Performance analysis of ERS using R134a - An experimental investigation (Part 2) – System design

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Abstract. The paper presents a mathematical model based on the 1-D gas dynamic equations used to calculate the design and dimensions of the ERS. In the model, the design of the heat exchangers i.e., generator, condenser and evaporator are calculated for the specified ejector geometry. Additionally, the fabrication methodology is also discussed in detail.

Keywords: Ejector, system design, heat exchanger.

1. Introduction
This paper is the continuation part of the previous paper in which the single-phase ejector is analyzed for the performance analysis and geometry by using 1-D mathematical model. This paper presents a mathematical model for designing an ejector refrigeration system (ERS) which can be further used for the experimental fabrication. It comprises the design of the condenser, generator and evaporator of the ERS. A very few researchers have presented the complete mathematical model for designing ERS in their research paper.

ERS is an attractive technology to harness the low-grade heat energy. There are number of various sources from which the low-grade heat energy can be easily obtained like thermal energy from solar radiations, industrial waste-heat, bio-gasifier, geothermal etc. There are a few important advantages of ERS i.e., simple design, low maintenance and operational cost. On the other hand, coefficient of performance of these systems is low and they are very sensitive to the atmospheric conditions. Apart from mathematical and CFD models, a few researchers have experimentally analyzed the system behavior and performance. To increase the performance of ERS, they focused on the geometrical and operational parameters.

Eames et al. [2] analyzed a single-phase double choked ERS using steam as a working fluid. The experiments were conducted for generator temperature between 120 to 140°C, evaporator temperature 5 to 10°C and condenser pressure from 34 to 60mBar; an average difference of 17% in COP was reported with that of the numerical findings. Aphornratana and Eames [3] conducted experiments using steam as a refrigerant, but with a movable primary nozzle. COP of the system was found to decrease and the critical condenser pressure to increase, when the nozzle exit position was moved away from the face of the constant area section and vice-versa. The generator conditions were...
maintained constant. Chunnanond and Aphornratana [4] experimentally investigated the effect of expansion angle, effective position and the shock wave position in an ejector refrigeration system while working with steam. The expansion angle is the angle at which the primary fluid fans out from the primary nozzle. The effective position is the position at which the secondary fluid gets choked and an imaginary nozzle is formed. The shock wave position was tracked by employing 12 pressure transducers. They found that the nozzle exit position controlled the position of effective area. When the primary nozzle was moved away from the mixing chamber, it reduced the primary fluid flow expansion inside the constant area section which in turn increased the available area for the secondary fluid mass flow rate. The increased secondary mass flow rate improved the entrainment ratio of the ejector, but it reduced the critical condenser pressure. The experimental results also revealed that increase in expansion angle decreased the entrainment ratio and thus shifted the shock wave position towards the primary nozzle. Rungtrakoon et al. [5] investigated the effect of eight different primary nozzles at the evaporator temperature of 7.5°C while the generator temperature was varied from 110°C to 150°C. Increase in primary nozzle throat diameter decreased the entrainment ratio of the ejector, but it could now be operated at higher condenser pressure.

2. Mathematic model
A mathematical model based on 1-D gas dynamic equations is used to calculate the dimensions of the various heat exchangers used in the ERS experimental setup. The complete program for finding the design of the ERS is given below:

