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Performance Estimation of Vapour Compression Refrigeration System using Real Gas Model

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Abstract. The performance of a vapour compression refrigeration system depends greatly on the spatial arrangement of the components, properties of the refrigerant and lubricant. Numerous models have been developed by using these properties to predict the performance of the vapour compression refrigeration system. The usual ideal gas model does not yield accurate results due to its limited range. In this study, an analytical real gas model has been developed for heat transfer in the evaporator and work output rate of compressor to achieve the coefficient of performance (COP) of the system. This real gas model is validated against the investigational data and the end result obtained was consistent with an error band of 10% for exhaust temperature and 7% for COP. This model can be used to predict the COP of any vapour compression refrigeration system, given that the thermodynamic properties of the working fluid and required dimensions of the system are known. However, it cannot be extended to compressor oils due to it does not differentiate between necessary properties like viscosity of refrigerator oil.

1. Introduction

The utilization of energy by the refrigerators is rising progressively due to their rapid usage. These ever-increasing energy requisite forces heighten the shortage in the energy and environment pollution. The previous researchers found the numerous models to determine the performance of a hermetically sealed compressor under dissimilar conditions. The refrigerant and lubricants plays a vital role in the working and performance of a compressor. To attain a high COP can be attained, high values of latent heat, liquid thermal conductivity in vapour density and low values of liquid viscosity and molecular weight are required. The experimental studies reveal usage of nanoparticles in lubricant has reduced power input to the compressor [1].

Due to rising environmental pollution concerns, bio-lubricants are projected as substitute to petroleum oil based lubricants. Vegetable oils have been considered for use in industries. But at high temperature and pressure, their performance is poor. However, plant oils are able of given that extremely low friction and wear coefficients under highly controlled test conditions. High lubricity, less lubricant consumption, energy efficiency, public health and cost effectiveness are some of the advantages of bio lubricants [2]. Previous studies show that the optimization methods provide the superior statistical prediction efficiency [3, 4]. Jatinder Gill et al [5] examined the vapour compression refrigeration system performance with the mixture of R134a and LPG refrigerant on various evaporator

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temperatures. The adaptive neuro-fuzzy inference system (ANFIS) model was s to predict the COP of the system. The superior statistical efficiency results with less than 0.3% percentage error was found in the evaluation of statistical analysis to mathematical using ANFIS models. Wang et al [6] presented the lubricant effect on the heat transfer characteristics on conventional refrigerants and natural refrigerant R-744. The results concluded that parameters like heat flux, saturation temperatures, vapour quality, oil concentrations and geometric configurations and affect the heat transfer coefficient. The present work the real gas mathematical model was developed to investigate the behaviour of heat exchangers in steady state vapour compression refrigeration system is validated against published data.

2. Experimental

2.1 Description of physical system

The test rig for conducting a performance test is newly developed as per the refrigeration test rig standards. Fig.1 shows the schematic diagram of simple vapour compression refrigeration system. The model of testing hermetically sealed reciprocating compressor, AQAW77X thermally protected for R134a was procured from shangzhure refrigeration Co. Ltd, the power input of the compressor has ½ HP and single phase. The condenser was air cooled and the capillary tube expansion valve was used for expansion process, the evaporator cabin dimensions are height 0.65m, length 6.54m and breadth 0.42m. The thermocouples used were calibrated T type for observing the temperature of the refrigerant.

