DESIGN OPTIMIZATION

OF

COKE PUSHER RAM

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By

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SCOPE AND CONTENTS:

In this thesis development of forces, structural analysis and development of a composite computer programme for the design optimization of coke pusher ram is covered. The major aim of the project is to demonstrate the applicability of optimization technique in the design of heavy machines for steel plants, and the development of a standard computer programme which can be used repeatedly in getting an optimized design of a machine element by supplying only the information available from the project designer. The so called composite computer programme developed here needs only the coke oven parameters and related data available from the project designer, and dimensions of the matching elements, to find out the optimum section of the coke pusher ram.

The forces developed theoretically by using emperical factors was verified by taking certain observations at the coke oven battery no. 3 of Steel Company of Canada, Hamilton. Two different optimization techniques are used to confirm the accuracy of the results.

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NOMENCLATURE

SYMBOL	DESCRIPTION
A	Area of cross-section of pusher ram without toothed and guide racks.
A_m	Current consumed by the electric motor of pushing mechanism.
a	Length of the coke oven from end to end.
a_1	Height of the pusher ram section (without toothed and guide rack).
b	Distance of the ram head face from the end of the oven brickwork (coke side) in the forward most position of the pusher ram.
b_r	Width of the toothed rack.
b_w	Minimum width of the oven.
b_1	Width of the ram section (without toothed and guide racks).
c	Distance of the ram head face from the end of the oven brickwork (machine side) in the initial position of the pusher ram.
c_1	Distance between the outside surfaces of the two web plates of the pusher ram section.
C_1	Distance between the pitch line of the toothed rack and outside surface of the bottom plate of ram.
C_2	Distance between the bottom surface of the ram and the bottom surface of the supporting shoe.

d — Distance between the ram head face and C.L. of the supporting shoe.

d_0 — Diameter of the rivet used to connect toothed rack with the pusher ram.

d_1 — Diameter of the rivet hole in the flange of the ram section.

e — Distance between the C.L. of first and second roller support.

E — Modulus of elasticity for structural steel.

F — Force exerted by the driving pinion on the toothed rack fitted with the pusher ram.

F_1 — Force of inertia of coke mass.

F_2 — Frictional resistance due to friction between coke and oven brickwork in normal condition.

F_3 — Total normal force of resistance acting axially on the pusher ram $(F_1 + F_2)$.

F_4 — Net maximum axial force acting on the pusher ram with sticker coke, in the first critical position.

F_5 — Maximum axial force acting on the pusher ram in the second critical position.

F_6 — Net maximum axial force acting on the pusher ram (including frictional force due to friction between oven sole and supporting shoe) in the second critical position.

F_7 — Frictional force due to friction between supporting shoe and oven sole in the third critical position.

F_{s1}	Maximum shear force in the first critical position of the pusher ram.
F_{s2}	Maximum shear force in the second critical position of the pusher ram.
F_{s3}	Maximum shear force in the third critical position of the pusher ram.
f	Distance between the C.L. of second and third roller support.
g	Distance between the end of oven brickwork (machine side) and C.L. of first roller support.
g_1	Accelaration due to gravity.
h	Usefull height of coke oven chamber.
I_x	Moment of inertia of the ram section (without toothed and guide racks) about X- axis.
I_y	Moment of inertia of the ram section (without toothed and guide racks) about Y- axis.
k	Factor for power conversion from H.P. to K.W.
K_1	Coefficient of friction between coke and oven brickwork.
K_2	Coefficient which takes into account the effect of friction of coke with the side wall of the oven.
K_3	Coefficient which takes into account the effect of excessive force of resistance created due to extraordinary stickiness of coke.
K_4	Coefficient of friction between the sole of oven and supporting shoe.

L	Total length of the pusher ram with the ram head.
L_r	Length of the pusher ram without ram head.
m	Length of flange extension of ram section beyond the web plate.
M_{max1}	Maximum bending moment in the first critical position of the pusher ram.
M_{max2}	Maximum bending moment in the second critical position of the pusher ram.
M_{max3}	Maximum bending moment in the third critical position of the pusher ram.
n	Speed of electric motor of the pushing mechanism (r.p.m.)
P_s	Frictional force between supporting shoe and sole of oven in the second critical position of the pusher ram.
q	Distance between the front end of the pusher ram (without ram head) and C.L. of the supporting shoe.
R_γ	Reduction ratio of the reduction gear of the pushing mechanism.
r	Pitch circle radius of the driving pinion of the pusher ram.
T	Total travel of the pusher ram.
T_i	Torque applied by the electric motor of the pushing mechanism.
T_o	Torque at the output shaft on which driving pinion is mounted.

T_2 Total travel of the pusher ram from initial to the second critical position.

t Time of accelaration of the coke mass.

t_a Thickness of the web plate of the pusher ram section.

t_b Thickness of the flange plate of the pusher ram section.

t_{r1} Rise in temperature of the pusher ram in the first critical position.

t_{r2} Rise in temperature of the pusher ram in the second critical position.

t_{r3} Rise in temperature of the pusher ram in the third critical position.

V Volume of coal, charged in the oven.

V_o Voltage employed for the pushing mechanism.

ν Maximum speed of the pusher ram.

W Self weight of the pusher ram including toothed and guide racks.

W_r Self weight of the toothed and guide racks combined.

W_1 Self weight of the pusher ram section (excluding toothed and guide racks).

W_2 Total weight of the coke mass.

W_3 Weight of the ram head.

W_4 Weight of the supporting shoe assembly.

Y.P. Yield point of the material used for the fabrication of the pusher ram.

y_1 Distance between the C.L. of pusher ram and the pitch line of the toothed rack.

y_2 Distance between the C.L. of pusher ram and bottom surface of the supporting shoe.

Z_x Net modulus of section of the pusher ram (excluding toothed and guide racks) about X-axis.

Z_y Net modulus of section of the pusher ram (excluding toothed and guide racks) about Y-axis.

GREEK SYMBOLS

α Coefficient of thermal expansion for structural steel.

ω Coke output per unit weight of coal charged.

η Transmission efficiency of the pushing mechanism.

ρ Density of coal charge.

ρ_s Density of structural steel.

σ_{b1} Maximum bending stress in the first critical position of the pusher ram.

σ_{b2} Maximum bending stress in the second critical position of the pusher ram.

σ_{b3} Maximum bending stress in the third critical position of the pusher ram.

σ_{d1} Direct compressive stress in the first critical position of the pusher ram.

σ_{d2} Direct compressive stress in the second critical position of the pusher ram.

σ_{d3} Direct compressive stress in the third critical position of the pusher ram.

σ_{max1}	Maximum compressive stress in the pusher ram in first critical position.
σ_{max2}	Maximum compressive stress in the pusher ram in second critical position.
σ_{max3}	Maximum compressive stress in the pusher ram in third critical position.
σ_{t1}	Thermal stress in the pusher ram in the first critical position.
σ_{t2}	Thermal stress in the pusher ram in the second critical position.
σ_{t3}	Thermal stress in the pusher ram in the third critical position.
τ_{s1}	Maximum shear stress in the web plate of the ram section in the first critical position.
τ_{s2}	Maximum shear stress in the web plate of the ram section in the second critical position.
τ_{s3}	Maximum shear stress in the web plate of the ram section in the third critical position.
Δ_{L1}	Deflection at the ram head end of the pusher ram in the first critical position.
Δ_{L2}	Deflection at the ram head end of the pusher ram in the second critical position.
Δ_{L3}	Deflection at the ram head end of the pusher ram in the third critical position.

1. INTRODUCTION

In the development of a new high capacity coke oven, the oven dimensions and all related parameters of the coke oven plant would be changed from existing designs, so all the machines serving the oven would be redesigned.

The present thesis demonstrates the use of non-linear optimization techniques and the computer in obtaining the optimum design of the heavy machines used in a coke oven plant. It also shows how the tedious work of redesigning a machine element could be made easier, faster and more reliable.

The technique is illustrated by optimizing the coke pusher ram which is a vital component of coke pusher machines. Besides the utilization of optimization techniques the thesis shows that for any element of the coke oven machine, a composite computer programme can be developed which can be used to obtain the optimum design with information and data provided by the project designer. The same computer deck can be used again and again by simply changing the input data card. Similar to this, a composite programme for the optimum design of each element of the machine can be developed, so whenever it is required to design these machines for a completely new oven, the work of redesigning will take much less time than it would have normally taken.

2. PRODUCTION OF COKE

2.1 Brief History

For centuries coke has been regarded as one of the major components in the production of iron. Coke was first successfully used in iron making in the United Kingdom in 1709. At that time it was produced by the same method as charcoal. Beehive ovens were introduced in about 1750; very good results being achieved with British coal. But this process did not prove very suitable for continental coal with its very high swelling index. So narrow coking chambers were designed in order to meet those difficulties. The first chambered coke oven battery was installed around 1852 in Belgium. After a long series of experiments, by-product recovery ovens were evolved and first operated by Otto around 1892. This system made possible a very marked increase in thermal efficiency and, later, the use of lean gas such as blast furnace gas, either by itself or mixed with coke oven gas, for the heating of ovens.

In the last few decades there had been rapid development in by-product coke oven plants. The developments were mostly in:

(a) the improvement of regenerators and combustion systems;

(b) the utilization of silica refractories in oven construction, which raised the coke yield per cubic meter;

(c) the improvement of the by-product recovery plant and recovery efficiency;

(d) the perfection of new methods of preparing coal for coking;

(e) an increase in coking chamber dimensions.

In the last decade the developments have been mainly confined to two aspects:

(a) full mechanization of the operation of the coke oven plant;

(b) an increase in oven capacity as far as possible.

A lot of work is being done in the U.S.A., U.S.S.R., U.K., and W. Germany to develop the design of high capacity ovens, their related serving machines and also to introduce maximum possible automation and mechanization in the operation of these plants.

2.2 Coke Oven Battery

The coke oven battery consists of a number of coking chambers, rectangular in section, varying generally from 30 ft. to 50 ft. in length, from 6 ft. to 14 ft. in height and 12 inches to 22 inches in width. From 10 to 100 ovens constitutes a battery of ovens, in which coking chambers alternate with heating chambers on each side of each coking chamber. The regenerative system is underneath the heating and coking chambers. Separating walls between regenerators also serve as foundation walls for the heating and coking chambers. The entire structure is supported either from the ground or by columns under a structural steel base. The coal is charged through openings in the top of the oven, and the coke is pushed out from one end by an electric power driven pusher ram, acting through the other end. Figure 1 shows a cross-section of the oven showing the position of different serving machines. All quenching of hot coke is done outside of the oven. During the coking period, the ends of the oven are closed by refractory lined doors, which must be constructed so as to affect complete hermetic sealing of the oven. To permit the escape of the volatile matter, which must undergo several different treatments to separate the various coal chemicals, an opening is provided through the top at one or both ends of the oven. This opening is fitted with an offtake pipe, which in turn connects with the gas collecting main.

Figure 2 shows a cross-section of a coke oven battery which clearly shows the construction of the oven, heating system, regenerator and foundation.

In normal modern coking practice the coal charge is heated out of contact with air for a period of about 18 hours at a temperature around 1000°C. The time and temperature of coking varies, depending upon the quality of coal.

2.2.1 Coke Pusher Machine

The function of the coke pusher machine is to travel along the length of the battery, remove and replace coke oven doors, push out coke from the oven, open and close leveller bar doors, level the coal charge in the oven, clean doors, frame and flash plates and degraphitize the oven ceiling.

The following mechanisms are installed in the coke pusher machine for serving the oven.

(a) Travelling mechanism

(b) Pushing mechanism

(c) Door latch unscrewing mechanism

(d) Door snatching mechanism

(e) Door turning mechanism

(f) Travelling mechanism for door extractor installation

(g) Door cleaning mechanism

(h) Frame and flash plate cleaning mechanism

(i) Levelling mechanism

(j) Leveller bar door opening mechanism

(k) Spill charge collecting system

(l) Degraphitizing mechanism.

The above are the major mechanisms provided in the coke pusher machine which perform all the operations for serving the oven. The total time taken by the coke pusher machine to serve one oven is about 12 minutes.

2.2.2 Coke Pusher Ram

The coke pusher ram is one of the vital elements of the pushing mechanism of the coke pusher machine. In every cycle of serving operation it pushes out hot coke from the oven. It consists of a welded box beam, a cast steel ram head, toothed rack and guide rack rivetted all along the length of the beam, and a supporting shoe. The whole pusher ram assembly is supported on three rollers which are mounted in three fabricated stanchion which are fixed on the coke pusher machine structure. The pusher ram is driven by an electric motor through reduction gears with a rack and pinion arrangement, the pinion being mounted on the output shaft of the reduction gear and rack rivetted on the bottom flange of the box beam. The driving pinion is mounted on the first stanchion and serves also as the supporting roller. The first stanchion has another roller mounted above the pinion with just sufficient clearance to pass the pusher ram between them. The two other stanchion have two side rollers also in addition to the top and bottom roller. The function of these rollers is to support the pusher ram, keep it straight and minimize resistance during its travel. Figure 3 clearly shows the installation with all its detail.

The box beam is fabricated from ordinary structural steel plate of standard thickness by welding flange plates to web plates. Some manufacturers of coke pusher machine use a H Section for the pusher ram. But it has been observed in practice that a box section has a better resistance against warping due to temperature change. Therefore a box section is preferred over H Section for the purpose. The present project work deals with the optimization of this box section which is to be designed for a specific coke pusher machine, selected for illustration as the one for a PK-2K type Russian designed coke oven of 21.6 cubic meter capacity. This oven has a capacity common for steel plants.

3. DEVELOPMENT OF FORCES ACTING ON THE PUSHER RAM

3.1 Introduction

The Russian PK-2K is anunderjet side fired compound oven. A similar type of oven exists in the Steel Company of Canada, in Hamilton. The dimensions and other characteristics, including size of pusher machine used, are very similar. This allowed the author to use actual results obtained from observations taken at Stelco for the justification of the theoretically calculated values. It will be shown later that the two values closely agree with each other.

As indicated earlier, one of the aims of this project is to develop a standard computer optimization programme. The input data of the computer programme which governs the complete design of the pusher ram are variables that vary from one battery to another. So if it is desired to get an optimized cross-section of the pusher ram for a specific coke oven battery, the parameters of that oven and other data which is normally available from the project drawings of the coke oven plant is simply fed into the computer programme along with the dimensions of the matching elements.

The first step is to determine the total force on the pusher ram in terms of these parameters. Figure 4 shows the cross-section of the coke oven along with the pusher machine, door extractor machine and coal charging machine. This figure corresponds to a project drawing and the parameters indicated in this drawing are normally available to the machine designer before he proceeds with the design of the machine. Table 1 shows the value of these parameters and value of matching dimensions which will be the input of the proposed composite computer programme. In the following formulations these parameters will be frequently used.

During the pushing of coke from the oven the ram must overcome the following resistance.

(a) Force of inertia of the static coke mass.

(b) Force of frictional resistance between the coke and the oven sole and wall.

(c) Force of frictional resistance in the bearings of the rollers and between the ram and the rollers.

In the present problem the force described in (c) will not be considered, since this is only used for calculating the motor power of the drive. Besides this the magnitude of this force is much less than the other two forces.

The force of inertia of the coke exists only at the beginning of the push but the frictional force due to resistance between the coke and brick-work remains throughout the travel of the ram; however it goes on reducing till the end of the travel. The variation of the forces can be seen in Figure 5. The graph shows the magnitude of force at different positions of the ram during its complete forward travel in the oven. The graph shown in Figure 5 has been developed with the help of the current recorder graph of the pushing mechanism of a similar coke pusher machine recorded and supplied by Koppers Co. Inc., Pittsburgh, U. S. A. This current recorder graph shown in Figure 6 was obtained by running the recorder drum at comparitively high speed so as to significantly record all the changes in the current requirement of the electric motor. The graph will be explained in detail in Section 3.6.

The total pushing force which the ram must exert on the coke varies during its travel. At the same time the force characteristics of different batteries are not the same. They not only differ with the ovens of different batteries, but also vary with the ovens of the same battery. This can be observed in the current recorder chart for the pushing mechanism of Stelco battery No. 3 and No. 4 in Figures 7 and Figure 8 respectively.

The magnitude of the required pushing force depends on many factors; the main factors can be summarized as follows.

(a) Quantity of coal charged in one oven.

(b) The quality of coke at the time of pushing (it may be perfectly dry coke or sticky coke due to insufficient shrinkage of the coke mass).

(c) Condition of coke oven battery (nature of deformation and type of carbon deposit).

(d) Quality of coal charged (tendency to swell).

(e) Coking process (temperature of coking, and maximum temperature difference along the height of the oven).

Factors such as condition of the coke oven chamber, uneven heating along the height of the oven, and others permanently effect the working, and are difficult to rectify. It is obvious that such effects increase the force necessary for pushing out the coke. In the determination of force for pushing, normally the following three possible cases are considered.

(a) Normal Travel of Coke Mass

This condition assumes that the condition of the oven chamber is satisfactory, i.e. neglecting the deformation of the oven, transverse shrinkage of the coke mass is normal and there is a clearance of 3/4" to 1 1/4" between the coke mass and oven wall.

(b) Close or Tight Travel of Coke Mass

This condition takes into account all the enumerated resisting forces as constantly acting, beside this it includes the condition that the coke is not dry but sticky and the coke shrinkage is comparatively small. It has been observed in practice that the force of pushing in this case is increased from 1.5 - 2.0 times the normal force. This can be seen in Figure 5 and Figure 7.

(c) Dead Stuck Coke

This condition, which occurs rarely in the oven, exists when the coke mass is extraordinarily sticky because of premature coking. If the coking is not complete all the tar and volatile contents are not removed, and this excess tar content creates this condition. In such cases the coke is hammered by the ram many times to push it out. The ramming force is limited by the maximum overload relay, which is adjusted for the magnitude of force which may not damage the oven brickwork. If by a few hammering blows the coke is not moved, then the doors are closed and further coking is continued until it is considered that the coke is dry.

3.2 Inertia Force of Coke Mass

Initially the coke is static before pushing. The pusher ram starts pushing it from one end and brings it to a maximum speed from rest. Figure 9 presents characteristics of speed at different positions during the forward travel of the ram. This was developed by the experimental section of "GIPROKOKS", the Russian coke oven design institute [1].

It should be noted from Figure 9 that the ram travels from its initial position A to B and further travels to reach the face of coke at C. From C it starts pressing the coke, and in doing so the velocity drops to D or D' and sometimes even near to zero. At this point the ram exerts its maximum force to release the whole coke mass from the grip of friction and cohesion between the coke and the oven sole. Due to application of such a high force the whole coke mass is suddenly freed from the grip and starts moving very fast. It has been observed in practice that the time taken to accelerate from D to E is about 0.5 - 0.7 seconds. Since no definite prediction is possible about the location of D, a safe calculation of inertia force would assume that D is on the x-axis. This is a conservative assumption. However, since the inertia force contributes only about 3 1/2% of the total force, its effect is not very considerable.

Force of inertia

$$F_1 = \frac{W_2}{g_1} \cdot \frac{\nu}{t} \qquad (3.1.1)$$

where

$$W_2 = V\rho\omega \qquad (3.1.2)$$

3.3 Frictional Force Due to Friction Between Coke and Oven Brickwork

The frictional force which will be experienced by the pusher ram during pushing of the coke mass depends on the coefficient of friction K_1 between the hot coke and oven sole, and the coefficient K_2 which takes into account the additional resistance arising from the friction between the coke and wall of the oven chamber. The values of these coefficients to a significant degree depends on the condition (degree of deformation) of the sole and wall of the oven chamber. Many design institutes in the Soviet Union and Coke Oven designing firms in the U. S. A. and Germany have performed experiments to find the value of these coefficients. It has been agreed that the values of K_1 vary between 0.5 - 0.75 and K_2 between 1.2 - 1.3.

Due to the shrinking quality of coke, there is always a gap between the coke mass and the walls of oven. It has been calculated and also found in practice that normally this is 3/4" to 1 1/4". But nevertheless there is significant friction between the coke mass and wall due to stickiness of the coke mass due to the presence of coal tar or due to premature coking.

In practice it has been observed that the coke mass does not start moving from the moment the ram touches the coke. The pushing ram first moves quite a distance while compressing the coke. This distance varies from 2 1/2 ft. to 7 1/2 ft. Only then the coke mass begins to move. The initial compression of the coke creates a pressure on the wall which creates an additional instantaneous resistance against pushing. It has been found that the factor K_2 of value 1.3 takes into account this additional resistance.

The pressing phenomenon of the coke is illustrated in Figure 10.

The normal frictional resistance F_2 can be given by the following expression.

$$F_2 = W_2.K_1.K_2 \qquad (3.2.1)$$

3.4 Maximum Possible Axial Force on Ram in Normal Condition

As discussed earlier, there are two types of resisting forces which act against the motion of the ram. They are the inertia force and the frictional force due to friction between the coke and brickwork. These forces are denoted by F_1 and F_2. So the total maximum force acting on the ram will be the addition of these two forces and expressed as

$$F_3 = F_1 + F_2 \tag{3.3.1}$$

3.5 Maximum Possible Axial Force on Ram in Any Condition

The derived force F_3 is the necessary force to be applied by the pushing ram to push out the coke in its normal condition. But there could be an instance when the pusher ram must overcome a force created due to tight or rough flow of the coke mass as explained above. Therefore it is necessary to multiply the normal force by a factor, which will take into account this unwanted excessive force created during the operation of the coke oven battery. It is difficult to estimate the exact value of this factor, since it varies with the type of coal, condition of oven, coking temperature and variation of temperature along the height of the oven.

So normally the factor is found by actually making an observation on the current consumption of drive motor in the coke oven plant. In this connection work was done by "GIPROKOKS", the Russian coke oven design institute, Kharkov Ukrain by an Engineer Y. M. Obukovskov, and he found that the value of this factor (K_3) varies from 1.5 to 2.0. In the present problem the value of K_3 is taken as 2.0.

Therefore the maximum force of resistance due to friction between the coke mass in the case of sticky conditons can be given as:

$$F_2' = F_2 . K_3 \tag{3.4.1}$$

And the maximum possible force exerted on the pushing ram in a sticky condition of coke is given as:

$$F_4 = F_1 + F_2'$$

or

$$F_4 = F_1 + F_2.K_3 \qquad (3.4.2)$$

The value of F4 in the present problem comes to around 60.2 kips. and that of F_3 as 31.2 kips.

3.6 Verification of the Calculated Values of the Forces

To justify the values of force of resistance obtained by calculation based on earlier experimental results, the author took some observations at the coke pusher machine of battery No. 3 of the Steel Co. of Canada, Hamilton. The author obtained the current recorder chart of the motor used with the drive of the pusher ram. From the characteristics of the pushing mechanism of this pusher machine, the force exerted by the electric motor in the pushing ram to overcome the resisting forces is computed by the following expression.

$$F = \frac{33000 \times V_o \times A_m \times R_Y \times n}{2\pi knr} \qquad (3.5.1)$$

The values supplied by Stelco for this mechanism are:

V_o = Voltage employed by the pushing mechanism = 230 volt D.C.

A_m = Current consumed by the electric motor of the pushing mechanism = variable.

R_Y = Reduction ratio of the reduction gear of the pushing mechanism = 39.2

η = Transmission efficiency of the pushing mechanism = 0.8.

k = Factor for power conversion from H.P. to K.W. = 0.7457.

n = Speed of electric motor of the pushing mechanism = 440 r.p.m.

r = Pitch circle radius of the driving pinion of the pusher ram.

Using equation (3.5.1) the force exerted by the electric motor on the ram was computed for current consumption from 5 amperes to 400 amperes with an interval of 5 amperes and presented in Table 2.

As explained earlier, detailed current characteristics of the electric motor of similar oven was obtained from the Koppers Co. Inc., Pittsburgh, U. S. A. with the help of these characteristics and computed value of the force on the pusher ram, the force characteristics for Stelco coke pusher were developed and are presented in Figure 5. These characteristics represent the behaviour of the coke resistance against the motion of the ram, which is explained as follows:

First the ram is moved from its original position and brought exactly on the edge of the oven brickwork. The ram is left in this position until the pusher operator gets a push out green signal from the coke guide operator. Figure 5 represents the force characteristics of the ram from this position to the forward most position. The initial pressing of coke is performed by the movement of the ram head from "0" to "a". There is a drop of "ab" because some times in the small part of the coke column which is being pressed, there occurs a slip which reduces the force requirement for a fraction of a second. Then pressing again starts until the grip of the coke offers maximum resistance up to "c", when suddenly this grip is released. This sudden motion of the coke attains inertia, and for further movement of coke the force requirement drops down, until the coke mass is decelerated by the time the ram reaches point "e". So, to again accelerate the last portion of the coke mass, the force requirement increases to "f" and remains more or less the same up to "g", and then suddenly drops at the end of its forward travel to reach "h".

The dotted curve bc'c denotes the emergency condition or maximum force requirement in case of rough or sticky coke. This peak "c'" is not normal but occurs occasionally.

It should be noted that the peak load "c" may not occur every time exactly after 9 ft. of travel inside the oven. It depends on the quality of coke and other factors. But normally the peak load lies between 5 ft. and 15 ft. of the ram travel from the edge of the oven brickwork.

Before going into detail of the experimental results it is important to note that the oven of battery No. 3 of the Steel Co. of Canada is nearly the same as that of the Russian battery PK-2K for which the pusher ram is being optimized. Table 3 shows different parameters of the two coke oven plants. From this table it can be seen that the ovens are nearly same. The oven for which the pusher ram is being designed is about 9%

bigger in volume than the Stelco oven.

The maximum current consumption at any time during a period of a few weeks was observed as 400 amperes. The force calculated as per the power consumed by the electric motor of the drive is 54.333 kips. (Table 2). This value is less than what was calculated for the Russian oven, since it is bigger than the Stelco oven. Similarly, normal consumption of current is about 225 amperes with a corresponding force of 30.562 kips, whereas the calculated value comes to 31.200 kips. This comparison shows that the values calculated theoretically are quite justified and somewhat on the safe side.

3.7 Vertical Forces Acting on the Pusher Ram

The following three forces are acting vertically on the pusher ram.

(a) Self weight of the pusher ram, W, which comprises of the weight of the ram section only, W_1, and the weight of the toothed rack and guide rack, W_r. Thus

$$W = W_1 + W_r \tag{3.6.1}$$

where

$$W_1 = \rho_s . A. \tag{3.6.2}$$

(b) Weight of the ram head, W_3

(c) Weight of the supporting shoe assembly, W_4.

4. STRUCTURAL ANALYSIS AND STRESS ANALYSIS OF PUSHER RAM

4.1 Introduction

The following forces are acting on the ram during its complete forward travel while pushing.

(a) Force of resistance of coke, F_3, which varies with the distance of travel. The maximum value of F_3 occurs at a place about 9 ft. inside the oven (Figure 5), and goes on reducing until it becomes zero at the end of travel. The maximum force due to sticky condition is F_4, which occurs occasionally.

(b) Weight of the ram head acting vertically.

(c) Weight of shoe assembly acting vertically.

(d) Self weight of pusher ram acting vertically.

(e) Force acting axially due to thermal stresses generated by a rise in temperature of the ram while it is restricted against axial expansion.

4.2 Critical Positions of Pusher Ram

There are three main positions which are considered critical for the design of the pusher ram.

