Chapter 2
Heat Losses

THE COMPUTATION OF HEAT REQUIREMENTS FOR BUILDINGS

Extracted from the I.H.V.E. GUIDE, 1959 Edition

SECTION II

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THE INSTITUTION OF HEATING AND VENTILATING ENGINEERS

THE COMPUTATION OF HEAT REQUIREMENTS FOR BUILDINGS TOGETHER WITH THERMAL CONDUCTIVITY AND RESISTIVITY TABLES FOR BUILDING MATERIALS

Being an extract from
the Guide to Current Practice

1959 Edition

SECTION II

Chapter 13

HEATING AND VENTILATING DESIGN

13.1 HEAT LOSSES

The discovery that heat was a form of energy was not made until 1798 when Rumford carried out his celebrated experiment on the boring of a cannon. In the concluding part of his Essay, Rumford writes:[(37)]

> "... these computations show how large a quantity of Heat might be produced, by proper mechanical contrivance, merely by the strength of a horse... . But no circumstances can be imagined in which this method of procuring Heat would not be disadvantageous; for more Heat might be obtained by using the fodder necessary for the support of a horse as fuel."

Joule's first direct determination of the mechanical equivalent of heat was made in 1843, and he succeeded thereby in establishing this fact. He obtained a value of 770 ft-lb/Btu at 60°F. Gradual refinements in experimental technique have given us more accurate values, but these differ only slightly from Joule's original value. The accepted figure is now 778 ft-lb/Btu. This figure is of course the basis for computing the heat gains to buildings from machinery and electrical apparatus.

Before Tredgold's time, the quantity of heat (or rather, the area of heating surface) necessary to warm a room was proportioned to the volume of the room, for example, 1 ft^2 of steam pipe per 200 ft^3, or 1 ft^3 of boiler per 2000 ft^3. Tredgold (1824) realised that there could be no universal ratio, and that heat requirements must depend on the structure, window area and ventilation. These requirements, he said[(64)] could always be measured in terms of a volume of air to be heated from the outdoor to the indoor temperature. His calculations were probably the first with any pretence to a rational scientific basis.

He starts from observations of the rate of cooling of a water filled vertical cylinder in air.* A cylinder contains c in^3 of water at a temperature of 180°F; it is allowed to cool in air in a room t°F (Fig. 13.1). The rate of cooling is observed to be n°F/min at the mean water temperature of T°F. The cooling effect of

*Although steam heating was largely used at that time, Tredgold thought the difficulty of observing the cooling of a steam pipe to be too great, and he turned to water instead.

the cylinder material is defined to be

$$\varepsilon = \frac{c\,n}{1728} \quad ft^3\ {}^oF/min$$

ε is thus the number of ^oF through which 1 ft^3 of water (or 2850 ft^3 of air)* would be cooled in 1 minute.

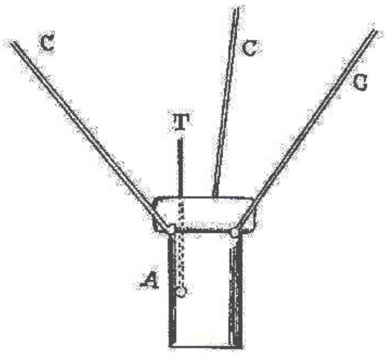

Fig. 13.1. Tredgold's heat loss experiment (1824).

Tredgold postulated that the cooling effect was proportional to the area of the cylinder s and to the mean temperature difference, i.e.

$$\varepsilon = A.s(T-t)$$

where A is a constant depending on the material. He found

for glass	A = 0.000644
cast iron	0.000738
tin plate	0.00041

(Péclet stated that Tredgold overestimated the heat loss, due to inadequate experimental precautions, though the ratios between various surface finishes were nearly correct. Clément's were better, but still larger than Péclet's own determinations from the condensation of steam in pipes.)

Tredgold assumed that this rate of heat loss would apply to the case of a window, and to the heat loss from pipes. He deduced that 1 ft^2 of window glass will cool $1\frac{1}{2}$ ft^3/min of air. By this device, Tredgold was able to express heat loss through glass in terms of the equivalent in fresh air supply.

