SEASONAL PERFORMANCE EVALUATION OF CHILLERS CHARACTERIZED BY NEW DESIGN SOLUTIONS IN AIR CONDITIONING PLANTS

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Abstract. *An experimental test has been performed to evaluate the operating conditions of water chilling packages characterized by new design solutions. In particular the innovations are about the lay-out of the refrigeration circuits and the control system. The Italian standard UNI 10963, recently introduced, has been used for the testing procedures and to elaborate the results. The fundamental scope is the investigation of the behaviour of the chillers in the real operating conditions which are normally characterized by a part load working influencing the efficiency of the machines. The working curves obtained in this way have permitted a comparison between the refrigeration machines in terms of steady state performances and also seasonal performances in the long term real working in a building air conditioning plant.*

Keywords: *Air conditioning, Part load, Chiller, Efficiency, Energy.*

1. INTRODUCTION

The performances of the refrigeration machines used in the air conditioning plants are the consequence not only of the operating thermal levels, but also of the requirement trends of the building which usually involve frequent reduction of the capacity. In fact these machines are normally sized for the peak load and therefore they normally work in part load conditions. Nowadays the consciousness of the importance of the part load working behaviour on the seasonal efficiency is increased. Consequently it is the effort to introduce new design solutions to improve the performances in part load conditions. To test the results obtained in these way the Italian standard UNI 10963, recently introduced, has been used to obtain in laboratory the part load working curves. 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. Starting from these data a specific method, recently proposed for a new standard (CTI 5/540), permits to calculate the seasonal efficiency for each application case. Therefore they have been implemented in a computer code able to perform long term simulations of the building-plant system. In these paper the procedure and the results of this analysis are presented and compared.

2. THE CALCULATION METHOD

In the most famous and diffuse computer programs for the dynamic simulation of the 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 standards suggest a generalized use of the following equation to calculate *PLF* (ARI, 1989):

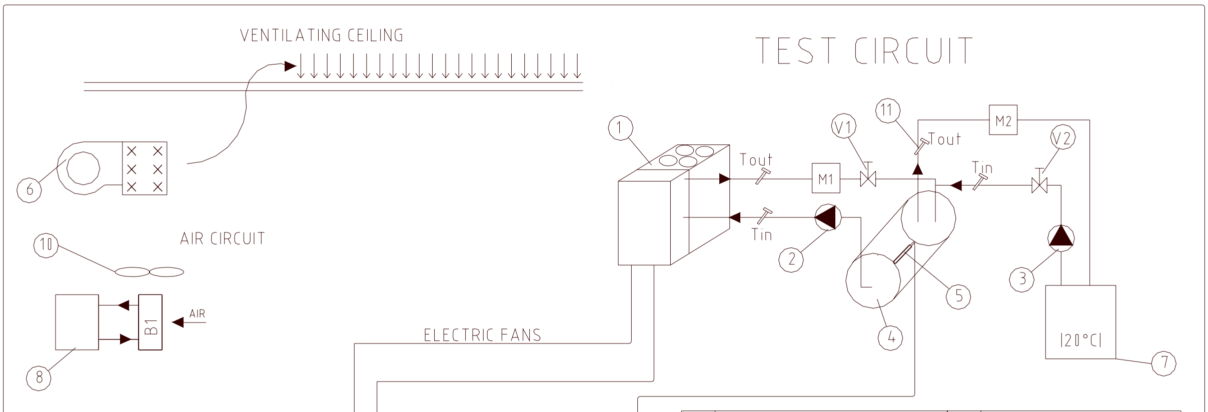
$$PLF = 1 - c_d \cdot (1 - PLR) \tag{1}$$

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.

