International Standard ISO 7547

INTERNATIONAL ORGANIZATION FOR STANDARDIZATION•МЕЖДУНАРОДНАЯ ОРГАНИЗАЦИЯ ПО СТАНДАРТИЗАЦИИ•ORGANISATION INTERNATIONALE DE NORMALISATION

Air-conditioning and ventilation of accommodation spaces on board ships — Design conditions and basis of calculations

Conditionnement d'air et ventilation des emménagements à bord des navires — Conditions de conception et bases de calcul

First edition — 1985-11-01

UDC 629.123.132 : 629.1.06 **Ref. No. ISO 7547-1985 (E)**

Descriptors : shipbuilding, ships, air conditioning, ventilation, design, specifications.

ISO 7547-1985 (E)

Price based on 8 pages

Foreword

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Draft International Standards adopted by the technical committees are circulated to the member bodies for approval before their acceptance as International Standards by the ISO Council. They are approved in accordance with ISO procedures requiring at least 75 % approval by the member bodies voting.

International Standard ISO 7547 was prepared by Technical Committee ISO/TC 8, *Shipbuilding and marine structures.*

Users should note that all International Standards undergo revision from time to time and that any reference made herein to any other International Standard implies its latest edition, unless otherwise stated.

Air-conditioning and ventilation of accommodation spaces on board ships — Design conditions and basis of calculations

1 Scope and field of application

This International Standard specifies design conditions and suitable methods of calculation for air-conditioning and ventilation of accommodation spaces and the radio cabin on board seagoing merchant ships for all conditions except those encountered in extremely cold or hot climates (i.e. with a lower or higher enthalpy than that stated in 4.2 and 4.3).

Annex A provides guidance and details of good practice in the design of ventilation and air-conditioning systems in ships.

Annex B gives the thermal conductivities of commonly used construction materials.

NOTE — Users of this International Standard should note that, while observing the requirements of the Standard, they should at the same time ensure compliance with such statutory requirements, rules and regulations as may be applicable to the individual ship concerned.

2 References

ISO 31/4, *Quantities and units of heat.*

ISO 3258, *Air distribution and air diffusion — Vocabulary.*

3 Definitions

For the purposes of this International Standard, the definitions given below, together with those in ISO 31/4 and ISO 3258, apply:

3.1 accommodation spaces: Spaces used as public rooms, cabins, offices, hospitals, cinemas, games and hobby rooms, hair-dressing saloons and pantries containing no cooking appliances.

3.2 air-conditioning: Form of air treatment whereby temperature, humidity, ventilation and air cleanliness are all controlled within limits prescribed for the enclosure to be air-conditioned.

3.3 ventilation: Provision of air to an enclosed space, sufficient for the needs of the occupants or the process.

3.4 relative humidity: In humid air, the ratio, expressed as a percentage, of the water vapour actual pressure to the saturated vapour pressure at the same dry bulb temperature.

3.5 dry bulb temperature: The temperature indicated by a dry temperature-sensing element (such as the bulb of a mercury-in-glass thermometer) shielded from the effects of radiation.

4 Design conditions

4.1 General

The system shall be designed for the indoor air conditions specified in 4.2 and 4.3 in all accommodation spaces defined in 3.1 at the stated outdoor air conditions and the outdoor supply airflow, ventilation, and air balance given in 6.2.1, 6.2.2 and 6.5 respectively.

NOTE — All temperatures stated are dry bulb temperatures.

4.2 Summer temperatures and humidities

Outdoor air: + 35 °C and 70 % relative humidity

Indoor air: + 27 °C and 50 % relative humidity

NOTE — In practice, the indoor air conditions obtained, especially humidity, can be different from those stated.

4.3 Winter temperatures

Outdoor air: − 20 °C

Indoor air: + 22 °C

NOTE — This International Standard does not specify requirements for humidification in winter.

4.4 Outdoor air

The minimum quantity of outdoor air shall be not less than 50 % of the total air supplied to the spaces concerned.

