UDC 621.56/.59.001.4

Ref. No. : ISO/R 916-1968 (E)

ISO

INTERNATIONAL ORGANIZATION FOR STANDARDIZATION

ISO RECOMMENDATION
R 916

TESTING OF REFRIGERATING SYSTEMS

1st EDITION

December 1968

For each individual country the only valid standard is the national standard of that country.

Printed in Switzerland

Also issued in French and Russian. Copies to be obtained through the national standards organizations.

This page intentionally left blank

BRIEF HISTORY

The ISO Recommendation R 916, *Testing of refrigerating systems,* was drawn up by Technical Committee ISO/TC 86, *Refrigeration,* the Secretariat of which is held by the British Standards Institution (BSI).

Work on this subject was entrusted to Sub-Committee ISO/TC 86/SC3, the Secretariat of which is held by Belgium. The work began in 1960, and was carried out using as a basis for discussion the "Recommendations for an international code for refrigerating machines"*, published in November 1957 by the International Institute of Refrigeration. The work led to the adoption of a Draft ISO Recommendation.

In March 1967, this Draft ISO Recommendation (No. 1153) was circulated to all the ISO Member Bodies for enquiry. It was approved, subject to a few modifications of an editorial nature, by the following Member Bodies :

Australia	Germany	Sweden
Belgium	Greece	Switzerland
Canada	Hungary	U.A.R.
Chile	Italy	United Kingdom
Czechoslovakia	Netherlands	Yugoslavia
Denmark	New Zealand	
France	Poland	

No Member Body opposed the approval of the Draft.

The Draft ISO Recommendation was then submitted by correspondence to the ISO Council, which decided, in December 1968, to accept it as an ISO RECOMMENDATION.

---

* **Bulletin IIF – 177 Boulevard Malesherbes, Paris 17^{e} – Volume XXXVIII, No. 1.**

CONTENTS

ISO Recommendation | R 916 | December 1968

TESTING OF REFRIGERATING SYSTEMS

INTRODUCTION

This ISO Recommendation has for its object the determination of the technical performance of a refrigerating system, but not of the functional duty of a complete installation or of the performance of its individual components.

The term *refrigerating system* implies the conventional vapour compression type consisting of compressing, condensing and evaporating apparatus, together with the interconnecting piping, and the accessories necessary to complete the refrigerant circuit.

The determination of technical performance for other refrigerating systems such as, for example, absorption machines and ejector type machines is not provided for in this ISO Recommendation, but may be dealt with in other ISO Recommendations.

The only tests envisaged are those of complete refrigeration systems operating normally and under steady working conditions (frequency, voltage, water supply, etc.), and where the refrigerant is entirely in a liquid state at entry to the expansion valve.

The direction of the tests should be entrusted only to persons possessing the necessary technical knowledge and experience.

When none of the combined methods given in this ISO Recommendation is practicable or acceptable, it may be possible to restrict the test to a determination of the performance of the compressor only, in accordance with ISO Recommendation R 917, *Testing of refrigerant compressors.*