3. DESCRIPTION OF THE MACHINES AND TESTS

The laboratory tests are regarded two reversible air to water heat pumps with two different refrigeration circuits. The second machine has been tested in presence of two different control systems. The first machine, here called machine 1, has a nominal capacity of 33.1 kW in cooling and 37.2 kW in heating mode. The machine is equipped with two equal scroll compressors in two independent refrigeration circuits working with R407C. The capacity control presents two step (50%,100%) based on the return water temperature at the inlet in order to ensure an outlet water temperature near a fixed value (7°C in cooling and 45°C in heating mode). The other reversible heat pump has a nominal capacity of 34 kW in cooling and 40.3 kW in heating mode. Its fundamental difference is the presence of two different size compressors installed in only one refrigeration circuit . The refrigeration fluid is again R407C. Referred to the total capacity the capacity ratios of the two compressors are 1/3 and 2/3 respectively. So we have now three capacity steps (33%,66%,100%). But another important advantage is the possibility to use always the total thermal exchange surface in the condenser and evaporator even if one compressor is stopped. In other terms during part load working the exchangers work oversized with consequent benefit due to the reduction of the thermal gap with the external fluids. Another innovation regards the control system. The outlet water temperature can be variable as a linear function of the thermal load of the building in alternative to the traditional fixed value. In detail between 15 and 7°C in cooling, between 35 and 45°C in heating mode when the building load varies between 25% and 100%. Both the control solutions have been tested. The results with the traditional control are here reported referred to machine 2 while the results in presence of the variable temperature control are here referred to machine 3. A test circuit, expressly built to stabilize the temperatures of the external fluids with which the machine exchanges heat, is used to simulate different load ratios simply by operating on the energy quota taken from the hydraulic circuit. The scheme of the test circuit is reported in figure 1. More details about the test apparatus are available in Gastaldello et al. (2004).



1	Tested machine	11	n. 4 thermometers
2	Pump	M1	Flow rate meter
3	Pump	M2	Flow rate meter
4	Water storage 420 l	V1	Flow rate regulator
5	n. 10 resistances	V2	Flow rate regulator
6	Resistances set 12 kW	C	Electric measures
7	Tank with water at 20°C	V	Electric measures
8	Chiller/heat pump	R	Electric measures
9	Electric board/measures	B1	Cooling/heating coil
10	Fan		

Figure 1- A sketch of the test circuit in laboratory.

For the machine 1 five tests were carried on both in cooling and heating mode. As foreseen by UNI 10963, first of all two full load measures: one in the nominal condition and the other in the so called auxiliary condition. In cooling mode it means inlet air temperature at 35°C or 28°C respectively, water temperature gap 12-7°C in both the cases. In heating mode it means air at 7°C or 12°C and water temperature gap always 40-45°C. As regards part load, both in heating and cooling mode, three tests near three different load percentages: 25%, 50%, 75% with an air temperature fixed to the auxiliary condition.

For the machine 2, the same tests were realized. For the machine 3, the full load tests are again the same of the other two machines but those ones at part load are characterized by an outlet water temperature depending from the part load ratio. In cooling this temperature is 15°C, 12.5°C, 10°C, in heating 35°C, 39°C, 43°C at 25%, 50%, 75% respectively.

In figure 2 the *COP* obtained by the tests are reported. A different behaviour for the three machines is clear. A load reduction for machine 1 involves a strong worsening of the performance. In spite of the machine 1 has a better *COP* than the others at full load; they present an higher *COP* at part load. The improvement is particular evident for machine 3 thanks to the variability of the outlet water temperature.

4. THE MODELS

As foreseen by UNI 10963 two points at full load, nominal and auxiliary conditions, are the minimum enough to obtain by linear interpolation the full load behaviour at different air temperature. Regarding part load working the standard proposes algorithms to calculate *PLF* as a function of *PLR* different from ARI. In order to obtain this mathematical model, the

