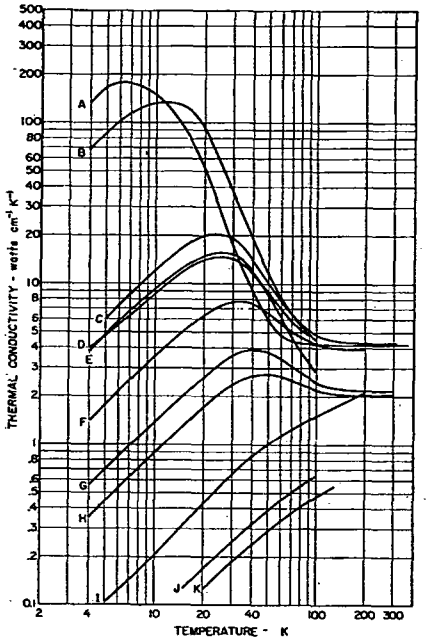
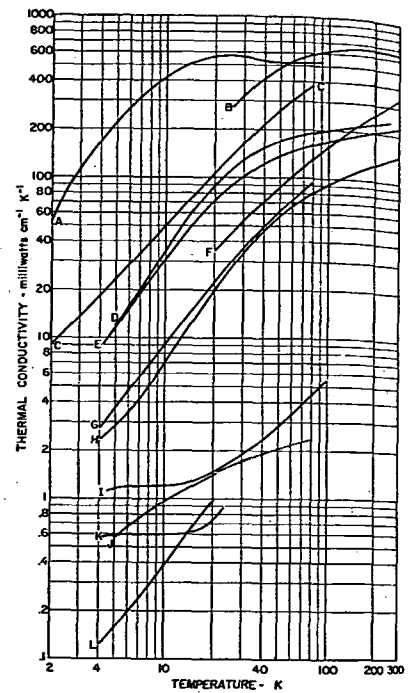


A—Silver 99.999% pure (16); B—high purity copper (30); C—coldrawn copper (30); D—copper, electrolytic tough pitch (30); E—aluminum single crystal (31); F—free-machining tellurium copper (30); G—aluminum, 1100 F (31); H—aluminum, 6063-T5 (31); I—copper, phosphorus deoxidized (30); J—aluminum, 2024-T4 (31); K—free-machining leaded brass (30).

Fig. 11 .... Low Temperatures Thermal Conductivities of Some Good Conductors



A—50-50 lead-tin solder (28); B—steel SAE 1020 (16); C—beryllium copper (32); D—constantan (16); E—Monel (16); F—silicon bronze (30); G—Inconel (16); H—type 347 stainless steel (16); I—fused quartz (16); J—polytetrafluoroethylene (Teflon) (29); K—polymethylmethacrylate (Perspex) (16); L—nylon (28).

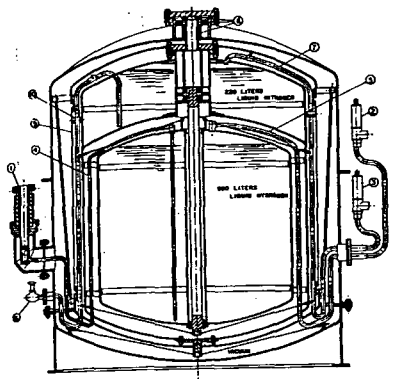
Fig. 12 .... Thermal Conductivity of Some Poor Conductors

pressing the efficiency of the energy exchange between the gas and the solid. For heat transfer between extended surfaces this quantity increases with decreasing temperature and may be taken as unity for all temperatures below the critical temperature of the gas. Of the common gases, only hydrogen and helium will have accommodation coefficients appreciably less than unity at cryogenic temperatures (see Table 3). Fig. 10 gives gaseous heat conduction for several typical gases. The formula for α applies to parallel plates, coaxial cylinders, or concentric spheres.

Conductivity of Supports

The heat conduction through supports and other connections between the warm and cold shells can become an appreciable percentage of the total if careful consideration is not given to their design and a good choice of material made. To minimize heat flow a material of high strength and low thermal conductivity is desired. Many other factors such as a tendency to outgas, brittleness, etc., most also be considered in making a final choice. Table 4 lists a number of materials suitable for use at low temperatures. A comparison of the net relative heat conduction can be made by dividing the yield strength by the average thermal conductivity value given. Figs. 11 and 12 show the variation of thermal conductivity with temperatures for a number of materials.