Program for finding tube lengths of heat exchangers in Ejector Refrigeration System (ERS)
"Known information"
\[ T_g = 357\text{[K]}; \ T_{evp} = 285\text{[K]}; \ d_t = 0.0019\text{[m]}; \ f_p = 0.9; \ f_m = 0.8; \ n_p = 0.99; \ n_s = 0.65; \ d_{p_1} = 0.00635\text{[m]}; \ d_3 = 0.0042\text{[m]}; \ T_c_s = 303\text{[K]}; \]
\[ P_{c_s} = \text{Pressure}(R134a, T = T_c_s, x = 0.9); \]
\[ P_{g} = \text{Pressure}(R134a, T = T_g_{sat}, x = 1); \]
\[ P_{evp} = \text{Pressure}(R134a, T = T_{evp_{sat}}, x = 1); \]
\[ T_{evp_{sat}} = T_{evp} - 2; \]
\[ T_g_{sat} = T_g - 4; \]
"Area calculation"
\[ A_3 = \frac{(\pi\times(d_3^2))}{4}; \]
\[ A_{p_1} = \frac{(\pi\times(d_{p_1}^2))}{4}; \]
\[ A_t = \frac{(\pi\times(d_t^2))}{4}; \]
"primary mass flow rate"
\[ m_p = \left(\frac{(P_g\times A_t)}{\sqrt{T_g}}\right)\times\left(\frac{\sqrt{\gamma/(\gamma - 1)}}{}\right)\times\left(\frac{\sqrt{n_p}}{}\right); \]
\[ \gamma = \frac{cp_g}{cv_g}; \]
\[ cv_g = \text{Cv}(R134a, T = T_g, x = 1); \]
\[ cp_g = \text{Cp}(R134a, T = T_g, x = 1); \]
\[ R = \frac{UGC}{MW}\times 1000; \ UGC = 8.314[J/gmol.K]; \]
\[ MW = \text{MolarMass}(R134a); \]
\[ \rho_{p} = \text{Density}(R134a, T = T_g, x = 1); \]
"Mach number & Pressure"
\[ (A_{p_1}/A_1)^2 = \frac{1}{(1 + ((M_{p_1}^2)/2))}\times((2/(\gamma + 1))\times((\gamma + 1)/(\gamma - 1))\times((\gamma + 1)/(\gamma - 1)))); \]
\[ (P_g/P_{p_1}) = ((1 + ((\gamma - 1)/2)\times((M_{p_1}^2)/2))\times((\gamma - 1)/(\gamma + 1))); \]
"Secondary fluid pressure at y-y"
\[ (P_{evp}/P_{s_y}) = (1 + ((\gamma - 1)/2))\times((\gamma/(\gamma - 1))); \]
\[ (P_{p_y}/P_{p_1}) = ((1 + ((\gamma - 1)/2)\times((M_{p_1}^2)/2))\times((\gamma - 1)/(\gamma + 1)))); \]
\[ \frac{A_y}{A_1} = \frac{(f_p/M_y)^*((2/(\Gamma+1)))*((1+(((\Gamma-1)/2)*(M_y^2))))^((\Gamma+1)/(2*(\Gamma-1)))}}{((1/M_1)^*((2/(\Gamma+1))(*(M_1^2))))^((\Gamma+1)/(2*(\Gamma-1)))}; \]

\[ P_y = P_m; \]

\[ A_3 = A_y + A_s; \]

"Mass flow rate of secondary fluid"

\[ m_s = \frac{(P_{evp}A_y)}{(\sqrt{T_{evp}})} \left( \frac{\sqrt{(\frac{\Gamma}{R})^{\frac{(\Gamma+1)}{\Gamma-1}}}}{\sqrt{n_s}} \right); \]

"Temperature of both fluids"

\[ T_g/T_y = (1+(((\Gamma-1)/2)*(M_y^2))); \]

\[ T_{evp}/T_s = (1+((\Gamma-1)/2)); \]

"Mixed flow at section m-m"

\[ f_m \left( (m_p v_p + m_s v_s) \right) = (m_p + m_s) v_m; \]

\[ v_p = M_p c_p; \]

\[ v_s = c_s; \]

\[ c_p = \sqrt{\Gamma R T_y}; \]

\[ c_s = \sqrt{\Gamma R T_s}; \]

\[ M_m = v_m/c_m; \]

\[ c_m = \sqrt{\sqrt{(\frac{\Gamma R T_m}{2})}}; \]

"Mixed flow across shock"

\[ \frac{P_3}{P_m} = (1+((2*\Gamma)/(\Gamma+1))*((M_m^2)-1)); \]

\[ (M_3) = (1+((\Gamma-1)/2)*(M_m^2)) \]

\[ P_c/P_3 = (1+(((\Gamma-1)/2)*(M_m^2)))*((\Gamma/(\Gamma-1)))/(\Gamma(M_m^2)); \]

\[ T_c = \text{Temperature(R134a, P=P, x=0)}; \]

