2002

# Testing Methodology For VRF Systems

J. Xia
*Tsinghua University*

E. Winandy
*University of Liege*

B. Georges
*University of Liege*

J. Lebrun
*University of Liege*

# TESTING METHODOLOGY FOR VRF SYSTEMS

*Jianjun XIA, Department of Building science, Tsinghua University
100084,Beijing, China; Tel: +86/10 62789761; Fax: +86/10 62773461
Email: Xiajjun99@mails.tsinghua.edu.cn *Author for Correspondence

Eric WINANDY, Bernard GEORGES and Jean LEBRUN,
Laboratory of Thermodynamic, University of Liège, Campus of Sart Tilman,
Building B49, Belgium; Tel: +32/4 3664800; Fax: +32/4 3664812
Email: eric.winandy@planetmail.com, b.georges@ulg.ac.be, j.lebrun@ulg.ac.be

## ABSTRACT

This paper presents a testing methodology applicable to Variable Refrigerant Flow (VRF) equipment. A test bench is presented: it consists in a set of 6 calorimeters, each one fully instrumented and controlled in such a way to compensate almost all combinations of sensible, latent, heating and cooling loads. This test bench has been used for a three-pipe VRF system with 5 indoor units and 1 outdoor unit. Examples of testing results are presented in the paper, in order to illustrate the methodology and also validate a simulation model.

The (heating or cooling) emission of each indoor unit is identified thanks to a very accurate "air" balance inside each calorimeter. Refrigerant side (pressure and temperature) measurements are used in order to identify the refrigerant flow rates, the characteristics of the compressors (isentropic effectivenesses) and the characteristics of the terminal units (heat transfer coefficients) in different regimes.

Examples of global performance evaluation are also presented in the paper.

## INTRODUCTION

Since the energy-efficient office building operation becomes more and more important all over the world, increasing attention now is being paid to air conditioning systems that can operate at part load with individual temperature control. The diversity of variable refrigerant flow rate systems ("VRF", "VRV" or "Multi-systems") commercially available is increasing every year. Compared with the conventional central air conditioning system, these systems can achieve a room-to-room temperature control, without air and water distribution circuits.

A three-pipe VRF system can also provide simultaneous cooling and heating; such service is welcome in many office buildings, during the temperate season and also in wintertime.

The three-pipe VRF system has four different operation modes [1]:

1) "Heating-all": All working indoor units are in the heating operation;
2) "Cooling-all": All working indoor units are in the cooling operation,;
3) "Mainly heating": Heating is the principal mode in the simultaneous cooling and heating operation;
4) "Mainly cooling": Cooling is the principal mode in the simultaneous cooling and heating operation.

The main inconvenient of a VRF in comparison with air and water distribution systems are:

1) Much longer refrigerant pipes and therefore, a higher refrigerant charge;
2) Higher refrigerant pressure drops along the pipes;

3) A more complex control system [2].

A test bench was built at the Thermodynamics Laboratory (University of Liège, Belgium) for a three-pipe VRF system, in order to verify its performance in the whole operation range.

## GENERAL DESCRIPTION OF THE TEST BENCH

### 2.1 System description

The VRF selected is a three-pipe system, charged with R22. It consists of 5 CH (cooling/heating changeover) indoor units, and one outdoor unit.

The outdoor unit is composed of a parallel set of two hermetic scroll compressors (one variable speed and one constant speed), two air-cooled heat exchangers, two variable speed fans, two electronic expansion valves, an accumulator and two reversing valves. The variable speed compressor is supplied by the frequency controller with a range of 20Hz to 115Hz.