The total heat loss from a room was then set down as the sum of the necessary ventilation, the infiltration through windows and doors, and the window losses in air equivalent. For the first, the fresh air supply per person was put at 4 ft^3/min; the infiltration was calculated by Tredgold to be 11 ft^3/min per window or door; and the window loss was equated to $1\frac{1}{2}$ ft^3/min per ft^2 of glass, say, a total of V ft^3/min.

He used the same formula $\varepsilon = A.s.(T - t_i)$ to work out the area of heating pipe needed, and to calculate the hourly quantity of steam needed, viz., 0.34 lb/h per ft^2 of heating pipe.

*This is Tredgold's figure. A more accurate value is 3225 ft^3 of air.

To estimate the hourly and annual fuel consumption, Tredgold uses the experimental result that to supply sufficient steam to 182 ft^2 of pipe in air at $60^\circ F$ would require the firing of 0.1 bushel* of coal per hour. If the internal temperature is to be $56^\circ F$, then in London heating is needed on 220 days a year, and the mean outdoor temperature during this period is $40^\circ F$ — an average temperature difference of 16°. The quantity of steam needed during the heating season is found to be

$$Q = 95\ V \text{ lb.}$$

and this requires the burning of 0.153 V bushels of coal.

This consumption refers to continuous heating. For intermittent operation, Tredgold proposed to assume some appropriate preheating period to be added to the duration of occupation to give the daily total hours of use. The fuel consumption would be proportional to this total.

He did in fact attempt to compute the preheating period, though his argument was unsound. He illustrated this by reference to a church of 100000 ft^3 volume, for which the heat loss equivalent was 4200 ft^3/min. With this rate of heat input, the time taken to heat the air from cold to the desired indoor temperature would then be 100000/4200 = 24 min. He also applied his theory to the ventilation of glasshouses to prevent overheating in summer.

Tredgold was dismayed that his rational approach was not adopted by his contemporaries. In a letter to Francis Bramah on December 31st 1827, he wrote:

> "I know there are few capable of discarding the idea of proportioning the heat to the space, instead of the cooling surface of glass — but if John Bull does not so readily comprehend this, his descendant Marcus Bull has got the right end of the story, and plumes himself not a little on the subject. Marcus built a place on purpose to experiment in.** He consulted all the wise heads of the Yankee colleges, and in fact did the thing in grand style. Faith, these spirited fellows will be giving us the go-by some of these days if we do not return to a more sober and frugal system."

In 1855, Hood wrote:(30)

> "No experiments on cooling are extant, that appear to be suitable to the present purpose (i.e. to the determination of heating surface), except some that were made by Tredgold, and these are erroneous in the applications he has made of them."

It is not, however, very clear why Hood discredited Tredgold's application, for his own approach was very similar. The only correction Hood made was to calculate the water equivalent of the vessel at its mean temperature, instead of the maximum temperature as Tredgold did.

Hood describes his own experiments on the rate of cooling of water in a 3-in o.d. horizontal cast iron pipe. He also studied the effect of surface finish on the rate of cooling. He deduced that the quantity of heat emitted by one foot of 4 in pipe at $\Delta T = 140^\circ F$ was sufficient to heat 222 ft^3 of air at the rate of $1^\circ F$/min (the temperature difference being $1^\circ F$). This figure is equivalent to an

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*An English measure of capacity. 1 bushel = 8 Imp. gallons, or 36.4 litres.

**According to Dufton, the experimental chamber built by Bull in America was for the purpose of comparing the performance of different stoves and fireplaces.

emission of 222 x 0.02 x 60 = 266 Btu/h. ft run; and may be compared with the presently accepted figure of *ca*. 400 Btu/h for a 4-in o.d. pipe.

In modern terms, Hood's results gave the following values for the surface coefficients:

rusty iron	1.64 Btu/ft^2h^oF
varnished iron	1.59
white painted iron	1.55
glass	1.59

Hood went on to show that the heat loss from a pipe was proportional to the square of the air velocity over it.

It was originally assumed that the losses through the walls, floor and roof were all negligible or zero. Although the fallacy of this assumption must have been suspected, no mention of the possibility of losses through these parts of a structure is made by either Tredgold or Hood. Tredgold's estimate of the losses through glass have already been given, and Hood too attempted to make a rough estimate of the loss of heat through glazed surfaces which he called the cooling effect of glass. He found that 1 ft^2 of glass would cool 1.279 ft^3 of air at the rate of 1^oF/min, when the air temperature on the two sides of the glass differed by one degree. (This corresponds to a thermal transmittance of 1.54 Btu/ft^2h^oF.) The experiments were conducted in still air; but Hood thought the data were generally applicable since high winds are not concurrent with very low outdoor temperature.

