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SIMPLIFIED MODELS TO SIMULATE PART LOAD PERFORMANCES OF AIR CONDITIONING EQUIPMENTS

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ABSTRACT

The performances of air conditioning equipments, like heat pumps, chillers or air conditioners, derive not only from the operative thermal levels, but also from the building requirement trend which normally involves frequent reductions of the full capacity. Nowadays an increasing consciousness of the importance of part load working influence on the equipment long term efficiency has stimulated the investigation about part load working behaviour of the machine. Owing to the incredible lack of experimental data about this topic, suitable new tests in laboratory and field monitoring, expressly carried on, have shown the necessity of new algorithms. In this paper these new mathematical models and their implementation in building-plant simulation codes are presented. The results obtained by their use for the calculation of the seasonal performances of HVAC systems in some typical applications are also discussed.

INTRODUCTION

In the most famous and diffuse computer programs for the dynamic simulation of building-plant system, besides the dynamic procedure to calculate the building thermal load we normally find a method to evaluate the average performance of the plant in steady-state conditions on a time interval equal to the time step of the simulation (usually one hour). As regards the inverse cycle machine, the procedure is based on the use of working curves from the manufacturer or from literature and in details it consists of two stages. First the calculation of the capacity and of the coefficient of performance (*COP*) at full load operating conditions normally as a functions of the thermal levels of the external fluids exchanging heat fluxes with the machine at the condenser and evaporator.

Then a mean part load ratio (*PLR*) is calculated as the ratio of the building requirement really supplied by the machine in the time step to the maximum energy which could be supplied in the same time interval in the case of continuous working at full capacity. The part load influence is taken into account by multiplying the full load *COP* for a part load factor (*PLF*) calculated as a function of *PLR*. The ARI standard suggests a generalized use of the following equation to calculate *PLF* (ARI, 1989):

$$PLF = 1 - c_d \cdot (1 - PLR) \qquad (1)$$

where c_d is a degradation coefficient specified by the manufacturer or taken to be 0.25 as a default value. This second event is more frequent and also in the following analysis here presented this value has been considered.

In effect, as presented later, the results of tests recently carried out, have shown how much unacceptable is the approximation of this simple correlation and its application for refrigeration machines also very different for operating and control modes. The exigence to have the correct algorithm, verified for each type of unit, leads to the elaboration of a standard which specifies the test conditions for the rating of commercial machines in order to estimate their behaviour also in part load working conditions. For this aim in Italy a new standard, UNI 10963, has been recently introduced and also proposed to ISO . The UNI 10963 "Air conditioners, chillers and heat pumps. Determination of the part load performances" has the merit of the novelty not only in Italy but also at international level. It establishes a procedure to evaluate in laboratory the minimum data necessary for the ratings of inverse cycle machines in part-load operating conditions. It also proposes a presentation of the test results in terms of calculation models useful for the building-plant system simulation distinguishing between on-off, modulating o multistage machines.

This activity has been obstructed by the deep lack of data in literature, especially experimental, about this topic. Therefore new experimental tests have been carried out expressly to develop this new standard. So at the University of Turin tests in laboratory on commercial air to air reversible heat pumps were performed, until now only on-off o modulating by inverter machines (Anglesio et al., 2001). For this reason the rating conditions and the presentation of the results foreseen by UNI 10963 are at present verified only for this kind of machines. The extension of the standard to the others and in particular to the water chilling packages is not validated yet and so it must be considered only as a first proposal to stimulate new experimentations.

In absence of laboratory tests, for chillers we have tried to use experimental data from long term monitoring of machines integrated in HVAC plants of real buildings. The mathematical models consequently elaborated have been used for the simulation of the seasonal performances of the building-plant system. The results obtained in some typical application cases are here presented and discussed.

THE MODELS FOR AIR CONDITIONERS

On the basis of the laboratory tests the UNI 10963 proposes a different algorithm to calculate *PLF* as a function of *PLR*. In order to obtain this mathematical model, the results of various tests at the same outside (28°C) and internal (27°C) air temperature are used: a full capacity test and different part load percentage tests. For the single stage unit the part load rating is obtained by the cycling of the compressor. In this case $PLF=COP_{cyc}/COP_{full}$ where COP_{cyc} is the coefficient of performance at part load working (cycling conditions) and COP_{full} is the full load *COP*. In the same way $PLR=Pc_{cyc}/Pc_{full}$ where Pc_{cyc} is the part load capacity and Pc_{full} is the full capacity at the same operative temperatures. A new parameter called *Z* is also introduced as the ratio of the electric consumption of the machine at part load working Pe_{cyc} to that one at full capacity Pe_{full}.

