PERFORMANCE OF WASTE HEAT ABSORPTION REFRIGERATION SYSTEM

Osama A. El Masry

Professor of Mechanical Engineering, Collage of Engineering, King Saud University
P.O. Box 800, Riyadh 11422, KSA

oelmasry@ksu.edu.sa

ABSTRACT

It is generally recognized that considerable energy savings could result from installing a waste heat recovery system in the exhaust of a Gas Turbine power plant. The present study proposes to use that recovered heat to supply heat to an absorption refrigeration system (ARS). Various sources and levels of extracted heat loads from exhaust gases were studied for possible utilization with the (ARS). Typical waste heat available loads were found to be in the range of 58-62% of the heat supplied to a frame 7 Gas Turbine unit. A comparative study of the Water-Multicomponent Salt Mixture, the conventional Water Ammonia and the Water-Lithium bromide systems was carried out for both single-stage and double-effect absorption cycles. The results showed that the Coefficient of Performance (C.O.P) is higher for Water-MCS mixture than that of the Water-LiBr solution. Water-MCS mixture can also be used for a wide range of operating loads and conditions; therefore it is suitable for use in absorption refrigeration systems associated with variable load waste heat sources. Water-MCS mixture is approximately five times cheaper than LiBr, which makes the cooling system more economical. A laboratory experiment for a simple single-stage absorption system utilizing the Water-MCS mixture was constructed and operated. The system was tested to determine the thermal performance under variable operating conditions. Actual system C.O.P. values were found to follow the theoretical analysis trend but with less values.

Keywords: *energy savings, waste heat recovery, absorption refrigeration, Multicomponent Salt Mixture, Water Ammonia Water-Lithium bromide system, system C.O.P*

الملخص

من المعروف أنه من الممكن الحصول على وفر كبير في الطاقة عند إضافة نظام لاسترجاع فاقد الحرارة من غازات العادم في محطات التوربينات الغازية. وتقترح الدراسة الحالية استخدام الحرارة المسترجعة لإمداد وحدة تبريد بالامتصاص. تم دراسة العديد من المستويات والأحمال الحرارية الناتجة من غازات العادم لاحتمال استخدامها مع نظام التبريد بالامتصاص. وقد وجد أن الأحمال الحرارية تتراوح بين ٥٨-٦٢% من الحرارة المضافة لتوربين غازي من المستوى ٧. تم كذلك دراسة أنظمة خليط الماء-الملح متعدد المكونات ومحلول الماء والنشادر-ومحلول الماء والليثيوم بروميد في دورة التبريد بالامتصاص أحادية المرحلة. وقد أظهرت النتائج أن معامل الأداء للدورة التي تستخدم خليط الماء والملح متعدد المكونات أعلى من مثيله في الدورة التي تستخدم خليط الماء-محلول الليثيوم بروميد. كما ان خليط الماء-الملح متعدد المكونات ممكن استخدامه في مدى أوسع من أحمال التشغيل وظروفه, لذلك فهو الأنسب للاستخدام مع نظام تبريد

بالامتصاص مرافق لمصدر حراري متغير الحمل. كما أن خليط الماء والملح متعدد المكونات أرخص من خليط الماء-محلول الليثيوم بروميد بنسبة ٥:١ مما يجعل نظام التبريد أفضل اقتصاديا. تم معملياً إنشاء وتشغيل وحدة تبريد بالامتصاص أحادية المرحلة تستخدم خليط الماء-الملح متعدد المكونات. تم إيجاد الأداء الحراري للوحدة تحت ظروف تشغيل متغيرة. وقد وجد أن قيم معامل الأداء تتبع الاتجاهات الناتجة من التحليل النظري ولكن بقيم أقل.

INTRODUCTION

Most power plants being installed in Saudi Arabia are Simple Cycle Gas Turbine plants that burn increasingly expensive Diesel and/or natural gas to produce electricity at up to 28-32% efficiency [Nagpal, 1980, E C, KSA, 1998 and El Masry; 2002]. These gas turbine plants exhaust gases at high enough temperature to be considered unnecessary waste. It is widely believed nowadays that considerable energy savings could result from installing a heat exchanger in the exhaust of the turbine [Boyen, 1980 and GE, 1978], if used to recover that heat. The recovered heat can be used in a variety of applications [Goldstick & Thumann, 1983]. On the other hand, great strides have been made to use the heat energy instead of work energy for producing refrigeration effect because it gives high system performance compared with machine operated with supply of work energy. Absorption refrigeration cycle is one of the oldest known cycles used for producing refrigeration effects [Alizadeh et al, 1979]. The present research aims at studying the technical feasibility of utilizing the waste heat of gas turbine power plants for the purpose of powering an absorption refrigeration system (ARS). The cooling effect of the ARS can be utilized in a variety of application among which is the cooling of the inlet air to the gas turbine itself [Fernández-Searaa, 1998].

