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DYNAMIC MODELLING OF A VAPOUR COMPRESSION AIR CONDITIONER SYSTEM

I. O. Ajagbe and **O. S. Ismail**

Department of Mechanical Engineering
University of Ibadan
Ibadan, Nigeria

ABSTRACT

Modelling is a very important tool for understanding complex problems. A model plays a fundamental role in the analysis, design and development of complex systems, especially when physical testing and experimentation are not feasible – or not possible at all. Modelling technology holds tremendous promise for reducing costs, improving quality, and shortening the time-to-market for manufactured goods. Unfortunately, this technology still remains largely underutilized by industries today, especially in Nigeria. Analyzing vapour compression air conditioners for their capability of achieving a desired performance level is a difficult and essential task that must be undertaken by the specifying engineer. This involves an understanding of what is required for the particular air conditioning application as well as what the particular grouping of equipment components into the system are or, more importantly, are not capable of doing. A detailed dynamic model of a vapour compression air conditioner system is therefore developed. The Model components are mathematically described, considering energy balances within the system. Ambient wet and dry bulb temperature and solar radiation are the few required inputs. Output measures of performance of the system such as system temperatures, energy flows, and coefficient of performance can be predicted. Simultaneously, a software package, "*AJBSIM*" has been developed to simulate the steady state performance of the equipment, ease and hasten the calculation expedience involved as well as enable parametric studies to be conducted.

KEYWORDS: Air conditioner, Dynamic model, Simulation, Refrigeration, Vapour compression, Coefficient of Performance.

INTRODUCTION

Analyzing vapour compression air conditioners for their capability of achieving a desired performance level is a difficult and essential task that must be undertaken by the specifying engineer. This involves an understanding of what is required for the particular air conditioning application as well as what the particular grouping of equipment components into the system are or, more importantly, are not capable of doing. An effective measure of evaluating a vapour compression air conditioner is therefore vital for the purpose of predicting the performance as well as helping the selection of the equipment. A valuable approach to predicting the thermal performance of the vapour compression air conditioner is dynamic modelling.

Correspondence: Dr. O. S. Ismail, Department of Mechanical Engineering, University of Ibadan, Ibadan, Oyo State, Nigeria. E-mail: os.ismail@mail.ui.edu.ng

Dynamic models of vapour compression air conditioner systems have been the topic of numerous papers. The models can be generally classified in terms of degree of complexity and empiricism. A review of literature reveals a few limitations of existing models.

Gordon and Ng (1994) proposed a simple thermodynamic model for reciprocating chillers. The model predicts the Coefficient of performance (COP) over a wide range of operating conditions from the inlet fluid temperatures and the cooling capacity, using three fitted parameters. The prediction of the COP is remarkably good for a range of different chillers. However, the model does not predict the cooling capacity, it is required as an input.

A chiller model presented by Shelton and Weber (1991) is similar in approach and also has the same limitation of requiring the cooling capacity as an input.

Stoecker and Jones (1982), Allen and Hamilton (1983), Hamilton and Miller (1990) have presented steady state equation-fit models of vapour compression refrigeration systems with reciprocating compressor. The Allen and Hamilton (1983) model utilises overall system data, for example, entering and leaving water temperatures and flow ratios. The models of Stoecker and Jones (1982) and Hamilton and Miller (1990) require more detailed data, such as internal refrigerant pressures and temperatures. Consequently, the latter two models will be difficult to use for engineers, who only have access to catalogue data.

Cecclinni and Marchal (1991) proposed a computer program for simulating refrigeration and air conditioning equipments of all types: air to air, air to water, water to water, water to air. Some parameters characterising the components require experimental data from equipment testing, such as the heat exchanger mean surface temperatures, the saturation pressures in both evaporator and condenser and superheating and sub cooling. Again these data are not typically provided in manufacturers' catalogue.

