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IMPROVING THE ENERGY EFFICIENCY OF A CONTROL CABINET AIR CONDITIONER THROUGH THE USE OF VARIABLE REFRIGERANT FLOW CAPACITY CONTROL

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ABSTRACT

Control cabinet air conditioners are used to regulate the temperature of an enclosure containing electrical equipment. Normally, these air conditioners are selected for worst-case conditions, and work on an on/off control.

This paper describes the work carried out to analyse the performance of an air conditioner (AC) with variable refrigerant flow (VRF) for a masters degree dissertation [1].

The performance of a standard on/off air conditioner was first measured. The implementation of VRF in a control cabinet was carried out successfully by installing a variable-speed compressor within a standard AC unit. Experiments performed showed that the energy savings are 14% at full load and between 8 and 32% at part load. For most conditions, the enclosure temperature could be controlled to a stable value with a flat enclosure temperature profile. Maintaining a stable enclosure temperature reduces electronic component failure.

A computer model was created using Microsoft Visual Basic for Applications, which could be used within Microsoft EXCEL. For a given set of ambient temperatures and enclosure loads, the model estimates the power consumption of a standard AC and a VRF AC and calculates the potential savings. When applied to various scenarios, savings of 18-25% were achieved.

The system efficiency can be improved further by other changes to the AC design. A mathematical software model of the AC was built using Visual Basic Express 2005, to evaluate these potential improvements. It was shown that the COP could be improved by increasing the air-flows and by controlling evaporator superheat. By using an electronic expansion valve, the degree of superheat could be accurately controlled. Changes in refrigerant charge were found to have more effect at high ambient temperatures, with the cooling capacity being maximised with only small changes in power consumption.

INTRODUCTION

"Heat from electronic devices is an integral part of information processing, and not a nuisance that can someday be eliminated. This is a physical principle that is independent of the device of information processing." This statement by Nakayama [2] sums the basic fact that all electrical and electronic processes produce heat. Thermal management at the component level has been extensively studied [3, 4, 5]. Racks of circuit boards are housed in small enclosures which are then placed in large enclosures. Excessive heat generation is the major factor affecting equipment performance and reliability [6]. Klebe [7] listed several trends in electronic packaging that influence enclosure design, including increased heat dissipation from electronic components, reduction in enclosure size, capital and operating costs and increased system reliability. Thermal management of enclosures is therefore becoming an important feature of enclosure system design.

One way to deal with this heat is through the use of Control Cabinet Air Conditioners. These are used when the temperature of an enclosure containing electrical equipment needs to be regulated. Normally, these air conditioners are selected for worst-case conditions, i.e. providing the required enclosure temperature with maximum enclosure load and at the highest ambient temperature possible. These conditions rarely occur and the air conditioner, working on an on/off control, will cycle its operation in order to maintain the temperature within set limits. Although generally perceived as cheap and reliable, the disadvantages of on/off control are numerous and include poor temperature control, reduced reliability and increased energy consumption.

The drive towards improved energy efficiency has led other sectors in the refrigeration and air-conditioning industry to implement variable refrigerant flow capacity control and in particular variable compressor speed capacity control which is considered to be the most efficient technique available. In these systems, the capacity of the refrigeration system is matched to the load by adjusting the speed of the compressor. Possible savings in the order of 20 to 40% have been demonstrated [8]. The control cabinet cooling sector has not followed and all products available on the market use single-speed compressors and on/off capacity control.

The objectives of this work were therefore to analyse the performance of an air conditioner (AC) with variable refrigerant flow (VRF), to demonstrate the potential efficiency improvements, to understand how this capacity control method behaves when cooling a control cabinet and to determine the design principles for optimum use.

NOMENCLATURE

AC		Air conditioner
COP		Coefficient of performance
EEV		Electronic expansion valve
Q_{TEST}	[W]	Test Cooling Capacity
TXV		Thermostatic expansion valve
VRF		Variable refrigerant flow
VSC		Variable speed compressor
W_{AC}	[W]	Electrical input to AC
W_{COMP}	[W]	Work input to AC compressor

EXPERIMENTAL SETUP FOR MEASURING AC PERFORMANCE

Figure 1 shows the test setup that was used. The AC under test was mounted to a test enclosure that is fitted with a bank of electric heaters and a recirculation fan providing sufficient air flow to give even distribution of the heat throughout the enclosure. The enclosure is placed within a temperature-controlled climatic chamber which simulates ambient conditions. For full-load operation, the electric heater is switched via a PID temperature regulator to maintain a fixed enclosure temperature at AC intake. For part-load tests, the voltage across the electric heater is adjusted so that the required heat load is attained.

The test unit selected is a KG-8277 control-cabinet air-conditioner for indoor applications manufactured by Seifert mtm Systems (Malta) Ltd. Figure 2 shows the external appearance of the air-conditioner together with an illustration of the air flow paths when mounted to an enclosure.

