THE HUMPHREY INTERNAL COMBUSTION PUMP.

I, Samuel Falkof, hereby certify that the thesis entitled "The Humphrey Internal Combustion Pump," which I have presented for the Degree of Master of Science in Engineering of the University of the Witwatersrand, is my own work and has not been submitted as a thesis for the Master's Degree of any other University.

Signed S. Falkof

University of the Witwatersrand
30th. December, 1935.

THE HUMPHREY INTERNAL COMBUSTION PUMP.

THESIS SUBMITTED BY

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FOR THE DEGREE OF

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SECTION 1.

GENERAL DESCRIPTION OF MODE OF ACTION.

In order to appreciate the advantages offered by the Humphrey Internal Combustion Pump over other types of pumps, it will be essential to begin with a theoretical discussion on internal combustion cycles in general.

The requirements of a perfect cycle are as follows:-

1. The suction stroke, during which a fresh charge of combustible mixture is introduced into the working cylinder.

2. The compression stroke, the length of which is dependent upon the compression pressure required.

3. The working or expansion stroke. This should be of sufficient length to allow of the gases expanding down to atmospheric pressure in order to fully utilise the work of expansion. In order to achieve this object, the length of this stroke must be considerably greater than that of the suction stroke. In the Otto cycle where these 2 strokes are of equal length, exhaust occurs at a pressure considerably greater than atmospheric and the work represented by the toe of the indicator diagram is lost. (Fig. 1).

4. The exhaust stroke. This is another long stroke comparable with the expansion strok , but its exact length depends on the clearance required.

Thus the perfect gas engine must be capable of meeting the above conditions, i.e. 4 strokes of unequal length.

L et us now consider the actual construction of a gas engine as it exists to-day. We have a solid piston with its water-cooling arrangements and piston rings; the connecting rod, cranks and crankshaft; the flywheel with its bearings and lubricating devices. In addition in the 4-stroke engine, we have the 2 to 1 gearing operating the cam-shaft, which actuates the valves in their correct sequence.

How/.........

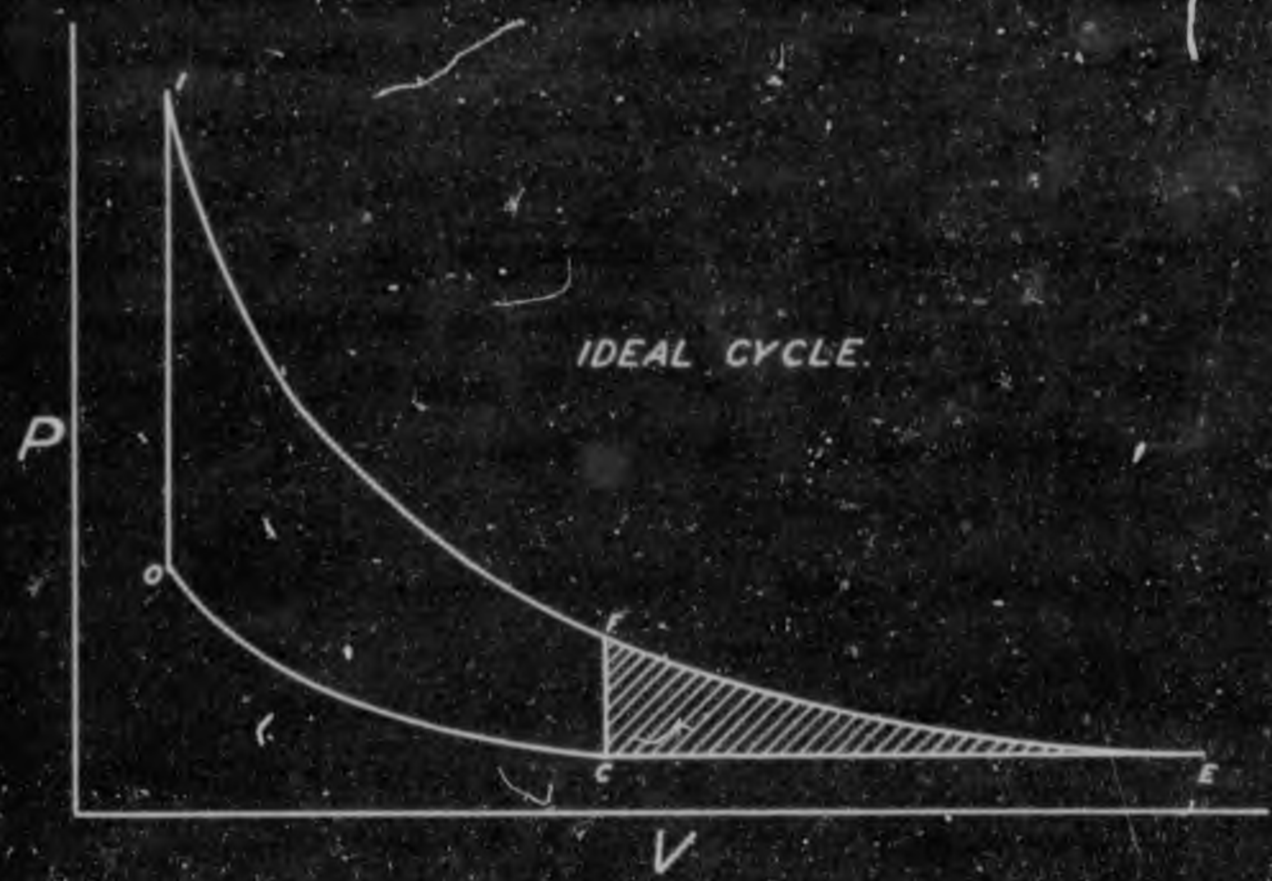


FIG. 1

Granted/........

How much of this is really necessary? The actual essentials required by a gas engine are:-

1. A working cylinder.
2. An inlet valve for mixture.
3. An exhaust vlave for burnt products.
4. An ignition device.
5. A flywheel.

Let us for a moment consider a gas-engine containing only the aforementioned essentials. We should then arrive at a gas-engine in which the energy was delivered direct to the flywheel without any intermediate gearing. In order to achieve this, the piston itself would have to be part and parcel of the flywheel and also fulfil its function of drawing in a fresh charge and compressing it.

We shall now enumerate the properties and component parts of our ideal gas engine in which everything is discarded which first principles do not demand as essential.

1. It must be capable of giving 4 unequal strokes per cycle.

2. It must utilise the whole range of expansion down to atmospheric.

3. The energy must be delivered direct to the flywheel without any intermediate gearing.

4. There must be internal cooling arrangements.

5. The suction and exhaust valves must be operated by pressure conditions in the cylinder.

6. There must be an automatic ignition device which shall only allow ignition to occur when maximum compression pressure has been attained.

7. When operating under reduced load, all 4 strokes must be capable of being shortened up, as it is wasteful to draw in a full charge of fuel under such conditions.

The problem that now confronts is: "To what extent can such an ideal arrangement be realised in practice?".

Granted/........

Granted that water is suitable for internal cooling, then surely the idea of using a water piston asserts itself. If this water piston is made heavy enough, it may well serve as a flywheel but the inertia effect would be reciprocating instead of rotary. Considering the oscillations of this flywheel, it is seen that on the working stroke a certain quantity of water is raised to a higher level or pressure. Our ideal gas engine becomes a pump and our gas power appears as water power. In addition the reciprocating motion of the water column will serve to draw in fresh mixture, compress it, and after the working stroke, expel the burnt products.

This ideal gas engine has been realised in practice by the Humphrey Internal Combustion Pump whose mode of action closely conforms with the ideal set of conditions enumerated.

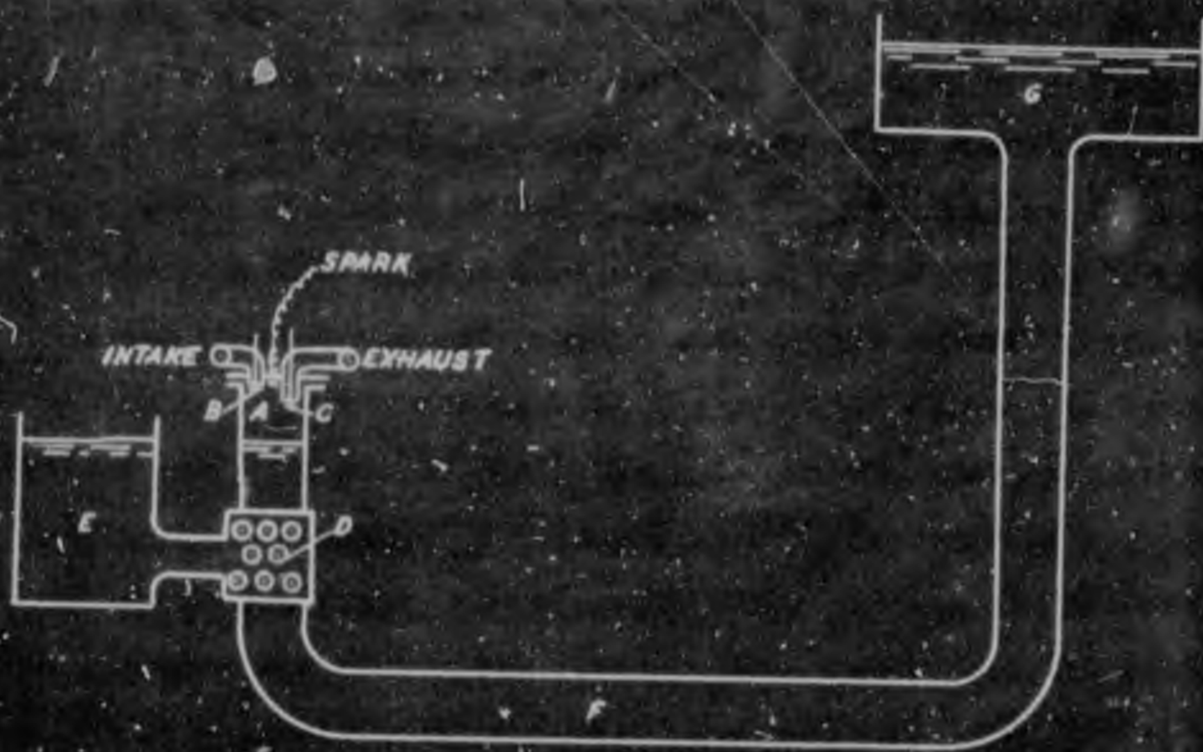
This remarkable piece of apparatus is the invention of Mr. Herbert A. Humphrey and bears his name. It was the outcome of years of research and experimental work on Internal Combustion Engine problems.

A description of the component parts and an explanation of the mode of action of a 4-stroke Humphrey Pump will now be given (Fig. 2). The diagram shows the arrangement in mere outline and is accompanied by an indicator diagram, showing the conditions of pressure existing in the pump throughout a complete cycle.

The pump consists essentially of a combustion chamber A, fitted with an inlet valve B for admission of combustible mixture and an exhaust valve C for burnt products; a water valve-box D for admission of water; a supply tank E and a play-pipe F leading to the reserviour G.

The action of the pump is as follows:- Consider a charge of mixture compressed in the combustion chamber. Its state point is represented by "C" on the P.V. diagram. The mixture is fired by a sparking plug and the gases rise in

pressure/......



4-STROKE HUMPHREY PUMP.

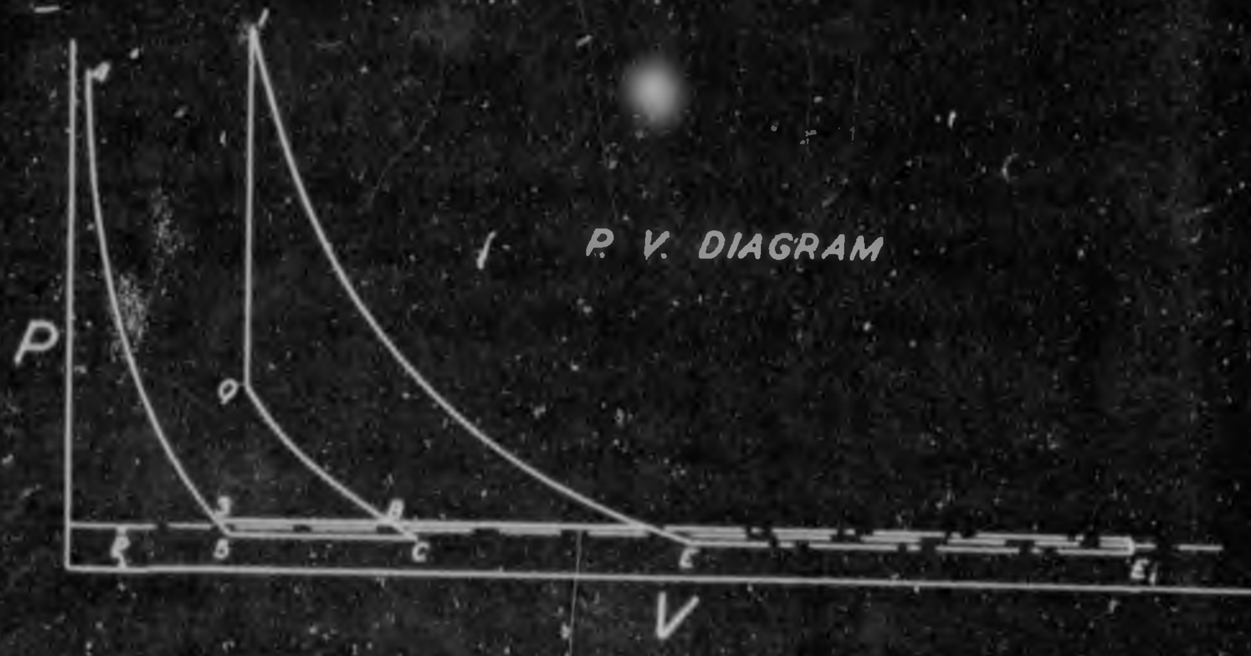


FIG. 2

pressure from "O" to "1". The gases then expand driving along the water column and discharging water into the reservoir. When the pressure has fallen to atmospheric as represented by "E", the exhaust and water valves open. Atmospheric conditions now exist in the combustion chamber. Water flows into the chamber, some of it following up the old water column, and as the motion of the original water column slows down, also filling the combustion chamber and driving out some of the exhaust products. At the point E, the momentum of the original column has been spent and it comes to rest, the water in the combustion chamber having in the meanwhile risen to a level represented by "B". On the return swing, the pressure tends to rise and the water suction valves shut. The rising column drives out the residual exhaust products until a level represented by point "3" is reached, which corresponds to the level of the exhaust valve, and a certain amount of gas is trapped in the combustion head. The cushioned gas is compressed by the momentum of the column until at a level represented by point "4" the compression pressure is sufficient to bring the column to rest. The gases now expand until a pressure slightly below atmospheric is reached. (Represented by point "5"). The admission valve now opens and a fresh charge of mixture is now drawn in. At a level represented by point "C" the column has again come to rest, and on the return swing, the rise in pressure causes the intake valve to shut. The mixture is then compressed to a pressure represented by "C" and a fresh cycle commenced.

Scavenging:

The pump is fitted with a simple scavenging device, so that practically pure air is left in the cushion space, enabling a richer mixture to be drawn in. The exhaust is fitted with 2 branches. One short branch is provided with a lig[illegible] eturn valve opening inwards for admission of scavenging air. The other branch is the exhaust pipe proper and is

provided/........

provided with a light non-return valve opening outwards for expulsion of exhaust products. The mode of action is as follows:- On the exhaust and water valves opening simultaneously the velocity of the water column is such that the level of the water in the combustion chamber falls at first, reducing the pressure slightly below atmospheric. The non-return valve in the short branch of the exhaust opens, admitting scavenging air, while the non-return valve in the exhaust pipe proper closes, preventing any outflow of air or burnt products of combustion. After a while, the water level in the chamber rises, increasing the pressure. This increase of pressure closes the scavenging-air valve and opens the valve in the exhaust pipe proper, thus allowing of expulsion of scavenged exhaust products. The residual gases are nearly all pure air.

Interlocking Gear:

The exhaust and intake valves are provided with an interlocking device which ensures that at any instant only one valve can open. One such device which does not impede easy movement of the valves will now be described.

The diagram (Fig. 3) shows a section through the gear employed by Humphrey during the Official Trials of his invention. The exhaust valve is situated below the level of the inlet valve. The valve stems each carry top and bottom circular nuts a, b, c and d. The latter serve to limit the opening of the valves by coming in contact with the rubber buffers, and the top nuts serve to lock the valves in their closed position when the bolt e slides between the under-side of the nut and the top of the brackets. The remaining gear will be sufficiently understood if its operation is described. Starting with all valves shut, ignition occurs followed by expansion. The locking bolts are at this time in such a position that the admission-valve is locked and the exhaust

valve/........

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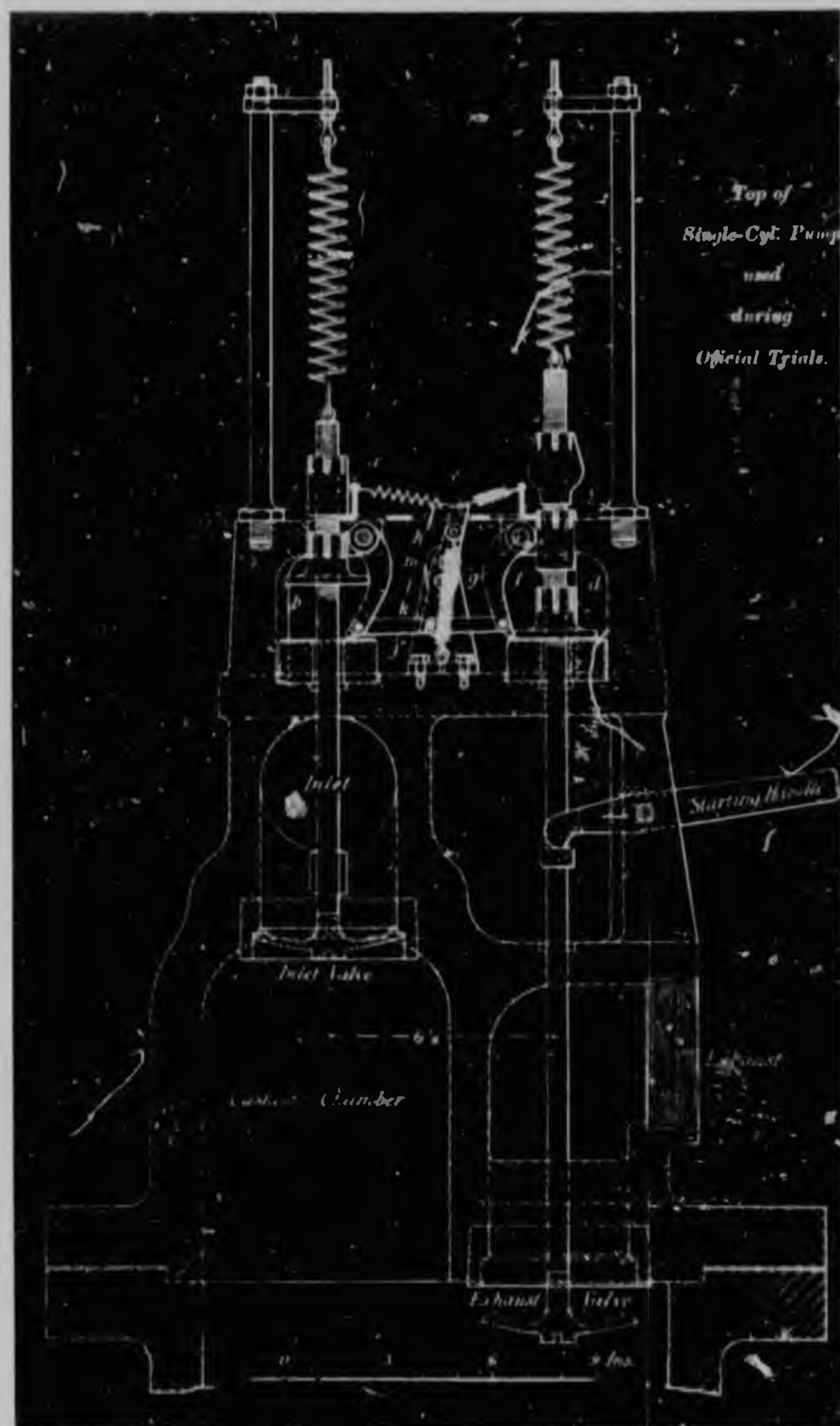


Top of
Single-Cyl. Pump
used
during
Official Trials.
Inlet
Starting Handle
Inlet Valve
Chamber
Exhaust
Exhaust Valve

FIG 3

free to open. The expansion is rapid, and as the point of atmospheric pressure is passed, suction causes the exhaust valve to open. On the return swing of the water column the exhaust-valve is shut by impact and the cushion and the cushion expansion stroke, followed by the suction stroke, results. It will, however, be noticed that when the exhaust valve shuts, the enlarged portion of its lower nut, bearing against the curved lever, f, which is hinged at g, forces this lever to the left. As another curved lever, h, is hinged to the first lever by means of the link j, this other lever has also to partake of the motion and move to the left, thus setting it in position to be acted upon by the lower nut on the inlet-valve. Link j carries a pin, k, which engages against a lever, l, hinged at m in such a manner that when link j moves to the left the top of lever l moves to the right. Two springs are shown attached to leverl, that on the left being stretched, and that on the right being closed, and as these two springs act upon the bolt e, in the position shown, the effect is to urge the bolt to the right. This bolt cannot, however, move until the top nut, c, has risen to its full height, which occurs when the exhaust valve shuts. The movement of the bolt then takes place and locks the exhaust shut, at the same time releasing the admission valve. During the admission stroke the opening and shutting of the admission-valve reverses the action of the gear, locks the admission and unlocks the exhaust ready for the next cycle.

Water Valves:

Instead of using a single water-suction valve which must needs be rather large and heavy in order to cope with the large volume of water drawn in, a number of smaller valves incorporated in a valve box is usually employed. The valve box is usually a continuation of the combustion chamber and

is/.......

is of approximately the same section. One arrangement is to dr. a number of holes in the valve box and screw in gun-metal-valves held on their seats by small springs. Another arrangement is to have a large number of flap valves in the valve box as in the case of the pumps at Chingford.

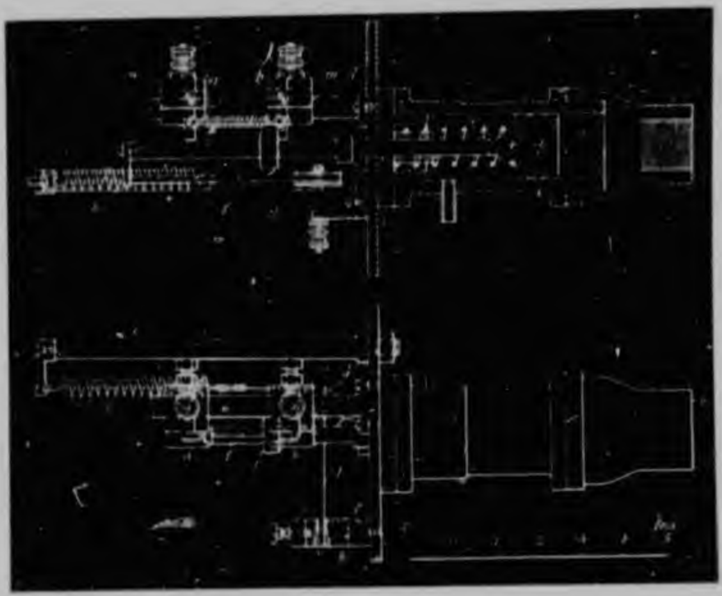
Ignition:

A description will now follow of an ignition device invented by Mr. Humphrey. It works on the high tension system with the use of an induction coil. (Fig. 4).

The low tension circuit supplying the coil is capable of being broken at three points by the switches a, b and c. When all three switches are closed simultaneously, sparking occurs. The piston "d" is acted upon by the water in the chamber giving compression of spring "e". The piston rod has attached to it one end of a band "f" which passes over a pulley "g" carrying a switch arm "j" which has its movement limited by two stops, "o" and "p". When passing between these stops, it rides over contact "k" and closes switch "c". On the piston rod is a collar "l" which at pressures below the desired range of compression moves switch lever "m" and breaks the circuit at "b", and at pressures above the range of compression, moves switch lever "n" and breaks the circuit at "a". Switch levers "m" and "n" are connected by a spring so that when collar "l" is between "m" and "n", switches "a" and "b" are both closed.

The action of the apparatus is as follows:- At atmospheric pressure switch arm "j" is against the stop "p", switc h "b" is open and "a" is closed. On further increase of pressure, collar "l" rides clear of switch lever "m", thus also closing switch "b". Before switch "b" closes, however, switch arm "j" moves over to stop "o", the band "f" slipping round pulley "g" as the pressure increases. When the compression pressure has reached its maximum and begins to decline slightly, this slight reduction in pressure causes the band "f" to reverse its former direction of motion and switch arm "j" moves over

from/.....



IGNITION GEAR.

FIG 4

from stop "o" to stop "p" riding over contact "k", thus closing switch "c". All three switches "a", "b" and "c" being now closed, ignition occurs. The sudden increase in pressure due to the explosion causes the switch arm "j" to move over to stop "o". The collar "l" also bears against switch lever "n" and opens switch "a". Expansion now occurs and the switch arm "j" passes over from stop "o" to stop "p" as the pressure diminishes. Switch "a" then closes and on further reduction of pressure, switch "b" is opened, so that we arrive at atmospheric pressure with everything in readiness for a fresh cycle. The cushion pressure also gives rise to a maximum, but being above the range of compression pressures, switch "a" is open and the passage of the switch arm "j" over the contact "k" does not produce a spark. This type of apparatus entirely safeguards against pre-ignition, since sparking can only occur just after the compression pressure has reached its maximum.

Supply Tank:

The level of the water in the combustion chamber corresponding to V_b (Fig. 2) tends to approach that of the water in the supply tank. It will be shown later that V_b is very nearly equal to V_c the charge volume. Thus by varying this level we are able to vary V_b and thus V_c, thereby varying the amount of energy developed per cycle. A float controlling the inlet to the supply tank is used to regulate the input and thus the power of the pump.

The Cushion Space:

The volume of the top of the combustion chamber, above the level of the exhaust-valve forms a clearance space comparable with the clearance space of an ordinary Otto cycle, and the scavenged products of combustion which remain in this space are mixed off with the fresh combustible charge and serve to dilute it. This fact limits the amount of clearance space in relation to a given charge volume and if this proportion is kept at about 20% as in the gas engine, it might happen that if

the/......

the pump is using larger charges of mixture per cycle than designed for, then the cushion pressure may well rise in excess of the maximum pressure for which the combustion head has been designed. To obviate this limitation on the flexibility of operation of the pump, the clearance space is made much larger than the customary 20% in gas engines and owing to the fact that the exhaust gases are well scavenged, a proportionally richer mixture drawn in.

Compression Ratio:

It will be shown later that the compression ratio of the Humphrey Pump Cycle depends only on the head pumped against. In order to pump against high heads, very large compression ratios are required. Although large compression ratios are conducive to high thermal efficiency, they lead to trouble as far as pre-ignition is concerned. The Humphrey Pump, working on the constant volume combustion cycle, is thus only capable of working against low and medium lifts. If solid fuel injection were employed, however, with combustion at constant pressure, much higher compression ratios could be utilised and the pump enabled to deliver against high heads as well.

Play-Pipe:

The length and cross-sectional area of the play-pipe play an important part in fixing the performance of the pump as will be indicated later. The cross-sectional area of the play-pipe is usually made the same as the combustion chamber. Sometimes the vertical portion or stand-pipe is made tapering, the area increasing with height, as in the pumps at Chingford. This is done to reduce the violent fluctuations of the column which would otherwise occur at the discharge end of the play-pipe.

