CONTROL METHODS UTILIZING ENERGY OPTIMIZING SCHEMES IN REFRIGERATION SYSTEMS

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Abstract

The potential energy savings in refrigeration systems using energy optimal control has been proved to be substantial. This however requires an intelligent control that drives the refrigeration system towards the energy optimal state. This paper proposes an approach for a control, which drives the condenser pressure towards an optimal state. The objective of this is to present a feasible method that can be used for energy optimizing control. A simulation model of a simple refrigeration system will be used as basis for testing the control method.

1 Nomenclature

c_w	Heat capacity water $\left[\frac{J}{kg \cdot K}\right]$
f_q	Heat loss coefficient compressor
h	Enthalpy $\left[\frac{J}{kg}\right]$
K_{cp}	Power constant condenser pump $\left[\frac{W \cdot s^3}{kg^3}\right]$
$\dot{m}_{ref}$	Massflow refrigerant $\left[\frac{kg}{s}\right]$
$\dot{m}_w$	Massflow water $\left[\frac{kg}{s}\right]$
N	Rotational speed [rpm]
OD	Opening Degree
P	Pressure [bar]
$\dot{Q}_e$	Cooling capacity [W]
SH	Superheat [K]
SC	Subcooling [K]
T	Temperature [K]
UA	Heat transfer coefficient $\left[\frac{kJ}{s \cdot K}\right]$
$\dot{W}$	Power consumption [W]
η	Efficiency

Indices

$_C$	Compressor
$_{CP}$	Condenser pump
$_{EP}$	Evaporator pump
$_c$	Condenser
$_e$	Evaporator
$_{is}$	Isentropic
$_{ie}$	Inlet evaporator
$_{oe}$	Outlet evaporator
$_{ic}$	Inlet condenser
$_{oc}$	Outlet condenser
$_{wic}$	Water inlet condenser
$_{woc}$	Water outlet condenser
$_{wie}$	Water inlet evapoator

2 INTRODUCTION

For an idealized refrigeration cycle it is possible by means of static models to calculate the energy optimal set points for the operation of the system under certain constrains (Jakobsen *et al.*, 2001). Hereby it has been shown that the potential energy savings, using these set points are substantial (Larsen and Thybo, 2002). The problem in utilizing a control scheme that enables such an energy optimizing procedure is that, it's not possible directly to measure how close the system is to an optimal state. This means that some sort of intelligent supervisory control must be established to find out what the optimal set points are under given circumstances.

The overall goal of energy optimization of refrigeration systems is to minimize the energy consumption, while keeping the cooling capacity ($\dot{Q}_e$) constant and upholding the specified temperature. For a simple 1:1 refrigeration system like the one shown in Figure 1, can the optimizing scheme, given the ambient temperatures (T_{wic} and T_{wie}), be written as:

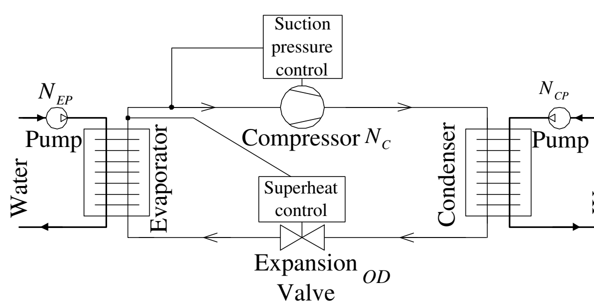

Figure 1: The system layout.

$$\min_{[N_C, N_{EP}, N_{CP}, OD | \dot{Q}_e = Const, SH = Const]} (\dot{W}_C + \dot{W}_{EP} + \dot{W}_{CP}) \tag{1}$$

N_C, N_{EP} and N_{CP} denotes the rotational speed of the individual components and OD the opening degree of the valve. It is assumed that a constant low superheat (SH) gives the highest possible evaporator efficiency.

The optimization scheme given by (1) results in a 2 degree of freedom problem, which means that 2 set points has to be estimated (P_e and P_c) to find the minimum. This can be simplified by assuming a constant evaporator pressure (P_e), controlled by an internal loop, hereby the problem is reduced to a 1 degree of freedom optimization problem. It has been shown in (Larsen and Thybo, 2002) that by optimizing the condenser pressure while keeping the evaporator pressure constant, the energy efficiency can be improved significantly. Another important point that is stated here is that the optimal condenser pressure practically is independent of the current evaporator pressure. This means that the evaporator and condenser pressure can be optimized individually as two 1 degree of freedom optimization problems in order to reach the global optimum (the maximal efficiency). The objective of this paper is thus to set up a method which optimizes the condenser pressure, that is solves *one* of the two 1 degree of freedom optimization problems.

