DESIGN ASPECTS FOR GLASS FURNACES

A CASE STUDY; EXPERIMENTAL FURNACE.

By

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PART (I): REVIEW OF SOME IMPORTANT ASPESTS

Chapter (I)

SOME ECONOMIC ASPECTS OF FURNACE DESIGN AND OPERATION

Various aspects and difficulties of making cost comparison are discussed. The uariety of situations ecountered does not permit the use of any standardised approach and the various methods used are illustrated by examples. These include comparison of different oil grades, determination of optimum furnace loading, comparison of different regenerator designs, the effect of heat loss savings, and fuel consumption analysis.

It is also appropriate to emphasise here that, following the escalation of fuel costs, any recent analyses will be dominant effect of the recent increase in oil prices.

Table 1. Glass praduction costs

	Year to March 1974 (%)	Year to March 1975 (%)
Raw materials and mixing	36	29
Fuel oil	17	34
Labour	13	10
Maintenance, repair, etc.	34	27

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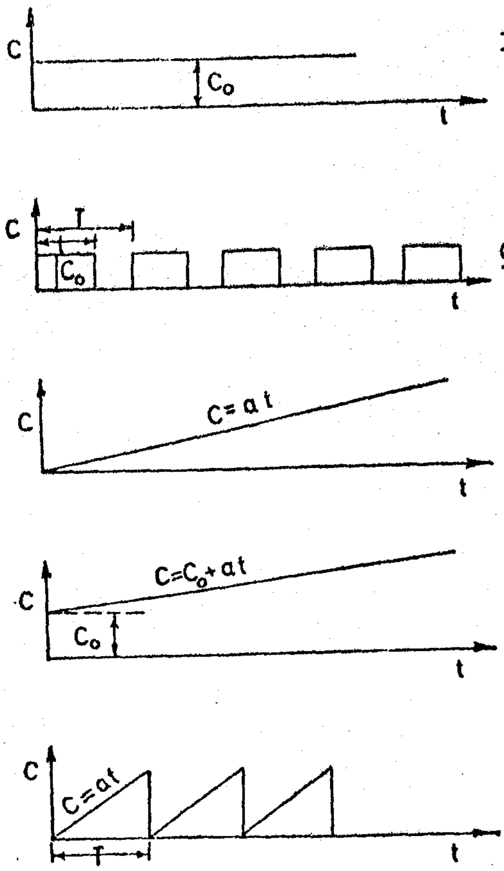
If C(t) is a continuous cost function (£/unit time) then the present value is

$$\int_0^{\infty} C(t)\,\exp(-pt)\,dt$$

which is, mathematically, the Laplace transform of C(t). It is then possible to give fairly simple solutions for the present value of various simple cost functions, as shown in Figure 1.



	Present value
(a) Continuous Constant Cost	C_o/p
(b) Periodic Constant Cost	$\frac{C_o}{P}\,\frac{1-\exp(-pt)}{1-\exp(PT)}$
(c) Linearly Increasing Cost	$\frac{a}{P^2}$
(d)	$\frac{C_o}{P}+\frac{a}{P^2}$
(e)	$\frac{a}{P}\,\frac{(1+pT)\exp(-pT)}{1-\exp(-pT)}$

Figure(1) Some cost functions and their present values

Heat loss sauings and fuel usage:

To offset the rapid increase in fuel costs, the most obvious course of action is to attempt to reduce heat losses from the furnace. However, it is important to realise that the effect of a reduction in heat loss does not show up directly as a saving of the equivalent amount of fuel. The reduction in fuel usage may be.

Table 2. The effect of heat loss savings on fuel consumption.

Sovrce of loss	Method of reducing loss	Fuel saving / Heat loss saving
Crown, Bottom, Sidewalls	Extra insulation	1.9-2.7
Port necks	Extra insulation	0.9-1.2
water cooled equipment	Use of uncooled equipment	2.2-2.8
Holes in furnace structure	Reduction of radiation loss	1.8-2.2
	Reduction of llush' losses	0.3-0.5

greater or less in equivalent thermal terms than the amount of heat saved.

Table(2) summarises the results of many different studies and shows how the actual fuel saving depends on how and where the actual heat loss saving is made.

Chapter (II)

THE CONTROL OF FUEL CONSUMPTION OF GLASS MELTING FURNACES

Statistical model:

The statistical model presented in that included an ageing term as well as a glass weight term

$$y = a_0 + a_1 x_1 \exp(kx_1) + a_2 x_2 \qquad (1)$$

where y is the energy consumption per unit of time, x_1 is the age of furnace in units of time, x_2 is the weight of glass per unit of time, and $a_0, a_1, a_2,$ and k are constants to be determined by fitting the equation to furnace campaign data. It is convenient in practical work to identify the individual terms of the equation as follows:

$$\text{constant term} = a_0 \qquad (2)$$

$$\text{time dependent term} = a_1 x_1 \exp(kx_1) \qquad (3)$$

$$\text{weifht dependent term} = a_2 x_2. \qquad (4)$$

The model was originally conceived for use in feedback control (Figure 1) by analysing the data for a complete campaign, fitting the model and using it to predict the subsequent campaign. It has been applied successfully

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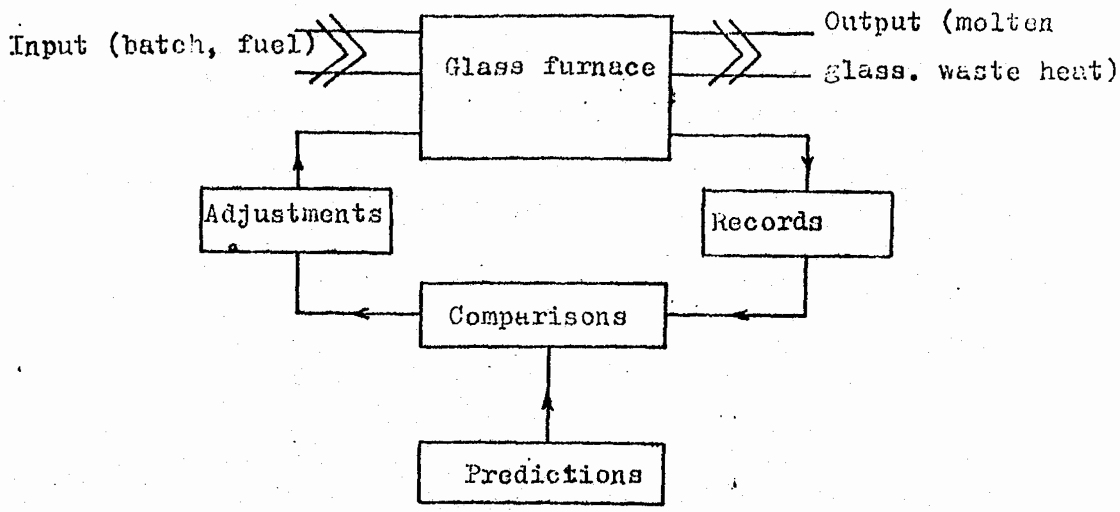


Figure (1) Schomatic of predictive fedback control.

in that role for the day-to-day and week-to-week control of energy consumption in glass melting, but another useful role has emerged that is relevant to the longer term decision procision decision processes of technical management. Using archival data, analyses of successive campaigns of the same furnace or of concurrent campaigns of separate furnaces may be undertaken so that differences in design and construction can be compared and so that the economic aspects of alternative fuels can be evaluated more accurately.

