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Comparing *COP* Optimization with Maximizing the Coefficient of System Performance for Refrigeration Systems in Supermarkets

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ABSTRACT

In recent years the energy usage of supermarkets, in particular that of their refrigeration systems, has been investigated using a variety of approaches, such as floating pressure set points and integrating the heating and refrigeration systems. Something which has not yet attracted much attention is the energy consumption of the dry condenser fans in refrigeration systems. This is surprising as it has been shown for comparable installations that including the energy consumption of these fans when optimizing the system efficiency was beneficial. To address this deficit, *COP* maximization has been compared to optimizing the Coefficient of System Performance (*COSP*). The simple refrigeration system used for this investigation was based on a commercially available R404A/CO_2 system comprising the basic components, with the condenser having extractor fans. The results show that, when the outdoor temperature is below about 15°C, there is no observable difference between these two approaches. However, when the ambient temperature increases beyond this threshold, the control method which optimizes *COSP* is significantly better for part load conditions. This indicates that maximizing the *COP* can lead to a sub-optimal system in terms of energy consumption under part load conditions. When the refrigeration system is at its full load point, however, both approaches produce similar results again.

1. INTRODUCTION

Supermarkets have been acknowledged as very energy intensive buildings with a large potential for efficiency improvements. As the refrigeration of a typical supermarket makes up approximately half of its energy usage, considerable efforts have been made to improve these systems (Arias and Lundqvist, 2006). One area of interest is the integration of the refrigeration system with the HVAC system of a supermarket. Arias and Lundqvist (2006) studied the difference in energy use between floating pressure control and heat reclaim systems. They found that, theoretically, it would be beneficial to implement both ideas at the same time, but they also point out that practical challenges need to be met before such a system can perform satisfactorily. A number of different integration topologies were investigated by Cecchinato *et al.* (2010) who concluded that any integration is superior to two stand-alone systems. Another question of interest is if and how mechanical sub-cooling may improve efficiency. Based on their research Thornton *et al.* (1994) suggest that the optimum amount of heat exchange area allocated to sub-cooling is about 10%. The two efficiency improvement suggestions summarized here so far, i.e. integrating the HVAC system with the refrigeration system and sub-cooling, were combined by Yang and Zhang (2010). Their results showed that the savings potential of such a complex system depends on careful design and on the ratio between HVAC load and the refrigeration system. At the other end of the complexity scale, low cost improvements have been also proposed and include suggestions such as cleaning the heat rejection area of a condenser and proper maintenance (Carbon Trust, 2011).

Despite this comprehensive approach to reducing energy use in supermarket refrigeration systems it seems that there has not been a thorough investigation of the interaction between the power consumption of dry condenser fans and the compressor power. The only mention of this that has been located claims that the "fan power is only a small fraction of the total power consumption" (Ge and Tassou, 2000). Research in related fields, though, suggests that there is some merit in investigating this interaction more closely. This interplay is well captured by Manske *et al.* (2001) who studied an industrial refrigeration system with an evaporative condenser. They point out that there is a trade-off between the energy used by the compressor and the energy consumption of the fans; which leads to an optimization problem. Their investigation finds that for a system with minimum overall energy use there is a strong, almost linear relationship between the outdoor wet-bulb temperature and the condenser pressure, but virtually no relationship with the cooling load. Yu and Chan have investigated the same interaction for chillers and have published extensively in this area. Their findings for various systems with dry condensers (Yu and Chan, 2005, Yu *et al.*, 2006, Yu and Chan, 2008) differ from those by Manske *et al.* (2001) for an evaporative one. Yu and Chan's results indicate that overall energy consumption depends, not only on the outdoor temperature, but also on the cooling load.

The above suggests that there might be a savings potential for supermarkets when the fan-compressor interaction is considered, but that this interplay has not been studied for supermarkets in great detail. Hence, this paper investigates this interplay and starts by defining *COP* and *COSP* and discussing the difference between them. This is followed by investigating the relationship between *COP* and *COSP* using a simple thermodynamic model based on a real supermarket refrigeration system by means of a Matlab program. The results for various cooling loads show that optimizing the *COP* of refrigeration system may lead to higher energy consumption for the overall system, particularly under part load conditions.

2. DEFINING AND COMPARING *COP* AND *COSP*

The term coefficient of performance (*COP*) is very familiar to both refrigeration engineers and researchers. Notwithstanding that, it seems advisable to review this term to understand the necessity for the coefficient of system performance (*COSP*). Hence, this section defines and contrasts both ways of describing the efficiency of a refrigeration system.