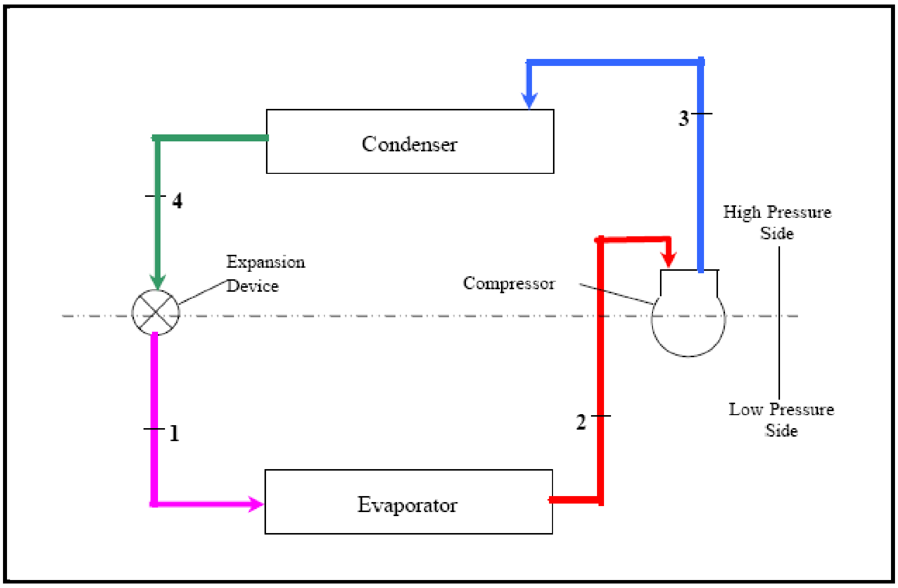

Fig.1 Schematic diagram of simple vapour compression refrigeration system

The temperatures were measured with thermocouples, read by 7-segment display of 4 digits. The pressures were measured using pressure gauges (absolute). The temperature and pressures were monitored and were recorded in steady state at the four thermodynamically significant points. Using a p-h chart, the enthalpy at those points were evaluated. From these enthalpies, the theoretical work input and evaporative heat were estimated. Also, the theoretical coefficient of performance was calculated. This experimental result was compared with the results obtained from the model.

2.2 Model Derivation and Evaluation

A mathematical model is used to predict the behaviour of the vapour compression refrigeration system. The execution of simulation program is based upon real gas mathematical models of the components of the refrigeration circuit including the compressor, heat exchangers, thermostatic expansion valve.

2.2.1 Heat Transfer in Constant Area Steady Flow

The following assumptions are made about the heat transfer in steady state condition. There is no heat transfer along the axis of the pipe. The flux, measured as energy per unit surface area per unit time, is constant. Pressure along the flow is constant. The surface tension energy is negligible. For such a pipe of constant area of cross-section, under steady state condition, the energy balance equation can be easily constructed from scratch. Since the fluid is a mixture of gas and liquid, the total enthalpy *dh* is the sum of changes in enthalpies in gas and liquid, weighted by a parameter called dryness fraction *x*.

Under constant pressure *dp=0,* and hence, the changes in temperatures are the only driving force for the changes in enthalpy. Thus, we write $(dhg) = cp,\ dT$ and $(dhf) = c,\ dT$ whereas, the dryness fraction *x* is defined as,

$$x = \frac{v}{v_g} = \frac{ZRT}{p} \times \frac{p}{Z_s R T_s} = \frac{ZT}{Z_s T_s} \tag{1}$$

Thus, the change in dryness fraction dx is expanded using logarithm in the following manner.

$$dx = \left(\frac{ZT}{Z_s T_s}\right) \times \left[\frac{dZ}{Z} + \frac{dT}{T} - \frac{dZ_s}{Z_s} - \frac{dT_s}{T_s}\right] \tag{2}$$

But, since, Ts = f(p) and Zs = g(Ts , p), dTs = 0 ; dZs = 0 since dp = 0. Hence

$$(dx)_p = \left(\frac{ZT}{Z_s T_s}\right)\left[\frac{dZ}{Z} + \frac{dT}{T}\right] = \left(\frac{ZT}{Z_s T_s}\right)\left[\frac{dZ}{dT}\frac{1}{Z} + \frac{1}{T}\right] dT \tag{3}$$

The total enthalpy difference (dh)p, however, is defined in terms of the above components, The isobar or constant pressure work done can be evaluated in the following way as

$$\delta w = p dv = p\left(\frac{\partial v}{\partial T}\right)_p dT = R\left[\left(\frac{\partial Z}{\partial T}\right)_p T + Z\right] dT \tag{4}$$

Velocity of the fluid flow, for a constant mass flux G is u = Gv. The infinitesimal change in velocity can be written in terms of the chosen variables

