FUEL/AIR RATIO CONTROL ON RECUPERATIVE GAS FIRED FURNACES

A thesis submitted to the
University of Aston in Birmingham
for the degree of Doctor of Philosophy.

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May, 1967.

SUMMARY

The application of recuperation by combustion air preheat to industrial gas fired furnaces can result in an appreciable saving in fuel. The application of such recuperation has, however, been limited, since changes in the combustion air temperature can cause the fuel/air ratio to vary widely from the optimum value. Such variations can nullify the potential fuel saving offered by the recuperator and, in extreme cases, cause burner instability.

In examining this problem the value of recuperation by combustion air preheat is assessed and data is presented giving the fuel saving that can be achieved with recuperators of various efficiencies operating over a range of furnace gas temperatures.

The general need to control fuel/air ratio to the stoichiometric value is illustrated and the cost, in terms of fuel wastage, of not doing so is calculated.

The effect of changes in combustion air temperature on the performance of air blast burner equipment is examined in detail and possible techniques for producing an air blast injector, that can maintain a constant air/gas ratio regardless of changes in throughput and combustion air preheat, have been studied.

A pneumatic bridge flow ratio controller has been developed which, applied to an air blast tunnel burner, is shown to maintain the air/gas ratio to within $\pm$ 5% of the stoichiometric value over a range of throughputs and combustion air temperatures.

Incidental information presented includes fundamental data on the performance of gas governors and a throughput control technique designed to have a linear effective characteristic.

ACKNOWLEDGEMENTS

The Author wishes to extend his appreciation to Dr. W.A. Simmonds, Director of Research, Gas Council's Midlands Research Station, and Dr. W.G.S. Parker, Professor of Chemistry at the University of Aston, for permission to carry out this work, and to Mr. P.G. Atkinson of the Gas Council's Midlands Research Station and Mr. A.R. Cooper of the University of Aston, for their guidance and encouragement.

The Author is also indebted to his friends and colleagues both at the University of Aston and the Midlands Research Station for their assistance in building the experimental equipment and in producing this thesis, and to Professor G.V. Jeffreys of the University of Aston in whose Department the work was completed.

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I INTRODUCTION

In any combustion fired furnace the products of combustion leave the furnace at a temperature equal to, or higher than, that of the stock. Sensible heat is thus lost from the furnace and the proportion of the initial heat input to the furnace which is available for heating the stock is thus reduced. The amount of heat lost in this way, and commonly termed "flue loss", is roughly proportional to furnace temperature, being negligibly small in the case of low temperature appliances such as hot water and steam raising boilers, where the flue gas temperature may be about 200°C, and amounting to almost the total heat input in the case of a gas fired furnace operating at about 1900°C.

Various means are available for reducing flue loss. These include:- (1) Reducing the total volume of combustion products flowing through the furnace per therm of heat released, for example, by burning the fuel with oxygen enriched air or with neat oxygen. (2) Recovery of a proportion of the waste heat and returning it to the furnace. (3) Utilization of the otherwise waste heat for ancillary processes such as steam raising. This latter technique, however, depends on the economics of the factory rather than the furnace, and is therefore not considered in this paper.

Of the alternative methods available for recovering waste heat the highest efficiency is achieved with load recuperation. With this method the load is preheated by the flue gases before it enters the furnace to be brought up to the temperature finally required. Load recuperation can conveniently be arranged on continuous furnaces, and, by having counter flow passage of the flue gases over the load, the flue gases can be reduced to a comparatively low temperature before final discharge to atmosphere. The successful application of load recuperation depends upon a continuous supply of work to the furnace, and upon the availability of sufficient floor space for the recuperative section.

An alternative method of waste heat recovery is to use the combustion products from the furnace to preheat the combustion air, or both the air and fuel, before it enters the furnace. This entails the use of either a recuperator, which is essentially a continuously operated heat exchanger, or a regenerator, which is a heat store and is operated cyclically. In the first part of such a cycle waste gases are passed through the regenerator to raise its temperature, and in the second the waste gases are turned off and the combustion air is passed through the same passages to recover the heat from the regenerator. By using two regenerators alternately, continuous waste heat recovery can be obtained, although some cyclic variation in the rate of heat recovery does occur.

The choice between recuperation and regeneration is one of duty, economics and convenience. The materials of construction are such that higher preheat temperatures can be obtained with regenerators, but they tend to be bulky, relatively expensive, and require valve gear to switch the flow of combustion air and flue gases. At present, recuperation is the most attractive means of recovering waste heat from industrial gas fired furnaces.

Fundamentally, recuperation can be used to preheat both the combustion air and fuel gas. However, since the volume flow ratio of air to town gas is normally about 4:1 the saving in fuel obtainable by preheating town gas is small compared with that achieved by preheating the combustion air to the same temperature, and, in practice, the additional complication of preheating both is not worth the relatively small additional return.

In addition to saving fuel, the use of preheated combustion air increases the flame temperature and thus the maximum working temperature of a furnace. In order to obtain furnace temperatures much above 1800°C it is obviously necessary to raise the flame temperature higher than its normal 2010°C and the choice of method lies between preheating the combustion air or using oxygen enriched air or pure oxygen for combustion. This choice is mainly an economic one depending upon the furnace usage, continuous operation for long periods favouring the use of recuperation, whilst occasional short runs tend to favour the use of oxygen.

The number of applications where it is necessary to fit a recuperator simply in order to obtain the required temperature are rather limited and the more common reason is the need to keep fuel costs to a minimum.

Providing the performance data of a recuperator are available, the saving in fuel costs can easily be calculated. Figure 1-1 shows the variation of percentage fuel saving with air preheat temperature for various furnace gas temperatures when burning a gas having a gross calorific value (C.V.) of 485 Btu/ft^3 with the stoichiometric air requirement. The calculations upon which figure 1-1 is based are presented in Appendix I of this paper.

In most furnace applications it is desirable that not only should the furnace attain the required temperature but that maximum fuel economy

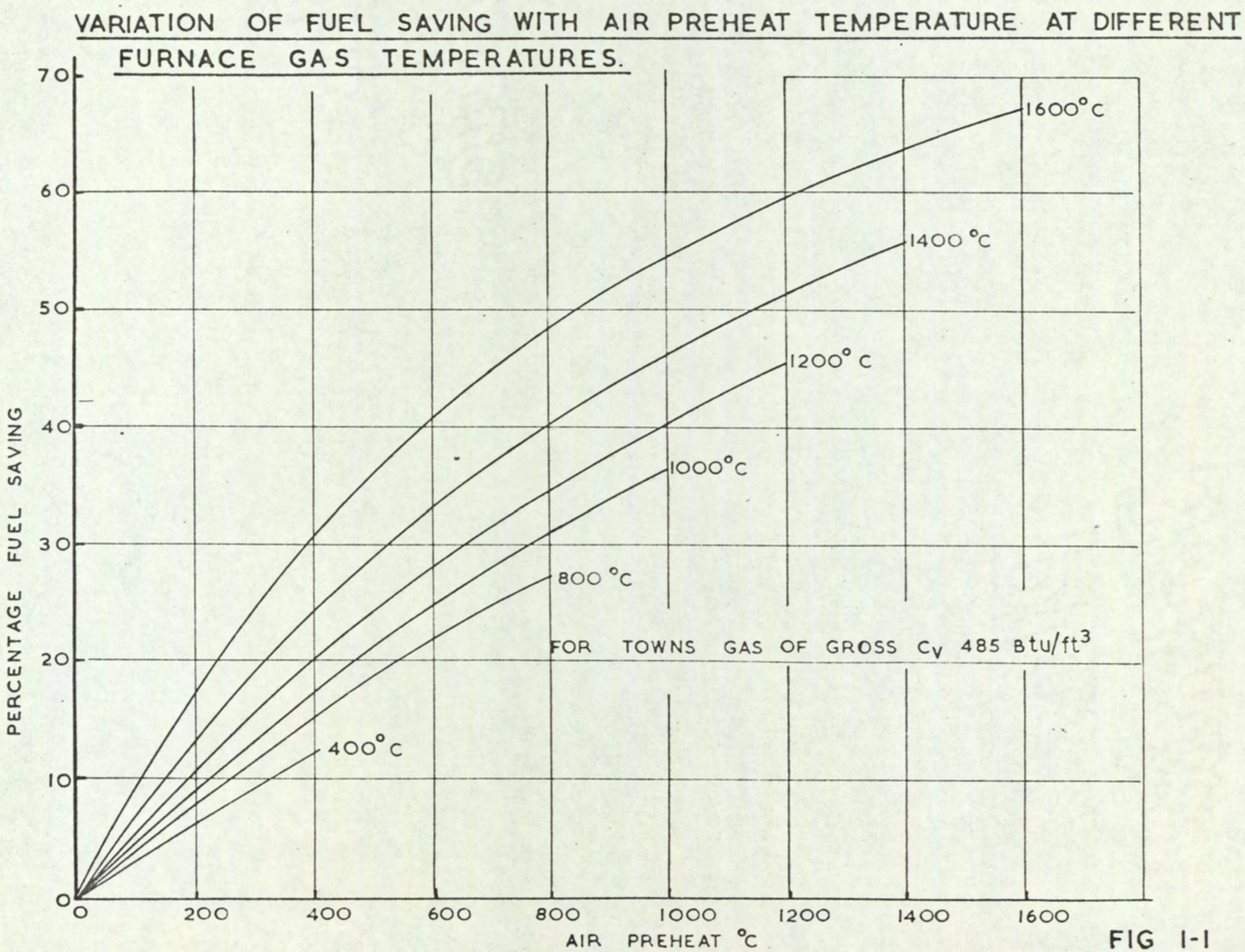

VARIATION OF FUEL SAVING WITH AIR PREHEAT TEMPERATURE AT DIFFERENT FURNACE GAS TEMPERATURES.
PERCENTAGE FUEL SAVING
0
10
20
30
40
50
60
70
AIR PREHEAT °C
0
200
400
600
800
1000
1200
1400
1600
1600°C
1400 °C
1200° C
1000°C
800 °C
400°C
FOR TOWNS GAS OF GROSS C_V 485 Btu/ft^3

FIG 1-1

should be achieved in its operation, and both these factors are considerably influenced by air/gas ratio. In any furnace, if gas and air are mixed and burned in stoichiometric proportions, i.e. the proportions which, if combustion were allowed to go to completion, would leave no combustibles unburned nor oxygen remaining, then there is a minimum wastage of sensible heat from the flue. The presence of either excess air or gas causes undue loss since, in either case, the excess air or fuel gas is raised to furnace temperature and discharged. Additional wastage occurs if excess gas is present, due to incomplete combustion of the available fuel resulting in failure to release all its potential heat.

Figure 1-2 shows the percentage of fuel wasted due to the deviation of the air/gas ratio from the stoichiometric value for various waste gas temperatures. Figure 1-2 relates to town gas of gross C.V. 485 Btu/ft^3 and stoichiometric ratio 4.15:1 and calculation of the data shown in the figure is presented in detail in Appendix II.

It can be seen that the percentage of fuel wasted due to the deviation of air/gas ratio from the stoichiometric is very dependent on waste gas temperature if excess air is present. On the other hand, if excess gas is present, the wastage is much less affected by waste gas temperature. This illustrates the need to maintain the air/gas ratio as close to stoichiometric as possible when firing a high temperature furnace and shows that when firing a low temperature appliance, such as a boiler, some excess air can be allowed with little loss of thermal efficiency. It should be noted that the actual value of the stoichiometric air/gas ratio is determined by the gas composition, however, for town gases a

PERCENTAGE OF FUEL WASTED V AIR/GAS RATIO (TOWNS GAS GROSS C.V. 485 Btu/ft³)

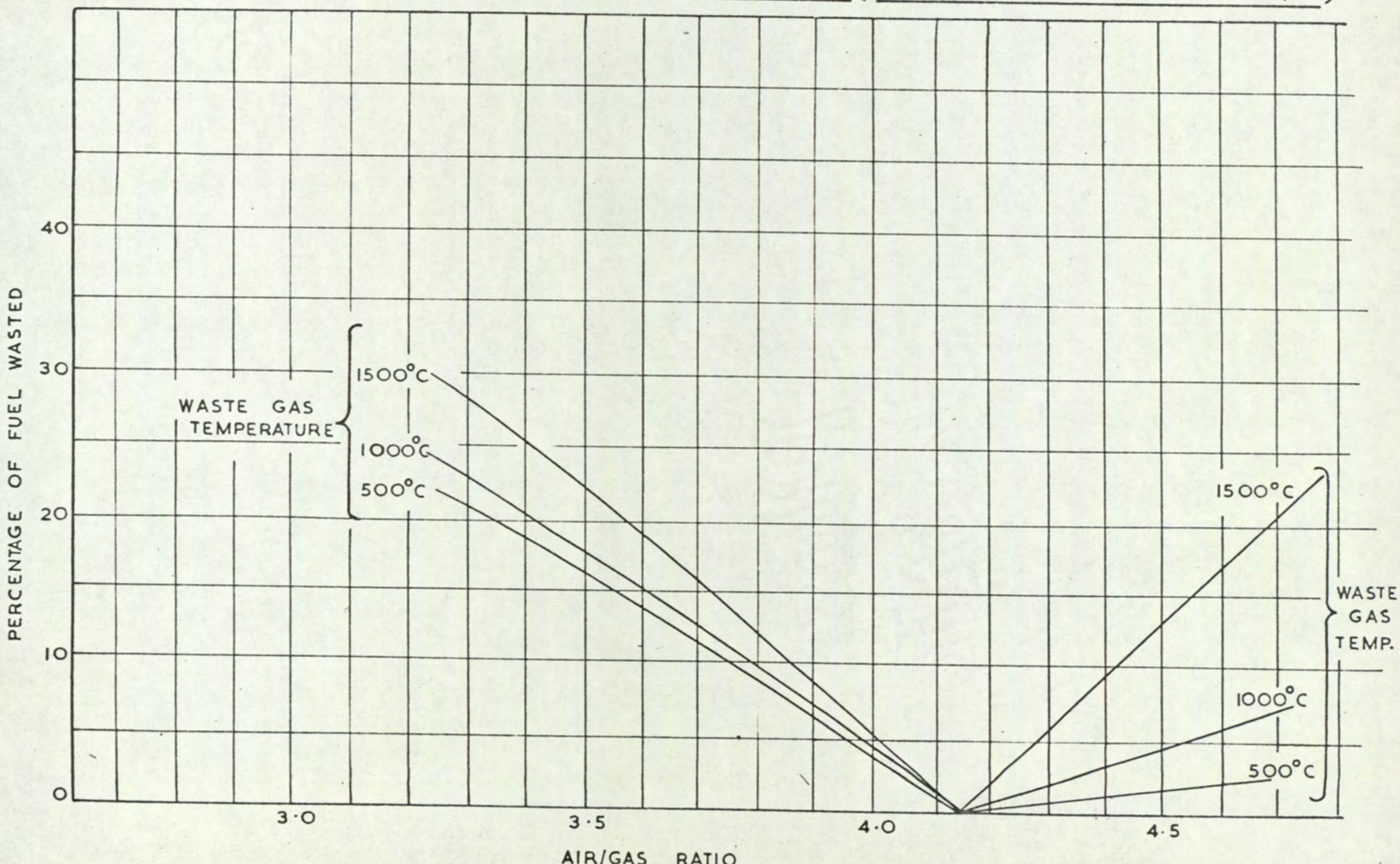


FIG. 1-2

practical value can be obtained by dividing the gross calorific value by 117. During the course of this work a number of changes in gas composition (increasing the calorific value) were made by the West Midlands Gas Board. As a result, various values of the stoichiometric air/gas ratio, ranging from 3.8:1 to 4.15:1, are referred to in this paper.

In addition to the above economic considerations two other factors often make it necessary to maintain a specific air/gas ratio under all firing conditions. One is burner stability; rich or weak combustion can cause certain types of burner to become unstable and blow off, and the other is the requirement to produce combustion products having a specific chemical composition. This latter requirement is particularly important in metallurgical processes where it is necessary to maintain either oxydizing, reducing or neutral conditions in a direct fired furnace, or where town gas is used to produce a prepared atmosphere for an indirectly heated furnace.

Many techniques are available for controlling air and gas to a burner and these have been considered in detail by the author elsewhere[1].

If the air and gas flow rates to a burner can be metered or the air/gas ratio measured, either from an analysis of the air/gas mixture, or from a flue gas analysis, then the required control can be achieved by manual adjustment of separate air and gas valves. Such a system can of course be made automatic by the use of conventional flow ratio controllers of the ring balance or diaphragm type used in chemical engineering practice, Controllers of this type however are generally too costly to fit to gas furnaces.

A method often adopted to achieve ratio control is to use linked valves in the gas and air lines to a burner, the valves being preset to give the desired flow ratio. This method can be satisfactory providing the valves are adjusted correctly, and where throughput is controlled on a high low basis this can readily be achieved. If throughput is to be controlled on a proportional basis then the valves must be matched at all levels of throughput and this can be difficult.

A common method of obtaining a constant air/gas ratio with varying throughput is to use an injector; either a gas blast type in which an expanding jet of gas entrains air from the atmosphere or an air blast type in which a jet of air entrains gas which is reduced to atmospheric pressure by means of a zero governor. The air/gas ratio of the resulting mixture is determined by the relative areas of the jet nozzle and throat of the injector and the density ratio of the air and gas. Theoretically injectors can be designed to produce a precise stoichiometric mixture when supplied with air and gas of constant known density. In practice, however, a means of trimming the air/gas ratio is provided to overcome dimensional tolerances in the injector and to allow for small variations in gas density.

The accuracy of ratio control that can be achieved with the above techniques obviously depends upon the precision with which the equipment is manufactured and set up. However, even if the equipment is absolutely accurate, errors in air/gas ratio can arise due to changes in the composition of the fuel gas. These changes can occur either due to a varying source of supply, where a consumer is on a grid distribution system, or due to changes of feedstock in the manufacturing process. In practice although the calorific value of the gas is held relatively constant its specific gravity can vary over a range of $\pm$ 0.05 (S.G. gas $\simeq$ 0.5, air = 1). Any control system therefore in which the flow ratio is determined by controlling the differential pressures across orifices, is subject to a practical limit of accuracy of approximately $\pm$ 5%. The linked valve technique described above can be used to control air/gas ratio to this order of accuracy over a throughput range of about 20 : 1 whilst the injector techniques can maintain a similar accuracy over a range of about 5 : 1.

Unfortunately neither the linked valve or injector techniques described above are satisfactory where the combustion air is preheated, since the changes in air density that occur as the preheat temperature varies causes deviations in the air/gas ratio.

Whilst the problem of maintaining a stoichiometric ratio under varying conditions of preheat temperature and throughput can be overcome technically by using a conventional flow ratio controller, and compensating for air temperature variation by means of ancillary control equipment, such a technique would generally be uneconomic. The fuel saving achieved by fitting a recuperator must be balanced against the capital and maintenance costs of the recuperator and the costs of any additional control equipment. These latter costs would be very high if equipment of the chemical process type was used.

In the past, a lack of air/gas proportioning equipment suitable for industrial gas furnaces and capable of use with preheated air has limited the application of waste heat recovery by air preheating.

Study of the literature since 1945 revealed no reference to techniques designed to overcome this problem, although over 120 abstracts associated with combustion control were noted. In view of the lack of information on this subject the operation of air blast injectors has been considered in some detail with a view to developing an injector which could compensate for changes in air temperature and produce a constant air/gas ratio regardless of air preheat and throughput changes. Techniques for doing this inexpensively have been evolved and should assist toward the wider application of recuperation on Industrial Furnaces.

II

THE EFFECT OF PREHEATED COMBUSTION AIR ON AIR BLAST BURNERS

In the introduction to this paper reference was made to air blast injectors in which an expanding jet of air, commonly at a supply pressure of 1 P.S.I.G., is used to entrain gas which is supplied at, or about, atmospheric pressure. In practice the air/gas mixture produced by the injector is fed to a burner head where combustion occurs, and the combination of injector and burner is generally known as an air blast burner. The injector, mixture pipe and burner head may be separate components, assembled as shown in Fig. 2-1, or the functions of the injector and mixture pipe may be combined by allowing the air jet to expand into a parallel mixture tube as shown in Fig. 2-2. The burner head is frequently a refractory tunnel, inside which combustion occurs, and from which the combustion products emerge at velocities often in excess of 400ft./sec. Much work has been done at the Midlands Research Station of the Gas Council on the study of air blast tunnel burners and their design and theory of operation is well understood.[2,3]

It can be shown that if the gas is supplied at atmospheric pressure, and the burner is firing into a furnace also at atmospheric pressure, then the air/gas ratio remains substantially constant regardless of changes in air supply pressure and throughput, the actual ratio depending upon the cross sectional area of the air nozzle, mixture tube and tunnel, and densities of the air and gas. In practice the gas is supplied at atmospheric pressure from a "zero" governor and a control valve or "gas restrictor" situated between this and the injector provides a means of

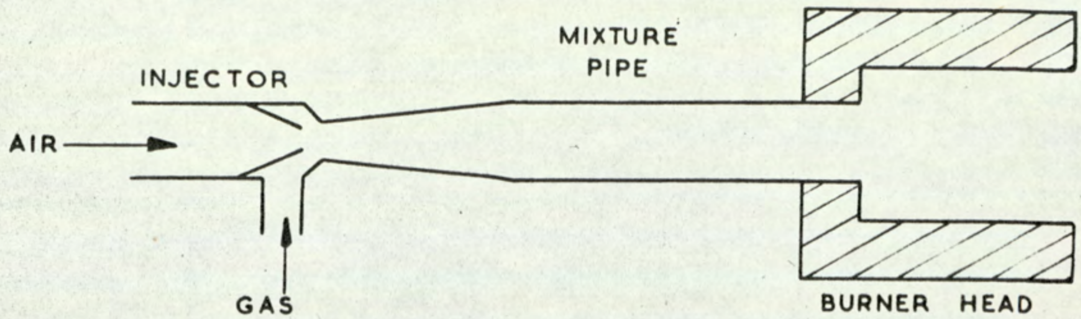

MIXTURE
PIPE
INJECTOR
AIR
GAS
BURNER HEAD

FIG. 2·1

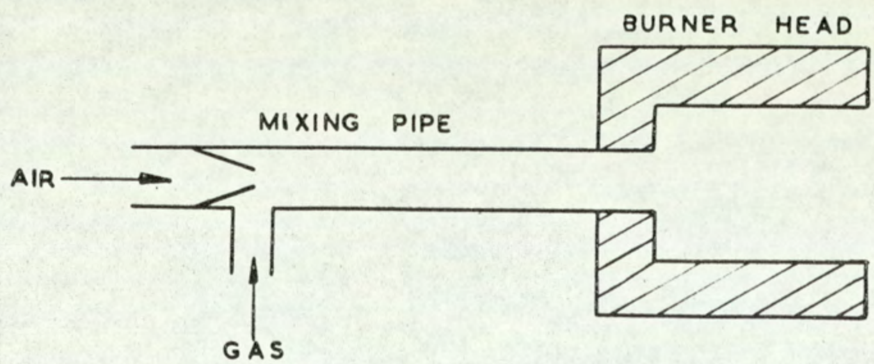

BURNER HEAD
MIXING PIPE
AIR
GAS

FIG. 2·2

trimming the air/gas ratio to allow for the dimensional tolerances of the burner. The gas restrictor is usually adjusted when the burner is commissioned and thereafter the burner throughput is controlled by means of a valve in the air supply line.

It is obvious that if recuperation is used to preheat the combustion air, the combustion air temperature will change with time, being room temperature on starting up and rising progressively as the associated furnace and recuperator system heats up. Similarly sudden large load changes, as may occur when a cold charge is introduced into the furnace, will also cause variation in the air preheat temperature.

The resulting changes in combustion air density thus cause the air/gas ratio to vary.

If the injector is designed and adjusted to give a stoichiometric air/gas ratio with air at room temperature, then the mixture will go progressively gas rich as the air temperature increases due to the increasing ratio of momentum to mass flow rate of the air jet. Conversely if the injector is designed and adjusted to give a stoichiometric air/gas ratio when supplied with hot air from the recuperator then the burner will run extremely weak during the initial heating up period when the combustion air is at room temperature. Apart from the considerable loss of thermal efficiency that can be caused by burning mixtures other than stoichiometric, as discussed above, it is possible for the deviation from stoichiometric to be sufficient to prevent satisfactory burner operation due to rich or lean blow off: rich blow off occuring at an air/gas ratio of about 2.5:1 and weak blow off between 5 - 10:1.

In view of this problem it is useful to consider the various adjustments that could be made to an injector burner system in order to maintain a constant mixture ratio as the combustion air temperature varies.

1) If the injector is designed to give a stoichiometric ratio when supplied with cold air, then as the air temperature rises and the mixture tends to become rich the gas restrictor could be closed progressively. The adjustment of the gas restrictor being determined either by a measurement of air temperature or by a comparison of the actual gas and air flows with some form of flow ratio measuring device.

2) By appropriate adjustment of the area ratio of air nozzle to throat; again the actual adjustment would be determined by either air temperature or flow ratio measurements.

3) By designing the injector to give a stoichiometric ratio with hot air when supplied with gas at atmospheric pressure and supplying the gas at a positive pressure at lower air temperatures to compensate for the reduced injection effect. (If at any particular air temperature the gas supply pressure were a constant fraction of the air pressure then a constant ratio could be maintained regardless of changes in throughput.)

A theoretical examination was made of each possible technique and these are described in turn below.

I <u>Temperature-variable gas restrictor</u>

It was thought initially that the simplest technique to engineer would be (1) above utilizing an adjustable gas restrictor operated by a temperature sensitive actuator such as a bimetal or mercury in steel thermometric element. Attention was therefore given to deriving the relationship between gas restrictor area and air preheat temperature necessary for a constant air/gas ratio for two types of air blast tunnel burner. The burners considered were a parallel mixture tube, parallel tunnel burner optimized for maximum pressure efficiency (Midlands Research Station Type A) and a parallel mixture tube converging tunnel burner also optimized for maximum pressure efficiency (Midlands Research Station Type C). The derivation which is presented in Appendix III of this paper, gave the following relationships:-

Type A Burner
(1in. dia. tunnel exit)

$$K_2A_2 = \sqrt{\frac{1}{1.255T_1 - 656.8}} \qquad (1)$$

Type C Burner
(1in. dia. tunnel exit)

$$K_2A_2 = \sqrt{\frac{1}{0.6874T_1 - 354.7}} \qquad (2)$$

where:- A_2 is the port area of the gas restrictor (ft^2)

K_2 is a constant involving the discharge and expansion coefficient of the restrictor.

T_1 is the absolute temperature ($^\circ R$).

It will be noticed that for both burners,

K_2A_2 is infinite when $T_1 \doteq 520^\circ R$.

K_2A_2 is imaginary when $T_1 < 520^\circ R$.

K_2A_2 is real and gets smaller with increasing T_1 when $T_1 > 520^\circ R$.

This is because the burners are designed to pass a stoichiometric air/gas mixture (3.8:1 at the time of calculations) with the air at room temperature and with the gas supply governed to atmospheric pressure at the gas inlet with no gas restrictor (i.e. an infinitely large gas restrictor). If the air temperature, therefore, is below room temperature the jet momentum is insufficient to produce a 3.8:1 mixture and the above equation is invalid. The variation of K_2A_2, as the air temperature increases above room temperature, is what would be expected, since, as the jet momentum increases the mixture will tend to become gas rich and A_2 must be made smaller to maintain a constant ratio of 3.8:1.

Equations (1) and (2) above have been used to determine the area of a gas restrictor required for burners having 1 in. dia. tunnel exits at a series of air temperatures, assuming that the restrictor is an orifice having a discharge coefficient of 0.6. The results, are shown in Fig. 2-3.

As stated above, burner manufacturing tolerances necessitate the fitting of a gas restrictor to a burner of optimized design. Calculations were therefore made to determine the relationship between restrictor area and temperature for A Type burners having slightly undersized air nozzles, and these results are also shown graphically in Fig. 2-3.

From these results two things are apparent. Firstly, that with a nominally optimized burner, restrictor area is very sensitive to air nozzle diameter when supplied with air near room temperature, and that the area / temperature characteristic is considerably affected by manufacturing tolerances. Secondly it would appear that if the air nozzle is deliberately undersized then very little adjustment of the gas restrictor

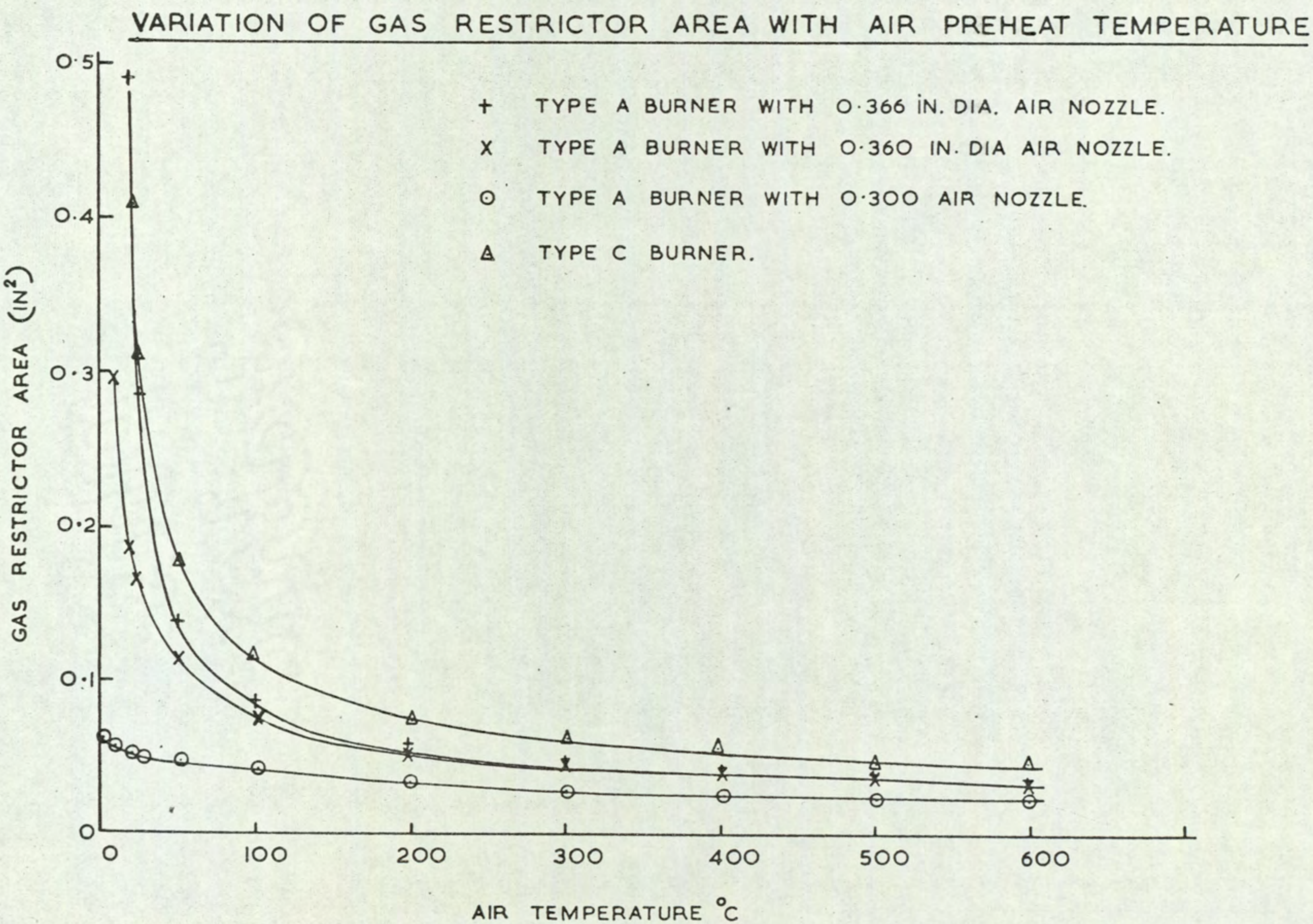

VARIATION OF GAS RESTRICTOR AREA WITH AIR PREHEAT TEMPERATURE
+ TYPE A BURNER WITH 0·366 IN. DIA. AIR NOZZLE.
X TYPE A BURNER WITH 0·360 IN. DIA AIR NOZZLE.
⊙ TYPE A BURNER WITH 0·300 AIR NOZZLE.
Δ TYPE C BURNER.
GAS RESTRICTOR AREA (IN²)
0·5
0·4
0·3
0·2
0·1
0
0
100
200
300
400
500
600
AIR TEMPERATURE °C

FIG. 2–3

is necessary to compensate for changes in air preheat. In view of this the variation of air/gas ratio with temperature was calculated for an A Type burner having an undersized air nozzle (0.300 in. dia. instead of 0.366 in. dia. for a 1 in. dia. tunnel) and for which the gas restrictor was adjusted to give a stoichiometric ratio at a preheat temperature of 500°C.

These calculations, which are presented in Appendix IV, showed that with air at room temperature an air/gas ratio of 6.44:1 would be obtained compared with a stoichiometric ratio of 3.8:1. It was thought that such a deviation from stoichiometric may be unimportant providing that the burner remains stable, since it only occurs when there is very little air preheat, i.e. when the waste gas temperature is low and that it may be possible to exploit this as a simple solution to the problem. Unfortunately however, further study of burner performance, and in particular the effect of preheating the combustion on air upon burner throughput, which is discussed below, showed that any control system in which an air nozzle of fixed dimensions is used, such as in either of the techniques outlined above, would unduly restrict the turndown range of the premix burners considered.

The turndown range of a premix burner is limited at high flow by the maximum throughput obtainable with the air supply pressure available, usually 1 P.S.I.G., and at low flow by the light-back rate of the burner i.e. the flow rate at which the mixture velocity through the mixture tube is approximately equal to the flame speed so causing the flame to light back down the mixture tube to the gas inlet.

It will be realised that if the combustion air is preheated then the subsequently higher flame speed will increase the burner light-back rate and so reduce the turndown of the burner. In the extreme, when the preheat temperature is sufficiently high, spontaneous combustion occurs when the air and gas mix; this sets a limit of about 500°C to the preheat temperature when using a premix burner.

In the case of a burner having a constant area air nozzle the turndown range is further reduced by the reduction in maximum throughput that occurs as the combustion air temperature is increased. The magnitude of this effect can be calculated since the mass flow rate of the air through the air nozzle for a particular air pressure is proportional to the square root of its density and thus inversely proportional to the square root of its absolute temperature. Fig. 2-4 shows the variation of throughput with air temperature for a constant air pressure.

