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Reducing power consumption in multi-compressor refrigeration systems

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ABSTRACT

An experimental analysis of compressor operation in a large refrigeration system was undertaken and a model for optimal compressor operation for energy efficiency was developed. The system used 5 screw compressors and ammonia as the refrigerant, with slide valves to regulate the compressors and match their refrigeration capacity with product freezing loads. Compressors in the existing system can operate simultaneously with reduced capacities, which results in reduced energy efficiency. Optimized operation was made both with and without a variable frequency drive. The results showed that the most electrical energy can be saved during days when not all of the tunnels were loaded. It is assumed that € 30 000–50 000 can be saved per year by optimizing the operation of the refrigeration system.

Diminution de la consommation d'électricité des systèmes frigorifiques à plusieurs compresseurs

Mots clés : système frigorifique ; système à compression ; compresseur à vis : modélisation ; optimisation : régulation ; vitesse variable ; économie d'énergie

1. Introduction

Throughout history, fishing and the seafood industry have made important contributions to Norwegian society. Today, seafood is one of Norway's largest exports (after oil, gas and metals), mostly in fresh, frozen or dried form, and exported primarily to Denmark, Russia and Japan (Statistics Norway, 2006). Because of the increasing interest in Norwegian seafood and its substantial development potential, the seafood industry has been designated as one of the government's priority areas. Norway's stationary energy system is to a large extent dependent on electricity, which is mainly based on hydropower and has been fairly inexpensive until this last

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Nomenclature

AC	alternating current
$a_{i,j}$	slope for part-load curve
$b_{i,j}$	intercept for part-load curve
COP	coefficient of performance
DC	direct current
$d_{i,j}$	operational variable
$f(x)$	object function in optimization problem
i	compressor (5 alternatives)
j	interval (5 alternatives)
K4, K5, K6, K9, K11	included compressors
P_{max}	power consumption at maximum capacity [kW]
PWM	pulse-width-modulated drive
Q_{0max}	maximum cooling capacity [kW]
Q_{0tot}	total cooling capacity [kW]
RSW	refrigerated seawater
$x_{i,j}$	relative cooling capacity

decade. Consequently, industry has not focused on electricity savings. This is clearly evident in the seafood industry, where the energy consumption per kilogram of processed fish varies significantly from one factory to another (Enova, 2002). Electricity prices are expected to continue to increase, which will force the industry to re-evaluate its energy systems and process controls. There is also an interest for more eco-products on the market. A reduction in electrical energy usage will not only be environmental friendly in itself, but would also lead to lower emissions of pollutants such as CO_2. The opportunities for energy savings in the seafood industry are many. Refrigeration systems are the main electrical consumer, and are used for chilling, ice production, cold storage and freezing of fish. Possible improvements include better dimensioning of the system (currently, the design cooling load is often higher than the operating cooling load), better system regulation of the components, and a more uniform use of energy over a 24-h period. A larger analysis showed that much could be gained by integrating refrigeration systems with heating systems (Gjøvåg, 2004; Aprea et al., 2007).

The objective of this paper was to use data from part-load operation (Widell and Eikevik, 2008) in a model that provided a more optimal operation of the compressor system, both with and without a variable frequency drive. The model gives the potential of electricity savings. The economic benefits of this approach as compared to today's systems have been calculated. We have not detailed how to implement this optimizing model in an actual control system. Examples of this can be found in Leducq et al. (2006). They have described a non-linear control algorithm for predicting optimal operation. Several objectives can be defined simultaneously and they have to be weighted against each other. The laboratory plant described in this paper has a single compressor with a variable speed drive, but the algorithm can be used on multi-unit systems also.