3 Energy Optimizing Control

In refrigeration systems a normal control scheme would be that the compressor(s) controls the suction pressure and the expansion valve(s) controls the superheat. Using this control scheme and assuming that the cooling load is constant will the cooling capacity be constant regardless of the condenser pressure (at steady state). Figure 2 shows the power consumption using this control scheme recorded at varying condenser pressures, as it can be seen can a minimum in the total energy consumption be found at a certain condenser pressure. Assuming that the above mentioned control scheme is used, this on the other hand means that the energy can be minimized by forcing the condenser pressure to its most favorable. Since the condenser pressure can be controlled by the condenser pump (N_{CP}), can the optimizing scheme therefore just be applied the condenser pump, while the control loops for the evaporation pressure and superheat keeps there preset set points, fulfilling the constraints given in (1). The sum-curve (the sum of the energy consumptions of individual components) is given by $\sum \dot{W}_x|_{P_e=const}$, where x denotes the relevant components. This sum can be split up into the individual contributions from each component as shown in Figure 2.

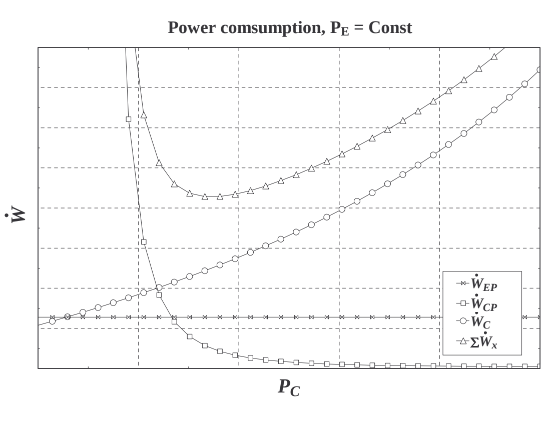

Figure 2: Power consumption in the individual components

The minimum can then be found as:

$$\begin{aligned} &\frac{d}{dP_c}\left(\sum \dot{W}_x|_{P_e=const}\right) = 0 \\ \Leftrightarrow &\left.\frac{d\dot{W}_C}{dP_c}\right|_{P_e=const} + \left.\frac{d\dot{W}_{CP}}{dP_c}\right|_{P_e=const} = 0 \end{aligned} \tag{2}$$

The power consumed by the evaporator pump is, as it can be seen from Figure 2, independent of the condenser pressure, therefore is the term $\frac{d\dot{W}_{EP}}{dP_c} = 0 \ \forall \ P_c$. Since $\dot{W}_C$ and $\dot{W}_{CP}$ are monotonically ascending and descending respectively and the optimization problem is convex, will the sign of $\frac{d\dot{W}_C}{dP_c} + \frac{d\dot{W}_{CP}}{dP_c}$ decide whether N_{CP} should be increased or decreased in order to reach minimum, and fulfill (2). That is if $\frac{d\dot{W}_C}{dP_c} + \frac{d\dot{W}_{CP}}{dP_c} < 0$ then N_{CP} should be decreased and vice versa. This means that if the gradient sum can be estimated, it is possible directly to use it as input to a controller, which drives it toward 0. The task by using this approach therefore is to estimate the gradient sum. A variety of different ways to estimate this sum can be relevant, the approach chosen in this paper is model based. By using simplified static models and estimating unknown parameters it is possible to predict the power gradient for a given set of inputs and subsequently drive it towards 0. The above mentioned optimizing procedure leads to a control hierarchy as depicted in Figure 3. The top layer consists of a parameter estimation routine, which estimates the parameters in the static model. The second layer is the optimizing routine, which estimates the power gradient and the reference to condenser pressure controller. The two top layers can be seen as a realtime reference governor, which predict the optimal set point based on the present state measurements. The third layer is the distributed control system including the superheat control and suction control. These are feed constant references, which ensure the (soft) constraints for the optimization are fulfilled (P_e, SH and Q_e should be constant). A further purpose of these control loops is to help ensuring stability of the system.

