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A SIMPLE CHILLER MODEL FOR HOURLY TIME STEP APPLICATIONS

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ABSTRACT

For the use in Swiss standards on energy performance of cooled buildings, a chiller model for an hourly time step calculation was needed. Unlike the models found in a literature review, the model aimed at should be based only on the information available from standard performance rating, i.e. COP values for 4 rating conditions.

To perform satisfactorily, the semi-physical/semi-empirical model had to be refined, which requires the addition of one additional COP value as an input. This refined model shows accurate results for most chiller types that were available for testing.

In this paper, the model and its development is explained, and its performance for a range of chiller types upon an application to an annual load profile of a real building is shown.

INTRODUCTION

The Swiss standards on energy performance of buildings distinguish between heated only buildings (mainly residential, but not exclusively) and buildings which need cooling and/or humidification/dehumidification. For the revision of the pair of standards covering the latter, it was decided to follow an hourly time step approach. The building side is based on the simplified dynamic room calculation in (EN ISO 13790 - 2008). For the system side, a comprehensive calculation of the system parts of the emission, the distribution and the generation of heat and cold was envisaged according to the requirements of (EN 15243 - 2007). Therefore, models for an application in an hourly time step were needed for the different components. One of the prominent ones is the chiller model.

Model requirements

The question can be asked why a new chiller model is needed. Numerous models can be found in literature. They were scanned in a review and checked against the main requirements for the application:

- A universal application for any chiller type (compressor type)
- A minimum need of input data
- A sufficiently good representation of the part load behaviour in an hourly time step application, also under the condition of varying chilled and cooling water temperatures

The last requirement is essential for the reason of discrimination: machines with a better performance shall result in a better value. The Swiss standard on ventilation and air conditioning system requirements demands chillers with a higher COP value in part load conditions that at full load, as shown in table 1. The effect of this on the energy consumption and the influence of the system operating conditions on it shall be visible in the calculation result.

Table 1
COP requirements for chillers according to Swiss standard

Total cooling power of the plant, kW	MINIMIM COP (HEAT REJECTION INCLUDED)			
	50 % part load		full load	
Value	limit	target	limit	target
1	3.2	4.0	3.2	4.0
10	4.4	5.2	3.3	4.1
20	4.8	5.8	3.5	4.3
50	5.5	6.6	3.8	4.6
100	6.0	7.3	4.1	4.9
200	6.2	8.0	4.2	5.0
500	6.2	8.2	4.2	5.0
1000	6.2	8.2	4.2	5.0

The requirement on minimum input data was important because in the local market it cannot be expected that manufacturers provide comprehensive data e.g. for the definition of a full performance map. While in (Hydeman and Zhou 2007) it is reported that full performance map data were provided during a performance based bid process – which is a highly appreciable procedure – this may be possible in special cases for large units. For the normal case, they can be expected to provide what existing standards require to measure for performance rating, i.e. (ARI 2003) or (Eurovent 2008), which defines the European Seasonal Energy Efficiency Ration, ESEER. A model based on these date would be most preferable.

The requirement of universality was not strict, but in many cases, the machine type would not be known at the time of calculation, Also, the performance rating standards do not distinguish between machine types except for the way of heat rejection.

Review results

The requirements are to some extent contradictory, and accordingly the result of the review was not promising. One part of the models, which fulfil the requirement of minimum input data, have either restrictions in covered chiller types, or they are not sufficiently representing the part load behaviour – or even both. In (EN 15243 – 2007) there is an annex referring to some simple model not really dedicated to an application in an hourly time step approach. (Bettanini et al 2003) propose a very simple model which considers the part load behaviour, but does not include variation of temperature levels (or indirectly via the part load factor) – a restriction which may be ok for a wide range of applications since many chilling circuits are still operated under constant temperatures. (Gordon and Ng 1994) proposed a semi empirical model, the restrictions of which are mentioned in (Solati et al 2003).