It will be seen therefore that a burner designed to pass 2000 s.cu.ft./hr. of air at room temperature with, say 1 P.S.I.G. air pressure, will only pass 1420 s.cu.ft./hr. of air at 300°C and 1220 s.cu.ft./hr. at 500°C. Thus at a preheat temperature of 500°C the throughput is reduced by 39% and the net heat input is reduced by 30%.

A burner sized, therefore, to give the appropriate heat input at a preheat temperature of 500°C would be larger than a burner designed for use with air at room temperature. The mixture tube would in fact be 20% larger in diameter and since it has been shown by Francis and Hoggarth[4] that lightback rate is proportional to mixture tube diameter raised to the power 2.863, it would subsequently have a 66% higher lightback rate. This represents a reduction in turndown ratio from 5:1 to 3:1.

VARIATION OF THROUGHPUT WITH PREHEAT TEMPERATURE

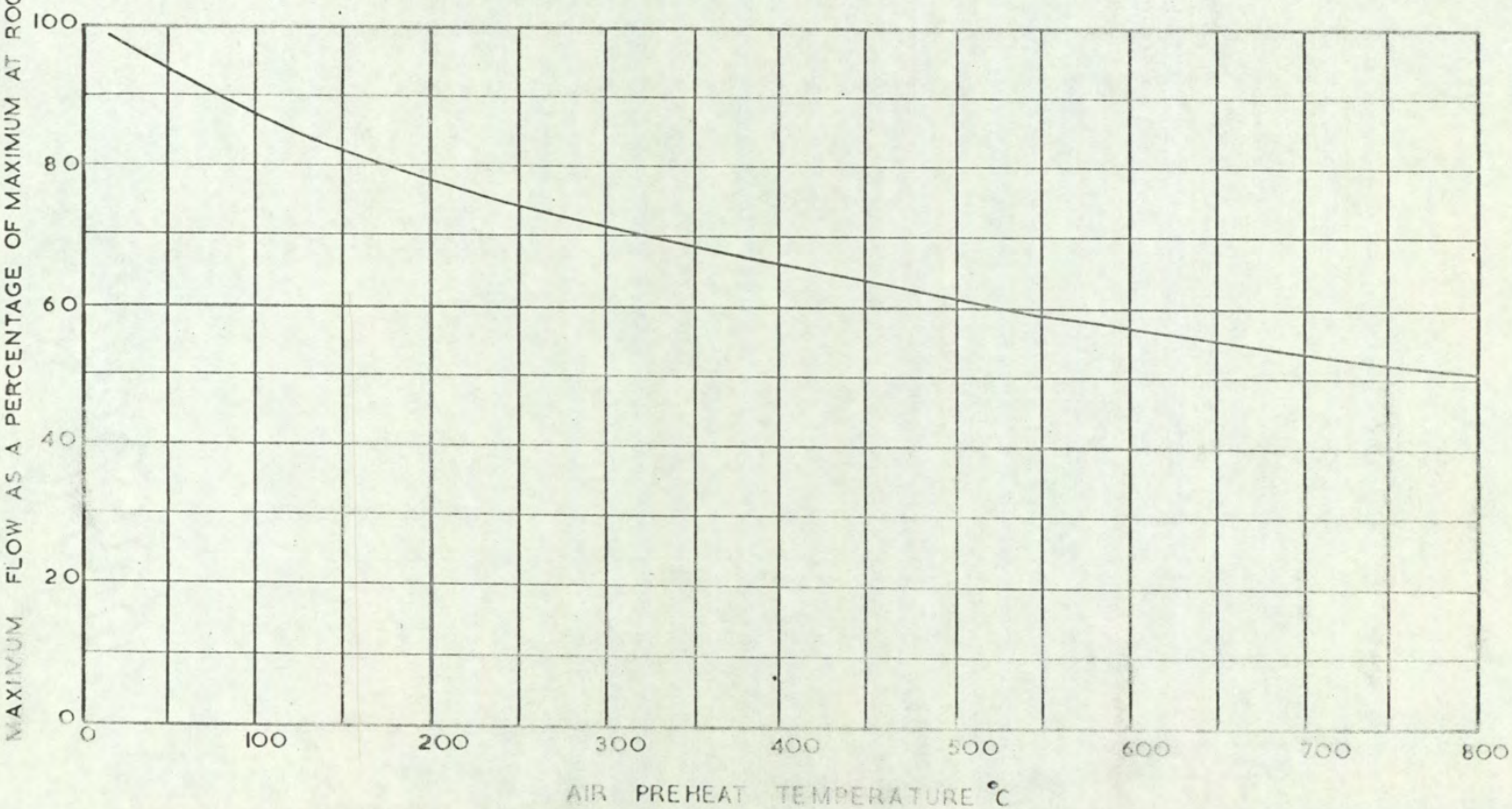


FIG. 2-4

Such a large reduction in turndown ratio would be undesirable in practice and attention was therefore directed to the second of the possible techniques listed above for maintaining a constant air/gas ratio with varying air temperature.

II Temperature-variable Air Nozzle.

This technique relies on appropriately adjusting the area ratio of the air nozzle to throat for a fixed setting of the gas restrictor. It was thought that this could be achieved relatively simply by varying the air nozzle area by means of a temperature sensitive actuator. With such a system there would be no undue reduction in turndown ratio since the throughput would not decrease with increasing air temperature. In addition, the total heat input for a particular air pressure increases with preheat. Thus, a burner sized to give a particular heat release at a preheat temperature of 500°C will be appreciably smaller than one designed for air at room temperature and will therefore have a correspondingly improved turndown ratio due to the smaller mixture tube diameter.

The relationship between air nozzle area and preheat temperature has been determined for 1 inch burners Type A and C and also for a non optimized 1 inch Type A burner having an air nozzle 0.300 in. diameter.

The calculations, which are presented in Appendix V, give the following relationships, the expressions being reduced to three significant figures.

1 inch Type A with a 0.366in. dia. air nozzle (optimized).

$$A_1 = \frac{49.41\ T_1}{159.6T_1 + 162.2 \times 10^3}$$

1 inch Type A with a 0.300in. dia. air nozzle (non-optimized).

$$A_1 = \frac{49.41\ T_1}{159.6\ T_1 + 280.6 \times 10^3}$$

1 inch Type C with a 0.406in. dia. air nozzle (optimized).

$$A_1 = \frac{35.96\ T_1}{108.0\ T_1 + 87.61 \times 10^3}$$

where A_1 is the air nozzle area in sq. in.

T_1 is the air temperature in $^{\circ}$R.

The above relationships are shown graphically in Fig. 2-5.

From the above calculations it appeared that if satisfactory ratio control was to be achieved using a thermometric actuator to operate a variable air nozzle, then the area/temperature relationship of the compensating mechanism would require characterizing both to suit each type of burner and also to compensate for manufacturing tolerances in burner dimensions.

In order to estimate the required accuracy of a compensating mechanism calculations were performed, which are presented in Appendix VI, to determine the effect on mixture ratio of small deviations of the air nozzle area from that required to give a stoichiometric air/gas ratio. It was found that in order to maintain an air/gas ratio within the

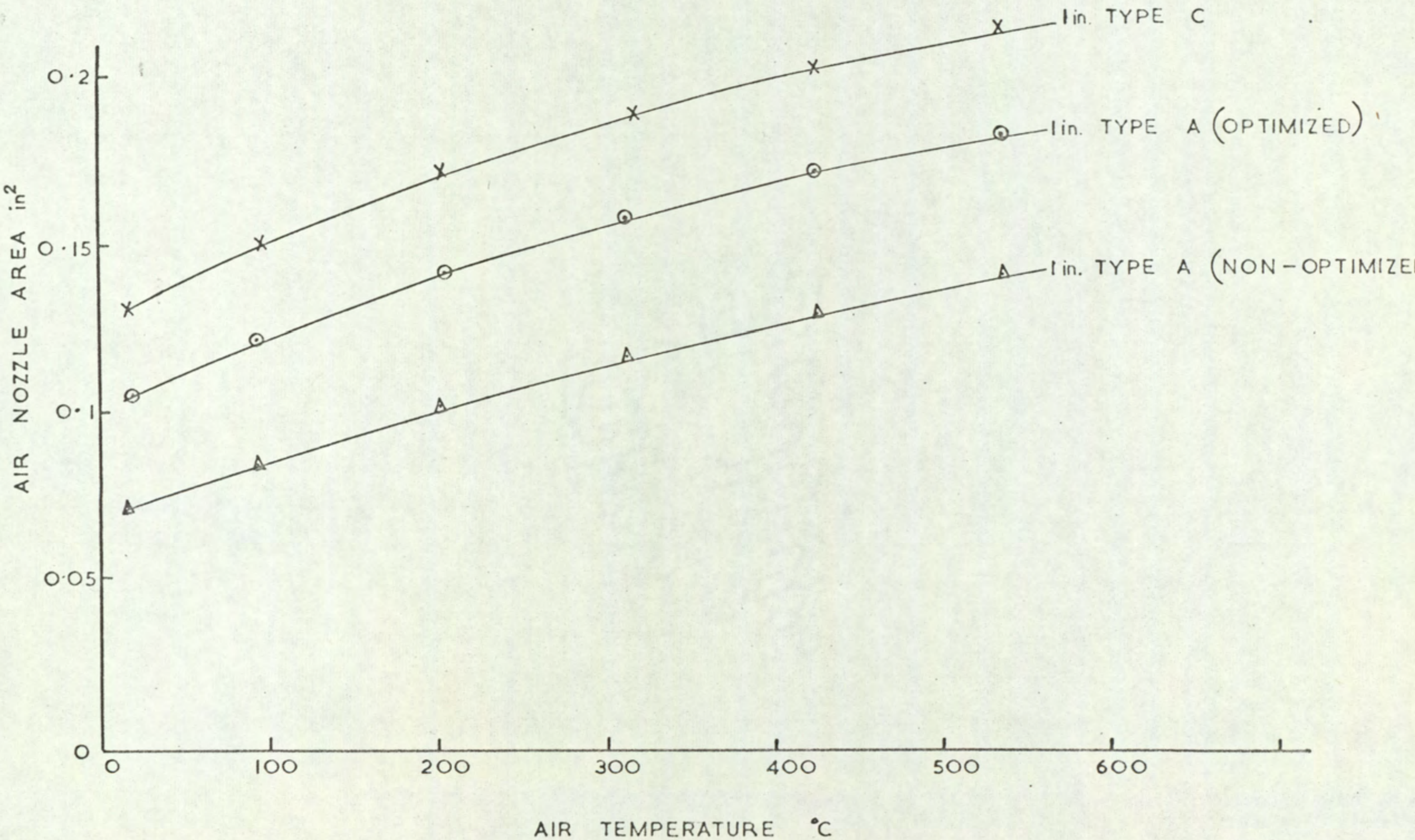

AIR NOZZLE AREA in^2
0·2
0·15
0·1
0·05
0
0
100
200
300
400
500
600
AIR TEMPERATURE °C
1in. TYPE C
1in. TYPE A (OPTIMIZED)
1in. TYPE A (NON-OPTIMIZED)

FIG. 2-5

desired limits of $\pm$ 5%, i.e. the limit of accuracy imposed by gas quality variation, then the air nozzle area must be maintained within $\pm$ 3.8% of the calculated area.

A survey was then made of commercially available thermometric actuators to find one which could be used to vary the area of the air nozzle, either by moving a tapered needle or by operating a sliding shutter. Since the range of preheat temperatures likely to be encountered lie between 15 - 500°C it was desirable that the compensating mechanism should operate over this range. This requirement limited the choice of actuator to bimetal devices and mercury in steel elements. Unfortunately these devices were found to have such limited actuating power and movement that it would have been extremely difficult to engineer a simple, industrially suitable system, particularly in view of the accuracy requirement referred to above. Attention was therefore turned to the third possible technique.

III <u>Temperature-variable Gas pressure</u>

Consider the performance of a conventional air blast burner, operating on preheated air and having an air nozzle optimized for air at maximum preheat temperature. If gas is supplied at atmospheric pressure the injector will produce a stoichiometric mixture at maximum preheat temperature and at lower temperatures will inject less gas so producing a weak mixture. However, if the inlet gas pressure could now be increased above atmospheric pressure then obviously it should be possible to achieve a stoichiometric ratio, the increase in gas supply pressure compensating for the reduced injection effect.

In order to determine the actual gas pressure required to achieve a stoichiometric ratio and how this varies with throughput and air temperature, the burner theory presented in Appendix I was developed further. The mathematics, which is presented in detail in Appendix VII, shows that for a 1 inch Type A burner optimized for an air temperature of 500°C:-

$$\frac{\psi}{Q_g^2} = 1.179 \times 10^{-3} - 8.407 \times 10^{-7}\,T_1$$

where ψ = inlet gas pressure(ie."Zero" governor outlet pressure)(in.w.g.)

Q_g = gas rate (ft.^{3}/hr.)

T_1 = Air temperature (°R)

Similarly for a 1 inch Type C burner it is shown that:-

$$\frac{\psi}{Q_g^2} = 6.367 \times 10^{-4} - 4.537 \times 10^{-7}\,T_1$$

It will be noticed that for any particular air temperature

$$\psi \propto Q_g^2$$

now $Q_a^2 \propto P_1$

where Q_a = air rate (s.ft.^{3}/hr.)

P_1 = air supply pressure.

Therefore a constant air/gas ratio will be maintained regardless of throughput providing the gas inlet pressure is a constant fraction of the air supply pressure.

In addition a constant air/gas ratio will be maintained regardless of changes in air temperature providing this fraction is varied with temperature according to the above equations.

Appendix VII shows that for a Type A burner optimized at an air temperature of 500°C, the gas pressure must be 43% of the air pressure if a stoichiometric ratio is to be produced with air at room temperature. Similarly, in the case of a Type C burner a gas pressure equal to 36% of the air pressure is required with air at room temperature.

Unfortunately in both cases a rather high gas pressure is required to obtain a stoichiometric mixture at reasonable throughputs. For example assuming that gas is available at the burner at the normal supply pressure of 2in.w.g. then the maximum gas rate obtainable on the 1 inch Type A burner whilst maintaining a stoichiometric ratio is only 118.8 ft^3/hr., compared with 200 cu.ft/hr. for a normal 1 inch Type A burner with air at 1 P.S.I.G.* and 60°F. Similarly the maximum gas rate obtainable with the 1 inch Type C burner is 161 ft^3/hr. compared with a normal 1 inch Type C burner which burns 275 ft^3/hr. of gas when supplied with air at 1 P.S.I.G. and 60°F.

In both cases the maximum throughput is unduly restricted and the turndown ratio reduced by almost half. The technique of maintaining a stoichiometric ratio regardless of air preheat by supplying gas at a positive pressure instead of atmospheric pressure would appear to be useful if gas is available at a pressure of about 6in.w.g. or more, either from the main or from a gas fan. Indeed, if a gas fan were used

* 1 P.S.I.G. ≡ 27.68in.w.g. is a common fan air supply pressure.

that would give an outlet pressure greater than 10.6 in.w.g., (i.e. 36% of 1 P.S.I.G.) then the maximum throughput of the 1 inch Type C burner with a 1 P.S.I.G. air supply, for example would be approximately 450 cu.ft/hr. with air at room temperature falling to about 275 cu.ft/hr. with air at 500°C; i.e. the throughput at room temperature would be approximately twice that obtained from the same sized burner optimized at room temperature and supplied with gas at atmospheric pressure.

However, for general application, the provision of a gas fan was considered an undue complication and expense, and in view of the need for developing a simple temperature compensating system attention was redirected to systems in which the air nozzle area was varied.

Flow ratio Controlled Injector

In the foregoing discussion attention has been given to compensating for combustion air temperature changes by measuring the actual temperature and making appropriate automatic adjustments to either the air nozzle area, gas restrictor setting or inlet gas pressure. The alternative technique referred to on page II - 3, is to compare the actual gas and air flows to the burner and, if the ratio is not correct, to make the necessary corrective adjustment. The problem thus becomes one of flow ratio control rather than temperature compensation. Attention was therefore directed towards designing an injector having a variable area air nozzle which could be automatically adjusted according to the actual flow ratio.

The technique that was ultimately developed involved controlling the air/gas ratio by equating the differential pressures developed across metering orifices in the air and gas supply pipes, the orifice in the air supply being situated upstream of the recuperator and therefore not subject to the air temperature and density changes occuring when preheat is used.

The control action was achieved by automatic adjustment of the air nozzle area by a pneumatic operator, any inequality in the differential pressures across the metering orifices resulting in appropriate adjustment of the air nozzle so as to restore the air/gas ratio to the desired value.

This technique has the advantages of the temperature-variable air nozzle technique described above but avoids the need for a compensating device characterised to a non-linear area/temperature relationship and applicable only to one type of burner.

The successful utilization of the technique obviously depends upon the availability of an air tight recuperator, however, this requirement was not considered unreasonable if it would enable a simple control system to be designed.

The design of a variable air nozzle injector based on the flow ratio control principle is described in the next section and the experimental investigation of the device and subsequent modifications are described in the following sections. In the course of evolving the final design a number of compensating injector systems were developed and subsequently superseded by improved versions. The mark system of reference has therefore been adopted in this paper, and the work described leads to the Mk. VI Compensating Injector which is the final industrialized version.

III

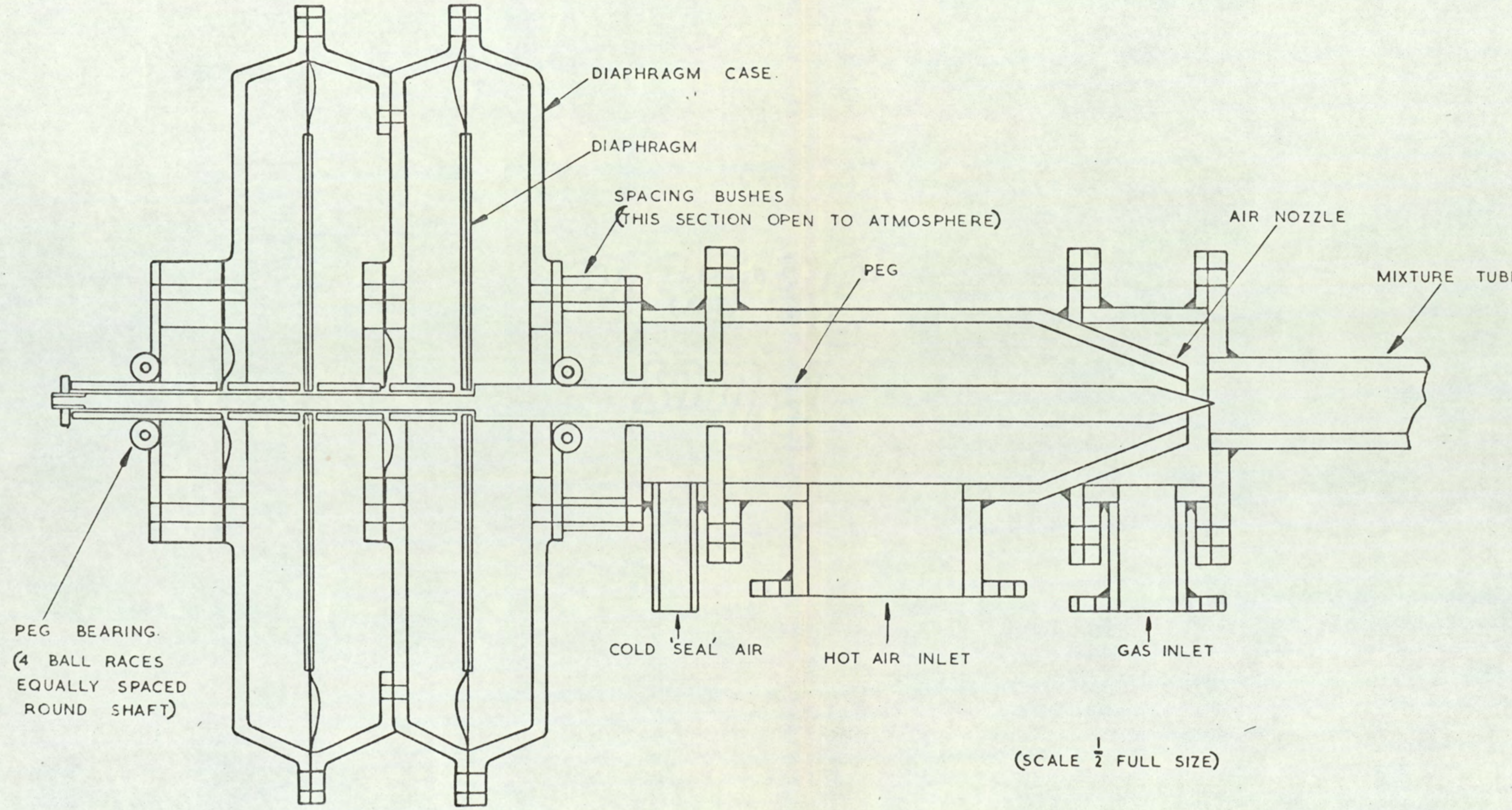
DEVELOPMENT OF THE MARK I COMPENSATING INJECTOR

In the previous Section a survey was made of the possible techniques for maintaining a constant air/gas ratio with varying air preheat temperatures when using an air blast burner or injector burner system. It was concluded that the most promising line of attack was the development of an air blast injector having a variable area air nozzle, which would be automatically adjusted so as to maintain the air and gas flow rates in stoichiometric proportions by comparison of the differential pressures developed across metering orifices in the gas and air supply lines; the air metering orifice being situated upstream of the recuperator and passing air of virtually constant temperature and density.

An injector based on this principle was designed and is shown in Fig. 3-1, whilst a complete burner system incorporating the injector is shown in Fig. 3-2. It will be seen that the area of the air nozzle can be varied by means of a tapered plug which is free to move in and out of the nozzle and is constrained only by the forces acting on the two diaphragms.

The actual design of the combined injector-burner assembly was that of a 1.5 in. Midlands Research Station Type A burner, having an air nozzle that could be varied in area over a range slightly in excess of that to maintain a stoichiometric air/gas ratio with combustion air preheat temperatures between 15 - 500°C.

COMPENSATING INJECTOR



(SCALE $\frac{1}{2}$ FULL SIZE)

FIG. 3-1

BURNER SYSTEM WITH COMPENSATING INJECTOR.

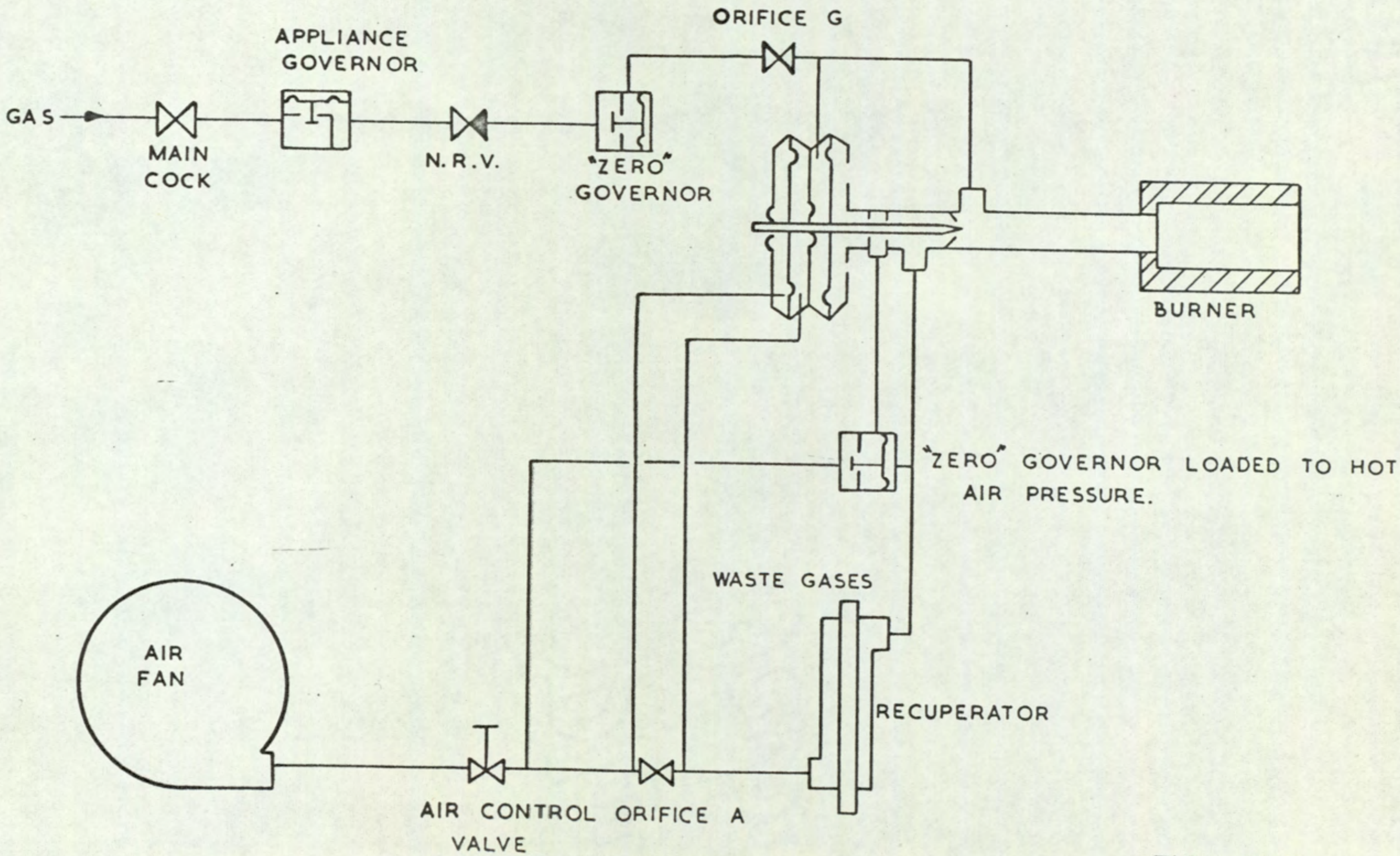


FIG. 3-2

This size of burner was selected as corresponding in throughput to the smallest size of commercial recuperator that was available and, with an applied air pressure of 1 P.S.I.G., the burner would pass approximately 2000 cu.ft./hr. of air. In addition, this size of burner was realistic from the point of view of industrial gas practice whilst being suitable for laboratory investigation.

The diaphragm assembly consisted of standard governor components which were readily available; the cases, leather diaphragms and diaphragm plates being parts manufactured for the Jeavons 1 inch Crawford range of governors.

The differential pressure developed across orifice A (see Fig. 3-2), which is proportional to the square of the combustion air mass flow, is applied to one diaphragm in a manner tending to move the plug into the air nozzle.

The gas flow through orifice G, i.e. the gas flow to the burner, depends upon the area of the orifice, its discharge coefficient and the differential pressure existing across it. Assuming the area to be constant, and neglecting changes in discharge coefficient with throughput, then the gas flow is entirely dependent on the differential pressure across the orifice. Since the gas supply is reduced to atmospheric pressure immediately upstream of orifice G, the gas flow will depend upon the negative pressure or suction developed by the injector. This negative pressure was applied to one side of the second diaphragm so as to tend to withdraw the plug from the air nozzle, the other side of the diaphragm being open to atmosphere.

If the air mass flow is constant then enlarging the air nozzle by withdrawing the plug tends to reduce the suction developed by the injector and conversely if the air nozzle area is decreased, the suction will be increased. The Mk. I injector was designed to operate as a force balance device so that if the suction did not equal the differential pressure across orifice A then the plug would move in the appropriate direction until it did.

When designing the device it was initially assumed that the only significant cause of error in equating the differential pressures would be friction in the plug bearings. Friction was therefore minimized by mounting the plug on ball bearings and by using a pneumatic seal to prevent leakage of hot air where the plug entered the air nozzle section.

The attainment of a satisfactory pneumatic seal depended upon cold air being supplied to the seal section at a pressure equal or almost equal to the pressure of the combustion air in the air nozzle section. Under these conditions negligeable cross leakage would occur through the narrow annular gap between the two sections; the only leakage would be that of the "seal" air to atmosphere which would provide a certain degree of cooling for the diaphragm assembly.

A typical coefficient of friction for ball bearings is 0.001, which, in the designed injector, could result in an inequality of 0.0006in.w.g. between the differential pressures. If the differential pressure across the air orifice A at maximum throughput is 2in.w.g. then at a 5:1 turndown this differential pressure will fall to 0.08in.w.g. Therefore if the required air/gas ratio is 4:1, the actual ratio on turndown (when maximum error should occur) was expected to lie between 3.97:1 and 4.02:2; an accuracy well within the desired limits of ± 5% as discussed in Section II.

It was stated above that the only significant cause of error was initially assumed to be due to friction in the movement. However, it became obvious from the design that some force would act on the tapered portion of the plug, due to the accelerating airflow through the nozzle, and due to the suction developed by the injector. Whilst it was thought that such forces would be negligible compared with the forces acting on the diaphragms, calculations were made to determine the magnitude of the forces acting on the plug and their influence on air/gas ratio. The calculations, which are presented in detail in Appendix VIII of this paper, showed that the maximum deviation from a desired ratio of 4:1, due to these forces, would be within the limits 3.99:1 and 4.10:1, thus, although the effect was not insignificant, the error was of a similar order to the estimated frictional error and was within the desired limits.

IV

DESCRIPTION OF TEST RIG

In order to investigate the performance of the injector described in the previous section it was necessary to provide the following facilities:-

1) Metered air and gas supplies to the injector.
2) An air heater capable of preheating the combustion air to any controlled temperature, between 15 - 500°C.
3) Manometers for the measurement of the various pressures and differential pressures in the system.
4) Measurement of actual air nozzle area whilst the injector was working.

A test rig providing these facilities was built in the Chemical Engineering Laboratory, Suffolk Street, and later rebuilt with slight modification in the Chemical Engineering Laboratories at the College of Advanced Technology, Gosta Green. A schematic diagram of the apparatus, including the injector, is shown in Fig. 4-1.

Combustion air was provided by a Keith Blackman two-stage centrifugal fan rated at 175 cu.ft./min. at 30in.w.g. outlet pressure. Gas was available in the laboratory from a 2in. B.S.P. supply. The flow rates of gas and combustion air to the injector could be measured on Letter Series Rotameters. The gas Rotameter was rated 50 - 500 cu.ft./hr. at 2in.w.g. pressure and gas density 0.5, and the air Rotameter rated 200 - 2000 cu.ft./hr. at 1 P.S.I.G. In order to obtain the maximum

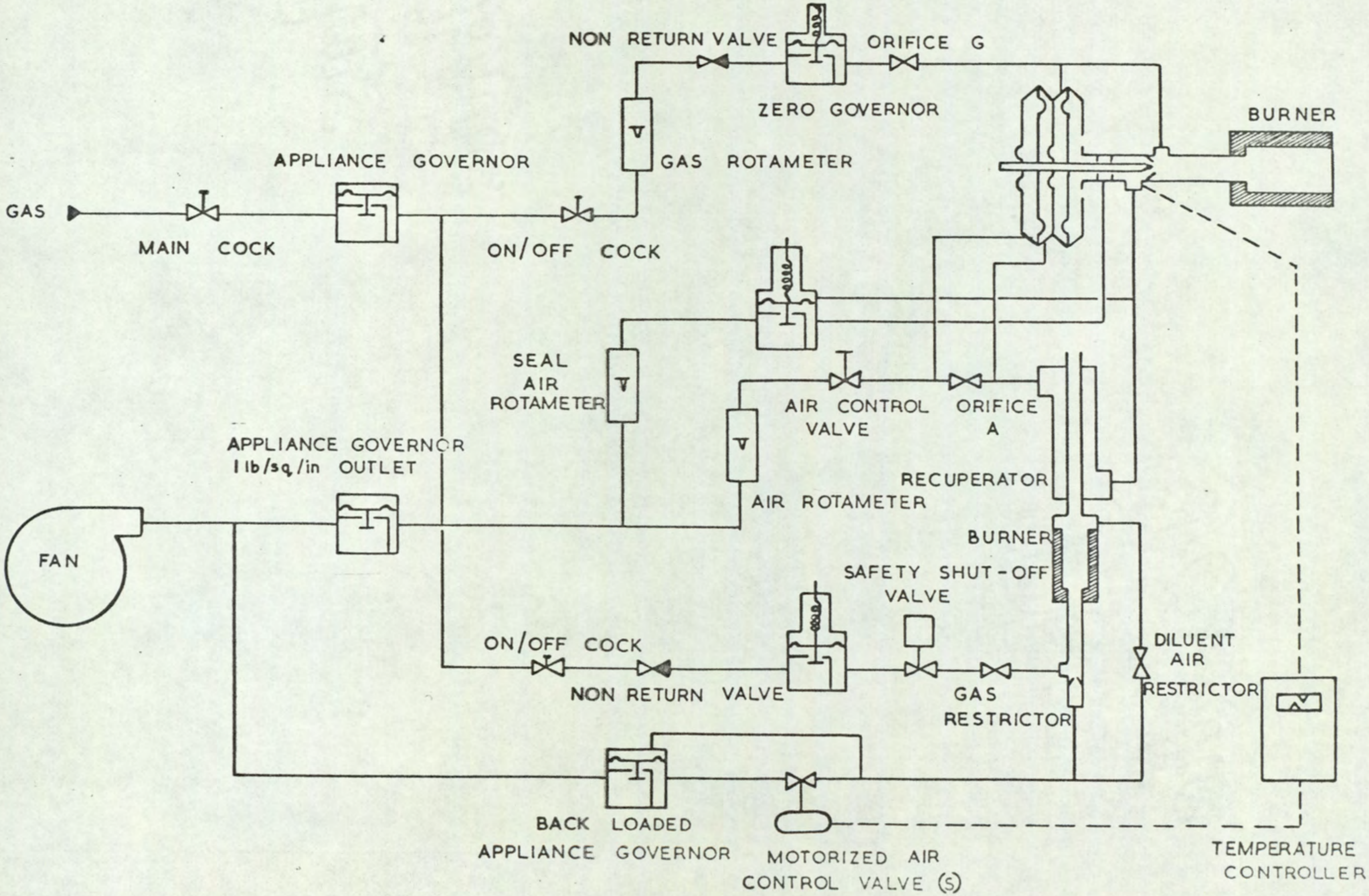

NON RETURN VALVE
ORIFICE G
ZERO GOVERNOR
BURNER
APPLIANCE GOVERNOR
GAS ROTAMETER
GAS
MAIN COCK
ON/OFF COCK
SEAL
AIR
ROTAMETER
AIR CONTROL
VALVE
ORIFICE
A
APPLIANCE GOVERNOR
1 lb/sq/in OUTLET
RECUPERATOR
AIR ROTAMETER
FAN
BURNER
SAFETY SHUT-OFF
VALVE
ON/OFF COCK
DILUENT
AIR
RESTRICTOR
NON RETURN VALVE
GAS
RESTRICTOR
BACK LOADED
APPLIANCE GOVERNOR
MOTORIZED AIR
CONTROL VALVE (S)
TEMPERATURE
CONTROLLER

FIG. 4-1

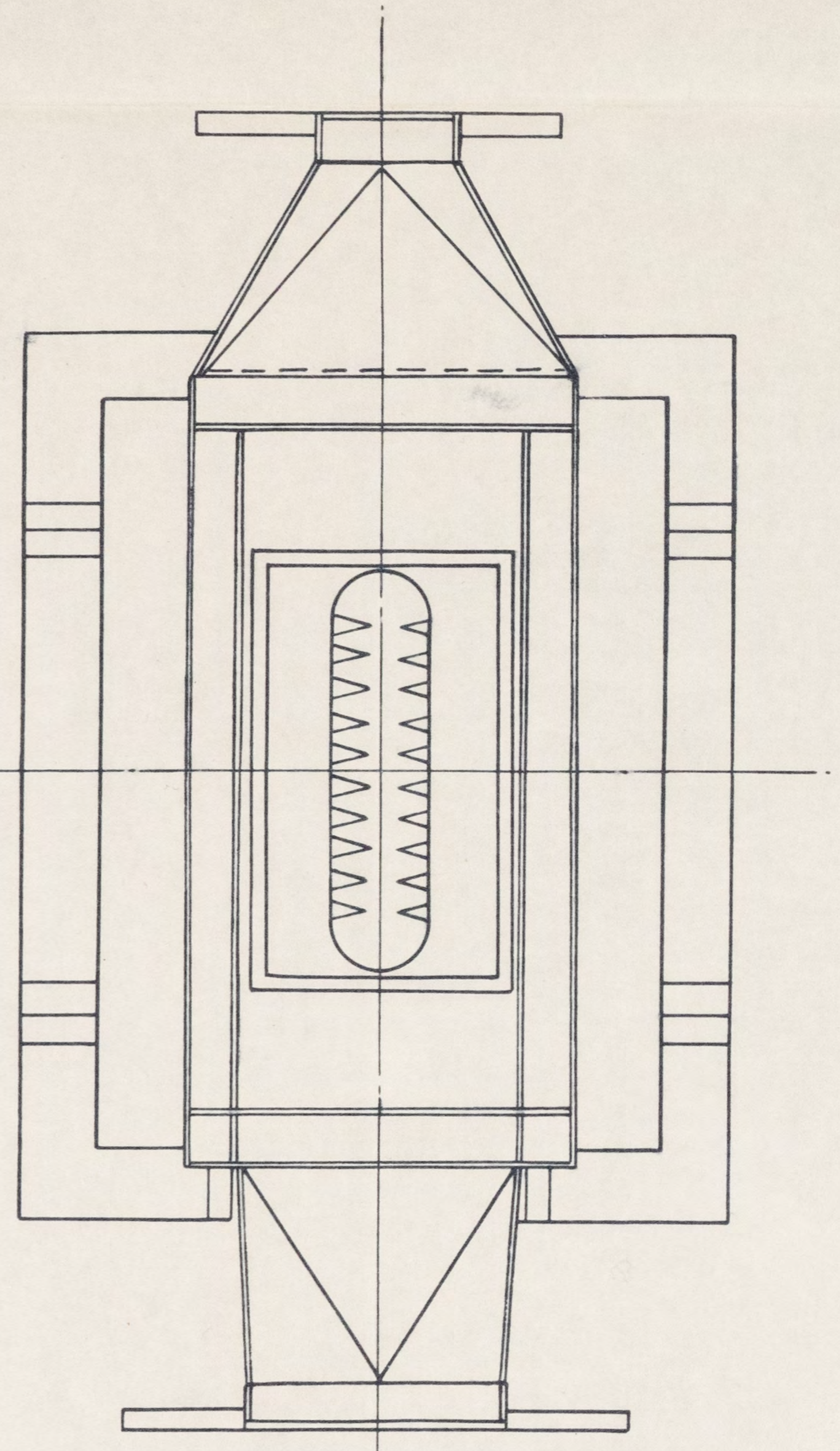
accuracy from these instruments the air and gas supplies were governed to 1 P.S.I.G. and 2 in. w.g. respectively at the inlet to the Rotameters and the flow control valves were situated downstream. Both instruments were thus operated under constant pressure conditions and at the calibrated pressures. The nominal accuracy of the Rotameters under these conditions is $\pm$ 2% of any reading.

