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DEVELOPMENT OF A NOVEL CONTROL STRATEGY FOR A MULTIPLE-CIRCUIT ROOF-TOP BUS AIR-CONDITIONING SYSTEM IN HOT HUMID COUNTRIES

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ABSTRACT

A novel control strategy to improve energy efficiency and to enhance passengers' thermal comfort of a new roof-top bus multiple-circuit air-conditioning (AC) system operating on partial load conditions is presented. The new multiple-circuit AC system consists of two evaporators, two condensers, two expansion valves, and two equal capacity compressors. Each one of the evaporators is split up into two halves, one-half of the evaporator coil of one circuit pairing with one-half of the other circuit. A novel strategy for an automatic control of the AC system was developed based on numerous experimental test runs at various operating conditions, taking into account energy saving and thermal comfort without sacrificing the proper cycling rate of the system compressor. For this task, more than 50 test runs were conducted at different set-point temperatures of 21, 22, and 23°C. Fanger's method was used to evaluate the passenger thermal comfort and the system energy consumption was also calculated. A performance comparison between that of the conventional AC system and the newly developed one has been conducted. The comparison revealed that the adopted control strategy introduces significant improvements in terms of thermal comfort and energy saving on various partial load conditions, with potential energy saving of up to 30.8% could be achieved. This results in a short payback period of 19 months. It was found from the economic analysis that the new system is able to save approximately 19.7% of the life cycle cost.

Keywords: Roof-top bus air-conditioning, Thermal comfort, Steady-state, Multiple-circuit system, Energy saving, Costing.

INTRODUCTION

The use of a roof-top passenger vehicle (buses or trains) air-conditioning (AC) system has been steadily growing in countries experiencing hot and humid tropical climate such as those in the south East Asia. It is quite common to see a bus AC compressor driven by an auxiliary engine to ensure a constant speed operation. Still, there are many that are belt-driven by the main engine. An AC system is the second biggest energy consumer component in either intercity or city buses. If the AC system is driven by the main engine, the driver can easily feel the drop in the vehicle power when the electromagnetic clutch is engaged. It is established through experience that the performance of an AC system is influenced by the variation of the daily cooling load which is dependent on the conditions outside and inside the bus. For instance, opening of a door, changing of sun load through the windshield and side glass windows, and number of passengers on board will change the load inside the bus cabin. A computer code has been developed by the authors according to ASHRAE 1997 to estimate the cooling load variation for tropical countries (see Figure 1).

As shown in Figure 1, the peak load occurred between 10.00 am to 3.00 pm and this is commanding the AC system to operate at maximum capacity. However, at other times when the system experiences partial load conditions (low sensible heat load), especially at night or when it rains, it still operates at maximum capacity subjecting the passengers to an uncomfortably cold condition. That is due to the absence of any provision to modulate the system capacity to match the drastic reduction in the imposed cooling load. On the other hand, AC systems are often over-designed first to ensure a fast response so that the cabin temperature drops quickly when the system is switched on and second to overcome the irregular and rare conditions of extremely high humidity and high atmospheric temperature. Thus, under normal conditions of low cooling load, a lot of the energy is unnecessarily wasted [1] and this also results in higher consumption of fuel. In 1998, Barbusse et al. [2] pointed out in their research that, under the environmental temperature of 30°C, the demand is for 3.1 liters of fuel for each 100 km journey, and increased to 3.8 liters at 40°C. The overall fuel consumption rate is reduced by a maximum of 20% in a vehicle driven without air conditioning.

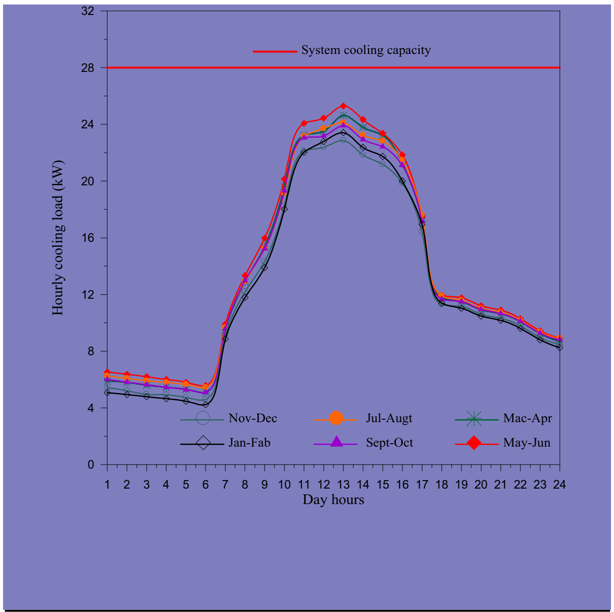

Figure 1:Hourly cooling load distribution at different typical day times around year

In addition, Global petroleum energy is estimated to be depleted in about 40 years and the recent instability in oil price is much related to the rapid development of the transportation and industrial sectors [3]. For this reason, researchers and AC system manufacturers seek to improve automobile AC systems design and technology to reduce the fuel consumption rate without forfeitting passenger thermal comfort.

Most AC systems for small scale vehicles have a thermostat control which enables the compressor to be disengaged from the vehicle engine when the comfortable set-point temperature is reached and reengaged when the cabin temperature increases. Unlike in large buses AC systems, this technology poses several drawbacks. Since the volume capacity of a bus compressor is quite large compared to that for AC system in small vehicles the compressor experiences a high inertia torque every time when it is started. If this on/off action is randomly and frequently repeated during a short interval, the system will undergo erratic behavior that will certainly get the compressor to break down quickly [4].

In practice, most of the conventional bus AC systems have no thermostat installed causing the compressor to work continuously at maximum capacity without any attention given to energy consumption or overcooling of the passenger compartment during partial load conditions. Thus, there is a need to develop an efficient bus AC system featured by economical operation in terms of energy saving and stable passenger thermal comfort at all atmospheric conditions.

Multiple-circuits system (MCS) offers an attractive solution for the bus AC system problems. In such systems more than one unit can be used, each unit consists of an independent compressor, condenser, and expansion valve, and shared evaporator. Each two or more units can share the evaporator face area; this is called face-to-face evaporator control. The main advantages of MCS are of its simplicity in installation and maintenance as well as the potential to conserve energy. However, to reap these benefits of the MCS an automatic controller should be designed properly and provided with excellent control strategy.