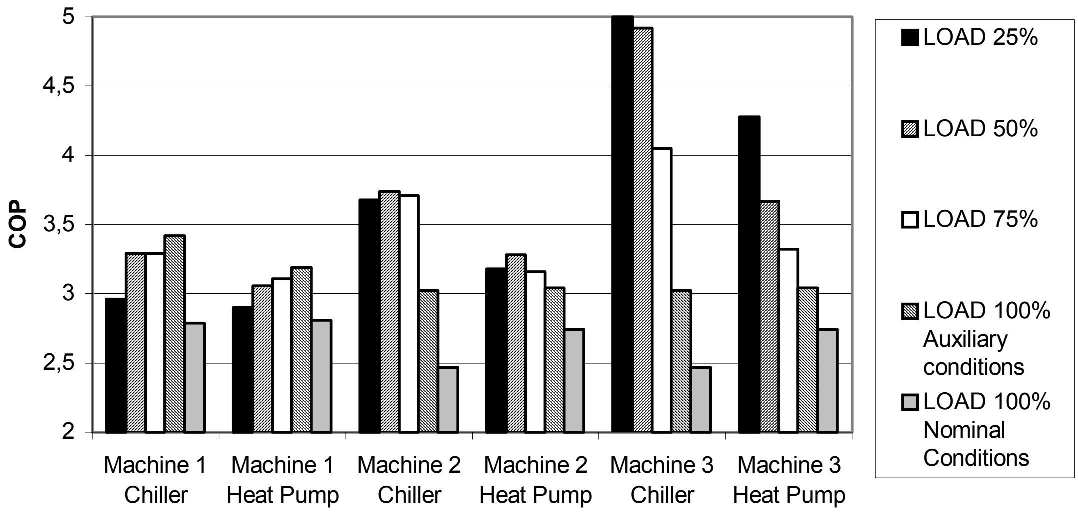


Figure 2 – Measured *COP* in the laboratory tests.

results of tests in the auxiliary condition are used: a full capacity test and different part load percentage tests. The part load rating is obtained by the cycling of the compressors. 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}.

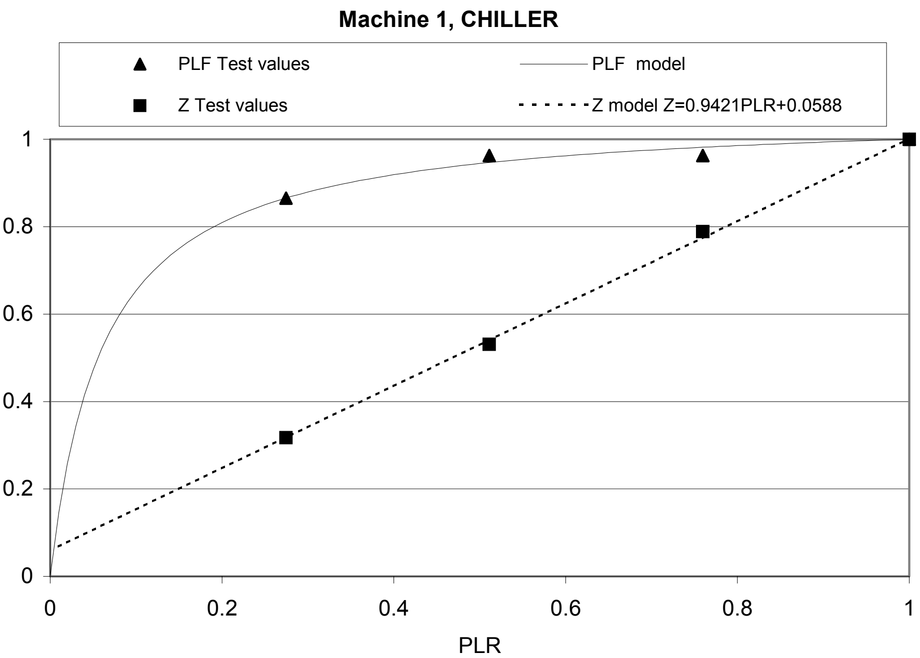


Figure 3 – Models of the parameters *PLF* and *Z* as a function of *PLR* and test values for machine 1 in cooling mode.

In figure 3 the experimental values, obtained for machine 1 in cooling mode, of the parameters *PLF* and *Z* are reported as a function of *PLR*. As recommended by UNI 10963, 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:

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

therefore:

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

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

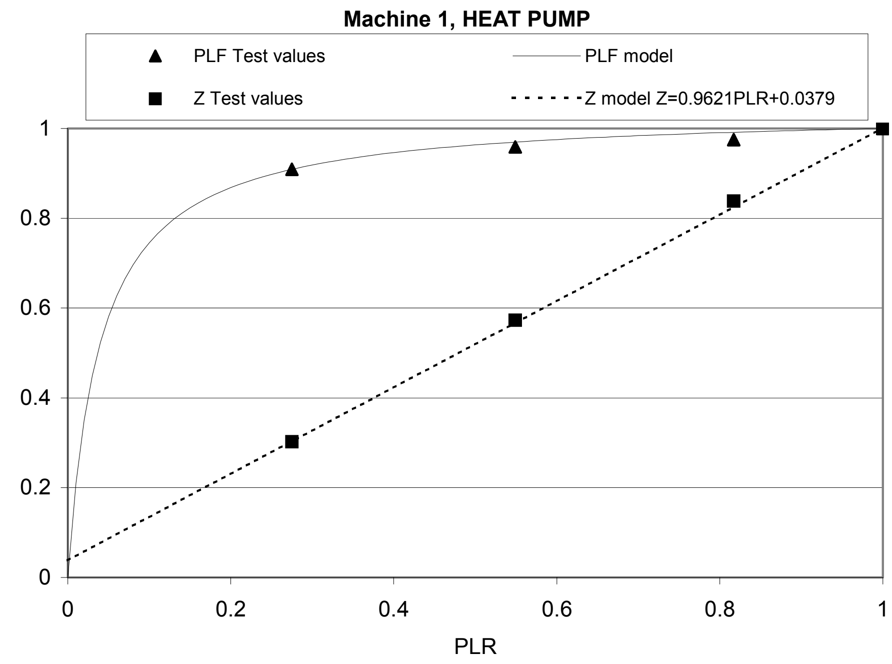


Figure 4 - Models of the parameters *PLF* and *Z* as a function of *PLR* and test values for machine 1 in heating mode.

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 2. Figure 3 shows the results in heating mode. It is important to notice that the coefficients of the models are different from those ones of figure 2. This means that it is necessary to perform independent tests for heating and cooling mode.

In figures 4, 5, 6, 7 the experimental results obtained for the machines 2 and 3 are reported in the same way for heating and cooling mode. The linear correlation between *Z* and *PLR* is confirmed again but now the term b of the correlation is a negative value. As consequence *Z* is equal 0 for *PLR* greater than 0. This solution is without sense and it means that the extrapolation of this linear correlation is not acceptable. The previous UNI model of *PLF* as the ratio of *PLR* to Z is not applicable now because diverging towards infinity at low *PLR* values. For this reason the decision is to use directly the experimental *PLF* values by a model simply based on the linear interpolation between them. The two point's interpolation model is practically coincident with the four points one. There are no tests near zero but under a *PLF* of 0.25 we have in any case an on-off capacity control based on the cycling of only one compressor with a fixed outlet water temperature. Therefore it is justified the use of

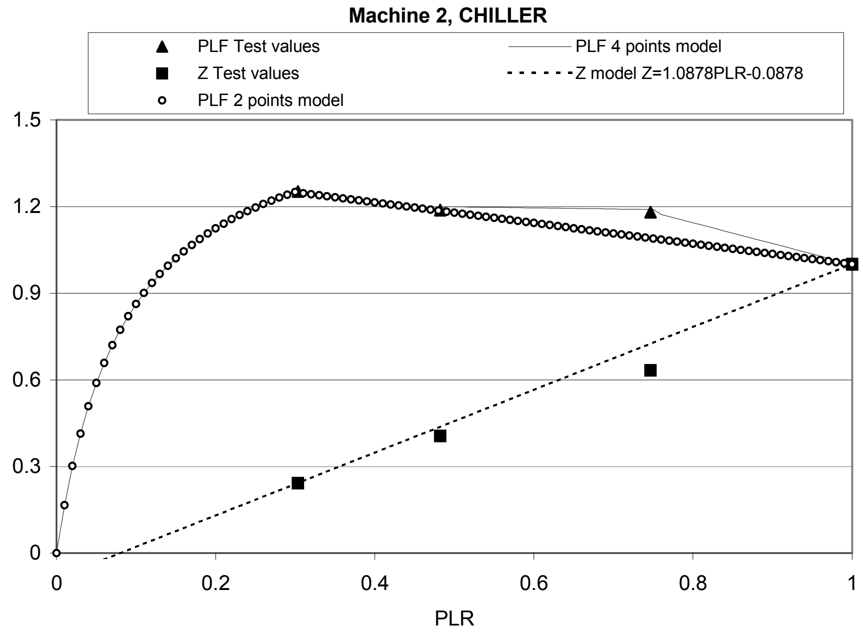


Figure 5 - Models of the parameters *PLF* and *Z* as a function of *PLR* and test values for machine 2 in cooling mode.