The standard compressor was replaced by one with a smaller capacity that matches the maximum performance of the variable speed compressor used in the VRF tests. The technical specifications of the test AC are:

Type KG-8277
Manufacturer Seifert mtm Systems (Malta) Ltd.
Size 1580 (H) x 395 (W) x 290 (D)
Power Supply 230V 50Hz
Assembly to enclosure All tests as externally mounted
Compressor Mitsubishi Electric type RB231GHA
Internal side fan ebm-papst type R2E225-BD92
Ambient side fan ebm-papst type R2E225-BD92
Controller Seifert proprietary
Evaporator Seifert proprietary, 21 rows, 4 tubes in 25x12.5mm layout, 3 circuits, 2.0mm fin spacing
Condenser Seifert proprietary, 28 rows, 5 tubes in 25x12.5mm layout, 2 circuits, 2.0mm fin spacing
Refrigerant control TXV, Honeywell type TLEX 2.0 with MOP +10°C
Filter drier De.Na. srl type SM3 50gr
Refrigerant R134a (CH_2FCF_3), 1180g

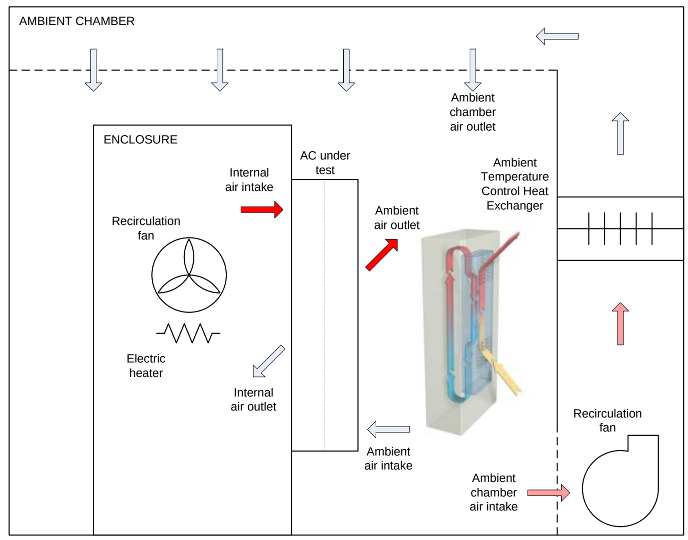

Figure 1 Sketch of test apparatus

Figure 2 KG-8277 control-cabinet air-conditioner

FULL-LOAD PERFORMANCE – EXPERIMENTAL RESULTS AND ANALYSIS

The scope of the tests was to determine the full-load performance characteristics of the AC, i.e. the useful cooling capacity and power consumption at various combinations of enclosure and ambient temperatures. Measurements were performed at combinations of ambient temperatures of 15, 25, 35, 45 and 55°C and enclosure temperatures of 20, 30, 35, 40 and 50°C for a total of 25 tests. Each condition was maintained for a minimum of 4 hours with the values over the last 2 hours being averaged. Figure 3 shows the COP of the air conditioner for difference combinations of ambient and enclosure temperatures. As can be seen, the COP decreases with lower enclosure temperatures and higher ambient temperatures.

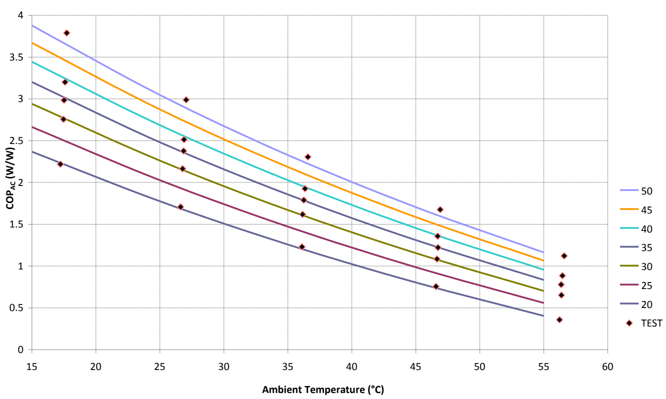

Figure 3 Plot of COP for ambient and enclosure temperature combinations

PART-LOAD PERFORMANCE – EXPERIMENTAL RESULTS AND ANALYSIS

Measurements were performed at combinations of ambient temperatures between 15 and 55°C and heat loads of 150, 500, 1000, 1500 and 2000W. Since the common control setting in the industry is 35°C, this was used as the standard. Each condition was maintained for a minimum of 4 hours with the values over the last 2 hours being averaged. Thirty-nine tests were performed. Figure 4 shows the results obtained.

