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Design and Installation of the First Geothermal Heat Pump in Iran

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ABSTRACT

Geothermal Heat pump is a new technology in A/C Systems in the world. The aim of this paper is to describe the procedure of design and installation of a geothermal heat pump in Iran. A 18000 BTU/hr air-to-air heat pump has been changed and designed to a geothermal heat pump system for the first time in Iran. Air-to-air condenser has been replaced by a Tube-in-Tube heat exchanger, the assembled system under the ARI-325 standard of the national energy lab of Iran has been tested and the results compared with the original system. The results show notable savings in energy consumption.

Local climate conditions and soil properties of Tabriz, located at the north-west of Iran, were used to design the geothermal coil. The coil was connected to the heat pump and average coefficient of performance (COP) of 2.57 has been recorded.

1. INTRODUCTION

Ground-source heat pumps (GSHPs), or geothermal heat pumps (GHPs) are considered as modern and suitable air conditioning and water heating systems. They provide comfort, leading to significant reduction of electrical energy use and demand, have very low levels of maintenance requirements and have lowest damage to environment (Kavanaugh, 1992, 1998).

Geothermal heat pumps are superior to other systems like air source heat pumps. The importance of these advantages are: a) lesser energy consumption during operation b) use of earth temperature as a more stable energy source (compared to air) c) low cost of design and maintenance d) no need to supplement heat during extremely low environment temperature e) less amount of refrigerant needed, (f) the place where temperature is to be controlled can be apart from where the system is installed. On the other hand, the main disadvantage of these systems which cause them not to be as widely used as one expects is the higher initial capital cost, being about 30-50% more expensive than air source units. This is due to the extra expenses and effort to install and bury heat exchangers in the soil or providing a well as an energy source or sink which has different costs in different places. In Iran, and at the place in which the ground coil installed for example, the cost for providing vertical wells is very much higher than horizontal one. However, once installed, the annual cost is less over the life of the system, resulting in energy and economical savings (Lund, 2001).

Many investigators have reviewed the world-wide application of direct utilization of geothermal energy. According to some of them, GHPs have had the largest growth since 1995 and about 9.7% annually. Most of this growth occurred in United States and Europe, though interest is developing in other countries. The installed capacity is more than 6900 MWt, and the annuall energy use is about 23287 TJ/yr at the beginning of 2000 in 27 countries (Lund, 2001).

It is estimated that the actual number of installed units is around 500,000, while the equivalent number of 12 KW units installed is over 570,000. The data however are incomplete (Lund, 2001).

There are numerous studies on different aspects of this industry regarding their standards, installation procedure, handbooks and case studies for GSHP systems. Besides these, the investigations and studies on these systems are extremely limited in Iran, though like its other neighbors, the temperature of the soil is stable in most areas and suitable for GSHP applications.

In developed countries, there are various organizations involved with GSHP industry such as utility companies, electric co-operatives, contractors, design engineers and hydro geological consultants, manufacturers of equipment, universities, research organizations and the government. The utilization of GHPs in residential buildings has not yet been done before in Iran, although they have been in use for years in developed countries. There is no Iranian GHPs manufacturer yet and neither any installation to date.

This study reports geothermal heat pump designed and installed for the fist time in Iran at Tabriz. There have been very limited studies about GSHP systems in some universities which were theoretical, but no practical effort has been done.

As the authors of this paper know, there is no standard issued on geothermal heat pumps by the Iranian Standard Institution.

The study reported here includes the performance evaluation of a horizontal GHP. This project is proposed by Tabriz Engineering Research Center and supported by the Ministry of Energy and Department of Renewable Energies.

In the next section, theoretical and experimental works will be described. The various tests of system performance in heating mode have been carried out. The results obtained from practical tests were in good agreement with GSHP's standard results.

2. SYSTEM DESCRIPTION

The Ground-Source Heat Pump has three main parts, the first one is heat pump circuit, second is ground coil heat exchanger and the third one is the interface heat exchanger which in this case is a tube-in-tube coaxial heat exchanger. The ground coil of heat pump was installed in about 500 m^2

area through a horizontally series system and a trench with depth of about 1.70 m on average. The tubes were 1½ inch polyethylene and has a strength capacity of about 1.5 ton with R 22 as refrigerant, the system is connected to a residential unit. The total length of pipes is about 280 meters and all connections are welded. For the design and installation of system, manuals' procedure and standards of IGSHPA (International Ground Source Heat Pump Association) were followed. Heating tests of the system have been done during June, 2004. An irrigation piping system was designed and installed above the pipe in the trench with a flow rate of 800-1000 lit/hr and is with nearly 60 sprinklers to increase moisture of soil around the pipes. When heat is absorbed or rejected from/to the earth, thermal conductivity of surrounded soil decreases that in turn decreases the performance of system.

