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ANALIZA RADNOG UČINA R12 ALTERNATIVA U ADIJABATSKOJ KAPILARNOJ CIJEVI RASHLADNOG SUSTAVA
PERFORMANCE ANALYSIS OF R12 ALTERNATIVES IN ADIABATIC CAPILLARY TUBES OF A REFRIGERATION SYSTEM

Bukola Olalekan Bolaji

Sažetak: *U ovom radu su eklsperimentalno ispitane performanse zamjenskih radnih tvari u adijabatskoj kapilarnoj cijevi kompresijskog rashladnog uređaja. Maseni protok je određen za različite temperature kondenzacije, razne stupnjeve pothlađivanja i različite duljine kapilarnih cijevi. Prosječni maseni protoci R152a i R134a su bili redom* 1,2 % *niži i* 1,9 % *viši nego za R12 pri istim radnim uvjetima. Rashladni množilac (COP) postignut korištenjem radnih tvari R134a i R152 je bio vrlo blizu onome radne tvari R12 smanjen redom* 2,6 % *i* 1,3 % *dok su rashladni množioci postignuti korištenjem tvari R23, R32 i R143a bile znatno niže. Učinak rashladnog uređaja varira s odabranim radnim tvarima. Razlike su veće za R143a, R32 i R23, a odstupanja od učinka korištenjem R12 rastu tim redoslijedom, dok su za R134a i R152a zanemariva. Najbolji sveukupni učinak postignut je korištenjem radne tvari R152a.*

Ključne riječi:
– zamjenske radne tvari
– kapilarna cijev
– maseni protok
– rashladni sustav
– učinak

Abstract: *In this work, the performance of alternative refrigerants in an adiabatic capillary tube was investigated experimentally in a vapour compression refrigeration system. The mass flow rate was determined at a series of condensing temperatures, varying degree of sub-cooling, and at various lengths of capillary tube. The average mass flow rate of R152a and R134a were* 1.2 % *lower and* 1.9 % *higher than that of R12 respectively, under the same operating conditions. The coefficient of performance (COP) obtained using R134a and R152a refrigerants was very close to that of R12 with only* 2.6 % *and* 1.3 % *reduction, respectively, while the COPs obtained using R23, R32 and R143a were significantly very low. The performance obtained in a refrigeration system differs among individual selected alternative refrigerants. The differences are larger for R143a, R32 and R23, and the deviation from the performance of R12 is in that order, while the differences are smaller or negligible for R134a and R152a. The best overall performance is obtained using R152a.*

Keywords:
– alternative refrigerants
– capillary tube
– mass flow rate
– refrigeration system
– performance

1. INTRODUCTION

The chlorofluorocarbon (CFCs) and hydro-chlorofluorocarbon (HCFCs) refrigerants are being replaced by hydro-fluorocarbons (HFCs) due to environmental concerns about depletion of the earth's protective stratospheric ozone layer and global climate change. A refrigeration system using new alternative refrigerants must be modified or newly designed because the thermo-physical properties of these alternative refrigerants differ from those of conventional refrigerants. In order to maintain or improve the performance of the cycle, the operating characteristics of individual components of the system should be clarified for use with the new alternative refrigerants [1, 2]. Capillary tubes, short tube orifices and thermostatic expansion valves have been used in refrigerators and heat pumps as refrigerant flow regulating devices in vapour compression plants for several years [3]. Capillary tubes and short tube orifices are constant area expansion devices. In some cases, they can substitute more expensive and complex thermostatic valves [4, 5]. The principle of operation of the capillary tube is the flow resistance caused by a long, narrow tube, throttling the refrigerant pressure.

Pressure falls gradually as the liquid flows through the tube, until it starts to evaporate in the tube. This vapour formation, called "vapour lock", causes a sudden pressure drop in approximately the last quarter of the length, down to the evaporator pressure. The capillary tube is a simple drawn copper tube with an inner diameter ranging from 0.5 to 2.0 mm and a length ranging from 400 to 2500 mm. The flow inside the capillary tube is complex and pressure drop through the capillary tube has a strong influence on the performance of the whole system [4, 6].