(a) First Position

The first position is shown in Figure 11, when the ram has entered about 10 ft. inside the oven and has the maximum cantilever condition from the concentrated load due to the weight of the ram head and shoe assembly, and the uniformly distributed self load. In addition, the ram has in this position a maximum magnitude of axial force required to push the coke. Figure 12 shows the loading condition with the loads acting on the pusher ram in the first position. It should be noted that, in practice, only two out of three rollers support the ram, because a clearance of about 1/8" is kept between the ram and rollers for free travel of the ram between these rollers. So for a safe design in this particular condition, it is assumed that the central roller support "B" is not giving any support to the ram. Figure 12 presents the actual mathematical model, showing all loads and reactions in the first position of ram travel.

(b) Second Position

The second position of the pusher ram is shown in Figure 13, when the ram has entered the coke guide. In this position the pusher ram has already pushed about 90% of the coke out of the oven. The last decelerated portion of coke remaining in the coke guide needs excess force for pushing. It can be observed in the graph of Figure 5 that after a drop of force requirement, it suddenly rises for the last portion of travel. This is because of the deceleration of the coke mass as explained earlier.

This is another position of the ram which needs attention. Here the axial force is about 60% of the maximum normal force required to push out the coke. So the axial force F_5 can be given by the expression:

$$F_5 = 0.6 F_2 \qquad (4.1.1)$$

The loading condition for this position of the ram is shown in Figure 14, with all loads and reactions. It should be noted that in this position, besides bending stress due to the shoe weight, the weight of the ram and ram head, there will be additional bending stress due to the frictional force caused by the friction between the shoe and the brickwork, plus the maximum thermal stress due to temperature rise of the pusher ram.

(c) Third Position

The third position of the pusher ram is shown in Figure 15, when the ram has reached the maximum forward position of its travel. At this position all the coke has been pushed out and the ram is about to start its backward travel, and the ram has its maximum span between two supports, giving a maximum bending moment due to its weight. There will be no major axial force except that due to the friction between the shoe and the oven brickwork. It should be noted that the third position is considered to occur a fraction of a second before it stops, so that shoe friction must be considered. Figure 16 illustrates this loading condition.

4.3 Calculation of Span Lengths of Pusher Ram in Different Critical Position

One of the aims of this project is to develop a composite computer programme in which all calculations are included in the programme with the input of only the oven parameters. This section provides the expressions for the calculations of different span lengths for different critical position of the pusher ram, in terms of the oven parameters.

(a) Span Lengths in First Position of Ram

$$L_r = L - (d - q) \quad (4.2.1)$$

$$L_{11} = g + q \quad (4.2.2)$$

$$L_{12} = g \quad (4.2.3)$$

$$L_{13} = e + f \quad (4.2.4)$$

$$L_{14} = L_r - (L_{11} + L_{13}) \quad (4.2.5)$$

(b) Span Lengths in Second Position of Ram

$$T_2 = 0.9T \quad (4.2.6)$$

$$L_{21} = q \quad (4.2.7)$$

$$L_{22} = g - (d + c) + T_2 \quad (4.2.8)$$

$$L_{23} = L_r - (L_{21} + L_{22}) \quad (4.2.9)$$

(c) Span Lengths in Third Position of Ram

$$L_{31} = q \quad (4.2.10)$$

$$L_{32} = g - (d + c) + T \quad (4.2.11)$$

$$L_{33} = L_r - (L_{31} + L_{32}) \quad (4.2.12)$$

4.4 Thermal Stress in the Ram

In every cycle of operation of 12 minutes the pusher ram enters the oven once. The forward and backward travel takes about 50 seconds. Thus the pusher ram is exposed to an oven temperature of 1800°F, and its temperature is always much higher than the atmospheric temperature. To save the ram from cyclic thermal shock, the ram is enclosed in a box lined with asbestos sheet. To find the magnitude of thermal stresses, it was necessary to know the exact rise in temperature of the pusher ram. For this the author took some observations on the coke pusher machine of Stelco coke oven battery No. 3. Surface temperatures at three points along the length of the ram were measured with a thermocouple just before entering the oven and immediately after it came out of the oven. The results are presented in Table 4. It can be noted that there was not more than 50°F rise in temperature at any point of the ram. So if we consider the temperature rise linear with time, which is a fair approximation, the total rise in temperature of the ram in its first critical position is estimated as 7°F, and in second and third critical position as 30°F.

Due to the rise in temperature of the ram, it will expand. This expansion is restricted by the axial force acting on the ram, developing a compressive stress in the ram.

The thermal stress can be expressed as:

$$\sigma_{t1} = \alpha E t_{r1} \quad (4.3.1)$$

$$\sigma_{t2} = \alpha E t_{r2} \quad (4.3.2)$$

$$\sigma_{t3} = \alpha E t_{r3}$$

where σ_{t1} = Thermal stress in the pusher ram in the first critical position.

σ_{t2} = Thermal stress in the pusher ram in the second critical position.

σ_{t3} = Thermal stress in the pusher ram in the third critical position.

α = Coefficient of thermal expansion for structural steel.

E = Modulus of elasticity for structural steel.

t_{r1} = Rise in temperature of the pusher ram in the first critical position.

t_{r2} = Rise in temperature of the pusher ram in the second critical position.

t_{r3} = Rise in temperature of the pusher ram in the third critical position. In this problem $t_{r3} = t_{r2}$.

4.5 Maximum Stress in the Pusher Ram

Since there are three critical positions of the pusher ram, the maximum stress in each case will be estimated separately.

(a) First Position

The structural analysis of the pusher ram in this position is given in Appendix A in which the maximum bending moment and maximum shear force are found. The maximum compressive stress σ_{max1} in the ram section can be expressed as

$$\sigma_{max1} = \sigma_{b1} + \sigma_{d1} + \sigma_{t1} \tag{4.4.1}$$

where

$$\sigma_{b1} = \frac{M_{max1}}{Z_x} \tag{4.4.2}$$

$$\sigma_{d1} = \frac{F_4}{A} \qquad (4.4.3)$$

$$\sigma_{t1} = \alpha E t_{r1} \qquad (4.4.4)$$

or

$$\sigma_{max1} = \frac{M_{max1}}{Z_x} + \frac{F_4}{A} + \alpha E t_{r1} \qquad (4.4.5)$$

where

σ_{b1} = Maximum bending stress.

σ_{d1} = Direct compressive stress.

M_{max1} = Maximum bending moment.

F_4 = Maximum axial force.

A = Area of cross-section of the pusher ram.

Z_x = Modulus of section of the pusher ram about x - x.

The maximum shear stress in the web of the section of the ram is given as

$$\tau_{s1} = \frac{F_{s1}}{I_x t_a} \left[t_b(b_1 - 2d_1)\left(\frac{a_1}{2} - \frac{t_b}{2}\right) + t_a\left(\frac{a_1}{2} - t_b\right)^2 \right] \qquad (4.4.6)$$

where

τ_{s1} = Maximum shear stress.

F_{s1} = Maximum shear force.

I_x = Moment of inertia of ram section about x - x.

a_1 = Height of the ram section.

b_1 = Width of the ram section.

t_a = Thickness of web plate of the ram section.

t_b = Thickness of flange plate of the ram section.

d_1 = Diameter of hole in the flange for rivet.

(b) Second Position

The structural analysis of the pusher ram in this position is given in Appendix B in which the maximum bending moment and shear force are again found. The maximum compressive stress in the ram section can be expressed as

$$\sigma_{max2} = \sigma_{b2} + \sigma_{d2} + \sigma_{t2} \qquad (4.4.7)$$

where

$$\sigma_{b2} = \frac{M_{max2}}{Z_x} \qquad (4.4.8)$$

$$\sigma_{d2} = \frac{F_6}{A} \qquad (4.4.9)$$

$$\sigma_{t2} = \alpha E t_{r2} \qquad (4.4.10)$$

If $\sigma_{t2}A$, which is the axial force due to thermal stress is more than F_6, then

$$\sigma_{t2} = \frac{F_6}{A} \qquad (4.4.11)$$

or

$$\sigma_{max2} = \frac{M_{max2}}{Z_x} + \frac{F_6}{A} + \sigma_{t2} \qquad (4.4.12)$$

The maximum shear stress in the web of the section of the ram is given as

$$\tau_{s2} = \frac{F_{s2}}{I_x t_a} [\ t_b(b_1 - 2d_1)\ (\frac{a_1}{2} - \frac{t_b}{2}) + t_a(\frac{a_1}{2} - t_b)^2] \qquad (4.4.13)$$

(c) Third Position

The structural analysis of the pusher ram in this position is given in Appendix C in which the maximum bending moment and shear force are found. The maximum compressive stress in the ram section can be expressed as

$$\sigma_{max3} = \sigma_{b3} + \sigma_{d3} + \sigma_{t3} \tag{4.4.14}$$

where

$$\sigma_{b3} = \frac{M_{max3}}{Z_x} \tag{4.4.15}$$

$$\sigma_{d3} = \frac{F_7}{A} \tag{4.4.16}$$

$$\sigma_{t3} = \alpha E t_{r3} \tag{4.4.17}$$

If $\sigma_{t3}.A$, which is the axial force due to thermal stress in more than F_7 then

$$\sigma_{t3} = \frac{F_7}{A} \tag{4.4.18}$$

$$\sigma_{max3} = \frac{M_{max3}}{Z_x} + \frac{F_7}{A} + \sigma_{t3} \tag{4.4.19}$$

The maximum shear stress in the web of the section of the ram is given as

$$\tau_{s3} = \frac{F_{s3}}{I_x t_a} [\ t_b(b_1 - 2d_1)\ (\frac{a_1}{2} - t_b)$$

$$+ \; t_a \left(\frac{a_1}{2} - t_b\right)^2] \qquad (4.4.20)$$

4.6 Deflection of the Pusher Ram

When the pusher ram enters the oven so that the supporting shoe is resting on the sole of the oven, there is a definite gap of about 7/8" kept between the sole of the oven and the bottom part of the ram head. Besides this, the bottom face of the ram head is designed such that it can play vertically (Figure 3) to allow any small obstacle to pass without interference. This is done because in an old oven, pits are formed in its sole, and the sharp rigid edge of a ram head may catch on these and destroy the brickwork of the sole. To avoid this while designing the pusher ram, a deflection more than 2.0" is not allowed at the ram head end of the pusher ram.

This deflection is found for the three positions.

(a) Deflection at the ram head end of the pusher ram in the first position, is calculated in Appendix A and expressed as:

$$\Delta L_1 = \frac{3}{3EI_x - F_4L_{13}L_{11}} [\{\frac{WL_{11}^4}{8} + \frac{W_3L_{11}^3}{3} + \frac{W_4}{6} (3L_{12}L_{11} - L_{12}^3)\}$$

$$- \{\frac{WL_{13}^3L_{11}}{24} - \frac{WL_{14}^4L_{13}L_{11}}{12} - \frac{L_{13}L_{11}}{3} (W_3L_{11} + W_4L_{12} + \frac{WL_{11}^2}{2}$$

$$- F_4y_1)\}] \qquad (4.5.1$$

(b) Deflection at the ram head end of the pusher ram in the second position, is calculated in Appendix B and expressed as:

$$\Delta L_2 = \frac{3}{3EI_x - F_5L_{22}L_{11}} \left[\frac{WL_{21}^4}{8} + \frac{W_3L_{21}^3}{3} - \frac{WL_{22}^3L_{21}}{24} + \frac{L_{22}L_{21}}{6}\right.$$
$$\left.\left(\frac{WL_{23}^2}{2} - F_6y_1\right) + \frac{L_{22}L_{11}}{3}\left(W_3L_{21} + \frac{WL_{11}^2}{2} + P_s\ y_2\right)\right] \qquad (4.5.2)$$

(c) Deflection at the ram head end of the pusher ram in the third position is calculated in Appendix C and expressed as:

$$\Delta L_3 = \frac{1}{EI_x}\left[\frac{WL_{31}^4}{8} + \frac{W_3L_{31}^3}{3} - \frac{WL_{32}^3L_{31}}{24} + \frac{W_3L_{32}L_{31}}{3}\right.$$
$$\left. + \frac{F_7y_2L_{32}}{3} + \frac{WL_{32}L_{31}}{6} + \frac{WL_{33}^2L_{32}}{12} + \frac{F_7y_1L_{32}}{6}\right]$$

4.7 Design Limits

The following are the design requirements for the pusher ram beam.

(a) The compressive stress should not exceed half the yield strength of the material at any time during the operation of the pushing mechanism.

(b) The shear stress at any point of the pusher ram section should not exceed one-fifth the yield strength (tensile) of the material in any position of the pusher ram.

(c) The maximum deflection on the ram head end of the pusher ram should not exceed 2.0 inches.

(d) The thickness of the web plate of the ram section should not be less than one-third the thickness of the flange plate, for welding considerations.

(e) The thickness of flange plate should not be less than the diameter of rivet used for fixing the toothed rack with the ram.

(f) The flange width of the ram section should not be less than the width of the toothed rack, and should not be more than 0.88 times the minimum width of oven.

(g) The height of the ram section should be such that it leaves at least one-fifth the height of oven gap under it so as to accommodate the supporting shoe.

4.8 Material Used

Most of the manufacturers of coke oven machines use low tensile strength carbon-silicon structural steel. In the present design problem low tensile strength carbon-silicon steel plate of standard ASTM A284-55T Grade A is used with the following specifications:

Tensile strength = 50000 psi

Yield point minimum = 25000 psi

Elongation in 8" minimum = 25%

Elongation in 2" minimum = 28%.

5. DESIGN OPTIMIZATION

5.1. Formulation of Optimization Problem

The optimization criteria for the pusher ram is weight reduction. The length of ram and density of structural steel being fixed, the objective function becomes the area of cross-section of the pusher ram.

The area of cross-section of the pusher ram is a function of four variables:

a_1 = height of the ram section in inches

b_1 = width of the ram section in inches

t_a = thickness of the web plate of the ram section in inches

t_b = thickness of the flange plate of the ram section in inches

and a constant

d_1 = diameter of the rivet hole in the flange of the ram section.

The objective function can be expressed as:

$$U = A = 2b_1t_b + 2(a_1 - 2b_1)t_a - 4t_bd_1 \qquad (5.1.1)$$

There are ten inequality constraints which define a feasible design as described in section 4.7.

5.1.1 Constraining Functions

I. Limit on the maximum compressive stress in the first critical position of the pusher ram.

$$\sigma_{max1} \leq \frac{Y.P.}{2} \tag{5.1.1.1}$$

II. Limit on the maximum compressive stress in the second critical position of the pusher ram.

$$\sigma_{max2} \leq \frac{Y.P.}{2} \tag{5.1.1.2}$$

III. Limit on the maximum compressive stress in the third critical position of the pusher ram.

$$\sigma_{max3} \leq \frac{Y.P.}{2} \tag{5.1.1.3}$$

IV. Limit on the maximum shear stress in the web plate of the pusher ram section.

$$\tau_{s1} \leq \frac{Y.P.}{5} \tag{5.1.1.4}$$

There are some limitations on dimensions of the ram section due to matching of parts which are already designed and to match with oven dimensions.

V. The width of the ram section should not be less than the width of the toothed rack.

$$b_1 \geq b_r \tag{5.1.1.5}$$

VI. The width of the ram section should not be more than 0.88 times the minimum width of the oven.

$$b_1 \leq 0.88\ b_w \tag{5.1.1.6}$$

VII. The maximum height of the ram section is limited in order to keep a clearance at least one-fifth of the useful height of the coke oven between the bottom face of the ram section and the sole of the oven, in order to permit easy installation of the supporting shoe.

$$\frac{a_1}{2} \leq \left(\frac{h}{3} - \frac{h}{5}\right) \qquad (5.1.1.7)$$

VIII. The thickness of the flange of the pusher ram section should not be less than the diameter of the rivet used to fix the toothed rack with the ram.

$$t_b \geq d_o \qquad (5.1.1.8)$$

IX. To avoid warping of the ram section after welding, the thickness of the web plate should not be less than one-third the thickness of the flange.

$$t_a \geq \frac{t_b}{3} \qquad (5.1.1.9)$$

X. To avoid the damage to the oven brickwork, the deflection of the ram head end of the pusher ram should not exceed 2.0 inches.

$$\Delta_{L1} \leq 2.0 \qquad (5.1.1.10)$$

It may be noted here that constraints to limit the shear stress in the web plate in the second and third critical positions were not formed because the shear force in these two positions were small compared to the shear force in the first position. Similarly, deflection at the ram head end of the pusher ram in the second and third critical positions were very insignificant and amounted to only 3% of the deflection in the first position.

The constraining functions can be summarized as follows, by expressing them in greater than or equal to zero functions, PHI(I).

$$\mathrm{PHI}(1) = \frac{Y.P.}{2} - \sigma_{max1}$$

$$\mathrm{PHI}(2) = \frac{Y.P.}{2} - \sigma_{max2}$$

$$\mathrm{PHI}(3) = \frac{Y.P.}{2} - \sigma_{max3}$$

$$\mathrm{PHI}(4) = \frac{Y.P.}{5} - \sigma_{s1}$$

$$\mathrm{PHI}(5) = b_1 - b_r$$

$$\mathrm{PHI}(6) = 0.88\ b_w - b_1$$

$$\mathrm{PHI}(7) = (\frac{h}{3} - \frac{h}{5}) - \frac{a_1}{2}$$

$$\mathrm{PHI}(8) = t_b - d_o$$

$$\mathrm{PHI}(9) = t_a - \frac{t_b}{3}$$

$$\mathrm{PHI}(10) = 2.0 - \Delta_{L1}$$

The object then, is to minimize the optimization function which is the area of cross-section of the pusher ram which in turn is a function of four variables subject to the condition that none of the ten limits are exceeded.

5.2 Direct Search Method

The direct search method is basically a sequential trial method in which is performed a sequential examination of trial solutions, which are obtained by the numerical evaluation of the objective and constraining functions.

In each iteration of trials, the value of the objective function is compared with the previous best value, and if improvement is observed the search continues in the same direction, otherwise the direction of search is changed. The search continues unilateral and with a pattern move until there is no further improvement in the objective function, showing that either the optimum has been reached or the movement is stuck on some constraint. This is known as stalling of the direct search, which can only be remedied by restarting the search from another point. This is the only limitation with this technique, otherwise it is a fast method for the optimization of nonlinear functions.

The method is not dealt with here in detail because it is already well documented elsewhere [11, 14].

The Direct Search Computer programme used in this problem is a subroutine from "OPTIPAC"*. The programme was modified to suit the problem and integrate it with the composite computer programme developed by the author.

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* "OPTIPAC" is a general composite computer program consisting of different subroutines employing various optimization technique to solve all types of linear and non-linear problems. This was prepared by Graduate Students (1968-69) of Mechanical Engineering Department, McMaster University, Hamilton, for a Design Optimization Course.

5.3 Successive Linear Approximation Technique

Successive linear approximation, which was originally called Method of Approximation Programming (MAP) introduced by Griffith and Stewart [15] to solve oil refinery problems. This method essentially consists of linearizing the non-linear optimization function and constraints followed by a linear programming solution of the linearized functions by the simplex method. This operation is applied repetitively so that the solution of the linear problem converges to the solution of the non-linear problem.

The method is explained as follows:

Let the objective function

$$U = U(x_1, x_2, \ldots\ldots, x_n) = \text{minimum}$$

subject to

$$\phi_j(x_1, x_2, \ldots\ldots, x_n) \leq b_j \qquad j = 1, m$$

$$\psi_k(x_1, x_2, \ldots\ldots, x_n) = d_k \qquad k = 1, p$$

First the functions are linearized at a point $x°$, by expanding them using Taylor's series and ignoring higher orders than linear. The above set of equations becomes:

$$U = U(x_1^{\circ}, \ldots, x_n^{\circ}) + \sum_{i=1}^{n} (x_i - x_i^{\circ}) \frac{\partial U(x_1^{\circ}, \ldots, x_n^{\circ})}{\partial x_i}$$

$$\phi_j(x_1^{\circ}, \ldots, x_n^{\circ}) + \sum_{i=1}^{n} (x_i - x_i^{\circ}) \frac{\partial \phi_j(x_1^{\circ}, \ldots, x_n^{\circ})}{\partial x_i} \leq b_j$$

$$\psi_k\,(x_1^\circ, \ldots, x_n^\circ) + \sum_{i=1}^{n} (x_i - x_i^\circ)\,\frac{\partial\psi_k(x_1^\circ, \ldots, x_n^\circ)}{\partial x_i} = d_k$$

Now let

$$\frac{\partial U(x_1^\circ, \ldots, x_n^\circ)}{\partial x_i} = C_i \quad \text{a constant}$$

$$U^\circ = U(x_1^\circ, \ldots, x_n^\circ)$$

$$\psi_k^\circ = \psi_k(x_1^\circ, \ldots, x_n^\circ)$$

$$\phi_j^\circ = \phi_j(x_1^\circ, \ldots, x_n^\circ)$$

$$\gamma_{ki} = \frac{\partial\psi_k(x_1^\circ, \ldots, x_n^\circ)}{\partial x_i}$$

$$S_{ji} = \frac{\partial\phi_j(x_1^\circ, \ldots, x_n^\circ)}{\partial x_i}$$

$$\delta x_i = (x_i - x_i^\circ)$$

From the above equations, we have:

$$U - U^\circ = \sum_{i=1}^{n} C_i\,\delta x_i = \text{minimum}$$

$$\sum_{i=1}^{n} \gamma_{ki}\,\delta x_i = d_k - \psi_k^\circ$$

$$\sum_{i=1}^{n} S_{ji}\,\delta x_i \leq b_j - \phi_j^\circ$$

The above equations form a problem which can be solved by the simplex method of linear programming.

However, since δx_i may be also a negative quantity,

δx_i is split in two parts as

$$\delta x_i = \delta x_i^+ - \delta x_i^-$$

Although this effectively doubles the number of variables, in the linear programming solution either δx_i^+ or δx_i^- will be zero. Before starting the linear programming, the change in x_i is limited to a small amount, to prevent the linearization from becoming invalid, i.e. $|\delta x_i| \leq m_i$.

The value of m_i is chosen by trial.

5.4. Composite Computer Programme

One of the main aims of this project is to develop a computer programme which can directly produce an optimum design of the pusher ram with the information available from the project drawings and drawings of the matching parts as the input.

The composite computer programme consists mainly of three parts, namely

(a) Composite Programme

(b) Subroutine SEEK

(c) Subroutine APPROX

5.4.1 Composite Programme

This programme first reads into the memory of the computer all of the necessary parameters which are given in Table 1, along with the starting values of the four variables, a_1, b_1, t_a and t_b which are the dimensions of the pusher ram section.

The subroutine SEEK is then called to find the optimized section of the pusher ram using the information available to the composite programme. SEEK finds the optimum by using the direct search method and returns it to the composite programme.

Next the subroutine APPROX is called by the composite programme with the same information that was supplied to subroutine SEEK. APPROX finds the optimum by using the successive linear approximation technique and returns the result to the composite programme.

After receiving an optimum cross-sectional area, and optimum dimensions of the pusher ram from SEEK and APPROX, the composite programme compares the two optimum values to find the best. First it rejects the poorer one, declaring that this method yielded an inferior optimum and then prints out from the method the cross-sectional area, and all the four dimensions of the pusher ram. The plate thicknesses are rounded off by 1/16", because standard steel plates are available only in this increment. Besides this, the width and height of the ram section are also rounded off by 1/16" for easy fabrication.

Then it accepts the best optimum value out of the two and declares it as the optimum design and prints out the cross-sectional area and all four dimensions of the pusher ram, again rounding off all the dimensions by 1/16".

5.4.2. Subrountine SEEK

This subroutine is called by the composite programme to find the optimum cross-sectional area of the pusher ram with the help of information available in the composite programme. It employs the direct search technique to find the optimum. As explained earlier this method does sequential trial search by making unilateral and pattern moves.

Subroutine OPTIMF

Subroutine OPTIMF is called by subroutine SEEK at every trial sequence in the process of search for the optimum point. This subroutine plays a very important part on the whole composite programme. It does the following operations:

(a) Calculates the span lengths of the pusher ram in different critical positions.

(b) Calculates the axial forces acting in the pusher ram in different critical positions.

(c) Calculates the cross-sectional area, moment of inertia, modulus of section and self weight of the pusher ram.

(d) Calculates the maximum bending moment, bending stress, thermal stress, and maximum compressive stress in all the three critical positions of the pusher ram.

(e) Calculates the maximum shear force and shear stress in the web plate of the pusher ram.

(f) Develops and evaluates the following constraints.

$$\text{PHI}(1) = \frac{\text{Y.P.}}{2} - \sigma_{max1}$$

$$\text{PHI}(2) = \frac{\text{Y.P.}}{2} - \sigma_{max2}$$

$$PHI(3) = \frac{Y.P.}{2} - \sigma_{max3}$$

$$PHI(4) = \frac{Y.P.}{5} - \tau s_1$$

$$PHI(5) = b_1 - b_r$$

$$PHI(6) = b_w - b_1$$

$$PHI(7) = (\frac{h}{3} - \frac{h}{5}) - \frac{a_1}{2}$$

$$PHI(8) = t_b - d_o$$

$$PHI(9) = t_a - \frac{t_b}{3}$$

$$PHI(10) = 2.0 - \Delta_{L1}$$

(g) Finally evaluates the objective functions, the cross-sectional area of the pusher ram, and tests for violations of the constraints. If any constraint is violated, its absolute value is multiplied by 10^{20} and added to the value of the objective function. This helps in bringing the search points from the infeasible to feasible region.

5.4.3 Subroutine APPROX

This subroutine is called by the composite programme to find the optimum cross-sectional area of the pusher ram with the help of information available in the composite programme. It employs the successive linear approximation technique to find the optimum design.

During the execution of this method it calls the following subroutines directly or indirectly.

(a) Subroutine MATRIX

(b) Subroutine REALU

(c) Subroutine ENEQ

(d) Subroutine ORDER

(e) Subroutine CONST

(f) Subroutine SIMPLE.

Subroutines a, b, d, and f are taken from the composite programme developed by Gurunathan [11] for his alternate search optimization technique. The subroutines were modified to suit the present problem.

Subroutine MATRIX

This subroutine called by APPROX sets up the simplex matrix from the linearized equations. At every step, the constraints and objective functions are evaluated by calling subrountines CONST, ENEQ and REALU. The values are denoted as ψ_k°, ϕ_j° and U°. Then a small increment STEPX(I) is given to each variable, and the new values for the equality constraints, inequality constraints and objective function ψ_k^1, ϕ_j^1 and U^1 are calculated at the new point. The partial derivatives of the equations are evaluated as explained earlier, and the entire matrix is set up with the slack variables included. To check for any of the B(I), in the matrix $AX = B$, becoming negative, subroutine ORDER is called, to rearrange the equations properly and include artificial variables if necessary.

Subroutine REALU

This subroutine, called by APPROX and MATRIX, calculates the value of the objective function.

Subroutine ENEQ

This subroutine, called by MATRIX, does the same job as subroutine OPTIMF except it does not test for the violation of the constraints as was done in OPTIMF of programme SEEK.

Subroutine ORDER

This subroutine is called by the subroutine MATRIX and tests for any of the B(I)s becoming negative. If any B(I) becomes negative it indicates that a constraint is being violated. ORDER arranges the violated inequality constraint in such a manner that the violated constraints are included in Phase I of the simplex programme.

Subroutine CONST.

The expression for equality constraints, denoted by PSI(I), is presented in this subroutine which is called by subroutine APPROX, MATRIX and REALU. The value of PSI(I) is computed here. In the present problem this subroutine is invalid since there is no equality constraints.

Subroutine SIMPLE

This subroutine is called by APPROX to solve the linearized function using the simplex method of linear programming. It consists of Phase I and Phase II of the standard simplex method, which is formulated and programmed in "Theory of Engineering Design" [10].

