A Positive Ventilation Air Chiller

"Re-engineering, design and analysis of a unit ideal for hot developing regions"

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Summary

This paper gives an outline of the work that has been carried out in developing a positive ventilation chiller. The demand for air cleaning equipment in public areas and work places (especially where smoking is permitted) had previously prompted the development of a highly innovative ventilator, featuring a combination of ventilation, recirculation and filtration of air. A refrigeration system was successfully retrofitted into the ventilation unit. The chiller is aimed at improving the working and living environment where modern air conditioners are beyond affordability. It will be particularly useful in developing countries where ambient temperature is often in excess of 30°C.

A prototype positive ventilation air chiller was designed, built and fully instrumented. Experimental investigations were made in a test chamber sectioned into indoor and outdoor rooms. The chiller demanded approximately 590W at outdoor temperatures of 30°C. A cooling power of 817 W was achieved at a useful coefficient of performance of 1.39 with a maximum air temperature drop of 10°C.

Keywords:

Air chiller, indoor air quality, positive ventilation, refrigeration, room air conditioning, ventilation.

Introduction

Most countries in hot climates are developing. They cannot afford the high investment and running costs of air conditioning equipment (Abdelrahman 1992; Parsons 1989). The high inflation and devaluation of local currency have further pushed HVAC units beyond the reach of most and diverted attention towards the very basic needs (food and shelter). The countries also fall in the regions that have been hit by natural disasters, droughts and floods. Lack of conditioned environment has limited industrial growth and threatened health in many of these countries.

A simple low cost ventilation (re-circulation) unit has been designed and marketed by Darwin (1998). It was further desirable to extend the design to incorporate air-cooling. In achieving this there was a need to bear in mind the cost limit (both manufacturing and operating). Other aspects considered were refrigerant choice, noise level, shape, size and mass.

It is becoming a global necessity (in some cases legislation) to install air-cleaning equipment in public areas and work places, especially where smoking is allowed. Some of these places are pubs, clubs, restaurants and hotels, homes and offices, schools and colleges.

Ventilators have been on the market for some time. These systems basically replace room air with outside air. Most ventilators (e.g. Xpelair, Air Vent) do not clean the air. Currently available room air conditioners (AC) consist of separated indoor and outdoor compartments (ASHRAE 1996). The proposed system seeks to bridge the gap between the two systems (simple ventilator, air conditioner). The system is designed to chill (or heat) room air while providing a cleaner atmosphere.

The aims of the research were therefore to (a) develop a compact low cost air chiller, (b) demonstrate the operation of the unit and (c) identify operating parameter ahead of unit commercialisation The paper outlines the rational behind the innovative positive ventilation air chiller. It is hoped that paper stimulates many to seek to provide comfortable and hygienic living and working environment with poor developing nations in mind.

Background

A highly innovative ventilator design, Fig 1, which heralds a new generation in air movement technology, is currently in production (Darwin 1998).

