THE REVIEW OF JET REFRIGERATION SYSTEM DESIGN & PERFORMANCE

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ABSTRACT

This system will be presented in this paper application are explain the steam jet refrigeration system the aim of this paper is to develop the jet refrigerator and the test under the actual ambient condition in the word it is driven by low grade thermal energy .refrigerator can operated actual ambient condition cooling water produced-chilling water from 10to 5about 1 parcet of water flowing through evaporator most be v porized analysis of refrigeration system .plant have a robust opretion and be easily manufactured at relatively low cost with off the shelf components . This paper presents a plant concept of a solar-driven steam jet ejector chiller with latent heat and cold storage. The concept will be realized in a first demonstration plant. The solar cooling plant will consist of a solar collector field based on evacuated tube collectors with a thermal output of 200 kW, a double stage steam jet ejector chiller with a cooling capacity of 80 kW and two thermal energy storage units, meaning a heat storage unit using polyethylene as latent heat storage medium and a cold storage unit using a paraffin/water dispersion as latent cold storage fluid. While designing the solar cooling plant some preliminary research works have been accomplished with special regard to the development and integration of the heat and cold storage units. The results of this research as well as the overall design of the system will be presented in this paper.

I. INTRODUCTION

The esteam jet refrigeration cycle is quite similar to more conventionl refrigeration cycle, with in evaporator, a compression devices, a condenar and refrigretor as the basic system components .this system instead of mechanical compression device characteristically employs a steam ejector booster to compress the refrigerant to the condenser pressure level Suparsonic ejector may be seen

Jet refrigeration system, which is one of the heat-powered refrigeration cycles, is of current interest. The distinctive point osuch cycle is that it is relatively simple to design, construct and operatecompared to the other types of heat-powered refrigeration systems. In this cycle, the importantequipment known as an ejector is used as a main driving part for the system. This is because an ejector isused to elevate refrigerant pressure similar to the use of a mechanical compressor as a modern evalution of the injector invented by henry Giffard in 1858 and used thereafter for feeding water to steam boilers (jet pump)or creating vacuum in rail brake system .in a refrigeration system the ejector may substitute the compressor and hence produce a heat driven machine with a boiler ,evaporator and condenser (fig.1)

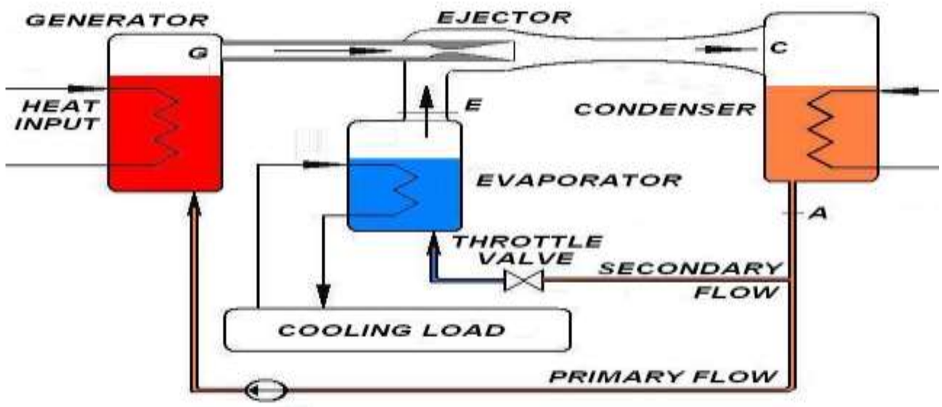


Fig.1 Steam Jet Refrigeration Cycle

High pressure vapor is produced in the genera (point G) .flow through the primary nozzle of the ejector and ,as it enters the mixing chamber .entrains the secondry fluid being drawn from the evaporator (point E).the combined flow is then compressed as it flow through the diffuser section of the ejector into the condenser (point C) the codenset (point A) is split into two stream ;one is expnded through a throttling valve and fed back to the evaporator whilst the other is returned via a feed pump to the boiler to maintain the refrigerator leval .The system COP .pump work usully modest ,is approximation given by:

CPO= $Q_s/$= $\omega \Delta h_s/$

Where $\omega = m_s/$ is the entrainment ratio between secondry and primary mass flow rate .for a given fluid ,therefor the system performance is mainly by the entrainment ratio which in turn is a fuction;

$\omega = f(\beta, \varphi, \varepsilon)$ pressure ratio = $p_c/$and = $p_G/$and and ,subscripts c,E and G referring to condenser ,evaporators and genretors while D and mixer and primary nozzle throats .hence the fundmentl problem when designing an ejector chiller is the optimization of the internal flow occurring within the ejector chillers ,entrainment ratio reported in the literature.