The other part, some of them also restricted to one chiller type, need too many input parameters, which are difficult to receive. (Peitsmann and Nicolaas 1988) proposed a chiller model in the early IEA Annex 10 work, which since then has been developed in Annex 17 and used in simulation programs. The polynomial part load approach is combined with physical heat exchanger representation requiring too much data for a wide range of applications. In (Bourdouxhe et al 1997) there is a range of different, rather detailed and more physically based models for different types of chillers. In (DOE 1980), chillers are represented by a set of three quadratic polynomial equations. Default curves for different chiller types are included. (Hydeman and Gillespie 2002) presented a development of the model, also in comparison with the Bourdouxhe and Gordon-Ng models, and showed the impressive effort of a comprehensive technique for its calibration. A comparison is also given in (Sreedharan and Haves 2001).

MODEL DESCRIPTION

Goal and Development

As already pointed out, the data, which can be expected to be available from the manufacturers, are the performance rating data according to the ARI (ARI 2003) or ESEER (Eurovent 2008) standards. They are shown in tables 2 and 3. The goal was to find a model approach which would not require more data for calibration than these four measuring points.

Table 2
ARI performance rating conditions for chillers

	ARI		
	Temperature °C		
PLR	Condenser-entering, air cooled	Condenser-entering, water cooled	Evaporator leaving
100%	35.0	29.4	6.4
75%	26.7	23.9	
50%	18.3	18.3	
25%	12.8	18.3	

Table 3
ESEER performance rating conditions for chillers

	ESEER		
	Temperature °C		
PLR	Condenser-entering, air cooled	Condenser-entering, water cooled	Evaporator leaving
100%	35.0	30.0	7.0
75%	30.0	26.0	
50%	25.0	22.0	
25%	20.0	18.0	

Table 4
Performance data (COP values) of chiller type 1

	PLR %	**100**	**75**	**50**	**25**
CONDENSER ENTRANCE TEMPERA-TURE, °C	**45**	3.8	4.10	4.3	3.8
	40	4.5	4.60	5.1	4.7
	35	**5.4**	5.60	*6.1*	5.8
	30	6.4	**6.90**	6.8	7.4
	25	7.7	8.50	**9.1**	10.3
	20	9.2	10.90	12.3	**14.1**

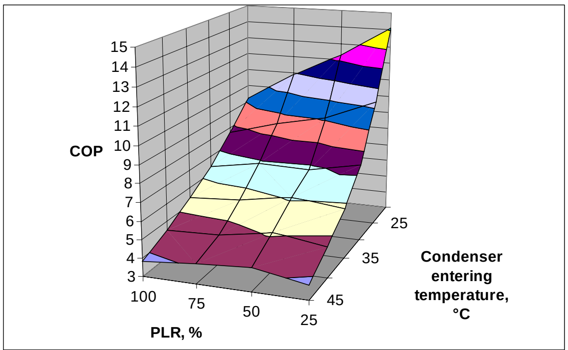

Figure 1 Performance map of chiller type 1

Table 5
Performance data (COP values) of chiller type 2

	PLR %	**100**	**75**	**50**	**25**
CONDENSER ENTRANCE TEMPERA-TURE, °C	**45**	3.8	4.2	4.4	2.7
	40	4.5	5.1	5.2	3.6
	35	**5.3**	5.9	*6.3*	4.3
	30	6.2	**6.7**	7.3	5.1
	25	7.3	7.9	**8.2**	**5.9**
	20				

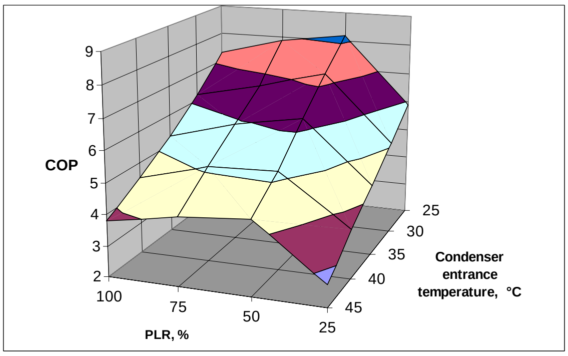

Figure 2 Performance map of chiller type 2

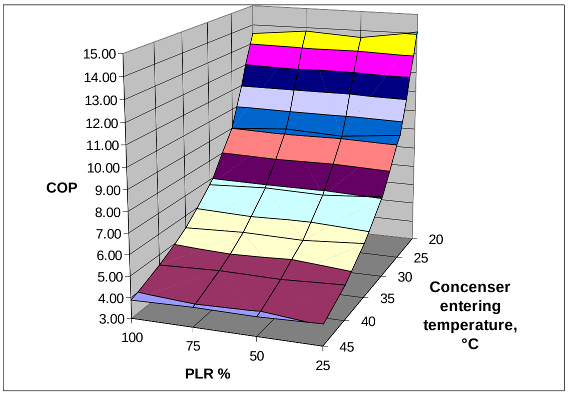

Figure 3 Performance map of chiller type 1, generated with model according to equation (1)