Experience in fitting Equation (1) in practice has led to interesting contrast between the relative importance of the time dependent and the weight dependent terms. In his previous paper the author introduced a term in the heat balance equation to allow for furnace ageing, and sough to find a satisfactory mathematical formulation of the ageing phenomenon and present a suitable method of numerical analysis. Subsequently the interest has shifted somewhat towards the weight dependent term (though the time dependent term remains essential in the heat balance). The linearcum-exponential form of the time dependent term (3) has been entirely satisfactory as a flexible means of fitting into a common formula the ageing characteristic of every furnace that has so far been studied. There has been no difficulty in arriving at suitable values for the constants a_1 and k so to provide a reasonable fit to the available data. In contrast, the evaluation of a_2 in the mathematically weight dependent term (4) has proved to be the more difficult, and more critical, aspect of fitting the model to furnace data.

The reason for this is explained in Figure 2. At any given stage in a furnace campaign the weight dependent term of the prediction equation can be represented as a straight line on a coordinate field of energy and weight. In determining the constants of Equation (1) the procedure is such that the line is constrained to pass through a point corresponding to the energy consumption at an average weight of glass melted, somewhere around the middle of the normal operating range. If the value of the constant a_2 in the weight dependent term is incorrectly estimated (for any reason whatsoever) the slope of the line will be modified.

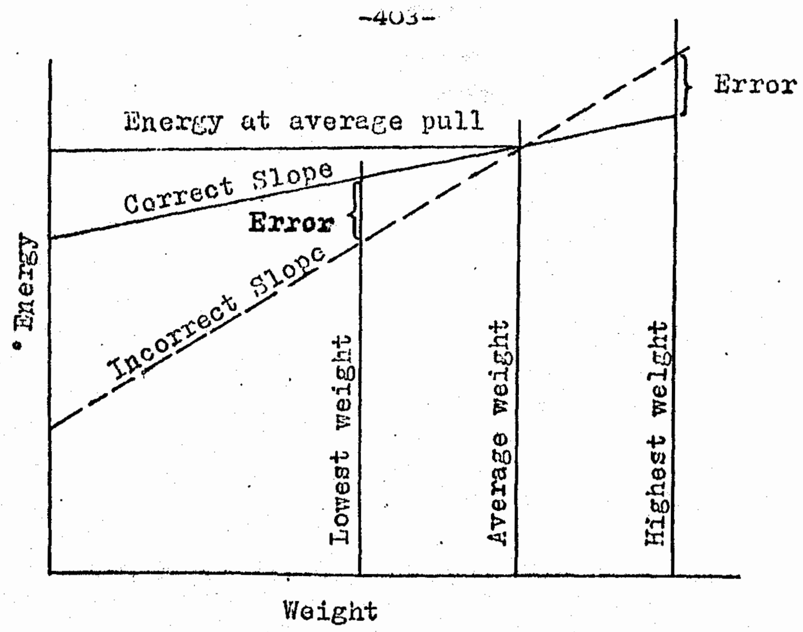


Figure (2) Effect of error in the weight coefficient

The prediction will remain true at average pull, but an error will appear in the prediction at the extremes of weight. If a strict control regime is being operated this error will have undesitable consequences. At one extreme of weight excessive consumption of fuel could actually be encouraged, while at the other there would be an inventive to reduce energy input below the required for good glass quality. The error will, of course, be more acute on a jobbing production line in which there are substantial changes of pull from time to time, than on a strady production line that has only small changes in weight.

It is therefore important that the weight coefficient should be determined with great care.

Chapter (III)

THE COST OF REGENERATOR EFFICIENCY

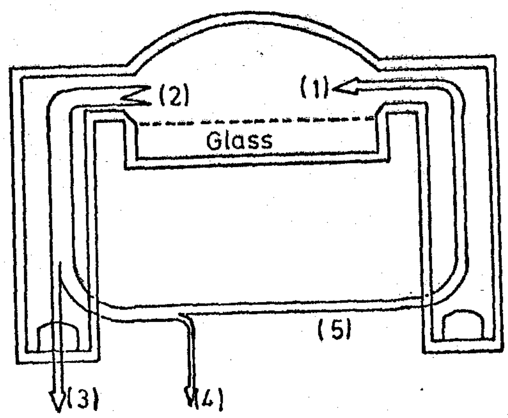
Theoretical background:

Because of the mathematical complexities associated with the full analysis of the regenerative cycle it has been usual to equate the process with a quasi steady state situation as indicated in Figure 1. Average values of temperatures and other parameters have then been applied to obtain an analysis of the regenerator behaviour. There have been two approaches to the analysis and these are to treat the problem as simple heat transfer to and from a plane surface or to consider the problem as analgous to the recuperator.

Heat transfer between a gas and a plane surface.

The applicable equation is similar to that of Gilbert & Mattocks".

$$Q = h.A. \Delta T \qquad (1)$$



(1) Preheat in (2) Heat out (3) Heat lost up chimney.

(4) Losses from regenerators (5) heat returned

Figure (1): the regenerative heat cycle.

in which the total coefficient of heat transfer, h, is a function of convective and radiative heat transfer between the gas and the brick surface. The temperature difference, ΔT, is considered to be the average temperature difference between gas and brick.

The treatment by Arrandale uses

$$h = h_c + h_r \quad (2)$$

$$h_c = 1.0 + 2.71\ p\ v\ \text{Btuh}^{-1}\ \text{ft}^{-2}\ \text{degF}^{-1} \quad (3)$$

$$h_r = 5.5\ \text{Btuh}^{-1}\ \text{ft}^{-2}\ \text{degF}^{-t}\ \text{(for waste gas)} \quad (4)$$

$$h_r = 0\ \text{(for combustion air).} \quad (5)$$

Arrandale estimated the value of h_r from the work of Schack, and it is in fair agreement with figures calculated by the author (Shown in Table 1) from Equation (6), which is discussed in Reference 4.

$$h_r = \frac{0.1713}{(T_G - T_s)}\ \varepsilon_s \left[\varepsilon_G \left(\frac{T_G}{100}\right)^4 - \alpha_G \left(\frac{T_s}{100}\right)^4 \right] \quad (6)$$

Table 1. Coefficient of heat transfer by radiation (Btuh^{-2} ft^{-1} deg F^{-2})

Flue diameter (in)	Arrandole	Equation(6)
6	5.5	4.3
9	-	5.6

Equation (3) is an empirical equation due to Trinks. An alteranative is Equation (7), which is derived in Reference 4 from the work of Nusselt, or Equation (8) due to Rummel:

$$h_c \propto v^{0.8}_{\circ} = \frac{0.32}{D^{0.25}} v^{0.8} \qquad (7)$$

$$h_c = (\text{constant}) \frac{v^{0.5}}{D^{0.33}} \qquad (8)$$

where v is in ft/s and D is in ft.

In all the above Equations the gas velocity is calculated to NTP from Equation (9).

$$v = \bar{\bar{V}}/a \qquad (9)$$

where,

$$\alpha = na_1. \qquad (10)$$

In this calculation the velocity is assumed to be uni-form throughout the regenerator and this is clearly not the real situation. Furthermore, a treatment of this nature ignores the influence of the thermal characteristics of the refractories on the heat transfer coefficiencts, and also the effects of the reversal period which, for instance. If too long can lead to thermal saturation of part of the checkerwork and hence to an effective reduction of A and the efficiency.