$u = Gv$ Implies $du = Gdv$ (or)

$$udu = G^2\left(\frac{ZRT}{p}\right)\left(\frac{\partial v}{\partial T}\right)_p dT = G^2\left(\frac{ZRT}{p}\right)\left(\frac{R}{p}\left[\left(\frac{\partial Z}{\partial T}\right)_p T + Z\right]\right) dT \tag{5}$$

Now, the heat balance equation can be written as $(dh_p) = (dh + udu + dw)_p$

Substituting for dh, udu & dw in above,

$$\delta p = \left(c_{pf} + c_{pg}\frac{ZT}{Z_s T_s} + \frac{\partial(ZT)}{\partial T}\left[\frac{\Delta h}{Z_s T_s} + (ZT)\left(\frac{G^2 R^2}{p^2}\right) + R\right]\right) dT \tag{6}$$

Substituting for constant flux and making change in heat as the subject in the above equation

$$\delta p = \frac{\dot{Q}\Pi}{GA} dl = \left(c_{pf} + c_{pg}\frac{ZT}{Z_s T_s} + \frac{\partial(ZT)}{\partial T}\left[\frac{\Delta h}{Z_s T_s} + (ZT)\frac{G^2 R^2}{p^2} + R\right]\right) dT \tag{7}$$

In other way, the above equation can be rearranged to to be written as

$$\frac{dT}{dl} = \frac{\dot{Q}\Pi}{GA}\left(c_{pf} + c_{pg}\frac{ZT}{Z_s T_s} + \frac{\partial(ZT)}{\partial T}\left[\frac{\Delta h}{Z_s T_s} + (ZT)\frac{G^2 R^2}{p^2} + R\right]\right)^{-1} \tag{8}$$

Alternatively, (7) can be integrated to obtain length – Temperature relationship, implicitly in terms of temperature. Thus, length *l* = f (T) only. However, this form does not allow closure in terms of length.

$$l = \left(\frac{GA}{\dot{Q}\Pi}\right) \int \left(c_{pf} + c_{pg}\frac{ZT}{Z_s T_s} + \frac{\partial(ZT)}{\partial T}\left[\frac{\Delta h}{Z_s T_s} + (ZT)\frac{G^2 R^2}{p^2} + R\right]\right) dT \tag{9}$$

Integrating the above, as an indefinite integral and introducing a constant of integration,

$$l = \left(\frac{GA}{\dot{Q}\Pi}\right)\left(c_{pf} T + \frac{\Delta h}{Z_s T_s} + R\right)(ZT) + \left(\frac{G^2 R^2}{2p^2}\right)(ZT)^2 + \frac{c_{pg}}{Z_s T_s}\int (ZT) dT + k \tag{10}$$

To calculate the integral in (10), we use integration by parts and simplify it.

$$l = \left(\frac{GA}{\dot{Q}\Pi}\right)\left(c_{pf} T + \frac{\Delta h}{Z_s T_s} + R\right)(ZT) + \left(\frac{G^2 R^2}{2p^2}\right)(ZT)^2 + \frac{c_{pg}}{Z_s T_s}[T \int Z dT + \int\int Z dT dT \tag{11}$$

However, it is enough to integrate the differential equation dh with respect to the temperature distribution to compute the change in heat along the duct. Thus,

$$\Delta q = \int_{h_0}^{h_L} dh = \int_{T_0}^{T_L} \frac{dh}{dT} dT = \int_0^L \frac{dh}{dT}\frac{dT}{dl} dl \tag{12}$$

The enthalpy change as written as

$$(dh)_p = c_{pf} dT + c_{pfg}\left(\frac{ZT}{Z_s T_s}\right) dT + \Delta h \left(\frac{ZT}{Z_s T_s}\right)\left[\left(\frac{\partial Z}{\partial T}\right)_p \times \frac{1}{Z} + \frac{1}{T}\right] dT \tag{13}$$

The third term in above equation can be simplified by taking LCM and observing that it is a term that can be simplified in a calculus method.

$$(dh)_p = c_{pf} dT + c_{pfg}\left(\frac{ZT}{Z_s T_s}\right) dT + \Delta h \left(\frac{1}{Z_s T_s}\right)\left[\left(\frac{\partial Z}{\partial T}\right)_p \times T + Z\right] dT$$

$$(dh)_p = \left(c_{pf} + ZT\left(\frac{c_{pfg}}{Z_s T_s}\right) + \left(\frac{\Delta h}{Z_s T_s}\right)\left[\frac{\partial}{\partial T}(ZT)_p\right]\right) dT \tag{14}$$