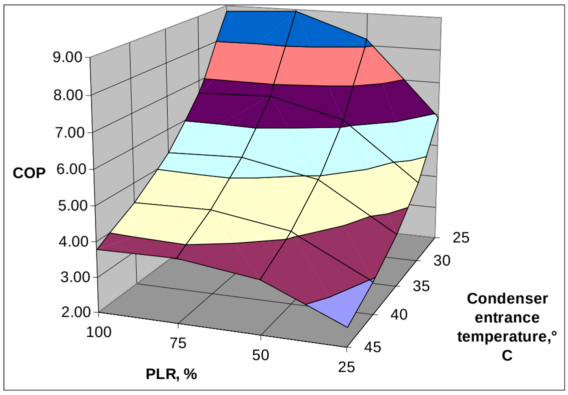

Figure 4 Performance map of chiller type 2, generated with model according to equation (1)

Data sets of two chiller types were made available by (Prochaska 2007) to check the performance of the model. The data are shown in table 4 and figure 1 for chiller type 1 and in table 5 and figure 2 for chiller type 2. It can be noticed that the two machines show a totally different behaviour. Type 1 has an almost perfect reaction on part load and on increased condenser entrance temperatures, whereas type 2 shows the more frequent non ideal behaviour, especially at low part load conditions. It can also be seen that the map of the two machines are not of equal size. Machine type 2 cannot be operated at condenser entrance temperatures below 25°C, therefore these data are missing.

The first approach tried for the model was the combination of a physically based Carnot-factor for the temperature dependency and a 3rd order polynom for the part load dependency as shown in equation (1).

$$\eta_{EER} = \eta_{EER,N} \cdot \frac{273 + \theta_{e,out}}{\theta_{c,in} - \theta_{e,out}} \cdot \left(a \cdot f_{PLR}^{\ 3} + b \cdot f_{PLR}^{\ 2} + c \cdot f_{PLR} + d \right) \quad (1)$$

with

$\eta_{EER,N}$ EER or COP at nominal conditions

$\theta_{e,out}$ Evaporator leaving chilled water temperature, °C

$\theta_{c,in}$ Condenser entering cooling water (or air) temperature, °C

f_{PLR} part load ratio

a, b, c, d calibration factors

The four coefficients a to d are derived from the four data points available by solving a 4 by 4 linear equation matrix.

This first approach was applied to the two chiller types, assuming that only four of the data points were available and ignoring the rest. The used data points are printed in bold in tables 4 and 5.

The results are shown in figures 3 and 4, to be compared to figures 1 and 2. They are not satisfactory in both cases. It becomes clear that the four points, lying on a diagonal line, do not form an enough reliable basis to support the full area of the performance map.

The points in bigger distance from the base diagonal (especially for low condenser temperatures and high PLR, the left-behind corner) are badly represented. This was a sign that the goal to find a model based only on the ARI/ESEER data might not be achieved.

Model Refinement

In a next step, the model was refined by adding correction temperature difference term $\Delta\theta_{cor}$ to equation (1), leading to equation (2).

$$\eta_{EER} = \eta_{EER,N} \cdot \frac{273 + \theta_{e,out} - \frac{\Delta\theta_{cor}}{2}}{\theta_{c,in} - \theta_{e,out} + \Delta\theta_{cor}} \cdot \left(a \cdot f_{PLR}^{\ 3} + b \cdot f_{PLR}^{\ 2} + c \cdot f_{PLR} + d \right) \quad (2)$$

The physical background behind this term is the difference which exists between the cooling and chilled water circuit temperatures and the temperatures of the refrigerant, as explained in figure 5 and equation (3).