Treatment as a recuperator:

The classical work of Rummel and his colleagues has provided the basis for much of the work being carried out today in which the regenerator is treated as a quasi steady state recuperator. It is convenient to refer here to the approximate equation derived by McAdams and quoted recently by Arrandale which provides as base for the calculation of heat transferred.

$$Q = \frac{A.(T_1-T_2)}{\left[\ln \frac{T_1}{T_2}\right]\left[\frac{1}{h_1\theta_1} + \frac{1}{h_2\theta_2} + \frac{1}{2.5c_b\, L\rho_b} + \frac{1}{K_s(\theta_1 + \theta_2)}\right]} \qquad (11)$$

where ΔT_1 and ΔT_2 are the average temperature differences between gas and air at the two ends of the regenerator.

Whilst Equation (11) represents a sound approach, the furnace designer is still left with the problems of determining reasonable values of ΔT_1, ΔT_2, h_1, and h_2 which would assume prior knowledge of the regenerator. In principle it may be reasonable to apply Equations (2)-(10) to estimate the heat transfer coefficients, but the process may be quite inaccurate because of the assumpitions made.

Chapter (IV)

APPLICATION OF THE FINITE ELEMENT METHOD TO CALCULATE FLOW PATTERNS IN GLASS TANK FURNACES.

A two dimensional tank furnace:

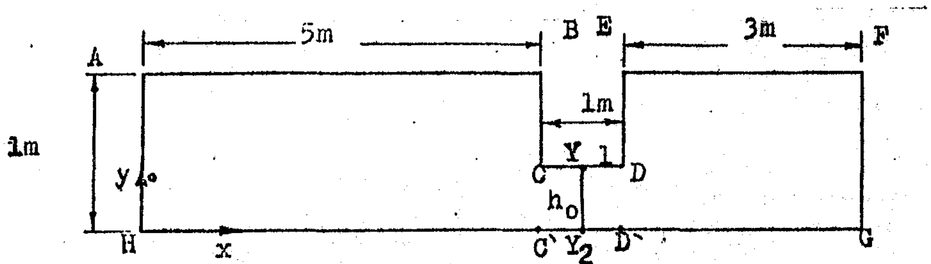


Figure (1) Representation of a longitudinal section of a tank furnace.

There in no great difficulty in applying either the finite difference or the finite element method to the more complicated geometry shown in Figure(1). This is intended to represent a longitudinal section of a container tank furnace. However, one must rememberthat we are at present limited to the solution of two dimensional problems.The solutions we obtain could only be of practical value if the throat extended across the full width of the furnace. We are not able to make even an approximate allowance for the fact that the glass has to flow inwards towards the central axis of the tank to pass through the throat.

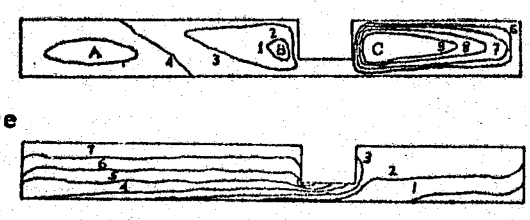
To deal with the problem properly it will be necessary to solve the complete three dimensional problem; this can be done, but at present the computing time required is large. The temperature boundary conditions which were specified for this problem are along AB($T = T_{max}$ (uniform)) at G($T = T_{min} = 1200°C$), along FE $[T = T_{min} + (T_{max} - T_{min})/3]$

at H$[T = T_{min} + (T_{max} - T_{min})/2,]$ and along AH, HG, GF, and EDCB the temperature varies linearly with position, AB and EF are free glass surfaces. Along all the other surfaces, the glass is in contact with the refractory.

Figures(2) and (3) show stream function and temperature contours for two problems in both of which $T_{max} - T_{min}$ = 40 degC, which corresponds to a Rayleight number of 10^5, Comparing the diagrams clearly shows the effect of changing the height of the throat from 0.3 to 0.5 m.

(a) Stream lines.

(b) Temperature contours.



Figure(2) Longitudinal contours of stream lines and isotherms (narrow throat)

Contour number	1	3	5	7	9
Stream line value	3.5	0.5	-0.5	-3.0	-6.0
Temperature value	0.125	0.375	0.625	0.875	-

(a) Stream lines.

(b) Temperature contours.

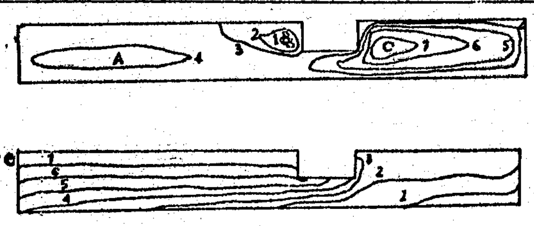


Figure (3) Longitudinal contours of streum lines and isotherms (wide throat).

Contour number	1	2	3	4	5	6	7
Stream line value	1.0	0.5	0.0	-1.0	-3.0	-6.0	-9.0
Temperature value	0.125	0.250	0.375	0.500	0.625	0.750	0.875

As is to be expectdd, the narrower throat causes more separation between the circulatory flows in the two sections of the tank. Table 2 shows the effect of Ray- -leigh number on the values of Ψ_{max} for the circulations A,B, and C for $h_o = 0.5$, A nonuniform triangular mesh was used, with smaller triangles near the boundaires and in the throat region. The total number of mesh points was 770. some further calculations were carried out in which there was a throughput of glass. The effect of this on

Table(1). Effects of Rayleigh number on (Circulation)

Ra	A	B	C
10^4	-0.687	0.488	-2.523
10^5	-1.975	1.171	-11.434
$2x10^5$	-1.397	3.600	-15.990
$3x10^5$	-1.608	4.240	-20.260

the flow pattern and temperature distribution was not very great for the order of throughputs encountered in practice. However, this could be a misleading result. Because of the channelling of glass through the throat which occurs in practice, one would expect that the pull on the tank will have a great effect on the flow pattern near the throat. This could only be investigated properly by carrying out a three dimensional calculation.

Flow in the throat.

It is interesting to compare the results of the finite element calculations of the velocity distribution in the throat with those obtained using the simple analytical equations derived by Peyches and Naruse. The former, which is based on the assumption of zero pull, is

$$v = \frac{\rho_0 g\beta}{6\eta} \left(\frac{T_1 - T_2}{l_0} \right) \left(y^2 - \frac{h_0^2}{4} \right) y \qquad (1)$$

where β is the cubical expansion coefficient of the glass, η is its viscosity, ρ_0 its density, g the acceleration due to gravity, T_1 and T_2 are mean temperatures at each end of the throat, and the meaning of the other symbols is clear from figure (4). The equation can be rewritten in a different

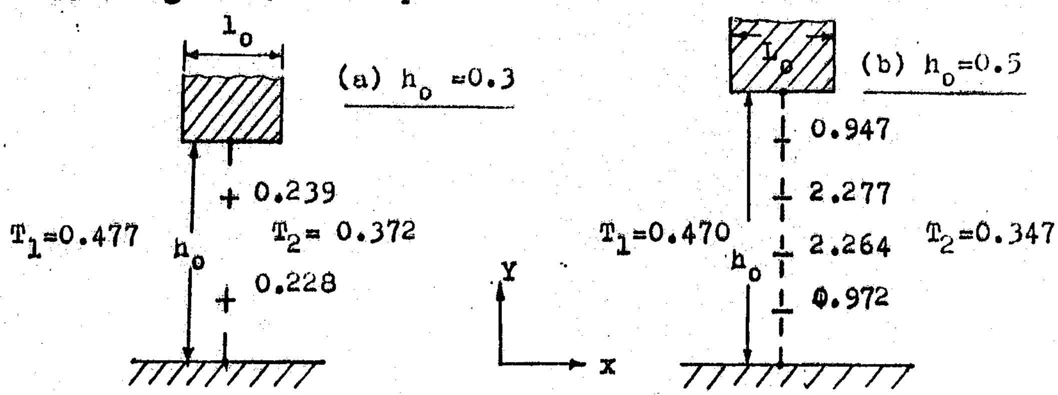


Figure (4) Stream function values for zero pull

from by introducing a dimensionless velocity V where ($V = vl_0/K$,) in which l_0 is the length of glass in the throat and K the thermal diffusivity of the glass. Equation (1) then becomes

$$V = \frac{g\beta\,(T_1 - T_2)}{6vK} \cdot y\,(y^2 - \frac{h_o^2}{4}) \qquad (2)$$

or, writing $t = y/h_o$,

$$V = \frac{Ra^*}{6}\,(t^3 - \frac{t}{4}) \qquad (3)$$

where Ra^* is a Rayleigh number characterising the flow in the throat, which should not be confused with the Rayleigh number used previously, which characterises flow in the whole furnace.