1. UNITS

Quantity	Symbol	International system (SI) units	Usual metric units	Usual non-metric units	Conversion factors
absolute temperature customary temperature	T, Θ t, θ	K	K °C	°R °F	t °C = T K – 273.15 t °F = T °R – 459.67
pressure	p	N/m²	kgf/cm²	lbf/in²	1 kgf/cm² = 98 066.5 N/m² 1 lbf/in² = 6894.76 N/m²
density (mass density)	ρ	kg/m³	kg/m³	lb/ft³	1 lb/ft³ = 16.0185 kg/m³
specific enthalpy	h	J/kg	$kcal_{IT}$/kg	Btu/lb	1 $kcal_{IT}$/kg = 4186.8 J/kg 1 Btu/lb = 2326 J/kg
specific entropy	s	J/(kg·K)	$kcal_{IT}$/(kg·K)	Btu/(lb·°R)	1 $kcal_{IT}$/(kg·K) = 4186.8 J/(kg·K) 1 Btu/(lb·°R) = 4186.8 J/(kg·K)
specific heat capacity	c	J/(kg·K)	$kcal_{IT}$/(kg·°C)	Btu/(lb·°F)	1 $kcal_{IT}$/(kg·°C) = 4186.8 J/(kg·K) 1 Btu/(lb·°F) = 4186.8 J/(kg·K)
specific latent heat of evaporation	l	J/kg	$kcal_{IT}$/kg	Btu/lb	1 $kcal_{IT}$/kg = 4186.8 J/kg 1 Btu/lb = 2326 J/kg
thermal conductivity	λ	W/(m·K)	$kcal_{IT}$/(h·m·°C)	Btu/(h·ft·°F)	1 $kcal_{IT}$/(h·m·°C) = 1.163 W/(m·K) 1 Btu/(h·ft·°F) = 1.730 73 W/(m·K)
surface coefficient of heat transfer	α	W/(m²·K)	$kcal_{IT}$/(h·m²·°C)	Btu/(h·ft²·°F)	1 $kcal_{IT}$/(h·m²·°C) = 1.163 W/(m²·K) 1 Btu/(h·ft²·°F) = 5.678 W/(m²·K)
overall coefficient of heat transfer	K	W/(m²·K)	$kcal_{IT}$/(h·m²·°C)	Btu/(h·ft²·°F)	1 $kcal_{IT}$/(h·m²·°C) = 1.163 W/(m²·K) 1 Btu/(h·ft²·°F) = 5.678 W/(m²·K)
kinematic viscosity	υ	m²/s	m²/s St	ft²/s	1 ft²/s = 0.092 903 0 m²/s 1 St = 0.0001 m²/s
mass flow rate	q_m	kg/s	kg/h	lb/h	1 lb/h = 126 × 10^{-6} kg/s
heat flow rate	Φ	W	$kcal_{IT}$/h	Btu/h	1 $kcal_{IT}$/h = 1.163 W 1 Btu/h = 0.2931 W
refrigerating capacity (overall, net, useful)	Φ_o	W	fg/h	ton	1 fg/h (= 1 $kcal_{15}$/h) = 1.163 W 1 ton of refrigeration (= a heat flow rate of 12 000 Btu/h removed by the refrigerating system from the cold body) = 3516.85 W
refrigerating performance (overall, net, useful)	ϵ	–	–	–	–
efficiency	η	–	–	–	–
power	P	W	kW ch	kW hp	1 ch = 735.499 W 1 hp = 745.700 W
area of an exchange surface	A	m²	m²	ft²	1 ft² = 0.092 903 0 m²
relative humidity	φ_p	–	–	–	–
specific humidity (mixture ratio)	x	–	–	–	–

It is recommended that the figures 1, 2, 3, etc. be used to indicate any state point of the refrigerant (see, for example, Fig. 1).

Inferior indexes

ambient atmosphere, air	a
water	w
heat transfer liquid (brine, alcohol, etc.)	f
refrigerant	(no index)
saturated	s

2. DEFINITIONS AND TEST DATA

2.1 Definitions

2.1.1 *Overall refrigerating capacity.* The rate at which heat is removed from external media by the refrigerant. The only heat quantities excluded from this refrigerating capacity are those which result from internal heat exchanges within the refrigerating circuit.

It should be noted that, in many cases, the overall refrigerating capacity can be obtained from the difference in specific enthalpy of the refrigerant entering the compressor and of the refrigerant leaving the condenser or the liquid sub-cooler, if any, multiplied by the mass flow of refrigerant circulated.

2.1.2 *Net refrigerating capacity.* The rate at which heat is removed by the refrigerant from the cooling medium which is used to transmit the refrigerating effect.

2.1.3 *Useful refrigerating capacity.* The rate at which heat is removed by the refrigerant or by the secondary cooling medium between two specific points, taking into account the conditions of utilisation.

2.2 Test data

2.2.1 One of the three refrigerating capacities defined under clause 2.1 should be stated.

2.2.2 In the case of refrigerating systems having several stages of evaporation and carrying out partial refrigerating duties, the intermediate temperatures should also be given.

2.2.3 In all cases, the following figures of consumption should be given :

(*a*) the intake of power (in terms of consumption of electricity, coal, steam, fuel oil, etc., together with the requisite data regarding characteristics);

(*b*) water for cooling, if used, together with full details of supply.

