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HEAT PUMP AND ENERGY RECOVERY IN AIR CONDITIONING SYSTEM

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Abstract

A theoretical analysis, and experimental work were carried out to study factors that affect the behaviour of an air-to-air heat pump.

The performance of the heat pump was studied under different outside air temperatures for cooling and heating modes. The analysis indicated the sensitive variation of performance of the air-to-air heat pump with the outside air temperature.

Nomenclature

t	temperature	^{o}C
W	Power	kW
Q	Heat transfer rate	kW
i	Enthalpy	kJ/kg
U	velocity	m/s
m	Mass flow rate	kg/s
A	Area	m^2
h	Heat transfer coefficient	W/m^2K
ρ	Density	kg/m^3
c_p	specific heat of air	$kJ/kg.^{o}C$
V	Voltage	volt
I	current	ampere
N	Number of rows	
R	Ratio	
cos ψ	Motor power factor	

SUBSCRIPTS

a	Outside air
e	Evaporation
c	Condensation
s	Surface
r	Refrigerant
d.a	Dew point of outside air
dp	Dew point of air leaving evaporator
com	compressor
cm	cooling mode
hm	meating mode
f	fan
Δ	difference

1. Introduction

The current energy crisis has prompted technologists to devise new energy saving ways, that help to reduce the consumption of energy. The area of residential heating has received a great attention with the development of heat pump, where the heat of condensation is utilized in heating purposes (1,2) instead of rejecting it to the atmosphere.

The most common and commercially available heat pump today, is the air-to-air heat pump (3,4) that uses the energy stored in the ambient air. From this point the performance of an air-to-air heat pump is greatly affected by the outside air temperature.

This paper presents an experimental study of the effect of the outside air temperature on the behaviour of the air-to-air heat pump.

2. Case Study

The flow diagram of the heat pump, employing refrigerant -12 is shown in Fig.1,while the thrmal operating condiditions are listed in table 1

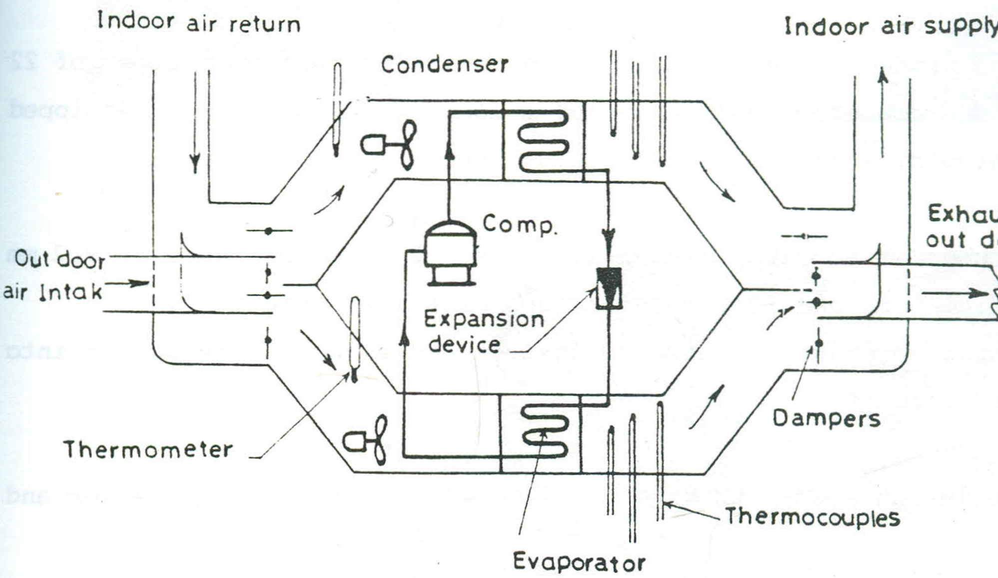

Figure 1 - Set-up of air-to-air heat pump for heating mode.

Table 1. Thermal Conditions

t_a	-	Outside air temperature	10 - 35 ^{o}C
t_e	-	Evaporation temperature	7 - 15 ^{o}C
t_c	-	Condensation temperature	33 - 46 ^{o}C

2.1 Experimental Apparatus

The experimental apparatus considered in this study is a vapor compression system, operating with refrigerant - 12.

