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Optimisation of VRF systems in buildings by monitoring

Stefan Vandaele, Hiroshi Aihara

Environmental Application Research Center

vandaele.s@daikineurope.com

aihara.h@bxl.daikineurope.com

Daikin Europe N.V.

Bergensesteenweg 1424, 1070 Brussel

Abstract

By monitoring VRF systems in buildings, we found that the energy efficiency and comfort reduces when oversizing the selected equipment in relation to the building load. It remains a challenge to assess the real building load accurately, because of the uncertainty of many parameters. In the past, after designing and selection, the equipment was installed and that was also the end of the process. There was no verification of the real performance in order to suggest possible improvements . Now, with new technologies, we can verify the result of the process of design – installation – commissioning and feedback to the design of the (refurbishment) equipment. In our research, we defined the real building load by monitoring the VRF system, after which the analysis of the data enables optimising the selection. of a new VRF system. By optimising the selected capacity, we increase energy performance and comfort of the VRF system in the building to its potential.

Methods :

For determination of cooling and heating capacity of the VRF system, we used the compressor curve method, calculating the performance by using the sensors and operational data of the unit.

Results:

Oversized VRF installations remain a challenge also in new buildings, and thus operate at lower than optimal conditions. By monitoring and analysing performance data, we can define the scale of oversizing and select an optimal system for refurbishment.

Keywords - refurbishment, VRF, optimisation, monitoring

1. Introduction

Towards 2020 there is a shift of judging performance of the HVAC equipment in test rooms according to standard conditions, towards judging performance of the buildings including the HVAC. It needs no explanation that performance in test rooms is based on specific conditions which do not always occur in buildings. However, we found that dimensioning of the HVAC compared to the load of the building, has a far bigger influence on the real life performance.

Currently, many designers of HVAC systems believe that oversizing would have a beneficial effect on the performance, since DC inverter technology, used to control the speed of the compressors, works more efficient at lower part load conditions. However, they are not sufficiently aware that below certain part load ratios, the efficiency is dropping.

As a result, for EPB, a gap is possible between the assessed energy efficiency ratio of equipment in buildings and the measured energy efficiency ratio after commissioning of the building. In this project we have tried to find the method to close this gap by bringing the real performance as close as possible to the calculated performance. We chose (Daikin)VRF (=VRV) equipment for this project, but this principle is basically applicable for all HVAC.

We want to emphasize that the VRF energy efficiency ratio as a product is not questioned here, but the conditions under which the equipment is running can be less favorable for the equipment and lead to lower efficiency in actual operation than what is expected based on calculated conditions. So, further in this paper, if we talk about improving efficiency of the VRF, it means that we change the conditions to conditions in which the product has a better efficiency.

Furthermore, energy consumption determines the performance of a building, not the energy efficiency of the HVAC system. A HVAC system can work at a very high efficiency, but the building can still have a bad energy performance or vice versa. Eg. High load operation of the HVAC due to a less performing building shell will show high efficiency of the equipment. An owner of a building with oversized HVAC, may find energy performance of his building high, because his energy consumption is lower than what he expected during the design phase, however, his HVAC may be operating at a bad efficiency due to the low part load operation. Optimising the efficiency of the HVAC by optimising selection to the load will decrease the energy consumption of the building, making the energy performance of the building better.

2. History

In 2011, Daikin was closely involved in the HVAC set-up of an nZEB building. The target of the project was to achieve cost optimal nZEB. Even though nZEB was reached, a big potential for energy saving was found by optimizing capacity and usage. At the same time, the comfort could be improved by monitoring and fine-tuning the equipment.