From, (12), we write

$$\Delta q = \int_T \left(c_{pf} + \frac{c_{pfg}}{Z_s T_s}(ZT) + \frac{\Delta h}{Z_s T_s}\left(\frac{\partial}{\partial T}(ZT)_p\right)\right) dT$$

$$= q_0 + c_{pf} T + \frac{Cpfg}{Z_s T_s}\int_T (ZT) dT + \frac{\Delta h}{Z_s T_s}(ZT) \tag{15}$$

The integral in (15) cannot be directly evaluated and in order to facilitate numerical integration, it is expanded using a mathematical technique called integration by parts. Also, as the expression thus obtained only qualifies for heat per unit mass while it is of general interest to obtain the rate of heat transfer to be computed. Thus, we multiply the above expression with rate of mass flow to obtain the gross heat transfer rate. Thus,

$$\dot{Q} = \dot{m}(q_o + c_{pf} T + \frac{c_{pfg}}{Z_s T_s}[T\int_T Z dT - \int_T \int_T Z dT dT + \frac{\Delta h}{Z_s T_s}(ZT) \tag{16}$$

The above equation gives insight on the nature of heat transfer. The integrals involved with the compression factor can be found numerically or analytically based on the model for compression factor used.

2.2.2 Work Transfer through Reciprocating Compressor

Compressor is the essential component of the vapour compression system. It creates the pressure difference that drives the working fluid through the system. In order to estimate the work transfer rate in the compressor, the compressor is a reciprocating compressor with a single slider crank mechanism is to be assumed. The single Slider-Crank Mechanism is shown in Fig.2.

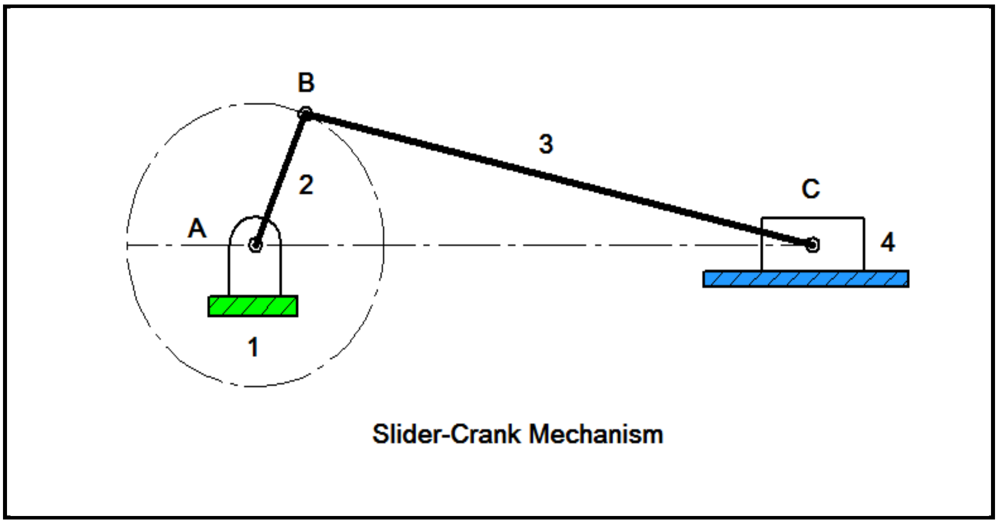

Fig.2 Single Slider-Crank Mechanism

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Let the crank be of radius r and connecting rod of length c, and are mass less for the sake of idealization. The mass of the piston be m. Then, the position of the piston from the origin is,

$$y = r\left(\sqrt{n^2 - sin^2\theta} - cos\theta\right) \tag{17}$$

where, n is the ratio of the lengths of connecting rod to that of the crank rod. As the position is known from the above equation, it is easy to compute the first and second derivatives of the above form, to obtain ẏ,&ÿ respectively.

