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R 916

TESTING OF REFRIGERATING SYSTEMS

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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.

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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)),

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

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

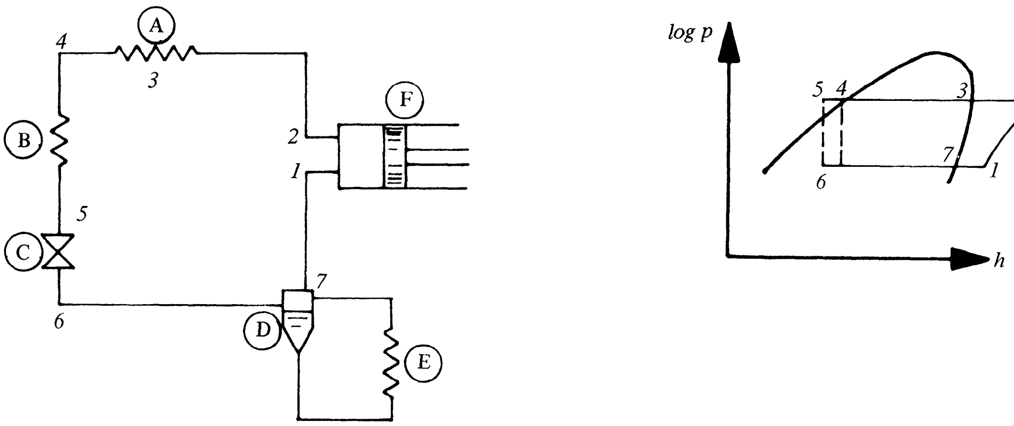
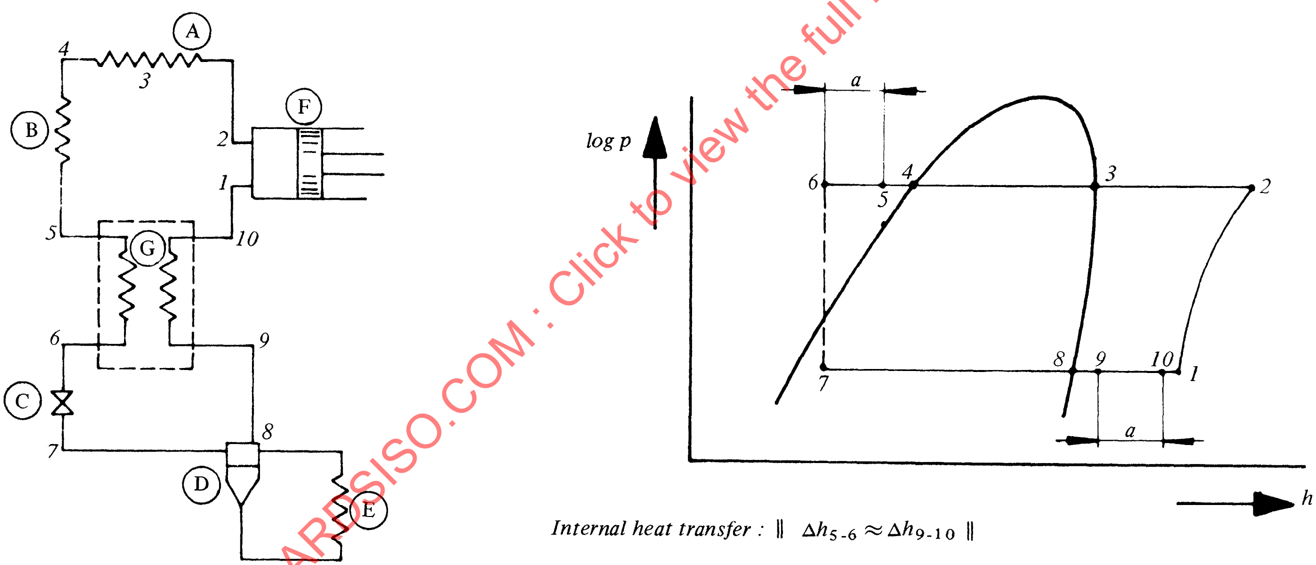


FIG. 1



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

FIG. 2

(A) Condenser

(B) Sub-cooler

(C) Expansion valve

(D) Separator

(E) Evaporator

(F) Compressor

(G) Internal heat exchanger

(*b*) If the condenser is followed by a sub-cooler, the heat balance should preferably be effected on the two pieces of apparatus taken together.

(*c*) For atmospheric condensers, the effects of evaporation often render the establishment of the heat balance difficult, but it is sometimes possible to reduce this difficulty by proceeding as follows. When the refrigerant is free from impurities, mass flow rate should be found from some other apparatus, generally the sub-cooler by itself. It is therefore recommended that provision should be made on the sub-cooler for taking the necessary temperature readings. For greater accuracy the flow of cooling liquid should be limited so that there is a minimum temperature difference of 3 °C between the inlet and outlet of the sub-cooler. This method is not recommended, however, unless the refrigerant is free from impurities.

Attention is drawn to the fact that this method requires the use of apparatus of the accuracy required in clause 8.4.1.