It lead to the start of the VRV monitoring project. The objective was to

1) find out how well VRV's in buildings are dimensioned to the cooling and/or heating load and what is the impact on the energy consumption.
2) Find methods to select optimal capacity VRV to the building, leading to the best possible performance

3. Monitoring setup

In general, the challenge in monitoring VRF systems can be described as follows. To start, the source and sink of the refrigerant system are both air, which makes it complicated and expensive to measure correctly the amount of energy delivered to the building in the airstream(Air enthalpy method). Additionally, on the indoor side, there are many different heat exchangers(indoor units), each operating at different conditions, according to the users preferences.

Therefore, rather than installing external monitoring equipment, we chose to use the unit's internal sensors and operational parameters instead, as can be seen in below figure.

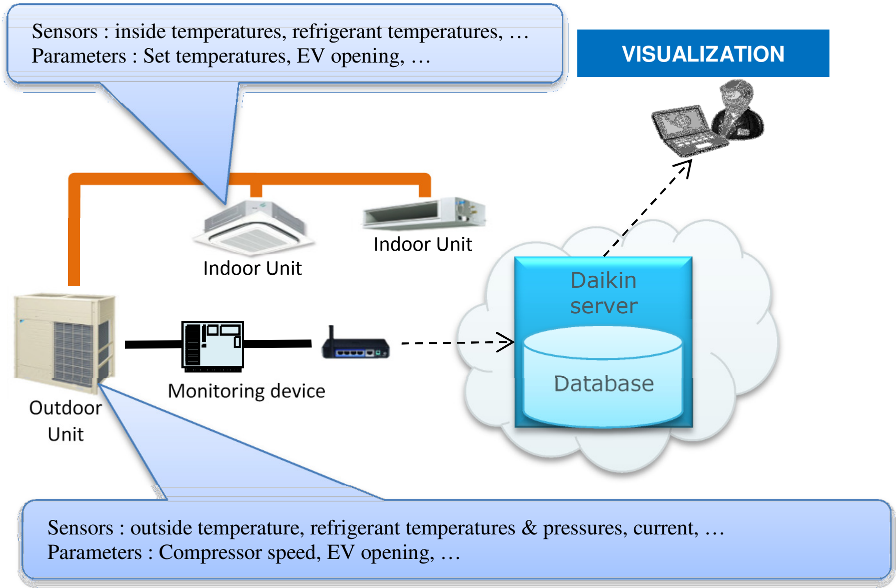

Figure 1 : Setup for monitoring

Power consumption can be deducted from the current sensor. However the units are not equipped with refrigerant flow meters. A different solution is to be found, see next chapter "Compressor Curve Method".

4. Compressor Curve Method

This method uses compressor characteristics to calculate refrigerant mass flow rate. Unlike methods such as "Air enthalpy method" that needs large and expensive equipment like air ducts, this method can calculate performance from refrigerant pipe side and thus, few sensors are needed and performance is easily calculated. To determine VRV efficiency in real buildings, it is the best method.

Research was done to verify the accuracy of the CC method.

Refer to :

1 - " **Performance Evaluation of VRF Systems using Compressor Curve Method "** (*Mugito KIKUCHI, Keisuke OHNO, Kiyoshi SAITO) 2011*

2 - "**The Simple Performance Evaluation Method of VRF System Using Volumetric Efficiency of Compressor** " (Naruhiro Sekine, Yuma Furuhashi, Shigeki Kametani) 2012

Conclusions:

CC method is a good alternative for AE method when only limited amount of sensors is present. A relatively high accuracy can be reached, depending on the amount of simplification….

Principle:

$$Q(h/c) \quad = \quad G \quad \times \quad \Delta i(h/c)$$

With

Qh = Heating Capacity

Qc = Cooling Capacity

G = mass flow rate of refrigerant

Δih = difference of enthalpy of heat exchanger (condenser)

Δic = difference of enthalpy of heat exchanger (evaporator)

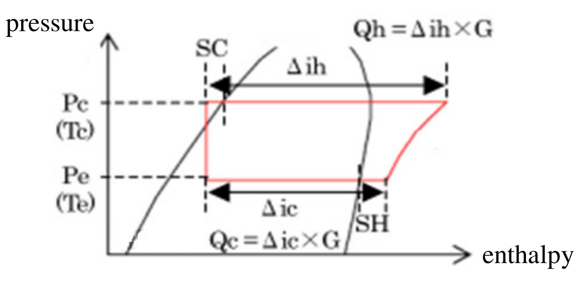

Figure 2 : Mollier diagram refrigerant

As can be seen from the figure, enthalpy can be deducted from measuring the pressures and temperatures in the refrigerant.

