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Exergy Analysis of the Solar-Driven Ejector Refrigeration System

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Abstract: *In the recent times, the use of solar energy for the refrigeration purpose has been increasing day by day. Generally, solar used for cooling purpose has proposed many advantageous features in the refrigeration field. The cooling path is basically depends on the amount of solar radiation. The more amount of solar radiation directly increases the C.O.P. of the system.*

This research paper, the solar cooling is done with the help of an ejector refrigeration system. The exergy analysis is used to determine the various losses. This paper deals with the exergy analysis to identify the losses and optimum conditions for driving such a refrigeration system. This paper also deals with the impact of losses on the performance of the system. To achieve this analysis for energy and exergy is carried out. The exergy analysis of a cycle identifies the performance of each component of the cycle.
Keywords: *Solar Collector, Ejector, Generator, Condenser, cooling load, exergy analysis*

I. The Solar Driven Ejector Refrigeration System-

The models of the various parts of the solar-driven ejector refrigeration system, which are used for simulation in this paper, are described in this. The schematic of the system is shown in Figure 1.1. A solar thermal collector is used to supply heat to the generator as a major energy source for the ejector refrigeration subsystem, via a thermal storage and an auxiliary heater. An evaporator provides cooling to the conditioned space. In this case, the cooling load is assumed to be the already introduced small 150 m^3 office building. Details of each subsystem are described in the following section, starting with the model of the solar collector subsystem, followed by the ejector refrigeration subsystem, and the cooling load.

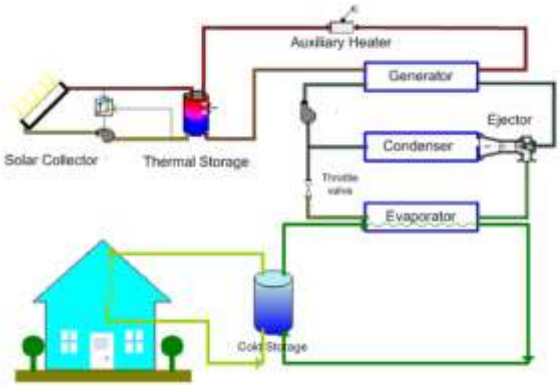

Figure 1.1

A Solar Driven Ejector Refrigeration System

In this study, TRNSYS and EES simulation tools are used to model and analyze the performance of a solar-driven ejector refrigeration system utilizing so-called co-solving. TRNSYS is a transient systems simulation program with a modular structure. It is widely used for the analysis of time dependent systems such as solar systems and HVAC systems. The whole system is modeled in the TRNSYS studio and it is divided into 3 main subsystems: solar collector subsystem, refrigeration subsystem and cooling load (building). The model of the ejector refrigeration subsystem is developed in Engineering Equations Solver, EES (Klein, 2004),

II. The Solar Collector Subsystem-

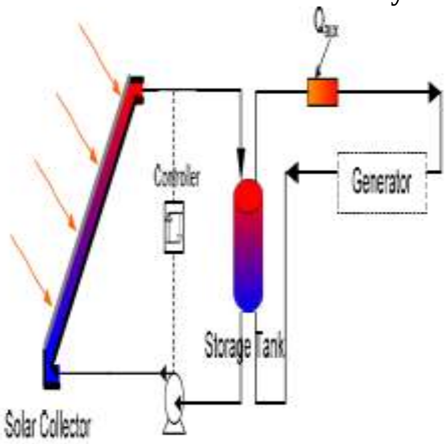

Figure 1.2

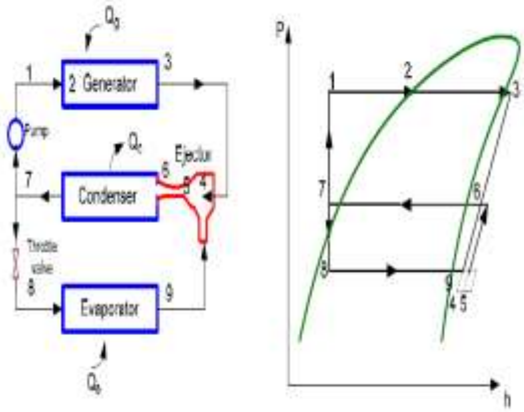

Figure 1.3

An Ejector Refrigeration Cycle

The processes of the ejector refrigeration subsystem are represented in a pressure-enthalpy diagram in Figure 1.3. The model of the ejector refrigeration subsystem is based on the thermodynamic states in each operating point according to Figure 1.3 and the following equations. In the following nomenclatures in this section (section), the numbers in the subscription refer to the condition according to Figure 1.3
Subscription 'm' refers to the condition in the mixing chamber of the ejector,
Subscription 'g' refers to the condition in the generator
Subscription 'c' refers to the condition in the condenser
Subscription 'e' refers to the condition in the evaporator
Subscription 'is' refers to the isentropic condition.

