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Testing of Refrigeration Compressors without Condensation

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ABSTRACT

This study presents the design procedure and the experimental results obtained on a new test bench for refrigeration compressors. The main components of the plant are: the compressor to be tested, a water cooled heat exchanger, an expansion valve and a refrigerant vessel.

The test bench can be used for: performance evaluation of the compressor (refrigerant flow rate can be found on a heat balance on the gas cooler and after it can be converted to cooling capacity), long live tests and quality control for compressors.

The advantages of a « fully gas cycle » are: low energy consumption refrigerant charge of the plant, small inertia of the plant.

NOMENCLATURE

AU - global heat transfer coefficient
c - specific heat
h - specific enthalpy
M - mass
 \dot{M} - mass flow rate
p - pressure
 \dot{Q} - heat flow rate
S - surface
t - temperature
U - internal energy
v - specific volume
V - volume
 \dot{V} - volumetric flow rate
W - power consumption

Greek:

Δ, δ - difference
 $\Delta\theta_m$ - mean log temperature difference
 ε - efficiency
 γ - isentropic coefficient
 τ - time
 ρ - density

Index:

amb - ambient
cp - compressor
cr - critical
ex - exhaust
HE - heat exchanger
HP - high pressure side
is - isentropic
LP - low pressure side
R - refrigerant
su - supply
sw - swept
vol - volumetric
w - water

INTRODUCTION

Performance evaluation of refrigeration compressors include estimation of cooling capacity for given setpoints. International Standard ISO 917 [1] describes 9 test methods for the determination of the cooling capacity:

- A. secondary fluid calorimeter in suction line
- B. flooded system refrigerant calorimeter in suction line
- C. dry system refrigerant calorimeter
- D. refrigerant vapour flow meter in the suction line or in the discharge line
- E. refrigerant liquid flowmeter
- F. water cooled condenser
- G. partial condensation of refrigerant
- H. calorimeter in the discharge line

For methods A, B and C, the cooling capacity is found by direct measurement (heat supplied to the evaporator). Methods D and E allow a direct measurement of refrigerant flow rate. Methods F, G and H estimate the refrigerant flow rate using an energy balance. Consequently, for methods D to H, the cooling capacity is found using the thermodynamic properties of the fluid.

The criteria of an efficient test bench for compressors should be:

- good precision
- facility of control
- low energy consumption
- small inertia (test point and steady state conditions should be reached as quickly as possible)

- simple layout of the test bench

Among the methods enumerated above, some of them are high energy consuming (A, B, C), others require a complex control (G) or are susceptible of introducing inaccuracies (G, D).

THEORETICAL APPROACH

From the test methods enumerated above, method G (partial condensation of refrigerant) has the advantage of minimal energy consumption. Its disadvantages are:

- only a fraction of the refrigerant flow rate is measured directly; the other fraction is estimated using a heat balance
- the design of the gas cooler shall be such that the outgoing vapour will not entrain droplets of liquid refrigerant
- complex control procedure

The layout of the plant and the functioning cycle are shown in Figure 1.

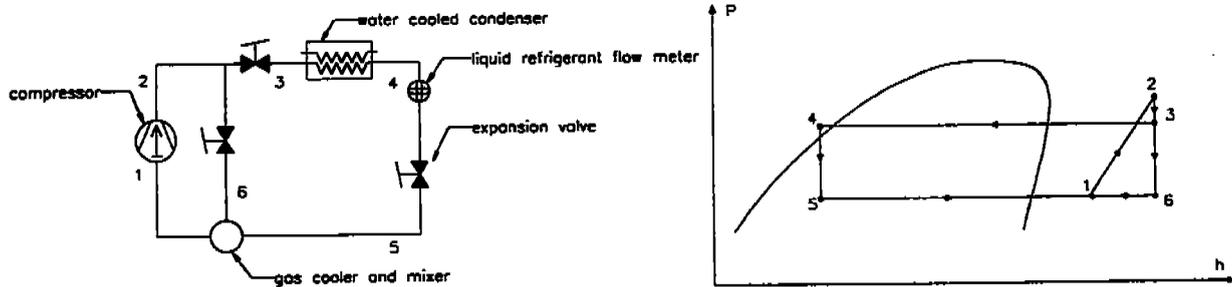


Figure 1: Partial condensation of refrigerant: plant layout and functioning cycle

This paper will introduce a refrigeration compressor testing facility with minimum energy consumption and easy control procedure: « fully gas cycle ».

