

MODELLING OF HVAC SYSTEM COMPONENTS FOR BUILDING DYNAMIC SIMULATION

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ABSTRACT

The reliable assessment of the annual energy demand has become necessary in view of building energy performance certification. Accurate models must be used to simulate the behaviour of HVAC components in real operation, usually characterized by a wide variation of building loads. In this context, this paper deals with the development and validation of an algorithm aimed at the assessment of part load performance of various kinds of controls for vapour compression based heat pumps and chillers, in particular referring to on-off, inverter-driven and multi-stage vapour compression. The reliability of this algorithm in the calculation of seasonal performances is checked against monitoring of heat pumps and chillers operating under real conditions.

INTRODUCTION

Nowadays the importance of part load operation in seasonal performance assessment has been acknowledged, especially with regard to heat pumps and chillers. As a matter of fact, these machines are usually sized for design conditions, therefore they normally work at part load conditions for the most of the season. For a correct evaluation of this behaviour it is necessary to consider the real heating/cooling profiles as well as to investigate how the machine works under such variable conditions. For this purpose, accurate software able to calculate heating/cooling loads of the building in dynamic conditions are today available on the market, also for professional purposes, and they will probably become more and more usual even in common design. At each simulation time step they may calculate the energy demand and the related efficiency of heat pumps and chillers as a function of both secondary temperatures and heating/cooling loads. For this aim some algorithms able to quantify the influence of the load factor on the efficiency of heat pumps and chillers have been developed and refined.

Furthermore, the Committees for Standardization of European Countries are completing and improving the technical Standards aimed at the energy certification of buildings. In this context, it is now needed to introduce an algorithm to calculate the seasonal efficiency of heat pumps and chillers,

defined as the the ratio of the total amount of heating/cooling energy provided by the unit during the seasonal operation to the total effective electric energy consumed during the same period.

In this paper, chillers will be considered in detail, whereas for heat pump analogous considerations may be assumed. In this case the energy efficiency is named *EER* (Energy Efficiency Ratio) and *SEER* is the Seasonal average of *EER*. One proposal refers back to the *IPLV* index introduced by ARI in the 1980s with Standard 550/590 (ARI, 2003), a performance rating Standard self-regulated through collaboration of associations, such as ARI, ASHRAE, and participating manufacturers. More recently, a new version of this index, *ESEER* (European *SEER*), has been introduced by Eurovent, more suitable for European climates and operating conditions than *IPLV*, as a voluntary certification among manufactures in Europe (Adnot et al. 2003). Both *IPLV* and *ESEER* characterize the behaviour of the machine in part load conditions, but regardless of the building where it is installed. This way, even if important to give more information about the actual behaviour of chillers under real operation conditions than the simple nominal *EER* at full load, *IPLV* and *ESEER* can not be considered an acceptable estimation of the actual *SEER* of a chiller installed in a particular building and climate. As a consequence, modifications are being introduced in the procedure to calculate *SEER*, as foreseen in EN 14825 (EN, 2012). Nevertheless, in Italy, two Standards already exist: UNI 11135 (UNI, 2004) and UNI 10963 (UNI, 2001) aimed at the assessment of the seasonal efficiency and at the definition of the minimum tests necessary to trace the behaviour of chillers working under part load conditions respectively. Unfortunately, these Standards are currently far from being applied, especially because they need the execution of specific laboratory tests under part load conditions, whereas index *ESEER* has met a large favour among European manufactures of chillers because of its simplicity. In this paper, a procedure is presented to calculate part load efficiency curves starting from the data used to calculate *ESEER*. This way, with no more tests, it is possible to achieve data to be used both in dynamic simulation and in building energy certifications. Obviously, the same procedure can use *IPLV* data instead of *ESEER* ones.

MODELLING

The procedure consists in a calculation procedure which is also normally used together with the dynamic simulation. In fact, in dynamic building energy simulations, the calculation by a dynamic algorithm of the building thermal demand is usually accompanied by a quasi steady-state evaluation of the plant behaviour.

