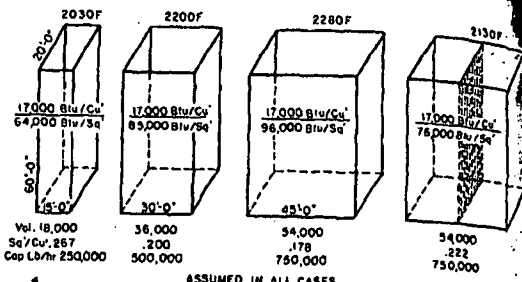
Why not write specifications, argues N J Connors, sales engineer, Babcock & Wilcox Co, *in words that enable comparison? Start with terms that relate heat release to performance; gas temperatures to measurement; ash to firing*



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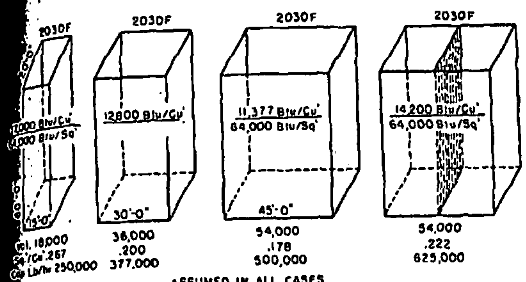

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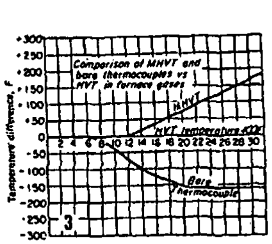

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Make Your Furnace-Design Specifications Work for You

▶ A LARGE NUMBER of specifications that are issued fail to accomplish their objective. *So far as furnace designs are* concerned, they surely fail if we assume that the objective is to receive bids that are truly comparable.

The furnace-design problem with all its interrelated variables is complex and difficult at best. Probably for this reason engineers who write specifications are somewhat reluctant to tie the designer's hands too tightly. But they usually do take a hand in the problem, at least to the extent of specifying a value of *Btu per cu ft that is not to be* exceeded. On some occasions they specify, as well, the maximum temperature of gases leaving the furnace. But they don't qualify by stating how the temperature is to be taken. Behind all this is the specification writer's desire to control *something*.

The furnace-design problem will probably always be a field of some difference of opinion. That is no reason, though, why we should not strive in every way possible to clear up thinking on the subject. At least let's avoid use of terms that tend to further misunderstanding.

Heat Release. What the specification writer tries to control, of course, is the temperature of gases at those points *where slagging trouble is first likely* to make itself felt. When he does this through the medium of a value of Btu per cu ft, not only is he doing it in a very indirect manner but he also gets no control. Until he further ties this value into a particular kind and extent of wall construction, and knows just how to vary this value for furnaces *of different sizes, he has no controlling* factor.

Almost every engineer realizes that the bounding surface does not change *proportionally with volume. Yet a great* number of engineers continue to use Btu per cu ft—a term that disregards this truth. Why is this?

For one thing, habits of thinking on furnace designs have been formed through a long period that knew no better guide. For a second, a large number of engineers do not appreciate the danger of too much reliance on Btu per cu ft as a measure of the furnace-design adequacy. It is surprising how few engineers really know what happens to this measuring stick of Btu per cu ft from size variations *alone. Let's simplify the problem to* the extent of removing the other variable, namely, degree of waterwall covering, and assume only completely water-cooled surfaces.

Working Example. To illustrate, take a furnace of cubical shape, having a dimension of 8 ft, 0 in. on all sides, and assumed to be watercooled on all six faces. This is not far from what one might find in marine practice. The volume is 512 cu ft and the en*velope area 384 sq ft. If we liberate* heat in this furnace at, say, 70,000 Btu per sq ft of watercooled furnace sur*face per hour, the Btu per cu ft is* 52,500 per hour. Now take a furnace of dimensions found in large central-station units, 45 ft, 0 in. x 60 ft, 0 in. x 20 ft, 0 in. Here the volume is 54,000 cu ft and the surface 9,600 sq ft. If we *liberate heat in this furnace at the* same rate, that is, 70,000 Btu per sq ft per hr, the Btu per cu ft is only 12,500. Yet in each case the same *amount of heat is being presented for* absorption to each square foot of furnace cooling surface. Does a yardstick that varies like this have *much* value?