"Heat exchanger design"

\[ h_{bar} = \text{Flow_Boiling_avg(Fluid$, T_{sat}, G, d_p, x, q, 'Horizontal'); Fluid$='R134a';} \]

\[ G_{boiler} = \frac{m_p}{A_p}; \]

\[ x_{in} = 0; \]

\[ x_{out} = 1; \]

\[ h_{boiler} = \text{Enthalpy(R134a, T=T_{sat}, x=0)}; \]

\[ h_{boiler} = \text{Enthalpy(R134a, T=T_{sat}, x=1)}; \]

\[ h_{fg} = h_{boiler} - h_{boiler}; \]

\[ q = m_p h_{fg}; \]

\[ k = \text{('Copper', T_{sat})}; \]

\[ \text{Call External_Flow_Cylinder(Fluid$, T_inf, T_surface, P_water, u_inf, D_outer, F_d, L, h_water, C_d, Nusselt, Re);} \]

\[ T_{inf} = 90+273; \]

\[ T_{surface} = 84+273; \]

\[ P_{water} = 101325\, \text{[Pa]}; \]

\[ u_{inf} = 0.7\, \text{[m/sec]}; \]

\[ D_{outer} = 0.00635\, \text{[m]}; \]

\[ q = (1/((1/(h_{bar}*(Pi*d_3*L_{boiler}))))=((ln(d_p_l/d_3))/(2*Pi*k*L_{boiler}))) + (1/(h_water*Pi*d_p_l*L_{boiler}))) \]

\[ \text{LMTD_boiler}; \]

\[ \text{LMTD_boiler} = 6\, \text{[K]}; \]

\[ q = \text{U_overall_boiler} + \text{Pi}*d_p_l*L_final boiler*LMTD_boiler; \]
\[ h_{evp\_1} = \text{Enthalpy}(R134a, T = T_{evp\_sat}, x = 0); \]
\[ h_{evp\_2} = \text{Enthalpy}(R134a, T = T_{evp\_sat}, x = 1); \]
\[ h_{fg\_evp} = h_{evp\_2} - h_{evp\_1}; \]
\[ q_{evp} = m_s \cdot h_{fg\_evp}; \]
\[ d_{evp} = 0.009525[m]; \]
\[ q_{evp} = U_{overall\_evp} \cdot \pi \cdot d_{evp} \cdot L_{final\_evp} \cdot \text{LMTD\_evp}; \]
\[ \text{LMTD\_evp} = 4[K]; \]
\[ h_{bar\_evp} = \text{Flow\_evp\_avg}(\text{Fluid\$}, T_{evp\_sat}, G_{evp}, d_{p\_1}, x_{in}, x_{out}, q_{evp}, \text{'Horizontal'}); \]
\[ G_{evp} = \frac{m_s}{A_{p\_1}}; \]
\[ \text{Call External\_Flow\_Cylinder}(\text{Fluid\$}, T_{inf\_evp}, T_{surface\_evp}, P_{water}, u_{inf}, D_{outer\_pipe}: F_d\_L, h_{water}) \]
\[ T_{inf\_evp} = 283; \]
\[ T_{surface\_evp} = 292; \]
\[ q_{evp} = \frac{1}{((1/(h_{bar\_evp}(\text{Pi}*d_{3\_evp}))) + ((\ln(d_{p\_1}/d_{3\_evp}))/2*Pi*k*L_{evp}) + (1/(h_{water}*\pi*d_{p\_1}*L_{evp}))))} \times \text{LMTD\_evp}; \]
\[ h_{cond\_1} = \text{Enthalpy}(R134a, T = T_{c\_s\_x}, x = 0); \]
\[ h_{cond\_2} = \text{Enthalpy}(R134a, T = T_{c\_s\_x}, x = 1); \]
\[ h_{fg\_cond} = h_{cond\_2} - h_{cond\_1}; \]
\[ m_t = m_s + m_p; \]
\[ q_{cond} = U_{overall\_cond} \cdot \pi \cdot d_{p\_1} \cdot L_{final\_cond} \cdot \text{LMTD\_cond}; \]
\[ \text{LMTD\_cond} = 5[K]; \]
\[ h_{bar\_cond} = \text{Flow\_cond\_avg}(\text{Fluid\$}, T_{cond\_sat}, G_{cond}, d_{p\_1}, x_{in}, x_{out}, q_{cond}, \text{'Horizontal'}); \]
\[ \text{Call External\_Flow\_Cylinder}(\text{Fluid\$}, T_{inf\_cond}, T_{surface\_cond}, P_{water}, u_{inf}, D_{outer\_pipe}: F_d\_L, h_{water}) \]
\[ T_{inf\_cond} = 308; \]
\[ T_{surface\_cond} = 303; \]
\[ q_{cond} = \frac{1}{((1/(h_{bar\_cond}(\text{Pi}*d_{3\_cond}))) + ((\ln(d_{p\_1}/d_{3\_cond}))/2*Pi*k*L_{cond}) + (1/(h_{water}*\pi*d_{p\_1}*L_{cond}))))} \times \text{LMTD\_cond}; \]