The equation derived by Naruse for the situation when there is also a pull on the tank is $V = V_c + V_v$ where V_c is the dimensionless velocity due to convection and V_p is the dimensionless velocity due to the pull V_c is given by Equation (2) and V_p by

$$V_p = \frac{-3Wl_o}{4\rho_o\,(\frac{h_o^3}{2})\,k}\,(y^2 - \frac{h_o^2}{4}) \qquad (4)$$

where W is the pull per unit width of the throat, which can easily be calculated from the mean residence time of glass in the furnace.

Table (2) Calculation of velocity values.

h_o	$t=y/h_o$	Nomerical	Peyches
0.3	+0.375	+2.391	1.94
	-0.375	-2.277	-1.94
0.5	+0.4	+9.474	9.19
	+0.2	+13.292	10.94
	-0.2	-12.919	-10.94
	-0.4	-9.723	-9.19
	0.0	-0.124	0.0

Figure(4) gives values of stream function calculated at the mid point of the throat for h_o = 0.3 and 0.5 for a situation in which the pull is zero. Table(2) compares the velocity values computed from the stream function values with those calculated from Equation (2). Figure (5) gives values of stream function calculated at the mid point of the throat for h_o = 0.3 and 0.5 for a situation in which the pull corresponds to a mean residence time of 36h.

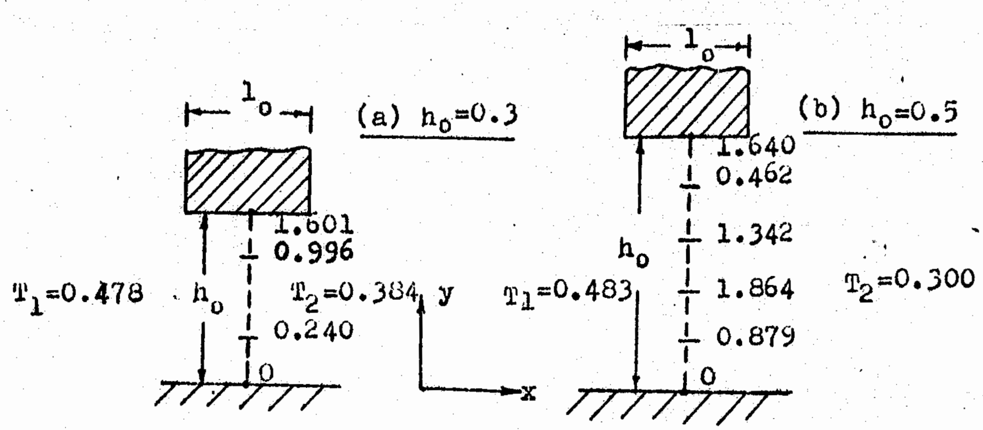


Figure (5) Stream function values for a finite pull.

Table(3) compares the velocity values computed from the stream function values with those calculated from the Naruse equation.

Table(3) Calculation of velocity values.

h_o	$t=y/h_o$	(numerical)	(Naruse)
0.3	+0.375	+6.055	+5.209
	0.0	+7.559	+8.007
	-0.375	-2.397	-1.793
0.5	+0.4	+11.780	+11.334
	+0.2	+18.044	+15.291
	0.0	+5.214	+4.919
	-0.2	-9.947	-7.029
	-0.4	-8.791	-7.792

PART (II) PROPOSED DESIGN PROCEDURE

Chapter (V)

THERMAL CALCULATION OF GLASS MELTING FURNACE

Data:-

(1) W = The weight of the out put Glass =100 kg /hr.

(2) t_1 = Degree of the inner furnace temperature=1400 C°

(3) t_2 = Degree of the outer surface of the furnace=100C°

(4) A_1 = The Melting end Area = 80 X 70 Cm^2

(5) A_2 = The working end Area = 50 X 70 Cm^2

(6) h = The inner hight of the furnace = 60 Cm.

(7) H = The total hight of the furnace = 100 Cm.

(8) Bricks ind which used in furnace:-

I-	For walls	(Silica) λ = 0.004
II-	For Bottom	(Chamotte,2) λ = 0.003
III-	For Chiminay	(Sellemenit) λ = 0.004
VI-	For Tanck	(muleet) λ = 1.1

(9) C.V.= Caloric value fuel = 10000 kcal/kg

(10) Percentage Glass Composition:-

Si 02	74.10 %
$Na_2O + K_2O$	16.69 %
Ca 0	4.65 %
MgO	3.23 %
Al_2O_3	0.33 %
Fe_2O_3	0.065%

(11) Sp= Mean Specific heat of the Colourless Glass=0.292

(A) From Table "Glass Table" The heat balance (The percentage).

1- Radiation and convection losses from the uppen furnace structure = 4.3 %

2- Radiation heat losses Furnace opening = 9 %

3- Sensible heat losses in waste gasses =75.9%

4- Potential heat losses in waste gasses due to incomplete cimbustion = 3.9 %

5- Heat transferred to the Glass = 6.9 %

(B) Percentage of heat lost by conduction through the walls of the lower structure from table

1- Side, Front and back walls containing Glass=49.9%

2- Bottom blocks incontact with Glass =9.1 %

3- Side, Front and back walls not in contact with Glasss = 17 %

4- Bottom structure not in contact with Glass=24%

1- <u>The heat use in Melting:-</u>

$$Qmelt. = W \times Sp \times t_1 \quad kcal/hr.$$

$$Qm = 100 \times 0.292 \times 14000 = 4088o \quad kcal./hr.$$

$$Qm = 6.9 \quad Qtotal$$

$$Qtotul = Qmelt. \times \frac{100}{6.9} \quad Kcal/hr.$$

$$\therefore Qt = 40880 \times \frac{100}{6.9} = \underline{592464} \quad Kcal/hr.$$

II- Total heat losses from Walls:-

$$Qloss. = 4.3\ Qt \quad Kcal/hr.$$

$$Ql = \frac{4.3 \times 592464}{100} = \underline{25476} \quad Kcal/hr.$$

III- Heat distribution:-

1- Side, front and back walls contining Glass=49.9 %

$$Q_1 = 49.9\ Ql \quad Kcal/hr.$$

$$Q_1 = \frac{49.9 \times 25476}{100} = 12712.5 \quad Kcal/hr.$$

Total Area $A_1 = 170\ Cm \times 20\ Cm \times 2 + 70\ Cm \times 20\ Cm \times 2 = 9600$

$$A_1 = 0.96\ mt^2$$

$$q_1 = \frac{Q_1}{A_1}, \quad q_1 = \frac{\Delta t}{Rth}\ Kcal/M^2.hr.$$

$$q_1 = \frac{12712.5}{0.96} = 13242 \quad Kcal/hrm^2$$

$$13242 = \frac{1300 - 400}{Rth} \quad \therefore Rth = \frac{900}{13242} = 0.0679$$

$$Rth = \frac{\delta}{\lambda} \quad \therefore \delta = Rth \times \lambda \quad mt$$

made of mulleat $\lambda = 1.1$

$$\delta_1 = 0.0679 \times 1.1 = 0.07469 \quad mt$$

$$\underline{\delta_1 = 7.5\ Cm}$$

2- Bottom blocks in contact with glass = 9.1%

$$Q_2 = 9.1\%\ Ql \quad Kcal/hr.$$

$$Q_2 = \frac{9.1 \times 25476}{100} = \underline{2319} \quad Kcal.$$

Total Area $A_2 = 70 \times 80 + 70 \times 50 = 9100 \ Cm^2$

$A_2 = 0.91 \ mt^2$

$q_2 = \frac{Q2}{A_2}$, $q_2 = \frac{\Delta t}{Rth.}$ Kcal /m^2. Hr.