4.5 Occupancy

The number of persons to be allowed for in the various accommodation spaces shall be as follows, unless otherwise stated by the purchaser:

Cabins: the maximum number of persons for which the cabin was designed.

Saloons, mess- or dining-rooms and recreation rooms: the number of persons who can be seated or, when this is not specified by the purchaser:

- 1 person/2 m^2 floor area for saloons;
- 1 person/1,5 m^2 floor area for mess- or dining-rooms;
- 1 person/5 m^2 floor area for recreation rooms.

Captain's and chief engineer's day room: 4 persons.

Chief officer's, first engineer's, chief steward's and other private day rooms: 3 persons.

Hospital: the number of beds plus 2.

Gymnasium, games room: 4 persons.

First aid room: 2 persons.

Offices: 2 persons.

5 Calculation of heat gains and losses

5.1 Applicability

For the calculation of summer conditions, 5.2 to 5.5 inclusive shall apply.

For the calculation of winter conditions, 5.2 only shall apply.

5.2 Heat transmission

5.2.1 Method of calculation

The following formula shall be used for calculating the transmission losses or gains Φ, in watts, for each separate surface.

$$\Phi = \Delta T\left[(k_v \cdot A_v) + (k_g \cdot A_g)\right]$$

where

ΔT is the difference in air temperature, in kelvins (for the difference of air temperature between air-conditioned and non-air-conditioned internal spaces, see 5.2.2);

k_v is the total heat transfer coefficient in watts per square metre kelvin, for the surface A_v (see 5.2.3);

A_v is the surface, in square metres, excluding side scuttles and rectangular windows (glazing + 200 mm) (see figures 1 and 2);

k_g is the total heat transfer coefficient in watts per square metre kelvin, for the surface A_g (see 5.2.3);

A_g is the area, in square metres, of side scuttles and rectangular windows (glazing + 200 mm) (see figures 1 and 2).

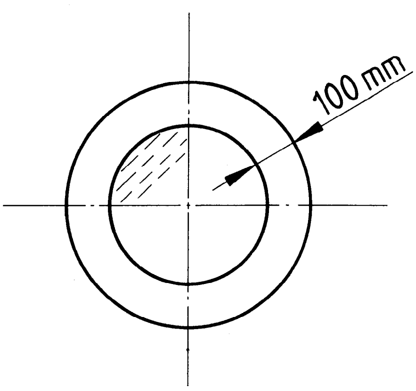

Figure 1 — Side scuttles

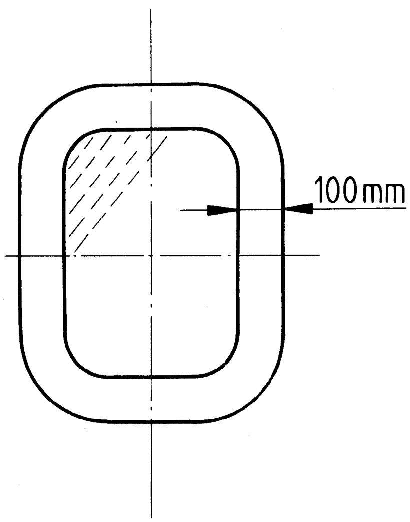

Figure 2 — Rectangular windows

5.2.2 Temperature differences between adjoining internal spaces

For differences of air temperature, ΔT in kelvins, between air-conditioned and non-air-conditioned internal spaces, see table 1.

Table 1 — Temperature differences between adjoining internal spaces

Deck or bulkhead	ΔT, K	
	Summer	**Winter**
Deck against tank provided with heating	43	17
Deck and bulkhead against boiler room	28	
Deck and bulkhead against engine room and against non-air-conditioned galley	18	
Deck and bulkhead against non-heated tanks, cargo spaces and equivalent	13	42
Deck and bulkhead against laundry	11	17
Deck and bulkhead against public sanitary space	6	0
Deck and bulkhead against private sanitary space		
a) with any part against exposed external surface	2	0
b) not exposed	1	0
c) with any part against engine/boiler room	6	0
Bulkhead against alleyway	2	5

NOTE — It is understood that means of heating are provided in exposed sanitary spaces.

