

Table 1 Heat Gain from People (for Marine Air Conditioning Loads)

Degree of Activity	Heat Rate (BTU/Hr)		
	Sensible	Latent	Total
Dancing	245	605	850
Persons Eating (Mess Rooms and Dining Rooms)	220	330	550
Waiters	300	700	1000
Moderate Activity (Lounge, ship's office, chart rooms, etc.)	200	250	450
Light Activity (Staterooms, crew's berthing, etc.)	195	205	400

effect of outdoor air reduces the operating load of the refrigeration plant.

Load Determination

The cooling load estimate for air conditioning consists of the usual factors, as discussed in Chapter 26 of the 1961 GUIDE AND DATA BOOK, including:

1. Solar radiation.
2. Heat transmission through decks and bulkheads.
3. Heat dissipations of occupants.
4. Heat gain due to lights.
5. Ventilation air.
6. Motors or other electrical heat producing equipment.

The heating load estimate for air conditioning should consist of the following:

1. Heat losses through decks and bulkheads.
2. Ventilation air.

No allowances are made for heat gain from warmer adjacent spaces.

Heat Transmission Coefficients. The overall heat transmission coefficient, U , between the conditioned space and the outside of the boundary in question depends upon the construction, material and insulation employed.

The composite structures common to shipboard construction do not lend themselves to theoretical derivation of such coefficients. Therefore, they are most commonly obtained from full scale panel tests.

Heat Dissipation from People. The rates at which heat and moisture are dissipated from people depend upon their state of activity. Table 1 gives values that can be used at 80 F room dry bulb.

Heat Load Calculations. Example 1 illustrates a detailed load calculation for a typical marine air-conditioning system.

Example 1: Determine the total load for air conditioning the officers' mess and lounge of a new cargo ship with the following design conditions.

Location	Design Temp., F		Total Heat, Btu/lb	Latent Heat, Gr/lb	Effective Temp.
	Dry-bulb	Wet-bulb			
Outdoor	95	82	46.0	146	88
Indoor	80	67	21.7	78	74

Volume = 3900 cu ft

People = 28, plus 2 waiters = 30

Lights = 1560 watts

Transmission Sources, Areas, and Conditions

Deck over = 520 sq ft, exposed to conditioned area.

Deck under = 40 sq ft, exposed to 85 F (no insulation). Remainder is exposed to conditioned area.

Outboard bulkhead = exposed to conditioned area.
Forward bulkhead = 221 sq ft, exposed to 120 F (2 in. insulation).
Aft bulkhead = 136 sq ft, exposed to 98 F (2 in. insulation).
Aft bulkhead = 51 sq ft, exposed to 98 F (no insulation).
Inboard bulkhead = 145 sq ft, exposed to 85 F (no insulation).
Glass = 45 sq ft

Solution: The heat transmission is tabulated for convenience.

Surface	Area, sq ft	Temp. Diff., F deg	U	Transmission, Btu/hr
Deck under	40	3	0.50	60
Forward bulkhead	221	38	0.23	1930
Aft bulkhead	136	16	0.23	500
Aft bulkhead	51	16	0.40	326
Inboard bulkhead	145	3	0.30	131
Glass	45 X 95 Btu/(hr)(sq ft)			4275

Total Transmission Load

The sensible and latent heat loads are calculated as follows:

Sensible Space Load		Btu/h
Transmission	=	7222
People = 28 X 220 Btu/h per person	=	6160
2 X 300 Btu/h per person	=	600
Lights = 1560 X 3.41 X 0.75 (use factor)	=	4000
		17,967

Sensible Outdoor Air Load

Minimum outdoor air = 15 cfm per person

30 people X 15 = 450 cfm

Air quantity required (based on a

temperature difference of 30 F deg) = 17,967 = 555 cfm

For 100 percent outdoor air, the sensible load is

555 X 1.08(95-80) = 9000 Btu/h

Total Sensible Heat Load = 17,967 + 9000 = 26,967 Btu/h

Latent Space Load

People = 28 X 330 Btu/h per person = 9240

= 2 X 700 Btu/h per person = 1400

10,650

Latent Outdoor Air Load = 555(0.67)(146-78) = 25,300 Btu/h

Total Latent Heat Load = 10,650 + 25,300 = 35,950 Btu/h

Total Heat Load = 26,967 + 35,930 = 62,917 Btu/h

= 5.25 tons of refrigeration

EQUIPMENT SELECTION

The principal equipment required for an air-conditioning system can be divided into four broad categories: (1) the air handling portion consisting of fans, filters, central heating and cooling coils and sound treatment; (2) the distribution network including air ductwork, water and steam piping as required; (3) the terminal treatment consisting of heating and cooling coils, terminal mixing units and diffusing outlets; and (4) the refrigeration equipment.

Factors to be considered in the selection of equipment are:

1. Installed first cost.
2. Space available in fan rooms, passageways, machinery rooms and staterooms.
3. Operating costs including system maintenance.
4. Noise levels.
5. Weight.

The wider use of unitary central equipment has reduced required fan room floor space, reduced operating noise levels and installation costs. Fans must be selected for stable performance over their full range of system operation and should be provided with adequate isolation to prevent transmission of vibrations to the deck. Effective sound treatment is es-

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sential since fan rooms are often located adjacent and near living quarters.