Parise (1986) developed a vapour compression air conditioner simulation model to predict the overall performance of a system; simple models for the components of the compression cycle were employed. Input data include compression speed, displacement volume, clearance ratio and other parameters for a comparatively detailed description of each component. Clearly, all that provide the advantages of the deterministic approach require detailed data beyond what is typically provided in manufacturers' catalogues. However, it would be useful to have a simple dynamic model, which provides a detailed description of the components but only requires fewer input data (not including manufacturers' catalogue data or detailed component information). This model may be derived entirely from the basic thermodynamic energy balance equations as well as fundamental correlations of heat transfer. However, the evaluation of air conditioner mathematical model relations could be rigorous and discouraging due to the computational complexity, input data and associated time requirements involved.

The use of appropriate computational techniques such as simulation software package (developed for the model) would not only simulate the air conditioner equipment's steady state performance, but also ease and hasten the calculation expedience involved. Such a model has been developed and is the main object of this paper.

1. DEVELOPMENT OF THE DYNAMIC MODEL

The vapour compression air conditioner model considered here is of conventional type. It utilises an actual basic vapour compression cycle. The air conditioning circuit includes the usual compression, expansion (throttle) valve, evaporator and condenser (both of cross flow type). Heat rejection in the condenser is accomplished by natural convection, while air from the conditioned space is circulated by natural convection and maintained at constant temperature within the limit of the design indoor dry bulb temperature. The model equipment diagram is illustrated in Figure 1.

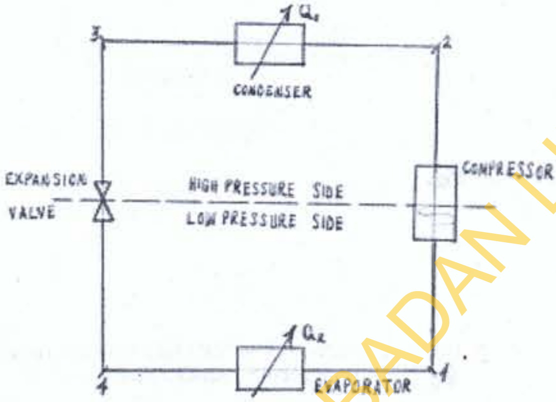

Figure 1. Equipment Diagram

2. MODEL DESCRIPTION

The actual air conditioning cycle considers heat transfer in the condenser and evaporator without pressure losses, a reversible adiabatic {isentropic} compression, and an adiabatic expansion throttle valve, connected by piping that has neither pressure loss nor heat transfer with the surroundings. With reference to cycle diagrams (displayed in Figures 2 and 3), the refrigerant leaves the evaporator at point 1 as a low pressure, low temperature, saturated vapour and gets superheated to point 1', before entering the compressor, where it is compressed isentropically. At point 2, it leaves the compressor as a high temperature, high pressure, superheated vapour and enters the condenser, where it is first de-superheated to point 2', and enters the condenser at constant pressure. At point 3, the refrigerant leaves the condenser as a high pressure, medium temperature, saturated liquid and gets sub-cooled before entering the expansion throttle valve, where it is expanded irreversibly and adiabatically{constant enthalpy}. At point 4, it leaves the expansion valve as a low pressure and temperature, low quality vapour and enters the evaporator, where it is evaporated reversibly at constant pressure to saturated state point 1.