There is no refrigerant mass flow sensor so it is determined by a “virtual sensor” based on a compressor flow map.

A compressor map is used to estimate refrigerant mass flow rate using input measurements of inlet and outlet pressure.

Refrigerant mass flow rate for a compressor can be represented using a polynomial equation for a specified amount of superheat.

G = func(Te, Tc, compressor speed)

The map at the specified superheat is corrected using the ratio of the compressor suction density at the actual superheat to the density at the fixed superheat.

As it is not the topic of this paper we will not elaborate in detail how the equation is made up.

More information can be found in following paper

“Evaluation of virtual refrigerant mass flow meters” (Woohyun Kim, James E. Braun)2012

5. Monitoring project – phase 1

In this phase, no special monitoring sites were selected, but we used monitoring data from existing sites all over Europe. Daikin has an internet service for customers, named iNET. It connects the equipment to internet and hourly operational data from the units is stored on a dataserver. This service is used for preventive maintenance. However, it also can serve for determination of the unit's capacity and performance. The "Compressor Curve Method" is used for this purpose (Refer to chapter 4, for more information about CC method)

26 systems(France: 6, Spain: 11, UK: 9) were investigated, for part load ratio and efficiency, during at least one year (2013)

Findings:

1) Partial load ratio : 80% of systems had a 95%tile partial load ratio below 50%.
2) Plotting seasonal efficiency as function of 95%tile part load ratio, showed a clear relation, see below figures

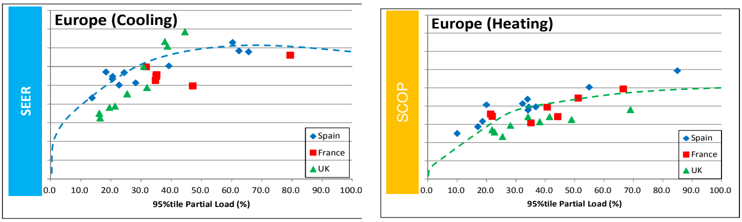

Figure 3 : Seasonal performance as function of 95%tile part load ratio

Each dot represents a VRV system in a building.

SEER/SCOP are the total delivered cooling/heating energy to the building divided by the total electrical consumption during the monitoring period (1 year).

95%tile partial load is the maximum occurring load of the VRV, in percentage of the maximum possible load, during the monitoring period, reduced by the top 5% for noise reduction. Eg. A 95%tile load of 50% means that during 95% of the time the part load ratio of the unit was 50% or lower. (100% partial load, means full capacity of the unit)

Conclusion:

It is said that the performance would be optimized at 40% to 60% partial load ratio. But most VRV's in buildings operate at less than optimal performance, caused by oversizing.
For the theoretical explanation of the reason behind this, refer to next chapter "Explanation for reduction of performance in oversized case".

6. Explanation for reduction performance in oversized case (theoretically)

Below figure shows a typical pattern of performance of a system with DC inverter compressor.