The absorption refrigeration system is heat-operated unit, which uses a refrigerant that is alternately absorbed by and liberated from the absorbent. Absorption units operate on the simple principle that under low absolute pressure, water will boil at a low temperature. The two-shell cooling units use heat to produce refrigeration efficiently. The lower shell contains an absorber and evaporator, while the upper shell consists of generator and condenser sections. The major question in the design of an absorption cooling cycles is the choice of working fluids due to the long list of available fluid combinations and the complex mixing characteristics of these mixtures [Fernández-Searaa, 1998].

Two working fluids are employed in the system, a refrigerant and an absorbent. It is important that both the refrigerant and the absorbent must have certain physical properties for an efficient working system. A general theoretical study on design and optimization of water-Lithium Bromide and the Ammonia-Water absorption refrigeration cycles has been carried out [Mansoori & Patel; 1979]. It has been shown that the cooling ratio defined as the ratio of the energy removed from the surroundings during the refrigeration phase to that supplied to the generator during the regeneration phase, increases as the condenser and absorber temperature decreases (for a fixed evaporator temperature). A comparative study of the Water-Multi-Component Salt mixture (MCS) and the conventional Water-Lithium Bromide mixture was reported [Kaushik et al, 1980], it has been found that the cooling C.O.P is higher for the Water-mixture than that for the Water-Lithium Bromide case. In separate studies of commercially available Water-Lithium Bromide absorption chillier [Ayyash, 1980 & Suri, & Ayyash, 1984], estimates of electrical energy saving have been made with the an absorption system when compared to the conventional vapor compression of equivalent cooling capacity.

Objectives of the present study include selecting the appropriate refrigerant for use with the waste heat loads of the Gas Turbine. Comparison of different refrigerants performance (namely: the NH_3- Water, Water-LiBr and Water-MCS) with the system, and experimental evaluation of the ARS to determine the coefficient of performance of the system within its application range.

ANALYSIS

1. AVAILABLE HEAT ENERGY

Two power plants in Riyadh City in the central region of the of Saudi Arabia namely power plant #7(PP7) and power plant #8 (PP8) are selected for the present study [E C, KSA, 1998]. Performance data including number of operating units, type of units, model, manufacturer and output power of the units are summarized in table (1). Available heat energy in the exhaust gases of operating gas turbine units in the two power plants are then calculated

Table (1) Performance Data for Unit in The Selected Power Plants [E C, KSA, 1998]

	Units					**Exhaust Gases**	
Power Plant	**No.**	**Manuf.**	**Model**	**ISO MW**	**Site MW**	**10^6 Ton/year**	**^oC**
PP7	12	GE	7001E	63.5	47.5	86.1	548
	10	GE	7001EA	77.5	57.5		
PP8	20	ABB	11D5	72.5	50	115.7	556
	10	GE	7001EA	77.5	57.5		

The gas turbines units considered in the present study are all of the (fram 7) type supplied by different manufacturers. Site ratings are in the range of 47-58 MW and the average thermal efficiencies of the different makes is 24.6%. Based on the plant data of total electrical output per year, type of fuel used, fuel consumption, flow rate of the exhaust gases, temperature of the exhaust gases and unit thermal efficiency, available heat energy in the exhaust gases is calculated using the following equations and assumptions:

Available heat in flow gases is expressed as [Cengel &Boles, 1989]:

$$Qg = m\ C_p\ (T_g - T_{dp}) \quad (1)$$

Where:

Q_g =heat rate in (kW)

m = mass flow rate(kg/s)

C_p = specific heat (kJ/kg ^{0}C), given in [Cengel &Boles, 1989]

T_{dp} = dew point temperature from [ASHRAE H B, 1985]

Assuming fuel calorific value for crude oil =43MJ/kg and for Diesel = 45 MJ/kg. The heat recovery from each gas turbine unit could amount to 58-62 % of the normal plant fuel consumption. This percetage if converted to heat means that there is waste heat as much as twice the output power,. i.e. 100-120 MW from each unit.The simplest and most economical way to recover and utilize this heat energy from the exhaust gases is to heat water at atmosheric pressure. Sinsible heat added to water is not as efficient as generating steam at high pressure, however it is less expensive, has simple technology, safe and could be applied to existing GT units. The output hot water at temperature< 100 ^{o}C will be used in driving a simple Absorption Refrigeration System (ARS).