2.2.4 It is advised that the following operating details should be included in the data :

(*a*) the refrigerant used;

(*b*) the speed of rotation of the compressor;

(*c*) if applicable, the pressure of the refrigerant at compressor suction, at the condenser inlet and at the evaporator outlet;

(*d*) when the overall refrigerating capacity is specified (see clause 2.1.1), the conditions of the refrigerant at the expansion valve and at the entry of the compressor;

(*e*) when the net refrigerating capacity is given (see clause 2.1.2),

- either the temperature of the heat transfer medium at the entry and exit of the condenser and of the evaporator,
- or the temperature of the heat transfer medium, either entering or leaving the condenser and the evaporator, together with the corresponding rate of flow. Preference should be given to the following :

  (1) *for an evaporative condenser* : the inlet temperature of the water, the temperature of the air and the relative humidity of ambient air (generally the temperature at inlet);

  (2) *for an air cooled evaporator* : the inlet temperature of the air and, if appropriate, its relative humidity;

  (3) *for a brine circulation evporator* : the outlet temperature of the brine.

2.2.5 It is not necessary to assess the flow of the heat transfer medium in an evaporator when its temperature should be practically uniform around the evaporator, in a space or in a reservoir (e.g. a brine tank).

3. DETERMINATION OF PERFORMANCE

3.1 The determination of the technical performance required in the Introduction concerns the following data in particular :

3.1.1 The refrigerating capacity given in clause 2.2.1 above, which should be so chosen that it is capable of practical verification.

3.1.2 The corresponding consumption given in clause 2.2.3.

3.1.3 The conditions of operation given in clause 2.2.4.

3.2 The data should be capable of verification under the conditions of operation laid down for the test.

3.2 As the test conditions are subject in practice to temporary unclassifiable variations, it is advised that the data be set out in such a way that they are applicable throughout the specified period of the test.

3.3.1 It is therefore advisable that the data in clauses 2.2.1 to 2.2.3 should provide for varying conditions in the neighbourhood of the conditions of operation in clause 2.2.4, and especially for different values in the neighbourhood of the temperatures given. For ease of interpolation, and in order to avoid adjustment by calculation, these values may be presented graphically, within the limits of fluctuation, for each pair of temperatures specified. Maximum permissible deviations should be laid down.

3.3.2 So far as the influence of temporary variations in other operating conditions is concerned, this should be the subject of an agreement between the interested parties.

4. ORGANIZATION OF TESTS

4.1 The tests refer exclusively to refrigerating plant operating under steady working conditions (see Introduction).

4.2 Preliminary tests for adjustment to specified conditions should be carried out before the official test is started. After this, only agreed adjustments should be made during the actual test period.

4.3 The tests should be made under the conditions defined in clause 4.4, which should be as close as possible to the working conditions.

4.4 The stability of operation (steady condition) should preferably be checked by plotting successive measurements over a sufficiently long time interval and until the initial and final states are the same for all quantities essential to the verification of the data.

4.5 Readings showing an excessive variation from the mean should be disregarded.

4.6 The number of readings used for a calculation should be at least ten. The readings should be regularly spaced at maximum intervals of 20 minutes.

4.7 All measurements should be made in conformity with international rules which may be in force, or, failing this, in conformity with the national rules accepted by those concerned. All measuring instruments should have been tested and certified for the purpose of the test.

4.8 The refrigerating system should be provided with the necessary thermometer and pressure gauge connections. These should be of a type suitable for the purpose for which they are to be used, so as to avoid errors in measurement (frosting, longitudinal heat flow along pipes, etc.).

4.9 All equipment required exclusively for these tests should in no way interfere with normal operation or accessibility.

4.10 It is advisable that a sightglass should be provided upstream of the expansion valve to serve as a means of determining the level of the refrigerant. Furthermore, it is necessary to ascertain that the plant has been purged before testing, and that the entrainment of lubricating oils is not excessive.

4.11 It is recommended that, wherever possible, two simultaneous tests should be made, with particular reference for the second test to the indirect methods described in clause 5.2.

If it is only possible to make use of one test method, two consecutive tests should be made, except in the case of a contrary agreement between the interested parties.