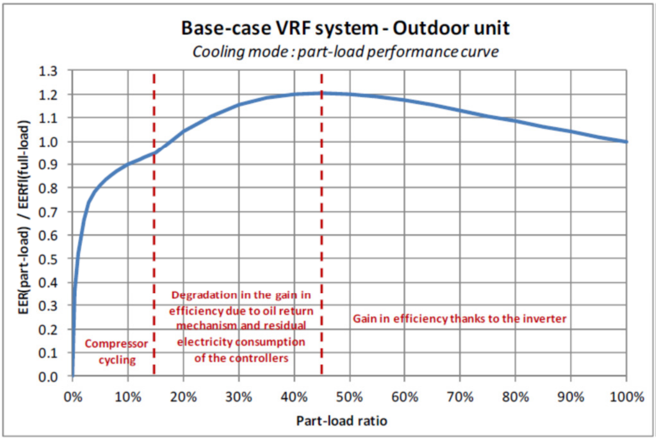

Figure 4 : Estimation of the cooling efficiency of VRF system at different part-load conditions (Final report of Task 4 – Air Conditioning products July 2012).

Well dimensioning of the unit's capacity to the building load, supposes 100 % part load ratio to occur at peak load, for cooling it means at the highest occurring outside temperature, for heating at the lowest occurring outside temperature. See below figure.

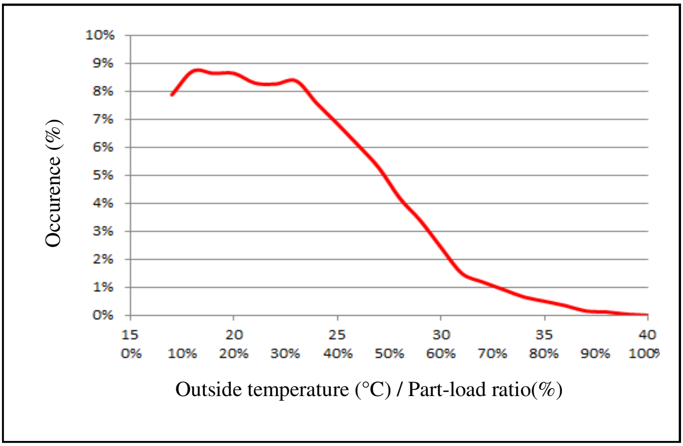

Figure 5 : Occurrence of outside temperatures(°C) corresponding to the reference cooling season in EN14825 and the corresponding part-load ratio(%)

Applying the curve of the cooling efficiency

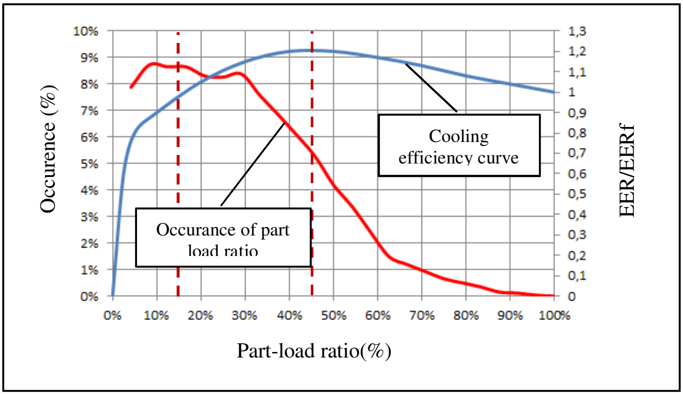

Figure 6 : Combining figure 4 and 5 for VRF dimensioned exactly to the building load

Optimal EER (around 40% to 60% part load ratio) occurs relatively much, while there is a small occurrence of compressor cycling. However, when unit capacity would be 2x the building load we get following curve.