2. Reducing refrigeration capacity with slide valve regulation

Screw compressors have often been used by the Norwegian seafood industry when large freezing capacity is needed and only one-stage compression is required. A one-stage screw compressor can work with a higher pressure ratio than a one-stage reciprocating compressor. Many screw compressors employ slide valve regulation, which should give continuous capacity control between 10% and 100%. A clear drawback of this system is its poor efficiency when refrigerating capacity is reduced below 100%. The power requirement is almost the same at part-load operation as it is at full load (Brendeng, 1979). The poor efficiency of part-load operation is mainly related to two factors, friction and volume ratio change. When the slide valve is in use, a slot opens for the refrigerant to vent back to the suction side. This leads to friction in the gas and a change in the volume ratio of the compressor (Stoecker, 1998). The design data for part-load efficiencies of screw compressors are often higher than they are in reality. Because the edge of the slide valve slot usually is situated at 80% of the swept volume, efficiencies fall rapidly as soon as the slide valve is opened. The slide valve position is not the same as the cooling capacity. At 80% of slide valve position, the cooling capacity can be as low as 50% of full capacity (Gosney, 1982). Fig. 1 illustrates this relationship. The curve is different for different types of compressors. Values from the compressor manufacturer have been used in our calculations. Another important aspect is the control system. Many fish processing industries have very simple control systems, where there is little communication between components. The compressors are controlled with a thermostat and more than one compressor can work at a partial load at the same time.

3. Reducing refrigeration capacity by varying compressor speed

A more efficient way to modify the refrigeration capacity than with the slide valve is to vary the compressor speed. This can be done with a variable frequency drive connected to the AC

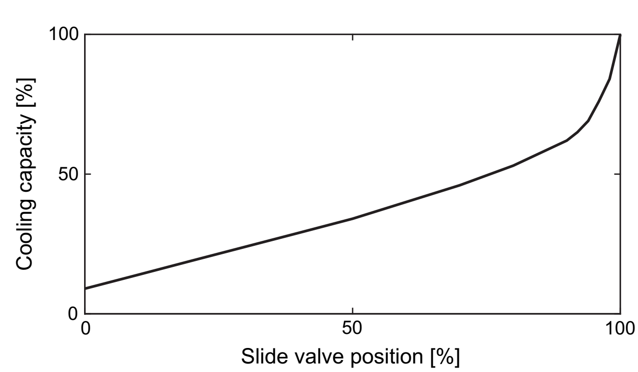

Fig. 1 – Theoretical curve showing cooling capacity as a function of compressor slide valve position.

motor. The most common one is a pulse-width-modulated drive (PWM drive). This operates firstly by converting the AC voltage into a fixed DC voltage. Secondly, the DC voltage is inverted back to AC in an inverter, where the frequency and voltage can be selected within a certain range (Avallone and Baumeister, 1996). Most of the literature found in this area analyses varying compressor speed using reciprocating or scroll compressors and not screw compressors. The frequency range for these types of compressors is 30–50 Hz for reciprocating compressors and 15–50 Hz for scroll compressors. Reciprocating compressors cannot use lower frequency than 30 Hz because of noise, vibration and lubrication troubles (Aprea and Renno, 2004; Aprea et al., 2006). We assume in this paper that screw compressors also have a lower frequency limit of 30 Hz. The suggested frequency range for the compressors at the processing plant (described in Section 4.1) is 30–67 Hz. Most motors are built for frequencies from 50 Hz to 67 Hz, so that it is possible increase the frequency of the compressors beyond 50 Hz.

4. Measurement and methods

A processing plant for pelagic fish was chosen for the analysis of the compressor regulation system. The plant is situated in the south-western part of Norway, and has an annual production volume of 50 000 ton of frozen fish and fish products. The plant was built in 1993.