It is unfortunate that the literature regarding automotive AC systems (AACS) particularly bus AC system is scarce due to the proprietary control of the knowledge and the rapidly increasing competition between automotive AC OEMs. A literature survey has revealed that indeed this perception is quite opportune. Conceicao et al. [5, 6] has developed an experimental AC test rig for a full scale bus to characterize the conditioned air flow around passengers and to evaluate the thermal comfort conditions perceived by passengers. In addition, the indoor air quality and air exchange rate in the passengers' compartment of an intercity bus was also studied. Jung et al. [7] studied the thermodynamic performance of supplementary/retrofit refrigerant mixtures for R12 automotive AC systems produced before 1995. Lee and Yoo [8] conducted performance analyses of the components of an automotive AC system and developed an integrated model to simulate the entire system. Ratts and Brown [9] experimentally analyzed the effect of refrigerant charge level on the performance of an automotive AC system. Al-Rabghi and Niyaz [10] retrofitted an R12 automotive AC system to use R134a and compared the coefficients of performance (COP) for the two refrigerants. Jabardo et al. [11] developed a steady-state computer simulation model for an automotive AC system with a variable capacity compressor and investigated its validity on an experimental unit. Joudi et al. [12] presented a computer model simulating the performance of an ideal automotive AC system working with several refrigerants. Kaynakli and Horuz [13] analyzed the experimental performance of an automotive AC system using R134a in order to find the optimum operating conditions. Figure 2 shows the schematic diagram of a conventional bus AC system which comprises of two cooling coils to provide the conditioned air to both rows of the passenger compartment, one compressor, a receiver drier, two condensers, and a thermostatic expansion valve. It should be noted that the two condensers are connected in series to allow the condensation process to occur in two stages whereas the evaporators are connected in parallel to allow equal cooling rate to both passenger rows.

The objective of this work is to develop a control strategy for an automatic controller to enable the new multiple-circuit AC system to respond to the variations of the cooling load imposed. The organization of this paper is accomplished as follows: a discussion of alternative control strategy methods is demonstrated, a description of the conventional and developed systems and the control strategy are presented and the experimental test rig and test procedures are described. This is followed by a discussion on the development of the analytical model which predicts the thermal comfort. Finally, a comparison between the developed system and conventional one in terms of the prospective energy savings, thermal comfort achievement, and costing is carried out.

DISCUSSION OF ALTERNATIVE CONTROL STRATEGY METHODS

In general, mobile air-conditioning systems have two basic controls objectives. The first is to maintain the cabin at the comfort conditions. The second is to prevent frost formation on the evaporator. Clutch cycling (or on-off cycling) is the most common method whereas variable-displacement compressors and electronic expansion valve are less common. All of these control methods are also applicable to stationary systems. Another control strategy used in some prototype stationary systems is a variable-speed compressor. This strategy is not practical for mobile systems [14]. The high initial cost of the variable-speed compressor and complexity of its installation have not attracted consumers as many as the manufactures expected. In addition, the long payback period for the initial cost often makes the customers reluctant to use it. Like the variable speed compressor, the electronic expansion valve (EEV) is characterized by a high expensive capital cost. The electronic expansion valve operates with a much more sophisticated control design. EEV controls the flow of refrigerant entering a direct expansion evaporator in response to signals sent to it by an automatic controller. A small stepper motor controlled by the microprocessor is used to open and close the valve port. The electronic signals sent by the microprocessor to the EEV are usually done by a thermistor connected to discharge airflow in the refrigerated case. EEV can perform effectively when installed with the variable speed compressor, causing amplification in the system capital cost, as well the cost of the EEV controller and sophisticated sensors. In addition, if the current supply to the controller is cut off suddenly during system operation, the compressor is apt to the floodback phenomenon (refrigerant liquid enters the compressor) because the EEV retain its position of opening port as dictated by the stepper motor. The refrigerant liquid floods the evaporator and the compressor and unless there is a solenoid valve prior to the EEV, the compressor will likely to get damaged rapidly.

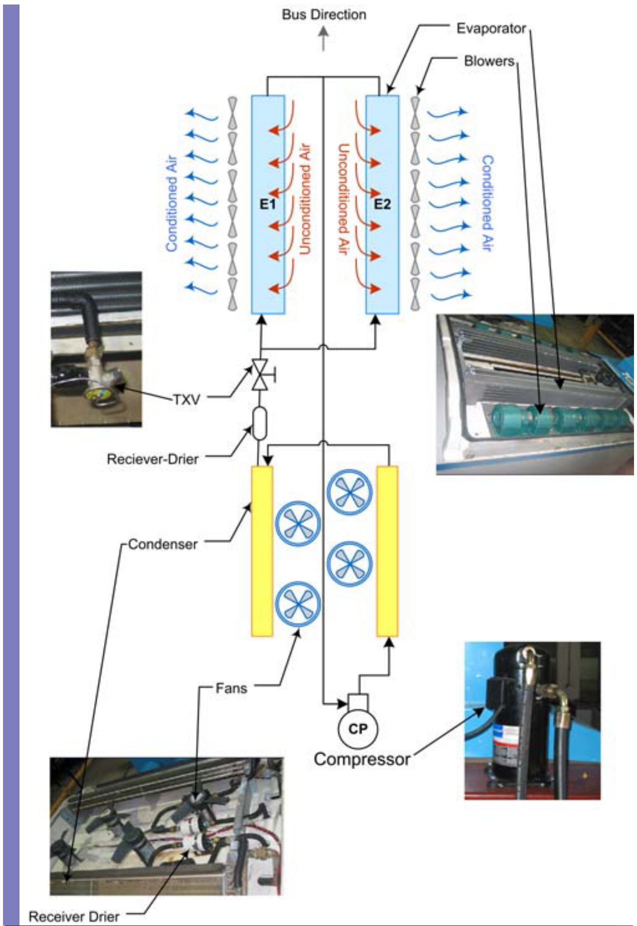

Figure 2: Schematic diagram of a conventional bus air-conditioning system

A variable-displacement compressor (VDC) changes its piston stroke length (or wobble plate angle), and consequently, the AC system will match the vehicle cooling demand. VDCs have the advantages of running continuously while responding to cooling load variation. However, these compressors are more complex than constant displacement compressors and can be less reliable. Additionally, it promotes the system to have a tendency to cause hunting phenomenon particularly when the condensing pressure decreases excessively away from its design value [15] which happens at low ambient temperature (e.g. at cloudy whether, at night or during rain time). The hunting phenomenon is a term used to identify the erratic system behaviour or instability due to an abrupt or sudden change in refrigerant flow.