2 ABSORPTION REFRIGERATION SYSTEM

A simple absorption system is illustrated in figure (1). The system consists of four basic components: an evaporator and absorber (which are located on the low-pressure side of the system) and a generator and a condenser (which are located on the high-pressure side of the system). Two working fluids are employed a refrigerant and an absorbent

In order to obtain thermal performance for the absorption refrigeration cycle, enthalpy data must be available for the working substances at all crucial points of the cycle. The basic system of equations used for each component of the system is as follows:

Mass Balance:

$$\Sigma\ m_i = \Sigma\ m_o \quad (1.a)$$

Energy balance:

$$\Sigma\ m_i\ h_i = \Sigma\ m_o\ h_o + Q_g \quad (1.b)$$

Where:

m is the mass flow rate, kg/s

h is the enthalpy, kJ/kg, and

Q_g is the heat given in the generator at temperature T_g,

Subscripts i &o donates inlet and exit respectively

Top

The system coefficient of performance is defined as:

$$C.O.P = \frac{refrigeration\ rate\,(Q_e)}{rate\ of\ \ heat\ addition\ to\ generator\ (Q_g)} \qquad (2)$$

Where:

Q_e is the absorbed heat from the evaporator at temperature T_e,

Also, Q_c is the discharged heat in the condenser at temperature T_c,

and Q_a is the heat discharged in the absorber at temperature T_a .

2.1 Analysis of Water-Ammonia AbsorptionRefrigeration System:

Ammonia-water systems are widely used in domestic refrigerators and in commercial and industrial systems where evaporator temperature is maintained close to or below zero ^oC .

With reference to figure (1), pure high pressure NH_3vapor enters the condenser at point (1) and leaves at point (2) as saturated NH_3 liquid. It is further throttled from the high pressure (P_H) to low pressure (P_L) to enter into the evaporator at point (3). The saturated vapor then leaves the evaporator at point (4) where it is absorbed by the weak NH_3 solution coming from generator through throttling valve and becomes stronger.

For the analysis and design of Water-Ammonia refrigerant system, it is necessary to represent the different processes and conditions of water on a Concentration-Enthalpy (*C-h*) chart passing through different components of the system [Arora & Domkundwar, 1995]. Since input and output of the system are in the form of heat (enthalpy), the system performance can then be calculated once the enthalpies at different points are known. Mass balance and energy balance of each component of the system are then carried out.

The major disadvantage of Ammonia-water system is that the absorbent (water) is reasonably volatile, and the refrigerant (ammonia) leaving the generator will usually contain appreciable water vapor as it enters the evaporator[ASHRAE HB, 1985.]. This could raise the evaporator temperature and reduce the refrigeration effect. The ammonia-water system efficiency can be improved using an analyzer and rectifier to remove the water vapor from the mixture leaving the generator.

2.2 Analysis of Water-Lithium Bromide and Water-Multicomponent Salt Mixture (MCS) Systems

Water-Lithium Bromide and Water-Multicomponent Salt Mixture (MCS) systems are used extensively in air conditioning and other high temperature applications, but with water as the refrigerant, they are not suitable for use in any application where the evaporator temperature is below zero ^oC.

One of the principal advantages of the Water-Lithium Bromide or Water-Multicomponent Salt Mixture (MCS) systems is that the absorbent is nonvolatile, so that there is no absorbent mixed with the refrigerant "water" vapor leaving the generator, and consequently, no analyzer or rectifiers is required in the system.

With reference to figure (1), the water evaporates in the evaporator and lowering its temperature. The water vapor will be added to the strong salt solution (cool), which is sprayed in the absorber maintaining low pressure (vacuum) in the evaporator. The weak solution is then pumped to the generator, where heated. The generator water vapor is further passed to the condenser. In the heat exchanger, weak hot liquid is cooled giving the heat to cold strong liquid before going into the generator. It reduces the heating load in the generator and cooling load in the absorber and reduces the overall operating cost of the system

Temperature-pressure concentration charts for LiBr-H_2O & MCS- H_2O solutions are used to define the relationship at different solution concentration. The charts are applied to saturated conditions where the solution is in equilibrium with the water vapor. The computation of the mass flow rate incorporates material balance using applicable concentrations of (LiBr or MCS) in the solution. Knowledge of the solution concentration and temperature enable the determination of the enthalpies at different points of the cycle. The enthalpies of water liquid and vapor are found from steam tables.