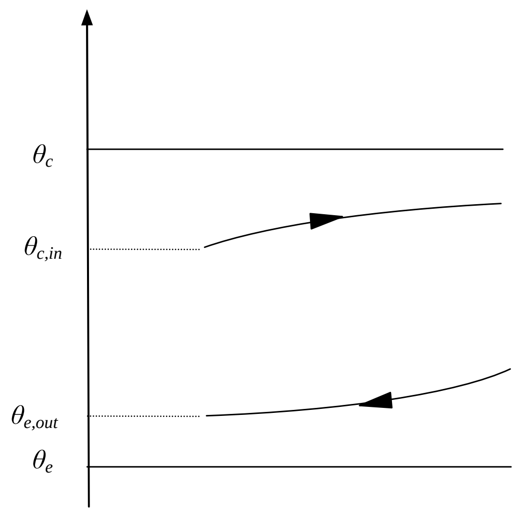

Figure 5 simplified Temperature diagram for chillers

$$\Delta\theta_{cor} = \left(\theta_c - \theta_e\right) - \left(\theta_{c,in} - \theta_{e,out}\right) \quad (3)$$

with

θ_c Refrigerant temperature in condenser, °C

θ_e Refrigerant temperature in evaporator, °C

This approach is still a strong simplification. It is assumed that the temperature difference is constant and equally distributed between the condenser and the evaporator. In practise, it can consist of two different parts for the two heat exchangers and vary with operating conditions.

The value of $\Delta\theta_{cor}$ can be calculated for a specific machine type, when the COP for one additional operating point is known. This is a deviation form the initial goal of relying only on the four rating points. It may be a necessary compromise, however, since – as shown above – the rating points are not in an ideal arrangement for this purpose. In addition, it was hoped that some typical values would be found to make this unnecessary.

If a 5th point is used, this will be ideally one as far as possible off the line drawn by the rating points. In the cases here, the point with a PLR of 50 % and the same condenser entrance temperature (35°C) as for the nominal (100 %) rating point was used (points printed in italics in tables 4 and 5).

The resulting performance maps are shown in figures 6 and 8 and should, again, be compared to figures 1 and 2, respectively. This is done in figures 7 and 9, which show the differences in COP between the model and the original data.

It can be seen that this model modification leads to a considerable improvement. Both map areas have now a comparable shape. The differences in COP can still reach values above 1.0 for machine type 1, for machine type 2 they are below this value. It can be expected that this model will perform quite well in an hourly calculation for these two machines.

Another model improvement was investigated: The $\Delta\theta_{cor}$ value will not remain constant, but rather change with the machine's actual cooling power or PLR. The heat exchangers must transfer higher or lower heat flows and therefore need higher or lower driving temperature differences. Considering this, the model gets the form according to equation (4).

$$\eta_{EER} = \eta_{EER,N} \cdot \frac{273 + \theta_{e,out} - \dfrac{\Delta\theta_{cor} \cdot \dfrac{f_{PLR}}{0.5}}{2}}{\theta_{c,in} - \theta_{e,out} + \Delta\theta_{cor} \cdot \dfrac{f_{PLR}}{0.5}} \cdot \left(a \cdot f_{PLR}^{\ 3} + b \cdot f_{PLR}^{\ 2} + c \cdot f_{PLR} + d\right) \qquad (4)$$

The results for the two machines with this modified model are shown in figures 10 and 12, with the differences to the original data being shown in figures 11 and 13. It can be recognised that there is a further improvement for machine type 1, the differences now being below 1.0 for all values. For machine type 2, however, the results are slightly worse.

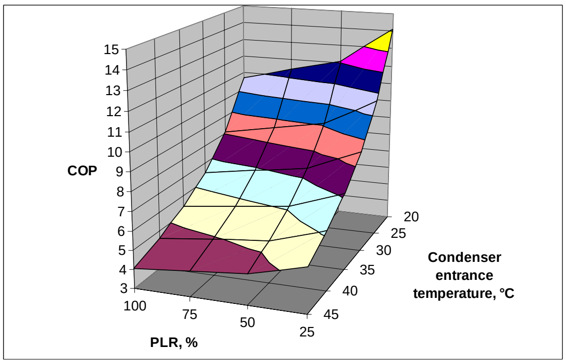

Figure 6 Performance map of chiller type 1, generated with model according to equation (2)

