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Ejector Refrigeration Cycle with Internal Cooling

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ABSTRACT: *This paper introduces a new refrigeration / thermodynamic cycle (Ejector refrigeration cycle with internal cooling) and its possible applications, as well as compares it with the absorption refrigeration cycle for the same generator temperature. As it is generally known, an ejector refrigeration system (ejector chiller) utilizes thermal energy (solar-thermal energy, geothermal energy, waste heat from thermal / nuclear power plants, as well as foundries, etc.) from a heat source for cooling buildings / industrial facilities. Compared to conventional chillers, ejector chillers have a pump and an ejector (thermal compressor) instead of conventional (mechanical) compressor. For the same refrigeration effect, the work of a pump is significantly lower than the work of a mechanical compressor. This results in reducing electric power input for a refrigeration system while maintaining the same cooling capacity. As the efficiency of the conventional ejector refrigeration system is too low, the conventional ejector (thermal compressor) is replaced with an ejector with internal cooling.*

KEYWORDS *–Ejector, Absorption, Refrigeration, Thermodynamic, Cycle / System, Chiller*

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I. INTRODUCTION

1.1. Absorption refrigeration system (ARS) driven by solar energy applied in Qatar 2022 FIFA World Cup Stadium Showcase

Absorption refrigeration system is applied in the prototype stadium Showcase at Al Thumama Stadium (refer to Figure 1) which will be used by GORD until 2022 to demonstrate using the power of the sun for stadium cooling, through their new hybrid super-efficient technology. The cutting edge cooling system uses solar collectors that track the sun, absorption chillers and eutectic storage tanks (refer to [1]).

Qatar 2022 FIFA Showcase is the world's most sustainable stadium, a radical piece of environmental architecture that was a major driver in Qatar's sustainability plan and World Cup bid.

1.2. Thermal compressor (ejector / injector)

In thermal compressors a high heat potential (enthalpy) of the motive stream of fluid is utilized to increase the pressure of a low heat potential suction stream of fluid (refrigerant at the outlet of an evaporator) in order to reach medium pressure (refer to [2]) .

In other words, a hot and high pressure motive fluid enters the nozzle of the injector, where it expands and its velocity and kinetic energy increase (its enthalpy decreases), as well as its pressure is reduced to the low suction pressure. Then a motive stream of vapor withdraws the cold stream of vapor and they are mixed in the mixing chamber of the injector. Then the mixture of vapors enters the diffuser where it is compressed to the medium pressure and its velocity and kinetic energy decrease (its enthalpy increases).

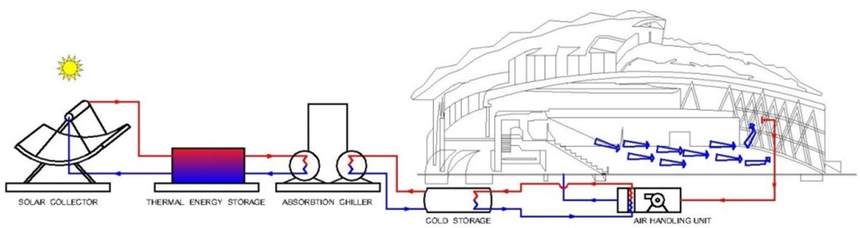

Figure1: Absorption refrigeration system applied in Qatar 2022 FIFA Showcase stadium

The 1st Law of Thermodynamics(refer to [3])for the processes (expansion and compression) in a thermal compressor (ejector, injector) states that the change in the heat potential (enthalpy) and the change in kinetic energy of a isolated system is equal to the amount of heat transferred into the system and the amount of technical work done by the system on its surroundings.

$q + l_t = h_A – h_3 + 1/2·(v_A^2 – v_3^2)$ (1)

where:

$q = 0$ ……….…. quantity of heat exchanged with its surroundings
$l_t = 0$ ……….…. quantity of technical work done by the system on its surroundings
h_i ……..……… enthalpy (heat potential) [kJ/kg]
v_i ……………. velocity [m/s]
$i = \{1,2,3,4,5,A,B,C,D,E,…\}$ ….. states of the refrigerant

$h_3 + v_3^2/2 = h_A + v_A^2/2 = E_{HP} + E_K = E_{Tot} = Const$ (2)

Also, the Law of Conservation of Energy states that the total energy of an isolated system is constant. Therefore, energy cannot be created nor destroyed. It can only be converted from one form to another.