The procedure can be divided into the following three calculations steps:

- 1- Evaluation of the declared capacity (DC) and related EER under full load conditions (EER_{DC}) as functions of the monthly mean temperatures of the secondary fluids exchanging heat at the evaporator and condenser. Performance data from the manufacturer may be used, and integrated by means of linear interpolation to adapt to current boundary conditions.
- 2- Assessment of the monthly mean capacity ratio CR as the ratio of the building cooling demand to the maximum energy which could be supplied in the month (the last one is obtained multiplying the mean declared capacity by the working hours of the chiller in the month).
- 3- The monthly mean EER is obtained multiplying EER_{DC} by a part load factor (PLF) aimed to take into account the influence of part load operation, being PLF a function of CR .

This calculation procedure may be applied at each time step of dynamic building energy simulations as well as in monthly energy calculations, using monthly means for temperatures and energy ratios, thus achieving monthly means for EER_{DC} and CR . As a consequence, it could be applied also to building energy certification calculations, based on monthly calculations. And $SEER$ can be calculated for any condition too. As a matter of fact, starting from the EER_i (with $EER_i = EER_{DC} \cdot PLF$) for each time-step i (one hour long dynamic building energy simulations or one month long in energy certification calculations) and indicating with n the number of time-steps in the season, we have:

$$SEER = \frac{Q_{c,s}}{Q_{e,s}} = \frac{\sum_{i=1}^n Q_{c,i}}{\sum_{i=1}^n Q_{e,i}} = \frac{\sum_{i=1}^n Q_{c,i}}{\sum_{i=1}^n \left(\frac{Q_{c,i}}{EER_i} \right)} \quad (1)$$

where:

- $Q_{c,s}$ and $Q_{c,i}$ are the cooling energies given in the season and in the single time-step respectively.
- $Q_{e,s}$ and $Q_{e,i}$ are the electric energies absorbed in the season and in the single time-step respectively.

PLF CALCULATION

In figure 1 some experimental values of PLF are reported as functions of CR for an air-to-water chiller with scroll compressor and on-off capacity control

(nominal cooling capacity: 10 kW). Experimental values of parameter Z are also shown.

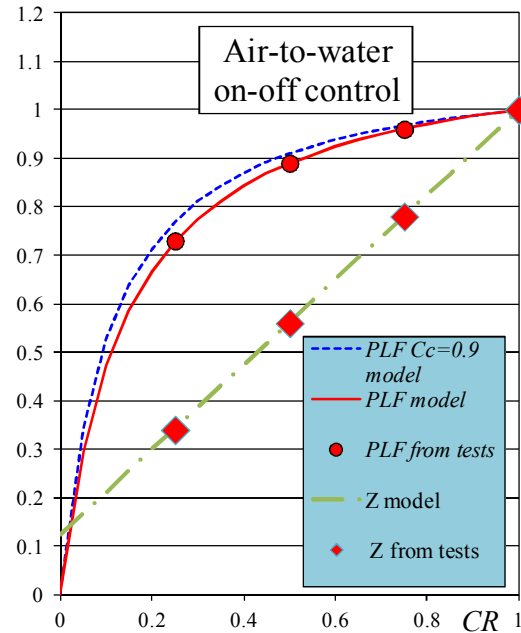


Figure 1. Curves Z - CR and PLF - CR and some values from tests of a chiller with on-off control