5.2.3 Total heat transfer coefficients

The values for the total heat transfer coefficients, k, in watts per square metre kelvin given in table 2 assume that adequate thermal insulation is provided on all surfaces exposed to outdoor conditions or adjoining hot or cold spaces, or hot equipment or pipework.

The values given in table 2 shall be used where appropriate, unless otherwise advised by the purchaser. For other cases a method of calculation of coefficient is given in 5.2.4.

5.2.4 Calculation of heat transfer coefficient

The heat transfer coefficient shall be calculated as follows:

$$\frac{1}{k} = \sum \frac{1}{\alpha} + \frac{\sum \frac{d}{\lambda} + M_L + M_b}{\mu}$$

where

k is the total heat transfer coefficient in watts per square metre kelvin;

α is the coefficient of heat transfer for surface air, in watts per square metre kelvin:

α = 80 W/(m²·K) for outer surface exposed to wind (20 m/s);

8 W/(m²·K) for inside surface not exposed to wind (0,5 m/s);

d is the thickness of material, in metres;

λ is the thermal conductivity, in watts per metre kelvin;

M_L is the thermal insulance for an air gap, in square metres kelvin per watt;

M_b is the thermal insulance between different layers of material, in square metres kelvin per watt;

μ is a correction factor for steel structure:

μ = 1,2 for insulation in accordance with figure 3;

1,45 for insulation in accordance with figure 4.

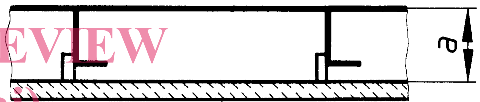

Figure 3 — Plane insulation of uniform thickness

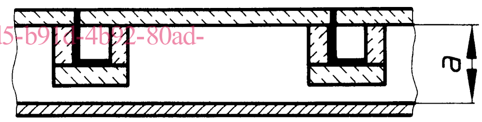

Figure 4 — Corrugated insulation of uniform thickness

Table 2 — Total heat transfer coefficients

Surfaces	Total heat transfer coefficient, k W/(m²·K)
Weather deck not exposed to sun's radiation and ship side and external bulkheads	0,9
Deck and bulkhead against engine room, cargo space or other non-air-conditioned spaces	0,8
Deck and bulkhead against boiler room or boiler in engine room	0,7
Deck against open air or weather deck exposed to sun's radiation and deck against hot tanks	0,6
Side scuttles and rectangular windows, single glazing	6,5
Side scuttles and rectangular windows, double glazing	3,5
Bulkhead against alleyway, non-sound reducing	2,5
Bulkhead against alleyway, sound reducing	0,9

Guidance on values of thermal conductivities of commonly used materials is given in annex B.

For thermal insulance, M_L, of non-ventilated air gaps, see table 3.

Table 3 — Thermal insulance of non-ventilated air gap

Boundary surfaces of air gap	Air gap thickness, a 1) mm	Thermal insulance 2) $m^2 \cdot K/W$
Both surfaces having high emissivity	5 20 200	0,11 0,15 0,16
One surface having high emissivity, other surface low emissivity	5 20 200	0,17 0,43 0,47
Both surfaces having low emissivity	5 20 200	0,18 0,47 0,51
High emissivity surfaces in contact 3)	0	0,09

1) See figures 3 and 4.

2) The term "thermal insulance" is used according to the definition given in ISO 31/4. In many countries this term is known as "thermal resistance" with a symbol R.

3) Aluminium foil and other polished surfaces are assumed to have low emissivity (0,2). All other surfaces are assumed to have high emissivity (0,9).

5.2.5 Measurement of transmission areas

The transmission area for bulkheads, deck and ship side shall be measured from steel to steel.

