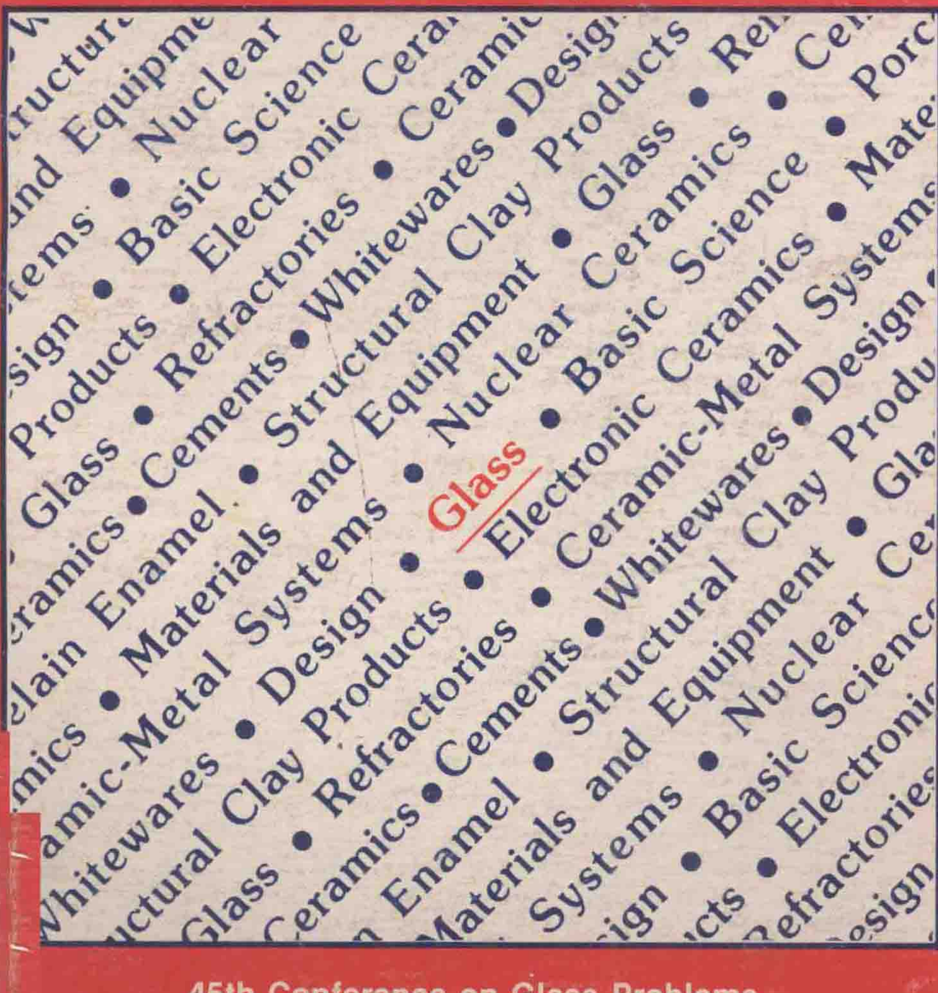
Ceramic Engineering and Science Proceedings

March–April 1985



45th Conference on Glass Problems

45th Conference on Glass Problems

Proceedings of the 45th Conference on Glass Problems

Charles H. Drummond III
Editor

A Collection of Papers Presented at the
45th Conference on Glass Problems
Sponsored by the
Departments of Ceramic Engineering
The Ohio State University
and
University of Illinois at Urbana-Champaign

November 13-14, 1984

The Ohio State University
Fawcett Center for Tomorrow
Columbus, Ohio

ISSN 0196-6219

Published by
The American Ceramic Society
65 Ceramic Drive
Columbus, Ohio 43214

Editorial and Subscription Offices: 65 Ceramic Drive, Columbus, Ohio 43214. Subscription $60 a year; single copies $12 (postage outside U.S. $2 additional). Published bimonthly. Printed in the United States of America. Allow six weeks for address changes. Missing copies will be replaced only if valid claims are received within six months from date of mailing. Replacements will not be allowed if the subscriber fails to notify the Society of a change of address.