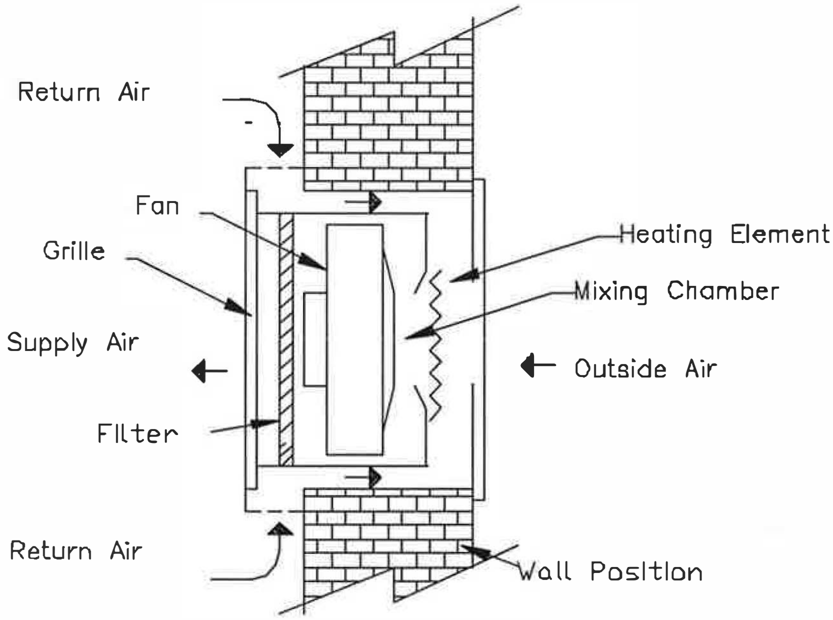

Fig. 1: A schematic diagram of the air ventilator

The unit engineered to European ventilation standards forces air into the room via a replaceable filter. It combines ventilation, re-circulation and filtration of air. This is a contribution towards the global demand to install air-cleaning equipment in public areas. The unit seeks to:

- get rid of stale and smoke laden air in pubs and clubs,
- provide fresh, clean and odour, dust and draught free air in restaurants,
- create an odour free, pollen free and hygiene environment in nursing homes and clinics,
- create a healthier working environment in offices, and training atmosphere in schools and colleges.

A variable speed blower offers a widely adjustable airflow. The unit features a telescopic construction to fit varying wall thickness, and offers an immaculate flush-fitting wall. An electric heater may be used when required. The re-circulation offers reduced heating costs of up to 30%.The ventilator has a maximum design air volumetric discharge of 1200 m^3/h at 2700 rpm. Electricity power requirement is 150 W.

Room air conditioners (ACs) are encased assemblies designed mainly for through wall or window mounting. The units consist of an indoor unit (cooling coils and air filter) and an outdoor unit. Figure 2 shows a typical room air conditioner in the cooling mode. Warm room air passes over the cooling coil and, in the process cools. The conditioned air is then steered back into the room by a fan. The room air is not mixed with external air, and ventilation is only in the form of infiltration.

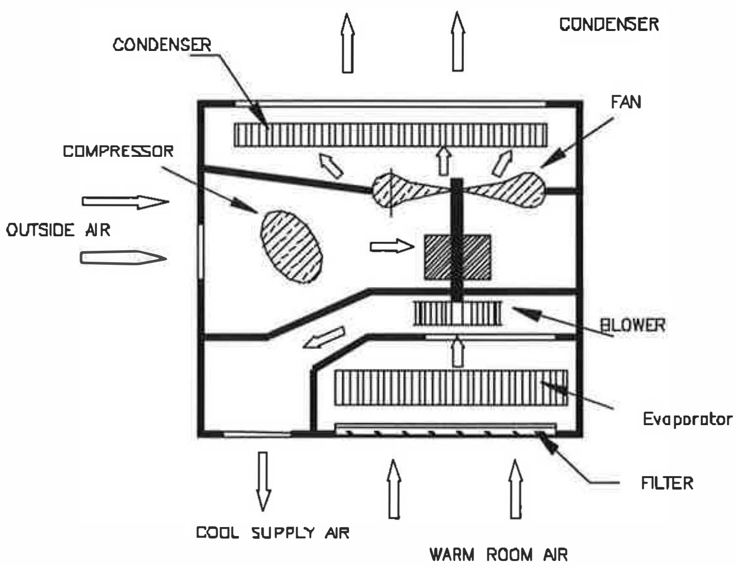

Fig 2: Schematic view of a window type room air conditioner.

The proposed air chiller seeks to bridge the gap between conventional ventilators and room air conditioners. This is achieved by chilling (or heating) both fresh air and room air while providing a clean atmosphere.

The targeted market is mainly the hot developing nations. Enquiries have been centred on India where the ventilator has already been widely marketed. The home market is also large though a system incorporating a heat pump would sell better. MBD (1998) reports a 38% sales increase over the years 1993-1997. It anticipated a growth in the UK air conditioner market of around 26% by 2002. The strength of the pound sterling however tends to favour import penetration. The demand in developing nations may soon exceed that in developed nations in which ambient conditions do not so clearly dictate their necessity. The African market is potentially high but constantly disturbed by ethnic conflicts and local currency depreciation.