$$\dot{y} = -r\dot{\theta}\left(sin\theta + \frac{sin2\theta}{\sqrt{n^2-\sin^2\theta}}\right) \text{ and } \ddot{y} = r\,\dot{\theta}^2\left(cos\theta + \frac{sin2\theta}{\sqrt{n^2-\sin^2\theta}} + \frac{cos2\theta}{(n^2-\sin^2\theta)^{\frac{3}{2}}}\right) \tag{18}$$

With the above definitions intact, we may now define the pressure exerted, using Newton's second law of motion.

$$F = m \times \ddot{y} = pA \tag{19}$$

In other way, we can rewrite the above equation as,

$$e = y - (c - r) \tag{20}$$

The instantaneous volume of the cylinder is realized with transforming the slider position into piston's displacement from the cylinder BDC.

$$\forall = Ae = A(y - c + r) \tag{21}$$

The instantaneous volume is then calculated as the product of y and the area of cross section A.

The thermodynamic work done is computed as, $\delta W = pdA$

From the above equation, $d\forall = Ady = A\frac{dy}{dt}dt = A\dot{y}dt$

From the equation for Newton's second law,

$$p = \frac{\dot{y}\dot{m}}{A} \Rightarrow \delta W = \frac{m}{A}\dot{y}y\,\ddot{A}dt = m\ddot{y}\dot{y}dt$$

In other way, we can rewrite the above equation as

$$\frac{dW}{dt} = W = m\dot{y}\ddot{y} \tag{22}$$

However, it is evident from the above equation that the rate of work transfer is dependent on the first and second derivatives (velocity and the acceleration) of the piston, and is time dependent.

2.2.3 Algorithm to Calculate the Coefficient of Performance

To calculate the coefficient of performance of the system in consideration, the heat transfer rate and the work transfer rate are the required data. However, due to the complex calculations involved, despite the software used, it is a complex task to be done, and revert to define the steps involved in the calculation. The function files TGrad.m, heat.m, xdot.m and xddot.m were written in Octave and Matlab are usable in format. The first two m-files also use another m-file named compFactor.m, that computes the compressibility factor at a given temperature and pressure. The inbuilt wrapper for lsode, a FORTRAN package traditionally used to handle to solve initial value problem of ODE is used in this paper. Though, the inbuilt function ode45 can also be used to obtain the same solution. Using the predefined values and the accessory m-files, with the given initial value, the lsode computes the temperature distribution for a given range of values. Since the nature of the solution is not known initially, a very small step size of 0.01 is assumed. Using the temperature distribution, the m-file heat.m is used to estimate the heat transfer rate which is saved to the internal memory. Again, the m-files xdot.m and xddot.m are used to compute velocity and acceleration of the piston that in turn are used to estimate the work transfer rate. These results are used to calculate the coefficient of performance of the system.