The air supply to the pneumatic seal of the injector was governed to the pressure of the air in the air nozzle section by means of a 1 inch back-loaded J 47 SZ zero governor. It was estimated that providing the governed pressure was within ½ in.w.g. of the required pressure, the ingress or egress of air from the air nozzle section of the injector would be less than 1% of the combustion air. The flow to the air seal could be measured by means of a Metric Series Rotameter calibrated in the range 45 - 450 ft^3/hr air.

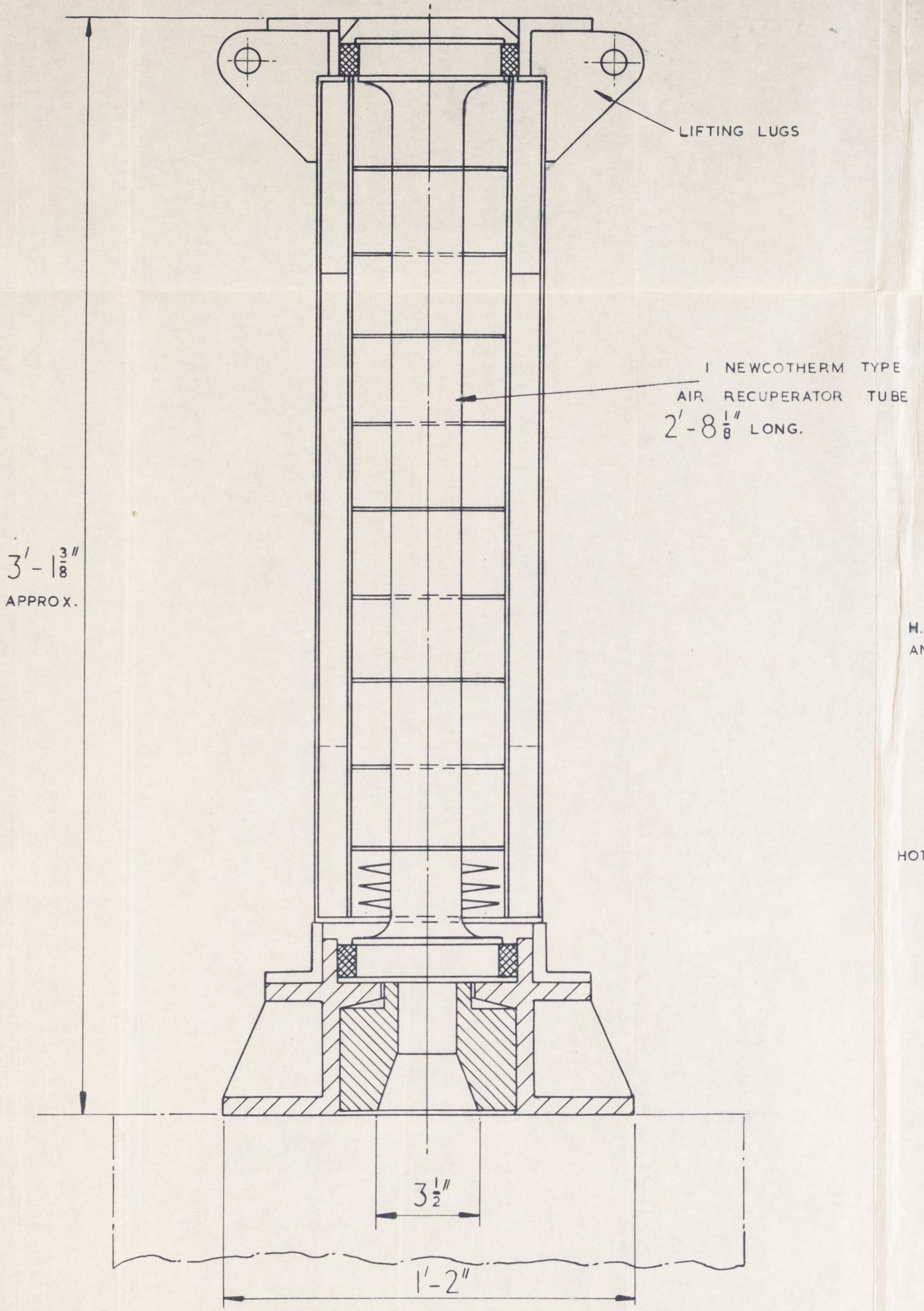
In order to preheat the combustion air, an air heater was incorporated in the apparatus. This consisted of a Newton Chambers needle tube recuperator fired by an auxilliary burner system. The actual recuperator, a detailed diagram of which is shown in Fig. 4-2, consisted of a single heat exchange tube having an extended needle surface both inside and out. The casework was such that the air made ten passes across the outside of the tube, whilst the waste gases made a single pass through the tube.

H.R.C.I. = HEAT RESISTING CAST IRON

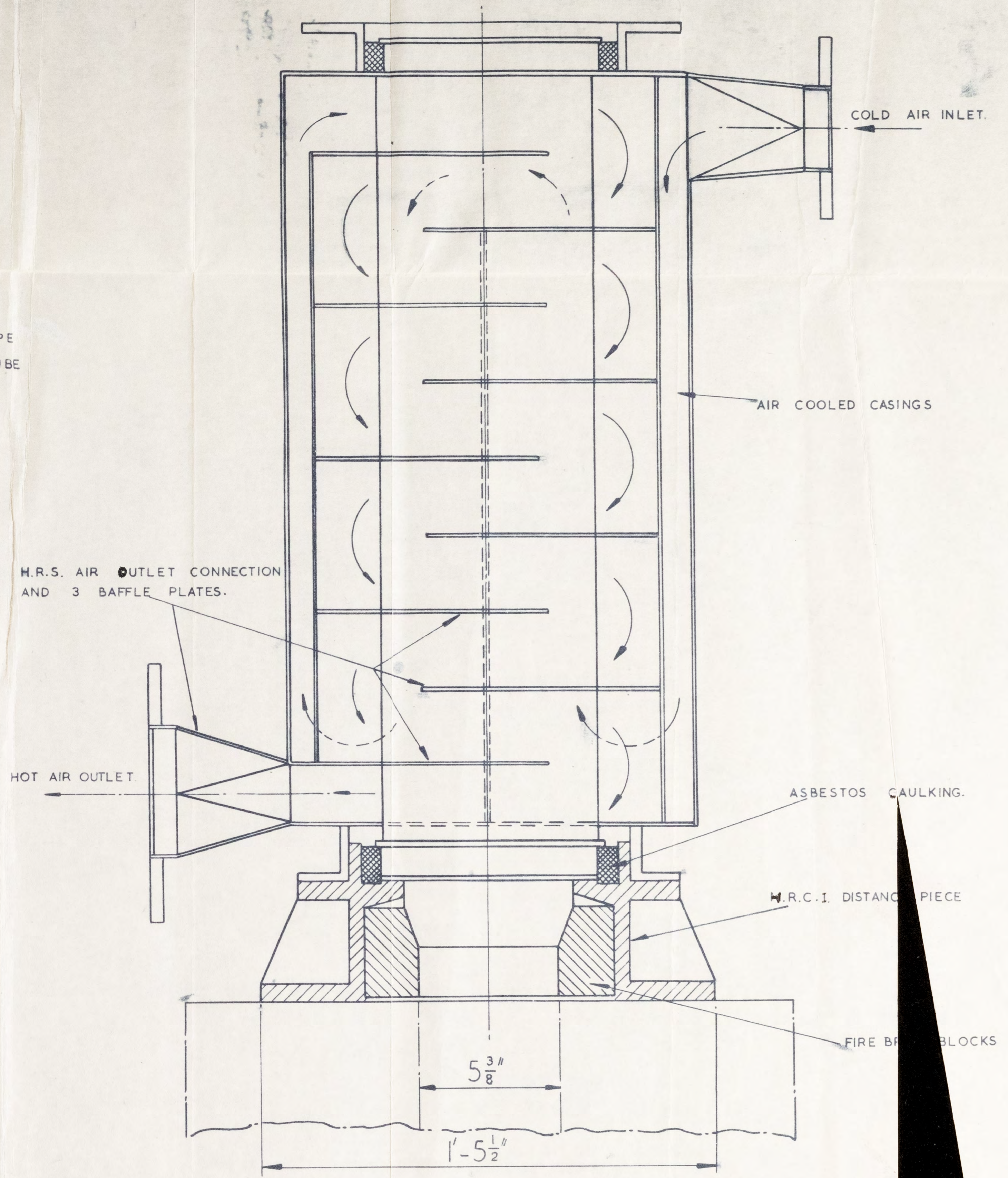
H.R.S. = HEAT RESISTING STEEL



SECTIONAL PLAN



CROSS SECTION



SECTIONAL ELEVATION

F

The manufacturers specification of the recuperator was as follows:-

Quantity of waste gas	2350 s.cu.ft./hr.
Waste gas inlet temperature	1000°C
Maximum safe waste gas temperature	1050°C
Waste gas outlet temperature	593°C
Quantity of air	2000 s.cu.ft./hr.
Air inlet temperature	15°C
Air outlet temperature	500°C
Resistance on waste gas side	0.22 in.w.g.
Resistance on air side	5.0 in.w.g.
Length of heat exchange element	2ft. 8¼in.
Approximate weight of recuperator	7 cwt.

The manufacturers stated that the air preheat temperature varied directly as the waste gas temperature: thus for the above recuperator

$$T_a \triangleq 0.5\ T_w$$

where T_a = air preheat temperature (°C)

T_w = waste gas temperature (°C)

When installed as an air heater the recuperator was fired by means of a ¾ inch Midlands Research Station Type B tunnel burner. In order to reduce the temperature of the combustion products emerging from the tunnel, from approximately 2000°C to below the safe maximum of 1050°C before entering the recuperator, secondary air was introduced into the refractory lined mixing chamber situated between the burner tunnel and the recuperator inlet. The layout of burner, mixing chamber and recuperator can be seen from Fig. 4-3.

The secondary, or diluent air was taken from the burner air supply and could be adjusted by means of a quadrant cock. However, since both burner and secondary air were taken from a common supply the quadrant cock was only necessary as a presetting device. Once set to give the required waste gas temperature the proportion of secondary air to high temperature combustion products remained substantially constant regardless of burner throughput, resulting in an almost constant waste gas temperature at the inlet to the recuperator.

A chromel/alumel thermocouple positioned in the gas stream at the inlet to the recuperator enabled the inlet waste gas temperature to be measured and, in association with an Ether Transitrol Indicating Temperature Controller, provided for automatic shut down of the heater should the waste gas temperature rise above the safe limit of 1050°C.

To facilitate accurate control of the outlet air temperature to any desired level in the range 0 - 500°C the heater was fitted with a three term proportional controller which enabled the heat input to the recuperator to be automatically adjusted according to a signal from a chromel/alumel thermocouple situated in the outlet air pipe from the recuperator. The equipment used was a Leeds and Northrup "Speedomax Type H" potentiometric temperature recorder and Series 60 Electrical Proportioning Control unit feeding a Honeywell Type 930 A Actuator.

The actuator was coupled to a 2 inch Selas valve (s) controlling the air supply to the recuperator burner.

SECTIONED ELEVATION OF THE AIR HEATER SHOWING LAYOUT OF BURNER, MIXING CHAMBER AND RECUPERATOR

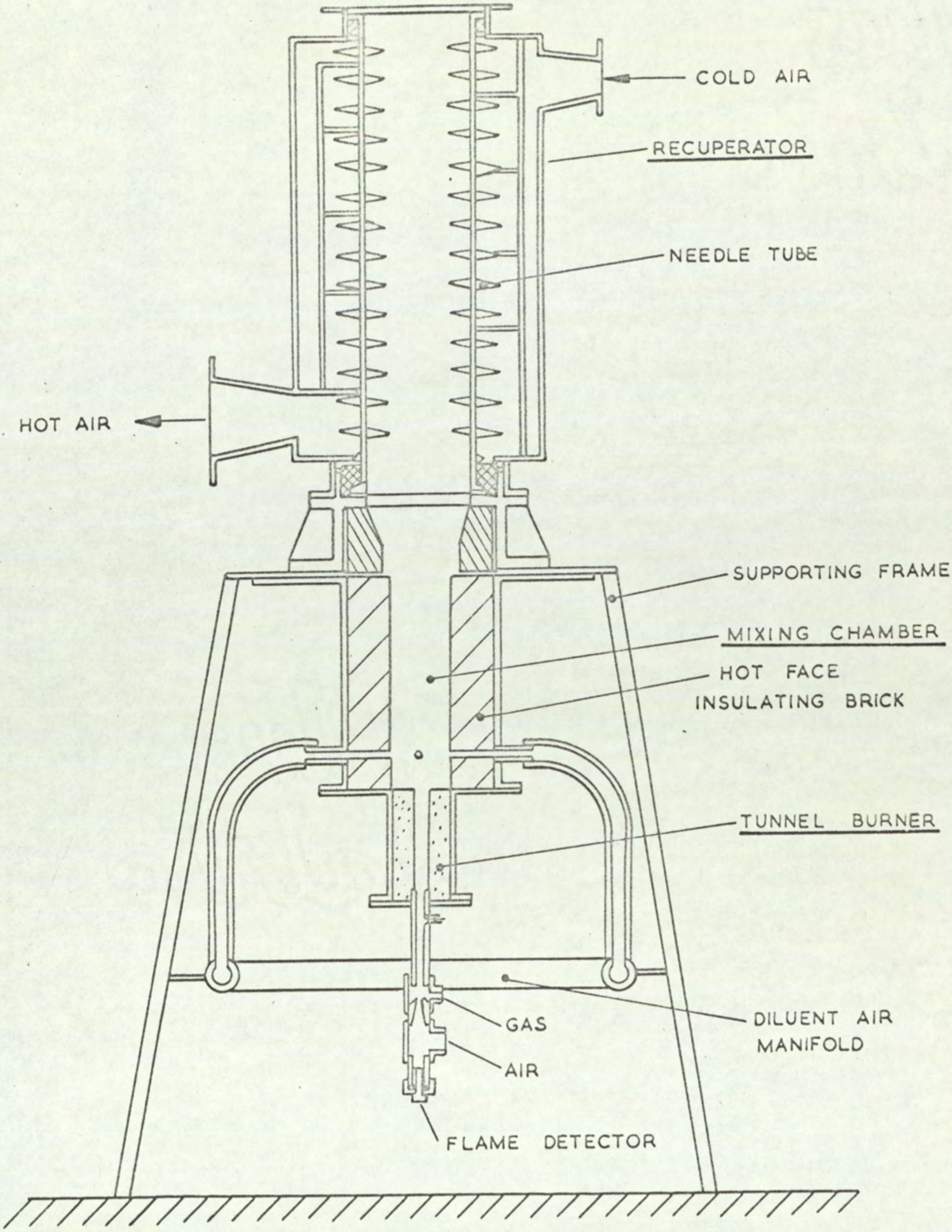


FIG. 4-3

It will be noticed that the differential pressure across the valve was maintained at a constant value regardless of throughput by means of a back loaded appliance governor. This technique was used in order to obtain a linear relationship between valve position and flow and is discussed in detail in Appendix IX of this paper.

In order to facilitate easy start-up, and safe unattended operation of the air heater, the recuperator burner was fitted with high-tension spark ignition and infra-red flame protection. A semi-automatic control system of the type described by Atkinson and the author[5] and employing an Ether Type 700 unit was used. A 1 inch Alcon Solenoid Valve (V) was employed for safety shut-off duty.

Provision was made to measure the various pressures in the injector system by means of inclined manometers. Two such instruments calibrated 0-3in.w.g. were used to measure the differential pressure across the air metering orifice and the negative pressure developed by the injector whilst a third, calibrated 0-1.5in.w.g. was used to measure the outlet pressure of the "zero governor".

A travelling microscope, mounted on a rigid table adjacent to the injector and located against it, enabled the peg position to be measured accurately throughout its stroke. This allowed for the actual air nozzle area to be calculated from measurements taken whilst the injector was working.

V

PRELIMINARY TESTING OF THE AIR HEATER

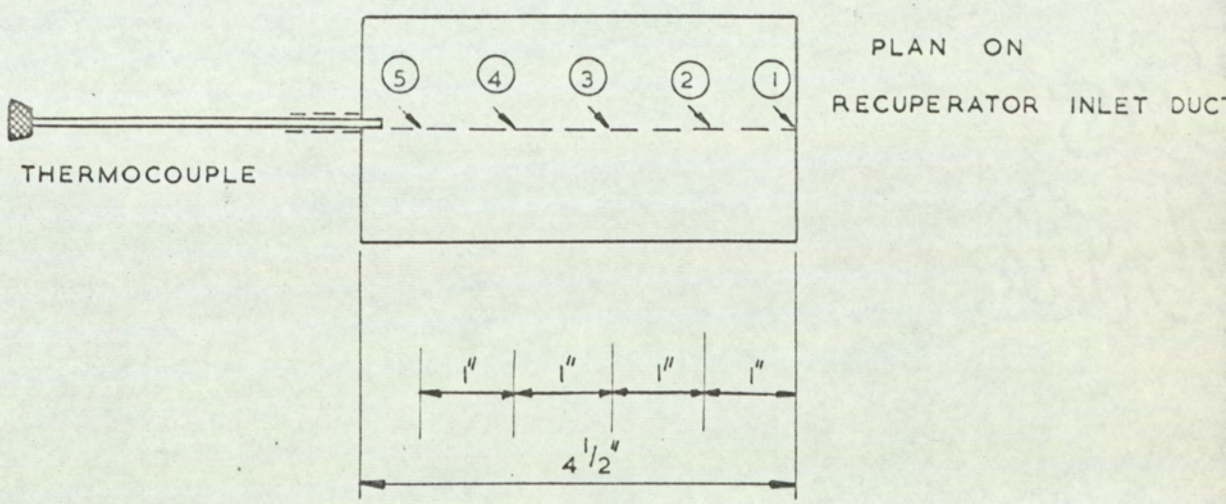
Before doing any experiments on the compensating injector it was decided to investigate the performance of the test equipment, partly in order to check that it would give the required output and partly to study the performance of the recuperator itself.

The air heater burner was set up in the usual manner. The air/gas ratio being set to stoichiometric, and at maximum throughput the diluent air was adjusted to give a combustion product temperature at the inlet to the recuperator of approximately 950°C, as indicated on the Ether Transitrol.

At this stage a check was made of the products temperature distribution across the inlet to the recuperator. This was done by sliding the thermocouple in and out along the major axis of the duct. The resulting distribution is shown in Fig. 5-1.

The skewed nature of the distribution was thought to be due to inaccuracies in temperature measurement, due to losses along the thermocouple, this being approximately ¼in. diameter. However, even if the distribution was real, it was not considered to be of sufficient magnitude to effect the overall efficiency of the recuperator, and therefore work was continued to carry out a heat balance on the system.

In order to do this, provision was made to measure the various temperatures in the system as follows:-



STATION	TEMPERATURE
1	960° C
2	970° C
3	970° C
4	953° C
5	890° C

FIG. 5-1

The inlet gas temperature to the recuperator was measured with the Ether Transitrol referred to above.

The outlet gas temperature from the recuperator was measured by means of an Atkinson Suction Pyrometer[8] positioned so as to measure the gas temperature at the centre of the outlet.

The inlet air was assumed to be at room temperature and the outlet air temperature could be measured by means of the chromel/alumel thermocouple feeding the Leeds and Northrup Recorder Controller.

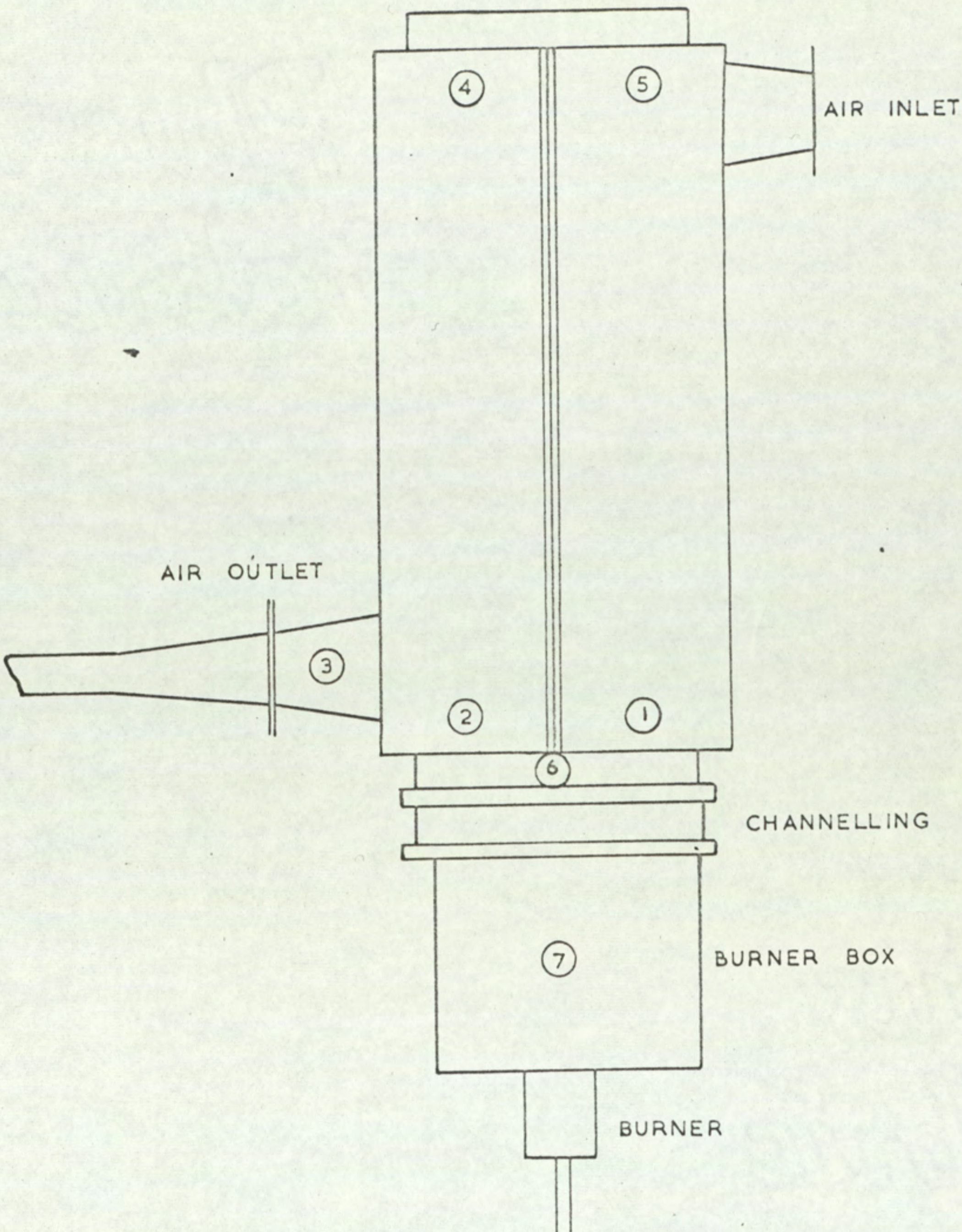
The surface temperature of the recuperator casework and air delivery pipe was measured by means of an Ether Surface Pyrometer.

The Stations at which surface temperature measurements were taken are indicated in Fig. 5-2.

The gas flow rate to the burner was measured by means of a dry meter situated in the gas supply line to the equipment, and the volume flow rate of products through the recuperator was estimated from knowledge of the gas flow rate, calorific value, temperature of combustion products entering the recuperator and published heat content data[6].

The air flow through the recuperator could be measured by means of the air rotameter.

Measurement were taken under conditions of low and high air flow, the low flow air rate being 250 cu.ft./hr. and the high flow air rate being 1750 cu.ft./hr. Under both conditions, the burner was operated at its maximum rate. The results obtained from these experiments are given below in Tables 5-1 and 5-2, and the results of the heat balance calculations, which are detailed in Appendix X of the paper, are also shown.



4
5
AIR INLET
AIR OUTLET
3
2
1
6
CHANNELLING
7
BURNER BOX
BURNER

FIG. 5-2

Table 5 - 1

Air flow rate	250 ft^3/hr.
Gas flow rate	196 ft^3/hr.
Flue gas temperature	576.5°C
Inlet gas temperature	970 °C
Inlet air temperature	17 °C
Outlet air temperature	258 °C
Air pipe temperature	120 °C

	Station	Temperature
Recuperator Case Temperature --	1	195°C
	2	220°C
	3	180°C
	4	130°C
	5	130°C
	6	205°C
	7	75°C

Heat Balances

Total heat input	79,000 Btu/hr.
Flue loss	44,400 Btu/hr.
Heat given up by combustion products	34,600 Btu/hr.
Heat gained by air	1,976 Btu/hr.
Wall losses	9,935 Btu/hr.

Table 5 - 2

Air flow rate	1,750 ft^3/hr.
Gas flow rate	196 ft^3/hr.
Flue gas temperature	510.6°C
Inlet gas temperature	975 °C
Inlet air temperature	17 °C
Outlet air temperature	385 °C
Air pipe temperature	120 °C

	Station	Temperature
Recuperator Case Temperature --	1	195°C
	2	220°C
	3	190°C
	4	115°C
	5	115°C
	6	198°C

Heat Balances

Total heat input	79,000 Btu/hr.
Flue loss	38,750 Btu/hr.
Heat given up by combustion products	40,250 Btu/hr.
Heat gained by air	21,500 Btu/hr.
Wall losses	9,467 Btu/hr.

It will be seen that in both cases a rather low air temperature was achieved compared with the required air temperature of 500°C, and also it will be noticed that a heat balance was not obtained. A further heat balance experiment was therefore carried out, in which a more detailed temperature survey of the recuperator casework was undertaken. Also since it was thought that the error in the above experiment may have been due to unsteady conditions, care was taken to ensure that the system had reached steady state before any measurements were taken. In addition, the inlet products temperature was checked by means of an Atkinson Suction Pyrometer, as well as the Ether Thermocouple, to obviate any possible errors due to radiation losses from the thermocouple. The outlet air temperature was similarly measured with a suction pyrometer, although it was felt that the velocity conditions existing in the outlet air pipe, where the thermocouple was situated, was sufficiently high to ensure an accurate temperature measurement. (In this condition, the convective heat transfer to the thermocouple is far larger than any radiation losses to the pipe walls).

The various points at which measurements of casework temperature were taken are shown in Fig. 5-3, and the results of experiments performed, are presented in Table 5-3.

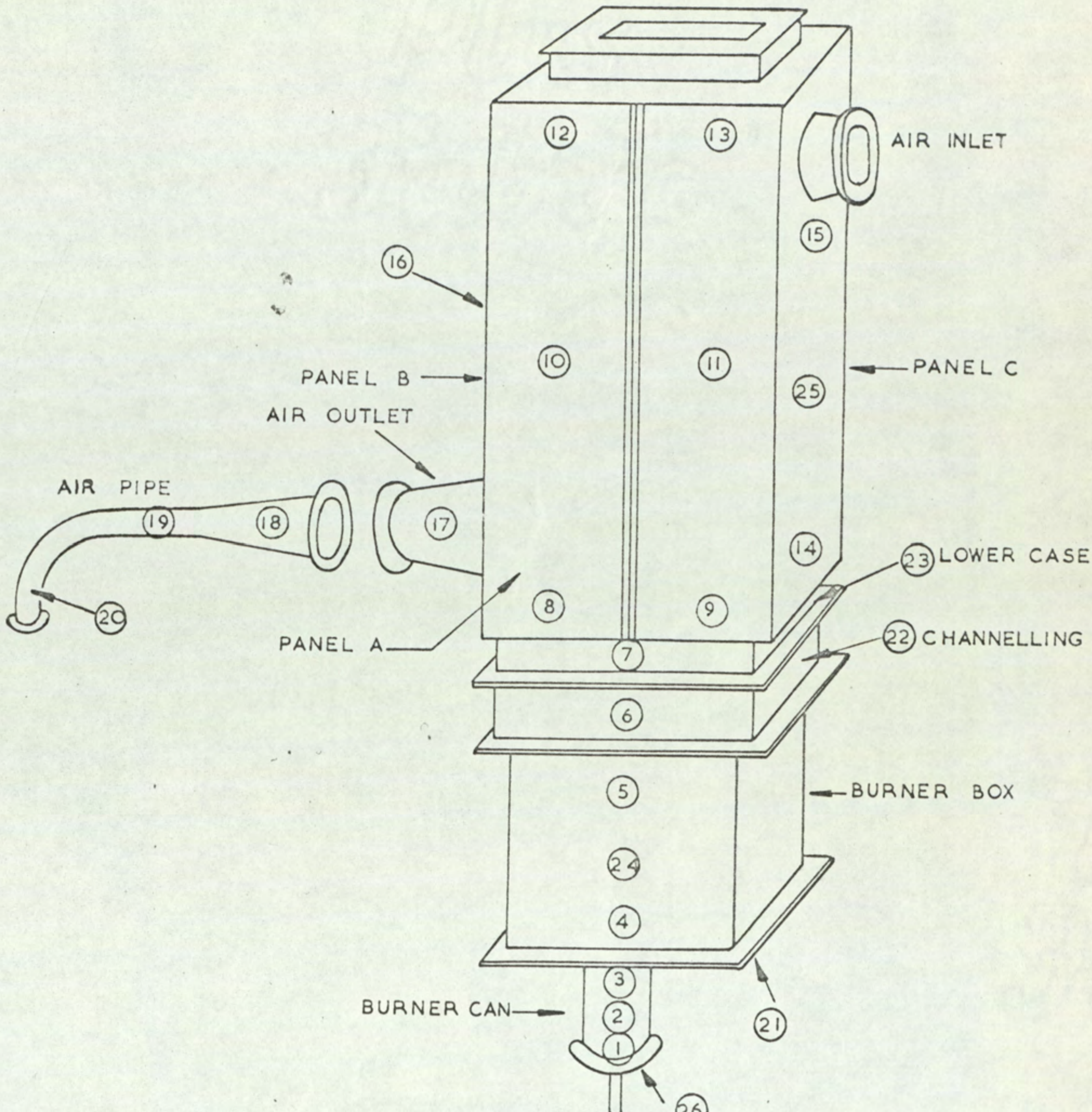

AIR INLET
PANEL B
AIR OUTLET
AIR PIPE
PANEL C
LOWER CASE
CHANNELLING
PANEL A
BURNER BOX
BURNER CAN
12
13
15
16
10
11
25
19
18
17
14
23
20
8
9
22
7
6
5
24
4
3
2
1
21
26

FIG. 5-3

Table 5 - 3

Air Flow Rate 250 ft^3/hr.