$q_2 = \frac{2319}{0.91} = 2548$ Kcal/m^2.hr.

$2548 = \frac{1200 - 400}{Rth}$, $Rth = \frac{800}{2548} = 0.31397$

$Rth = \frac{\delta}{\lambda}$ $\delta = Rth \times \lambda$ mt

$\delta_2 = 0.397 \times 1.1 = 0.345$ mt. $\underline{\delta_2 = 35 \ cm}$

3- Side, front and back walls not in contact with

Glass (Using Silica) = 17%

$Q_3 = 0.17 \ Q1$

$Q_3 = \frac{17 \times 25476}{100} = 4331$ Kcal /hr.

Total Area $A_t = A_a + A_b + A_c \ mt^2$

$A_a = 170 \ cm \times 50 \ cm \times 2$ walls $= 17000 \ cm^2$

$A_b = 110 \ cm \times 50 \ cm \times 2$ walls $= 11000 \ cm^2$

$A_c = \frac{3.14 \times 150}{2} \times 170 \ cm = 40100 \ cm^2$

$Atotal = 1.70 + 1.10 + 4.01 = \underline{6.81} \ mt^2$

$$Q_a = \frac{1.7}{6.81} \times Q_3 = 0.2496 \times 4331 = 1081 \text{ Kcal/hr.}$$

$$Q_b = \frac{1.1}{6.81} \times Q_3 = 0.165 \times 4331 = 699 \text{ Kcal/hr.}$$

$$Q_c = \frac{4.01}{6.81} \times Q_3 = 0.5889 \times 4331 = 2551 \text{ Kcal/hr.}$$

$$q_a = \frac{Q_a}{A_a} = \frac{1081}{1.7} = 635.9 \text{ Kcal /m}^2\text{.hr.}$$

$$q_b = \frac{Q_a}{A_b} = \frac{699}{1.1} = 635.5 \text{ Kcal/m}^2\text{.hr.}$$

$$q_c = \frac{Q_c}{A_c} = \frac{2551}{4.01} = 636.2 \text{ Kcal/m}^2\text{. hr.}$$

$$q = \frac{\Delta t}{Rth}, \quad Rth = \frac{\delta}{\lambda} \qquad \delta = Rth \times \lambda \text{ mt}$$

$$635.9 = \frac{480 - 80}{(Rth)_a} \qquad (Rth)_a = \frac{400}{635.9} = 0.629$$

$$\delta_a = 0.629 \times 0;04 = 0;025 \text{ mt.}$$

$$\underline{\delta_a = 2.5 \text{ cm.}}$$

$$635.5 = \frac{480 - 80}{(Rth)_b} \qquad (Rth)_b = \frac{400}{635.5} = 0.6295$$

$$\delta_b = 0.6295 \times 0.04 = 0.025 \text{ mt}$$

$$\underline{\delta_b = 2.5 \text{ Cm.}}$$

$$636.2 = \frac{1300 - 80}{(Rth)_c} \qquad (Rth)_c = \frac{1220}{636.2} = 1.9176$$

$$\delta_c = 1.9176 \times 0.04 = 0.077 \text{ mt}$$

$$\underline{\delta_c = 7.7 \text{ Cm.}}$$

4- Bottom structure not in contact with Glass=24 %

(Using chamotte no (2))

$Q_4 = 0.24 \; Q1$ Kcal /hr.

$Q_4 = \frac{24 \times 25476}{100} = 6115$ Kcal/hr.

$A_4 = 170$ cm X 110 Cm = 1.87 mt^2

$q_4 = \frac{Q_4}{A_4} = \frac{6115}{1.87} = 3270$ Kcal/m^2.hr.

$q_4 = \frac{\Delta t}{(Rth)_4}$ $\delta_4 = (Rth) \times \lambda$ mt

$3270 = \frac{480 - 80}{Rth.}$ $Rth = \frac{400}{3270} = 0.1223$

$\delta_4 = 0.1223 \times 0.03 = 0.03669$ mt. $\delta_4 = 3.6$ Cm.

The weight of fuel used melting:-

$Q_{total} = Q_{fuel}$ Kcal/hr.

$Q_{Fuel} = 592464$ Kcal /hr.

$Q_{fuel} = (C.V.) \times G$

$Q_{fuel} = 10000 \times G = 592464$

$G = 59.3$ Kg/hr.

$G = V \times \rho$ kg $\rho_{sol.} = 0.83$

$59.3 = V \times 0.83$

$V = \frac{59.3}{0.83} = 71$ lt/hours.