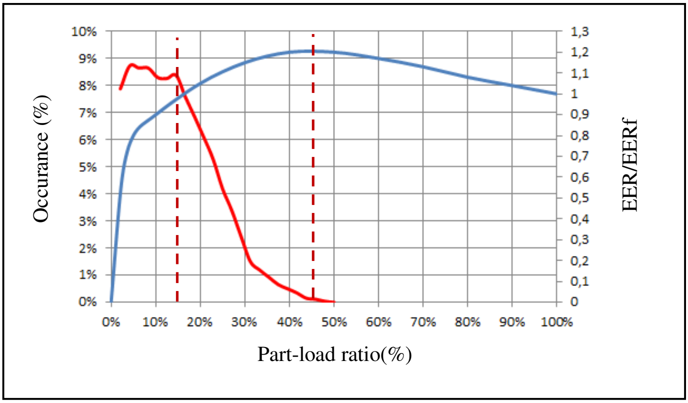

Figure 7 : same as figure 6 but here the VRF capacity is 2x the building load

Now, optimal EER practically never occurs, while there is a big occurrence of compressor cycling. Seasonal EER of this case will be much lower than the first case. Additionally we expect a lower comfort feeling due to the high compressor cycling occurrence.

7. Monitoring project – phase 2

In this phase, which is still ongoing, several sites were selected (currently 13) for deeper analysis of the unit performance and to accurately measure the part load ratio and related efficiency as described in the previous chapter.
For this purpose, minutely data is captured.
Merit : start-stop conditions(compressor cycling), defrost cycles, etc. can be analyzed.

The monitored sites were mainly offices in southern Europe(focus on cooling), but also some were located in more northern countries and for other applications like shops and hotels.

Findings:

- We confirm the oversizing as in phase 1, even though some sites were also undersized
- Besides dimensioning issues to the load, we found that many sites suffer from comfort issues, related to human mistakes during commissioning and use of the equipment.
- In some cases we detected issues due to improper installation of the equipment.

For further analysis we split up in 3 interests:

1) What is the performance of the system.
2) How well is the system dimensioned to the building load.
3) What is the comfort in the building.

In many cases there is a relation between the 3.

1) Performance

It is not in our scope to investigate the performance of the building shell, so our performance investigation is limited to the system, which is defined by the EER/COP. As seen in chapter 6, EER/COP is related to partial load. So, both are to be visualized for analysis.

Below figure shows real performance data of a system in a building (System 1).
Optimal EER occurs at a part load ratio of 35 - 40%. As this part load occurs relatively much (almost 10%) As a consequence this site has a good seasonal efficiency.

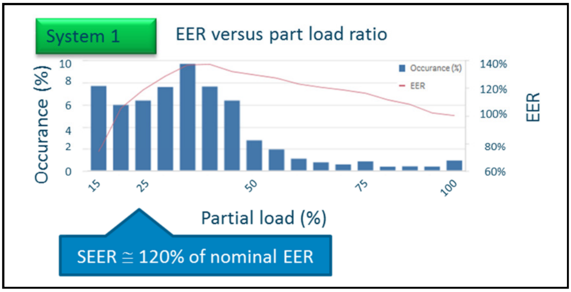

Figure 8 : France, shop. Period 01/2015 to 08/2015

Partial load (%) is defined by compressor speed. 100% partial load means maximum compressor speed.
EER is expressed in percentage of nominal EER. Nominal EER occurs at 100% partial load.

Below figure, of another building (System 2), shows that for the occurring part load ratios, the EER is similar as previous unit (even slightly better), but because it mainly operates at minimum partial load(5%) ,with lower EER, the seasonal EER is much lower than the first one.

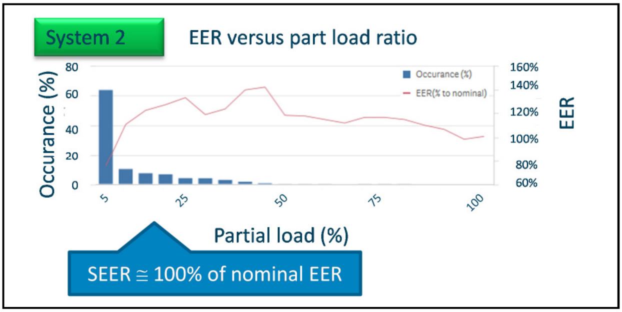

Figure 9 : France, office. Period 01/2015 to 08/2015

Below figure, of another building (System 3), shows a case of under sizing.