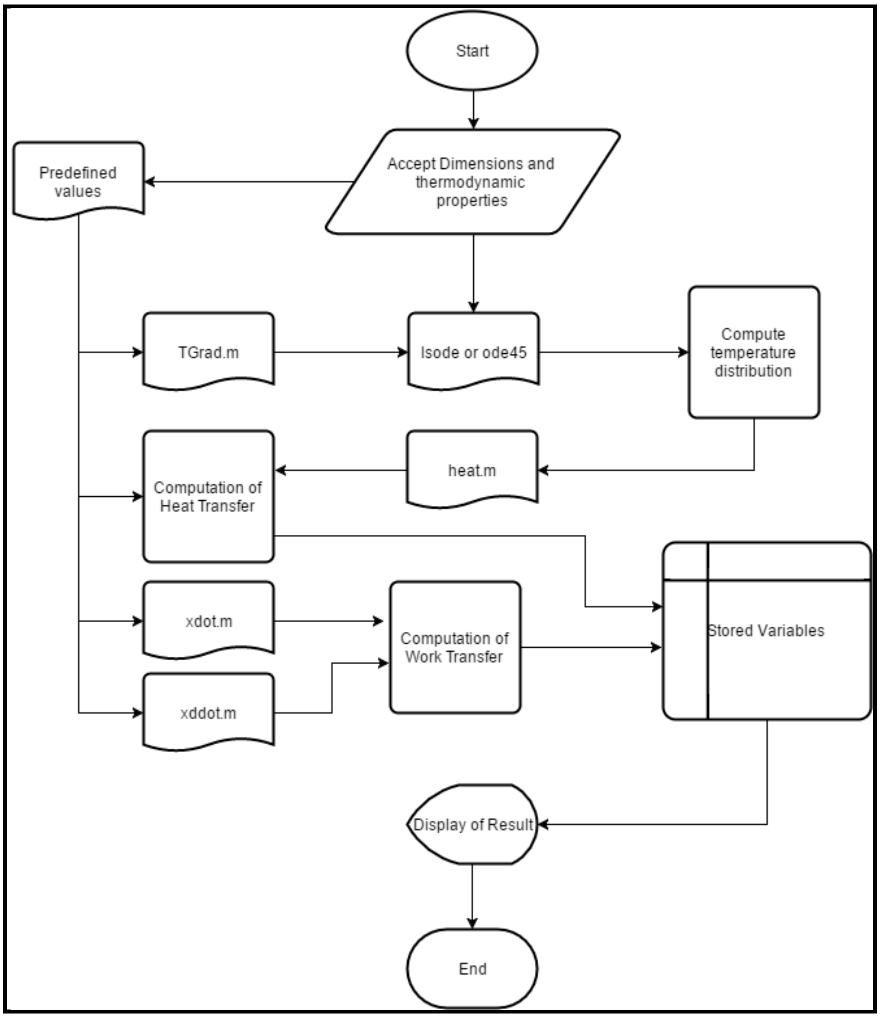

Fig.3. Computational flow to calculate COP

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3. Results and Discussion

In this study, the steady-state behaviour and simulation of the refrigeration system using R134a and synthetic refrigeration oil was presented. Throughout the process, the heat and work changes of the system were analyzed.

3.1 Heat Transfer and Work Transfer Rate

In this study, experimental data were collected for the performance analysis of R134a synthetic refrigeration oil (ISO grade POE68). The results obtained from experimentation and model is given in the Tables.2 and 3. From the Fig.4,for this particular case, it can be shown that the temperature gradient does not depend upon the length, which is obvious from the definition of the gradient, in equation [22], apparent that the gradient changes direction of propagation around some temperature between 300 K and 400 K. The pressure of the evaporator is altered according to the difference.