In a handbook on refrigeration fundamentals (ASHRAE, 1997) the generic definition of the *COP* is given as:

$$COP \equiv \frac{\text{Useful refrigeration effect}}{\text{Net energy supplied from external sources}} \tag{1}$$

The denominator in this equation can be defined in a number of different ways. For instance, the same source (on page 1.8) analyses a theoretical single-stage cycle and equates the net supplied energy with the mass of the refrigerant multiplied by its enthalpy change. Obviously this does not take any compressor or motor losses into account. Probably this is why ASHRAE also gives compressor specific definitions of the energy (or rather power) supplied. In the handbook *HVAC System and Equipment* (ASHRAE, 2002), the power input is defined as either the electric power supplied to the motor terminals (for hermetic or semi-hermetic compressors) or as the mechanical power acting on the compressor shaft (for an open compressor). This short discussion shows that (a) the well-known term *COP* may lead to misunderstandings as it can mean different things to different people, and (b) it does not consider any other energy requirements of the wider refrigeration system.

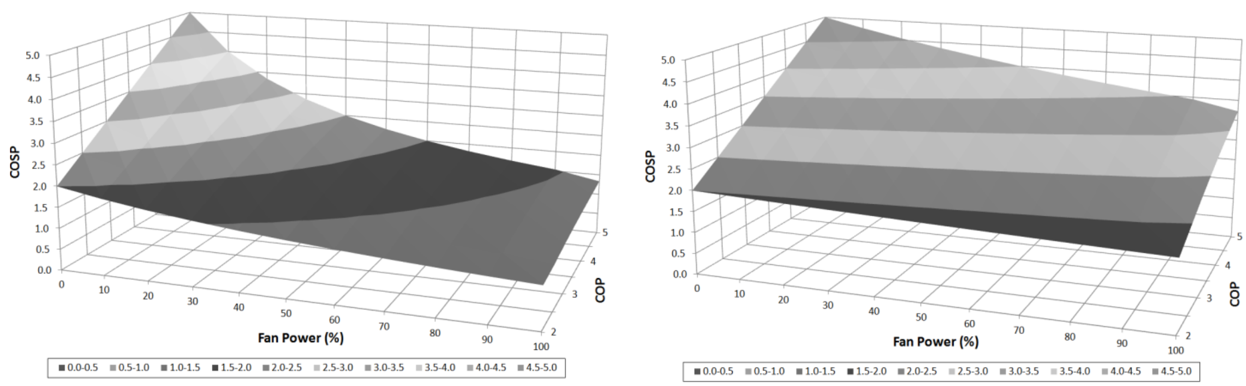
The performance figure *COSP* is used to clearly distinguish between the efficiency of the base refrigeration system, which may be characterized by a *COP* number, and the efficiency of the whole refrigeration plant. In other words, *COSP* includes the additional power consumption of equipment such as pumps and condenser fans as indicated in equation (2) (Evans, 2008).

$$COSP = \frac{\dot{Q}_e}{E_{total}} = \frac{\dot{Q}_e}{E_{comp} + E_{fan} + E_{other}} \tag{2}$$

It is also possible to create a relationship between those two efficiency coefficients as shown in equation (3). For this equation it was assumed that E_{other} is much smaller than both E_{comp} and E_{fan} and, for this reason, can be neglected. Furthermore it was assumed that the compressor is semi-hermetic (as in the refrigeration system used in the subsequent analysis) and therefore the *COP* equals $\dot{Q}_e / E_{comp}$.

$$COSP \cong \frac{COP}{1 + COP \, \frac{E_{fan}}{\dot{Q}_e}} \tag{3}$$

When analyzing equation (3) it is obvious that when the fans are switched off, the *COP* is equal to the *COSP*. A less apparent result is that when $\dot{Q}_e$ (the cooling load) increases, the relative importance of the energy use of condenser fans diminishes. On the other hand, as the *COP* increases, so does the influence of fan power consumption. Figure 1 visualizes the results of this equation for one part load point ($\dot{Q}_e$ = 20% $\dot{Q}_{e,max}$) and for full load (the maximum fan power is assumed to be 10% of $\dot{Q}_{e,max}$). It clearly shows that the *COSP* is influenced by the condenser fan, particular under part-load conditions.



Cooling load = 20% of maximum   Cooling load = 100% of maximum

Figure 1: The influence of fan power consumption and *COP* on *COSP*

3. METHOD OF MODELING

The Matlab program used to investigate the different power requirements for *COP* and *COSP* optimized refrigeration systems for the temperature range from 0°C to 35°C under four load conditions (7.5kW, 15kW, 30kW and 60kW) is shown in Figure 2. This program took into account the interaction amongst *COP*, E_{fan} and $\dot{Q}_e$, something not considered in the previous section where these variables were treated as mutually independent. It models the simple vapor compression cycle depicted in the left-hand panel in Figure 3, below, consisting of an ideal evaporator, compressor, expansion device and an ideal condenser with a 2.9kW variable speed fan (GEA Searle, 2013).

