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ALGORITHM-BASED OPTIMIZATION FOR ENERGY-EFFICIENT OPERATION OF REFRIGERATION SYSTEMS

Jörg Bentz, Felix Loistl, Christian Schweigler
CENERGIE - Research Center Energy-efficient Buildings and Districts
University of Applied Sciences Munich
Munich, 80335, Germany, joerg.bentz@hm.edu

ABSTRACT

In refrigeration systems, a set of devices is applied for the provision of cooling, for the transfer of reject heat to ambient and for circulation of the respective heat carrier fluids. Aiming at energy-efficient operation of the entire system, a complex optimization exercise is found, characterized by the number of degrees of freedom and the interdependence of the different components. In order to form a realistic model of the system, the operating characteristics of all system components must be modeled in sufficient quality. In order to allow for sufficient precision of the models, e.g. for representation of part load behavior, a nonlinear optimization has to be applied. Overall, a considerable effort for the development and application of a mathematical optimization of complex refrigeration systems is required. Within the framework of this study, a method is shown to make the application of optimization methods also available for smaller refrigeration systems.

Keywords: refrigeration systems, energy efficiency, performance, modelling, optimization

1. INTRODUCTION

This paper describes the development of an optimization algorithm for a significant reduction of the power consumption of refrigeration systems. Refrigeration systems are among the most important power consumers. They use about 14% of the total electrical power in Germany. (VDMA, 2011)

In refrigeration systems several individual devices interact, which leads to a high number of degrees of freedom in their mode of operation. A high number of interdependencies occurs when chillers are operated in conjunction with their peripherals like heat rejection units, pumps and other refrigeration units. An example is shown in Fig. 1. An optimal setting of operating parameters is essential for finding the most efficient point of operation and therefore a reduction of the power consumption. A mathematical optimization procedure is needed to find the best constellation of parameter values. (D'Antonio *et al.*, 2005) Depending on the complexity and type of the objective, different optimization procedures can be used. For this purpose, the operating characteristics of all system components must be modeled in sufficient quality (Thiem and M, 2017). Overall, a considerable effort has to be undertaken for the configuration and application of a mathematical optimization of complex refrigeration systems.

Within the framework of this study, an approach is shown to make the application of optimization methods also available for smaller refrigeration systems.

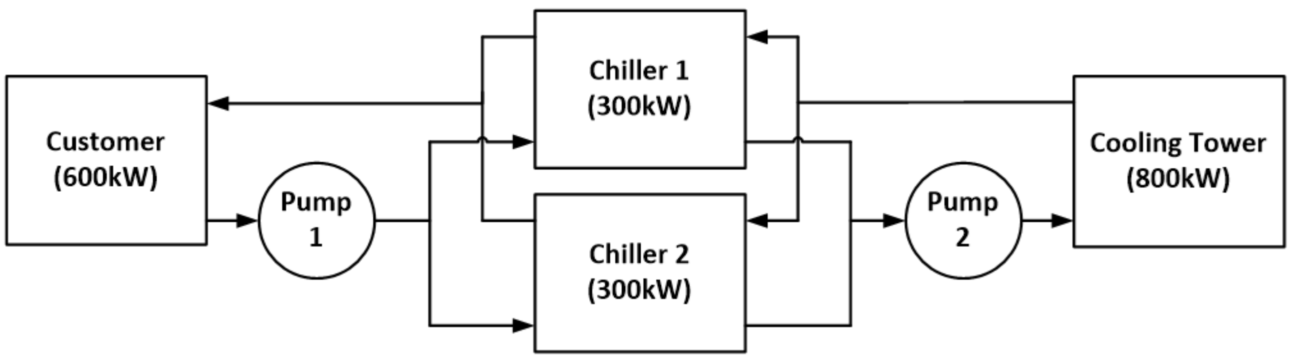

Figure 1: Example of examined refrigeration system

2. MODELING

Simulation models with different model accuracy for the different components of refrigeration systems are developed. In this case, a detailed semi-empirical physical model and a highly simplified and generalized model are used. These different component models are applied in conjunction with suitable optimization algorithms to determine which model accuracy is required in order to obtain a reliable statement about the optimal parameter setting of the refrigeration system with the reasonable computational effort.