The equations that determinate the state of the mixture after mixing of two vapor streams (motive and suction) in the mixing chamber and with respect to the Law of Conservation of Energy(refer to [2]):

$h_C · (m_3 + m_5) = h_3 · m_3 + h_5 · m_5$ (3)

$s_C · (m_3 + m_5) = s_3 · m_3 + s_5 · m_5$ (4)

where:

m_i……....mass fluid flow(kg/s)
p_i ……. pressure [bar]
s_i ……..entropy [kJ/(kg·K)]
t_i …….. temperature [°C]

1.3. Thermal compressor (ejector/ injector) with internal cooling

Efficiency of the conventional injector is less than 35%. The main reason of low efficiency of a thermal compressor is the increase of entropy of the mixture of vapors. The friction between fluid particles (molecules) and between them and the walls of a thermal compressor causes losses in kinetic energy (entropy generation).

So the heat potential energy (enthalpy) of a motive stream of fluid is mainly used to increase the entropy of mixture instead of its pressure.

Actually increasing the enthalpy of a suction stream of fluid is not a goal. So, in order to increase its pressure in an efficient manner, a high density fluid can be injected into the fluid mixture (refer to Figure 3). In this way, the compression of the mixture in a mixing chamber takes place simultaneously with mixing with a high density fluid. So the kinetic energy of both streams, in the first place, is used for increasing the pressure of the secondary mixture, while reducing the entropy of the primary mixture, as well as rising the entropy of the secondary cooling stream (with high density fluid). In this way, the compression of the secondary mixture is intensified. Finally, this results in the increase of the overall efficiency of an injector.

Actually, a part of liquid from the secondary cooling stream evaporates and cools the secondary mixture. In other words, the entropy generated by friction cannot be destroyed, but it is only transferred from the primary mixture to the secondary cooling stream (refer to [2]).

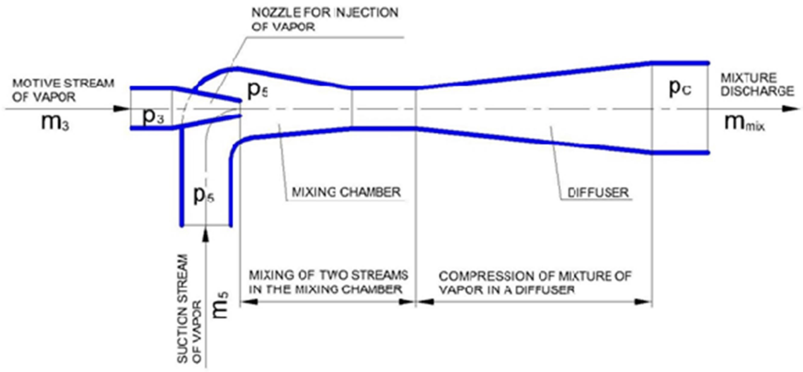

Figure 2: Conventional thermal compressor (ejector /injector)

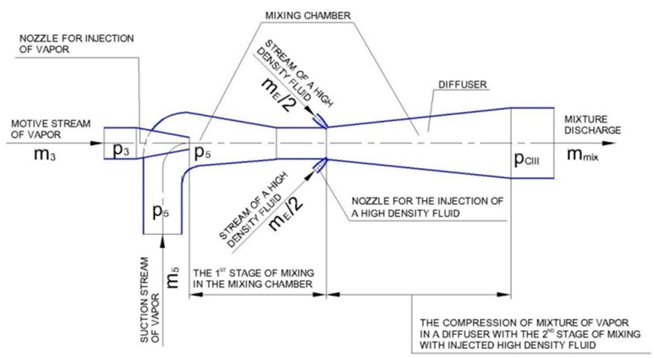

Figure 3: Thermal compressor (ejector / injector) with internal cooling.

II. SYSTEM DESCRIPTION AND THE CALCULATION MODEL

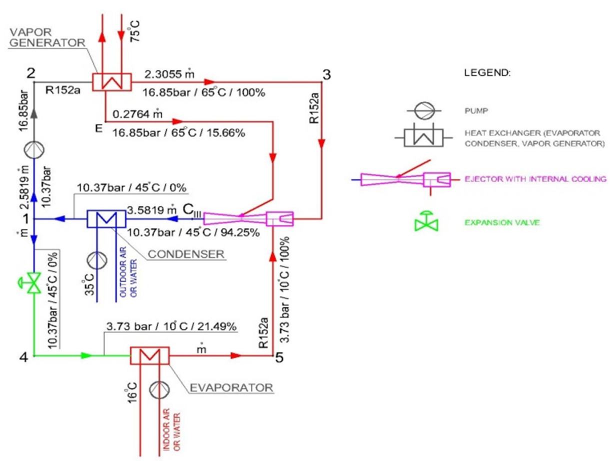

Figure 4: Schematic diagram of the ejector refrigeration system (ERS) with internal cooling for the refrigerant R152a.