The procedure, then, in the mid-19th century, was to work out the total area of glass and evaluate the volume of air which this glass would cool. To this was added the requirement for ventilation and/or leakage; and the area of heating surface or length of pipe calculated. It was perhaps fortunate that the transmission through glass was over-estimated, thus affording some sort of correction for the neglect of other structural losses.

The design outdoor temperature was chosen to correspond with the usage of the building. In the case of buildings for daytime use only, 25^oF was recommended; but where buildings were to be heated both by night and by day, as for a forcing-house for instance, "perhaps 10^oF will not be too low to calculate from". On very exposed sites, a temperature of 0^oF was recommended by Hood.

What is possibly the first attempt at a guide to current practice is given by Hood, when he suggests values for the heating surface required for various types of building:

Building	Inside temperature ^oF	Heating surface (ft of 4-in pipe per 1000 ft^3)
Church	55	5
Large public rooms	55	5
Dwelling rooms	65	10
-do-	70	12-14
Halls, shops, waiting rooms	55-60	7-8
Factories and workrooms	50-55	5-6
Greenhouses	55	35
Graperies	65-70	45
Pineries and hothouses	80	55

HEAT-LOSS CALCULATIONS

The following example is given for the benefit of those who are not familiar with heat-loss calculations.

The heat loss from a single-storey factory is to be determined. Suppose the building is 200 ft × 100 ft × 12 ft at eaves and 20 ft at ridge. North-light construction. Walls of 9 in brick, unplastered ; 10 windows down each long side, 50 sq ft each ; floor, concrete on hardcore and earth ; roof, corrugated iron lined with insulating board to meet requirements of the Thermal Insulation (Industrial Buildings) Act, 1957 (see Section XV) ; area of corrugated sheeting, say, 20 000 sq ft ; area of roof glass, say, 10 000 sq ft.

Air Change.

The mean height of the factory is 16 ft. From page 43 the rate of air change required if occupied to the legal limit and 1500 cu ft of air per person allowed is 2 changes per hour. The rate of infiltration is given (page 42) as between $1\frac{1}{2}$ and 2 changes per hour, hence, the rate of air change required on the basis of 1500 cu ft per hour per person is adopted, i.e. 2 changes per hour.

For purposes of heat-loss calculations the specific heat of air is taken at 0·02 B.t.u./cu ft. deg. F.

Structure Losses.

It will be seen from page 44 that unless this building is on a hill site, the coast, or riverside, the degree of exposure will be in the " normal " category. Let it be assumed that at the time of estimating the orientation of the building is not known. In such a case the figures between the heavy lines on page 56 would be used.

Allowance for Height of Building.

See page 50. The percentage addition to the total losses from structure and air change will be 4 per cent. for hot-water radiators or 6 per cent. for steam radiators or unit heaters.

Details of Calculation.

Structure	*Area*		*U*	*B.t.u./h. deg. F.*
Side windows	1 000	×	1·0	1 000
9 in outer wall	6 200	×	0·47	2 900
North-lights	10 000	×	1·0	10 000
Roof sheeting	20 000	×	0·34*	6 800
Floor	20 000	×	0·037†	740
				21 440

	Cube		*Air Change per hour*		*Specific Heat*	
Air	320 000	×	2	×	0·02	12 800
						34 240
Allowance for Height	..		6 per cent. of this total			2 050
						36 290

If the occupants are largely employed on sedentary work a temperature of 65°F is required and if the outside is taken as 30°F, i.e. 35 deg. F rise, the heat input will, therefore, be 36 290 × 35 = 1 270 150 B.t.u./h.

NOTES :

* The regulations under the Thermal Insulation (Industrial Buildings) Act, 1957, require the heat loss not to exceed 12 B.t.u./sq ft. h when the outside temperature is 30°F. Hence for an inside temperature of 65°F :

$$\text{“U” must not exceed } \frac{12}{65 - 30} = 0{\cdot}343$$

From page 74 it will be seen that insulating board has a conductivity figure of about 0·39. Corrugated iron, lined with $\frac{1}{2}$ in insulating board and applied on battens, thus forming an air space, will have an overall coefficient U = 0·34 (see page 54).