Mass-Flow Balance of the system gives:

$$m_2 = m_1 - m_3 \qquad (4.a)$$

$$m_1 x_1 = m_2 x_2 \qquad (4.b)$$

$$m_3 = m_1 \frac{x_1}{x_3} \qquad \& \qquad m_2 = m_1 \frac{x_1}{x_2} \qquad (4.c)$$

where x is the solution mass fraction and 1 & 2 are the inlet and outlet conditions of the generator and 3 represent the water vapor.

By solving the two balance equations simultaneously we can determine the mass flow rate in the absorption cycle.

and the cycle *C.O.P.* is:

$$C.O.P = (Q_e/Q_g) \qquad (5)$$

Similarly, the mass and heat balances of the double-effect absorption system can be derived.

A computer program is prepared using Engineering Equation Solver EES and different digitized properties data of the three different pairs of salts [H_2O-NH_3, (MCS)-H_2 O and LiBr-H_2O]. The output of the program gives the performance of the three cycles.

3. RESULTS OF THE THEORETICAL ANALYSIS

A comparison between the performances of the three refrigerant systems considered in the present study is summarized. The different performances are presented as a function of the main controlling parameter which is the heat recovery load expressed here as the generator temperature (T_g). The range of the generator temperature is set between 60-85 ^oC which is the temperature of the subcooled heated water at atmospheric pressure. As expected, the cycle *C.O.P.* increases as the generator temperature (T_g) increases. This is according to the Carnot principles, however, the increase will differ from one refrigerant to the other. Water-MCS mixture has the highest *C.O.P.* followed by the Water-Lithiurn Bromide and then the Ammonia-Water. This is shown in figure (2), which also shows the effect of the condenser temperature (T_c) on the cycle *C.O.P.* Two different values for the condenser temperature are selected as 25 and 30 ^oC respectively. The figure shows that the cycle *C.O.P.* increases as the condenser temperature decreases from 30 ^oC to 25 ^oC due to the decrease of the condenser pressure (P_c) which in turn results in an increase of the refrigerant flow rate (m_3) and a decrease in the pressure difference between the high and low pressure sides of the cycle. The figure also indicates that, the *C.O.P.* always increases with the generator temperature irrespective of the condenser pressure.

Since the absorption cycle is mainly used in cooling the GT inlet air or air conditioning applications as stated before, the evaporator temperature ranges is set at 5-10 ^oC. To study the effect of evaporator temperature (T_e) on the cycle *C.O.P.*, two different values are selected as 5 and 10 ^oC respectively as shown in figure (3). It was found that the *C.O.P.* increases as the evaporator temperature does due to the increase of the evaporator pressure (P_e) resulting in a decrease of the pressure difference between the high and low pressure sides. Water-MCS system has the highest *C.O.P.* followed by the Water-Lithium Bromide and then the Ammonia-Water systems. This result is more pronounced at low generator temperature. The use of the water as a refrigerant in the MCS mixture is probably more suitable than Ammonia for the selected evaporator temperature range since Ammonia is best used in a cycle with an evaporator temperature close to or below zero ^oC .

The purpose of the present investigation is to couple the two technologies of waste heat recovery and absorption refrigeration system. The theoretical analysis for both the GT heat recovery systems and the absorption refrigeration cycle show that the suggested inexpensive heat recovery load would be in the form of hot water with an average temperature of 60–85 ^oC, which will be the operating range of the absorption cycle.

Fig.(4) illustrates the suggested operating range for the selected Water-MCS solution as a working fluid. The wide range of operating conditions is suitable for the absorption cooling systems associated with GT heat recovery system without any major crystallization problem.

EXPERIMENTAL

1. SELECTION OF H_2O-MCS MIXTURE FOR THE SYSTEM

Choice of the H_2O-MCS mixture as an experimental working fluid is based on the theoretical analysis given above, which indicates that the H_2O-MCS mixture can be used for a wide range of operating conditions with a C.O.P. higher than that of the Water-Lithium Bromide. Since the solution has low specific heat, the boiling temperature range of the mixture, at relatively low pressure, is (50 -80^{0}C). This range can be easily achieved from a simple water heater placed across the exhaust gases of the Gas Turbine. The wide range can also accommodate the variable heat load of the Gas Turbine. The investigated refrigerant-absorbent pair [H_2O-MCS mixture] consists of: the refrigerant which is the water (H_2O) and the absorbent which is the MCS mixture {1.2 mole % LiCl, 0.75 mole % $CaCl_2$ and 0.08 mole % $Zn(NO_3)_2$}.