To make the investigation as comparable as possible, the selection of a suitable reference system is required. For this purpose, the boundary conditions such as the required cooling capacity, the topology of the plant or the technologies and components used have to be defined. When mapping the system characteristic or modeling the system, the applied modeling method is kept as simple as possible. Since any increase in the level of detail induces more effort in the creation and parameterization of the model and requires a higher computing capacity during application. Consequently, a main subject of the investigation is the clarification to what extent an increase of the model complexity makes sense with regard to accuracy and computational effort.

As an example for the modelling of the system components, in this section the characterization of a vapor compression chiller is presented. Fig. 2 shows the scheme of a chiller with its four key components, i.e. expansion valve, evaporator, compressor and condenser. The modelling of the cooling tower and the pump periphery is not dealt with within this study.

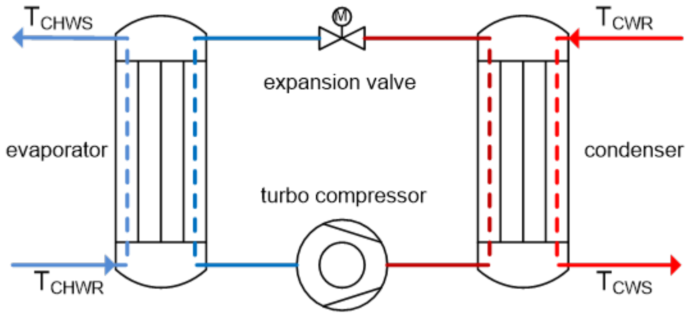

Figure 2: Scheme of a water cooled turbo compressor chiller

2.1. Semi-empirical physical model

In the semi-empirical physical model, the relevant components of the compression chiller are modeled partly physically and partly on the basis of empirical data. The heat transfer at the heat exchangers and the refrigerant states throughout the cycle are described physically. Hence, the calculation of the refrigeration cycle requires access to a refrigerant property database. For the modelling of the compressor a multidimensional polynomial is used. Coefficients for the polynomial characterization of the compressors are provided by most compressor manufacturers according to the standard (AHRI, 2015). Based on the norm AHRI-540, the electrical consumption of the compressor can be represented as a 3rd degree polynomial, as given by Eq. (1), depending on the evaporation temperature T_0 and condensation temperature T_1.

$$P_{el} = C_1 + C_2 \cdot T_0 + C_3 \cdot T_1 + C_4 \cdot T_0^2 + C_5 \cdot T_0 \cdot T_1 + C_6 \cdot T_1 + C_7 \cdot T_0^3 + C_8 \cdot T_1 \cdot T_0^2 + C_9 \cdot T_0 \cdot T_1^2 + C_{10} \cdot T_1^3 \qquad \text{Eq. (1)}$$

The values C_1 to C_{10} represent the coefficients valid for the particular compressor model. In the case of a speed-controlled compressor, these coefficients additionally vary as a function of the current frequency (C(f)) of rotation of the compressor. This characteristic can also be represented by a square polynomial. However, this leads to increased complexity of the models.

For the modelling of the heat exchangers, a distinction of different zones according to the state of aggregation of the refrigerant is applied (VDI, 2013). In each section, an energy balance for steady-state operation can be solved. For simplicity, it is assumed that fluid properties and operating characteristics such as the heat transfer coefficient k remain constant throughout the entire zone. This leads to a division into two sections for the evaporator: the two-phase section (evaporation) and the gaseous section (superheating) after evaporation. Whereas the condenser must be divided into three areas: gaseous (desuperheating), two-phase (condensing) and liquid (subcooling).