After a medium pressure (10.37 bar) liquid refrigerant R152a (3.5819 m) leaves the condenser (point 1), it is divided into two streams. The first one (m) goes through the expansion valve (the cold stream) where its pressure is reduced to 3.73 bar and begins to expand and evaporate (21.49%). Its evaporation ends (100%) in the evaporator (point 5). The other medium pressure stream (2.5819 m) enters the pump which increases its pressure to 16.85 bar. Then the liquid refrigerant enters vapor generator (point 2) where it evaporates. This hot stream is also divided into two streams. When 15.66% of liquid evaporates, the first stream (0.2764 m) is taken from the vapor generator and injected into the inlet of the diffuser (mixing chamber). The rest of the hot stream (2.3055

m) continues to evaporate until there is no liquid in it (the state of 100% of vapor). Then such hot and high pressure fluid enters the nozzle of the injector, where it expands to the state A and its velocity and kinetic energy increase (its enthalpy decreases), as well as its pressure is reduced to the pressure 3.73 bar.

Then a motive stream of vapor withdraws the cold stream of vapor and they are mixed in the mixing chamber of the injector (state B_I). Then the mixture of vapors ($m_{BI}=m_3+m_5$) enters the diffuser where a high density fluid is also injected through the nozzle. After that the process of further mixing with a high density fluid takes place simultaneously with the process of compression of the mixture in a diffuser. As the high density fluid is going through the nozzle, it accelerates and its heat potential (enthalpy) decreases, but its kinetic energy increases. This increase in kinetic energy is later used to increase the pressure of this fraction (m_E) of the working fluid. At last, the new mixture ($m_{mix}=m_3+m_5+m_E$) is compressed to the state C_{III} and its velocity and kinetic energy decrease (its enthalpy increases), as well as its pressure increases to the condensing pressure 10.37 bar (C_{III}). After that, the vapor mixture enters the condenser where it is condensed.

The initial assumptions for the calculation of a real ejector refrigeration system (ERS) with internal cooling and with all irreversible processes (i.e. with friction):

- Evaporating temperature in a vapor generator: $T_{GE}= 65℃$
- Evaporating temperature in a evaporator: $T_{EE}= 10℃$
- Condensing temperature in a condenser: $T_{CC}= 45℃$
- A high density fluid of the state E is injected into the inlet of a diffuser of an injector. It expands in the nozzle to the state D.
- The velocity of a high density fluid and a vapor mixture entering the diffuser are similar: $v_D \approx v_{BI}$.
- The compression of a mixture in a diffuser takes place simultaneously with the mixing with a high density fluid injected into a diffuser. The final state of the secondary mixture ($m_3+m_5+m_E$) is C_{III}.
- The final enthalpy of the secondary mixture ($m_3+m_5+m_E$) is the same as the final enthalpy of the primary mixture (m_3+m_{13}), i.e. $h_{CIII}=h_{BI}$ (isenthalpic process).

The equations that determinate the states of a mixture with respect to the Law of Conservation of Energy (refer to [2]):

- for the 1st mixing of two vapor streams (motive and suction) in the mixing chamber:

$h_{CII} \cdot (m_3 + m_5) = h_3 \cdot m_3 + h_5 \cdot m_5$ (5)

- for the 2nd mixing of a vapor mixture and a high density fluid in a diffuser:

$h_{CIII} \cdot (m_E + m_3 + m_5) = h_{CII} \cdot (m_3 + m_5) + h_E \cdot m_E$ (6)

The equations that determinate the states of a mixture with respect to The 1st Law of Thermodynamics (refer to [3]):

$h_3 + v_3^2/2 = h_A + v_A^2/2$ (7)

$h_{CII} + v_{CII}^2/2 = h_{Br} + v_{Br}^2/2 = h_{BI} + v_{BI}^2/2$ (8)

$h_E + v_E^2/2 = h_D + v_D^2/2$ (9)

III. RESULTS AND DISCUSSION

For the following values of the flow ratio:

$k_I = m_3/m_5 = 2.3055$ (10)

$k_{II} = m_E/m_{BI} = 0.0836$ (11)

the following values for h_{CIII} and s_{CIII} are calculated:

$h_{CIII} = (k_I \cdot h_3 + h_5 + h_E \cdot k_{II} \cdot (k_I + 1))/((k_I + 1) \cdot (k_{II} + 1)) = 519.3680$ kJ/kg (12)