The chiller

A cooling load of 1 kW was chosen to design, size and select appropriate components for the refrigeration circuit. The system was designed to fit into the ventilator so that it will chill the supply air.

The refrigeration layout is as shown in Fig. 3. The circuit consists of condensing unit, evaporator, expansion device, filter drier, sight glass and flexible hoses. The condenser, compressor and receiver make up the condensing unit.

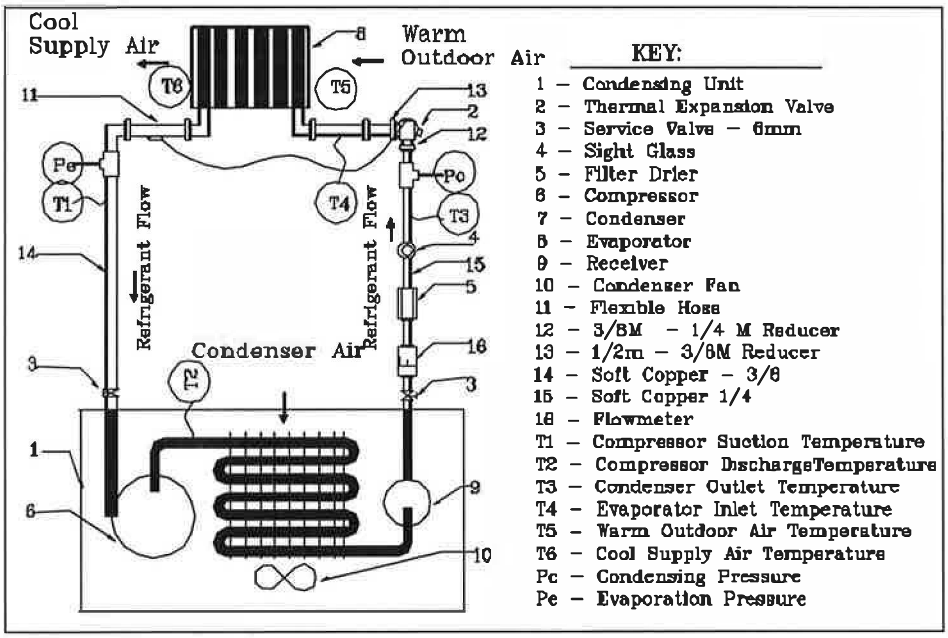

Fig. 3: Refrigeration circuit

A tubeless form constructed coil was selected for the evaporator. It gives a high conductance. The evaporator is rated at 1 235 W.

A $^{3}/_{8}$ hp 134a Electrolux condensing unit is used. The unit is designed for a cooling capacity of approximately 1120 W at 5°C, evaporating temperature. The compressor's maximum start-up power is 590 W while the condenser fan is rated 40W. It has an overall energy ratio 1.78. The maximum recommended operating ambient temperature is 43°C. The unit weighs about 20kg.

For the prototype an R134a internal equalising 'Alco' thermostatic expansion valve (TXV) with a cage assembly was used. In commercial production a capillary tube would be used.

A hydrofluorocarbon (HFC) refrigerant R134a is used, which is currently the world option for small units. The HFC have zero ozone depleting potential but still have a substantial global warming effect. The system was charged with approximately 900 g of refrigerant.

The system was completed with appropriate line components, which included a suction drier and a sight-glass. A flow meter was also fitted (purely for laboratory test purposes) to enable a more accurate determination of the cooling load. Flexible hoses were incorporated to enable access to air filter.

It is increasingly becoming an international requirement to minimise the running cost of air conditioning units. This is mainly due to environmental awareness and the escalating cost of electricity.

Experimentation

The prototype positive ventilation air chiller was installed in a test chamber. The outdoor air temperature was varied between 26°C and 42°C. This was in order to give a practical indication of the chiller performance as well as a practical evaluation of the design. Results were also required to validate computational simulations on room air distribution.