Fuel:

Gaseous fuel is usually employed in Humphrey Pumps. The installation usually includes gas producers, supplying the necessary fuel. With the use of a suitable carburettor, petrol

can/......

can be efficiently used. As mentioned previously, a pump employing solid fuel has potentialities as far as high heads are concerned.

It was found that at the Official Trials of the invention, the contamination of the water due to coming in contact with the gaseous fuel and the exhaust products was absolutely negligible.

With reference to sparking plugs, Mr. Humphrey employed a plug which was not affected by immersion in water for a short space of time during each cycle. Of course, an ordinary plug could be used, situated at the top of the combustion head, the high cushion pressure being an effective safeguard against splashing.

SECTION II.

HUMPHREY PUMP INSTALLATIONS.

The Chingford Plant.

When Mr. Humphrey delivered his original paper before the Institution of Engineers in the Autumn of 1909, the Metropolitan Water Board had in progress at Chingford, the construction of a large reservoir for the purpose of impounding the flood waters of the Lea. These floods came down with great rapidity and a very powerful pumping plant was necessary to cope with the large quantity of water. The lift being low and the quantity great, the use of ordinary centrifugal pumps was barred by consideration of capital cost, which would be in no way compensated for by a low operating cost, since low lift reciprocating pumps have by no means a high efficiency. Centrifugal pumps also have a low efficiency when working under low heads.

Struck by the remarkable results recorded in Mr. Humphrey's paper, Mr. W.B. Bryan, the Chief Engineer of the Metropolitan Water Board, decided to invite a tender from the Pump and Power Co., Ltd., the firm manufacturing Humphrey Pumps, in competition with tenders from centrifugal pump manufacturers. The conditions required the set of pumps to have an aggregate capacity of 180 X 10^6 gallons per diem against a head of 30 feet.

The tenders, when opened, showed that the plant submitted by Mr. Humphrey would cost £19,000 less than the next lowest tender and that there was, moreover, an enormous difference in the guaranteed fuel costs. Mr. Humphrey guaranteed a figure not exceeding 1.1 lb. of anthracite per pump horse-power hour, a penalty of £1,000 being attached to every $\frac{1}{10}$ lb. consumption in excess of this figure. This fuel figure was from 2 to 3 times below that of the next tender. Another clause in the contract imposed a penalty of £20,000 should the plant prove a failure.

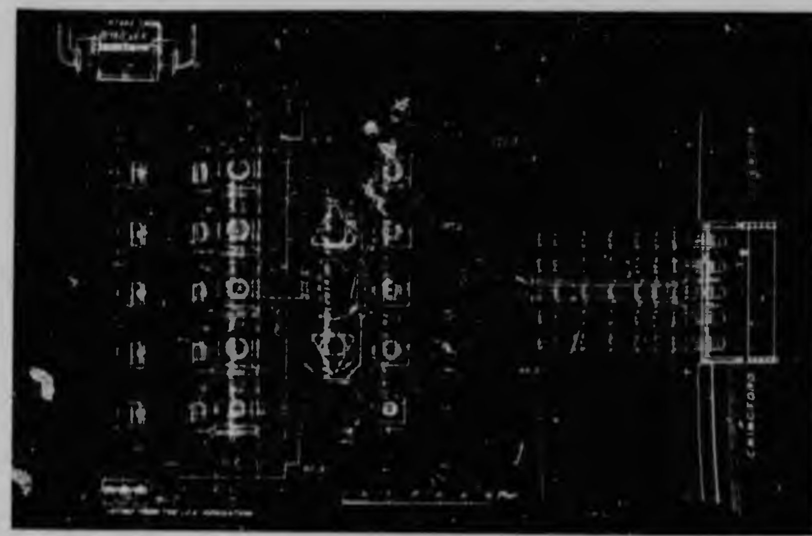
It was a somewhat bold step on Mr. Humphrey's part

to/.......

to accept a contract under such heavy penalties for pumps each developing between 200 and 300 H.P. on the basis of an experimental pump delivering only 35 H.P., but the results have thoroughly justified his confidence in the capabilities of his invention. As a matter of fact, the pumps substantially exceeded their guaranteed output when put into operation.

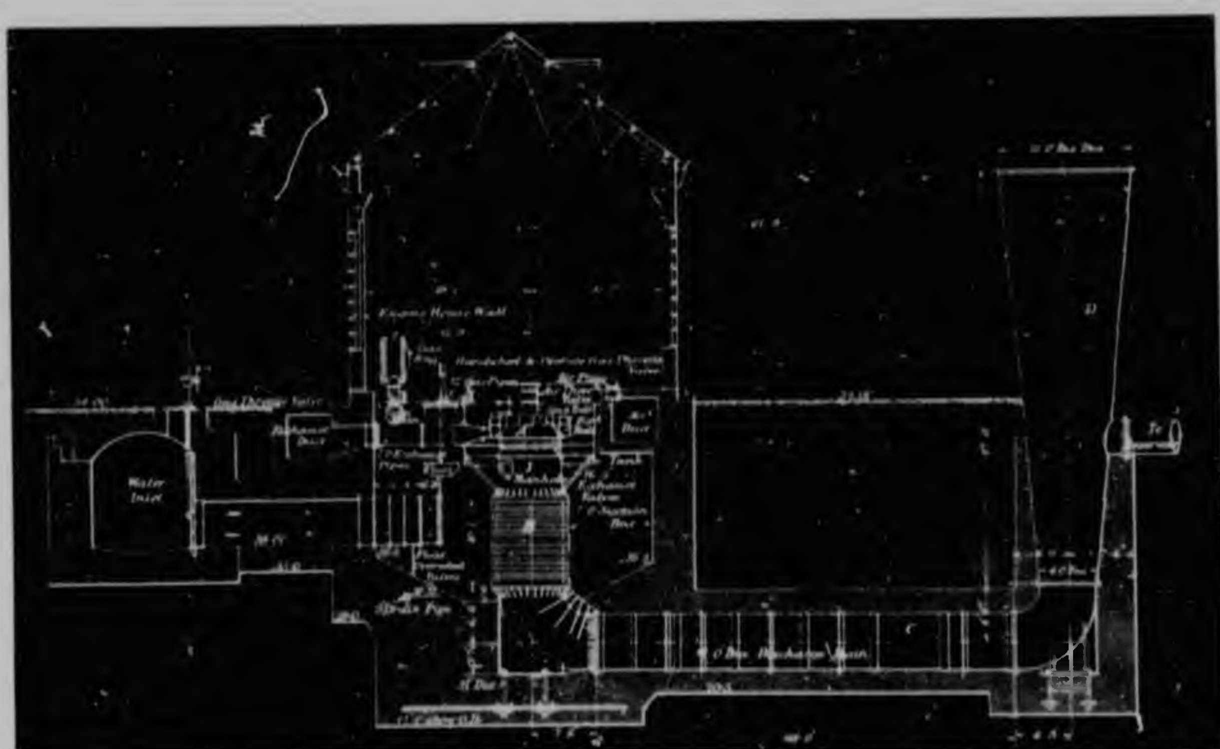
The plant consists of four 40×10^6 gallon per day units and one of 20×10^6 gallons capacity. The diagram (Fig. 5) shows a plan of the layout. The larger units have pump cylinders 7 feet in diameter while the smaller unit has a bore of 5 feet. Fig. 6 shows a section through one of the 40×10^6 gallon units. Each pump consists of a cast steel combustion head A, a water valve box B from which the play-pipe C leads to the base of a tapering water-tower D, built up of rivetted plates. From each of these towers 4 feet diameter pipes lead over the top of the reservoir embankment and discharge into the uppermost of a series of water cushions extending to the bottom of the reservoir. The supply of water is led along the back of the pump house through a conduit formed in the concrete as shown in the figure. From this, a cast iron pipe, embedded in concrete is led into each of the five pump pits. Sluice valves are provided,by means of which each pit may be isolated when so desired. A working platform is fixed round the pump head, giving access to the valves and other operating gear. The air-duct for the pump is arranged below the floor level on the right of the building, and the exhaust duct is on the left. The play-pipe C is 6 feet in diameter and is embedded in concrete. The bend which connects the bottom of the water vlave box to the play-pipe is a steel casting and serves to anchor the pump chamber. securely to the foundations against the uplift caused by the fact that the full pressure of the explosion is not all used in the mass acceleration of the water column. Owing to the

gases/......



CHINGFORD INSTALLATION.

FIG. 5



SECTION THROUGH 40×10^{6} GALLON UNIT.

FIG. 6

gases being exhausted at atmosphere and the ample precautions taken in design, the pump works practically without noise, and the smooth running, absence of shock and freedom from vibration are truly remarkable.

The mode of operation of the pumps is the same as that of the simple 4-stroke pump described previously. Separate scavenging valves interlinked with the exhaust valves are employed. In place of single large valves for admitting exhausting and scavenging, a number of smaller valves are provided. The interlocking gear between the admission and exhaust valves consists of an oscillating ring operated by a relay connected up to the water in the pump chamber. Each pair of valves is provided with an interlocking device, each of which is connected by a rod to a lug on the oscillating ring.

The water-valve box consists of 2 cast-iron castings bolted together. A large number of simple flap valves are used. Both valves and seats are of manganese bronze and are bolted to the valve box casting. These valves operate most efficiently and are remarkably tight, the water being capable of being held up for a full 24 hours.

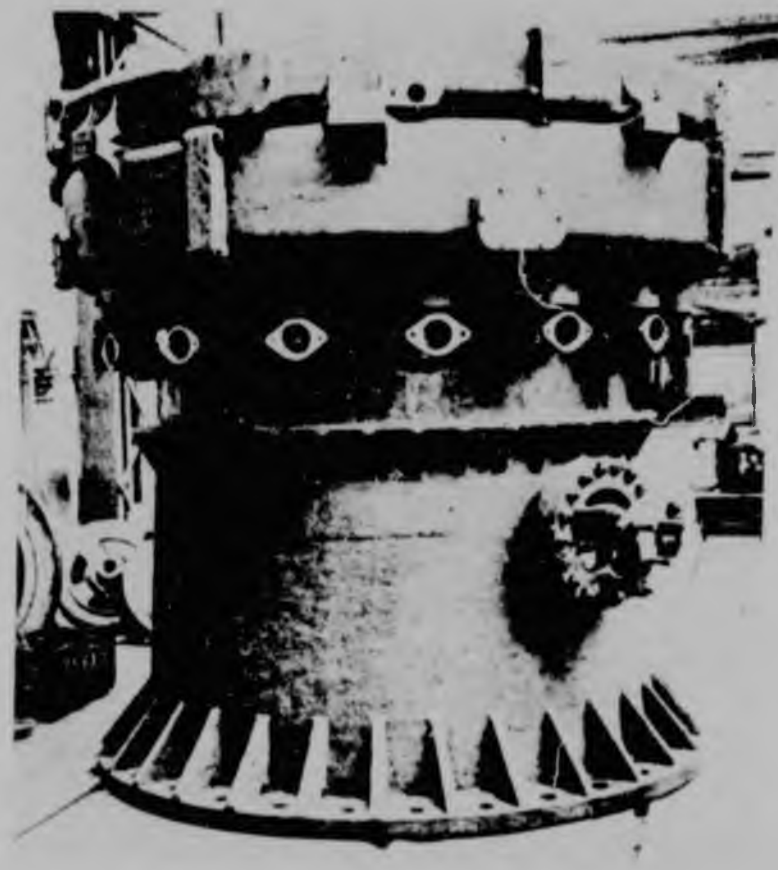
Fig. 7 is a photograph of the combustion head for one of the 40 X 10^6 gallon units, fitted with gas-belt and exhaust belt.

Fig. 8 shows a half section and plan of the valve arrangements for one of the larger pumps. The interlocking arrangement may be clearly understood from the diagram.

Fig. 9 shows the water-valve box for the 20 X 10^6 gallon pump. Some of the flap valves may be seen bolted in place.

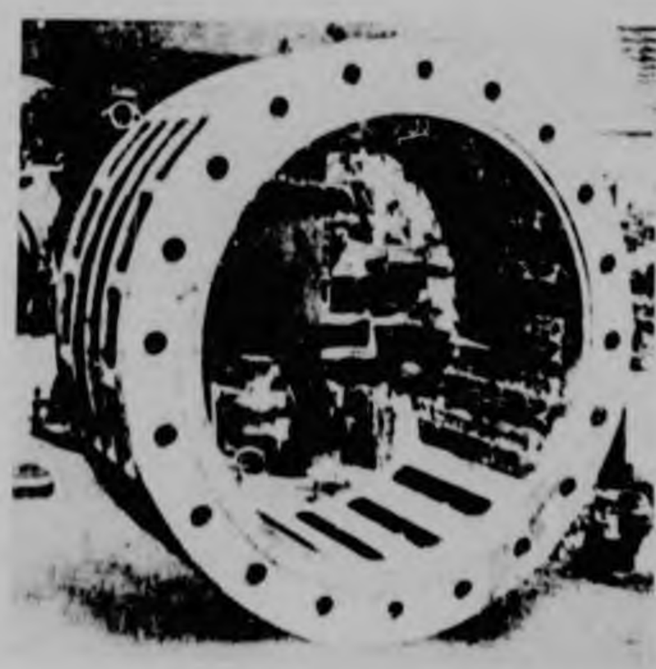
Fig. 10 is a photograph of the interior of the pump house. The five pumps are clearly seen in their pits. The pumps are remarkably silent in operation. All one hears is the click-click of the valves.

Fig./.......



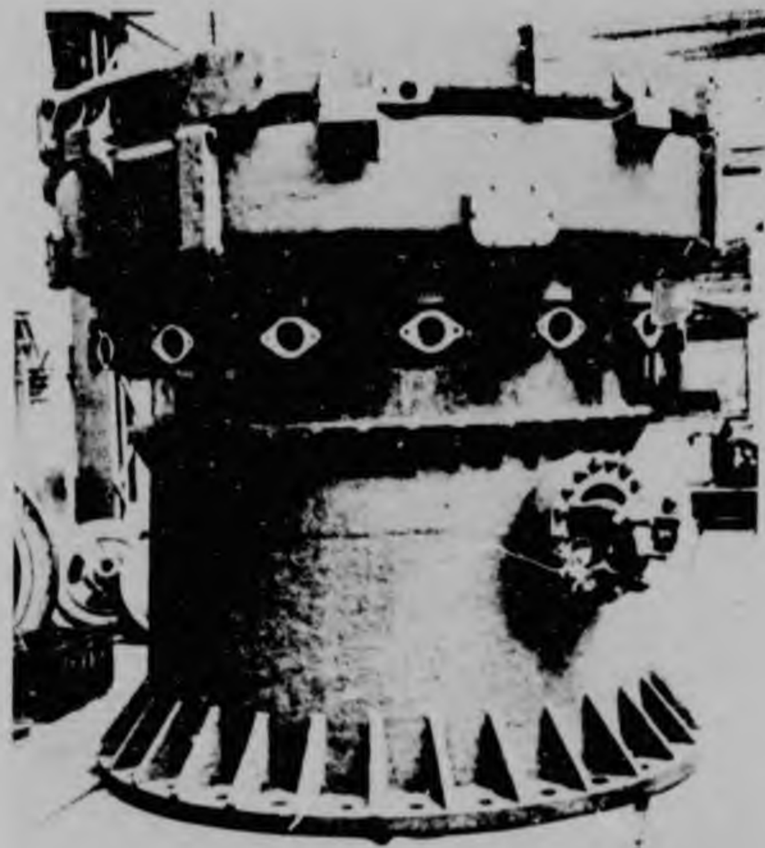
COMBUSTION HEAD FOR 40×10^6 GALLON PUMP.

FIG. 7



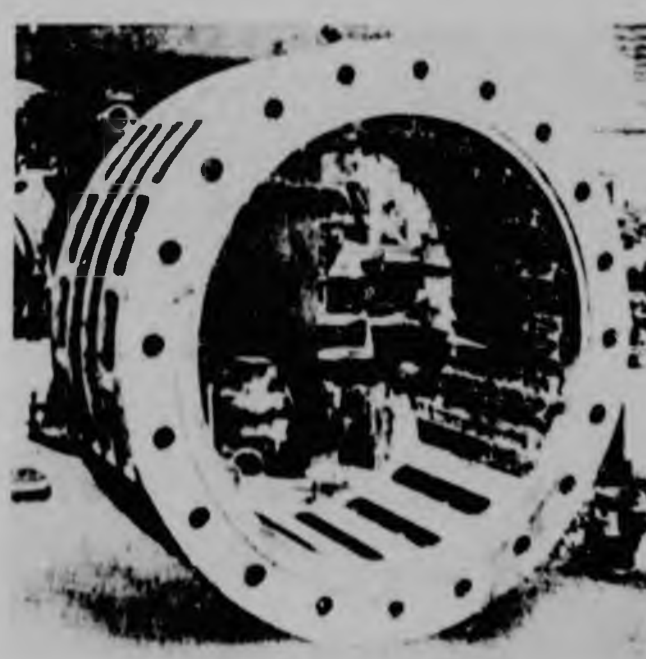
WATER VALVE BOX FOR 20×10^6 GALLON PUMP.

FIG. 9



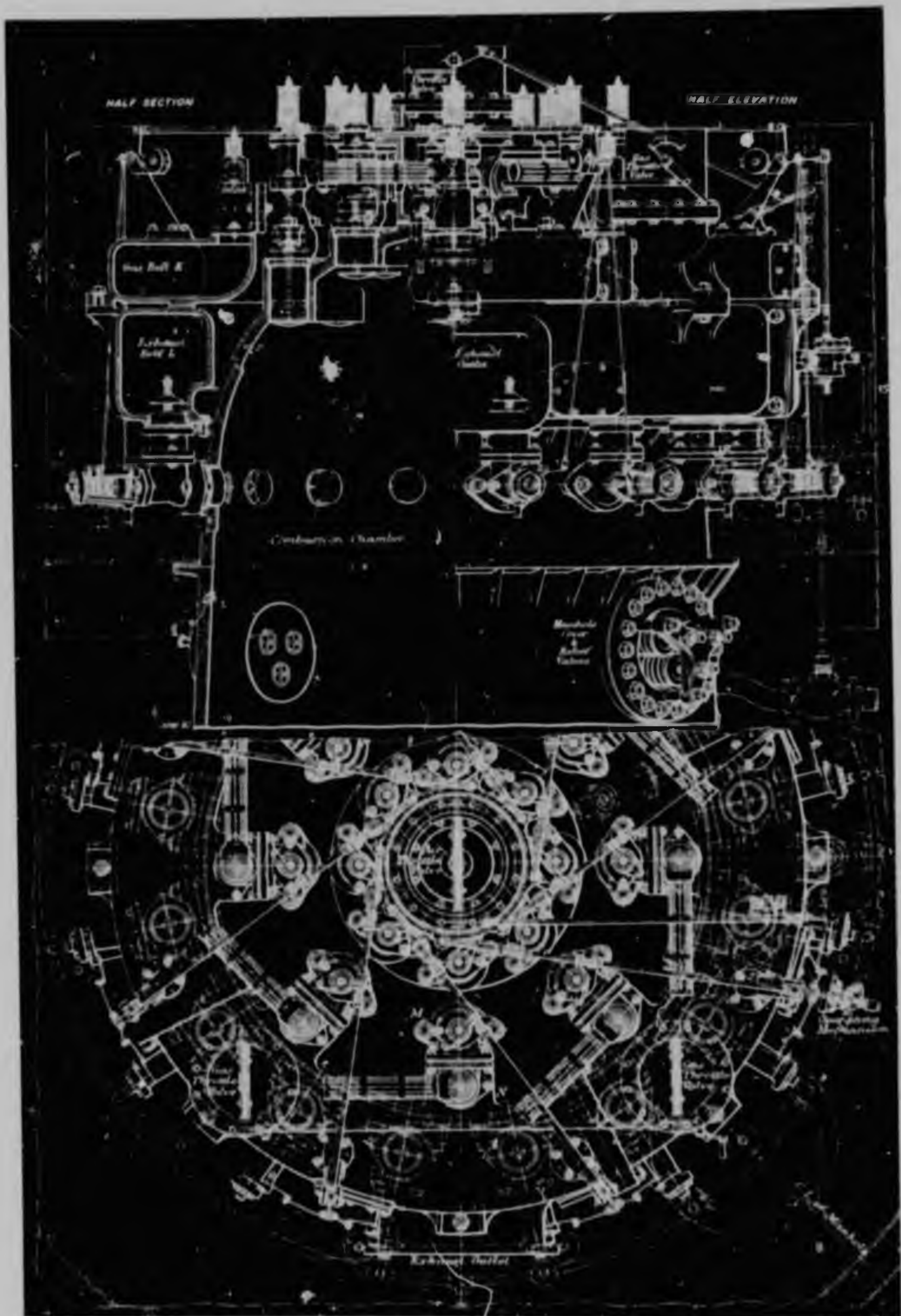
COMBUSTION HEAD FOR 40×10^6 GALLON PUMP.

FIG. 7



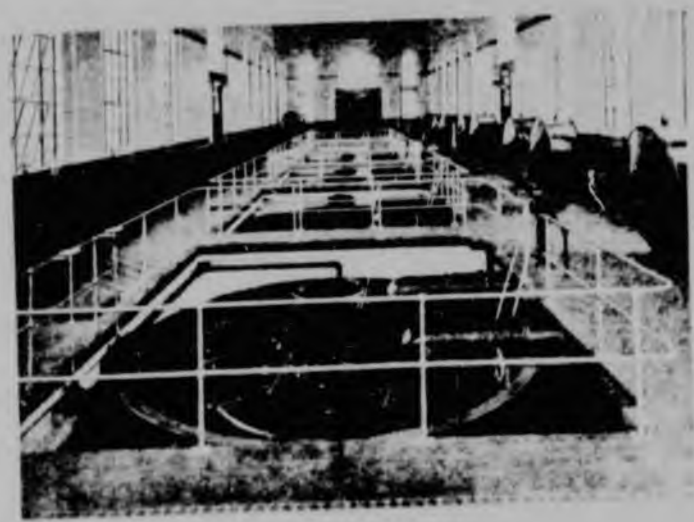
WATER VALVE BOX FOR 20×10^6 GALLON PUMP.

FIG. 9



VALVE ARRANGEMENTS

FIG 8



INTERIOR OF PUMP HOUSE.

FIG. 10

DISCHARGE FROM TWO PUMPS.

FIG. 11

Fig. 11 is a photograph showing the reservoir embankment. Two pumps are working. The water cushions may be clearly seen.

The fuel required for operating the pumps is supplied by a Dowson Gas Plant. This consists of 4 producers, three rated to convert 370 lbs. of anthracite per hour into gas and the other having a rating of 138 lbs. A cooling pipe leads from the top of each producer to coke scrubbers. After passing the coke scrubbers, the gas is sent through a sawdust scrubber and is then led through an underground conduit to a gas-holder, 24 feet in diameter and 12 feet high. The steam required is supplied by 2 vertical boilers. A governor gear is fitted by which the rise of the gas-holder automatically regulates the steam supply to the producers, thus keeping the gas production and consumption in balance. Figs. 12 and 13 show various views of the arrangement.

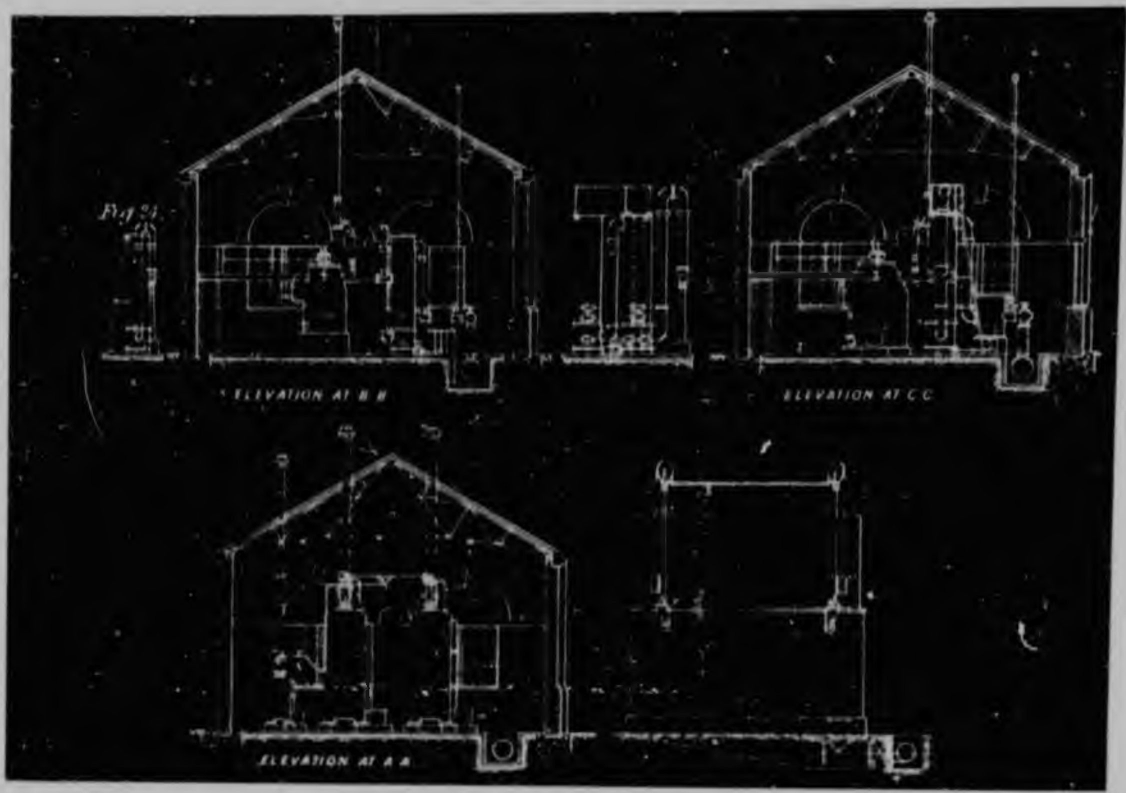
To start the pumps, a charge of gas and air is delivered from a compressor to the combustion chamber and forces down the level of the water until the required charge volume is reached. A hand-switch is then operated causing sparks to pass and the first outstroke is made. From this moment the action of the pump is entirely automatic and it picks up its load and soon attains to its normal output.

The following is a summary of results obtained at the official tests of the plant:-

No. of Pump	Gals./24 hrs.	Thermal efficiency.	lbs. of coal per P.H.P.hr.
1	48×10^6	22.39	0.946
2	47×10^6	22.19	0.957
3	47.6×10^6	22.33	0.949
4	47×10^6	24.07	0.881
5	26×10^6	25.63	0.796

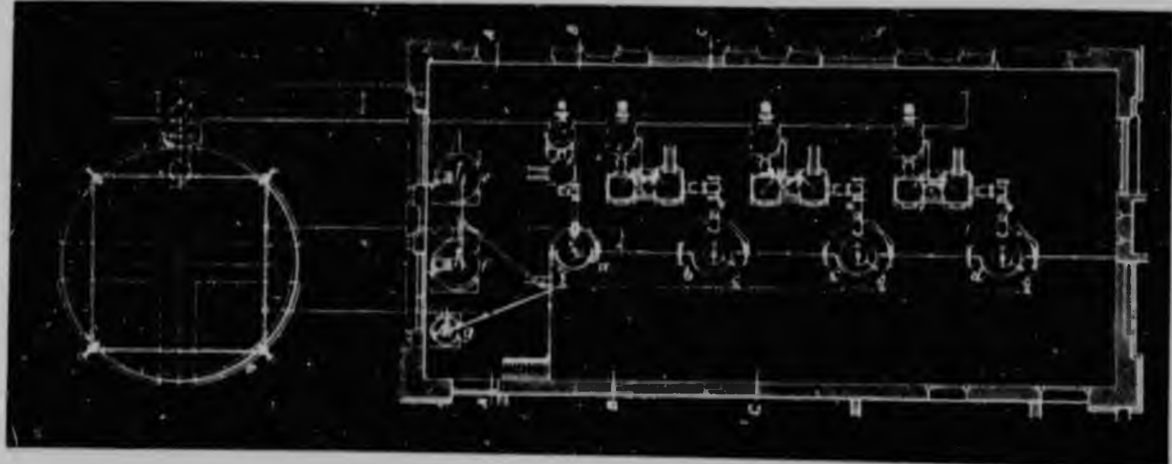
The figures show that the guaranteed output of the

pumps/.....