The refrigeration system in Figure 3, below, is based on the primary side of a commercial R404A/CO_2 refrigeration system for supermarkets (Searle Manufacturing Company, 2009) and has a maximum cooling load capacity of 60kW. Although the refrigerant R404A is a zeotropic mixture, its behavior in the 2-phase region was approximated by a pure substance so that almost all of the usual simplifying assumptions could be applied (for a list see, for instance, Ameen (2006)). The most notable exception was that the refrigerant was allowed to enter the sub-cooled region. The right-hand panel of Figure 3 shows the corresponding *p-h* diagram. At the starting point of the refrigeration cycle the refrigerant is a saturated vapor at -9°C. This vapor is then isentropically compressed from h_1 to h_2. This requires a power input of E_{comp} for the semi-hermetic compressor. The heat $\dot{Q}_{air}$ is rejected from the refrigerant into the air stream at constant condenser pressure. The minimum condenser pressure $p_{c,\,min}$ corresponds roughly to the minimum pressure across the expansion device for the proper operation of the real system. The maximum condenser pressure of 18.5bar_a is when the high pressure switch of the commercial system trips. After the heat is rejected the refrigerant is either a saturated (h_3) or sub-cooled liquid (h_3'). Next, this liquid undergoes an adiabatic expansion process from p_c to p_e before it absorbs heat from the evaporator. This transforms the refrigerant back into a saturated vapor at the exit of the evaporator.