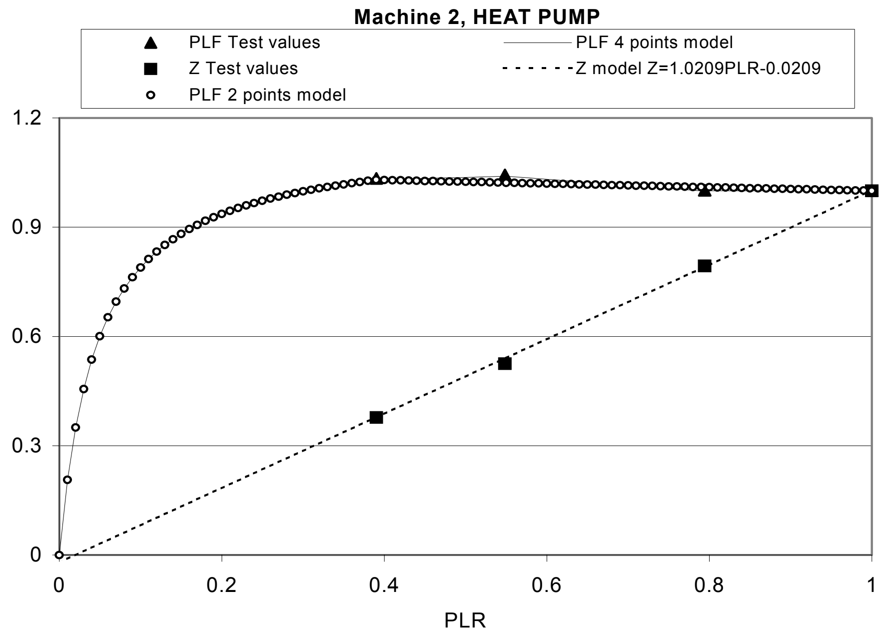


Figure 6 - Models of the parameters *PLF* and *Z* as a function of *PLR* and test values for machine 2 in heating mode.

a UNI model built for the interval between zero and the point at *PLR* near 0.25. Also for the machines 2 and 3 we can conclude that only two tests at the auxiliary condition of temperature, one at full load and another at 25% of the load are enough for an acceptable description of the part load behaviour. The different experimental values confirm again the necessity of independent tests for heating and cooling mode.

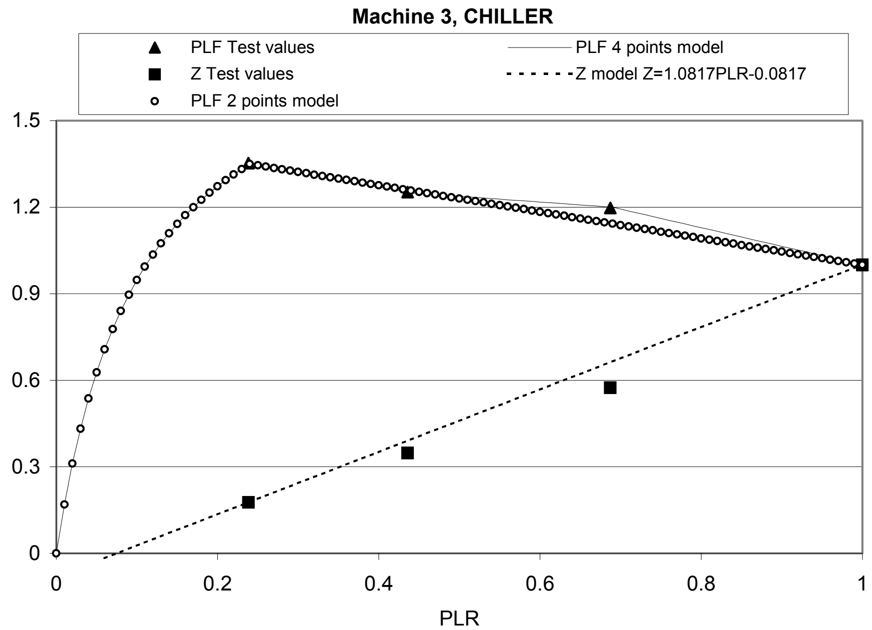


Figure 7 - Models of the parameters *PLF* and *Z* as a function of *PLR* and test values for machine 3 in cooling mode.