6. DISCUSSION

The results obtained from the successive linear approximation technique were very satisfactory and consistent. The optimum always converged to the same point from any starting position. This is presumed to be due to its ability to proceed along a constraining function without stalling. On the other hand the direct search method did not produce consistent results. From almost every starting point the convergence was different. Besides this the step size had a significant effect on direct search, whereas successive linear approximation was only affected to the extent that the convergence time differed, without changing the optimum result.

One advantage of direct search is that the search can be started from even an infeasible region, which is not possible in the case of successive linear approximation. It is also faster, and if by a number of trials the starting point and step size can be selected properly, this method may be preferable. But on the other hand a considerable amount of computer time is wasted in making trial runs to locate a good starting point and step size. Therefore it would seem that the successive linear approximation technique is generally superior to direct search in the present problem. The results from the two methods starting from different points are presented in Table 5 for comparison.

An optional way of using the optimizing technique can be set up by adding and changing a few cards. Direct search is just used to find an optimum starting from any arbitrary point even in the infeasible region. This is then internally used as a starting point for the successive linear approximation method. In this particular problem, this approach was slightly quicker than using the methods in parallel. However no comparative change on the optimum is obtained. The change in the computer programme for this is given in the last page of Appendix E.

Table 6 presents the results from the composite computer programme, after starting from the best selected point.

The optimization of the pusher ram achieved a reduction of about 25% in the weight over the existing design. There will be a further saving in engineering time; a design engineer will spend at least 20 days on the complete analysis and design of the pusher ram, where as the composite computer programme does all that work in 10 seconds. The cost of computer time will be around $2.00 compared to the design engineer who will be paid about $500.00.

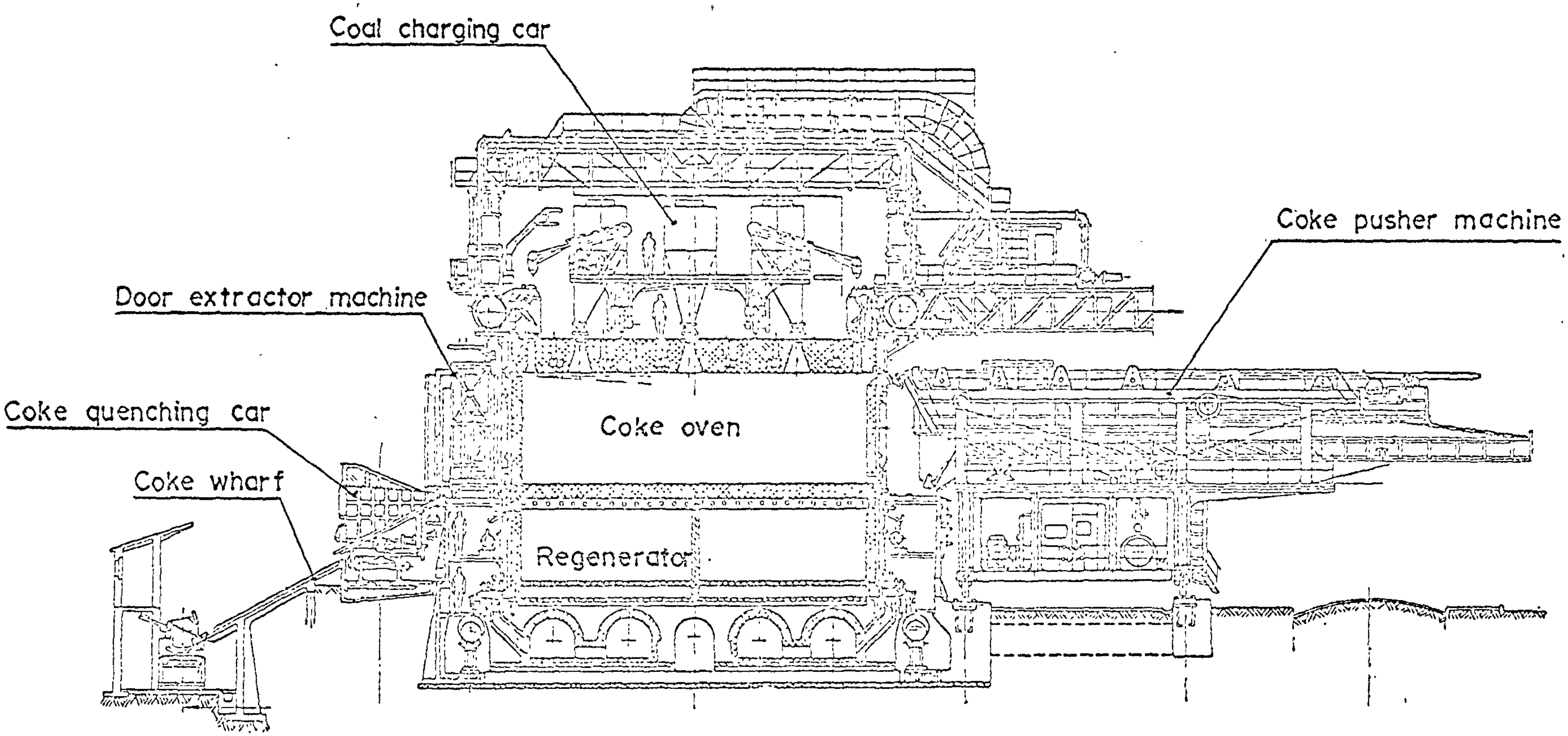
7. CONCLUSIONS

This project demonstrates the use of optimization technique and computerization of the design of heavy machine elements. In this example one component of a coke pusher machine is optimized along with complete computerization of the design process. Similarly, a composite programme for individual major elements could be developed. It is also not difficult to combine the composite programmes of individual elements in one programme to make a design optimization package for a complete machine. Such computerized design optimization techniques can revolutionize the design and development of heavy machines for coke ovens, blast furnaces, steel melting and other plants of the iron and steel making complex.

It is a laborious job preparing such big computer programmes, but once completed it can be used again and again and will be much more economical than designing the machines every time for new ovens.

END OF CONTEXT

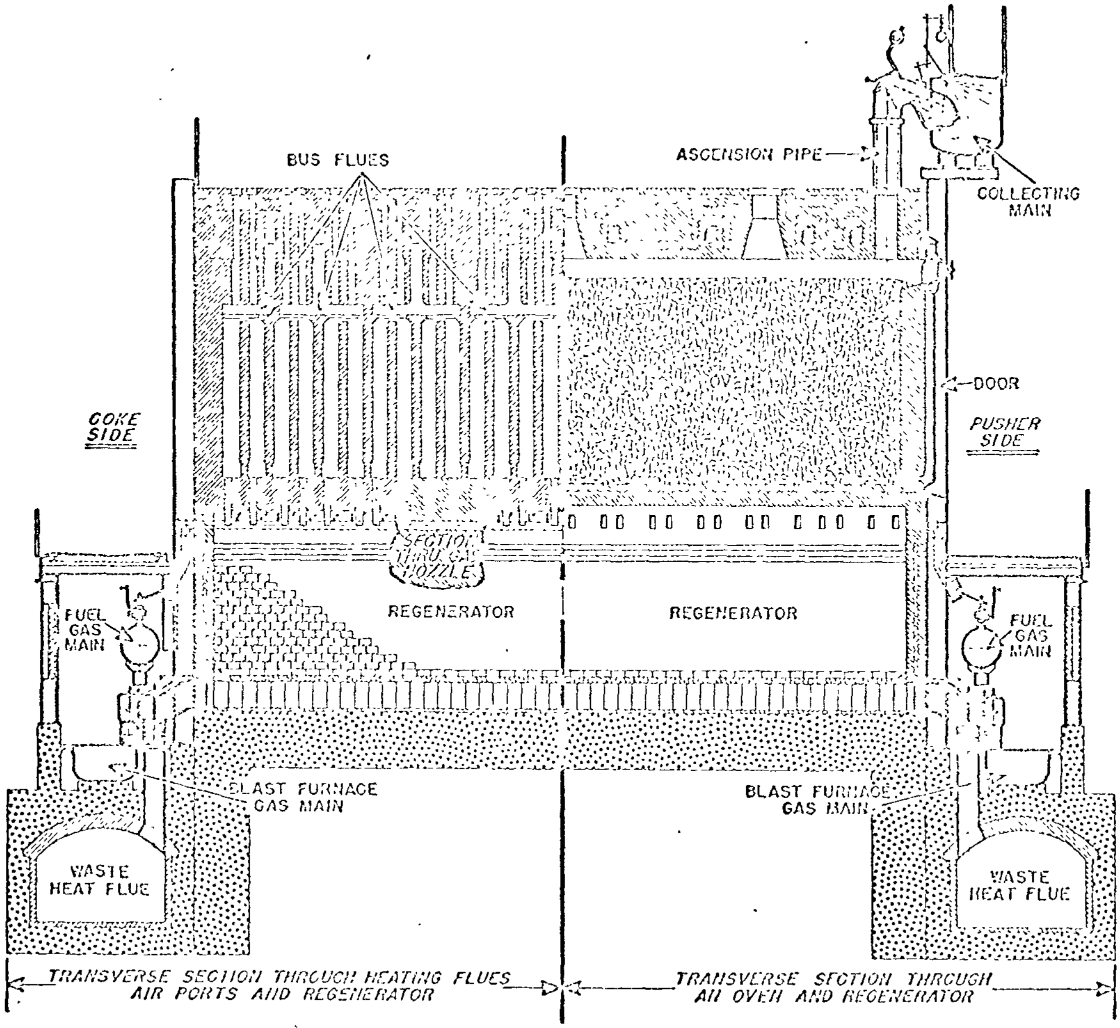
I L L U S T R A T I O N S

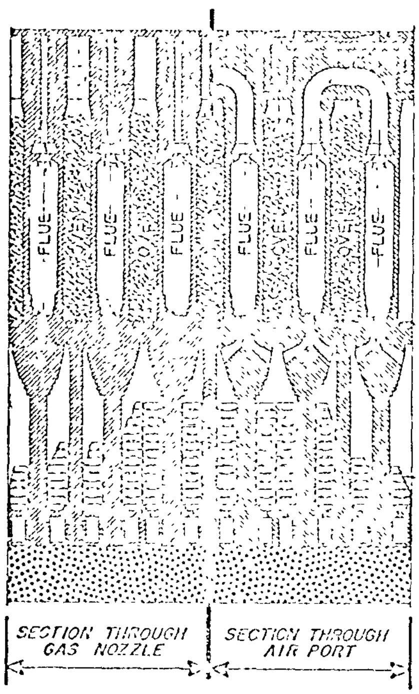


CROSS-SECTION OF COKE OVEN BATTERY

(RUSSIAN DESIGN · PK- 2K)

Figure - I

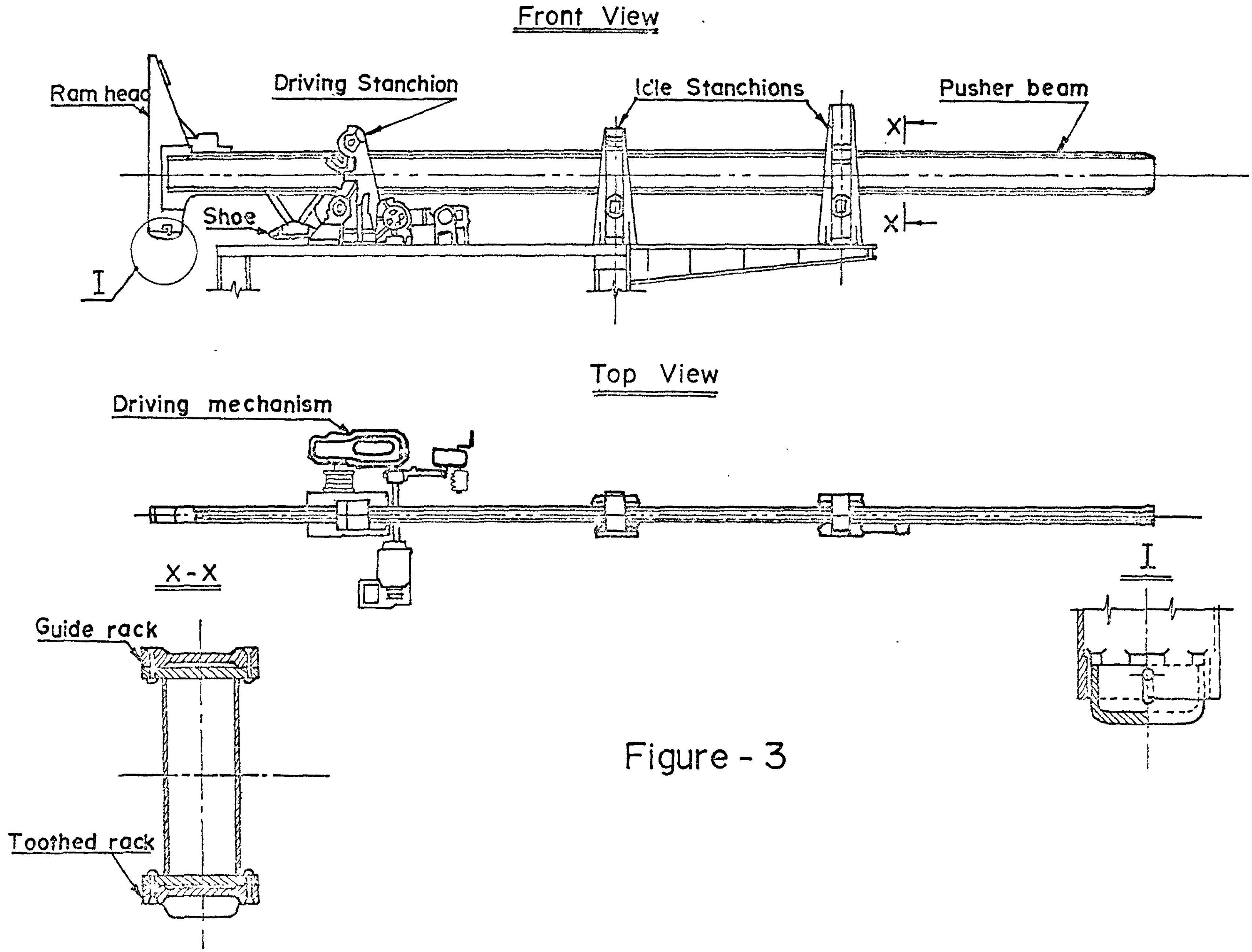




CROSS-SECTION OF A COKE OVEN BATTERY WITH DETAIL OF HEATING SYSTEM

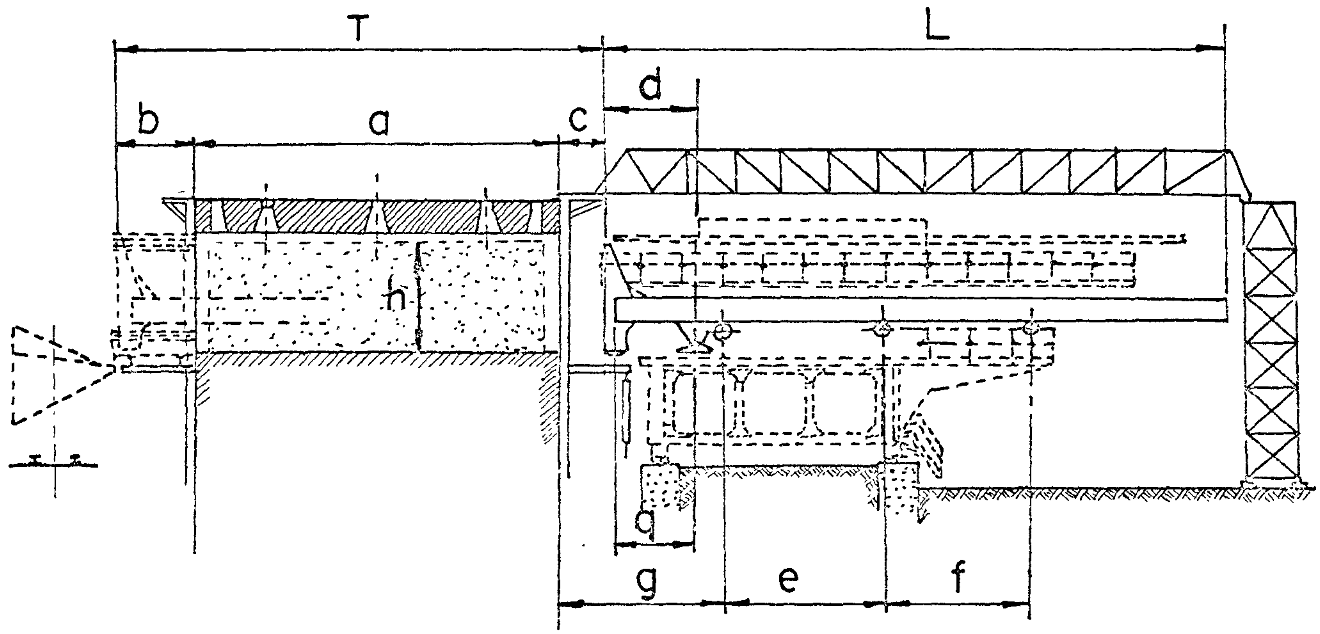
(Reference 5)

Figure - 2



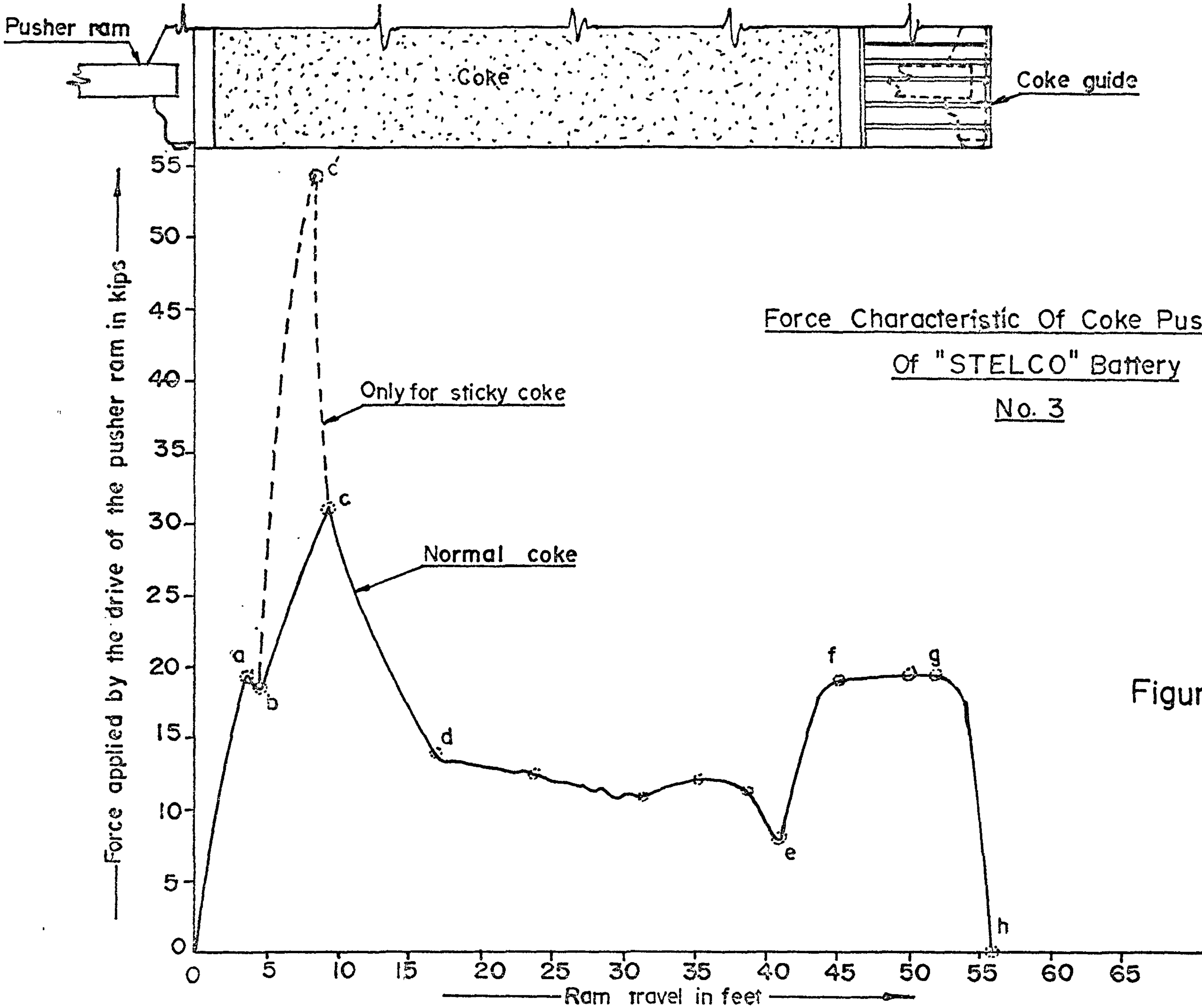
Front View
Ram head
Driving Stanchion
Idle Stanchions
Pusher beam
X
X
Shoe
I
Top View
Driving mechanism
X - X
Guide rack
Toothed rack
I