† See page 58 for floor of 200 ft × 100 ft.

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Tomlinson[63] writing after Hood, quotes Arnott's estimates of the heat requirements of buildings. He assumed an inside temperature of 60°F, an outdoor temperature of 22°F, and steam heating surface at 200°F. One ft^2 of heating surface was allowed for each 6 ft^2 of window or for 120 ft^2 of ceiling, wall or roof constructed of ordinary materials, and for each 6 ft^3/min of ventilating air. Arnott[3] also noted that double windows had only one-quarter of the heat loss of a single window.

Tomlinson also used Hood's data on the heating effect of 4-in pipe. In habitable buildings, he assumed that the ventilation requirements amounted to 3½ ft^3/min per person. To this was added 1¼ ft^3/min as being the equivalent of each ft^2 of glass. The total length of 4-in pipe required was then

$$\frac{125\ T}{t_p - t_i} \times \frac{n}{222} \quad \text{ft}^*$$

Here t_p is the temperature of the pipe, t_i the indoor temperature, T the desired temperature difference between indoors and outdoors, and n the total number of ft^3 of air to be warmed (not the volume of the space). For churches, where the heat output of the occupants is relatively large, Tomlinson gave an empirical rule:

ft of 4-in pipe = Volume/200

Constantine (1881) was quite satisfied with such empirical procedures:[17]

> "Of late years, the makers of boilers, pipes, bends and other fittings for the circulating system have exercised considerable ingenuity in providing everything that is necessary for reducing or enlarging the flow pipes, turning corners, dipping or ascending, so that the execution of this class work has been greatly simplified, and now almost any respectable plumber can do hot-water fitting, and they all know that one foot of 4-in pipe warms 100 ft of air; there is, therefore, no necessity for giving elaborate tables as was the custom in books written 20 years ago. Another simple rule should never be lost sight of, that is, always have a good margin of heating power both in boiler and pipes."

Perkins' high pressure systems were also designed on an empirical basis. In 1904, Walter Jones was:[33]

> "not aware of any work yet published that gives the length of pipes and the boiler power required to obtain various temperatures by high-pressure heating".

He suggests that, based on experience, for offices to be warmed to 60-65°F (outdoor temperature 30°F), 18-22 feet of ⅞-in bore pipe are required per 1000 ft^3 of room volume. The furnace coil is to be equal to one-tenth of the room pipe length.

Nevertheless, the theory of high temperature hot water heating had been given by Einbeck in 1877, and Rietschel[52] gives detailed calculations for the furnace coil and for the room heating surface. On the assumption that the furnace temperature is 1000°C and the combustion gases leave at 200°C, the length of the furnace coil is to be