A picture of the heat pump manufactured in the research center is shown in Fig.1. All of the important elements of the system have been labeled in the figure. The base of the heat pump systems is the same for an ideal refrigeration compression cycle. There is a compressor, evaporator, condenser and expansion valve. The elements of cycle, are connected to each other by copper tubes in a closed loop. The pressure of the refrigerant increases in the compressor and this high temperature, high pressure fluid enters the condenser and dissipates heat at a constant pressure.

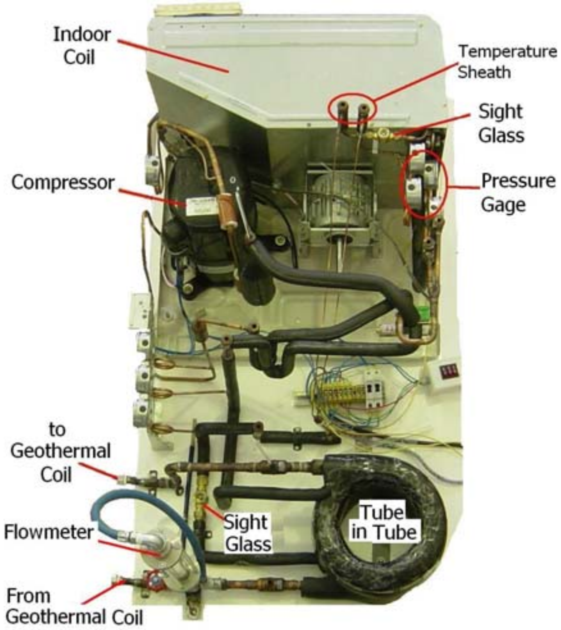

Figure 1: Designed heat pump cycle.

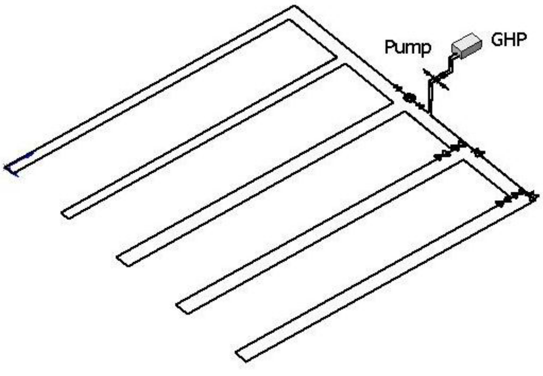

Figure 2: A schematic view of ground heat exchanger and its arrangement.

Figure 3: Installed circulator pump.

Figure 4: Arrangement of the polyethylene tubes in a trench of loop.

Leaving the condenser, the refrigerant enters an expansion valve where total pressure drops to evaporator pressure to facilitate evaporating and heat gaining of the refrigerant.

The Tube-in-Tube heat exchanger consists of two coaxial tubes. Water flows within the inner tube and refrigerant passes through the annulus between the two tubes. With connection of ground coil and tube-in-tube heat exchanger, heat transfer will be possible. Refrigerant will absorb heat from earth through the ground coil. In the ground coil, water and antifreeze circulate continuously during the working period and provide the stable temperature difference for heat transfer. To avoid freezing the water under the working condition and during the winter, a 10% ethylene glycol mixture by weight is prepared. Fig.3 shows placement of circulating pump and Fig. 4 depicts the arrangement of the polyethylene tubes in a trench of loop at a depth of about 1.7 meter.

3. WEATHER DATA

The "classical bin method " was used for calculating the average heat gained and lost per hour because this method is satisfactory for residential buildings where a linear load profile exists and only one shift is considered (Bose, 1985). Table 1 shows the climate conditions for Tabriz city. The bin method data is outlined in Table 2. These data are from climatologic station of Tabriz.

