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Performance of a Small Window Type Air Conditioner using a Thermostatic Expansion Valve and a Capillary Tube

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ABSTRACT

Electrical energy conservation is an important element of energy policy. Energy conservation reduces the energy consumption and energy demand per capita, thus offsets some of the growth in energy supply needed to keep up with population growth. Reduction of energy consumption is a major concern in the vapor compression refrigeration system especially in the area with hot weather conditions where window-air-conditioners are usually used to cool homes. This kind of weather condition has a major influence in the performance characteristics of air_conditioning units. In most of the cases, electrical power consumption increases considerably in such weather condition. Though some research has been focused on optimizing medium to large capacity refrigeration systems, considering both materials cost and operating costs, not much study has yet been done for a small system like the window air conditioner to find the optimum operating condition i.e. the minimum power requirement and hence the minimum cost of operation during typical summer days. Hence the present study includes an

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economic comparison between thermostatic expansion device and capillary tube expansion device as well as the optimum operating condition for a small window type air conditioner in hot climates.

Keywords: *Window air conditioner, Optimization, Power consumption, Cost of operation, Thermostatic Expansion valve and capillary tube.*

1 INTRODUCTION

The increase in use of air conditioning over the years has been identified as a contributor to increasing comfort and occupant's working efficiency. Most of the air conditioning units in service provide comfort air conditioning [1], the purpose of which is to provide comfortable conditions for people. Summer cooling systems have become a standard utility in large buildings throughout the world. Even in climates where summer temperatures are not high, large buildings may have to be cooled in order to remove the heat generated internally by people, lights, and other electrical equipments. In hot climates, summer cooling systems is essential for high efficiency of workers' performance and their comfort. Some form of central system usually serves large buildings. It may consist of one or more water-chilling plants and a water heater located in the plant room. The conditioned spaces are served by one or more air-supply and return systems with hot or chilled water supplied by pipes to heat exchangers (air handling/ fan-coil units) in the conditioned space.

During the past few decades, efforts have been devoted to performance of air conditioning systems. Available experimental information on the performance test of air conditioning systems has been focused on the industrial chillers and large air conditioners. An improved mathematical model based on static equilibrium of valves and some other components with reasonable simplifying assumptions had been made by Saraf G.R. and Dhar P.L. (1978) [2] to simulate the compressor performance. Later on some systematic and scientific procedure had been followed for optimum design of a liquid chilling plant to predict the performance of its components [3]. A new design for small window air conditioner was introduced and experimentally investigated by Ebrahim H. (2007) [4] with high commercialization potential where power consumption decreased by about 16% and the coefficient of performance increased by about 55%.

2 METHODOLOGY AND EXPERIMENTAL SETUP

For the present work, a window type air conditioner with rated cooling capacity of 2.0TR (7.034 kW) and using R-22 as the refrigerant has been used. A twin parallel capillary tube connecting the condenser and the evaporator headers have been used as the expansion device. Experiments were carried out with the capillary tubes; later a thermostatic expansion valve (TXV) was installed in conjunction with the capillary tubes so that either could be used independently as the expansion device. Experiments were also carried out with the TXV, keeping the capillary tube shut off. The important specifications of the air conditioner are given below:

Manufacturer:	General (assembled in Thailand)
Cooling Capacity:	24,000 Btu/h (2.0TR)
Refrigerant type:	R-22
Amount of refrigerant charged:	1.0kg (approx.)
Maximum refrigerant flow rate:	2.0 kg/min
Volts:	220V (Single phase)
Hertz:	50 Hz
Compressor type:	Hermetic–rotary type