One of the principal advantages of the H_2O-MCS system is that the absorbent is nonvolatile, so that there is no absorbent mixed with the refrigerant "water" vapor leaving the generator, and consequently, no analyzer or rectifiers are required in the system. In addition, the solution is non-corrosive in nature (pH about 6.7).

2 SPECIFICATION AND OPERATION OF THE EXPERIMENTAL SINGLE-STAGE SINGLE-EFFECT REFRIGERATION ABSORPTION SYSTEM

Experimental apparatus for a single-stage single-effect absorption refrigeration system utilizing Multi-component Salt Mixture (MCS) was designed and constructed at the thermal laboratory of Alexandria University [Shehata, 1997]. The apparatus was used for the purpose of testing the effects of simulated thermal loads on the performance of the selected ARS. Loads expected from an experimental solar pond were first examined [Shehata, 1997]. In the present work an extension of the parameters ranges is carried out simulating low temperature hot water loads generated by the gas turbine exhaust gases. Figure (5) illustrates the experimental setup of the absorption system. It consists of four basic components: an evaporator and absorber, which are located on the low-pressure side of the system, and a generator and condenser, which are located on the high-pressure side of the system and put in the same shell. The flow cycle of (water) is from the condenser to the evaporator to the absorber to the generator and back to the condenser, while the Multi-component Salt Mixture (MCS) passes from the absorber to the generator and back to the absorber.

<u>Design features of the system are</u>:

The shell containing the generator and condenser has the following specification:

Shell length = 30 cm Number of the generator tubes = 30

Number of the condenser tubes = 40

Top

The shell containing the evaporator and absorber has the following specification:

Shell length = 35 cm Number of the evaporator tubes = 25

Number of the absorber tubes = 35

All tubes are of Copper, tube length = 50 cm, tube O.D. = 12.5 mrn

<u>Thermal operating parameters of the experiment are</u>:

Maximum generator heat load is set at 2 kW, achieved with hot water at temperature range of (60 to 80^{o}C). An external electrical heating element is used for initial start. Temperature difference between inlet & outlet hot water is not to exceed 8 ^{o}C. Condenser cooling water temperature difference between inlet & outlet is not exceeding 5 ^{o}C. Evaporator chilled water temperature difference between inlet & outlet = 5 -10^{o}C.

With reference to figure (5), the operating sequence is to flow high-pressure condensate water from the condenser into the evaporator through an expansion device that reduces the pressure of the water to the low-pressure existing in the evaporator [Yen-Hsiung, 1982]. The water vaporizes in the evaporator by absorbing latent heat from the chilled water being cooled, and the resulting low-pressure vapor then passes into the absorber, where it is absorbed by, and goes into the solution with the MCS mixture. In the absorbing water vapor, the (MCS) solutions are diluted in the absorber and then re-concentrate it. This weak solution is then pumped to the generator, where it is heated by the hot water taken from a small storage tank simulating the deriving heat load where part of water is vaporizes and makes the strong (MCS) solution. This solution is again passed into the absorber. The generator water vapor is further passed into the condenser. Water is then passed into the evaporator to compensate for the evaporated water in the evaporator and completes the cycle.

3 FILLING AND PREPARATION OF THE EXPERIMENTAL SET-UP:

1- the system must pass a vacuum leak test to ensure that the system is air free.
2- the strong solution is then prepared with 60 % concentration ratio (weight percentage) and the weak solution with 20 % concentration ratio. The generator side is then filled with the strong solution and the absorber side with the weak solution.
3- the generator/condenser side (high pressure side) is evacuated to 4.27 kPa. Corresponding to minimum operating condenser temperature (30 ^{o}C).
4- the evaporator/absorber side (low pressure side) is evacuated to 1.23 kPa. Corresponding to minimum operating condenser temperature (10 ^{o}C),
5- the hot water storage tank is put into service by switching its electrical heater. The water temperature rise is preciously controlled to the specified load temperature (60–80 ^{o}C).
6- Adjustment of the return strong solution flow rate from the generator to the absorber is done.

Top

7- Adjustment of the condensate water flow rate from the condenser to the evaporator is done.