DESCRIPTION OF THE AC SYSTEM AND CONTROL STRATEGY

The system used for the experimental work is a prototype version of a bus AC system with a total nominal cooling capacity of 28 kW. The refrigerant used in the AC system is R134a. The system components were mounted on a frame in a horizontal orientation, quite similar to the normal orientation of a roof-top bus AC system. The system is made up of two identical refrigerant circuits. Each circuit consists of a scroll compressor (of half the capacity of a conventional bus AC compressor), finned-tube condenser, an internally equalized thermostatic expansion valve, a liquid receiver/drier, and a finned tube evaporator divided into two sections as shown in Figure 3. The main advantages for using two identical units are the reliability and flexibility in term of control and if one unit becomes inoperative, the other unit can rather hold the load and supply some cooled air to passengers until the damaged unit is repaired. In this study, the control elements for the developed system are the number of compressors and evaporator blowers. The developed bus AC system has four different operating modes; all system off and only evaporator blowers on (0^{th} mode), two compressors running with full blowers capacity (1^{st} mode), one compressor running with full blowers capacity (2^{nd} mode), and one compressor running with partial blowers' capacity (3^{rd} mode).

The concept of the control strategy relies on the chronological operation between the control actuators to adapt the system capacity to match the imposed cooling load. The criterion of this configuration is the achievement of energy savings and thermal comfort without impairing the durability of the system compressors. The philosophy of the novel strategy is to express the imposed cooling load in the passenger cabin by the temperature difference (TD) between the evaporator inlet air temperature and the set-point temperature. This will bring the automatic controller to a more reliable, strict, and efficient cooling load recognition, which will lead the system to a faster response to the changing thermal loads.

DESCRIPTION OF THE EXPERIMENTAL SET-UP

The experimental set-up shown in Figure 4 consists of original components from a bus AC system, arranged in a way that emulates those in the actual bus. It should be noted that the two refrigeration circuits are identical and hence only one circuit is shown in the schematic diagram of Figure 4. The geometrical configurations for both condenser and evaporator are listed in Table 1. Both the evaporator and the condenser were mounted above the simulated passengers' compartment. In order to simulate the cooling load imposed on the passengers' compartment, an electric heater was immersed in the main air duct upstream to the cooling coils. The evaporator inlet air temperature can be controlled by the electric heater to achieve the sensible cooling load while the latent load was obtained by mixing streams of external air with cooled air from the coil, as indicated in Figure 4. The compressor was driven by a three phase 4.5 kW electric motor with a constant rotational speed of 2900 rpm to simulate the task of the auxiliary engine. The individual evaporator coil was serviced by six blowers (each pair of the blowers is connected together by one motor) while the condenser coolant air was provided by two centrifugal fans. Both the evaporator blowers and the condenser fans motors were energized by a direct current power source of 24 volt. Flexible rubber hoses were used to connect the evaporator and condenser with the compressor. The air ducts and passengers compartment were insulated by polyurethane foam with a thickness of 5 cm. The refrigerant lines of the system were made from copper tubing and insulated by an elastomeric material. Temperature, pressure, mass flow rate, and relative humidity were measured at the locations indicated in Figure 4.

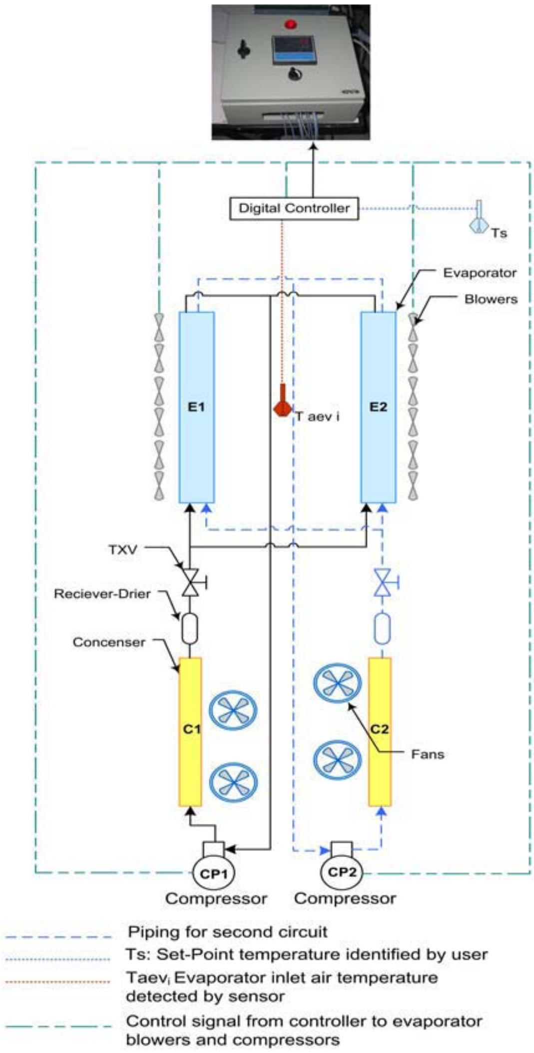

Figure 3: Schematic diagram of the developed multiple-circuit bus air-conditioning

The refrigerant and the air temperatures at various points of the system were monitored by K type thermocouples. The thermocouples for the refrigerant temperature were inserted inside the copper tubes. The dry bulb temperature and relative humidity of the air stream at the inlet and the outlet of the evaporator were measured. The thermocouple which is responsible of measuring of the evaporator inlet air temperature was insulated and shielded with balanced, low-pass filtered differential amplifiers (to avoid noise contamination and precision error). The wall interior temperatures of the simulated passengers' compartment were also measured in order to determine the mean radiant temperature inside the passengers' compartment. The Pressures at various points of the refrigerant circuit were measured by Bourdon tube gauges. The refrigerant mass flow was measured by using a refrigerant flow meter for R-134a. The mass flow rate of air passing over the evaporator was determined by measuring the average air velocity in the return air duct using an anemometer. The density of the air at the evaporator inlet was determined with the help of the dry bulb temperature and relative humidity measurements and evaluating them in the continuity equation along with the flow area of the duct. The evaporator inlet and exit air relative humidity were also measure by a digital hygrometer. The electric signal generated by the transducers (thermocouples or the humidity and velocity sensors) were logged by a data acquisition system which is connected to a PC as shown in Figure 4. Some features of the instrumentation are summarized in Table 2.