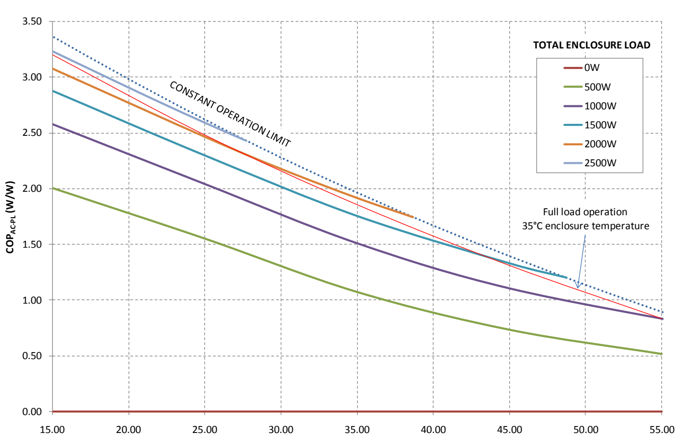

Figure 4 Plot of COP for ambient temperature and load combinations

Figure 5 shows an example of the temperature fluctuations inside the cabinet for an enclosure load of 1000W and various ambient temperatures. Temperature fluctuation varies between 5.3°C and 7.7°C and increases with ambient temperature. For low ambient temperatures, heat is lost from the enclosure and therefore the rate of temperature rise is less than that occurring at higher ambient temperatures where additional heat enters the enclosure. When compared to the expected temperature rise of an empty enclosure heated with the equivalent load, a faster rate of temperature rise is observed. This is however slower than the expected rise if only the air inside the enclosure was being heated. This behaviour can be due to the fact that the thermal inertia of the walls delays the heating of the walls, and the fact that the AC still provides some cooling effect after the compressor has stopped, until all the refrigerant in the evaporator has evaporated and reached the enclosure temperature. In all cases (except 15°C ambient) the rapid enclosure temperature rise means that the controller set-point temperature is exceeded in around 1 minute. The controller prevents the compressor from starting in order to maintain the minimum off time and therefore the compressor always starts after the minimum off-time has elapsed. During this additional delay, the enclosure temperature continues to rise. For the 15°C ambient conditions, the enclosure temperature is significantly less than for other conditions. Here the useful cooling capacity of the AC is very large and the temperature of the enclosure drops very fast. Since the minimum operating time has to be observed, the enclosure is cooled below the off control point. The temperature fluctuations are however in the same order as for higher ambient temperatures. A possible consequence of this is that if the temperature is reduced too low, condensation may occur on the equipment and walls of the enclosure with harmful effects.

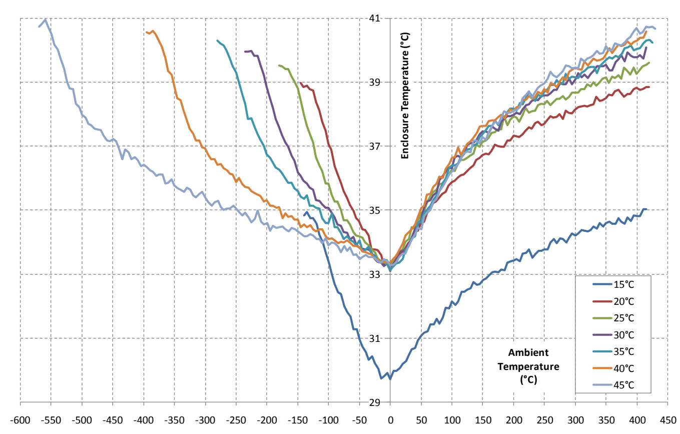

Figure 5 Enclosure temperature during cycling, 1000W load and different ambient conditions

EFFECT OF CONTROL SET-POINT ON POWER CONSUMPTION

Although 35°C is the standard set-point, some users choose to operate the AC at higher or lower set-points. To determine the effect of set-point on the power consumption and cyclic

COP, measurements were performed for a 1000W enclosure load cooled at different ambient temperatures and set-points of 25°C and 45°C which were compared to those for 35°C. The results are shown in figure 6.

At 25°C set-point, power consumption is 40% higher than at a 35°C set-point. A smaller change is observed for a 45°C set-point, the consumption being an average of 9% less than with a 35°C set-point.

The results show that a lower control set-point than is necessary will increase the operating power consumption and hence operating costs significantly. Maintenance costs would also be increased since the longer operation time will require more frequent cleaning and lifetime is reduced. Moreover, a low enclosure temperature in a humid environment will result in excessive condensation formation (with useful cooling capacity being lost as latent heat transfer) and the risk of condensation forming on the equipment and enclosure walls.