In figure 1 some experimental values of the parameters *PLF* and *Z* are reported as a function of *PLR* (Anglesio e al., 2001 second part). All the tests suggest a linear correlation between *Z* and *PLR* and this observation permits to elaborate a simple mathematical model for *PLF* as a function of *PLR* as follows:

$$COP_{cyc}=\frac{Pc_{cyc}}{Pe_{cyc}} \qquad COP_{full}=\frac{Pc_{full}}{Pe_{full}} \qquad Z=a\ PLR+b \tag{2}$$

therefore:

$$PLF=\frac{COP_{cyc}}{COP_{full}}=\frac{Pc_{cyc}}{Pe_{cyc}}\frac{Pe_{full}}{Pc_{full}}=\frac{PLR}{\dfrac{Pe_{cyc}}{Pe_{full}}}= \tag{3}$$

$$=\frac{PLR}{Z}=\frac{PLR}{a\,PLR+b}$$

The fundamental conclusion is that only one part load test besides one full load test, is sufficient to calculate the linear correlation between *Z* and *PLR* and consequently the algorithm of *PLF* as a function of *PLR*. These models are also reported in figure 1.

For the modulating machine, the capacity control is based on the changing of the rotational speed of the electric motor due to the variation of the frequency of the electric supply by an inverter between the nominal maximum frequency and a minimum value. Below this value a further reduction of the capacity can be obtained again by on-off cycles as for the single stage machine.

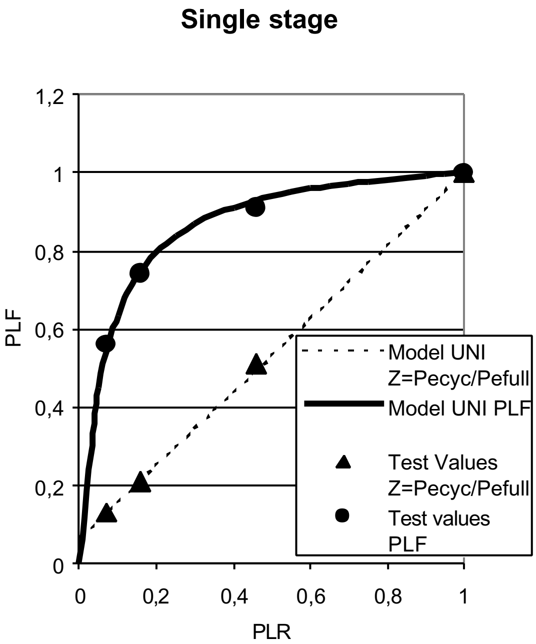

Figure 1. UNI models for the parameters *Z* and *PLF* as a function of *PLR* and some experimental values from laboratory tests for a single stage machine.

In figure 2 some experimental values of the couple *PLR* and *PLF* are reported (table III B in Anglesio et al., 2001 second part). For the algorithm of *PLF*, the standard suggests a simple linear correlation between the point at nominal frequency and the point at minimum frequency i.e. in the interval of the real modulation of the capacity. For lower *PLR* it is possible to elaborate the same model seen for the single stage machine. It is important to notice that the global curve of *PLF* as a function of *PLR* is now obtained by utilizing only three test points: one full load test at the maximum frequency, one full load test at the minimum frequency, one on-off test at the minimum frequency. Another fundamental remark is that the reduction of the motor revolutions results in a strong increase of the *PLF* above 1 i.e. a strong improvement of *COP* whose positive consequences continue also in the first interval of on-off working. In fact a lower refrigerant flow rate moved by the compressor, due to the motor speed reduction, causes inferior thermal fluxes exchanged at the evaporator and condenser and then smaller thermal differences between the refrigerant and the outside.

THE MODELS FOR WATER CHILLING PACKAGES

As mentioned above, for air cooled reversible chillers in absence of data from laboratory tests, we have used experimental data from long term monitoring campaigns of units installed in buildings. In comparison with laboratory tests, this approach is very different.