CESPDK Vol. 6, No. 3-4, pp. 121-298, 1985

Foreword

The conference was sponsored by the Departments of Ceramic Engineering of The Ohio State University and the University of Illinois at Urbana-Champaign.

Director of the conference was Dr. Charles H. Drummond III, Associate Professor, Department of Ceramic Engineering, The Ohio State University.

Associate Dean Sunder H. Advani of the College of Engineering, The Ohio State University, gave the welcoming address, and Dr. Dennis W. Readey, Chairman of the Department of Ceramic Engineering, gave the departmental greetings.

The themes and chairmen of the three half-day sessions were:

Furnace Design
Carl W. Hibscher
Toledo Engineering Co., Toledo, OH

Plant Operations
John McConnell
PPG Industries, Pittsburgh, PA

Combustion
John. A. Priestley
Fiberglas Canada, Sarnia, Ontario

Selected Topics
Everett A. Thomas
Didier Taylor Refractories
Cincinnati, OH

Presiding at the banquet was Professor Clifton Bergeron, Professor of Ceramic Engineering, University of Illinois at Urbana-Champaign. The banquet speaker was Beuther Schmidt of Lancaster, OH. His address was entitled "Stop, Look and Listen."

The conference was held at the Fawcett Center for Tomorrow, The Ohio State University, Columbus, OH.

Preface

In the tradition of previous conferences, the papers presented at the 45th Annual Conference on Glass Problems have been collected and published as the 1984 edition of The Collected Papers.

The manuscripts are reproduced as furnished by the authors but were reviewed prior to presentation by the respective session chairmen. Editing was done by C. H. Drummond. The Ohio State University is not responsible for the statements and opinions expressed in this publication.

Charles H. Drummond III
Columbus, OH
December 1984

Acknowledgments

It is a pleasure to acknowledge the assistance and advice provided by the members of the Program Advisory Committee:

Clifton G. Bergeron
University of Illinois at Urbana-Champaign
Eugene D. Bowman
Owens-Illinois
Douglas H. Davis
FMC
Carl W. Hibscher
Toledo Engineering Co.
John F. McConnell
PPG Industries
Michael McDonald
Corning Glass Works
John A. Priestley
Fiberglas Canada, Ltd.
Dennis W. Readey
The Ohio State University
Subhash H. Risbud
University of Illinois at Urbana-Champaign
C. Philip Ross, Jr.
Kerr Glass Manufacturing
Everett A. Thomas
Didier Taylor Refractories Corp.

Charles H. Drummond III
Conference Director

Table of Contents

Furnace Analysis Applied to Glass Tanks

ROBERT H. ESSENHIGH

Department of Mechanical Engineering
The Ohio State University
206 W. 18th Ave., Columbus, OH 43210

Introduction

Furnace analysis is the name given to a particular procedure for determining the thermal performance and thermal efficiency of furnaces. Specifically, it results in a set of three performance equations that describe: the firing curve (the variation of firing rate with pull); the thermal efficiency curve; and an intrinsic efficiency line. Interest in efficiency derives, of course, from the truism that management is primarily interested in product quality at acceptable cost, and thermal efficiency is generally an important component of cost.

Since the analysis is general it applies, in principle, as much to glass tanks as to any other furnace types (or engines).[1] The value of the analysis is that it can identify and quantitatively rank the principal factors that control thermal efficiency. Thus, it can be applied to an operating glass tank to determine the point of optimum performance (peak efficiency); to quantify the losses from off-peak operation; to quantify the fuel excess and costs due to over-high excess air; and to monitor the possible degradation of performance with time due to wall erosion (higher wall losses), regenerator blockage (lower preheat and higher pressure losses), and the like. It can also be used for tank design in conjunction with more detailed calculations of internal (mostly radiative) heat transfer.