Table1:The condenser and evaporator characteristics

Parameters	**Condenser**	**Evaporator**
Tube Spacing (mm×mm)	38.1×31.75	25.4×22.23
Tube inner diameter (mm)	15.67	8.865
Tube outer diameter (mm)	15.875	9.525
Height (mm)	457.2	177.8
Length (mm)	1250	1425
Fin pitch (fin/mm)	0.512	0.551
Number of rows	4	6
Number of circuits	6	3
Number of tubes per row	12	8
Fin thickness (mm)	0.15	0.15
Face air velocity (m/s)	2.5	2.25
Saturation temperature (°C)	55	10

Table 2: Characteristics of some of the experimental set-up components

Measured variable	**Instrument**	Instrument range	Uncertainty
Temperature	Type K thermocouple	-50/100 °C	0.3°C
Pressure	Bourdon gauge	0/1000, 10/2500 kPa	10/50 kPa
Air flow rate	velocity meter	0.1/20 m/s	±1%
Refrigerant flow rate	rotameter	10/110 g/s	+1%
Relative humidity	Digital hygrometer	0/100 %	±1%

DESCRIPTION OF THE TESTING PROCEDURE

The experimental work consists of a series of experiments conducted at three set –point temperatures of 21°C, 22°C, and 23°C. These temperatures are selected based on the typical temperature of a conditioned space. The aim of these experiments is to provide a reference data for the control strategy design. These experiments were conducted and controlled manually, with the controller disengaged, according to the following procedure:

1. Select the set-point temperature.
2. Heat the inlet evaporator air.
3. Turn on the developed system with a specified operating mode.
4. Record the required data and readings.
5. Determine the cycling on time taken for the inlet evaporator air temperature to decrease to the set-point temperature and the cycling off time.

Calculate the energy consumption and passenger thermal comfort indexes.

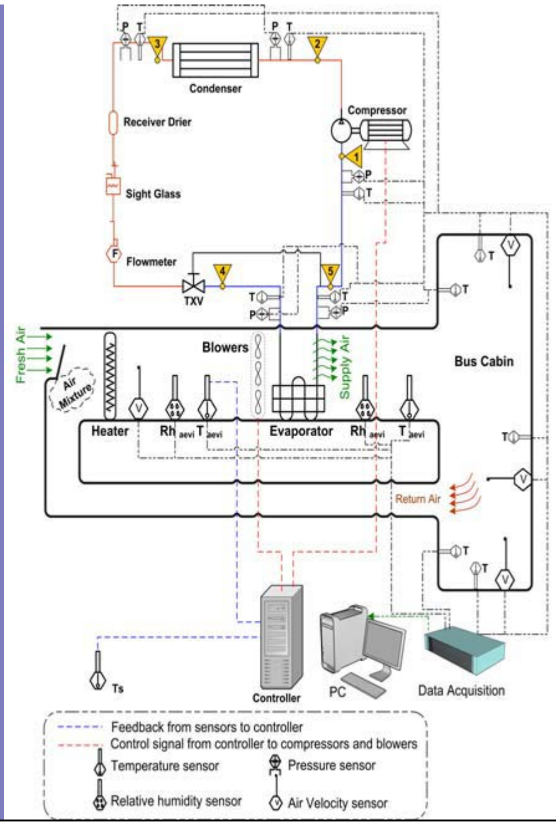

Figure 4: Schematic diagram of the experimental apparatus

ENERGY CONSUMPTION ANALYSIS

The evaporator inlet air temperature for the bus AC system is subject to variation interval of one hour as a result of the change in thermal load conditions. Thus, the quasi-steady-state concept will be employed here in order to calculate the energy consumption. That because the dynamic response of most systems is much more rapid than the one-hour time step used in thermal load calculations [16]. Quasi-steady-state refers to the steady-state operation at a certain hour. As a result, the energy consumption can be calculated cumulatively from the start of the system using this relationship:

$$E_{n+1} = E_n + \frac{\overline{P}_n . \Delta t}{60} \times N_{cy} \qquad (1)$$

Where,

Δt = running or on time of the compressor in minutes during one-hour step.

$\overline{P}_n$ = average system power input during the running time, kW

N_{cy} = number of compressor cycles per hour

At n = 0 (the starting of the system), E_0 = 0, N_{cy} is determined by dividing 60 minutes (one hour) over the sum of the running and off times. The consumed power input P_n can be calculated from the following formula:

$$P_n = \sqrt{3}\ \text{V.I.}\cos\phi$$

The average of the power consumption during cycle running time is used to determine the average input power. The daily energy consumption for both systems (conventional and developed system) is shown in Figures 5 and 6. It should be noted that the conventional system has a continuous compressor operation resulting in constant energy consumption for any temperature setting as shown in Figure 5. As seen from this Figure, the conventional system has the largest value of the daily energy consumption. As expected, if the developed AC system operates on the third mode during all operation time, the system will consume the least energy. However, the passenger thermal comfort will be affected as a result of insufficient cooling of the AC system. The selection of the appropriate system operating mode by considering the minimum energy consumption and thermal comfort for the developed AC system operation will be discussed in the next sections.