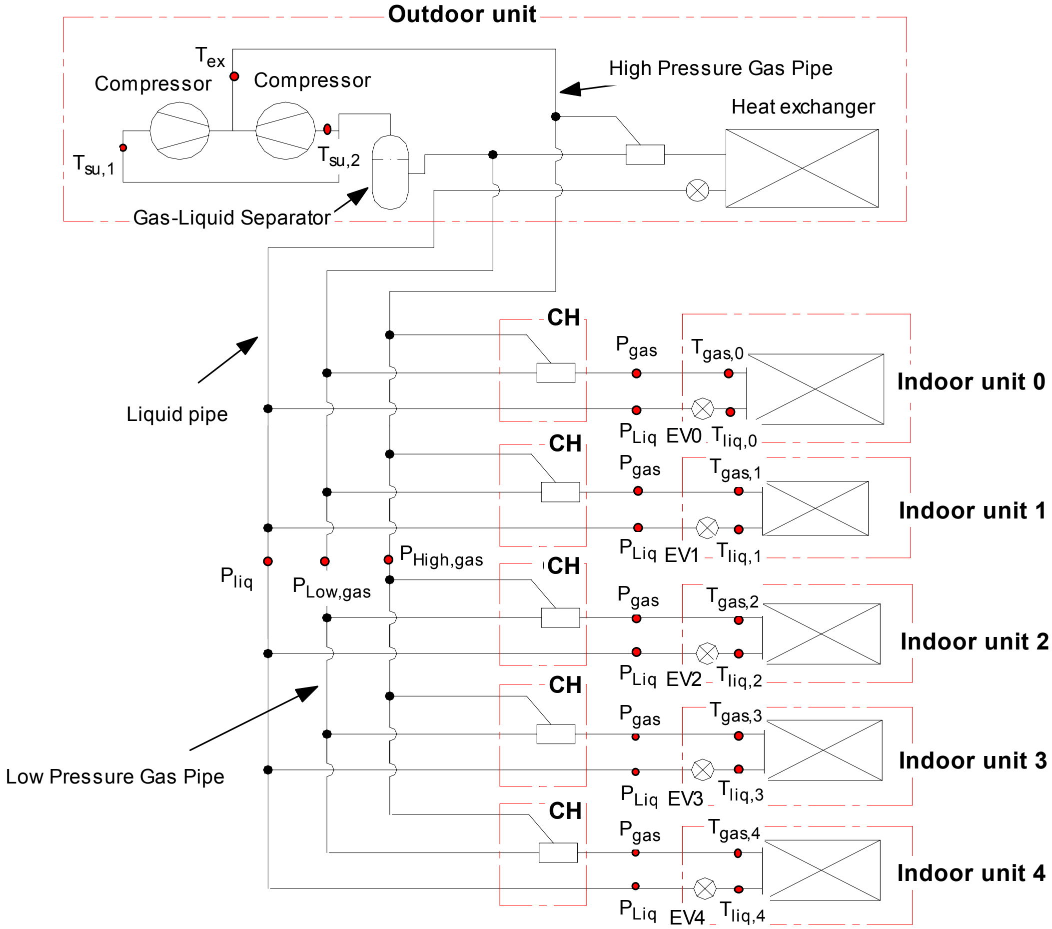


**Figure 1 Scheme of the refrigerant circuit of the VRF system**

Each indoor unit is composed of an air-cooled heat exchanger, a three-speed fan and an electronic expansion valve.

Figure 1 shows the connection principle diagram.

### 2.2 Indoor unit calorimeters

In the laboratory test bench, each indoor unit is enclosed in a calorimeter box representing the room. Auxiliary heating/cooling systems are also included in the way to generate a thermal load. The calorimeter permits to obtain good thermal balances by reducing as much as possible the ambient exchanges. In the same way, the outdoor unit is also enclosed in a calorimeter with its auxiliary heating/cooling systems.

Figure 2 shows us the detail view of the calorimeters

All the arrangements ensure the air having a good distribution to the indoor unit fan sucking zone.

Instead of measuring the airflow rate of the indoor unit, the energy balance inside of the indoor unit calorimeter is used here to calculate the thermal output of each indoor unit. (Figure 3)