Two ducts (60cm x 45cm x 37.5cm) were placed at both sides of the air conditioner casing on the condenser side of ambient air inlet grilles so that uniform flow can be maintained passing over the condenser. Before doing any further modification, the refrigerant had been discharged from the system. . Two air pre-heaters of 1 kW capacity were placed on both sides of the ambient air inlet to condenser, screwed to the ducts as shown in **Fig.1**. The capacity of the air pre heater is taken as 1kW in order to increase the ambient temperature by about 10ºC. Also a variac of 3.0 kW capacity with suitable range was selected to supply required power to the pre-heaters. Two independent watt meters of different capacity were connected with the compressor and fan power line to read the output directly from the meters. A refrigerant flow meter was placed in the liquid line to measure the flow rate of the refrigerant. A selector switch was used to measure the temperature at all relevant points in the air conditioner. A liquid receiver was placed to control the refrigerant pressure in the evaporator. To measure the condenser pressure and the evaporator pressure in the system, a high range pressure gauge and a low range pressure gauge were installed to get direct readings. The system was first completely evacuated by a vacuum pump and then charged with the required amount of R-22 using a charging system. To check whether there was any leakage or not in the line, soap foams were used at the

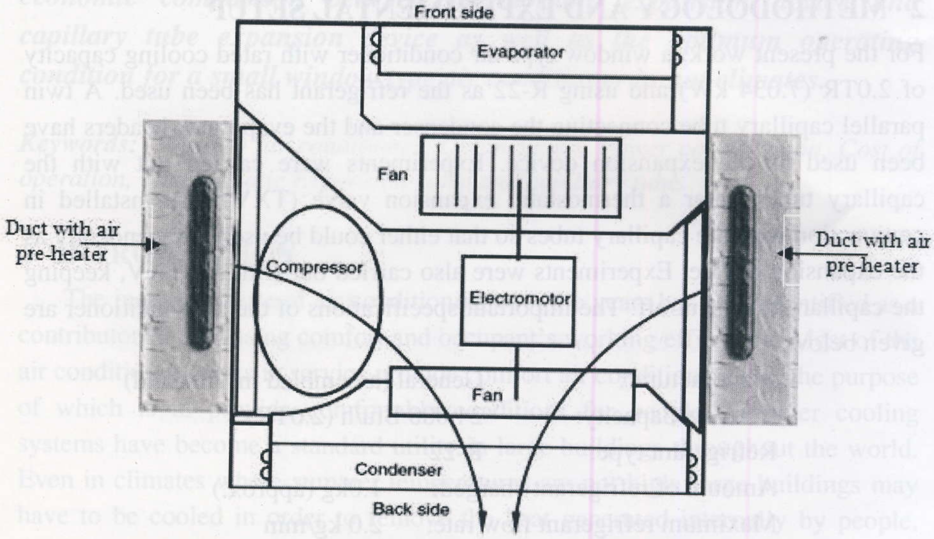

Figure 1: Schematic Diagram of the Air Conditioner.

joints. After the setup was ready for operation, experiments were started. Initial readings were taken without changing the temperature of the condenser cooling air. Then the variable transformer was adjusted to increase the heat input to the air pre-heaters, and a second set of readings were taken. In this way several readings were taken. After that, the system was switched to thermostatic expansion valve from capillary tube and the same procedure followed. The dry bulb and wet bulb temperatures of the air were measured at both condenser and evaporator side inlets and outlets using four different sets of dry-bulb and wet-bulb thermometers.

Equations Used

The refrigeration cycle is shown in **Fig.2**.

Heat absorbed in the evaporator (cooling capacity),

$$Q_e = m_r\,(h_1 - h_4) \qquad (1)$$

Heat rejected in the condenser,

$$Q_c = m_r\,(h_2 - h_3)\ (2) \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (2)$$

Power required for compression,

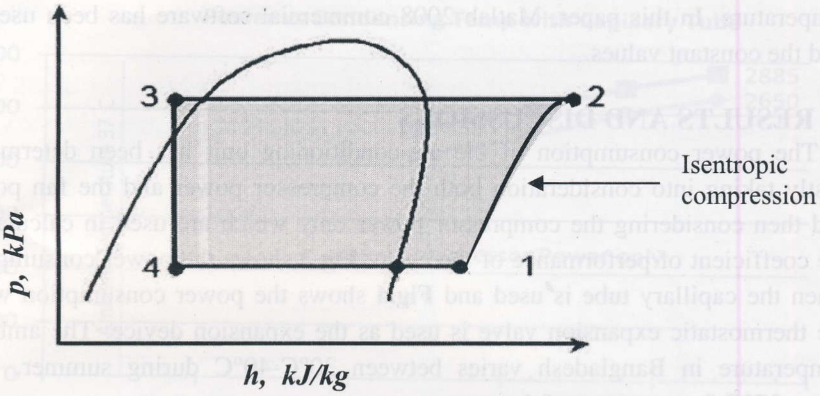

Fig 2: Refrigeration cycle on p-h diagram.