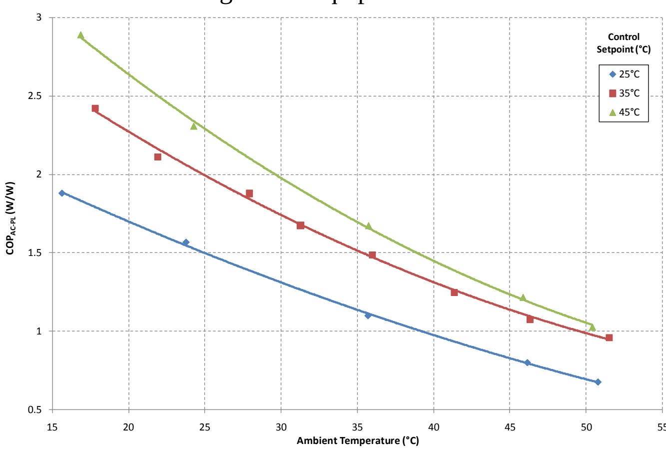

Figure 6 Variation of COP for different set-points, 1000W load

DEVELOPMENT OF SOFTWARE MODEL FOR ESTIMATING POWER CONSUMPTION

To estimate the true energy cost of the AC over a diverse range of operating conditions, a software tool was developed. The purpose of this software model was to compare the performance of the variable compressor speed capacity control with on/off capacity control and hence determine the improvements in energy efficiency and potential savings and benefits. This was to be sufficiently simple to allow its use by persons making enclosure load estimations. It was decided to create a Visual Basic for Applications (VBA) macro which could be integrated into Microsoft Excel.

To verify the model, a test measurement was set up with a fixed heat load of 1445W. The ambient temperature was changed every 4 hours. The start of every temperature cycle was taken when the climatic chamber set-point was modified. This means that the ambient temperature change to the new set-point occurs at the start of the cycle. The average temperature for the cycle was used as the input to the model. The resultant temperature differences are shown in figure 7.

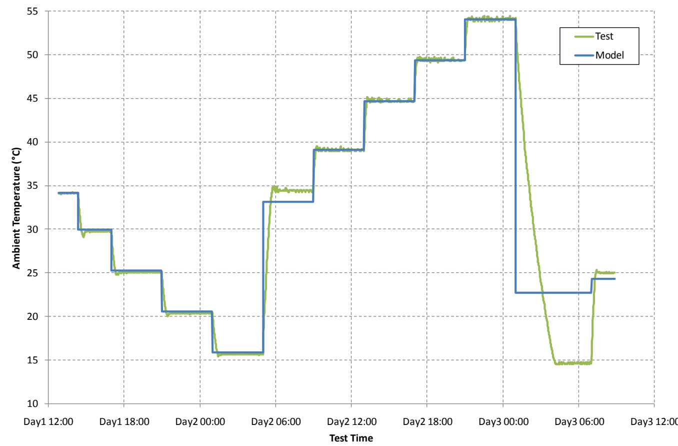

Figure 7 Comparison of actual test and model ambient temperatures

The test consumption was 38.65kWhr compared to 36.8kWhr of the model. This is an under-prediction of 4.8%. Generally most of the values of power consumption are under-predicted. The largest errors are obtained when the ambient temperature is falling when, while the model assumes an immediate change to the lower value, in actual fact the AC still operates at higher temperatures for some time. Since the time between temperature changes is not allowed for, errors are to be expected. Modeling errors could be reduced by decreasing the time interval further. However the overall error is sufficiently small and suitable for the intended purpose of the model, as outlined above.

PERFORMANCE OF A VARIABLE COMPRESSOR SPEED CAPACITY CONTROLLED CONTROL CABINET AIR-CONDITIONER – EXPERIMENTAL RESULTS AND ANALYSIS

To have a direct comparison, an AC unit identical to the one used in the tests discussed earlier on in this paper, was fitted with a variable-speed compressor. The specifications are compared to those of the standard fixed-speed compressor in Table 2. The VSC operation is controlled through a digital inverter driving module.

Feature	Fixed-speed	Variable-speed
Manufacturer	Mitsubishi Electric	Mitsubishi Electric
Type	RB231GHA	VSC
Design	Single rotary	Twin rotary
Displacement	23.1cm^3	13.0cm^3
Motor type	Permanent Split Capacitor	Brushless DC Motor
Speed	2860 rpm	1200 – 4800 rpm
Speed controller	-	Mitsubishi Electric proprietary
Rated capacity	2730W	1888W at 3600 rpm

Table 2 Comparison of fixed and variable speed compressors

Since the operation of the AC is always at full-load conditions for the particular compressor speed and temperatures, only full-load performance tests were needed. Measurements were performed at combinations of ambient temperatures of 15, 25, 35, 45 and 55°C, enclosure temperatures of 20, 35 and 50°C and compressor speeds of 20, 40, 60 and 80Hz. A total of 60 tests were performed, each condition being maintained for a minimum of 4 hours with the values over the last 2 hours averaged.

Figures 8 to 11 show the COP charts for the various compressor speeds and combinations of ambient and enclosure temperatures. For 20 and 40Hz, some conditions exist where the capacity of the AC is insufficient to overcome the load due to the thermal losses across the enclosure walls and therefore cooling of an additional load is not possible.