In general, construction of all equipment is considerably heavier than required for land applications and must be capable of withstanding the corrosive environment of the salt air. Materials such as stainless steel, bronze alloys, steel hot dip galvanized after fabrication, etc., are used extensively.

Centrifugal refrigeration machinery using Refrigerants 11 and 114 for air conditioning loads in excess of 150 tons usually proves most economical. For the majority of cases where the load is under 150 tons, one or more reciprocating machines are employed. The compressors, in general, must be capable of operating satisfactorily under shipboard list, trim, pitch and roll conditions and be constructed for marine service. The use of direct-connected reciprocating compressors has become standard practice due primarily to savings in space over belt-driven units. Hermetic centrifugal machines offer a similar advantage over open compressors and usually result in a substantially lower first cost. To date, these have only been used to a limited extent.

Steam-jet refrigeration has been used on foreign passenger liners, but has never been applied to any American ships, except a few naval vessels. The absorption machine designed for chilled water service, utilizing either low pressure or high pressure steam, with lithium bromide as the absorbent and water as the refrigerant, offers many interesting possibilities for shipboard application, particularly where very quiet operation is necessary on military vessels.

With regard to the refrigerant employed on merchant ships, Refrigerant 12 is most widely used, due to its availability in all parts of the world. However, Refrigerant 22, used extensively on land applications, is now more readily available at ports around the world. An increased use of this refrigerant should be expected because of the following advantages:

1. Since Refrigerant 22 has a lower specific volume than Refrigerant 12, the same compressor has approximately twice the capacity of Refrigerant 22. This permits savings in compressor costs, space, and weight.
2. The size of refrigerant lines and valves will be reduced.
3. Liquid suction heat exchangers commonly used with Refrigerant 12 systems to increase the thermodynamic efficiency of the cycle will not be required with Refrigerant 22.

Sheet Metal Ducts

Sheet metal ductwork on merchant ships has been constructed to various gages of steel and aluminum alloys. For low pressure ductwork, hot dipped galvanized, copper bearing steel has been used most extensively. Where protected, the minimum thickness of the material for non-watertight ducts is usually determined by the width or diameter of ducts according to the following:

Size	Steel
Less than 6 in.	22 USSG (0.0313 in.)
6-12 in.	20 USSG (0.0375 in.)
12 1/4-24 in.	18 USSG (0.050 in.)
24 1/4 in. and over	16 USSG (0.0625 in.)

It is important that the fabrication of the ductwork and transitions sections be such that all seams are airtight.

With the increased use of high velocity systems on merchant ships, round, small diameter, prefabricated pipe and fittings are being employed. One of the effective ways of minimizing air leakage for these high pressure systems is the use of an economical pressure-sealed clamp for joining fittings to air piping.

The principal advantage of the round, small diameter duct-

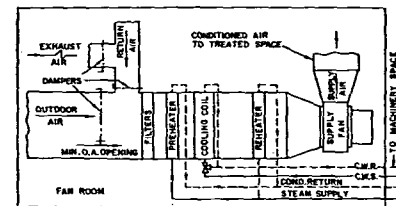


Fig. 1 Single Zone Central (Type A) System

work is that it occupies a fraction of the space required for the larger sheet metal duct. Saving in weight also is an important consideration.

TYPICAL SYSTEMS

Comfort air-conditioning systems installed on shipboard are classified as:

- (1) those serving passenger staterooms, (2) those serving crew's quarters and similar small spaces, and (3) those serving public spaces.

A brief description of the types of systems used is given in the following sections.

Single-Zone Central System

Public spaces are treated as one zone and can effectively be handled by this simple arrangement. Exceptionally large spaces may require two systems. Central station systems of either the built-up type or factory-assembled fan-coil type are generally employed for large public spaces. Fig. 1 is a schematic diagram of a typical system, which also is known as the Type A system. Under certain conditions it is found desirable to use all outdoor air, avoid the use of return ducts, and penalize the refrigeration plant accordingly. Outdoor air and return air are both filtered before being preheated, cooled and reheated as required at the central station unit. A room thermostat maintains the desired temperature by modulating the valve regulating the flow of steam to the reheat coil. During mild weather the dampers frequently are arranged to automatically admit 100 percent outdoor air.

Multi-Zone Central System

This system (Fig. 2) is also known as the Type C system. Spaces are divided into zones in accordance with similarity of loads and exposures. Each zone has a reheat coil to supply air at temperature adequate for all spaces served. A thermostat regulates the amount of steam or hot water circulated to the reheat coil. Manual control of air volume is the only means for occupant control of conditions. Filtering is same as for Type A system.

The internal sensible heat load components of the spaces, particularly solar and lights, in any one zone are subject to large variance. Also, the methods used for thermostatic control of reheaters cannot compensate for these large variations. Therefore, it is impossible for this system to always satisfy individual space requirements. Also, volume control is conducive to noise, drafts and odors.

Terminal Reheat System

The terminal reheat system is commonly known as the Type D system. A diagram of this system is shown in Fig. 3. In this