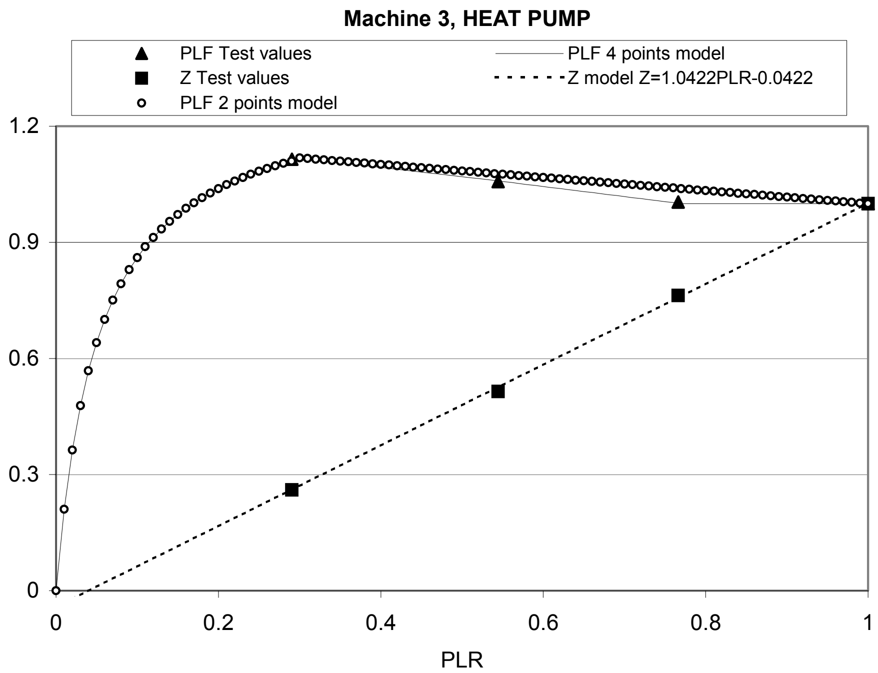


Figure 8 - Models of the parameters *PLF* and *Z* as a function of *PLR* and test values for machine 3 in heating mode.

5. THE SEASONAL RESULTS

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, 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 20 W/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 has been used (European Commission, 1985).