Table 1: Climatic data and relative humidity for Tabriz at different months of year

Month	Average ambient temperature (°C)	Maximum ambient temperature (°C)	Minimum ambient temperature (°C)	Relative humidity (%)
January	0.0	10.0	-10.0	71.8
February	1.9	12.8	-9.0	62.6
March	10.8	22.2	-0.7	51.2
April	15.0	25.4	4.6	45.5
May	19.5	29.6	9.4	42.3
June	23.4	34.6	12.2	31.6
July	25.9	36.8	15.0	35.2
August	27.9	37.8	18.0	31.6
September	21.9	31.6	12.2	32.5
October	14.7	26.0	3.4	46.4
November	6.4	19.6	-6.8	66.6
December	3.2	13.2	-6.9	69.1

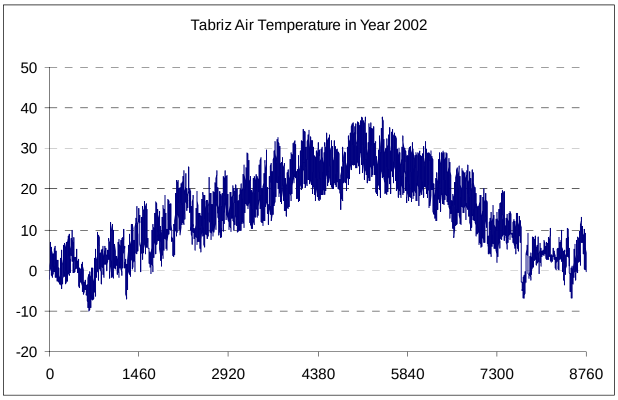

Figure 5: Hourly air temperature swing.

As Table 2 shows, required energy for heating has its maximum value at January, while for cooling mode, the most energy intensive month is July. Bin method has been used in both heating and cooling mode of heat pump design.

Table 2: Bin weather data for city of Tabriz

Range	Temperature bin °C
-15 To -12	0
-12 To -9	7
-9 To -6	77
-6 To -3	193
-3 To -0	476
0 To 3	771
3 To 6	857
6 To 9	749
9 To 12	708
12 To 15	743
15 To 18	750
18 To 21	878
21 To 24	859
24 To 27	649
27 To 30	572
30 To 33	297
33 To 36	138
36 To 39	36
39 To 42	0

4. SOIL CHARACTERISTICS

In GSHP applications, deposition or extraction of thermal energy from the ground is accomplished by using a ground heat exchanger (GHE). The transfer of heat between the GHE and adjoining soil is primarily by heat conduction and to a certain degree by moisture migration. Therefore it depends strongly on the soil type, temperature and moisture gradients (Hepbasli, 2002).

The soil thermal diffusivity (α_s) is a defined property and is the ratio of the thermal conductivity (k_s) and the heat capacity ($\rho_s C_s$). Therefore, the three soil properties, k_s ρ_s , C_s must be known or estimated to predict the thermal behavior of GHEs. Obtaining accurate values for the thermal properties of the soil require detailed site survey and knowledge of contents of the soil as well as its moisture amount. In Fig. 6, annual variations of soil temperature in depth from 5 cm to 100 cm of ground surface are shown. The ground temperature in a suitable depth is independent from ambient temperature but is a strong function of soil type.

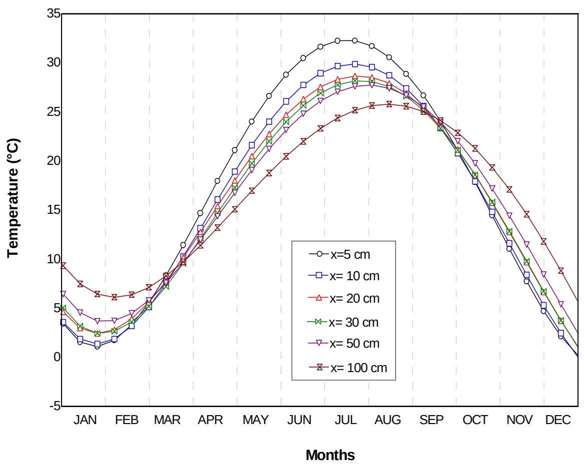

Figure 6: Variation of annual soil temperatures at several depth for Tabriz, Azerbaijan