The test chamber, Fig 4, measuring 3.6m x 2.4m x 2.4m was used for experimental tests. It was equally partitioned into an outside (outdoor) room and an indoor room (air-conditioned space). Fresh air was supplied into this room from the outdoor room. The air was passed across the evaporator, reducing its temperature. The indoor room side temperature was controlled by means of a mechanical thermostat.

Fig: 4a. Indoor (left) and outdoor sides of the test chamber

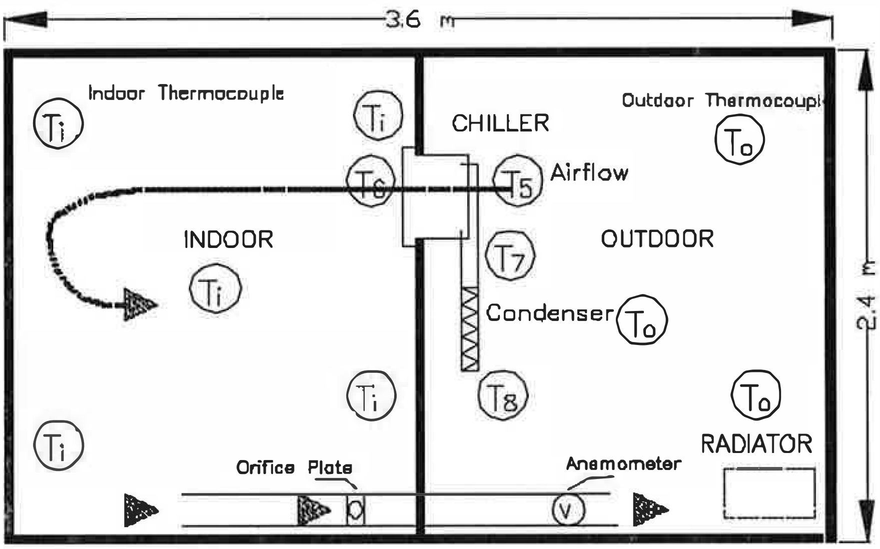

Fig. 4b: Test chamber layout.

The outdoor room housed the condensing unit, the thermal expansion valve and the system line components. The temperature in the outdoor room was maintained by means of an electric fan heater that was controlled by electronic thermostat.

A 6" pipe was installed through the wall dividing the two rooms. The indoor room was sealed airtight using high-grade silicon sealant and some duct tape. The airflow through the pipe could therefore be measured in order to obtain the flow rate. The airflow rate was determined using a 'hot-wire' anemometer, a vane anemometer and an orifice plate.

Instrumentation

Temperatures around the refrigeration circuit and in the air stream were measured using type K thermocouples. Bourdon gauges were used to measure refrigerant pressures. Manometers were used to measure various room pressures. A 'Platon' rotameter, calibrated for liquid refrigerant R134a, was used for the refrigerant flow rate. A 'NORMA' Power Analyser D5135 was used to measure the electric power. A 'VelociCalc® TSI Model 8360' anemometer was used for airflow and temperature measurements.

The Ventilation Fan

Fans drive the room air to maintain and achieve the desired indoor ventilation and thermal comfort. As recommended by ASHRAE (1997) for the pressure drops which are expected across the filter and evaporator. This is a centrifugal fan with backward-curved blades.

Airflow

The velocity profile across the centre of the evaporator as the fan speed setting is increased from 0 to 10 is shown in Fig 5.