5.1.1.2 MEASUREMENT BY CALIBRATED FLOW-METER. The mass flow rate of the refrigerant can also be measured by tested and approved flow-meters, arranged either at a point where the refrigerant is entirely in the liquid state, or where it is entirely in the form of superheated vapour in the case where antipulsating devices are provided or where there are no pulsations.

Measurement of flow of superheated vapour is generally to be preferred because the influence of certain impurities is less marked than in the liquid phase. In any case, the conditions laid down in the standard rules in force for such devices should be satisfied.

5.1.2 *Net refrigerating capacity*

5.1.2.1 SECONDARY COOLING MEDIUM (LIQUID). Two methods of measurement are described below.

5.1.2.1.1 The first method of measurement is based on the use of the formula :

$$\Phi_{o} = q_{mf}\, c_{f}\, (\Delta t_{f}) + \Phi_{c} \qquad \ldots\ldots \quad (5)$$

where the inferior index f refers to the secondary cooling medium.

(*a*) The mass flow q_{mf} should be measured by one of the methods in common use either at the inlet or the outlet of the evaporator. The use of a calibrated tank is recommended for the measurement of small flows, and that of a volumetric flow-meter for large flows.

(*b*) The values of specific heat c_f for secondary cooling media in common use are given in the publications referred to in clause 10.1.

(*c*) The fall in temperature Δt_f of the secondary cooling medium between the inlet and the outlet of the evaporator should be at least 3 °C. This method is therefore not applicable when such a temperature range is incompatible with the specified conditions or when it cannot be realized.

(*d*) The corrective term Φ_c is usually small and can therefore be determined approximately, bearing in mind, however, the tolerances laid down in clause 8.4.1. This corrective term comprises

– the heat equivalent of the power absorbed by any auxiliaries which may be located between the points of measurement in the secondary cooling circuit (circulating pumps, agitators, etc.);

– a term Φ_c' which should be applied whenever the secondary cooling medium in the evaporator is not completely insulated from the surrounding atmosphere. This term can be calculated by the relation

$$\Phi_c' = K A (t_a - t_m)$$

where

K is the overall coefficient of heat transfer between the surrounding atmosphere and the secondary cooling medium; this coefficient can be calculated by the use of equation (4);

A is the area of the external surface of the evaporator exposed to the surrounding atmosphere;

t_m is a mean temperature which is

– the arithmetic mean of the inlet and outlet temperatures of the secondary cooling medium for forced circulation apparatus (counter-current apparatus, parallel flow apparatus, etc.);

– the outlet temperature for brine tanks with suitable agitation.

It should be noted that the term Φ_c' refers only to the thermal effect of the surrounding atmosphere on the secondary refrigerant which is being cooled, and not to the thermal effect of the atmosphere on the refrigerant. When the latter effect occurs, as for instance in apparatus in which the refrigerant is in contact with the outer walls of the external apparatus, it should not be taken into account in calculating the net refrigerating capacity, in accordance with the definition in clause 2.1.2.

5.1.2.1.2 Another method, namely the calorimeter method, involves the replacement of the normal heat loading of the evaporator by an alternative measurable heat source which is capable of maintaining the steady working conditions required.

(*a*) If steam is used to provide the alternative heat source, the condensate should be weighed. To ensure accuracy, it is recommended that the steam should be slightly superheated before coming into contact with the evaporator, and that the condensate should be sub-cooled below its saturation temperature. Care should be taken to avoid freezing of the condensate, and the formation of pockets of water in the steam coil should be avoided by arranging the coil with a continuous slope.

(*b*) Hot water, a special brine, or electrical heating, can also be used for the heat source.

5.1.2.2 SECONDARY COOLING MEDIUM (GASEOUS). In this case, measurement of the net refrigerating capacity necessitates determination of the flow and of the hygrometric state of the gas or air cooling medium, a fact which makes accurate measurement somewhat difficult.

However, for certain coolers with forced circulation of air (or gas), measurements which are acceptable and compatible with the tolerances laid down in clause 8.4.1 are sometimes possible.

The net refrigerating capacity can then be calculated, using the total enthalpy change H, by the following equation :

$$\Phi_o = H_1 - H_2 + \Phi_c$$

$$= q_{m_1} h_1 - (q_{m_2} h_2 + q_{m\ liq} h_{liq} + q_{m\ sol} h_{sol}) + \Phi_c \qquad \ldots\ldots \quad (6)$$

where, in the inlet section of the evaporator,

$$H_1 = h_1 q_{m_1} = h_{1\,dry} q_{m_1\,dry}$$

and similarly, in the outlet section of the evaporator,

$$H_2 = h_2 q_{m_2} = h_{2\ dry} q_{m_2\ dry}$$

depending on whether the specific enthalpy of the secondary cooling medium is used, or the enthalpy related to a kilogramme of dry gas of this medium. This flow may also be measured using the standard rules.