The functioning cycle of a compressor running in a fully gas cycle is shown in Figure 2. If such a functioning cycle is foreseen, the layout of the plant will be as the one shown in Figure 3.

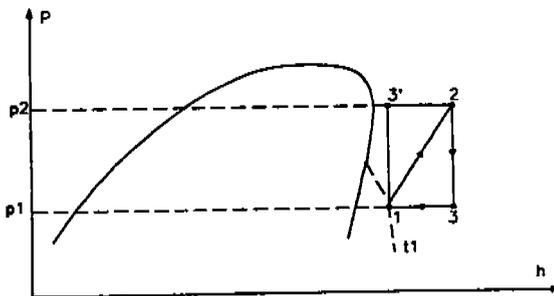


Figure 2: Fully gas cycle

A test point for refrigeration compressors is obtained by imposing three physical variables:

- pressure at compressor supply (p_1)
- temperature at compressor supply (t_1)
- pressure at compressor discharge (p_2)

On the plant shown in Figure 3 we dispose only of two parameters that can be tuned:

- the opening of the expansion valve
- the temperature (or flow rate) of the cooling water (t_{wsu} or \dot{M}_w)

therefore the system has only two degrees of freedom. If we want to reach any set point for the compressor we must add another degree of freedom. This third degree of freedom is given by the refrigerant charge of the system. The corresponding layout of the plant is shown in Figure 4.

The refrigerant charge of the plant can be increased by opening the solenoid valve on the low pressure side: the refrigerant will migrate from the vessel to the plant. If the charge of the plant is to be reduced, the solenoid valve on the high pressure side should be opened. The capillary tubes limit the flow rate from / towards the vessel.

The gas cooler is installed on the low pressure side of the refrigerant circuit due to the following reasons:

- the risk of condensation is easier to avoid (cooling water must have a temperature superior to the saturation temperature of the refrigerant in order to avoid condensation);

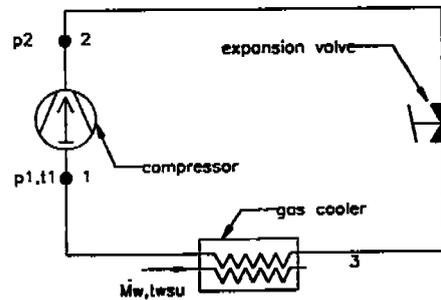


Figure 3: Initial plant layout

- suction temperature of the compressor is easier to control: if a high efficiency heat exchanger is used, the cooling water supply temperature must be set to the desired inlet temperature of the compressor. (If the heat exchanger is installed on the high pressure side, the water supply temperature must be computed using refrigerant properties);
- lower heat exchange between the gas cooler and the ambience.

The only disadvantage introduced by a low pressure cooler is that the expansion valves are bigger than the ones used in case of a high pressure gas cooler (the specific volume at the inlet of the expansion valves is higher).

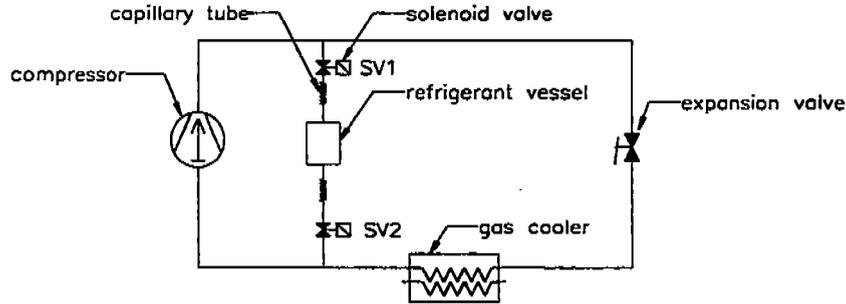


Figure 4: Testing facility layout

The equations that describe the behaviour of the system are:

- for the compressor:

$$\dot{V}_R = \dot{M}_R v_{su_{cp}} = \varepsilon_{vol} \dot{V}_{sw} \quad \varepsilon_{vol} = f(p_{su_{cp}}, t_{su_{cp}}, p_{ex_{cp}})$$

$$\varepsilon_{is} = \frac{\dot{M}_R \Delta h_{is}}{\dot{W}_{el}} \quad \varepsilon_{is} = f(p_{su_{cp}}, t_{su_{cp}}, p_{ex_{cp}})$$

- for the expansion valve:

$$\dot{M}_R = f(S, p_{ex_{cp}}, t_{ex_{cp}}, \Delta p) \quad \Delta p = p_{ex_{cp}} - \max(p_{su_{cp}}, p_{cr}) \quad \text{if } p_{su_{cp}} < p_{cr} \text{ choked flow}$$

- for the heat exchanger:

$$\dot{Q}_{HE} = \dot{M}_R (h_{su_{HE}} - h_{ex_{HE}}) = \dot{M}_w c_{p_w} (t_{w_{ex}} - t_{w_{su}}) = AU \Delta \theta_m$$

- mass conservation:

$$\dot{M}_R = \sum \dot{M}_{R_i} = \rho_{LP} V_{LP} + \rho_{HP} V_{HP}$$

For a given refrigerant charge of the circuit and supply conditions of the compressor (pressure and temperature) the above equations can be solved for the discharge pressure and the opening degree of the valve. Furthermore, by imposing water supply temperature, we can compute the water flow rate and its discharge temperature. Sample results (high pressure and low pressure as a function the opening degree of the expansion valve for given refrigerant charges) are given in Figure 5.

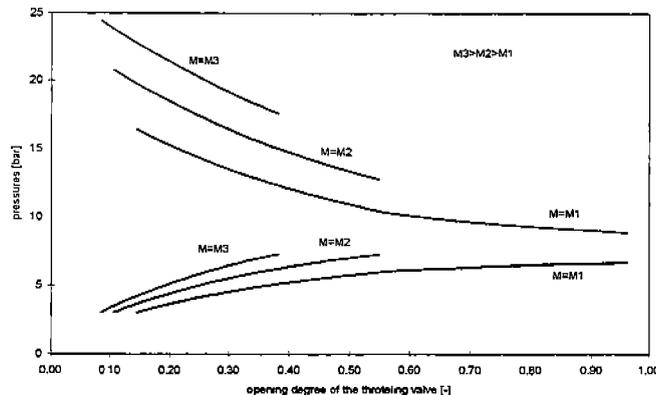


Figure 5: System behaviour

EXPERIMENTAL SET-UP

The test facility is shown in Figure 6. The differences between the experimental set-up and the theoretical one are:

- the expansion valve was replaced with a downstream pressure regulator for control purposes
- the gas cooler is placed on the high pressure side (this was the initial design)

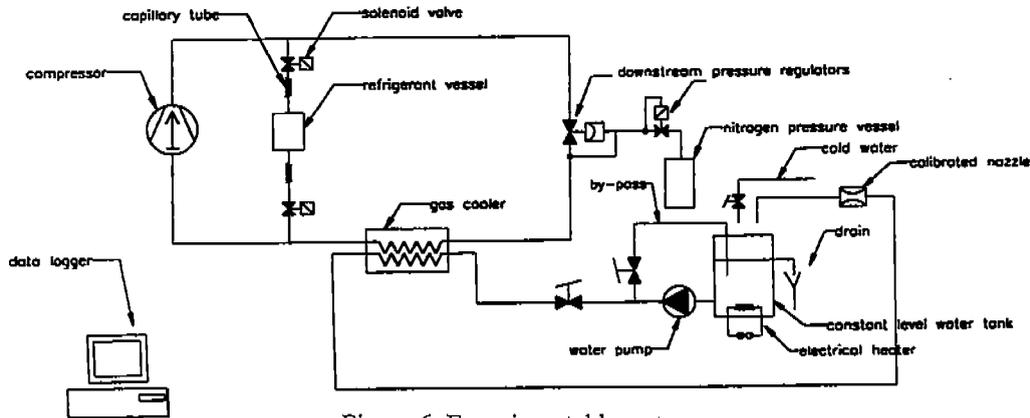


Figure 6: Experimental layout

The downstream pressure regulator is built from a thermostatic expansion valve, modified in order to take under control the pressure inside the bulb. Water supply temperature is obtained with an ON/OFF control of the heating resistance. The water by-pass is used to obtain a homogenous water temperature inside the tank. Water flow rate is measured with a calibrated nozzle.