4.12 Attention is drawn to the causes of inaccuracy in measurements of liquid or vapour flow by calibrated flow-meters (pulsation in pipelines, oil entrainment, impurities in the circuit).

5. MEASUREMENT OF REFRIGERATING CAPACITY

5.1 Direct methods

5.1.1 *Overall refrigerating capacity.* When the refrigerant vapour in circulation is dry saturated or superheated at the compressor inlet, i.e. without liquid entrained or in suspension, the overall refrigerating effect can be calculated by the equation :

$$\Phi_o = q_m (h_1 - h_5) \qquad \ldots\ldots \quad (1)$$

State 1 is the state at the inlet flange of the compressor, and state 5 is the state at the outlet from the sub-cooler (to be exact, the inlet flange on the expansion valve or the inlet flange of the internal heat exchanger on the high pressure side, as shown in Fig. 1 or 2).

Specific enthalpies for the more common refrigerants are given in the tables and diagrams referred to in clause 10.1.

The measurement of the mass flow rate of refrigerant in the low pressure circuit should be made either by heat balance (see clause 5.1.1.1) or by calibrated flow-meter (see clause 5.1.1.2).

5.1.1.1 MEASUREMENT BY HEAT BALANCE. In principle, the mass flow rate can be found from the heat balance of any apparatus in the circuit, provided that the same flow passes through it. If any of the refrigerant has been bled off previously into subsidiary circuits, this quantity should be taken into account.

(*a*) For single stage installations, the apparatus most suitable for establishing a heat balance is the condenser, when this is arranged for cooling by a liquid without evaporation. The flow rate is then given by the equation :

$$q_{\mathrm{m}} = \frac{q_{\mathrm{m\,w}}\, c_{\mathrm{w}} \Delta t_{\mathrm{w}} + \Phi_{\mathrm{c}}}{\Delta h} \qquad \ldots\ldots \quad (2)$$

where the inferior index w refers to the cooling liquid (in general, water).

Δh represents the drop in specific enthalpy or the refrigerant in passing through the condenser.

The mass flow rate $q_{\mathrm{m\,w}}$ of the liquid is obtained by one of the methods in common use for the measurement of flow (calibrated tanks, orifices, etc.).

The heat flow Φ_{c} is a corrective term which should be employed whenever the temperature of the external surface of the apparatus is different from the ambient temperature. This correction is given by the formula :

$$\Phi_{\mathrm{c}} = K\,A\,(t_{\mathrm{m}} - t_{\mathrm{a}}) \qquad \ldots\ldots \quad (3)$$

where

- K is the overall coefficient of heat transfer between the fluid circulating in the external passage of the apparatus and the surrounding atmosphere; as Φ_{c} is merely a corrective term, it will be sufficiently accurate to use the approximate value $K = 7\ \mathrm{W/(m^2 \cdot K)}$ [$K \approx 6\ \mathrm{kcal_{IT}/h \cdot m^2 \cdot {}^\circ C}$] when the apparatus is not insulated;

- A is the surface area of the apparatus in contact with the surrounding atmosphere;
- t_{m} is the mean temperature of the external surface, taken for this corrective term to be the temperature of the fluid in the part of the circulation system immediately adjacent to it;
- t_{a} is the ambient temperature.

The corrective term Φ_{c}, positive or negative, as the case may be, should be small relative to the other terms in the heat balance since its determination is only approximate. In this case, it should be decided, according to the tolerance laid down in clause 8.4.1, whether it is necessary to insulate the apparatus in order to reduce the value of this term still further.

If so, the value of K will be determined by the approximate formula for flat plates, which is as follows :

$$\frac{1}{K} = \frac{1}{\alpha} + \frac{e}{\lambda} \qquad \ldots\ldots \quad (4)$$

where, according to the units selected (see definition for K, below equation (3)),

$$\alpha = 7\ \mathrm{W/(m^2 \cdot K)} \quad \text{or} \quad \alpha = 6\ \mathrm{kcal_{IT}/(h \cdot m^2 \cdot {}^\circ C)}$$

and e and λ represent respectively the thickness of the insulation and its coefficient of thermal conductivity under the prevailing conditions.