The system is comparable to the most commercial air-to-air heat pump. As shown in Fig. 2. and Fig. 3, the unit has the following main components:

Hermetically sealed compressor Danfoss type Sc 15 BX, with standard refrigeration capacity of 680 Watt.

Air cooled finned tubes condenser, consisting of 36 copper tubes of 22 mm outside diameter and of 0.265 m long. The condenser is equipped with 66 aluminium fins of 0.2 mm thickness.

Finned tubes evaporator, consisting of 27 copper tubes of 9.2 mm outside diameter, and 90 aluminium fins of 0.2 mm thickness.
Thermostatic expansion valve to control the flow of refrigerant into the evaporator.

The unit is equipped with electrical panel, refrigerant receiver and filter dryer.

The low and high pressures were measured by pressure gauges located at suction and discharge lines of compressor. The velocities of air passing through condenser and evaporator, were measured by a velometer.

The air temperatures at inlet and outlet of condenser and evaporator,

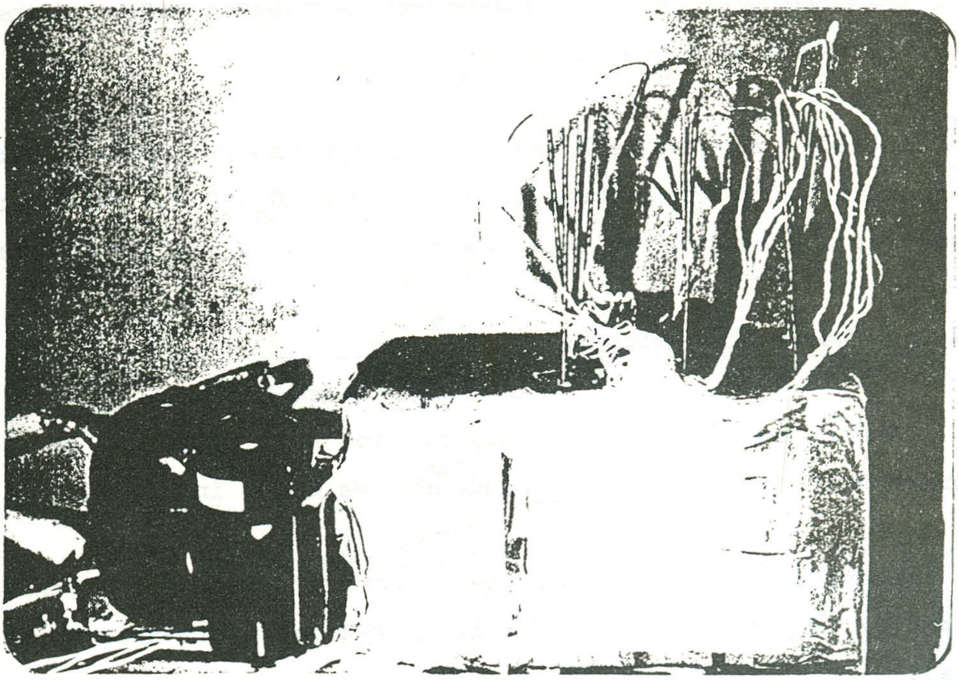

Figure 2 - Heat recovery condenser with suspending thermocouples for measuring outlet air temperatures.