Gas Flow Rate 183 ft^3/hr.

Flue gas temperature	631.4°C
Inlet gas temperature	
Ether thermocouple	1,010 °C
Suction pyrometer	1,201 °C
Inlet air temperature	24 °C
Outlet air temperature	
at recuperator outlet	405 °C
at air pipe outlet suction pyrometer	295 °C
Chromel Alumel thermocouple	287 °C

Recuperator Case Temperature --

Station	Temperature
1	240°C
2	300°C
3	280°C
4	90°C
5	105°C
6	230°C
7	250°C
8	260°C
9	245°C
10	195°C
11	190°C
12	170°C
13	160°C
14	175°C
15	95°C
16	170°C

Table 5 - 3 (Cont).

Station	Temperature
17	180°C
18	140°C
19	120°C
20	90°C
21	200°C
22	200°C
23	235°C

Heat Balances

Total heat input	=	73,800 Btu/hr.
Heat input to recuperator	=	71,553 Btu/hr.
Flue loss	=	35,200 Btu/hr.
Heat given up by combustion products	=	36,353 Btu/hr.
Heat gained by air	=	3,120 Btu/hr.
Wall loss	=	16,766 Btu/hr.

The results of heat balance calculations, detailed in Appendix X, again showed considerable error, and critical examination of the possible causes led to a suspicion of the accuracy of the particular surface pyrometer used. This instrument was therefore compared with other instruments of similar type, and was found to be in error. A calibration curve was therefore plotted and the true casework temperatures were estimated from it.

Recalculation of the heat balance, also detailed in Appendix X, gave a resulting balance agreeing to within 2%. The final result of the heat balance calculations is given below in Table 5-4.

Table 5-4

Air Flow 250 ft^3/hr.

Total heat input	=	73,800 Btu/hr.
Heat input to recuperator	=	69,552 Btu/hr.
Flue loss	=	34,220 Btu/hr.
Heat given up by combustion products	=	35,332 Btu/hr.
Heat gained by air	=	3,120 Btu/hr.
Wall loss	=	31,543 Btu/hr.

In view of the uncertainty in these results, due to the need to estimate temperature, the experiment was repeated; additional measurements being taken of casework temperature by means of

(1) A surface pyrometer probe feeding a Honeywell recorder, calibrated 0 - 1200°C, and

(2) Cr/Al thermocouples slipped under the casing bolts, also feeding the Honeywell recorder.

These measurements were in addition to measurements taken with the inaccurate surface pyrometer.

The results of the repeated experiments are given below in Tables 5-5 and 5-6.

Table 5-5

Air flow rate 250 ft^3/hr.

Gas flow rate 180.2 ft^3/hr.

Flue gas temperature		701.1°C
Inlet gas temperature	(Ether thermocouple)	990 °C
	(Suction pyrometer)	1,150 °C
Inlet air temperature		22.5°C
Outlet air temperature	(at recuperator outlet)	403 °C
	(at pipe outlet)	
	(Suction pyrometer)	245 °C
	(Cr/Al thermocouple)	284 °C

	Station	Ether Surface Pyrometer °C	Honeywell Surface Pyrometer °C	Honeywell Casing Bolt Temperature °C
Recuperator case Temperature	2	350	327	
	24	140	140	
	17	230	208*	
	19	145	155	
	25	185	184	
	16	210	200	
	8	262	274	304
	12	220	228	235
	21	165	184	
	26	205	219	
	6			280

* Insufficient time of contact.

Table 5-6

Air flow rate	1,750 ft^3/hr.
Gas flow rate	181 ft^3/hr.

Flue gas temperature		574.8°C
Inlet gas temperature	(Ether thermocouple)	990 °C
	(Suction pyrometer)	1,144.5°C
Inlet air temperature		22.5°C
Outlet air temperature	(at recuperator outlet)	395 °C
	(at pipe outlet)	
	(Suction pyrometer)	372 °C
	(Cr/Al thermocouple)	369 °C

	Station	Ether Surface Pyrometer °C	Honeywell Surface Pyrometer °C	Honeywell Casing Bolt Temperature °C
Recuperator case Temperature	2	310	325	
	24	135	145	
	17	220	217	
	19	185	195	
	25	82	87	
	16	115	132	
	8	155	172	218
	12	105	125	140
	21	165	177	
	26	210	222	
	6			265

The results of the heat balance calculations, presented in Appendix X, are given in Tables 5-7 and 5-8 below, and it can be seen that the balance agreed to within 4% in the case of the low air rate experiment, and within 3% in the case of the high air rate experiment.

Table 5-7

Air Flow 250 ft^3/hr.

Heat given up by combustion products	29,100 Btu/hr.
Heat gained by air	3,122 Btu/hr.
Wall losses	27,321 Btu/hr.

Table 5-8

Air Flow 1,750 ft^3/hr.

Heat given up by combustion products	37,100 Btu/hr.
Heat gained by air	21,420 Btu/hr.
Wall losses	16,817 Btu/hr.

It will be noticed that in both cases the heat losses from the casing were large compared with the heat transferred to the air stream, and this was thought to be the cause of the comparatively low maximum air temperature achieved, i.e. 403°C with an air flow of 250 ft^3/hr. and 395°C with an air flow of 1,750 ft^3/hr.

Calculations showed that with an air flow of 1,750 ft^3./hr. it would only be necessary to increase the heat transfer to the air stream by 6,750 Btu/hr. in order to raise the temperature to 500°C. Since this rate of heat transfer was less than half the casing loss, it was thought possible to increase the air temperature considerably by lagging the outside of the recuperator.

The recuperator was therefore lagged with one layer of 2 inch diameter asbestos rope, and the air delivery pipe with one layer of ¾ inch diameter asbestos rope.

The heat balance experiments were repeated at both high and low flow air rates. The casing temperature being monitored, to avoid possible overheating. This was done by means of Cr/Al thermocouples slipped under the casing bolts. An approximate measurement was made of the surface temperature of the asbestos rope in order to estimate casing losses. The results of the experiments are given in Tables 5-9 and 5-10 below.

Table 5-9

Air flow rate 250 ft^3/hr.

Gas flow rate 181 ft^3/hr.

Flue gas temperature		774°C
Inlet gas temperature	(Suction pyrometer	1,225°C
	(Cr/Al thermocouple	1,035°C
Inlet air temperature		20°C
Outlet air temperature	(at recuperator outlet	535°C
	(at air pipe outlet	366°C

(cont.)

Table 5-9 (cont.)

	Station	Casing bolt Temperature
Recuperator case Temperature	8	563°C
	12	524°C
	6	397°C
Asbestos surface temperature		155°C

Table 5-10

Air flow rate	1,750 ft^3/hr.
Gas flow rate	185.3 ft^3/hr.

Flue gas temperature		625.6°C
Inlet gas temperature		
(Suction pyrometer		1,266 °C
(Cr/Al thermocouple		1,040 °C
Inlet air temperature		25 °C
Outlet air temperature		
(at recuperator outlet		472 °C
(at air pipe outlet		442 °C
	Station	**Casing bolt Temperature**
Recuperator case Temperature	8	384°C
	12	282°C
	6	444°C
Asbestos surface temperature		114°C

In the case of the low flow experiment the measurements were made before steady state conditions were reached, since it was felt that damage to the recuperator might occur if the casework temperatures were allowed to rise any higher. When the measurements were taken, these temperatures were rising slowly.

The above experiment showed that an adequate air temperature could be achieved with an air flow of 250 cu.ft./hr., the outlet temperature being 535°C (c.f. 403°C unlagged) and the maximum casing temperature being 563°C (c.f. 304°C unlagged).

With an air flow of 1,750 cu.ft./hr. the steady state outlet air temperature was 472°C (c.f. 395°C unlagged) and the maximum casing temperature was 384°C (c.f. 172°C unlagged).

Heat balance calculations showed considerable error, which at the time was thought to be due to inaccurate measurement of the surface temperatures, but, in the light of later experience may have been due to air leakage from the recuperator.

Having demonstrated that the equipment was capable of delivering air at the required temperature, the system was put onto automatic control. The burner throughput was automatically adjusted to give the required air temperature by the Leeds and Northrup three term controller. By appropriate adjustment of the proportional band and the rate and reset functions a high degree of control was achieved.

With an air flow of 250 cu.ft./hr. through the recuperator and the outlet air temperature controlled to 500°C the maximum steady state casing temperature was found to be 504°C. In view of this rather high temperature, it was decided to restrict the initial experiments on the compensating injector to lower air temperatures, and only proceed to higher air temperatures when much of the work had been completed, and a short recuperator life was unimportant.

VI

EXPERIMENTS ON THE MARK I INJECTOR

In the first instance, the compensating injector described in Section III was given a preliminary test on the experimental rig described in Section IV. In setting up the injector/burner system the technique normally adopted in setting up an air blast burner was used, in that the burner was lit, and with the air supplied at a temperature of 500°C, the gas restrictor was adjusted to give a 4:1 air/gas ratio at a throughput of 500 cu.ft./hr. of gas. Throughput was then reduced and the zero governor outlet pressure was adjusted to give a 4:1 air/gas ratio with a throughput of 75 cu.ft./hr. of gas.

As a preliminary experiment, the throughput was then adjusted to approximately 200 cu.ft./hr. of gas and the air temperature progressively reduced. The results of this simple experiment are given below in Table 6-1 from which it can be seen that the air/gas ratio changed considerably as the air temperature was reduced.

Table 6-1

Air Preheat Temperature °C	Air Flow Rate ft^3/hr.	Gas Flow Rate ft^3/hr.	Air/Gas Ratio
437	800	194	4.12
216	800	148	5.4
196	800	148	5.4

Experiments were therefore performed to check the ability of the compensating injector to equalize the differential pressures developed across the air metering orifice and the gas restrictor. For this experiment the zero governor was adjusted to give atmospheric pressure, the gas restrictor being left at the same setting as in the previous experiment. Experiments were carried out with air at room temperature, at a temperature of approximately 300°C, and at a temperature of approximately 240°C. Burner throughput was adjusted over the working range of the burner and measurements taken of the differential pressure across the air metering orifice, the outlet pressure from the zero governor and the suction developed by the injector. Also noted was the relative position of the tapered plug and air nozzle. The results of these experiments are presented below in Tables 6-2, 6-3 and 6-4.

Table 6-2

Air at Room Temperature

Air Flow $ft^3/hr.$	ΔPc in.w.g.	Suction in.w.g.	Zero governor Outlet pressure in.w.g.	ΔPg in.w.g.	$\frac{\Delta Pa}{\Delta Pg}$
500	0.10	0.045	0.02	0.065	2.2
750	0.23	0.13	0.02	0.15	1.77
1,000	0.41	0.24	0.01	0.25	1.70
1,250	0.65	0.40	-0.005	0.395	1.625
1,500	0.95	0.61	-0.01	0.60	1.56
1,750	1.25	0.84	-0.01	0.83	1.512
2,000	1.67	1.12	-0.02	1.10	1.49

Table 6-3

Air Temperature approx. 300°C

Air Flow $ft^3/hr.$	Air Pressure in.w.g.	Air Temp. °C	ΔPa in.w.g.	Suction in.w.g.	Zero governor Outlet pressure in.w.g.	ΔPg in.w.g.	$\frac{\Delta Pa}{\Delta Pg}$
500	1.38	302	0.105	0.08	0.02	0.10	1.05
750	2.94	310	0.24	0.24	0.01	0.25	0.96
1,000	4.95	308	0.41	0.46	-0.003	0.457	0.895
1,250	7.90	311	0.65	0.75	-0.01	0.74	0.88
1,500	11.25	315	0.94	1.10	-0.015	1.085	0.856
1,750	14.95	318	1.26	1.50	-0.020	1.48	0.851
2,000	19.3	318	1.66	2.03	-0.035	1.995	0.831

Plug retracted 1.3 cm.

Table 6-4

Air Temperature approx. 240°C

Air Flow $ft^3/hr.$	Gas Flow $ft^3/hr.$	Air/Gas Ratio	Air Pressure in.w.g.	Air Temp. °C	ΔPa in.w.g.	Suction in.w.g.	Zero Governor Outlet Pressure in.w.g.	ΔPg in.w.g.	$\frac{\Delta Pa}{\Delta Pg}$
500	95	5.26	1.25	230	0.10	0.06	0.02	0.08	1.25
750	148	5.06	2.72	238	0.24	0.20	0.02	0.22	1.09
1,000	200	5.00	4.55	236	0.41	0.38	0.018	0.398	1.03
1,250	257	4.86	7.15	238	0.65	0.63	0.000	0.63	1.03
1,500	310	4.94	10.1	238	0.94	0.92	-0.007	0.913	1.03
1,750	361	4.85	13.5	240	1.25	1.27	-0.01	1.26	0.992
2,000	418	4.79	17.75	240	1.68	1.71	-0.02	1.69	0.995

Plug retracted 1.28 cm.

A further experiment to determine the variation of air/gas ratio with temperature was performed, the ratio being set to 3.88:1 at an air temperature of 385°C. Measurements were again taken of air and gas flow rates, the differential pressure across the air orifice, the suction developed by the injector and the zero governor outlet pressure. The results of this experiment are given in Table 6-5.

Table 6-5

Air Flow $ft^3/hr.$	Gas Flow $ft^3/hr.$	Air/Gas Ratio	Air Temp. °C	ΔPa in.w.g.	Suction in.w.g.	Zero Governor Outlet Pressure in.w.g.	ΔPg in.w.g.	$\frac{\Delta Pa}{\Delta Pg}$
1,000	258	3.88	385	0.41	0.425	0.000	0.425	0.965
1,020	228	4.47	260	0.44	0.32	0.01	0.33	1.33
1,030	218	4.72	220	0.44	0.28	0.02	0.30	1.47
1,040	213	4.89	189	0.45	0.26	0.02	0.28	1.61
1,025	204	5.03	162	0.44	0.25	0.02	0.27	1.63
1,030	196	5.25	150	0.44	0.225	0.02	0.227	1.94

From the results quoted in Tables 6-1 to 6-5, it can be seen that the injector was not controlling as expected, a considerable difference existing between the differential pressure across the air orifice and that across the gas restrictor. From the magnitude of the errors it was thought that one or more of the diaphragm chambers may have been leaky, and these were therefore pressure tested to 1 P.S.I.G. A leak was in fact found in one chamber, the leak occuring where the plug passed through a small diaphragm. This was rectified and the subsequent test showed the chamber to be adequately pressure tight. The maximum leak rate from any chamber was 0.016 cu.ft./hr. at 1 P.S.I.G. When the initial experiments were repeated however, very similar results were obtained; indicating that the error in ratio control was not in fact due to the leak. Detailed experiments were therefore carried out to study the forces acting upon the diaphragm assembly, and to determine the cause of error in control performance.

In designing the injector, it had been assumed that the only forces which would restrict movement of the plug, and so cause error in balancing the differential pressures across the air metering orifice and gas restrictor, would be those due to friction in the plug bearings which was assumed to be negligibly small. However, in view of the results discussed above, it was thought that friction may in fact be greater than had been estimated and that there may possibly be additional forces due to elasticity of the diaphragms, particularly the secondary diaphragms, since they were extended fully at the extreme position of the plug.

Initially therefore an experiment was performed to determine the frictional forces acting on the plug, and this was done simply by inclining the injector assembly to the horizontal and increasing the angle of inclination until movement of the plug was noticed. The angle at which movement just commenced was measured and from this the coefficient of limiting friction was calculated. The coefficient of friction μ thus found was 0.024; compared with a typical value for ball bearings of 0.001. By comparison with the friction observed when the diaphragms were removed it was observed that the high value of μ was due largely to stiffness. The frictional force also appeared to vary according to the position of the plug, presumably due to the varying convolutions of the diaphragms. The value of $\mu = 0.024$ corresponded to a differential pressure acting across the diaphragms of 0.03in.w.g., and on the basis of a 5:1 turndown in throughput would cause an error in air/gas ratio on turndown such that if the required ratio was 4:1 the actual ratio would lie between 3.42:1 and 5.06:1, whilst at maximum throughput the ratio would lie between 3.985:1 and 4.04:1. Comparing this error with that observed in the experiments at high flow above, it was obvious that friction was comparatively unimportant and that other factors, possibly elasticity in the diaphragms, must be the cause.

In order to study this effect an experiment was arranged in which the injector diaphragms could be loaded with various pressures and the position of the injector plug measured with a travelling microscope, without any air flow through the injector. This technique eliminated from the measurements any forces acting on the plug due either to aerodynamic drag or to changing section where the plug entered the air nozzle or thirdly to the annular jet from the air seal impinging on the front diaphragm.

A schematic diagram of the apparatus is shown in Fig. 6-1. It can be seen that the pressures were applied so that the slack in both diaphragms took up the position existing if the injector were operated normally. A negative pressure of about -0.5in.w.g. was applied to chamber 2 and a positive pressure of about 1in.w.g. applied to chamber 4. The pressures were measured accurately by means of inclined manometers and the position of the plug was noted. The pressure applied to chamber 4

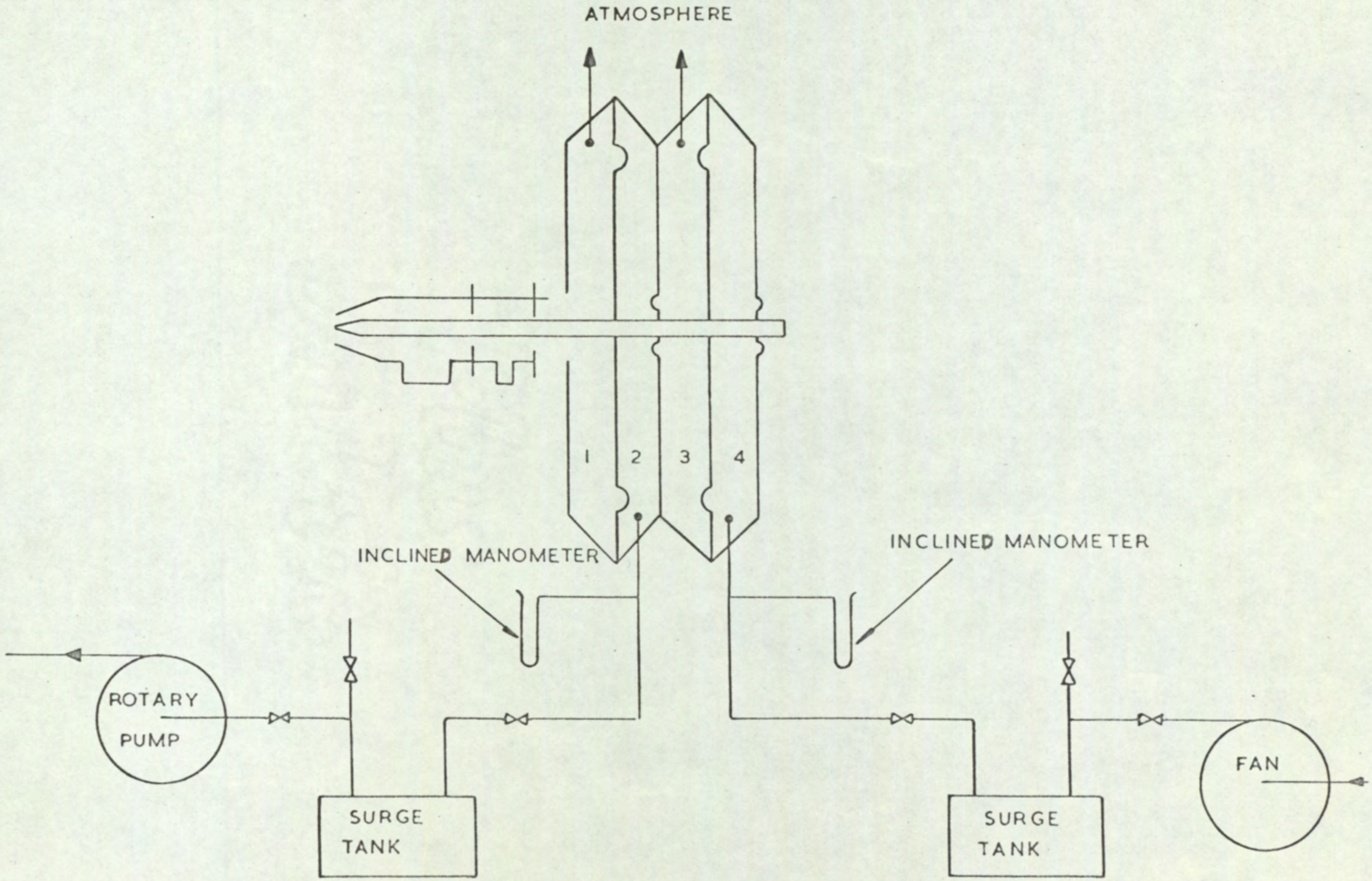

ATMOSPHERE
1
2
3
4
INCLINED MANOMETER
INCLINED MANOMETER
ROTARY PUMP
SURGE TANK
SURGE TANK
FAN

FIG. 6-1

was then slightly reduced and the measurements repeated. This process was continued until the pressure applied to chamber 4 was about 0.25in. w.g. when the process was reversed and measurements taken at a series of increasing pressures applied to chamber 4.

The resultant force on the plug was proportional to the numerical difference between the pressure applied to chamber 4 and the suction applied to chamber 2. Fig. 6-2 is a plot of plug position versus pressure difference.

Ideally this should be a step function or, allowing for friction in the movement, a step function with some hysteresis. The results shown in Fig. 6-2 suggested considerable elasticity in the diaphragms in addition to friction.

The above results were obtained using polymer diaphragms throughout, and it was thought that these may be contributing to the apparent elasticity in the movement, and it was therefore decided to replace the polymer diaphragms with leather, since although leather may be slightly stiffer than the polymer it is less elastic.

After re-assembly and leak testing the injector, the above experiment was repeated, the results being shown in Fig. 6-3. It will be noted from the increased slope of the curve and the greater hysteresis, that there appeared to be considerably less elasticity in the movement, but that friction had been increased.

It was concluded that the large diaphragms contributed most of the frictional force whilst the small diaphragms which suffered comparatively

VARIATION OF PLUG POSITION WITH PRESSURE DIFFERENCE USING POLYMER DIAPHRAGMS

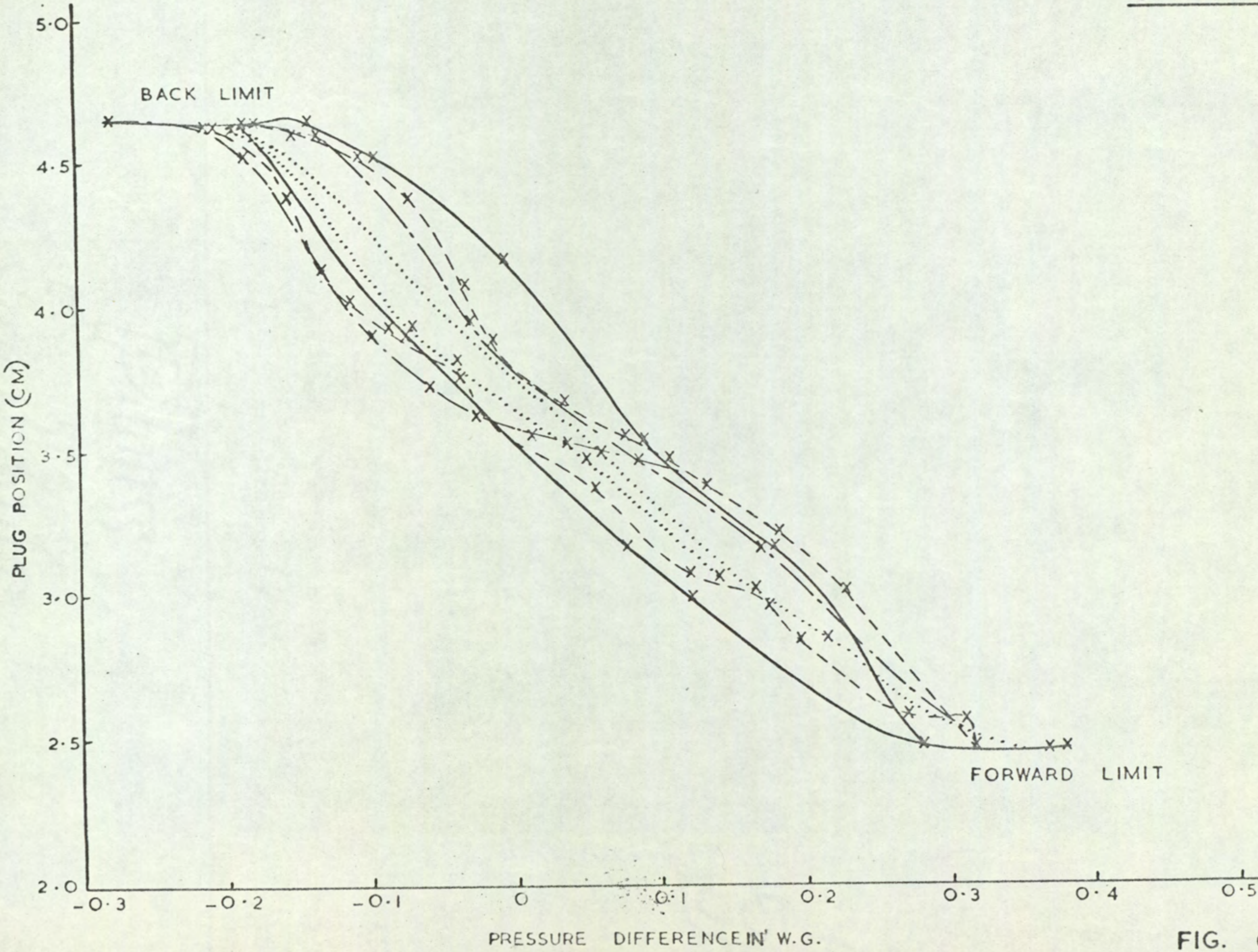


FIG. 6-2

A VARIATION OF PLUG POSITION WITH PRESSURE DIFFERENCE USING LEATHER DIAPHRAGMS

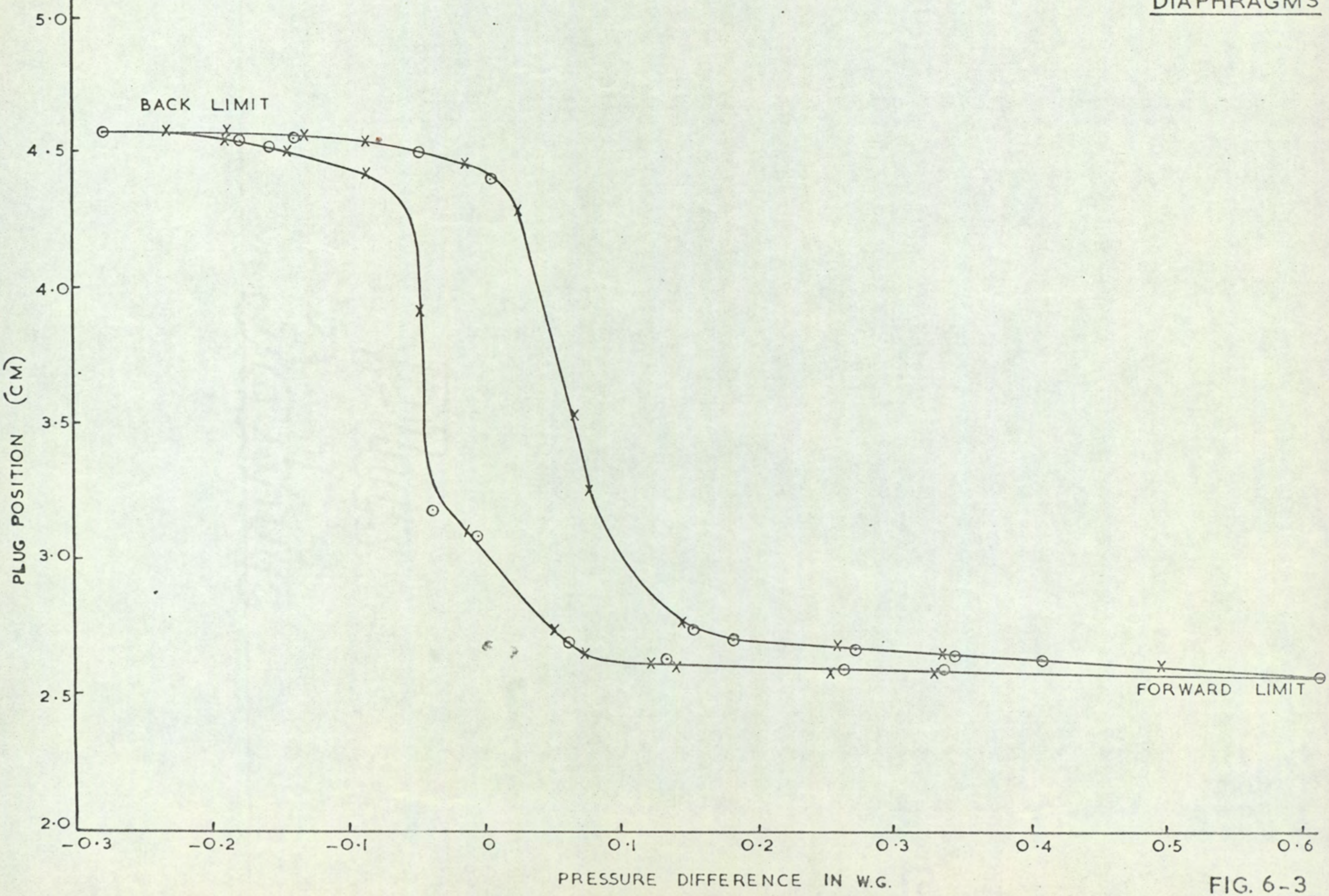


FIG. 6-3

much larger deflections contributed most of the elastic force. The injector was therefore re-assembled with polymer main diaphragms and leather secondary diaphragms and the above experiment was again repeated. Fig. 6-4 shows a plot of plug position versus pressure difference. The results obtained were practically identical to those obtained when polymer diaphragms were used throughout. Friction was in fact slightly reduced, being equivalent to 0.06in.w.g. compared with 0.10in.w.g., but the elasticity was exactly the same, thus eliminating the possibility that this was due to excessive deflection of the polymer secondary diaphragms considered above.

Consideration of the above results led to the conclusion that the "elasticity" in the movement may only be apparent and may in fact be due to changes in the effective area of the main diaphragms throughout the stroke of the plug. Since the injector was a force balance device this would mean that the differential pressures across the diaphragms would only be equal when the plug was in its mid position and would be unequal in all other positions. Also, since the diaphragms operated back to back, one would expect the variation of plug position with difference in differential pressures to be linear regardless of the characteristic of each individual diaphragm, in an analagous manner to a push-pull electronic amplifier, thus accounting for the linearity in Figs. 6-2, 6-3 and 6-4.

An experiment was therefore performed to find the variation in plug position when a variable differential pressure was applied to each diaphragm separately. To do this the injector was mounted vertically,

VARIATION OF PLUG POSITION WITH PRESSURE DIFFERENCE USING POLYMER MAIN AND LEATHER SECONDARY DIAPHRAGM.

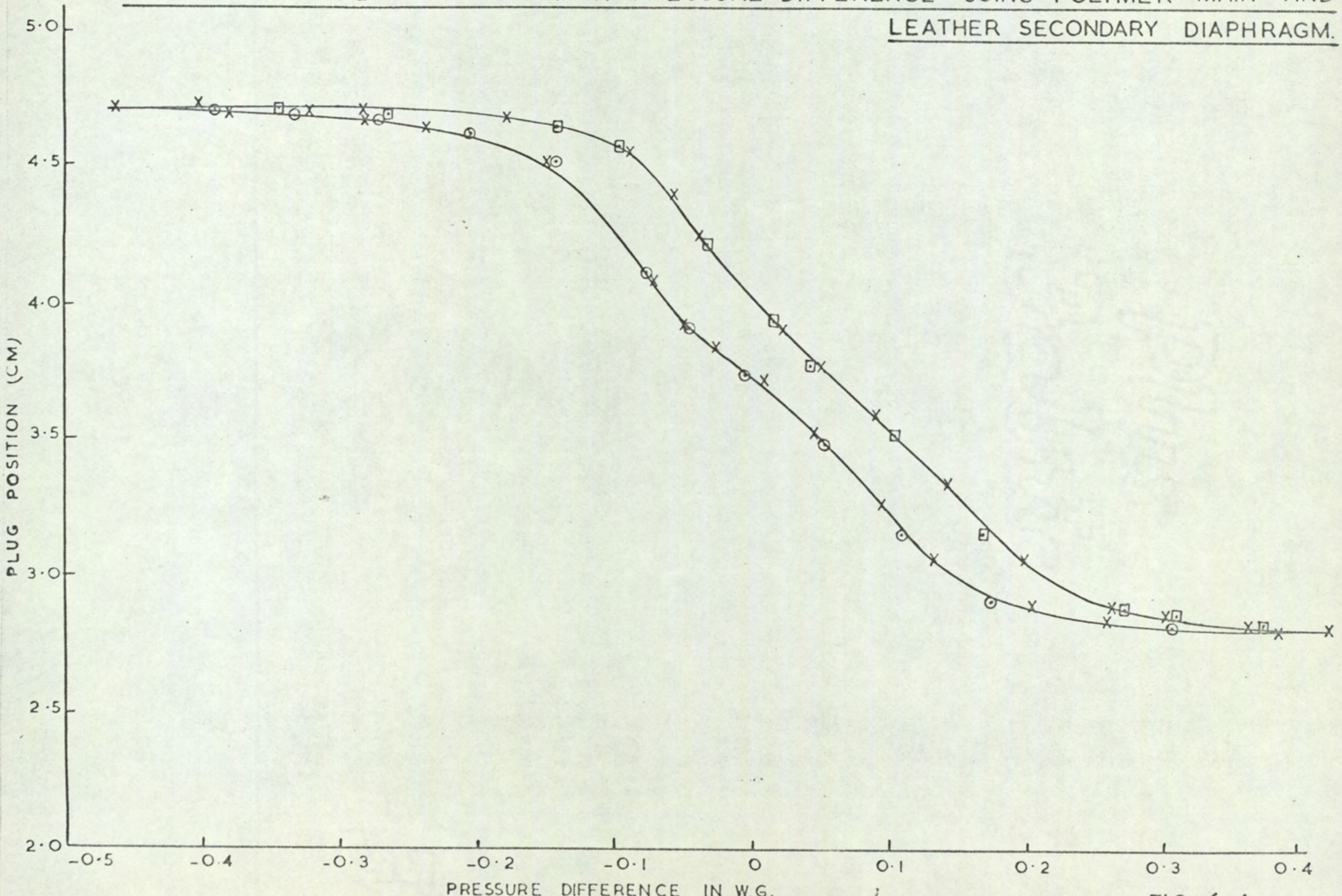


FIG. 6-4

see Fig. 6-5, and the plug position measured with a travelling microscope. In the case of the air diaphragm a measured pressure was applied to the underside of the diaphragm and in the case of the gas diaphragm a measured suction was applied to the upper side of the diaphragm. Thus both diaphragms took up the shape they would have when the injector was working normally.