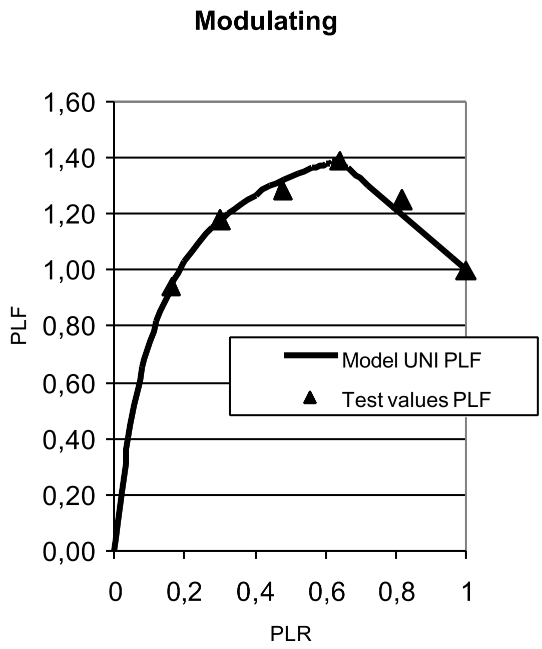

Figure 2. UNI model of the parameter *PLF* as a function of *PLR* and some experimental values from laboratory tests for the modulating machine.

First of all the measured points are not obtained in steady state conditions, but instead during the real working which is continuously transient. Therefore it is not possible to expect results perfectly reproducible. In this case the multiplicity of the test results can provide a correlation between the parameters which is more evident if referred to average values, on hourly or daily basis. Consequently this correlation cannot be used to evaluate effectively the instantaneous behaviour. On the other side it can be even more reliable than the algorithm elaborated from laboratory tests in order to estimate the long term mean performance. Indeed our fundamental scope is the prediction of the seasonal performances. A new difficulty instead can be the lack of full capacity tests necessary to calculate the denominator of the parameter *PLF*. The use of data available in the technical bulletins from the manufacturer often reveals to be unreliable for a specific machine. Furthermore also the measured full capacity working points are often so few (sometimes absent) to make very hard the elaboration of a specific algorithm which permits to calculate the full capacity *COP* as a function of the thermal levels of the external fluids, air and water, which the machine exchanges heat with.

As regards small size units we have utilized here the results obtained from a monitoring campaign of two reversible heat pumps, each of them installed to serve a small building in order to simulate the thermal requirements of a typical small residential unit (about 60 m^2) in Milan (Pettorossi et al.,2002). Both the plants were equipped with a monitoring system able to provide the continuous record of the working conditions of the machines in terms of thermal levels, electric consumptions and consequently the supplied energy and *COP* for different weather and operating conditions.

The two machines are single stage with a scroll compressor and R22 refrigerant. The control is on-off. The first machine (machine n°1) has a nominal capacity of 4.7 kW in cooling and 5.5 kW in heating . The other (machine n°2) is bigger with a nominal capacity of 7.1 kW in cooling and 8.2 kW in heating. Both the machines proved to be oversized for the building requirements. The final units are fan-coils directly supplied by a distribution circuit of minimum length. A water storage is absent. In other terms the thermal inertia of the hydraulic circuits is very modest. The monitoring as heat pump have been performed in the winter 1999-2000. The tests in air conditioning mode during the following summer.

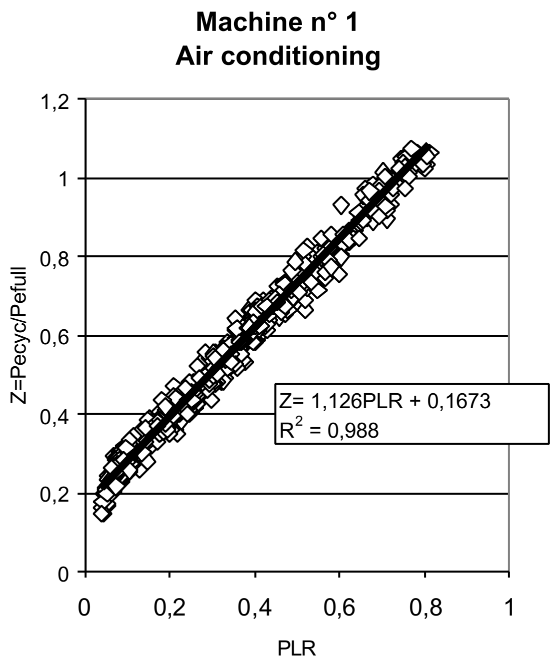

Figure 3. UNI model of the parameter *Z* as a function of *PLR* and hourly mean values from the monitoring of the machine n°1 in air conditioning mode.