The energy balance of the zones comprises the heat duty in the internal cycle (see Eq. (3)), the performance of the heat transfer from the refrigerant to the external fluid with the heat transfer coefficient u, the transfer area A and the logarithmic temperature difference ΔT_{log} (Eq.4)), as well as the capacity of the external water cycle (Eq. 5).

$$EER = \frac{\dot{Q}_0}{P_{el}} \qquad \text{Eq. (2)}$$

$$\dot{Q} = \dot{m}_{ref} \cdot \Delta h \qquad \text{Eq. (3)}$$

$$\dot{Q} = k \cdot A \cdot \Delta T_{log} \qquad \text{Eq. (4)}$$

$$\dot{Q} = \dot{m}_{water} \cdot c_p \cdot \Delta T \qquad \text{Eq. (5)}$$

The performance of the internal cycle, which is calculated by the refrigerant mass flow m_{ref} and the enthalpy difference Δh, requires the input of physical fluid property data. To further determine the u-value for the heat transfer, different correlations have to be applied for the respective aggregate states. The correlations include operating parameters, property data and dimensionless numbers such as the Reynolds number or the Prandtl number.

The coupling of the component models with the overall optimization procedure can lead to a substantial computational effort. Additionally, due to the high complexity of the mathematical problem a poor stability of the solving process may be obtained. Additionally, a detailed model needs a deep understanding of the system, as an enormous number of parameters (e.g. heat exchanger area, heat transfer coefficients, internal geometry of the heat exchangers) must be defined. These parameters are often unknown or not provided by the manufacturer. As a consequence, many parameters must be estimated, possibly leading to errors or inaccuracies in the calculation. In order to achieve an efficient coupling of the optimization procedure with the calculation models with a reduced number of input parameters, it becomes essential to simplify the model as far as possible.

2.2. Simplified and generalized model

The generalized model consists of fundamental relationships which describe the characteristics of the system components referring to the underlying principles of operation. In contrast to the complex calculation method, a simplified calculation based on the Carnot coefficient of performance is applied, following a publication of the AMEV (AMEV, 2007). Accordingly, the current EER of the chiller is calculated by the Carnot coefficient of performance multiplied by a system efficiency η^* and a correction factor f for the consideration of the refrigerant in use (see Eq. (6)). If the compressor is speed-controlled, the constant system efficiency η^* must be replaced by a square polynomial in order to characterize the partial load behavior of the compressor. This behavior can be seen when a change in load causes a change in the performance at constant temperature conditions (see Fig. 7).

$$EER = \frac{T_0}{T_1 - T_0} \cdot \eta^* \cdot f \qquad \text{Eq. (6)}$$

$$\eta^* = C_1 + C_2 \cdot PLF_0 + C_3 \cdot PLF_0^2 \qquad \text{Eq. (7)}$$

As in the detailed approach, the heat exchange at evaporator and condenser must be modelled for the determination of the temperatures of the internal cycle T_0 (evaporating temperature) and T_1 (condensing temperature) in conjunction with the temperatures of the external heat carriers. However, in contrast to the detailed model, a simplified description of the heat exchangers is applied. Instead of the mean logarithmic temperature difference a closest approach temperature model is used (see Eq. (8+9)). The closest approach temperature (CAT) describes the smallest temperature difference between internal and external temperatures. With this approach only one relation per heat exchanger is needed and a division into different zones is not necessary. For a distinct state of operation, the internal temperatures T_0 and T_1 are obtained by adding the current CAT_0 to the chilled water outlet temperature TCHWS (Eq. (8)) or subtracting CAT_1 from the cooling water outlet temperature TCWS (Eq. (9), figute 8). The current temperature difference $CAT_{0/1}$ is a result of the temperature difference at the design point $CAT_{0/1,ref}$ multiplied by a part load factor $PLF_{0/1}$ for the actual refrigeration capacity and a part flow factor $PFF_{0/1}$ which corresponds to the current volume flow $\dot{V}_{0/1}$ of the chilled water or the cooling medium (Eq. (10)).

Thereby $PLF_{0/1}$ describes the current part load state with capacity $\dot{Q}_{0/1}$ relative to the capacity $\dot{Q}_{0/1,ref}$ at the design point (Eq.(11)). In the same manner $PFF_{0/1}$ is given by the ratio of the current volume flow $V_{0/1}$ and the volume flow at the design point $\dot{V}_{0/1,ref}$ (Eq.(12)).