Figure - 3



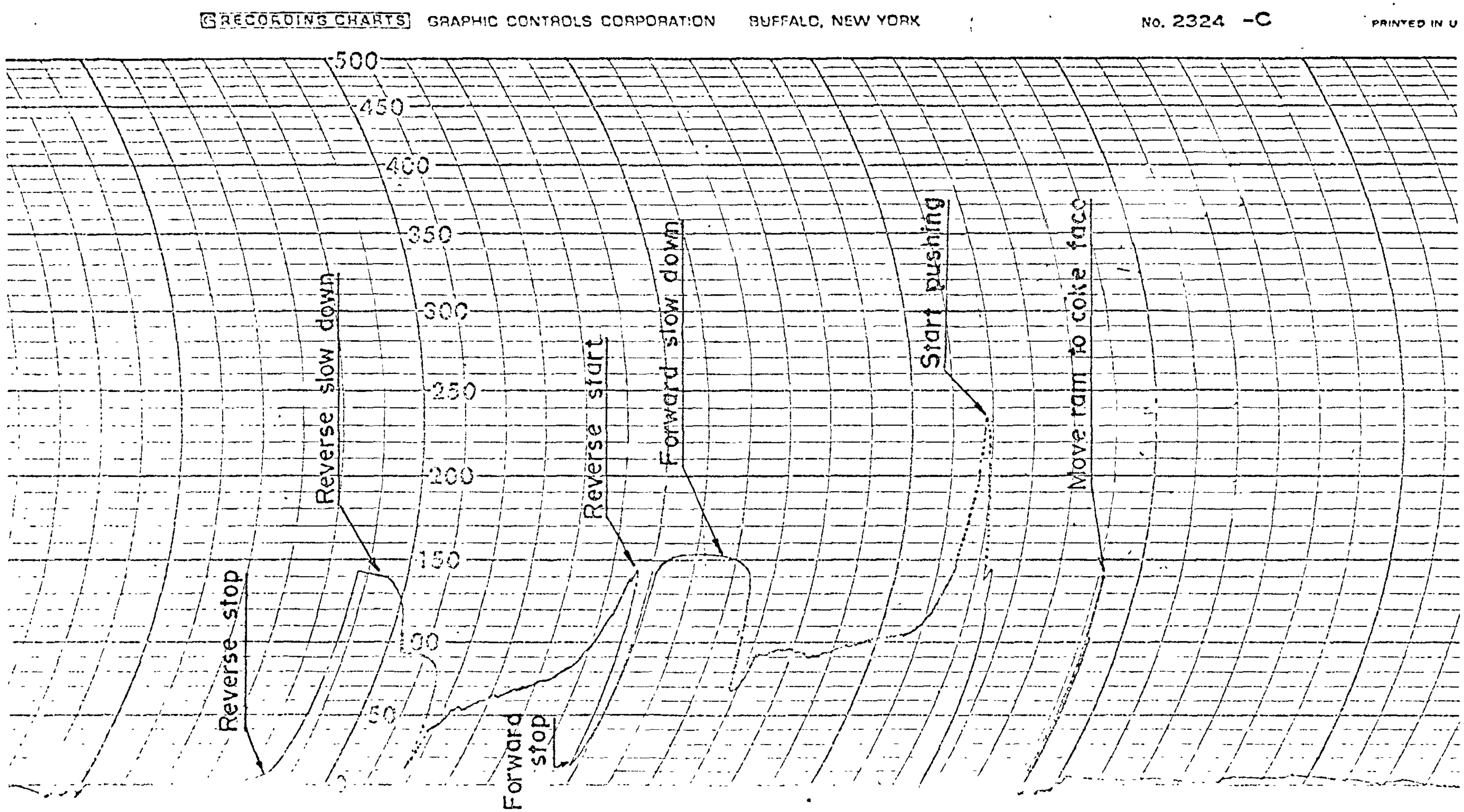
Project Drawing Of Coke Oven Plant

Figure - 4



Force Characteristic Of Coke Pusher Ram
Of "STELCO" Battery
No. 3

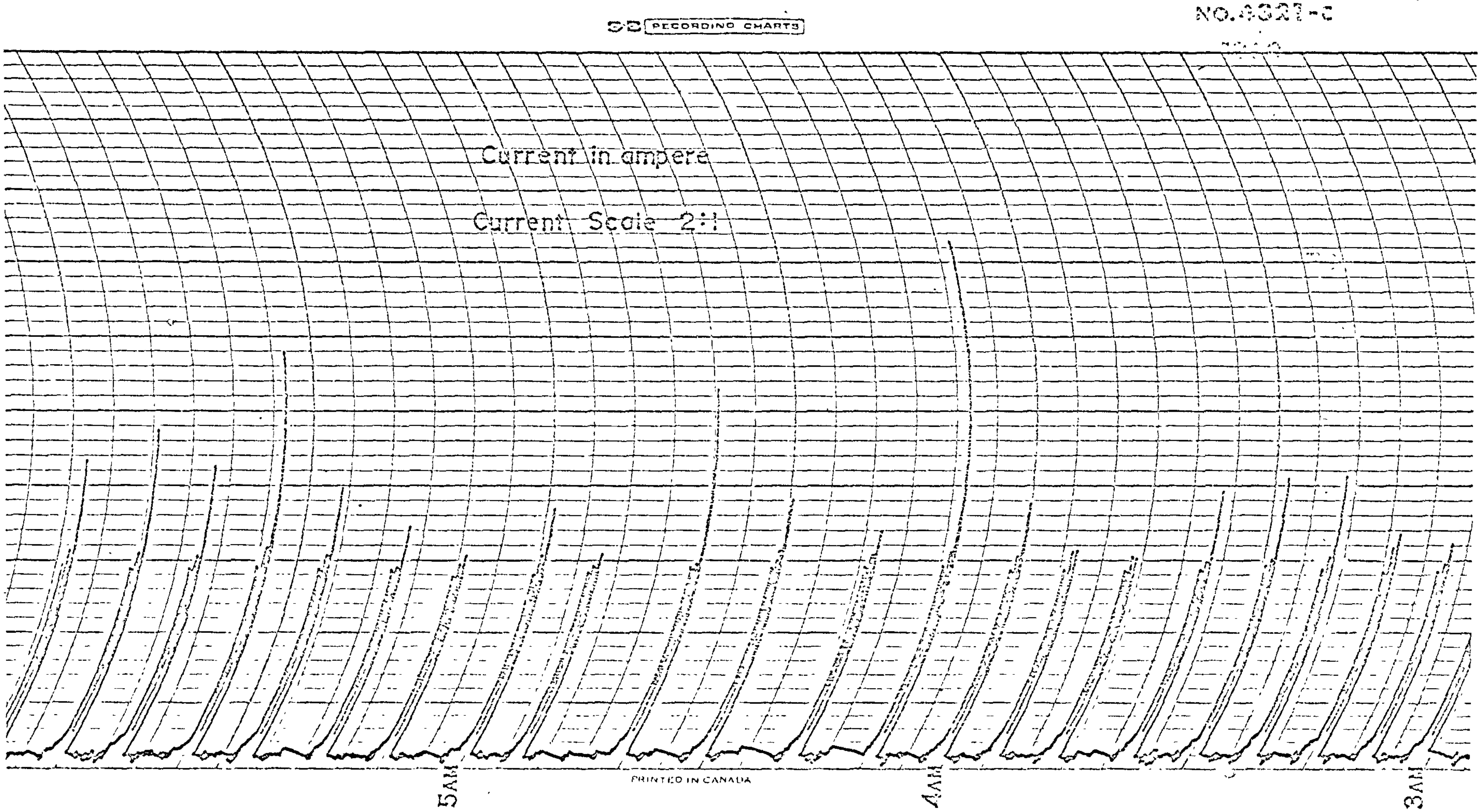
Figure - 5



CURRENT RECORD OF PUSHING MECHANISM

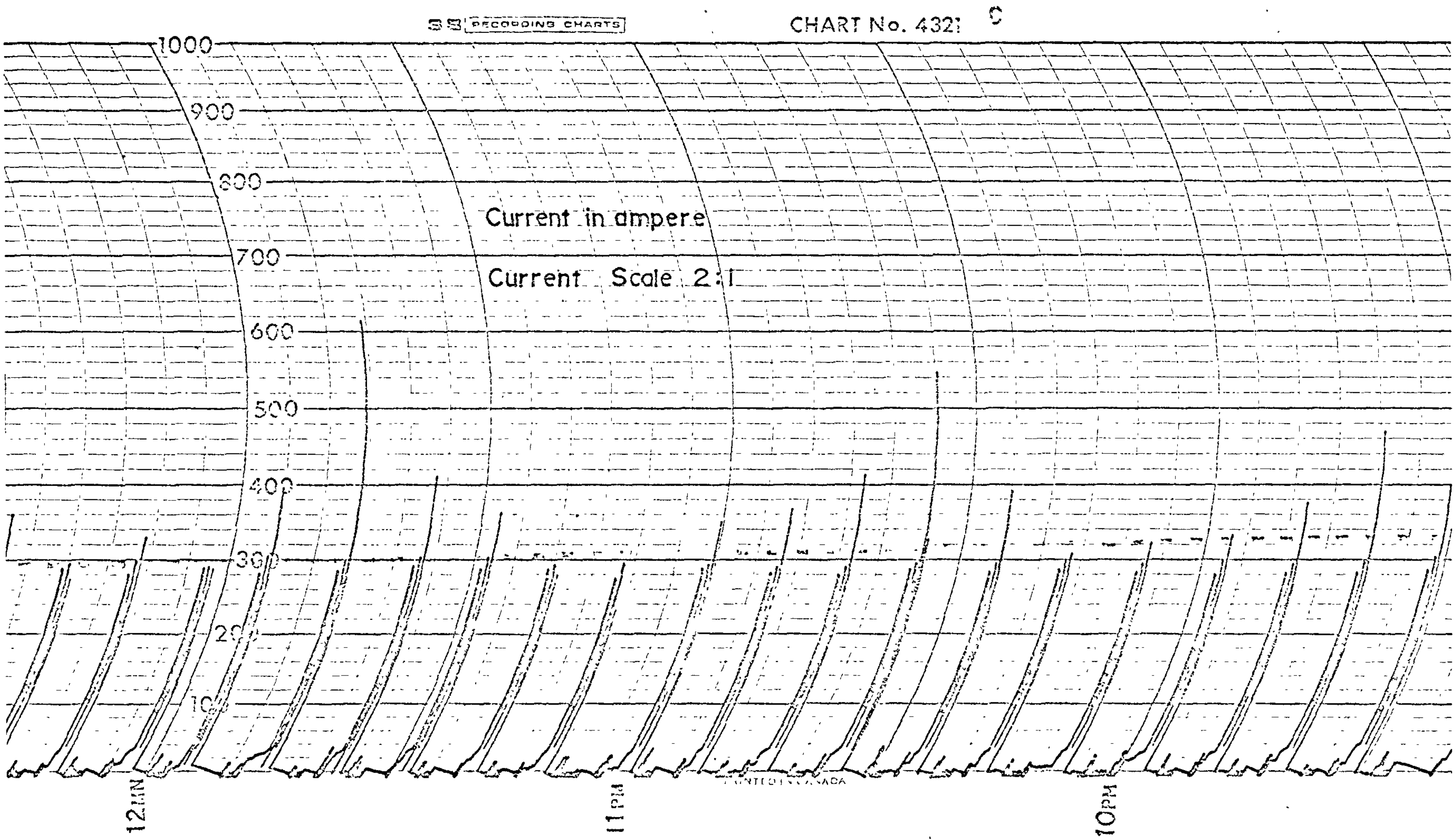
(Courtsey Koppers Co. Inc. Pittsburgh U.S.A.)

Figure - 6



CURRENT RECORD OF PUSHING MECHANISM OF COKE PUSHER
MACHINE OF STELCO BATTERY NO: 3.

Figure - 7



CURRENT RECORD OF PUSHING MECHANISM OF COKE PUSHER MACHINE OF STE[illegible] BATTERY NO: 4.

Figure - 8

Speed Characteristic Of Pusher Ram (Reference 1)

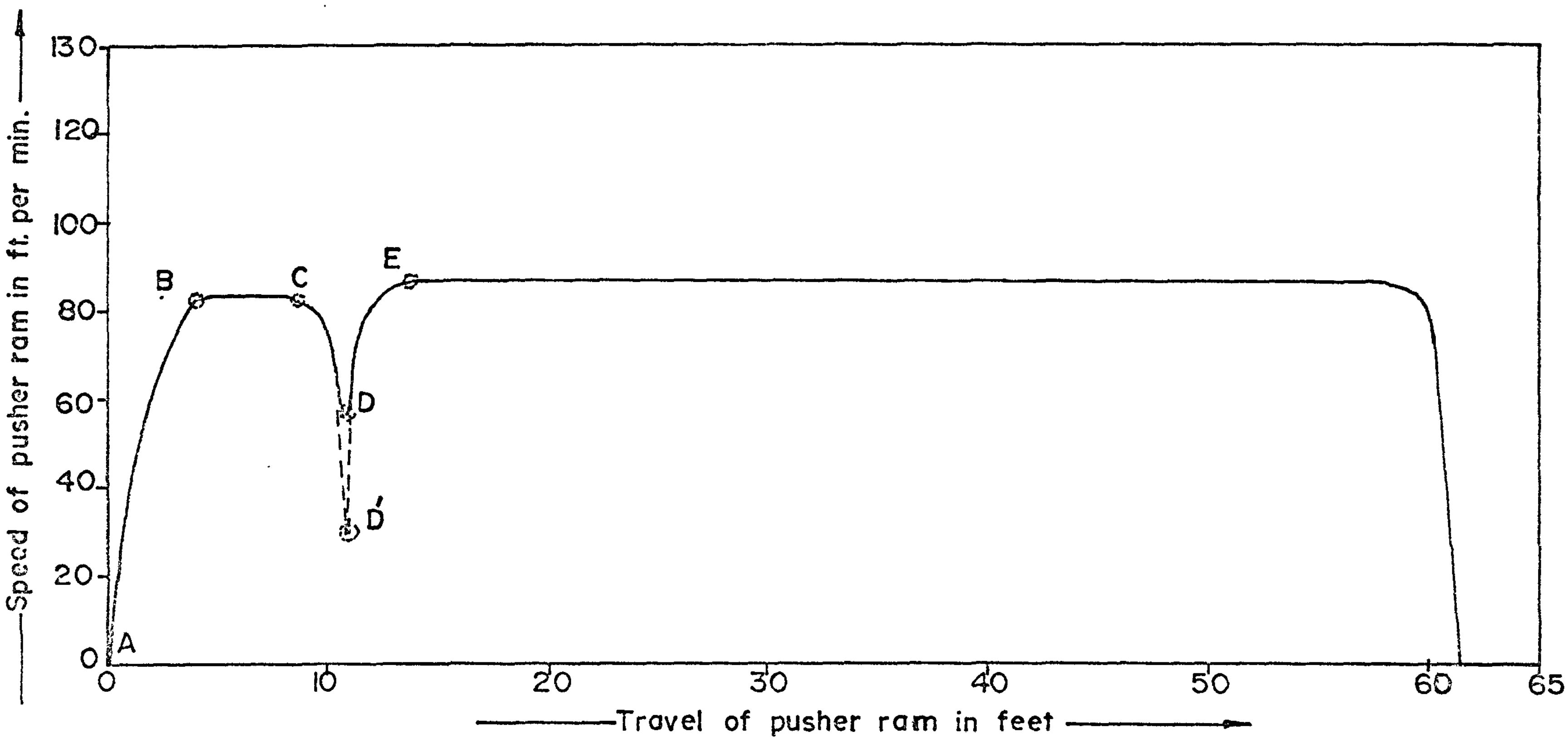
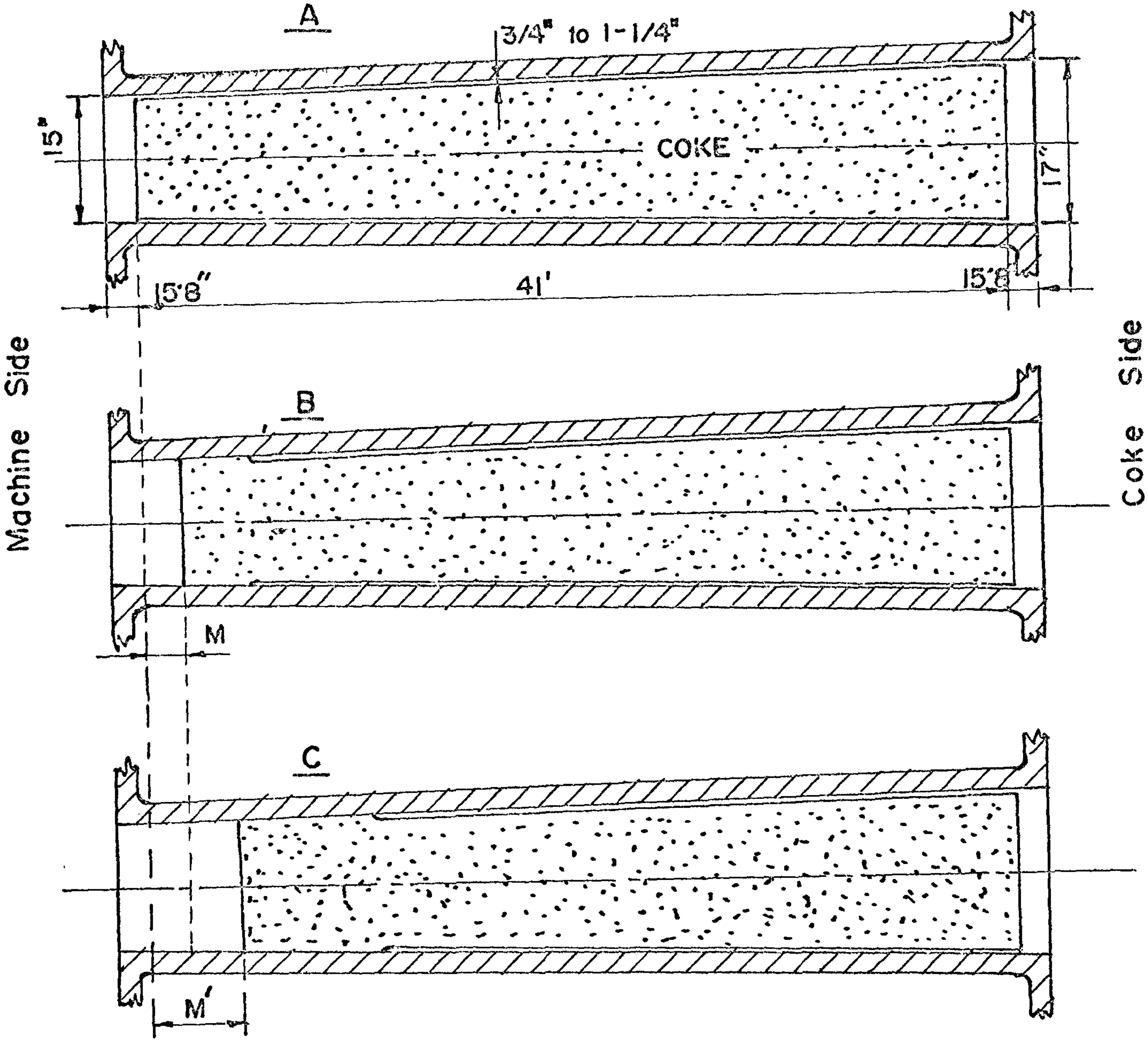


Figure - 9

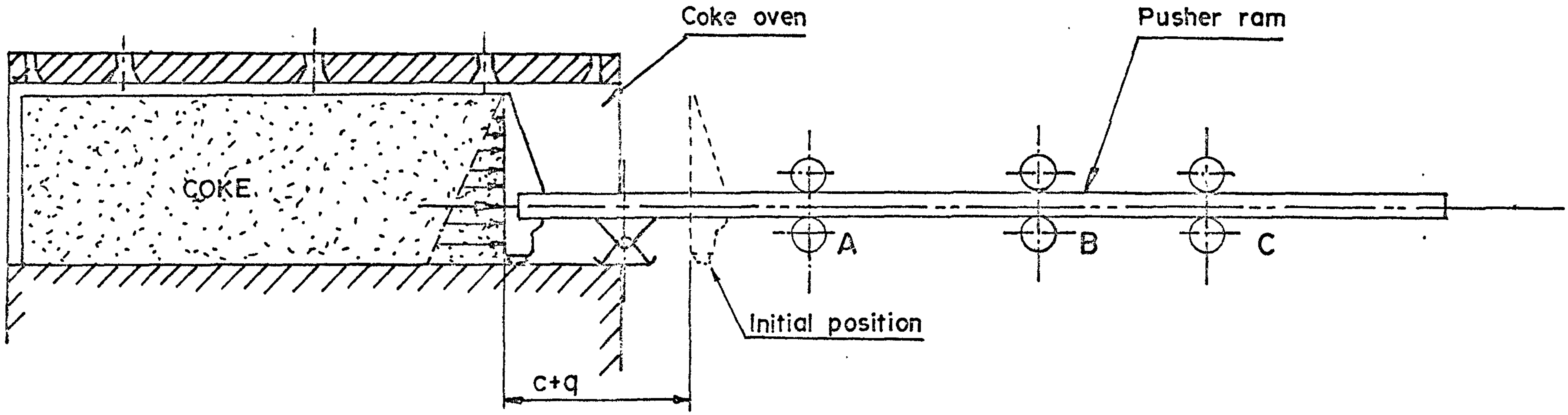


A - Coke oven chamber with coke before pushing.
B - Coke oven chamber with coke pressed to some depth.
C - Coke oven chamber with coke pressed to full extent and is about to move.

M - First pressing length.
M' - Second and last pressing length.

Figure - 10

(Not to scale)



FIRST CRITICAL POSITION OF THE PUSHER RAM

Figure - 11

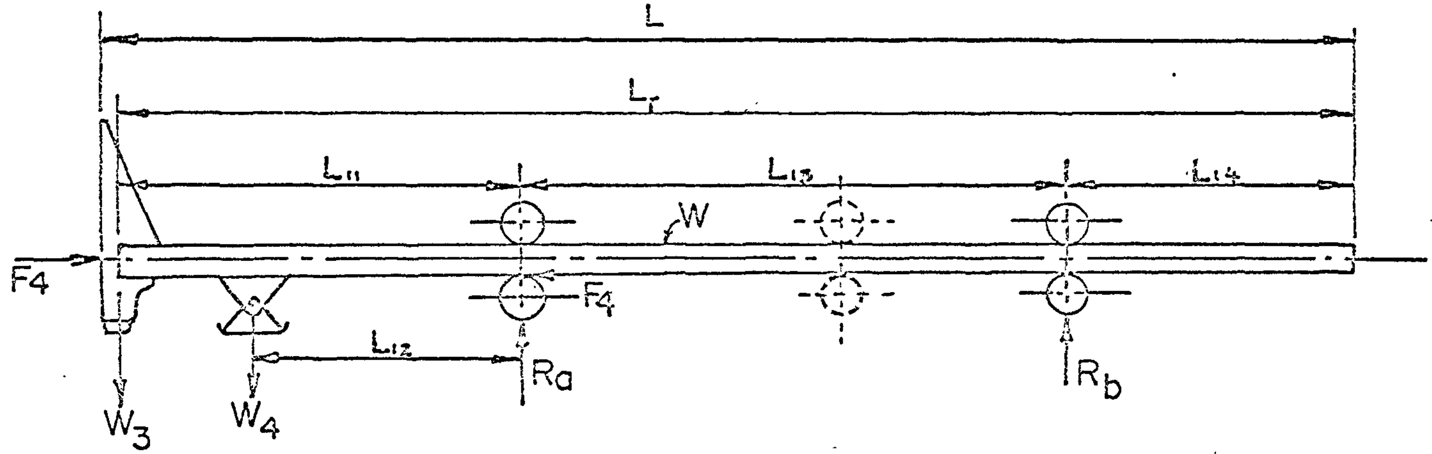
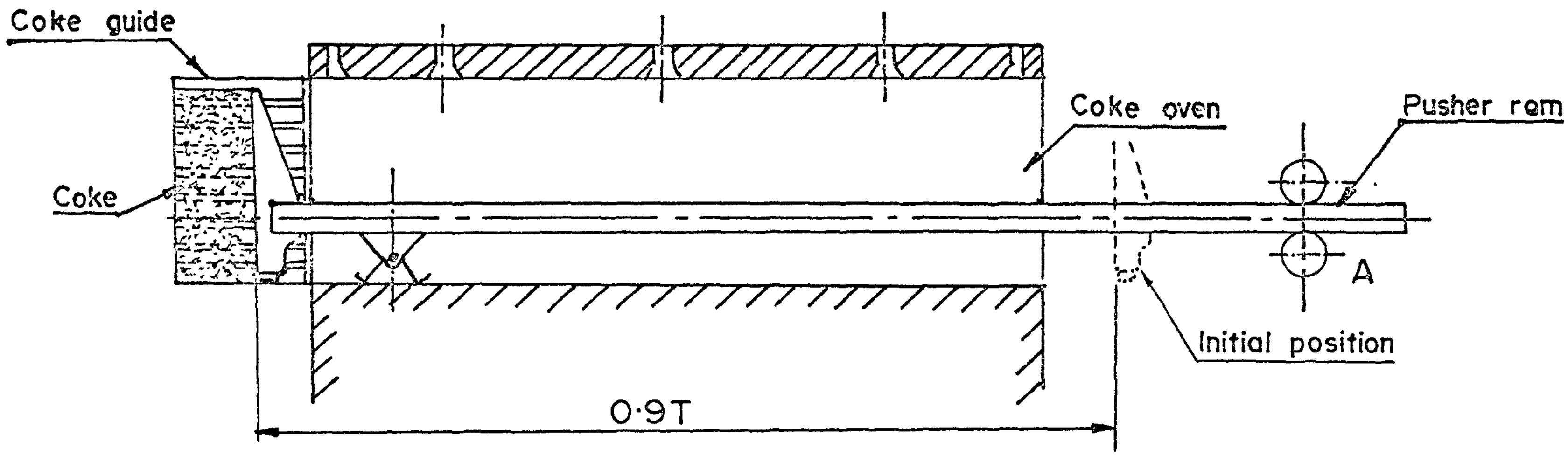


Figure - 12



SECOND CRITICAL POSITION OF THE PUSHER RAM

Figure - 13

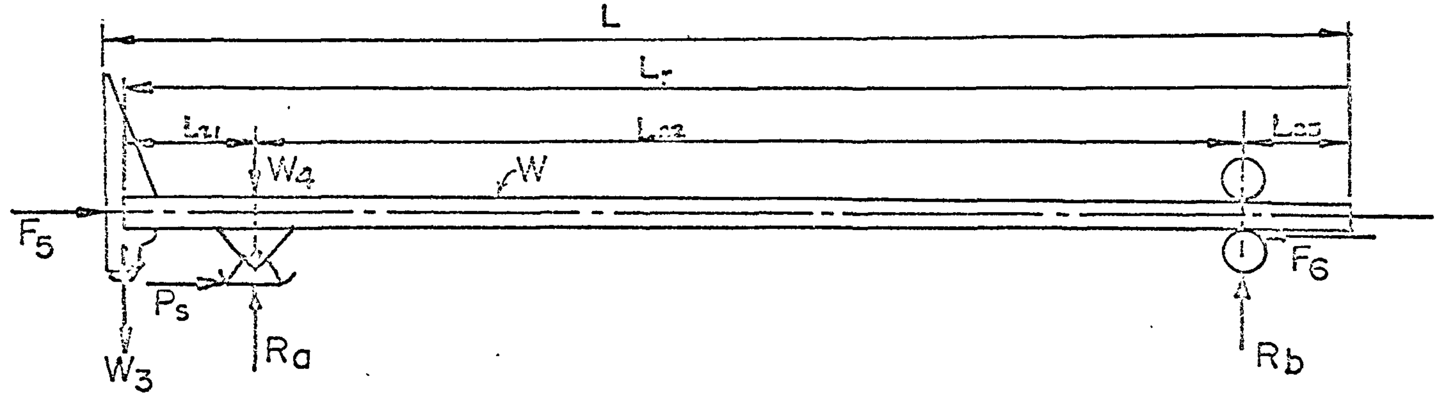
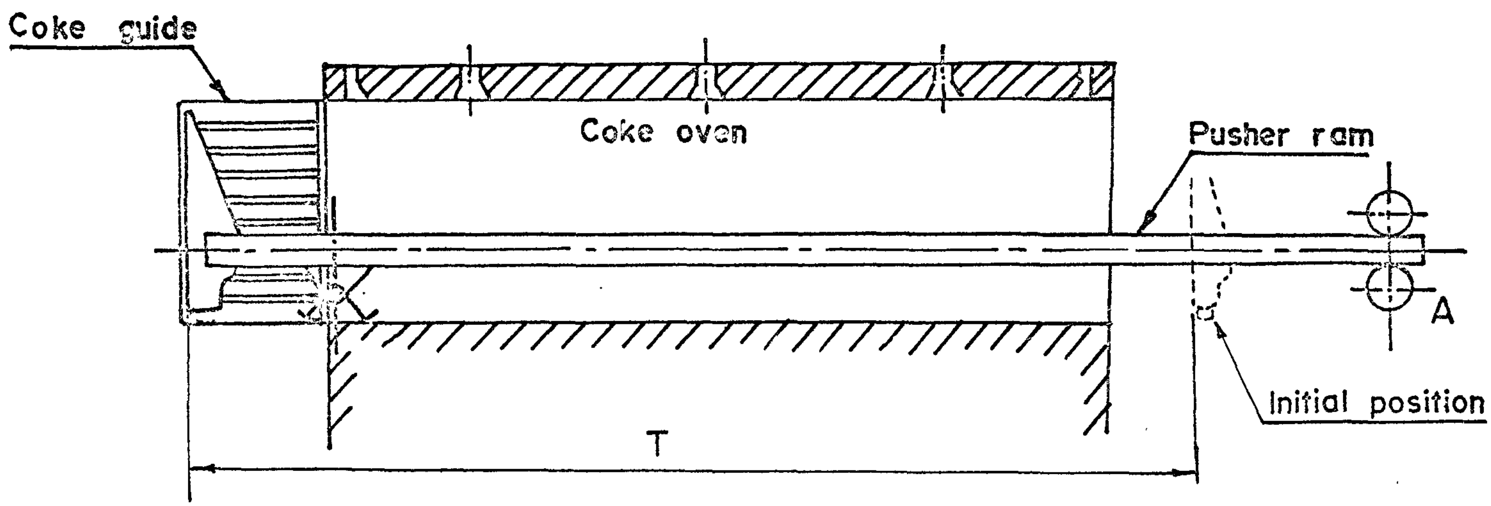


Figure - 14



THIRD CRITICAL POSITION OF THE PUSHER RAM

Figure - 15

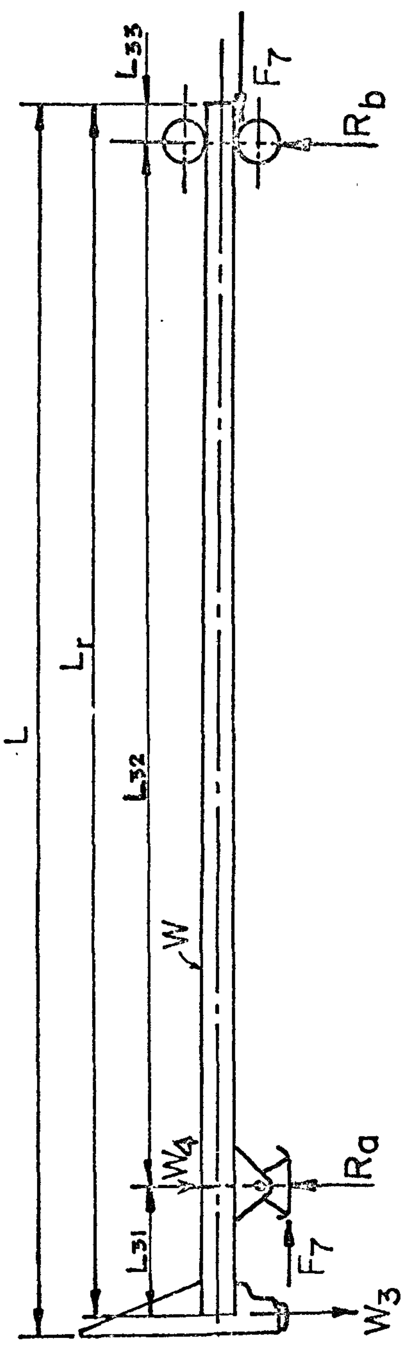


Figure - 16

T A B L E S

TABLE-1

S.No.	Description	Symbol	Unit	Value
1	Length of coke oven from end to end.	a	ins.	562.0
2	Distance of the ram head face from the end of the oven brickwork in the forward most position of the pusher ram.	b	ins.	109.0
3	Distance of the ram head face from the end of the oven brickwork in the initial position of the pusher ram.	c	ins.	64.0
4	Distance between the ram head face and the C.L. of the supporting shoe.	d	ins.	130.0
5	Distance between the C.L. of first and second roller support.	e	ins.	244.0
6	Distance between the C.L. of second and third roller support.	f	ins.	197.0
7	Distance between the end of oven brickwork and the C.L. of first roller support.	g	ins.	233.0
8	Usefull height of the coke oven chamber.	h	ins.	157.0
9	Total travel of the pusher ram.	T	ins.	735.0
10	Total length of the pusher ram with ram head.	L	ins.	940.0

Table 1 continued...