Starting from an acquisition every three seconds and their record at one minute interval, the mean hourly values of the working parameters have been evaluated. These data have been elaborated as required by UNI 10963. For the machine n°1, the figure 3 shows the hourly mean values of parameter Z as a function of the corresponding *PLR* in the case of the air conditioning

The linear correlation just noted in laboratory for the single stage machine remains again absolutely evident. The validity of the linear regression is quantified by the high value of the correlation coefficient R^2:

$$R^2 = \frac{\sum(Z_{calculated} - \overline{Z})^2}{\sum(Z_{real} - \overline{Z})^2} \qquad (4)$$

where $\overline{Z}$ is the mean value of all the real measured Z (or *PLF*). Afterwards the algorithm to calculate the parameter *PLF* as a function of *PLR* has been elaborated by utilizing the same modelling just seen for the air to air single stage machines. The curve obtained in this way for the air conditioning mode is reported in figure 4 together with the mean hourly experimental values from the monitoring.

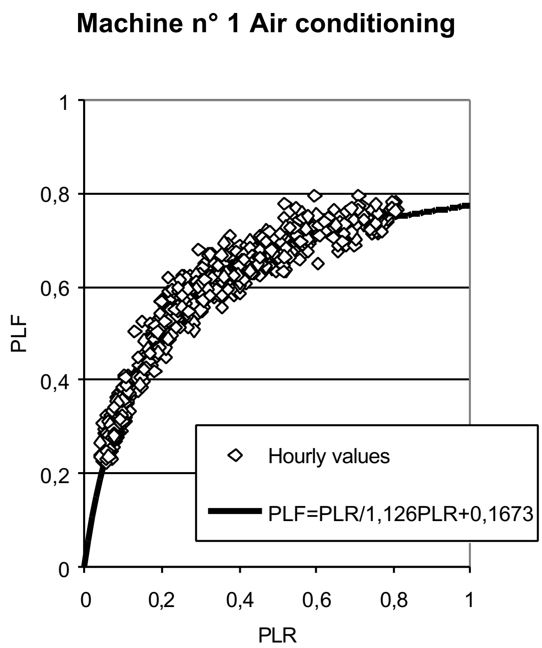

Figure 4. UNI model of the parameter *PLF* as a function of *PLR* and hourly mean values from the monitoring of the machine n°1 in air conditioning mode.

The unavoidable dispersion of the experimental points does not prevent from the observation of the substantial validity of the curves introduced.
In the heating mode, the machine n°1 provides results analogous which are reported in the figures 5 and 6.
But the numerical coefficients of the two correlations which give *Z* and *PLF* as a function of *PLR* are different of those ones found for the air conditioning mode. In other terms for the same value of the part load factor *PLR*, the correction coefficient *PLF* results different in heating or cooling mode. The results obtained for the machine n°2 are absolutely similar as the figures 7, 8 and 9 show. The only difference regards the experimental points characterized by a stronger capacity reduction due to a larger oversized with respects to the real building requirement.

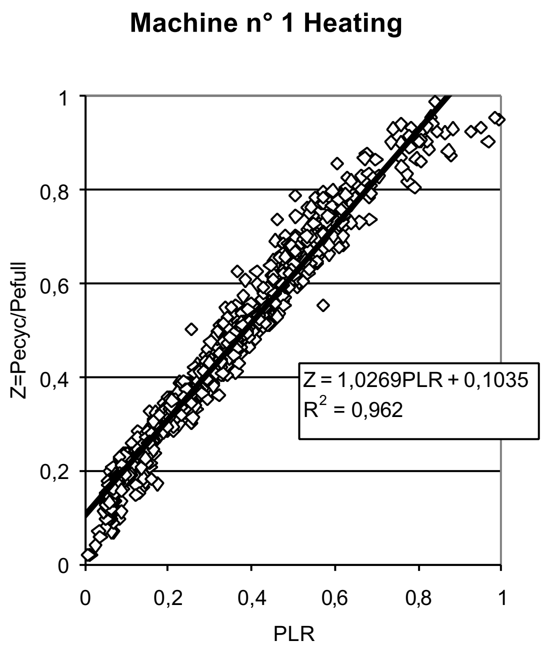

Figure 5 UNI model of the parameter *Z* as a function of *PLR* and hourly mean values from the monitoring of the machine n°1 in the heating mode.