The capillary tube has the advantages of simplicity, inexpensiveness, and the requirement of a low starting torque of the compressor [2]. However, reduction in cycle efficiency when load condition changes is one of the major disadvantages in adopting the capillary tube as an expansion device because the capillary tube does not have the function to actively adjust to this change. Practical redesign of the system by adopting a different capillary tube for alternative refrigerants requires the determination of the length and diameter of the capillary tube for a given refrigeration capacity and operating conditions [7, 8].

In the past, capillary tube performances for R22 alternatives were extensively studied by many researchers and recently, Kim et al. [2] investigated the performance of R22, R407C and R410A in several capillary tubes for air conditioners. Singh et al. [9] studied the performance of R134a flow through short tube orifices. Wongwises and Chan [5] studied the two-phase separated flow model of refrigerants flowing through capillary tubes. Sinpiboon and Wongwises [10] developed a numerical study of the refrigerant flow through non-adiabatic capillary tubes. A model of the capillary flow was represented by Garcia-Valladares et al. [11]. Their model represents the solution of the capillary flow with all four partial regions (two thermodynamic equilibrium and two meta-stable) in an adiabatic and non-adiabatic capillary tube.

Zhang and Ding [12] represented the approximate analytic solution of the capillary tube valuable for theoretical analysis and engineering calculation. Two solutions of adiabatic capillary behaviour were developed. The first solution is the explicit function for the capillary tube length prediction. The second solution is represented by the explicit function estimating the refrigerant mass flow rate. Bansal and Wang [13] presented a homogeneous simulation model for choked flow conditions for pure refrigerants (R134a and R600a) in adiabatic capillary tubes. Their model was based on the first principles of thermodynamics and fluid mechanics and some empirical relations.

However, experimental studies of performance of alternative refrigerants in the adiabatic capillary tube of the refrigeration system are seldom reported in available literature. Most of the studies reported are theoretical studies of capillary tube performance and models for calculating the mass flow rate through capillary tubes and short orifices. Therefore, the major aim of this study is to investigate experimentally the performance of alternative single-fluid refrigerants for R12 substitution in the vapour compression refrigeration system. Also in this study, the effect of refrigerant mass flow rate at various operating conditions and capillary tube length for different alternative refrigerants is analysed.

2. MATERIAL AND METHODS

2.1. Selection of Alternative Refrigerants

The range of possible alternative fluids is extensive; hydro-fluorocarbons (HFCs), refrigerant mixtures, and natural fluids. Among these groups of alternatives, HFCs and natural fluids are the most useful. According to Calm [14], possible environment-friendly refrigerants with zero ODP and lower global warming potential (GWP) could be selected from derivatives of methane and ethane. In this work, a full array of methane and ethane derivatives were considered and the trade-off in flammability, toxicity and chemical stability concerning atmospheric lifetime with changes in molecular chlorine, fluorine and hydrogen content were carried out [8]. Therefore, five single-fluid alternative refrigerants (R23, R32, R134a, R143a and R152a) that contain no chlorine were selected for the investigation.

2.2. Refrigeration System Performance

The p-h diagram shown in Figure 1 is frequently used in the analysis of the vapour compression refrigeration cycle.

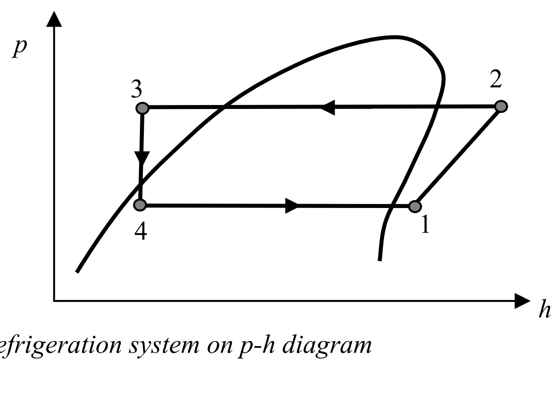

Figure 1. Vapour compression refrigeration system on p-h diagram

In the refrigeration system, the representative performance characteristics are compressor power (W_c, kW), refrigerating effect (Q_e, kW) and coefficient of performance (COP). The compressor power is expressed as:

$$W_c = m_r(h_2 - h_1) \tag{1}$$

where m_r = mass flow rate of refrigerant; (kg/s); h_1 = enthalpy of refrigerant at the inlet of compressor (kJ/kg); and h_2 = is the enthalpy of the refrigerant at the outlet of compressor (kJ/kg). The refrigerant mass flow rate is obtained using Eq. (2), assuming the isentropic compression to be[15]:

$$m_r = \left[1 + C - C\left(\frac{p_{dis}}{p_{suc}}\right)^{\frac{1}{\gamma}}\right] \cdot \frac{A \cdot N}{v_{suc}} \tag{2}$$

where C = compressor clearance volume ratio; p_{dis} = compressor discharge pressure (kN/m^2); p_{suc} = compressor suction pressure (kN/m^2); v_{suc} = specific index for the refrigerant under suction conditions. The refrigerating effect (kJ/s) is the heat absorbed by the refrigerant in the evaporator, given as: volume of the refrigerant at the compressor suction (m^3/kg); A = area of the cross-section of the compressor piston (m/s); and γ = isentropic

$$Q_e = m_r(h_1 - h_4) \tag{3}$$

where h_4 = enthalpy of the refrigerant at the inlet of the evaporator (kJ/kg).

From the first law of thermodynamic point of view, the measure of performance of the refrigeration cycle is the coefficient of performance (COP) and it is the refrigerating effect produced per unit of work required [16]. It is expressed as:

$$COP = \frac{Q_e}{W_c} \tag{4}$$

2.4 Experimental Set-up and Test Procedure

The test rig is an experimental apparatus of a vapour compression refrigeration system developed for drop-in tests for R12 substitutes. The schematic diagram of the system is shown in Figure 2.

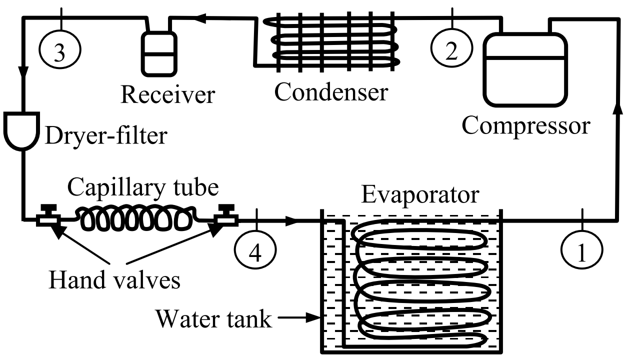

Figure 2. Experimental apparatus of vapour compression refrigeration system

Temperature and pressure measurements were made at the four different points indicated in Figure 2. Temperatures were measured with copper-constantan thermocouples with an accuracy of ± 0.1 ^{o}C. Pressures were measured with 24 V DC pressure transducers with an accuracy of ± 0.5 kPa. The energy consumption was measured with a watt-hour meter with an accuracy of ± 0.2 kWh.

The system was composed mainly of a hermetic compressor equipped with receiver, condenser, dry-filter, evaporator and exchangeable expansion device (capillary tube); polyester oil was used during the testing. In order to facilitate the capillary tube replacement, two hand valves were added to the refrigeration loop in the series between the ends of the liquid line and the evaporator inlet. The capillary tube length mounted between the two valves can be changed as needed. The system was designed based on a refrigerating capacity of 3.75 kW at a condensing temperature of 40 ^oC, evaporating temperature of 5 ^oC, and with R12 as a working fluid. The main components in the system are listed as follows:

(i) ***Compressor*****:** reciprocating type, power 0.746 kW, refrigerant R12, and swept volume 30.4 cm^3.

(ii) ***Condenser*****:** finned-tube, pipe inner diameter 6.8 mm, outer diameter 8.6 mm, tube length 8.4 m.

(iii) ***Evaporator*****:** bare coil, immersed in water being cooled, pipe inner diameter 6.8 mm, outer diameter 8.6 mm, tube length 6.6 m. Water tank 1.97 x 10^{-2} m^3.

(iv) ***Expansion device*****:** coiled capillary tube, pipe inner diameter 1.2 mm, outer diameter 1.9 mm, tube lengths 500, 750, 1000, 1250, and 1500 mm and coiled diameter 52 mm.

The system was tested with different lengths of capillary tubes under different conditions. The temperature difference across the evaporator and condenser coils was controlled to achieve the desired conditions. In order to minimize the heat loss to the surroundings, each capillary tube was insulated with layers of pipe insulation with a thermal conductivity of 0.39 W/m.K. This was necessary to ensure that the testing of the capillary tube was under adiabatic conditions.