"{End of Program}"

Figure 4.3 Schematic diagram of ERS [6]
3. Experimental test bench

An experimental test rig has been fabricated using R-134a refrigerant. The actual photograph of the set-up is shown in figure 4.2. The experimental setup consists of four main components: generator, condenser, evaporator, and ejector. The test facility is having various measuring devices i.e., temperature sensors, pressure gauges, ammeter, voltmeter, indicator, etc.

3.1 Heat exchangers

The heat exchangers i.e., evaporator, condenser and generator are designed in tube in shell type heat exchangers. A 6.35mm diameter Copper tube in coiled shape has been used for the fabrication of the condenser and the generator, but the evaporator has 9.52mm diameter copper tube. The length of the copper tube in the evaporator, generator, and the condenser is 15.24, 24.384 and 30.48meters respectively. These copper tubes are spiraled and then placed in three insulated containers/shells. The copper tube has been used for the refrigerant, and shell for the external fluid in all the heat exchangers. A 4.5kW electric heater with a current controller has been placed in the generator to provide heat to the external fluid, which is 40% concentrated glycol solution. A mechanical recirculating pump is used as a stirrer fitted in the generator. It recirculates the glycol solution to make uniform temperature distribution among the container. To minimize the heat interaction between the heat exchangers and the surrounding, 9mm thick industrial insulation is used to insulate all the connecting pipes used in the experimental test facility. The shell of the evaporator has a known quantity of water, which was continuously cooled by the refrigerant flowing in the coils of the evaporator. An electric resistance heater of 3kW capacity with a current controller has been used to provide the heating effect, same as the cooling effect produced in the evaporator. This way the temperature of water in the evaporator shell does not change and so the energy consumed by the heater of the evaporator is reasonably assumed to be equal to the cooling effect produced by the ERS. The uniform temperature of the water present in the evaporator is maintained by a mechanical stirrer. To remove the latent heat of the refrigerant in the condenser fresh water at a constant temperature is circulated. And after gaining the heat from the condenser this water is drained. The liquid refrigerant inlet in the evaporator and the generator has been provided at the bottom of the coil, but in the condenser, as the refrigerant is in vapor phase at its inlet, it was provided at the top of the coil.

3.2 Ejector

The ejector is made up of four different parts i.e., primary nozzle, constant area section, suction chamber and diffuser. These parts are interchangeable and have been made of brass metal. To assemble these parts standard threaded connections have been used to make a complete ejector. The convergent-divergent supersonic primary nozzle made-up of brass material, perfectly in line with CAS has been placed facing the center of the constant area section. There is total three primary nozzles, all having the same inlet area ($A_{p1}$), but different throat ($A_{p2}$) and exit ($A_{p3}$) areas. As the throat and the exit areas of the primary/motive nozzle are different, the nozzle divergence angle of all the three nozzles is also different. The constant area section (CAS) of the ejector has been made up of two different diameters and lengths. Dimensions of all the six ejectors used in the investigation are shown in table 1.