5.3 Solar heat gain

Solar heat gain, Φ_s, is calculated, in watts, as follows:

$$\Phi_s = \Sigma(A_v \cdot k \cdot \Delta T_r) + \Sigma(A_g \cdot G_s)$$

where

A_v is the surface exposed to solar radiation in square metres (side scuttles and rectangular windows are not included);

k is the total heat transfer coefficient in accordance with 5.2.3 or 5.2.4 for a ship structure (deck, outer bulkhead, etc.) within the surface A_v;

ΔT_r is the excess temperature (above the outside temperature of + 35 °C) caused by solar radiation on surfaces as below:

ΔT_r = 16 K for vertical light surfaces;

29 K for vertical dark surfaces;

12 K for horizontal light surfaces;

32 K for horizontal dark surfaces;

A_g is the glass surface (clear opening) exposed to solar radiation in square metres;

G_s is the heat gain per square metre from glass surfaces:

G_s = 350 W/m² for clear glass surfaces;

240 W/m² for clear glass surfaces with interior shading.

For corner cabins the surface which gives the highest Φ_s shall be chosen for calculation of the heat gain.

Surfaces not included in A_v because of shadow from overhanging deck or other means of sun protection shall be calculated at a sun angle of 45°.

NOTES

1 If solar radiation reflecting glass is used, G_s may be reduced.

2 The excess temperatures for vertical and horizontal surfaces and the additional heat gain from glass surfaces caused by solar radiation are based on the most extreme average temperatures in subtropical climate and give the "worst-condition" occurring during a day.

5.4 Heat gain from persons

Values of sensible and latent heat emitted by a person at an indoor temperature of 27 °C are given in table 4.

Table 4 — Body activity and heat emission

Activity	Type of heat	Emission W	
Seated at rest	Sensible heat	55	135
	Latent heat	80	
Medium/heavy work	Sensible heat	140	390
	Latent heat	250	

5.5 Heat gain from lighting and other sources

In spaces with daylight, additional heat gain from lighting shall be ignored.

In spaces without daylight, the heat gain from lighting shall be calculated from the rated wattage of the lighting, as advised by the purchaser or as specified by the appropriate authority. Where the rated output is not specified by the purchaser or the appropriate authority, the heat gain from general lighting shall be taken as stated in table 5, with consideration given to special lighting requirements.

Table 5 — Heat gain from general lighting

Space	Heat gain from general lighting W/m²	
	Incandescent	Fluorescent
Cabins, etc.	15	8
Mess- or dining-rooms	20	10
Gymnasiums, etc.	40	20

Refrigerator output shall be taken as 0,3 W/l storage capacity, unless otherwise specified by the purchaser.

Other sources of heat gain, such as from appliances that are in operation for considerable periods during the day, shall only be taken into consideration if specified by the purchaser.

Temporary electrical appliances such as radio and television sets, hot water urns, etc. shall be ignored.

The heat gain from equipment, etc. in the radio cabin shall be taken as 2,5 kW, unless otherwise specified by the purchaser.

Heat gain from fans shall be taken to give a rise in the temperature of the air of 1 °C/kPa pressure rise.

The rise in the temperature of the air in ducts shall be limited to + 2 °C.

6 Airflow calculation

6.1 Volume of space

Volume of furniture, wardrobes, stationary equipment, etc. shall not be deducted in calculating the gross volume of cabins and other spaces.

6.2 Supply airflow

6.2.1 Air supply for air-conditioning

The air supply to each air-conditioned space shall be calculated using whichever of the following criteria gives the highest value:

a) airflow to maintain the conditions of 4.2;

b) airflow to maintain the conditions of 4.3;

c) outdoor supply airflow not less than 0,008 m^3/s per person for which the space is designed.

The air supply to cabins with a private sanitary room (bath, shower or W.C.) shall be at least 10 % higher than the exhaust air from the sanitary room.

NOTE — It is to be observed that there are national regulations specifying a minimum number of air changes.