4.1. *The processing plant refrigeration system*

The refrigerant system consists of two subsystems, which together have 3 reciprocating and 8 screw compressors, ranging in cooling capacity size from 180 kW to 795 kW (−40/+20 C). The reciprocating compressors work in two stages and an economizer is operated with the screw compressors. The refrigeration system supplies cooling to 5 batch freezing tunnels, 10 plate freezers, 3 spiral freezers and a large freezing storage area. The plant also has RSW units and ice machines. The system contains 30 ton of ammonia as the refrigerant. The freezing capacity is 625 ton of fish per day with the storage capacity at 10 000 ton (Solheim, 2006). The screw compressors are divided into two subsystems that have different pressure levels. The first has an evaporator temperature of −38 C and contains 5 screw compressors. The other subsystem includes 3 screw compressors and has an evaporator temperature level at approximately −42 C. The first subsystem (−38 C) and its compressors have been evaluated in this paper. A simplified sketch of the system can be seen in Fig. 2. Since the conditions in a large industrial plant are complex, making measurements is quite complicated and it is not possible to measure all variables. The plant is equipped with a control and monitoring system that continuously logs temperatures, pressures and other data from the refrigeration system. Two periods of 9 days each have been selected and data from these periods have been analysed. The refrigeration system was operated at full capacity during most of the first period and at reduced capacity on the second. The focus was on the compressors.

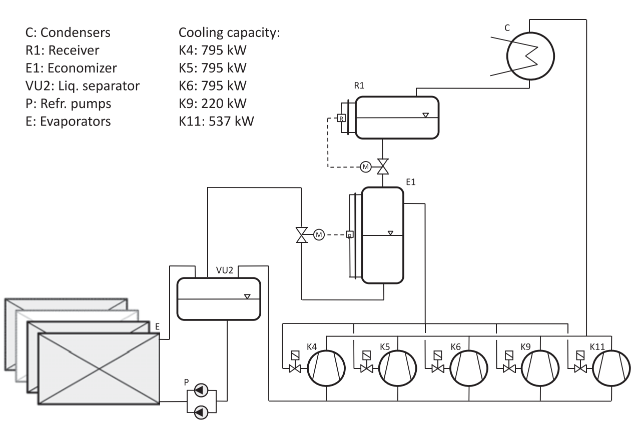

Fig. 2 – A simplified model of a part of the refrigeration plant. Only one of the two subsystems was analysed in this paper and it includes 5 screw compressors.

4.2. *Part-load capacity curves of the screw compressors*

The behaviour of the compressors was analysed and described by Widell and Eikevik (2008) and part-load capacity curves were constructed for each of the screw compressors in the system. These curves show the relative power consumption by relative cooling capacity and are used in the optimization. The closer to the ideal line, the better is the efficiency of the system. The part-load capacity curve for a compressor with variable frequency drive is given by York (Solheim, 2006) and is illustrated in Fig. 3. It can be seen from this that varying compressor speed gives a more energy efficient operation than slide valve regulation.

4.3. *Optimization model*

A linear programming model was developed for analyzing the difference between the actual (measured) system and a system with optimized operation of each compressor. The objective was to minimize power consumption, provided that the total refrigeration load requirement was met. The refrigeration loads entered into the model were from the measured period and each measured value was analysed with the model. The object function was:

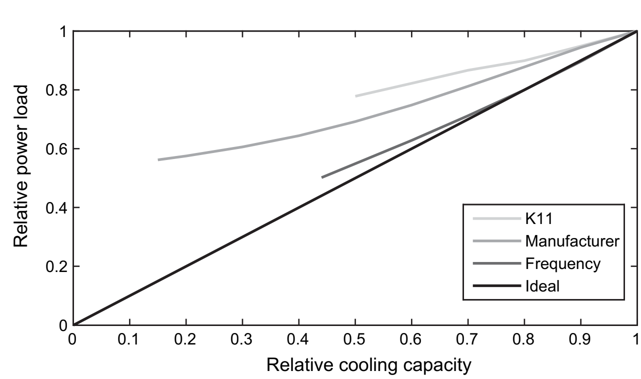

Fig. 3 – Part-load capacity curves for compressor No. 11, with calculated values, values from the manufacturer and with a variable frequency drive.