DOWSON GAS PLANT.

FIG. 12



DOWSON GAS PLANT.

FIG. 13

pumps is substantially exceeded, and the fuel consumption well below the guarantee figure of 1.1 lbs. per P.H.P. hr. The figures for thermal efficiency may not at first sight appear outstanding, but considering the fact that a compression ratio of 3 was employed, and that the theoretical thermal efficiency increases with increase of compression ratio, the figures are much better than for any internal combustion engine using the same fuel and compression ratio.

Although put into operation over 20 years ago, the Chingford plant is still working and, we are told, even better than when first installed owing, in large measure, to the efforts and skill of the engineer, Mr. Wood, and his assistants who have given close study to the pumps under their care. The Chingford plant inspires all who visit it with awe, and is a mighty tribute to the genius of that great engineer and inventor, Mr. Humphrey.

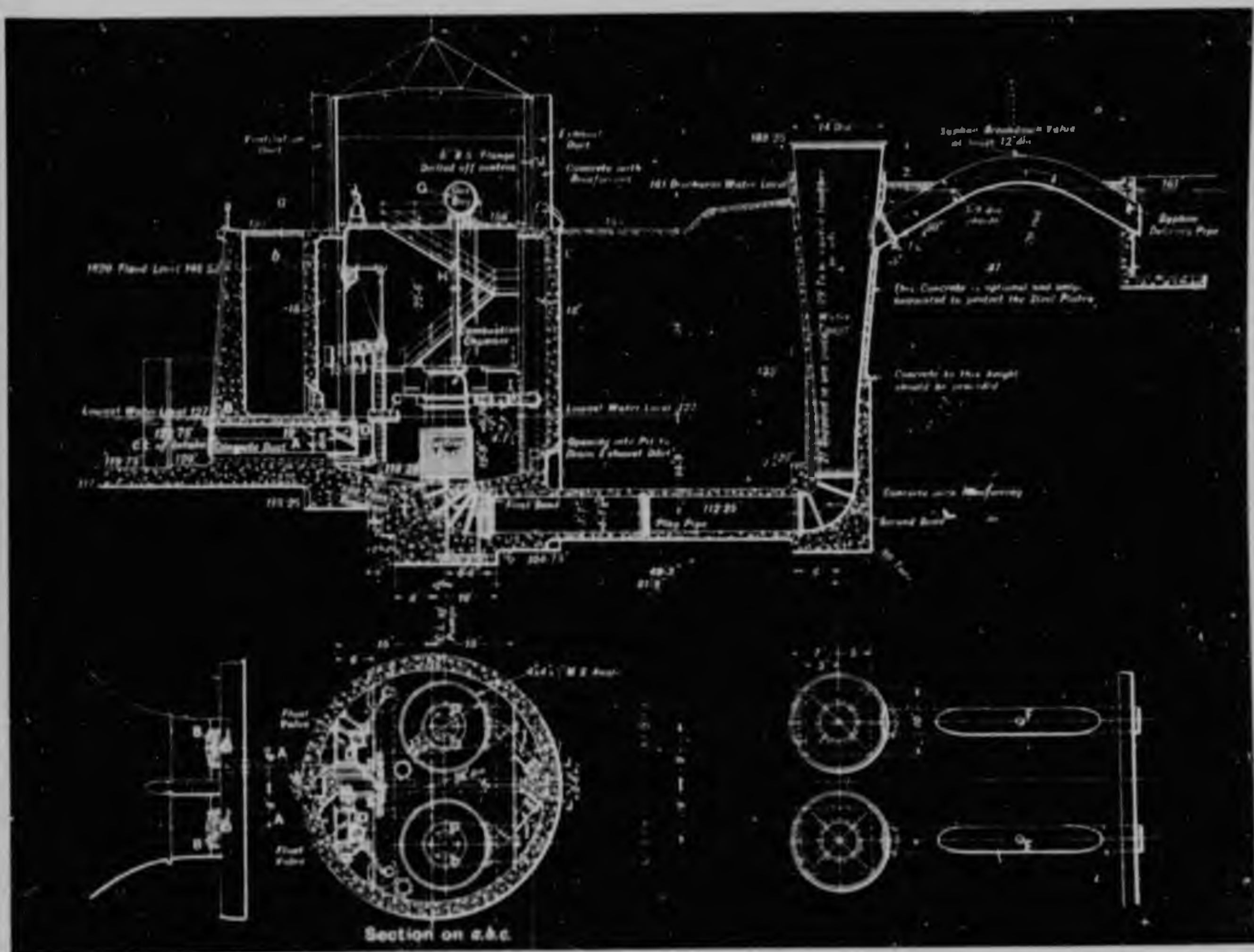
Installation at Cordobla, South Australia.

In 1924, William Beardmore and Co., Ltd., supplied a large Humphrey Pump plant to the South Australian Government. The installation consists of 2 identical pumps with a combined output of 2,875,000 gallons per hour. They are somewhat smaller than the large Chingford units.

The pumps are used for irrigation purposes at Cardobla, which is situated about 180 miles up the River Murray. Owing to the fact that the river level rises and falls a matter of 20 feet according to the season, the combustion chambers are placed in a circular concrete caisson, 36 feet internal diameter, in the bottom of which are formed circular sumps.

Fig. 14 shows the lay-out of the installation. The water is led through the concrete ducts A from the river to the pump sumps, penstock valves being provided at B, to isolate the sumps from the river. The water-level in the sumps is automatically controlled by floats, shown at C, which operate the

butterfly/......



INSTALLATION AT CORDOBLA.

FIG 14

butterfly valves D, and a dash-pot is incorporated in each float governed valve to prevent sudden movements which might give rise to severe water shock in the concrete ducts. The water is finally delivered from the water towers through syphons to the irrigation canal E. By opening the air valves at F, shown on top of the syphons, air is permitted to enter, thus destroying the syphon action and completely isolating the pumps from the delivery canal without the necessity of employing sluice valves.

The pumps themselves are of the 4-stroke type and the proportions are kept very much the same as those at Chingford. A few departures in detail are made, however. Since no excessive pressures have ever been recorded with the Chingford pumps, cast-iron is employed in the combustion head. Owing to the low temperatures which prevail and the fact that the chamber is flushed with water, all the valves in the combustion chamber are provided with rubber facings. The valves are thus rendered extremely tight and all shock is eliminated. Instead of having to regrind the valves and seats, all that is necessary is to renew the rubber faces. Fig. 15 is a photograph of a combustion chamber bolted on a water-suction box. The enormous size of these pumps may be gauged by the size of the man.

The Chingford plant and the plant at Cardobla are the only 2 large Humphrey Pump installations in existence. It is rather surprising that Mr. Humphrey's invention has not found more application than it has, especially after having so convincingly proved its worth in practice. Nevertheless, the author is quite convinced that in the course of time, Mr. Humphrey's remarkable contribution to the progress of Mechanical Engineering will be universally recognised and appreciated.

COMBUSTION CHAMBER ON WATER SUCTION BOX

FIG. 15

SECTION III.

INTERNAL COMBUSTION ENGINE DESIGN CHART (G).

From the literature on the Humphrey Pump it would seem that in the past each pump was designed from first principles. The labour involved in the determination of the periodic rate of action of the pump is very heavy, and the computation of the various quantities required in the design of the pump is a matter of great difficulty. The author has developed an analysis, employing similarity principles whereby the main quantities required in design may be very easily deduced from charts. The charts may be applied to all Humphrey pumps designed to work within compression ratios ranging from 2 to 6 and employing fuels whose mixture strengths vary in calorific value from 40×10^4 foot lbs. per lb. to 80×10^4 foot lbs. per lb. The analysis employs true variable specific heats and in order to determine the various thermodynamic quantities required, the chart which will now be described, was devised.

In the following analysis a linear relationship between Specific Heat and Temperature will be assumed.

$$K_p = A + ST$$

$$K_v = B + ST$$

where K_p and K_v are the specific heats of a gas at constant pressure and constant volume respectively, the units being foot lbs./lb./°C; T is the temperature expressed in °C (abs.); and A, B and S are constants of any particular gas.

$$\text{put} \quad \frac{A}{B} = m$$

$$\text{and} \quad \frac{S}{B} = \lambda$$

Then for adiabatic (isentropic) conditions of the gas the following relationship between temperature and volume

holds/.......

holds:-

$$T\,V^{m-1}\,e^{\lambda T} = \text{constant.}$$

The proof may be found in any modern book on Heat Engines.

If suffix "i" refers to initial conditions of the gas then:-

$$T_i\,V_i^{m-1}e^{\lambda T_i} = T\,V^{m-1}e^{\lambda T}$$

and putting $R_V = \dfrac{V}{V_i}$

$R_T = \dfrac{T}{T_i}$

the above expression becomes

$$-(m-1)\log_e R_V = \log_e R_T + \lambda T_i (R_T - 1) \quad \ldots\ldots\ldots(1)$$

Further putting $R_p = \dfrac{P}{P_i}$,

it follows from the gas law, $PV = RT$

that $R_p = \dfrac{R_T}{R_V}$

Adding $(m-1)\log_e R_T$ to both sides of equation (1) and putting $\dfrac{R_T}{R_V} = R_p$, we obtain:-

$$(m-1)\log_e R_p = m\log_e R_T + \lambda T_i (R_T - 1) \quad \ldots\ldots(2)$$

Let us first consider the case of adiabatic compression. In this case R_T is > 1 and $1 > R_V > 0$; $R_p > 1$. R_V^{-1} is clearly the compression ratio r.

From (1)

$$(m-1)\log_e R_V^{-1} = \log_e R_T + \lambda T_i (R_T - 1) \quad \ldots\ldots(3)$$

and from (2)

$$(m-1)\log_e R_p = m\log_e R_T + \lambda T_i (R_T - 1) \quad \ldots\ldots\ldots(4)$$

Considering the right hand side of (3) we may easily plot $\log_e R_T$ to a base of R_T . The expression $\lambda T_i(R_T - 1)$ on being plotted to a base of R_T will yield a family of straight lines passing through the point (1,0), the slopes all being positi. and proportional to the parameter T_i .

By/.........

By adding the ordinates of these 2 curves for any value of the abscissa R_T and the parameter T_1 we can obtain the value of the right-hand side of equation (1). We shall call this quantity Y_1 .

i.e. $Y_1 = \log_e R_T + \lambda T_1 (R_T - 1)$ ....(5).

Similarly we may plot $m \log_e R_T$ to a base of R_T. The term $\lambda T_1 (R_T - 1)$ in (4) is identical with the term $\lambda T_1 (R_T - 1)$ in (3).

Hence by adding the ordinates of the curves of $m \log_e R_T$ and $\lambda T_1 (R_T - 1)$ we may obtain the value of the right-hand of (4) for any value of R_T and T_1 . We shall call this quantity Y_2 .

i.e. $Y_2 = m \log_e R_T + \lambda T_1 (R_T - 1)$ ....(6).

Let us now consider the case of adiabatic expansion. Here:-

$1 > R_T > 0$; $1 > R_p > 0$; and $R_v > 1$

From (1)

$(m-1) \log_e R_v = -\log_e R_T + \lambda T_1 (1 - R_T)$ ......(7),

and from (2)

$(m-1) \log_e R_p^{-1} = -m \log_e R_T + \lambda T_1 (1 - R_T)$.........(8).

As before we may easily plot curves of $-\log_e R_T$ and $-m \log_e R_T$ to a base of R_T . The expression $\lambda T_1 (1 - R_T)$ which occurs on the right-hand side of (5) and (6) on being plotted to a base of R_T will yield a family of straight lines passing through the point (1,0) and having negative slopes proportional to the parameter T_1 . Thus by adding the ordinates of the $-\log_e R_T$ and $\lambda T_1 (1 - R_T)$ curves in the case of (5) and by adding the ordinates of the $-m \log_e R_T$ and $\lambda T_1 (1 - R_T)$ curves in the case of (6) we can obtain the values of the right-hand sides of equations (5) and (6) for any value of R_T and the parameter T_1 . We shall put

$Y_3 = -\log_e R_T + \lambda T_1 (1 - R_T)$................(9).

and $Y_4 = -m \log_e R_T + \lambda T_1 (1 - R_T)$..................(10).

It/........

It is also clear from (3), (4), (5), (6), (7), (8), (9) and (10) that :-

$$Y_1 = (m-1) \log_e R_v^{-1}$$

$$Y_2 = (m-1) \log_e R_p$$

$$Y_3 = (m-1) \log_e R_v$$

$$Y_4 = (m-1) \log_e R_p^{-1} .$$

Thus if we plot a curve of

$$Y = (m-1) \log_e x$$

on a base of x where the scale of Y is the same as the scale of the $\log_e R_T$ v. R_T etc. curves, then the abscissae of this curve for values of $Y = Y_1$, Y_2 , Y_3 and Y_4 respectively will yield the corresponding values of R_v^{-1}, R_p , R_v and R_p^{-1} corresponding to any values given to R_T and T_1 in equations (3), (4), (7) and (8).

In chart G the following constants were adopted on the recommendation of Professor Walker, as good average figures for a gas engine mixture.

$K_p = 320 + 0.067\ T$ ft. lbs./lb./°C.

$K_v = 224 + 0.067\ T$ ft. lbs./lb./°C.

$$m = \frac{A}{B} = 1.43$$

$$\lambda = \frac{S}{B} = 0.0003$$

Chart G is divided into 2 parts; the right-hand part being devoted to compression and the left to expansion. The $\log_e R_T$ and $m \log_e R_T$ v. R_T curves for compression are drawn on the right for a range of R_T from 1 to 6. The scale of ordinates is not marked but is: 1 unit is represented by [illegible].5 decimetres.

In order to plot the family $\lambda T_1 (R_T - 1)$ v. R_T , $\lambda T_1 (R_T - 1)$ was taken as one unit. Then for $T_1 = 1000$°C (abs.), which is the highest initial compression ~~pressure~~ temperature considered

R_T/..........

$$R_T = 1 + \frac{1}{\lambda T_1} = 1 + \frac{1}{0.0003 \times 1000}$$

$$= 4.333.$$

At $R_T = 4.333$ a vertical line (T_c) is erected 1 unit (2.5 decimetres) in length and divided up into a scale of T_c, intial compression temperatures, as shown. A new origin O_c (1, 1) is chosen. Then for any value of T_c, if O_c is joined to that value of T_c on the vertical line T_c, the joining line is clearly a mirror image of the line $\lambda T_c (R_T - 1)$ v. R_T. Hence if we draw a parallel to this line through any point on the $\log_e R_T$ or $m \log_e R_T$ v. R_T curves for any value of R_T, then the intercept of this line on the vertical axis will give the value of $\log_e R_T + \lambda T_c (R_T - 1)$ (Y_1) is $m \log_e R_T + \lambda T_c (R_T - 1)$ (Y_2) for the particular value of T_c and R_T considered. This clearly follows from the fact that the slope of the line $\lambda T_c (R_T - 1)$ v. R_T is proportional to the initial compression temperature T_c.

The left-hand side of the chart is devoted to expansion. The curves of $-\log_e R_T$ and $-m \log_e R_T$ are plotted to a base of R_T for values of R_T ranging from 1 to 0.1. The scale of ordinates which is again not marked is also equal to: 1 unit is represented by 2.5 decimetres.

In order to plot the family $\lambda T_1 (1 - R_T)$ v. R_T. $\lambda T_1 (1 - R_T)$ was taken as equal to 0.6 units. Then for $T_1 = 3000^{\circ}C$ (abs.) which is the highest initial expansion temperature considered $R_T = 1 - \frac{0.6}{\lambda T_1} = 1 - \frac{0.6}{0.0003 \times 3000}$

$$= 0.333$$

At $R_T = 0.333$ a vertical line (T_1) is erected and marked off into a scale of T_1, initial expansion temperatures, as shown. A new origin O_1, (1, 0.6) is chosen on the vertical axis. Then for any value of T_1, if O_1 is joined to that value on the T_1 line, the joining line is the mirror image

of/........

of the line $\lambda T_1(1 - R_T)$ v. R_T. Hence if we draw a parallel to this line through any point on the $- \log_e R_T$ and $-m \log_e R_T$ curves corresponding to any value of R_T then the intercept on the vertical axis will give the values of $- \log_e R_T + \lambda T_1 (1 - R_T)$ (Y_3) and $- m \log_e R_T + \lambda T_1 (1 - R_T)$ (Y_4) for the values of T_1 and R_T considered. As in the case of compression this clearly follows from the fact that the slope of the $\lambda T_1 (1 - R_T)$ v. R_T lines are proportional to T_1.

In order to obtain the values of R_v, R_v^{-1}, R_p and R_p^{-1} once Y_1, Y_2, Y_3 and Y_4 have been ascertained, the whole horizontal axis is divided up into a scale of x. Curves of $Y = (m - 1) \log_e x$ are drawn on this base. The scale of ordinates which is not marked is the same for all the curves (three in number) and is the same as the $\log_e R_T$ v. R_T etc. curves, i.e. 1 unit of Y is represented by 2.5 decimetres. Curve x(1) has a range of x from 0.1 to 20. The horizontal scale of x(2) is $2\frac{1}{2}$ times as great and that of x(3) is 10 times as great, giving a total range of x from 1.0 to 200.

Nomographic scales T_c^1 and T_1^1 are provided so as to give the value of temperature T corresponding to any initial temperature and value of R_T. The vertical axis is first divided up into a scale of absolute temperatures. Considering compression, we choose any value of R_T, say 2.5, for convenience. Then for an initial temperature of 1000°C (abs) the corresponding final temperature for $R_T = 2.5$ is 2500°C abs. We plot this point, and mark it 1000°C (abs.) on the T_c^1 scale. The ordinate is then divided up into a scale of initial temperatures as shown. If we join up any value of T_c on the centre and T_c^1 scales then it is clear that the ordinate under this line, corresponding to any value of R_T, read off on the centre scale will give the temperature corresponding to the values of T_c and R_T considered. This clearly follows from the fact that there is obviously a linear law between the final temperature and R_T for any value of T_c.

/In.....

In the case of expansion we choose for convenience a value of $T_1 = 3000^oC$ (abs.) and $R_T = 0.5$ giving a final temperature of 1500^oC (abs.) We plot this value and letter it 3000^oC on the $T_1{}^1$ scale. The ordinate is then divided up into a scale of initial temperatures as shown. As before, if we join up any value of T_1 on the centre and $T_1{}^1$ scales, then the ordinate read off on the centre scale will give the value of the temperature corresponding to the value of T_1 and R_T chosen.

The horizontal scale on the top of the diagram is divided up into a scale of E and I. E and I are then plotted against T^oC (abs.) (centre scale).

E, the internal energy per lb. of gas, and I, the total energy, are deduced for various values of T^oC (abs.) from the relations:-

$$E = \int_0^T K_v dT = BT + \tfrac{1}{2} ST^2$$

$$\text{and} \quad I = \int_0^T K_p dT = AT + \tfrac{1}{2} ST^2$$

The construction of the chart is now complete. The ranges employed are ample for the majority of problems likely to be met with. Similar charts employing different constants may be rapidly constructed on the lines previously indicated.

Previously, it was indicated how for given values of T_c and R_T in the case of compression, and of T_1 and R_T in the case of expansion the corresponding values of $R_v{}^{-1}$, R_p and R_v, $R_p{}^{-1}$ respectively may be deduced. Alternatively we may wish to find values of R_T corresponding to various values of R_v and various initial temperatures in the cases of compression and expansion. This may be done by reversing the process mentioned previously. Having obtained R_T, R_p may be found from the relation $R_p = \frac{R_T}{R_v}$ or may be found directly from the chart as explained previously. Having obtained R_T, the corresponding temperature and internal and total energies may easily/.....

In the case of expansion we choose for convenience a value of T_1 = 3000°C (abs.) and R_T = 0.5 giving a final temperature of 1500°C (abs.) We plot this value and letter it 3000°C on the $T_1{}^1$ scale. The ordinate is then divided up into a scale of initial temperatures as shown. As before, if we join up any value of T_1 on the centre and $T_1{}^1$ scales, then the ordinate read off on the centre scale will give the value of the temperature corresponding to the value of T_1 and R_T chosen.

The horizontal scale on the top of the diagram is divided up into a scale of E and I. E and I are then plotted against T°C (abs.) (centre scale).

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The construction of the chart is now complete. The ranges employed are ample for the majority of problems likely to be met with. Similar charts employing different constants may be rapidly constructed on the lines previously indicated.

Previously, it was indicated how for given values of T_c and R_T in the case of compression, and of T_1 and R_T in the case of expansion the corresponding values of $R_v{}^{-1}$, R_p and R_v, $R_p{}^{-1}$ respectively may be deduced. Alternatively we may wish to find values of R_T corresponding to various values of R_v and various initial temperatures in the cases of compression and expansion. This may be done by reversing the process mentioned previously. Having obtained R_T, R_p may be found from the relation $R_p = \frac{R_T}{R_v}$ or may be found directly from the chart as explained previously. Having obtained R_T, the corresponding temperature and internal and total energies may

easily/.....

easily be deduced. This flexibility of manipulation enables the chart to be put to a wide variety of uses. Once the intrinsic theory of the chart has been thoroughly grasped, its manipulation should present no difficulty. Also owing to the fact that the various quantities required may be determined directly without the necessity of constructing subsidiary curves or interpolation, a very high degree of accuracy is assured.

DATA SHEET FOR CHART G (GAS CHART)

$\log_e R_T$ v. R_T etc. curves.

R_T (com.)	R_T (exp.)	$\log_e R_T$ (com.) $-\log_e R_T$ (exp.)	$m \log_e R_T$ (com.) $-m \log_e R_T$ (exp.)
1.0	1	0	0
1.05	0.9524	0.0488	0.0698
1.1	0.9530	0.0953	0.1363
1.2	0.8333	0.1823	0.2606
1.3	0.7692	0.2624	0.3765
1.4	0.7143	0.3365	0.4820
1.5	0.6667	0.4055	0.5800
1.6	0.6250	0.4700	0.6725
1.8	0.5556	0.5878	0.8420
2.0	0.5000	0.6931	0.9920
2.2	0.4545	0.7885	1.1270
2.4	0.4167	0.8755	1.2520
2.6	0.3846	0.9555	1.3660
2.8	0.3571	1.0296	1.4730
3.0	0.3333	1.0986	1.5700
3.2	0.3125	1.1632	1.6650
3.4	0.2941	1.2238	1.7500
3.6	0.2778	1.2809	1.8320
3.8	0.2632	1.3350	1.9100
4.0	0.2500	1.3863	1.9810
4.5	0.2222	1.5041	2.1500
5.0	0.2000	1.6094	2.3000
5.5	0.1818	1.7047	2.4400
6.0	0.1667	1.7918	2.5610
6.5	0.1538	1.8718	2.6600
7.0	0.1429	1.9459	2.7810
7.5	0.1333	2.0149	2.8800
8.0	0.1250	2.0794	2.9750
8.5	0.1176	2.1401	3.0620
9.0	0.1111	2.1972	3.1400
10.0	0.1000	2.3026	3.3000

Y/.....

Y = (m - 1) loge x curve. (m = 1.43).

x	y	x	y
1.00	0	14.0	1.1320
1.25	0.0961	15.0	1.1670
1.50	0.1741	16.0	1.1910
1.75	0.2404	17.0	1.2170
2.00	0.2980	18.0	1.2420
2.50	0.3940	19.0	1.2680
3.0	0.4730	20.0	1.2880
3.5	0.5380	22.5	1.3390
4.0	0.5960	25.0	1.3820
4.5	0.6470	27.5	1.4260
5.0	0.6917	30.0	1.4620
6.0	0.7713	32.5	1.4960
7.0	0.8370	35.0	1.5270
8.0	0.8940	37.5	1.5580
9.0	0.9450	40.0	1.5890
10.0	0.9920	42.5	1.6110
11.0	1.0400	45.0	1.6400
12.0	1.0700	47.5	1.6600
13.0	1.1050	50.0	1.6820
60	1.7600	130	2.0970
70	1.8220	140	2.1240
80	1.8860	150	2.1590
90	1.9330	160	2.1830
100	1.9800	170	2.2090
110	2.0250	180	2.2340
120	2.0600	190	2.2600
		200	2.2800

E and I CURVES.

$$E = BT + \tfrac{1}{2}ST^2 = 224T + \tfrac{1}{2} \times 0.067T^2$$
ft. lbs./lb.

$$I = AT + \tfrac{1}{2}ST^2 = 320T + \tfrac{1}{2} \times 0.067T^2$$
ft. lbs./lb.

T°C (abs.)	$E \times 10^{-4}$	$I \times 10^{-4}$
0	0	0
200	4.6144	6.5344
400	9.4976	13.3376
600	14.6496	20.4096
800	20.0704	27.7504
1000	25.7600	35.3600
1200	31.7184	43.2384
1400	37.9456	51.3856
1600	44.4416	59.8016
1800	51.2064	68.4864
2000	58.2400	77.4400
2200	65.5424	86.6624
2400	73.0600	96.1000
2600	80.9536	105.9316
2800	89.0624	115.9424
3000	97.4400	126.2400

SECTION IV.

THEORETICAL COMPUTATION OF DESIGN DATA FOR HUMPHREY PUMPS.

In this analysis, the following ideal set of conditions will be assumed:

1. The head will be assumed to remain constant throughout a cycle.

2. The pump admits and exhausts at atmosphere.

3. An intake temperature T_c of 300°C (abs.) will be assumed.

4. The initial cushion compression temperature T_3 will lso be taken as 300°C (abs.)

5. The cushion volume V_3 (Fig. 2) will be assumed as 40% of the stroke volume V_c (per lb. of mixture).

6. There are no mechanical or hydraulic losses in the pump.

7. There are no thermal losses due to radiation, etc.

From these considerations, it naturally follows that the water horse-power will be assumed equal to the thermal indicated horse-power.

Hence the indicater diagram shown in Fig. 15 will be assumed. P_c will be taken as atmospheric pressure. From C to O we have adiabatic compression of mixture, the water column being at rest at C and O; from O to 1, combustion at constant volume. (The column is also at rest at 1); from 1 to E, we have adiabatic expansion down to atmosphere. At E, the water-valves open and the water column continues on to E_1, atmospheric conditions existing in the pump chamber in the meanwhile. On the original water column reaching E_1, the level of the water in the pump barrel has risen to B. At B, the augmented water column is at rest. The exhaust is

completed/......

FIG. 15

completed at atmosphere from B to 3 when the exhaust valve shuts by impact. From 3 to 4 we have adiabatic compression of the cushion. At 4, the column is again at rest. From 4 to 5, the cushion expands to atmosphere. At 5, the admission valve opens and mixture admitted at atmospheric until C is reached when the column is at rest once more and a fresh cycle commenced.

It is clear from the indicater diagram that points 3 and 5 will coincide. Also $P_E = P_{E_1} = P_5 = P_c$. The point B has not so far been determined, but the following considerations will show that, in our ideal case, it coincides with point C.

The area under the curve B 3 4 (Fig. 15) represents the work done by the head on the gases in exhausting and compressing the cushion, (per lb. if mixture admitted), while the area under the curve 4 5 C represents the W.D. by the gases against the head in the cushion expansion and the admission of fresh mixture. Since the column is at rest at 4, B and C, it is clear that these areas must be the same and hence points B and C must coincide.