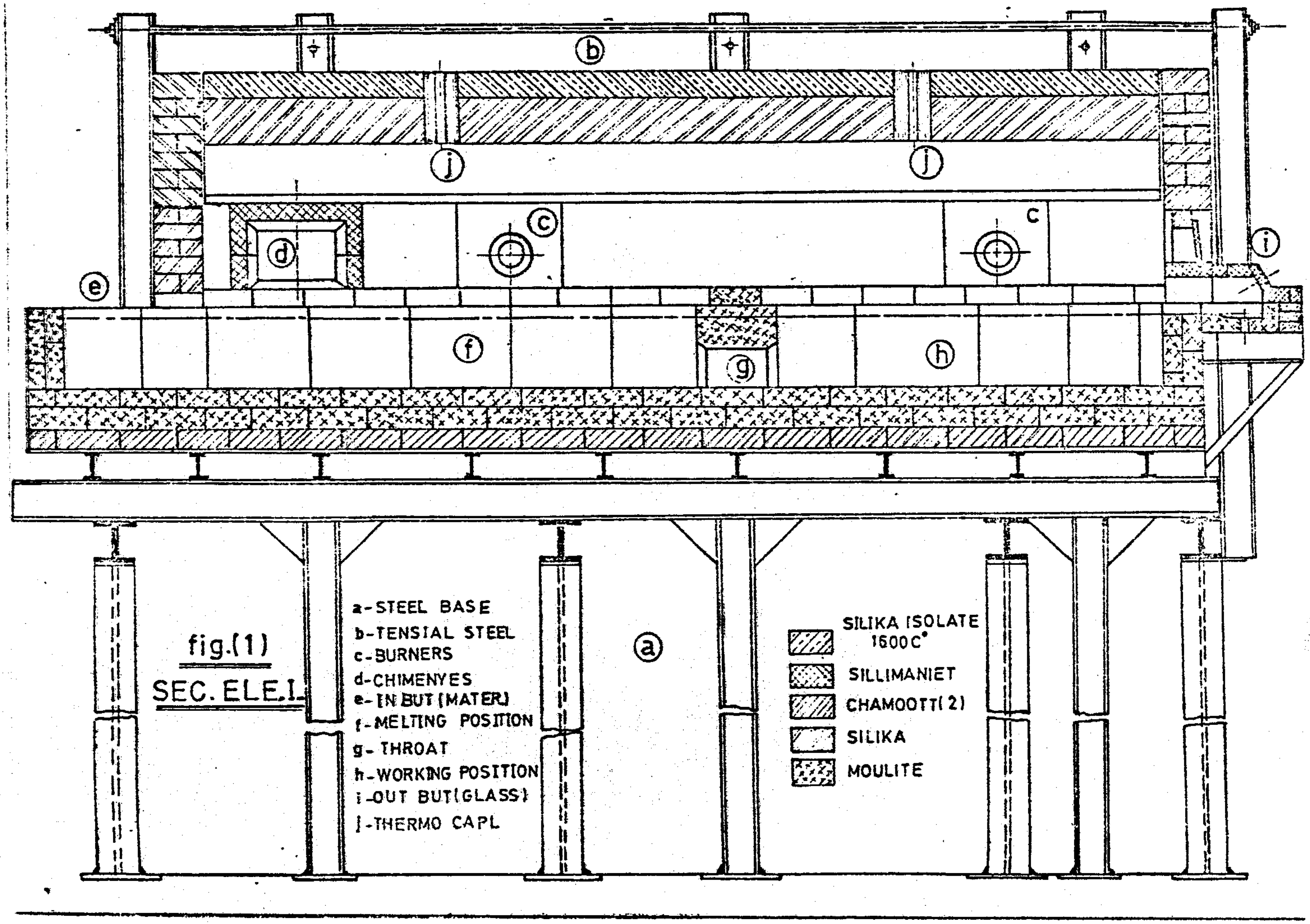


fig.(1)
SEC. ELE.I.

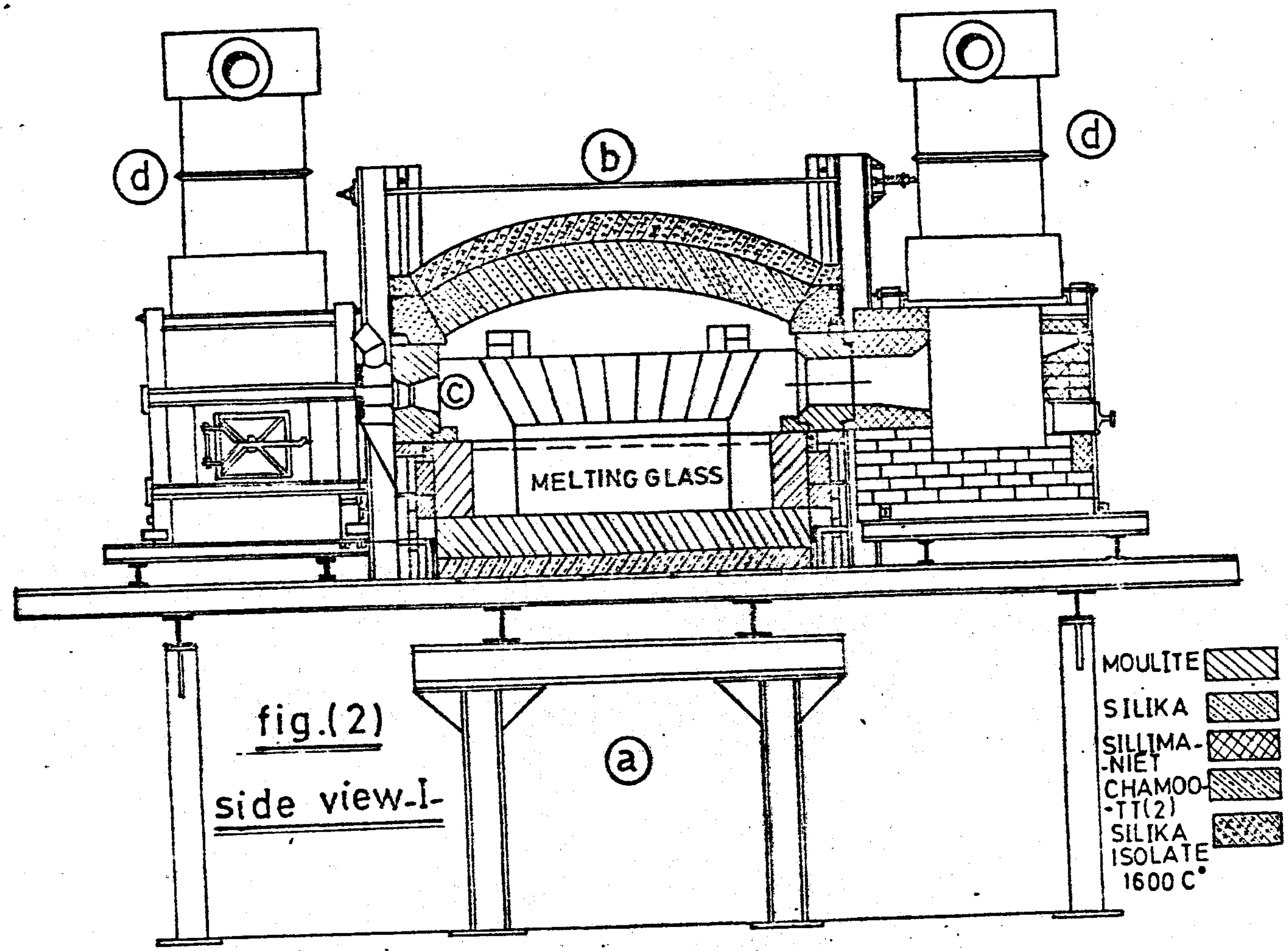
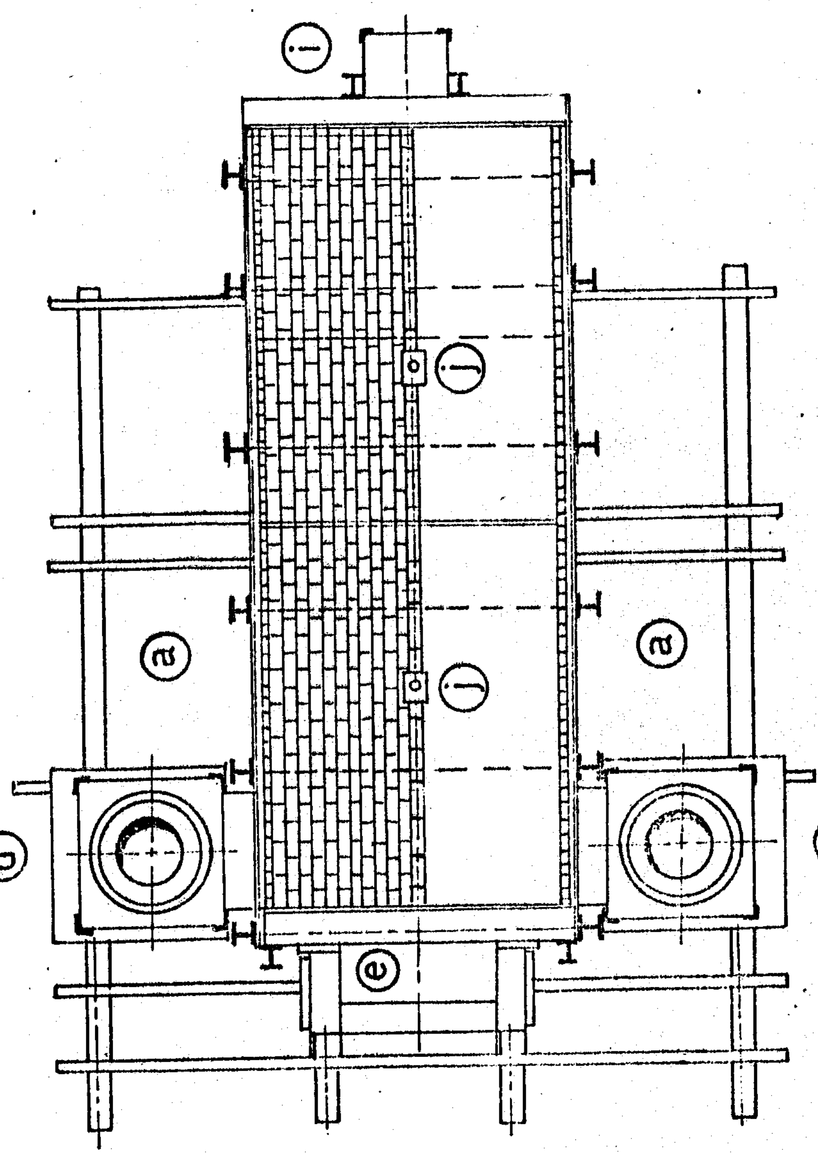