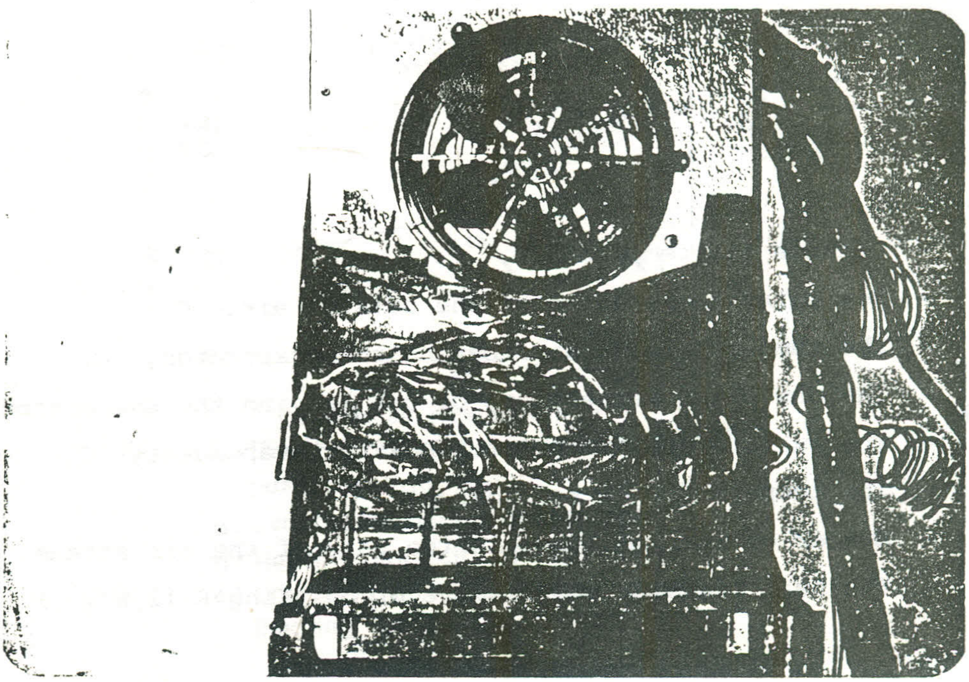

Figure 3 - Evaporator with suspending thermocouples for measuring outlet air temperatures.

were measured by shielded copper-constant thermocouples, suspending at various levels of connected ducts.

The outside air temperature was measured by means of two Hg thermometers. The input power to compressor (W_{com}) and fans motors of condenser and evaporator (W_f), were measured by volt-ampere meter.

2.2 The Theoretical Analysis

The theoretical analysis of the studied heat pump, had been done for evaporator, comperessor, and condenser as shown in Fig. 4.

The theoretical analysis of the evaporator, was done by the humidity method/5/ for the climate condition of Zagazig city, is shown in Fig. (4-a).

The heat transfer rate of the evaporator is given as follows:

$$Q_e = \frac{h_e}{R_s} \cdot A_d \cdot N\,(t_s - t_r) \qquad \text{kW} \qquad (1)$$

where :

t_a - Outside air temperature, ^{o}C.

t_d - Dew point temperature of the outside air, ^{o}C

t_{da} - The temperature of air leaving the evaporator, ^{o}C.

d_{dp} - The dew point temperature of air leaving the evaporator, ^{o}C

t_s - The surface coil temperature of the evaporator, ^{o}C.

t_r - The temperature of refrigerant, ^{o}C

N - Number of rows of tubes in the path of the air stream.