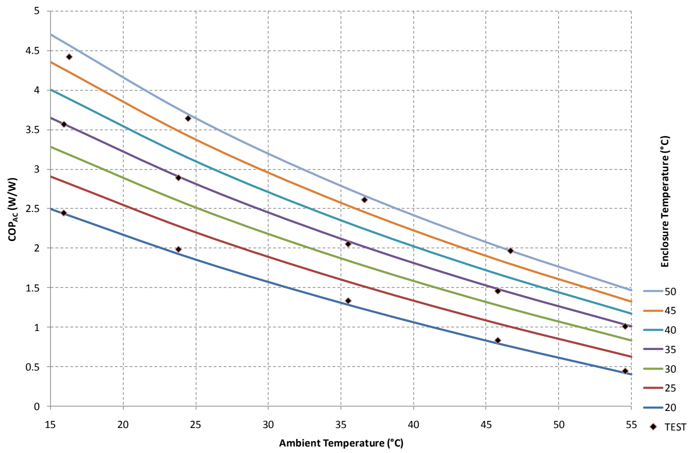

Figure 8 Plot of COP for ambient and enclosure temperature combinations for 80Hz

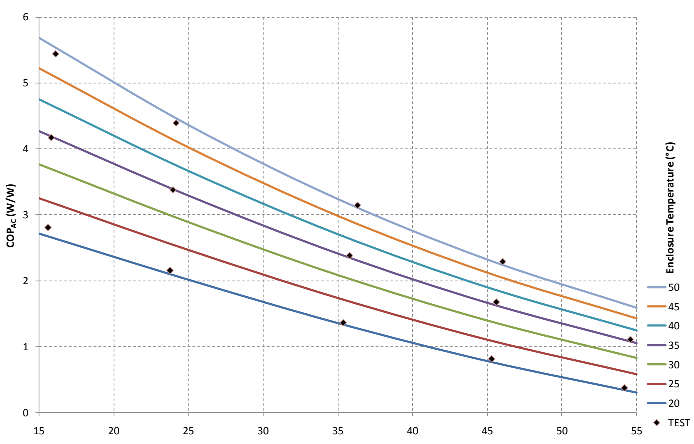

Figure 9 Plot of COP for ambient and enclosure temperature combinations for 60Hz

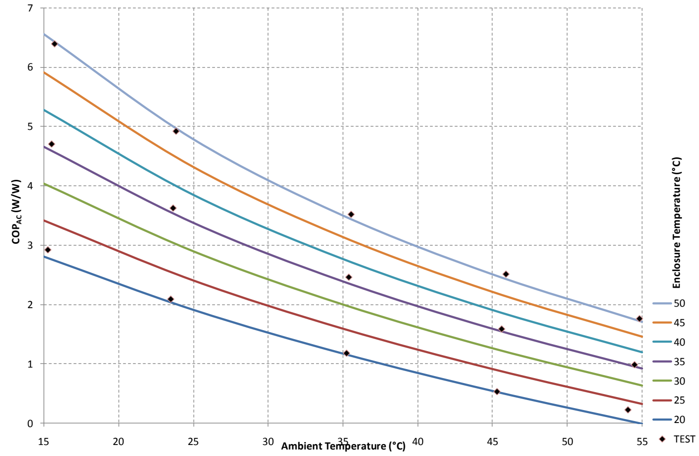

Figure 10 Plot of COP for ambient and enclosure temperature combinations for 40Hz

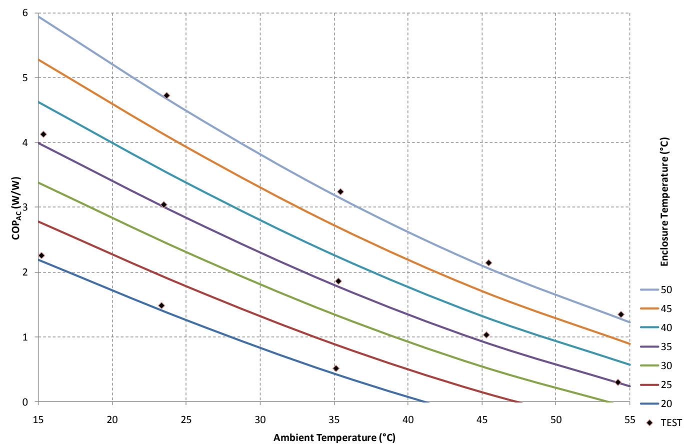

Figure 11 Plot of COP for ambient and enclosure temperature combinations for 20Hz

COMPARISON OF EXPERIMENTAL RESULTS FOR ON/OFF AND VARIABLE COMPRESSOR SPEED CAPACITY-CONTROL TECHNIQUES

An initial comparison was performed for the different VSC settings and the performance with the fixed-speed compressor operating at full-load with 35°C enclosure temperature. This is shown in figure 12. At 80Hz, the AC performance matches closely that of the same unit with a fixed-speed compressor. This is expected since the refrigerant delivery volume (compressor speed multiplied by displacement) at these conditions is approximately equal.