6.2.2 Air supply for ventilation

Supply of conditioned air to ventilated spaces, such as those listed in a) to e), shall be provided directly or by transfer of less vitiated air from an adjacent space, and shall be sufficient to permit the exhaust airflow requirements of 6.4 to be achieved:

a) public sanitary rooms (bath, shower, urinal or W.C.);

b) laundry;

c) drying and ironing rooms;

d) changing rooms;

e) cleaning lockers.

NOTE — It is assumed that supplementary means of heating are provided for ventilated spaces where necessary.

6.3 Temperature of supply airflow

The temperature of the air supplied to the space shall not be more than 10 °C lower than the average temperature nor, for the heating mode, more than 23 °C higher than the average temperature of the space.

6.4 Exhaust airflow

6.4.1 Volume of airflow

The exhaust airflow in saloons, mess- and dining-rooms and common day rooms shall be the same as the supply airflow.

The exhaust airflow in hospitals and pantries shall be at least 20 % more than the supply airflow.

The exhaust airflow in private sanitary rooms (bath, shower or W.C.) shall be 0,02 m^3/s or a minimum of 10 air changes per hour, whichever gives the highest value.

The exhaust airflow in common sanitary rooms (bath, shower, urinal or W.C.), laundries, and drying and ironing rooms shall be a minimum of 15 air changes per hour and in changing rooms, wash rooms and cleaning lockers a minimum of 10 air changes per hour.

Public sanitary rooms in passenger ships including ferries shall be given special consideration. The exhaust airflow shall be a minimum of 30 air changes per hour.

6.4.2 Exhaust system

The exhaust air from the spaces listed in a) and b) below shall be fed directly to the open air, and not used for recirculation. Additionally, the exhaust systems for each of these spaces or group of spaces shall be separate from each other:

a) hospitals;

b) sanitary rooms, laundry, pantry, etc.

6.5 Air balance

The system must be positively balanced. It shall be applicable on every deck.

In rooms where there are tumbler dryer(s), the balance between supply and exhaust air shall be taken into account in consultation with the manufacturer.

Hospitals and pantries shall be maintained at a slightly lower pressure than that in the adjoining accommodation.

Annex A

Guidance and good practice

(This annex does not form an integral part of the Standard.)

A.1 System and ducting

The layout of the plant and the duct sizes should allow air supply without recirculation.

A.2 Supply air

In hospitals, a non-return flap should be installed in the supply air duct.

A.3 Exhaust air

In laundries and drying and ironing rooms, exhaust air devices should be installed over areas with high heat emission and high humidity.

A.4 Air movement in the occupied areas

The air movement in the occupied areas should be within limits shown in figure 5.

Air velocity for the upper value is applicable only in spaces where people are active.

