporator and condenser temperatures of 40°C and 30°C, respectively, ORC gives only 2.5% for the two working fluids while the ejector system gives 21.5% and 27.5% for ammonia and propane, respectively. The difference in efficiency reduces as the evaporator temperature increases.

Geometric parameters of the ejector that will provide the required operating velocities and pressures need to be designed in further studies. The ejector system with ammonia and propane as working fluids offers environment friendly energy conversion of heat from low-temperature sources at improved efficiency.

5. NOMENCLATURE

ORC organic Rankine cycle
OTEC Ocean Thermal Energy Conversion

h	enthalpy
KE	kinetic energy
q	heat input
s	entropy
T	temperature
w	work
x	quality

Greek Letters

η	efficiency
Δ	change

Subscripts

l	liquid
g	vapor
con	condenser
diff	diffuser
eje	ejector
evap	evaporator
noz	nozzle
cp	condensate pump
p	pump
tur	turbine
in	inlet
out	outlet

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