3.3 Measuring instruments and supporting devices

RTD temperature sensor having accuracy ±0.1°C have been used at various locations to measure the temperature. Bourdon-tube type pressure gauges accurate up to ±6kPa have been fitted to measure pressure at different locations in the facility. Voltage and current supplied to the compressor, and the heating elements in the generator and the evaporator are measured using analog voltmeters and ammeters with an uncertainty of ±0.5V and ±0.05A respectively. Total uncertainty of the experimental setup in the determination of COP of the system has been found to be less than 5%. The details of the measuring instruments used in the experimental setup are shown in table 2.
3.4 Experimental procedure and the operating parameters

The atmospheric temperature in hot weather conditions is generally higher than the saturation temperature of the refrigerant in the setup in off condition; hence the refrigerant in the setup exists in vapor phase at the time of starting. The liquid pump installed after the condenser is never designed to work with the vapor, so an auxiliary compressor has been used to increase pressure in the condenser to condense the vapor.

### Table 1 Geometries of ejector used in the experiments

| Sr. No | Ejector | Primary nozzle geometry | Secondary nozzle geometry | Area Ratio ($\frac{A_4}{A_{p2}}$) |
|-------|---------|-------------------------|---------------------------|---------------------------------|
|       |         | Inlet Area ($A_{p1}$) (mm$^2$) | Throat Area ($A_{p2}$) (mm$^2$) | Exit Area ($A_{p3}$) (mm$^2$) | CAS Area ($A_4$) (mm$^2$) | CAS’s Length (L) (mm) | (NXP) (mm) |
| 1     | I       | 31.7                    | 3.14                      | 10.3                           | 31.7 | 80 | 2.54 | 10.08 |
| 2     | II      | 31.7                    | 3.14                      | 10.3                           | 31.7 | 120 | 2.54 | 10.08 |
| 3     | III     | 31.7                    | 4.91                      | 15.9                           | 31.7 | 80 | 2.54 | 6.45 |
| 4     | IV      | 31.7                    | 4.91                      | 15.9                           | 31.7 | 120 | 2.54 | 6.45 |
| 5     | V       | 31.7                    | 7.07                      | 20.6                           | 67.2 | 80 | 2.54 | 9.51 |
| 6     | VI      | 31.7                    | 7.07                      | 20.6                           | 67.2 | 120 | 2.54 | 9.51 |

### Table 2 Instruments used in the experimental setup

| Sr. No. | Parameter | Instrument type | Unit | Uncertainty | Operating Range |
|---------|-----------|-----------------|------|-------------|-----------------|
| 1.      | Generator/condenser pressure | Bourdon-tube type pressure gauge | PSI | ± 2 PSI | 0 – 500 PSI (Gauge pressure) |
| 2.      | Evaporator pressure | Bourdon-tube type pressure gauge | PSI | ± 2 PSI | 0 to 250 PSI (Absolute pressure) |
| 3.      | Temperature | Digital display unit with RTD sensor | °C | ± 0.1°C | -50°C to 200°C |
| 4.      | Voltage    | Analog          | Volts | ± 0.5 V | 0 – 300 V, 50 Hz |
| 5.      | Current    | Analog          | Ampere | ± 0.05 A | 0 – 20 A, 50 Hz |

As the pump started receiving liquid refrigerant, the auxiliary compressor was switched off from the system. Figure 4.3 shows a schematic diagram of the experimental setup. The setup takes approximately an hour to maintain the steady-state conditions with refrigerant R134a. The complete experimental procedure is explained using a flow chart as shown in the figure 4.4. The condensed refrigerant has been pumped to the generator where it got converted into high-pressure vapor refrigerant. The glycol solution in the generator has been maintained at 90°C and the temperature of the vapor refrigerant coming out of the generator has been maintained at 84°C. Superheating of 4°C has been ensured in all the experiments to consider heat loss in the connecting pipe to the ejector. The temperature of refrigerant in the evaporator has been varied
from 10 to 17.5°C in a gap of 2.5°C. The temperature of the condenser was maintained by adjusting the mass flow rate of the tap water circulated through it and the sub-cooling of 1°C has been secured in the condenser for all the experiments.

4. Conclusion
The paper presents a 1-D mathematical model for designing and finding the geometries of various heat exchangers to be used in an ERS. The design of the heat exchanger is based on the geometry of the ejector calculated in the previous research paper of the sequence.

Acknowledgement
This research is funded under the “Visvesvaraya Research Promotion Scheme” funded by Dr. A. P. J. Abdul Kalam Technical University, Lucknow, India.

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