The volume of water pumped per cycle is clearly $V_{E_1} - V_c$ per lb. of mixture and the indicated work done against the head is $Wh(V_{E_1} - V_c)$ where h is the water head in feet and W the weight of a cubic foot of water. The indicated work done by the gases per cycle is the area I E C O, and hence it is clear that area I E C O = $Wh(V_{E_1} - V_c)$.

✓

Computation/.....

COMPUTATION OF THE MAIN STATE POINTS ON THE P.V. DIAGRAM.

In order to compute the conditions of pressure, volume and temperature existing at the main state points C, O, 1, E, E_1, 3 and 4, it will be necessary to consider the variation in internal energy of the working fluid throughout a complete cycle. It is well known that the change in internal energy of a gas under adiabatic conditions is a measure of the external work done on or by the gas as the case may be. Also, since we have assumed the head to remain constant throughout a complete cycle it is clear that the work done by or in the head for any displacement of the column is directly proportional to the displacement and hence the volume change of the working fluid (assuming the volume of fluid to be of constant cross-section). Thus if curves of E and W.D. by or on the total head (per lb. of gas) are drawn to a base of relative volumes, then the ordinates intercepted between the E and W.D. curves are clearly equal to the excess energy which appears in the form of K.E. of the water column (per lb. of gas) at the points considered. Thus for our particular cycle, energy v. volume relationships, in addition to pressure v. volume relationships, are of great use.

Fig. 16 shows such an Energy v. relative volume diagram. The lines representing W.D. on or by the head are clearly straight lines since the W.D. on or by the head is proportional to the displacement of the column. Since the K.E. of the column is zero at C and O, the straight line representing the work done by the head clearly passes through C and O. But the work done by the head per lb. of gas in compressing the gases from C to O is WH $(V_c - V_o)$, where H is the total head in feet, i.e. WH = Wh + P_c. Hence slope of line CO = $\dfrac{WH(V_c - V_o)}{\dfrac{V_c - V_o}{V_c}}$ = W.H.V_c. It is clear then, that

/the.......

COMPUTATION OF THE MAIN STATE POINTS ON THE P.V. DIAGRAM.

In order to compute the conditions of pressure, volume and temperature existing at the main state points C, O, 1, E, E_1, 3 and 4, it will be necessary to consider the variation in internal energy of the working fluid throughout a complete cycle. It is well known that the change in internal energy of a gas under adiabatic conditions is a measure of the external work done on or by the gas as the case may be. Also, since we have assumed the head to remain constant throughout a complete cycle it is clear that the work done by or in the head for any displacement of the column is directly proportional to the displacement and hence the volume change of the working fluid (assuming the volume of fluid to be of constant cross-section). Thus if curves of E and W.D. by or on the total head (per lb. of gas) are drawn to a base of relative volumes, then the ordinates intercepted between the E and W.D. curves are clearly equal to the excess energy which appears in the form of K.E. of the water column (per lb. of gas) at the points considered. Thus for our particular cycle, energy v. volume relationships, in addition to pressure v. volume relationships, are of great use.

Fig. 16 shows such an Energy v. relative volume diagram. The lines representing W.D. on or by the head are clearly straight lines since the W.D. on or by the head is proportional to the displacement of the column. Since the K.E. of the column is zero at C and O, the straight line representing the work done by the head clearly passes through C and O. But the work done by the head per lb. of gas in compressing the gases from C to O is WH $(V_c - V_o)$, where H is the total head in feet, i.e. WH = Wh + P_c. Hence slope of line CO = $\dfrac{WH(V_c - V_o)}{\frac{V_c - V_o}{V_c}}$ = W.H.V_c. It is clear then, that

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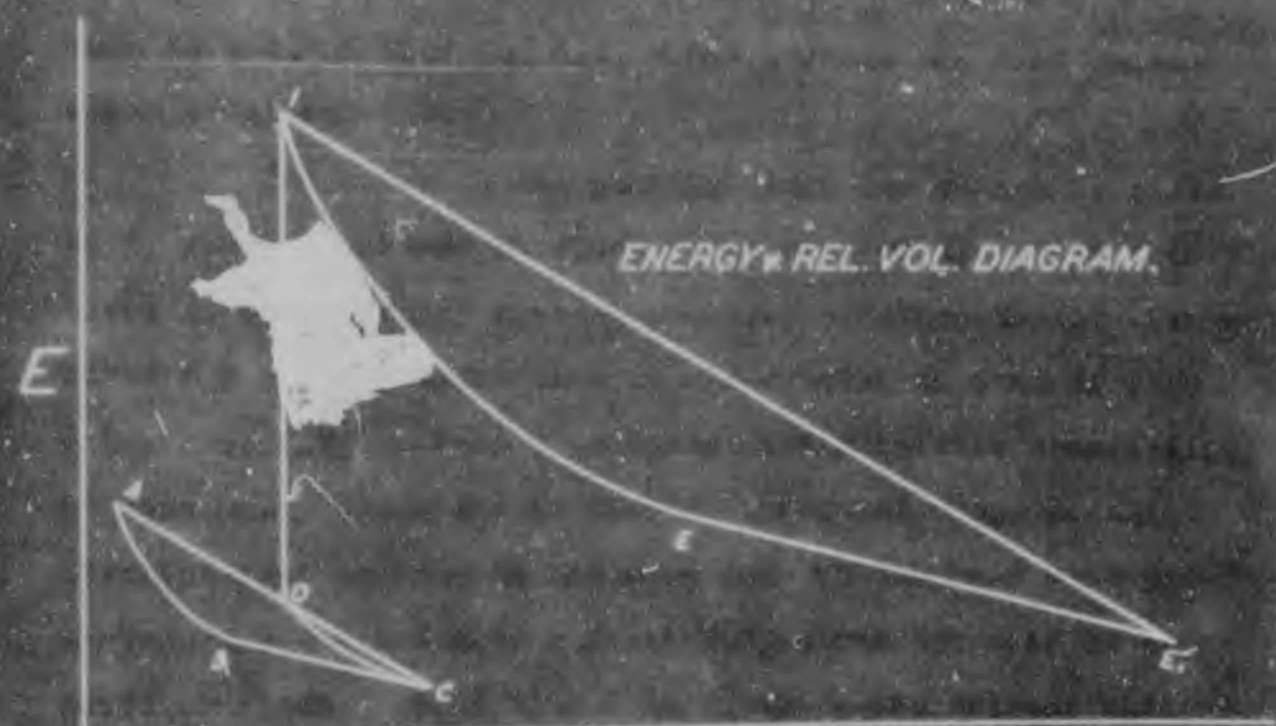

FIG. 16

the slope of this line for any fuel depends only on the total head.

From 0 to 1 the internal energy of the gas is increased by the addition of the calorific value of the mixture. From 0 to E the gas expands adiabatically down to atmosphere doing work against the head. The work done against the head from E to E_1 is clearly $P_c(V_{E_1} - V_E)$ since atmospheric conditions exist in the pump cylinder. It then follows that curve E E_1 is a straight line of slope $\dfrac{P_c(V_{E1} - V_E)}{\dfrac{V_{E1} - V_E}{V_c}} = P_cV_c$.

At 1 and E_1 the column is at rest. Hence the straight line indicating the amount of work done against the head must pass through 1 and E_1. The slope of line 1 E_1 is clearly the same as that of line CO, i.e. $W.H.V_c$, since the head is assumed constant.

The water in the pump cylinder has meanwhile risen to C, and the augmented column is at rest at C. From C to 3 there is expulsion of exhaust products at atmosphere. Hence curve C 3 is a straight line of slope equal to that of line E E_1, i.e. P_cV_c. From 3 to 4, we have adiabatic compression of cushion. Since the column is at rest at C and 4, the straight line indicating the work done by the head must pass through C and 4 and have a slope WHV_c since the head is assumed constant. From 4 to 3 to C, the gases retrace their path C to 3 to 4. Hence the diagrams for these two strokes coincide.

the slope of this line for any fuel depends only on the total head.

From O to 1 the internal energy of the gas is increased by the addition of the calorific value of the mixture. From O to E the gas expands adiabatically down to atmosphere doing work against the head. The work done against the head from E to E_1 is clearly $P_c(V_{E_1} - V_E)$ since atmospheric conditions exist in the pump cylinder. It then follows that curve E E_1 is a straight line of slope $\frac{P_c(V_{E1} - V_E)}{\frac{V_{E1} - V_E}{V_c}} = P_cV_c$.
At 1 and E_1 the column is at rest. Hence the straight line indicating the amount of work done against the head must pass through 1 and E_1. The slope of line 1 E_1 is clearly the same as that of line CO, i.e. $W.H.V_c$, since the head is assumed constant.

The water in the pump cylinder has meanwhile risen to C, and the augmented column is at rest at C. From C to 3 there is expulsion of exhaust products at atmosphere. Hence curve C 3 is a straight line of slope equal to that of line E E_1, i.e. P_cV_c. From 3 to 4, we have adiabatic compression of cushion. Since the column is at rest at C and 4, the straight line indicating the work done by the head must pass through C and 4 and have a slope WHV_c since the head is assumed constant. From 4 to 3 to C, the gases retrace their path C to 3 to 4. Hence the diagrams for these two strokes coincide.

Computation/.....

COMPUTATION OF STATE POINT, O, AND RELATION BETWEEN HEAD AND COMPRESSION RATIO (r).

Considering the compression stroke CO, the values of $R_T \left(\frac{T_o}{T_c}\right)$ for various values of $r \left(\frac{V_c}{V_o}\right)$ are obtained from chart G , by considering adiabatic compression of a gas with an initial compression temperature (T_c) of 300°C (abs.). Knowing the various values of $r(R_v^{-1})$ and R_T, $R_p\left(\frac{P_o}{P_c}\right)$ is easily deduced from the relationship: $R_p = \frac{R_T}{R_v}$. Since T_c is 300°C (abs.), T_o is easily calculated and E_o found from chart G. Now, since the column is at rest at C and O (Fig. 16) the work done by the head per lb. of mixture, $WH(V_c - V_o)$ is equal to the adiabatic work of compression, $E_c - E_o$.

Let us denote the slope of the line CO by m_r, r taking the values 1, 2,3, etc.

Hence:

$$
\begin{aligned}
WH(V_c - V_o) &= E_o - E_c \\
&= m_r\left(\frac{V_c - V_o}{V_c}\right) \\
&= m_r\left(1 - \frac{1}{r}\right).
\end{aligned}
$$

Knowing $E_o - E_c$ and r, m_r may be easily deduced for various values of r.

From the above it is also clear that $m_r = W.H.V_c$.

Hence for any fuel the total head pumped against is the only factor governing the compression ratio of any pump.

In this notation it is clear that m_1 (corresponding to unity compression ratio, i.e. no compression) the slope of lines E E_1 and C 3

$$
\begin{aligned}
&= P_cV_c = RT_c \\
&= 96 \times 300 \\
&= 2.88 \times 10^4 \text{ ft. lbs./lb.}
\end{aligned}
$$

The total head corresponding to various values of r may be expressed in atmosphere as follows:

$$\frac{m_r}{m_1} = \frac{W.H.V_c}{P_c.V_c}.$$

Hence/.....

Hence W.H. = $\frac{m_r}{m_1}$ atmospheres and Wh, the water head

= $\frac{m_r}{m_1}$ - 1 atmospheres

since W.H. = Wh + P_c.

Design Chart 1 (D.C.(1)) shows the relationship between head of water in atmospheres and compression ratio.

An examination of this chart will show that the maximum water head pumped against with gas as a fuel is limited to about 2.5 atmospheres. Higher heads are bound to lead to trouble as far as pre-ignition is concerned.

COMPUTED VALUES FOR POINT 0.

r	1	2	3	4	5	6
$\frac{V_o}{V_c}$	1	0.5	0.333	0.25	0.2	0.167
$\frac{T_o}{T_c}$	1	1.315	1.526	1.710	1.854	1.982
T_o	300	395	458	513	558	595
$E_o x 10^{-4}$	7.05	9.35	11.00	12.35	13.52	14.50
$\frac{P_o}{P_c}$	1	2.630	4.578	6.840	9.270	11.892
$m_r x 10^{-4}$	2.88	4.60	5.93	7.07	8.10	8.90
$\frac{WH}{P_c}$	1	1.60	2.06	2.45	2.81	3.09
$\frac{Wh}{P_c}$	0	0.60	1.06	1.45	1.81	2.09

Computation/.....

Hence W.H. = $\frac{m_r}{m_1}$ atmospheres and Wh, the water head

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Design Chart 1 (D.C.(1)) shows the relationship between head of water in atmospheres and compression ratio.

An examination of this chart will show that the maximum water head pumped against with gas as a fuel is limited to about 2.5 atmospheres. Higher heads are bound to lead to trouble as far as pre-ignition is concerned.

COMPUTED VALUES FOR POINT O.

r	1	2	3	4	5	6
$\frac{V_o}{V_c}$	1	0.5	0.333	0.25	0.2	0.167
$\frac{T_o}{T_c}$	1	1.315	1.526	1.710	1.854	1.982
T_o	300	395	458	513	558	595
$E_o x10^{-4}$	7.05	9.35	11.00	12.35	13.52	14.50
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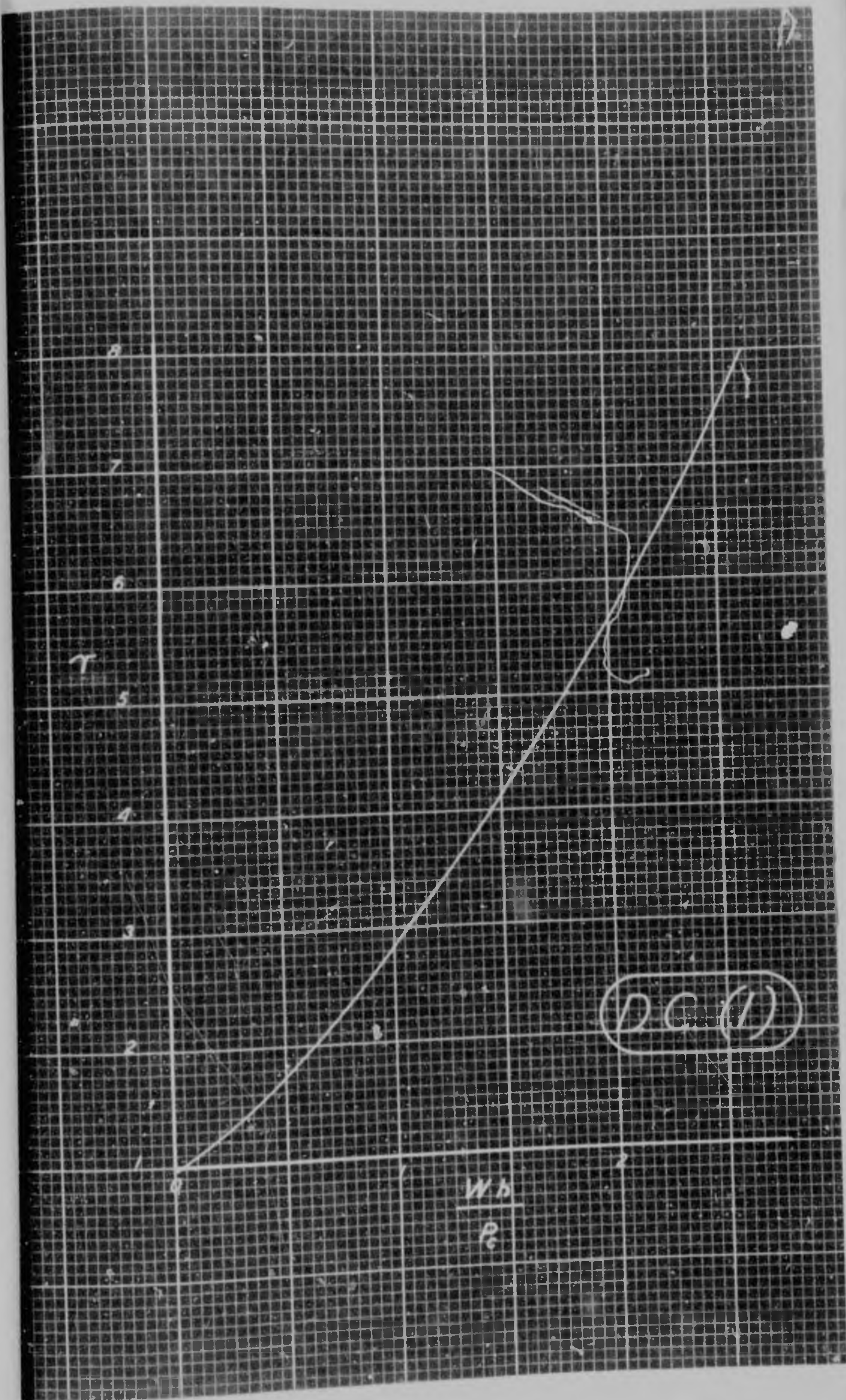
Computation/.....



8
7
6
T
5
4
3
2
1
0
1
2
Wb
Pc
DC (1)

COMPUTATION OF STATE POINT 1.

0 to 1 represents the addition of heat to the gas at constant volume. Since there is no external work done, the amount of heat added (which is equal to the calorific value of the mixture) is entirely used in increasing the store of internal energy of the gas.

Hence if C.V. is the calorific value of the mixture in ft. lbs./lb., then:

$$C.V. = E_1 - E_o.$$

E_1 is hence easily deduced for various values of C.V. and E_o. Knowing E_1, T_1 may be found from chart G and hence $\frac{T_1}{T_c}$. Knowing $\frac{T_1}{T_c}$ and $\frac{V_1}{V_c}$ $(= \frac{V_o}{V_c})$, $\frac{P_1}{P_c}$ may be calculated from the relationship:

$$\frac{P_1}{P_c} = \frac{\frac{T_1}{T_c}}{\frac{V_1}{V_c}}$$

These quantities were deduced for values of r ranging from 2 to 6, and values of C.V. ranging from 40×10^4 ft. lbs./lb. to 80×10^4 ft. lbs./lb.

Design Chart 2 (D.C. (2)) was then drawn showing the relationship between T_1, the maximum explosion temperature and the water head in atmospheres.

Design Chart 3 (D.C. (3)) shows the relationship between $\frac{P_1}{P_c}$, the maximum explosion pressure expressed in atmospheres, and the water head.

It should be noted that owing to the cooling effect of the water in the combustion chamber, the maximum temperature will be considerably less in practice.

COMPUTATION OF STATE POINT 1.

O to 1 represents the addition of heat to the gas at constant volume. Since there is no external work done, the amount of heat added (which is equal to the calorific value of the mixture) is entirely used in increasing the store of internal energy of the gas.

Hence if C.V. is the calorific value of the mixture in ft. lbs./lb., then:

$$C.V. = E_1 - E_o.$$

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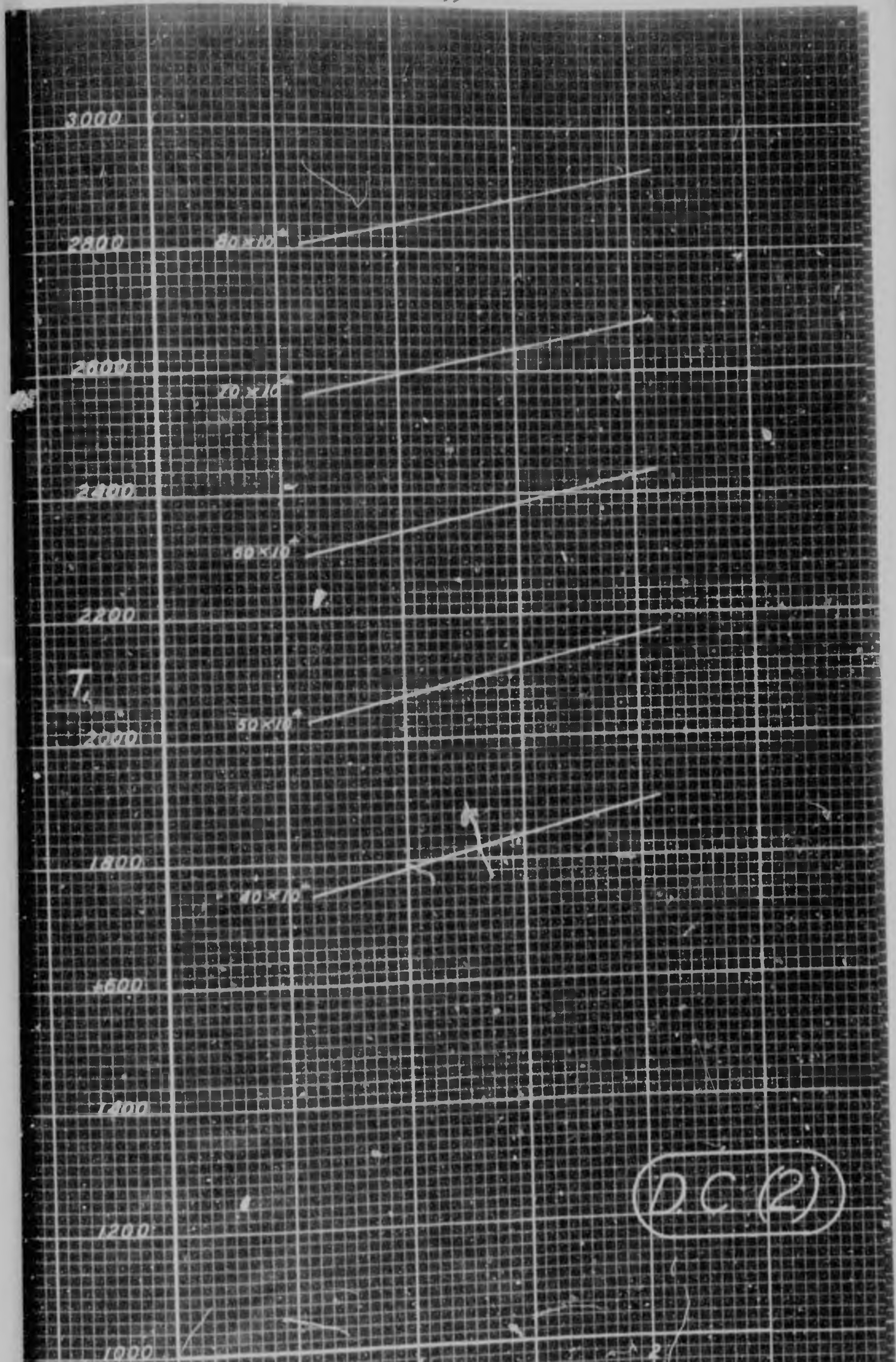
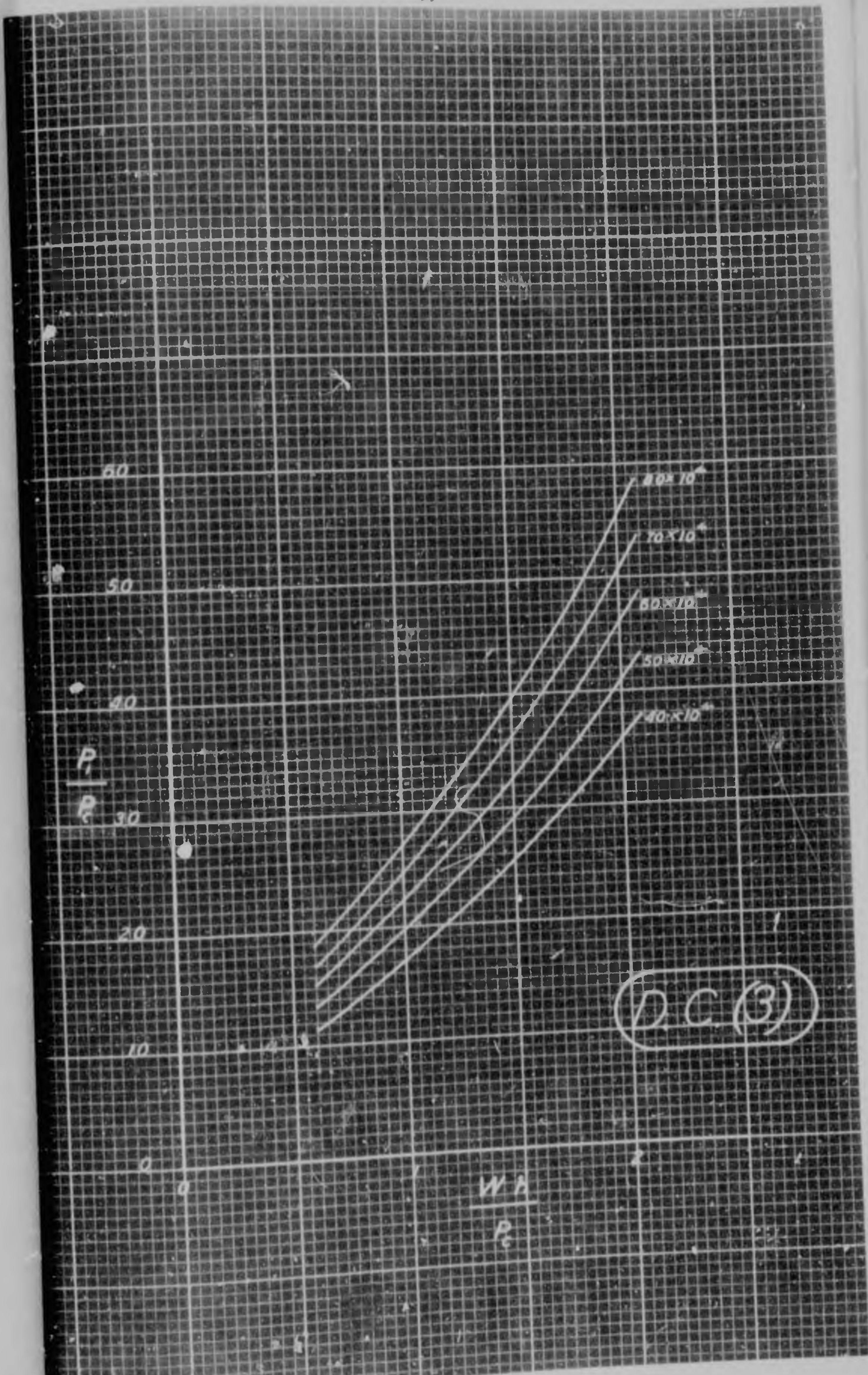

3000
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
T_1
80×10
70×10
60×10
50×10
40×10
0
1
2
$\frac{W_b}{P_c}$
D.C.(2)


80×10⁴
70×10⁴
60×10⁴
50×10⁴
40×10⁴
60
50
40
30
20
10
0
P₁/P_c
0
1
2
WA/P_c
D.C.(3)

COMPUTED VALUES FOR POINT 1.

r = 2. $E_o = 9.35 \times 10^4$ ft.lbs./lb.

C.V. x 10^{-4}	$E_1 \times 10^{-4}$	T_1	$\frac{P_1}{P_c}$
40	49.35	1745	11.62
50	59.35	2030	13.53
60	69.35	2300	15.33
70	79.35	2560	17.07
80	89.35	2805	18.70

r = 3. E_o = 11.00 ft./lbs./lb.

C.V. x 10^{-4}	$E_1 \times 10^{-4}$	T_1	$\frac{P_1}{P_c}$
40	51	1795	17.95
50	61	2075	20.75
60	71	2340	23.40
70	81	2600	26.00
80	91	2845	28.45

r = 4. E_o = 12.35 ft. lbs./lb.

C.V. x 10^{-4}	$E_1 \times 10^{-4}$	T_1	$\frac{P_1}{P_c}$
40	52.35	1835	24.50
50	62.35	2110	28.15
60	72.35	2380	31.75
70	82.35	2635	35.20
80	92.35	2880	38.40

r = 5/.....

[illegible] = 5. E_o = 13.52 x 10^4 ft.lbs./lb.