The refrigerant vessel can be designed in two hypothesis:

- its content is only in vapour state. The result will be a bigger vessel with no cooling
- its content is in saturated state (liquid + vapours). The result is a smaller vessel but cooling of the vessel should be provided.

In a first approach, the vessel was designed for a vapour content. Its volume is chosen in such a way that, no matter what the functioning regime of the plant is, the pressure inside the vessel is between the low pressure and the high pressure. In this way the charge of the circuit can always be varied.

If an industrial application of the method is foreseen, the optimum design will contain a central refrigerant vessel to which all the testing facilities are connected through solenoid valves. This time, the refrigerant reserve will be kept in saturated state (cooling of the central refrigerant vessel should be provided).

The first compressor installed on the test bench was a 10 kW (2.85 tons) refrigeration capacity scroll compressor.

EXPERIMENTAL RESULTS

In a first phase, the tests that were performed had as objective to show that any desired test point can be reached. It was not paid a special attention to the estimation of cooling capacity. For this reason the circuit does not contain an oil separator and the refrigerant mass flow rate can be identified only with one method (heat balance).

Figures 7 and 8 show sample evolution trends for pressure and temperature.

The set point of the two tests are:

Test 1:

- condensing temperature: 50°C (122°F)
- evaporating temperature: -15°C (5°F)
- temperature at compressor inlet: 15°C (59°F)

Test 2:

- condensing temperature: 40°C (104°F)
- evaporating temperature: -10°C (14°F)
- temperature at compressor inlet: 15°C (59°F)

The tolerances imposed on both set points are: $\pm 2^\circ\text{C}$ (3.6°F) for the condensing temperature, $\pm 1^\circ\text{C}$ (1.8°F) for the evaporating temperature, $\pm 2^\circ\text{C}$ (3.6°F) for the supply temperature of the compressor. Both tests were inside the imposed limits.

The evolution trends show the small inertia of the system: a set point was obtained in around 15 minutes.

The refrigerant charge used to produce the test was around 0.2 kg (0.44 lb.). If a classical circuit would have been used to produce the same test (method A for example), the refrigerant charge of the circuit can easily reach 2 kg (4.4 lb.).

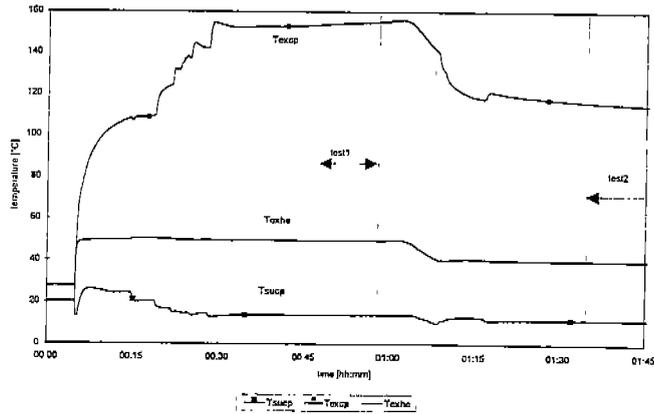


Figure 7: Pressure evolution

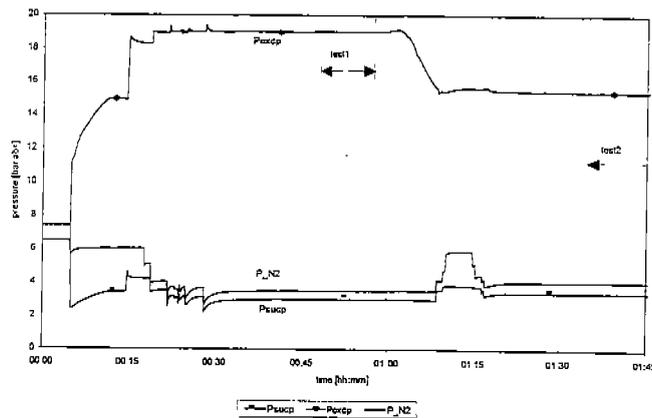


Figure 8: Temperature evolution

COOLING CAPACITY EVALUATION

As mentioned in International Standard ISO 917: « all tests shall comprise two test methods, ..., which shall be carried out simultaneously ». On the test bench that was previously introduced, the following methods can be used in order to find refrigerant mass flow rate:

- direct measurement (methods D from ISO 917)
- indirect estimation:
 - heat balance on the cooler
 - heat balance on the compressor [2]

Figure 9 shows the gas cooler used as thermal flow meter.