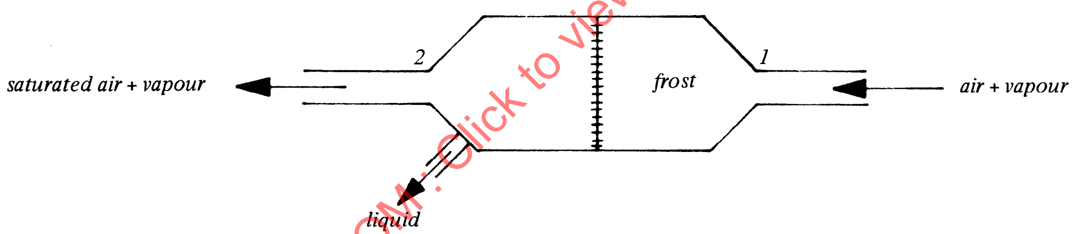


FIG. 3

Equation (6) can also be used in the following form :

$$\Phi_o = q_{m_2}(h_1 - h_2) + q_{m\ liq}(h_1 - h_{liq}) + q_{m\ sol}(h_1 - h_{sol}) + \Phi_c \qquad \ldots\ldots \quad (7)$$

account being taken of the constancy of mass flow rate

$$q_{m_1} = q_{m_2} + q_{m\ liq} + q_{m\ sol}$$

The enthalpies $h_1, h_{1\ dry}, h_2, h_{2\ dry}$, are determined by calculation or by means of a psychrometric chart, from measurement of temperature and of relative humidity in each uniform section of the vein, at the inlet and at the outlet.

If the air is saturated, the difference of enthalpy can be determined from the isothermal lines in the fog zone of the same diagram.

If the fan is located between the two points of measurement, the capacity measured is the useful refrigerating capacity. The net refrigerating capacity can be obtained by adding the thermal equivalent of the power input to the fan.

The corrective term Φ_c is determined as indicated in clause 5.1.2.1.1 (d) for the correction Φ_c'.

5.1.3 *Useful refrigerating capacity.* The useful refrigerating capacity, when it is in the form of a quantitative measurement between two specified points in the secondary cooling medium circuit, can be ascertained by the methods described in clause 5.1.2 for net refrigerating capacity, but taking into account any corrections necessitated by the conditions of utilization.

The qualitative characteristics of utilization form no part of this ISO Recommendation in its present form.

5.2 Indirect methods

It is recommended that indirect methods should be used

– when direct methods are impractical or less precise than indirect methods;

– as a means of verification of tests by direct methods (see clause 4.11).

5.2.1 *Calibrated compressor method.* By this is understood a test of the capacity of the compressor itself, usually carried out in the maker's works, under conditions (especially evaporating and condensing temperatures) corresponding to the conditions of utilization.

This procedure is frequently employed by reason of its convenience. It is actually a direct method for testing the compressor as an isolated component, as prescribed in ISO Recommendation R 917, *Testing of refrigerant compressors.* It is, however, included here among the indirect methods because this ISO Recommendation in its present form is concerned principally with tests of complete systems (see Introduction).

5.2.2 *Methods based on the measurement of the overall refrigerating capacity.* It is also possible to derive the net refrigerating capacity (see clause 5.1.2) from a measurement of the overall refrigerating capacity (see clause 5.1.1) whenever means exist for ascertaining the insulation losses between the expansion valve and the compressor suction.

5.2.3 *Method based on the overall heat balance of the refrigerating system.* This method is classed as an indirect method as it does provide means of checking the primary direct tests, and it should be carried out whenever possible.

If substantial differences should appear, the reasons for these should be investigated.

In the case of a refrigerating system working on direct expansion and having a condenser cooled by water, without evaporation of the water, the following equation for the heat balance applies :

$$\Phi_o = \Phi_I + \Phi_{II} + \Phi_{III} - P + \Phi_{IV} \qquad \ldots\ldots \quad (8)$$

or, in the case of a single stage of compression,

$$\Phi_o = \frac{h_1 - h_5}{h_2 - h_1} (P - \Phi_{II} - \Phi_{IV}) \qquad \ldots\ldots \quad (9)$$

All the terms of equations (8) and (9) should be expressed in the same system of units.

In the above formulae,

- Φ_o is the overall refrigerating capacity;
- Φ_I is the rate of heat rejection to the water and to the surrounding air in the condenser and the sub-cooler;
- Φ_{II} is the rate of heat rejection to the cooling water in the cylinder jackets and, when applicable, in the inter-coolers (multi-stage plants) and in the auxiliaries;
- Φ_{III} is the rate of heat loss from hot delivery pipes;
- P is the power input to the shaft of the compressor;
- Φ_{IV} is the rate of heat dissipation from the compressor, excluding the heat allowed for in Φ_{III};

inferior index 1 refers to compressor inlet conditions;

inferior index 2 refers to compressor outlet conditions;

inferior index 5 refers to the outlet of the sub-cooler or, if there is no sub-cooler, of the condenser.

The expression $\dfrac{\Phi_o}{h_1 - h_5}$ is equal to the refrigerant mean flow rate q_m.

The value of Φ_{III} can be arrived at by taking the enthalpy of the refrigerant at each end of the pipe system and measuring the mass flow rate of refrigerant. The rate of flow can be ascertained by the method of clause 5.1.1.1.

Since Φ_{II}, Φ_{IV} and Φ_{III} are only corrective terms, their determination can be approximate.