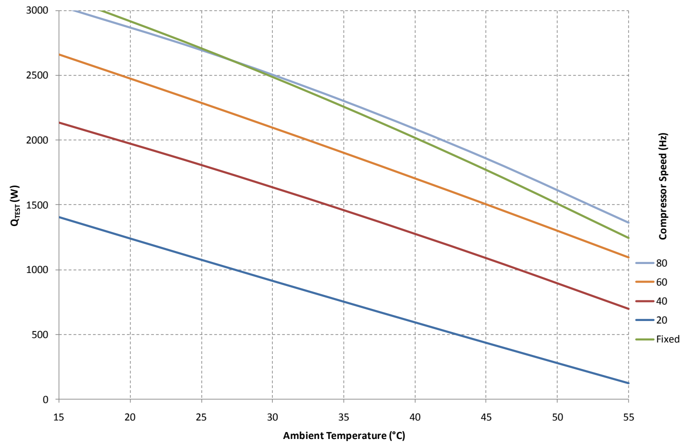

Figure 12 Plot of Q_{TEST} for ambient temperature and compressor speed combinations for 35°C enclosure temperature

The performance chart also shows that the cooling capacity of the AC at the lowest compressor speed is still rather high at lower temperatures and it is necessary to adopt a further on/off capacity-control under these conditions.

A first difference is observed when comparing the electrical consumption (figure 13). The power consumption of the VSC at 80Hz is constantly less than that of the fixed-speed compressor. Since the compressor capacity is almost identical, this change should be due to the compressor motor being itself more efficient and the smaller twin-rotary compression mechanism generating less frictional resistance than the larger single-rotary mechanism used in the fixed-speed compressor.

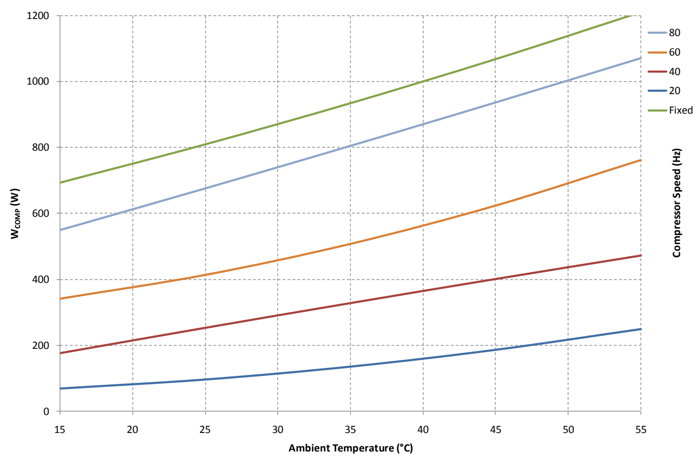

Figure 13 Plot of W_{COMP} for ambient temperature and compressor speed combinations for 35°C enclosure temperature

The resultant COP is therefore also improved. For the fixed-speed compressor system the COP was 1.86. This is improved to 2.12 with the variable-speed compressor. The COP plot in figure 14 shows a great variation in the AC operating COP between the different speeds. At 40 and 60Hz, the best COP values are obtained. The COP at 20Hz degrades considerably with an increase in ambient temperature. This is due to the proportion in the total consumption of the electrical consumption of the enclosure and ambient fans (fixed at 284W) which increases as the compressor consumption decreases. This effect is more significant at low compressor speeds where the introduction of variable-speed fans can help to improve the COP. Figure 15 plots the COP based only on the consumption of the compressor and the total cooling capacity of the AC and shows that the compressor COP always increases with lower compressor speed. The total cooling capacity of the AC includes the heat losses of the enclosure and the work input to the AC fan.

Figures 14 and 15 indicate a possible problem with the interpretation of AC performance at low speed and high ambient temperatures through functions. Both charts show that whereas higher compressor speeds follow a trend, this is not so for 20Hz speed. This can be attributed to the following factors which should be investigated further:

- The power consumption and cooling capacity are very low leading to larger relative measurement errors.
- Losses through the enclosure walls are proportionately large and errors in the heat transfer coefficient estimation have a larger effect.
- The tendency for the functions to smooth off the data can result in large errors at the extremes.
- Refrigerant mass flow is very low at these conditions. The performance of the compressor and TXV may be adversely affected.

Due to the very low loads and high temperatures, these conditions should only occur over a minimal amount of time.