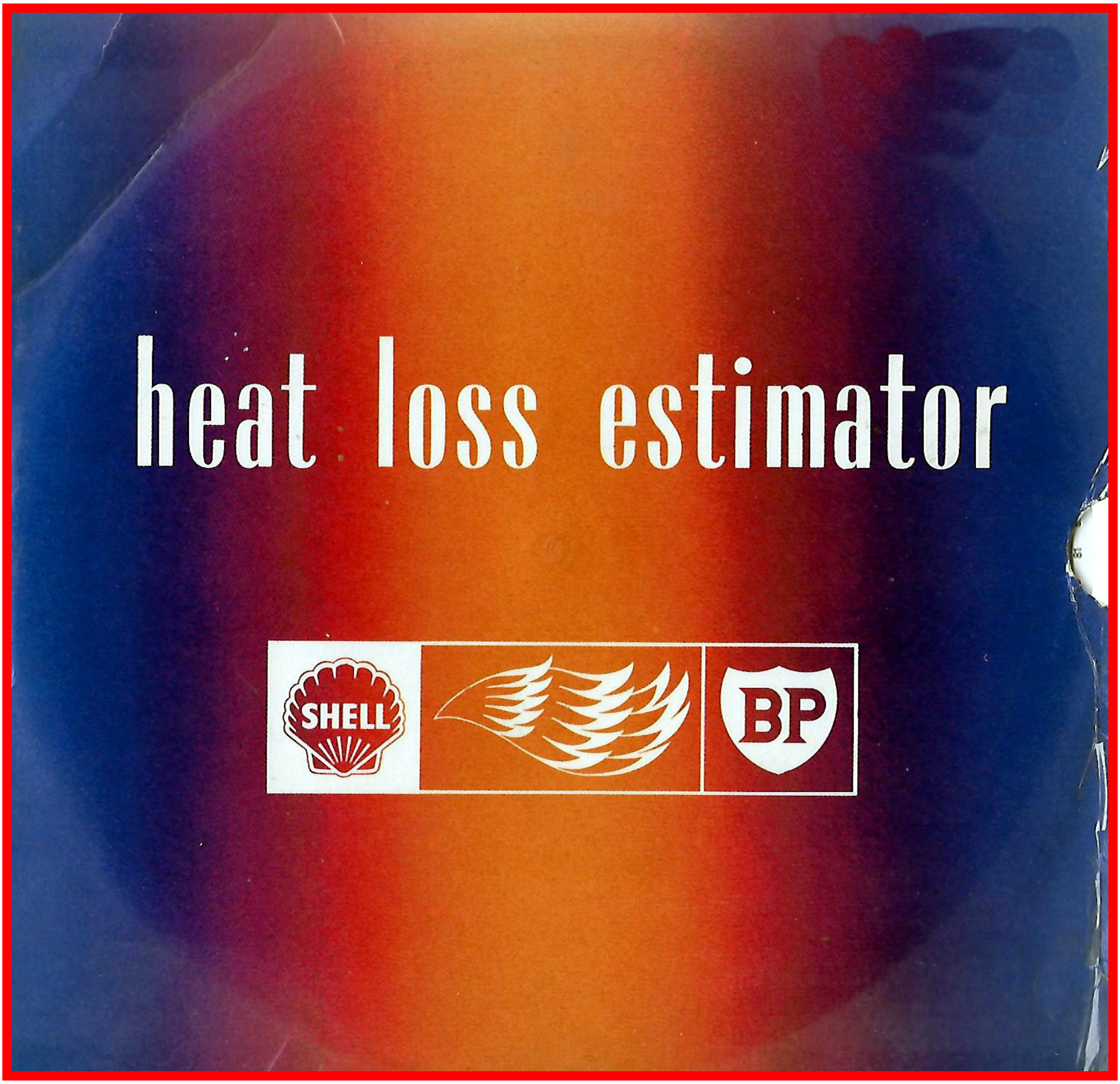
$$l = 0.002 \quad C \quad \text{metre,}$$

where C is the design heat loss, kcal/h. The length of the room heating surface is calculated from

*The figure 222 is Hood's value for the volume of air which can be warmed by 1 ft run of pipe.

$$l_2 = \frac{10\ C}{11.5\ (\frac{1}{2}\ (t' + t'') - t_{air})}$$

t' and t'' being the flow and return temperatures (150° and $80^{\circ}C$), the factor 11.5 being the pipe emission (the value given here is appropriate only to the temperature quoted).