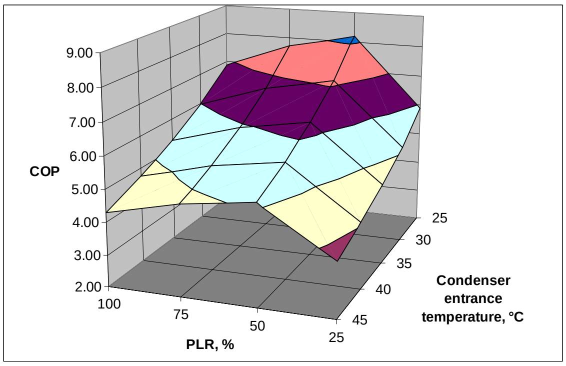

Figure 8 Performance map of chiller type 2, generated with model according to equation (2)

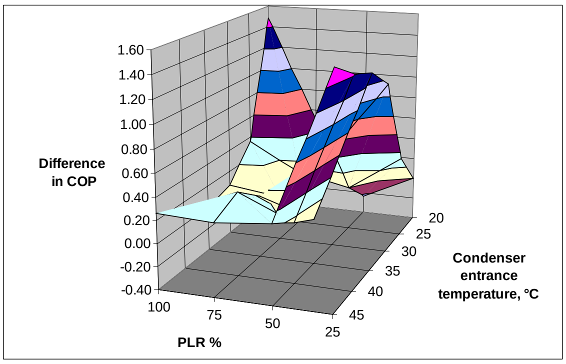

Figure 7 Differences of COP generated with model according to equation (2) against original performance data for chiller type 1

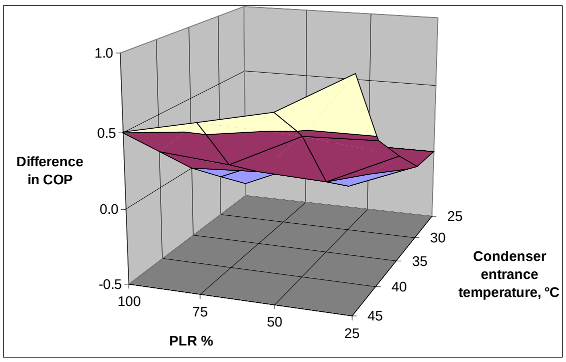

Figure 9 Differences of COP generated with model according to equation (2) against original performance data for chiller type 2

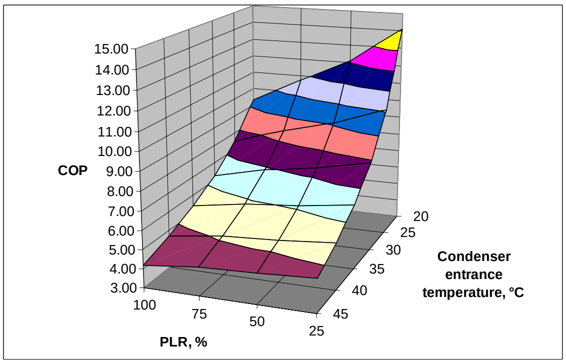

Figure 10 Performance map of chiller type 1, generated with model according to equation (4)

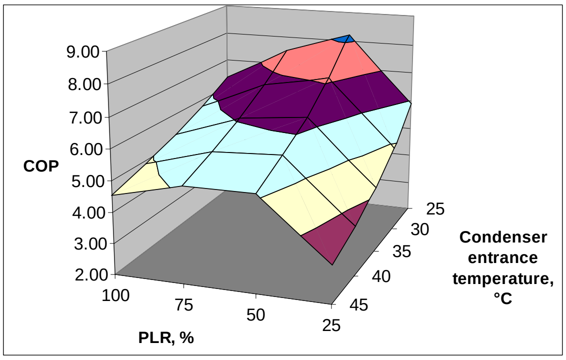

Figure 12 Performance map of chiller type 2, generated with model according to equation (4)