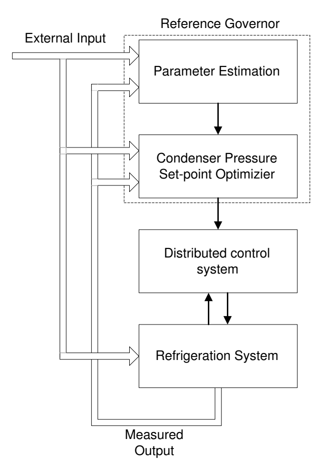

Figure 3: The Control Hierarchy

4 STATIC MODEL

As previously mentioned is the power gradient generated based on a static model. Using a static model to estimate parameters in a dynamic system off course introduces some dynamic errors, which though tends to zero at steady state. A way to avoid these dynamic "disturbances" on the estimates, would obviously be to use a dynamic model for the parameter estimation instead, this would however require a priori knowledge of the dynamics in the system. Anyhow it should be kept in mind that the requirements for the bandwidth of the optimizing control is very low, a static model will therefore be sufficient. Later on this statement will be supported by dynamic simulations.
The static model that has been used, is described by the following 6 equations:

$$\begin{aligned}
\dot{W}_C &= \frac{1}{1-f_q}\cdot \dot{m}_{ref}(h_{ic}-h_{oe}) && (3)\\
\dot{W}_{CP} &= K_{CP}\cdot(\dot{m}_w)^3 && (4)\\
T_{woc} &= T_c + (T_{wic}-T_c)\cdot \exp\left(-\frac{UA}{\dot{m}_w\cdot c_w}\right) && (5)\\
Q_e &= \dot{m}_{ref}(h_{oe}-h_{oc}) && (6)\\
0 &= \dot{m}_{ref}(h_{ic}-h_{oc}) - \dot{m}_w c_w (T_{woc}-T_{wic}) && (7)\\
\eta_{is} &= \frac{\dot{m}_{ref}(h_{is}-h_{oe})}{\dot{W}_C} && (8)
\end{aligned}$$

The abbreviations can be found in the nomenclature.

Equation (3) describes the power consumption in the compressor assuming a constant heat loss coefficient f_q.

Equation (4) describes the power consumption in the condenser pump.

Equation (5) can be derived assuming a lumped temperature of the wall between the refrigerant and the water and constant condensing temperature all through the condenser.

Equation (6) describes the cooling capacity of the evaporator.

Equation (7) describes the conservation of energy across the condenser wall.

Equation (8) describes the isentropic efficiency

In the equations above it is implied that the enthalpies (h) are functions of the respective pressures and temperatures. The equations are used as basis for the parameter and power gradient estimation.

5 PARAMETER ESTIMATION

The parameter estimation is carried out by using the MIT rule. Using this adaptive parameter adjustment routine, the parameters can be tuned by minimizing the error between the measurements and the model, that is in accordance to following equation (Åström and Wittenmark, 1989):

$$\frac{\partial \theta}{\partial t} = -\gamma e \frac{\partial e}{\partial \theta}, \qquad (9)$$

where e denotes the model error and θ the parameter estimate. The parameter γ determines the adaption rate.
Rewriting (3)-(8) following parameter dependent error can be obtained:

$$e(\theta) = \begin{bmatrix} \frac{\dot{m}_w c_w (T_{woc}-T_{wic})}{h_{ic}-h_{oc}} - \widehat{\dot{m}}_{ref} \\ \frac{h_{ic}-h_{oe}}{\dot{W}_C} - \frac{1-\widehat{f}_q}{\widehat{\dot{m}}_{ref}} \\ \frac{\dot{W}_{CP}}{(\dot{m}_w)^3} - \widehat{K}_{CP} \\ \dot{m}_w c_w \ln\left(\frac{T_c-T_{wic}}{T_c-T_{woc}}\right) - \widehat{UA} \\ h_{oe}-h_{oc} - \frac{\widehat{Q}_e}{\widehat{\dot{m}}_{ref}} \\ \frac{h_{is}-h_{oe}}{\dot{W}_C} - \frac{\widehat{\eta}_{is}}{\widehat{\dot{m}}_{ref}} \end{bmatrix}, \qquad (10)$$

where $\theta = [\widehat{\dot{m}}_{ref}, \widehat{f}_q, \widehat{K}_{CP}, \widehat{UA}, \widehat{Q}_e, \widehat{\eta}_{is}]^T$, the remaining variables and constants are assumed to be either known or measured.
From (10) the derivative can be derived:

$$\frac{\partial e}{\partial \theta} = \begin{bmatrix} -1 & 0 & 0 & 0 & 0 & 0 \\ \frac{1-\widehat{f}_q}{(\widehat{\dot{m}}_{ref})^2} & \frac{1}{\widehat{\dot{m}}_{ref}} & 0 & 0 & 0 & 0 \\ 0 & 0 & -1 & 0 & 0 & 0 \\ 0 & 0 & 0 & -1 & 0 & 0 \\ \frac{\widehat{Q}_e}{(\widehat{\dot{m}}_{ref})^2} & 0 & 0 & 0 & -\frac{1}{\widehat{\dot{m}}_{ref}} & 0 \\ \frac{\widehat{\eta}_{is}}{(\widehat{\dot{m}}_{ref})^2} & 0 & 0 & 0 & 0 & -\frac{1}{\widehat{\dot{m}}_{ref}} \end{bmatrix} \qquad (11)$$