[illegible].V. x 10^{-4}	E_1 x 10^{-4}	T_1	$\frac{P_1}{P_c}$
40	53.52	1865	31.1
50	63.52	2140	35.7
60	73.52	2410	40.2
70	83.52	2660	44.3
[illegible]	93.52	2905	48.4

[illegible] = 6. E_o = 14.50 x 10^4 ft.lbs./lb.

[illegible].V. x 10^{-4}	E_1 x 10^{-4}	T_1	$\frac{P_1}{P_c}$
40	54.50	1895	37.9
50	64.50	2170	43.4
60	74.50	2435	48.7
70	84.50	2685	53.7
80	94.50	2930	58.6

Computation/.....

COMPUTATION OF STATE POINT E.

From 1 to E, the gases expand down to atmosphere. Hence for the expansion 1 to E, $R_p^{-1} = \frac{P_1}{P_E} = \frac{P_1}{P_c}$ which we have already deduced. The initial temperatures, T_1, have also been deduced. From the chart G, the corresponding values of $R_T = \frac{T_E}{T_1}$ are found. Knowing R_T, T_E and E_E are easily found. $R_V = \frac{V_E}{V_1}$ is found from the relationship: $R_p = \frac{R_T}{R_V}$.

V_E may be expressed in terms of V_c:-

$$\frac{V_E}{V_c} = \frac{V_E}{V_1} \cdot \frac{V_1}{V_c} = \frac{\frac{V_E}{V_1}}{r}.$$

All the above-mentioned quantities were computed for the values of r and C.V. considered.

COMPUTED VALUES FOR POINT E.

r = 2.

C.V. x 10^{-4}	T_E	E_E x 10^{-4}	$\frac{V_E}{V_c}$
40	981	25.1	3.27
50	1125	29.3	3.75
60	1249	33.1	4.16
70	1395	37.9	4.65
80	1510	41.4	5.03

r = 3.

C.V. x 10^{-4}	T_E	E_E x 10^{-4}	$\frac{V_E}{V_c}$
40	903	22.9	3.01
50	1032	26.5	3.44
60	1161	30.4	3.87
70	1284	34.2	4.28
80	1410	38.3	4.70

r = 4/....

r = 4.

C.V. x 10^{-4}	T_E	E_E x 10^{-4}	$\frac{V_E}{V_c}$
40	852	21.5	2.84
50	981	25.1	3.27
60	1098	28.5	3.66
70	1224	32.4	4.08
80	1335	35.8	4.45

r = 5.

40	828	20.8	2.76
50	942	24	3.14
60	1056	27.2	3.52
70	1164	30.6	3.88
80	1275	34.0	4.25

r = 6.

40	810	20.3	2.70
50	930	23.5	3.10
60	1023	26.3	3.41
70	1128	29.4	3.76
80	1236	32.5	4.12

Computation/.....

COMPUTATION OF STATE POINT E_1.

In what follows, we shall denote the intercept between the curve showing the change of internal energy of the gas (E) and the straight line representing the work done by or on the total head, by E^1 (Fig. 16). It was shown previously that E^1 is a measure of the K.E. of the water column per lb. of mixture.

At E, atmospheric conditions exist in the pump cylinder. Hence slope of line EE_1, as mentioned previously, is equal to m_1, and slope of line $1E_1 = m_r$.

From Fig. 16 it is clear that:

$$E^1{}_E = \frac{V_{E_1} - V_E}{V_c}(m_r - m_1)$$

$$\text{and} = (E_1 - E_E) - \frac{V_E - V_1}{V_c} m_r$$

and since $\dfrac{V_1}{V_c} = \dfrac{1}{r}$

$$\frac{V_{E_1}}{V_c} = \frac{V_E}{V_c} + \frac{(E_1 - E_E) - \left(\frac{V_E}{V_c} - \frac{1}{r}\right) m_r}{(m_r - m_1)} .$$

Since we have ascertained all the quantities on the right-hand side of the above equation it is a simple matter to calculate the corresponding values of $\dfrac{V_{E_1}}{V_c}$

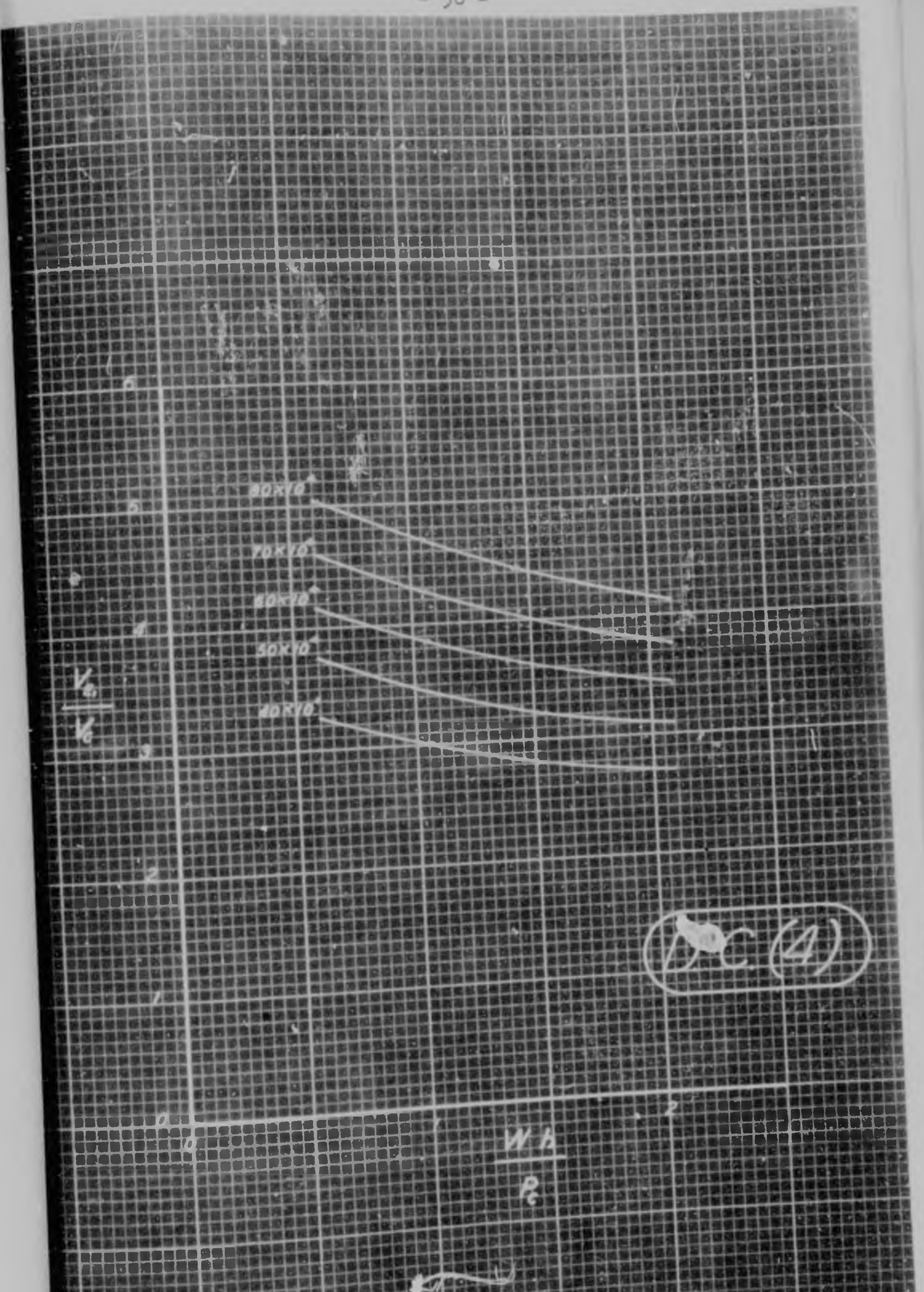
Design Chart 4 (D.C. (4)) shows the relationship between $\dfrac{V_{E1}}{V_c}$ and head for various values of C.V.

If L_c is the length of the charging stroke corresponding to volume V_c, then L_{E_1}, the maximum distance travelled by the water column is given by $L_c \dfrac{V_{E1}}{V_c}$.

Hence D.C. (4) yields values of L_{E_1} in terms of L_c, which serve as indications of where to place the water valves.

It should be noted that the length of the various strokes expressed in terms of L_c depend only on the head and the mixture strength used.

Computed/....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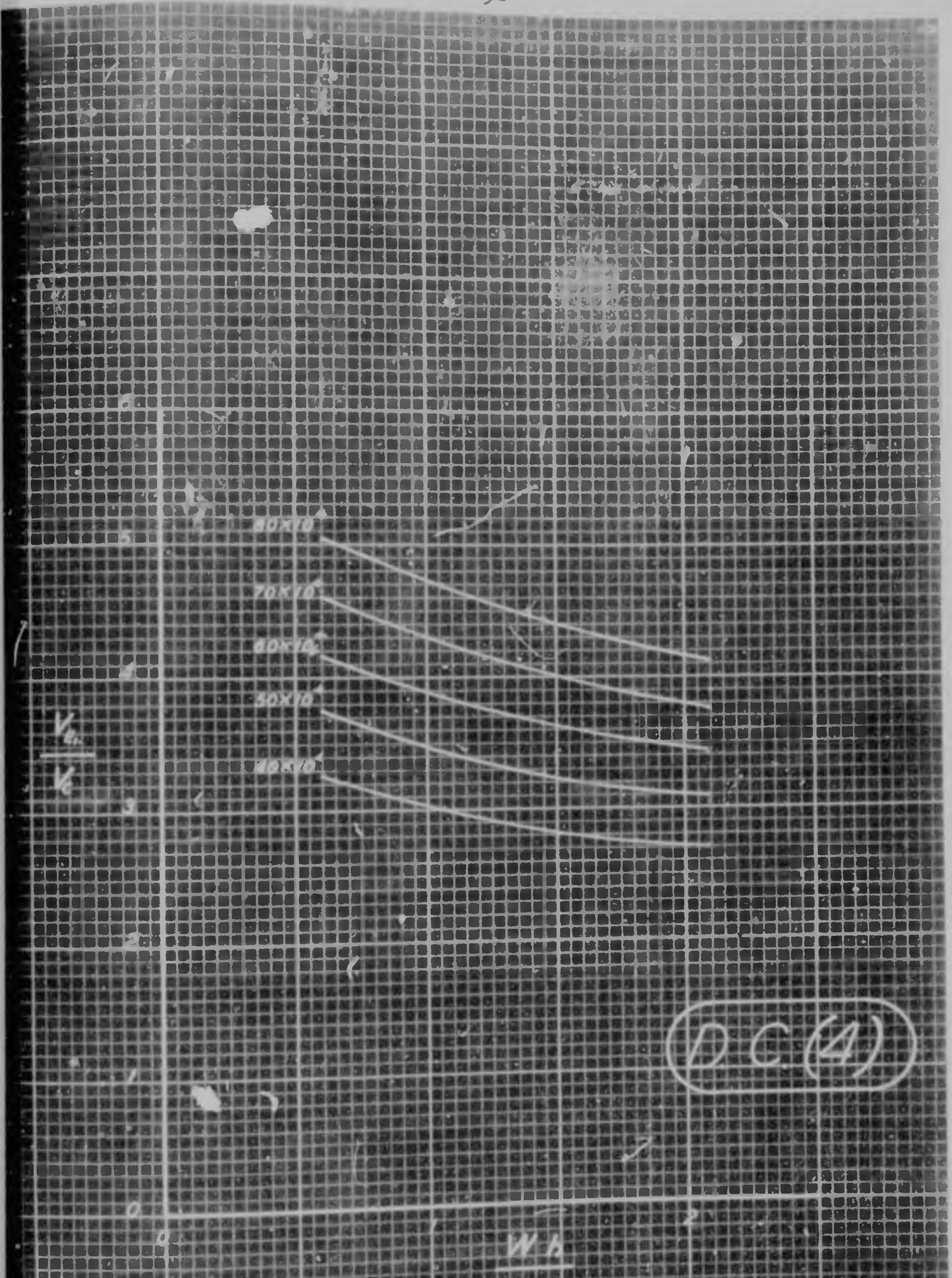
DC (4)


80X10
70X10
60X10
50X10
40X10
DC(4)
0
1
2
3
4
5
6
Wh
Pc

COMPUTED VALUES FOR POINT E_1.

Values of $E^1{}_E \times 10^{-4}$.

C.V. x 10^{-4} r	40	50	60	70	80
2	11.50	15.1	19.4	22.4	27.1
3	12.4	16.0	19.6	23.4	26.2
4	12.6	15.9	19.8	22.9	26.9
5	12.0	15.7	19.4	23.1	26.7
6	11.7	14.9	19.4	23.2	26.8

Values of $\frac{V_{E_1}}{V_c}$.

C.V. x 10^{-4} r	40	50	60	70	80
2	9.91	12.56	15.02	17.75	20.43
3	7.00	8.59	10.25	11.98	13.43
4	5.96	7.20	8.46	9.80	10.97
5	5.10	6.16	7.19	8.26	9.32
6	4.64	5.63	6.59	7.54	8.50

Computation/.....

COMPUTATION OF STATE POINT 3.

From C to 3 we have expulsion of exhaust products at atmospheric pressure. We have assumed $\frac{V_3}{V_c} = 0.4$.

From Fig. 16, it is clear that $E^1{}_3$

$$= \left(\frac{V_c}{V_c} - \frac{V_3}{V_c}\right)(m_r - m_1)$$

$$= 0.6\ (m_r - m_1).$$

COMPUTED VALUES OF $E^1{}_3$.

r	$E^1{}_3 \times 10^{-4}$
2	1.032
3	1.830
4	2.514
5	3.132
6	3.612

Computation/.....

COMPUTATION OF STATE POINT 3.

From C to 3 we have expulsion of exhaust products at atmospheric pressure. We have assumed $\frac{V_3}{V_c} = 0.4$.

From Fig. 16, it is clear that E^1_3

$$= \left(\frac{V_c}{V_c} - \frac{V_3}{V_c}\right)(m_r - m_1)$$

$$= 0.6\ (m_r - m_1).$$

COMPUTED VALUES OF E^1_3.

r	$E^1_3 \times 10^{-4}$
2	1.032
3	1.830
4	2.514
5	3.132
6	3.612

Computation/.....

COMPUTATION OF STATE POINT 4.

From 3 to 4 we have adiabatic compression of 0.4 lbs. of gas since 0.6 lbs. have been expelled to exhaust for every lb. of gas admitted. As stated previously, we shall assume T_3, the initial cushion compression temperature, as 300°C (abs.), since the exhaust gases have been well scavenged.

$\frac{V_4}{V_3}$ for various values of r cannot be obtained analytically and hence recourse will be had to graphical methods.

Referring to Chart θ_{34}, curve E was drawn to a base of $\frac{V}{V_c}$. This curve corresponds to curve 34 in Fig. 16, and shows the variation in internal energy of 0.4 lbs. of gas with change of relative volume.

Owing to the small range in the value of the internal energy, the accuracy afforded by Chart G was not sufficient and the various points on this curve were calculated from first principles.

Various temperatures were considered and the corresponding values of E (per lb.) and R_V^{-1} calculated. The values of $\frac{V}{V_c}$ corresponding to $R_V^{-1}(\frac{V_3}{V})$ are deduced from the relationship:

$$R_V^{-1} = \frac{V_3}{V} = \frac{V_c}{V}.\frac{V_3}{V_c} = \frac{V_c}{V}(0.4).$$

$$\text{Hence } \frac{V}{V_c} = \frac{0.4}{R_V^{-1}}.$$

The values of E (per lb.) were multiplied by 0.4 to give internal energies of 0.4 lbs. of gas, and reduced to a datum of the energy at $\frac{V_3}{V_c}$, i.e. at 300°C. (abs.)

Straight lines H_2, H_3, etc. with slopes m_2, m_3, etc. were then drawn through points (0.4, E_3^1). These lines clear-
ly correspon straight line CO4 in Fig. 16. It is
then clear th the intersection of the lines H_2, H_3, etc.
with curve E must yield us values of $\frac{V_4}{V_c}$ for the various com-
pression ratios considered. From these values of $\frac{V_4}{V_c}$ we de-
duce the corresponding values of R_V^{-1} $(\frac{V_3}{V_4})$ and from Chart G,

the/.....

COMPUTATION OF STATE POINT 4.

From 3 to 4 we have adiabatic compression of 0.4 lbs. of gas since 0.6 lbs. have been expelled to exhaust for every lb. of gas admitted. As stated previously, we shall assume T_3, the initial cushion compression temperature, as 300°C (abs.), since the exhaust gases have been well scavenged.

$\frac{V_4}{V_3}$ for various values of r cannot be obtained analytically and hence recourse will be had to graphical methods.

Referring to Chart θ 34, curve E was drawn to a base of $\frac{V}{V_c}$. This curve corresponds to curve 34 in Fig. 16, and shows the variation in internal energy of 0.4 lbs. of gas with change of relative volume.

Owing to the small range in the value of the internal energy, the accuracy afforded by Chart G was not sufficient and the various points on this curve were calculated from first principles.

Various temperatures were considered and the corresponding values of E (per lb.) and R_V^{-1} calculated. The values of $\frac{V}{V_c}$ corresponding to $R_V^{-1}(\frac{V_3}{V})$ are deduced from the relationship:

$$R_V^{-1} = \frac{V_3}{V} = \frac{V_c}{V}.\frac{V_3}{V_c} = \frac{V_c}{V}(0.4).$$

$$\text{Hence } \frac{V}{V_c} = \frac{0.4}{R_V^{-1}}.$$

The values of E (per lb.) were multiplied by 0.4 to give internal energies of 0.4 lbs. of gas, and reduced to a datum of the energy at $\frac{V_3}{V_c}$, i.e. at 300°C. (abs.)

Straight lines H_2, H_3, etc. with slopes m_2, m_3, etc. were then drawn through points (0.4, E_3^1). These lines clearly correspond to the straight line CO4 in Fig. 16. It is then clear that the intersection of the lines H_2, H_3, etc. with curve E must yield us values of $\frac{V_4}{V_c}$ for the various compression ratios considered. From these values of $\frac{V_4}{V_c}$ we deduce the corresponding values of R_V^{-1} $(\frac{V_3}{V_4})$ and from Chart G,

the/.....

the values of R_T ($\frac{T_4}{T_3}$) are easily found. R_p ($\frac{P_4}{P_3} = \frac{P_4}{P_c}$) is then deduced from the relationship $R_p = \frac{R_T}{R_v}$.

Design Chart 5 (D.C. (5)) shows the variation in value of $\frac{P_4}{P_c}$, the maximum cushion pressure in atmospheres, with head. An examination of D.C. (5) will show that for strong mixture strengths and heads up to about 1.5 atmospheres, the explosion pressure is greater than the maximum cushion pressure, while for heads greater than 1.5 atmospheres, the maximum cushion pressure is in excess of the explosion pressure. For weak mixture strengths, the cushion maximum is usually in excess of the explosion maximum.

DATA SHEET FOR CHART θ_{34}

E_3 = 7.050 ft.lbs./lb.

T^oC (abs.)	E x 10^{-4}	$0.4(E-E_3) \times 10^{-4}$	R_v^{-1}	$\frac{V}{V_c}$
300	7.050	0	1.000	0.400
315	7.391	0.147	1.132	0.354
330	7.759	0.294	1.275	0.314
345	8.128	0.442	1.428	0.281
360	8.500	0.591	1.593	0.252
375	8.874	0.740	1.770	0.227
390	9.248	0.890	1.960	0.205
405	9.624	1.040	2.160	0.186
420	10.002	1.191	2.338	0.172
435	10.382	1.343	2.606	0.153
450	10.762	1.495	2.850	0.141
465	11.143	1.648	3.110	0.129
480	11.528	1.802	3.380	0.119
495	11.914	1.956	3.670	0.109
510	12.298	2.110	3.980	0.101
525	12.688	2.276	4.30	0.093
540	13.077	2.421	4.64	0.086

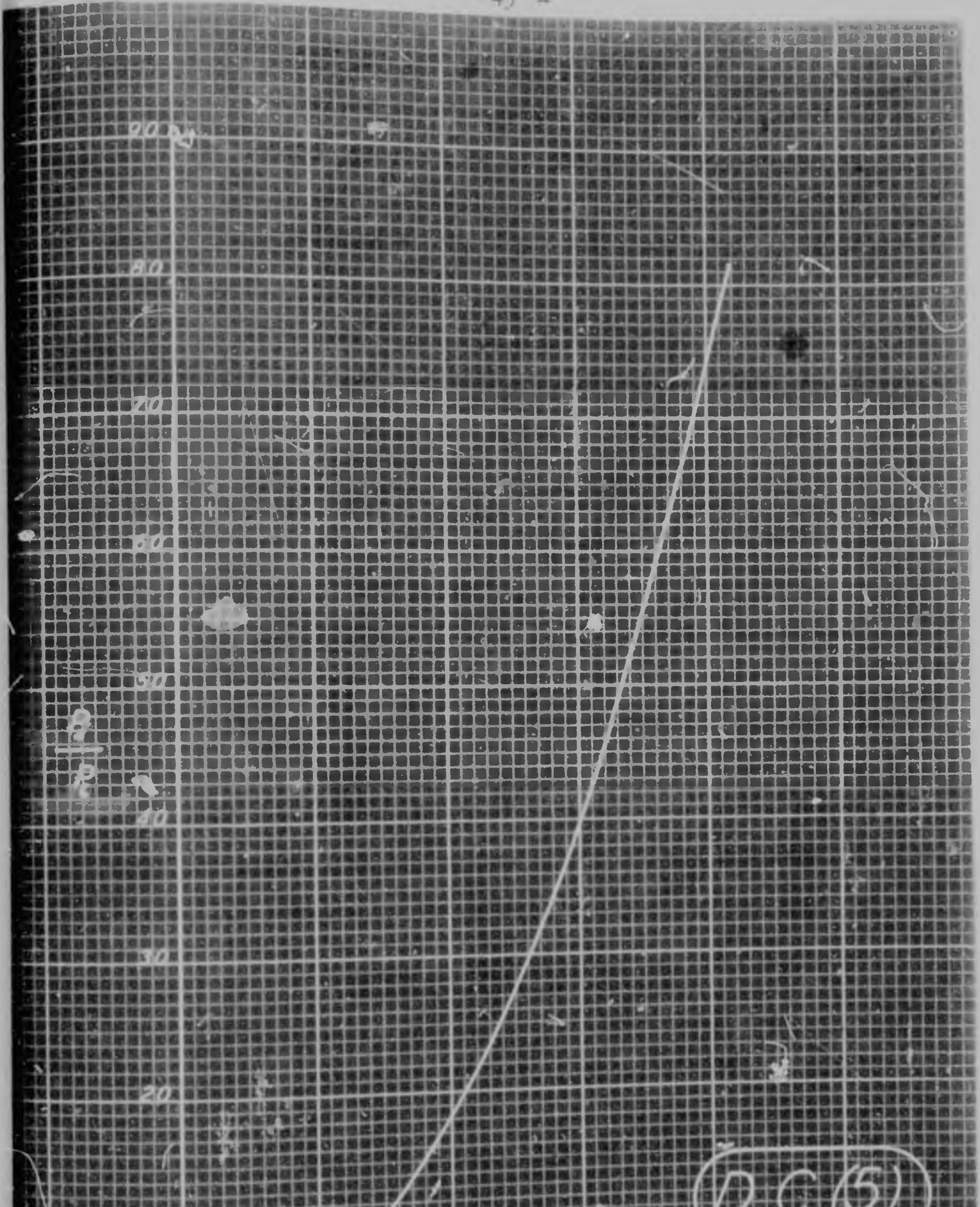
cont./.....

T°C (abs)	E x 10^{-4}	0.4(E-E_2) x 10^{-4}	n_V^{-1}	$\frac{V}{V_c}$
555	[illegible]	2.5[illegible]	5.00	0.080
570	[illegible].853	2.734	5.36	0.075
[illegible]	[illegible]4.251	2.891	5.76	0.070
[illegible]	[illegible]4.650	3.051	6.17	0.065
[illegible]	15.05	3.211	6.63	0.060
[illegible]	15.450	3.371	7.06	0.057
645	[illegible]8.84[illegible]	3.525	7.54	0.053
66[illegible]	[illegible]	3.691	8.00	0.050
From [illegible]				
	[illegible]	3.78	8.41	0.475
	17.69	4.26	10.00	0.040
	19.48	4.98	13.33	0.030
	22.79	6.30	20.00	0.020
	25.20	7.26	26.67	0.015
	29.80	9.10	40	0.01

COMPUTED VALUES FOR POINT 4.

r	$\frac{V_4}{V_c}$	$\frac{P_4}{P_c}$
[illegible]	0.084	7.76
3	0.045	20.26
4	0.029	36.80
[illegible]	0.020	60.40
[illegible]	[illegible].016	81.50

Thermal/.....


DC(5)

THERMAL EFFICIENCY OF THE CYCLE.

The amount of heat supplied per lb. of gas per cycle is clearly equal to C.V. The heat expelled during exhaust occurs at constant pressure from E to C (Fig.15) and is clearly equal to $(I_E - I_c)$ the loss in total energy from E to C.

Hence the work done per lb. of gas per cycle = C.V. - $(I_E - I_c)$ and the thermal efficiency of the cycle,

$$= \frac{C.V. - (I_E - I_c)}{C.V.}$$

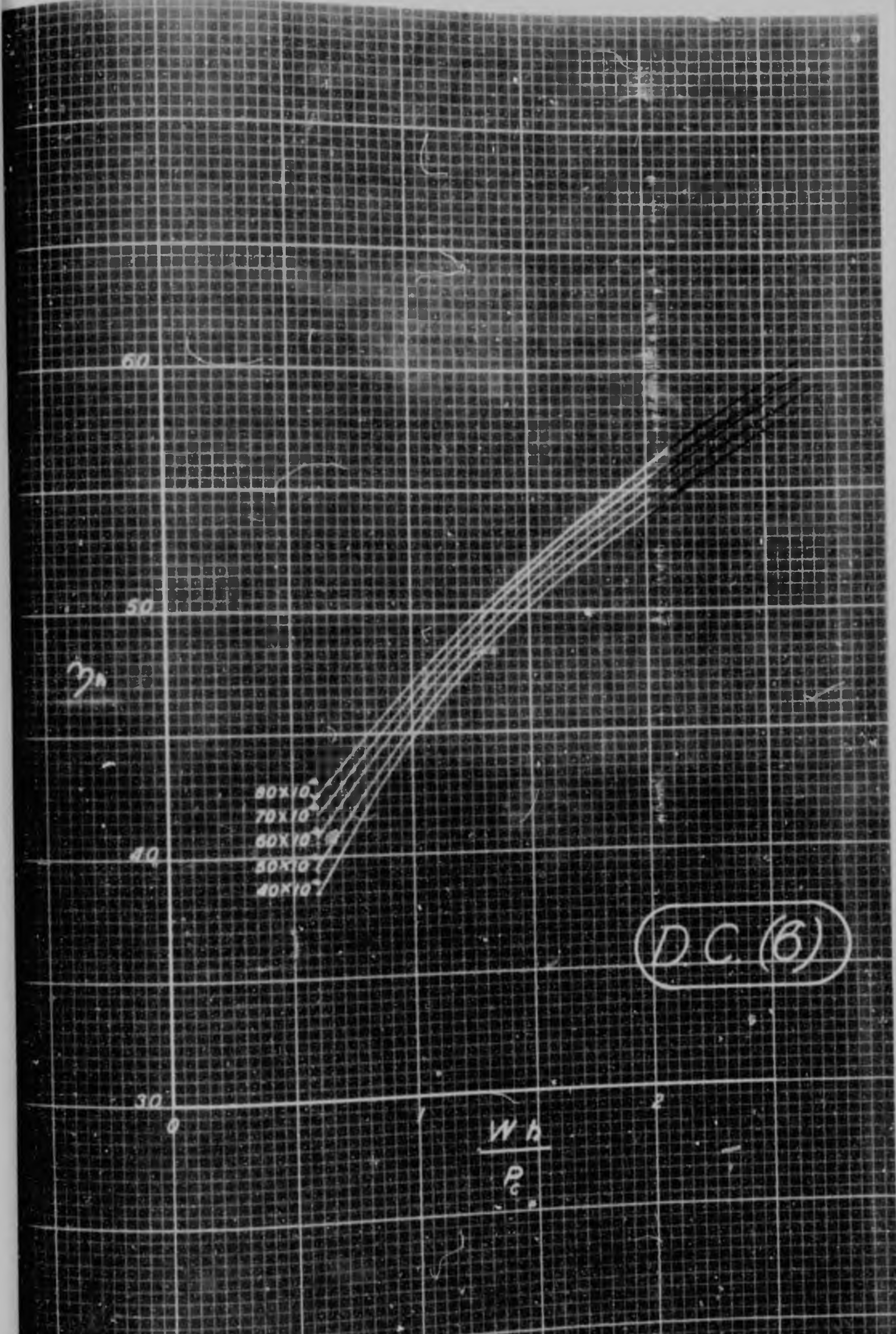
Since T_E and T_c for the various values of r and C.V. are known, the corresponding values of I_E and I_c may be deduced from Chart G and η_h calculated.