$$W_c = m_r\,(h_2 - h_1) \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \qquad (3)$$

C.O.P = cooling produced / power supplied,

$$C.O.P = (h_1 - h_4)/(h_2 - h_1) \quad \ldots \quad \ldots \quad \ldots \quad \ldots \qquad (4)$$

where,

m_r = mass flow rate of the refrigerant (kg/s)

h_1 = specific enthalpy at compressor entry, point 1 (kJ/kg)

h_2 = specific enthalpy at compressor exit, point 2 (kJ/kg)

h_3 = specific enthalpy at condenser exit, point 3 (kJ/kg)

h_4 = specific enthalpy at evaporator inlet, point 4 (kJ/kg).

Mathematical analysis

To analysis the system performance using mathematical equations, first step is to determine the power consumption equation for a reciprocating compressor that represent the experimental performance data (et. al. 13)

$$P = c_1 + c_2 t_e + c_3 t_e^2 + c_4 t_c + c_5 t_c^2 + c_6 t_e t_c + c_7 t_e^2 t_c + c_8 t_e t_c^2 + c_9 t_e^2 t_c^2 \qquad (5)$$

Where,

P = power required by compressor, kW

t_e = evaporating temperature, °C

t_c = condensing temperature, °C

The constants in **Eq.5** are determined by equation fitting procedures, e.g. the method of least squares or from taking nine experimental readings for the system and substituting it in the above equation to develop a set of nine simultaneous equations for solving the constants.

After calculating the nine constants for the empirical equation, the next step is to put the constant values in the equation and for a particular condensing temperature; we can get the power consumption for different evaporating temperature. In this paper, Matlab 2008 commercial software has been used to find the constant values.

3 RESULTS AND DISCUSSIONS

The power consumption of the air conditioning unit has been determined firstly taking into consideration both the compressor power and the fan power and then considering the compressor power only which are used in calculating the coefficient of performance of the cycle. **Fig.3** shows the power consumption when the capillary tube is used and **Fig.4** shows the power consumption when the thermostatic expansion valve is used as the expansion device. The ambient temperature in Bangladesh varies between 30°C-40°C during summer, with about 37°C for most part of the summer season.

Both **Fig.3** and **Fig.4** show that as the condenser temp is increasing, the power consumption is also rising. In **Fig.3**, the maximum power consumption reaches to a value of 2.88 kW which is 395W higher than that in **Fig.4.** So, the thermostatic expansion valve reduces the power consumption by 13.7%. This happens because of relatively lower evaporating pressure when the capillary tube is used as compared to the thermostatic expansion valve.

Fig.5 shows the change of refrigerating capacity with increase of condensing temperature. It is seen that the refrigeration capacity for both the capillary tube and the thermostatic expansion valve. The light line indicates the refrigeration capacity vs. condensing temperature when thermostatic expansion valve is used and the deep line indicates the refrigeration capacity with the capillary tube.

An attempt has been made to find the optimum operating condition of the air conditioning unit i.e. the minimum power requirement and hence the minimum cost of operation during typical summer days when the ambient temperature was about 35°C. A successive simulation process is used to verify how these experimental results vary with the results obtained from simulation.

Fig.6 and **Fig.7** show the variations of the coefficient of performance with change of condensing temperatures. The coefficient of performance of the system decreases as the condenser temperature increases. Initially, the decrease rate is rather small with increase of condensing temperature, but after a certain value, the COP reduces sharply for both the cases. When the fan power is considered, the coefficient of performance of the system will clearly be lower than when the fan power is omitted.