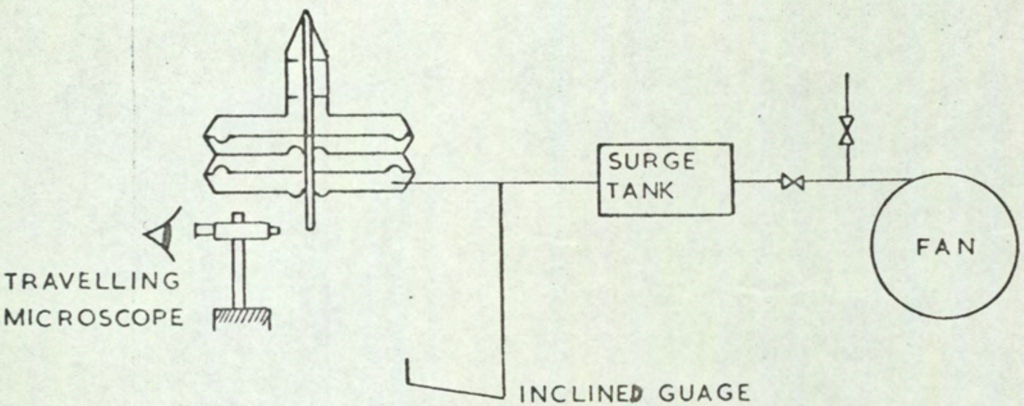
If there is no change in effective diaphragm area then a plot of peg position versus applied pressure (or suction) should be a step function, the step occuring when the applied pressure just balances the weight of the moving parts.

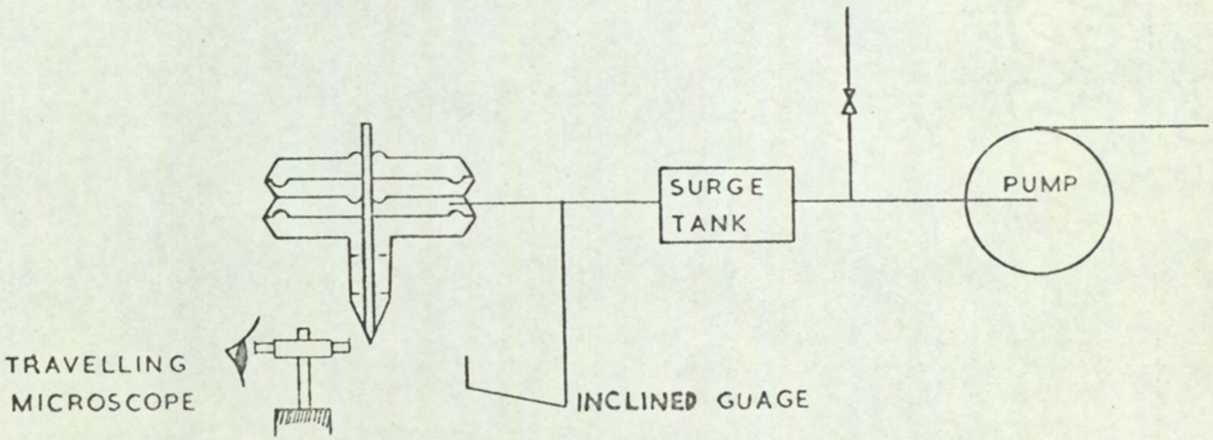
Figs. 6-6 and 6-7 show the actual variation of peg position with applied pressure (or suction). It can be seen that the plots are neither step functions nor straight lines; however the curves are of the same shape and thus with the two diaphragms acting back to back the variation of deflection with the difference in the differential pressures will be linear.

An experiment was then performed to measure the actual elasticity in the movement. The injector was again mounted vertically and the peg position could be measured by means of the travelling microscope. The injector movement was suspended on a nylon thread which was passed over two ball races (see Fig. 6-8) and to the other end of which was attached a balance pan and weights.

The position of the plug was then measured as weights were added to, and removed from, the balance pan. Fig. 6-9 is a plot of the variation of plug position with the effective differential pressure

APPARATUS FOR MEASURING VARIATION OF PLUG POSITION WITH PRESSURE APPLIED TO AIR DIAPHRAGM.





APPARATUS FOR MEASURING VARIATION OF PLUG POSITION WITH SUCTION APPLIED TO GAS DIAPHRAGM.

FIG. 6-5

VARIATION OF PLUG POSITION WITH PRESSURE

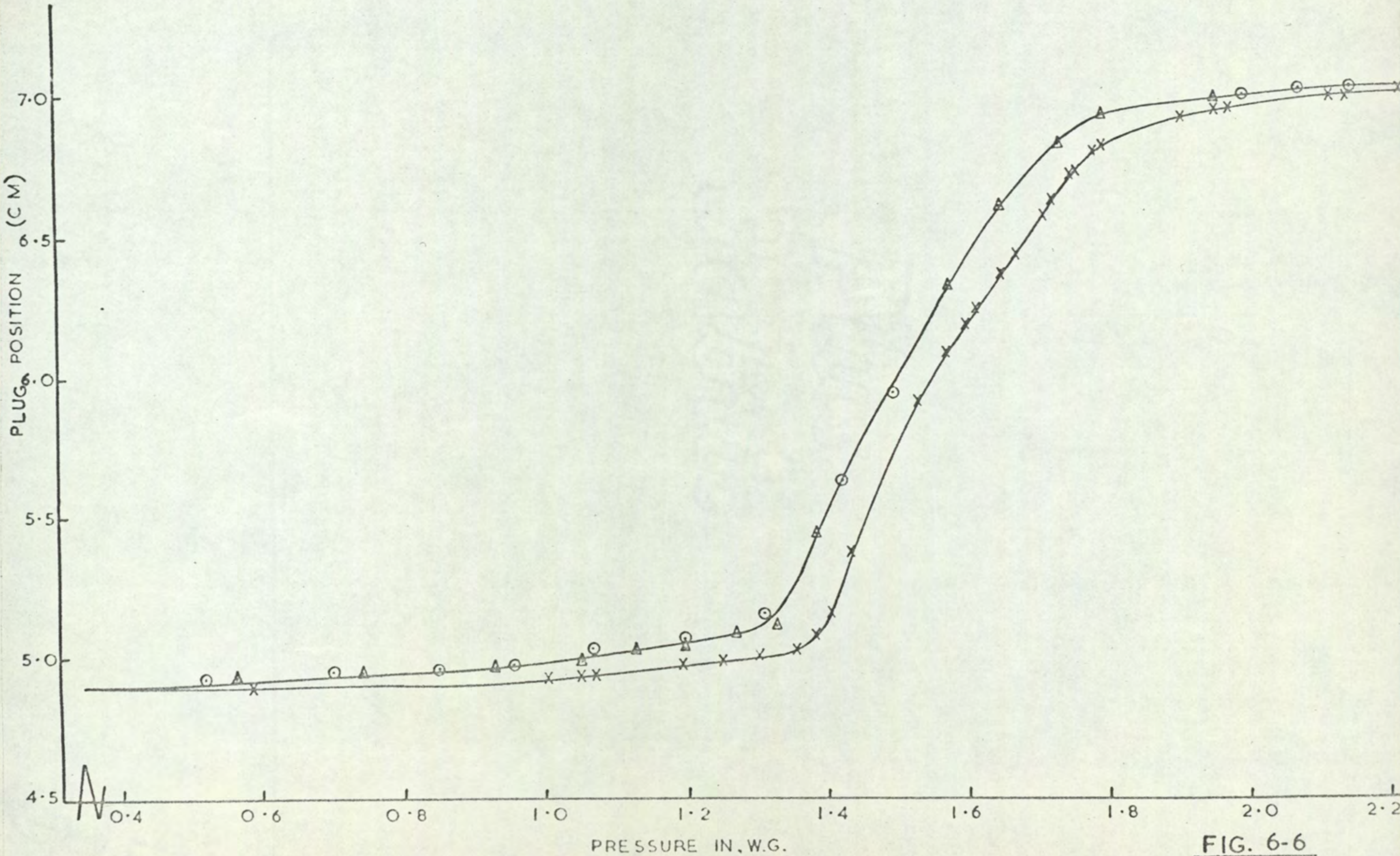


FIG. 6-6

VARIATION OF PLUG POSITION WITH SUCTION

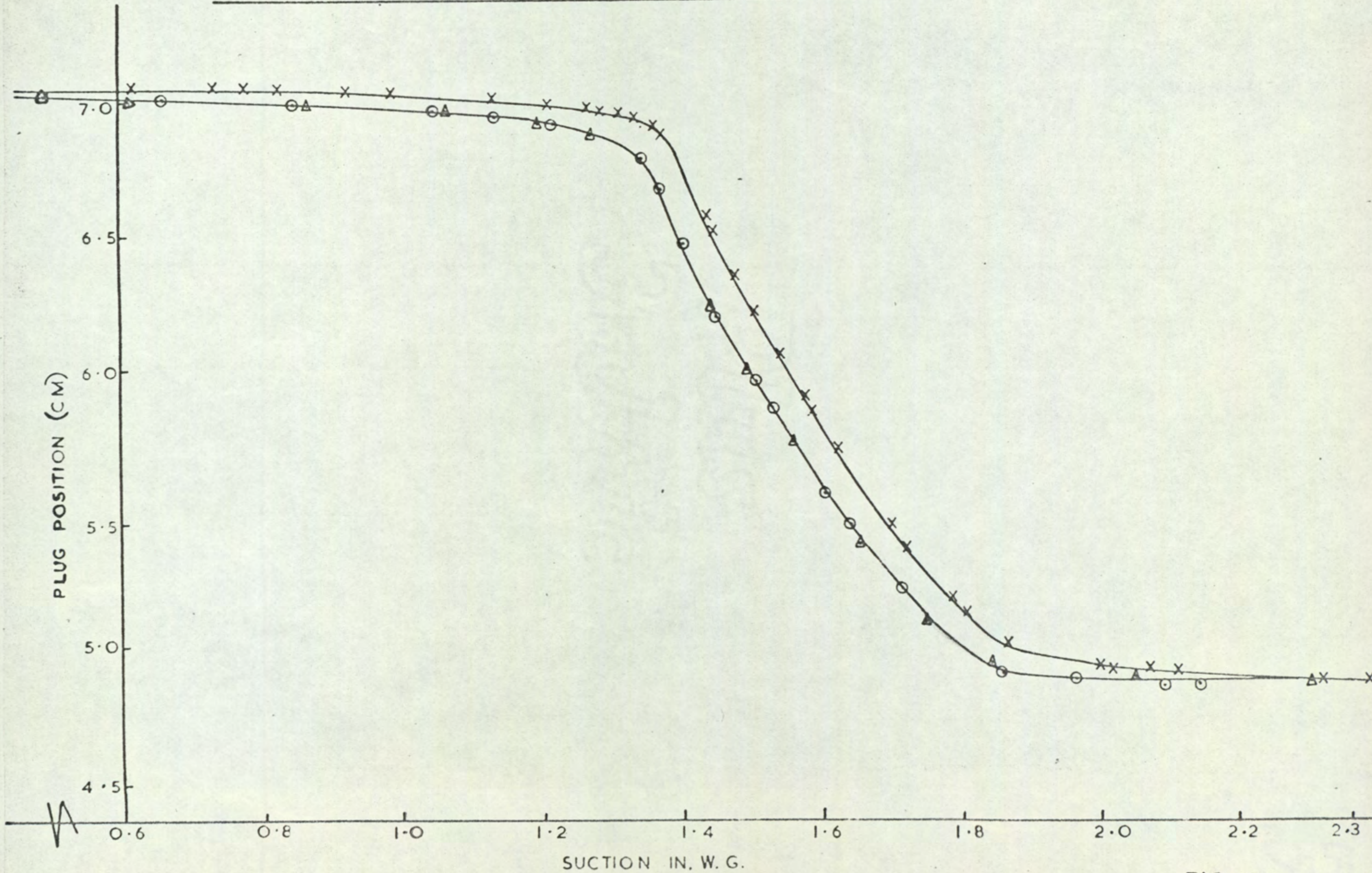


FIG 6-7

APPARATUS FOR MEASURING ACTUAL ELASTICITY.

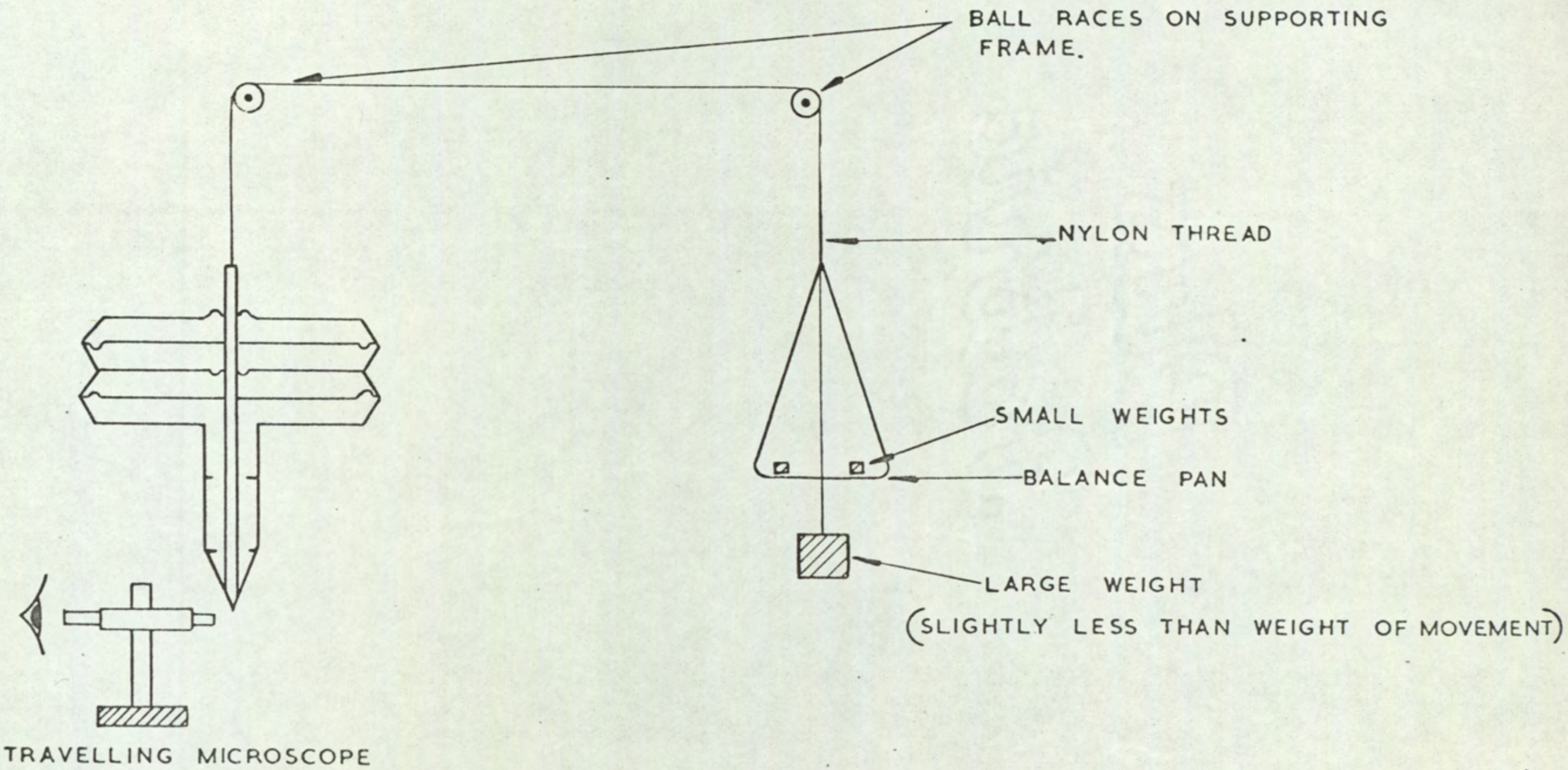


FIG. 6-8

VARIATION OF PLUG POSITION WITH EQUIVALENT PRESSURE.

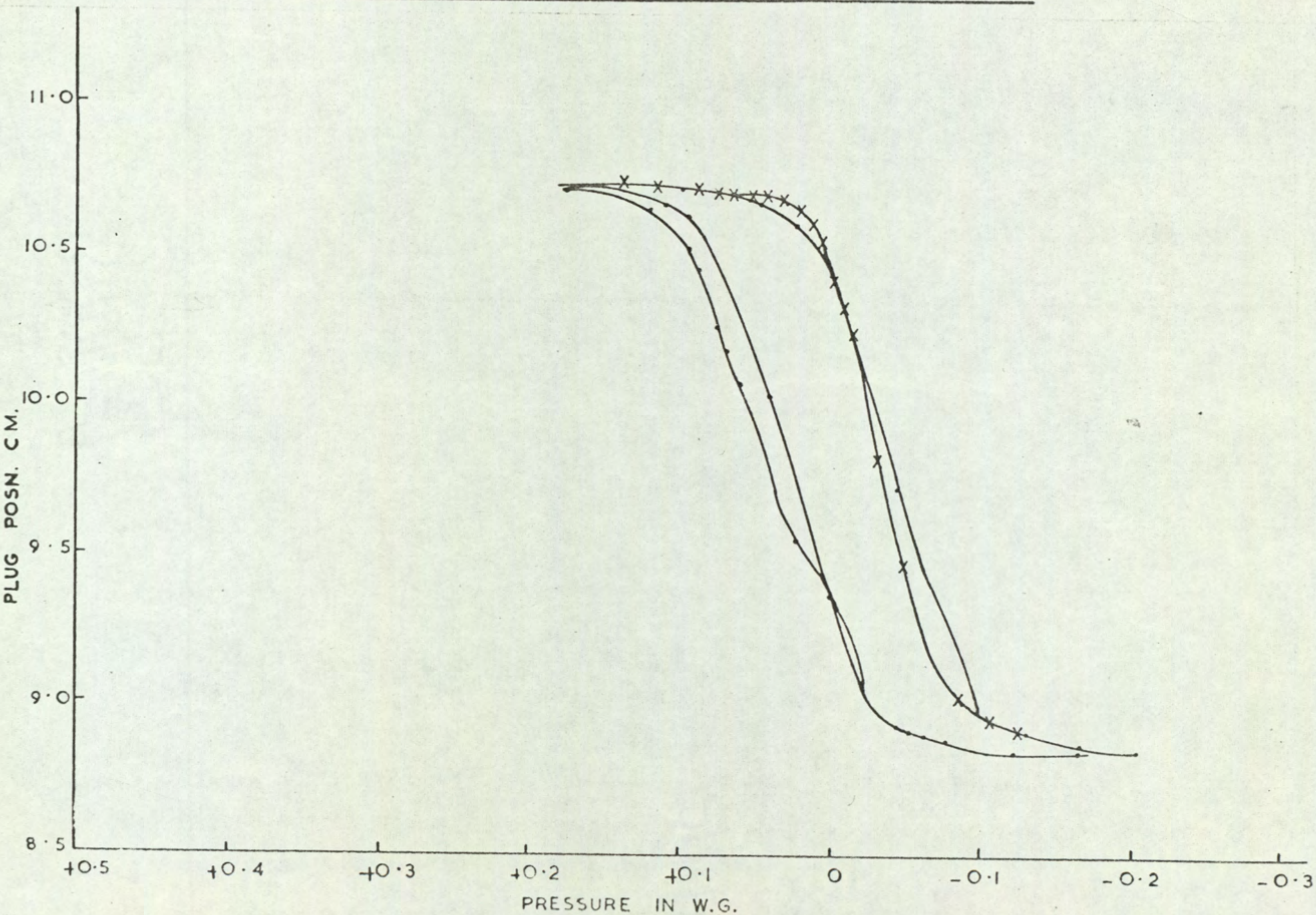


FIG. 6-9

difference. This figure can be compared directly with Fig. 6-4 and it can be seen that true elasticity only accounts for a small proportion of the apparent elasticity in the movement.

Consider now the force acting on one diaphragm when a differential pressure ΔP is acting across it. Fig. 6-10 is a scale drawing of the cross section of the diaphragm chamber showing the diaphragm in three positions; 1 and 3 being the extreme positions of the diaphragm and 2 being the mid position. The force acting on the diaphragm can be considered in two parts; firstly the effect of ΔP on the diaphragm plate, which is constant whatever the position of the diaphragm, and secondly the effect of ΔP on the annulus of leather (or polymer) diaphragm. Here the effect of ΔP is to produce a tension T in the material of the annulus and since the magnitude of T depends partly upon the radius of curvature of the annular section it will vary with diaphragm position. In addition the resultant force acting on the diaphragm plate due to T will also vary with changing diaphragm position due to the changing line of action of T.

From knowledge of the dimensions of the system and weight of the moving parts it has been possible to calculate the differential pressure (applied to one diaphragm) required to support the plug (injector mounted vertically) at various positions throughout its stroke. Details of the actual calculations are included in Appendix XI and the results are presented in Fig. 6-11. Also plotted on Fig. 6-11 is a second curve displaying the net effect of the calculated change in effective area plus the actual elasticity as found experimentally. This curve may be compared directly with Fig. 6-7 and to aid this comparison the experimental

SCALE DRAWING OF CROSS SECTION OF DIAPHRAGM CHAMBER.

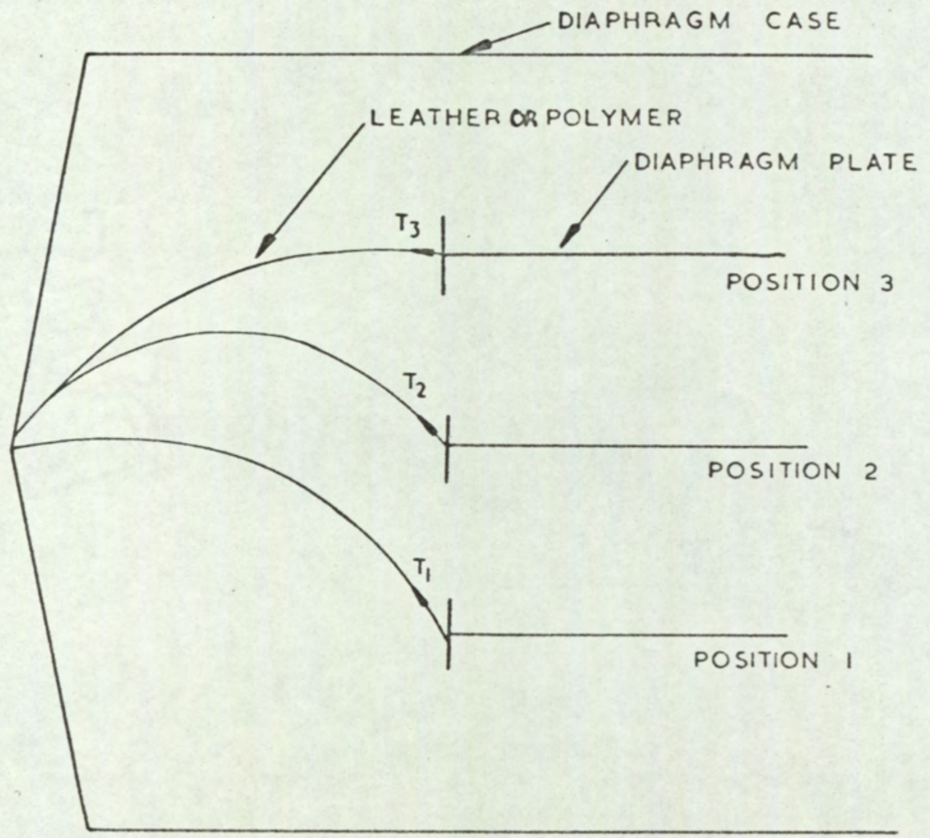


FIG. 6-10

VARIATION OF PLUG POSITION WITH SUCTION

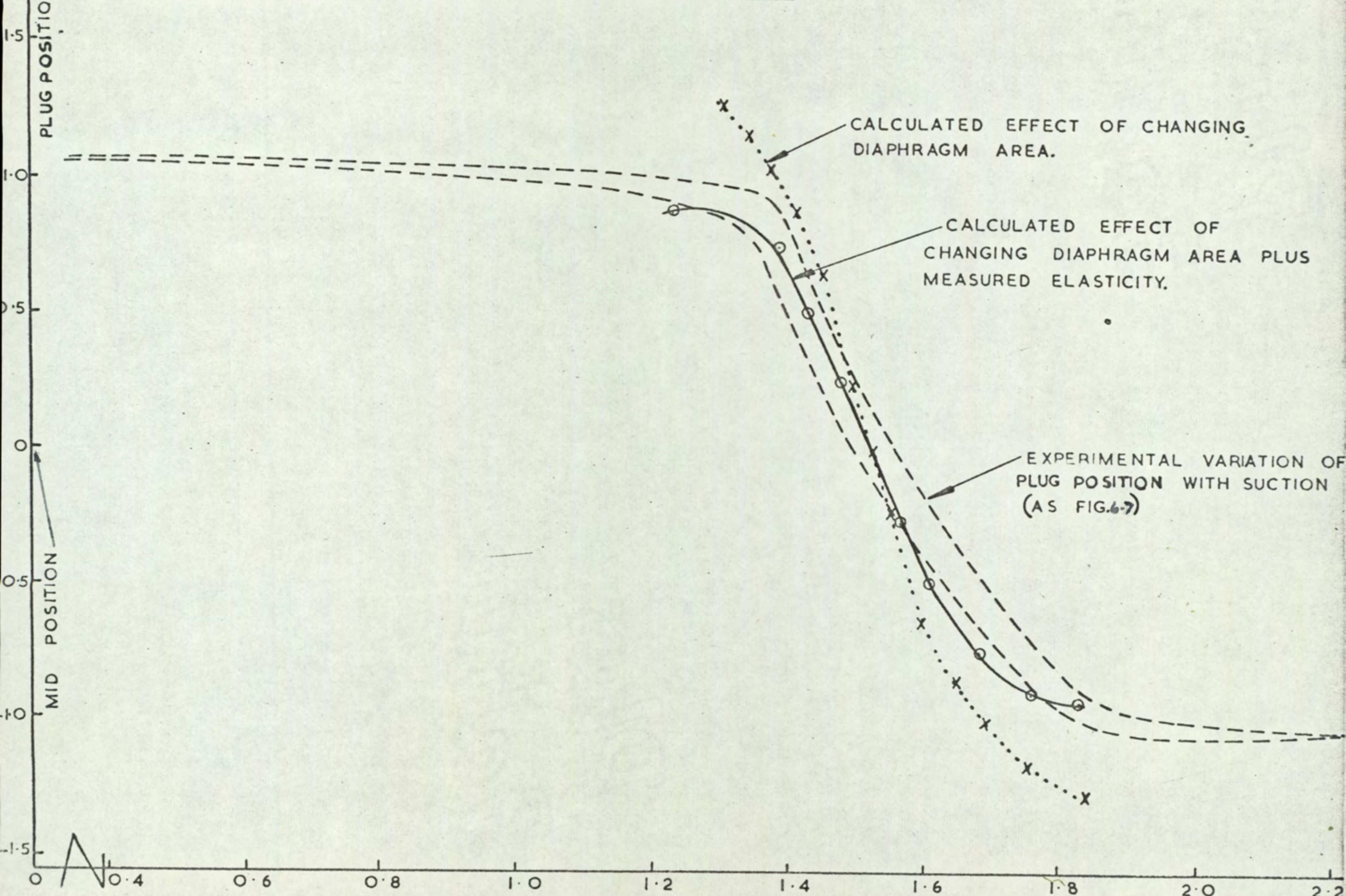


FIG. 6-11

curve of Fig. 6-7 has been re-plotted on Fig. 6-11.

It was felt that the close agreement between these two curves proved that the apparent elasticity in the moving parts of the injector was in fact a combined effect of actual elasticity plus an apparent elasticity due to the effective area of the diaphragms changing throughout the stroke of the plug.

Although it may have been possible to reduce both these effects by various means, for instance, by employing rolling diaphragms of very thin material, it was felt that it would be difficult to eliminate the effect completely and that attention should be turned to eliminating the effect of changing effective diaphragm area by making the injector a null balance device rather than a force balance device; i.e. by making the differential pressure across the diaphragms (or diaphragm) zero in the balanced position.

VII

DEVELOPMENT OF BRIDGE CONTROLLED INJECTORS

Mk.II Injector

The work described in the preceding section revealed the need to develop a null balance injector in order to overcome the difficulties inherent in the force balance system, caused by changing effective diaphragm area with plug stroke.

A technique was therefore devised in which a pneumatic bridge network was used to compare the air and gas flow rates to the injector and in which the ultimate control device tended toward a null balance condition. The principal of operation can be seen from fig.7-1.

The air flows through orifice A before reaching the recuperator and injector and the gas through orifice G. A small quantity of air is allowed to bleed through the pneumatic resistors R, which consist of equal lengths of capillary tube and which are sized so that the amount of air added to the gas stream is negligibly small.

If the air and gas supply pressures are P_a and P_g respectively and the pressure drops across the metering orifices are likewise ΔP_a and ΔP_g then the pressures P_1 and P_2 at points (1) and (2) in the system are:-

$$P_1 = \frac{(P_a - \Delta P_a) + P_g}{2} = \frac{P_a + P_g}{2} - \frac{\Delta P_a}{2}$$

$$P_2 = \frac{P_a + (P_g - \Delta P_g)}{2} = \frac{P_a + P_g}{2} - \frac{\Delta P_g}{2}$$

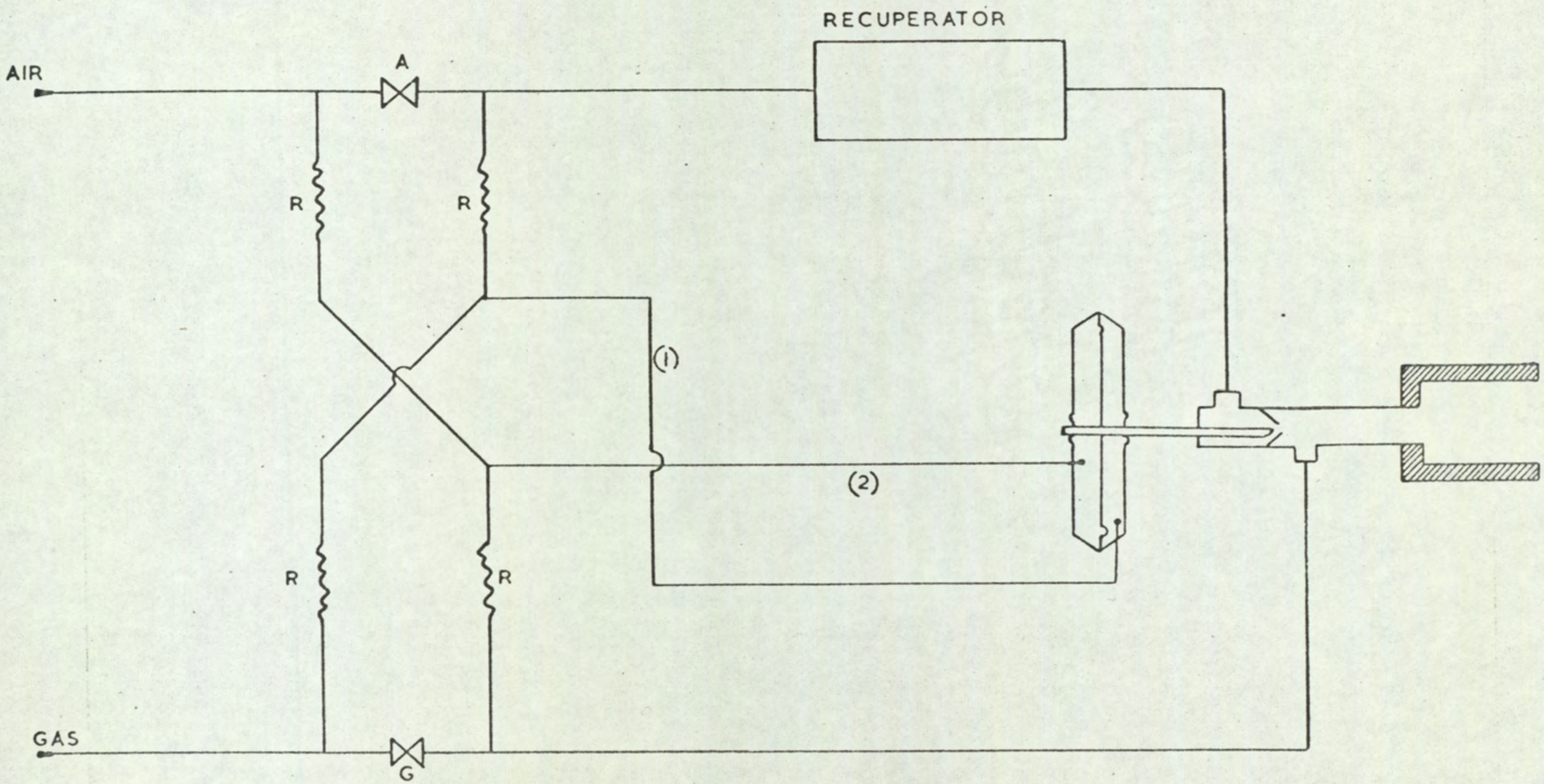


FIG. 7-1

From which it can be seen that $P_1 = P_2$ if $\Delta P_g = \Delta P_a$, whilst if $\Delta P_g \neq \Delta P_a$ the plug will move so as to make them equal.

This system was examined experimentally by applying pressures P_1 and P_2 across one of the main diaphragms of the compensating injector, the other diaphragm chamber being left open to atmosphere. This experiment showed that although changes in the effective area of the main diaphragm were eliminated there remained a force balance effect between the two secondary diaphragms and that the changing effective area of these introduced a considerable error.

Mk. 111 Injector

After considering various methods of eliminating this effect it was decided that a number of advantages would accrue if P_1 and P_2 were compared in a pilot instrument which could control a supply of air to the main diaphragm of the injector, the latter simply acting as a pneumatic motor.

For example there would be :-

1. Ample power available to drive the plug resulting in:-
   (a) Possible reduction in size of motor diaphragm,
   (b) Elimination of ball bearings for the plug and possible elimination of the air-seal.
2. More rapid response to changes in air preheat due to a reduction in the full stroke displacement of the sensing diaphragm.
3. The need for the injector to be mounted horizontally would be eliminated.