S.No.	Description	Symbol	Unit	Value
11	Distance between the front end of the pusher ram (without ram head) and the C.L. of the supporting shoe.	q	ins.	114.0
12	Volume of coal charged in the oven.	V	Cu. ft.	760.0
13	Minimum width of the oven.	b_w	ins.	15.0
14	Weight per unit length of toothed rack and guide rack combined.	W_r	lb/in	0.0192
15	Weight of the ram head.	W_3	Kips	5.0
16	Weight of the supporting shoe assly.	W_4	Kips	2.0
17	Mean velocity of the pusher ram.	ν	ft/min	85.0
18	Time of accelaration of pusher ram in attaining maximum speed.	t	Sec.	0.6
19	Diameter of the rivet for rivetting toothed and guide rack to the pusher ram.	d_o	ins.	0.8125
20	Distance between the pitch line of toothed rack and outside surface of the bottom plate of ram.	C_1	ins.	3.0
21	Distance between the bottom surface of the ram and the bottom surface of the shoe.	C_2	ins.	39.4

Table 1 continued....

S.No	Description	Symbol	Unit	Value
22	Width of the toothed rack.	b_r	ins.	12.5
23	Rise in temperature of the pusher ram in the first critical position.	t_1	^oF	7.0
24	Rise in temperature of the pusher ram in the second critical position.	t_2	^oF	30.0

TABLE-2

Serial No.	Current consumed by electric motor in amperes.	Force exerted by the electric motor on the pusher ram in Kips.
1	5	0.6791
2	10	1.3583
3	15	2.0374
4	20	2.7166
5	25	3.3958
6	30	4.0749
7	35	4.7541
8	40	5.4332
9	45	6.1124
10	50	6.7916
11	55	7.4707
12	60	8.1499
13	65	8.8290
14	70	9.5082
15	75	10.1874
16	80	10.8665
17	85	11.5457
18	90	12.2248
19	95	12.9040
20	100	13.5832
21	105	14.2623
22	110	14.9415
23	115	15.6206
24	120	16.2998
25	125	16.9790
26	130	17.6581
27	135	18.3373
28	140	19.0165
29	145	19.6956

Table 2 continued.

Serial No.	Current consumed by electric motor in amperes.	Force exerted by the electric motor on the pusher ram in Kips.
30	150	20.3748
31	155	21.0539
32	160	21.7331
33	165	22.4123
34	170	23.0914
35	175	23.7706
36	180	24.4497
37	185	25.1289
38	190	25.8081
39	195	26.4872
40	200	27.1664
41	205	27.8455
42	210	28.5247
43	215	29.2039
44	220	29.8830
45	225	30.5622
46	230	31.2413
47	235	31.9205
48	240	32.5997
49	245	33.2788
50	250	33.9580
51	255	34.6371
52	260	35.3163
53	265	35.9955
54	270	36.6674
55	275	37.3538
56	280	38.0330
57	285	38.7121

Table 2 continued.

Serial No.	Current consumed by electric motor in amperes.	Force exerted by the electric motor on the pusher ram in Kips.
58	290	39.3913
59	295	40.0704
60	300	40.7496
61	305	41.4288
62	310	42.1079
63	315	42.7871
64	320	43.4662
65	325	44.1454
66	330	44.8246
67	335	45.5037
68	340	46.1829
69	345	46.8620
70	350	47.5412
71	355	48.2204
72	360	48.8995
73	365	49.5787
74	370	50.2578
75	375	50.9370
76	380	51.6162
77	385	52.2953
78	390	52.9745
79	395	53.6537
80	400	54.3339

T A B L E - 3

S.No.	Description	Unit	PK-2K Russian oven.	STELCO Coke oven battery no.3
1	Length of oven.	ft.	43.4	40.55
2	Usefull height of the oven.	ft.	13.1	12.25
3	Mean width of the oven.	ins.	16.0	16.0
4	Weight of coke produced per oven per cycle.	lbs.	29750.0	26380.0
5	Coking temperature.	^oF	1850	2150
6	Coking period.	hrs	17.0	15.23
7	Length of the pusher ram.	ft.	78.3	73.2
8	Power of motor used for pushing the coke.	KW	60.0	48.0
9	Voltage of power supply	Volt	220	230
10	Speed of motor.	rpm	470	440
11	Reduction ratio of the reduction gear.	-	37.0	39.2

TABLE-4

S. No.	Point No. 1 5 ft. from the ram head			Point No. 2 15 ft. from the ram head.			Point No. 3 25 ft. from the ram head		
	Initial temp. °F	Final temp. °F	Rise in temp. °F	Initial temp. °F	Final temp. °F	Rise in temp. °F	Initial temp. °F	Final temp. °F	Rise in temp. °F
1	250	295	45	240	280	40	220	255	35
2	250	300	50	235	280	45	225	260	35
3	245	295	50	235	285	50	220	260	40
Mean temperature rise in °F			48.3	Mean temperature rise in °F		45	Mean temperature rise in °F		36.6

TABLE- 5

S. No.	Starting Point				Optimum Value by APPROX					Optimum Value by SEEK				
	a_1 ins.	b_1 ins.	t_b ins.	t_a ins.	A in^2	a_1 ins.	b_1 ins.	t_b ins.	t_a ins.	A in^2	a_1 ins.	b_1 ins.	t_b ins.	t_a ins.
1	40.0	13.0	1.5	0.75	34.9953	30.1875	12.5	0.8125	0.3125	37.1625	27.6250	12.6250	1.00	0.3750
2	30.5	12.75	1.0	0.5	34.9953	30.1875	12.5	0.8125	0.3125	42.2656	27.4375	12.5625	0.9375	0.4375
3	29.0	12.8	1.1	0.5	34.9953	30.1875	12.5	0.8125	0.3125	42.5687	26.2500	12.6250	1.00	0.4375
4	27.5	12.7	1.0	0.4	34.9953	30.1875	12.5	0.8125	0.3125	42.5687	26.2500	12.6250	1.00	0.4375
5	25.0	12.5	0.82	0.27	34.9953	30.1875	12.5	0.8125	0.3125	36.2984	29.9375	12.6250	0.8750	0.3125

TABLE-6

Method	Starting Point				Final Optimum Result				
	a_1 ins.	b_1 ins.	t_b ins.	t_a ins.	A in^2	a_1 ins.	b_1 ins.	t_b ins.	t_a ins.
Successive Linear Approximation.	25.0	12.5	0.82	0.27	34.9953	30.1875	12.5	0.8125	0.3125
Direct Search.	25.0	12.5	0.82	0.27	36.2984	29.9375	12.625	0.8750	0.3125

A P P E N D I C E S

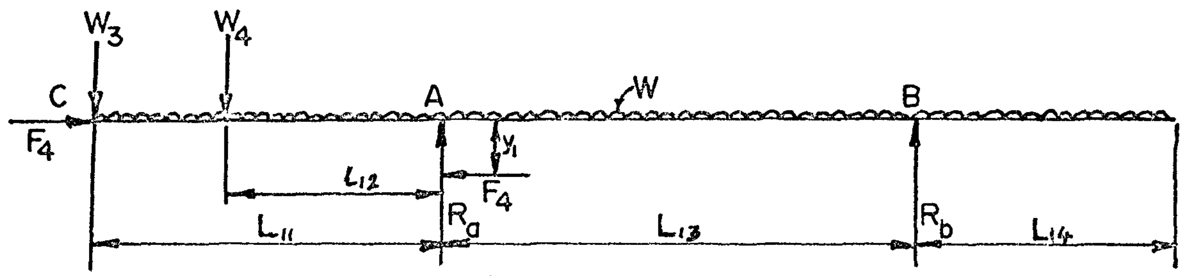
APPENDIX - A

CALCULATION OF MAXIMUM BENDING MOMENT
SHEAR FORCE AND DEFLECTION OF THE
PUSHER RAM IN FIRST CRITICAL
POSITION

Calculation of Maximum Bending Moment

Shear Force and Deflection of the

Pusher Ram in First Critical Position



The figure shown above is the loading condition of the pusher ram in its first critical position as described earlier. The bending moment at point A can be expressed as:

$$M_1 = W_3L_{11} + W_4L_{12} - F_4y_1 + F_4\Delta_{L1}$$

and bending moment at point B is

$$M_2 = \frac{WL_{14}^{\;2}}{2}$$

Δ_{L1} is the deflection at point C and is derived in the following.

By using the method of super position, the deflection Δ_{L1} at the point C can be expressed as [6].

Δ_{L1} = (Deflection at point C due to load W_3, W_4 and W considering span AC as cantilever)

$$- L_{11}\,\theta_1$$

where θ_1 = slope angle at support A.

It should be noted that the use of the superposition method to find the deflection at C is justified because the L/r ratio for the pusher ram is much less than 60.

$$\theta_1 = \frac{WL_{13}^3}{24EI_x} - \frac{M_1L_{13}}{3EI_x} - \frac{M_2L_{13}}{6EI_x}$$

Deflection at point C due to load W_3, W_4 and W

$$= \frac{WL_{11}^4}{8EI_x} + \frac{W_3L_{11}^3}{3EI_x} + \frac{W_4}{6EI_x}(3L_{12}^2L_{11} - L_{12}^3)$$

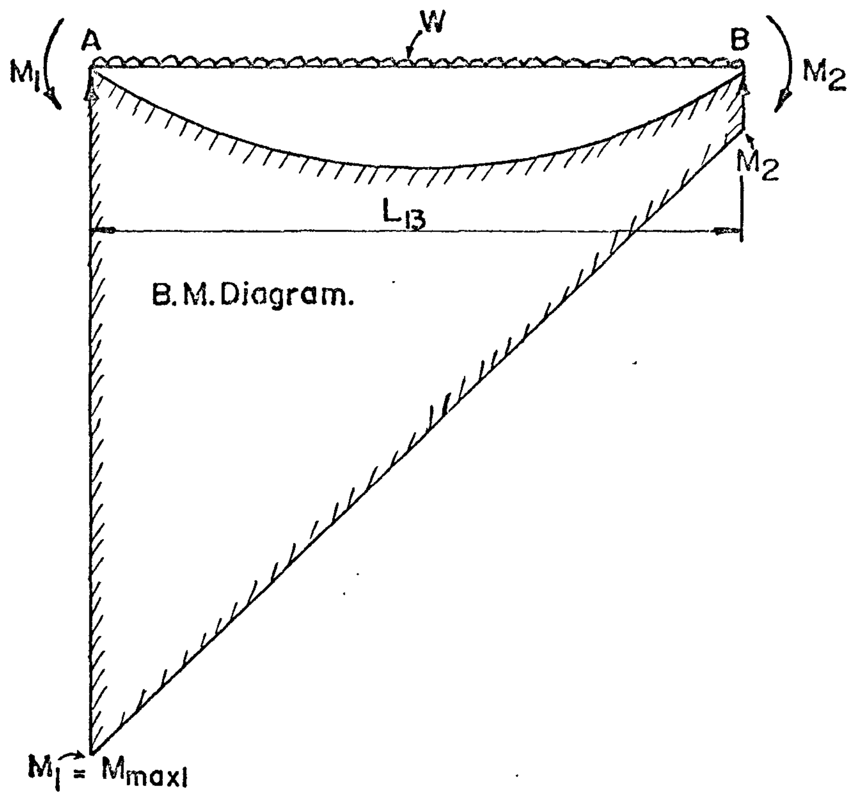
Therefore

$$\Delta_{L1} = \{\frac{WL_{11}^4}{8EI_x} + \frac{W_3L_{11}^3}{3EI_x} + \frac{W_4}{6EI_x}(3L_{12}^2L_{11} - L_{12}^3)\}$$
$$- (\frac{WL_{13}^3L_{11}}{24EI_x} - \frac{M_1L_{13}L_{11}}{3EI_x} - \frac{M_2L_{13}L_{11}}{6EI_x})$$

Substituting the value of M_1 and M_2 and solving we get

$$\Delta_{L1} = \frac{3}{3EI_x - F_4L_{13}L_{11}}[\{\frac{WL_{11}^4}{8} + \frac{W_3L_{11}^3}{3} + \frac{W_4}{6}(3L_{12}L_{11} - L_{12}^3)\}$$
$$- \{\frac{WL_{13}^3L_{11}}{24} - \frac{WL_{14}^4L_{13}L_{11}}{12} - \frac{L_{13}L_{11}}{3}(W_3L_{11} + W_4L_{12}$$
$$+ \frac{WL_{11}^2}{2} - F_4y_1)\}]$$

The figure below shows the portion of the pusher ram between point A and B as a free body along with the bending moment diagram.



The maximum bending moment in the pusher ram occurs at point A in this position and is expressed as:

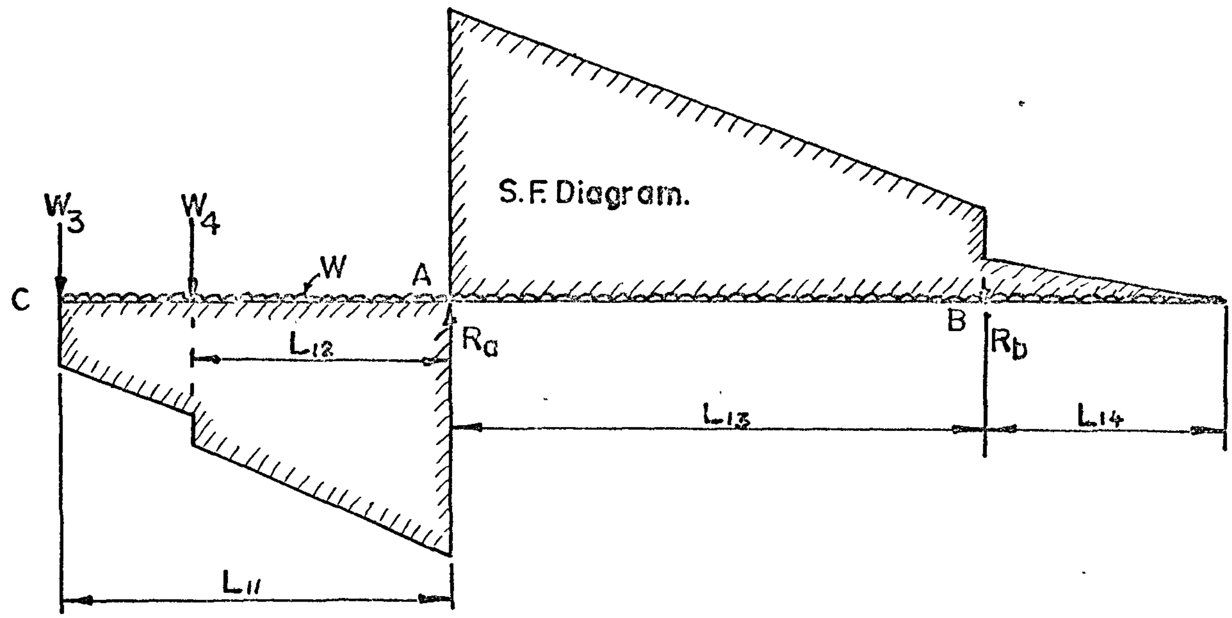
$$M_{max1} = W_3L_{11} + W_4L_{12} - F_4y_1 + F_4\Delta_{L1}$$

The figure below shows the shear force diagram in the first position of the pusher ram.

The maximum shear force will occur at point A and is given as

$$F_{s1} = W_3 + W_4 + WL_{11} \qquad (1)$$

or $$F_{s1} = R_a - (W_3 + W_4 + WL_{11}) \qquad (2)$$



Where

$$R_a = \frac{W_3(L_{11} + L_{13})}{L_{13}} + W_4(L_{12} + L_{13}) + \frac{W(L_{11} + L_{13})^2}{2}$$

$$- \frac{WL_{14}^{\ 2}}{2} - F_4y_1$$

The values of shear force in equation (1) and (2) must be compared and the greater value taken as the maximum shear force.

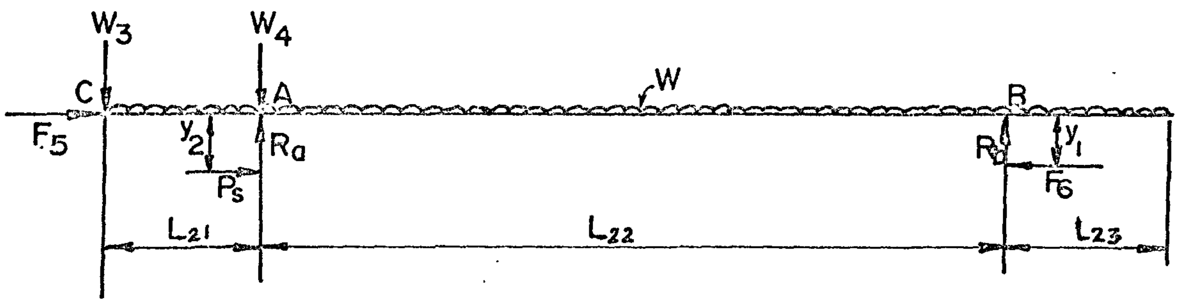
APPENDIX - B

CALCULATION OF MAXIMUM BENDING MOMENT SHEAR FORCE AND DEFLECTION OF THE PUSHER RAM IN SECOND CRITICAL POSITION

Calculation of Maximum Bending Moment

Shear Force and Deflection of the Pusher

Ram in Second Critical Position



The figure shown above is the loading condition of the pusher ram in the second critical position as described earlier.

The bending moment at point A is given as

$$M_1 = W_3 L_{21} + \frac{W L_{21}^2}{2} + F_5 \Delta_{L2} + P_s y_2 \tag{1}$$

and bending moment at point B is

$$M_2 = \frac{W L_{23}^2}{2} + F_6 y_1 \tag{2}$$

The deflection Δ_{L2} at point C is found by the method of superposition as described in Appendix - A.

Deflection Δ_{L2} at point C can be expressed as

Δ_{L2} = (Deflection $\Delta_L{}'$ at C due to load W_3 and W considering span AC as cantilever) - $L_{21}\theta_1$

θ_1 = slope angle at point A

$$= \frac{WL_{22}^3}{24EI_x} - \frac{M_1L_{22}}{3EI_x} - \frac{M_2L_{22}}{6EI_x}$$

Therefore $\Delta_{L2} = \Delta_L' - L_{21}\theta_1$

$$\text{therefore } \Delta_L' = \frac{WL_{21}^4}{8EI_x} + \frac{W_3L_{21}^3}{3EI_x}$$

$$\text{therefore } \Delta_{L2} = \frac{WL_{21}^4}{8EI_x} + \frac{W_3L_{21}^3}{3EI_x} - \frac{WL_{22}^3L_{21}}{24EI_x} + \frac{M_1L_{22}L_{21}}{3EI_x} + \frac{M_2L_{22}L_{21}}{6EI_x}$$

Substituting the value of M_1 and M_2 from equation (1) and (2) to the above equation and solving we get

$$\Delta_{L2} = \frac{3}{3EI_x - F_5L_{22}L_{11}}\left[\frac{WL_{21}^4}{8} + \frac{W_3L_{21}^3}{3} - \frac{WL_{22}^3L_{21}}{24} + \frac{L_{22}L_{21}}{6}\left(\frac{WL_{23}^2}{2} - F_6y_1\right) + \frac{L_{22}L_{11}}{3}\left(W_3L_{21} + \frac{WL_{11}^2}{2} + P_sy_2\right)\right]$$

The expression for the frictional force P_s can be derived as follows.

Taking moments about point B we have

$$- W_3(L_{21} + L_{22}) - W_4L_{22} + R_aL_{22} - P_sy_2 - \frac{W(L_{21} + L_{22})^2}{2}$$

$$+ \frac{WL_{23}^{2}}{2} + (F_5 + P_s)y_1 = 0$$

or

$$R_a L_{22} = W_3(L_{21} + L_{22}) + W_4 L_{22} + P_s y_2 + \frac{W(L_{21} + L_{22})^2}{2}$$

$$- \frac{WL_{23}^{2}}{2} - F_5 y_1 - P_s y_2$$

and $P_s = K_4 R_a$ or $R_a = \frac{P_s}{K_4}$

therefore

$$\frac{P_s L_{22}}{K_4} = W_3(L_{21} + L_{22}) + W_4 L_{22} + P_s(y_2 - y_1)$$

$$+ \frac{W(L_{21} + L_{22})^2}{2} - \frac{WL_{23}^{2}}{2} - F_5 y_1$$

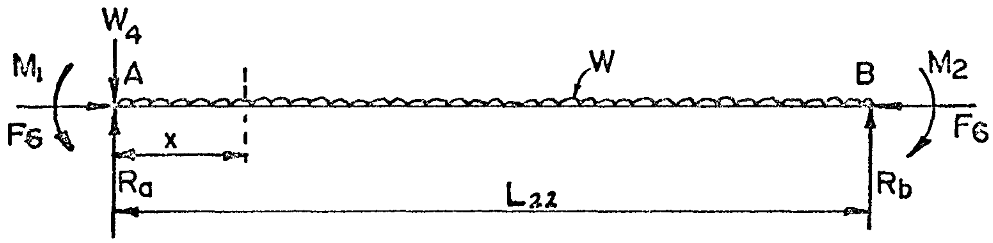
Solving the above equation we get

$$P_s = \frac{K_4}{(L_{22} - K_4(y_2 - y_1))} [W_3(L_{21} + L_{22}) + W_4 L_{22} + \frac{W(L_{21} + L_{22})^2}{2}$$

$$- \frac{WL_{23}^{2}}{2} - F_5 y_1]$$

The following figure shows the portion of the ram between point A and B as a free body. The analysis of this will give the maximum bending moment.

The loading system of the pusher ram is converted into an equivalent system as shown in the figure above. The analysis of the above system is given as follows [4].



At any point a distance x from the beam end, the expression for the bending moment can be given as:

$$M = M_1 - \left[\frac{(M_1 - M_2)}{L_{22}} + \frac{WL_{22}}{2} \right] x + \frac{Wx^2}{2} - F_6 y$$

and

$$R_a = \frac{(M_1 - M_2)}{L_{22}} + \frac{WL_{22}}{2}$$

We know

$$M = EI \frac{d^2y}{dx^2}$$

Differentiating equation (B.3) twice with respect to x, we get -

$$\frac{d^2M}{dx^2} + \frac{F_6}{EI_x} M = W \qquad (1)$$

let

$$j = \sqrt{\frac{EI_x}{F_6}} \qquad \text{or} \qquad \frac{F_6}{EI_x} = \frac{1}{j^2} \qquad (2)$$

Substituting (2) in equation (1) we get -

$$\frac{d^2M}{dx^2} + \frac{1}{j^2} M = W \qquad (3)$$

The solution of the above differential equation is

$$M = C_1 \operatorname{Sin} x/j + C_2 \operatorname{Cos} x/j + Wj^2 \qquad (4)$$

where C_1 and C_2 are constants of integration and Sin x/j and Cos x/j are the limits of an infinite series of variable x/j

when $\quad x = 0 \quad M = M_1$

and $\quad x = L_{22} \quad M = M_2$

$$\text{therefore } C_1 = \frac{M_2 - Wj^2}{\operatorname{Sin}L_{22}/j} - \frac{M_1 - Wj^2}{\tan L_{22}/j}$$

$$\text{or} \qquad C_1 = \frac{M_2 - Wj^2 - (M_1 - Wj^2) \operatorname{Cos} L_{22}/j}{\operatorname{Sin} L_{22}/j}$$

and $\quad C_2 = M_1 - Wj^2$

let $\quad D_1 = M_1 - Wj^2$

and $\quad D_2 = M_2 - Wj^2$

$$\text{therefore } M = \frac{D_2 - D_1 \operatorname{Cos} L_{22}/j}{\operatorname{Sin} L_{22}/j} \operatorname{Sin} x/j + D_1 \operatorname{Cos} x/j + Wj^2$$

To find the position of the maximum bending moment equation (4) is differentiated and equated to zero.

$$\frac{dM}{dx} = \frac{C_1}{j} \operatorname{Cos} x/j - \frac{C_2}{j} \operatorname{Sin} x/j = 0$$

$$\text{or} \qquad \tan x/j = \frac{C_1}{C_2} = \frac{D_2 - D_1 \operatorname{Cos} L_{22}/j}{D_1 \operatorname{Sin} L_{22}/j}$$

$$\text{therefore} \quad \tan x/j = \frac{D_2 - D_1 \cos L_{22}/j}{D_1 \sin L_{22}/j}$$

The value of x must fall within $x = 0$ and $x = L_{22}$ otherwise M_1 or M_2 is the maximum bending moment.

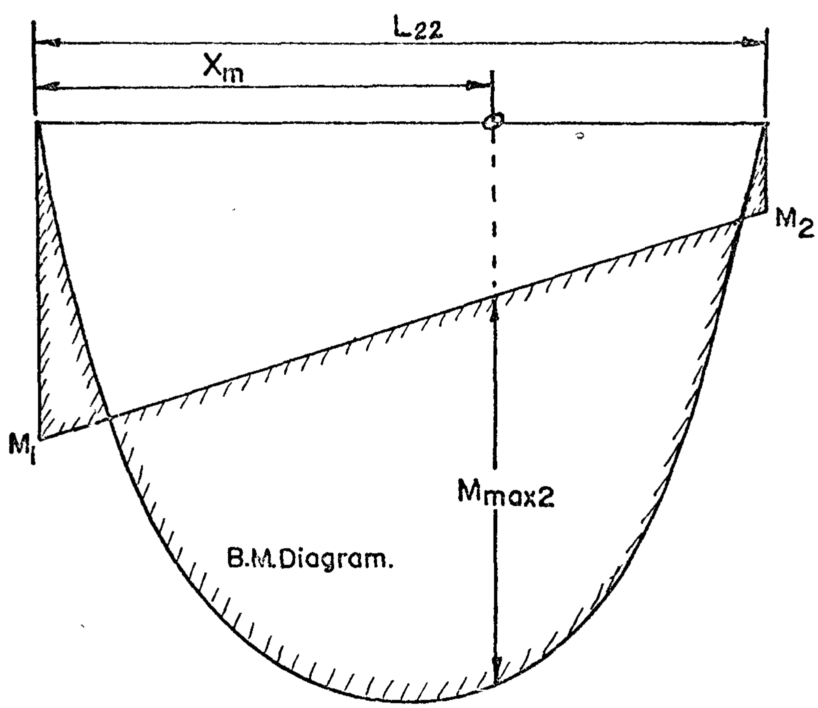
Let x_m is the point of maximum bending moment.

$$\text{Then } x_m = j \tan^{-1} \left(\frac{D_2 - D_1 \cos L_{22}/j}{D_1 \sin L_{22}/j} \right)$$

and the maximum bending moment is

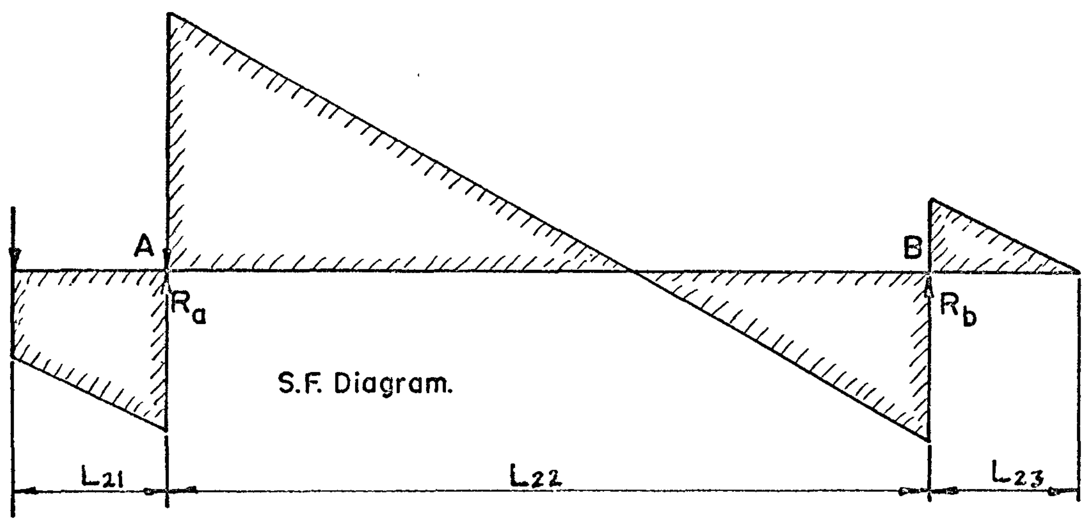
$$M_{max2} = \frac{D_1}{\cos x_m/j} + Wj^2$$

The following figure shows the bending moment diagram of the portion AB of the pusher ram, showing the point of maximum bending.