8- Adjustment of the hot water, cold water and chilled water flow rates is done.

9- the additional electrical heater is put off.

Estimation of measurement errors:

The relative errors of the measurement are obtained experimentally with a confidence level of 95% for the following parameters:

The different watercircuits are controlled through measuring the flow rate values, using rotary flow meters and control valves. The flow meters inaccuracy never exceeded ±2% according to the calibration curves obtained.

Input and output temperatures for the apparatus components are measured using accurate temperature instruments. Stationary thermocouples type (T) and a read-out electronic potentiometer is used to measure and record the temperature at different points of the apparatus as indicated in figure (5). Calibration of the temperature measuring equipment is performed under similar range of temperature as of the experiment according to the manufacturer recommendation. Errors are found to be limited to $\pm\ 0.5^0$ C.

A double U-tube manometers with the 0.25 class of accuracy are used for pressure measurements in the condenser and evaporator since these measurements are performed practically in a two-phase flow.

Enthalpy data are obtained from the appropriate charts and tables. Non-linear interpolation from the data source has an inherent error of ±3%.

Overall inaccuracy in the C.O.P. calculation is believed to be within ±5%.

Experimental range:

The following tables summarize the runs performed on the apparatus for the present study:

Table (2) Operating Conditions for the Experimental Apparatus

Run #	Heat Load kW	**Generator Side Heating Water**			**Evaporator Side Cooling Water**		
		Flow rate, l/s	Temp. in	Temp. out	Flow rate, l/s	Temp. in	Temp. out
A	**2.07**	0.062	$88\ ^0$C	80^0 C	0.068	$22\ ^0$C	$16\ ^0$C
B	**1.91**	0.051	$83\ ^0$C	$74\ ^0$C	0.064	$23\ ^0$C	$18\ ^0$C
C	**1.70**	0.058	$82\ ^0$C	$75\ ^0$C	0.063	$23\ ^0$C	$18\ ^0$C
D	**1.64**	0.056	$78\ ^0$C	$71\ ^0$C	0.061	$23\ ^0$C	$19.5\ ^0$C
E	**1.20**	0.048	$75\ ^0$C	$69\ ^0$C	0.058	$23.5\ ^0$C	$19\ ^0$C

In table (3), Readings of the evaporator and generator absolute pressures and corresponding temperatures according to run #B are given. The results are shown graphically at Figure (6).

Table (3) Recorded Readings Of Run # B: Evaporator Pressure is Held Constant and Generator Temperature is Variable

Readings		1	2	3	4	5	6	7	8
Evaporator	Pressure, kPa	117 ± 4							
	Temperature, ^{0}C	6.1	5.9	5.7	5.8	5.6	5.6	5.9	6.0
Generator	Pressure, kPa	406.3	411.1	416.2	421.1	423.5	426.0	427.2	429.9
	Temperature, ^{0}C	63.5	65.2	76.3	70.2	73.7	74.8	78.1	80.3

4 EXPERIMENTAL RESULTS AND DISCUSSION

The results of the thermal performance tests for the indoor single-stage single-effect absorption system are generally presented by plotting the system *C.O.P* vs. the generator temperature or the evaporator temperature. Other *C.O.P.* curves may be obtained as a function of the mass flow rate, which reflects the thermal load. Samples of experimental results are presented in figurer (6).

The system *C.O.P.* increases as the generator temperature increases. The theoretical calculation indicates that the *C.O.P.* approaches asymptotic value of 0.74 at 68 ^{o}C or higher, however, the experimental calculation over the same operating conditions showed a gradual increase up to 75. ^{o}C Fig.(6) shows the system *C.O.P* as a function of generator temperature while values of other operating parameters are kept constant. The data are compared with the theoretical model calculation. The experimental data show the same curve trend, however, the experimental results have lower values than that of the theoretical one due to the many thermal and hydraulic losses involved in the experiment. The same experimental apparatus was used in another study [Shehata, 1997] to evaluate the effect of changing the thermal load extracted from a solar pond on the performance of the absorption refrigeration system More experimental data on the performance of the system are given in.[Shehata, 1997].