Entrainment Ratio and Coefficient of Performance

The mass ratio or so called the entrainment ratio can be written as,

$$\omega = \frac{m_e}{m_g} = \sqrt{(\eta_N \cdot \eta_D) \cdot \left(\frac{h_3 - h_{4,is}}{h_{6,is} - h_5}\right)} - 1$$

The product of the isentropic efficiency of the nozzle (η_N) and the isentropic efficiency of the diffuser (η_D) may be referred to as the ejector isentropic efficiency (λ),

$$\lambda = \eta_N \cdot \eta_D$$

Another important criterion for the ejector is the compression ratio, which is defined as the pressure ratio between the condenser and the evaporator

$$r_p = \frac{p_c}{p_e}$$

The efficiency of the ejector cooling subsystem is generally expressed in terms of both the entrainment ratio, ω, and a coefficient of performance (COP_{ejc}). The COP of an ideal thermal refrigeration system can be written as the product of the Carnot heat engine efficiency and the ideal COP of the Carnot refrigeration cycle, as previously defined in Equation. Neglecting the work input to the pump, the thermal COP of the ejector refrigeration system is defined as the ratio between cooling capacity and necessary heat input, as shown in Equation.

$$COP_{ejc} = \frac{Q_e}{Q_g}$$

$$COP_{ejc} = \frac{m_e(h_9 - h_8)}{m_g(h_3 - h_1)}$$

Second law analysis

The exergy analysis is used to determine the various losses. This chapter deals with the exergy analysis to identify the losses and optimum conditions for driving such a refrigeration system. This chapter also deals with the impact of losses on the performance of the system. To achieve this analysis for energy and exergy is carried out. The exergy analysis of a cycle identifies the performance of each component of the cycle. Exergy at a given state is equal to the maximum work that can be obtained when operating reversibly between the given state and the reference state. As we know that the energy inputs to a solar-driven ejector refrigeration system are solar radiation, heat & electricity. Three different forms of energy used in the equation: heat, radiation and electricity leading to three possible definitions of system performance:

$$SP_1 = \frac{Q_e}{Q_{su} + W_{p,el}}$$

Or,

$$SP_2 = \frac{Q_e}{Q_{su}}$$

Or,

$$SP_3 = \frac{Q_e}{W_{p,el}}$$

Where:-

Q_e = Cooling capacity (kW)

Q_{su} = Useful solar radiation input to the collector (kW)

$W_{p,el}$ = Electricity input to the pump (kW)

Solar Collector

The exergy input to the collector per unit area in terms of exergy emitted from the sun and the radiation emitted from the solar collector is given by following relation

$$E_s = f\sigma T_{sun}^4 + (1-f)\sigma T_{planet}^4 - \sigma T_{sc}^4$$

Where f is the sunlight dilution factor, which is equal to 2.16 x 10^{-5} on the earth,

σ = Stefan-Boltzmann constant.

Followings are the inputs to the system

$E_{s,h}$ = Incident radiation on the solar collector

E_e = Exergy heat load to the evaporator

$W_{p,el}$ = Electricity used by the pump

And followings are the outputs from the system

$E_{c,out}$ = Rejected exergy from the condenser

I_{total} =Total irreversibilities from every component in the system
The exergy balance for the system can be written as:

$$E_{s,h} + E_e + W_{p,el} = E_{c,out} + I_{total}$$

The heat coming from sun in the form of solar radiation is partly absorbed by the thermal fluid and the surrounding equipment and partly lost to the environment. The absorbed solar radiation is converted in to the available heat (Q_{ava}). The transformation of thermal energy into exergy is given by the second law of thermodynamics as per the following relation

$$E_{s,h} = Q_{ava} \cdot (1-(T_{ref} / T_{sc}))$$

Where

T_{ref} = Reference temperature
T_{sc} = Average temperature of the solar collector
The exergy loss during the conversion from solar radiation to heat on the solar collector is given as

$$I_{sc,r} = E_s - E_{s,h}$$

Assuming zero loss of energy transferred from the absorbing plate to the thermal fluid ($Q_u = Q_{ava}$), the exergy gain by the solar collector is

$$E_{su} = Q_u \cdot (1-(T_{ref} / T_{sc}))$$

Where:-
Q_u = Useful steady state energy gain to the solar collector.
Q_u is calculated by the Hottel-Whillier-Bliss equation as