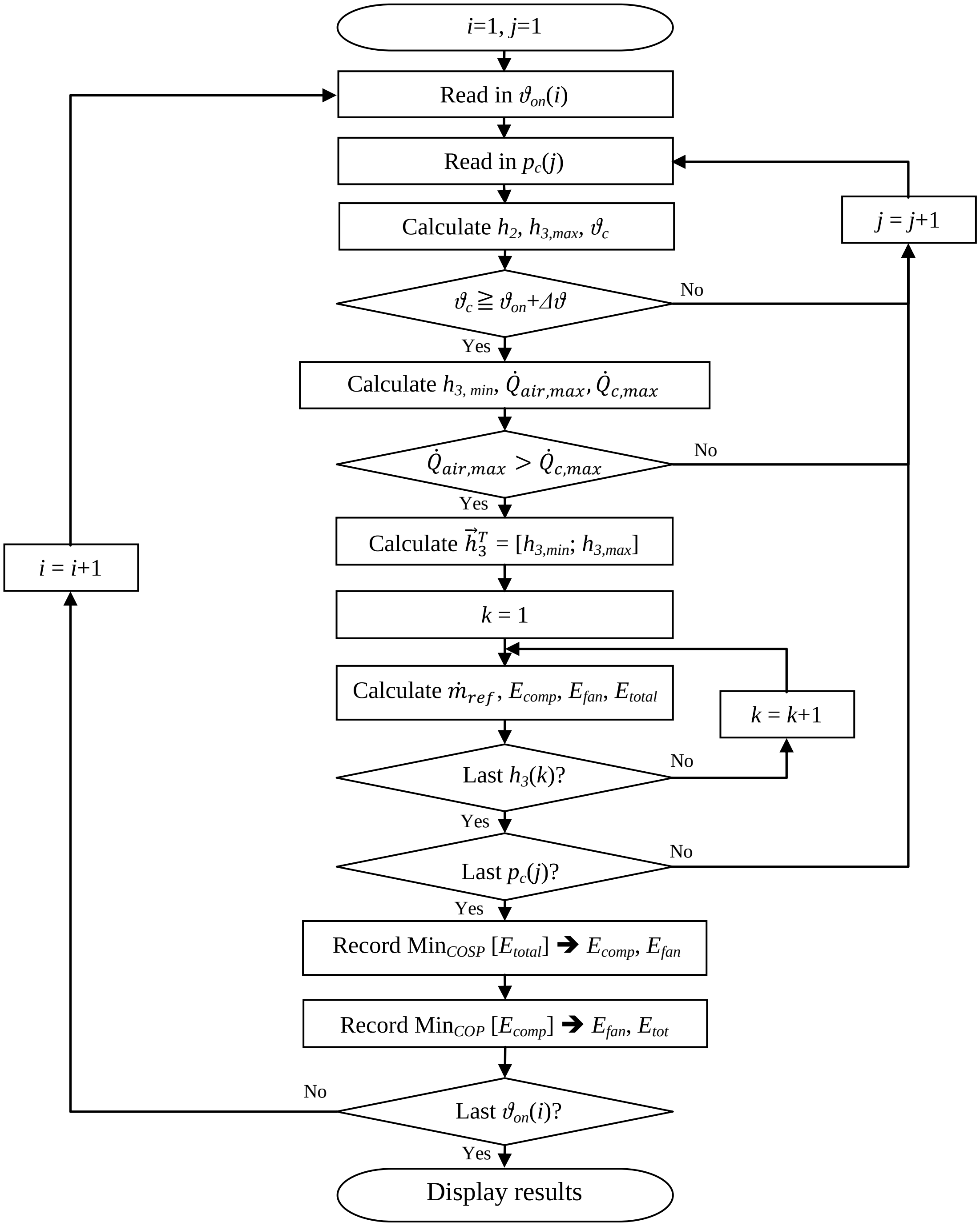


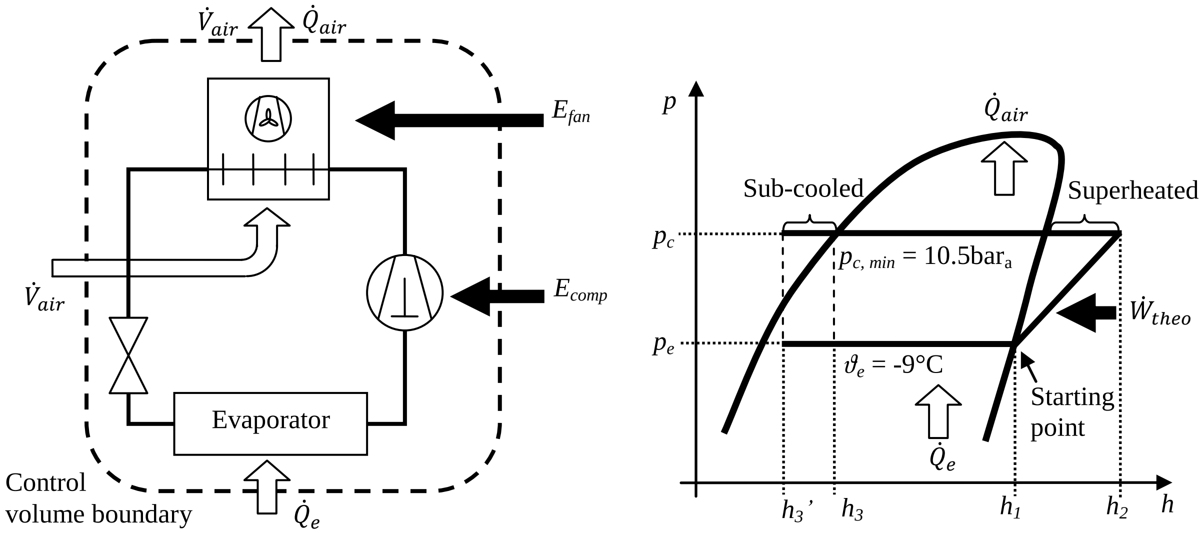
Figure 2: Flowchart – Matlab program for comparing *COP* and *COSP* optimization at a specific load point

The data necessary to compute the enthalpies came from an R404A *p-h* diagram. The enthalpy h_2 was computed using the line of constant entropy originating from the saturated vapor point at -9°C and was modeled by equation (4). The enthalpy $h_{3,\,max}$, (the enthalpy of the saturated liquid at the condenser pressure) was calculated with equation (5) which was a good approximation over the pressure range of interest.

(4)

$$h_2 = 1.394 \, \frac{\mathrm{kJ}}{\mathrm{kg\ bar_a}} \, p_c + 365.93 \, \frac{\mathrm{kJ}}{\mathrm{kg}}$$

$$h_{3,max} = 4.642 \, \frac{\mathrm{kJ}}{\mathrm{kg\ bar_a}} \, p_c + 179.51 \, \frac{\mathrm{kJ}}{\mathrm{kg}} \tag{5}$$



Refrigeration system *p-h* diagram

Figure 3: Simple refrigeration system and its *p-h* diagram

The possible minimum enthalpy $h_{3,\,min}$ of the refrigerant at the exit of the condenser was determined with an averaged specific heat constant of 1.6kJ/kg/K for the sub-cooled region. The air temperature leaving the condenser in equation (6) (i.e. ϑ_{off}) is the condenser temperature approximated by equation (7) minus an arbitrary constant of 2K which acknowledges that ϑ_{off} cannot reach the condenser temperature ϑ_c.

$$h_{3,min} = h_{3,max} - c_{ref} \left(\vartheta_{off} - \vartheta_{on} \right) \tag{6}$$

$$\vartheta_c = 2.742 \, \frac{\mathrm{°C}}{\mathrm{bar_a}} \, p_c - 9.793\mathrm{°C} \tag{7}$$

The compressor power was calculated with equation (8) in which η is an efficiency constant taking various losses into consideration. The constant was set to 0.3 to give results comparable to the readings of the real system. Equations in (9) make use of the fan law to calculate the power used by the fans, E_{fan}, and assume that the air flow rate is kept to a minimum. The total power of the system is the sum of E_{comp} and E_{fan}.

$$E_{comp} = \dot{Q}_e \, \frac{h_2 - h_1}{h_1 - h_3} / \eta \tag{8}$$

$$\frac{P_{fan}}{P_{max}} = \left(\frac{\dot{V}_{air}}{\dot{V}_{max}} \right)^3 \Rightarrow P_{fan} = P_{max} \left(\frac{\dot{m}_{ref} \, (h_2 - h_3)}{\dot{V}_{max} \, \varrho_{air} \, c_{p,air} \left(\vartheta_{off} - \vartheta_{on} \right)} \right)^3 = P_{max} \left(\frac{\dot{m}_{ref} \, (h_2 - h_3)}{\dot{Q}_{air,max}} \right)^3 \tag{9}$$

For the four cooling loads 7.5kW, 15kW, 30kW and 60kW (full load), E_{fan}, E_{comp} and E_{total} were calculated at 0.5°C temperature steps for each possible condenser pressure and h_3 and recorded in arrays. The minimum of E_{comp} and E_{total} for each temperature step were located in their respective arrays and the other corresponding power values selected to produce the graphs in Section 4 'Results of software model'.