Following each test, the system was drained and evacuated and charged with the appropriate refrigerant. Similar procedures were followed when testing the various capillary tube lengths. All baseline tests for R12 were performed as a reference for comparison purposes. The primary parameters observed during the course of this study were; pressure, temperature, and power consumption for the alternative refrigerants under investigation.

In order to obtain the effect of sub-cooling and condensing temperature on the mass flow rate, the refrigerant pressure at the capillary tube inlet was adjusted to the saturation pressure corresponding to the condensing temperatures of 40, 42, 44, 46 and 48 ^oC. The degree of sub-cooling at the capillary tube inlet was changed for each condensing temperature and was selected as 2, 4, 6, 8 and 10 ^oC for the condensing temperature of 44^oC and the capillary tube of a length of 1000 mm. The capillary tubes in most refrigeration systems are coiled, therefore, in this experimental study, five different lengths of 500, 750, 1000, 1250 and 1500 mm were coiled with the same coiled diameter of 52 mm. The influences of the length of capillary tube on the mass flow rate were determined for the condensing temperature of 44 ^oC and degree of sub-cooling of 6 ^oC. The evaporation temperature varies between temperatures of 6 and 8 ^oC (or 7 ± 1 ^oC) throughout the experiments, which prevented freezing of water in the evaporator compartment.

3. RESULTS AND DISCUSSION

Figure 3 represents the influence of the degree of sub-cooling on the mass flow rate of R12, R32, R23, R134a, R143a and R152a for the condensing temperature of 44 ^oC and a capillary tube length of 1000 mm.

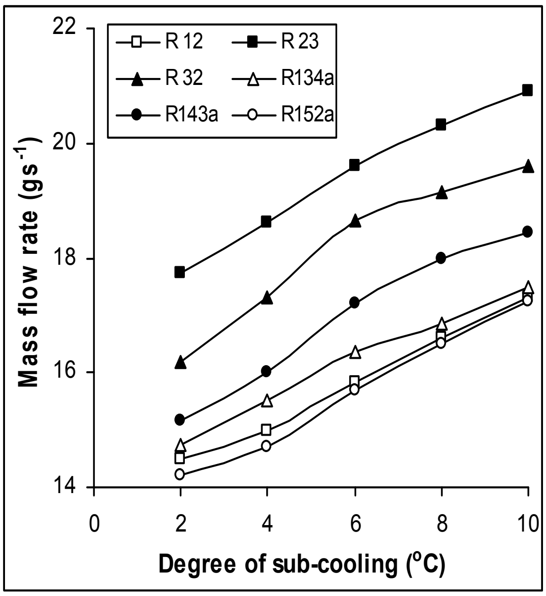

Figure 3. Effect of degree of sub-cooling on the mass flow rate of refrigerants

As shown in the figure 4, mass flow rate increases as the sub-cooling temperature at the capillary tube inlet increases, which implies a greater liquid portion in the capillary tube. This will retard the flashing of refrigerant and will reduce the mass quality of refrigerant vapour at the exit of the capillary tube; therefore, a low mass flow rate is desired. R152a offers the lowest mass flow rate.
Figure 4 shows the influence of condensing temperature on the mass flow rate of the selected alternative refrigerants for a degree of sub-cooling of 6 ^{o}C and a capillary tube length of 1000 mm.