The analysis as originally conceived and reported in the literature is essentially steady state. There is now some thought, however, that the analysis can be extended or used on a perturbation basis to describe unsteady state behavior. This may be important in determining excess fuel consumption when changing pull on a tank if restabilization at the new firing rate takes too long. This is not a point that is thought to have been given too much attention in the past compared with other factors, but some evidence is now available to suggest that it can make a useful second-order contribution to thermal efficiency, and thus to costs.

This article has two objectives. The first is to summarize the performance equations, with an appropriate discussion of typical steady state behavior. The second is to examine some aspects of unsteady state behavior and the possible implications for control and management of load (pull) changes which, as will be seen, start to focus attention on the dynamic thermal response of the furnace walls.

Literature Sources

Some key references are identified here for background; no attempt will be made to review the literature on furnace analysis in detail; much of this provided in Ref. 1.

Furnace analysis applies, as indicated, to all types of thermal devices, including engines as well as furnaces. The origins of the analysis are usually attributed to a paper on boiler performance by Hudson[2] in 1890 with a follow-up by Orrock[3] in 1926. In 1927 Armstrong[4] published a particularly significant paper titled "Characteristics of Furnaces Curves as an Aid to Fuel Control" which appears to be the first recognition that the output rate is one of the most significant factors controlling efficiency.

In the developments following, the analyses were generally centered on the internal heat transfer characteristics of the specific thermal devices by using idealized flame descriptions and heat transfer models, as described, for example, by Hottel and Sarofim.[5] In this type of approach, thermal inputs and outputs (firing rate and pull) then appeared as boundary conditions to the internal heat transfer problem; accordingly, it is sometimes referred to as the "Internal Analysis" though the method is more synthetic than analytic. This approach has led in the last decade or so to the development of extensive codes for predicting heat transfer profiles in furnaces. A significant problem of such codes, however, is the extent to which they are dependent on numerous approximations and assumptions whose validity is not always certain. The codes also tend to be difficult and/or expensive to use for routine on-line analysis of furnace performance. Comparative evaluation has also been a problem since many of the codes are proprietary.

The alternative approach emphasized in this present article focusses more on the overall performance of the furnace, based essentially on a first law analysis of the gross enthalpy values carried by a limited number of entering or leaving streams: these are, the total heat supplied by the fuel which then equates to the energy leaving in the stock (glass, steel, stream, etc.), plus the exhaust flue gas (products of combustion or POC), and plus the wall loss. In glass tanks, of course, the wall loss is particularly significant, being in the range of 20 to 35% of input, compared to loss from boilers where it is generally less than 5% and frequently has been found to be under 1%. The theoretical approach is thus more analytical in the sense of examining operational data in the light of the relatively simple set of performance equations.

One of the more significant contributions that permitted effective development of the external analysis was provided by Thring and Reber[6] who observed that wall losses tended to be insensitive to firing rate or output, and this provided a simple theoretical means of handling that component of the energy streams. They did use a crude internal analysis based on the so-called long furnace to provide a missing equation, but this was found to be no better, and less general, than a heuristic assumption regarding the specific enthalpy of the exhaust stream introduced by Essenhigh and Tsai[7] to close the equation set. The elements of the approach are summarized in Ref. 1.

There have been a number of applications of the analysis in the last decade, one significant application being to an aluminum re-melt furnace.[8] This incorporated a new extension for the effect of oxygen enrichment, and also it introduced the use of the analysis in costing.