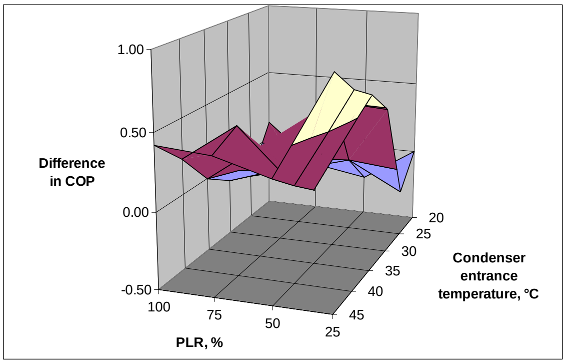

Figure 11 Differences of COP generated with model according to equation (4) against original performance data for chiller type 1,

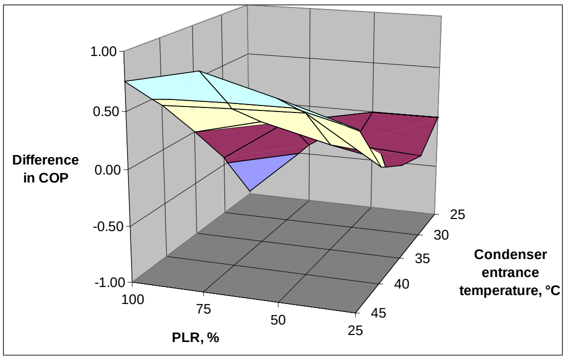

Figure 13 Differences of COP generated with model according to equation (4) against original performance data for chiller type 2

It seems that the last modification improves the model especially for machines, which have a good controllability in part load conditions, whereas for machines with more restrictions and non-idealities, it leads rather to a degradation.

Application to Other Machine Types

For a set of five more machines, there were data available to check the model. However, as opposed to the two machines shown above, there was no full performance map available for these. Apart from the four rating points, COP values for a constant condenser temperature of 32°C and the four PLR of 25, 50 and 75 % were provided. Since 5 points are needed to calibrate the model, there are only two points remaining for checking, in addition to a qualitative-visual examination of the generated performance maps.

A set of two machine types out of the five, type B and D, was selected for reporting in this paper. For both types, a 3D performance map generated with the model and a line chart with COP curves for the different condenser entrance temperatures are shown. Figures 14 to 21 show these graphs for both machine types and for the two model versions equation (2) and (4).

In the line charts (figures 15, 17, 19 and 21) the comparison of the model values with the base vaules can be seen. The four rating points (solid triangles) are on the COP line for the respective temperature. So does the 50 %/32°C value (solid rectangle). An indication of the quality of the model representation is the position of the 75 %/32°C and 25 %/32°C vaules (solid rectangles). Ideally, these should be on the 32°C line, and the closer they are, the better the model.

Machine type B (figures 14 and 15) was chosen as an example of very good representaion. The two points are almost exactly on the line with only a small difference for the 75 % value. Machine type D (figures 16 and 17), however, is an example of a bad representation. There are substancial differences for both values. The 3D charts for both machines (figures 14 and 16) do not give any indication of an unplausible shape.

For these two machines it was also investigated, whether the model refinement from equation (2) to equation (4) would bring an improvement. The results are shown in figures 18 and 19 (machine type B) and 20 and 21 (machine type D). It can be noticed that there is rather a decrease in quality for both machine types. The two check points are for both cases in bigger distance to the respective curve, and the shape of the curves looks rather less realistic, especially for machine type D (figures 20 and 21). Similar findings can be reported for the other 3 machine types A, C and E not shown here. This is another reason for not further relying on the equation (4) model refinement, although it may give improvements in special cases.

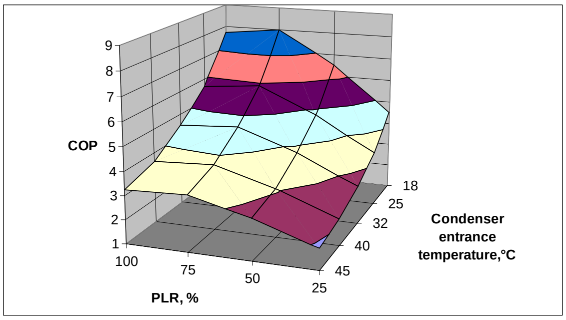

Figure 14 Performance map of chiller type B, generated with model according to equation (2)