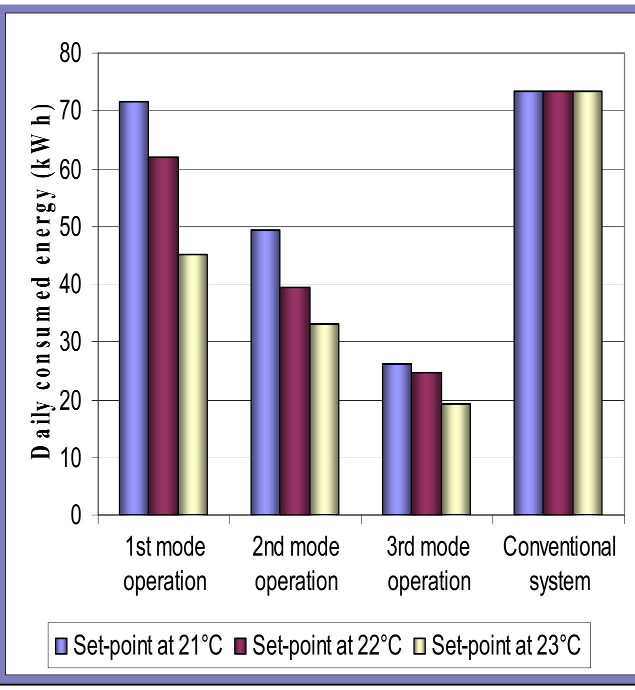

Figure 5: Daily energy consumption for both systems

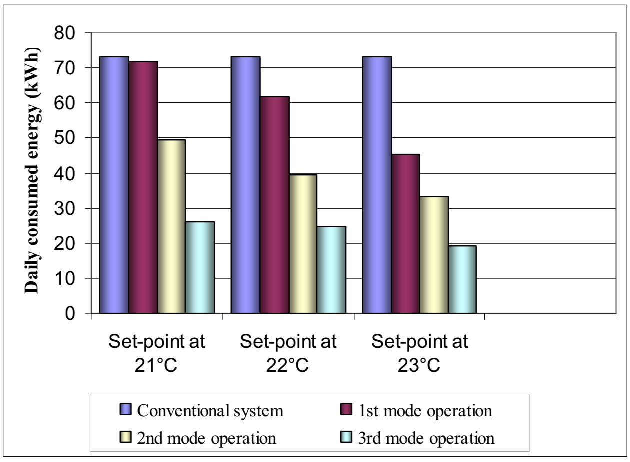

Figure 6: Daily energy consumption for both systems at different set-point temperatures

THERMAL COMFORT ANALYSIS

The human sense of thermal comfort is very complex. It relates both the physiological and the psychological states of a person under specific environmental conditions. To quantify thermal comfort in fabricated enclosures such as houses, offices, cars and ships, Fanger [17] suggested that body activity, thermal resistance of clothing, air temperature, mean radiant temperature, relative air velocity, and humidity are the most important variables and are taken into account in his proposed thermal comfort equation, in terms of "Predicted Mean Vote (PMV)". The details of this equation development are given by Kumar in [18]. While the thermal comfort equation provides the general degree of discomfort for a group of people, thermal sensation varies from one individual to another. To quantify the thermal comfort, it is more reasonable and practical to indicate the percentage of persons who can be expected to be thermally comfortable. Therefore, in the present study, the "Predicted Percentage of Dissatisfied (PPD)" was used to evaluate thermal comfort. These indices were derived from research studies carried out with a large number of human subjects inside a climatic chamber who were subjected to different climatic conditions and asked to express their thermal sensations on a conventional seven-point scale (−3 = cold, −2 = cool, −1 = slightly cool, 0 = neutral, 1 = slightly warm, 2 = warm, 3 = hot). The environment is considered comfortable if $-0.5 < PMV < 0.5$, which implies $PPD < 10\%$. The most important parameters which specify the degree of thermal comfort are the space or interior air temperature and the relative humidity. The computation of space air temperature and relative humidity requires the knowledge of the exit air psychometric conditions from the AC system and solving the three coupled governing equations. These equations are the dry air mass balance, vapour mass balance, and air energy balance. The experimental work provides the analytical model of the thermal comfort calculation by providing the exit air psychometric properties for the three different operating modes of the system. The analytical model then computes the space air temperature and relative humidity. The results of the analytical model are presented in Figure 7. The procedure for solving the three governing equations is detailed by Chi-Chang in [19]. The other parameters such as the average air velocity and mean temperature were also determined. The average air velocity was measured inside the simulated passengers' compartment while the mean radiant temperature was determined using the method which is described by ASHRAE in [20]. In this study, passengers are defined as resting quietly with "causal" clothing. This produces a metabolic rate of 1 met and a clothing insulation of 0.8 clo.

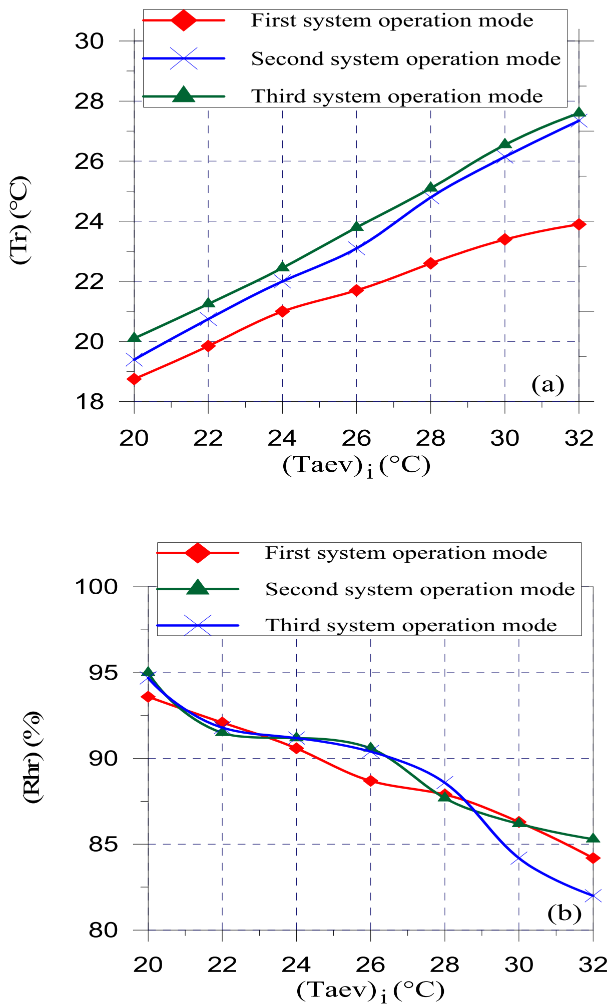

Figure 7: Effect of evaporator inlet air temperature on: (a) space air temperature, and (b) space air relative humidity at the three different modes of the newly developed AC system