The Steady-State Performance Curves

Firing Curves

The thermal performance of any furnace is dominated by the firing curve which represents the variation of firing rate with output or pull. Figure 1 illustrates such a firing curve for a cross-fired glass tank with firing rate in cubic feet of gas per day and pull in tons of glass per day. When the firing rate and pull are expressed in thermal rate terms (Btu or kJ per hour or day), written as H_f and H_s, respectively, the firing curve can be described by the following firing equation:

$$H_f = H_f^\circ + \frac{H_s}{\alpha^\circ(1 - H_s/H_s^m)} \tag{1}$$

This is the equation of a concave-upwards curve with the following additional characteristics:

— When the furnace is idle (no pull) a minimum firing rate, H_f°,—the idle heat—is still required to balance wall losses to maintain the furnace at operating temperature.

— For a sufficiently small range of outputs the denominator in the last term of Eq. 1 can be written: H_s/α°; and for this small output range the firing rate rises linearly with output. This expresses the variation described by the empirical SGT[9] and HAPF[10] equations for glass tanks.

— In the range of the linear approximation, the heat supplied splits in a roughly constant proportion between the load (glass) and the products of combustion (POC). If this approximation held accurately it would also mean that the exhaust gas temperature was invariant with output, and this conclusion is not supported by experiment.

— At higher outputs and firing rates the curve is non-linear since the added heat increasingly disproportionates between the POC and the glass, with an increasing proportion going into the exhaust; on the same account the exhaust gas temperature also rises as output increases. The physical reason for this disproportionation is that the transit time of the combustion gases drops as the firing rate increases so there is less time for heat transfer to the glass (or other load).

— The limiting theoretical result of the disproportionation in heat transfer is an experimentally unattainable condition in which the firing rate would rise to infinity and the POC temperature would rise to the adiabatic flame temperature—a finite, not infinite value. By the same token the furnace would also be at, or close to the adiabatic flame temperature.

— Since there is a theoretical maximum furnace temperature, even at infinite firing rate, there is also a finite maximum rate of heat transfer to the glass, and this maximum sets the maximum output, H_s^m.

Figure 1 illustrates several of these characteristics. There is a clear idle or holding heat. The best-fit line through the experimental points, based on additional data obtained from regenerator measurements, is a curve, as shown (solid line), obeying Eq. 1. Without prior, independent data and analysis, however, there would be no reason not to use a straight line as an empirical best fit (dotted line), in accordance with the SGT and HAPF equations[9,10]—and also in agreement with Eq. 1 for $H_s/H_s^m << 1$.

Another unusual characteristic of Fig. 1 is the availability of data points at low output rates. Characteristically, available data, e.g., from Ref. 11, tend to be limited to a relatively small range at the high output end so that the ratio of scatter to range is large, and determinations of curvature are difficult or impossible. This would have been the case in Fig. 1, for example, if the data were limited to the output range 90 to 180 tonne/d (100 to 200 ton/d).

Thermal Efficiency

Thermal efficiency is conventionally defined as the ratio of useful output to total input in consistent units or, in the thermal units used for Eq. 1:

$$\eta = H_s/H_h \tag{2}$$

This is inversely proportional to the common alternative for specifying efficiency as: Btu/ton (or Btu/sq.ft.ton). Applying Eq. 2 to the trend lines of Fig. 1 yields the two efficiency curves of Fig. 2. (To make the thermal-units conversion the heat of combustion was taken as 1000 Btu/cu.ft., and the specific enthalpy of the glass as 1.9*E6 Btu/ton).

In Fig. 2 the solid/dashed line is that obtained using Eq. 1 with Eq. 2. The dotted line is that obtained from treating the curve of Fig. 1 as a straight line (SGT/HAPF empirical formula). The curves have the following characteristics:

- — The solid-dashed line is characteristic of all thermal devices where there is zero efficiency at two points: zero output and some (theoretical) maximum output; and there is a maximum efficiency at some point between.
- — The initial rise to the maximum is dominated by wall losses. If wall losses can be reduced, the maximum efficiency increases, and the optimum output for maximum efficiency decreases (i.e. moves to the left). If there were no wall losses, the maximum efficiency would be at zero output, and it would fall steadily with increasing output (in practice some boilers with very low wall losses get surprisingly close to this condition).
- — The fall in efficiency beyond the maximum is dominated by the transit time of the combustion gases, with the transit time falling (theoretically) to zero as output rises to the theoretical maximum.
- — If the firing rate is assumed to rise linearly with output, as described by the SGT/HAPF equations, the rise in efficiency (dotted line) closely matches the previous calculation up to the maximum, but this alternative yields no maximum; instead, as shown by the dotted line it rises monotonically to an asymptotic value at infinite output. By the same token there is no maximum output although purely physical reasoning shows that there must be one.