$$Q_u = AF_R\left[G(\tau\alpha)_e - U_L(T_i - T_a)\right]$$

F_R = Collector heat removal factor,
G = Solar flux density,
U_L = Heat loss coefficient,
T_i = Temperature of the working medium entering the solar collector,
T_a = Ambient temperature,
α = Absorptivity factor,
τ = Transmissivity factor.
The exergy loss related to the heat transfer in the collector, from the exergy absorbed by the solar collector to the exergy supplied to the working fluid can be calculated using the equation

$$I_{sc} = E_{s,h} - E_{su}$$

Equations Used in the Exergy Analysis

Different component of the solar-driven ejector refrigeration system are solar collector, generator, ejector, condenser, pump, expansion device, evaporator. Following equations are used to calculate exergy losses by performing an exergy balance for each component..

Performance

The alterative definitons of COP for the refrigeration subsystem are as

$$COP_{el} = \frac{Q_e}{W}$$

$$COP_{thermal} = \frac{Q_e}{Q_g}$$

$$\eta_{carnot} = \left(\frac{T_g - T_1}{T_g}\right) \cdot \left(\frac{T_2}{T_1 - T_2}\right)$$

The Energy performance of the system is defined as the system thermal ratio (STR),

$$STR = \frac{Q_e}{G \cdot A} = \frac{Q_e}{Q_g} \times \frac{Q_g}{G \cdot A} = COP \times \eta_{sc}$$

III. Methodology

The exergy analysis of the solar-driven ejector refrigeration system was performed assuming the following standard conditions:

- The incident solar radiation is 700 W/m^2 , and a double-glazed flat-plate solar collector is used with an area of 25 m^2, $F_R(\tau\alpha)e = 0.8$ and $F_R U_L = 3.5$ W/m^2 K^1,
- Water is used as the heat carrier between the solar collector and the generator,
- The outlet temperature of the solar collector is assumed to be 10 K above the generating temperature,
- The cooling capacity is 5 kW,
- The ambient air temperature is 27°C which also is used as the reference temperature for the analysis,
- The generating temperature is 90°C,
- The condensing temperature is 37°C,
- The evaporation temperature is 10°C,

<u>Table 1.1 Equations Used in the Exergy Analysis</u>

Solar collector	
Exergy (radiation) input	$E_s = f\sigma T_{sun}^4 + (1-f)\sigma T_p^4 - \sigma T_{sc}^4$
Exergy (heat) input	$E_{s,h} = Q_{out} \cdot (1 - (T_{ref} / T_{sc}))$
Loss (transformation process)	$I_{sc,r} = E_s - E_{s,h}$
Useful exergy gain	$E_{su} = Q_u \cdot (1 - (T_{ref} / T_{sc}))$
Loss (heat transfer)	$I_{sc} = E_{s,h} - E_{su}$
Generator	
Exergy available	$E_{gin} = Q_g \cdot (1 - (T_{ref} / T_g))$
Exergy loss	$I_s = T_{ref} \cdot (m_g(S_3 - S_1) + m_{sc}(S_{g,sc\_out} - S_{g,sc\_in}))$
Ejector	
Exergy loss	$I_j = T_{ref} \cdot [\ (m_e + m_g) \cdot S_6 - m_g \cdot S_3 - m_e \cdot S_9\ ]$
Condenser	
Exergy loss	$I_c = T_{ref}\left[(m_e + m_g) \cdot (S_7 - S_6) + (Q_c / T_{ref})\right]$
Pump	
Exergy loss	$I_p = W_{pump} + m_g \cdot ((h_1 - h_7) - T_{ref} \cdot (S_1 - S_7))$
Expansion device	
Exergy loss	$I_{exp} = m_e \cdot (T_{ref} \cdot (S_8 - S_7))$
Evaporator	
Exergy delivered	$E_e = Q_e \cdot (1 - (T_{ref} / T_{room}))$
Exergy loss	$I_e = T_{ref}\left(m_e \cdot (S_9 - S_8) - (Q_e / T_{room})\right)$
Total irreversibility	$I_{total} = I_{sc} + I_{gen} + I_c + I_p + I_{exp} + I_e$

IV. Results

- The overall thermal energy efficiency or the system thermal ratio (STR) at a generating temperature of 90°C is about 5.8%.
- The coefficient of performance (COP) of the ejector cycle is 0.13.
- The solar collector efficiency is about 35% for the conditions simulated.
- The Refrigeration cycle efficiency is about 22% for the conditions simulated.
- The system thermal ratio (STR) is quite low.