The program in Figure 2, above, compared the possible maximum heat absorbed by the air, $\dot{Q}_{air,max}$, with the maximum heat rejected by the condenser. These values were calculated with equations (10) and (11).

$$\dot{Q}_{air,max} = \dot{V}_{max}\, \varrho_{air}\, c_{p,air}\left(\vartheta_{off} - \vartheta_{on}\right) \tag{10}$$

$$\dot{Q}_{c,max} = \dot{Q}_e \frac{h_2 - h_{3,max}}{h_1 - h_{3,max}} \tag{11}$$

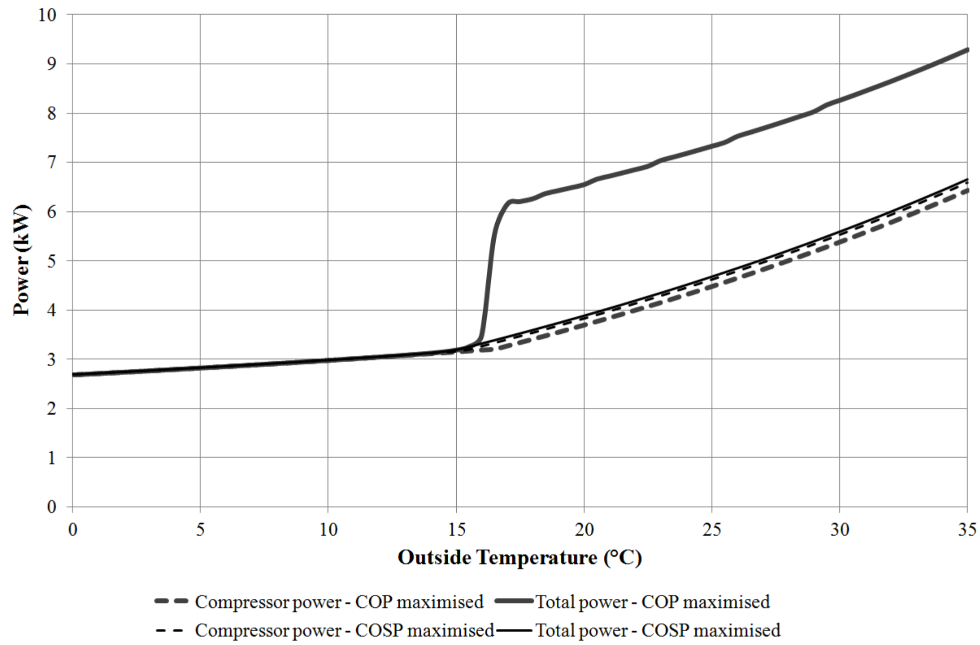


Figure 4: Comparing compressor and total power of *COP* and *COSP* optimized systems at a cooling load of 7.5kW