Parameter Z is the ratio of the real electric consumption of the chiller to the total electric consumption under full load, with the same temperatures of the secondary fluids. Tests in figure 1 suggest a linear trend of Z against CR , thus resulting in a simple mathematical model for PLF (Bettanini et al., 2003):

$$Z = \frac{Pe}{Pe_{DC}} = a \cdot CR + b \quad (2)$$

$$EER = \frac{Pc}{Pe} \quad EER_{DC} = \frac{Pc_{DC}}{Pe_{DC}} \quad (3)$$

thus achieving:

$$PLF = \frac{EER}{EER_{DC}} = \frac{Pc}{Pe} \frac{Pe_{DC}}{Pc_{DC}} = \frac{CR}{\frac{Pe}{Pe_{DC}}} = \quad (4)$$

$$= \frac{CR}{Z} = \frac{CR}{a \cdot CR + b}$$

The final conclusion is that only one test under part load conditions in addition to the one at full loads is sufficient to define the correlation PLF - CR in case of on-off capacity control. As a matter of fact, figure 1 shows this model fits quite well the monitored values. In particular, in case of on-off chillers, EN 14825 proposes the following correlation:

$$PLF = \frac{CR}{C_c \cdot CR + (1 - C_c)} \quad (5)$$

Where C_c is a degradation coefficient. In absence of test values, the default value of C_c is 0.9. As shown in figure 1, this assumption leads to an acceptable deviation from the EN model. In case of variable capacity control chillers, such as multistage or inverter-driven compressors, the PLF - CR curve is obtained by linear interpolation between part load data rated in laboratory tests. For CR values falling below the minimum percentage of continuous modulation the on-off behavior takes place, with consequent adaption of the above seen model.

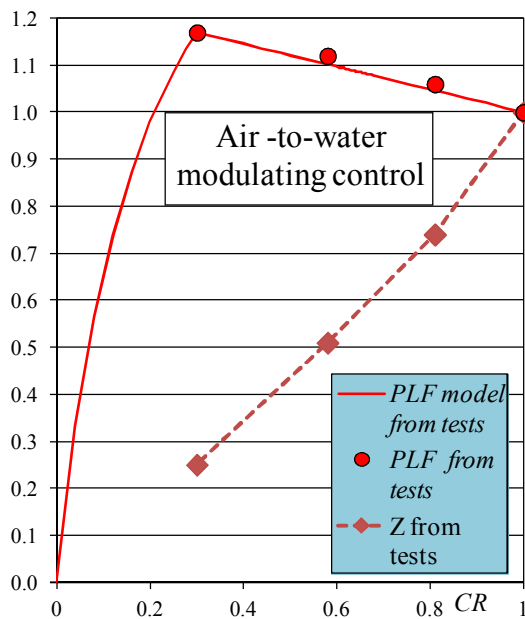


Figure 2. Curves Z - CR and PLF - CR and values from tests with modulating control

In figure 2 experimental values of PLF and Z are shown for an air-condensed chiller equipped with scroll compressor, driven by inverter down to 30% of the nominal capacity (16 kW).

In this case, the trend of Z is not linear, but a linear correlation between $CR = 0.3$ and $CR = 1.0$ gives an acceptable estimation for PLF in this interval. Laboratory tests provided analogous results also for air conditioners, such as split systems (Bettanini et al., 2001).

ESEER CALCULATION

Normalized index $ESEER$ (European SEER) proposed by Eurovent is introduced to compare part load performances of chillers, but this value can not represent the seasonal efficiency of a real chiller installed in a specific building, because it is calculated referring to predefined and conventional cooling load profiles and secondary fluid temperatures. The cooling load profile as well as the reference secondary fluid temperatures widely vary

depending on building intended use, HVAC terminal units, and HVAC water heat storages. Anyway, this index allows a consistent comparison among homologous products available on the market, because it highlights the behaviour of chillers under part load conditions. The $ESEER$ is calculated through the following equation:

$$ESEER = A \cdot EER_A + B \cdot EER_B + C \cdot EER_C + D \cdot EER_D \quad (6)$$

where:

- $EER_A, EER_B, EER_C, EER_D$ are the Energy Efficiency Ratios at the four partial load working conditions forward defined; EER is the ratio of the actual cooling capacity (equal to the cooling load) to the actual power consumption.

- A, B, C, D are the weighting factors of the corresponding part load $EERs$.