The following figure shows the shear force diagram of the pusher ram in the second critical position. The maximum shear force will occur at A and can be expressed as:

$$F_{s2} = R_a - (W_3 + WL_{21})$$



APPENDIX - C

CALCULATION OF MAXIMUM BENDING MOMENT SHEAR FORCE AND DEFLECTION OF THE PUSHER RAM IN THIRD CRITICAL POSITION

Calculation of Maximum Bending Moment Shear Force and Deflection of the Pusher Ram In Third Critical Position

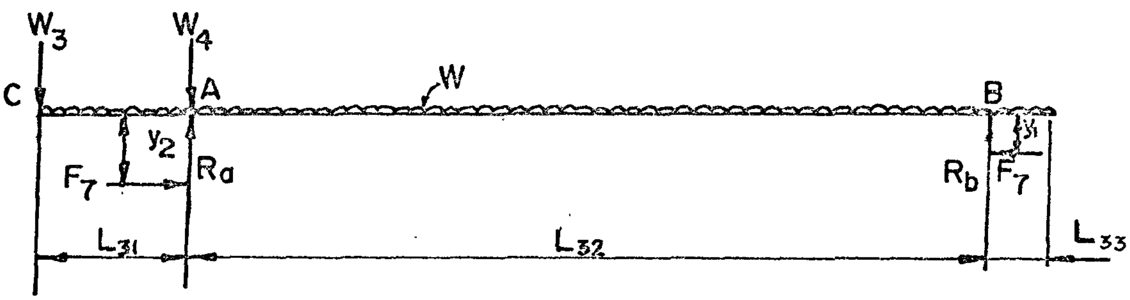


Figure shown above is the loading conditon of the pusher ram in the third critical position as described earlier.

The force due to friction between the supporting shoe and the oven brickwork (F_7) can be derived as follows:

Taking moments at point B we get

$$- W_3(L_{31} + L_{32}) - W_4L_{32} + R_aL_{32} - F_7y_2 - \frac{W(L_{31} + L_{32})^2}{2}$$

$$+ \ F_7y_1 \ + \ \frac{WL_{33}^{\ 2}}{2} \ = \ 0$$

$$\text{or} \quad R_a = \frac{W_3(L_{31} + L_{32})}{L_{32}} + W_4 + \frac{F_7(y_2 - y_1)}{L_{32}} + \frac{W(L_{31} + L_{32})^2}{2L_{32}}$$

$$+ \frac{WL_{33}^{2}}{2L_{32}}$$

But $R_a = \frac{F_7}{K_4}$

therefore $$\frac{F_7}{K_4} - \frac{F_7(y_2 - y_1)}{L_{32}} = \frac{W_3(L_{31} + L_{32})}{L_{32}} + W_4$$

$$+ \frac{W(L_{31} + L_{32})^2}{2L_{32}} + \frac{WL_{33}^{2}}{2L_{32}}$$

or $$F_7 = \left[\frac{K_4L_{32}}{L_{32} - K_4(y_2 - y_1)} \right] \left[\frac{W_3(L_{31} + L_{32})}{L_{32}} + W_4 \right.$$

$$\left. + \frac{W(L_{31} + L_{32})^2}{2L_{32}} + \frac{WL_{33}^{2}}{2L_{32}} \right]$$

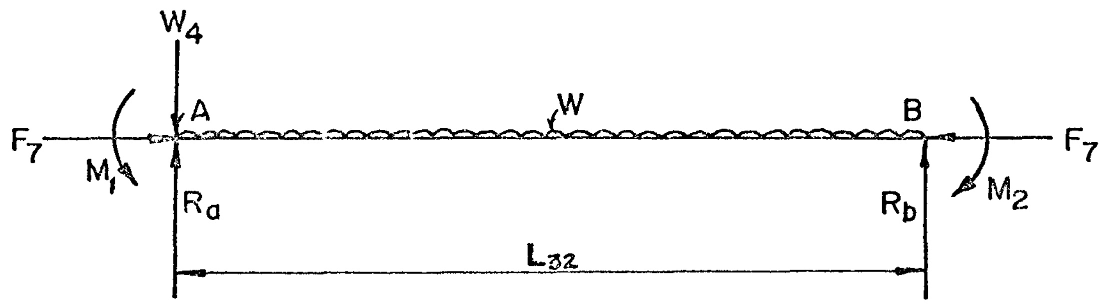
The bending moment at point A can be given as

$$M_1 = W_3L_{31} + F_7y_2 + \frac{WL_{31}^{2}}{2} \qquad (1)$$

and bending moment at point B is

$$M_2 = \frac{WL_{33}^{2}}{2} + F_7y_1 \qquad (2)$$

The following figure shows the free body diagram of the portion AB of the pusher ram. The analysis for the maximum bending moment will be exactly similar to that described in Appendix - B.



Using the same notation used in Appendix - B and following the same analysis [4] we get -

$$C_1 = \frac{M_2 - Wj^2 - (M_1 - Wj^2) \text{ Cos } L_{32}/j}{\text{Sin } L_{32}/j}$$

and $C_2 = M_1 - Wj^2$

let $D_1 = M_1 - Wj^2$ and $D_2 = M_2 - Wj^2$

The final equation of bending moment at any point at a distance x from the left support can be given as:

$$M_x = \frac{(D_2 - D_1 \text{ Cos } L_{32}/j)}{\text{Sin } L_{32}/j} \text{ Sin } x/j + D_1 \text{ Cos } x/j + Wj^2$$

The maximum bending moment will occur at a point, x_m distance from the left support.

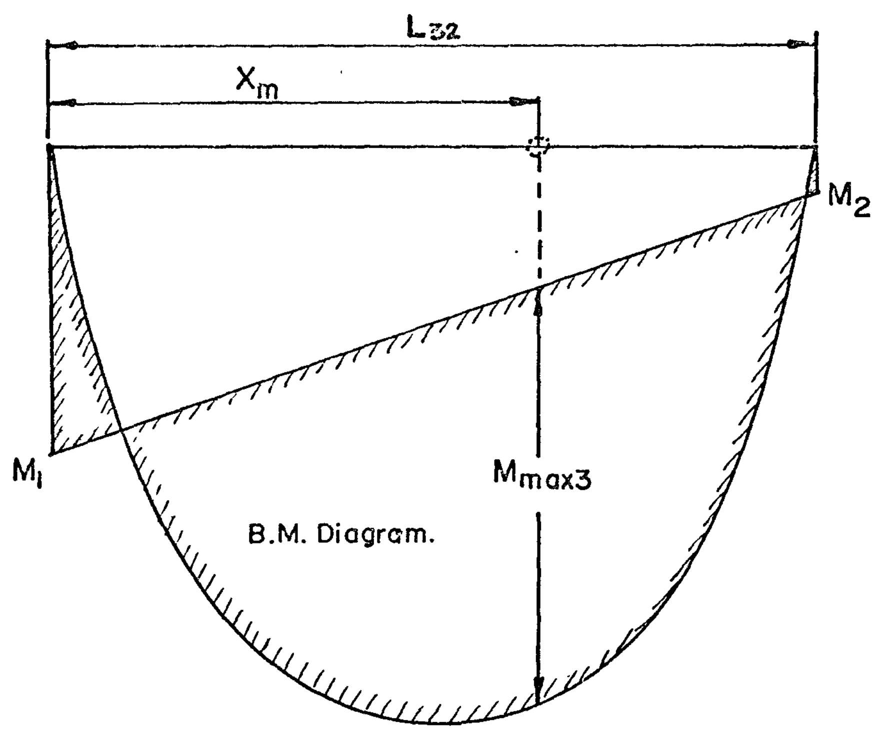
$$x_m = j \tan^{-1} \left[\frac{(D_2 - D_1 \text{ Cos } L_{32}/j)}{D_1 \text{ Sin } L_{32}/j} \right]$$

The value of the maximum bending moment will be

$$M_{max3} = \frac{D_1}{\text{Cos } x_m/j} + Wj^2$$

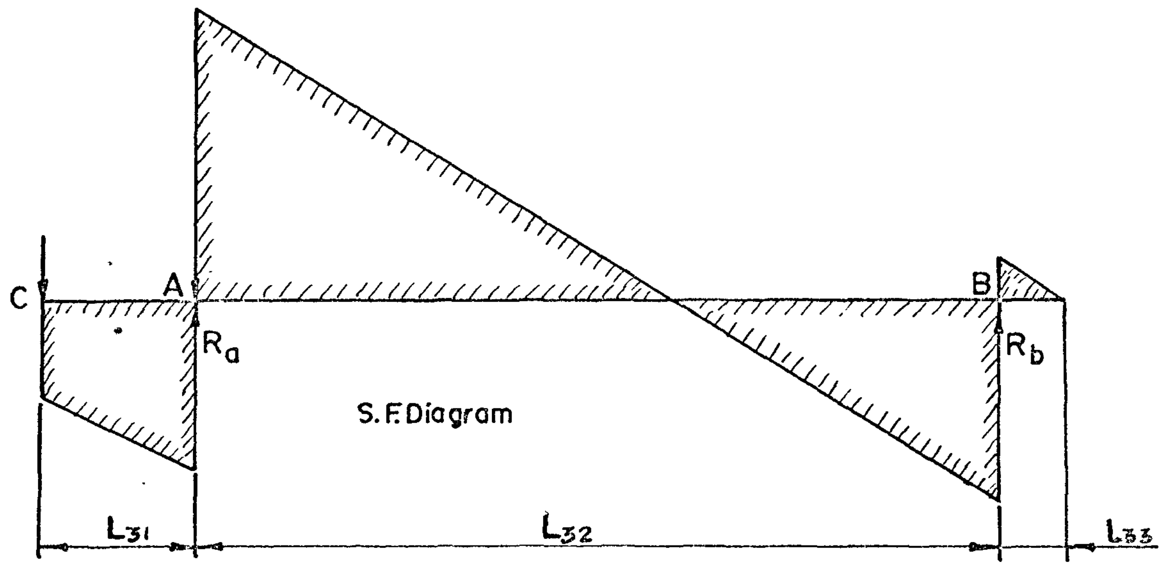
If the value of x_m is not $0 < x_m < L_{32}$, then the maximum bending moment is either M_1 or M_2.

The following figure shows the bending moment diagram of the portion AB of the pusher ram.



The following figure shows the shear force diagram of the pusher ram in the third critical position. The maximum shear force will occur at A and can be expressed as

$$Fs_3 = R_a - (W_3 + WL_{31})$$



The deflection Δ_{L3} at point C is found by method of superposition as was done in Appendix - A and Appendix - B.

Following the same analysis of Appendix - B, for finding the deflection, we have

Δ_{L3} = (Deflection $\Delta_L{}'$ at C due to load W_3 and W considering span AC as cantilever) - $L_{31}\theta_1$.

θ_1 = slope at point A

$$= \frac{WL_{32}^{3}}{24EI_x} - \frac{M_1L_{32}}{3EI_x} - \frac{M_2L_{32}}{6EI_x}$$

$$\Delta_{L3} = \Delta_L{}' - L_{31}\theta_1$$

$$\Delta_L{}' = \frac{WL_{31}^{4}}{8EI_x} + \frac{W_3L_{31}^{3}}{3EI_x}$$

therefore

$$\Delta_{L3} = \frac{WL_{31}^{4}}{8EI_x} + \frac{W_3L_{31}^{3}}{3EI_x} - \frac{WL_{32}^{3}}{24EI_x} + \frac{M_1L_{32}}{3EI_x} + \frac{M_2L_{32}}{6EI_x}$$

substituting the value of M_1 and M_2 from equation (1) and (2) and solving we get

$$\Delta_{L3} = \frac{1}{EI_x}\Big[\frac{WL_{31}^{4}}{8} + \frac{W_3L_{31}^{3}}{3} - \frac{WL_{32}^{3}L_{31}}{24} + \frac{W_3L_{32}L_{31}}{3}$$

$$+ \frac{F_7y_2L_{32}}{3} + \frac{WL_{32}L_{31}}{6} + \frac{WL_{33}^{2}L_{32}}{12} + \frac{F_7y_1L_{32}}{6}\Big]$$

APPENDIX - D

CALCULATION OF MAXIMUM SHEAR AND BENDING STRESS IN THE PUSHER RAM

Calculation of Maximum Shear and Bending Stress in the Pusher Ram

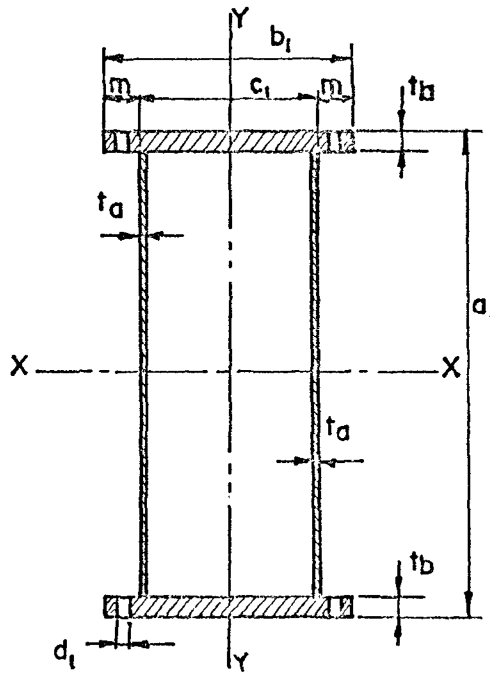


Figure above shows the cross-section of the pusher ram beam with all necessary dimensions. The area of cross-section is expressed as:

$$A = 2b_1 t_b + 2t_a(a_1 - 2t_b) - 4t_b d_1$$

The moment of inertia about x-x can be given as:

$$I_x = \frac{b_1 a_1^3}{12} - \frac{(c_1 - 2t_a)(a_1 - 2t_b)^3}{12} - \frac{m(a_1 - 2t_b)^3}{6} - \frac{d_1 t_b^3}{3} - 4d_1 t_b\left(\frac{a_1}{2} - \frac{t_b}{2}\right)^2$$

and moment of inertia about y-y is

$$I_y = \frac{a_1 b_1^3}{12} - \frac{(a_1 - 2t_b)(c_1 - 2t_a)^3}{12} - \frac{(a_1 - 2t_b)m^3}{6}$$

$$- \frac{t_b d_1^3}{3} - 4 t_b d_1 (\frac{c_1}{2} + \frac{m}{2})^2$$

Modulus of section about x-x is

$$Z_x = \frac{2I_x}{a_1}$$

and modulus of section about y-y is

$$Z_y = \frac{2I_y}{b_1}$$

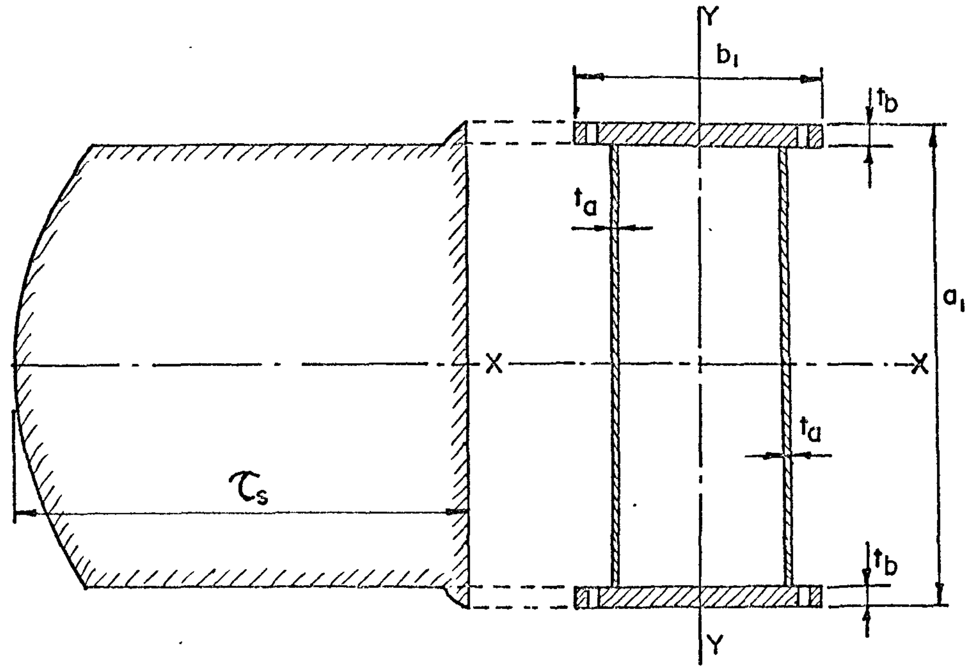
The maximum bending stress in the three critical positions of the pusher ram will be,

$$\sigma_{b1} = \frac{M_{max1}}{Z_x}$$

$$\sigma_{b2} = \frac{M_{max2}}{Z_x}$$

$$\sigma_{b3} = \frac{M_{max3}}{Z_x}$$

Following figure shows the shear stress distribution of the pusher ram section. The maximum shear stress will occur in the centre of the web plate. The general equation for shear stress in any section is



$$\tau_s = \frac{F_s A\bar{y}}{I b_o}$$

Where

τ_s = Shear stress at a point in the section.

A = Area of the section above that point.

$\bar{y}$ = Distance between the centroid of area above that point and that point.

b_o = Least width of the section above that point.

F_s = The maximum shear force in the beam.

I = Moment of inertia of the section about x-x.

The maximum shear stress will occur in the centre of the web plate and it can be expressed as:

$$\tau_s = \frac{F_s}{I_x t_a} [\ t_b(b_1 - 2d_1)(\frac{a_1}{2} - \frac{t_b}{2}) + t_a(\frac{a_1}{2} - t_b)^2\ ]$$

The maximum shear force in the three positions of the pusher ram is F_{s1} , F_{s2} and F_{s3} in first, second and third respectively.

Therefore the maximum shear stress in the pusher ram in three positions can be given as:

$$\tau_{s1} = \frac{F_{s1}}{I_x t_a} [\ t_b(b_1 - 2d_1)(\frac{a_1}{2} - \frac{t_b}{2}) + t_a(\frac{a_1}{2} - t_b)^2\ ]$$

$$\tau_{s2} = \frac{F_{s2}}{I_x t_a} [\ t_b(b_1 - 2d_1)(\frac{a_1}{2} - \frac{t_b}{2}) + t_a(\frac{a_1}{2} - t_b)^2\ ]$$

$$\tau_{s3} = \frac{F_{s3}}{I_x t_a} [\ t_b(b_1 - 2d_1)(\frac{a_1}{2} - \frac{t_b}{2}) + t_a(\frac{a_1}{2} - t_b)^2\ ]$$