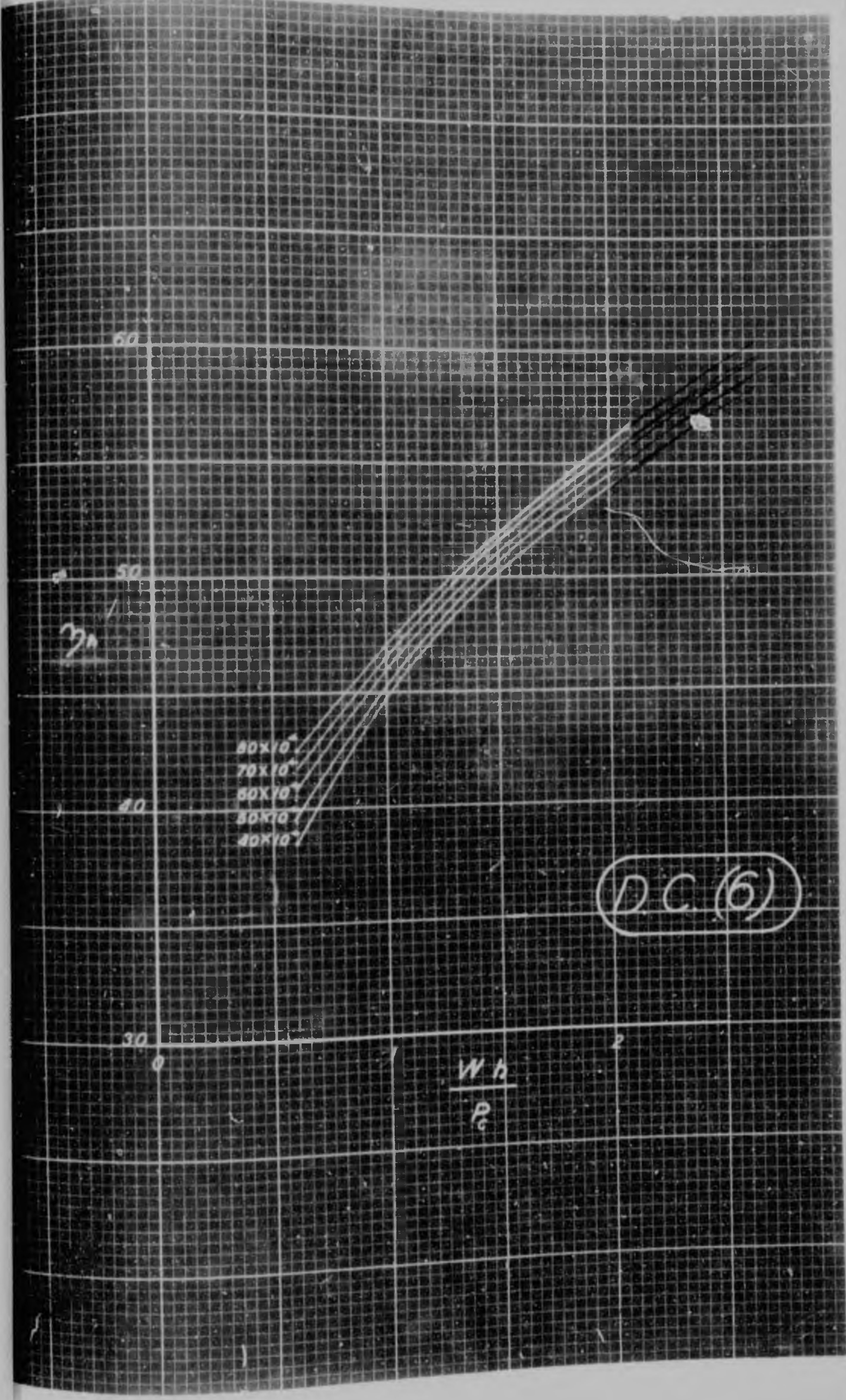
Design Chart 6 (D.C. (6)) shows the relationship between η_h and head for the various values of C.V. considered. It is seen that η_h is only affected to a very small degree by the mixture strength, while the main factor, as one would expect, is the head. These figures are higher than for engines employing the same fuel and compression ratio, but working on the Otto Cycle.

COMPUTED VALUES OF η_h (%).

C.V. x10^{-4} / r	40	50	60	70	80
2	38.5	39.4	46.6	41.	42.5
3	45.7	46.8	47.1	47.4	47.6
4	50.8	50.8	51.2	51.4	51.5
5	53	53.6	54.2	54.5	54.5
6.	54.5	54.6	56.1	56.6	56.6

Periodic/.....


60
50
40
30
ηn
80×10
70×10
60×10
50×10
40×10
0
1
2
Wh
Pc
D.C. (B)


60
50
40
30
η_h
80×10
70×10
60×10
50×10
40×10
0
1
2
$\frac{W h}{P_c}$
D.C.(6)

PERIODIC RATE OF ACTION OF PUMP.

As mentioned previously, the intercepts E^1 (Fig.16) between the curves of W.D. by or on the gas, and W.D. by or against the head, give a measure of the K.E. of the column per lb. of mixture.

In order to simplify the analysis in the case where the cross-sectional area of the piping is not constant from point to point, we shall reduce the column to one having the same K.E. as the original column at any instant, but with uniform cross-section throughout equal to that of the pump barrel.

If A is the cross-sectional area of pump barrel and "a" the cross-section at any section of elementary length dx, and if "v" is the velocity of flow of the water in the barrel at any instant, then the K.E. of the element of the column of length dx:

$$d(K.E) = \frac{W a dx}{2g}\left(v^2 \frac{A^2}{a^2}\right) \text{ (Engineers' units).}$$

If the total length of piping is L_T, then total K.E. of column

$$K.E. = \frac{W A^2 v^2}{2g} \int_0^{L_T} \frac{dx}{a}$$

the integration being carried out from $x = 0$ to $x = L_T$.

The above expression may be written:

$$\frac{W A v^2}{2g} \left[A \int_0^{L_T} \frac{dx}{a} \right]$$

The expression

$$\left[A \int_0^{L_T} \frac{dx}{a} \right]$$

clearly denotes the length of a water column of constant cross-section A having the same total K.E. as the column considered. We shall call this quantity L_E, the equivalent inertia length of the column.

Thus:-

K.E./.....

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$$K.E. = \frac{W A^2 v^2}{2g} \int_0^{L_T} \frac{dx}{a}$$

the integration being carried out from $x = o$ to $x = L_T$.

The above expression may be written:

$$\frac{W A v^2}{2g} \left[A \int_0^{L_T} \frac{dx}{a} \right]$$

The expression

$$\left[A \int_0^{L_T} \frac{dx}{a} \right]$$

clearly denotes the length of a water column of constant cross-section A having the same total K.E. as the column considered. We shall call this quantity L_E, the equivalent inertia length of the column.

Thus:-

K.E./.....

$$K.E. = \frac{WAL_E v^2}{2g}.$$

Further:-

$$K.E. = E^1 \times (\text{number of lbs. of mixture in cylinder})$$

$$= E^1 \times \frac{A\ L_c}{V_c}$$

(where L_c is the length of the combustion chamber corresponding to point c).

Hence

$$\frac{W\ A\ L_E\ v^2}{2g} = \frac{E^1\ A\ L_c}{V_c}.$$

Also

$$v = \frac{dx}{dt} = L_c\ \frac{d(\frac{V}{V_c})}{dt}$$

and V_c is more conveniently expressed as:-

$$V_c = \frac{R\ T_c}{P_c} = \frac{m_1}{P_c}$$

Hence from above:-

$$v = \left[\frac{2g\ P_c\ L_c}{W\ L_E\ m_1}\right]^{\frac{1}{2}} \times \sqrt{E^1} = L_c\ \frac{d(\frac{V}{V_c})}{dt}.$$

$$\therefore dt = \left[\frac{W\ L_E\ L_c\ m_1}{2g\ P_c}\right]^{\frac{1}{2}} \times \frac{d(\frac{V}{V_c})}{\sqrt{E^1}}.$$

Hence T, the periodic rate of action of the pump:

$$T = \left[\frac{W\ L_E\ L_c\ m_1}{2g\ P_c}\right]^{\frac{1}{2}} \times \oint \frac{d(\frac{V}{V_c})}{E^1}$$

where the integration is performed over a complete cycle.

The expression $\oint \frac{d(\frac{V}{V_c})}{E^1}$

will be denoted by θ, the period coefficient of the pump

$$\therefore T = \left[\frac{W\ L_E\ L_c\ m_1}{2g\ P_c}\right]^{\frac{1}{2}} \theta$$

Also:

/.....

$$\oint \frac{d(\frac{V}{V_c})}{\sqrt{E^1}} = \int_o^c \frac{d(\frac{V}{V_c})}{\sqrt{E^1}} + \int_1^E \frac{d(\frac{V}{V_c})}{\sqrt{E^1}}$$

$$+ \int_E^{E_1} \frac{d(\frac{V}{V_c})}{\sqrt{E^1}} + 2\int_c^3 \frac{d(\frac{V}{V_c})}{\sqrt{E^1}} + 2\int_3^4 \frac{d(\frac{V}{V_c})}{\sqrt{E^1}} .$$

In the "θ" notation

$$\theta = \theta_{CO} + \theta_{1E} + \theta_{EE_1} + 2\theta_{C3} + 2\theta_{34}.$$

These integrals were computed for the values of r and C.V. considered. In Design Chart 7 (D.C. (7)) these values are plotted to a base of water head in atmospheres.

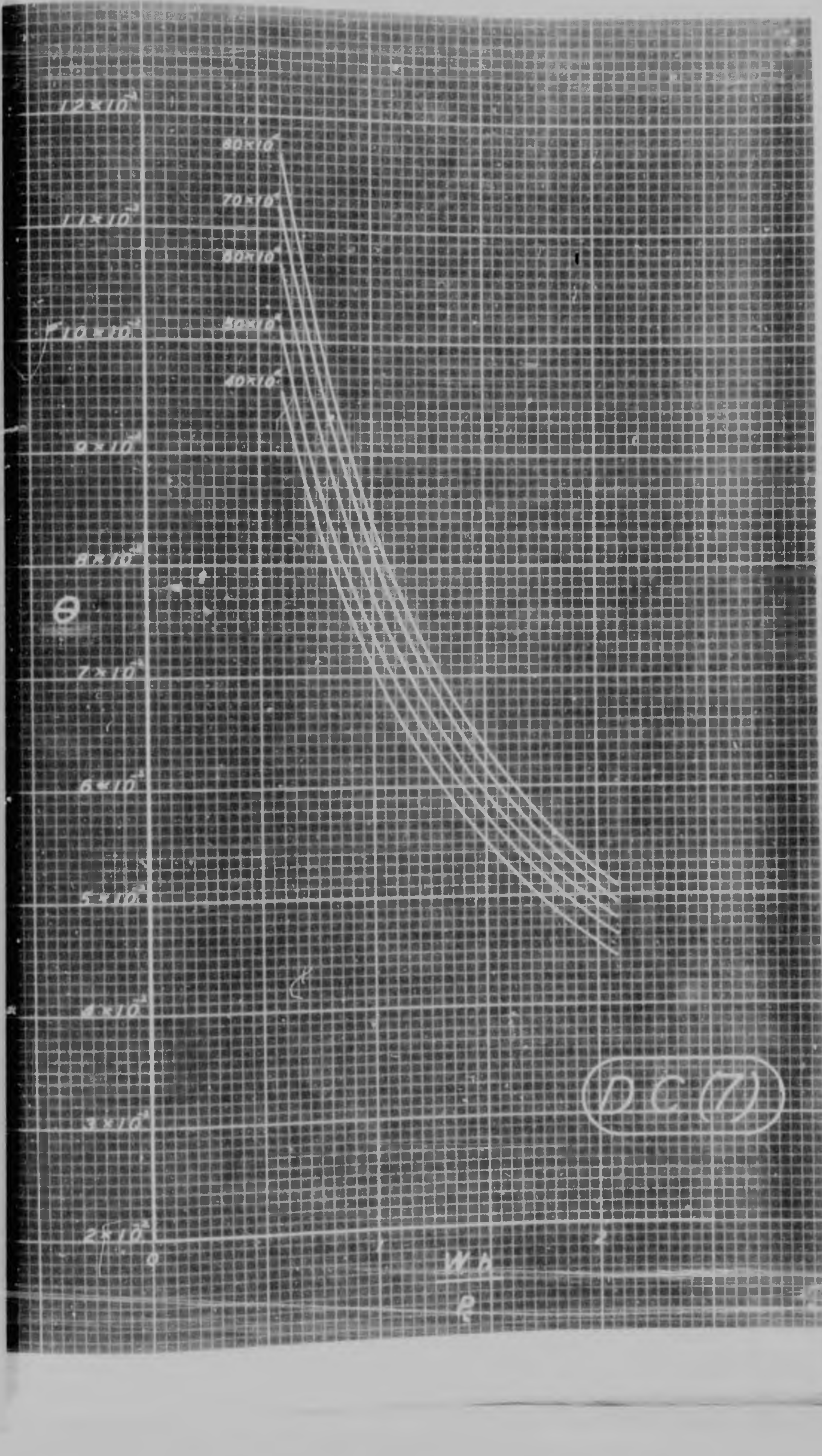
It should be noted that the expression for the cyclic period T, consists of 2 factors. The first,

$$\left[\frac{W\, L_E\, L_c\, m_1}{2g\, P_o}\right]^{\frac{1}{2}}$$

depends only on the dimensions of the pump and the height of the barometer. The second factor, θ, depends only on the head in atmospheres and the fuel mixture strength used. It should also be noted that the period is independent of the cross-sectional area of the combustion chamber.

An examination of D.C. (7) will show that the fuel mixture strength has only a small effect on the period. This might be expected from the fact that the oscillations of the water column may be likened to that of a pendulum, the period of which, for small oscillations, is practically independent of the amplitude. The richer fuel mixture strengths give rise to a greater amplitude of vibration (From D.C. (4) it is seen that $\frac{V_{E1}}{V_c}$ increases with increase of C.V.) but have only a small effect on the period of the cycle. The curves are hyperbolic in shape, indicating that for unity compression ratio, the period would be infinite, an indication which is obvious, since under these

conditions/.....


12×10
11×10
10×10
9×10
8×10
Θ
7×10
6×10
5×10
4×10
3×10
2×10
80×10
70×10
60×10
50×10
40×10
0
1
2
Wh
k
DC (7)

conditions, the head of water is zero and the pump does not function. For infinite compression, the period would obviously be zero.

The ratio $\frac{\theta_{c3}}{\theta}$ gives the fraction of the period devoted to admission of mixture. This quantity is of use in calculating the gas velocities during admission in order to calculate suitable sizes of admission ports.

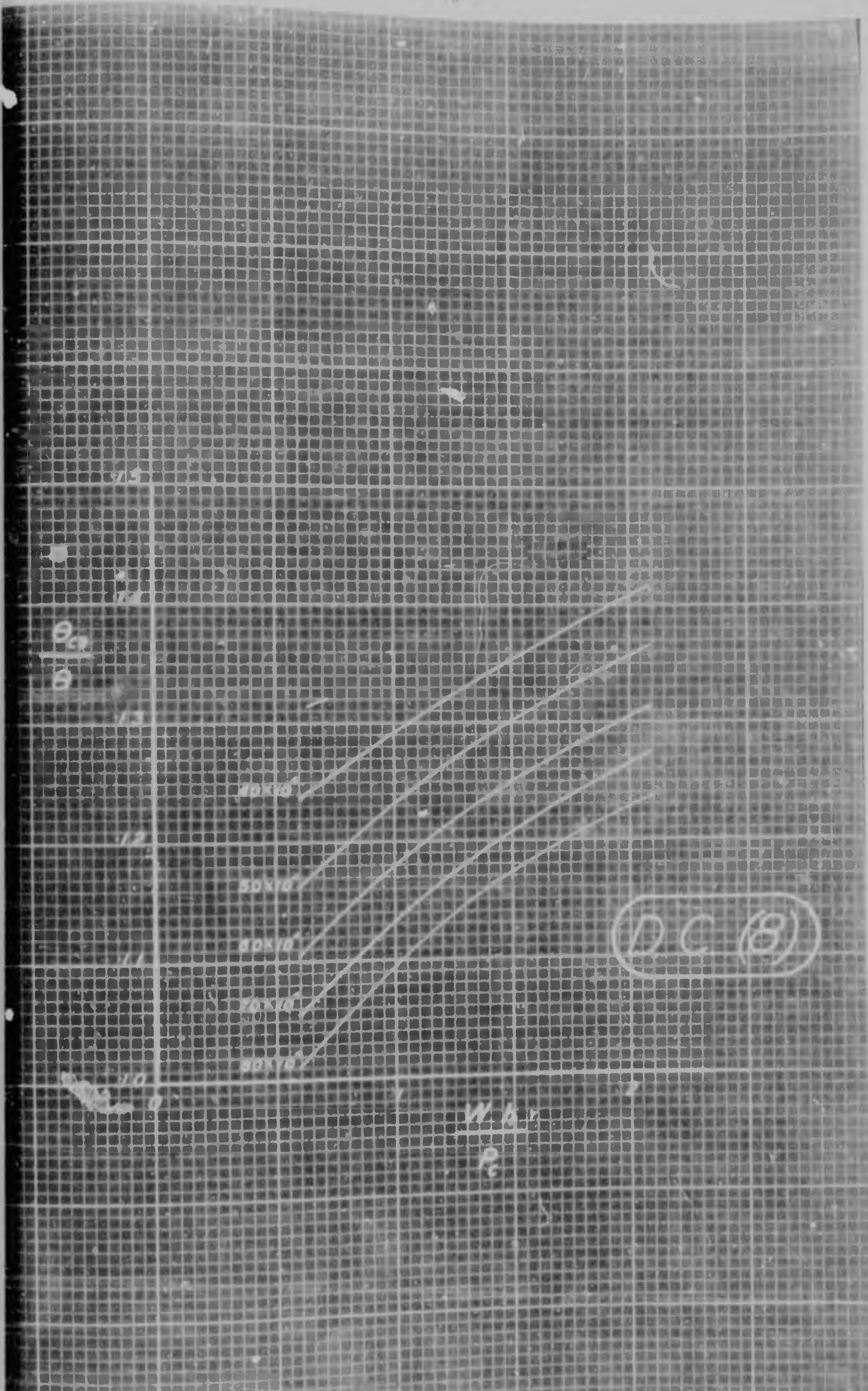
In Design Chart 8 (D.C. (8)) this ratio is plotted to a base of water head in atmospheres. An examination of this chart shows that $\frac{\theta_{c3}}{\theta}$ varies only through a small range of values.

COMPUTATION OF θ_{co}.

In order to compute this quantity, Chart θ_{co} was drawn. Curve E is a curve of internal energy (E) per lb. of mixture drawn to a base of relative volumes $(\frac{V}{V_c})$. The internal energy at C, E_c, was chosen as datum. The various points on this curve were obtained from the data sheet for Chart θ_{34}. ($T_3 = T_c = 300^{\circ}$C (abs.). Values of $\frac{V}{V_c}(\frac{1}{r})$ for the values of r considered were chosen and the corresponding points on curve E joined by straight lines H_2, H_3, etc. to the point (1, 0). These lines correspond to the line CO in Fig. 16, and are clearly of slope m_2, m_3, etc.

The intercepts E^1 between these lines were measured for various values of $\frac{V}{V_c}$. The vorresponding values of $\frac{1}{\sqrt{E^1}}$ were then calculated and plotted to the same base as the E curve. The curves are clearly asymptotic at $\frac{V}{V_c} = 1$ and $\frac{V}{V_c} = \frac{1}{r}$ and hence it is not possible to plot points in the immediate neighbourhoods of these points. In order to overcome this difficulty the curve E was approximated to a straight line in the vicinity of these points. The slopes of these lines were taken as the slope of the curve E at the points $\frac{V}{V_c} = 1$, and $\frac{V}{V_c} = \frac{1}{r}$.

Now/.....


1.5
1.4
1.3
1.2
1.1
1.0
0
1
2
40x10
50x10
60x10
70x10
80x10
DC (B)
θ
θ
Wb
P_c

Now, since $E = \int PdV$, the slope of curve E at any point =

$$\frac{d}{d(\frac{V}{V_c})}\left[\int PdV\right]$$

$$= P\,V_c$$

$$= \frac{P\,V_c}{P_cV_c}\cdot P_cV_c$$

$$= \frac{P}{P_c}\cdot m_1 = m \text{ (say)}.$$

Consider first the neighbourhood $\frac{V}{V_c} = 1$. Here $P = P_c$ and $m = m_1$, the slope of the atmospheric line.

Hence we can express

$$E^1 = (m_r - m_1)\left[\frac{V_c - V}{V_c}\right]$$

in the neighbourhood of $\frac{V}{V_c} = 1$.

In the vicinity of $\frac{V}{V_c} = \frac{1}{r}$, $m = \frac{P_o}{P_c}\cdot m_1$,

and ∴ $E^1 = (\frac{P_o}{P_c}\cdot m_1 - m_r)(\frac{V - V_o}{V_c})$.

The areas under the curves $\frac{1}{\sqrt{E^1}}$ V. $\frac{V}{V_c}$ were then accurately measured by a planimeter and yielded values of θ_{co} $[\theta^{1}_{co}]$ except for the integrals in the vicinity of $\frac{V}{V_c} = 1$ and $\frac{V}{V_c} = \frac{1}{r}$.

These integrals were obtained as follows:

If the small range of $\frac{V}{V_c}$ considered, be denoted by $\Delta_c\frac{V}{V_c}$ in the neighbourhood of $\frac{V}{V_c} = 1$ and by $\Delta_o\frac{V}{V_c}$ in the neighbourhood of $\frac{V}{V_c} = \frac{1}{r}$, then $\Delta_c\theta_{co}$ the value of the integral in the neighbourhood of $\frac{V}{V_c} = 1$:-

$$\Delta_c\theta_{co} = \int_0^{c(\frac{V}{V_c})}\frac{d(\frac{V_c - V}{V_c})}{\left[(m_r - m_1)\,\frac{V_c - V}{V_c}\right]^{\frac{1}{2}}}$$

$$= \frac{2(\Delta_c\,\frac{V}{V_c})^{\frac{1}{2}}}{(m_r - m_1)^{\frac{1}{2}}}$$

and/.....

and $\Delta_o \theta_{co}$, the value of the integral in the neighbourhood of $\frac{V}{V_c} = \frac{1}{r}$:-

$$\Delta_o \theta_{co} = \int_{o}^{o\left(\frac{V}{V_c}\right)} \frac{d\left(\frac{V - V_o}{V_c}\right)}{\left[\left(\frac{P_o}{P_c} m_1 - m_r\right) \frac{V - V_o}{V_c}\right]^{\frac{1}{2}}}$$

$$= \frac{2\left[\Delta_o \left(\frac{V}{V_c}\right)\right]^{\frac{1}{2}}}{\left[\frac{P_o}{P_c} m_1 - m_r\right]^{\frac{1}{2}}}$$

DATA SHEET FOR CHART θ_{co}.

$(E - E_c) \times 10^{-4}$	$\frac{V}{V_c}$	$(E - E_c) \times 10^{-4}$	$\frac{V}{V_c}$
0	1.000	4.891	0.273
0.367	0.884	5.275	0.251
0.735	0.784	5.665	0.233
1.105	0.700	6.054	0.216
1.477	0.628	6.444	0.200
1.851	0.565	6.835	0.187
2.225	0.510	7.228	0.174
2.601	0.465	7.628	0.162
2.979	0.428	8.028	0.151
3.359	0.384	8.427	0.142
3.739	0.351	8.812	0.133
4.120	0.322	9.227	0.125
4.505	0.296		

Curves/.....

Curves $\frac{1}{\sqrt{E'}}$ v. $\frac{V}{V_c}$.

$\frac{V}{V_c}$	$\frac{1}{\sqrt{E'}} \times 10^2$				
	r = 2	3	4	5	6
0.99	5.76	5	4.26	4.08	3.78
0.98	5	4.08	3.33	3.015	2.885
0.96	3.65	2.885	2.425	2.18	2.04
0.94	3.16	2.355	2	1.767	1.665
0.92	2.775	2.06	1.74	1.562	1.443
0.90	2.5	1.875	1.58	1.40	1.308
0.85	2.135	1.561	1.29	1.156	1.066
0.8	1.925	1.386	1.14	1.04	0.94
0.7	1.857	1.222	0.98	0.87	0.795
0.6	2.135	1.17	0.913	0.79	0.718
0.55	2.775	-	-	-	-
0.53	3.535	-	-	-	-
0.52	3.92	-	-	-	-
0.51	5	-	-	-	-
0.5	-	1.26	0.705	0.767	0.683
0.45	-	1.373	-	-	-
0.4	-	1.666	0.966	0.773	0.676
0.38	-	1.89	-	-	-
0.36	-	2.39	-	-	-
0.35	-	2.885	1.065	-	-
0.34	-	4.47	-	-	-
0.3	-	-	1.375	0.905	0.738
0.28	-	-	1.74	-	-
0.26	-	-	2.885	-	-
0.25	-	-	-	1.162	0.847
0.24	-	-	-	1.27	-
0.23	-	-	-	1.415	-
0.21	-	-	-	2.293	-
0.20	-	-	-	-	1.207
0.19	-	-	-	-	1.413
0.18	-	-	-	-	1.825
0.17	-	-	-	-	3.53

Cont.

COMPUTED VALUES OF θ_{co}

$\Delta_c \left(\frac{V}{V_c}\right)$ = 0.01 in each case.