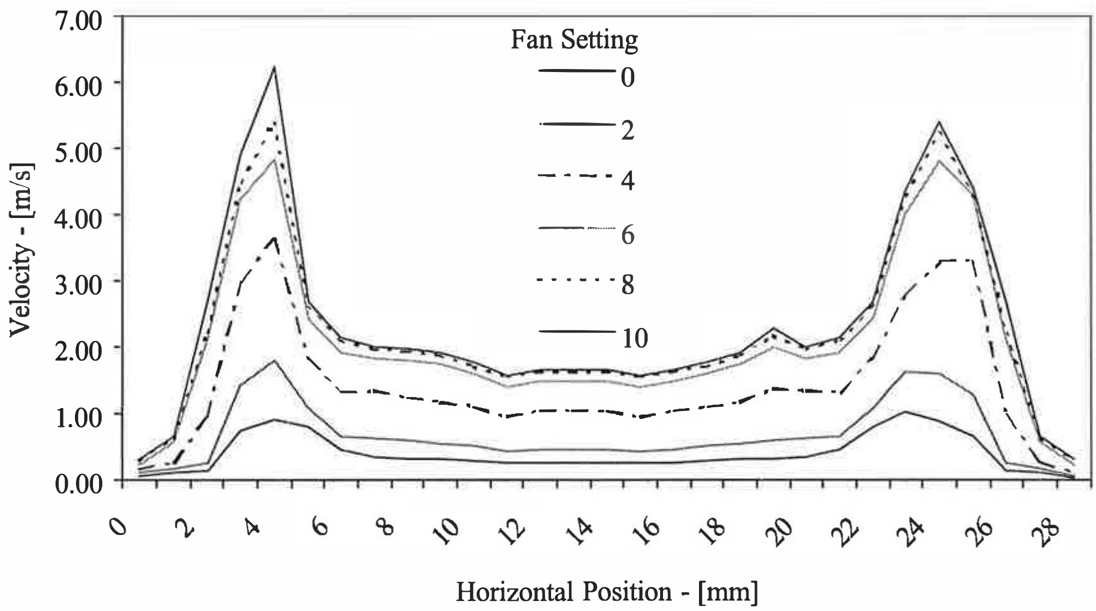

Fig. 5: Velocity profile across the centre of the evaporator

The volumetric flow rate was measured to be 348 m^3/h with chiller back grille in place. The mass flowrate was then computed to be 0.12 kg/s.

Refrigeration

Refrigeration performance tests were carried out with and without allowing air recirculation. The performance was evaluated with the aid of a refrigerant library 'Xmedlib'. At 30 °C and maximum fan speed refrigeration performance was obtained as in Table 1.

	With Recirculation	Without Recirculation
Refrigeration load, W	817.4	862.1
Compressor Power, W	360.0	335.7
Cycle COP	2.30	2.57
Useful COP	1.39	1.43

Table 1: Refrigeration performance with and without air recirculation.

The useful coefficient of performance (COP) takes into account the total electricity requirements for the chiller.

Figure 6 shows a typical trend of the compressor inlet temperature (T1), compressor discharge temperature (T2) and the expansion valve inlet temperature (T3) as the outdoor temperature is increased from 26 °C to 42 °C.