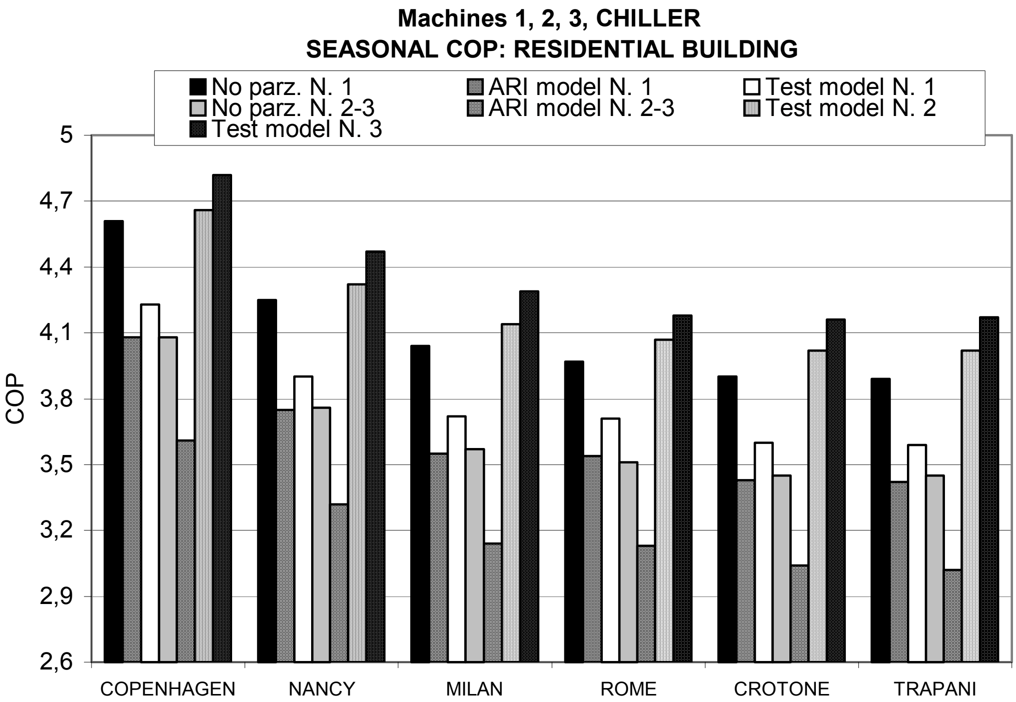


Figure 9 – Seasonal *COP* for the three machine calculated with three different part load model: no penalization, ARI model or test model in air conditioning of the residential building.

Figure 9 shows the seasonal *COP* obtained neglecting the part load effect (no part load), by using ARI model or the models from tests for the three machines in the residential application case and cooling mode. The differences with the various models are remarkable. The seasonal *COP* error introduced neglecting the part load influence is significant. The ARI model is not able to evaluate the different behaviour between the machines 2 and 3.

The necessity to use specific models deduced from experimental investigation is confirmed. In figure 10 the seasonal *COP* of the three machines are compared for the air conditioning period in the residential and office case. The experimental curves were used. It is rather evident the improvement connected with the circuit solution introduced for the machine 2 and further enlarged by the flowing temperature control of the machine 3. It is anyway important to underline that this kind of temperature control is to be used with care. In fact a temperature increase can cause a reduction till the impossibility of the dehumidification capacity. In the event of a precise humidity control the ventilation air handling will require a cooling capacity produced by another refrigeration machine at lower temperature or the limitation of temperature increment to satisfy the hygrometric control. For the machine 3 the manufacturer foresees a global supervisory system able to regulate the inlet flow rate to the plant terminals as a function of the water temperature to balance the event of a different variability of the loads in the various rooms of the building.

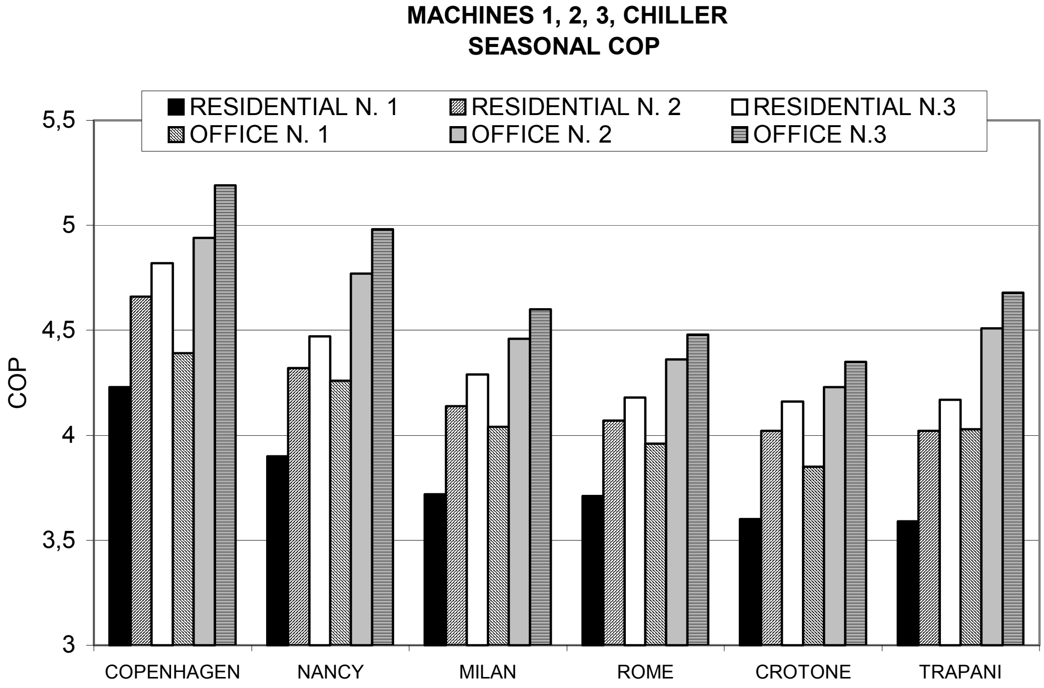


Figure 10 – Seasonal *COP* in cooling mode for the three machines calculated with the test models for the office and residential buildings.