Using (10) and (11) the parameter estimator can be implemented using the MIT rule given by (9) as shown in Figure 4.

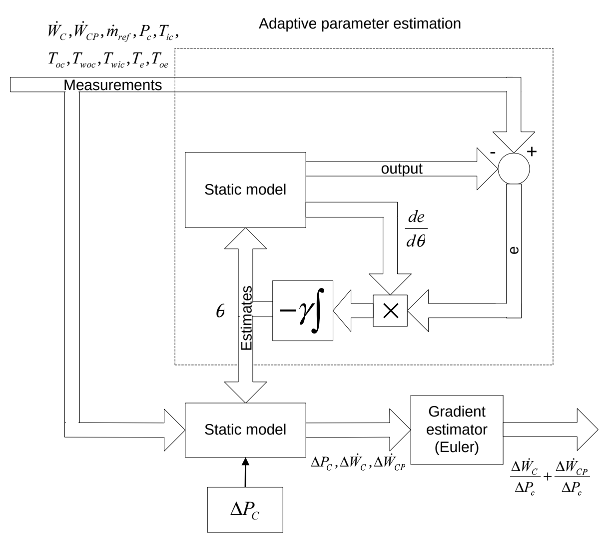

Figure 4: Implementation of parameter estimator and condenser pressure set point optimizer.

Using this approach a lowpass filtering of the measurements through the integrator is obtained smoothing parameter estimates. Furthermore the parameters, which though are assumed constant, are enabled to adapt to un-modelled changes in the system.

6 CONDENSER PRESSURE SET POINT OPTIMIZER

In order to estimate the power gradient (2), the static model is perturbed with ΔP_c (from the present state), whereby the power gradient can be estimated as $\frac{\Delta \dot{W}_C}{\Delta P_c} + \frac{\Delta \dot{W}_{CP}}{\Delta P_c}$.
From (3) and (6) it can be seen that $\Delta \dot{W}_C$ can be found directly. Slight greater difficulties is encountered in finding $\Delta \dot{W}_{CP}$. $\dot{W}_{CP}$ depends as it can be seen in (4) on $\dot{m}_w$ and $\dot{m}_w$ can because of the non-linearities not directly be isolated from (5) and (7). An iterative method is therefore enquired to determine $\dot{m}_w$, for this purpose the Newton Method has been used. By using a small perturbation $\dot{m}_w$ thus can be found with a small error within a few iterations, which enables a short processing time. Subsequently insertion in (4) produces $\Delta \dot{W}_{CP}$. Now all the result are found that is needed to generate the power gradient $\frac{\Delta \dot{W}_C}{\Delta P_c} + \frac{\Delta \dot{W}_{CP}}{\Delta P_c}$. This signal is fed to the condenser pressure controller, which is implemented as a cascade controller, showed in Figure 5. This

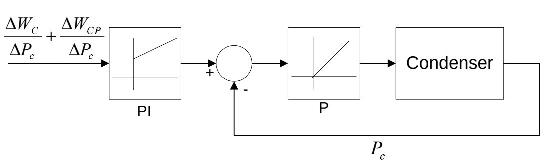

Figure 5: The Condenser Pressure Control

control setup enables relatively fast dynamics in the inner loop, which helps keeping the system on the right track by suppressing disturbances. The slow integral action is moved to outer loop, firstly because only slow performance of the optimizing control is required and secondly because the estimated value of power gradient actually only holds true near steady state.