**Figure 2 Indoor unit calorimeter**

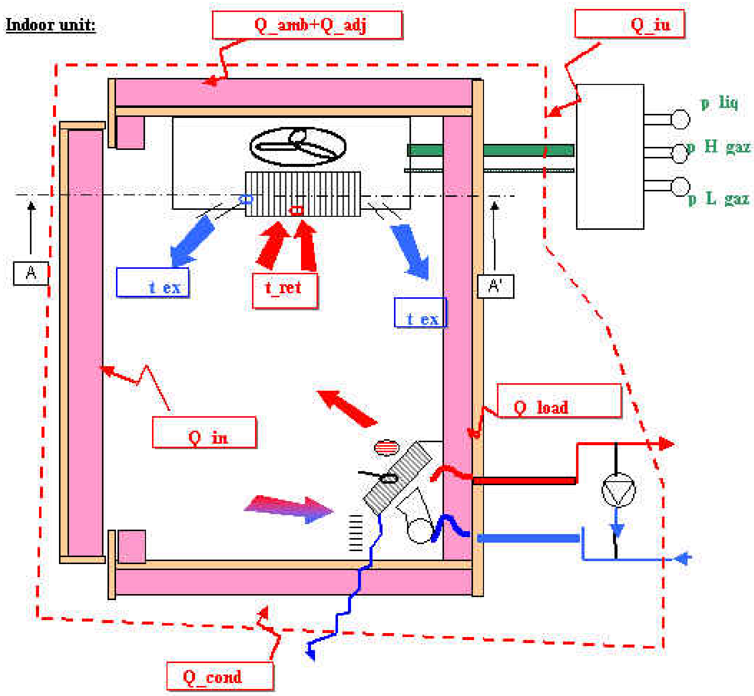


**Figure 3 Energy balance of the indoor unit calorimeter**

The energy balance inside the calorimeters can be defined by the following equation:

$$\dot{Q}_{iu} + \dot{Q}_{load} + \dot{Q}_{amb} + \dot{Q}_{adj} + \dot{Q}_{fan} = \dot{Q}_{in} \qquad (1)$$

Where:

$\dot{Q}_{iu}$ is the thermal output of the indoor unit;

$\dot{Q}_{load}$ is the thermal load of the building, which can be provided by the electric heater the fan coil and the steam humidifier.

When provided by the fan coil, this term can be calculated on water side:

$$\dot{Q}_{load} = \dot{M}_w \cdot C_{p,w} \cdot (t_{w,su} - t_{w,ex}) \qquad (2)$$

$\dot{Q}_{amb}$ is the heat transfer from the ambient to the calorimeter, which can be calculated by:

$$\dot{Q}_{amb} = AU_{amb} \cdot (t_{a,amb} - t_{a,in}) \qquad (3)$$

$\dot{Q}_{adj}$ is the heat transfer from each adjacent box to the calorimeter, which can be calculated by:

$$\dot{Q}_{adj} = AU_{adj} \cdot (t_{a,adj,in} - t - a_{in}) \qquad (4)$$

$\dot{Q}_{fan}$ **is** the electric consumption of the fans of the indoor unit fan and of the fan-coil, which is measured by the electric counter.

$\dot{Q}_{in}$ **is** the rate of thermal storage inside the calorimeter.

In order to avoid the thermal disturbance from the ambient, the walls of the calorimeters are well insolated. The heat transfer coefficient between the ambient and each calorimeter is around 5 W/ K, according to the calibration. During all the tests, the temperatures maintained inside the indoor unit calorimeters are about 22 °C, which is very near to the ambient temperature (inside the laboratory). Thus the heat transfer to the ambient is quite small, compared with the thermal output of the indoor unit. The identified thermal mass of each calorimeter is around 150 kJ/K, so the rate of thermal storage is of about 4 W for 1 K/h of temperature drift. This term cannot be ignored in the energy balance calculation.

2.3 Outdoor unit calorimeter (Figure 4).

The outdoor calorimeter is located upper the indoor boxes. It contains the outdoor unit and the heating/cooling systems.

The same energy balance is made as for other calorimeters. Two air handling units, equipped with heating/cooling coils, are used to maintain the outdoor temperature. They are supplied with hot or cool water according to the needed conditions.

2.4 Measurements

Nearly 150 thermocouples (type T, copper copper-nickel) are used here to measure the temperatures of the VRF system. All the thermocouples were calibrated, with the help of a hot water bath, with the reference temperature 0 °C given by a water-ice mixture. The final accuracy is estimated to ±0.1 and ±0.3K, in relative and absolute values respectively.