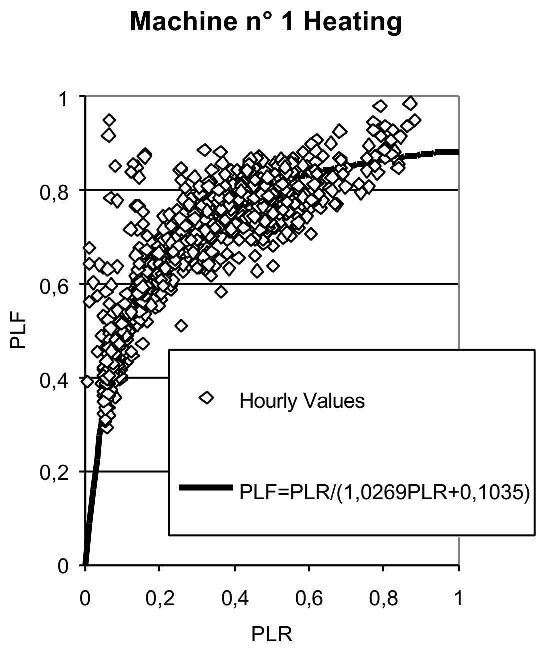

Figure 6 UNI model of the parameter *PLF* as a function of *PLR* and hourly mean values from the monitoring of the machine n°1 in the heating mode.

Besides it is important to underline the difference of the numerical coefficients in the formulas for the two machines which confirms the necessity to carry out individual tests for each model of machine.

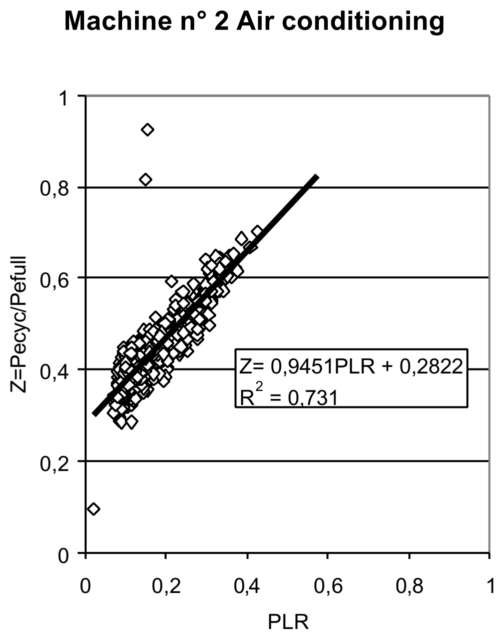

Figure 7 UNI model of the parameter Z as a function of *PLR* and hourly mean values from the monitoring of the machine n°2 in the air conditioning mode.

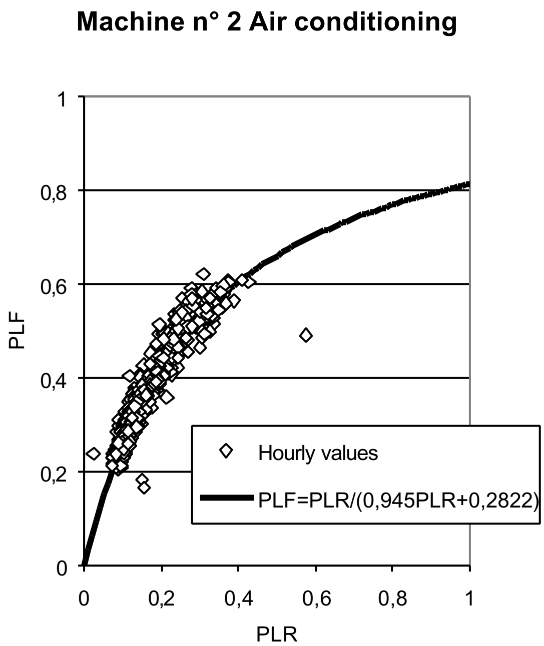

Figure 8 UNI model of the parameter *PLF* as a function of *PLR* and hourly mean values from the monitoring of the machine n°2 in the air conditioning mode.