fig.(2)

side view-I-



figuer(3): PLAN-I-

CALCULATION OF AIR USED FOR GLASS FURNACE

The constant Used to calculate
Gaseous specific heats

Gas	A	B
Co_2	0.0451	$1.8x1\bar{0}^5$
H_2o	0.0389	$0.777x1\bar{0}^5$
O_2	0.0349	$0.583x1\bar{0}^5$
N_2	0.0339	$0.409x1\bar{0}^5$

The mean specific heat between temperatures t_2 and T_2 is given by :-

$$\bar{C_p} = A+\frac{B}{2}(T_2+T_1)+\frac{C}{3}(T_2^2+T_2T_1+T_1^2)$$

The equations which used to calculocte Gaseous specific heats:- at(1400 $\overset{o}{C}$) = 1400+273=1673$\overset{o}{K}$

$$C_p(Co_2) = 0.4014+0.1602x1\bar{0}^3t-0.368x1\bar{0}^7t^2$$
$$= 0.4014+0.1602c1\bar{0}^3x1673-0.368x1\bar{0}^7(1673)^2$$
$$= \underline{0.5664}\ Kcal/m^3/\overset{o}{K}$$

$$C_p(H_2o) = 0.3462+6.91x1\bar{0}^5t-0.051x1\bar{0}^7t^2$$
$$= 0.3462+6.91x1\bar{0}^5x1673-0.051x1\bar{0}^7x(1673)^2$$
$$= \underline{0.4475}\ Kcal/m^3/\overset{o}{K}$$

$C_p(O_2) = 0.3101+5.187\times10^{-5}t-1.145\times10^{-8}t^2$

$= 0.3101+5.187\times10^{-5}(1673)-1.145\times10^{-8}\times(1673)^2$

$= \underline{0.3648}$ Kcal/m^3/°K

$C_p(N_2) = 0.3019+3.635\times10^{-5}t-0.513\times10^{-8}t^2$

$= 0.3019+3.635\times10^{-5}\times1673-0.513\times10^{-8}\times(1673)^2$

$= \underline{0.3484}$ Kcal /m^3 / °K

Gas	Specific heats(C_p) kcal/m^3/°K
Co_2	0.5664
H_2o	0.4475
O_2	0.3648
N_2	0.3484

The calculation of air used in combustion:-

Carbon moleculic weight = 12 (C)

Hydrogen moleculic Weight = 1 (H_2)

Volume percentage of Oil: C = 85% H_2= 15%

assume 100 K_g fuel for complet combustion

$$\frac{85}{12}C+\frac{15}{2}H_2+x(O_2+3.76\,N_2$$

$$= L\,Co_2+Y\,H_2o+M\,N_2$$

Where :

L,Y,M,X the comustion constant

The oxygen and nytrogen percentage in air :

For $1m^3$ air

$O_2 = 0.21\ m^3$

$N_2 = 0.79\ m^3$

For 1 Kg air

$O_2 = 0.23$ Kg

$N_2 = 0.77$ Kg

To colculate the specific heat of waste Gases:

at $1400^{\circ}c$

$$Cp\ Gass = V_{Co_2}\%\ C_{p_{co_2}} + V_{H_20} + V_{N2}\%\ C_{p_{N2}}$$

$C_{p_{co2}} = 0.55664$ $Kcal/m^3/\mathring{K}$

$C_{p_{H_2o}} = 0.4475$ $Kcal/m^3/\mathring{K}$

$C_{p_{N_2}} = 0.3484$ $Kcal/m^3/K^{\circ}$

$$V_{co2}\ \% = \frac{L}{L+y+M} = \frac{7.08}{7.08+7.5+40.73} = 0.128$$

$$V_{H_2o}\ \% = \frac{Y}{L+y+M} = \frac{7.5}{7.08+7.5+40.73} = 0.136$$

$$V_{N_2}\ \% = \frac{M}{L+Y+M} = \frac{40.73}{7.08+7.5+40.73} = 0.736$$

$$C_p\ Gases = 0.55664x0.128+0.4475x0.136+0.3484x0.736$$

$$= 0.3885\ Kcal/m^3/\mathring{K}$$

To calculate the specific heat for air:

$$C_{p_{air}} = V_{O_2}\%\ C_{p_{o_2}} + V_{n_2}\%\ C_{p_{N_2}}$$

at(t = 27°) at(t = 27°)

$C_p(O_2)$ at(t = 27°)

$T = 27+273 = 300\ \mathring{c}$

$C_p(O_2) = 0.3101+5.187x10^{-5}x300-1.145x10^{-8}x(300)^2 = \underline{0.3246}$

$C_p(N_2) = 0.3019+3.635x10^{-5}x300-0.513x10^{-8}x(300)^2$

$= \underline{0.3123}$

$C_{p_{air}} = 0.21x0.3246+0.79x0.3123 = \underline{0.3148}$

The combustion equation:

$$\frac{85}{12}C+\frac{15}{2}H_2+x(O_2+3.76N_2)$$

$$= LCo_2+YH_2o+M\ N_2$$

Balance for (C) :

$$L=\frac{85}{12}=7.08$$

Balance for (H_2) :

$$\frac{15}{2}x2 = Yx2$$

$$Y=\frac{15}{2}=7.5$$

Blance for (N_2) :

$$2x3.76x = 2M$$

$$M = 3.76x$$

Balance of (O_2) :

$$2x= 2L+Y$$

$$2x= 2x\frac{85}{12}+\frac{15}{2}$$

$$X= 10.83$$

$$M= 3.76x\frac{65}{6} = 40.73$$

$$\frac{\dot{V}_a}{\dot{V}_f}=\frac{X\ x32\ x\ 100}{21\ x\ 120.482}$$

Where:

0.21 = Oxygen percent in $1m^3$ of air

120.482 = Volume of 100 K_g of Oil

32 = Oxygen molecalic weight

X = oxygen constant in combustion

$$\frac{\dot{V}_a}{\dot{V}_f}=\frac{10.83\ x\ 32\ x100}{21\ x\ 120.482}= 13.7\ M^3{}_{air}/\ m^3_f$$

With exess air 10%

$\dot{V}_a$ = 1.1 x 13.7 = 15.1 m^3air /m^3f

$\dot{V}$ oil = 71.38 lit /hour = 71.38x10^{-3} m^3/ hour

$\dot{V}_a$/hour = 15;1x71.38 = 1077.83 m^3/ hour

$\dot{V}_{Gas}$ = Vair/hr+ $\dot{V}_f$/hr = 1077.83+71.83 = 1149.21m^3/hr

RESULTS

1- Heating up time: 18 hrs.