$$\min_x f(x) = \sum_i \sum_j P_{\max,i} \cdot \left[x_{i,j} \cdot a_{i,j} + d_{i,j} \cdot b_{i,j}\right] \quad (1)$$

where $P_{\max,i}$ is the power load of compressor i at maximum capacity, i stands for the different compressors and j is the interval in the part-load curve. $x_{i,j}$ is a matrix of the relative power consumption, $a_{i,j}$ and $b_{i,j}$ are the slope and the intercept from the part-load curves for each compressor. $d_{i,j}$ is a matrix of binary variables that is found together with $x_{i,j}$. It is also used in the constraints and it controls that the computer program does not allow a solution where one compressor is operated in more than one interval. The optimization problem has both equality and inequality constraints. The equality constraint is:

$$Q_{0,\mathrm{tot}} = \sum_i \sum_j x_{i,j} \cdot Q_{0\max,i} \quad (2)$$

where $Q_{0,\mathrm{tot}}$ is the total cooling capacity. This is the input for the optimization problem. $Q_{0\max,i}$ is the maximum compressor capacity for each compressor. The capacity curves are divided into intervals so that each interval has its own set of linear regression constants. The first two inequality constraints set the limits for x in each interval. The third inequality constraint sets the sum of the d-variables for one instant equal to or less than 1, which ensures that a compressor is only operating in one interval at the time. When the compressors with variable frequency drives were simulated, the data from Fig. 3 were used to find a and b. Several combinations of compressors and variable frequency drives were evaluated, but only the best ones have been presented.

4.4. *Limitations*

In reality, to avoid compressor damage, there are limitations on how often, or the times between when a compressor can be stopped and started again. This is not included in the optimization equations since these only apply to one cooling capacity at the time and not to a vector of numbers. The error from excluding this from the model is estimated to be low, due to the fact that the three of the compressors were the same size. Another limitation is that the compressors are always started and stopped in a certain order and this is also not included in the optimization. This means that a compressor with lower efficiency can be chosen instead of one with higher efficiency. It would be better for the actual system to have a more flexible way of selecting which compressor to start or stop.

5. Results and discussion

5.1. *Optimized system performance*

The part-load capacity curves were calculated based on measurement data from the first period of nine days (Widell and Eikevik, 2008). Values from both periods were used to find optimized operation of the system. The cooling capacity was calculated from the slide valve position, maximum cooling capacity and a conversion graph given by Sabroe (Solheim, 2006). Cooling capacities for both periods are shown in Fig. 4. The power consumption was calculated from relative cooling capacity, part-load capacity curves and a maximum power consumption. This simplified approach was used to make the comparison between the measured and optimized system more reliable. It was important that we compared relative values (multiplied with constant maximum values) instead of measured absolute values, since these latter varies with different factors, such as seawater temperature and mass flow in the condensers.

The first period was during the fishing season when the plant was under full production for most days. The processing plant operates all year round, but uses the freezing tunnels for only about 60 days a year. At peak production time, the large freezing tunnels are full, with a maximum capacity of 625 ton per 24 h. The first period was selected because of its large