As mentioned earlier, the temperature in Bangladesh normally varies between 30°C to 40°C in the summer. The condensing temperature may be

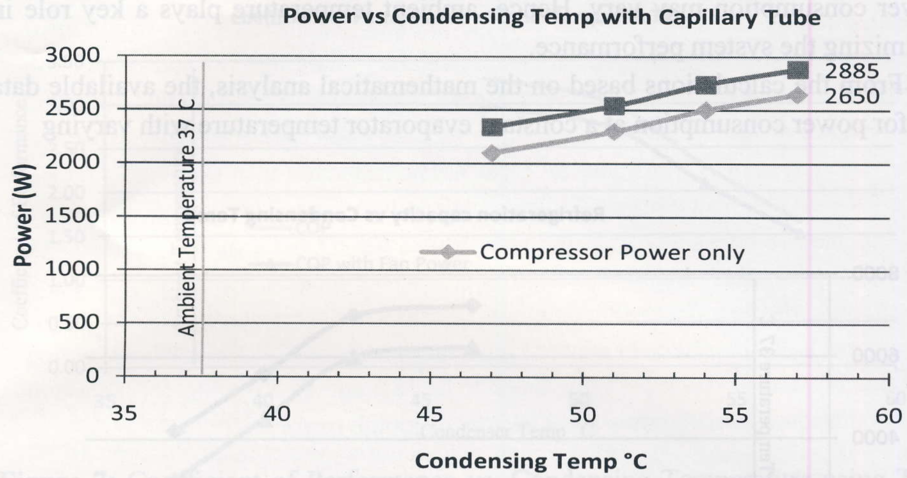

Figure 3: Power vs. Condensing Temperature with Capillary tube as expansion device, ambient temp 37°C.

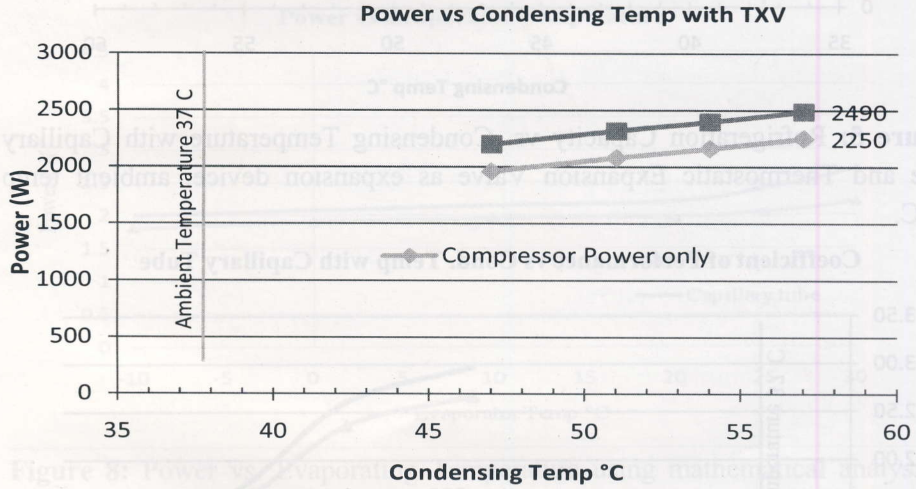

Figure 4: Power vs. Condensing Temperature with Thermostatic Expansion Device, ambient temp 37°C.

10°C~15°C higher than the ambient temperature. So, it is clear that power consumption is in the lower side when the ambient temperature is lower. It should be mentioned here that if the same air conditioner is tested for performance at some different country or at some other season the optimum power consumption may vary. Hence, ambient temperature plays a key role in optimizing the system performance.

From the calculations based on the mathematical analysis, the available data are for power consumption at a constant evaporator temperature with varying