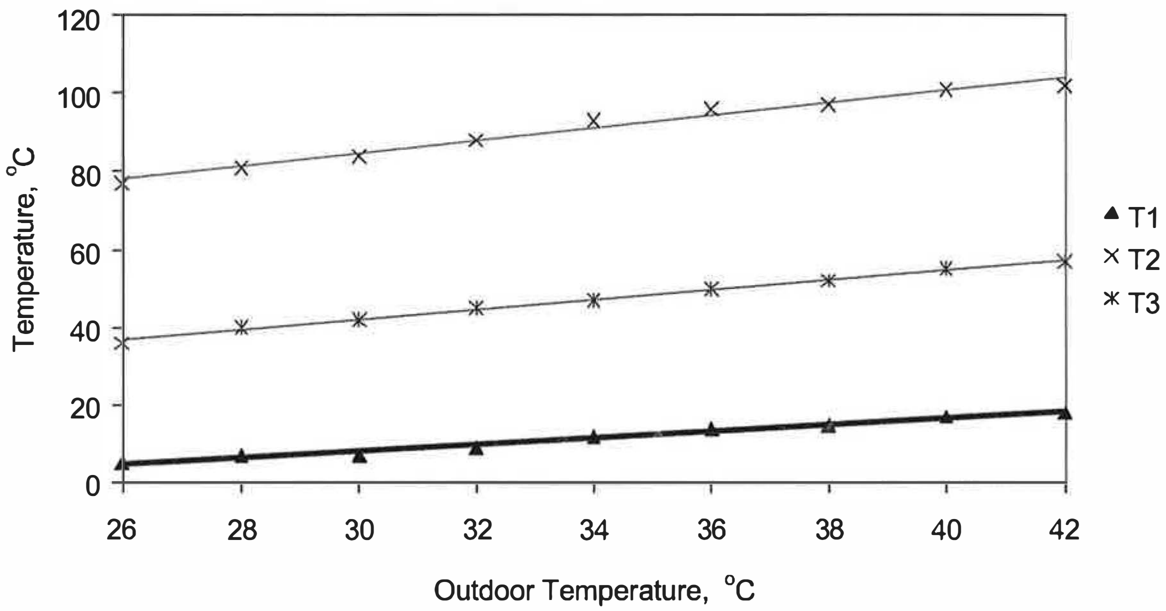

Fig. 6: Refrigeration line temperatures as a function of outdoor temperature.

Evaporating temperature increased from 0.7 °C to 8.2 °C as the outdoor temperature was increased through to 42 °C. The evaporation effect decreased with the outside temperature. The coefficient of performance (COP) decreased as the outdoor temperature was increased, Fig. 7.

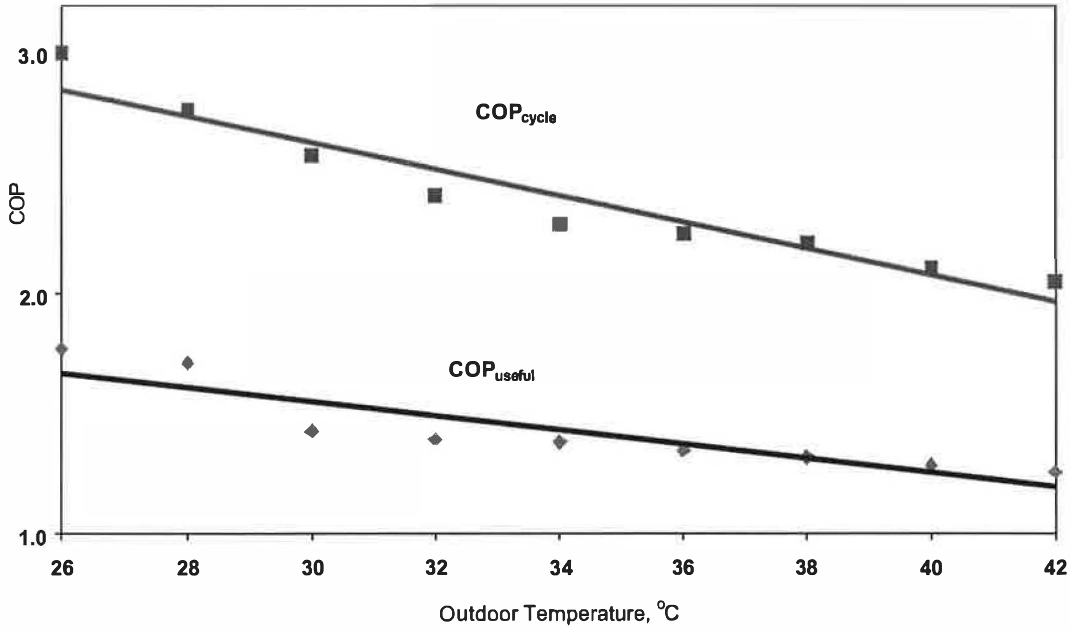

Fig. 7: COP as a function of outdoor temperature.

The maximum refrigerant circulation rate was approximately 9 g/s. The compressor power at start up was approximately 590W. At normal operation the running power at high cooling demand was approximately 445W.

The compressor power at different fan speeds is shown in table 2.

Fan Speed Position	0	2	4	6	8	10
Power [W]	425	435	440	442	444	445

Table 2: Power supply to compressor

The compressor discharge temperature ranged from 70 °C to 88 °C with air recirculation but rose to 102°C when the ventilator was run without air re-circulation.

Air temperature drop

Figure 8 shows the distribution of the air temperature drop across the evaporator area.