The results from the monitoring of two large size chillers, nominal capacity of 510 kW, are now reported. They are installed in a HVAC plant which has other two chillers of the same capacity but with the possibility of total heat recovery from the condensers.
The plant is in an office building with a large computer room which requires a strong cooling also in the winter. Water storages are present both to increase the thermal inertia of the hydraulic circuit and as cold storage useful in the case of temporary breakdown of the refrigeration machines or electric failure. The plant can be therefore considered characterized by high thermal inertia. For sake of brevity the reference is (Schibuola 1999) for a detailed description of the plant and of the monitoring campaign carried out there. For the scope of this paper, the working data of the two chillers operating essentially in summer have been reported. They are typical water chilling packages with four reciprocating compressors, refrigerant R22 and air cooled. Each unit presents four control step of the capacity. At the moment of the monitoring the machines have been working 8 years long.

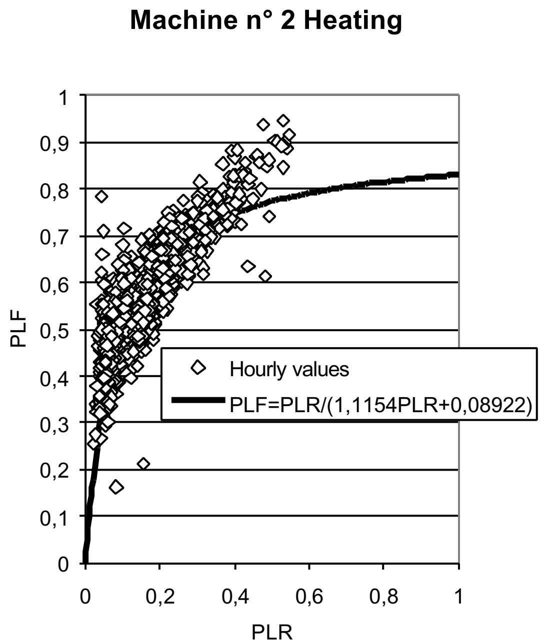

Figure 9 UNI model of the parameter *PLF* as a function of *PLR* and hourly mean values from the monitoring of the machine n°2 in the heating mode.

The figures 10 and 11 show the UNI models of the parameters Z and *PLF* respectively as a function of *PLR* elaborated starting from the hourly mean performances from summer monitoring for one of the two machines (machine n°3). For the other machine (machine n°4), the model of the parameter *PLF* is directly reported in the figure 12.
It can be notice that for all the tested machines a clear experimental correlation between the mean hourly values of the part load ratio *PLR* and the part load factor *PLF* has been always confirmed. It means that the only parameter *PLR* is really able to determine the influence of the part load working on the hourly mean behaviour of the machine. With the same hourly mean *PLR*, different distributions of the cooling demand certainly present in the monitoring periods do not cause differences in the hourly mean

PLF. This result is of the maximum importance because it validates the use of the models introduced for dynamic simulation of the building-plant system with one hour time step.

Machine n° 3 Air cooling

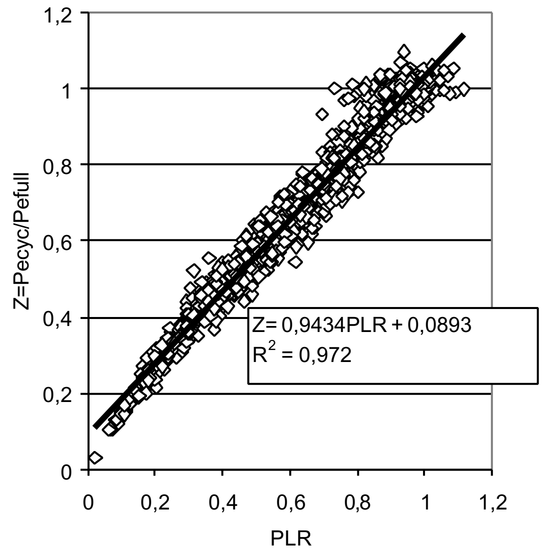

Figure 10 UNI model of the parameter Z as a function of *PLR* and hourly mean values from the monitoring of the machine n°3 in the air conditioning mode.