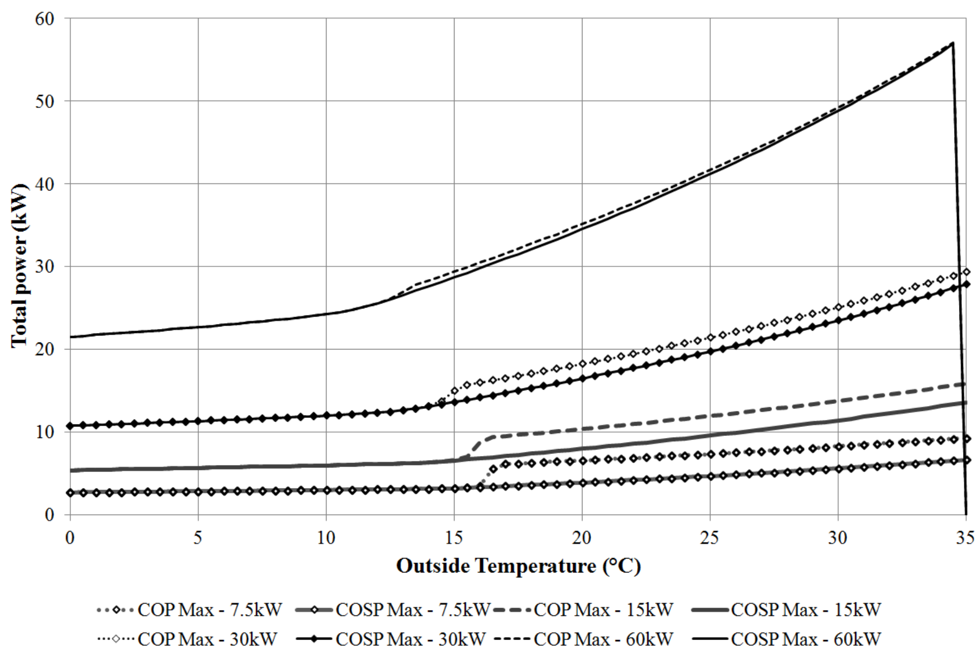
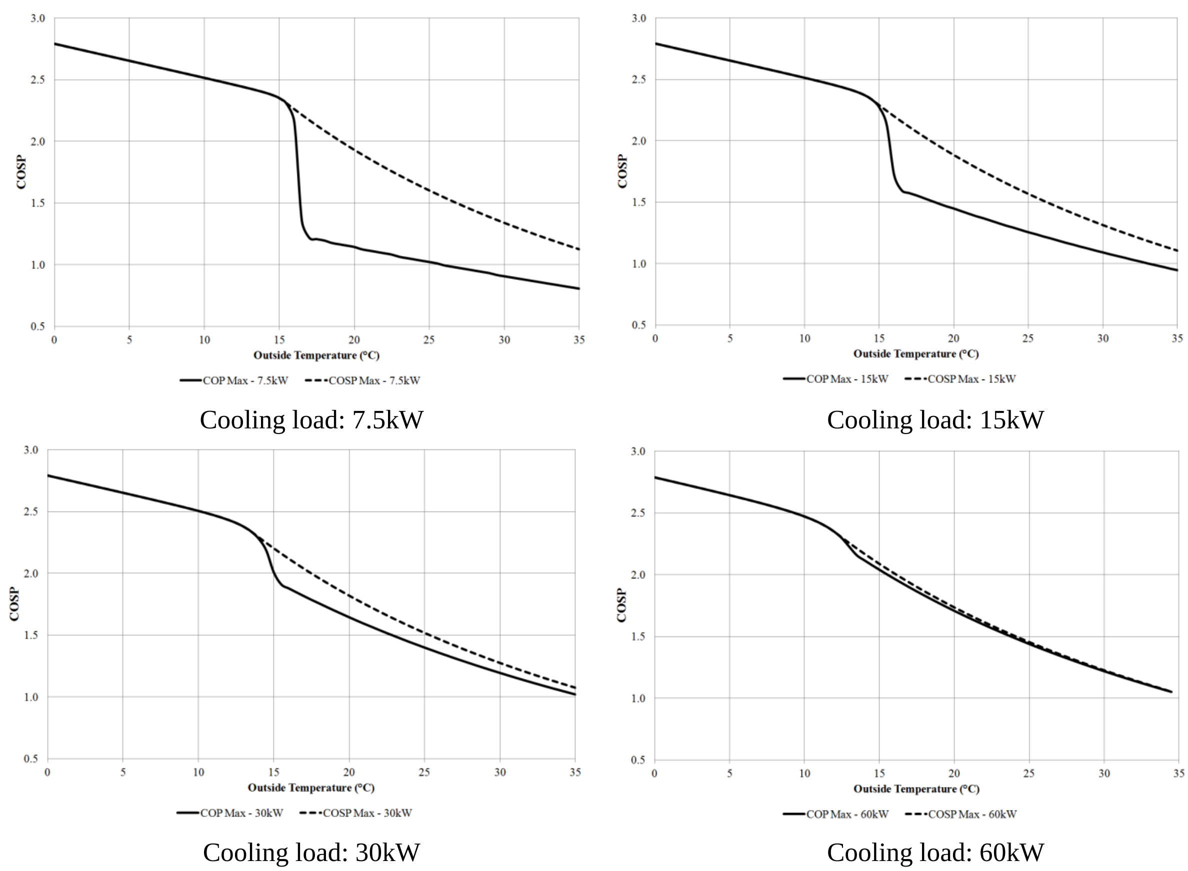


Figure 5: Comparing total power consumption of *COP* and *COSP* optimization systems at all four load conditions

4. RESULTS OF SOFTWARE MODEL

The figures discussed here concentrate mainly on the power consumption of the complete system under the two optimization approaches. Figure 4, above, is an exception as it also includes traces of compressor power and was included to show the effect of both methods more clearly. In this figure all four lines follow each other up to an outside temperature of approximately 16°C. After that they separate, showing two pairs of lines. The top and bottom traces relate to the *COP* optimized system. The thinner lines in between them are those for the *COSP* optimization approach. These graphs show that the compressor power for the maximum *COP* is indeed lower than that for the *COSP* optimized system, albeit only marginally. The difference in total energy input to gain this advance can be gauged by the two solid lines. The thicker of these lines illustrates that to increase the *COP* a relatively high penalty has to be paid in the second half of the temperature range. On the other hand, the thinner line for the *COSP* optimized system is just above its compressor consumption graph. The difference between compressor consumption and total power required is the fan power. The fan power for the *COP* optimized system starts to increase sharply at about 16°C and reaches its maximum shortly thereafter. Only after that is it allowed to drive the compressor harder (otherwise the system cannot be considered *COP* optimized). The sharpness of the increase in E_{fan} is due to the following two relationships: (a) the volumetric flow rate and the temperature difference $\vartheta_{off} - \vartheta_{on}$ have an inverse relationship, and (b) the fan power and the volumetric flow rate are related by a cubic equation (see Equation 9). The overall conclusion for Figure 4 is that by driving the compressor slightly harder the power requirements for the whole system can be significantly reduced for an outside temperature above approximately 16°C.