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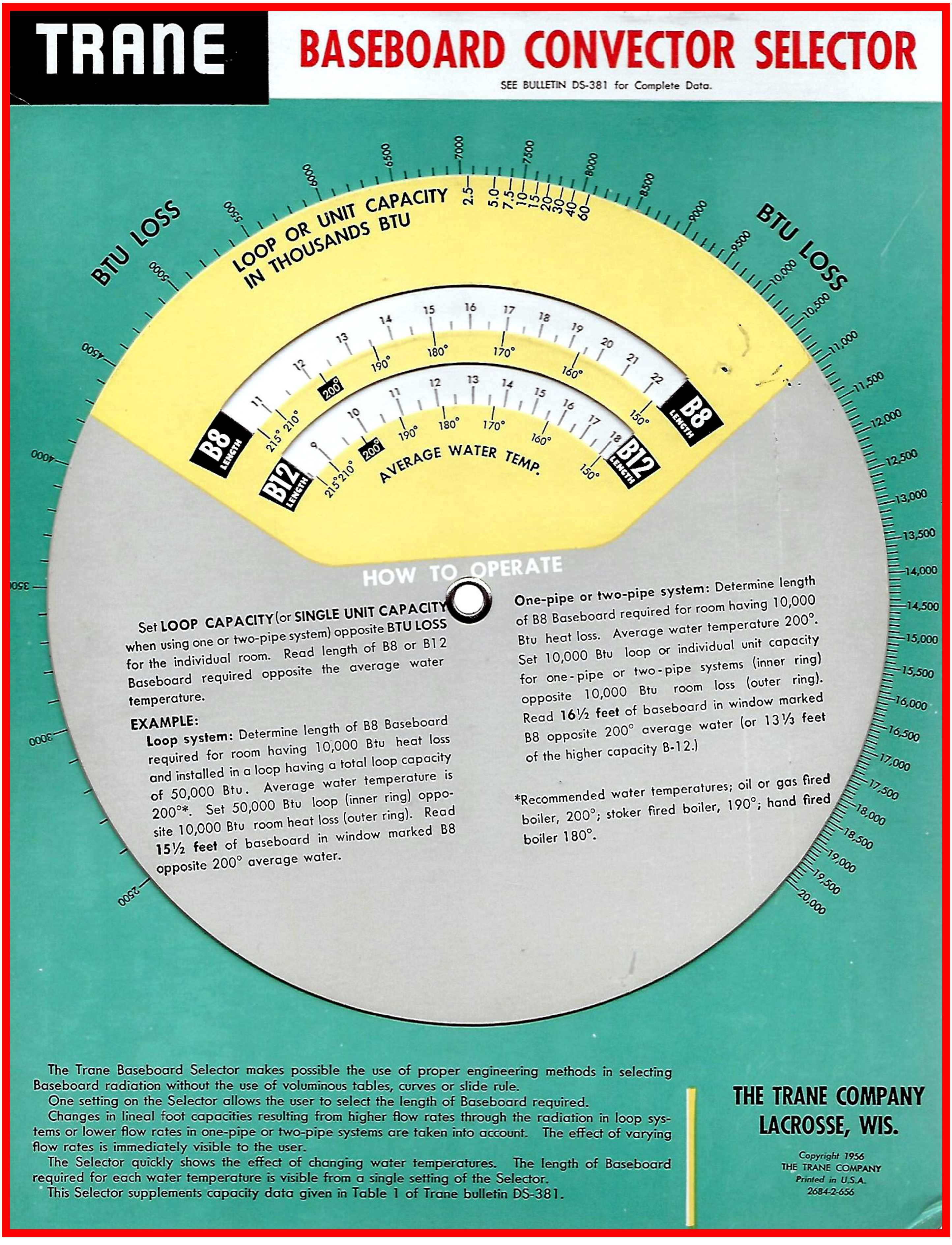
TRANE
BASEBOARD CONVECTOR SELECTOR
SEE BULLETIN DS-381 for Complete Data.
BTU LOSS
LOOP OR UNIT CAPACITY IN THOUSANDS BTU
2.5 5.0 7.5 10 15 20 30 40 60
B8 LENGTH
B12 LENGTH
AVERAGE WATER TEMP.
BTU LOSS
HOW TO OPERATE
Set LOOP CAPACITY (or SINGLE UNIT CAPACITY when using one or two-pipe system) opposite BTU LOSS for the individual room. Read length of B8 or B12 Baseboard required opposite the average water temperature.
EXAMPLE:
Loop system: Determine length of B8 Baseboard required for room having 10,000 Btu heat loss and installed in a loop having a total loop capacity of 50,000 Btu. Average water temperature is 200°*. Set 50,000 Btu loop (inner ring) opposite 10,000 Btu room heat loss (outer ring). Read 15½ feet of baseboard in window marked B8 opposite 200° average water.
One-pipe or two-pipe system: Determine length of B8 Baseboard required for room having 10,000 Btu heat loss. Average water temperature 200°. Set 10,000 Btu loop or individual unit capacity for one-pipe or two-pipe systems (inner ring) opposite 10,000 Btu room loss (outer ring). Read 16½ feet of baseboard in window marked B8 opposite 200° average water (or 13⅓ feet of the higher capacity B-12.)