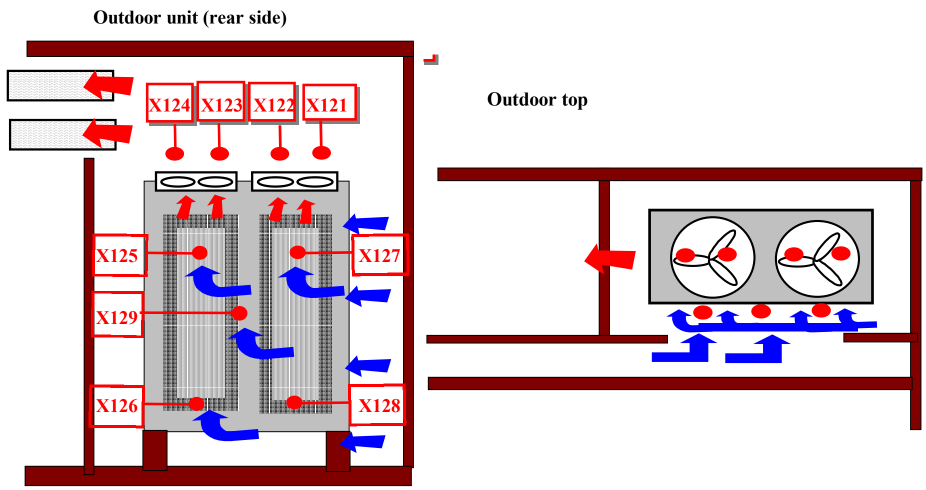


**Figure 4 Location of the outdoor unit air temperature measurement**

Five electric power counters are used to measure the electric power input of the five calorimeters. Each measurement includes the electric powers consumed by the fan coil, the electric heater, and the fan of the indoor unit. The measuring range of each counter was tested from 150W to about 3200W, and the error was identified to less than 1%.

Five water counters measure the water flow-rates injected in the water circuits of the fan-coils located in the indoor calorimeters. Another water counter measures the water flow-rate injected in the water circuit of the air handing units located inside the outdoor calorimeter. All those counters were calibrated with an accurate balance.

Electrical vapor generators are used to produce latent load in the indoor calorimeters. They are calibrated by weighing the generator, while producing vapor and measuring the consumed power. The emission of the vapor generator depends on the supply voltage in the laboratory (often between 213 and 222V) and the efficiency can be considered as 91±1%, according to the calibration. A maximum emission error of 100W is expected, due to voltage variation. Condensed water is collected outside of the calorimeter and weighed, in order for comparison with the mass injected by the vapor generators.

A global error of ±2% is found for the thermal capacity of each indoor unit during these tests.

## RESULTS

Figure 5 shows the global COP of the system at the different part load ratios and at different outdoor temperatures. The Global COP is defined by using total thermal load divided by the total electric consumption of the system (compressor and fans).

All the tests presented here are performed in "cooling all" mode and without latent load (SHR = 1.0). From the figure we can find that the COP does not vary too much according to the part load ratio. This can be explained by the use of two compressors in "tandem", which gives very good part load performance. The COP decreases a little with the increasing of the outdoor air temperature.

Figure 6 shows the compressor operation conditions during all the tests. We can find that a very low constant evaporating pressure 5.5 Bar (evaporating temperature of 3.1°C) is maintained at the suction of compressor, even during the non dehumidification operation. This is due to the system control strategy: with the low evaporating pressure, by control the mass flow rate of refrigerant of each indoor unit (using expansion valve), the system can fulfill the thermal need of each calorimeter. By using a liquid by-pass, the system controls the global superheating to a reasonable range. This control is much easier to realize but it increase the energy consumption of the compressors, because not all the tests conditions need the system to maintain such a low evaporating pressure.

The global heat transfer coefficient of each heat exchanger is calculated with ε – NTU method, doing as if the refrigerant was isothermal (i.e. without taking the overheating into account):

$$\varepsilon_{iu} = 1 - \exp(NTU_{iu}) \tag{5}$$

$$NTU_{iu} = \frac{AUiu}{\dot{c}_{a,\,iu}} \tag{6}$$

$$\varepsilon_{iu} = \frac{t_{a,\,ret,\,iu} - t_{a,\,ex,\,iu}}{t_{a,\,ret,\,iu} - t_{su,\,iu}} \tag{7}$$

Where $t_{a,ex,iu}$, $t_{a,ret,iu}$ are the exhaust and return air temperature of the indoor unit, $t_{su,iu}$ is the evaporating temperature of the refrigerant at the supply of the indoor unit.