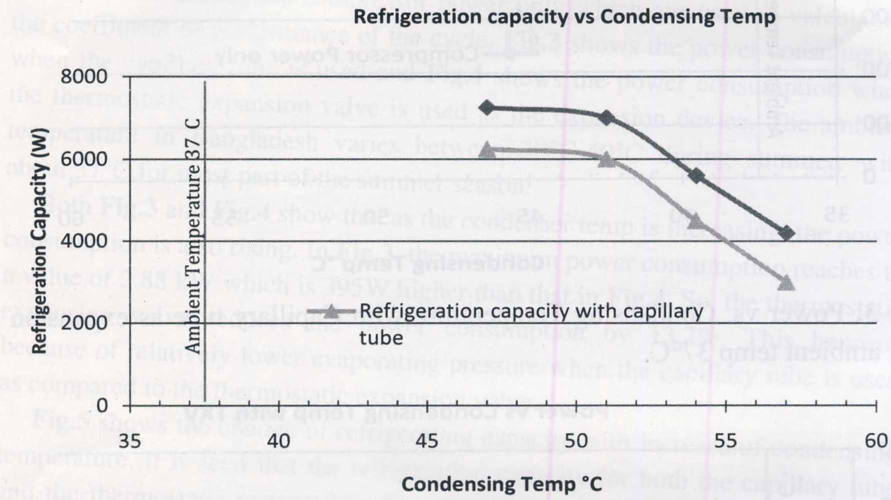

Figure 5: Refrigeration Capacity vs. Condensing Temperature with Capillary tube and Thermostatic Expansion Valve as expansion device, ambient temp 37°C.

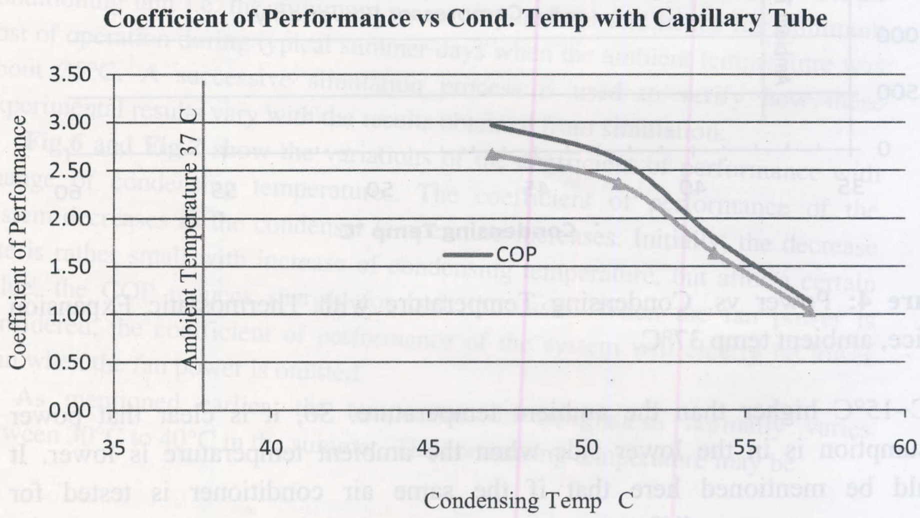

Figure 6: Coefficient of Performance vs. Condensing Temperature using Capillary tube as the expansion device, ambient temp 37° C.

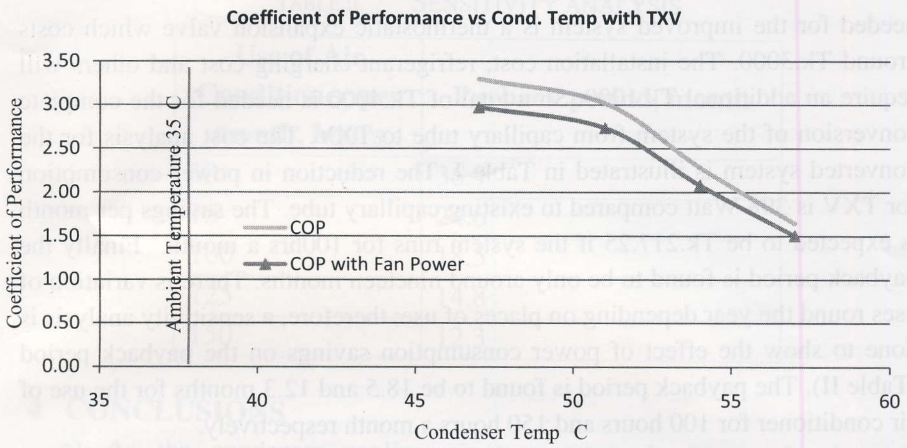

Figure 7: Coefficient of Performance vs. Condensing Temperature using TXV as the expansion device, ambient temp 37° C.