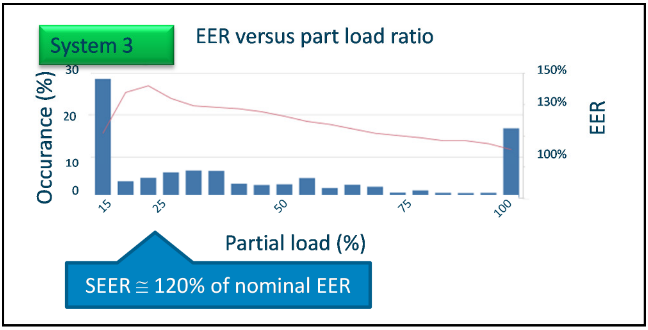

Figure 10 : Italy, shop. Period 01/2015 to 08/2015

It has to be noted that in this building, night set back function is activated (forced low part load operation during the night). It explains the high occurrence of low part load ratio. On the other hand, it shows also high occurrence of 100% part load, which occurs during many days. Nevertheless SEER is still 120% of the EER.
We can conclude that oversizing has significant negative effect on the performance, while under sizing will not affect the performance that much. On the other hand, under sizing, will have significant negative effect on the comfort (see later)

2) Building load

To get an idea of the building load, the data has to be visualized differently. Therefore we depict the part load as function of the outside temperature.

Setting out raw data results in following graph. Due to the dynamic behavior of both the building and the climate and the reaction of the unit to it, the data is very scattered.

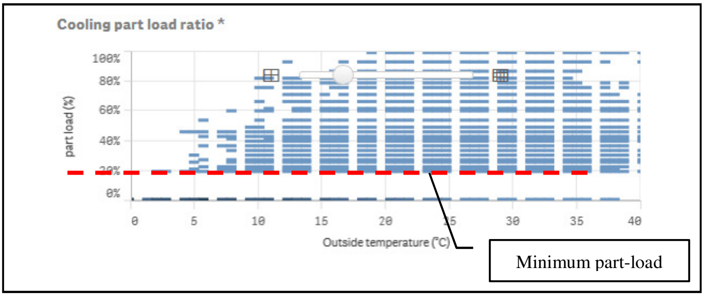

Figure 11 : France, shop. Period 01/2015 to 08/2015 (minutely data)

Making a moving average of the data, eliminates the big fluctuations in the unit's capacity. In below graphs is the result of an hourly moving average of system 1. See below figure.

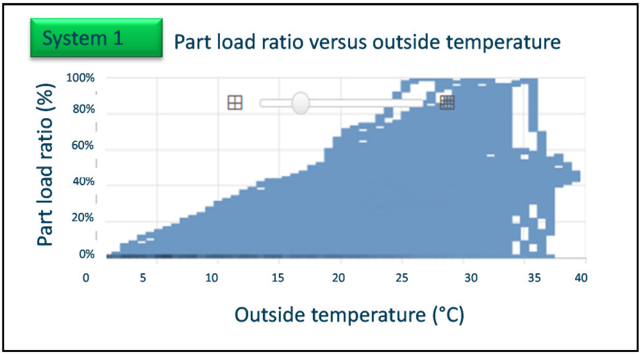

Figure 12 : France, shop. Period 01/2015 to 08/2015 (hourly moving average data)

As can be seen, at around 2°C, there is already a (very small) cooling demand, which increases with increasing outside temperatures. The reason why highest part load does not occur at the very highest outside temperature is due to thermal response time of the building and orientation of the windows (irradiance).

For sytem 2 we get a similar trend. There are 2 differences. There is only cooling demand above 12°C, also part load ratio remains lower. See below figure.