$s_{CIII} = (k_I \cdot s_3 + s_5 + s_E \cdot k_{II} \cdot (k_I + 1))/((k_I + 1) \cdot (k_{II} + 1)) = 2.371062$ kJ/(kg·K) (13)

The technical work done by the pump:

$l_t = k_2 \cdot l_{t1\text{-}2} = (k_I + k_{II} \cdot (k_I + 1)) \cdot l_{t1\text{-}2} = 2.84$ kJ/kg (14)

The heat absorbed by the technical system:

$q_{Gen} = (k_I + k_{II} \cdot (k_I+1)) \cdot q_{2\text{-}E} + k_I \cdot q_{3E} = 618.06$ kJ/kg (15)

$q_{Evap} = m_5 \cdot q_{4\text{-}5} = q_{4\text{-}5} = 232.79$ kJ/kg (16)

The heat rejected by the technical system:

$q_{out} = k_{tot} \cdot q_{CIII\text{-}1} = (k_I + 1) \cdot (k_{II} + 1) \cdot q_{cIII\text{-}1} = -853.69$ kJ/kg (17)

Table I: States of the refrigerant R152a and its parameters for the ejector refrigeration system with internal cooling.

Refrigerant	***R152a***								
State	***t***	***p***	***h***	***s***	***x***	***ρ***	***η***	***l_t***	***q***
	°C	*bar*	*kJ/kg*	*kJ/(kgK)*	*-*	*kg/m³*	*-*	*kJ/kg*	*kJ/kg*
1	45.00	10.37	281.03	1.6237	0.0000	845.00			
2r	45.40	16.85	281.80	1.6261				0.77	
2	45.58	16.85	282.13	1.6272			0.70	1.10	
3'	65.00	16.85	320.03	1.7402	0.0000	782.00			
3	65.00	16.85	541.51	2.3952	1.0000	54.94			186.79
4	10.00	3.73	281.03	1.6393	0.2149				
5	10.00	3.73	513.82	2.4614	1.0000	11.65			232.79
5'	10.00	3.73	217.30	1.4142	0.0000	936.00			
1"	45.00	10.37	533.90	2.4185	1.0000	32.41			
Ar	10.00	3.73	495.08	2.3952				46.44	
A	10.00	3.73	502.04	2.4198	0.9603		0.85	39.47	
Br	10.00	3.73	513.93	2.4617					
B_I	10.00	3.73	519.37	2.4808			0.50		
Dr	10.00	3.73	338.66	1.8428				16.06	
D	10.00	3.73	341.07	1.8513	0.4174		0.85	13.65	
E	65.00	16.85	354.72	1.8428	0.1566				72.59
C_{III}	45.00	10.37	519.37	2.3711	0.9425				-238.34

The coefficient of performance of the ejector refrigeration system with respect to work applied to it (≈ electric energy input):

$COP_I = q_{Evap} / l_t = 81.9672 = 8196.72$ % (18)

The coefficient of performance of the ejector refrigeration system with respect to thermal energy applied to it (heat input to the vapor generator):

$COP_{Gen} = q_{Evap} / q_{Gen} = 0.3766 = 37.66$ % (19)

The total coefficient of performance of the ejector refrigeration system:

$COP_{Tot} = q_{Evap} / (l_t + q_{Gen}) = 0.3749 = 37.49$ % (20)

As it is generally known, in the ejector refrigeration system the conventional (mechanical) compressor is replaced with a pump and an ejector (injector, thermal compressor with no moving parts). For the same refrigeration effect, the work of a pump is significantly lower (more than 25 times in a case of the advanced ejector) than the work of a mechanical compressor. This results in reducing electric power input for a refrigeration system while maintaining the same cooling capacity.

The coefficient of performance of the advanced ejector refrigeration system with respect to work applied to it (≈ electric energy input of the pump) is:

$COP_I = q_{Evap} / l_t = 81{,}9672 = 8196{,}72$ %

This means that the cooling capacity of the advanced ERC is over 80 times larger than the pump power. If the electric power inputs of external pump (e.g. for a water circuit through solar panels) and condenser fans are taken into account, then the COP_I for the entire system would be more than 20 (2000%). This is still at least 7 times more than in a case of current chillers with $COP_I \approx 3$ (300%).