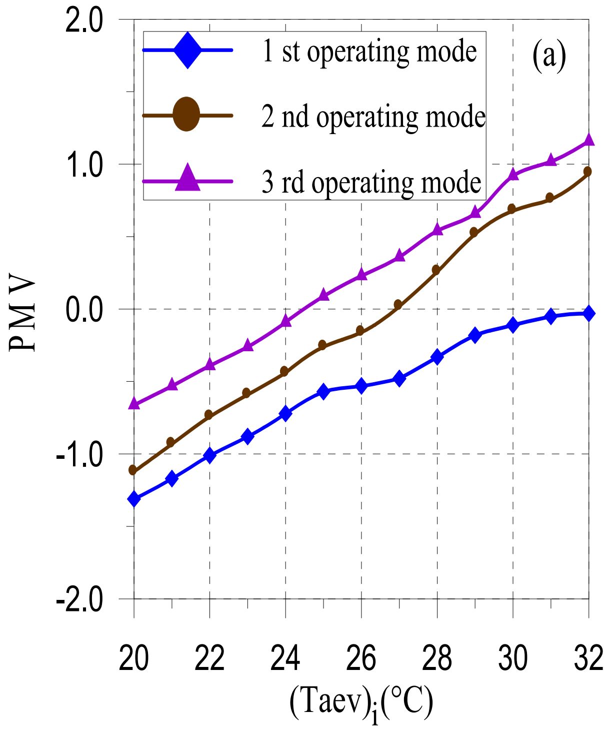

Figure 8: Evaporator inlet air temperature effect on passenger thermal comfort in terms of (a) PMV (b) PPD

Figure 8 shows the degree of thermal comfort in terms of PMV and PDD for the three different modes of the newly developed AC system. It can be seen that the system in the first mode operation is capable of providing a thermal comfort condition inside the bus compartment only above the evaporator inlet air temperature of 27°C. Below 27°, the conditions inside the compartment is characterized by overcooling causing an increase in the PPD. Alternatively, the operation of the system in the second mode at temperature range below 27°C and above 23°C is characterized by a relatively achievement for the thermal comfort condition inside the compartment. As for the third mode operation, the region below 24°C indicates a comfortable condition inside the bus cabin. Therefore, the temperature of 27°C is the key point of switching the system from the first mode to other modes and vice versa. However, it is observed that operating the system in the second mode at a temperature below 24°C results in a PMV value that is away from the comfortable zone (-0.5 <= PMV <0). Accordingly, the temperature of 24°C is the border line between the second mode and the third one. This is the general rule for creating the control strategy but there is another decision variable that governs the design of the control strategy which is the number of cycling per hour for the system compressors. This depends on the difference between the evaporator inlet air temperature and the set-point temperature as well as the system capacity. The sum of cycling on and off times is called cycling rate and the common cycling rate recommended by compressor manufacturers' for the large compressor capacity is 3 or 4 cycles/hour. More than 50 test runs were conducted at the set-point temperatures of 21, 22 and 23°C to determine the proper control strategy which satisfies the recommended cycling rate. The control strategy is plotted in Figure 9. The principle of the control strategy is based on the temperature difference (TD) between the set-point temperature, which is determined by the controller user (usually the bus driver) and the value of indoor air temperature, which is detected by the controller temperature sensor.

CONTROL STRATEGY

When the bus driver turns on the controller button during start up, one compressor will commence to work with all blowers operating. The driver then selects a set-point temperature. The controller detects the evaporator inlet air temperature in milliseconds and by comparing the inlet evaporator air temperature and set-point temperature, the controller can recognize the cooling load and proceed with the decision making procedure. If the second compressor engagement is essential to cover the imposed cooling load, it will be on within one minute. The decision making process undertaken by the controller include the evaporator blowers as well; one third of the blowers is equipped to be independently off of the system, while all-blowers-on operation can be executed during 0^{th}, 1^{st}, and 2^{nd} operating modes. The control strategy shown in figure 9 presents a distinct scenario for different set-point temperatures, and for different evaporator inlet air temperature. As an example for how the control strategy works; for a cooling load represented by a set-point temperature of 21°C and DT <4 (i.e. T_{evai} <25 °C) the controller shall keep one compressor on and one third of the blowers off (3^{rd} mode). However, if a cooling load represented by 4≤TD<6 and the same set-point temperature of 21°C is recognized by the controller, all the evaporator blowers will be on while still keeping one compressor off (2^{nd} mode). If the cooling load imposed exceeds the representation of TD<6, the system will work with full capacity (1^{st} mode). On the best cases, when the evaporator inlet air temperature equals the set-point temperature, all the system will be off, and the evaporator blowers will be on (0^{th} mode). The TD values are selected precisely to satisfy proper cycling rate for both compressors and also to avoid short-interval cycling.