As suggested in Section 2 the relative importance of the fan power with respect to the change in *COSP* diminishes with increasing cooling load. This conclusion is also supported by Figure 5, above, in which only the total power curves are displayed. This plot shows that both the relative and absolute distances between the pairs of lines decrease as the cooling load increases from 7.5kW to full load. As a matter of fact the two lines for full load become increasingly indistinguishable as temperature increases. The sharp drop of the full load lines after 34.5°C emulates the shutdown of the real refrigeration plant when the high temperature switch has tripped.



Cooling load: 7.5kW

Cooling load: 15kW

Cooling load: 30kW

Cooling load: 60kW

Figure 6: *COSP* for different cooling loads and for *COSP* and *COP* optimization

The four panels in Figure 6 show the *COSP* for the cooling loads 7.5kW, 15kW, 30kW and 60kW. For these graphs the respective cooling loads were divided by the relevant total power. All graphs start at approximately 2.8 and for the first few degrees afterwards fall at about the same rate. Then it can be observed that the lower the cooling load is, the longer the two lines stay together. When these two lines separate the drop of the *COP* optimized system is steeper and deeper or, in other words, the efficiency drops more significantly for such a system under low load conditions. The best system is *COSP* maximized with a cooling load of 7.5kW, but even this system only achieves a *COSP* of just over 1.1 at 35°C.

As Figure 4, above, shows, the main difference in total power consumption between the two optimization approaches is not so much the difference in compressor power, but in the condenser fan power. Therefore Figure 7 displays the fan power traces to show more clearly how their power use differs under different load conditions. It can be seen that, although the temperature has some influence, in actual fact the cooling load is the determining factor. All graphs show a 'kink' which corresponds to the point at which the minimum condenser pressure is no longer sufficient and needs to be increased. The full load curve shows a further steep rise at the end of the temperature range when the maximum condenser pressure has been reached.

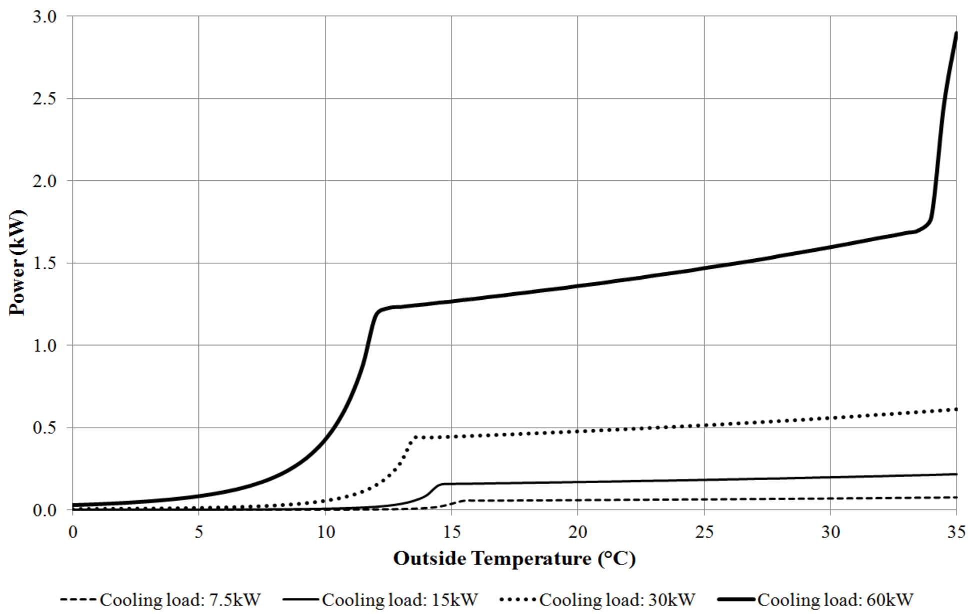


Figure 7: Power consumption of condenser fans for the *COSP* optimized system with different cooling loads

5. DISCUSSION

The *COSP* equation derived in Section 2 indicates that the fan power has an appreciable effect on the overall efficiency, but that its relative importance diminishes with increasing cooling load. The software model used above agrees with this at temperatures above approximately 15°C. Below this there is no appreciable difference between a *COP* and a *COSP* maximized system. This is so because a real supermarket refrigeration system has to provide a certain pressure for the proper operation of the expansion device regardless of the outside temperature. When this pressure is no longer sufficient a marked difference between the two optimizing approaches is apparent for all examined part load conditions. This demonstrates that, under those conditions, the *COSP* equation holds. In addition, both the equation and the software model show that with growing cooling load the difference in energy consumption between *COP* and *COSP* optimization diminishes. For full load this difference has virtually vanished.