Machine n° 3 Air conditioning

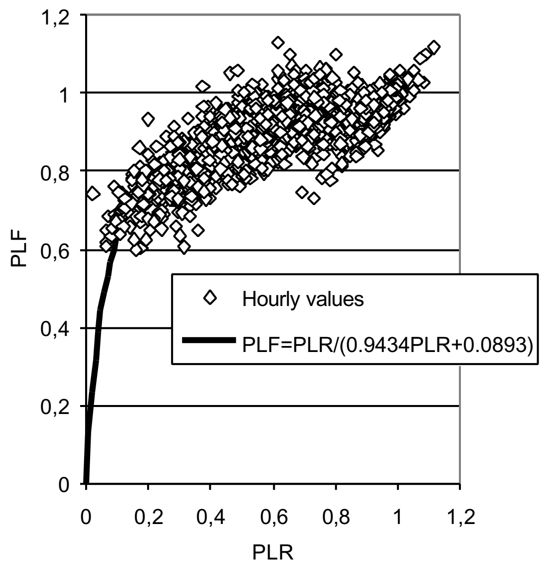

Figure 11 UNI model of the parameter *PLF* as a function of *PLR* and hourly mean values from the monitoring of the machine n°3 in the air conditioning mode.

In order to have a further experimental validation, the hourly mean thermal levels and building requirement from monitoring have been considered. As they were obtained by a dynamic simulation, they are used to calculate *PLR* parameter on the basis of the reference working curves of the machine at full capacity. Consequently the hourly mean *COP* has been calculated and corrected by the use of the model proposed for *PLF*. Dividing the mean hourly capacity by this *COP* the mean hourly electric consumption is calculated. The seasonal mean COP is obtained as the ratio between the values integrated in the season period of the satisfied requirement and the electric consumption. Table I reports the comparison between the seasonal *COP* so calculated and the real measured *COP* from monitoring of the machines here considered. The percentage error is always less than 1%.

Machine n° 4 Air conditioning

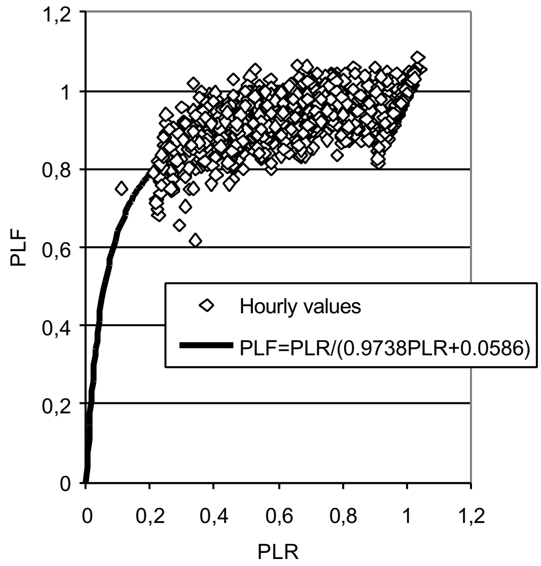

Figure 12 UNI model of the parameter *PLF* as a function of *PLR* and hourly mean values from the monitoring of the machine n°4 in the air conditioning mode.

Table I Comparison between calculated and measured seasonal COP.

	COP Measured	COP calculated	Error %
Machine n°1 Air conditioning	2.469	2.463	-0.24
Machine n°1 Heating	3.041	3.073	1.05
Machine n°2 Air conditioning	1.516	1.514	-0.13
Machine n° 2 Heating	2.053	2.051	-0.10
Machine n° 3 Air conditioning	2.580	2.581	0.4
Machine n° 4 Air conditioning	2.618	2.617	-0.04

Furthermore the models from summer monitoring has been used for the winter and vice versa. For the machine n°1, in the first case the previous comparison shows an error of 18%, in the second case 23%. This bad result indicates that it is absolutely necessary distinguish the heating and cooling models of the parameter *PLF* for the same machine, probably because of the different thermal levels in cooling and heating modes.

APPLICATION OF THE MODELS

A long term dynamic simulation of building-plant system is able to provide a correct estimation of the seasonal performances of a refrigeration machine installed in a HVAC plant. For this aim, between the various commercial computer programs today widely diffuse, DOE program, release 2.1E, has been here used (DOE, 1994).

Typical applications cases have been analysed (Bettanini et al., 2001). So we have considered a flat of five rooms situated on the intermediate floor of a block, 108 m^2, height 3 m and with three external walls. Typical envelope structures have been assumed, for example brick walls well insulated. About 15% of the external surface is double panel glazing. Internal heat gains have been neglected here and an average natural infiltration rate has been estimated equal to 0.5 vol h^{-1}. Intermittent plant working from 7 am to 9 pm (14 h) is investigated. An office case has also been considered. The only envelope difference is a more extended glazing equal to 50% of the external walls. The plant working is from 7 am to 7 pm (12 h). Forced ventilation equal to 2 vol h^{-1} is present only in this period. Internal heat gains for persons (maximum 1 person 10 m^{-2}), lighting (maximum 20W/m^2, fluorescent) and computers (maximum 12 W/m^2), with typical daily profiles are foreseen for the office case. The air conditioning is present from May to the end of September. Various climatic conditions well distributed in Europe have been considered: Copenhagen, Nancy, Milan, Rome, Crotone and Trapani. For these localities Test Reference Years (TRY) provided by European Commission (European Commission Directorate, 1985) have been used.