CONCLUSION

The purpose of the present investigation was to investigate methods and means of utilizing the gas turbine waste heat in driving an absorption refrigeration system. According to the Gas Turbine waste heat analysis applied to PP7 &PP8 of (SEC-Center), each GT of frame 7 type could produce up to 50-60 MW of cooling effect The present study demonstrated that coupling the two technologies of recovery of waste heat to heat water and the operation of an absorption refrigeration system on that hot water could be a successful approach. The theoretical analysis for both the GT heat recovery system and the absorption refrigeration cycle showed that the suggested inexpensive heat recovery load would be in the form of hot water with an average temperature of 60 –85 ^{o}C, which will be the operating range of the absorption refrigeration cycle. Water-MCS system has the highest *C.O.P.* followed by the Water-Lithium Bromide and then the Ammonia-Water systems . The wide range of operating conditions is suitable for the absorption cooling systems associated with GT heat recovery system without any major crystallization problem.

A limited experiment was performed to obtain the operating range of a single-stage single-effect absorption refrigeration system utilizing Multi-component Salt Mixture (MCS) The experiment demonstrates clearly the possibility of building a simple and economical absorption refrigeration system utilizing (MCS) to use in association with heat recovery low temperature water heaters. The successful system gives a C.O.P. value up to 0.7 and accommodates driving temperatures between 60-75 ^{o}C. This proposed system could generate cooling chilled water at 4-6 ^{o}C, which can be utilized in different application [El Masry; O., 2002] in and around the Gas Turbine power plant. The full-integrated system deserves further study and economical assessment.

REFERENCES

1. Alizadeh; S., Bahar; F., and Geoola; F., 1979 "Design and Optimization of an Absorption Refrigeration System Operated By Solar Energy ", *Solar Energy*, 22, 149- 154.
2. Arora S. C.and Domkundwar S., 1995, A Course in Refrigeration and Air Conditioning, Dhanpat Rai & Sons.
3. ASHRAE Hand Book of Fundamentals, 1985.
4. Ayyash, S, 1980, "Power Saving In Solar Absorption Cooling Systems ", 3rd MICAES, Florida. U.S.A.
5. Boyen J. L, 1980, Thermal Energy Recovery, John Wiley &Sons Inc.
6. Cengel, Y. A. &Michael A.Boles, 1989, Thermodynamics, McGraw-Hill.
7. Electricity Corporation, 1998, "Long Term Electrification Plan for the Kingdom of Saudi Arabia", Report on Generation Planning (UPDATE), , Shawal 1418H, Riyadh, KSA.
8. El Masry; O., 2002, "Heat Recovery In Gas Turbine Power Plants", 6th Power Generation Symposium, SEC-Central, Riyadh, KSA.
9. Fernández-Searaa J.,Valesa A. and Vázqueza A., 1998, "Heat recovery system to power an on board NH_3-H_2O absorption refrigeration plant in trawler chiller fishing vessels" *Applied Thermal Engineering*, Volume 18-12, pp 1189-1205.
10. General Electric Company, 1987, "Gas Turbine & Heat Recovery Cycles Flexibility", (GE Report# 3421).
11. Goldstick Robert & Albert Thumann, 1983, The Waste Heat Recovery Hand book, The Fairmont Press.
12. Kaushik; S. S., Gadhi, S. M. B., and Kumari: R., 1980, "Modeling and Simulation Studies On Single/Double-Effect Absorption Cycle Using Water-Multicomponent Salt (MCS) Mixture", *Solar Energy*, Vol 49, pp 431-441.

Top

13. Mansoori; A., and Patel; V., 1979, "Thermodynamic Basis for the Choice of Working Fluids for Solar Absorption Cooling System", *Solar Energy*, Vol 22, pp 483 –491.
14. Nagpal G.R., 1980, "Power Plant Engineering", Romesh Chander Khanna, New Delhi.
15. Suri, R. K., and Ayyash, S., 1984, "Choice of Thermal Energy System For Solar Absorption Cooling", *Solar Energy* , Vol 32. pp 181-187,
16. Shehata, M. S., 1997, "Utilization of Solar Pond Extracted Heat in Refrigeration Absorption System", Ph. D. thesis, Alexandria University, Egypt.
17. Yen-Hsiung Kiang, Waste Energy Utilization Technology, 1982. Marcel Dekker.