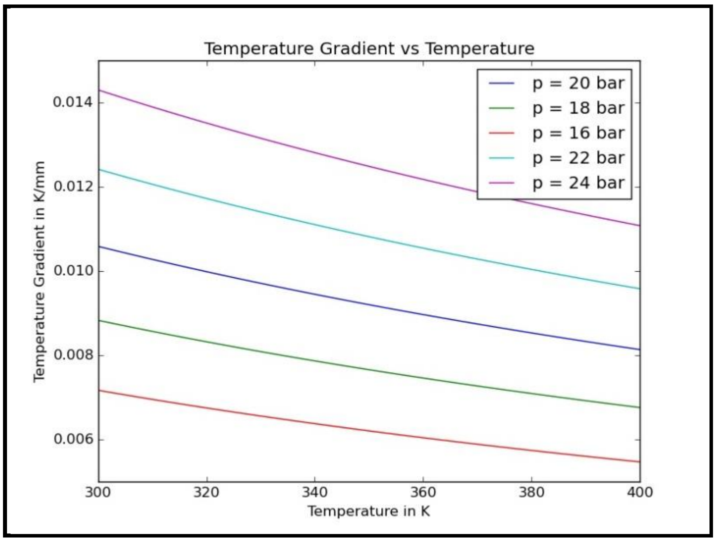

Fig.4 Temperature gradient vs Temperature

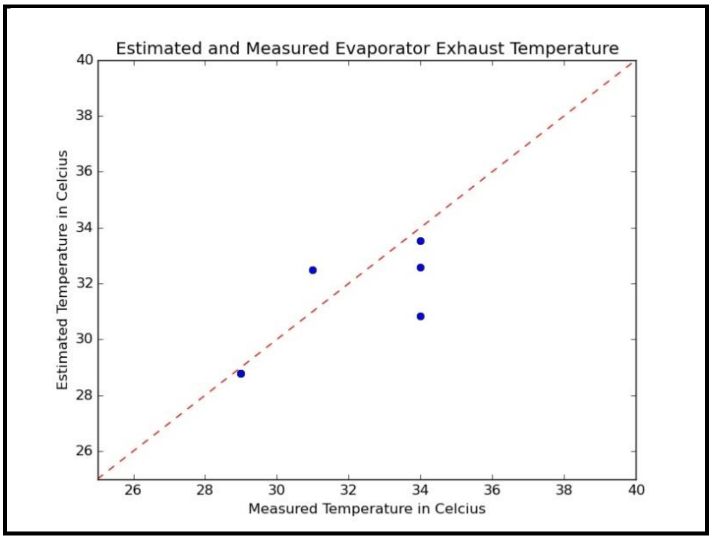

Fig.5 Estimated against measured evaporator exhaust temperature

Table.1 Temperatures and Pressures at Points of Interests

	Evaporator Suction		Compressor Suction		Compressor Exhaust	
S.No	Temperature (^{0}C)	Pressure psi	Temperature (^{0}C)	Pressure psi	Temperature (^{0}C)	Pressure psi
1	21	21	31	11	51	125
2	19	19	29	15	49	156
3	19	19	29	17	49	127
4	22	18	34	10	57	140
5	23	20	34	11	57	150
6	23	20	34	14	55	140

Table.2 Predicted and Measured Evaporator Temperatures

S.No	Evaporator Suction Temperature (^{0}C)	Evaporator Suction Pressure psi	Estimated Temperature at Evaporator Exhaust (^{0}C)	Measured Evaporator Temperature (^{0}C)	Error (%)
1	21	21	32.498	31	-4.832
2	19	19	28.785	29	0.741
3	19	19	28.785	29	0.741
4	22	18	30.840	34	1.294
5	23	20	33.541	34	1.350
6	23	20	32.572	34	4.200

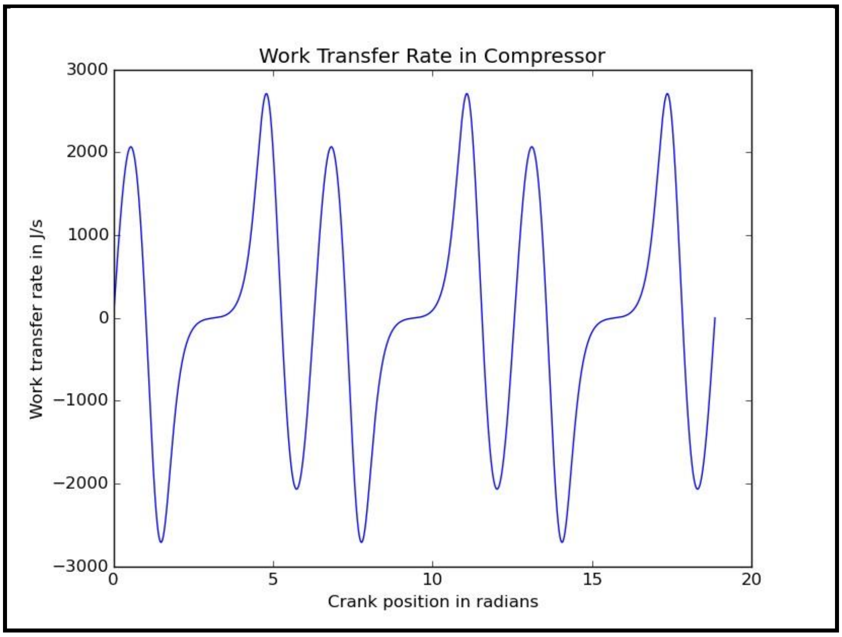

Fig.6 Work Transfer Rate in Compressor

Table.3 Heat and Work Calculated from Experimental Data

S.No	Heat Transfer (kJkg)	Work Transfer (kJkg)	Coefficient of Performance
1	9.811	4.594	2.135
2	9.047	4.586	1.972
3	8.802	4.695	1.874
4	11.248	6.211	1.810
5	10.534	5.042	2.091
6	10.196	4.571	2.230