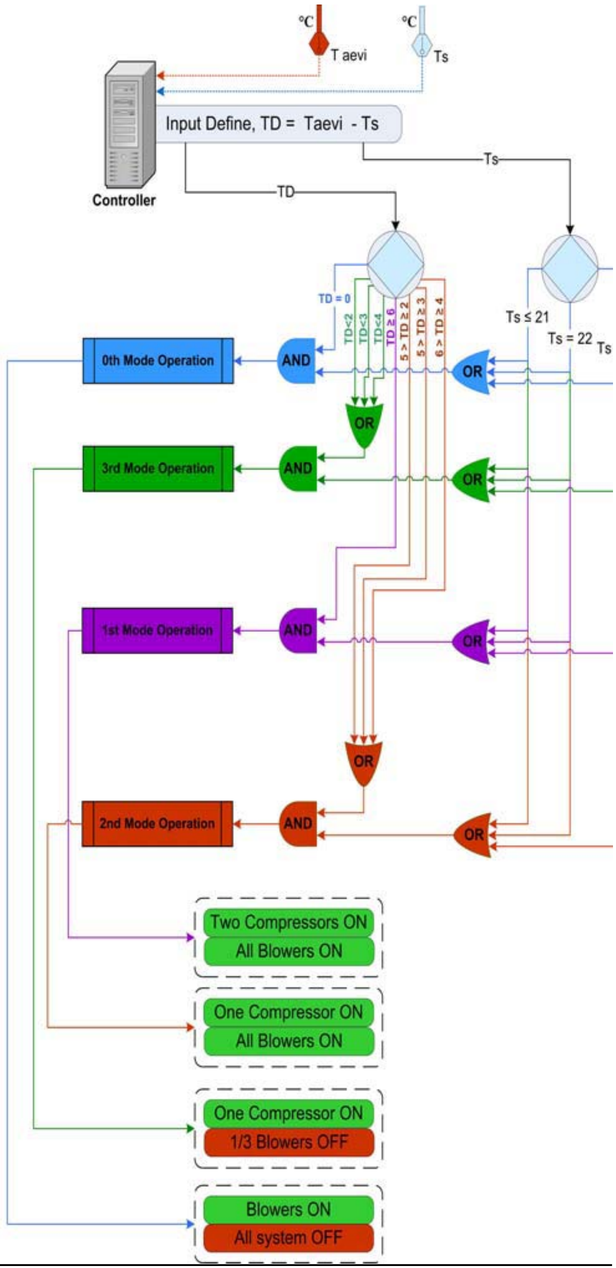

Figure 9: Control strategy

POSSIBLE ENERGY SAVINGS

This section demonstrates possible energy savings when the developed system is used. The daily usage of a bus AC system in a tropical country such as Malaysia is more than fifteen hours per day from 7 am to 11 pm. Assuming that the bus AC system runs 25 days a month resulting in a monthly operation of 375 hours. For one year, the total operation hours for the system are 4500 hours. Figure 10 shows the annual energy consumption for the conventional system and the developed one for the three temperatures settings and the possible energy saving could be reaped from the developed system.

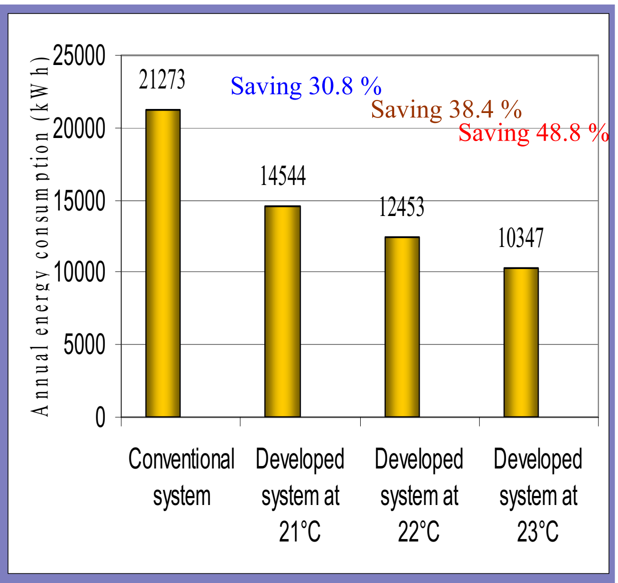

Figure 10: Comparison of yearly energy consumption between the conventional and the developed system for the three temperature settings

It can be seen from Figure 10 that the potential saving in energy consumption from the developed system is approximately 30.8% when the set-point temperature for the controller is 21°C. The potential saving for 22°C and 23°C are 38.4% and 48.8 % respectively.

COSTING

This section addresses the economic merit of the developed AC system over the conventional system. This can be evaluated by calculating and comparing various economic indicators, such as life cycle cost and payback period (PP). The calculation of these indicators involves two major cost categories: owing cost and operating cost. Owing costs comprise of initial costs, salvage value, property taxes, rents, and insurance. On the other hand, the annual system expenditures resulting from the actual use of the system are referred to as operating costs. Operating costs include costs for energy and maintenance. For the purpose of evaluating the developed system, it is assumed that owing costs include initial (capital) costs only, whereas the operating costs involve costs for energy and periodic maintenance only. Thus, the effects of salvage value, property taxes, and insurance on owing costs are neglected. The annual operating cost can be obtained by summing the annually energy cost and adding the annual maintenance cost. The payback period method is the most commonly used economic analysis technique for evaluating energy alternatives [21]. It is defined as the capital cost of implementation divided by the annual cost savings at today's cost. The major advantages of the technique are simplicity and that it is based on facts that are known today and not on some future projections and assumptions. The life cycle cost (LCC) method is often used to promote and enhance expensive energy-related alternatives [21]. The capital cost (first cost) can be added to the present value of annual operating costs to yield the present value of life cycle cost for the system evaluated over the expected life. When comparing between the two systems, the lowest present value is most desirable since it represents the minimum present cost of initial capital outlay and the sum of all future operating costs over the period of the economic analysis.

Table 3 and Figure 11 summarize the comparison between the two AC systems cost. As shown in Figure 11 and Table 3 that the initial cost for the new system is more expensive than that for the old one by 1000 USD as a result of replacing the big compressor with two compressors of identical capacities and a new automatic controller including the installation and commission. However, the saving in energy consumption makes the new system profitable by an annual saving of 617 USD as a result of the reduction in the energy consumption by 30.8% when the set-point temperature is 21°C. That corresponds to a payback period of one year and seven months. On the other hand, the payback period for a temperature setting of 22°C is one year and three months while for a temperature setting of 23°C is only a year. The newly developed system has a relatively short payback period (less than two years) for any of the three temperature settings. The combination of these temperature settings during the system operating hours will further shorten the payback period as a result of the considerable reduction in energy consumption.
In order to determine the life cycle cost for the two systems, the life cycle time and investment rate should be firstly identified. Assuming that the life cycle time for both systems is 10 years and the investment interest is 5%, the resulting running cost for the new and existing systems are 19557 and 28272 USD respectively. This indicates the new system is more economical and has a lower life cycle cost as shown in Figure 11 to encourage the customers to accept the new system.