A simple pilot instrument, shown in Fig. 7-2, was constructed by mechanically linking two pilot governors of the type fitted to Keith Blackman zero governors. In view of the high sensitivity of these devices it was expected that only a very small difference between P_1 and P_2 would be required for full stroke operation of the pilot valves. Fig. 7-3 shows the intended method of using the pilot instrument to actuate the plug of the compensating injector.

Although it was realized that the pilot instrument was a force balance device and may be expected to suffer from changing effective diaphragm areas, it was anticipated that, since the stroke of the device was very small, this effect would be negligable.

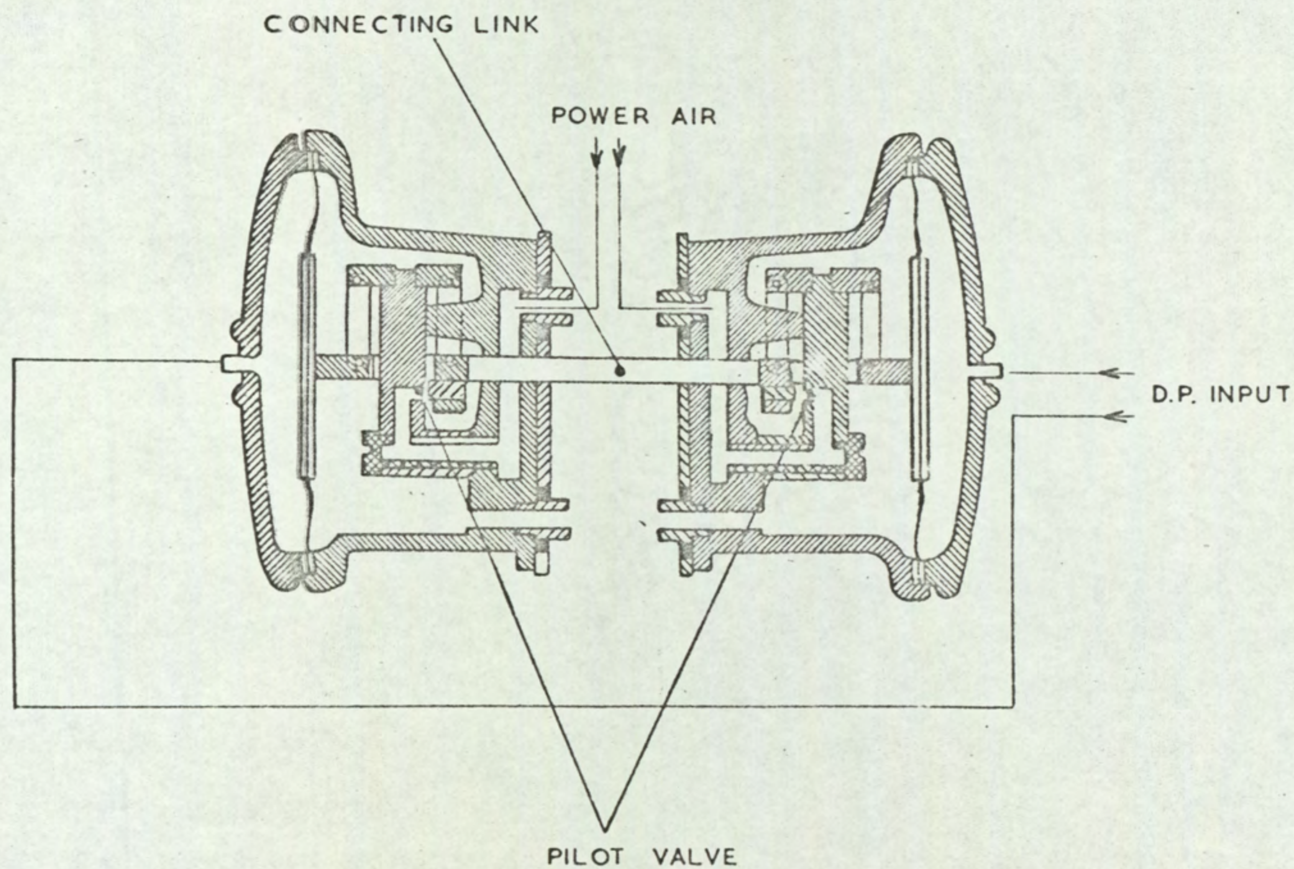


FIG. 7-2

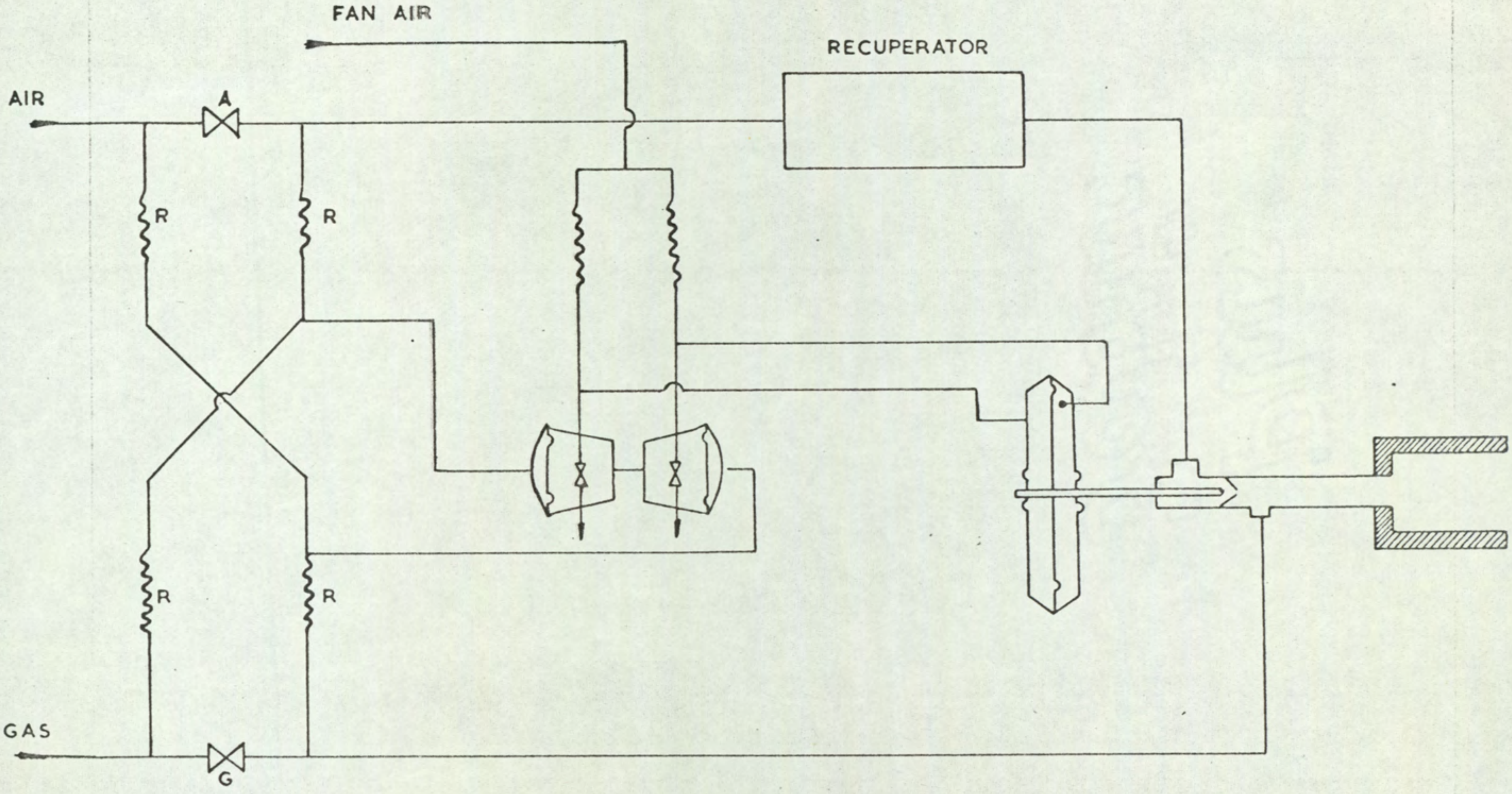


FIG. 7-3

When the pilot control system described above was tested it was found that the sensitivity of the pilot instrument was affected by the magnitude of the pressures applied to the two diaphragms and that although the linkage was free to move when no pressure was applied, a differential pressure of several tenths of an inch w.g. was required for full stroke operation if the applied pressures were about 7 in.w.g. This indicated that the apparent stiffness of the movement was not due to friction of the linkage bearings but was caused in some way by the diaphragms; probably a combined effect of changing effective area and rolling resistance in the annular portion. In order to overcome these effects, attention was directed at producing a null balance pilot instrument using one Keith Blackman pilot governor.

Mk.IV Injector

The problem of developing a null balance pilot system was the development of a differential pressure amplifier having adequate sensitivity. It was obvious that if sufficient sensitivity were to be obtained, mechanical friction must be kept to a minimum and the use of secondary diaphragms avoided. These requirements were met by the instrument shown in Fig.7-4 which consisted of a modified Keith Blackman Governor. With this instrument the pilot valve could be fully opened or closed by a differential pressure of a few hundredths of an inch w.g. acting appropriately across the diaphragm. However, as can be seen from the Figure, the need for any secondary diaphragms was avoided by allowing the linkage shaft from the main diaphragm to the pilot valve to pass through a small clearance hole in the sealing disc. Thus, unless special precautions are taken, the

MODIFIED K.B. PILOT GOVERNOR

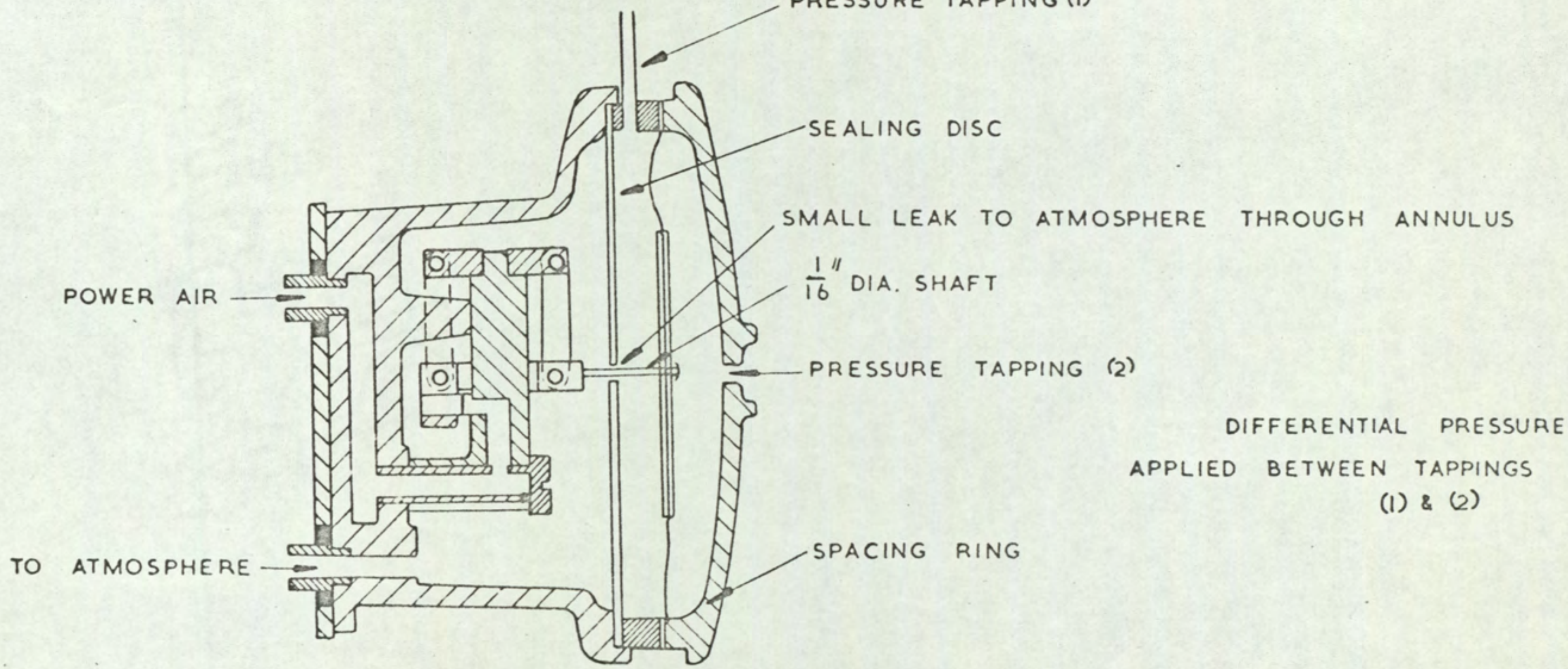


FIG. 7-4

leak to atmosphere through the small annulus from the diaphragm chamber can cause an error in any measurement of differential pressure. This problem was overcome by using the instrument in the pneumatic bridge system shown in Fig.7-5.

It can be seen that in this system the pilot instrument valve formed one leg of a pneumatic bridge connected across the diaphragm of the compensating injector. Each arm of the bridge had a pneumatic resistance approximately equal to that of the pilot instrument valve when the latter was in its mid position. Thus, actuation of the pilot instrument valve could apply up to half the fan air pressure to drive the injector plug to and fro.

Each resistance R_1, R_2, R_3, R_4, of the measuring bridge consisted of a glass capillary 3 mm bore and 3 in long, and with an air pressure of 24 in.w.g. the chain flow was about 30 cu.ft./hr. compared with the leakage from the pilot governor of about 6 cu.ft./hr. (in effect this leak was through a high resistance r). In order to enable the bridge to be balanced a small variable resistor R_5 (a pet cock), was placed in series with resistor R_4 so that at balance the series/parallel combination of R_4, R_5 and r had a resistance $R = R_1 = R_2 = R_3$.

The flow of air down the chain was much greater than in the previous systems and it will be noticed that the air flowing through R_3 and R flows to atmosphere.

Consider the flow down each chain to be q and the flow through air and gas orifices to be Q_a and Q_g when the differential pressures across each are equal. It can be seen from Fig.7-5 that the quantity of metered

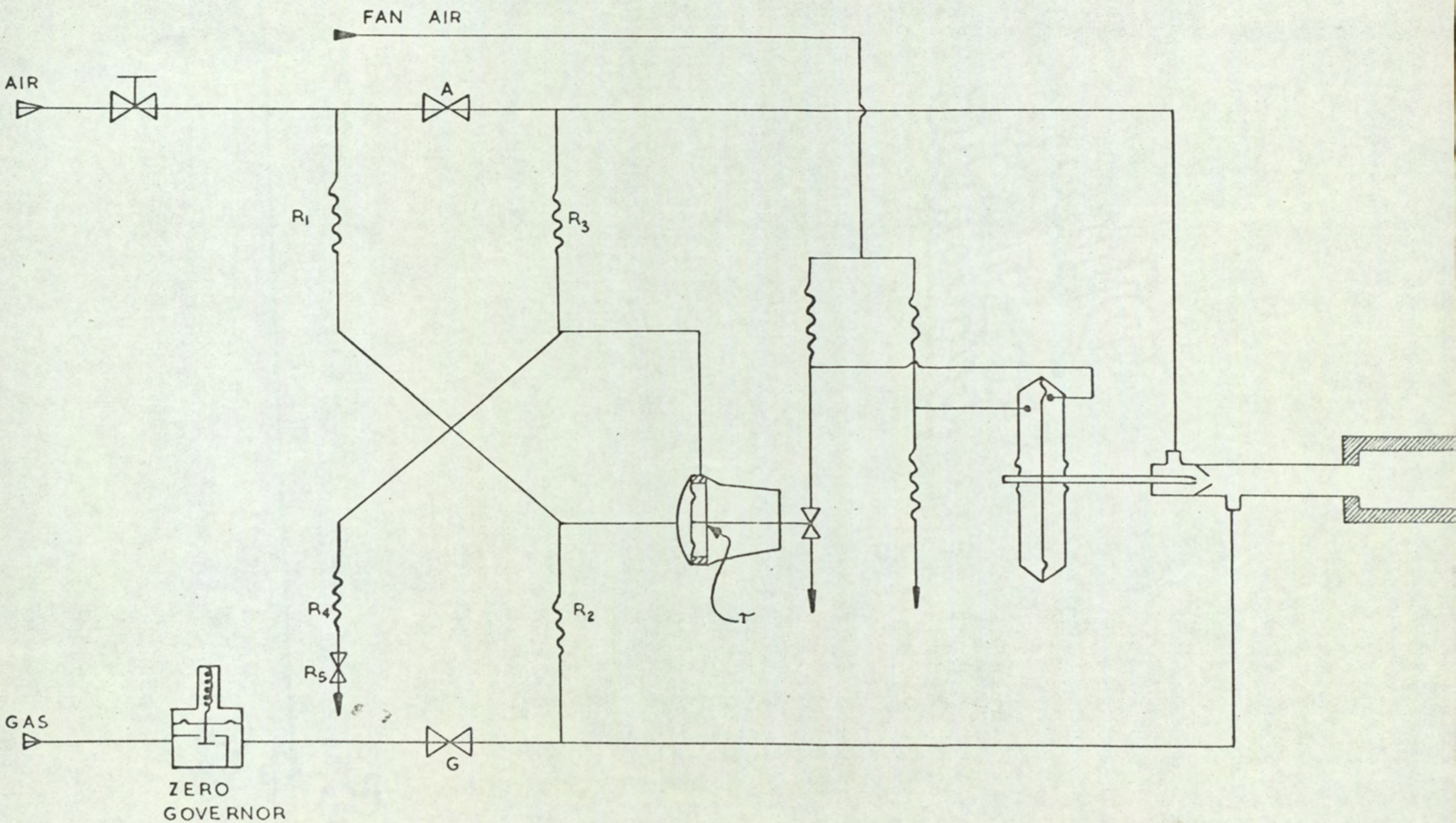


FIG. 7-5

air lost through R_3 and R is equal to that added to the gas stream through R_1 and R_2, thus the quantity of air passing through the burner air nozzle is $Q_a - q$ whilst Q_g of gas and q of air enter through the gas inlet. The presence of air in the gas stream presents no problems providing the mixture is not inflammable (for example if $q < 1.8\ Qg$ for the town gas used during this work) even so, in practice, it may be convenient to connect R_2 directly to the injector, in which case no mixture would flow through the gas supply pipe.

If $R_1 = R_2 = R_3 = R$ then the effect of the air and gas orifices will, in fact, cause q to be slightly different for each chain. However, with air pressures of about 1 P.S.I.G. and a 2 in. differential pressure across the air and gas orifices, this results in excess air at the burner of 0.2 q. This amount of excess air is negligible, i.e. 0.3%, and in any event, since q is proportional to air pressure, it is constant and can be allowed for when initially setting the ratio.

Using the test rig described in Section 4 preliminary experiments were carried out to check the performance of the complete system by measuring the flow of air and gas to the burner at varying preheat temperatures. Fig. 7-6 shows the variation of air/gas ratio with preheat that was obtained for various throughputs, and the conclusions that were drawn from these results are enumerated as follows:-

1. In the middle flow ranges the air/gas ratio was maintained constant as the air preheat was varied. The pilot instrument appeared to be working satisfactorily.
2. The results revealed a decreasing air/gas ratio on turndown, usually indicative of a positive zero governor outlet pressure.
3. At the lowest rate of throughput the air/gas ratio increased.

VARIATION OF AIR/GAS RATIO WITH PREHEAT AT VARIOUS THROUGHPUTS.

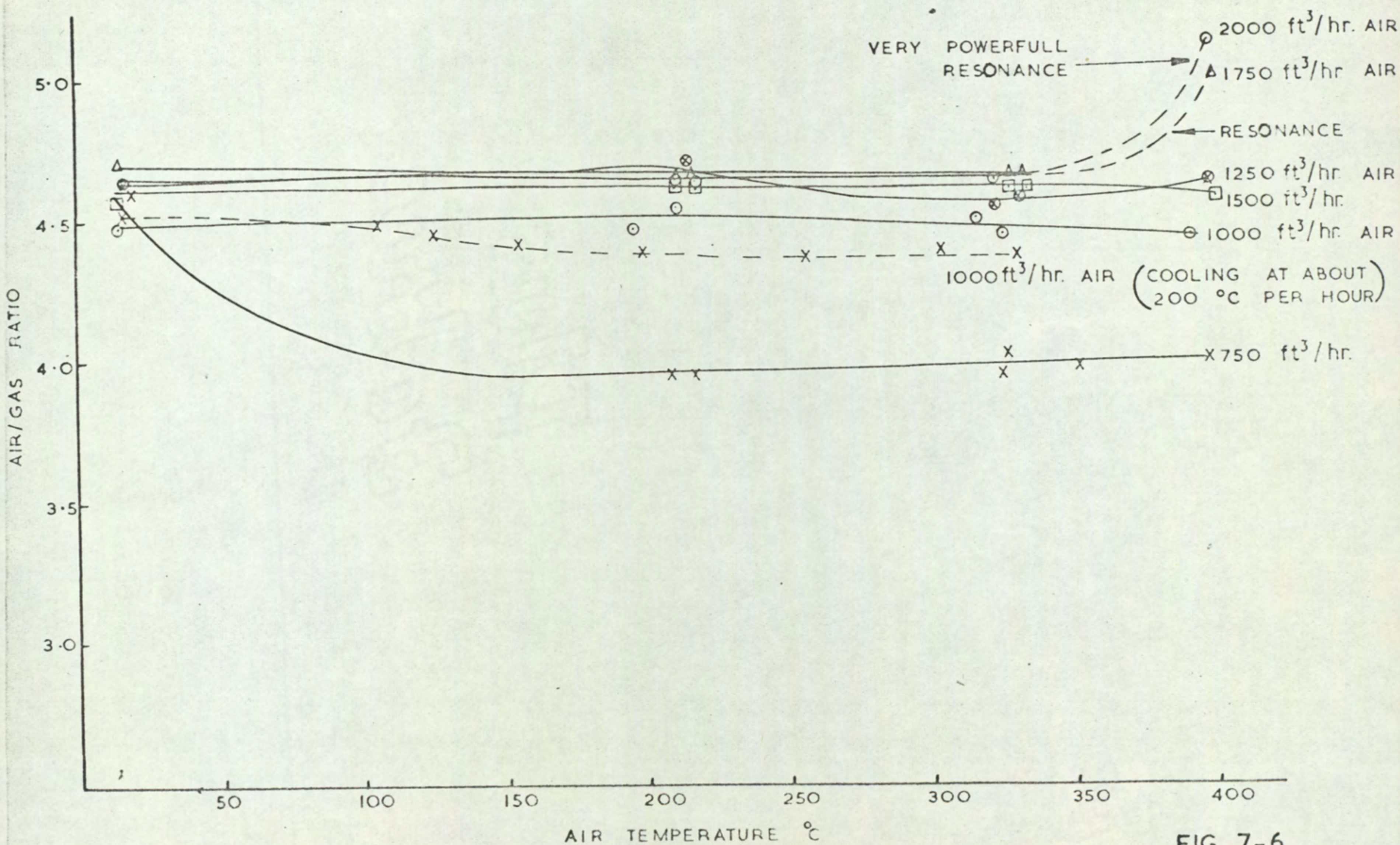


FIG. 7-6

However, as the preheat temperature was reduced below 150°C it was observed that the plug was in the fully forward position, i.e. the air nozzle area was at its minimum and the system not controlling.

4. At the maximum flow rates and preheat temperatures intense burner resonance occurred, on the fringe of the resonant region it caused the control system to hunt and in the intense region the plug was driven fully back.

5. It was noted that the controlled ratio was about 4.65 : 1 as measured by the rotameters. From the appearance of the burner flame however, the ratio was stoichiometric or perhaps slightly gas rich. Obviously a considerable amount of air was being lost from the system and a subsequent investigation revealed considerable leakage from the recuperator.

In view of 4 above, the burner tunnel was shortened in order to eliminate or reduce resonance; the tunnel being in fact shortened from 10 inches to 9 inches.

In view of 5 above the recuperator was leak tested and it was found, with an air pressure of about 1 P.S.I.G. applied to the air side of the recuperator, air leaked to the waste gas side at a rate of approximately 400 cu.ft./hr. The recuperator was therefore dismantled and reassembled with fresh sealing between the needle tube and the casing. Originally this sealing consisted of asbestos caulking and it was through this that the air leak was occurring. On reassembly the bottom end of the needle tube was set into a refractory concrete and the top end of the tube sealed with "Caprosil" and sodium silicate. By this means it was hoped that any relative movement between the needle tube and the case due to expansion would therefore occur only at the top seal which, being exposed, could be readily remade if necessary. However, it was hoped that the jointing material used would be sufficiently resilient to take up any movement without cracking.

Examination of the needle tube and the casing whilst the recuperator was dismantled revealed no deterioration in metal condition even though the recuperator had been run slightly hotter than the maximum temperature recommended by the Manufacturers.

After reassembly the recuperator was tested for leaks and it was found that the leakage rate was about 1% of the air flow rate compared with 20% prior to dismantling. This leakage rate was considered acceptably low and the control system and burner were therefore reassembled and tested. On operating the burner up to its maximum throughput with air preheats up to 300°C, no trace of resonance was noticed.

Experiments were performed to determine the variation of air/gas ratio with changing air temperature at two levels of throughput and the results of this experiment are shown in Fig.7-7. It will be seen that the control ratio was 4.1:1, a value corresponding to the ratio estimated from the appearance of the flame.

Further experiments were performed to determine the variation of air/gas ratio with throughput; the air being maintained at virtually constant temperature. The results of these experiments are shown in Fig.7-8 from which it can be seen that a marked variation of ratio with throughput occurred. This variation was thought to have two possible

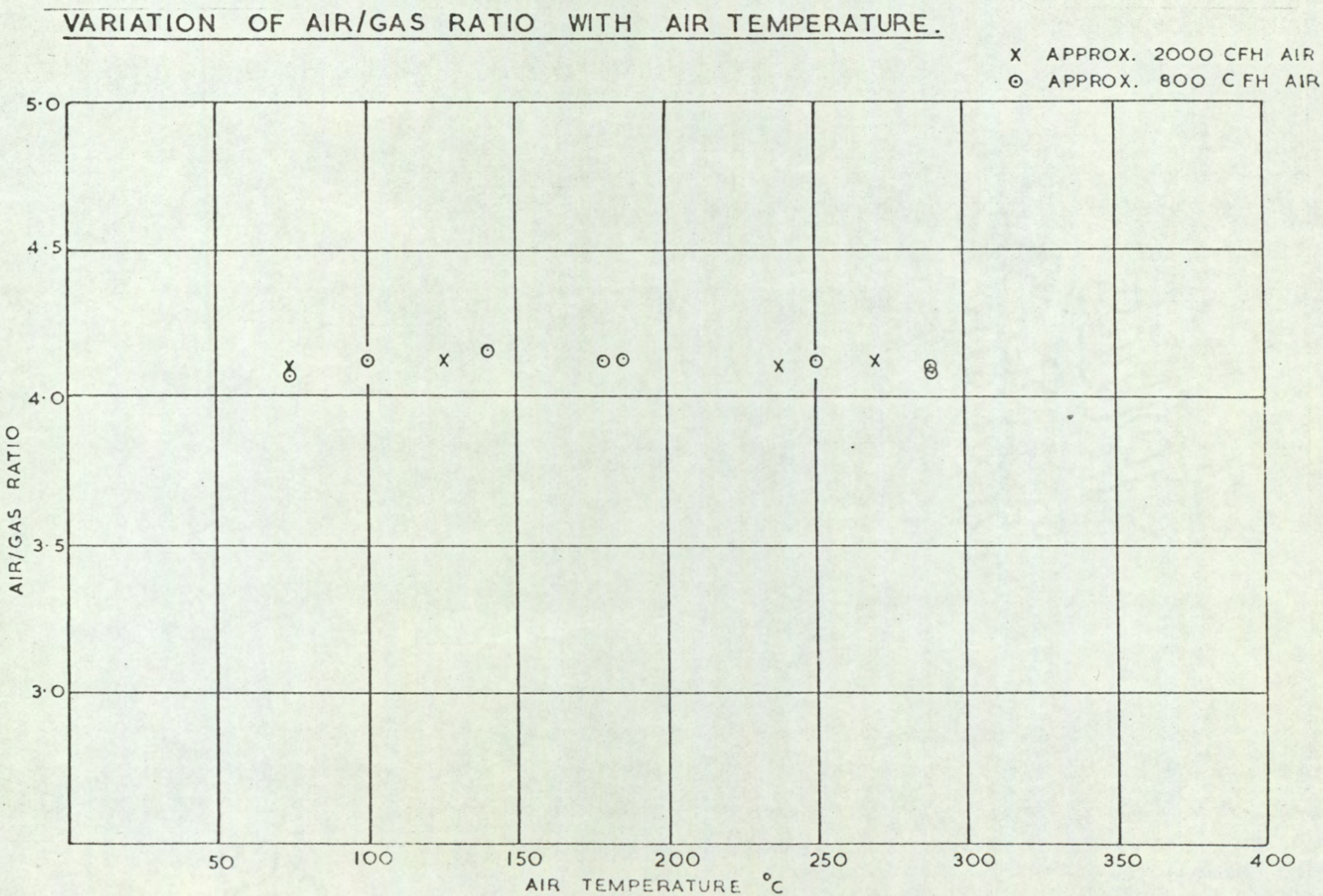

VARIATION OF AIR/GAS RATIO WITH AIR TEMPERATURE.
X APPROX. 2000 CFH AIR
⊙ APPROX. 800 CFH AIR
5·0
4·5
4·0
3·5
3·0
AIR/GAS RATIO
50
100
150
200
250
300
350
400
AIR TEMPERATURE °C

FIG. 7-7

VARIATION OF AIR/GAS RATIO WITH THROUGHPUT.

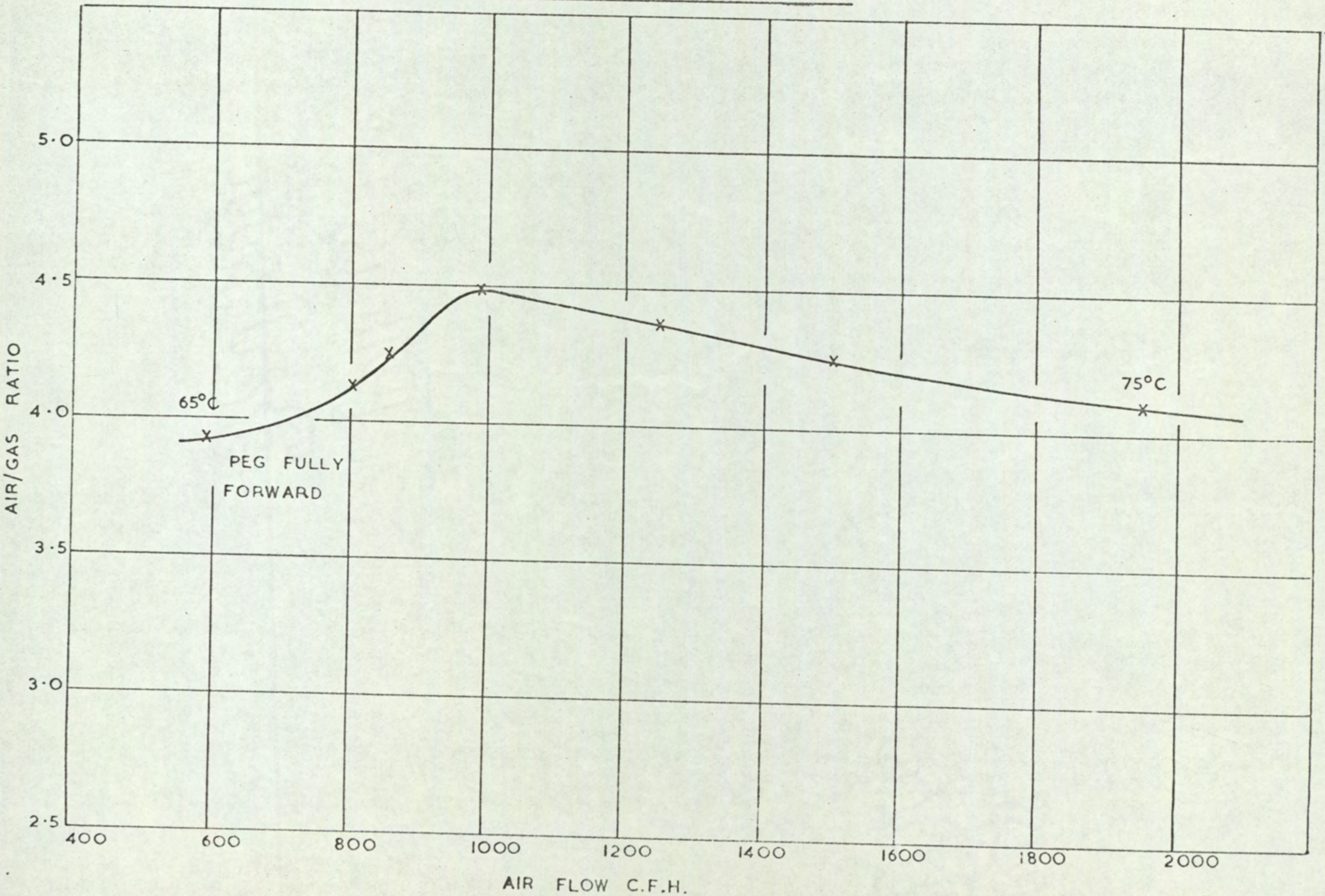


FIG. 6 - 8

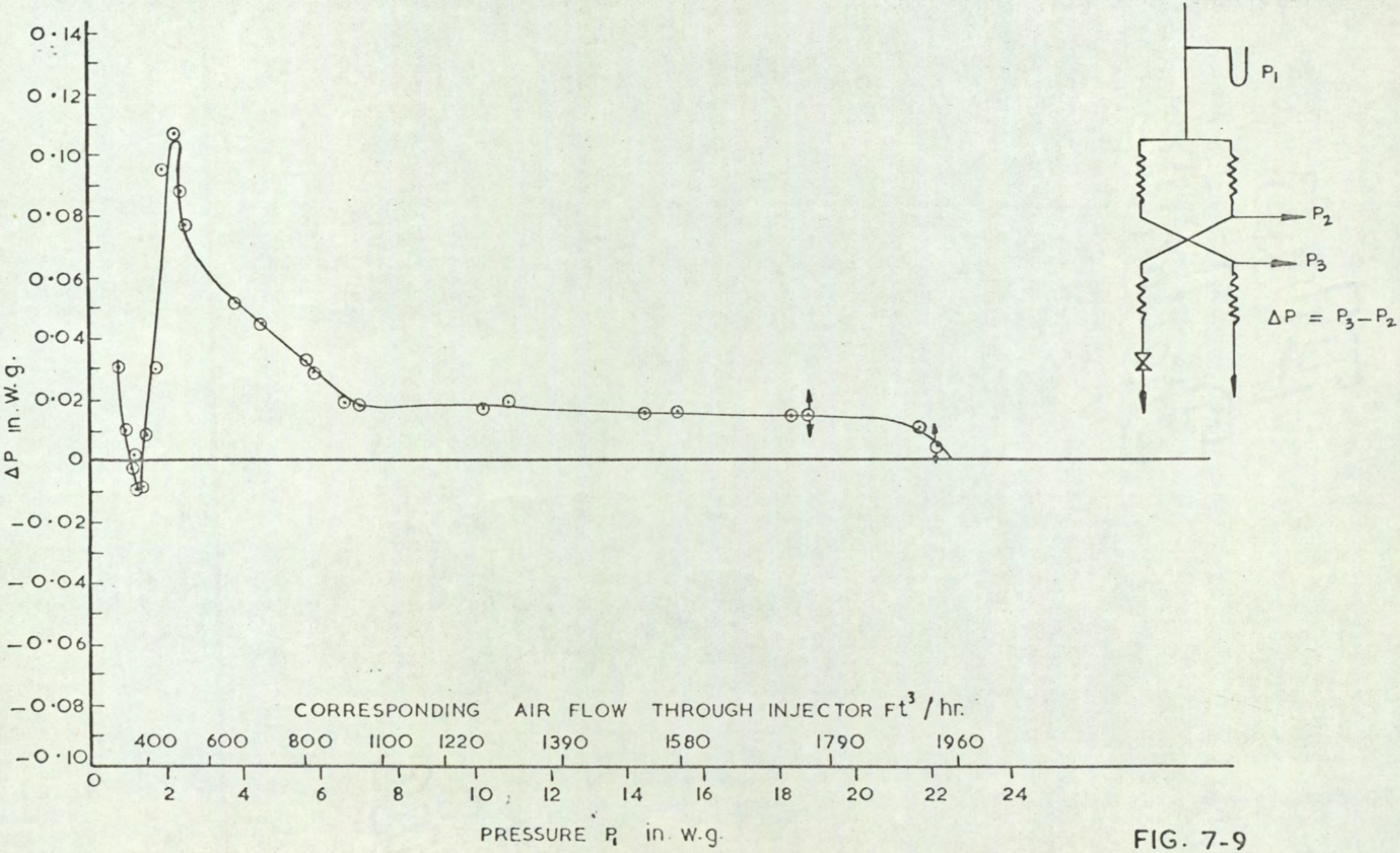
causes:-

(a) Unbalance of the measuring bridge causing the differential pressures across the air and gas orifices to be unequal at some flow rates.