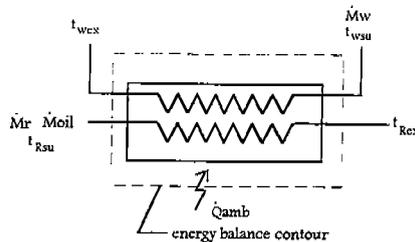


Figure 9: Gas cooler thermal flow meter

The energy balance equation is:

$$\frac{dU}{dt} = \dot{Q}_{amb} + \dot{M}_R (h_{Rsu} - h_{Re x}) + \dot{M}_{oil} c_{oil} (t_{Rsu} - t_{Re x}) + \dot{M}_w c_w (t_{wsu} - t_{wex})$$

In steady state regime, refrigerant mass flow rate can be found as:

$$\dot{M}_R = \frac{\dot{M}_{oil} c_{oil} (t_{Rsu} - t_{Re x}) + \dot{M}_w c_w (t_{wsu} - t_{wex}) + \dot{Q}_{amb}}{h_{Rsu} - h_{Re x}}$$

$$\dot{M}_R \cong \frac{\dot{M}_{oil} c_{oil} (t_{Rsu} - t_{Re x})}{\bar{c}_R (t_{Rsu} - t_{Re x})} + \frac{\dot{M}_w c_w (t_{wsu} - t_{wex}) + \dot{Q}_{amb}}{h_{Rsu} - h_{Re x}} \cong \frac{\dot{M}_{oil} c_{oil}}{\bar{c}_R} + \frac{\dot{M}_w c_w (t_{wsu} - t_{wex}) + \dot{Q}_{amb}}{h_{Rsu} - h_{Re x}}$$

If oil circulation is neglected the error made on refrigerant flow rate is given by:

$$\frac{\delta \dot{M}_R}{\dot{M}_R} = \frac{\dot{M}_{oil} c_{oil}}{\dot{M}_R \bar{c}_R}$$

ISO 917 allows up to 1.5% in mass oil circulation. For refrigerants with low specific heat in superheated state (R22 for example), the ratio $\frac{c_{oil}}{\bar{c}_R}$ can reach 3. If the gas cooler is to be used as thermal flow meter, the errors (due to oil circulation) in estimating refrigerant mass flow rate can reach 4.5%.

CONCLUSIONS

Test results showed that with the new layout is possible to cover a large domain of set points used for performance evaluation of compressors.

The advantages of the « fully gas cycle » method are:

- low energy consumption of the testing facility
- facility of control
- simple layout of the plant
- small refrigerant charge
- small inertia

The disadvantages of the method are:

- the suction temperature of the compressor has a minimum limit imposed either by the risk of condensation (if the high pressure gas cooler is used) or by the temperature of cold water (if the low pressure gas cooler is used)
- error analysis shows that even for low percentage of oil circulation (1.5%), the inaccuracy of gas cooler energy balance can reach important values (4.5%) if oil circulation is neglected. This disadvantage is specific to all methods using a gas cooler / heater as thermal flow meter.

Taking in account the above facts it can be concluded that the method is particularly suitable for long live tests and quality control of compressors. Due to the low charge of refrigerant, the method can be a useful tool in investigating compressors behaviour when using alternative refrigerants for which refrigerant charge of the plant is submitted to prescriptions (hydrocarbons for example).

REFERENCES

- [1] International Standard ISO917: « Testing of refrigerant compressors »
 [2] R. Dirlea, J. Hannay, J. Lebrun, K. Lopes and J. Pimenta: « Testing of refrigeration compressors » ; Proceedings of the 19th International Congress of Refrigeration, The Hague (The Netherlands), vol. IIIa, p 277, 1995