On this study, analytical equations have been used for determining thermal properties, density and moisture content of soil. Thermal conductivity coefficient of soil, k_s (W/mk) is calculated from the following equations (Hepbasli , 2002):

$$k_s = 0.14423(0.9\log\omega - 0.2)10^{0.000624\rho_{sd}} \quad (1)$$

For silt and clay soils

$$k_s = 0.14423(0.7\log\omega + 0.4)10^{0.000624\rho_{sd}} \quad (2)$$

For sand soils

In which ω is moisture content with weight percentage and ρ_{sd} is density with dimension of kg/m^3

Thermal diffusivity coefficient is very sensitive to moisture content. Since the thermal capacity of 4.18 kJ/kg·K is much greater than heat capacity of soil and rock 0.04-1.05 kJ/kg·K, for determining αs :

$$\alpha_s = \frac{k_s}{C_{psc}\rho_{sc}} \quad (3)$$

$$C_{PSC} = [\omega C_{PW} + (100-\omega)C_{Psd}]/100 \quad (4)$$

$$\rho_{SC} = [\omega\rho_{PW} + (100-\omega)\rho_{sd}]/100 \quad (5)$$

Where C_{PSC} is specific heat of modified soil ρ_{sc} is density of modified soil, C_{PW} specific heat of water, ρ_W is water density. Soil can include sand, clay and silt.

C_{PSD} is assumed to be 0.84 kJ/kg·K . Using equations 1 to 5, coefficient of thermal diffusivity at a depth of 2 meters was calculated to be 0.0026 m^2/hr. The other calculated values are shown in Table 3.

Table 3: Physical Properties of Soil

	Symbol	Unit	2 m in depth
Thermal conductivity	k_s	$Wm^{-1}K^{-1}$	1.082
Dry density	ρ_{sd}	kg m^{-3}	1382
Moisture	ω	% by weight	7.9
Dry specific heat	C_{psd}	kJ kg^{-1} K^{-1}	0.84
Corrected density	ρ_{sc}	kg m^{-3}	1352
Corrected specific heat	C_{psc}	kJ kg^{-1} K^{-1}	1.105
Thermal diffusivity	α_s	m^2 h^{-1}	0.0026

5. ANALYSIS METHOD

The performance of the water source heat pump unit can be checked by measuring the water flow rate and water or anti-freeze solution temperature change on the liquid side and the electric power input needed.

The amount of heat transfer in tube-in-tube heat exchanger is calculated by the following equation:

$$\dot{Q} = \dot{m}_w \cdot C_{pw} \cdot (t_{wo} - t_{wi}) \quad (6)$$

Where C_{pw} is the specific heat of water-antifreeze solution, $\dot{m}_w$ is the mass flow rate of water and $(t_{wo} - t_{wi})$ is the temperature difference between the outlet and inlet of the GHE.

The power input to the compressor, $\dot{W}_C$ and the circulating pump $\dot{W}_P$ and required power of fan $\dot{W}_f$ are calculated by the following equations respectively:

$$\dot{W}_C = I_C \cdot V_C \cdot Cos\varphi \quad (7)$$

$$\dot{W}_P = I_P \cdot V_P \cdot Cos\varphi \quad (8)$$

$$\dot{W}_f = I_f \cdot V_f \cdot Cos\varphi \quad (9)$$

The coefficient of performance is defined as:

$$COP = \frac{\dot{Q}_{cond}}{\dot{W}_C + \dot{W}_P + \dot{W}_f} \quad (10)$$

Where $\dot{Q}_{cond}$ obtains by using the following equation:

$$\dot{Q}_{cond} = \rho_a \cdot \dot{V}_a \cdot C_{pa} \cdot (t_{ao} - t_{ai}) \quad (11)$$

6. MEASURMENT METHODS

The following data can be recorded:

(a) Mass flow rate of water using a rotameter

(b) Temperature of water entering and leaving the geothermal coil using thermometer

(c) Condenser and evaporator pressures using bourdon type manometers

(d) Outdoor and indoor air temperature using thermometers

(e) Electrical power input to compressor and circulating pump using a wattmeter

(f) Dry bulb temperature of air at the inlet and outlet of indoor coil unit using thermometer

Figure 7 illustrates test scheme of the system and all points for which data acquisition have been done. In this system, pressure gauges were installed for pressure measurement purposes. Cases for mounting thermometer on the needed points are also shown in the figure.