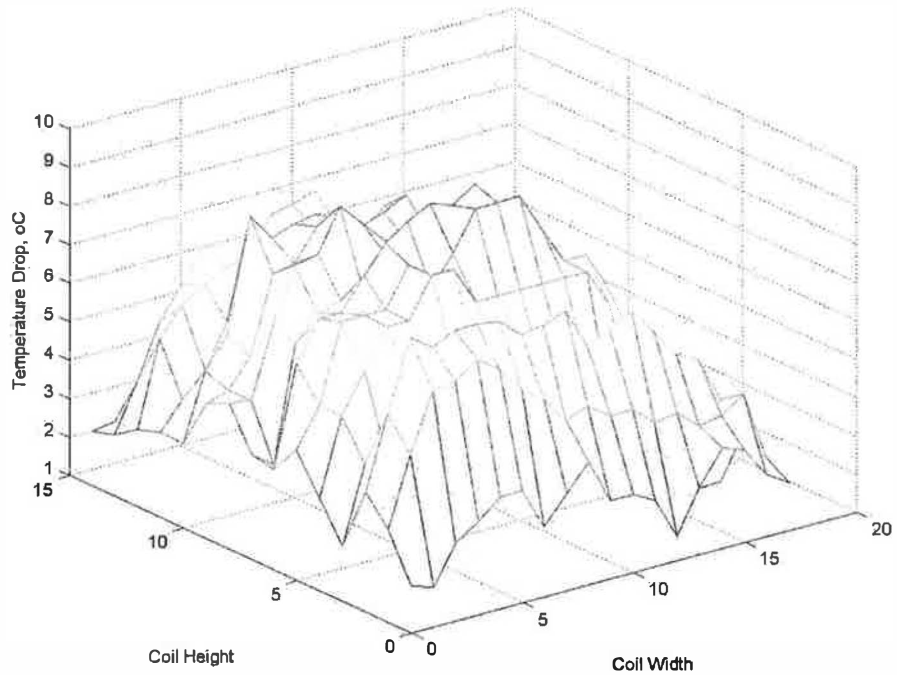

Fig 8: Air temperature drop across the across the evaporator.

The average air temperature drop without recirculation is as shown in Fig. 9.

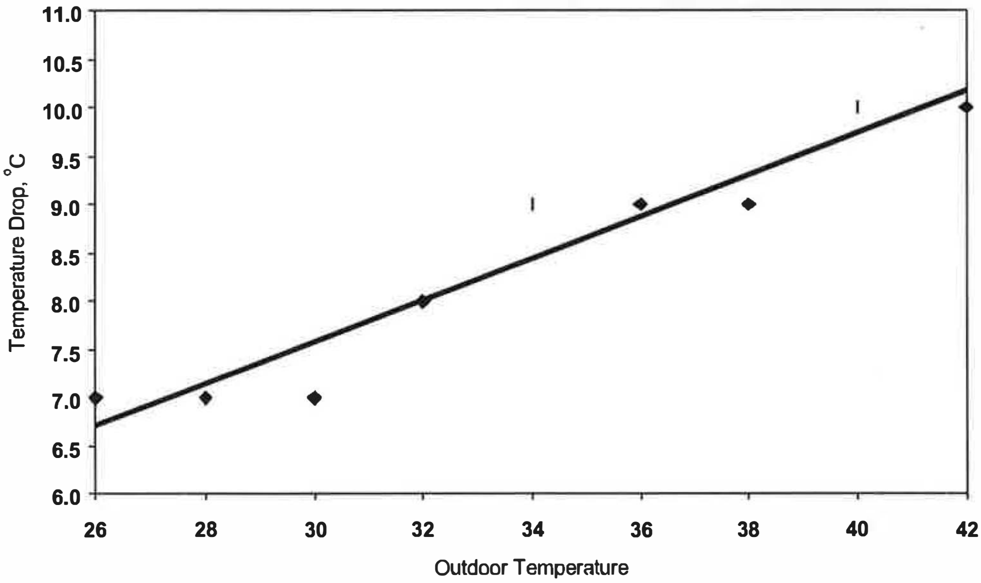

Fig.9: The air temperature drop across the evaporator coil as a function of outdoor temperature.

Energy

In order to effectively determine the room conditions and carry out energy balance several thermocouples were mounted in the outdoor and indoor rooms.