Green house gas (GHG) emission that is generated during electricity generation is released increasingly to the environment .this reflects the fect that refrigeration and air –conditioning systems are one of the high emitters of GHGto the environment. To reduce the demand of electricity for refrigeration application, alternative refrigeration system that can be opreted by using thermal energy,heat-powered refrigeration cycle, is introduced at present time.

II. PRINCIPALE OF STEAM JET REFRIGERATION SYSTEM:-

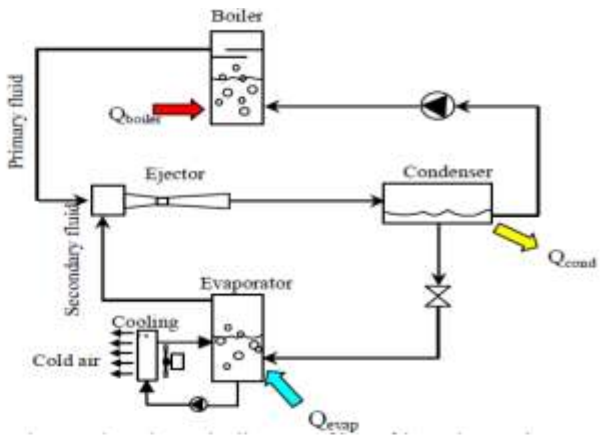


Fig.2 Steam Jet Refrigeration System

When external pressure on a liqud changes,its boiling point also changes. Normally, atmospheric pressure act on the surface on liquid .howevere, if by some means this pressure is reduced then liquid wiil start boling at lower temperature.this basic principal of boiling of liquid at low temperature dy reducing the pressure on it surface is used in steam jet refrigeration system. The pure water boil at 100 C at standard pressure of 760mm of Hg (1.013bar).it has been observed that water boils at 12 ^{0}C and 7^{0}C if the pressure on the surface of water are kept at 0.014bar and 0.01bar respectively.the very low pressure or high vacuum on the surface of the water can be maintained by throttling the steam through the jets or nozzles.

III. DESCRIPTION OF SYSTEM COMPONENTS

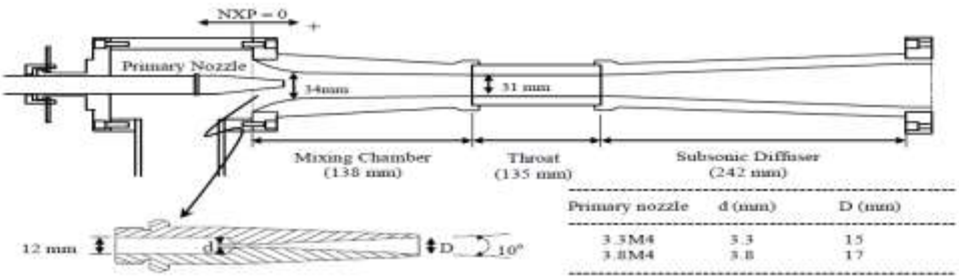
The component of steam jet refrigeration system is brifly described below:-