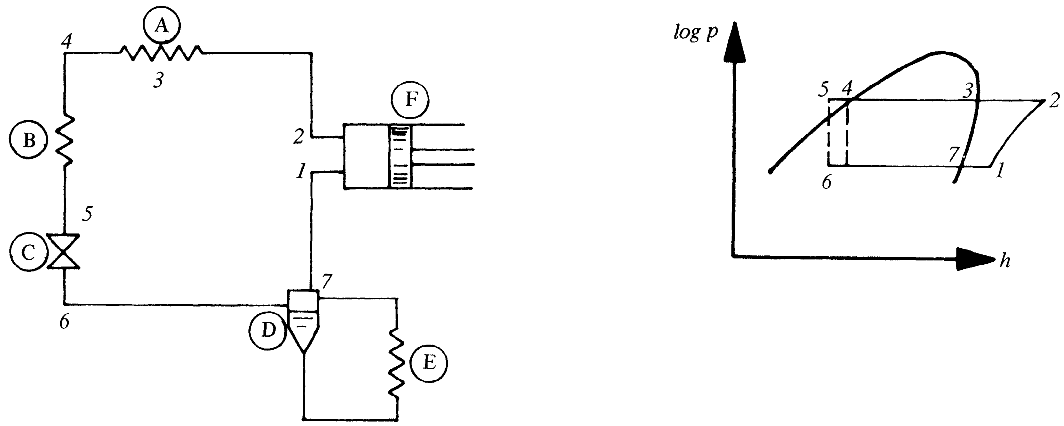
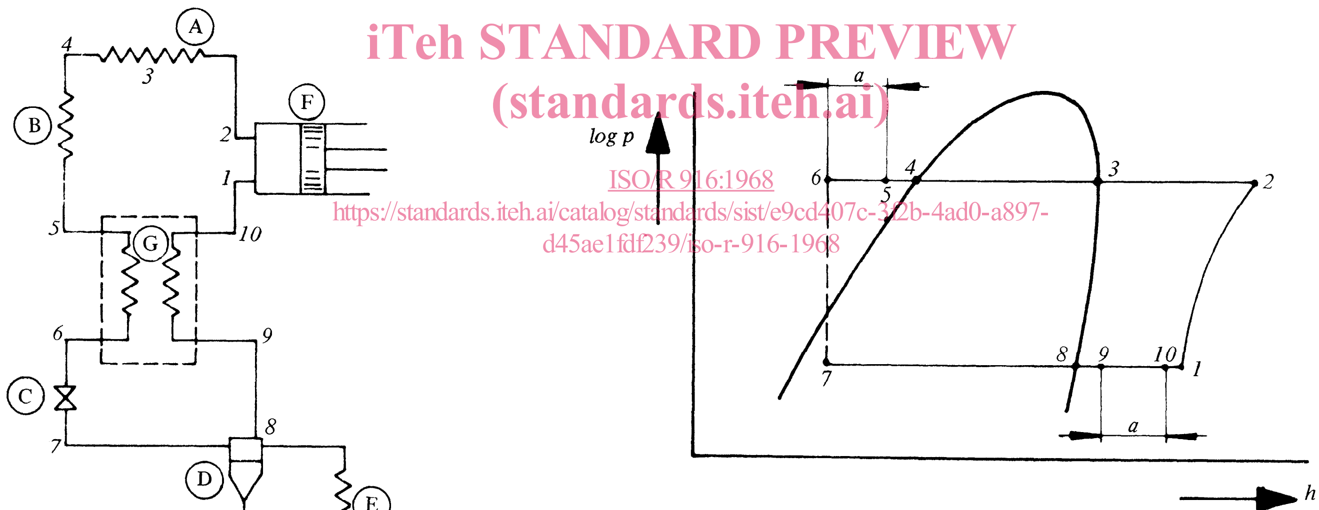


FIG. 1



Internal heat transfer : $\| \Delta h_{5-6} \approx \Delta h_{9-10} \|$

FIG. 2

(A) Condenser

(B) Sub-cooler

(C) Expansion valve

(D) Separator

(E) Evaporator

(F) Compressor

(G) Internal heat exchanger