The test conditions to calculate the part load $EERs$ are shown in Table 1 according with the Standard, depending on the type of chiller or air-conditioner. In Table 1, Inlet fluid temperatures (evaporator or condenser) are indicated with “i”, outlet fluid temperatures (evaporator) with “o”. For example, air-condensed chillers at test conditions “A” work with entering air temperature at 35°C and leaving water temperature at 7°C, in case of fan-coils, or 12°C, in case of cool floors. For every test the Standard indicates also the inlet–outlet temperature differences in the heat exchangers.

Table 1. Inlet (t_i) and outlet (t_o) temperatures for part load tests for different chillers and air-conditioners

Type:		air-air		water-air	
Test	Load	t_i , Outdoor air	t_i , Indoor air	t_i , Water tower/sink	t_i , Indoor air
A	100%	35	27	30/15	27
B	75%	30	27	26/15	27
C	50%	25	27	22/15	27
D	25%	20	27	18/15	27

Type:		air-water		water-water	
Test	Load	t_i , Outdoor air	t_o , Water fan-coil/panels	t_i , Water tower/sink	t_o , Water fan-coil/panels
A	100%	35	7/18	30/15	7/18
B	75%	30	7/18	26/15	7/18
C	50%	25	7/18	22/15	7/18
D	25%	20	7/18	18/15	7/18

In Table 2 the weighting factors are summarized on the basis of the type of cooling unit. In particular, two different series of values are considered, depending on the external secondary fluid. In some cases, the Standard admits the calculation of the part load efficiency starting from mere full load tests. For the sake of brevity, these cases are not presented here.

Table 2. Weighting factors for part load $EERs$ chosen to calculate $ESEER$

		Type:			
Test	Load	air-air	water-air	air-water	water-water
A	100%	4%	4%	3%	3%
B	75%	26%	26%	33%	33%
C	50%	40%	40%	41%	41%
D	25%	30%	30%	23%	23%

relevance to focus the attention of the HVAC plant designer not only on the full load nominal efficiency but even on the behaviour under part load conditions and allows the HVAC plant designer to compare more consistently different models available on the market. But *ESEER* can not represent the real seasonal efficiency of a particular chiller or air-conditioner in a specific context. For example, in case of air-condensed chillers, the calculation of *ESEER* is based on the simplification that the cooling load is a linear function of the only outdoor air temperature, whereas it greatly depends on internal heat gains, latent loads, solar radiation and occupancy schedules. Obviously this is not sufficient for a reliable assessment of the seasonal efficiency of machine chiller or air-conditioner installed in a specific context. Therefore it is necessary to introduce another procedure for this purpose.

PLF FROM ESEER DATA

As an reference, in this paper the frequent case of air-condensed chillers and air-conditioners may be considered. At fixed temperatures for the internal secondary fluid, *EER* depends on the outside air temperature and on the degree of part load. The experimental tests provide typical *EER* profiles as the ones shown in figure 3. To reduce the number of tests needed to draw such profiles, the *PLF-CR* curve (where *PLF* is the ratio between *EER* and *EER_{DC}*) is assumed as fixed for different outside air temperatures. In particular, the *PLF* values required to build this curve can be obtained at different thermal levels, thus leading to the use of *EER_A.....EER_D* obtained for the *ESEER* calculation and adapt them to calculate also four more values of *PLF*, thus allowing the estimation of the *PLF-CR* curve also for other temperature conditions.