h_e - The heat transfer coefficient of boilling R-12, $W/m^2.K$

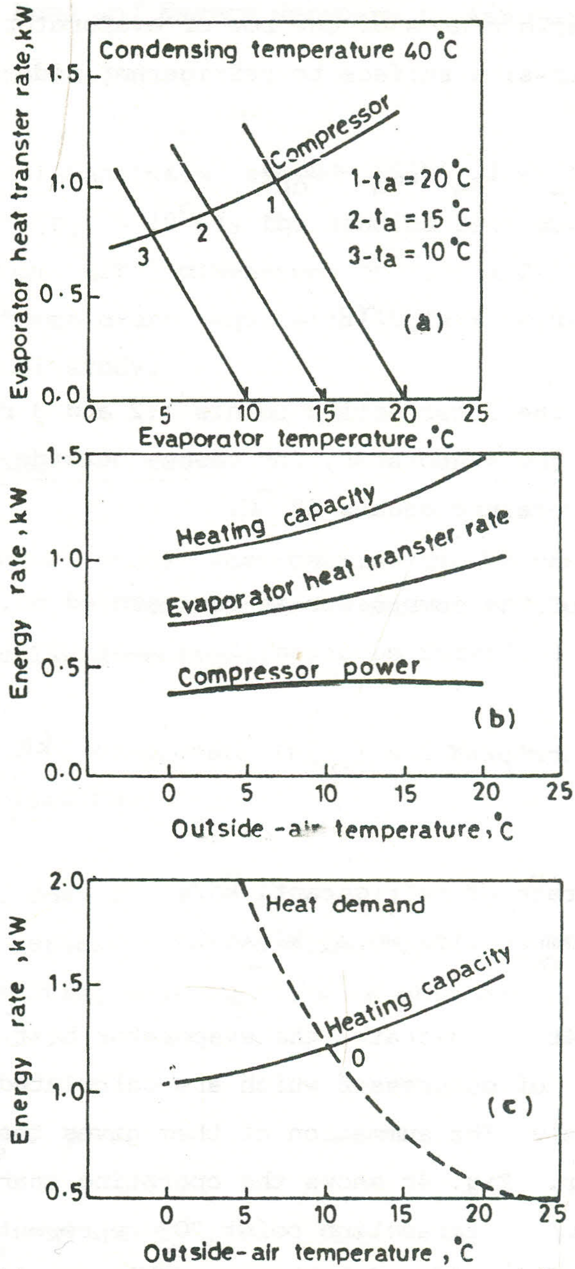

Figure 4 - The theoretical performance of heat pump (heating mode)

a) Evaporator heat transfer rate

b) Heating capacity of heat-pump

c) Operating condition of heat pump (heating mode)

A_d Air-side surface area of one row of evaporator tubes, m^2

R_s - Ratio of air-side surface to refrigerant side surface.

$$R = (t_a - t_{da})/(t_a - t_{dp})$$

and

$$t_s = (R.t_d - t_a)/(R-1)$$

From Fig. (4a) the intersection points 1.2 and 3 represent the heat transfer rates of the evaporator, for vaious outside air temperatures, at condensing temperature equals 40 ^{o}C.

The input power of the compressor is represented on, Fig 4 b, and was obtained from the following equation.

$$W_{com} = m_r . w \qquad kW \qquad (2)$$

where:

m_r - Mass flow rate of refrigerant, kg/s

w - specific compression work, kJ/kg

Figures 4a and 4b illustrate the evaporator heat transfer rate and the input power of compressor which are calculated by equations (1) and (2) respectively. The summation of them gives the heating capacity of the condenser. Fig. 4c shows the operating characteristic of the studied heat pump. Intersection point "O" represents the theoretical operating point of considered heat pump. This point corresponds to an outside air temperature equals 10 ^{o}C.

At an outside air temperature less than 10^{o}C, a supplementary source of heat will be needed.

The theoretical analysis showed that, for the worst condition of Zagazig City ($t_a = 10^{o}C$), the studied heat pump is suitable for heating a room with dimensions of 2.23 x 2.23x 3 m, maintaining an inside air temperature equals $25^{o}C$. This result was considered in the experimental study.

2.3 The Experimental Study

The experimental study was carried out by means of the arrangement shown on Fig. 1, for heating mode. For cooling mode the dampers, will have perpendicular direction.

The input power of compressor (W_{com}) and fans motors (W_f), were obtained as following:

$$W = V.\ I.\ \cos\ \phi\ , \qquad kW \tag{3}$$

The cooling capacity for cooling mode (Q_{cm}), and the heating capacity for heating mode (Q_{hm}), were obtained by equations (4) and (5) respectively.

$$Q_{cm} = m_e\ .\ \Delta\ i\ , \qquad kW \tag{4}$$

$$Q_{hm} = m_c .\ Cp\ .\ \Delta\ t_c\ , \qquad kW \tag{5}$$

where:

m_e, m_c - mass flow rate of air passing through, evaporator and condenser respectively, kg/s

$m = U.A.\rho$ kg/s

U - mean velocity of air through duct, m/s

A - cross-sectional area of duct, m^2

ρ - density of air, kg/m^3

Δi - specific enthalpy difference of air passing through evaporator, kj/kg.

Δt_c - the inlet-outlet temperature difference of air passing through the condenser, ^oC.

C_p - specific heat of air , kJ/(kg. ^oC)

The experimental data of Q_{cm} and Q_{hm} are plotted on Fig. 5. The heating and cooling demand were determined for the considered room (2.23x2.23x3m) and comfort condition (6)