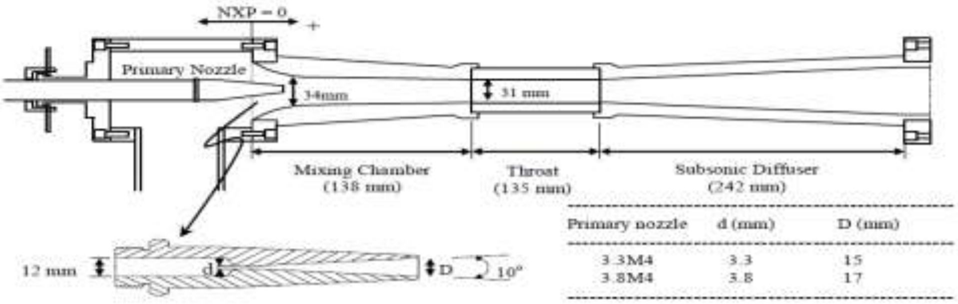
3.1 Evaporator

The evaporator orflash chamber is a large vessel and is heavily insulated to avoid the rise in temperature of water due to highy ambient temperature. It is fitted with perforated pipesfoor spraying water.the warm water coming out of the refrigerator space is sprayed into flash water chamber where some of which is converted into vapour after absorbing the latent heat. Thereby cooling the rest of water.

3.2 Steam Jet Ejector

High pressure steam from the boiler(generally colled primary fluid or motive steam/vapour) is admitted to the steam chest and expanded in convergent –divergent nozzle to a very low pressure and the attains supersonic velocities in the range of 1000m/s to 1350m/s the flash chamber is connected to the region of low pressure of the ejector the water vepour from the flash chamber is entrained in the highy velocity jet of steam and both are mixed in the mixing section at the constent pressure .the men velocity of mixture will be super sonic after the completion of the mixing process. This supersonic steam get a normal shock in constent area throt of diffuser,resulting in rise of pressure and subsonic flow.the diverting portion of the diffuser velocity head is recovered as pressure head and finaly high pressure steam is condensed in condenser.





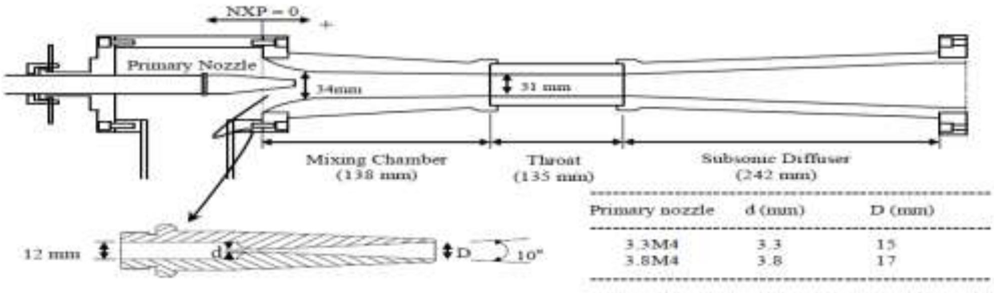


Fig.3 The Significant Dimention of The Ejectior and Nozzel Used For Testing The Refrigerator.

3.3 Condenser

It is employed because of its compact size and ability to be located anywhere including .inside the building in this case steam consumption is higher and larger surface condenser and cooling water will be required ite is used to advantage in those installation which do not have space limitation and whear it is economically feasible to treat the evaporator and boiler water on a onece through basis.

3.4 Air Ejector

The air or non condensable gas ejector typically employs two small steam powered ejector in series an inter and after condenser is often employed to improve economy in a tight system ,an air ejector unit will have capacity of 0.05kg of air per tone of refrigerator.

IV. THE COFFICIENT OF PERFORMANCE WITH THE VARIATTON OF COOLING CAPACITY

In order to analyze the system performance of the jet refrigerator, the well known term ,the coefficient of performance (COP), is necessary to know because it indicates the overall performance of the refrigerator .the coefficient of performance of the jet refrigrerator can be defined by equation COP= $Q_{coolina}/Q_{bo}$ Where Q_{cooli} is the cooling load produced at the evaporator Q_{boi} Is the heat supplied to the boiler The heat supplied to the boiler can be calculated by equation (5)

$$Q_{boiler} = m_b(h_{g@boi} - h_{f@cc})$$

Where, $h_{g@boi}$ is the saturated vapour enthalpy produced by boiler (kj/kg),

$h_{f@cond}$ is the saturatLiquid enthalpy at the condenser (kj/kg)

$\dot{m}$ Is the primary fluid mass flow rate (kg/s)