The experimental working curves for the heat pump mode are used to calculate the seasonal *COP* of figure 11. Different commutation temperatures were considered for the heat pumps: -2, 0, 2, 4°C. In fact the electric heat pumps which use as cold source the outside air normally operate only over an external temperature (commutation temperature) which ensures a *COP* economically advantageous with respect to the boiler otherwise working in alternative. This commutation temperature is not generalizable because it depends from the comparison between the electric tariffs and the cost of the fuel used for the boiler. For sake of brevity the only results of a commutation temperature (Tset) equal to 2°C are reported here. For the complete results the reference is (Gastaldello, 2003)

In the figure 11 it is possible to notice that when the heating load is higher, i.e. in cold climates and residential buildings, the seasonal *COP* are similar with a light better performance for machine 1 which, as seen in figure 2, presents better full load *COP*. For the office building instead, owing to the internal heat gains, the requirements decrease and the part load working favours the seasonal *COP* of the machine 2 and 3. The same effect is caused by mild climates.

6. CONCLUSIONS

Starting from tests expressly performed as requested by the Italian standard UNI 10963 the models to describe the behaviour of the machines have been obtained and used to calculate the seasonal performances. The results confirm the validity of the innovations introduced and point out the necessity of taking into account the effect of part load working in order to carry out a correct energy analysis of HVAC plants.

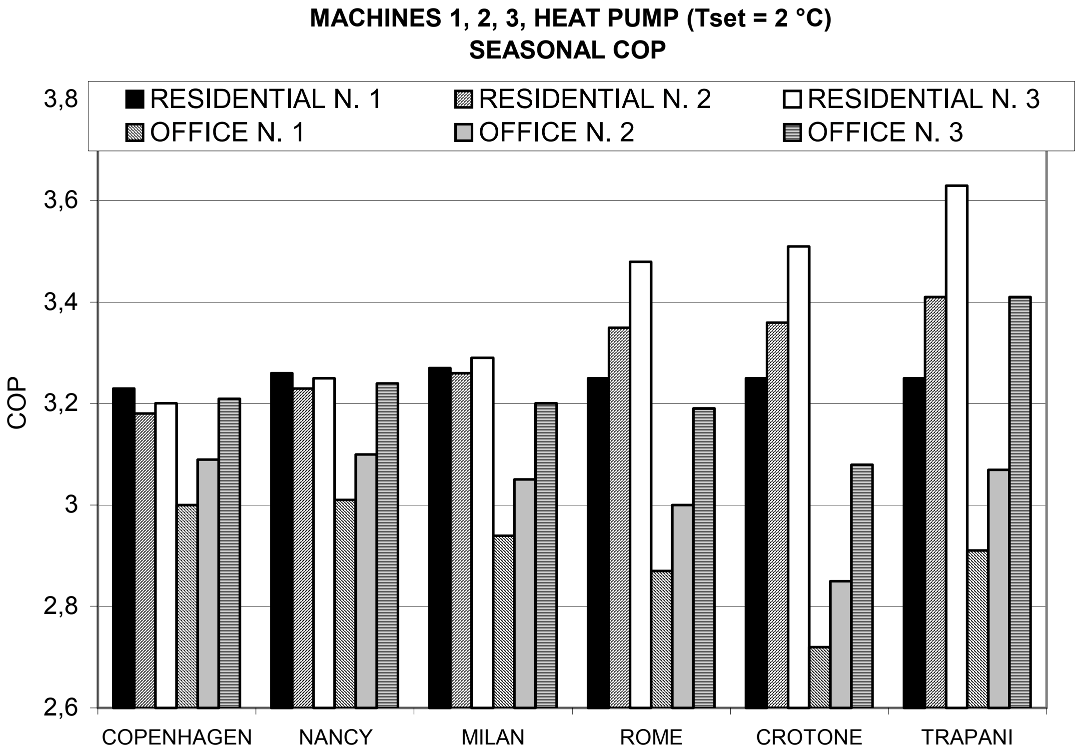


Figure 11 – Seasonal *COP* in cooling mode for the three machines calculated with the test models for the office and residential buildings.

The question still open 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 a 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.

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