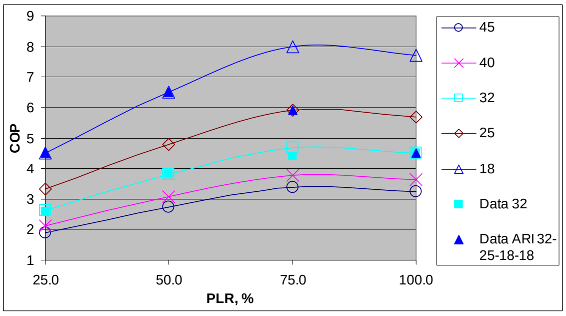

Figure 15 COP curves for different condenser temperatures of chiller type B, generated with model according to equation (2)

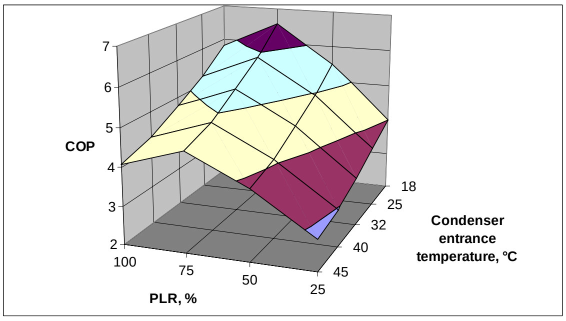

Figure 16 Performance map of chiller type D, generated with model according to equation (2)

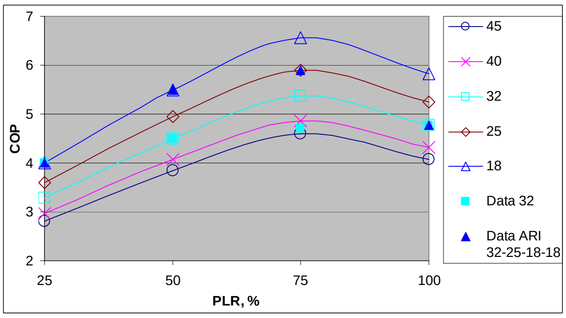

Figure 17 COP curves for different condenser temperatures of chiller type D, generated with model according to equation (2)

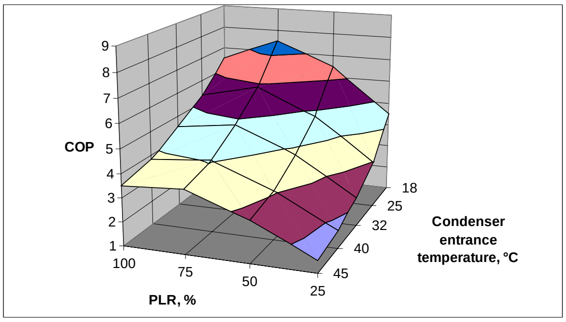

Figure 18 Performance map of chiller type B, generated with model according to equation (4)

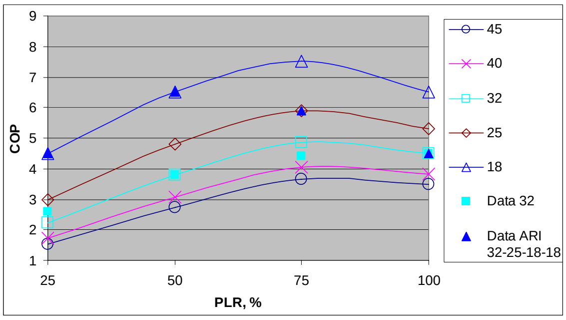

Figure 19 COP curves for different condenser temperatures of chiller type B, generated with model according to equation (4)

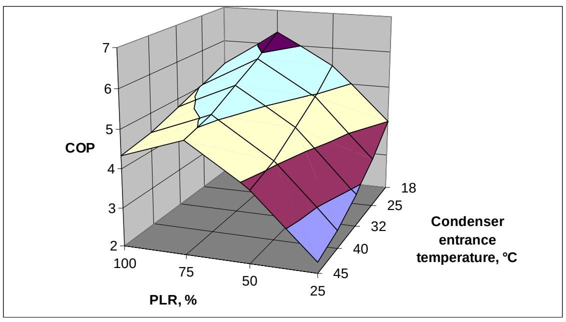

Figure 20 Performance map of chiller type D, generated with model according to equation (4)