For the particular furnace for which these data were obtained it happens, as can be seen, that a maximum efficiency was evidently reached, and at the highest output just beyond the optimum, the efficiency was a few points below expectation from the asymtotic curve. This obviously has implications for design as tank size is increased. If Fig. 2 is regarded for the moment as indicative of the change in efficiency by increasing tank size (a sufficiently acceptable though not totally accurate assumption) it is clear that the two methods of extrapolation lead to significant differences in expected firing rate requirements beyond the 135 tonne/d (150 ton/d) output level. If the asymtotic curve is used as the basis for design, but the peaking curve is the more accurate representation, it can mean that at higher outputs the required firing rate will have been underestimated. By the same token, there will be corresponding underestimates of all other related factors: fan requirements, pipe sizes, gas flow volumes, regenerator dimensions, and the like. Correspondingly, if the design requirements are based on the peaking curve, an efficiency decrease must be accepted as an associated penalty of increased output, or else additional heat recovery equipment must be added with, of course, its added costs.

Heat Recovery

Heat recovery is an essential component of any high temperature furnace, generally for reasons of thermal efficiency. In the case of high-temperature systems such as the glass tank, open hearth, and blast furnace, the more important factor is the increase in flame temperature, as otherwise melting would not be achieved, or else output would be too small to support the industrial level that exists.

Where the efficiency of the heat exchanger is roughly constant the effect of heat recovery is not to change the functional form of Eq. 1 but to alter the values of the three equations coefficients: H_f°, H_s°, and α°. The analysis then shows that the coefficients are functions of heat exchanger efficiency. The results are that: the idle heat, H_f°, is reduced, the other two parameters are increased. Overall, of course, the required fuel rate is reduced for a particular value of output (H_s); analytically, the reduction results from the changes in all three of the equation coefficients. A particularly interesting effect is the significant increase in the maximum output (H_s^m) with the result that the term (H_s/H_s^m) in Eq. 1 is correspondingly less important, and a straight line becomes a reasonably acceptable first approximation to the actual curve over the range of operations.

For the data as given, the output maximum is estimated (see Fig. 2) as 450 tonne/d (500 ton/d) with the regenerator efficiency, based on gas temperature measurements, estimated at 75%. Without the heat recovery, the estimated maximum output would have been about 72 tonne/d (80 ton/d), and the operating maximum output would probably have been in the region of 27 tonne/d (30 ton/d) requiring about 50% greater firing rate. These values clearly show the dominant importance of heat recovery. They also indicate the potential value of additional heat recovery if larger tanks are to be built and operated much beyond their optimum output for maximum efficiency.

The temperature measurements used to determine the heat exchanger efficiency showed that the gas temperature leaving the furnace rose with output. It was this information that showed that the firing curve was not a straight line. Had it been a straight line, this would have meant that the heat released by combustion was always proportional in the same ratio between the exhaust and the glass, and in such a case the exhaust gas temperature would have been a constant.