$\Delta_o \frac{V}{V_c}$ = 0.01 in each case.

r	2	3	4	5	6
$\theta^1_{co} \times 10^2$	1.2328	1.0845	0.9601	0.8300	0.8092
$\Delta_c\theta_{co} \times 10^2$	0.1525	0.1141	0.0980	0.0876	0.0816
$\Delta_o\theta_{co} \times 10^2$	0.1163	0.0858	0.0035	0.0518	0.0442
$\theta_{co} \times 10^2$	1.5016	1.2844	1.1216	0.9694	0.9350

COMPUTATION OF θ_{1E}

θ_{1E} was also deduced graphically. For this purpose, the Cuarts θ_{2-1E}, θ_{3-1E}, etc. corresponding to r = 2, 3, etc. were drawn. The curves E_{40}, E_{50}, etc. correspond to the curve 1E in Fig. 16 for C.V. corresponding to 40 x 10^4, 50 x 10^5, etc. ft. lbs./lb. The points on these curves were obtained from Chart G. The range of $\frac{V}{V_c}$ is from $\frac{V_1}{V_c}$ to $\frac{V_E}{V_c}$. The ordinates are internal energy (E) per lb. of mixture the datum being chosen at 0°C (abs.). Lines H_2, H_3, etc. of slope m_2, m_3, etc. were then drawn through the points $(\frac{V_1}{V_c}, E_1)$. These lines clearly correspond to line $1E_1$ in Fig. 16. The intercepts between "H" lines and the corresponding E curves were measured for various values of $\frac{V}{V_c}$ and the corresponding values of $\frac{1}{\sqrt{E^1}}$ deduced. Curves $\frac{1}{\sqrt{E^1_{40}}}$, $\frac{1}{\sqrt{E^1_{50}}}$, etc. are then drawn to a base of $\frac{V}{V_c}$. The areas under these curves yield the values of θ_{1E} required. Since $E^1_1 = 0$, the curves will be asymptotic at $\frac{V}{V_c} = \frac{V_1}{V_c}$. Making the same assumptions as in the case of θ_{co}, the value of the integral over a range of $\frac{V}{V_c} = \Delta_1 \frac{V}{V_c}$ in this neighbourhood

$$\Delta_1\theta_{1E} = \frac{2\left(\Delta_1 \frac{V}{V_c}\right)^{\frac{1}{2}}}{\left(m_1 \cdot \frac{P_1}{P_c} - m_r\right)^{\frac{1}{2}}}$$

The/.....

COMPUTED VALUES OF θ_{co}

$\Delta_c (\frac{V}{V_c})$ = 0.01 in each case.

$\Delta_o \frac{V}{V_c}$ = 0.01 in each case.

r	2	3	4	5	6
$\theta^1_{co} \times 10^2$	1.2328	1.0845	0.9601	0.8300	0.8092
$\Delta_c\theta_{co} \times 10^2$	0.1525	0.1141	0.0980	0.0876	0.0816
$\Delta_o\theta_{co} \times 10^2$	0.1163	0.0858	0.0035	0.0518	0.0442
$\theta_{co} \times 10^2$	1.5016	1.2844	1.1216	0.9694	0.9350

COMPUTATION OF θ_{1E}

θ_{1E} was also deduced graphically. For this purpose, the Charts θ_{2-1E}, θ_{3-1E}, etc. corresponding to r = 2, 3, etc. were drawn. The curves E_{40}, E_{50}, etc. correspond to the curve 1E in Fig. 16 for C.V. corresponding to 40 x 10^4, 50 x 10^5, etc. ft. lbs./lb. The points on these curves were obtained from Chart G. The range of $\frac{V}{V_c}$ is from $\frac{V_1}{V_c}$ to $\frac{V_E}{V_c}$. The ordinates are internal energy (E) per lb. of mixture the datum being chosen at 0°C (abs.). Lines H_2, H_3, etc. of slope m_2, m_3, etc. were then drawn through the points ($\frac{V_1}{V_c}$, E_1). These lines clearly correspond to line $1E_1$ in Fig. 16. The intercepts between "H" lines and the corresponding E curves were measured for various values of $\frac{V}{V_c}$ and the corresponding values of $\frac{1}{\sqrt{E^1}}$ deduced. Curves $\frac{1}{\sqrt{E^1_{40}}}$, $\frac{1}{\sqrt{E^1_{50}}}$, etc. are then drawn to a base of $\frac{V}{V_c}$. The areas under these curves yield the values of θ_{1E} required. Since $E^1_1 = 0$, the curves will be asymptotic at $\frac{V}{V_c} = \frac{V_1}{V_c}$. Making the same assumptions as in the case of θ_{co}, the value of the integral over a range of $\frac{V}{V_c} = \Delta_1 \frac{V}{V_c}$ in this neighbourhood

$$\Delta_1\theta_{1E} = \frac{2\left(\Delta_1 \frac{V}{V_c}\right)^{\frac{1}{2}}}{\left(m_1 \cdot \frac{P_1}{P_c} - m_r\right)^{\frac{1}{2}}}$$

The/.....

The areas under these curves except in the neighbourhood of $\frac{V_1}{V_c}$ were carefully measured by planimeter and yielded integrals θ^1_{1E}. θ_{1E} the total integral is equal to $\theta^1_{1E} + \Delta_1 \theta_{1E}$.

DATA SHEET FOR CHART $\theta_{2.1E}$.

(r = 2).

Values of E multiplied by 10^{-4}.

$\frac{V}{V_1}$	$\frac{V}{V_c}$	E_{40}	E_{50}	E_{60}	E_{70}	E_{80}
1	0.5	49.35	59.35	69.35	79.35	89.35
1.2	0.6	46.3	55.9	65.7	75.2	84.7
1.4	0.7	44.0	53	62.3	71.4	80.6
1.7	0.85	41.1	49.6	58.4	67.2	75.7
2	1	38.7	46.9	55.3	63.7	72.0
3	1.5	33.6	40.9	48.45	55.9	63.8
4	2	30.1	36.9	44	50.7	57.8
5	2.5	27.8	34.1	40.6	47.0	53.8
6	3	26.0	31.9	38.1	44.0	50.4
6.54	3.27	2.51	-	-	-	-
7	3.5	-	30.2	36	41.8	47.8
7.5	3.75	-	29.3	-	-	-
8	4	-	-	34.3	40	45.6
3.32	4.16	-	-	33.8	-	-
9	4.5	-	-	-	-	43.6
9.3	4.65	-	-	-	37.8	-
10.06	5.03	-	-	-	-	42

Curves/...

Curves $\frac{1}{\sqrt{E'}}$ v. $\frac{V}{V_c}$

$\frac{V}{V_c}$	$\frac{1}{\sqrt{E'}} \times 10^2$				
	C.V. = 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
0.6	0.633	0.577	0.546	0.513	0.477
0.7	0.472	0.430	0.403	0.385	0.356
0.8	0.408	0.371	0.344	0.326	0.304
0.9	0.370	0.339	0.312	0.295	0.276
1.0	0.346	0.315	0.292	0.274	0.259
1.25	0.315	0.285	0.263	0.246	0.233
1.5	0.299	0.269	0.249	0.231	0.217
1.75	0.291	0.258	0.239	0.221	0.208
2	0.286	0.253	0.233	0.215	0.202
2.5	0.286	0.249	0.227	0.208	0.195
3	0.292	0.251	0.225	0.206	0.191
3.27	0.296	-	-	-	-
3.5	-	0.255	0.227	0.206	0.190
3.75	-	0.257	-	-	-
4	-	-	-	0.207	0.190
4.16	-	-	0.231	-	-
4.5	-	-	-	0.211	0.191
4.65	-	-	-	-	-
5.03	-	-	-	-	0.194

Date/....

DATA SHEET FOR CHART $\theta_{3.1E'}$

(r = 3).

Value of E multiplied by 10^{-4}.

$\frac{V}{V_1}$	$\frac{V}{V_c}$	E_{40}	E_{50}	E_{60}	E_{70}	E_{80}
1	0.333	51	61	71	81	91
1.2	0.4	47.9	57.7	67	76.6	86.1
1.5	0.5	44.3	53.3	62.2	71.2	80.3
1.8	0.6	41.6	50	58.4	67.2	75.8
2.1	0.7	39.5	47.5	55.6	64	72.2
2.4	0.8	37.6	45.4	53.4	61.1	69.4
2.7	0.9	36.1	43.7	51.2	59.1	66.8
3	1	34.9	42	49.5	57	64.6
3.6	1.2	32.6	39.4	46.4	53.6	61
4.2	1.4	30.7	37.4	44	50.9	58
5.4	1.8	28	34.2	40.2	46.6	53.2
6.6	2.2	25.9	31.8	37.5	43.3	49.6
7.8	2.6	24.3	29.8	35.4	41	47
9	3	-	28.2	33.5	39	44.6
9.04	3.01	23.0	-	-	-	-
10.20	3.4	-	-	32.1	37.2	42.8
10.32	3.44	-	26.8	-	-	-
11.4	3.8	-	-	-	35.8	41.2
11.6	3.87	-	-	30.6	-	-
12.6	4.2	-	-	-	-	40
12.84	4.28	-	-	-	34.2	-
14.1	4.70	-	-	-	-	38.3

Curves $\frac{1}{\sqrt{E^1}}$ v $\frac{V}{V_c}$.

$\frac{V}{V_c}$	$\frac{1}{\sqrt{E^1}} \times 10^2$				
	C.V. = 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
0.4	0.609	0.568	0.527	0.491	0.481
0.45	0.485	0.445	0.410	0.392	0.369
0.5	0.421	0.386	0.358	0.341	0.320
0.6	0.360	0.327	0.303	0.287	0.271
0.8	0.308	0.280	0.259	0.244	0.229
1	0.287	0.258	0.239	0.224	0.212
1.25	0.272	0.246	0.225	0.210	0.199
1.5	0.266	0.238	0.217	0.203	0.191
2	0.265	0.234	0.212	0.195	0.183
2.5	0.272	0.236	0.212	0.194	0.181
3	0.287	0.243	0.215	0.195	0.181
3.44	-	0.252	-	-	-
3.5	-	-	0.221	0.198	0.183
3.867	-	-	0.2267	-	-
4	-	-	-	0.204	0.187
4.28	-	-	-	0.207	-
4.7	-	-	-	-	0.194

Data/...

DATA SHEET FOR CHART $\theta_{4.1E}$.

Values of E multiplied by 10^4.

$\frac{V}{V_1}$	$\frac{V}{V_c}$	E_{40}	E_{50}	E_{60}	E_{70}	E_{80}
1	0.25	52.35	62.35	72.35	82.35	92.35
1.2	0.3	49.4	58.8	68.5	78	87.6
1.6	0.4	44.6	53.2	62.2	70.9	79.9
2	0.5	41.2	49.3	57.7	66.2	74.6
2.4	0.6	38.8	46.4	54.4	62.2	70.6
2.8	0.7	36.7	44	51.7	69.3	67.4
3.2	0.8	35	42	49.3	56.6	64.6
3.6	0.9	33.6	40.4	47.4	54.5	62.2
4	1	32.2	38.9	45.7	52.6	60
5	1.25	29.8	35.9	42.3	48.6	55.6
6	1.5	27.7	33.6	39.6	45.6	52.2
8	2	24.8	30.1	35.8	41.2	47.4
10	2.5	22.8	27.8	33	38.1	44
11.5	2.875	21.7	-	-	-	-
12.	3	-	26	30.9	35.8	41.2
13.1	3.275	-	25.2	-	-	-
14	3.5	-	-	-	33.8	39
14.65	3.66	-	-	28.8	-	-
16	4	-	-	-	-	37.4
16.3	4.075	-	-	-	32	-
17.8	4.45	-	-	-	-	36.1

/Curves...

CURVES $\frac{1}{\sqrt{\varepsilon'}}$ v $\frac{V}{V_c}$.

$\frac{V}{V_c}$	$\frac{1}{\sqrt{\varepsilon'}} \times 10^2$				
	C.V. = 40 x 10^4	50 x 10^4	60 x 10^4	70 x 10^4	80 x 10^4
0.35	0.452	0.413	0.408	0.365	0.349
0.4	0.389	0.356	0.338	0.311	0.298
0.5	0.329	0.297	0.282	0.261	0.251
0.6	0.398	0.271	0.254	0.237	0.228
0.8	0.271	0.242	0.229	0.214	0.205
1.0	0.259	0.234	0.215	0.202	0.192
1.25	0.252	0.226	0.208	0.193	0.182
1.5	0.250	0.223	0.203	0.188	0.178
2	0.254	0.222	0.202	0.185	0.174
2.5	0.264	0.228	0.204	0.186	0.174
2.835	0.277	-	-	-	-
3	-	0.239	0.211	0.190	0.176
3.27	-	0.247	-	-	-
3.5	-	-	0.219	0.195	0.18
3.66	-	-	0.223	-	-
4	-	-	-	-	0.185
4.075	-	-	-	0.204	-
4.45	-	-	-	-	0.191

/Data.....

DATA SHEET FOR CHART θ 5.1E.

Values of E multiplied by 10^{-4}.

$\frac{V}{V_1}$	$\frac{V}{V_c}$	E_{40}	E_{50}	E_{60}	E_{70}	E_{80}
1	0.2	53.52	63.52	73.52	83.52	93.52
1.5	0.3	40.5	55.3	64.8	73.5	82.6
2	0.4	42	50.2	58.8	67.3	75.4
2.5	0.5	39	46.6	54.7	62.6	70.6
3	0.6	36.5	43.8	51.6	59.1	66.9
3.5	0.7	34.6	41.5	49	56.1	63.5
4	0.8	33.0	39.7	46.7	53.6	60.8
4.5	0.9	31.6	37.9	44.8	51.5	58.3
5	1.0	30.4	36.7	43.3	49.5	56.4
6	1.2	28.3	34.2	40.0	46.6	53.0
8	1.6	25.4	30.8	36.7	42.2	48.1
10	2	23.3	28.3	33.8	39	44.6
12	2.4	-	26.4	31.6	36.6	41.8
13.8	2.76	20.7	-	-	-	-
14	2.8	-	2[illegible]	30	34.6	39.6
15.7	3.14	-	24	-	-	-
16	3.2	-	-	28.5	33.1	37.9
17.6	3.52	-	-	27.5	-	-
18	3.6	-	-	-	31.6	36.3
19.4	3.88	-	-	-	30.8	-
20	4	-	-	-	-	35 0
21.23	4.25	-	-	-	-	34.2

Curves/.....

[illegible] $\frac{1}{\sqrt{E^1}}$ v $\frac{V}{V_c}$.

$\frac{V}{V_c}$	$\frac{1}{\sqrt{E^1}} \times 10^{[illegible]}$				
	S.V. = 40 x 10^4	50 x 10^4	60 x 10^4	70 x 10^4	80 x 10^4
0.3	[illegible]	0.374	0.352	0.332	0.316
0.4	0.321	0.29[illegible]	0.278	0.267	0.250
0.5	0.296	0.264	0.257	0.233	0.225
0.6	0.272	0.247	0.232	0.216	0.206
0.8	0.253	0.231	0.214	0.198	0.190
1	0.24[illegible]	0.222	0.206	0.192	0.181
1.25	0.24[illegible]	0.217	0.201	0.186	0.175
1.5	0.243	0.216	0.198	0.183	0.172
2	[illegible]	0.221	0.199	0.183	0.171
2.5	0.272	[illegible]	0.205	0.187	0.172
2.76	0.28[illegible]	-	-	-	-
3	-	-	0.214	0.193	0.177
3.14	[illegible]	0.252	-	-	-
3.5	-	-	-	0.201	0.183
3.52	-	-	0.235	-	-
3.88	-	-	-	0.209	-
4.25	-	-	-	-	0.194

Data/.....

DATA SHEET FOR CHART $\theta_{6.1E}$.

Values of E multiplied by 10^{-4}.

$\frac{V}{V_1}$	$\frac{V}{V_c}$	E_{40}	E_{50}	E_{60}	E_{70}	E_{80}
1	0.167	54.5	64.5	74.5	84.5	94.5
1.2	0.2	51.4	61.0	70.5	80.0	89.6
1.8	0.3	44.5	53.0	61.8	70.2	79.5
2.4	0.4	40.4	48.4	56.2	64.2	72.4
3	0.5	37.3	44.6	52.0	59.8	67.5
3.6	0.6	35.0	4[illegible]	49.0	56.2	63.7
4.8	0.8	31.0	[illegible]	44.3	51	57.8
6	1	28.[illegible]	35.0	41.1	47.3	53.6
7.2	1.2	27.[illegible]	32.8	38.6	44.4	50.9
9	1.5	24.8	30.1	35.5	41.1	46.6
12	2.1	22.0	27.1	32.0	37.1	42.3
15	2.5	-	25.0	29.4	34.3	39.2
16.2	2.7	20.1	-	-	-	-
18	3.0	-	-	27.6	32.2	36.7
18.65	3.11	-	23.2	-	-	-
20.45	3.41	-	-	26.4	-	-
21	3.5	-	-	-	40.4	34.
22.57	3.76	-	-	-	29.7	-
24.7	4.12	-	-	-	-	32.8

Curves/.....

CURVES $\frac{1}{\sqrt{E^1}}$ v $\frac{V}{V_c}$.

$\frac{V}{V_c}$	$\frac{1}{\sqrt{E^1}} \times 10^2$				
	C.V. = 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
0.25	0.401	0.386	0.386	0.331	0.316
0.3	0.337	0.320	0.305	0.276	0.268
0.4	0.289	0.265	0.248	0.234	0.226
0.5	0.267	0.244	0.226	0.214	0.206
0.6	0.254	0.232	0.215	0.203	0.194
0.8	0.241	0.220	0.203	0.196	0.180
1.0	0.236	0.214	0.197	0.184	0.173
1.25	0.235	0.211	0.194	0.180	0.169
1.5	0.238	0.212	0.193	0.179	0.167
2	0.254	0.219	0.196	0.180	0.167
2.5	-	0.232	0.204	0.185	0.170
2.7	0.293	-	-	-	-
3	-	-	0.215	0.192	0.175
3.1	-	0.256	-	-	-
3.41	-	-	0.229	-	-
3.5	-	-	-	0.203	0.183
3.76	-	-	-	0.209	-
4.12	-	-	-	-	0.195

Computed/.....

COMPUTED VALUES OF θ_{1E}.

$\Delta_1 (\frac{V}{V_c})$ = 0.1 in each case.

r = 2.

C.V. x 10^{-4}	$\theta^1_{1E} \times 10^2$	$\Delta_1\theta_{1E} \times 10^2$	$\theta_{1E} \times 10^2$
40	0.8880	0.1180	1.0060
50	0.9154	0.1077	1.0231
60	0.9401	0.1003	1.0404
70	0.9714	0.0944	1.0658
80	0.9829	0.0901	1.0730

r = 3.

C.V. x 10^{-4}	$\theta^1_{1E} \times 10^2$	$\Delta_1\theta_{1E} \times 10^2$	$\theta_{1E} \times 10^2$
40	0.7775	0.0921	0.8696
50	0.8027	0.0861	0.8888
60	0.8264	0.0806	0.9070
70	0.8525	0.0751	0.9276
80	0.8741	0.0724	0.9465

r = 4.

C.V. x 10^{-4}	$\theta^1_{1E} \times 10^2$	$\Delta_1\theta_{1E} \times 10^2$	$\theta_{1E} \times 10^2$
40	0.6992	0.0794	0.7786
50	0.7322	0.0735	0.8057
60	0.7562	0.0682	0.8244
70	0.7794	0.0650	0.8444
80	0.8085	0.0702	0.8787

r = 5.

C.V. x 10^{-4}	$\theta^1_{1E} \times 10^2$	$\Delta_1\theta_{1E} \times 10^2$	$\theta_{1E} \times 10^2$
40	0.6627	0.0700	0.7327
50	0.6917	0.0650	0.7567
60	0.7244	0.0610	0.78536
70	0.7387	0.0580	0.7967
80	0.7602	0.0553	0.8155

r = 6/.....

r = 6.

C.V. x 10^{-4}	$\theta^{1}_{1E} \times 10^2$	$\Delta_1 \theta_{1E} \times 10^2$	$\theta_{1E} \times 10^2$
40	0.6540	0.0632	0.7172
50	0.6816	0.0587	0.7403
60	0.6843	0.0552	0.7395
70	0.7041	0.0524	0.7565
80	0.7297	0.0500	0.7797

COMPUTATION OF θ_{EE_1}.

This integral is easily obtained analytically, because E^1 for this range may be expressed as a function of $\frac{V}{V_c}$ (Fig. 16).

$$\theta_{EE_1} = \int_0^{\frac{V_{E_1} - V_E}{V_c}} \frac{d\left(\frac{V}{V_c} - \frac{V_E}{V_c}\right)}{\left[(m_r - m_1)\left(\frac{V}{V_c} - \frac{V_E}{V_c}\right)\right]^{\frac{1}{2}}}$$

$$= \frac{2\left(\frac{V_{E_1} - V_E}{V_c}\right)^{\frac{1}{2}}}{(m_r - m_1)^{\frac{1}{2}}}$$

COMPUTED VALUES OF θ_{EE_1}.

C.V. x 10^{-4}	$\theta_{EE_1} \times 10^2$				
	r = 2	3	4	5	6
40	3.935	2.290	1.728	1.338	1.132
50	4.500	2.600	1.938	1.520	1.294
60	5.030	2.895	2.140	1.170	1.453
70	5.502	3.185	2.340	1.827	1.585
80	5.980	3.385	2.500	1.966	1.705

Computation/.....

r = 6.

C.V. x 10^{-4}	$\theta^{1}_{1E} \times 10^2$	$\Delta_1 \theta_{1E} \times 10^2$	$\theta_{1E} \times 10^2$
40	0.6540	0.0632	0.7172
50	0.6816	0.0587	0.7403
60	0.6843	0.0552	0.7395
70	0.7041	0.0524	0.7565
80	0.7297	0.0500	0.7797

COMPUTATION OF θ_{EE_1}.

This integral is easily obtained analytically, because E^1 for this range may be expressed as a function of $\frac{V}{V_c}$ (Fig. 16).

$$\theta_{EE_1} = \int_0^{\frac{V_{E_1} - V_E}{V_c}} \frac{d\left(\frac{V}{V_c} - \frac{V_E}{V_c}\right)}{\left[(m_r - m_1)\left(\frac{V}{V_c} - \frac{V_E}{V_c}\right)\right]^{\frac{1}{2}}}$$

$$= \frac{2\left(\frac{V_{E_1} - V_E}{V_c}\right)^{\frac{1}{2}}}{(m_r - m_1)^{\frac{1}{2}}}$$

COMPUTED VALUES OF θ_{EE_1}.

C.V. x 10^{-4}	$\theta_{EE_1} \times 10^2$				
	r = 2	3	4	5	6
40	3.935	2.290	1.728	1.338	1.132
50	4.500	2.600	1.938	1.520	1.294
60	5.030	2.895	2.140	1.170	1.453
70	5.502	3.185	2.340	1.827	1.585
80	5.980	3.385	2.500	1.966	1.705

Computation/.....

COMPUTATION OF θ_{C3}.

This integral is also easily obtained analytically (Ref. Fig. 10).

$$\theta_{C3} = \int_{1}^{0.6} \frac{d\left(1-\frac{V}{V_c}\right)}{\left[(m_r - m_1)\left(1-\frac{V}{V_c}\right)\right]^{\frac{1}{2}}}$$

$$= \frac{2\,(0.6)^{\frac{1}{2}}}{(m_r - m_1)^{\frac{1}{2}}}$$

COMPUTED VALUES OF θ_{C3}.

r	$\theta_{C3} \times 10^2$
2	1.180
3	0.886
4	0.756
5	0.676
6	0.630

COMPUTATION OF θ_{34}.

Chart θ_{34}, the construction of part of which was described previously, is also used in the computation of θ_{34}. The intercepts between the "H" lines and the curve E were measured for various values of $\frac{V}{V_c}$ and the corresponding values of $\frac{1}{\sqrt{E^1}}$ calculated. These values were plotted to a base of $\frac{V}{V_c}$ (Curves $\frac{1}{\sqrt{E_2^1}}$ etc.) and the area under these curves measured (except in the vicinity of $\frac{V_4}{V_3}$ where the curves are asymptotic) thus yielding values of θ^1_{34}.

The integral $\Delta_4\theta_{34}$ over the range $\Delta_4\left(\frac{V}{V_c}\right)$ in the

vicinity/.....

vicinity of $\frac{V_4}{V_c}$ is approximately given by the expression:

$$\Delta_4 \theta_{34} = \frac{2\left[\Delta_4 \left(\frac{V}{V_c}\right)\right]^{\frac{1}{2}}}{\left(m_1 \cdot \frac{P_4}{P_c} - m_r\right)^{\frac{1}{2}}}$$

(Refer to computation of $\Delta_1 \theta_{1E}$ etc.)

It is clear that $\theta_{34} = \theta^1_{34} + \Delta_4 \theta_{34}$.

DATA SHEET FOR CHART θ_{34}.

$\frac{V}{V_c}$	$\frac{1}{\sqrt{g^1}} \times 10^2$.				
	r = 2	3	4	5	6
0.4	0.985	0.739	0.630	0.570	0.526
0.35	0.953	0.744	0.608	0.548	0.506
0.3	0.935	0.695	0.590	0.631	0.490
0.25	0.948	0.686	0.578	0.519	0.478
0.2	0.994	0.691	0.577	0.514	0.474
0.15	1.118	0.72	0.586	0.518	0.474
0.1	1.875	0.818	0.630	0.543	0.490
0.095	2.18	-	-	-	-
0.09	2.885	-	-	-	-
0.08	-	0.932	0.680	0.570	0.510
0.06	-	1.290	0.774	0.620	0.550
0.055	-	1.542	-	-	-
0.04	-	-	1.18	0.758	0.624
0.035	-	-	1.458	-	-
0.03	-	-	-	0.980	0.738
0.025	-	-	-	-	0.848

Computed/......

vicinity of $\frac{V_4}{V_c}$ is approximately given by the expression:

$$\Delta_4 \theta_{34} = \frac{2\left[\Delta_4 \left(\frac{V}{V_c}\right)\right]^{\frac{1}{2}}}{\left(m_1 \cdot \frac{P_4}{P_c} - m_r\right)^{\frac{1}{2}}}$$

(Refer to computation of $\Delta_1 \theta_{1E}$ etc.)

It is clear that $\theta_{34} = \theta^1_{34} + \Delta_4 \theta_{34}$.

DATA SHEET FOR CHART θ_{34}.

$\frac{V}{V_c}$	$\frac{1}{\sqrt{E^1}} \times 10^2$.				
	r = 2	3	4	5	6
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0.08	-	0.932	0.680	0.570	0.510
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0.055	-	1.542	-	-	-
0.04	-	-	1.18	0.758	0.624
0.035	-	-	1.458	-	-
0.03	-	-	-	0.980	0.738
0.025	-	-	-	-	0.848

Computed/......

COMPUTED VALUES OF θ_{34}.

$\Delta_4 (\frac{V}{V_c}) = \frac{1}{100}$ in each case.

r	$\theta^1_{34} \times 10^2$	$\Delta_4 \theta_{34} \times 10^2$	$\theta_{34} \times 10^2$
2	0.3238	0.0474	0.3712
3	0.2625	0.0276	0.2901
4	0.2296	0.0202	0.2498
5	0.2079	0.0155	0.2234
6	0.1914	0.0133	0.2047

COMPUTED VALUES OF θ.

r	$\theta \times 10^2$				
	C.V. = 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
2	9.545	10.127	10.674	11.172	11.658
3	6.796	7.125	7.439	7.749	7.968
4	5.640	5.877	6.098	6.318	6.512
5	4.839	5.045	5.224	5.392	5.560
6	4.454	4.639	4.797	4.946	5.089

COMPUTED VALUES OF $\frac{\theta_{c3}}{\theta}$.

r	$\frac{\theta_{c3}}{\theta}$				
	C.V. 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
2	0.124	0.116	0.110	0.106	0.101
3	0.130	0.124	0.119	0.114	0.111
4	0.134	0.128	0.124	0.119	0.116
5	0.140	0.134	0.129	0.125	0.122
6	0.141	0.136	0.131	0.127	0.124

Computation/.....

COMPUTATION OF RATE OF DISCHARGE OF PUMP.

The quantity of water delivered per cycle

$$= \left(\frac{V_{E_1} - V_c}{V_c}\right) A.L_c \text{ cu. ft.}$$

The number of cycles per second

$$= \frac{1}{T} = \left(\frac{2.g.P_c.}{W.L_c.L_E.m_1}\right)^{\frac{1}{2}}.\frac{1}{\theta}$$

∴ D (Discharge of pump in cusecs)

$$= \left[\frac{V_{E_1} - V_c}{V_c}\right]\left[\frac{2.g.P_c}{W.L_c.L_E.m_1}\right] \frac{A\ L_c}{\theta} \text{ cusecs.}$$

$$= \left[\frac{A^2.2\ g\ P_c\ L_c}{W\ L_E\ m_1}\right]^{\frac{1}{2}} \triangle$$

where $\triangle$ will be referred to as the discharge coefficient.