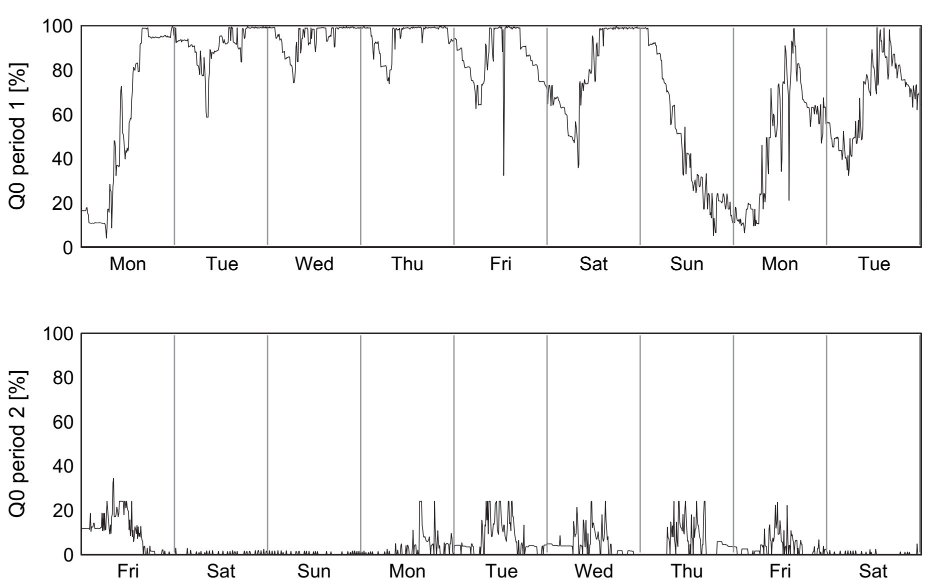

Fig. 4 – The relative total cooling capacity during periods 1 and 2.

production load. Large production leads to a more stable system with more compressors operating at full load, which was beneficial for the part-load capacity curves. We see that the greatest number of improvements can be made when the system is at part-load operation, which is during days when not all of the tunnels are loaded. Systems like this use a great deal of energy even at lower production rates because the compressors are designed for their most efficient operation at a high load.

The freezing tunnels were not in operation during the second period, which can be seen in Fig. 4. This period illustrates how the system is operated for most of the year. There is no production during the weekends; the only cooling capacity needed then is to support a small storage freezer.

The system was first optimized without variable frequency drives. All of the compressors were regulated with slide valves. In these simulations, more than one compressor could be at part-load operation, but then often above 90%. It seems to be more efficient to have two compressors at a high part-load operation than one compressor at a low part-load operation. The measured system had often more than one compressor at a low part-load operation.

The next optimizations included different alternatives for variable frequency drives. The system has 5 screw compressors, of which three are of the same size. Frequency drives were simulated in compressors K5, K9 and K11. The part-load characteristic shown in Fig. 3 was used for all of the compressors. Three different combinations of two compressors with variable frequency drives were also analysed; these were K5 + K9, K5 + K11 and K9 + K11. The results showed that it could perhaps be profitable to install one variable frequency drive, but not two.

5.2. *Coefficient of performance*

The coefficient of performance (COP) was calculated from the cooling capacity and the power consumption. Average values were calculated for each day. Graphs for the COP for both the measured and the optimized system are shown in Fig. 5 for period 1 and Fig. 6 for period 2. The greatest improvements can be made during days when not all of the tunnels were loaded, but not when the system is almost off, as it is during the weekends in period 2. Here we do not have enough data to calculate a reliable average COP. During these days the cooling load varied between zero and very low (typically with one compressor at part-load below 50%). A zero COP is not included in the average values and the optimization model could not find a correct value when the part-load was below 50%.

Cooling capacities and COPs have been sorted by cooling capacities, as can be seen in Fig. 7. From this we see that the COPs for the optimized system (gray rings and crosses) are much higher and more even than for the measured system (black dots). For high cooling capacities, COPs for both optimized and measured system are high. When cooling capacities are low, there are larger differences between the optimized system and the measured system. In the latter the COP can be different at one cooling load; this means in reality that a compressor with lower efficiency was used instead of one with higher efficiency.

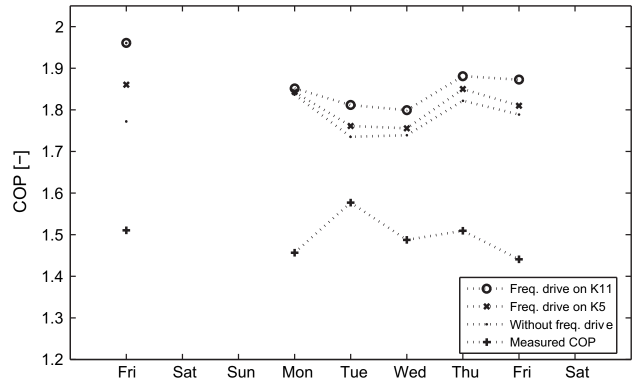

Fig. 6 – Daily average of COP for the system for period 2. The lines have been drawn to better illustrate the connections and differences between the values.