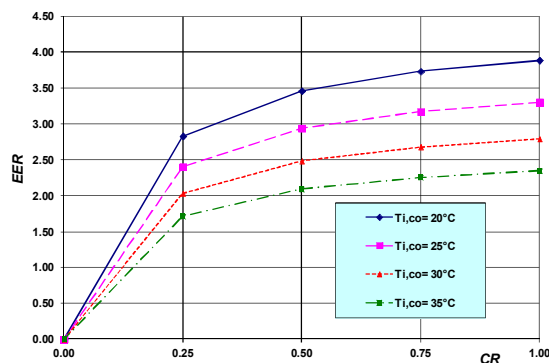


Figure 3. *EER-CR* for various condenser inlet air temperatures (*T_{i,co}*) for an air-cooled chillert

In detail:

$$PLF_r = \frac{EER_r}{EER_{DCr}} \quad (7)$$

where:

- *EER_r* is available from the data required by EN 14825 (with *r* = *A*, *B*, *C*, *D*);

- *EER_{DCr}* can be obtained from the tables of full capacity performances provided by manufacturers in usual technical documentation.

Full load							
	CR (%)	To,ev (°C)	Ti,co (°C)	Pc (kW)	Pe (kW)	EER _{DC}	
A	100	7	30	13.10	3.17	4.13	
B	100	7	26	13.71	2.85	4.81	
C	100	7	22	14.01	2.70	5.19	
D	100	7	18	14.22	2.60	5.47	
Part load							
	CR (%)	To,ev (°C)	Ti,co (°C)	Pc (kW)	Pe (kW)	EER	PLF
A	100	7	30	13.10	3.17	4.13	1.00
B	75	7	26	10.28	2.22	4.63	0.96
C	50	7	22	7.01	1.50	4.66	0.90
D	25	7	18	3.56	0.87	4.10	0.75

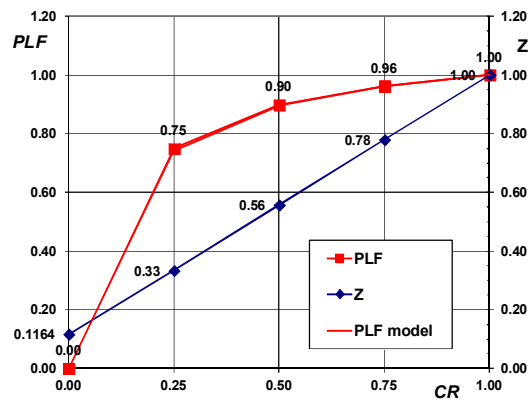


Figure 4. Full and part load performance and consequent *PLF-CR* and *Z-CR* curves for chiller 1 (water-to-water)

The part load coefficient *PLF* mostly depends on the part load effect, whereas secondary fluid temperatures have lower influence. The four values of *PLF_r* with *r* = *A*, *B*, *C*, *D* correspond to *CR* = 1.00, 0.75, 0.50, 0.25 respectively. From these four values, the values of the part load coefficient can be obtained by simple interpolation for any capacity ratio *CR*. The application of the procedure to some typical commercial chillers is presented here. For each chiller a table is presented with working data under full and part load conditions. In deeper detail, the cooling capacity *P_c*, absorbed electric power *P_e* and consequent *EER* are given for the secondary fluid temperatures provided in the *SEER* procedure. The ratio of *EER* to *EER_{DC}* gives the corresponding *PLF* value. In figure 4, a water-condensed chiller (machine 1) is considered with just one refrigeration circuit equipped with a scroll compressor. The set of the full and part load performances and the corresponding *Z-CR* and *PLF-CR* profiles are shown in figure 4. For this on-off chiller the linear trend of *Z* is confirmed. The performance data at full load have been obtained from the available technical documentation. The part load performance is available from the manufacturer as well. In particular, it is obtained by certified tests done in authorized laboratories as required by Eurovent/CEN and used by the manufacturer to calculate the *ESEER*

reported in the commercial catalogue. For this chiller *ESEER* is 4.51.