- The decrease of the Carnot efficiency is due to the heat losses in the system. The solar collector efficiency decreases when the output temperature increases due to the heat loss in to the environment.
- The STR decreases slightly after the generator temperature reaches levels above 60°C. This decrease in STR is due to a decrease in solar collector efficiency at high generating temperatures.

Table 1.2 Energy Balance

	Exergy Received (kW)	Exergy Delivered (kW)	Exergy Losses (kW)	Exergy Effici. (%)
Solar collector	38.26	6.98	31.28	18
Refrig cycle	3.2	0.09	3.11	3
Overall	41.46	0.09		0.22

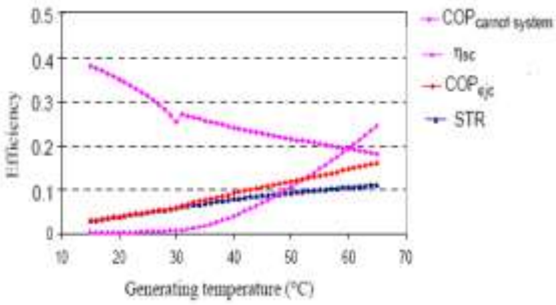

Figure 1.4
System Energy Efficiency

Table 1.3 Exergy Balance

	Energy Received (kW)	Energy Delivered (kW)	Energy Losses (kW)	Energy Efficie (%)
Solar collector	38.26	13.391	24.869	35
Refrig. cycle	13.6	3	0.06	22
Overall	51.86	3		5.8

Overall exergetic efficiency is less than 1%. The extremely low exergetic efficiency is due to the fact that cooling is performed at a temperature level near the reference state for the exergy analysis. This indicates that the concept of overall exergetic efficiency is better suited to prime movers.

To reduce the losses better design of ejector is required. Frictional losses of the flow inside the ejector through the converging diverging nozzle is the main cause for the exergy loss in the ejector. However the performance of the ejector increases hugely at increased evaporator temperatures. A more efficient ejector design would give less losses in the system. This would imply that optimum efficiency would be obtained at lower generating temperatures since the collector losses would be reduced.

The losses in the pump are related with the electrical and mechanical inefficiencies. The expansion device is used in the refrigeration cycle to decrease the pressure of the refrigerant from the condenser pressure to the evaporator pressure. The throttling process in the expansion valve causes a loss that is the smallest one in this system.

A higher mass flow rate thus needs a higher energy supply to the generator. For generator temperatures over 80°C the losses in the ejector decrease, but the losses in the solar collector increase due to the decrease in solar collector efficiency, as shown in Figure 5.5 and Figure 5.6. The optimum generator temperature thus depends on the evaporation temperature and other operating conditions such as condensing temperature, working fluid, and the ejector efficiency.

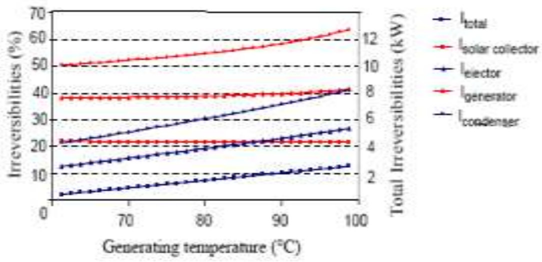

Figure1.5
The Percentage of the Exergy Loss in Each Component as a function of the Generator Temperature, at the Condensing Temperature of 37°C and the Evaporting Temperature of 10°C

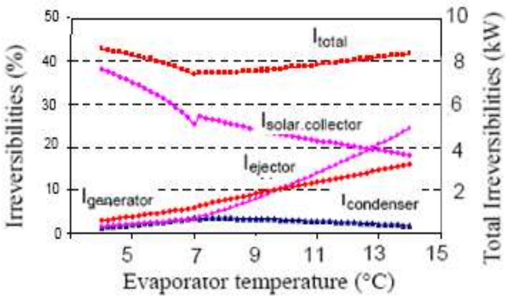

Figure 1.6
Percentage of Irreversibility in Each Component as a function of the Evaporation Temperature, at the Condensing Temperature of 37°C and the Generating Temperature of 90°C

V. Conclusion

This paper Exergy analysis is used as a tool in analyzing the performance of an ejector refrigeration cycle driven by solar energy. The analysis is based on the following conditions- a solar radiation of 700 W/m^2, an evaporator temperature of 10°C, a cooling capacity of 5 kW,an ambient temperature of 30°C, and the reference temperature of 30°C (the same as the ambient temperature).

Irreversibilities occur among components and depend on the operating temperatures. The largest portion of the exergy is lost in the solar collector followed by the ejector, the generator, the condenser, the evaporator and the expansion valve.

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