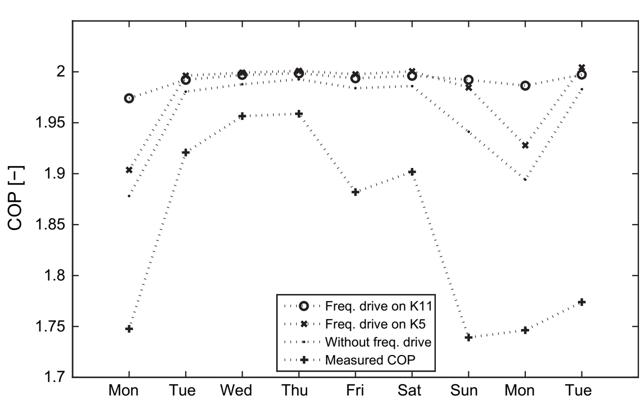

Fig. 5 – Daily average of COP for the system for period 1. The lines have been drawn to better illustrate the connections and differences between the values.

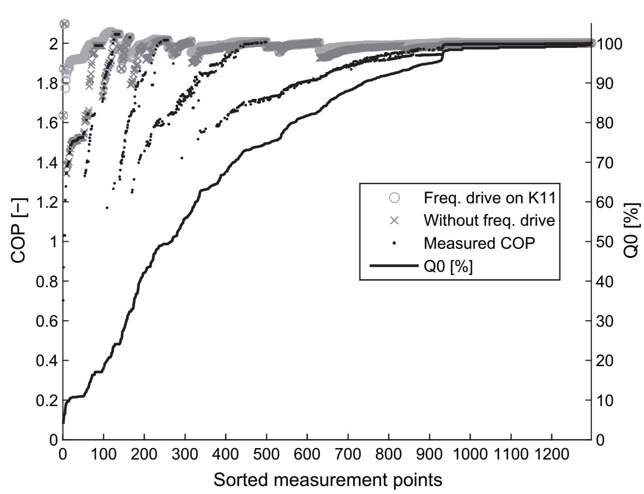

Fig. 7 – Values for COP sorted by the increasing relative cooling capacity (relative to maximum load).

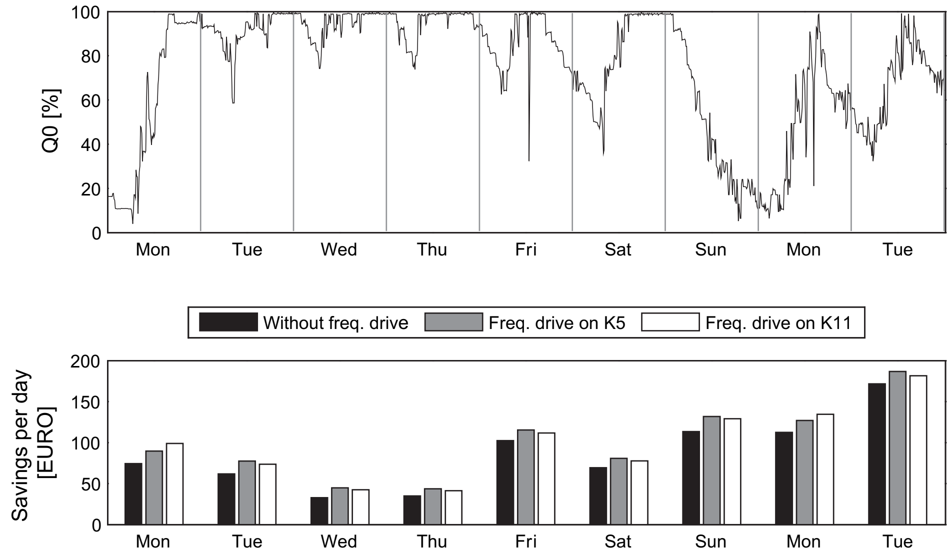

Fig. 8 – Savings per day [€] for period 1, compared with measured system, for three different optimization alternatives. Cooling capacities [kW] for the period are included.