There are various typical laboratory size Dewar vessels. Fig. 13 shows a semi-portable supply type vessel for liquid hydrogen or helium.



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1 Liquid H_2 Valve
2 H_2 Relief Valve
3 H_2 Relief Valve
4 Liquid H_2 Fill & Empty
5 H_2 Vent Line
6 Liquid N_2 Fill & Empty
7 N_2 Vent Line
8 Multiple-Contact Insulating Supports
9 Radiation Shield
10 Teflon Bumper
X Aluminum to Stainless Soldered Connections

Fig. 13 .... A 500 Liter Liquid Hydrogen (or Helium) Supply Dewar Vessel

The thermal loss to the hydrogen container will boil away about 1 liquid liter per day. The radiation shield loss is about 15 liters of liquid nitrogen per day.

Multiple Layer Insulation

This relatively new type of insulation for cryogenic applications has yielded the lowest overall heat conduction into Dewar type vessels and is often referred to as super-insulation. It is the application of many floating radiation shields closely spaced but thermally separated by a poor conducting fiber in the evacuated space for great reduction in the radiation heat leak. This insulation was first developed by P. Petersen[1] of the Swedish Technical Research Council in 1951 and has been further developed and improved for cryogenic application by many others.[2,3] When properly applied this insulation has an apparent mean thermal conductivity between 540 R and 36 R of 2 to 4 $\times 10^{-6}$ Btu/(hr) (square foot) (R deg/ft).

For a theoretical evaluation of the effectiveness of multiple radiation shields the following equation may be used:

$$Q = \frac{E\sigma A}{n+1}(T_2^4 - T_1^4) \qquad (2)$$

where

σ = the Stefan-Boltzmann constant.
A = is the area.
E = the effective emissivity.
n = the number of floating radiation shields.
T_1 and T_2 = the absolute temperatures of the hot and cold boundaries.

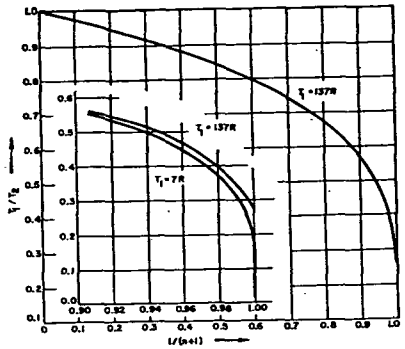


Fig. 14 .... Temperature Distribution Through a System of n Floating Radiation Shields.

(Legends: I denotes any shield between 1 and n, T = 540 R is the warm boundary temperature, and T_1 is the cold boundary temperature.)

Fig. 14 shows the theoretical temperature distribution for radiation shields separated by a vacuum between parallel walls at 540 R and 137 R and parallel walls of 540 R and 7 R. The curves were obtained from the relation:

$$T_i = \left[T_2^4 - \frac{i}{n+1}(T_2^4 - T_1^4)\right]^{1/4} \qquad (3)$$

where

T_i = temperature of the ith shield.
i = number of the ith shield counting from the hot wall.
n = total number of shields in the system.

To arrive at an apparent mean thermal conductivity (k) for multi-layer insulations, however, the well-known Fourier equation is used:

$$Q_T = \frac{kA(T_2 - T_1)}{t} \qquad (4)$$

where

t = thickness of the insulation.

There are a number of choices of reflective foils and fiber mats but the lowest thermal conductivity values have been achieved with aluminum foil separated by glass fiber paper. Table 5 gives a number of values of k reported by Kropschot[2] for a number of material combinations and conditions.

Gaps between adjacent shields at corners or joints can add significantly to the total heat transport. Therefore, design considerations and techniques of application are of utmost importance. For this reason, a cylindrical shape is to be preferred over a spherical shape. In addition, since an interstitial gas pressure lower than 10^{-4} mm Hg is required to attain low conductivities, the insulation must be installed clean and dry. A small super-insulated Dewar vessel for liquefied gases in 175 liter size is obtainable. The loss of liquid is about 1 liter nitrogen or 3 liters of hydrogen per day.