Intrinsic Efficiency

There is a third performance equation to be obtained from the analysis which leads to a measure for intrinsic or design efficiency. Design efficiency is used here to include all such factors as the influence of relative dimensions (L/D ratios), number and placement of burners, type of fuel, type of flame (luminous/nonluminous), firing method, etc. It excludes such construction factors as type of brick, thickness of wall, degree of further insulation, etc., as these all control the idle heat parameter, H_f°. Use of Eq. 2 to make comparisons of operational efficiency (of different tanks, or firing methods, for example) can be misleading because of the substantial influence of the construction factors. The intrinsic efficiency equation subtracts out of the effects. It is based on Thring and Reber's observation[6] that the wall loss (plus the associated fraction of the stack gas loss) is essentially constant so that the net heat available for transfer to the load is not H_f but $(H_f - H_f^\circ)$. This available heat must then be split only between the load and exhaust. Thring and Reber introduced the fraction going to the load as the heat utilization factor (HUF), α. Using their definition with Eq. 1 we obtain:

$$\alpha = H_s/(H_f - H_f^\circ) = \alpha^\circ(1 - H_s/H_s^m) \qquad (3)$$

which defines $\alpha°$. From this it can be seen that $\alpha°$ is a limiting (maximum) intrinsic efficiency after eliminating the construction factor effects. It can also be seen that the variable, α, declines linearly with output, going (theoretically) to zero as output reaches its (theoretical) maximum.

The limiting or intrinsic efficiency, $\alpha°$, is thus a dimensionless parameter representing essentially the effects of design, independently of construction, by which different design effects can be compared. For the tank that generated the data of Fig. 1 the intrinsic efficiency was determined as 65%. This value can be compared with other values cited in Ref. 1 for glass tanks. Six values ranging from 40% to 60% were obtained from the asymtotic limit procedure using the SGT/HAPF basis; the other two values using the procedure here were 51% and 60%. For comparison, boilers with low steam temperatures and/ or high levels of heat recovery have intrinsic efficiencies in the region of 95%. This suggests that there is still scope for improvement in the glass tank efficiencies, probably by the addition of further heat recovery equipment (if this has cost-benefit validity).

Transient Conditions

General Properties

Transient conditions result from changes in load or pull with associated changes in firing rate and thus in the temperatures of the flame gases. The refractories temperatures then adjust to suit with a time response usually of hours depending on conditions. Even without the lagged-time response of the refractories, the change of firing rate with pull requires knowledge of the firing curve if the firing rate is to be reset for the new pull without hunting. If the reset for the new pull is too high or too low, the glass temperature will slowly become too high or too low in response (the whole furnace is heating up or cooling down); either way, it presents a condition that can again take some time (generally hours) to correct. This can result in excessive fuel use, and production of glass of uncertain quality until the furnace restabilizes. This emphasizes the need to know something about the response time of the furnace to transient conditions.

The firing rate reset is further complicated by the initial lagged time response of the refractories. If the firing rate is increased in response to increased pull, the time for the refractories to initially respond and increase their surface temperature results in an increased thermal load on the flame, thus reducing the heat transferred to the load. If a firing curve is developed under such non-equilibrium conditions, a hysteresis loop is obtained instead of a monotonic curve, with firing requirements on the high side for continually increasing output and on the low side for continually decreasing output.

The transient effects are particularly marked when a cold furnace is initially fired up at full fire; this is not treatment ever given to a glass tank but the response is instructive nevertheless. In experiments on a refractory-walled furnace with water cooled tubes on the floor to measure thermal efficiency, the immediate response to full fire in the cold furnace was a jump in thermal efficiency from zero to about 10%. This represented the contribution to heating from the flame alone. As the walls heated up, the thermal efficiency gradually increased [following roughly a (1−exp) curve] over a period of 7 or 8 h to an asymtotic value of about 30%. The additional 20 percentage points in efficiency represented the additional contribution of the hot walls, showing that the roof and walls were contributing about ⅔ of the heat to about ⅓ from the flame. Similar behavior will always occur in a hot wall furnace on a microscale when there is a change in load and an associated change in firing rate.