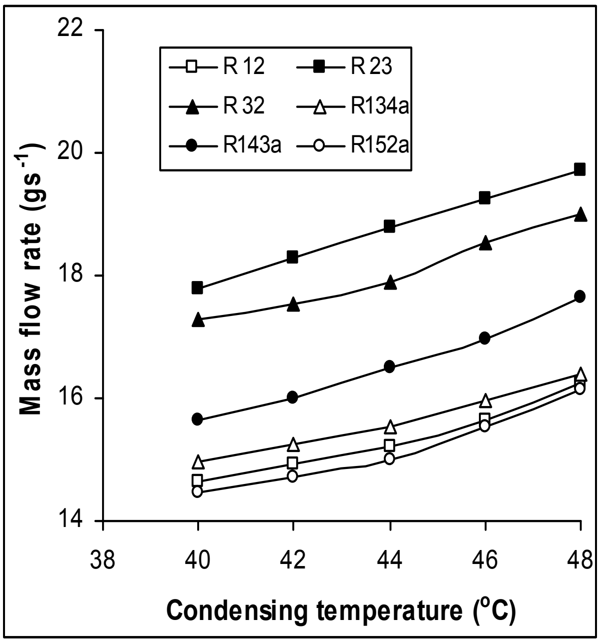

Figure 4. Effect of condensing temperature on the mass flow rate of refrigerants

As shown in this figure, a higher condensing temperature will increase the mass flow rate. An increase in the condensing temperature affects other components in the system; it increases the evaporating temperature and the refrigerant mass flow rate. R23 and R32 show the highest mass flow rate for a given condensing temperature mainly because of their higher vapour pressure as compared to others [17]. In this figure, there is not much difference between the mass flow rates of R152a, R134a and R12 refrigerants. The average mass flow rate of R152a and R134a are 1.2 % lower and 1.9 % higher than that of the R12 refrigerant, respectively. From these observations, it is clearly seen that the mass flow rate increases with an increase in the degree of sub-cooling and an increase in condensing temperature (or corresponding condensing pressure). These trends were also observed in previous research carried out with R12, R22 and R134a by other researchers [18-21].
Figure 5 represents the influence of the length of capillary tubes on the mass flow rates for a condensing temperature of 44 ^{o}C and a degree of sub cooling of 6^{o}C.

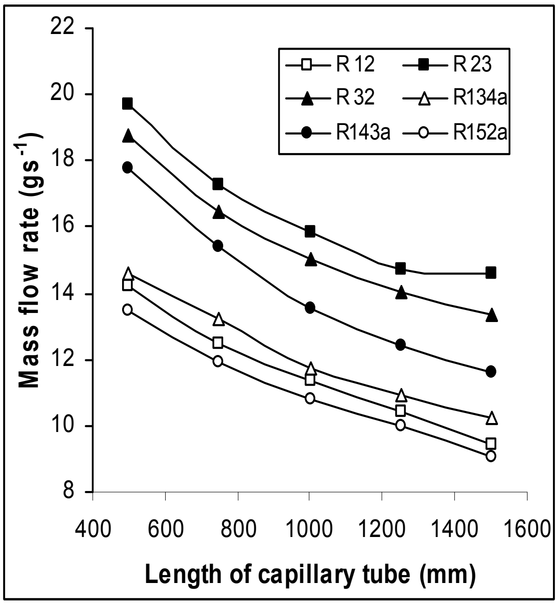

Figure 5. Effect of capillary tube length on the mass flow rate of refrigerants

As shown in this figure, the mass flow rate is approximately inversely proportional to the length of the capillary tube. An increase in the length of the capillary tube will increase the friction loss in the tube, which will reduce the mass flow rate of the refrigerant.

Figure 6 shows the comparison between the COPs of alternative refrigerants and R12.

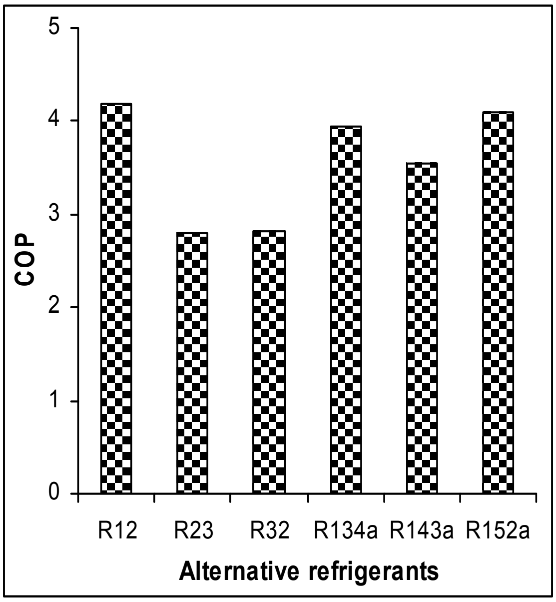

Figure 6. Coefficient of performance (COP) of alternative refrigerants

From the figure, it can be seen that the COP of all refrigerants was smaller than that of R12, but among the five refrigerants that are being considered as alternative to R12, two refrigerants (R134a and R152a) have a COP similar to that of R12, with only 2.6 % and 1.3 % reduction, respectively. The COP of the last three refrigerants (R32, R23 and R143a) were 39.3 %, 37.9 % and 17.8 % smaller than that of R12, respectively.