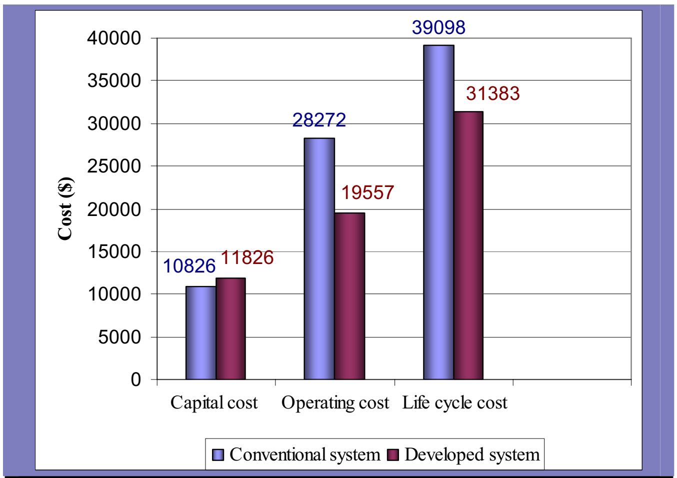

Figure 11: Comparison between the conventional and developed system in term of cost aspect

Table 3 Economic analysis of the newly developed and conventional AC systems

Initial cost	Conventional system	Developed System	
Cost of equipment ($)	10526	11526	
Cost of installation ($)	300	300	
Total initial cost ($)	10826	11826	*Difference = 1000 $*
Annual operating cost			
Energy consumption (kWh)	21971	15198	*Energy saving = 30.8%*
Cost of Energy ($)	2325	1608	*Difference = 717 $*
Cost of maintenance ($)	1000	1100	*Difference = 617$*
Payback period and present value			
Payback period	0	1.6	
Present value of annual operating cost ($)	28272	19557	*Difference = 8715$*
Life cycle cost ($)	39098	31383	*Difference = 7715$*

THERMAL COMFORT ACHIEVEMENT

As aforementioned, the operation of the conventional bus AC system could cause thermal discomfort to the passengers particularly during low sensible load as a result of the overcooling condition inside the bus compartment. The developed bus AC system is able to match the imposed thermal load and thus prevents the condition of the thermal discomfort to the passengers. Figure 12 shows that the developed system is within the comfort zone and reduces the Predicated Percentage of Dissatisfied passengers PPD significantly. As shown in Figure 12 the operating range of the PMV for the developed system is from −0.66 to + 0.02 corresponds to a PPD range of 15.3 % to 5.1%. On the other hand, the operating range of the PMV for the conventional system is mostly a way from the comfort level except above the evaporator inlet air temperature of 26°C as shown in Figure 12.

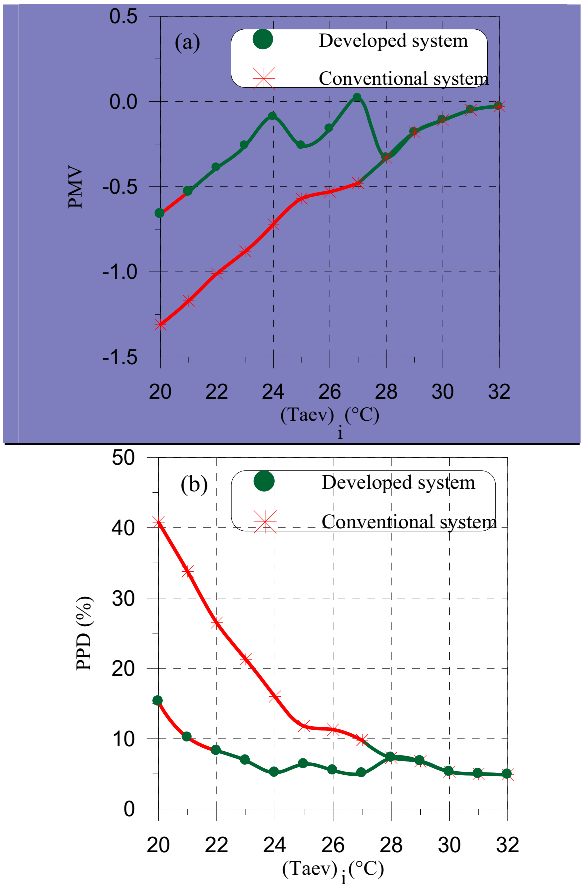

Figure12: Effect of the evaporator inlet air temperature on (a) PMV (b) PPD of the developed and conventional systems

CONCLUSION

A new energy-saving system for bus air-conditioning in tropical countries is presented. The developed system is able to achieve adequate thermal comfort in the passenger cabin of the bus especially at low sensible load conditions with the compressor cycling rates not exceeding 4 cycles/hour. A novel control strategy for the new multiple-circuit AC system is presented. This novel strategy depends on the precise representation of cooling load in term of temperature difference to an automatic controller. Based on the experimental evidences, the following conclusions can be drawn:

- The developed system is capable of saving energy with 30.8% saving for 4500 hours of operation per year at a set-point temperature of 21°C. This corresponds to an annual saving of 617USD. Moreover, the life cycle cost for the developed AC system at this setting is 19.73% lower than that for the conventional AC system. The payback period in this case is one year and seven months.
- The developed system is able to provide thermal comfort conditions for the passengers at different load modes particularly at low sensible loads. For an indoor temperature of 25°C, the PMV is +0.01 (PPD of 8 %) for the developed system while PMV is -0.66 % (PPD of 12.2%) for the conventional AC system. The problem of overcooling will not prevail with the developed system.
- By expressing the cooling load through the evaporator inlet air temperature and selecting the desired set-point temperature, the AC system is automatically controlled for thermal comfort by the controller allowing the bus driver to concentrate on driving.
- The developed AC system has an advantage in that if one unit is unable to operate, the other unit can still provide conditioned air to the passengers compartment until proper maintenance is performed.

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