(b) Changes in Reynold's Numbers in the air and gas orifices having a considerable effect on the orifice discharge coefficients, this would mean that even if the differential pressures were controlled accurately, the air/gas ratio would alter with throughput.

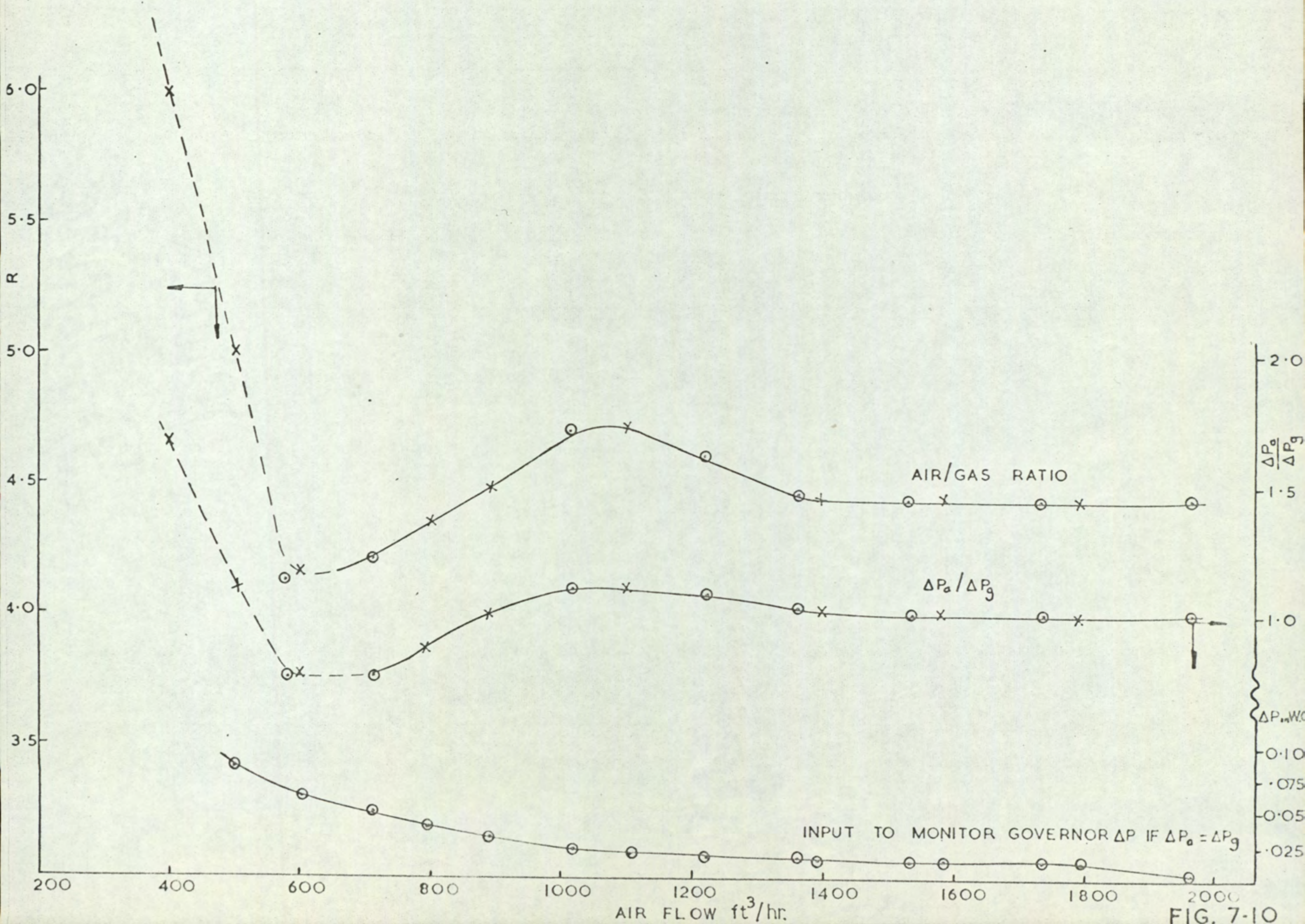
In order to investigate the first of these possibilities, tests were carried out to determine the accuracy of bridge balance with changing pressure across the bridge. To do this the bridge was set up as shown in the inset to Fig.7-9 and balanced with the maximum pressure applied, the unbalance ΔP being measured on an inclined manometer. Fig.7-9 shows the variation of ΔP with applied pressure and also with the corresponding air flow through the injector. Calculation showed that the Reynolds Nos. in the capillary tubes lay between 5200 (max.P_1) and 1020 ($P_1 \simeq 1.5$ in.w.g.) and that over this range changes in friction factor occurred that could well account for the bridge going out of balance on turndown.

Experiments were then carried out on the complete system in which the air/gas ratio and the ratio $\frac{\Delta P_a}{\Delta P_g}$ were measured over the full range of throughputs. The results of these experiments are presented in Figs.7-10 and 7-11 and show a marked correlation between air/gas ratio and $\frac{\Delta P_a}{\Delta P_g}$. The difference in the shape of the curves in Figs.7-10 and 7-11 is thought to be due to differences in bridge adjustment. For example, the results shown in Fig.7-10 were obtained with the bridge balanced as described above



0·14
0·12
0·10
0·08
0·06
0·04
0·02
0
−0·02
−0·04
−0·06
−0·08
−0·10
ΔP in. W.g.
CORRESPONDING AIR FLOW THROUGH INJECTOR Ft³ / hr.
400 600 800 1100 1220 1390 1580 1790 1960
0 2 4 6 8 10 12 14 16 18 20 22 24
PRESSURE P1 in. W.g.
P1
P2
P3
ΔP = P3 − P2

FIG. 7-9



6·0
5·5
5·0
4·5
4·0
3·5
R
AIR/GAS RATIO
$\Delta P_a / \Delta P_g$
INPUT TO MONITOR GOVERNOR ΔP IF $\Delta P_a = \Delta P_g$
$\frac{\Delta P_a}{\Delta P_g}$
2·0
1·5
1·0
ΔP in W.G.
0·10
·075
0·05
·025
200
400
600
800
1000
1200
1400
1600
1800
2000
AIR FLOW $ft^3/hr.$

FIG. 7-10

VARIATION OF RATIO WITH THROUGHPUT.

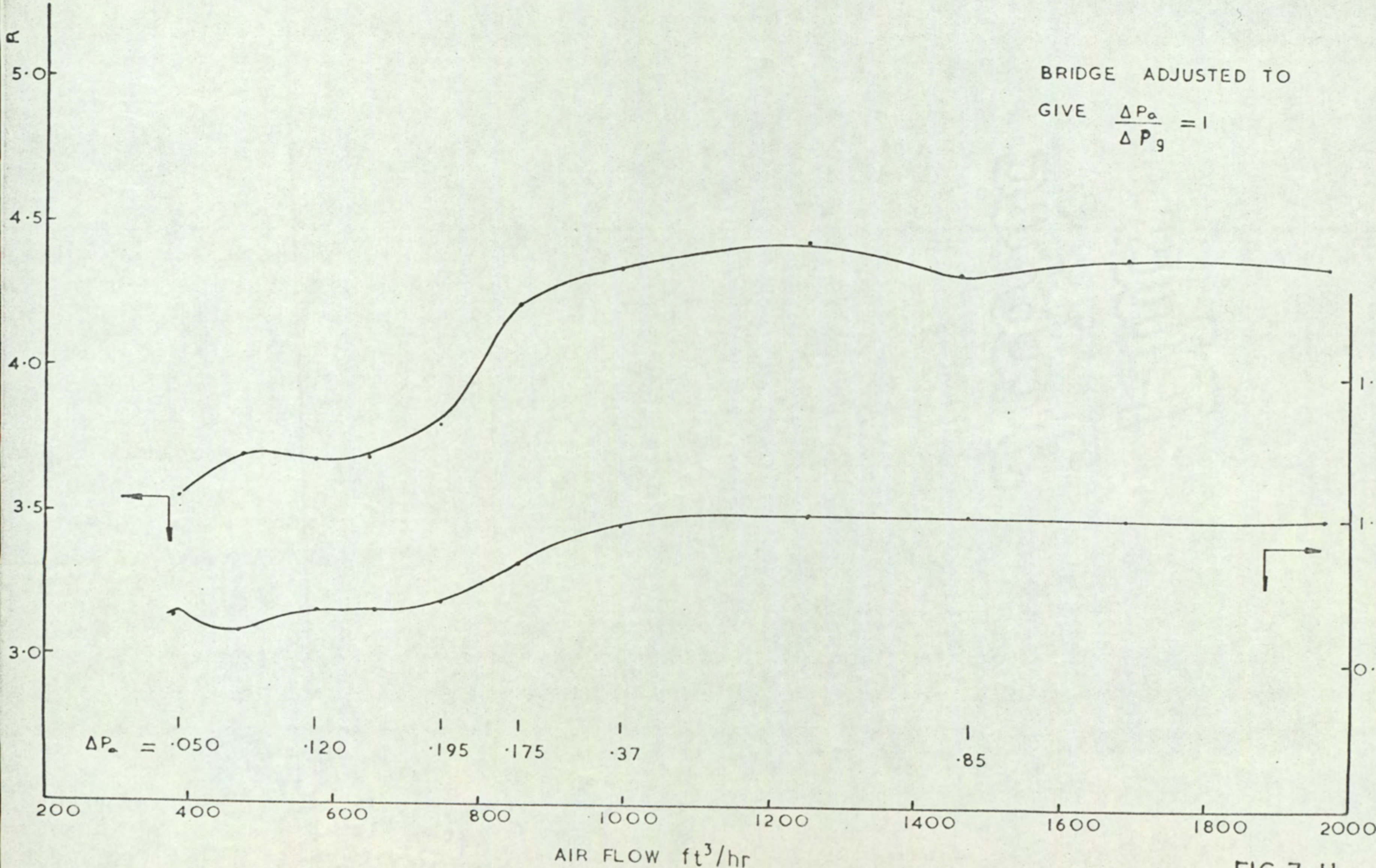


FIG 7-11

and the results shown in Fig.7-11 were obtained after the system had been disturbed and the bridge readjusted to give $\frac{\Delta P_a}{\Delta P_g}$ = 1 at maximum throughput.

Since the variation of air/gas ratio with throughput appeared to be due to unbalance of the bridge, and since this was probably caused by changes in friction factor within the bridge arms, an experiment was performed which was designed to maintain the Reynolds numbers, and hence the friction factors, within the arms constant. This was done by controlling the air supply to the burner downstream of the air metering orifice so that the bridge was subject to an approximately constant pressure at all burner throughputs. By this means the flow through the bridge was maintained approximately constant.

The results of this experiment, together with those of a comparative experiment in which the burner air supply was controlled upstream of the air metering orifice, are shown in Fig.7-12.

It should be noted that the air/gas ratio was set to an arbitary value of 5.2:1 and no attempt was made to set the ratio accurately to the stoichiometric value.

It will be seen from Fig.7-12 that controlling the air supply downstream of the metering orifice effected no improvement in performance over the comparative experiment and that over the range of air throughputs 850 - 1700 ft^3/hr the deviation of the air/gas ratio from the set value was in fact greater than in the case of the comparative experiment.

Since the reason for this was not clear, attention was turned to the second of the possible causes of variation of ratio with throughput, (b) above.

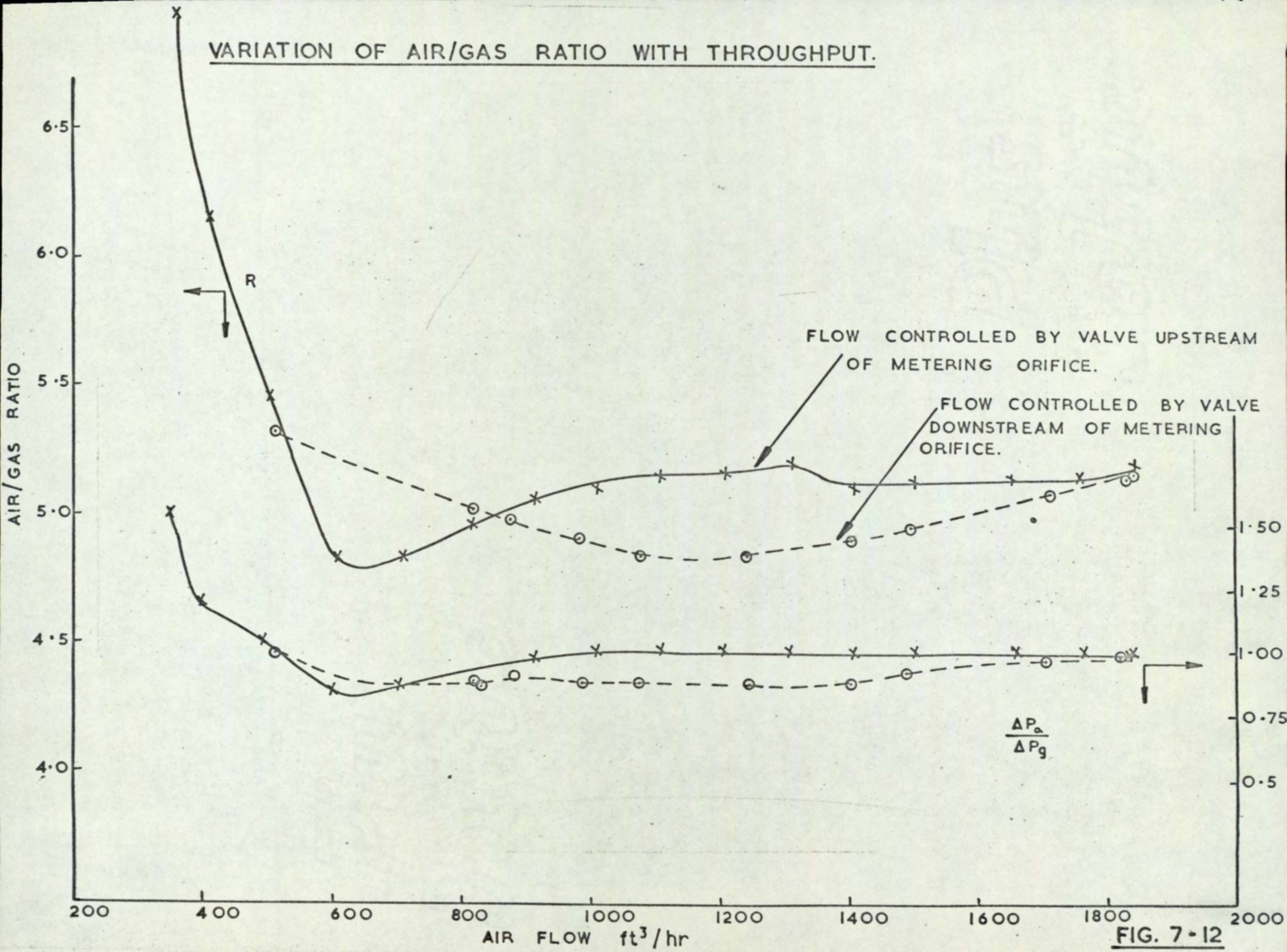

VARIATION OF AIR/GAS RATIO WITH THROUGHPUT.
AIR/GAS RATIO
6·5
6·0
5·5
5·0
4·5
4·0
R
FLOW CONTROLLED BY VALVE UPSTREAM OF METERING ORIFICE.
FLOW CONTROLLED BY VALVE DOWNSTREAM OF METERING ORIFICE.
1·50
1·25
1·00
0·75
0·5
$\frac{\Delta P_a}{\Delta P_g}$
200
400
600
800
1000
1200
1400
1600
1800
2000
AIR FLOW ft^3/hr

FIG. 7-12

Data obtained during the above experiments were used to plot Fig.7-13. This shows the relationship between the square root of the differential pressure $\sqrt{\Delta Pg}$ developed across the gas orifice and the gas flow rate Qg and also the relationship between the square root of the differential pressure $\sqrt{\Delta Pa}$ developed across the air orifice and the air flow rate Qa. In Fig.7-13 $\sqrt{\Delta Pa}$ is in fact plotted against $\frac{Qa}{5.2}$ since 5.2:1 was the air/gas ratio at the particular setting of the gas orifice at which the pressure and flow measurements were made. Thus if the flow coefficients of both orifices are constant with varying throughput then both sets of results should be on the same straight line.

It is clear from Fig.7-13 that there is no significant variation in the flow coefficients of either orifice over the major portion of the flow range. Some deviation between the two curves occurs at low flows but this was not considered important since, for example, at a gas rate of 100 ft^3/hr this deviation would cause the air/gas ratio to deviate from the desired value by only 4%. Gas rates below 100 ft^3/hr would not normally be practicable due to the onset of burner instability because of lightback; the increasing deviation below this rate is therefore unimportant.

From the above experiments it was concluded that despite the somewhat anomolous results shown in Fig.7-12 the variation of air/gas ratio with throughput was in fact due to unbalance of the bridge caused by changes in friction factor within the bridge arms.

Attention was therefore directed at eliminating the effect of changing friction factor within the bridge arms.

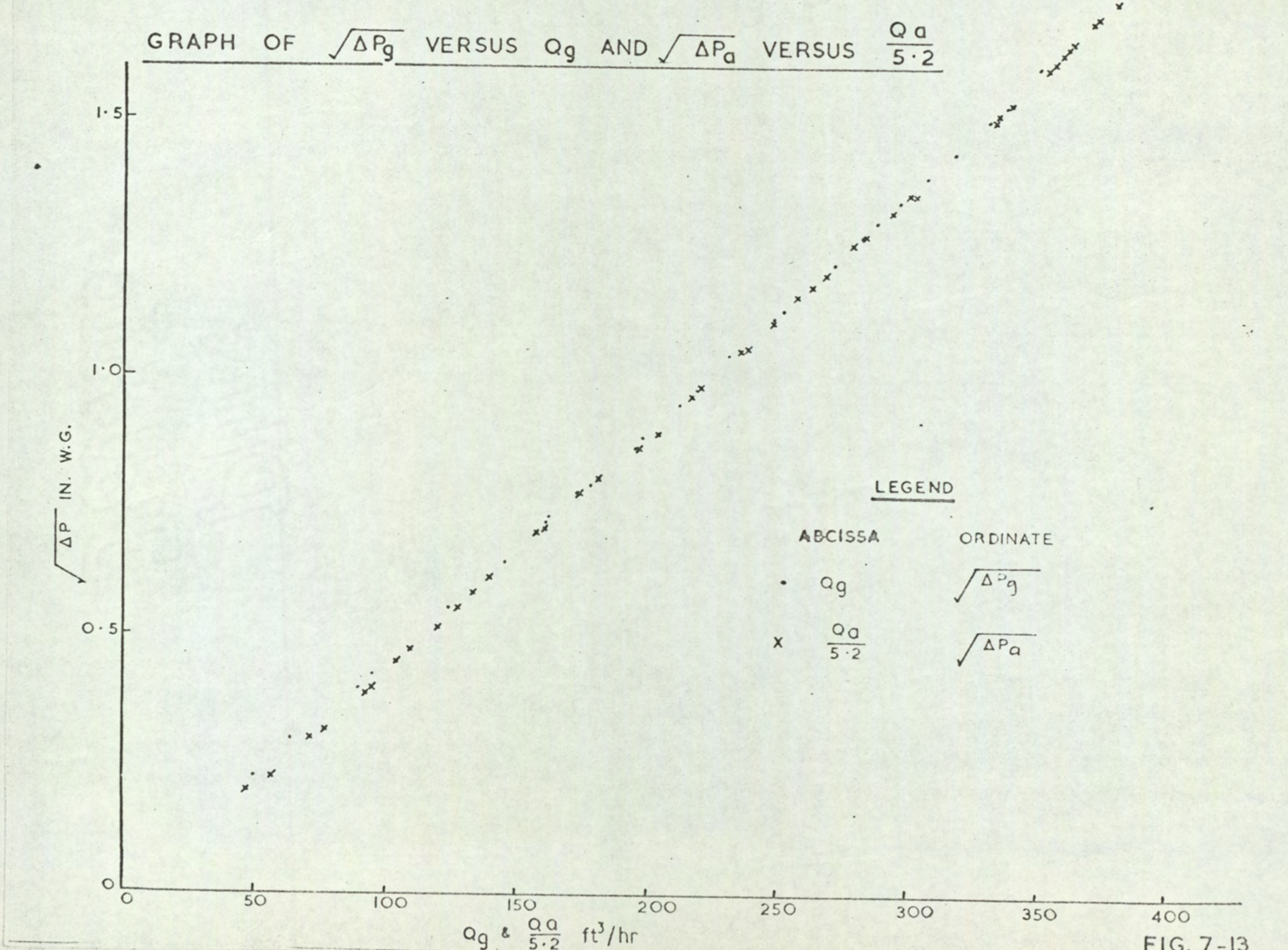

GRAPH OF $\sqrt{\Delta P_g}$ VERSUS Q_g AND $\sqrt{\Delta P_a}$ VERSUS $\frac{Q_a}{5 \cdot 2}$
$\sqrt{\Delta P}$ IN. W.G.
1·5
1·0
0·5
0
0
50
100
150
200
250
300
350
400
Q_g & $\frac{Q_a}{5 \cdot 2}$ ft^3/hr
LEGEND
ABCISSA
ORDINATE
Q_g
$\sqrt{\Delta P_g}$
X
$\frac{Q_a}{5 \cdot 2}$
$\sqrt{\Delta P_a}$

FIG. 7-13

Mk.V Injector

In order to eliminate the effect of friction factor within the bridge the capillary tubes were replaced by small orifices as the resistance elements in the bridge arms.

Orifices sized to give the same chain flow as the capillary bridge were chosen and it was anticipated that over the range of Reynolds Numbers involved 1400 - 6000 the discharge coefficients would be reasonably constant.

The orifices consisted of 0.098 in.dia. holes drilled through 9/16 in. discs of 18 gauge copper, these discs being conveniently clamped between fibre washers by 3/8 B.S.P. nipples screwed into short sockets.

The first orifice bridge was tested in a similar manner to the capillary bridge, i.e. the bridge was balanced at the maximum pressure by a series pet cock in one arm, and the out of balance pressure ΔP was measured as the applied pressure was reduced. The result of this experiment is shown in Curve 1 Fig.7-14 and can be compared with the capillary bridge curve in Fig.7-9 (Note the different ordinate scales).

The erratic result was thought to be due possibly to the relatively short distance (3.8 pipe diameters) of the pressure tappings downstream of the orifices. This distance was therefore increased to 10 pipe diameters. The experiment was then repeated and the result is shown in Curve (2) Fig.7-14. It can be seen that this modification made no significant improvement.

At this stage it was noticed that the bridge resonated (emitting a faint high frequency whistle), and as the pressure was altered different modes of resonance together with quiet zones were observed. It was also

VARIATION OF ΔP WITH VARIATION IN PRESSURE P_1 APPLIED TO BRIDGE

ORIFICE BRIDGE BALANCED AT 25 in. w.g. (ie. max. P_1)

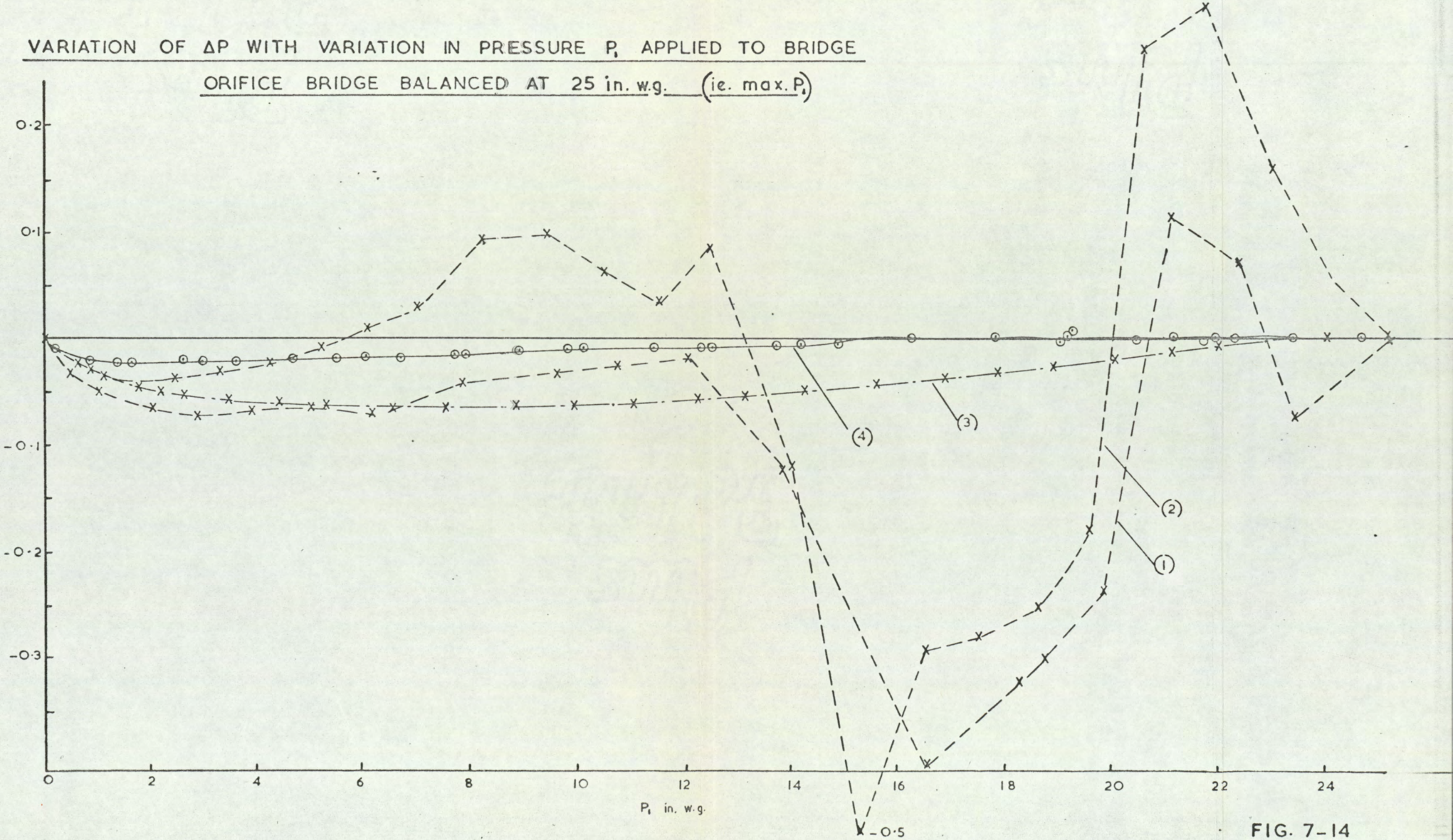


FIG. 7-14

observed that tuning the bridge by cupping the hand round one of the bridge outlets could affect the resonance and considerably affect ΔP even though care was taken not to impede the flow out of the bridge arm.

Subsequent experiments revealed that resonance was caused by the actual orifice edge and could be prevented by chamfering one side of the plate so as to produce a knife edged orifice.

The bridge was re-assembled and tested, and the result is shown in Curve (3) Fig.7-14. This was obviously a great improvement over the initial orifices but the results were still not as good as with the capillary bridge.

It appeared that since all the orifices were equal in size with virtually the same flow through each that the errors must be due to a change of discharge co-efficient with flow in either the annular leak in the pilot governor, or in the adjusting cock, or perhaps in both. Calculation showed that the change in Reynolds number in the annular leak would not seriously affect its discharge coefficient but that the series adjusting cock may well be affected due to the fairly low maximum pressure drop across it. In order therefore to increase the Reynolds Number in the adjuster, the bridge was modified so that the adjuster was fitted in parallel to the orifice rather than in series. In this position the maximum pressure drop across the adjuster was greatly increased. The use of a parallel adjuster necessitated the use of a slightly smaller orifice (0.081 in.dia.) in the adjustable arm so that the parallel combination of orifice, leak and adjuster should have the same pneumatic resistance as the other orifices in the bridge.

The modified bridge was tested and the resulting output is shown in Curve 4 Fig.7-14. This output appeared to be satisfactory, being rather better than that of the capillary bridge.

Mk.VI Injector

The bridge system described above having the parallel adjuster was used as a basis for the Mk.VI Injector, a schematic diagram of which is shown in Fig.7-15. The injector system was assembled on the test rig and experiments were performed to examine the variation of air/gas ratio with throughput, the air being at room temperature and the bridge balanced at maximum air pressure. Fig.7-16 shows the variation of air/gas ratio with throughput and also the variation of $\frac{\Delta P_a}{\Delta P_g}$ (i.e. the ratio of the differential pressures across the air and gas metering orifices). It should be noted that an arbitrary air/gas ratio was used for the test and no attempt was made to set the ratio to the stoichiometric value. Fig. 7-16 showed that the mixture went progressively weak as the throughput was reduced and the correlation between the variation of R and $\frac{\Delta P_a}{\Delta P_g}$ suggested that the error was due to unbalance of the bridge at low flows. However, since R deviated progressively as the throughput was reduced it was thought that if the zero governor outlet pressure were set slightly positive it may be possible to compensate for the bridge error.