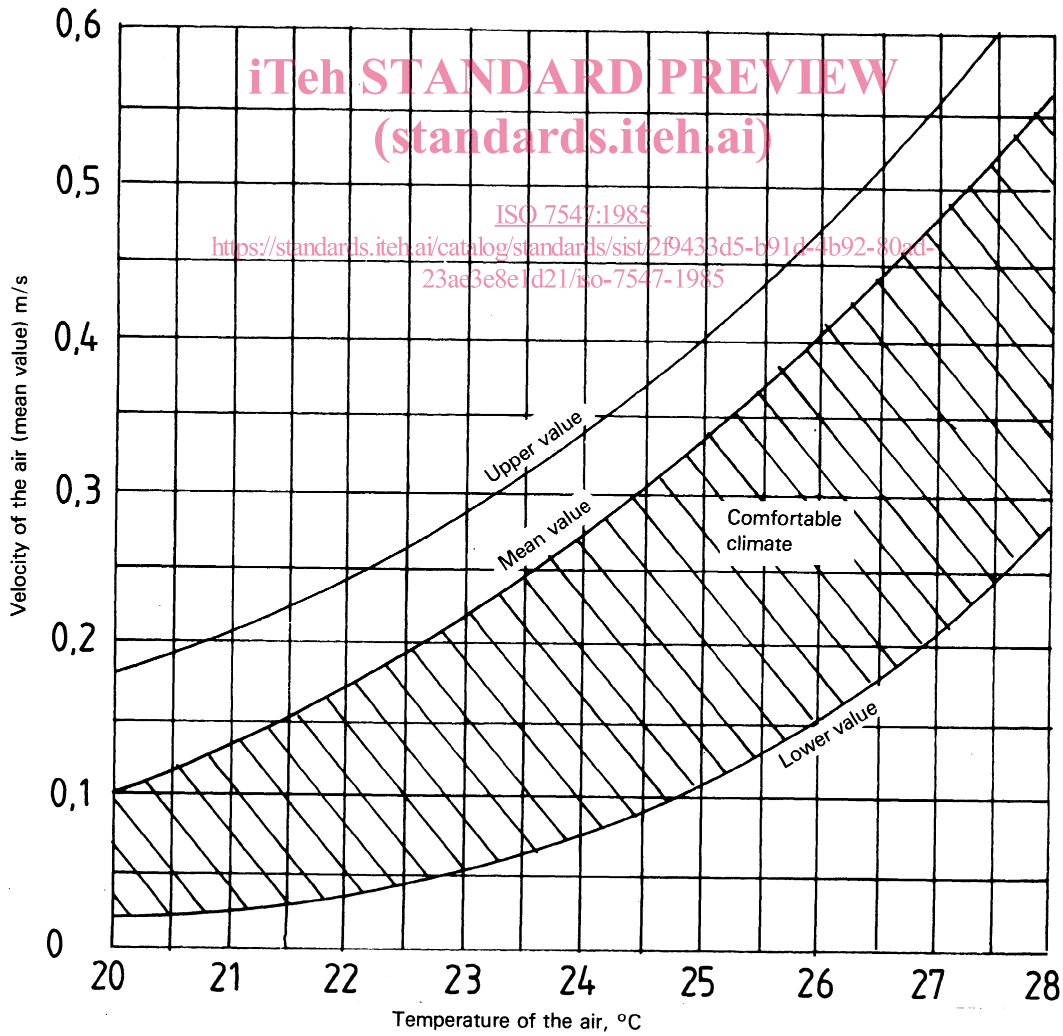

Figure 5 — Air movement in occupied areas

NOTE — For normal applications for human comfort, the occupied areas are geometrically limited to 0,15 m from all room surfaces with a height of 1,80 m above the floor.

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A.5 Temperature variation in the occupied areas

The maximum difference in temperature between any points within the occupied areas (see A.4) should not exceed 2 K.

A.6 Refrigerating machinery

For a sea-water system, the size of the condenser should be based on an inlet water temperature of + 33 °C, and the compressor motor should be capable of driving the compressor at an inlet water temperature of + 35 °C.

For an indirect cooling system the condenser should be designed for + 36 °C on inlet cooling water and the compressor motor for + 38 °C.

When calculating the total heat transfer coefficient of the condenser, a fouling factor of 0,000 09 $m^2 \cdot K/W$ should be used.

When calculating the cooling effect, a specific mass of air of 1,20 kg/m^3 should be used.

A.7 Sound

The system should be so designed that the A-weighted sound pressure level from the air distributing system measured 1 m from the air terminal device should not exceed 55 dB (A).

A.8 Temperature control

Individual temperature control should be fitted to each accommodation space. This can be obtained by controlling the air flow.

A.9 Humidification during winter

With humidification during the winter, it is strongly recommended that the upper level of humidification be limited to 35 % relative humidity and that the humidification be so controlled that it only takes place during long periods of cold and dry weather. The risk of condensation on cold surfaces and thereby the risk of formation of ice in the insulation should be taken into consideration. Where insulation is fitted an surfaces exposed to the atmosphere, care should be taken to ensure a complete vapour seal, to avoid penetration of warm humid air.