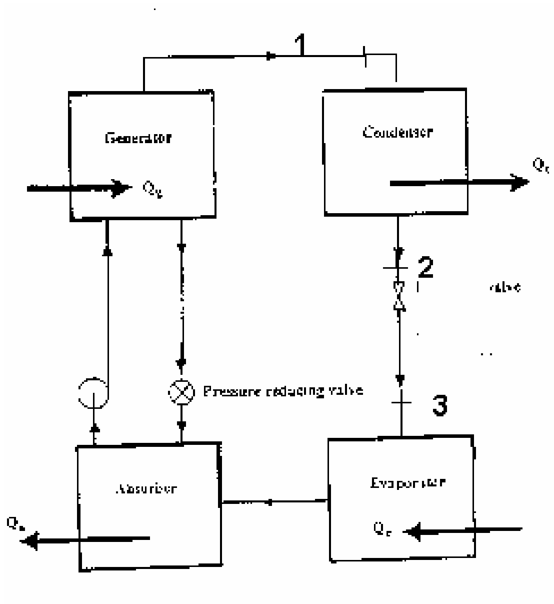

Figure (1) Simple Diagram for an Absorption Refrigeration System

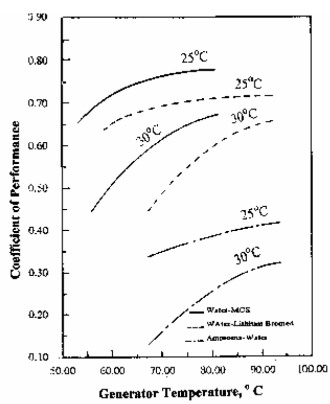

Figure (2) A Comparison between the Performances of the three refrigerant systems [H_20-NH_3, (MCS)-H_2O and L_iB_r-H_2O].as a Function of the Condenser Temperature

Top

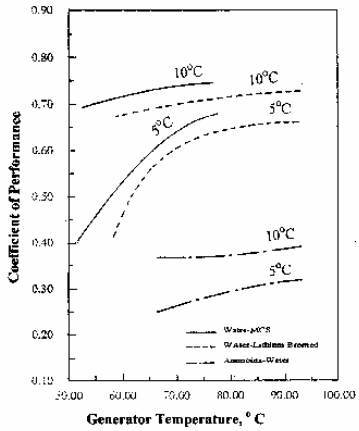

Figure (3) A Comparison between the Performances of the three refrigerant systems [H_20-NH_3, (MCS)-H_2O and L_iB_r-H_2O].as a Function of the Evaporator Temperature.

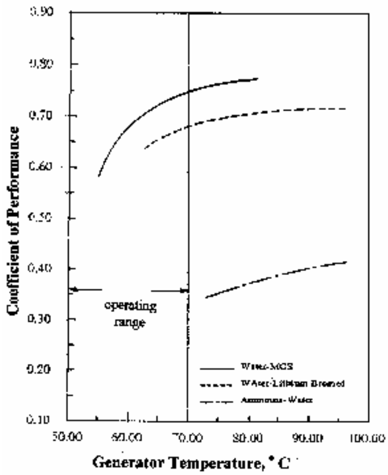

Figure (4) Suggested Operating Range for the Selected Water-MCS Solution as a Working Fluid

Top

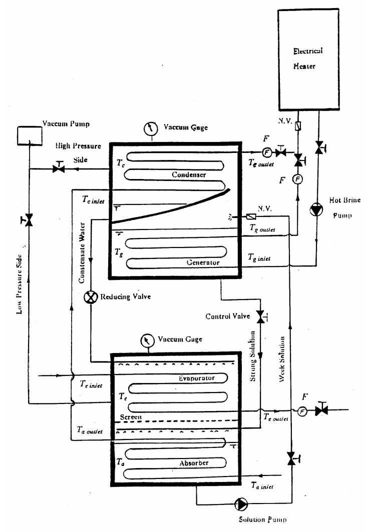

Figure (5) Experimental Setup of the Single-Stage Single-Effect Absorption System

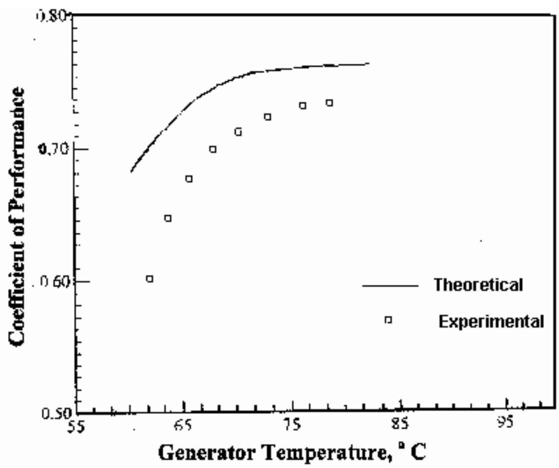

Figure (6) Experimental Data for the System C.O.P vs. the Generator Temperature