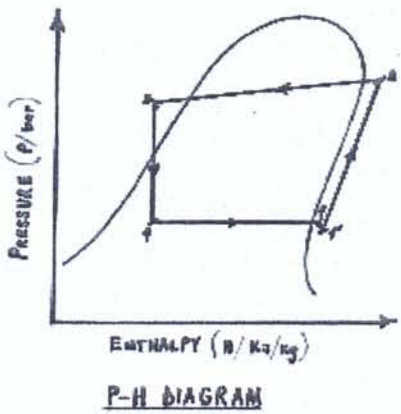

Figure 2: Pressure – Enthalpy Diagram

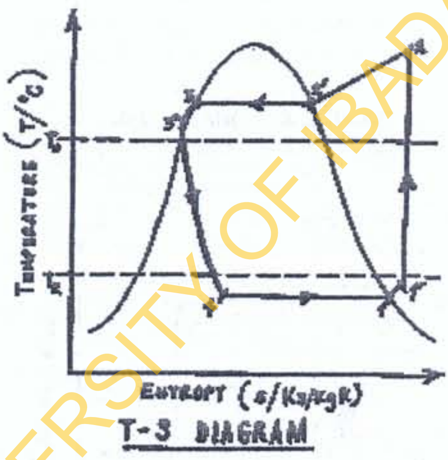

Figure 3: Temperature – Entropy Diagram

3. SELECTION OF INPUT PARAMETERS

The few inputs required for the model analysis include hourly variations of ambient wet and dry bulb temperature condition as well as hourly variations of solar radiation, both of which depend on measured meteorological data for the geographical location.

However, additional input parameters such as transport (or state point) properties of the selected refrigerant (working fluid of the air conditioner model) are required together with the basic inputs in solving the model equations in other to yield the output measures of performance. These transport properties of the refrigerant were estimated using appropriate thermodynamic relations and with reference to the cycle diagrams above (Figures 2 and 3).

The solar radiation input could be explained as having an indirect impact on the air conditioner unit, as it affects the space being conditioned (which surrounds the equipment) via the building construction materials such as the roof, walls, glass, floor, etc. The space being conditioned in turn affects the equipment. Thus, the building energy (or the building cooling load) is a measure of the solar radiation input into the building.

In this research, a case study (an industrial building of known specifications) in Ibadan metropolis in Nigeria was considered. Approximate values of ambient wet and dry bulb temperature variations were reasonably culled from daily meteorological temperature data for Ibadan location.

With the use of standard calculation procedures, together with the location's weather data and data in respect of load due to building material specifications, the variations in building cooling load was estimated. These estimates were carried out based on the design month (when the solar radiation intensity is at maximum) for the location considered. The design month is March for Ibadan. The basic inputs to be used in the analysis of the air conditioner model are as presented in Table 1 below.

Basic Input Parameters									
{Steady state} Hourly time step	Outdoor Conditions					Indoor Conditions			Estimated Air conditioning Load
	Dry bulb Temp.		Wet bulb Temp.		Moisture Content	Dry bulb Temp.		Moisture Content	
(hr)	°C	°F	°C	°F	g/lb	°C	°F	g/lb	KJ/s
5.00 -6.00A.M	27.30	81.14	20.80	68.14	5.352	25	77	5.079	73.2030810
6.00 - 7.00A.M	28.00	82.40	23.00	73.40	7.075	25	77	5.079	104.894219
7.00 - 8.00A.M	29.00	84.20	23.56	74.40	7.302	25	77	5.079	130.573724
8.00 - 9.00A.M	30.80	85.28	24.96	76.93	8.160	25	77	5.079	161.577684
9.00 - 10.00A.M	31.60	87.44	24.98	76.97	7.891	25	77	5.079	178.191914
10.00 - 11.00A.M	33.70	88.88	26.16	79.08	8.753	25	77	5.079	199.599956
11.00 - 12.00A.M	33.90	92.60	26.48	79.66	8.617	25	77	5.079	213.669011
12.00 - 13.00P.M	35.00	92.84	27.13	80.84	8.798	25	77	5.079	227.480999
13.00 - 14.00P.M	32.80	95.00	28.78	83.80	9.705	25	77	5.079	248.668289
14.00 - 15.00P.M	31.500	91.04	26.95	80.50	9.116	25	77	5.079	239.670367
15.00 - 16.00P.M	31.50	88.70	26.14	79.05	8.708	25	77	5.079	225.218330
16.00 - 17.00P.M	30.00	86.54	24.94	76.89	8.073	25	77	5.079	199.717060
17.00 -18.00P.M	29.80	85.64	24.81	76.75	8.163	25	77	5.079	172.839935

Table 1. Basic Input Parameters

4. MATHEMATICAL MODEL EQUATIONS OF SYSTEM COMPONENTS

The analysis of the air conditioner model involves the application of thermodynamic principles to the actual basic vapour compression of the developed model. In predicting the performance, or otherwise, to investigate the effect of the basic input variations on the performance of the vapour compression air conditioner, appropriate relationships between the two sets of parameters (input and output measures of performance) are sought.