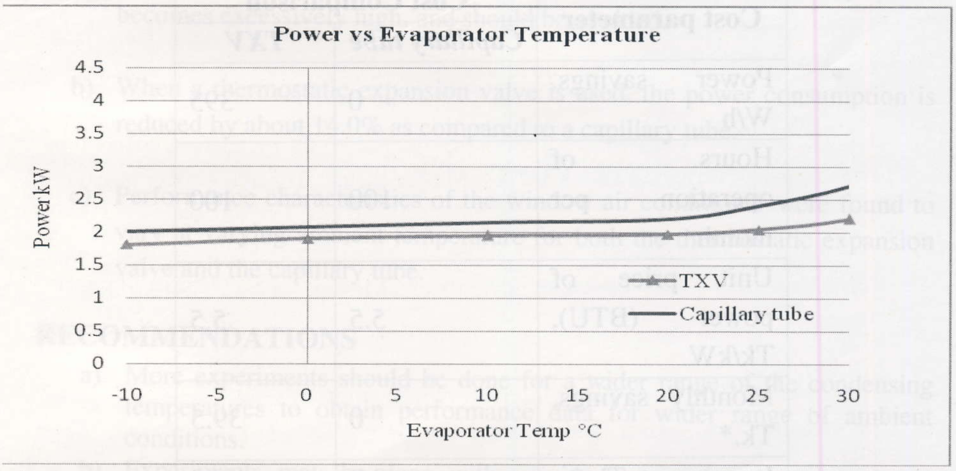

Figure 8: Power vs. Evaporating Temperature using mathematical analysis at condensing temp 50° C.

condenser temperature. The **Fig.8** also shows that the power consumption increases with increasing evaporating temperature and the minimum value is about 2.0kW at -10°C evaporating temperature.

Cost comparison of capillary tube and TXV

Cost comparison is done to calculate the payback period for efficient operation after modification with thermostatic expansion valve. The additional investment needed for the improved system is a thermostatic expansion valve which costs around Tk.3000. The installation cost, refrigerant charging cost and others will require an additional Tk.1000. So a total of Tk.4000 is needed for the complete conversion of the system from capillary tube to TXV. The cost analysis for the converted system is illustrated in Table I. The reduction in power consumption for TXV is 398 Watt compared to existing capillary tube. The savings per month is expected to be Tk.217.25 if the system runs for 100hrs a month. Finally the payback period is found to be only around nineteen months. There is variation of uses round the year depending on places of use; therefore, a sensitivity analysis is done to show the effect of power consumption savings on the payback period (Table II). The payback period is found to be 18.5 and 12.3 months for the use of air conditioner for 100 hours and 150 hours a month respectively.

TABLE I. PAYBACK PERIOD ANALYSIS OF TXV SYSTEM TYPE

Cost parameter	**Cost Comparison**	
	Capillary tube	***TXV***
Power savings, W/h	0	395
Hours of operation per month	100	100
Unit price of power (BTU), Tk/kW	5.5	5.5
Monthly savings, Tk.*	0	39.5
Investment, Tk.*	0.0	4000
Annual savings, Tk.*	0	474
Pay back period, month	--	18.41

*74 BDT Tk. Is equivalent to approx. 1 US Dollar.

TABLE II. SENSITIVITY ANALYSIS

Use of Air Conditioner per month, hours	Payback period, month
50	36.9
75	24.6
100	18.5
125	14.8
150	12.3

4 CONCLUSIONS

a) As the condenser cooling air temperature is increased when the evaporating temperature remains fairly constant, the cooling capacity decreases and the compressor power increases (hence cop decreases). Above 57°C condensing temperatures, the condensing temperature becomes excessively high, and should be avoided.

b) When a thermostatic expansion valve is used, the power consumption is reduced by about 14.0% as compared to a capillary tube.

c) Performance characteristics of the window air conditioner were found to vary at varying ambient temperature for both the thermostatic expansion valve and the capillary tube.

RECOMMENDATIONS

a) More experiments should be done for a wider range of the condensing temperatures to obtain performance data for wider range of ambient conditions.

b) Experiments may be done with varying evaporating temperatures at constant condensing temperatures.

c) Performance of window air conditioner with refrigerant other than R-22 should be studied and compared.

d) Optimization with inclusion of materials costs for the complete system may be studied.

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