5.3. *Electrical energy savings and cost differences*

The total electrical energy load and the electricity cost were calculated for the selected periods. The electrical energy cost was set to 0.06 €/kWh. The savings in € per day for period 1 can be seen in Fig. 8. Three different optimization alternatives were chosen, one without a variable frequency drive and two with a variable frequency drive, in compressors K5 and K11. By looking at the cooling load also displayed in the graph, we can conclude that the smallest differences between the measured and optimized system are when the production is high. It is during days when not all of the tunnels were loaded that most of the savings can be had. The difference between the optimized system with and without frequency drives is not significant. Variable frequency drive in the largest compressor, K5, seems to be favourable for most of the days during period 1. However, it cannot be concluded from this that K5 is the best compressor to connect to a variable speed drive. This kind of operation only happens for about 16 days a year. An analysis over the usage during a whole year must be undertaken.

Since the system is very complex, with many uncertainties and variation during the year, it is not easy to make a suggestion for possible savings over a whole year's operation. We have to make assumptions about the number of days with different kinds of cooling loads and assume how much

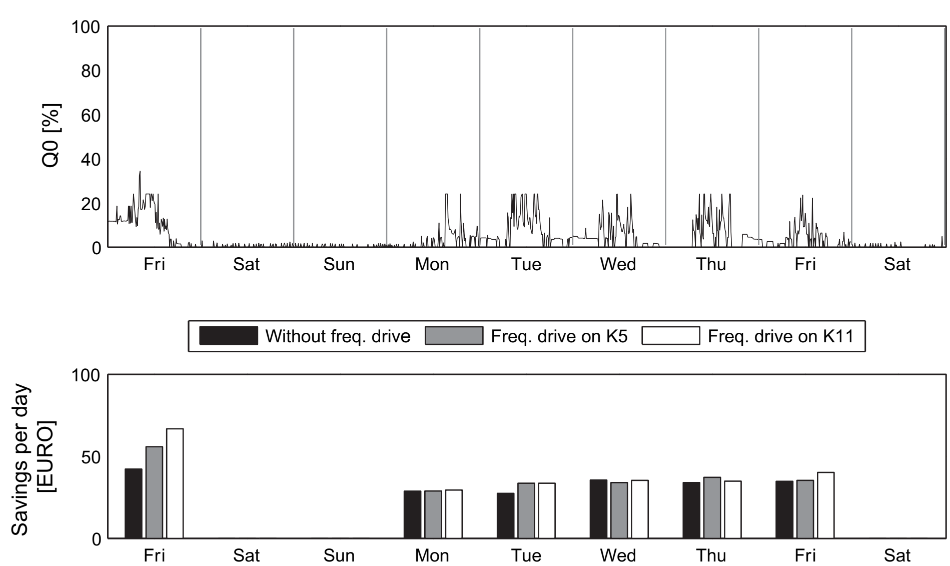

Fig. 9 – Savings per day [€] for period 2, compared with measured system, for three different optimization alternatives. Cooling capacities [kW] for the period are included. Three days did not have enough data for the graph.

can be typically saved during these days. Both periods have to be used for this analysis. The results from this analysis vary too much to provide a precise answer. The system operates much like the working days in period 2 during most of the year (about 200 days). Fig. 9 shows how much can be saved during these days. It is assumed that € 30 000–50 000 can be saved per year by optimizing the operation. There does not seem to be a large difference between optimizing with and without a variable frequency drive, but the difference is probably larger in reality. There are limitations on how often a compressor can be stopped and started, but this has not been included in the optimization model, as described in Section 4.3. The inability to include the stop/start limitations leads to more optimistic values for the system without a variable speed drive. This error is probably lower for the alternatives with a variable speed drive, since this gives better and more flexible regulation. The system tends to regulate only one compressor.