Applying the steady flow equation (a statement of the first law of thermodynamics) to each of the components of the vapour compression cycle, the following energy balance relations were derived.

1. Energy flow in Evaporator: Air conditioning load is

$$Q_R = m(h_1 - h_4);\ \text{in KJ/s} \tag{1}$$

Thus, refrigerant flow rate is

$$m = \frac{Q_R}{(h_1 - h_4)};\ \text{in Kg/s} \tag{2}$$

2. Energy flow in Condenser: Condenser heat transfer rate is

$$Q_O = m(h_2 - h_3);\ \text{in KJ/s} \tag{3}$$

3. Energy flow in Compressor: At balance point of the vapour compression equipment

$$W_{comp} = Q_O - Q_R \tag{4}$$

Also, compressor work transfer rate is

$$W_{comp} = m(h_2 - h_1);\ \text{in KJ/s} \tag{5}$$

4. Energy flow in Expansion {throttle} valve: The input work transfer in the expansion {throttle} valve is assumed negligible, as it is small compared to the energy flows in the other components. This implies that

$$h_3 = h_4 \tag{6}$$

5. Coefficient of performance (C.O.P) of the system:
C.O.P = refrigeration effect / compressor work rate

$$= \frac{Q_R}{W_{comp}} \tag{7}$$

6. Also due to Irreversibility in the system, Energy degraded or Irreversibility is

$$\text{Irrev.} = T_0\ \{\Delta S_R + \Delta S_0\}$$

$$= T_O\left\{-\frac{Q_R}{T_R} + \frac{Q_O}{T_O}\right\} \tag{8}$$

5. APPLICATION OF "*AJBSIM*" SOFTWARE PACKAGE

The mathematical equations for the air conditioner model, the equations for the refrigerant's transport properties as well as the set of equations for the cooling load computation were firstly programmed using FORTRAN. The programme was then imported to Visual Basic environment, thus enabling simulation of the equipment via a user-friendly interface. The combination of the two programming languages has been used to develop "*AJBSIM*"- an air conditioner Simulation software package.

However, several assumptions have been invoked to ease and simplify the analysis of the air conditioner model. These assumptions however define the limit and scope of validity of the analysis results. The results from the dynamic simulation run, using the software, is as shown in Table 2 below.

Time step (Hr)	flow rate, m(Kg/s)	evaporator load Q_R(KJ/s)	condenser load Q_O(KJ/s)	compressor load W_{comp}.(KJ/s)	Irreversibility Irrev,(KJ/s)	System's C.O.P C.O.P
5.00- 6.00	0.5203870	73.2030810	78.808474	30.0064180	-1.1292910	2.439581
6.00- 7.00	0.7494320	104.894219	113.64014	43.7106590	-3.8413850	2.399740
7.00- 8.00	0.9396700	130.573724	142.746404	55.6972520	-8.7191160	2.344348
8.00- 9.00	1.1704300	161.577684	142.746404	70.3746270	-15.153824	2.295965
9.00-10.00	1.2993180	178.191914	198.029183	79.2345710	-21.503255	2.248916
10.00-11.00	1.4640900	199.599956	223.449915	90.383274	-28.844429	2.208373
11.00-12.00	1.5924130	213.669011	243.859817	101.413805	-44.166010	2.106903
12.00-13.00	1.6966440	227.480999	259.863737	108.209733	-47.690574	2.102223
13.00-14.00	1.8718230	248.668289	287.252422	121.473558	-60.883182	2.047098
14.00-15.00	1.7740090	239.670367	271.273977	111.493728	-43.173544	2.149630
15.00-16.00	1.6507710	225.218330	251.899502	101.753944	-31.875594	2.213362
16.00-17.00	1.4509410	199.717060	220.936556	87.791845	-21.120521	2.274893
17.00-18.00	1.2510980	172.839935	190.332822	75.1061950	-15.692381	2.301274

Table 2. Simulated Performance Results

6. RESULTS

For the given set of basic input parameters, performance results from the simulation run, using the software package, are as presented graphically in Charts 1, 2, 3 and 4 below.