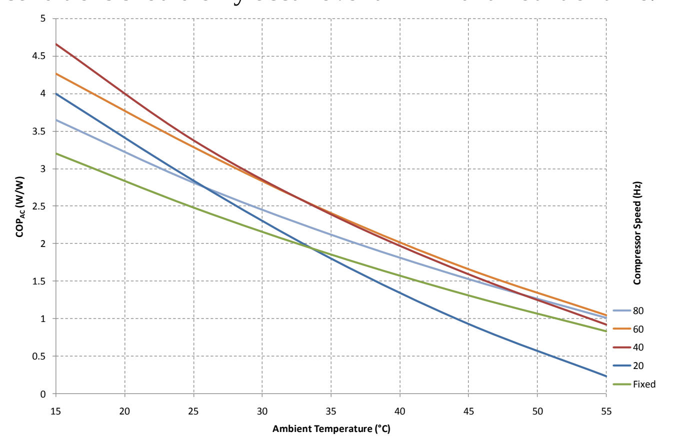

Figure 14 Plot of COP for ambient temperature and compressor speed combinations for 35°C enclosure temperature

Figures 16 and 17 show the difference in total AC consumption and compressor consumption respectively with varying load for a fixed ambient temperature. These indicate that:

- As the load increases, the difference decreases. This is expected since with increasing load, the variable-speed compressor will run at a faster speed, consuming more energy and approaching the point where the two compressors both work at full speed.

- As the load decreases, the difference tends to increase. This is due to the proportionately larger consumption of the fans. Figure 17 shows that the difference in compressor consumption continues to increase but is offset by the consumption of the fans.
- For every ambient temperature there exists a load rating at which the difference is maximised.
- The difference increases with decreasing ambient temperature.
- The difference ranges from 9% for 55°C and 1400W load, to 32% for 15°C temperature and 1900W load. Considering the difference only in the compressor consumption, this ranges from 11% for 55°C and 1400W load, to 68% for 15°C temperature and 1400W load.

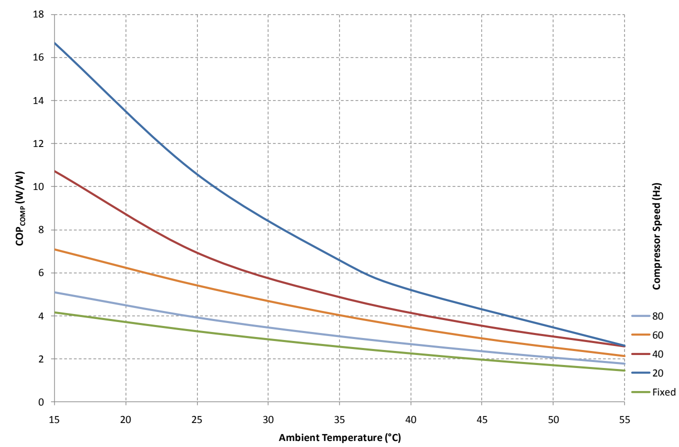

Figure 15 Plot of COP_{COMP} for ambient temperature and compressor speed combinations for 35°C enclosure temperature

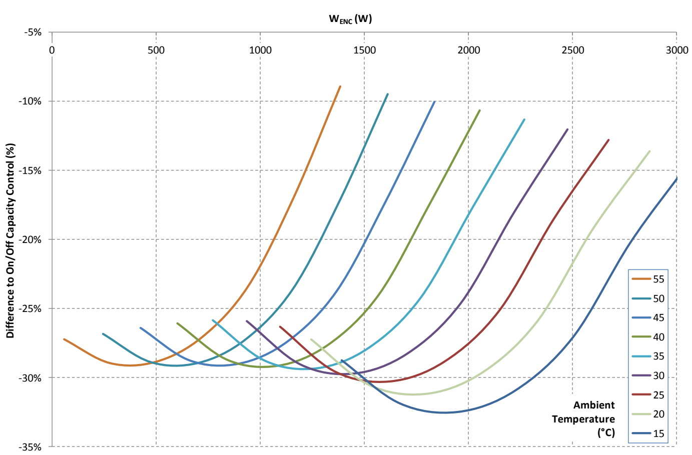

Figure 16 Plot of % difference of W_{AC} between variable compressor speed and on/off capacity-control for constant ambient temperature

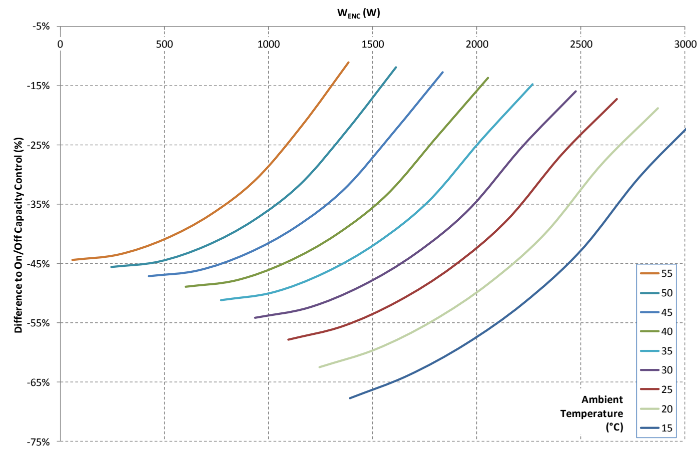

Figure 17 Plot of % difference of W_{COMP} between variable compressor speed and on/off capacity-control for constant ambient temperature