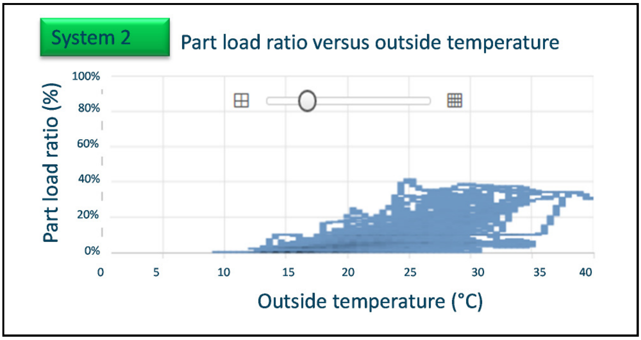

Figure 13 : France, office. Period 01/2015 to 08/2015 (hourly moving average data)

System 2 is oversized to the building load. Highest part load ratio occurring (moving hourly average) is 40%.

3) Comfort

This topic is more difficult because comfort is decided by many parameters (temperature, humidity, draft, CO_2, etc.). With the used monitoring system it is not possible yet to monitor all those parameters so our comfort analysis is limited to the indoor temperature and the deviation of it to the set temperature. The results of the comfort analysis are very different for each building and discomfort can be due to many reasons.

In most cases it concerned human errors which could be easily solved by changing settings. In some cases we detected discomfort due to under sizing of indoor units, and even some cases of undersized outdoor units.

Below figure is the example of system 3.

We set out comfort as function of part load ratio. Comfort is expressed in percentage. We judged that comfort is OK when the actual temperature is within 1 degree of the target temperature. Then, eg. a comfort level of 90% means that 90% of time during the monitoring period, the actual temperature was within 1 degree of the target temperature.

At low part load operation, comfort is bad, which is to be expected due to the forced low part load operation at night(so, to be discarded), but between partial load of 20% up to 60%, the system can maintain a comfort level of 90%. For higher part load operation, the comfort level starts to go down because the indoor units cannot achieve the target temperature anymore. At partial load of 100%, the omfort has reduced to 70%.

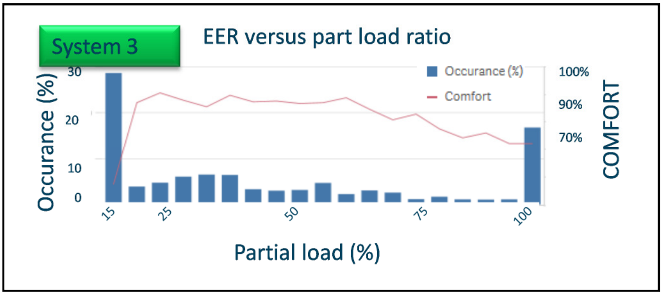

Figure 14 : Italy, shop. Period 01/2015 to 08/2015

We conclude that selecting optimal capacity requires a balance between performance and comfort. While the capacity of the system should be as close as possible to the real load of the building we must avoid reduced comfort due to under sizing of (indoor) unit(s).

We could not find clear cases of discomfort due to oversizing. This is probably because indoor temperature is measured at the suction side of the indoor unit and/or at the room thermostat, these are locations where cold draft is less likely to be sensed. For near future we plan to monitor also indoor unit discharge air temperatures. This should allow us to detect cold draft due to compressor cycling in case of oversized unit.

8. Conclusion

In order to close the gap between calculated performance and real performance of equipment in buildings, it is required to dimension the capacity of the HVAC equipment well to the building load. Close monitoring of the system during its lifetime is most useful to optimise the equipment operation. Based on the findings, energy saving measures can be done, but also, the data can be used for replacement to optimal capacity unit at end of life, or as benchmark for other buildings.

9. Nomenclature

Q : Capacity, kW
G : mass flow rate refrigerant, kg/s
Δic : difference of enthalpy of evaporator, kJ/kg
Δih : difference of enthalpy of condensor, kJ/kg
Tc : Saturation temperature to condensing pressure (°C)
Te : Saturation temperature to evaporating pressure (°C)
SC : Refrigerant subcool (°C)
SH : Refrigerant superheat(°C)