In this case, the primary fluid mass flow rate produced by the boiler can be calculated by equation (6)

$$\dot{m} = A^* P_o \sqrt{k/}(2/k+1)^{k+1/[2(k-1]}$$

Where, A is the cross section area of the primary nozzle throat(m^2) is the boiler pressure(kPa) is the boiler tempreture(K) ,K is the specific heat ratio,R is the ideal gas constant (kj/kg,k)

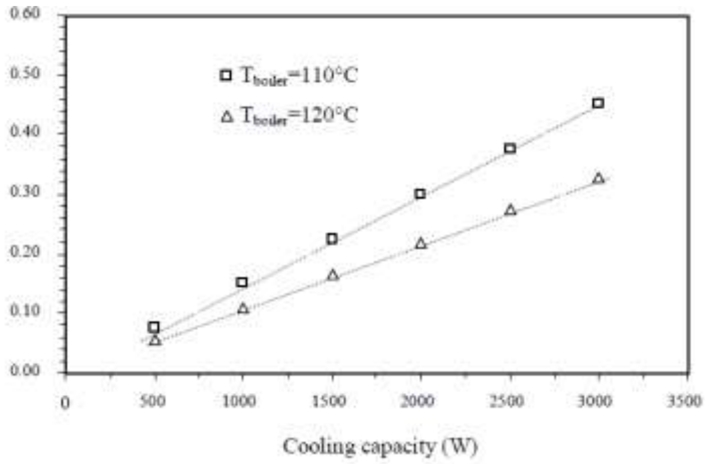


Fig.4 Variation of The COP With A Cooling Capacity

It can see from fig. that with a fixed boiler temperature the COP of the jet refrigeration is increased linearly with an increase in the cooling capacity. The lowest COP for the test is of 0.03for the cooling capacity of 500W.Meanwhile, the highest COPof 0.45 is obtained at the cooling capacity of 3000W.an increase of COP with cooling capacity is the result of in increase in cooling capacity while the heat supplied to the boiler is kept constant due to fixed boiler temperature.therefore,COP of the jet refrigerature is increased with cooling capacity. It comes together with an undersired cooling.temperature, due to increase in the evaporator temperature.

IV. DESIGN OF STEAM JET REFRIGERATION SYSTEM

According to the results gathered with the first two ejectors, an increase in the mixing zone and diffuser lengthcan be favorable, as long as theimprovement in terms of momentum exchange between primary and secondary flow balances the increased friction loss. Hence the third ejector was designed with length between the flanges increased from 740 to 950 mm. The shape is a scaled up and modified version of a CRMC profile designed by Ian Eames. The last part is shaped as a straight cone with 4° half angle. The inlet has a fillet that allows a smooth transition from the cylindrical pipe upstream. The primary nozzle, for the moment, was left unchanged. The number of pressure ports was increased to 9. Extensive testing has been carried out on this ejector. Stable operation is obtained with saturation temperatures of 5°C at evaporator and 100°C at generator. Primary nozzle position has been optimized and the best results have been obtained with a displacement of 8 mm to the right with reference to the position shown in. shows the COP as a function of condenser saturation temperature. In this condition the critical condenser temperature is around36.2°C. Fig. 8b shows how the increase in condenser temperature and pressure modifies the static pressure along theejector. At low condenser pressure (lower curve) the flow is still supersonic at the diffuser throat and hence reexpandsafterwards. This produces a normal shock somewhere in the divergent part of the diffuser. At intermediatecondenser pressurethe normal shock moves leftwards (intermediate curves) until, at still higher pressure the shockovercomes the throat and the ejectorceases to be supersonic. This corresponds to the last point on the right of Fig.8a and indicates a highly unstable working condition.When evaporator temperature is raised to 10°C, the COP increases to 0.29. This result is lower than that reportedby Eames et al. in , but they used an higher generator temperature (110°C) and a bigger area ratio, whichincreases COP but lowers critical condenser temperature. On the other hand, the experimental results given in terms of entrainment ratio by Scott et al. in , when converted to COP, indicate a lower performance.