Full load							
	CR (%)	To,ev (°C)	Ti,co (°C)	Pc (kW)	Pe (kW)	EER _{DC}	
A	100	7	35	10.40	4.43	2.35	
B	100	7	30	11.08	3.97	2.79	
C	100	7	25	11.69	3.54	3.30	
D	100	7	20	12.28	3.16	3.89	
Part load							
	CR (%)	To,ev (°C)	Ti,co (°C)	Pc (kW)	Pe (kW)	EER	PLF
A	100	7	35	10.40	4.43	2.35	1.00
B	75	7	30	8.31	3.10	2.68	0.96
C	50	7	25	5.85	1.99	2.94	0.89
D	25	7	20	3.07	1.08	2.83	0.73

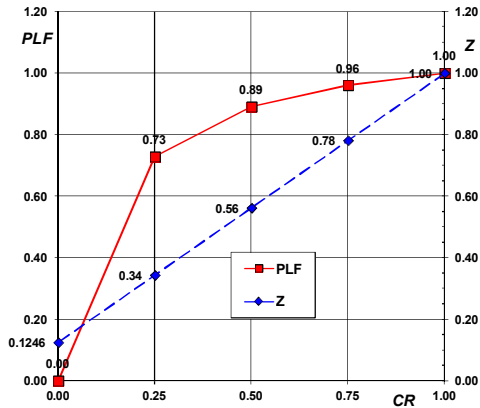


Figure 5. Full and part load performance and consequent PLF-CR and Z-CR curves for chiller 2 (air-cooled)

In figure 5 the data and the consequent profiles of parameters Z and Y are presented for an air-condensed chiller with only one circuit and one scroll compressor (machine 2). As often happens, in this case the full load capacity at 20°C is not present in the technical documentation provided by the manufacturer.

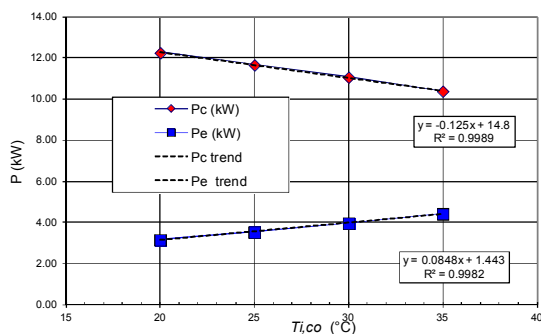


Figure 6. Cooling capacities Pc and absorbed electric power Pe as functions of outdoor temperature (Ti,co) for chiller 2 (air-cooled).

It is although possible to use a mathematical extrapolation, because the cooling capacities and absorbed electric powers have usually a linear profile, as shown in figure 6. The validity of the linear regression is quantified by the high value of the correlation coefficient R^2 .

$$R^2 = \frac{\sum(Pc_{calculated} - \bar{Pc})^2}{\sum(Pc_{real} - \bar{Pc})^2} \quad (8)$$

where \bar{Pc} is the mean value of all the real measured Pc (or Pe).

For chillers 1 and 2, characterized by on-off capacity control, the linear profile of parameter Z confirms the possibility to use the model of equation (4) as an alternative of the interpolation among the four points from tests. But this simple model is no more usable in presence of multistage or invert-driven compressors. A chiller provided with multistage capacity control is now presented in figure 7, consisting in an air-condensed chiller with four scroll compressors and two refrigerating circuits (machine 3), with *ESEER* equal to 4.40. Also in this case the correlation of Z is not linear but the four values of PLF permit an acceptable evaluation of this factor in the whole range.

Starting from the certified values of *EER*, available because needed to calculate the *SEER*, the manufacturer can immediately calculate four values of the Y coefficient for four load percentages (25%, 50%, 75%, and 100%). Therefore it is sufficient that the manufacturer introduces these four values in the technical documentation besides the *ESEER* value. This way, with no more tests, the HVAC plant designer will be able to apply the proposed procedure to calculate the seasonal efficiency, thus achieving the electric energy demand for cooling both in building energy dynamic simulations and in steady-state procedures for building energy certification.