The air conditions in the outdoor and indoor rooms were as tabulated below.

	Dry bulb DB [°C]	Wet bulb WB [°C]	Enthalpy h [kJ/kg]	Dew Point DP [°C]	Humidity Ratio ω [g/kg]	Relative Humidity RH [%]	Specific Volume v [m^3/kg]
Outdoor	30	20	57	14.8	10.5	39.7	0.872
Indoor	23	17	47.6	13.5	9.6	55	0.851

Table 3: Outdoor and in-door side air conditions

Using the enthalpy method, with the outdoor and indoor conditions as in Table 3 energy removed from the air was 1103 W.

$$Q_a = m_a \Delta h_a$$

Where m_a is the air mass flow rate and Δh_a is the air enthalpy change across the evaporator.

This energy agrees with heat that was absorbed by the refrigerant in the evaporator (1198 W). Given by:

$$Q_e = m_r \Delta h_r$$

Where m_r is the refrigerant mass flow rate and Δh_r is the refrigerant enthalpy gain flowing through the evaporator.

Noise

The fan manufacturer quotes the noise level produced as 71 dBA. The sound was however measured to be 76dBA maximum. The measurements were taken about 0.5 m from the unit and 1.5m above floor level according to ASHRAE specifications.

Discussion

A refrigeration unit has been successfully designed, constructed and commissioned. The performance of the refrigeration unit was satisfactory to ASHRAE (1986) standards. It was demonstrated that the designed unit is an ideal retrofit to the air ventilator. It achieves the desired conditions. Temperature drops in the order of 10 °C were achieved.

Approximately 348 m^3/h of air was delivered into the indoor room without recirculation. The flowrate increased to 600 m^3/h when the unit was run without a filter and without an evaporator. A pressure drop of 25 mmH_2O was experienced across the filter/evaporator. Approximately one third of the pressure developed by the fan is used to overcome the grille impedance. An adjustable damper with opposed-operating blades is proposed. This will enable control of the airflow while protecting the fan from air-borne particles.

A cooling load capacity around 817 W was achieved. The compressor performed within design specifications. The refrigeration COP was 2.30. The overall refrigeration COP was 1.39. The equivalent heat pump COP for the cycle would be 2.23

The chiller demands 590 W at 6.14p (UK kWh charge) running cost of which is therefore 29 p per day assuming that the unit is run 8 hours. Charges in Zimbabwe are Z$2.50 (4.2p) per kWh. This amounts to a running cost of Z$11.80 (20p) per 8 hour unit running day. Electricity is generally available to most of the developing nation at least to those who can afford buying the unit.

It was impossible to determine accurately the energy losses in the open system due mainly to unavailable component data (e.g. overall heat transfer, pressure drop), which are commercially confidential. It is recommended that co-operation with component manufacturers be established before recommending (or experimenting with) their products.

Available packaged air conditioners require well skilled maintenance technicians while the proposed unit does not. The unit is simple and easy to service. It can be manufactured from local (developing countries) components at local manpower rates making it widely available and affordable.

Several ventilation design guidelines have been developed, e.g. for displacement ventilation mainly in Scandinavia (Svensson 1989, Nielsen 1995; Sandberg and Blovmqvist 1989), but these cannot be used in hot countries with confidence, since the regions have high cooling loads, and employ different heating systems. Design rules (e.g. operating chiller temperatures, chiller and vent location) will be established with the use of computational fluid dynamics (CFD).

Conclusion

Though the HVAC industry is old there is an open end to new system designs. An air chiller unit has been successfully designed, built and tested. This chiller has been designed to provide clean cooler air. By combining filtration and recirculation, indoor air quality and comfort are achieved at lower running costs.

The unique compact positive ventilation unit capable of delivering up to 600 m^3/hr has been developed. The chiller is ideal for domestic, office and industrial use. Cooling effects exceeding 1 kW were achieved. Air temperature drop across the evaporator of up to 10°C were achieved. Computational investigations on airflow patterns, pressure drop and temperature distribution are underway.

Acknowledgements

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