Just one more example to drive home the point. Consider the large furnace in the last example. Without changing the dimensions, let us put a division wall down through the furnace. This wall substantially divides the furnace into halves and receives heat on both sides. Now our surface has changed to 12,000 sq ft, an increase in radiant-heat absorbing surface of more than 30%. If we introduce no more total *heat than before, no engineer would* have much trouble deciding which of these two furnaces is more effective in reducing gas temperature. But our commonly used yardstick, Btu per cu ft, which *is* supposed to *tell us something* about what is happening in a furnace, fails utterly to recognize this drastic change in surface. The Btu per cu ft is exactly the same after adding the division wall as it was before.

Perhaps this Btu-per-cu-ft weakness can be cleared up by *closely examining* the diagrams above.

In these diagrams are shown *four furnaces* assumed to be *watercooled* on all six faces. Each one is dimensioned realistically for a unit of the capacity labeled under each sketch. Also given the ratio of surface to volume and the volume itself.

Furnace Sizes. Now let us assume that all these widely varying sizes of furnaces are sized at the same Btu per cu ft. If this is the case and with all *of the surfaces operated* under one set of conditions as *to coal*, combustion and ash coating, we know from actual high-velocity thermocouple tests that the leaving temperature of the gases for this set of conditions is as shown above each sketch. It will be seen that these temperatures vary widely even though the Btu per cu ft is always the same.

Now what we really want is to have control of these temperatures. This at least is what we think we are doing in specifying a certain Btu per cu ft. Let's see, in the diagrams above, what happens to these same furnaces *and the capacity of* the boilers they serve when we really control this exit temperature.

We want to hold this temperature at 2030 F. To do this we have seen from the diagrams *on* p 102 we cannot present to the furnace surfaces more than 64,000 Btu per sq ft per hr. Notice what this does to the Btu per cu ft value and, worse still, what it does to the capacity we thought we had when we were placing so much reliance on Btu per cu ft in these diagrams.

In conclusion, Btu per cu ft is safe to use only in comparisons of units of *approximately the same size.* It is very dangerous and misleading to use it over a wide range of sizes. It *is* dangerous because for many years most of us have been living with the comfortable feeling that our pet value of Btu per cu ft was somehow synonymous with a satisfactory leaving temperature. *For* some one size of unit this is true. Diagrams *above* and *on* p 102, though, show how far from true it really is over a range of sizes.

A New Standard. A large measure of the misunderstanding now prevalent in the furnace-design problem would be removed, if a determined effort were *made to* (1) refrain from using Btu per cu ft and (2) adopt as a yardstick Btu available per square *foot of exposed* watercooled surface per hour. Just how will this come about?

The most important function of a furnace is to burn the fuel and properly condition the gases before entering the boiler. Only watercooled surface can reduce gas temperature. The engineer can assure himself that the comparisons he makes are true, not only between units *of* approximately the same size, but also between units of widely varying sizes, if he specifies that the design must conform to a maximum *Btu* released and available per square foot of *projected watercooled-furnace* envelope area per hour.

Consider for a moment how unscientific and wholly inadequate is the universally used yardstick "Btu per cu ft" in describing a furnace design. A measure of the volume tells absolutely *nothing* about the kind of surface by which it is enclosed (see above). And remember, only watercooled surface appreciably cools gas by absorption of radiant heat. As you increase a volume, the ratio of *bounding* surface area to volume diminishes. Btu per cu ft then can be used as a yardstick for comparison only if volumes are the same. Just what steps can be taken to find out how this yardstick should change for different volumes? This is the point *where opinion* and guess work come into the picture to plague and confuse thinking.

Btu available per sq ft per hr is a fundamentally sound unit for design comparisons. Why not use it? *With so many* natural variables in the pulverized coal-fired furnace problem, why go out of the way to compound difficulties by introducing an artificial one?

Gas Temperature. Occasionally engineers specify limits of furnace design desired by using a furnace exit-gas temperature as a measure of design limit. Usually when this device is employed, no description is given of the method by which the temperature is to be taken. This practice is not conducive to clear thinking or to obtaining comparable *furnace* offerings.

And here is why. It is known from work with high-velocity thermocouples over the last eight to ten years that in the range of temperature from 2000 to 3000 F *a bare metal thermocouple* exposed to furnace gases will register values of the order of 250 to 300 F below the true gas temperature.

The curve shown in Fig. 3 gives (1) high-velocity thermocouple (*HVT*) temperatures as a base and (2) a comparative reading of the bare metal thermo-