2- Melting time : 8 hrs.

3- Continious output glass

4- Max. Temperature: 1380 °C

5- Temp. at working end: 1250 °C

6- Glass Melt:- Buble Free.

7- Loss of composition: less than 3%

RECOMMENDATION

The design procedure and the construction of the furnace can be used to design glass furnace with satisfactory results.

References

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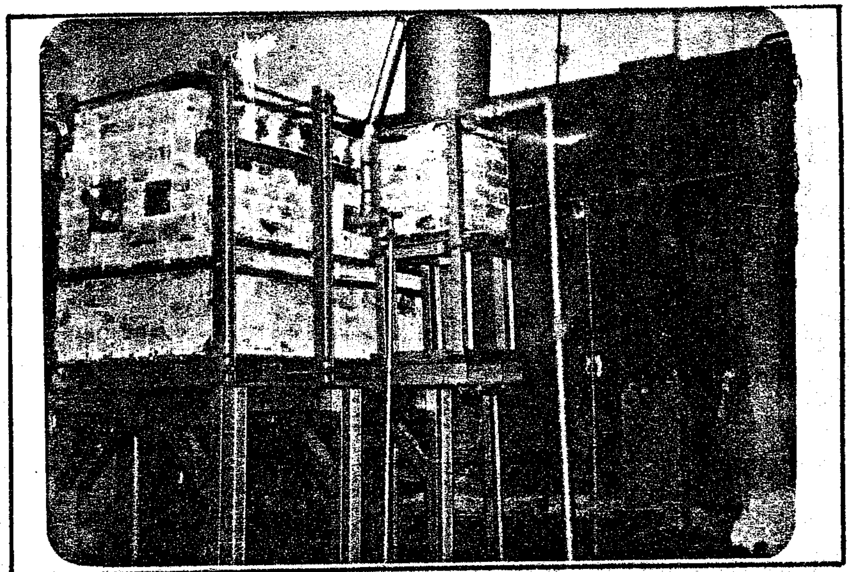

Figure (A): GENERAL FURNACE VIEW

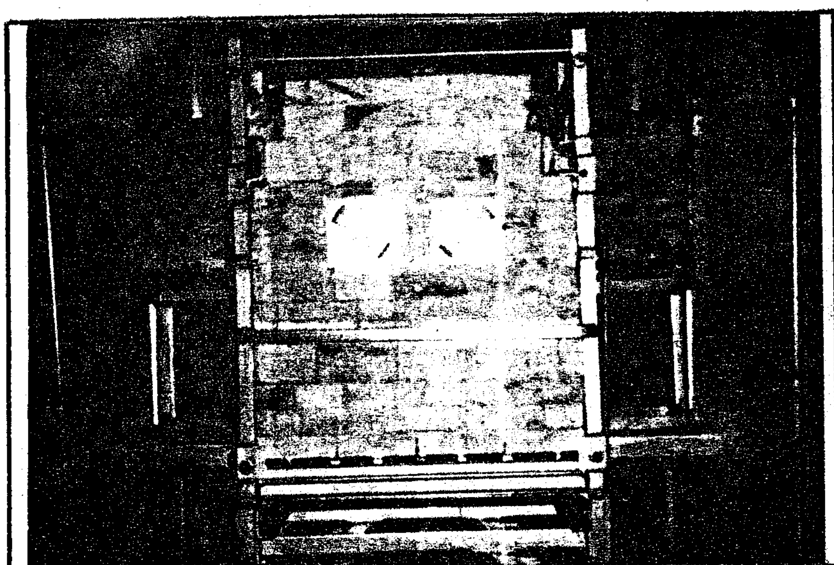

Figure (B): REAR VIEW DIRECTION OF WORKING END.

تصميم فرن معملى لصهر الزجـــاج

أ ٠د ٠ عبد الهادى عبد البارى ناصر ، د ٠ لطفى لويز سيفين ، د ٠ سعاد محمد سراج
م٠ محمد محمود جادو

تم دراسة طرق تصميم أفران الزجاج بالتوازن الحرارى والنماذج الرياضيـــة المتكاملة التى تحل بالحاسبات الالكترونيـــة واستحدث طريقة لتصميم الأفران تتلخـــص فيما يلى :

١ ــ تحديد كمية الحرارة اللازمة لصهر الزجاج طبقا لطاقة الصهر المطلوبـــة ٠
٢ ــ تحديد كمية الحرارة المفقودة من أجزاء الفـــرن المختلفـــة ٠
٣ ــ حساب كميـــة الحرارة المكتسبـــة للزجاج المنصهـــــــــر ٠
٤ ــ تحديد الحرارة المفقودة فى مداخن الأفـــران ٠
٥ ــ حساب كميـــة الوقـــود اللازمـــة للصهـــــر ٠
٦ ــ تغيير أبعاد الفرن الرئيسية حتى الوصول الى الأبعاد التى تفى بالتـــوازن الحرارى بين الحرارة المفقودة والمكتسبة من الزجاج والوقود المستخدم للصهر ٠

وعادة يتم افتراض نسبة بين العرض والطول وفى معظم الأحيان يفضل تثبيت عرض الفرن وتغيير الطول حتى استقرار التوازن ٠

وقد طبقت الطريقة السابقـــة لتصميم فرن معملى يعطى طاقة صهر ١٠٠ كجم لكـــل ساعة وكانت أبعاد الفرن الرئيسية ١٠٠ سنتيمتر عرض ١٧٠ سنتيمتر طول وعمق حـــوض الزجاج ٤٠ سنتيمتر والارتفاع الكلى للفرن ٩٠ سنتيمتر وكانت كمية الحرارة الكليـــــة اللازمة للصهر ٥٩٢٤٦٤ كيلو كالورى / ساعة تعادل ٧١ لتر من الوقود الخفيف فـــى الساعـــة الواحدة ٠

وقد أعطى الفرن النتائج التاليـــة :

١ ــ زمن التسخين ١٨ ساعـــة ٠
٢ ــ زمن الصهر للزجاج ٨ ساعـــات ٠
٣ ــ امداد مستمر من الزجاج المنصهر ٠
٤ ــ أقصى درجة حرارة للزجاج المنصهر ١٣٨٠ $^{\circ}$م ٠
٥ ــ درجة حرارة منطقـــة التشغيل ١٢٥٠ $^{\circ}$م ٠
٦ ــ الزجاج المنصهر خالى من الفقاقيع الهوائيـــة والعيوب الظاهريـــة ٠

٧ ـ نسبة التغير فى التركيب الكيميائى ٣٪ ٠

٨ ـ تركيب الزجاج كما يلـى :

أكسيد سليكون	١٠ر٧٤ ٪
أكسيد صوديوم + أكسيد بوتاسيوم	١٦ر٦٩ ٪
أكسيد كالسيـوم	٤ر٦٥ ٪
أكسيد ماغنسيـوم	٣ر٢٣ ٪
أكسيد ألمنيـوم	٠ر٣٣ ٪
أكسيد حديـد	٠ر٠٦٥ ٪

التوصيـات :

تصلح الطريقـة المستخدمة لتصميم الأفران ونرى استخدامها فى تصميم أفـران الزجاج لصناعة الزجاج المقترحة فى محافظـة المنوفيـة ٠