Chart 2, which depicts the energy flows against time step within the respective components of the model, expectedly depicts that the times of maximum energy flows within the evaporator, condenser and compressor all coincide at the peak time, which in this case, occurs at a steady state clock period of 13.00-14.00Hrs. These maximum energy flows specify the optimum duty (load) of the respective components.

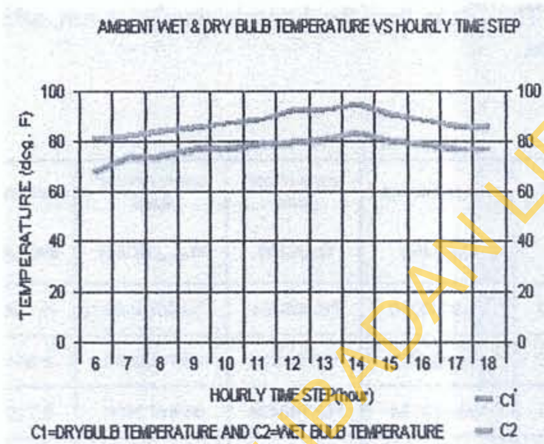

Chart 1. Ambient temperature against time step

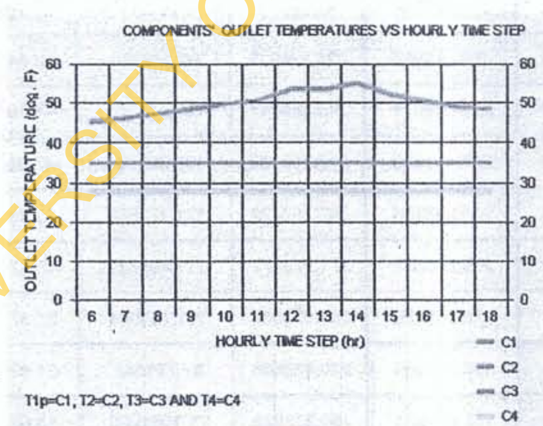

Chart 2. Energy flows within system components

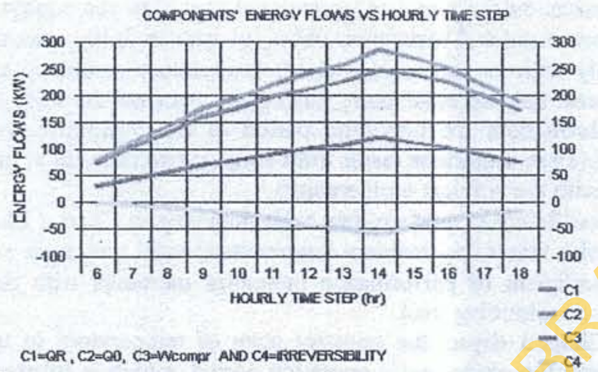

Chart 3. Components' outlet temperature against time step

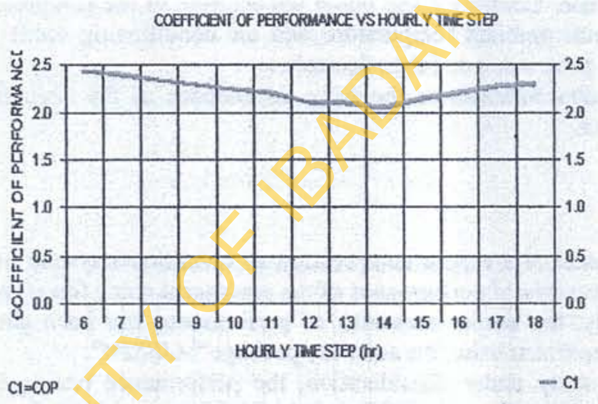

Chart 4. Coefficient of performance against time step

However, it will also be seen from Chart 2 that throughout the operation period of the equipment, the compressor load always "leads" the air conditioning load (Q_R) and the compressor work (W_{comp}). This could be explained by the fact that an energy balance exists (i.e. the heat rejected by the refrigerant in the condenser is always equal to the sum of the heat absorbed by the refrigerant in the evaporator and the work of the compressor). More so, it confirms the cyclic nature (operation) of the vapour compression air conditioner.