Steady State Wall Storage

When load and firing rate change, the associated change in wall temperature requires the addition or removal of stored heat. The time response of this behavior is responsible, of course, for the time delay to re-equilibration. One measure of the wall effects can be obtained by comparing wall storage with wall loss rates. If a wall of thickness L, density σ, uniform specific heat C_p, and thermal conductivity K, is at steady state with an inside wall temperature T_w and an outside wall temperature T_s, the total heat stored in a segment of the wall of unit cross-section is:

$$Q=\sigma C_P L(T_w+T_s)/2 \tag{4}$$

If the external (ambient) temperature is T_o with an external heat transfer coefficient, h_o, between wall and ambient, the rate of heat loss is:

$$\begin{aligned} H&=\bar{h}(T_w-T_o)\\ &=(k/L)(T_w-T_s)=h_o(T_s-T_o) \end{aligned} \tag{5}$$

where h is the overall heat transfer coefficient between the inside wall and the ambient.

A useful measure of the relative importance of wall storage compared with wall loss is the characteristic time (t_c) required for the steady state wall loss to equal the initial storage: i.e., $Q=H.t_c$.

Since T_o is close enough to zero (°F or C) compared with T_w we find by rearranging Eqs. 4 and 5:

$$t_c=(\sigma C_p/k)(L^2/2)(1+2k/Lh_o) \tag{6}$$

which is independent of all temperatures. For firebrick, the group $(2k/Lh_o)$ is about 0.4 for a 0.3m- (12 in.) thick wall (taking h_o=2.5 Btu/sq.ft.hr.F.), and is half that for a 0.6m- (24 in.) wall. The reciprocal thermal diffusivity group $(\sigma C_p/k)$ has a value of about 700 h/m^2 (65 h/ft^2). For a 0.3m- (1 ft.) thick wall this yields about 45 h for t_c or nearly 2 d (and t_c then increases almost as the square of the wall thickness): that is to say, that it takes about 2 d steady operation at temperature for the wall loss to equal the heat stored in start up. For a continuous tank with a campaign of months or years, this is nothing, but it is not unimportant in day furnaces.

For an 0.5 m (18 in.) wall the 45 h increases to 65 h; even for a 0.15m (6 in.) wall it is about 15 h. Thus a 1% change in wall storage (rise or fall in temperature by 10° per 1000) is equivalent to 1.5, 4.5, and 6.5 h of wall loss from 0.15 m (6 in.), 0.3m (1 ft.), and 0.5m (1.5 ft.) walls, respectively. This indictes the overall response times required for re-equilibration after load changes.

Unsteady State Requirements

The above calculations only compare storage with steady state loss although the time required for a 1% change in storage is strongly indicative. The response time is alternatively determined from the unsteady state equation for conduction (Fourier) heat transfer which in one dimension has the form:

$$\partial T/\partial t=(k/\sigma C_p)(\partial^2 T/\partial x^2) \tag{7}$$

where x is the distance through the furnace wall from the inside. If x is normalized by dividing by the wall thickness, L, the multiplier group (which is the thermal diffisuvity) then has the dimensions of time and can be written as another characteristic response time:

$$\tau = \sigma C_p L^2 / k \tag{8}$$

This is also a main component term of Eq. 5. This establishes the close relationship between the wall storage, wall loss rates, and transient response characteristics. The numerical values of t_c and τ are also very close so that the time estimates of the previous section can be used here as a first approximation.

Theoretically, of course, the time required to reach steady state is infinite and even in practice it is generally common to see temperatures continuing to change even after hours or days of operation. For purposes of calculation it is usual to use a criterion of 99% of the theoretical limit as a measure of time-to-equilibrium, and that is found to a multiple of τ. Masuda[12] has shown that typically the inner-third of the wall reacts to changes most actively, so that the characteristic time for significant responses is reduced by about 1/9; by the same token, however, this time only represents the time for a fraction 1/e of the total change. The times required to reach 95 and 99% of the final equilibrium temperatures are then 3 times and 4.6 times the characteristic times, respectively. For a 0.3m- (1 ft)-thick wall, the characteristic time is about 7 h; the half time is 5 h; and the 95 and 99% recovery times are about 20 and 30 h, respectively. Thus, in any operation where the pull is changed once a day or more the re-equilibration time of the furnace is likely to be important and should be taken into account in resetting the firing rate.