$$T_0 = T_{CHWS} + CAT_0 \qquad \text{Eq. (8)}$$

$$T_1 = T_{CWS} - CAT_1 \qquad \text{Eq. (9)}$$

$$CAT_{0/1} = CAT_{0/1,ref} \cdot PLF_{0/1} \cdot PFF_{0/1} \qquad \text{Eq. (10)}$$

$$PLF_{0/1} = \frac{\dot{Q}_{0/1}}{\dot{Q}_{0/1,ref}} \qquad \text{Eq. (11)}$$

$$PFF_{0/1} = \frac{\dot{V}_{0/1}}{\dot{V}_{0/1,ref}} \qquad \text{Eq. (12)}$$

With this calculation method, the chiller model is based on design data only, that are usually provided by the manufacturer. Only the polynomial for the description of the partial load behavior (Eq.(7), figure 7) must be selected depending on the respective compressor technology.

2.3. Comparison of the component models

For validation of the component models, a comparison of the coefficient of performance (EER) of the chiller resulting from the two models is conducted, as shown in Figure 3. Operation with evaporator capacities of 200 and 300 kW obtained by control of the compressor speed and evaporation temperature varying from -5 to 10 °C is investigated. Obviously, the simplified model is in very good agreement with the results of the complex semi-empirical physical model in the expected operating range with evaporation temperatures from 0°C to 8°C. Only at the limits of the operating envelope, a marginal deviation occurs between the models. Nevertheless, the simplified model can very well represent the characteristics of the chiller, showing an increase of the EER with increasing evaporation temperature. Figure 4 additionally illustrates the relative error of the EER value over the entire operating range, exhibiting low coefficients of performance at low evaporation temperatures and high EER values at high evaporation temperatures. The error is less than 4 % in the relevant operating range and less than 10 % in the entire range.

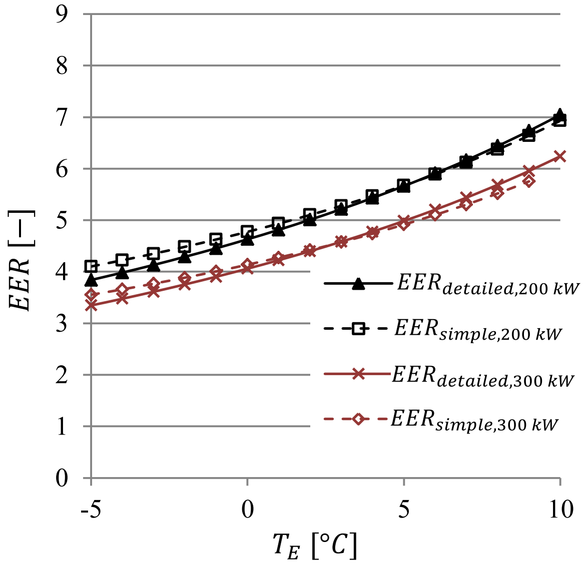

Figure 3: Comparison of EER values obtained from detailed and simple model at varying evaporation temperatures

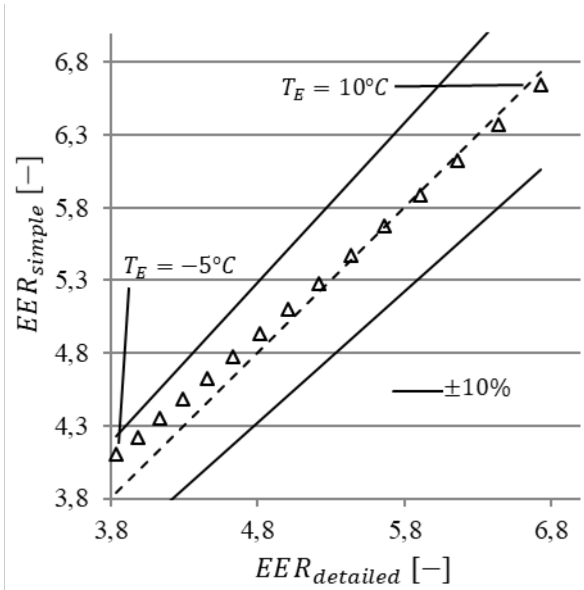

Figure 4: Deviation of the simple and the detailed model at varying evaporation temperatures

A similar behavior with an acceptable deviation of the detailed and simple model is found for the EER in dependence on the condensing temperature (see Figure 5). Again, the operating characteristics of the chiller, i.e. falling EER with rising condensing temperature, is represented clearly. Figure 6 shows only a small deviation of the investigated EER values over the entire range of condensing temperature. Only at very high operation temperatures (>55 °C) leading to low EER values, a higher deviation of more than 10 % occurs. However, condensation temperatures above 50 °C are not expected in conventional air-cooled chiller applications.