*Recommended water temperatures; oil or gas fired boiler, 200°; stoker fired boiler, 190°; hand fired boiler 180°.
The Trane Baseboard Selector makes possible the use of proper engineering methods in selecting Baseboard radiation without the use of voluminous tables, curves or slide rule.
One setting on the Selector allows the user to select the length of Baseboard required.
Changes in lineal foot capacities resulting from higher flow rates through the radiation in loop systems or lower flow rates in one-pipe or two-pipe systems are taken into account. The effect of varying flow rates is immediately visible to the user.
The Selector quickly shows the effect of changing water temperatures. The length of Baseboard required for each water temperature is visible from a single setting of the Selector.
This Selector supplements capacity data given in Table 1 of Trane bulletin DS-381.
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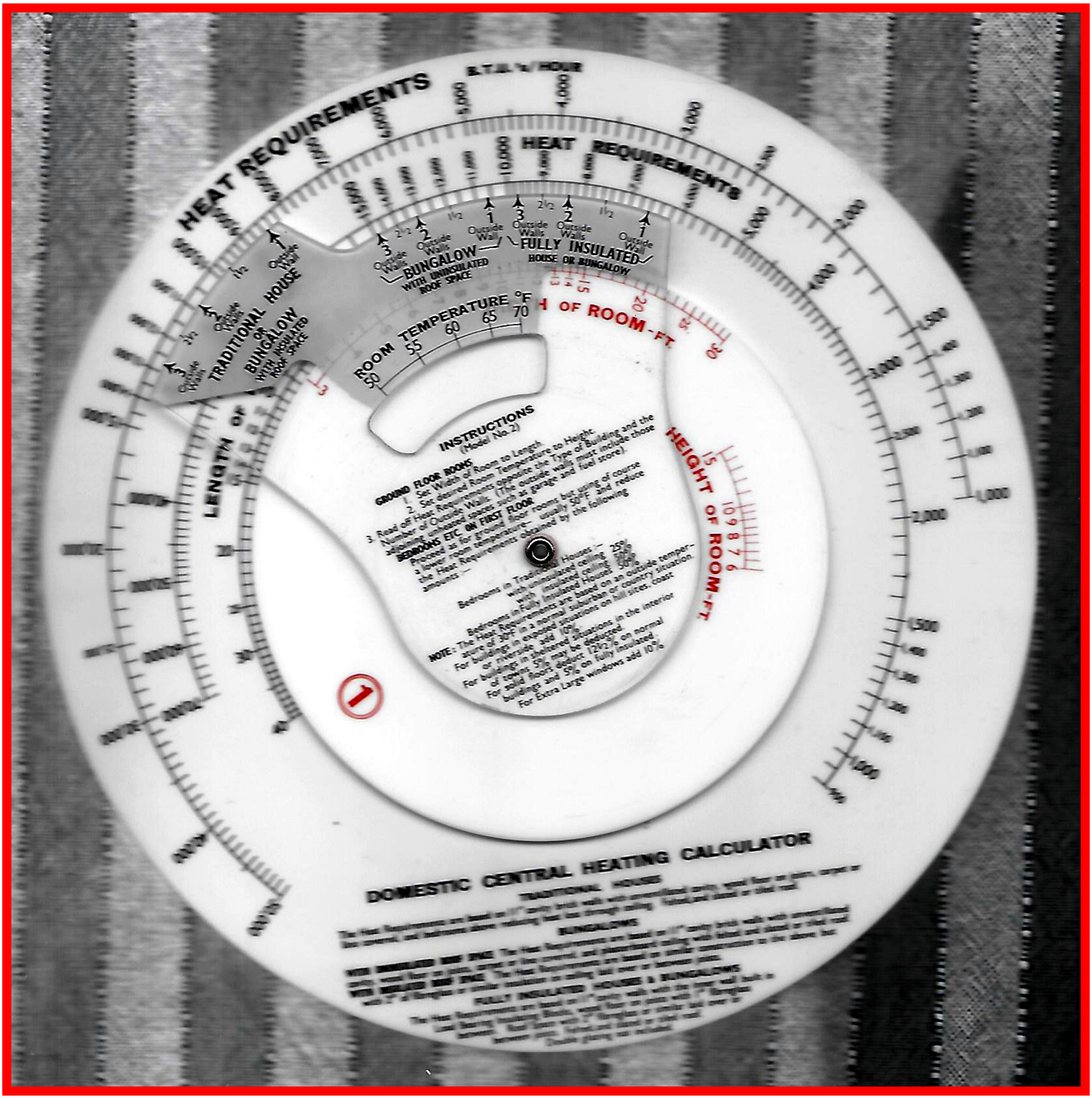