Practical Implications

These estimates of response times have some implications for furnace management. In one case known to the writer (though the details are confidential), the management analyzed the firing rate changes in a glass tank with time that occurred after load changes and found that there was a substantial hunting over a period of some h to re-establish equilibrium. A cost estimate of the excess gas used during the period of hunting showed that more careful management of the changes in firing rate was likely to be cost-effective. This was found to be the case. The procedure used was to review the past records of fuel requirements and melting rates to construct a firing curve similar to Fig. 1, and this was used to provide the operators with the estimated firing rate for the new load requirements. Under these operating conditions the time required to reestablish equilibrium was markedly reduced together with improvement in product quality during the changeover period.

A systematic procedure should also take into account the theoretically infinite time requirement to reach equilibrium. If the firing rate is reset to the level required after equilibrium has been established it will take several hours to reach even an acceptable approximation. The better procedure is to overfire the furnace if melting rate is to be increased, or underfire if it is to be decreased, until the conditions as indicated by an appropriate sensor are near equilibrium, and then to cut back the firing rate, or to increase it as appropriate, to the predetermined level obtained from the firing curve.

Discussion and Conclusions

Although fuel prices have stabilized in the last 2 yr (1982-1984) and some projections are for continued stability or only slow further rises in the next decade compared with the 1970s, the cost of fuel is at least an order of magnitude more important than it was until 1970. When the fuel cost for a single tank runs upwards to over $1 million per year, a 1% saving is $10000, or $100 000 for 10 tanks. Thus the potential cost-benefit value from improved fuel efficiency is likely to be substantial and well worth investigating. The furnace analysis procedure for establishing potential savings is particularly appealing as it relies in the first instance on appropriate analysis of existing data records and this is, itself, a relatively low cost procedure.

What is outlined in this paper is the elements of the analysis to illustrate the procedure. It emphasized primarily the role of output as the prime factor governing efficiency. Other factors that have been identified include[13], in some order of significance:

(1) Output
(2) Excess air
(3) Processing temperature
(4) Flame/load T (or flame temperature)
(5) Heat recovery
(6) Wall loss
(7) Load emissivity
(8) Flame emissivity

The effect of output is the prime focus, of course, of the analytical procedure, and its effect is summarized by Eqs. 1 and 2, or by Figs. 1 and 2. The effects of heat recovery and wall loss have also been indicated in the preceding discussion, as components of the analysis—wall loss is evident in the factor H_f. Excess air is a major factor now included in some unpublished extensions of the analysis (see also Ref. 14). Of the remaining factors, processing temperature is essentially uncontrollable, as may also be the case with load emissivity though control of batch that might float out on the melt surface, for example, might be important. When processing temperature is high, a high flame temperature or furnace-enclosure temperature is required to obtain a sufficiently high T for heat exchange to be sufficiently rapid. This, of course, is aided by heat recovery used to increase flame temperature (independently of the value of heat recovery just to improve efficiency), as discussed above. This shows the strong relationship between entries three, four, and five in the above list of efficiency factors.

Flame emissivity would appear to have the smallest influence on thermal efficiency. This is somewhat contrary to accepted views but it would appear to be the result of a trade-off between increased heat transfer from the flame as it becomes optically blacker, and reduced heat transfer from the roof and walls by greater interception of radiation by the blacker flame. Indeed, in some recent experiments[15] a highly radiant flame (from a coal-water slurry) fired in a hot-wall furnace with water-cooled tubes (for load) yielded lower thermal efficiencies than No. 1 oil or gas, and these results have recently been supported with additional measurements using mixtures of No. 1 and No. 6 oils. These results would suggest that some other common views would be worth reexamination, in addition to the question of flame emissivity, such as the superiority of over, or under, or side-port firing over each alternative. If the data