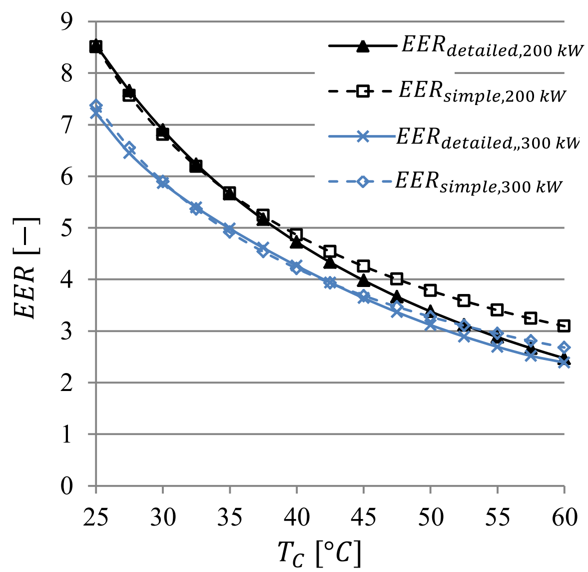

Figure 5: Comparison of EER values obtained from detailed and simple model at varying condensation temperatures

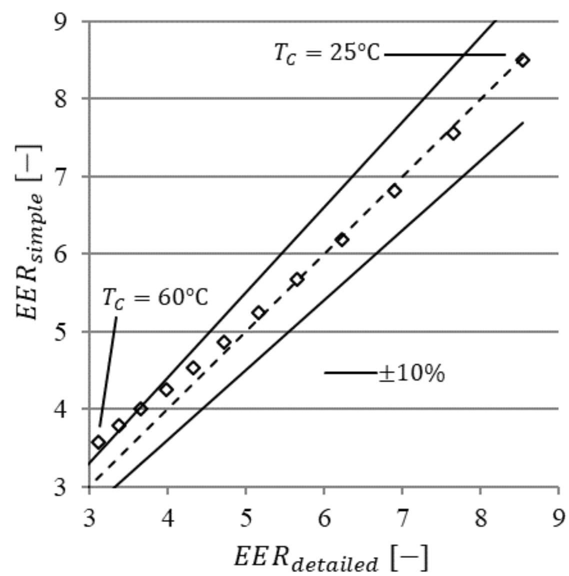

Figure 6: Deviation of the simple and the detailed model at varying condensation temperatures

Figure 7 shows the EER as a function of the third degree of freedom of the chiller; the variation of the cooling capacity. This curve corresponds to the typical behavior of a speed-controlled turbo compressor, which exhibits maximum efficiency at low partial load and an almost continuous decrease towards higher load states. The comparison shows that the simplified model has excellent agreement with the complex model. As a result, the maximum error is limited to a few percent.

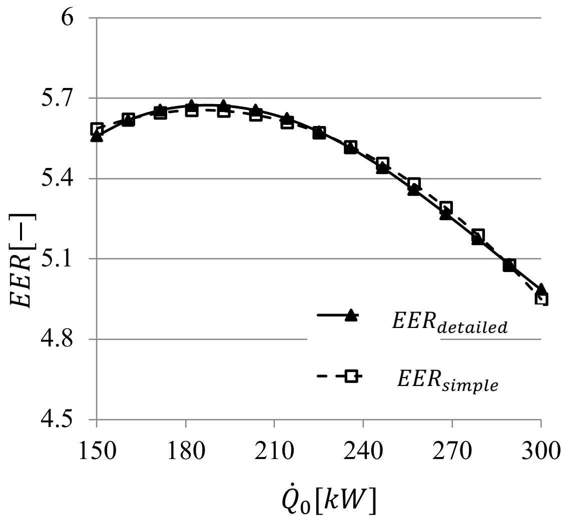

Figure 7: EER values obtained from detailed and simplified model as a function of the cooling capacity