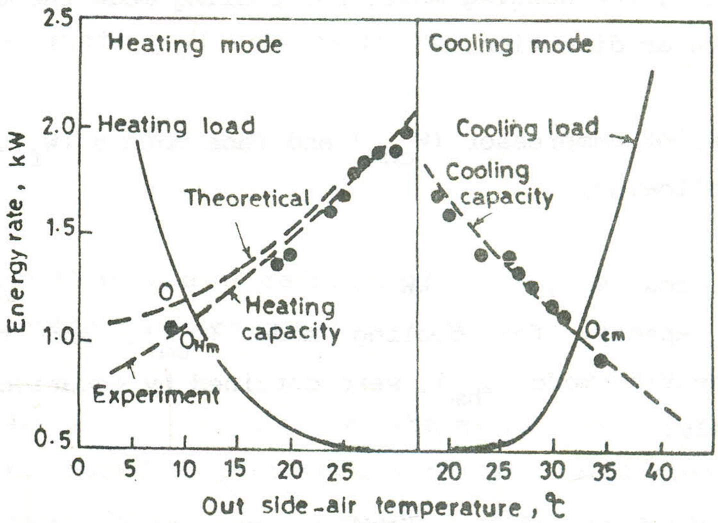

Figure 5. Air-to-air heat pump operating performance

3. Analysis and Discussion

From Fig. 5, it is clear that:

For heating mode the intersection point of heating capacity and heating demand is represented by point (Q_{Hm}), that corresponds to an outside temperature equals 12°C. This point is the optimum actual operating point, which indicates that, the heat pump ensures the heating of the considered room at comfort condition. If the ouside air temperature drops below 12°C, a supplementary source of heat must be added [7,8]. The experimental heating capacity of the studied heat pump, is smaller than the theoretical heating capacity by 10% at outside air temperature equals 10°C. This deviation increases as the outside air temperature drops, and decreases as the outside air temperature rises.

For cooling mode, the optimum operating point "Q_{cm}" corresponds to an outside air temperature equals 32.5°C. For higher temperature, the cooling load will increase and consequently greater machine must be considered.

The coefficient of performance COP, and the gross coefficient "GCOP", are given by:

$$COP = Q/W_{com} \tag{6}$$

$$GCOP = Q/(W_{com} + W_f) \tag{7}$$

where

Q	-	The cooling or heating capacity	kW
W_{com}	-	The input power of compressor	kW
W_f	-	The input power of fans motor	kW

Fig. 6 illustrates the behaviour of the coefficients COP and GCOP with the outside air temperature, from which it is clear that:

For heating mode, the curves show steep rise in the coefficients COP and GCOP with rising of the outside air temperature.

This is due to the decreasing temperature difference between the heat source and the heating fluid.

For cooling mode, the curves indicate a steep drop in COP_{cm} and $GCOP_{cm}$ with rising outside air temperature. This is due to the increase of condensation-evaporation temperature difference and consequently the delivered power to compressor.

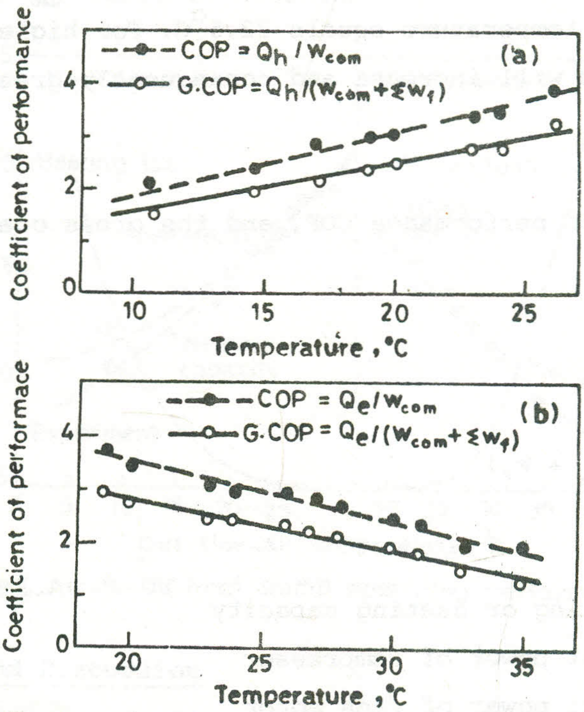

Figure 6_ (COP) and (G.COP) for heat pump as a function of the outside air temperature.
a)-Heating mode. b)-Cooling mode

4. Conclusion

The balance point of air heat pump influences the energy consumption. For heating mode, the additional heating is only switched in below this point. For cooling mode additional cooling capacity should be considered, above the corresponding balance point. The coefficients of performance COP and GCOP are also very dependent on operating condition, so the energy consumption must be calculated for each outside air temperature and relevant operating condition.

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