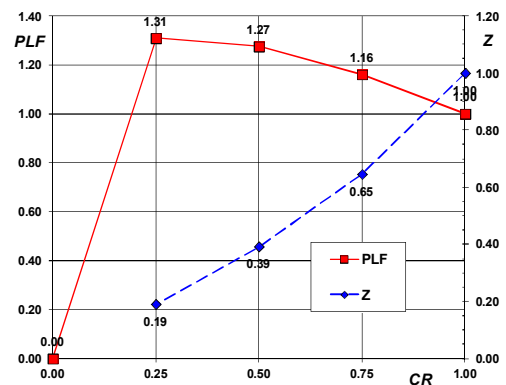


Figure 7. Full and part load performance and consequent PLF-CR and Z-CR curves for chiller 3 (air-cooled).

SIMULATION VS MONITORING

A comparison between simulation results and data from monitoring of a chiller installed in a residential building is presented. The unit is an air-cooled chiller with one scroll compressor and on-off control integrated into a HVAC plant based on fan-coil terminal units. The nominal capacity is 5 kW and the cold water is normally produced at 7°C. The *ESEER* of the unit is 3.23. By using the procedure previously

presented, the four values of $PLFs$ are calculated from $ESEER$ data provided by the manufacturer.

The monitoring system has collected the chiller inlet/outlet water temperatures, the outdoor air temperature and the electric energy consumption of the unit with a one-minute interval for a summer in Italy (from June to September). The water flow rate to the evaporator was constant and it was measured. Starting from these measures, the hourly average of the cooling capacity and of the electric consumption have been calculated and the consequent the EER values. By interpolation between the data from the manufacturer, the EER_{DC} are evaluated as a function of the measured water and outdoor air temperatures. This way experimental $PLFs$ (ratio EER/EER_{DC}) have been obtained.

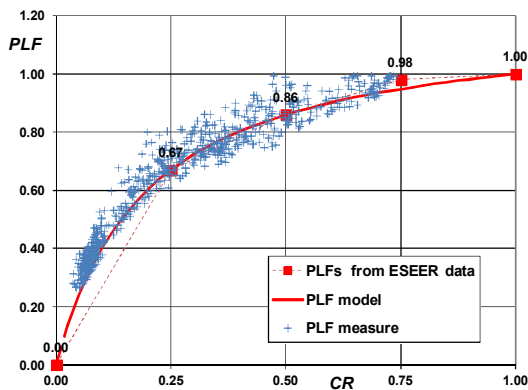


Figure 8. PLF measures for the monitored chiller and PLF model obtained by means of $PLFs$ from $ESEER$ data.

In figure 8 a comparison between measured $PLFs$ and calculated ones (by linear interpolation between the fourth values calculated from $ESEER$ data) is shown. For low CR values better correlation with the mathematical model of equation (4), also reported in figure 8, is identified, as the unit presents an on-off capacity control. A validation of the procedure to simulate the seasonal performance has been carried on, considering, for the whole summer, the actual cooling energy measured in every hour. For each hour, on the basis of the water and outdoor air temperatures, the chiller performance at full load are calculated as well as the CR and consequent PLF .

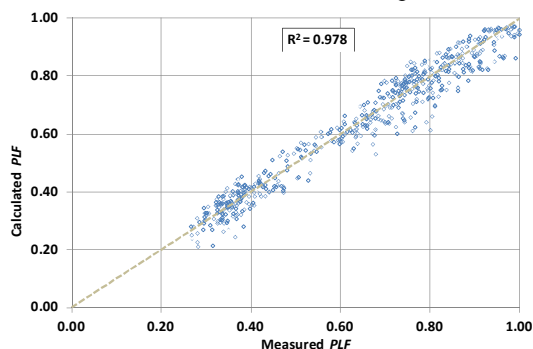


Figure 9. Comparison between PLF measures and calculated $PLFs$ for the monitored chiller.