From the figure 7 we can find that, with the decreasing of the thermal output, the superheating after each heat exchanger is increasing (about 9~15°C) and the heat transfer coefficient of the fictitious semi-isothermal heat exchanger decreases rapidly.

Figure 8 shows that the power consumed by the compressor can be correlated through a linear regression with the corresponding isentropic power. [3].

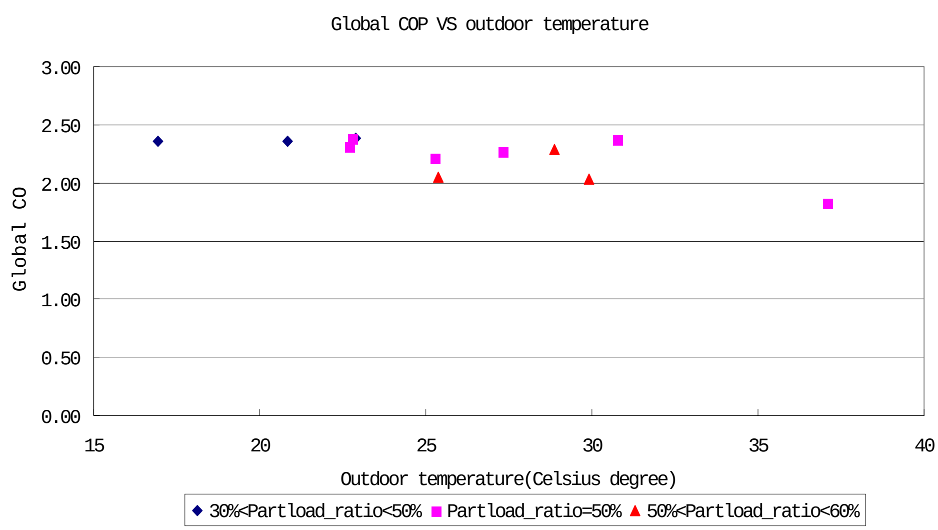


**Figure 5 Global COP of the cooling all mode**

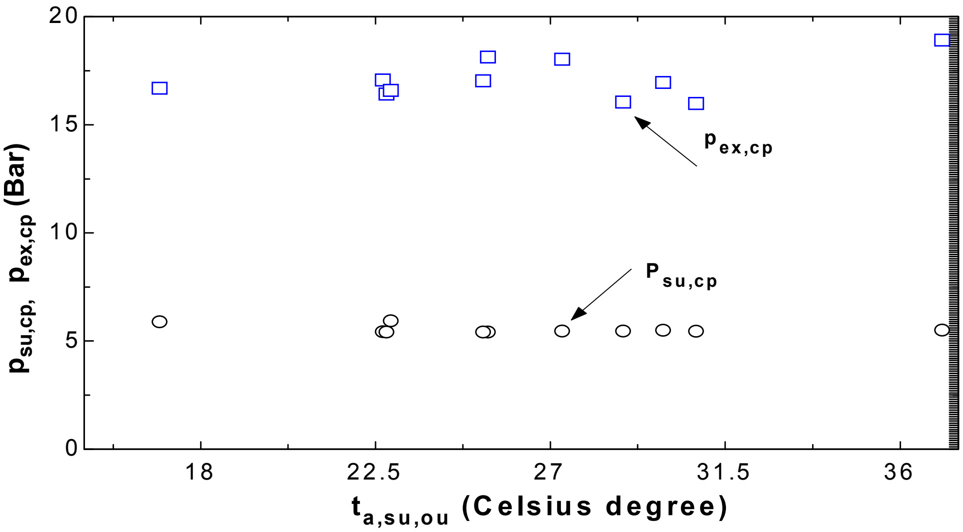


**Figure 6 Compressor inlet and outlet pressure**

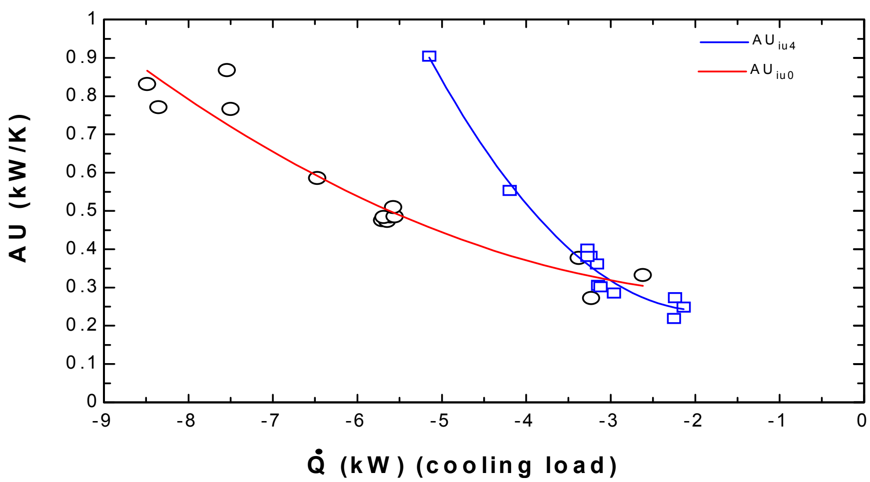


**Figure 7 Heat transfer coefficient of the indoor unit heat exchanger VS thermal output**

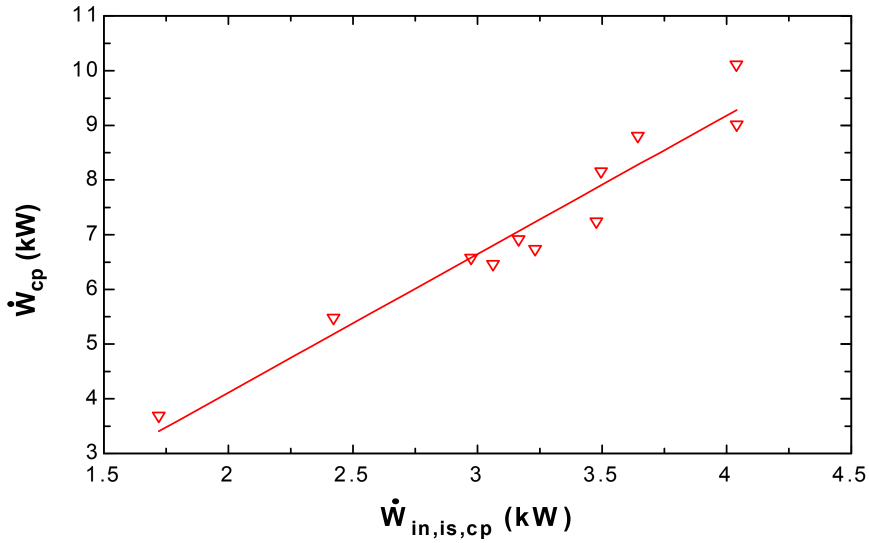


**Figure 8 the power consumed by the compressor**

## CONCLUSIONS

A test bench of VRF system is built in the laboratory. By using the accurate energy balance inside each calorimeter, we get the thermal output of each indoor unit and outdoor unit instead of by using air enthalpy method, which would require to measure the dry and wet bulb temperatures of air entering and leaving the unit and the airflow rate. With this method, a global error of ±2% is found for the thermal capacity calculation.

A VRF system product provided by the manufacturer is test in the test bench. A global COP of 1.9~2.4 is found during the sensible cooling regime. A constant evaporating pressure is found during all the tests, due to the control strategy of the system.


## ACKNOWLEDGMENTS

The authors wish to thank the financial support of Electrabel and DTC. This study has been possible thanks to the cooperation program between the Tsinghua University –China and the University of Liege – Belgium.


## REFERENCE


[1] H. Lijima, N. Tanaka, et al, Development of a new multi-system air conditioner with concurrent heating and cooling operation. ASHRAE Trans. 1991, Vol.97, Part 2, p309-315

[2] M. Masuda, K.Wakahara, et al, Development of a multi-system air conditioner for residential use. ASHRAE Trans, 1991, vol.97, Part 1, p127-131

[3] Winandy E., Saavedra C. and Lebrun J., Experimental analysis and simplified modeling of a hermetic scroll refrigeration compressor, Applied Thermal Engineering, Volume 22, Issue 2, February 2002, Pages 107-120.