The total coefficient of performance of the ejector refrigeration system:

$COP_{Tot} = q_{Evap} / (q_{Gen} + l_t) = 0{,}3749 = 37{,}49\%$

This means that the initial investment in such system is significantly higher, but the total electric power input is at least 7 times less than in a case of current chillers (with mechanical compressors). In the long run the total costs (investment and operational) will decrease after a certain period of system operation (e.g. 3 to 8 years depending on a type of a heat source, architectural design, a price of electricity, etc.).

The best type of a heat source for ejector refrigeration system (ERS) would be solar-thermal energy, geothermal energy or waste heat from thermal power plants, as well as foundries.

The following types of the advanced ejector refrigeration systems will be mainly applied in the near future:

1. The ejector refrigeration system driven by solar energy (Figure 5).

In this case each part of the building envelope, except windows, can be used for installation of solar collectors, including parapets. In this way the heat absorbed by the building envelope with solar collectors will be utilized for cooling of the building, i.e. extracting the heat (entered the building through the windows) from the building. So, new panels for building envelope should be developed, so they would look nice while being able to collect solar energy.

In that way, building envelope would become active and collect solar energy that will be utilized for cooling the building.

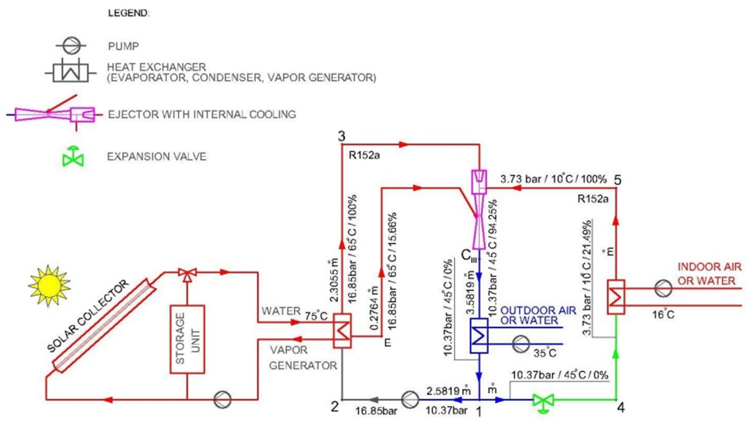

Figure 5: Schematic diagram of the ejector refrigeration system driven by solar energy.

Compared to absorption refrigeration systems (ARS), ejector refrigeration systems (ERS) can operate at lower generator temperature. For instance, coefficient of performance (COP) of ARS is very low or even 0 for generator temperature of 65˚C, while COP of ERS (with internal cooling) is around 40% for the same generator temperature.

2. The ejector refrigeration system driven by waste heat from a power plant (Figure 6).

As you can see from the Fig. 6, a district heating system can be also used as a district cooling system in Summer.

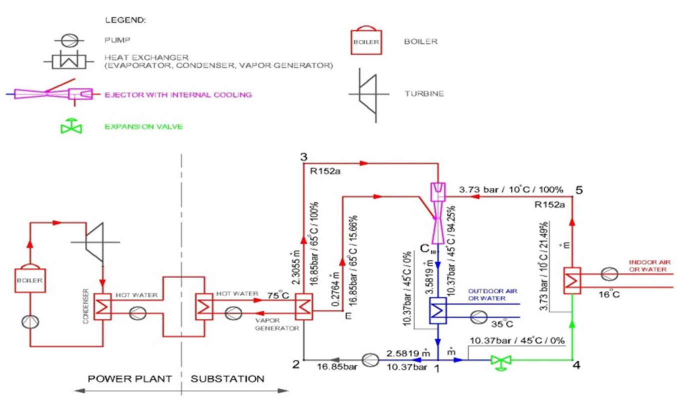

Figure 6: Schematic diagram of the ejector refrigeration system driven by waste heat from a power plant.

IV. CONCLUSION

In conclusion, ejector refrigeration systems (ejector chillers) with internal cooling can utilize mostly solar thermal energy or waste heat from thermal power plants or boiler plants instead of electric power.

Compared to absorption chillers, ejector chillers with internal cooling can operate at lower generator temperature. Therefore, ejector chillers with internal cooling can be applied in solar thermal systems at lower temperature of a heat source.

As the ejector refrigeration system with internal cooling is simpler and more efficient than the absorption refrigeration system for the same generator temperature, ejector chillers can be used in the trigeneration (combined cooling, heat and power – CCHP) system for generating electricity, heat and chilled water instead of absorption chillers.

Therefore, when developed fully, ejector chillers with internal cooling could replace both conventional chillers with mechanical compressors (with high electric power consumption) and absorption chillers for cooling buildings and industrial facilities.

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