MODELING THE AC COMPONENTS

For the implementation of a VSC into an AC unit, the rest of the refrigeration circuit and other components have been largely unchanged. The tests carried out show that the efficiency of the AC is improved with the speed-controlled design. To further enhance the efficiency, as well as improve that of the standard on/off system, other changes to the AC design can be implemented. The main areas of improvement are reduction in parasitic and thermal losses, fans and air-flows, evaporator superheat control and heat-exchanger design. To evaluate some of the proposed improvements, a numerical modeling approach was adopted with some possible solutions then being evaluated experimentally. The model developed in this work was intended to provide an indication of the changes in system performance and efficiency before the solution is verified experimentally.

A mathematical software model of the AC was designed and programmed using Visual Basic Express 2005. The aim of the model was to evaluate the improvement possibilities numerically without the need for extensive testing. The model errors compared to test results were acceptable considering the relative simplicity of the model and the purpose of its use. Details of the model can be found in [1].

The model was used to simulate various changes in system parameters. It was shown that the COP could be improved by increasing the air-flows and in particular those in the ambient circuit. These results were verified with tests which also showed that larger improvements of COP could be obtained at reduced air-flows since the reduction in fan power consumption was more significant than the reduction in cooling capacity. The control of evaporator superheat could also contribute to improving the COP. It was proposed that by using an EEV the degree of superheat could be accurately controlled eliminating the problems associated with TXV control and giving improved performance. The savings however may not justify the additional costs of installing the EEV and required accessories for its function.

Changes in the refrigerant charge were found to have more effect at high ambient temperatures, with the cooling capacity being maximised with only small changes in power consumption. The highest ambient temperature operation limit could however be adversely affected by overheating of the compressor.

CONCLUSION

In the absence of documented research, the measurements carried out on a standard on/off AC have shown that the COP is reduced to values considerably below that of constant operation – and rated – conditions. In part-load conditions, power consumption does not decrease proportionally to the changes in load, but the COP reaches very low values. This is more so at low enclosure loads and high ambient temperatures. The operation also showed large temperature variations inside the enclosure – this was made worse by limits on the rate of compressor cycling imposed as a protection to its lifetime. For low loads and high ambient temperatures, some problems with restart were observed although a time interval is already allowed for pressure rebalance. It was also shown that low control set-points for enclosure temperature can significantly increase the power consumption.

The implementation of VRF in an AC has been carried out with success. The installation of a variable-speed compressor within a standard AC unit and the experiments performed show that the energy savings that can be achieved are considerable. For the rated full-load condition of L35L35, the COP was improved from 1.86 to 2.12 – a savings of 138W or 14%. This improvement increases at part load where the compressor slows down and operates in a more efficient manner. The savings range between 8 and 32% and 105 to 280W. It was shown that compressor consumption was reduced between 11 and 68%, but the percentage savings are less for the total consumption due to the fixed consumptions of the air circulation fans.

For most conditions, the enclosure temperature could be controlled to a stable value with a flat enclosure temperature profile. The tests however also showed that the capacity of the AC will still be too large for some load and ambient temperature combinations and therefore on/off capacity-control will still be necessary, resulting in some temperature instability. However, since the compressor would have switched off at its lowest speed, the pressures within the system are low enough to allow restart in the minimum time suggested by the compressor manufacturer and the effects on enclosure temperature can be minimized. The tests also showed that superheat control by the TXV was inaccurate and hunting behaviour was observed in some conditions.

Since power consumption is highly dependent on the enclosure load and ambient temperature, and these are expected to vary during the AC's operation life, a model is necessary to estimate the operating condition and consumption of the AC. A software model was created using Microsoft Visual Basic for Applications and which could be used within Microsoft Excel. For a given set of ambient temperatures and enclosure loads, the model estimates the power consumption of a standard and a VRF AC and calculates the savings possible. For the standard tested conditions the model was accurate to within 5%. When applied to various scenarios, savings of 18-25% were achieved.

Besides those attributed to the compressor, the system efficiency can be improved further by other changes to the AC design – the main areas of improvement are reduction in parasitic and thermal losses, fans and air-flows, evaporator superheat control and heat-exchanger design. A mathematical software model of the AC was developed to evaluate the improvement possibilities numerically without the need for extensive testing. The model errors compared to test results were acceptable considering the relative simplicity of the model and the purpose of its use.

The work carried out in this project has shown that for a varied range of scenarios an energy saving of 18 to 25% is possible for a VRF unit compared to a traditional AC. The potential for further improvement in efficiency was also demonstrated.

Savings are also indirectly obtained through reduced electronic component failure associated with the stable temperature maintained. Considering also ever increasing energy costs and greater consumer environmental awareness, the possible economical and commercial benefits of this project are evident.

ACKNOWLEDGEMENTS

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