On the contrary, the results illustrate that the irreversibility (energy degraded) of the system decreases from start of operation at (5.00 - 6.00 a.m.), gets to a minimum at peak time and then increases until the end of operational period of the equipment. This trend is attributed to the condenser and expansion valve, as irreversibility within each component changes significantly with variations in ambient temperature compared to irreversibility in the other components. For the condenser, this occurs because the inlet state of condenser remains constant throughout the operation period of the equipment. In the case of the expansion valve, the large variations result from temperature changes at the expansion valve outlet (which vary with the ambient temperature).

The plots for coefficient of performance with time-step in Chart 4 illustrates a decrease in the parameter (with increasing ambient temperature) until minimum value is attained at peak time. The coefficient of performance thereafter increases with decreasing ambient temperature and air conditioning load.

The plots of Chart 3 depict the constant state of temperature in the condenser and evaporator throughout the equipment's operation period, which is substantiated by the fact that transition of the refrigerant from one phase to the other occurs at a constant temperature. In addition, the heat transfer across the components occurs through a finite temperature difference. Contrarily, the outlet temperature of the compressor and expansion valve increases (with ambient temperature and air conditioning load) from the start of operation, gets to a peak value and then decreases.

The above results demonstrate the large dependence of the simulated results on the system's basic inputs.

CONCLUSIONS

A dynamic model of a vapour compression air conditioner system has been developed to predict output measures of performance of the equipment using fewer inputs.

Simultaneously, the output measures of performance has been simulated (based on steady state considerations) using the software package "*AJBSIM*".

For the case study under consideration, the performance results from the dynamic simulation run shows that the energy flows within the components peaks (i.e. when they are at maximum) at steady state period 13.00-14.00 p.m. To assist the selection of air conditioner equipment (for the case study under consideration), the optimum load specifications of the model (from the simulation results) are under listed.

Evaporator Duty – 248.67KW
Condenser Duty – 287.25KW
Compressor Duty – 121.47KW
Required air conditioning capacity (peak cooling load) – 70.72tons
C.O.P of system – 2.4396
Energy degraded within system – 60.88KW

Furthermore, the large dependence of the simulated results on the system's basic inputs has been demonstrated.

In the model presented above, fewer input parameters (such as ambient temperature and solar radiation variation) has been used to evaluate the performance of the air conditioner equipment, compared to detailed component information or manufacturers' catalogue data used/required by earlier existing models.

Parametric studies could be conducted using the software package. The versatility of the developed software has been tested through simulation of different set of input parameters (from appropriate data source). Extremely useful results have been obtained in most cases testing the applicability of the model to the air conditioner system.

NOMENCLATURE

SYMBOLS

m	refrigerant flow rate, Kg/s
1, 1′, 2, 2′, 3, 3′, 4	cycle state points
1	evaporator outlet
1′	compressor inlet
2	compressor outlet, condenser inlet
2′	condenser intermediate point
3	expansion valve inlet
3′	condenser outlet
4	expansion valve outlet, evaporator inlet
T_o	ambient dry bulb temperature, ° C, F
T_R	indoor temperature, ° C, F
Irrev.	energy degraded, KJ/s, KW
COP	coefficient of performance
W_{comp}	compressor duty {load}, KJ/kg
Q_o	condenser duty {load}, KJ/kg
Q_R	building cooling load, KJ/kg; air conditioning load, evaporator duty {load}
h	enthalpy, KJ/kg
S	entropy, KJ/kg

SUBSCRIPTS

R	conditioned space, Indoor
o	ambient, outdoor

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