As application of the models just introduced, the results of dynamic simulation are here presented for an air to air conditioners with on-off or modulating capacity control and for a chiller referring to the models of the machine n°1.

In figure 13, for the air to air machine, the seasonal *COP* obtained by simulation for the two buildings and for the various cities is reported assuming for the part load influence: no influence, influence valuated with ARI formula (1) or with UNI model. The modulating control effect has also been investigated with corresponding UNI model just seen. First of all one can observe the fundamental influence of the part load working on the seasonal *COP* which cannot be neglected and instead it have to be estimated with a more precise correlation than the simplified one proposed by ARI. Furthermore remarkable is the increase of the performances with the models of the modulating conditioner which permits to quantify the advantages of an efficacious capacity control.

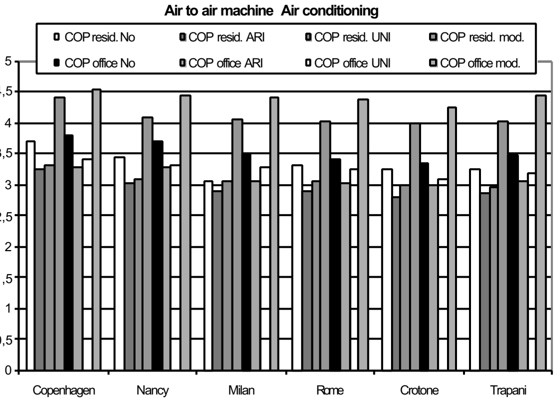

Figure 13 Comparison between the seasonal *COP* obtained by simulation for the air conditioner and utilizing different models: no influence, ARI or UNI model for the on-off control and UNI model for the modulating control, in the various cases studied.

In the same way the figure 14, for the machine n°1 (single stage and on-off control), shows the seasonal *COP* obtained by considering the part load influence in accordance with the same three models in the various application cases.

Also for chillers the necessity of a correct choice of the calculation model is confirmed because of the large difference of the results obtained in the three modes.

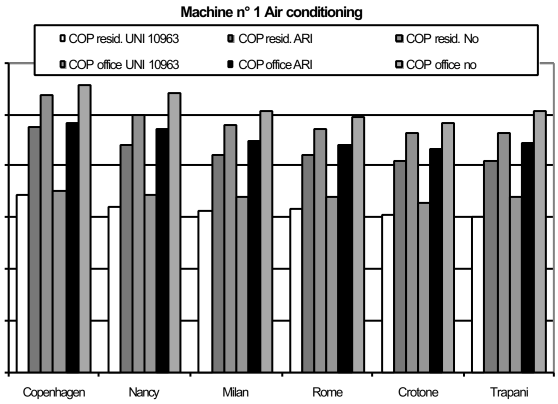

Figure 14 Comparison between the seasonal *COP* obtained by simulation and utilizing different models: no influence, ARI or UNI model for the machine n°1, in the various cases studied.

CONCLUSIONS

The models proposed for an extended use by UNI 10963, but really developed on the basis of laboratory tests of only air to air split machines, are resulted valid also to explain the working data of chillers obtained from the monitoring of real plants.

However this fact does not mean that it is possible to give up to the laboratory tests in steady state conditions and to use only data from monitoring. In both the case the question to answer is if and how the model built for a machine in a particular application condition can be used again for the same machine but in different application context. It is evident that this problem is fundamentally for the chillers which interact with the distribution circuit and its thermal inertia due especially to its water contents.

Another conclusion regards the real efficacy of the models introduced by UNI 10963 in a calculation based on hourly means of *PLR* and *PLF* and therefore their usefulness in the calculation procedures, simplified or comprehensive, today available for the evaluation of the seasonal performances of the building-plant system.

The application examples reported have confirmed the necessity of the evaluation of the part load working influence in order to carry out a correct energy analysis of HVAC plants.

AKNOWLEDGEMENTS

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