V. CASE STUDY OF REFRIGERATION SYSTEM

There are vast quantities of cooling requirements around Tushan Cogeneration Company, a power plant in An-hui province of China. The total cooling load of the company reaches the number of 69780kW while the amount of surplus steam quantities up to 240 ton per hour. The pressure of the surplus steam is1.27Mpa and the temperature is 304℃. We are going to design an absorption system and a combined cycle system respectively for district cooling in the power plant and then compare the results obtained. In China's current industry standard, the driving steam of the double-effect Li-Br absorption refrigerator can be saturated steam at the pressure of 0.4MPa, 0.6MPa and 0.8MPa. We call the pressure at which the steam entering the absorption refrigerator as the mid-pressure, and we have to carry out the thermodynamic calculation of combined cycle systems with different mid-pressures. In the calculation, the total cooling load is divided into 20 units so that the capacity of each refrigerator is reasonable. The design conditions in the thermodynamic calculationThe result of thermodynamic calculation and the consumption of superheated steam under the designed working conditions are listed in table.2.Synthetic refrigeration coefficient means the refrigeration coefficient of the whole system: it is a combined action of absorption refrigerator and temperature and pressure reducer in an absorption system, while it is a combined action of compression refrigerator, superheated steam saturator and absorption refrigerator in a combined cycle system. It reflects the refrigeration capacity of the whole system. It can be observed that combined cycle system with the midpressure of 0.8MPa has the strongest refrigeration capacity. As the transfer efficiency can reach up to 0.8 in actual operation, the energy conservation potential of combined cycle system can be predicted.[10]

VI. HEAT STORAGE MATERIAL OF THE PLANT

The function of the heat storage unit in the system is to ensure the constant supply of motive steam tothe SJEC. Furthermore, the space for the erection of the heat storage unit is limited due to the intendedinstallatiointo the container, so that the volume of the heat storage unit should be as small as possible.This leads to the use of a phase changematerial (PCM) to realize a latent heat storage unit. The PCM isused as storage material which stores heat energy in form of latent heat of fusion during a phase transitionover a temperature shift. The latent heat of fusion of a material is higher than the sensible heat capacity ofthe material so that the use of a PCM allows increasing the heat density of a storage unit and therefore toreduce its volume.In a first approach the operating temperature of the heat storage unit was assigned to be in the rangefrom 120 °C to 150 °C. In view of this temperature range the listed storage materials in Table 1 wereconsidered for the heat storage unit.

Storage material	Temperature in °C	Specific heat of fusion in kJ/kg
Water (mentioned as reference)	120-150	*128.5* (sensible heat!)
Polyethylene PE-HD [GHR 8110]	122-133	150-200
Polyethylene PE-UHM [GUR4120]	124-134	150-200
Polyethylene Licocene PE 4201	125-130	246
Polypropylene Licocene PP 6102	142-148	70
Polypropylene Licocene PP 7502	155-165	106
KNO_3-$NaNO_2$-$NaNO_3$ (53-40-7)	142	80
Mannitol	164-167	294-325
Palatinitol (Isomalt)	145	170

In some preliminary tests the heat of fusion and the thermal stability of the material were investigated.The preliminary tests were conducted in an oven and include heating-up and afterwards cooling down thematerial. This procedure wasconducted 30 times, with constant measuring of the temperatures of theoven and the material. The maximum temperature was 180 °C. Especially polyethylene shows a highspecific heat of fusion in combination with a good thermal stability and market availability so thatpolyethylene was chosen as PCM for the heat storage unit. The temperature curves and additionaldifferential scanning calorimetry (DSC) measurements before and after the tests indicate no change of thetemperature range and the specific heat of fusion. But as depicted in a visual change of the materialcan is observed so that it was decided to limit the maximum temperature of the heat storage unit of the SJEC plant to 145 °C.[11]

VII. CONCLUSIONS

It is found that the prototype steam jet refrigerator can completely be operated with variousrange of the operating-condition and primary nozzle used which is suitable for air-conditioning system(chilled water produced by refrigerator between 5°C and 17°C). The refrigerator can provide themaximum COP of 0.45 at the cooling capacity of 3000W, evaporator temperature of 17°C and boiler temperature of 110°C.

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