Clearly

$$\triangle = \frac{\frac{V_{E_1} - V_c}{V_c}}{\theta}$$

$$= \frac{\frac{V_{E_1}}{V_c} - 1}{\theta}$$

The expression for D, as in the case of T, the period of a cycle, consists of 2 factors. The first

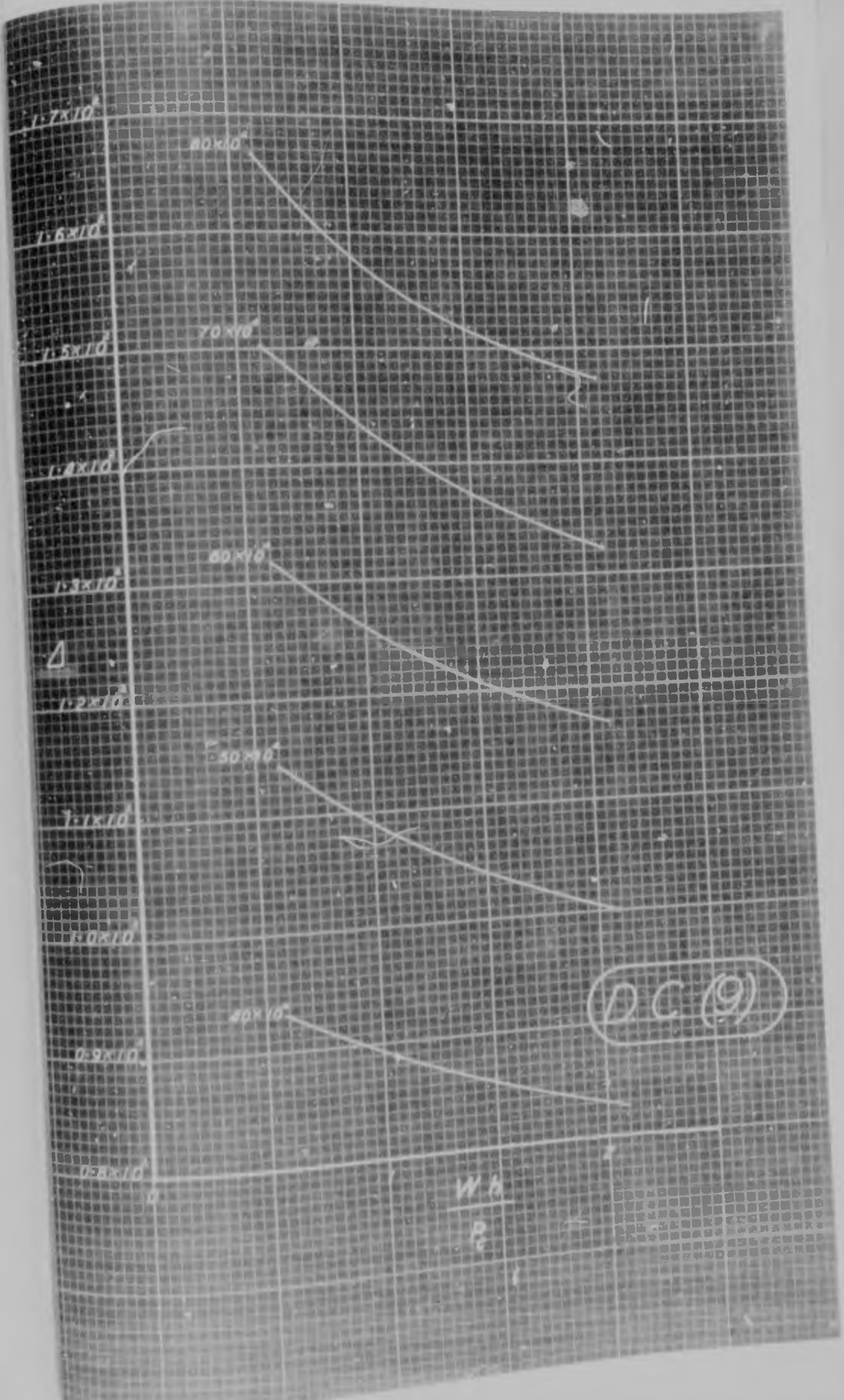
$$\left[\frac{A^2\ 2\ g\ P_c\ L_c}{W\ L_E\ m_1}\right]^{\frac{1}{2}}$$

depends only on the dimensions of the pump and the height of the barometer. The second, $\triangle$, depends only on the head in atmospheres and the fuel mixture strength used.

Design Chart 9 (D.C. (9)) gives values of $\triangle$ to a base of water head in atmospheres.

An examination of this Chart will show that the variation in the value of $\triangle$ with head is small. This is due to the fact that the period as well as $\frac{V_{E_1} - V_c}{V_c}$ decreases with increase of head.

Computed/.....


1·7×10
80×10
1·6×10
70×10
1·5×10
1·4×10
60×10
1·3×10
Δ
1·2×10
50×10
1·1×10
1·0×10
D.C.(9)
40×10
0·9×10
0·8×10
Wh
Pc

[illegible] Δ.

[illegible]	$\Delta \times 10^{[illegible]}$				
	C.V. = 40 x 10^4	50 x 10^4	60 x 10^4	70 x 10^4	80 x 10^4
[illegible]	[illegible]	[illegible]	1.315	1.500	1.666
[illegible]	[illegible]	1.064	1.243	1.420	1.560
[illegible]	[illegible]	1.0[illegible]4	1.221	1.390	1.532
[illegible]	[illegible].365	[illegible]	1.18[illegible]	1.345	1.495
[illegible]	[illegible]	[illegible]	1.164	1.322	1.472

Computation/.....

COMPUTATION OF THE MAXIMUM VELOCITY.

We have already shown that the total K.E. of the water column at any instant:

$$K.E. = \frac{W\ A\ L_E\ V^2}{2\ g} = \frac{E^1\ A\ L_c}{V_c} = \frac{E^1\ A\ L_c\ P_c}{m_1},$$

Hence

$$V = \left[\frac{2\ g\ L_c\ P_c}{W\ L_E\ m_1}\right]^{\frac{1}{2}} \times \sqrt{E^1}.$$

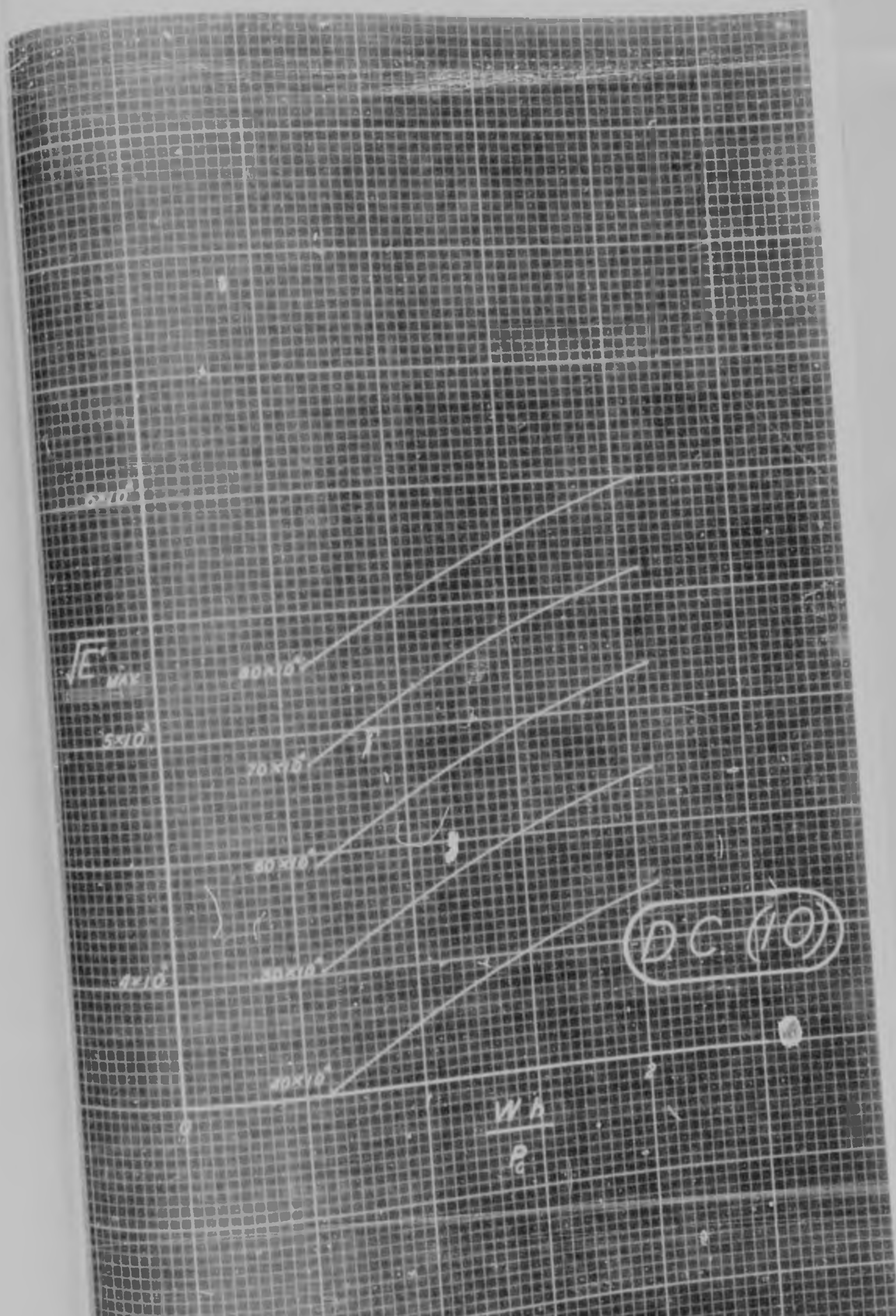
From an examination of the θ charts, the maximum values of E^1 and hence $\sqrt{E^1}$ are seen to occur towards the end of the stroke 1 E. These maximum values of E^1 are read off the θ_{1E} charts and the corresponding values of $\sqrt{E^1}$ max. calculated.

Design Chart 10 (D.C. (10)) gives these values to a base of water head in atmospheres.

COMPUTED VALUES OF $\sqrt{E^1}$ max.

r	$\sqrt{E^1}$ max. x 10^{-2}				
	C.V. = 40 x 10^4	50 x 10^4	60 x 10^4	70 x 10^4	80 x 10^4
2	3.52	4.01	4.45	4.87	5.27
3	3.78	4.27	4.73	5.17	5.54
4	4.00	4.51	4.96	5.41	5.75
5	4.13	4.62	5.05	5.48	5.86
6	4.25	4.73	5.19	5.60	5.99

Computation/.....



80×10
70×10
60×10
50×10
40×10
5×10
DC (10)
Wh
Pc

COMPUTATION OF FRICTIONAL LOSSES.

The following analysis is only approximate as the velocities employed are based upon zero losses in the pump and are thus greater than if losses were taken into account.

Consider an element of the play-pipe of length dx.

Let M be the hydraulic mean depth of the section of the pump barrel,

and let m be the hydraulic mean depth of the section considered.

Let H_f be the total friction head.

Then

$$d\ (H_f) = \frac{4\ f\ dx}{2\ g\ m}\left[V^2\ \frac{A^2}{a^2}\right]$$

(where V, as before, is the velocity of the water in the barrel)

and

$$H_f = \frac{4\ f\ V^2}{2\ g\ M}\left[M\ A^2 \int_0^{LT} \frac{dx}{m\ a^2}\right]$$

We shall denote the quantity

$$\left[M\ A^2 \int_0^{LT} \frac{dx}{m\ a^2}\right]$$

by L_F, the equivalent friction length of piping. (If the play-pipe is of uniform section equal to that of the pump barrel throughout then $L_F = L_T$).

We have already shown that the total K.E. of the column at any instant:

$$K.E. = \frac{E^1\ A\ L_c}{V_c} = \frac{W\ A\ L_E\ V^2}{2\ g}$$

$$\therefore \frac{V^2}{2\ g} = \frac{E^1\ L_c}{W\ L_E\ V_c}$$

Substituting for $\frac{V^2}{2\ g}$ in the expression for H_f:

H_f/.....

$$H_f = \frac{4 f L_F}{M} \cdot \frac{E^1 L_c}{W L_E V_c}$$

Increment of work done against friction per lb. of mixture:-

$$d\,(W.F.) = W H_f \, d\,(\text{volume})$$

$$= \frac{W\,4 f L_F}{M} \; \frac{E^1 L_c}{W L_E V_c} V_c \, d\left(\frac{V}{V_c}\right).$$

∴ the work done against friction per lb. of mixture per cycle,

$$= \frac{4 f L_F L_c}{M L_E} \oint E^1 \, d\left(\frac{V}{V_c}\right).$$

Now, we have already shown that the work done against the head per lb. of mixture per cycle

$$= \left[C.V. - (I_E - I_c) \right]$$

∴ % friction losses F:-

$$F = \frac{\frac{4 f L_F L_c}{M L_E} \oint E^1 \, d\left(\frac{V}{V_c}\right)}{\left[C.V. - (I_E - I_c) \right]} \times 100$$

$$= \left[\frac{400 f L_F L_c}{M L_E} \right] \quad \phi$$

where ϕ will be referred to as the friction coefficient of the pump.

Clearly

$$\phi = \frac{\oint E^1 \, d\left(\frac{V}{V_c}\right)}{\left[C.V. - (I_E - I_c) \right]}$$

The expression

$$\left[\frac{400 f L_F L_c}{M L_E} \right]$$

is independent of the total length of piping, since $\frac{L_F}{L_E}$ may be expressed as a number depending only on the proportional configuration of the piping. Hence the % friction losses for pumps employing the same mixture strengths and pumping against the same head are independent of the length of the

play-pipe/....

play-pipe.

The integral $\oint E^1 \, d(\frac{V}{V_c})$

$$= \int_{1}^{\frac{V_o}{V_c}} E^1 \, d(\frac{V}{V_c}) + \int_{\frac{V_1}{V_c}}^{\frac{V_E}{V_c}} E^1 \, d(\frac{V}{V_c}) + \int_{\frac{V_E}{V_c}}^{\frac{V_{E_1}}{V_c}} E^1 \, d(\frac{V}{V_c})$$

$$+ 2\int_{1}^{\frac{V_3}{V_c}} E^1 \, d(\frac{V}{V_c}) + 2\int_{\frac{V_3}{V_c}}^{\frac{V_4}{V_c}} E^1 \, d(\frac{V}{V_c}) .$$

The integrals

$$\int_{1}^{\frac{V_o}{V_c}} E^1 \, d(\frac{V}{V_c}), \quad \int_{\frac{V_1}{V_c}}^{\frac{V_E}{V_c}} E^1 \, d(\frac{V}{V_c}), \quad \text{and} \quad \int_{\frac{V_3}{V_c}}^{\frac{V_4}{V_c}} E^1 \, d(\frac{V}{V_c})$$

are obtained graphically by measuring the areas intercepted between the "E" curves and the "H" lines in charts θ_{co}, θ_{1E} and θ_{34} respectively.

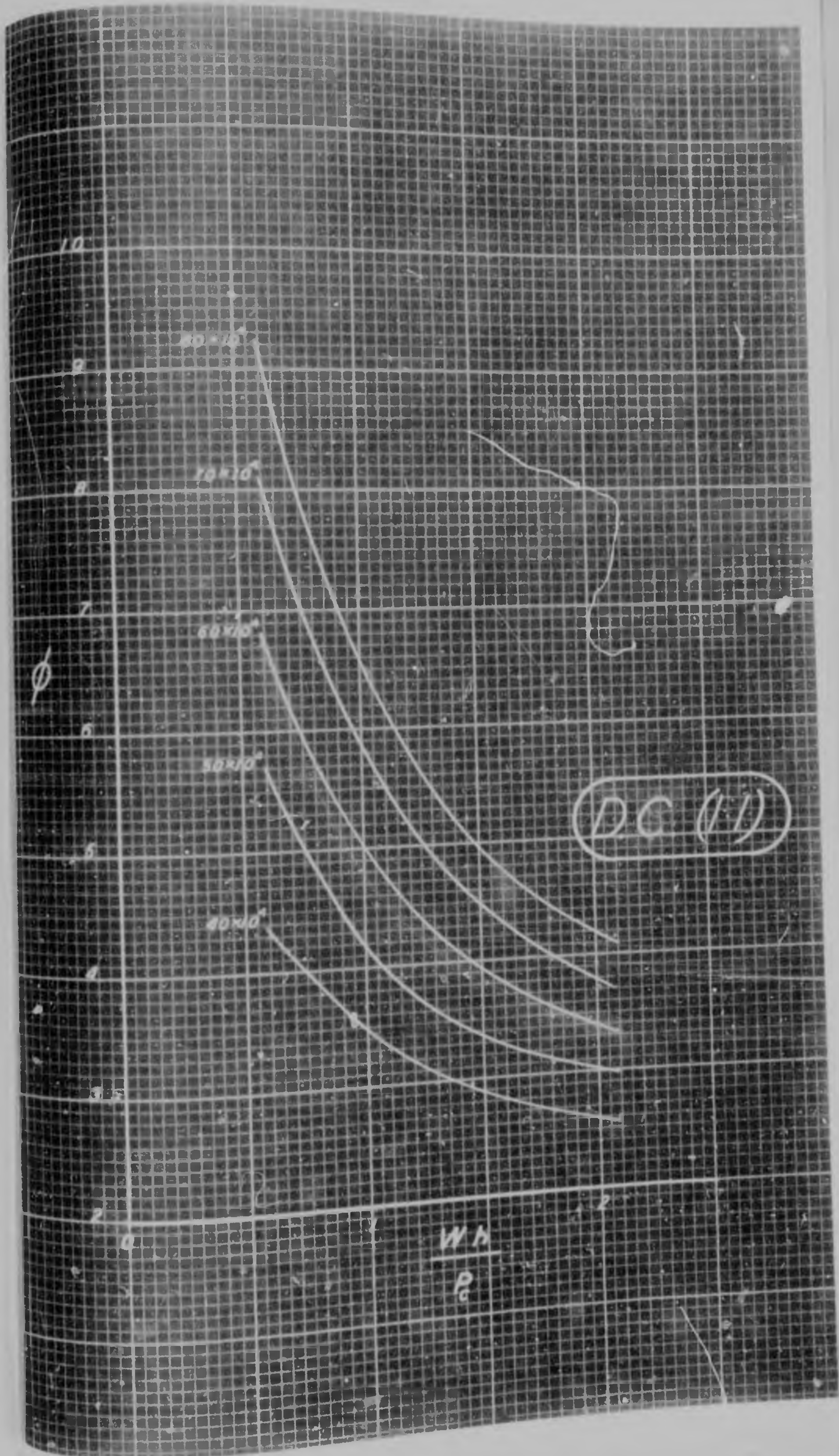
From Fig. 16 it is clear:-

$$\int_{\frac{V_E}{V_c}}^{\frac{V_{E_1}}{V_c}} E^1 \, d(\frac{V}{V_c}) = \frac{1}{2} \left[\frac{V_{E_1} - V_E}{V_c} \right] E^1_E$$

and $$\int_{1}^{\frac{V_3}{V_c}} E^1 \, d(\frac{V}{V_c}) = \frac{1}{2} (0.6) \, E^1_3 .$$

Design Chart 11 (D.C. (11)) gives values of ϕ to

a base/.....


DC (II)
φ
Wh/Pc
10
9
8
7
6
5
4
3
2
0
1
2

a base of water head in atmospheres.

Since the velocities assumed are greater than would occur in practice, the figures for friction would be somewhat on the large side, but although not exact they nevertheless are of use in comparing different pumps.

COMPUTED VALUES OF $\int_{1}^{\frac{V_o}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)$

r	$\left[\int_{1}^{\frac{V_o}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)\right] \times 10^{-4}$
2	0.095
3	0.315
4	0.614
5	0.956
6	1.204

COMPUTED VALUES OF $\int_{\frac{V_1}{V_c}}^{\frac{V_E}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)$

r	$\left[\int_{\frac{V_1}{V_c}}^{\frac{V_E}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)\right] \times 10^{-4}$				
	C.V. = 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
2	28.634	44.109	61.287	83.929	107.656
3	32.752	48.712	68.434	91.306	116.492
4	35.490	53.010	73.058	97.168	121.626
5	37.244	53.819	73.107	95.974	116.043
6	38.252	55.916	74.927	97.228	123.447

Computed/......

a base of water head in atmospheres.

Since the velocities assumed are greater than would occur in practice, the figures for friction would be somewhat on the large side, but although not exact they nevertheless are of use in comparing different pumps.

COMPUTED VALUES OF $\int_{1}^{\frac{V_o}{V_c}} E^1 \, d(\frac{V}{V_c})$

r	$\left[\int_{1}^{\frac{V_o}{V_c}} E^1 \, d(\frac{V}{V_c})\right] \times 10^{-4}$
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3	0.315
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COMPUTED VALUES OF $\int_{\frac{V_1}{V_c}}^{\frac{V_E}{V_c}} E^1 \, d(\frac{V}{V_c})$

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2	28.634	44.109	61.287	83.929	107.656
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4	35.490	53.010	73.058	97.188	121.626
5	37.244	53.819	73.107	95.974	116.043
6	38.252	55.916	74.327	97.228	123.447

Computed/......

COMPUTED VALUES OF $\displaystyle\int_{\frac{V_E}{V_c}}^{\frac{V_{S_1}}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)$

r	$\left[\displaystyle\int_{\frac{V_E}{V_c}}^{\frac{V_{E_1}}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)\right] \times 10^{-4}$.				
	C.V. = 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
2	37.8	66.7	101.5	146.8	204.0
3	24.2	40.4	62.1	90.4	116.3
4	20.3	32.3	48.3	68.6	89.1
5	14.3	23.8	35.0	50.1	67.2
6	11.3	19.2	30.4	43.1	57.6

COMPUTED VALUES OF $\displaystyle\int_{1}^{\frac{V_3}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)$ and $\displaystyle\int_{\frac{V_3}{V_c}}^{\frac{V_4}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)$.

r	$\left[\displaystyle\int_{1}^{\frac{V_3}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)\right] \times 10^{-4}$	$\left[\displaystyle\int_{\frac{V_3}{V_c}}^{\frac{V_4}{V_c}} E^1 \, d\left(\frac{V}{V_c}\right)\right] \times 10^{-4}$
2	0.350	0.297
3	0.550	0.648
4	0.755	0.973
5	0.940	1.260
6	1.082	1.517

Computed/......

COMPUTED VALUES OF $\oint E^1 \, d(\frac{V}{V_c})$

r	$\left[\oint E^1 \, d(\frac{V}{V_c})\right] \times 10^{-4}$.				
	C.V. = 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
2	67.82	112.20	164.18	232.11	313.06
3	59.66	91.82	133.24	184.42	235.50
4	59.86	89.38	125.43	169.86	214.80
5	56.90	82.93	113.46	151.43	188.60
6	55.91	81.52	111.73	146.73	187.45

COMPUTED VALUES OF ϕ.

r	ϕ				
	C.V. = 40×10^4	50×10^4	60×10^4	70×10^4	80×10^4
2	4.30	5.70	6.73	8.10	9.20
3	3.26	3.92	4.52	5.55	6.35
4	2.95	3.51	4.08	4.80	5.21
5	2.68	3.09	3.49	3.98	4.32
6	2.56	2.99	3.32	3.70	4.15

Section V/.....

SECTION V.

DESIGN OF HUMPHREY PUMPS.

With the aid of the design charts drawn up, the main quantities required in the design of Humphrey Pumps may be very easily deduced.

As a preliminary to design proper, a proportional configuration of the piping should be decided upon. It is advisable to have a tapering stand pipe in order to dampen out the violent fluctuations of the column which would otherwise occur at the discharge end if this provision were not made. From this configuration one should determine the ratios $\frac{L_E}{L_T}$ and $\frac{L_F}{L_T}$.

$$\left[L_E = A \int_0^{L_T} \frac{dx}{a} ; \quad L_F = M A^2 \int_0^{L_T} \frac{dx}{ma^2} \right]$$

The cross section of the system is most conveniently chosen as circular. Hence $M = \frac{D}{4}$ and $A = \frac{\pi}{4} D^2$, where D is the diameter of the barrel.

The discharge D, the height of the barometer and the head are included in the specification for design. The fuel mixture strength, C.V., is decided upon by the type of fuel available at the site of the installation.

The period of the pump should next be decided upon. This should be of sufficient duration to allow of proper functioning of the valves. A good figure for medium sized pumps should be about 3 or 4 seconds.

The % frictional losses (F) should next be decided upon. From the expression for F, it is seen that the larger the pump bore, the smaller the frictional losses. But pumps

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$$\left[L_E = A \int_0^{L_T} \frac{dx}{a} \; ; \quad L_F = M A^2 \int_0^{L_T} \frac{dx}{ma^2} \right]$$

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The % frictional losses (F) should next be decided upon. From the expression for F, it is seen that the larger the pump bore, the smaller the frictional losses. But pumps

of/.....

[illegible] mean greater cost of construction. Hence one must arrive at a compromise between efficiency and initial costs in deciding upon F. It should be noted that the frictional losses in Humphrey Pumps are invariably small and the maximum value of F one need consider is about 10%.

Now consider the expressions:-

$$D = \left[\frac{A^2 \; 2 \; g \; P_c \; L_c}{W \; L_E \; m_1}\right]^{\frac{1}{2}} \Delta \; \ldots \; \ldots \; \ldots \; \ldots \quad (1).$$

$$T = \left[\frac{W \; L_E \; L_c \; m_1}{2 \; g \; P_c}\right]^{\frac{1}{2}} . \; \theta \quad \ldots \; \ldots \; \ldots \quad (2).$$

$$F = \left[\frac{400 \; f \; L_F \; L_c}{M \; L_E}\right] . \; \phi \quad \ldots \; \ldots \; \ldots \quad (3).$$

(Express L_E and L_F in terms of L_T).

Since C.V. and $\frac{W \; h}{P_c}$ is known, Δ, θ, and ϕ may be found from Charts (9), (7) and (11). Having decided on D, T and F, equations (1), (2) and (3) enable us to solve for A, L_c and L_T, the three most important quantities required in design.

Having ascertained L_c, the clearance space is decided upon from the fact that we have taken $\frac{V_3}{V_c} = 0.4$.

The necessary strength of the combustion chamber is determined from a knowledge of the maximum temperature T_1, and the maximum pressure $\frac{P_1}{P_c}$ or $\frac{P_4}{P_c}$ as the case may be. These quantities may be found from Design Charts 2, 3 and 5.

The size of the admission posts may be ascertained from a knowledge of $\frac{\theta_{c3}}{\theta}$ as mentioned previously. The exhaust posts should be made of the same size as the maximum exhaust velocities occur from C to 3.

The distance L_{E1} is deduced from Design Chart 4, since $L_{E1} = L_c \frac{V_{E1}}{V_c}$. This enables us to place the water valves in the most suitable position.

A knowledge/.....

A knowledge of the thermal efficiency, obtained from Design Chart 6, enables us to calculate the gas input into the pump and hence the size of gas producers required.

The design should be checked from Design Chart 10 to see whether the maximum velocities encountered are not excessive.

Owing to its simplicity of construction, low initial costs, economical running and low maintenance costs, the Humphrey Pump would be ideal for use on irrigation schemes and farms in general. The author is quite convinced that its potentialities, which at present seem to be misunderstood, will be clearly realised and exploited to the full.

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