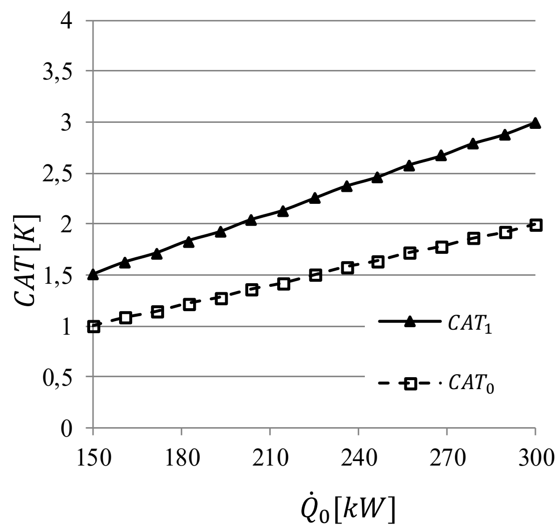

Figure 8: Variation of the CATs according to the part load ratio

2.4. Transfer to polynomial representation

Both, the complex and the highly simplified models are mathematically still highly complex for most of the solvers to solve the conventional optimization tasks. Mostly it even requires an iterative solution. To this end, it is expedient to bring the operating characteristics of the individual system components into a mathematical form, that allows most solvers to process the optimization routine with little effort.

The polynomial representation is particularly useful for implementing the component models in a system optimization algorithm. With polynomials an overall system model can be set up, incorporating all variables and parameters involved. At the same time, it is possible to adapt the operating characteristics determined by the coefficients of the polynomial during execution of the optimization algorithm. The polynomials are created by application of the simplified models performing a simulation of behavior of the respective system component covering all possible operating constellations. This procedure creates a set of operating states, which can be converted into a linear or non-linear system of equations (polynomial) using an appropriate regression method. The complexity of the polynomials is chosen in such a way that a compromise is achieved in terms of model accuracy and calculation effort.

3. CONCLUSION

In this work, a method was shown how refrigeration supply systems can be modeled and optimized with little mathematical effort. The system configuration and the function of the individual components, such as refrigeration systems and dry coolers, are described using mathematical models. The search for the optimum set of operating parameters is then carried out by means of software applications, so-called solvers, which solve the optimization task taking into account all secondary conditions.

In order to limit the computing effort, simplified models are used, which despite the strong reduction in complexity compared to physical modeling, can represent the given processes in the cooling supply system with sufficient quality. Due to the general shape of the models, the enormous variety of systems in refrigeration technology can be covered with a limited number of model variants. For use in the optimization process, the simplified models are converted into higher-order polynomials, which can then be processed by the optimization solver. This allows to find a setting for optimum operation of the cooling supply system. Compared to linear models applied earlier, a significantly higher model accuracy is achieved. On/off switching of components and the operation of modular systems can be taken into account using mixed-integer approaches.

Within a R&D project, the component functions are currently being bundled in a software solution "ENOS" that shall be used as control tool for the online optimization of refrigeration systems. Coupled with the central control system, the ENOS tool shall enable dynamic optimization of the system control of refrigeration systems. A first pilot installation is in preparation. Based on the results, statements about the savings potential will be made using plant-specific models and optimization processes. In the course of the research activity, the functions of the ENOS software will be further developed, accompanied by an expansion of the model library and the refinement of the component models.

NOMENCLATURE

A	Heat Exchange Area (m^2)	0	Evaporation
EER	Energy Efficiency Ratio $(-)$	1	Condensation
u	HeatTransfer Coefficient $(\mathrm{W}/(\mathrm{m}^2 \cdot \mathrm{K}))$	CWS	Cooling Water Supply
P	Electrical Power (kW)	CWR	Cooling Water Return
PLF	Part load factor $(-)$	$CHWS$	Chilled Water Supply
PFF	Part load flow factor $(-)$	$CHWR$	Chilled Water Return
T	Temperature (°C)	el	Electrical
$\dot{Q}$	Energy Flow (kW)	ch	Chiller
ΔT	Temperature difference (K)	ref	Reference Point

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