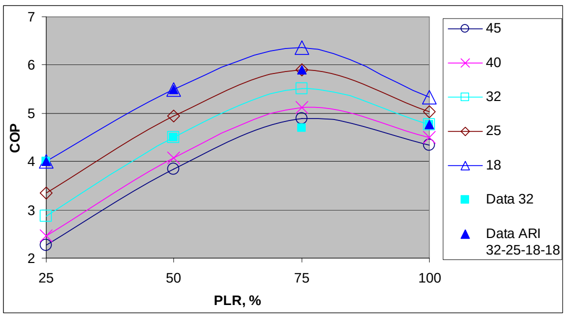

Figure 21 COP curves for different condenser temperatures of chiller type D, generated with model according to equation (4)

APPLICATION TO AN ANNUAL CALCULATION

Example Building Load Profile

To evaluate the model's performance on an annual calculation, it was applied to an annual load profile of an office building, the details of which cannot be shown here. A picture of the building is shown in figure 22. The frequency distribution of the hourly cooling loads is shown in table 6 in comparison with the weighting factors used for the performance rating of the ARI and ESEER standards.

Figure 22 Example building used for annual calculations

Table 6
Relative load frequency for the used object compared to ARI and ESEER weighting factors

	RELATIVE FREQUENCY	WEIGHTING FACTOR	
	OBJECT	ARI	ESEER
PLR>0.75	0.002	0.01	0.03
0.5<PLR≤0.75	0.012	0.42	0.33
0.25<PLR≤0.5	0.048	0.45	0.41
PLR≤0.25	0.938	0.12	0.23

Table 6 shows that the used object has a considerably higher number of hours with low PLR values (and vice versa) than the performance ratings of the standards propose. This is, among other factors, due to a continuously running data centre using a relatively low portion of the total cooling power. It is, however, considered a typical case for the Swiss market, showing that the general validity of the rating factors is questionable.

Annual Performance Factor Results

The application of the model performance maps to the hourly load profile of the building results in an annual performance factor. In order to evaluate the precision of this value, it has to be compared to a calculation using the original full performance map data (from tables 4 and 5 and figures 1 and 2). To enable this, these maps had to be brought into a form of a function. A 5^{th} by 3^{rd} order (for machine type 1) or 4^{th} by 3^{rd} order (for machine type 2) two dimensional polynom was used to represent them. These functions were applied to the same hourly building load profile. The result is shown in table 7. In addition – for completeness – the results of applying the standard rating weighting factors to the two machine types are reported. Only the ARI and model values can be shown for machines B and D.

Table 7 Annual performance factors compared to calculations with full original performance map data and with ARI and ESEER weighting factors

	ORIGINAL	MODEL	ARI	ESEER
TYPE 1	12.5	12.7	10.8	10.3
TYPE 2	5.8	6.2	7.7	7.3
TYPE B		5.0	6.0	
TYPE D		4.0	5.5	

The results in table 7 show a very good performance of the model for both machine types compared to the original data. There are substantial differences to the ARI and ESEER rating values. These differences are in opposite direction for the two machine types. The rating underestimates the performance of machine type 1, but overestimates machine type 2. A similar difference is seen for the two machines type B and D, but there is no "true" value to compare with for these.

CONCLUSION

The goal of developing a chiller model, representing the whole performance map based only on the information of four rating values of the COP according to the ARI or ESEER standards could not be achieved. However, by adding only one more COP value for another measuring point, the model gives, as could be seen so far, very accurate results for most types of machines. There are some types where the model does not represent correctly all areas of the part load and condenser entrance temperature ranges. However, these are restricted areas, some of them with rather low numbers of expected operation hours. On a whole year hourly simulation, these deviations will most probably equal out to a great extent.

The model has only been tested on one specific load profile. Further tests with different machines and load profiles will be necessary (and carried out during the test phase of the software where the model is implemented in) to get an enough reliable base of security.

In comparison with the standard ratings, there are considerable differences in seasonal performance factors. The fact, that the four rating points are not sufficient to support the generation of a full performance map, is partly due to their arrangement. They are arranged too much on a diagonal of the performance map. It is, however, not sure that another arrangement of the four points would be sufficient for the generation of performance maps. Knowing, that this is not the purpose of these rating points, it would nevertheless be of help if there could be a development in the direction of a better support for this.

ACKNOWLEDGEMENT

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