Figure 7 represents the overall heat transfer coefficient of the condenser versus the mass flow rate of refrigerants. The figure shows that the overall coefficient of heat transfer increases with a mass flow rate for all the refrigerants.

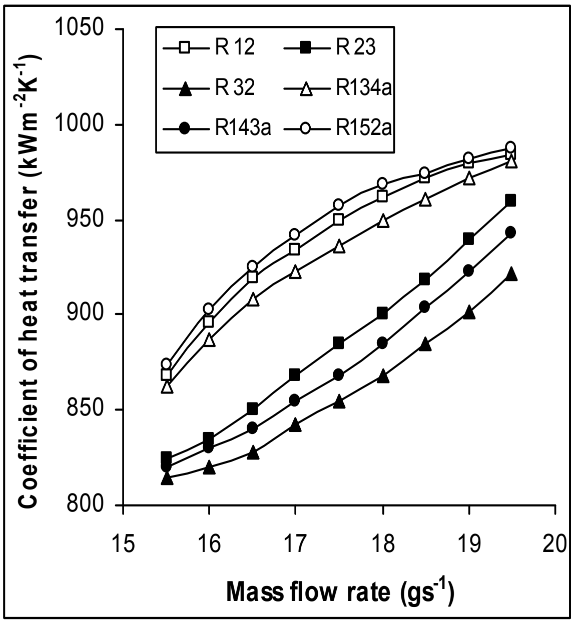

Figure 7. Effect of mass flow rate of refrigerants on overall coefficient of heat transfer of condenser

As shown in the figure, the increase in the coefficient of heat transfer for R12, R134a and R152a is greater than that of R143a, R32 and R23. R134a and R152a refrigerants have approximately the same performance with R12.

4. CONCLUSION

The phase-out of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) has resulted in the performance of chlorine-free, partially fluorinated hydrofluorocarbons (HFCs) being investigated. In this work, the performance of alternative refrigerants in the adiabatic capillary tube was investigated experimentally, with the aim of selecting single-fluid alternative refrigerants that have the same performance characteristics with R12 in refrigeration systems. A complete vapour compression refrigeration system was developed as an experimental apparatus. The system was tested with selected refrigerants (R23, R32, R134a, R143a and R152a) under different conditions. All baseline tests for R12 were performed as a reference for comparison. The mass flow rate was determined at a series of condensing temperatures, varying degree of subcooling, and at various lengths of capillary tube. The mass flow rate of R134a and R152a were 1.9 % higher and 1.2 % lower than that of R12, respectively, and under the same operating conditions. The COP of R12 was higher than all of the selected alternatives. Only R134a and R152a refrigerants have COPs that were very close to that of R12 with only 1.6 % and 1.3 % reduction, respectively. In conclusion, the performance obtained in a refrigeration system differs among individually selected alternative refrigerants. The differences are larger for R143a, R32 and R23, and the deviation from the properties of R12 is in that order, while the differences are smaller or negligible for R134a and R152a. As a result, R134a and R152a refrigerants are good substitutes for R12 in the vapour compression refrigeration system. The best performance was obtained from the use of R152a in the system.

5. LIST OF SYMBOLS

area	A,	m^2
coefficient of performance	COP,	-
compressor clearance volume ratio	C,	-
compressor discharge pressure	p_{dis},	kN/m^2
compressor suction pressure	p_{suc},	kN/m^2
compressor speed	N,	m/s
compressor work input	W_c,	kJ/s
enthalpy at the inlet of compressor	h_1,	kJ/kg
enthalpy at the outlet of compressor	h_2,	kJ/kg
enthalpy at the inlet of evaporator	h_4,	kJ/kg
isentropic index	γ,	-
mass flow rate of refrigerant	m_r,	kg/s
refrigerating effect	Q_e,	kJ/s
specif. volume at compr. suction	v_{suc}	m^3/kg

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Technical note

Adresa autora / Author's address
Bukola Olalekan Bolaji
Department of Mechanical Engineering,
University of Agriculture,
P.M.B. 2240, Abeokuta, Nigeria
bobbolaji2007@yahoo.com