APPENDIX - E

COMPOSITE COMPUTER PROGRAMME
FOR THE OPTIMIZATION OF COKE
PUSHER RAM

```
C       ************************            ****************************
C
C       COMPOSITE COMPUTER PROGRAMME FOR THE OPTIMIZATION OF COKE PUSHER
C                                     RAM.
C
C       ************************            ****************************
C
C
C**** THE FOLLOWING PARAMETERS ARE THE INPUT TO THIS COMPOSITE
C       COMPUTER PROGRAMME FOR THE DETERMINATION OF OPTIMIZED
C       SECTION OF THE COKE PUSHER RAM.
C
C
C
C**** THE FOLLOWING ARE THE PARAMETERS OF COKE OVEN PLANT NORMALLY
C       DETERMINED BY THE PROJECT DESIGNER AND MADE AVAILABLE TO THE
C       MACHINE DESIGNER.
C
C
C       A= LENGTH OF THE COKE OVEN FROM END TO END.
C
C       B= DISTANCE OF THE RAM HEAD FACE FROM THE END OF OVEN
C          BRICKWORK IN THE FORWARD MOST POSITION OF THE PUSHER RAM.
C
C       C= DISTANCE OF THE RAM HEAD FACE FROM THE END OF THE OVEN
C          BRICKWORK IN THE INITIAL POSITION OF THE PUSHER RAM.
C
C       D= DISTANCE BETWEEN THE RAM HEAD FACE AND THE C.L. OF THE
C          SUPPORTING SHOE.
C
C       E= DISTANCE BETWEEN THE C.L. OF FIRST AND SECOND
C          ROLLER SUPPORT.
C
C       F= DISTANCE BETWEEN THE C.L. OF SECOND AND THIRD SUPPORTS.
C
C       G= DISTANCE BETWEEN THE END OF OVEN BRICKWORK AND THE C.L.
C          OF FIRST ROLLER SUPPORT.
C
C       H= USEFULL HEIGHT OF THE COKE OVEN CHAMBER.
C
C       T= TOTAL TRAVEL OF THE PUSHER RAM.
C
C       CL= TOTAL LENGTH OF THE PUSHER RAM WITH RAM HEAD.
C
C       Q= DISTANCE BETWEEN THE FRONT END OF THE RAM (WITHOUT RAM
C          HEAD) AND THE C.L. OF THE SUPPORTING SHOE.
C
C       V= VOLUME OF COAL CHARGED IN THE OVEN (IN CU. FT.).
C
C       WO= MINIMUM WIDTH OF THE OVEN.
```

```
C**** THE FOLLOWING PARAMETERS ARE OF THE RELATED ELEMENTS
C     DECIDED EARLIER BY THE MACHINE DESIGNER.
C
C
C     WR= WEIGHT PER UNIT LENGTH OF RACK AND GUIDE COMBINED.
C
C     W3= WEIGHT OF RAM HEAD.
C
C     W4= WEIGHT OF SHOE ASSEMBLY.
C
C     V1= MEAN VELOCITY OF RAM.
C
C     TI= TIME OF ACCELARATION IN ATTAINING MAXIMUM SPEED.
C
C     SD= DIAMETER OF THE RIVET.
C
C     C1= DISTANCE BETWEEN THE PITCH LINE OF TOOTHED RACK AND
C         OUTSIDE SURFACE OF THE BOTTOM PLATE OF RAM.
C
C     C2= DISTANCE BETWEEN THE BOTTOM SURFACE OF THE RAM AND THE
C         BOTTOM SURFACE OF THE SUPPORTING SHOE.
C
C     WP= WIDTH OF THE TOOTHED RACK.
C
C     RT1= RISE IN TEMPERATURE OF THE RAM IN THE FIRST POSITION.
C
C     RT2= RISE IN TEMPERATURE OF THE RAM IN THE SECOND POSITION.
C
C
C**********************************************************************
C
C
C**** THE FOLLOWING ARE THE PARAMETERS WHICH ARE TRUE FOR ALL
C     COKE OVEN PLANT.
C
C
C     ALPHA= COEFFICIENT OF LINEAR EXPANSION OF STRUCTURAL STEEL.
C
C     CE= MODULUS OF ELASTICITY OF STRUCTURAL STEEL.
C
C     ROS= DENSITY OF STRUCTURAL STEEL IN LBS./CUB. IN.
C
C     RO= DENSITY OF COAL CHARGE IN KIP/CU.FT.
C
C     ETA= COKE OUTPUT PER UNIT WEIGHT OF COAL CHARGED.
C
C     GR= ACCELARATION DUE TO GRAVITY.
C
C     CK1= COEFFICIENT OF FRICTION BETWEEN COKE AND THE OVEN
C          BRICKWORK.
```

```
C      CK2= COEFFICIENT OF FRICTION BETWEEN THE SHOE AND OVEN SOLE.

C      CK3= COEFFICIENT WHICH TAKES INTO ACCOUNT THE EFFECT OF
C           FRICTION OF COKE WITH SIDE WALL OF THE OVEN.

C      CK4= COEFFICIENT WHICH TAKES INTO ACCOUNT THE EFFECT OF
C           EXCESSIVE FORCE CREATED DUE TO EXTRAORDINARY
C           STICKINESS OF COKE.

C**********************************************************************

C**** DIMENSIONS OF THE RAM SECTION.

C      SA= HEIGHT OF THE RAM SECTION.

C      SB= WIDTH OF THE FLANGE OF THE RAM SECTION.

C      STA= THICKNESS OF THE WEB PLATE.

C      STB= THICKNESS OF THE FLANGE PLATE.

C      AREA= NET AREA OF CROSS-SECTION OF THE PUSHER RAM.

C**********************************************************************

C**** THE FOLLOWING PARAMETERS ARE CALCULATED IN THE
C      PROGRAMME ITSELF.

C**** SPAN LENGTHS OF THE PUSHER RAM IN DIFFERENT POSITIONS.

C      C1L1= DISTANCE BETWEEN THE C.G. OF RAM HEAD AND C.L. OF FIRST
C            ROLLER SUPPORT IN THE FIRST POSITION OF THE PUSHER RAM.

C      C1L2= DISTANCE BETWEEN THE C.L. OF SHOE AND FIRST ROLLER
C            SUPPORT OF THE RAM IN THE FIRST POSITION OF
C            THE PUSHER RAM.

C      C1L3= DISTANCE BETWEEN THE C.L. OF FIRST AND THIRD SUPPORT
C            IN THE FIRST POSITION OF THE PUSHER RAM.

C      C1L4= THE LENGTH OF THE TAIL OVERHANG OF THE RAM IN THE
C            FIRST POSITION OF THE PUSHER RAM.

C      C2L1= DISTANCE BETWEEN THE C.G. OF RAM HEAD AND C.L. OF
C            SHOE IN THE SECOND POSITION OF THE PUSHER RAM.
```

```
C     C2L2= DISTANCE BETWEEN THE C.L. OF SHOE AND FIRST ROLLER
C           SUPPORT IN THE SECOND POSITION OF THE PUSHER RAM.

C     C2L3= THE LENGTH OF THE TAIL OVERHANG OF THE RAM IN THE
C           SECOND POSITION OF THE PUSHER RAM.

C     C3L1= DISTANCE BETWEEN THE C.G. OF RAM HEAD AND C.L. OF
C           THE SHOE IN THE THIRD POSITION OF THE PUSHER RAM.

C     C3L2= DISTANCE BETWEEN THE C.L. OF SHOE AND THE FIRST
C           SUPPORT IN THE THIRD POSITION OF THE PUSHER RAM.

C     C3L3= THE LENGTH OF THE TAIL OVERHANG OF THE RAM IN THE
C           THIRD POSITION OF THE PUSHER RAM.


C**** DIFFERENT FORCES ACTING ON THE PUSHER RAM.


C     F1= FORCE OF INERTIA OF COKE MASS.

C     F2= FRICTIONAL RESISTANCE DUE TO FRICTION BETWEEN COKE
C         AND OVEN BRICKWORK IN NORMAL CONDITION.

C     F3= TOTAL NORMAL FORCE OF RESISTANCE ACTING ON THE RAM(F1+F2).

C     F4= TOTAL MAXIMUM FORCE OF RESISTANCE ACTING ON THE RAM WITH
C         STICKER COKE IN THE FIRST CRITICAL POSITION.

C     F5= MAXIMUM AXIAL FORCE ACTING ON THE RAM IN THE SECOND
C         CRITICAL POSITION.

C     F6= NET MAXIMUM AXIAL FORCE ACTING ON RAM INCLUDING FRICTIONAL
C         FORCE DUE TO FRICTION BETWEEN SHOE AND SOLE OF OVEN IN THE
C         SECOND POSITION. (F5+FS)

      F7=  FRICTIONAL FORCE DUE TO FRICTION BETWEEN SHOE AND
C          OVEN SOLE IN THE THIRD POSITION OF THE PUSHER RAM.

C     FS2= FRICTIONAL FORCE DUE TO FRICTION BETWEEN SHOE AND
C          OVEN SOLE IN THE SECOND POSITION OF THE PUSHER RAM.


C**** THE REST OF THE PARAMETERS CALCULATED IN THE PROGRAMME
 C    IS GIVEN BELOW.


C     CLR= LENGTH OF THE RAM WITHOUT RAM HEAD.
```

```
C      T2= TOTAL TRAVEL OF THE PUSHER RAM FROM INITIAL POSITION
C          TO THE SECOND POSITION.

C      W1= WEIGHT PER UNIT LENGTH OF ONLY RAM SECTION.

C      W2= TOTAL WEIGHT OF COKE MASS.

C      W= WEIGHT PER UNIT LENGTH OF RAM INCLUDING RACK AND GUIDE.

C      SM= FLANGE EXTENSION BEYOND WEB PLATE.

C      SC= DISTANCE BETWEEN OUTSIDE SURFACE OF WEB PLATES.

C      WOA= MAXIMUM ALLOWED WIDTH OF THE RAM.

C      CS= CLEARANCE FOR SHOE FITTING.

C      SD1= DIAMETER OF THE RIVET HOLE.

C      Y1= DISTANCE BETWEEN THE C.L. OF RAM AND THE PITCH LINE
C          OF THE TOOTHED RACK.

C      Y2= DISTANCE BETWEEN THE C.L. OF THE RAM AND THE BASE
C          SURFACE OF THE SUPPORTING SHOE.

C      ZX= NET MODULUS OF SECTION OF THE RAM ABOUT X-AXIS.

C      ZY= NET MODULUS OF SECTION OF THE RAM ABOUT Y-AXIS.

C      SIY= NET MOMENT OF INERTIA OF THE RAM SECTION ABOUT Y-AXIS.

C      SIX= NET MOMENT OF INERTIA OF THE RAM SECTION ABOUT X-AXIS.

C      C1XM= DISTANCE OF THE POINT OF MAXIMUM BENDING MOMENT
C            FROM THE LEFT SUPPORT IN THE FIRST POSITION
C            OF THE PUSHER RAM.

C      C1MMAX= MAXIMUM BENDING MOMENT IN THE FIRST POSITION
C              OF THE PUSHER RAM.

C      C2MMAX= MAXIMUM BENDING MOMENT IN THE SECOND POSITION
C              OF THE PUSHER RAM.

C      C3MMAX= MAXIMUM BENDING MOMENT IN THE THIRD POSITION
C              OF THE PUSHER RAM.

C      S1MAX= MAXIMUM COMPRESSIVE STRESS IN THE FIRST POSITION
C             OF THE PUSHER RAM.

C      S2MAX= MAXIMUM COMPRESSIVE STRESS IN THE SECOND POSITION
C             OF THE PUSHER RAM.
```

```
C      S3MAX= MAXIMUM COMPRESSIVE AND TENSILE STRESS IN THE
C             THIRD POSITION OF THE PUSHER RAM.
C
C*************************************************************************
C
C
      DIMENSION XSEEK(10),XAPPROX(10),XFINAL(10),Z(10),JJ(10),J(10),XA(1
     10),XB(10),XS(10),YZ(10)
C
      COMMON/A1/A,B,C,D,E,F,G,H,T,CL,Q,V,WO,WR,W3,W4,V1,TI,SD,C1,C2,WP,R
     1T1,RT2
C
C      THE STARTING VALUE FOR OPTIMIZATION OF RAM SECTION SHOULD BE
C      SELECTED BY THE MACHINE DESIGNER.
C
      SA=25.0
      SB=12.5
      STB=0.82
      STA=0.27
C
C      THE FOLLOWING DATA IS TO BE SUPPLIED BY THE PROJECT DESIGNER.
C
      DATA A,B,C,D,E,F,G,H,T,CL,Q,V,WO/562.,109.,64.,130.,244.,197.,233.
     1,157.,735.,940.,114.,760.,15./
      DATA WR,W3,W4,V1,TI,SD,C1,C2,WP,RT1,RT2/0.0192,5.0,2.0,85.,0.6,0.8
     1125,3.0,39.4,12.5,7.,30./
      WRITE(6,5)
    5 FORMAT(1H0,///,22X,*RESULTS OF THE OPTIMIZATION OF COKE PUSHER RAM
     1 BY DIRECT SEARCH AND SUCCESSIVE LINEAR*,/,53X,*APPROXIMATION TECH
     2NIQUE.*,////)
C
      CALL SEEK(SA,SB,STA,STB,XS,U)
C
      USEEK=U
      DO 20 I=1,4
   20 XSEEK(I)=XS(I)
C
     *CALL APPROX(SA,SB,STA,STB,YZ,UR)
C
      UAPPROX=UR
      XAPPROX(1)=YZ(1)
      XAPPROX(2)=YZ(2)
      XAPPROX(3)=YZ(3)
      XAPPROX(4)=YZ(4)
      IF(USEEK.LT.UAPPROX) GO TO 15
      WRITE(6,105)
  105 FORMAT(1H0,10X,*DIRECT SEARCH METHOD FAILED TO PRODUCE BETTER RESU
     1LT THAN SUCCESSIVE LINEAR APPROXIMATION*,//,11X,*THE RESULT FROM D
     2IRECT SEARCH METHOD IS-*,/)
      DO 19 I=1,4
```

```
C     ROUNDING OFF THE VARIABLES IN MULTIPLE OF 1/16 INCH.
C
      Z(I)=XSEEK(I)*16.
      JJ(I)=Z(I)
      J(I)=JJ(I)+1
      XB(I)=J(I)
      XB(I)=XB(I)/16.0
   19 CONTINUE
      SA=XB(1)
      SB=XB(2)
      STA=XB(3)
      STB=XB(4)
      U=2.*STB*SB+2.*(SA-2.*STB)*STA-4.*STB*SD*1.2
      WRITE(6,110) U
      WRITE(6,115) SA,SB,STB,STA
      WRITE(6,85)
   85 FORMAT(1H0,30X,*RESULTS FROM THE SUCCESSIVE LINEAR APPROXIMATION T
     1ECHNIQUE,*,/,50X,*WHICH IS THE OPTIMUM*,//)
      DO 25 I=1,4
   25 XFINAL(I)=XAPPROX(I)
      GO TO 37
   15 WRITE(6,100)
  100 FORMAT(1H0,10X,*SUCCESSIVE LINEAR APPROXIMATION METHOD FAILED TO P
     1RODUCE BETTER RESULT THAN DIRECT SEARCH*,//,11X,*THE RESULT FROM S
     2UCCESSIVE LINEAR APPROXIMATION METHOD IS-*,/)
C
C     ROUNDING OFF THE VARIABLES IN MULTIPLE OF 1/16 INCH.
C
      DO 29 I=1,4
      Z(I)=XAPPROX(I)*16.
      JJ(I)=Z(I)
      J(I)=JJ(I)+1
      XA(I)=J(I)
      XA(I)=XA(I)/16.
   29 CONTINUE
      SA=XA(1)
      SB=XA(2)
      STA=XA(3)
      STB=XA(4)
      U=2.*STB*SB+2.*(SA-2.*STB)*STA-4.*STB*SD*1.2
      WRITE(6,110) U
      WRITE(6,115) SA,SB,STB,STA
      WRITE(6,95)
   95 FORMAT(1H0,30X,*RESULTS FROM THE DIRECT SEARCH TECHNIQUE,*,/,46X,*
     1WHICH IS THE OPTIMUM*,//)
      DO 35 I=1,4
   35 XFINAL(I)=XSEEK(I)
C
C     ROUNDING OFF THE VARIABLES IN MULTIPLE OF 1/16 INCH.
```

```
 37 DO 75 I=1,4
    Z(I)=XFINAL(I)*16.
    JJ(I)=Z(I)
    J(I)=JJ(I)+1
    XFINAL(I)=J(I)
    XFINAL(I)=XFINAL(I)/16.
75  CONTINUE
    SA=XFINAL(1)
    SB=XFINAL(2)
    STA=XFINAL(3)
    STB=XFINAL(4)
    U=2.*STB*SB+2.*(SA-2.*STB)*STA-4.*STB*SD*1.2
 45 WRITE(6,55) U
 55 FORMAT(1H0,10X,*THE OPTIMUM CROSS-SECTIONAL AREA OF THE PUSHER RAM
   1 IN SQ. INCH =*,F8.4)
    WRITE(6,65) SA,SB,STB,STA
 65 FORMAT(1H0,10X,*OPTIMUM DIMENSIONS OF THE RAM SECTION.*,//,11X,*HE
   1IGHT IN INCHES          =*,F8.4,/,11X,*WIDTH IN INCHES           =*,
   2F8.4,/,11X,*FLANGE THICKNESS IN INCH =*,F8.4,/,11X,*WEB THICKNESS
   3IN INCH    =*,F8.4)
110 FORMAT(1H0,10X,*THE CROSS-SECTIONAL AREA OF THE PUSHER RAM IN SQ.
   1INCH =*,F8.4)
115 FORMAT(1H0,10X,*DIMENSIONS OF THE RAM SECTION.*,//,11X,*HEIGHT IN
   1INCHES          =*,F8.4,/,11X,*WIDTH IN INCHES           =*,F8.4,/,1
   21X,*FLANGE THICKNESS IN INCH =*,F8.4,/,11X,*WEB THICKNESS IN INCH
   3   =*,F8.4,///////)
    STOP
    END
```

```
      SUBROUTINE SEEK(SA,SB,STA,STB,XS,U)
C
C
C     DIRECT SEARCH OPTIMIZATION TECHNIQUE.
C
C
      DIMENSION XS(10),XSTRT(10),PHI(20),XLAM(10),DX(10),XO(10),DXS(10)
     1XN(10)
      XSTRT(1)=SA
      XSTRT(2)=SB
      XSTRT(3)=STA
      XSTRT(4)=STB
      DX(1)=0.2
      DX(2)=0.1
      DX(3)=0.1
      DX(4)=0.1
      M=4
      N=10
      IFENCE= 1
      ICT=6
      EPS        =  1.0E-8
      LEVEL      =  1
      GS=3.0
      PDO=0.3
      PL=1.75
      IPRINT=0
      MAXM       =  50
C     PARAMETERS=
C
C     LEVEL=       INDEX FOR USER'S LEVEL,
C          LEVEL=0   FOR UNSOPHISTICATED USER.
C          LEVEL=1  FOR SOPHISTICATED USER

C     IPRINT=    PRINTING INDEX
C          IPRINT=J  PRINT EVERY J STEP.

C     M=   NUMBER OF INDEPENDENT VARIABLES.

C     N= NUMBER OF CONSTRAINTS.

C     MAXM=       MAXIMUM NUMBER OF COMPLETE MOVES ALLOWED IN THE SEARCH.

C     XLAM(I)=    STEP SIZE MULTIPLIER FOR EACH VARIABLE, (I=1,M),FOR
C           SOPHISTICATED USER.

C     GS=  STEP SIZE MULTIPLIER , FOR UNSOPHISTICATED USER.

C     PD=    PATTERN MOVE COEFFICIENT.

C     PL=     MULTIPLIER FOR PATTERN MOVE COEFFICIENT.
```

```
C       XSTRT(I)=     STARTING VALUE FOR EACH VARIABLE,(I=1,M) EACH IS SET
C             EQUAL TO 1.E-6 FOR UNSOPHISTICATED USER.
C
C       XS(I)=       THE OPTIMUM OUTPUT VALUE FOR EACH VARIABLE,(I=1,M)

C       U=      OPTIMUM OUTPUT VALUE OF THE OPTIMIZATION FUNCTION.

C       KO=      RESULT INDICATOR,
C            KO=0   ACCEPTABLE RESULT
C            KO=1    OPTIMUM IS NOT FOUND IN MAXM CYCLES.

C       OPTIMF=      ARTIFICIAL OPTIMIZATION SUBROUTINE CALLED.
C
C       INITIALIZE,SET STARTING CONDITION
        DO 100 I  =  1,M
        XLAM(I)      =  GS
    100 CONTINUE
        KT         =  0
        KO         =  0
        IP         =  0
        NSEAR=0
        PD         =  PDO
        DO 2 I=1,M
        XS(I)      =  XSTRT(I)
        XO(I)=XS(I)
        DXS(I)=DX(I)
  2     CONTINUE
   61   CALL OPTIMF(XS,U,PHI,M,N)
        UF=U
        US=U
     60 I=1
C       SEARCH IN ONE DIRECTION AT THE TIME
  3     XS(I)=XS(I)+DX(I)
        CALL OPTIMF(XS,U,PHI,M,N)
        IF(U.LT.US) GO TO 4
        XS(I)=XS(I)-2.*DX(I)
        CALL OPTIMF(XS,U,PHI,M,N)
        IF(U.LT.US) GO TO 5
        XS(I)=XS(I)+DX(I)
   63   IF(I.EQ.M) GO TO 6
        I=I+1
        GO TO 3
C       SUCCESSFULL IN MOVING TO POSITIVE DIRECTION
C       INCREASE STEPSIZE
  4     CONTINUE
        US=U
        DX(I)=DX(I)*XLAM(I)
        XS(I)=XS(I)+DX(I)
        CALL OPTIMF(XS,U,PHI,M,N)
        IF(U.LT.US) GO TO 4
```

```
C       RETURN TO LAST POSITION DUE TO FAILURE
        XS(I)=XS(I)-DX(I)
C       SET STEP LENGTH EQUAL TO ITS ORIGINAL VALUE
        DX(I)=DXS(I)
        IF(IFENCE.EQ.0) GO TO 3
        GO TO 63

C       SUCCESSFULL IN MOVING IN NEGATIVE DIRECTION

  5     CONTINUE
        DX(I)=-DX(I)
        GO TO 4
  6     CONTINUE

C       TEST TO SEE IF ANY VARIABLE HAS BEEN CHANGED

        CALL OPTIMF(XS,U,PHI,M,N)
        TEST=ABS((U-UF)/UF)
        IF(U.LE.1.0E-20 ) TEST=U
        IF(TEST.LE.EPS.AND.KT.LT.ICT) GO TO 7
        IF (TEST .LT. EPS .AND. KT .GE. ICT) GO TO 30
        GO TO 8
C
C       DECREASE  THE STEP SIZES BY A FACTOR OF 10.
C
      7 DO 18 I   =   1,M
        DX(I)     =   DX(I)/10.0
     18 DXS(I)    =   DX(I)
        KT        =   KT + 1
        GO TO 61
  8     CONTINUE

C       FIRST STEP OF PATTERN MOVE

        US=U
        PD=PDO
        DO 42 I=1,M
        XN(I)=XS(I)
 42     CONTINUE
     15 CONTINUE

C       MAKE A PATTERN MOVE

        DO 9  I  =  1,M
        XN(I)  =  XN(I) + (XS(I) - XO(I)) * PD
      9 CONTINUE
        CALL OPTIMF(XS,U,PHI,M,N)
        IF (U .LT.US) GO TO 14
     10 CONTINUE
```

```
C     CHECK IF THE LAST MOVE WAS IN THE POSITIVE OR NEGATIVE DIRECTION

      IF (PD.LT.0.) GO TO 13

C     RETURN TO LAST POSITION DUE TO FAILURE

      DO 40 I=1,M
      XN(I)=XN(I)-(XS(I)-XO(I))*PD
 40   CONTINUE
      PD  =  -PDO
      GO TO 15
   13 CONTINUE
      DO 16 I =1,M
      XS(I)=XN(I)-(XS(I)-XO(I))*PD
 16   XO(I) =XS(I)
      UF  =  US
      NSEAR=NSEAR+1
      IF (NSEAR.GT.MAXM) GO TO 20
      IP        =  IP + 1
      IF (IPRINT .EQ. 0) GO TO 61
      IT        =  (IP/IPRINT)*IPRINT
      IF (IT .EQ. IP) WRITE (6,22) IP,UF,(I,XS(I), I = 1,M)
   22 FORMAT (1H0,6HEND OF,I3,35H CYCLES, VALUE OF OBJECT FUNCTION =,
     1 E16.8/25X,18HVALUE OF VARIABLES/(3H X(,I3,2H)=,E16.8,4H  X(,I3,2H
     2)=,E16.8,4H  X(,I3,2H)=,E16.8,4H  X(,I3,2H)=,E16.8))
      GO TO 61

C     FURTHER PATTERN MOVES IN THE SAME DIRECTION

   14 CONTINUE
      PD  =  PD * PL
      DO 11  I  =  1,M
      XN(I)  =  XN(I) + (XS(I) - XO(I)) * PD
   11 CONTINUE
      US  =  U
      CALL OPTIMF(XS,U,PHI,M,N)
      IF (U .LT.US) GO TO 14
C
C     RETURN TO LAST POSITION DUE TO FAILURE

      DO 41 I=1,M
      XN(I)=XN(I)-(XS(I)-XO(I))*PD
 41   CONTINUE
      PD=PDO
      GO TO 15
   20 CONTINUE
       WRITE(6,33) MAXM
 33   FORMAT(1H , 29H  OPTIMUM CANNOT BE FOUND IN ,I3,7H CYCLES)
      KO        =  1
      GO TO 1000
```

```
C     SENSITIVITY ANALYSIS PERFORMED FROM HERE TO LOCATE A BETTER OPTIMUM
C     POINT.
C
   30 US=U
      DELX=0.001
  55  DO 28 I=1,4
   28 XS(I)=XS(I)-DELX
      CALL OPTIMF(XS,U,PHI,M,N)
      IF(U.GT.US) GO TO 29
      US=U
      GO TO 55
   29 DO 44 I=1,4
   44 XS(I)=XS(I)+DELX
      CALL OPTIMF(XS,U,PHI,M,N)
   45 US=U
      XS(1)=XS(1)-DELX
      CALL OPTIMF(XS,U,PHI,M,N)
      IF(U.LT.US) GO TO 45
      XS(1)=XS(1)+DELX
      CALL OPTIMF(XS,U,PHI,M,N)
   46 US=U
      XS(2)=XS(2)-DELX
      CALL OPTIMF(XS,U,PHI,M,N)
      IF(U.LT.US) GO TO 46
      XS(2)=XS(2)+DELX
      CALL OPTIMF(XS,U,PHI,M,N)
   47 US=U
      XS(3)=XS(3)-DELX
      CALL OPTIMF(XS,U,PHI,M,N)
      IF(U.LT.US) GO TO 47
      XS(3)=XS(3)+DELX
      CALL OPTIMF(XS,U,PHI,M,N)
   48 US=U
      XS(4)=XS(4)-DELX
      CALL OPTIMF(XS,U,PHI,M,N)
      IF(U.LT.US) GO TO 48
      XS(4)=XS(4)+DELX
      CALL OPTIMF(XS,U,PHI,M,N)
 1000 RETURN
      END
```

```
      SUBROUTINE OPTIMF(XS,U,PHI,M,N)
C
C     THIS SUBROUTINE DOES ALL NECESSARY DESIGN CALCULATION.
C
      DIMENSION XS(10),XSTRT(10),PHI(20),XLAM(10),DX(10),XO(10),DXS(10),
     1XN(10)
C
      COMMON/A1/A,B,C,D,E,F,G,H,T,CL,Q,V,WO,WR,W3,W4,V1,TI,SD,C1,C2,WP,R
     1T1,RT2
C
C     THE FOLLOWING DATA IS UNIVERSAL CONSTANTS OR FIXED PARAMETERS
C     FOR ALL COKE OVEN PLANTS.
C
      DATA RO,ETA,GR,CK1,CK2,CK3,CK4,ROS,CE,ALPHA/0.05,0.75,32.,0.75,0.5
     1,1.3,2.0,0.284,30000.,0.0000061/
C
      SA=XS(1)
      SB=XS(2)
      STA=XS(3)
      STB=XS(4)
C
C     CALCULATION OF THE SPANS OF THE PUSHER RAM IN THE FIRST POSITION.
C
      T2=0.9*T
      CLR=CL-(D-Q)
      C1L1=G+Q
      C1L2=G
      C1L3=E+F
      C1L4=CLR-(C1L1+C1L3)
C
C     CALCULATION OF THE SPANS OF THE PUSHER RAM IN THE SECOND POSITION.
C
      C2L1=Q
      C2L2=G-(D+C)+T2
      C2L3=CLR-(C2L1+C2L2)
C
C     CALCULATION OF THE SPANS OF THE PUSHER RAM IN THE THIRD POSITION.
C
      C3L1=Q
      C3L2=G-(D+C)+T
      C3L3=CLR-(C3L1+C3L2)
C
C     CALCULATION OF AXIAL FORCES.
C
      W2=V*RO*ETA
      F1=(W2*V1)/(60.*TI*GR)
      F2=W2*CK1*CK3
      F3=F1+F2
      F4=F1+CK4*F2
      F5=0.6*F2
```

```
C       CALCULATION OF THE AREA OF CROSS-SECTION, MOMENT OF INERTIA AND
C       MODULUS OF SECTION OF THE PUSHER RAM.
C
        SM=3.5*SD
        SD1=1.2*SD
        SC=SB-2.*SM
        AREA=2.*SB*STB+2.*(SA-2.*STB)*STA-4.*STB*SD1
        SIX=SB*SA**3/12.-(SC-2.*STA)*(SA-2.*STB)**3/12.-SM*(SA-2.*STB)**3/
       16.-SD1*STB**3/3.-4.*SD1*STB*(SA/2.-STB/2.)**2
        SIY=SA*SB**3/12.-(SA-2.*STB)*(SC-2.*STA)**3/12.-(SA-2.*STB)*SM**3/
       16.-STB*SD1**3/3.-4.*STB*SD1*(SC/2.+SM/2.)**2
        ZX=2.*SIX/SA
        ZY=2.*SIY/SB
C
C       CALCULATION OF THE SELF WEIGHT OF THE PUSHER RAM ONLY.
C
        W1=ROS*AREA/1000.
        W=W1+WR
C
C       CALCULATION OF THE MAXIMUM BENDING MOMENT  IN THE PUSHER RAM IN
C       THE FIRST POSITION.
C
        Y1=C1+SA/2.
        C1M2=W*C1L4**2/2.
        DEL1=((3.*CE*SIX)/(3.*CE*SIX-F4*C1L3*C1L1))*((W*C1L1**4)/(8.*CE*SI
       1X)+(W3*C1L1**3)/(3.*CE*SIX)+W4*(3.*C1L2*C1L1-C1L2**3)/(6.*CE*SIX))
       2-((W*C1L3**3*C1L1)/(24.*CE*SIX)-(C1M2*C1L3*C1L1)/(6.*CE*SIX)-((C1L
       33*C1L1)/(3.*CE*SIX))*(W3*C1L1+W4*C1L2+W*C1L1**2/2.-F4*Y1))
        C1M1=W3*C1L1+W4*C1L2+W*C1L1**2/2.-F4*Y1+F4*DEL1
        C1XM=C1L3/2.+(C1M1-C1M2)/(W*C1L3)
        C1MMAX=W*C1L3*C1XM/2.-W*C1XM**2/2.-C1M2-(C1M1-C1M2)*(C1L3-C1XM)
        C1MMAX=C1M1
        S1MAX=F4/AREA+C1MMAX/ZX+ALPHA*CE*RT1
C
C       CALCULATION OF MAXIMUM BENDING MOMENT AND COMPRESSIVE STRESS FOR T
C       SECOND POSITION OF THE PUSHER RAM.
C
        Y2=C2+SA/2.
        FS2=(CK2/(C2L2-CK2*(Y2-Y1)))*(W3*(C2L1+C2L2)+W4*C2L2+W*(C2L1+C2L2)
       1**2/2.-W*C2L3**2/2.-F5*Y1)
        F6=F5+FS2
        C2M2=W*C2L3**2/2.+F6*Y1
        DEL2=((3.*CE*SIX)/(3.*CE*SIX-F5*C2L2*C2L1))*(W*C2L1**4/(8.*CE*SIX)
       1+W3*C2L1**3/(3.*CE*SIX)-W*C2L2**3*C2L1/(24.*CE*SIX)+C2M2*C2L2*C2L1
       2/(6.*CE*SIX)+(C2L2*C2L1/(3.*CE*SIX))*(W3*C2L1+W*C2L1**2/2.+FS2*Y2)
       3)
        C2M1=W3*C2L1+W*C2L1**2/2.+FS2*Y2+F5*DEL2
        CJ=SQRT(CE*SIX/F6)
        D1=C2M1-W*CJ**2
        D2=C2M2-W*CJ**2
```

```
      C2XM=CJ*ATAN((D2-D1*COS(C2L2/CJ))/D1*SIN(C2L2/CJ))
      C2MMAX=D1/COS(C2XM/CJ)+W*CJ**2
      S2=ALPHA*CE*RT2
      FT2=S2*AREA
      IF(FT2.GT.F6) GO TO 5
      STEM2=S2
      GO TO 7
    5 STEM2=F6/AREA
    7 S2MAX=C2MMAX/ZX+F6/AREA+STEM2
C
C     CALCULATION OF MAXIMUM BENDING MOMENT AND STRESS IN THE THIRD
C     POSITION OF THE PUSHER RAM.
C
      F7=(W3*(C3L1+C3L2)/C3L2+W4+W*(C3L1+C3L2)**2/(2.*C3L2)+W*C3L3**2/(2
     1.*C3L2))*(CK2*C3L2/(C3L2-CK2*(Y2-Y1)))
      C3M1=W3*C3L1+F7*Y2+W*C3L1**2/2.
      C3M2=W*C3L3**2/2.+F7*Y1
      CJ1=SQRT(CE*SIX/F7)
      D11=C3M1-W*CJ1**2
      D22=C3M2-W*CJ1**2
      C3XM=CJ1*ATAN((D22-D11*COS(C3L2/CJ1))/D11*SIN(C3L2/CJ1))
      C3MMAX=D11/COS(C3XM/CJ1)+W*CJ1**2
      IF(FT2.GT.F7) GO TO 25
      STEM3=S2
      GO TO 27
   25 STEM3=F7/AREA
   27 S3MAX=C3MMAX/ZX+F7/AREA+STEM3
C
C     CALCULATION OF MAXIMUM SHEAR FORCE AND SHEAR STRESS IN THE WEB
C     PLATE OF THE RAM SECTION.
C
      RA=(1./C1L3)*(W3*(C1L1+C1L3)+W4*(C1L2+C1L3)+W*(C1L1+C1L3)**2/2.-W*
     1C1L4**2/2.-F4*Y1)
      ASHF=W3+W4+W*C1L1
      RSHF=RA-ASHF
      SHF=RSHF
      IF(ASHF.GT.RSHF) SHF=ASHF
      SHS=(SHF/(SIX*STA))*(STB*(SB-2.*SD1)*(SA/2.-STB/2.)+STA*(SA/2.-STB
     1)**2)
      WOA=0.88*WO
      CS=H/5.
C
   50 PHI(1)=12.5-S1MAX
      PHI(2)=12.5-S2MAX
      PHI(3)=12.5-S3MAX
      PHI(4)=5.0-SHS
      PHI(5)=SB-WP
      PHI(6)=WOA-SB
      PHI(7)=(H/3.-CS)-SA/2.
      PHI(8)=STB-SD
```

```
      PHI(9)=STA-STB/3.
      PHI(10)=2.0-DEL1
      U=2.*STB*SB+2.*(SA-2.*STB)*STA-4.*STB*SD1
      DO 250 I=1,N
250   IF(PHI(I).GE.0.00001) PHI(I)=0.0
      DO 255 I=1,N
  255 U=U+ABS(PHI(I))*1.E+20
   60 RETURN
      END
```

```
    ** SUBROUTINE APPROX(SA,SB,STA,STB,Z,UR)
C
C      OPTIMIZATION TECHNIQUE USING SUCCESSIVE LINEAR APPROXIMATION.
C
       DIMENSION Z(20),STEPX(20),X(40),S(40),DELX(20),A(40,40),B(40),
      1BB(40),C(40),II(40),III(40),V(20),XUP(20),XLO(20),PSI(20)
C    †
C
C      NUM,NUMR          - NUMBER OF EQUALITY CONSTRAINTS.

C      NEQ               - NUMBER OF INEQUALITY CONSTRAINTS.

C      K                 - NUMBER OF REAL VARIABLES.

C      NMAX              - MAXIMUM ALLOWED NUMBER OF ITERATIONS IN A
C                          SIMPLEX CYCLE.

C      INDEXI            - INDICATOR FOR PHASE I OR II OF THE SIMPLEX CYCLE.

C      TES               - CRITERION FOR OPTIMUM.

C      M                 - TOTAL NUMBER OF CONSTRAINTS FOR THE SIMLEX SOLUTION

C      Z(I),X(I)         - VARIABLE NAME.

C      BB(I),B(I)        - CONTROLLED STEP SIZE FOR THE VARIABLES DURING
C                          SIMPLEX OPERATION.

C      STEPX(I)          - A SMALL INCREMENT FOR EACH VARIABLE.

C      NCYCLE            - COUNTER FOR THE NUMBER OF SIMPLEX CYCLE.

C      UR                - REAL OPTIMUM VALUE.

C      A(I,J)            - SIMPLEX MATRIX.

C      C,S               - CO-EFFICIENTS IN THE OBJECTIVE FUNCTION.

C      PHI               - INEQUALITY CONSTRAINT FUNCTION.
C
C
C      READ IN AND STORE THE PROBLEM VARIABLES IN COMPUTOR MEMORY

       READ(5,200) K,NUM,NEQ,NMAX,INDEXI
       TES=0.00001
       IMM=2*K
       IM=IMM+NEQ
       IN=IM+IMM
       M=IM+NUM
    ⊕  Z(1)=SA
```

```
      Z(2)=SB
      Z(3)=STA
      Z(4)=STB
      READ(5,203) (BB(I), I=1,IMM)
      READ(5,205) (STEPX(I), I=1,K)
C     TRANSFER INITIAL VALUES FROM STORAGE TO WORKING LOCATIONS
      DO 5 I=1,M
    5 III(I)=IMM+I
      NCYCLE=0
    1 N=IN+NUM
      MM=IMM
      NUMR=NUM
      INDEX=INDEXI
      DO 2 I=1,M
    2 II(I)=III(I)
C
C     SET UP THE MATRIX FOR LINEAR APPROXIMATION
C
      CALL MATRIX(A,B,BB,Z,STEPX,C,NUMR,N,M,MM,X,S,II,K,NEQ)
C
C     CALCULATE INITIAL OPTIMUM VALUE
C
      CALL REALU(UR,Z,UI)
      NCYCLE=NCYCLE+1
C
C     START LINEAR APPROXIMATION ROUTINE
C
C     SIMPLEX OPERATION
      CALL SIMPLE(A,B,C,NUMR,N,M,MM,INDEX,X,NMAX,II,S)
C
C     CALCULATE NEW VALUES FOR BASIC VARIABLES
C
      DO 31 I=1,K
      Z(I)=Z(I)+X(2*I-1)-X(2*I)
      DELX(I)=X(2*I-1)-X(2*I)
   31 CONTINUE
C
C     CHECK FOR FINAL OPTIMUM VALUE
C
      CALL REALU(UR,Z,UP)
      CALL CONST(PSI,Z)
      IF(ABS(UI-UP).LT.TES) GO TO 1000
      GO TO 1
 1000 CONTINUE
 200  FORMAT(8I3)
  203 FORMAT(8F10.5)
  205 FORMAT(8F10.5)
      RETURN
      END
```

```
      SUBROUTINE MATRIX(A,B,BB,Z,STEPX,C,NUMR,N,M,MM,X,S,II,K,NEQ)

      DIMENSION Z(20),STEPX(20),X(40),PSI(20),PHI(50),PSIN(20),PHIN(20),
     1A(40,40),B(40),BB(40),C(40),II(40),S(40)
C
C     SET UP INCREMENTS AND THE MATRIX
C
C     SET UP COEFFICIENTS OF OBJECTIVE FUNCTION
C
      DO 5 I=1,M
      DO 5 J=1,N
    5 A(I,J)=0.0
      CALL REALU(U, Z, UK)
      IF(NUMR.NE.0) CALL CONST(PSI, Z)
      CALL ENEQ(PHI,Z)
      DO 10 I=1,K
      Z(I)=Z(I)+STEPX(I)
      CALL REALU(UN,Z,UK)
      IJ=2*I-1
      S(IJ)=(UN-U)/STEPX(I)
      S(IJ+1)=-S(IJ)
      IF(NUMR.NE.0) CALL CONST(PSIN,Z)
      CALL ENEQ(PHIN,Z)
      DO 9 J=1,NEQ
      JI=J+MM
      A(JI,IJ)=-(PHIN(J)-PHI(J))/STEPX(I)
      A(JI,IJ+1)=-A(JI,IJ)
    9 CONTINUE
      IF(NUMR.EQ.0) GO TO 11
      DO 6 J=1,NUMR
      JI=J+MM+NEQ
      A(JI,IJ)=(PSIN(J)-PSI(J))/STEPX(I)
      A(JI,IJ+1)=-A(JI,IJ)
    6 CONTINUE
   11 CONTINUE
      Z(I)=Z(I)-STEPX(I)
   10 CONTINUE
C
C     SET UP EQUATIONS FOR UPPER AND LOWER LIMITS
C
      MMK=MM-1
      J=0
      DO 12 I=1,MMK,2
      J=J+1
      JJ=2*J-1
      A(I,JJ)=1.0
      A(I+1,JJ)=-1.0
      A(I+1,JJ+1)=1.0
   12 A(I,JJ+1)=-1.0
```

```
C     SET UP B(I)
      DO 20 I=1,MM
   20 B(I)=BB(I)
      MP=MM+1
      MEQ=MM+NEQ
      DO 19 I=MP,MEQ
      J=I-MM
      IF(ABS(PHI(J)).LE.0.001) PHI(J)=0.0
   19 B(I)=PHI(J)
      IF(NUMR.EQ.0) GO TO 16
      MEQI=MEQ+1
      DO 21 I=MEQI,M
      J=I-MEQ
   21 B(I)=-PSI(J)
   16 CONTINUE
C
C     SET UP SLACK VARIABLES
C
      DO 22 I=1,M
      DO 15 J=MP,N
      S(J)=0.0
   15 A(I,J)=0.0
      MI=MM+I
C
C     CHECK FOR NEGATIVE VALUES OF B AND REARRANGE IF NECESSARY

   22 A(I,MI)=1.0
C
      CALL ORDER(A,B,NUMR,N,M,MM,K,II)
C
C     SET INITIAL FEASIBLE BASIS
C
      DO 30 I=1,MM
   30 X(I)=0.0
      DO 35 I=1,M
      MMI=MM+I
   35 X(MMI)=B(I)
      RETURN
      END
```

```
      SUBROUTINE REALU(U,X,UR)

      DIMENSION X(20),PSI(20)
      COMMON/A1/A,B,C,D,E,F,G,H,T,CL,Q,V,WO,WR,W3,W4,V1,TI,SD,C1,C2,WP,R
     1T1,RT2
      UR=2.*X(4)*X(2)+2.*(X(1)-2.*X(4))*X(3)-4.*X(4)*1.2*SD
      U=UR
      RETURN
      END




      SUBROUTINE CONST(PSI,X)

      DIMENSION X(20), PSI(20)
      PSI(1)=0.0
      RETURN
      END
```

```
      SUBROUTINE ENEQ(PHI,X)
C
C     THIS SUBROUTINE DOES ALL NECESSARY DESIGN CALCULATION.
C
      DIMENSION X(20), PHI(20)
      COMMON/A1/A,B,C,D,E,F,G,H,T,CL,Q,V,WO,WR,W3,W4,V1,TI,SD,C1,C2,WP,R
     1T1,RT2
C
C     THE FOLLOWING DATA IS UNIVERSAL CONSTANTS OR FIXED PARAMETERS
C     FOR ALL COKE OVEN PLANTS.
C
      DATA RO,ETA,GR,CK1,CK2,CK3,CK4,ROS,CE,ALPHA/0.05,0.75,32.,0.75,0.5
     1,1.3,2.0,0.284,30000.,0.0000061/
C
      SA=X(1)
      SB=X(2)
      STA=X(3)
      STB=X(4)
C
C     CALCULATION OF THE SPANS OF THE PUSHER RAM IN THE FIRST POSITION.
C
      T2=0.9*T
      CLR=CL-(D-Q)
      C1L1=G+Q
      C1L2=G
      C1L3=E+F
      C1L4=CLR-(C1L1+C1L3)
C
C     CALCULATION OF THE SPANS OF THE PUSHER RAM IN THE SECOND POSITION.
C
      C2L1=Q
      C2L2=G-(D+C)+T2
      C2L3=CLR-(C2L1+C2L2)
C
C     CALCULATION OF THE SPANS OF THE PUSHER RAM IN THE THIRD POSITION.
C
      C3L1=Q
      C3L2=G-(D+C)+T
      C3L3=CLR-(C3L1+C3L2)
C
C     CALCULATION OF AXIAL FORCES.
C
      W2=V*RO*ETA
      F1=(W2*V1)/(60.*TI*GR)
      F2=W2*CK1*CK3
      F3=F1+F2
      F4=F1+CK4*F2
      F5=0.6*F2
C
C     CALCULATION OF THE AREA OF CROSS-SECTION, MOMENT OF INERTIA AND
C     MODULUS OF SECTION OF THE PUSHER RAM.
```

```
      SM=3.5*SD
      SD1=1.2*SD
      SC=SB-2.*SM
      AREA=2.*SB*STB+2.*(SA-2.*STB)*STA-4.*STB*SD1
      SIX=SB*SA**3/12.-(SC-2.*STA)*(SA-2.*STB)**3/12.-SM*(SA-2.*STB)**3/
     16.-SD1*STB**3/3.-4.*SD1*STB*(SA/2.-STB/2.)**2
      SIY=SA*SB**3/12.-(SA-2.*STB)*(SC-2.*STA)**3/12.-(SA-2.*STB)*SM**3/
     16.-STB*SD1**3/3.-4.*STB*SD1*(SC/2.+SM/2.)**2
      ZX=2.*SIX/SA
      ZY=2.*SIY/SB
C
C     CALCULATION OF THE SELF WEIGHT OF THE PUSHER RAM ONLY.
C
      W1=ROS*AREA/1000.
      W=W1+WR
C
C     CALCULATION OF THE MAXIMUM BENDING MOMENT  IN THE PUSHER RAM IN
C     THE FIRST POSITION.
C
      Y1=C1+SA/2.
      C1M2=W*C1L4**2/2.
      DEL1=((3.*CE*SIX)/(3.*CE*SIX-F4*C1L3*C1L1))*((W*C1L1**4)/(8.*CE*SI
     1X)+(W3*C1L1**3)/(3.*CE*SIX)+W4*(3.*C1L2*C1L1-C1L2**3)/(6.*CE*SIX))
     2-((W*C1L3**3*C1L1)/(24.*CE*SIX)-(C1M2*C1L3*C1L1)/(6.*CE*SIX)-((C1L
     33*C1L1)/(3.*CE*SIX))*(W3*C1L1+W4*C1L2+W*C1L1**2/2.-F4*Y1))
      C1M1=W3*C1L1+W4*C1L2+W*C1L1**2/2.-F4*Y1+F4*DEL1
      C1XM=C1L3/2.+(C1M1-C1M2)/(W*C1L3)
      C1MMAX=W*C1L3*C1XM/2.-W*C1XM**2/2.-C1M2-(C1M1-C1M2)*(C1L3-C1XM)
      C1MMAX=C1M1
      S1MAX=F4/AREA+C1MMAX/ZX+ALPHA*CE*RT1
C
C     CALCULATION OF MAXIMUM BENDING MOMENT AND COMPRESSIVE STRESS FOR TH
C     SECOND POSITION OF THE PUSHER RAM.
C
      Y2=C2+SA/2.
      FS2=(CK2/(C2L2-CK2*(Y2-Y1)))*(W3*(C2L1+C2L2)+W4*C2L2+W*(C2L1+C2L2)
     1**2/2.-W*C2L3**2/2.-F5*Y1)
      F6=F5+FS2
      C2M2=W*C2L3**2/2.+F6*Y1
      DEL2=((3.*CE*SIX)/(3.*CE*SIX-F5*C2L2*C2L1))*(W*C2L1**4/(8.*CE*SIX)
     1+W3*C2L1**3/(3.*CE*SIX)-W*C2L2**3*C2L1/(24.*CE*SIX)+C2M2*C2L2*C2L1
     2/(6.*CE*SIX)+(C2L2*C2L1/(3.*CE*SIX))*(W3*C2L1+W*C2L1**2/2.+FS2*Y2)
     3)
      C2M1=W3*C2L1+W*C2L1**2/2.+FS2*Y2+F5*DEL2
      CJ=SQRT(CE*SIX/F6)
      D1=C2M1-W*CJ**2
      D2=C2M2-W*CJ**2
      C2XM=CJ*ATAN((D2-D1*COS(C2L2/CJ))/D1*SIN(C2L2/CJ))
      C2MMAX=D1/COS(C2XM/CJ)+W*CJ**2
      S2=ALPHA*CE*RT2
```

```
      FT2=S2*AREA
      IF(FT2.GT.F6) GO TO 5
      STEM2=S2
      GO TO 7
    5 STEM2=F6/AREA
    7 S2MAX=C2MMAX/ZX+F6/AREA+STEM2
C
C     CALCULATION OF MAXIMUM BENDING MOMENT AND STRESS IN THE THIRD
C     POSITION OF THE PUSHER RAM.
C
      F7=(W3*(C3L1+C3L2)/C3L2+W4+W*(C3L1+C3L2)**2/(2.*C3L2)+W*C3L3**2/(2
     1.*C3L2))*(CK2*C3L2/(C3L2-CK2*(Y2-Y1)))
      C3M1=W3*C3L1+F7*Y2+W*C3L1**2/2.
      C3M2=W*C3L3**2/2.+F7*Y1
      CJ1=SQRT(CE*SIX/F7)
      D11=C3M1-W*CJ1**2
      D22=C3M2-W*CJ1**2
      C3XM=CJ1*ATAN((D22-D11*COS(C3L2/CJ1))/D11*SIN(C3L2/CJ1))
      C3MMAX=D11/COS(C3XM/CJ1)+W*CJ1**2
      IF(FT2.GT.F7) GO TO 25
      STEM3=S2
      GO TO 27
   25 STEM3=F7/AREA
   27 S3MAX=C3MMAX/ZX+F7/AREA+STEM3
C
C     CALCULATION OF MAXIMUM SHEAR FORCE AND SHEAR STRESS IN THE WEB
C     PLATE OF THE RAM SECTION.
C
      RA=(1./C1L3)*(W3*(C1L1+C1L3)+W4*(C1L2+C1L3)+W*(C1L1+C1L3)**2/2.-W*
     1C1L4**2/2.-F4*Y1)
      ASHF=W3+W4+W*C1L1
      RSHF=RA-ASHF
      SHF=RSHF
      IF(ASHF.GT.RSHF) SHF=ASHF
      SHS=(SHF/(SIX*STA))*(STB*(SB-2.*SD1)*(SA/2.-STB/2.)+STA*(SA/2.-STB
     1)**2)
      WOA=0.88*WO
      CS=H/5.
C
C     TESTING OF THE VALUES OF VARIABLE UNDER THE GIVEN CONSTRAINTS.
C
      PHI(1)=12.5-S1MAX
      PHI(2)=12.5-S2MAX
      PHI(3)=12.5-S3MAX
      PHI(4)=5.0-SHS
      PHI(5)=SB-WP
      PHI(6)=WOA-SB
      PHI(7)=(H/3.-CS)-SA/2.
      PHI(8)=STB-SD
      PHI(9)=STA-STB/3.
      PHI(10)=2.0-DEL1
      RETURN
      END
```

```
      SUBROUTINE ORDER(A,B,NNN,N,M,MM,K,LL)
C
      DIMENSION A(40,40),B(40),KL(20),LL(20)
C
C     ROWS WITH SLACKS CHECKED
C
      NK=0
      KN=M-NNN
      DO 21 I=1,KN
      IF(B(I).GT.(-1.0E-06)) GO TO 21
C
C     STORE NEGATIVE B ROWS
C
      NK=NK+1
      KL(NK)=I
   21 CONTINUE
C
C     IF ALL B ARE POSITIVE CHECK THE ROWS WITH ARTIFICIAL VARIABLES
C
      IF(NK.EQ.0) GO TO 25
      ML=KN-NK+1
      ND=0
      DO 22 I=ML,KN
      ND=ND+1
C
C     CHECK IF INTERCHANGE OF ROWS IS NECESSARY
C
      DO 23 J=1,NK
   23 IF(KL(J).EQ.I) GO TO 24
C
C     INTERCHANGE ROWS AND ALTER THE SIGNS OF A AND B
C
      IE=KL(ND)
      DO 26 JJ=1,MM
      TEMP=A(I,JJ)
      A(I,JJ)=-A(IE,JJ)
   26 A(IE,JJ)=TEMP
      TEMP=B(I)
      B(I)=-B(IE)
      B(IE)=TEMP
      GO TO 27
C
C     INTERCHANGE OF ROWS NOT NECESSARY, CHANGE SIGNS OF A AND B.
C
  24  DO 29 JK=1,MM
   29 A(I,JK)=-A(I,JK)
      B(I)=-B(I)
C
C     SHIFT THE NOW NEGATIVE SLACK VARIABLES OF BASIS
C
   27 MD=MM+ND
```

```
      A(I,MD)=-1.0
   22 CONTINUE
      IF(NNN.EQ.0) GO TO 40
C
C     CHECK THE ROWS WITH ARTIFICIAL VARIABLES.
C
   25 IF(NNN.EQ.0) GO TO 32
      MP=KN+1
      DO 31 I=MP,M
      IF(B(I).GE.(-1.0E-06)) GO TO 32
C
C     B IS NEGATIVE, ALTER THE SIGNS OF B AND ALL A EXCEPT THE ARTIFICIA
C
      DO 33 J=1,MM
   33 A(I,J)=-A(I,J)
      B(I)=-B(I)
      GO TO 31
   32 IF(B(I).LT.0.0) B(I)=0.0
   31 CONTINUE
   40 CONTINUE
C
C     CHECK IF CHANGE IN BASIS IS REQUIRED.
C
      IF(NK.EQ.0) GO TO 50
C
C     CHANGE THE BASIS
C
      DO 35 I=1,M
      MMI=MM+I
      A(I,MMI)=0.0
      DO 36 J=1,NK
      NJ=N+J
   36 A(I,NJ)=0.0
      MIL=MMI+NK
      LL(I)=MIL
      A(I,MIL)=1.0
   35 CONTINUE
      N=N+NK
      NNN=NNN+NK
      MM=MM+NK
   50 CONTINUE
      RETURN
      END
```

```
      SUBROUTINE SIMPLE(A,B,C,NN,N,M,MM,INDEX,X,NMAX,II,S)
      DIMENSION S(40)
      DIMENSION A(40,40),B(40),C(40),II(50),X(40)
C
C     PHASE 1 OR 2 OF LINEAR PROGRAMING STANDARD SIMPLEX
C
      NCYCLE=1
C
C     INDEX=0 FOR PHASE 2   INDEX=1 FOR PHASE 1
C
      IF(INDEX.NE.1) GO TO 8
C
C     CALCULATION OF ALL C(J) FOR VARIABLES NOT IN BASIS.
C
      MM=N-M
      MMM=M+1-NN
      DO 1 J=1,MM
    1 C(J)=0.
      IF(NN.EQ.0) GO TO 3
      DO 5 J=1,MM
      DO 5 I=MMM,M
   5  C(J)=C(J)-A(I,J)
    3 CONTINUE
C
C     SET C(J)=1.E10 FOR VARIABLES IN BASIS.
C
      MA=MM+1
      DO 4 J=MA,N
   4  C(J)=1.E10
C
C     CALCULATE INITIAL U0
C
      U0=0.
      IF(NN.EQ.0) GO TO 7
      DO 6 I=MMM,M
   6  U0=U0+B(I)
    7  CONTINUE
      GO TO 9
    8 MB=M+1
      DO 12 J=1,N
   12 C(J)=S(J)
      U0=0.0
C
C     SELECT SMALL C(J) WHICH IS C(L)
C
    9 SMALL=C(1)
      L=1
      DO 10 I=2,N
      IF(C(I).GE.SMALL) GO TO 10
      SMALL=C(I)
```

```
      L=I
   10 CONTINUE
C
C     TESTING FOR OPTIMUM NOTE ALLOWANCE FOR ROUND OFF ERROR
C
      IF(C(L)+1.E-5.GE.0.) GO TO 100
C
C     TESTING FOR FINITE OPTIMUM  ALLOWANCE FOR ROUND OFF ERROR.
C
      DO 15 I=1,M
      IF(A(I,L).GT.1.E-5) GO TO 16
   15 CONTINUE
      WRITE(6,210)
      GO TO 101
C
C     SELECT SMALLEST RATIO FOR WHICH A(I,L) GT.0. GIVING EQN. (LL)
C     IN WHICH VARIABLE IS DROPPED.
C
  16  SMALL=1.0E+10
      LL=1
      DO 18 I=1,M
      IF(A(I,L).LE.1.E-5) GO TO 18
      IF(B(I)/A(I,L).GT.SMALL) GC TO 18
      SMALL=B(I)/A(I,L)
      LL=I
  18  CONTINUE
C
C     BRINGING C(K) BACK TO 0 BEFORE CONVERTING TO NEW CANNONICAL FORM
C
      K=II(LL)
      C(K)=0.
C
C     CONVERTING TO NEW CANNONICAL FORM.
C
      B(LL)=B(LL)/A(LL,L)
      U0=U0+B(LL)*C(L)
      DO 30 J=1,N
      IF(J.EQ.L) GO TO 30
      A(LL,J)=A(LL,J)/A(LL,L)
      C(J)=C(J)-A(LL,J)*C(L)
   30 CONTINUE
      A(LL,L)=1.
      DO 33 I=1,M
      IF(I.EQ.LL) GO TO 33
      Y=A(I,L)
      B(I)=B(I)-B(LL)*A(I,L)
      DO 31 J=1,N
  31  A(I,J)=A(I,J)-A(LL,J)*Y
   33 CONTINUE
C
C     SWITCH BASIS TAGS ON LL EQN.
```

```
      C(L)=1.E10
      KK=II(LL)
       II(LL)=L
C
C     SETTING OLD VARIABLE IN BASIS =0
C
      X(KK)=0.
C
C     RECORD NEW VALUES OF X IN MEMORY. VARIABLES NOT IN BASIS ARE
C     ALREADY 0 IN THE MEMORY.
C
      DO 40 I=1,M
      K=II(I)
   40 X(K)=B(I)
C
C     ITERATION COMMAND.
C
      NCYCLE=NCYCLE+1
      IF(NCYCLE.EQ.NMAX) GO TO 110
      GO TO 9
C
C     OUTPUT.
C
  100 CONTINUE
      IF(INDEX.NE.1) GO TO 101
C
C     CALCULATION OF CANNONICAL FORM OF OPT. EQN. FOR INITIAL FEASIBLE
C     BASIS.
C
  102 N=N-NN
      MC=M+1
      DO 94 J=MC,N
   94 S(J)=0.0
      DO 95 J=1,N
   95 C(J)=S(J)
      U0=0.
      DO 90 I=1,M
      K=II(I)
      Q=C(K)
      U0=U0+B(I)*Q
      DO 90 J=1,N
   90 C(J)=C(J)-A(I,J)*Q
      INDEX=0
      DO 91 I=1,M
      K=II(I)
   91 C(K)=1.E10
      GO TO 9
  101 RETURN
  110 WRITE(6,211) NCYCLE
  111 STOP
```

```
200  FORMAT(2X,4HUU= ,E11.5)
 201 FORMAT(2X,8HA MATRIX,/,(1X,10F11.5))
 202 FORMAT(2X,22HVARIABLES IN BASIS ARE,/,(2X,30I3))
 206 FORMAT(2X,28HPHASE II OF SIMPLEX SOLUTION,//)
 208 FORMAT(2X,8HC MATRIX,/,(2X,8E13.5))
 210 FORMAT(2X,17HNO FINITE OPTIMUM)
 211 FORMAT(2X,30HPROCESS DID NOT CONVERGE AFTER,2X,E12.5,2X,6HCYCLES)
     END
```

```
     INPUT IN THE SUBROUTINE APPROX.

     K, NUM, NEQ, NMAX, INDEXI  IN FORMAT  I3

004000010099001

     BB(I),  I=1,8  IN FORMAT  8F10.5

0.5       0.5       0.1       0.1       0.1       0.1       0.1       0.

     STEPX(I),  I=1,4  IN FORMAT 4F10.5

0.00001   0.00001   0.00001  0.00001
```