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The results from Fig.4 and Fig.5 show that the condenser parameters largely influence the discharge pressure and the compressor mass flow rate, while the evaporator parameters mostly have an effect on the refrigerant mass flow rate and the heat transfer rates which was also concluded by Harms [7].It is to be noted that the work transfer rate is very dynamic and the frequency is too high such that its peak value can be considered to be its steady state value. Due its complex form it is to be estimated only from graphical data. In this case, the work transfer rate is determined to be 2698 J/s from the Fig.6. Table.3 shows the heat and work transfer calculated from the experimental data.Fig.6 depicts the work transfer in the compressor that will influence by the pressure drop between condenser and evaporator. The work of compression decreases due to decreased condenser heat capacity increases for same cooling capacity of the system. The error occurred in the case of evaporator exhaust temperature estimation it is ±10%.

3.2 Coefficient of Performance

The variation in coefficient of performance with respect to the rate of heat transfer in (kW) is calculated from the experimental results and validated against the mathematical model. The results be evidence and as the same trend provided Akintunde et al [8]. Table.4 shows the predicted COP against calculated COP and the data for validation are compared are shown.

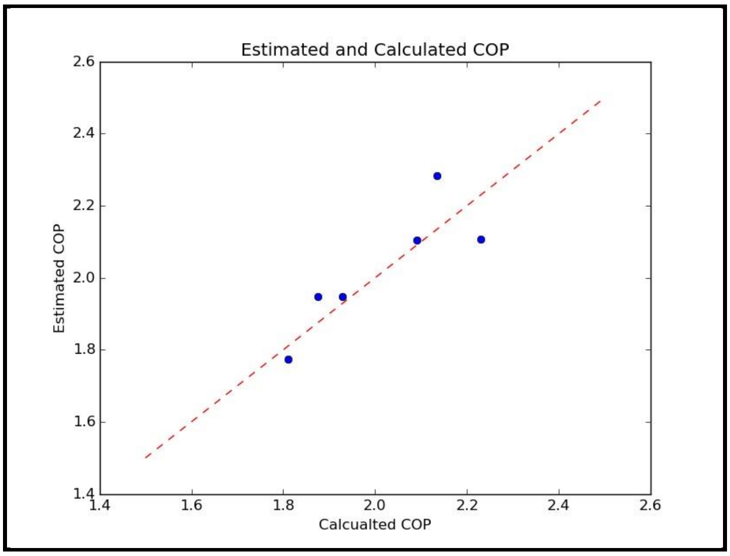

Fig.7 Estimated and Calculated COP

Table 4. Predicted COP against Calculated COP

S.No	Heat Transfer Rate (J/s)	Work Transfer Rate (J/s)	Coefficient of Performance Estimated	Observed	Error (%)
1	6160.9	2698	2.284	2.135	-6.978
2	5255.6	2698	1.948	1.928	-1.037
3	5255.6	2698	1.948	1.875	-3.893
4	4782.5	2698	1.773	1.810	2.044
5	5678.5	2698	2.105	2.091	-0.669
6	5678.5	2698	2.108	2.230	5.470

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The compressor work consumption reduction thus increases the COP. It is observable that the error in estimation of COP is ±7% and is well within the acceptable range when compared with the results of by Shurooq Talib Remedhan Al-Hemeri. Fig.7 depicts the estimated and calculated COP. The COP of the system increases due to an increase in the refrigeration capacity resulting from increases in the evaporator efficiency [10].

4. Conclusions

A mathematical model has been developed, analytic in nature in order to predict the evaporator exhaust temperature, the work transfer rate and COP of the system, given the evaporator suction temperature. The thus developed model when compared with experimental results, the model yields consistent results with an error band of ±10% and ±7% for temperature exhaust temperature and COP respectively. This model can be applied to any vapour compression refrigeration System to estimate its performance index. However, this model does not differentiate between the various parameters like volumetric efficiency and Viscosity of the refrigeration oil in the compressor for work transfer, and is based solely on the piston design and its angular velocity. Therefore, this model cannot be used to predict the changes when the compressor oil is changed.

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