Based on the results presented here, it can be concluded that there is an appreciable savings potential when the condenser fan power consumption is considered, especially if one considers that refrigeration systems frequently work in the part load region. This conclusion is not the same as the one presented by Ge and Tassou (2000) who suggest that the condenser fan power is only a small part of the overall energy consumption. However, if it is assumed that these researchers only considered a system under full load, then their conclusion is comparable with the results here. On the other hand, the conclusion that the fan power can make up a significant part of the overall consumption under part load conditions agrees with the results for water chillers with dry condensers. In such a system, energy efficiency is not only dependant on the outdoor temperature, but also diminishes with load increase. As the results in the papers by Yu and Chan have been presented with respect to load (Yu and Chan, 2005, Yu *et al.*, 2006, Yu and Chan, 2008) and not with respect to outside temperature, it is difficult to completely correlate their results with the findings here.

All of the above suggests that investigating the interplay between the power consumptions of condenser fans and compressors should result in energy savings. However, this conclusion is only based on a software model. Therefore this model should be rigorously validated against real consumption data to verify the results above. In addition the control algorithm for the condenser fans should be examined because the one used in this study monitors essentially only the condenser pressure (Resource Data Management Ltd, 2013). This has the effect that when the condenser set point is reached the fans switch on regardless of the cooling load. This is contrary to what Figure 7 demonstrates, which is that the control algorithm for the fans of a *COSP* optimized system should take into consideration the outside temperature and the cooling load. To compute this load (or, alternatively, the required heat rejection rate at the condenser) condenser pressure readings are insufficient and the mass flow rate of the refrigerant is also needed. This leads to the conclusion that the current algorithm is sub-optimal and should be improved.

6. CONCLUSIONS

The main conclusions of this work can be are summarized as follows:

- Optimizing a refrigeration plant for the *COP* of the core system may not minimize the overall energy consumption.
- When under part load conditions and at higher ambient temperatures (of approximately 15°C and above), the *COSP* maximized system investigated here uses appreciably less energy than its *COP* optimized counterpart.
- The software model should be validated against real consumption data.
- To minimize overall energy consumption condenser fan controllers should use ambient temperature and cooling load (or condenser load) as control inputs.
- Further work should suggest improved control algorithms for condenser fans.

NOMENCLATURE

$c_{p,\,air}$	Specific heat capacity of air	(kJ/kg/K)
c_{ref}	Average specific heat capacity of refrigerant in sub-cooled region	(kJ/kg/K)
COP	Coefficient of performance	(-)
$COSP$	Coefficient of system performance	(-)
h_1	Specific enthalpy at compressor input port	(kJ/kg)
h_2	Specific enthalpy at compressor output port	(kJ/kg)
h_3, h_3'	Specific enthalpy at condenser output port	(kJ/kg)
$h_{3,\,max}$	Possible maximum of specific enthalpy at condenser output port	(kJ/kg)
$h_{3,\,min}$	Possible minimum of specific enthalpy at condenser output port	(kJ/kg)
$\dot{m}_{ref}$	Mass flow rate of refrigerant	(m^3/s)
i, j, k	Loop index	(-)
E_{comp}	Power input into compressor(s)	(kW)
E_{fan}	Power input into condenser fan(s)	(kW)
E_{other}	Power input into other devices	(kW)
$\dot{W}_{theo}$	Theoretical power into refrigeration cycle	(kW)
E_{total}	Total power of system	(kW)
p_e	Evaporator pressure	(bar)
p_c	Condenser pressure	(bar)
$\dot{Q}_{air}$	Heat rejection rate to air	(kW)
$\dot{Q}_{air,max}$	Possible maximum of heat absorbed by air	(kW)
$\dot{Q}_c$	Heat rejection rate	(kW)
$\dot{Q}_{c,max}$	Possible maximum of heat rejection rate required	(kW)

$\dot{Q}_e$	Cooling load	(kW)
$\dot{V}_{air}$	Volumetric flow rate of air through condenser	(m^3/s)
$\dot{V}_{max}$	Maximum volumetric flow rate of air through condenser	(m^3/s)

Greek letters

η	Efficiency of motor and associated devices	(-)
ϑ_e	Evaporator temperature	(°C)
ϑ_c	Condenser temperature	(°C)
ϑ_{on}	Temperature of air entering condenser	(°C)
ϑ_{off}	Temperature of air leaving condenser	(°C)
$\Delta\vartheta$	Arbitrary temperature offset	(K)
ϱ_{air}	Air density	(kg/m^3)

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