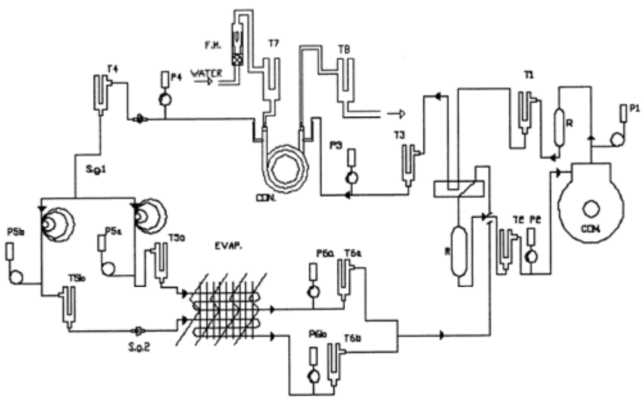

Figure 7: Data Acquisition Points of the System

Sample data from heating test of system are presented in Table 4. In this table, heat capacity and coefficient of performance of the system have been calculated using the above mentioned equations.

Table 4: calculated and measured parameters of experimental performance of system

Item	Value	Unit
Measured Parameter		
Evaporation temperature	7.2	°C
Condensation temperature	54.5	°C
Evaporation pressure	625.4	KPa
Condensation pressure	2159	KPa
Temperature of water at tube-in-tube inlet	11	°C
Temperature of water at tube-in-tube outlet	7.6	°C
Volumetric flow rate of water	19	Lit/m
Air temperature at condenser inlet	25.1	°C
Air temperature at condenser outlet	47.7	°C
Volumetric flow rate of air	344	CFM
Calculated parameters		
Power input to compressor	1770	Watt
Power input to circulator pump	405	Watt
Power input to fan	175	Watt
Total Power consumption	2350	Watt
Heating capacity	21424	Btu/h
Heating COP of heat pump	2.672	COP

7. RESULTS AND CONCLUSION

GHP systems are a promising new energy technology that has shown an increase in usage over the past ten years, although there is never any previous work in Iran and some other countries.

Fig. 8 & 9 depict the results of a work cycle in T-S and P-h diagrams respectively. These diagrams show a good agreement between refrigeration cycle and the current system. The refrigerant in the outlet side of condenser is completely in a liquid state (as can be observed with sight glass) while at the inlet of compressor, it is completely vapor.

Using measured data and calculating the thermodynamic conditions of cycle, power input to circulating pump is calculated to be 405 Watts. With regard to Table 5, it can be seen that there was an improper selection of circulating pump and if we chose a proper pump with a grade of Good, pump power would be 125.6 Watts and coefficient of performance will be 10.35. The average coefficient of performance of system is 8.768 (see Figs. 10, 11 and 12). So coefficient of performance can be improved by 18 % if pump is improved. Since the fan was selected for both indoor and outdoor heat exchangers of system, therefore after removing the outdoor coil, we can select a smaller fan thereby reducing power input to fan by 40 %. With this improvement in the system, coefficient of performance can be improved by 7 %.

Although Fig. 13 shows acceptable but lower agreement between coefficient of performance of the system and ARI-325 standard, choosing a compressor with higher EER, will lead to higher performance.