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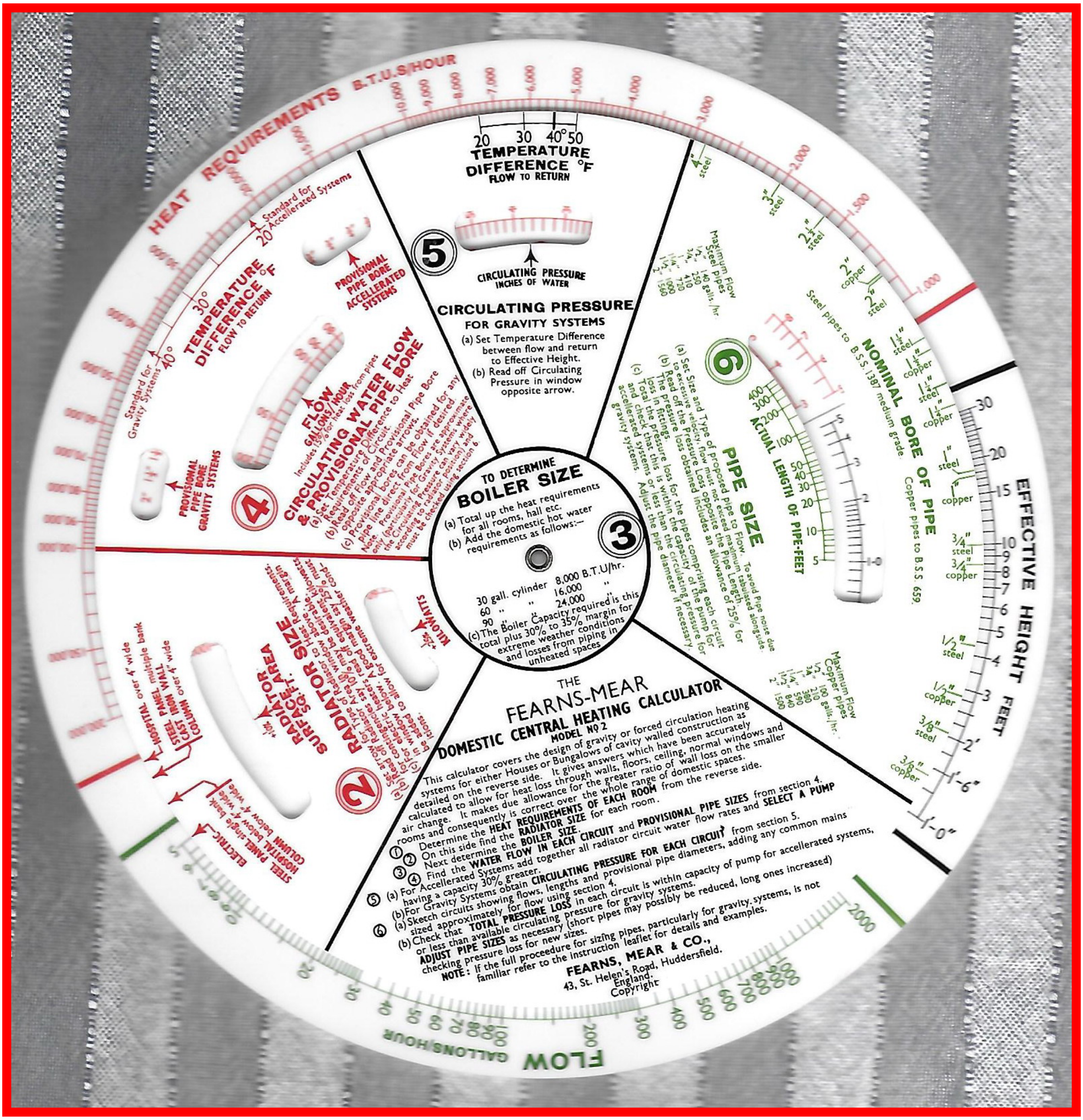

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