The results showed that most electrical energy can be saved during days when not all of the tunnels were loaded. This is not the case for more than about 20–40 days per year. Most typical operation is below 10% of the total cooling capacity. Our calculations gave very different results for these days, but the savings per day for compressor K11 are best in most cases. We conclude that this compressor should have a variable speed drive. This is also logical from a cooling capacity view, as it is in mid-range (see also Fig. 2).

6. Conclusions

A large amount of the electricity used in refrigeration systems goes to operate the compressors. This research has therefore focused on energy optimization in the compressor system. Since screw compressors with slide valve regulation often are used in the fish processing industry in Norway, improvements in this area can result in large energy savings for the whole industry. One of the challenges in the freezing plant studied was to use the already installed cooling capacity in a more optimal way. The compressors were often running at part-load operation, which resulted in high energy consumption per tonne of processed fish. Optimal control of the compressors was an important factor for reducing the refrigeration energy demand.

A linear programming model was developed to give the optimized operation for each compressor in the system, with minimized power consumption, provided that the total refrigeration load requirement was met. The model was used to optimize the operation of the compressors during two measurement periods. There was high production during the first period and low production during the second. The optimization was done both with and without a variable frequency drive.

The model calculated power consumptions for all of the optimization alternatives. These power consumptions were used together with the cooling loads to find the coefficient of performance (COP) for the system. Costs in € were also calculated and the difference between measured and optimized system provided possible savings amounts. Savings per day varied with different cooling loads. The results showed that most electrical energy can be saved during days when not all of the tunnels were loaded. Systems like this use a great deal of energy even at lower production rates because the compressors are designed for their most efficient operation at a high load.

The most typical operation is below 10% of total cooling capacity. Our calculations gave very different results for these days, so very precise calculations for savings per year were not possible. It is assumed that € 30 000–50 000 can be saved per year by optimizing the operation of the refrigeration system. There does not seem to be a large difference between optimizing with and without a variable frequency drive, but the difference is probably larger in reality. Our calculations show that the mid-range compressor should have a variable speed drive.

References

Aprea, C., Mastrullo, R., Renno, C., 2006. Experimental analysis of the scroll compressor performances varying its speed. Applied Thermal Engineering, 983–992.

Aprea, C., Renno, C., 2004. An experimental analysis of a thermodynamic model of a vapour compression refrigeration plant on varying the compressor speed. International Journal of Energy Research, 537–549.

Aprea, C., Renno, C., de Rossi, F., 2007. Optimization of the variable speed compressor performances. In: The 22nd International Congress of Refrigeration. IIR/IIF, Beijing iCR07-B2-797.

Avallone, E.A., Baumeister III, T., 1996. Marks' Standard Handbook for Mechanical Engineers, tenth ed. McGraw-Hill.

Brendeng, E., 1979. Reciprocating compressors or screw compressors? International Journal of Refrigeration 2, 163–170.

Enova, 2002. Resultater fra industrinettverket. Available from internet: http://www.enova.no (accessed 18.10.05).

Gjøvåg, G., 2004. Energibruk og -utnyttelse ved industrielle kuldeanlegg. Master's thesis, NTNU.

Gosney, W.B., 1982. Principles of Refrigeration. Cambridge University Press.

Leducq, D., Guilpart, J., Trystram, G., 2006. Non-linear predictive control of a vapour compression cycle. International Journal of Refrigeration 29 (5), 761–772.

Solheim, O., 2006. Energy efficiency in fish processing plant. Master's thesis, NTNU.

Statistics Norway, 2006. Eksport av fisk og fiskeprodukt, etter mottakarland og varegruppe. Available from internet: http://www.ssb.no/emner/10/05/nos_fiskeri (accessed 21.02.08).

Stoecker, W.F., 1998. Industrial Refrigeration Handbook. Mc Graw Hill.

Widell, K.N., Eikevik, T., 2008. Reducing power load in multi-compressor refrigeration systems by limiting part-load operation. In: IIR Gustav Lorentzen Conference.