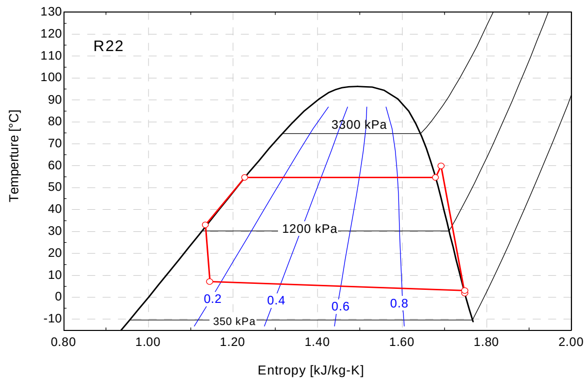

Figure 8: Measured Points of GHP cycle in the T-S diagram

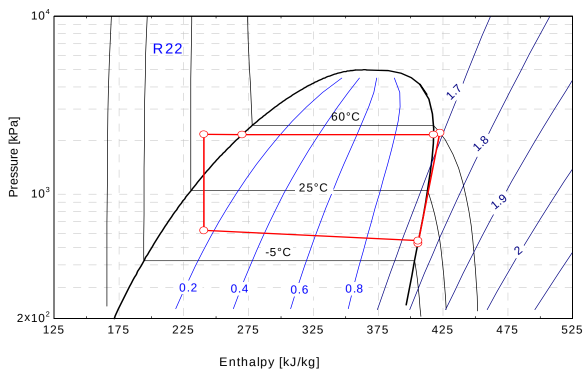

Figure 9: Measured Points of GHP cycle in the P-h diagram

Table 5: Benchmarks for GSHP system pumping efficiency required pump power (Kavanaugh 1997)

Performance		Watts Input	
Grade	Efficiency	Per KW	Per ton
A:	Efficient	14 or Less	50 or
B: Good	Acceptable systems	14-21	50-75
C: Mediate		21-28	75-100
D: Poor	Inefficient systems	28-24	100-150
E: Bad		Greater	Greater

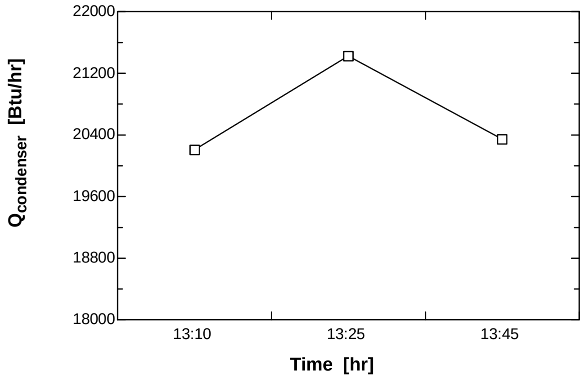

Figure 10: The hourly variation of heating capacity

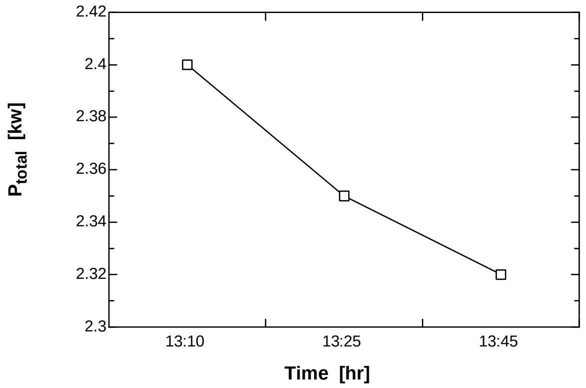

Figure 11: The hourly variation of total power consumption

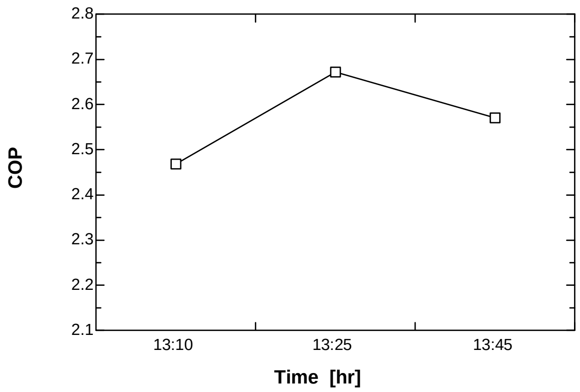

Figure 12: The hourly variation of coefficient of performance (COP)

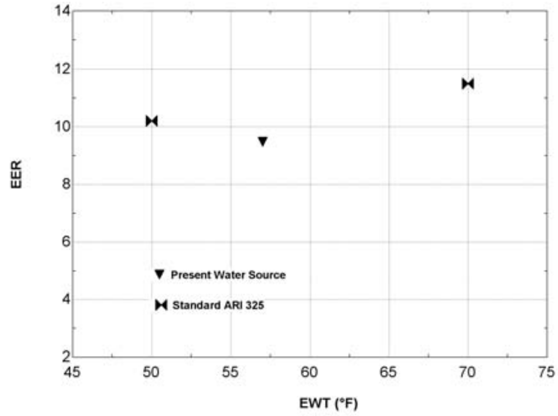

Figure 13: Comparison of coefficient of system according to ARI-325 standard

8. ACKNOWLEDGEMENTS

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