RESULTS OF THE OPTIMIZATION OF COKE PUSHER RAM BY DIRECT SEARCH AND SUCCESSIVE LINEAR APPROXIMATION TECHNIQUE

DIRECT SEARCH METHOD FAILED TO PRODUCE BETTER RESULT THAN SUCCESSIVE LINEAR APPROXIMATION.

THE RESULT FROM DIRECT SEARCH METHOD IS

THE CROSS-SECTIONAL AREA OF THE PUSHER RAM IN SQ. INCH = 36.2984

DIMENSIONS OF THE RAM SECTION.

HEIGHT IN INCHES	= 29.9375
WIDTH IN INCHES	= 12.6250
FLANGE THICKNESS IN INCH	= 0.8750
WEB THICKNESS IN INCH	= 0.3125

RESULTS FROM THE SUCCESSIVE LINEAR APPROXIMATION TECHNIQUE WHICH IS THE OPTIMUM

THE OPTIMUM CROSS-SECTIONAL AREA OF THE

PUSHER RAM IN SQ. INCH = 34.9953

OPTIMUM DIMENSIONS OF THE RAM SECTION.

HEIGHT IN INCHES	= 30.1875
WIDTH IN INCHES	= 12.5000
FLANGE THICKNESS IN INCH	= 0.8125
WEB THICKNESS IN INCH	= 0.3125

THE FOLLOWING CARD SHOULD BE ADDED OR EXCHANGED AS MARKED TO COMBINE THE SUBROUTINES SEEK AND APPROX.

THE FOLLOWING CARD SHOULD BE EXCHANGED AS MARKED IN THE COMPOSITE PROGRAMME.

```
*CALL APPROX(XS,YZ,UR)
```

THE FOLLOWING CARDS SHOULD BE EXCHANGED AS MARKED IN SUBROUTINE APPROX

```
**SUBROUTINE APPROX(XS,Z,UR)
 ⊕DO 3 I=1,4
3 ⊕Z(I)=XS(I)
```

THE FOLOWING CARD SHOULD BE ADDED IN THE APPROPRIATE PLACE IN SUBROUTINE APPROX AS MARKED.

```
†DIMENSION XS(10)
```

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