7 RESULTS

A dynamic model of the refrigeration cycle like shown in Figure 1 has been used in the simulation. The model consists of a lumped parameter moving boundary model of the evaporator (a plate heat exchanger), a lumped parameter model of the condenser (a shell and tube condenser) and static models of the expansion valve (a step motor controlled valve) and the compressor (a scroll compressor). A detailed description of the model can be found in (Larsen and Holm, 2002).
On Figure 6 is the power consumed by the compressor and condenser pump using optimizing control compared with a constant condenser pressure control. The minimal power consumption using the exact optimal set points is also indicated. The system is started under the following conditions:

Reference P_e	Reference SH	T_{wic}	T_{wie}
4.22 [bar]	5 [K]	17 [^oC]	27 [^oC]

After 10000 sec, the temperature T_{wic} is altered by a step from 17 to 7 ^oC. Hereby as well the static as the dynamic properties of the control can be examined. In systems with air-cooled condensers (which are normally placed outside), will the ambient temperature be comparable with T_{wic}. A change in T_{wic} is therefore comparable with changes in the ambient temperature.
When the process settles after start-up, deviates the power

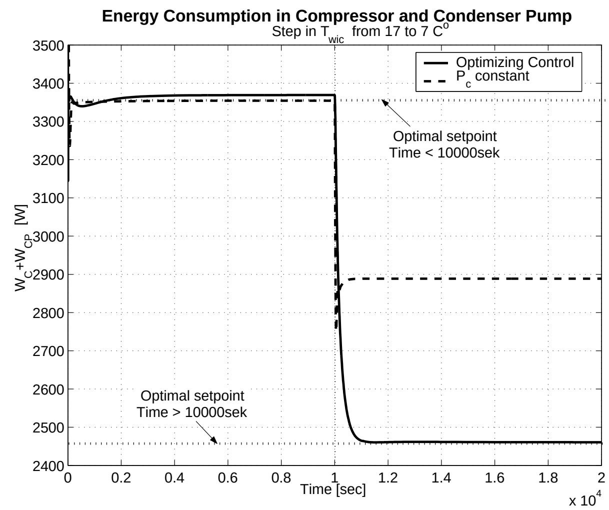

Figure 6: Power consumption using the energy optimizing strategy compared to keeping the condenser pressure P_c constant. After 10000 sec is T_{wic} altered from 17 to 7^oC

consumption of the optimizing control 0.43% from the optimal set point and after the step is the deviation 0.03%. It is therefore possible within a relatively narrow margin to operate the system in the optimal state (under the given conditions). Furthermore it can be seen, that though the model is static it does have any impact on the dynamic response of the power consumption, as previously stated.
The optimizing control has been compared to a constant condenser pressure control strategy, which is a strategy widely used. It can be seen that even though the constant condenser pressure control has been started-up at an optimal set-point, the potential energy saving, after the step in T_{wie} is around 14%. In the light of this the deviations from the optimal set-point using the optimizing control are insignificant.
In the figure below 7, there has been made a step change after 5000 sec in the inlet temperature to the evaporator (T_{wie}) from 27 ^oC to 22^oC, hereby the cooling capacity (Q_e) changes. This means that a new value has to be estimated since it enters into the static model. As it can be seen from the dynamic response this adaption of the parameters (based on a static model) does not initiate any foul behavior. This is as previously mentioned because the adaption rate is much slower that the dynamics in the underlying distributed control systems.

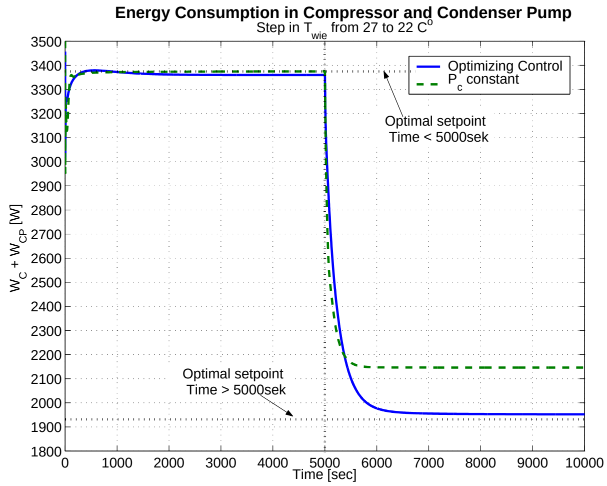

Figure 7: Power consumption using the energy optimizing strategy compared to keeping the condenser pressure P_c constant. After 5000 sec is T_{wie} altered from 27 to 22^oC

8 Conclusions

In this paper a method for minimizing the energy consumption in a refrigeration system has been presented. Based on a non-linear steady state model, the power gradient has been estimated and used as input to a control of the condenser pressure. Simulation on a dynamic (non-linear) model has shown that this approach makes it possible to achieve good estimates, which enables the control to drive the system very close the optimal set point.

9 Acknowledgments

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