In figure 9 a comparison between the calculated PLF values and the corresponding measured ones is proposed. You can note the high correspondence quantified by a high correlation coefficient R^2 . On the other hand, the validity of the mathematical model in presence of on-off capacity control is confirmed also by the good prediction of Z shown in figure 10. Furthermore, in each hourly time step, the mean electric consumption is calculated as the ratio of the mean measured cooling capacity provided to the corresponding calculated PLF in the same interval. Figure 11 shows the calculated mean electric consumption values versus the corresponding measured ones. Finally, in table 3, the seasonal results are resumed, starting from the same measured cooling energy provided to the building.

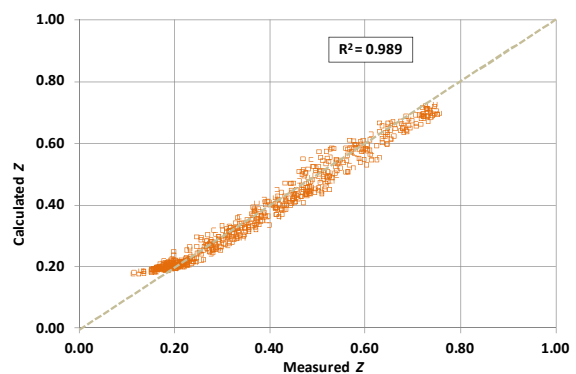


Figure 10. Comparison between Z measures and calculated Z values for the monitored chiller

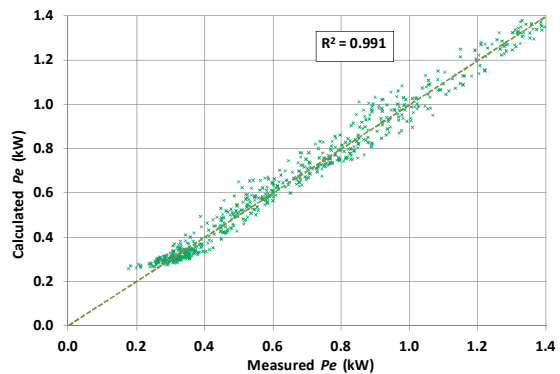


Figure 11. Comparison between electric absorption Pe (kW), measured and calculated, for the monitored chiller

The good prediction of the real $SEER$ can be appreciated in table 3.

Obviously, this analysis is not a validation of the global model of the building-plant system, but only a test for a procedure to assess $SEER$ to be implemented in simulation models. Indeed the validation of building models is not the scope of this paper.

Table 3. Comparison between measured and simulated seasonal performance of the monitored chiller

	Measured	Simulated
Cold demand (kWh)	1071	
Electric consumption (kWh)	434	437
SEER	2.47	2.45

CONCLUSION

A simple algorithm to simulate the part load efficiency of chillers and air-conditioners has been presented and validated in a case study. Its fundamental prerogative is the capacity to evaluate the behaviour of the unit in part load conditions. The data required to characterize each particular unit can be derived from information usually available in manufacturer's catalogues and specifications sheets. The goal of the model is to allow a correct assessment of the seasonal performance of chillers and air-conditioners.

NOMENCLATURE

CR	= capacity ratio
C_c	= degradation factor
EER	= energy efficiency ratio
EER_{DC}	= energy efficiency ratio at full load (DC=Declared Capacity)
$SEER$	= seasonal energy efficiency ratio
Pe	= absorbed electric power, kW
Pe_{DC}	= absorbed electric power at full load, kW
P_c	= cooling capacity, kW
P_{cDC}	= cooling capacity at full load, kW
PLF	= part load factor
Q_c	= cooling energy produced, kWh
$Q_{c,s}$	= seasonal cooling energy produced, kWh
Q_e	= electric consumption, kWh
$Q_{e,s}$	= seasonal electric consumption, kWh
$T_{i,co}$	= secondary fluid inlet temperature at the condenser, °C
$T_{o,ev}$	= secondary fluid outlet temperature at the evaporator, °C

z = ratio of electric power absorption to the electric absorption at full load with the same thermal levels of the secondary fluids

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