An experiment was therefore performed in which the air/gas ratio was set to 4.15:1 at the maximum throughput by adjusting the gas restrictor and at the minimum throughput (i.e. air flow rate of 600 cu.ft./hr) by means of the zero governor. This technique is, incidently, the conventional

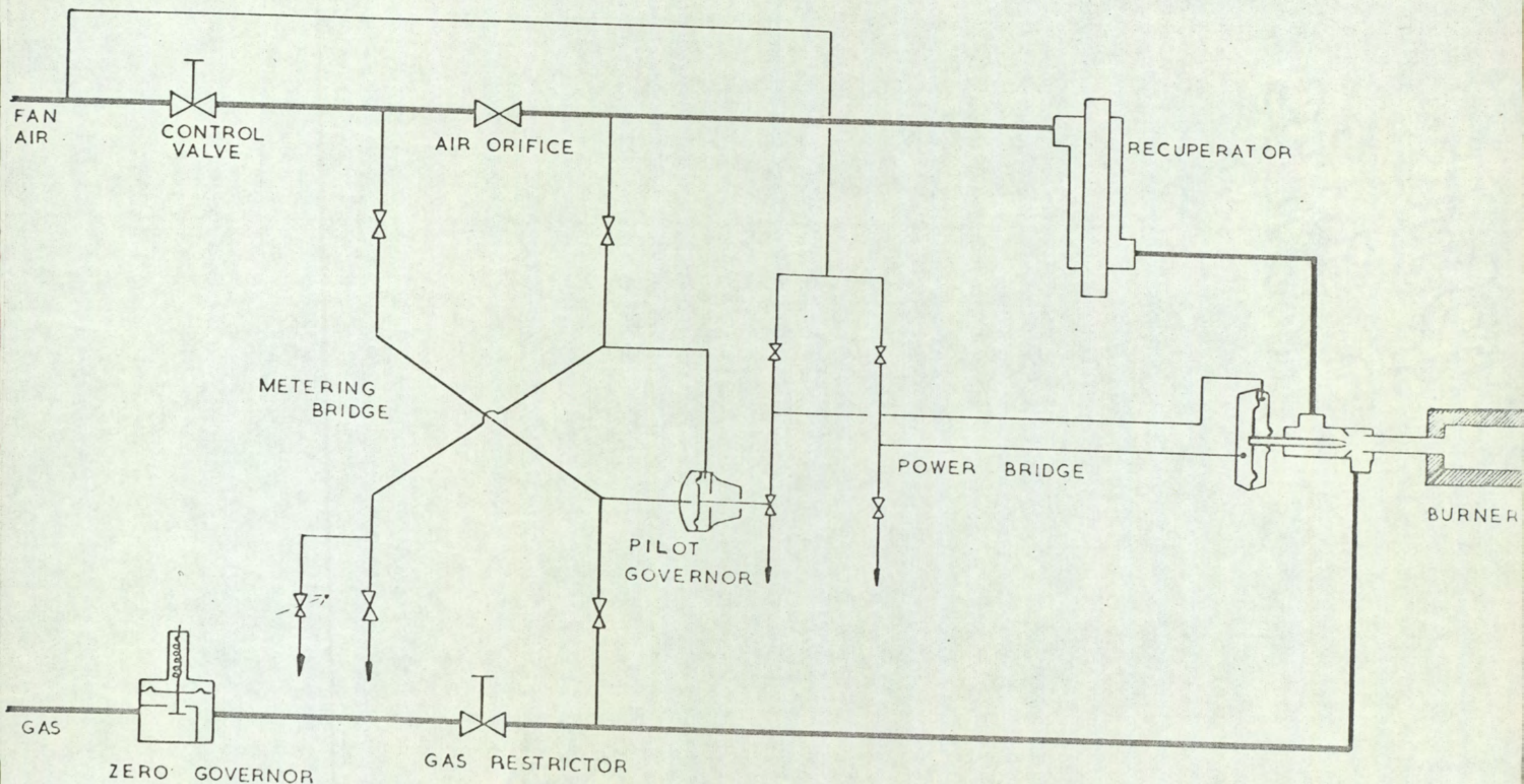

FAN
AIR
CONTROL
VALVE
AIR ORIFICE
RECUPERATOR
METERING
BRIDGE
POWER BRIDGE
PILOT
GOVERNOR
BURNER
GAS
ZERO GOVERNOR
GAS RESTRICTOR

FIG. 7-15

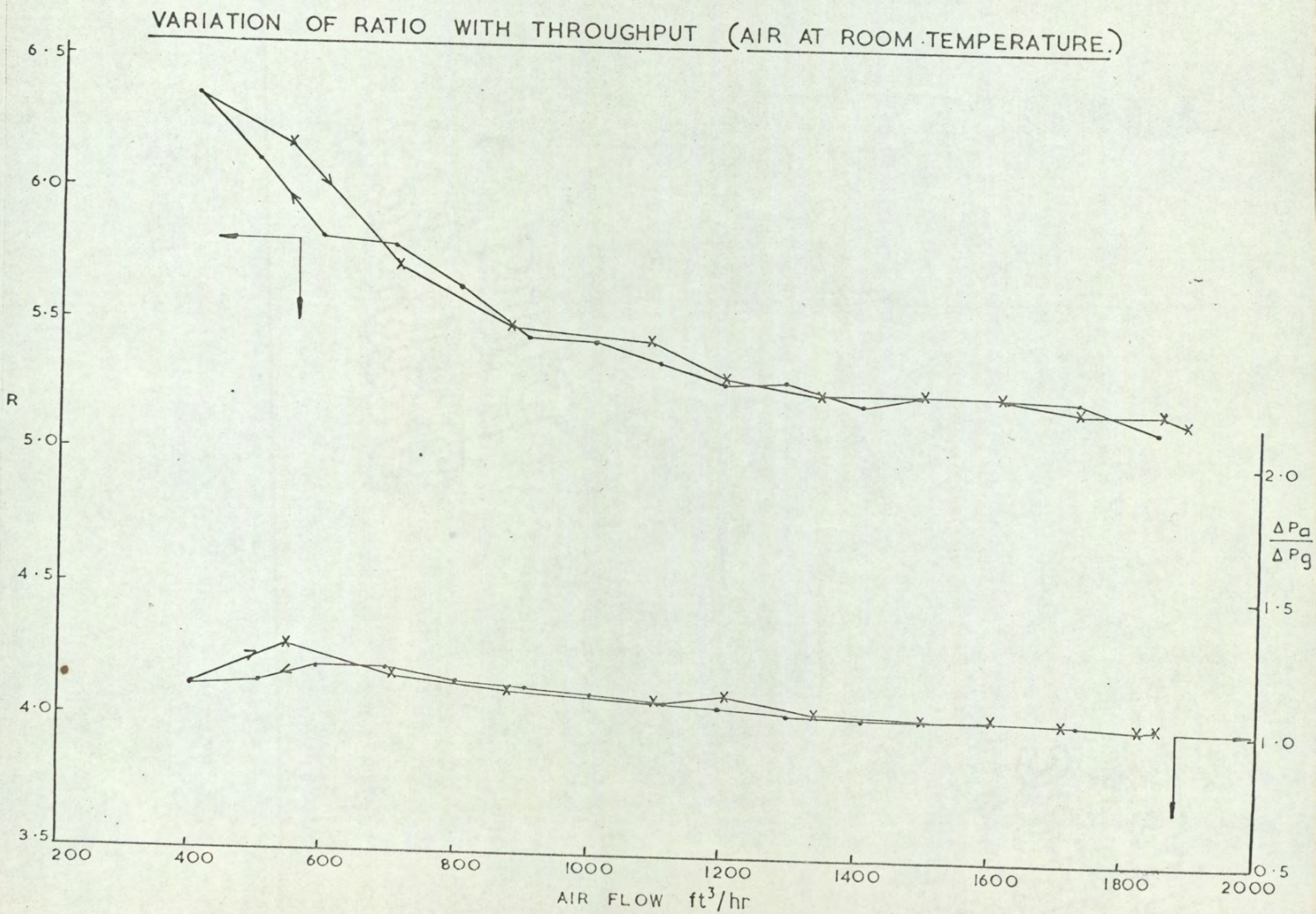

VARIATION OF RATIO WITH THROUGHPUT (AIR AT ROOM TEMPERATURE.)
R
6·5
6·0
5·5
5·0
4·5
4·0
3·5
ΔPa / ΔPg
2·0
1·5
1·0
0·5
200
400
600
800
1000
1200
1400
1600
1800
2000
AIR FLOW ft³/hr

FIG. 7-16

method of setting up an air blast burner in industrial practice. When correctly adjusted the zero governor was found to deliver gas at a pressure of + 0.090 in.w.g.

Fig.7-17 shows the variation of air/gas ratio with throughput and also the variation of $\frac{\Delta P_a}{\Delta P_g}$ with throughput.

The scatter of the experimental points was thought to be within the accuracy normally expected of air blast equipment and in view of the encouraging nature of these results further experiments were performed to determine the variation of air/gas ratio with throughput at a series of air preheat temperatures. The results of these latter experiments are shown in Fig.7-18.

It will be noticed that although the value of R differed slightly from the value obtained in the previous experiment, as shown in Fig.7-17, it was reasonably constant both for variations in throughput and air preheat temperature.

In order to check the system for repeatability, two further experiments were performed at fortnightly intervals to determine the variation of air/gas ratio with throughput at a series of air preheat temperatures. The settings of the bridge adjustment, gas restrictor and zero governor were left untouched from the experiment described above and the results obtained are shown in Figs. 7-19 and 7-20, these results are directly comparable with Fig.7-18. It will be noticed that the results shown in Fig.7-19 show rather more scatter than those of the other experiments but that the average air/gas ratio is reasonably constant in all cases.

VARIATION OF RATIO WITH THROUGHPUT (AIR AT ROOM TEMPERATURE.)

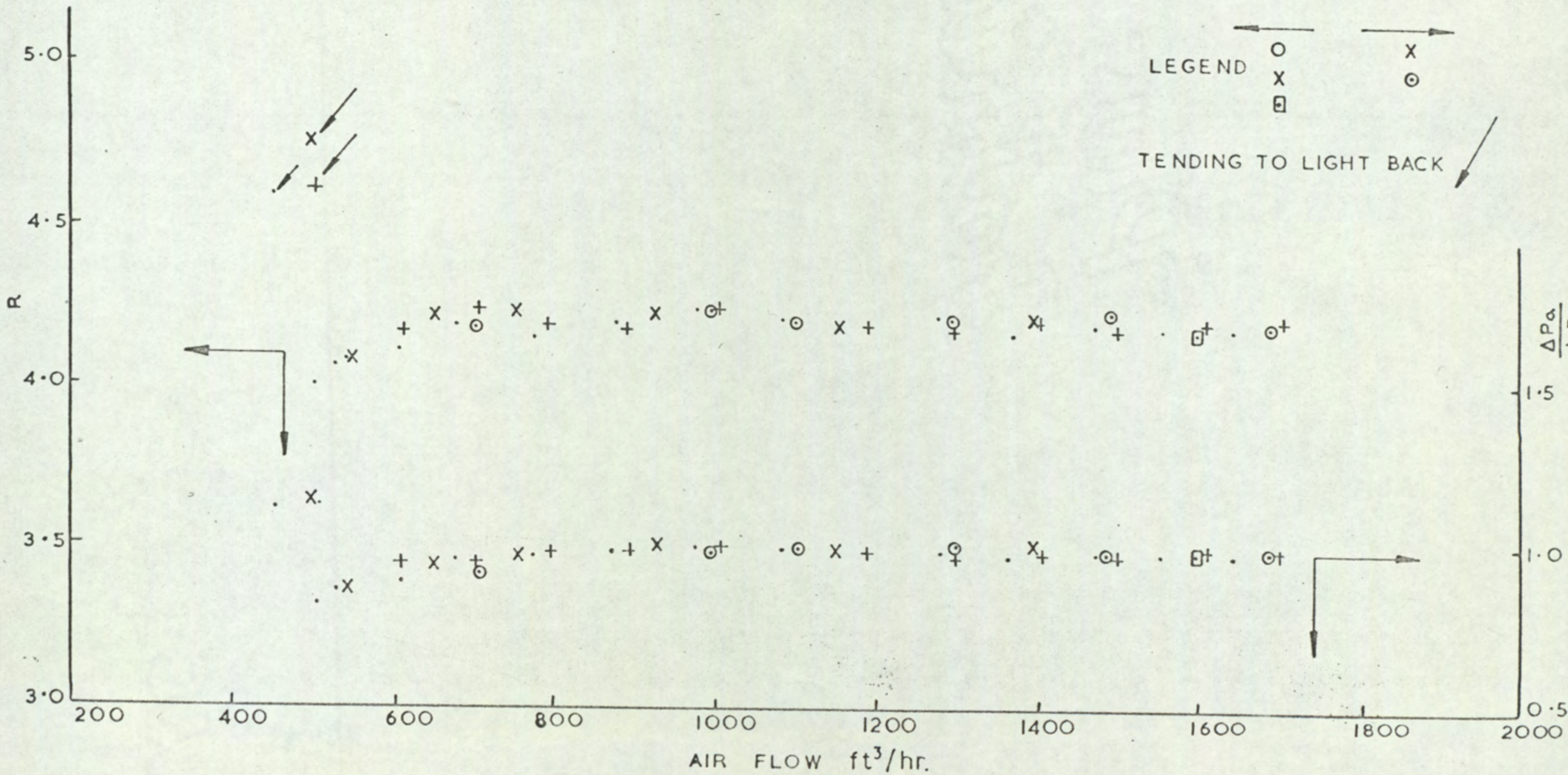


FIG. 7-17

VARIATION OF AIR/GAS RATIO WITH THROUGHPUT AT A SERIES OF TEMPERATURES.

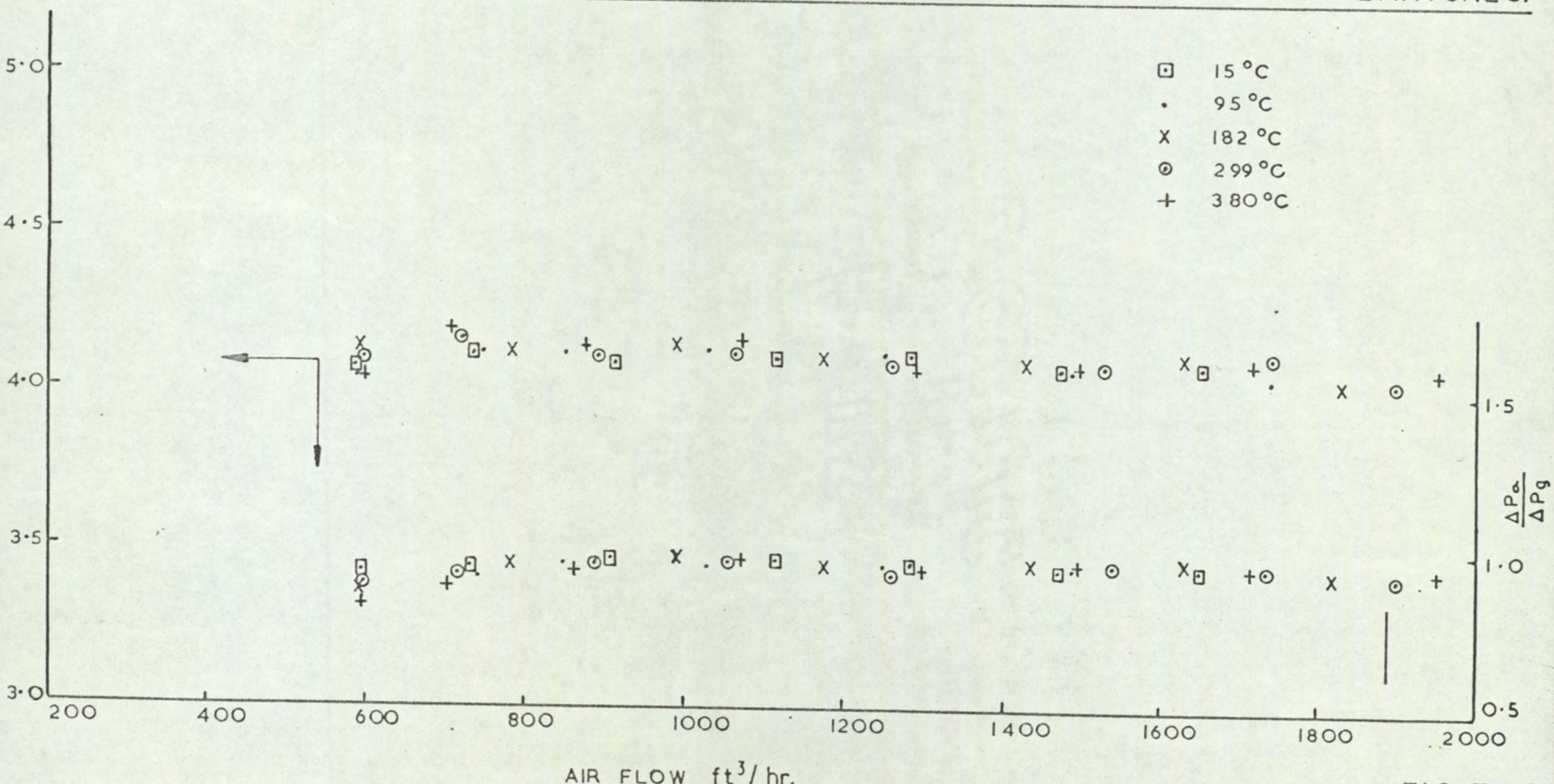


FIG. 7-18

VARIATION OF AIR/GAS RATIO WITH THROUGHPUT AT A SERIES OF TEMPERATURES.

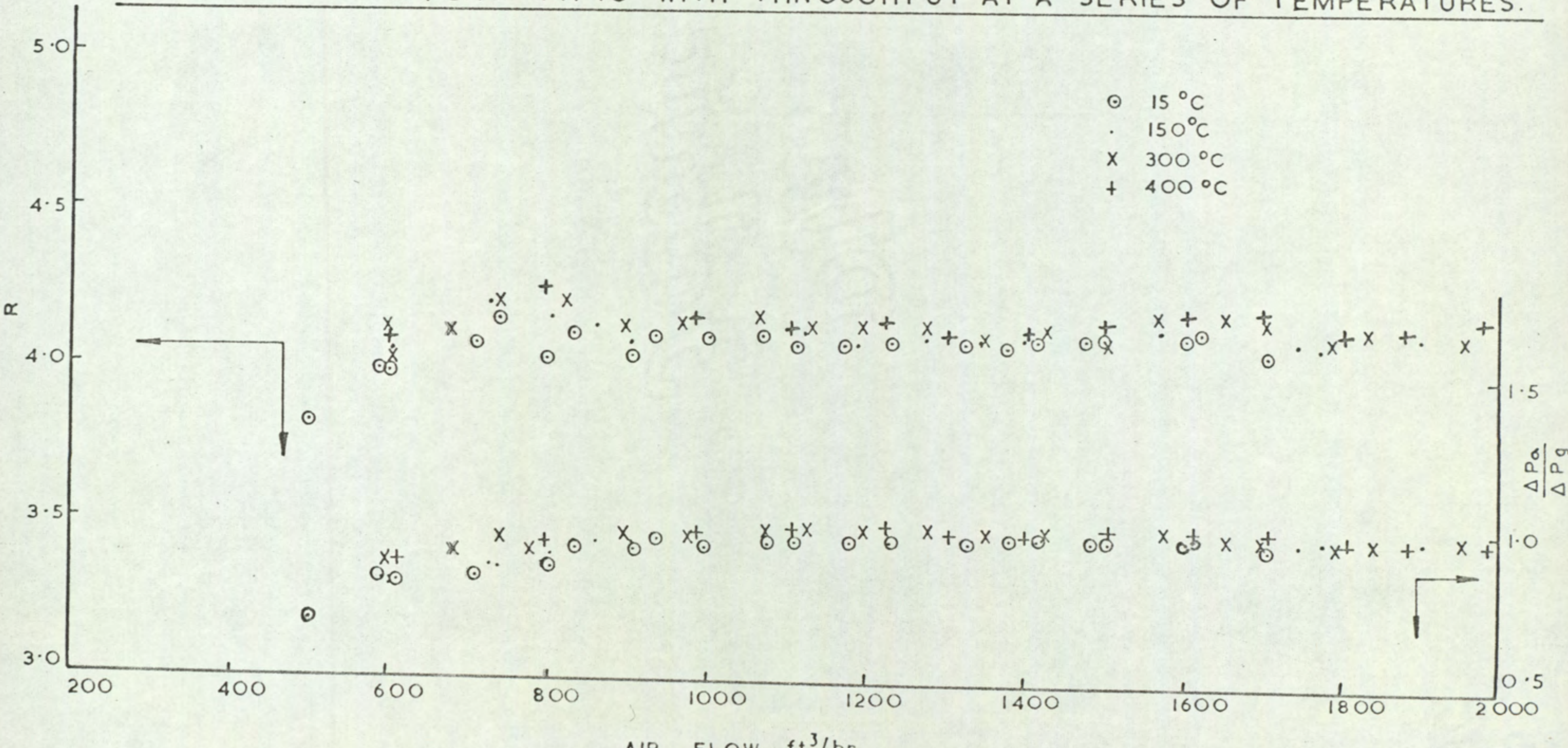


FIG. 7-19

VARIATION OF AIR/GAS RATIO WITH THROUGHPUT AT A SERIES OF TEMPERATURES.

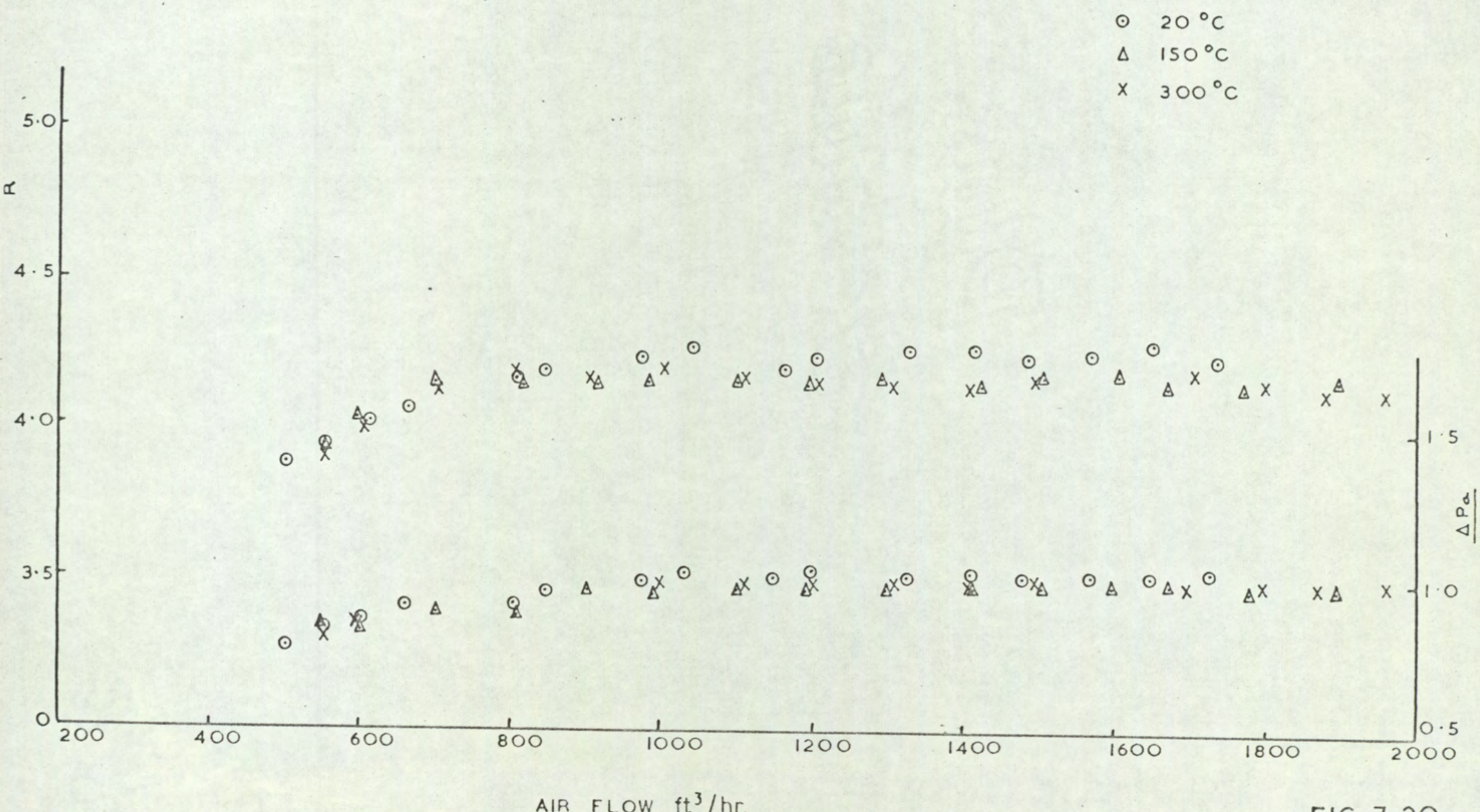


FIG. 7-20

The air/gas ratio obtained with air at room temperature (20°C) shown in Fig.7-20 are noticeably lower than those obtained at higher temperatures, 4.25:1 compared with 4.15:1, and this is thought to be due to the failure of the system to balance due to the plug being jammed fully forward (at all flows almost full forward power was applied to the plug). This appeared to be due to air leakage from the recuperator requiring the injector to entrain proportionally more gas than it was designed to do at room temperature. In fact, during the experiment, particularly at the higher air temperatures, it was noticed from the appearance of the flame that the mixture was gas rich even though the metered air and gas supplies were in approximately stoichiometric proportions.

However since the system itself was satisfactory, attention was directed at producing a prototype injector system suitable for industrial use.

Mk.VII Injector

In redesigning an injector system consisting of a measuring bridge, power bridge and injector, for industrial use the following factors were considered:-

1. Reliability. The system must have few moving parts and be capable of working in adverse conditions i.e. high ambient temperatures and dust laden atmospheres.
2. Simplicity. The injector must be relatively easy to commission and have long term stability.

3. Cost. The price of the equipment must be very low if the economic advantages of recuperation are to be realized.

Bridge System

The redesigned measuring bridge and power bridge were based on "Enots" solderless fittings manufactured by Messrs Benton and Stone and are shown in Fig.7-21. These fittings were chosen as they were inexpensive, compact and could conveniently be piped up with 3/8 in.o.d. and 1/4 in.o.d. annealed copper tubing.

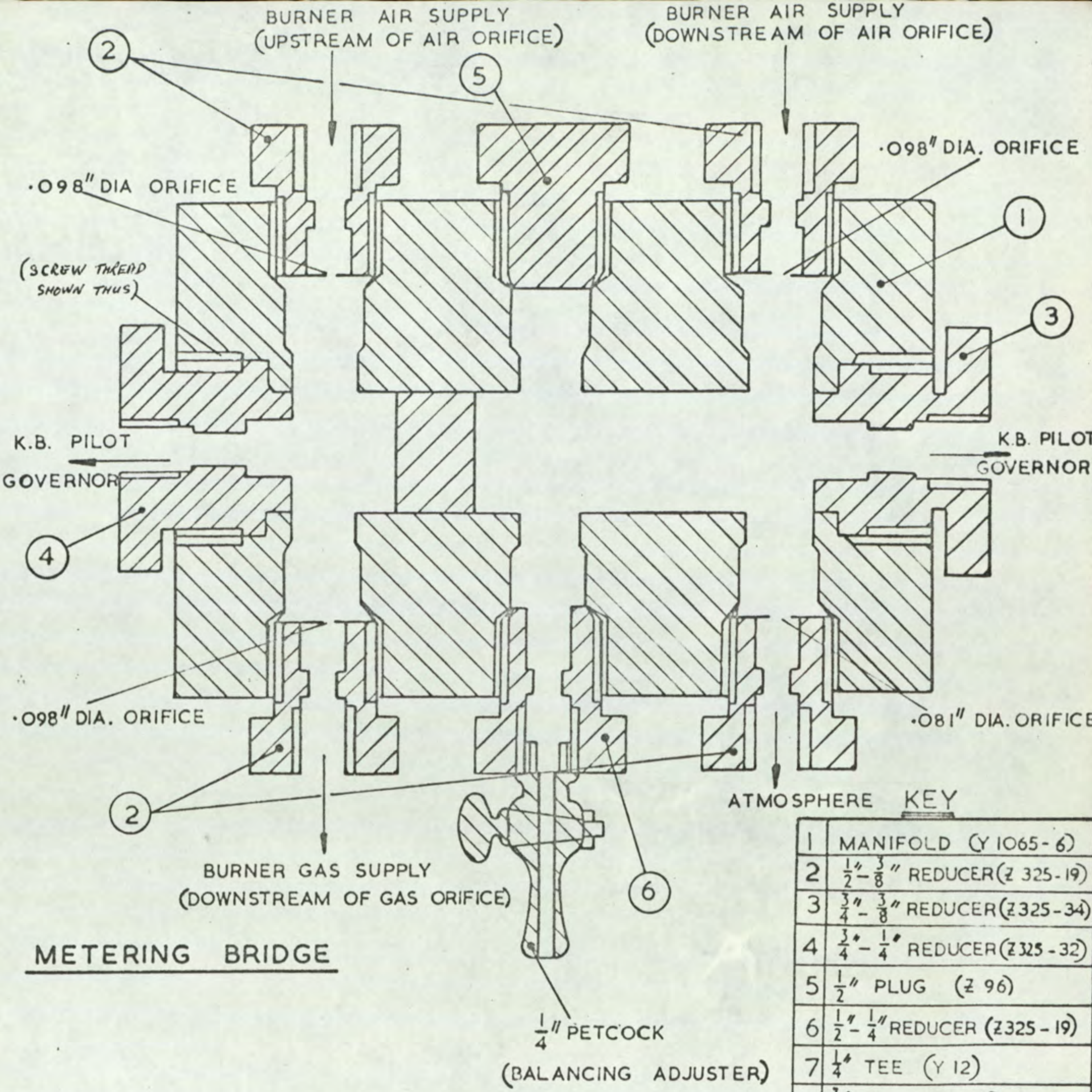
The orifice plates in the measuring bridge were identical to the orifices used in the Mark VI bridges i.e. three orifices 0.098 in.dia (No.40 drill) and one 0.081 in.dia. (No.46 drill), all orifices having chamfered exits and drilled through 18 guage copper discs 11/16 in.dia.

The orifice plates in the power bridge were designed to have the same flow characteristics as the Kieth Blackman pilot valve when the latter was in its mid position. These orifices had chamfered exits and were drilled in 18 gauge copper discs, two discs being 1/2 in.dia. and one 3/8 in.dia.

A photograph of the measuring bridge, Kieth Blackman pilot governor and power bridge installed on the test rig is shown in Fig.7-22.

Injector

The use of a metering bridge and power amplifier technique enabled the design of the actual injector to be greatly simplified. Firstly only a single "motor" diaphragm was required to drive the plug into and out of the air nozzle. Secondly the availability of ample



METERING BRIDGE

KEY

1	MANIFOLD (Y 1065-6)
2	$\frac{1}{2}$"–$\frac{3}{8}$" REDUCER (Z 325-19)
3	$\frac{3}{4}$"–$\frac{3}{8}$" REDUCER (Z 325-34)
4	$\frac{3}{4}$"–$\frac{1}{4}$" REDUCER (Z 325-32)
5	$\frac{1}{2}$" PLUG (Z 96)
6	$\frac{1}{2}$"–$\frac{1}{4}$" REDUCER (Z 325-19)
7	$\frac{1}{4}$" TEE (Y 12)
8	$\frac{3}{8}$" TEE (Y 908)
9	$\frac{1}{4}$" NIPPLE (Z 373-2)
10	$\frac{3}{8}$"–$\frac{1}{4}$" REDUCER (Z 325-32)
11	$\frac{1}{4}$" TUBE NUT (Z 17)

POWER BRIDGE

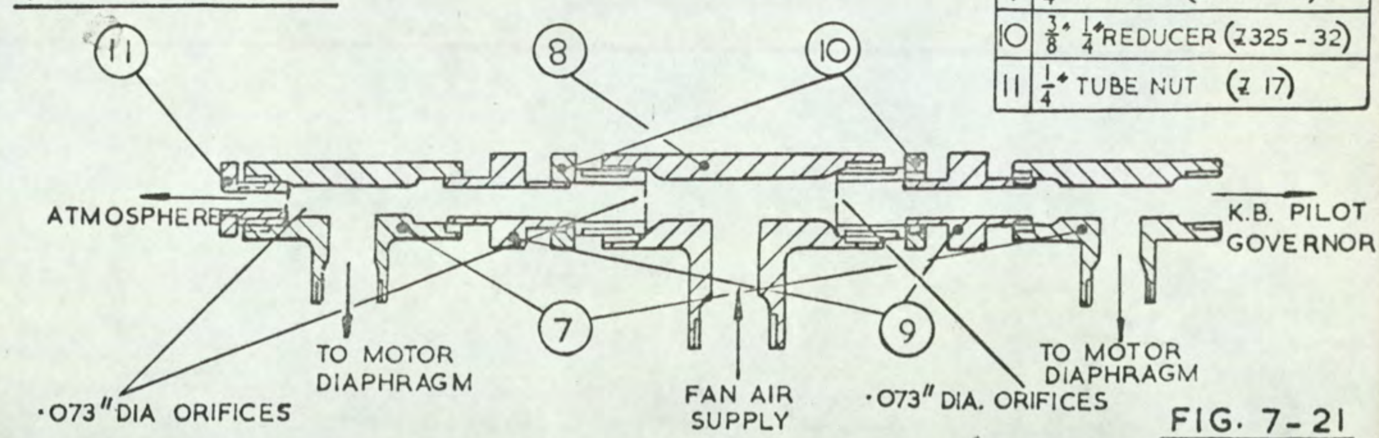


FIG. 7-21

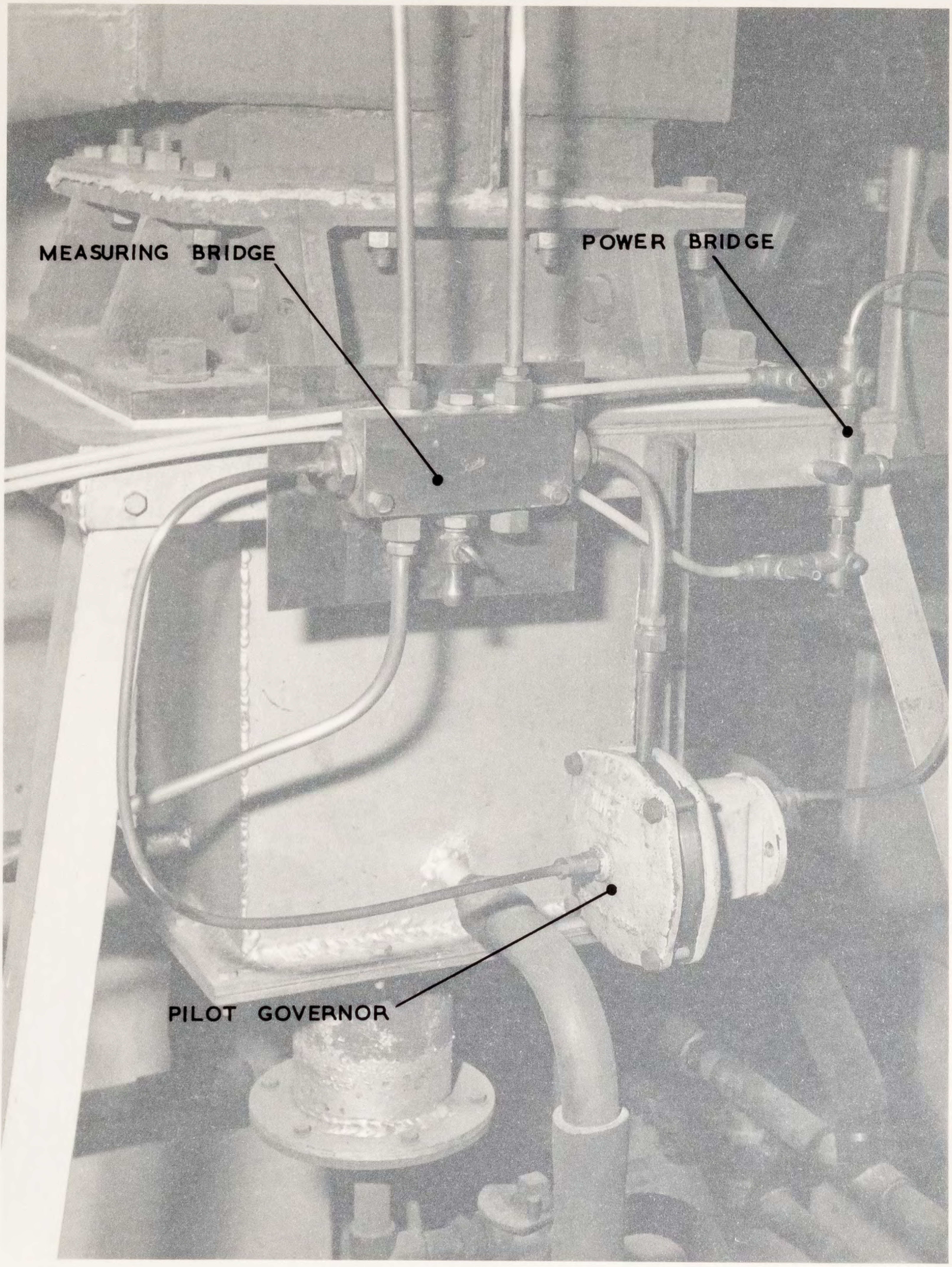
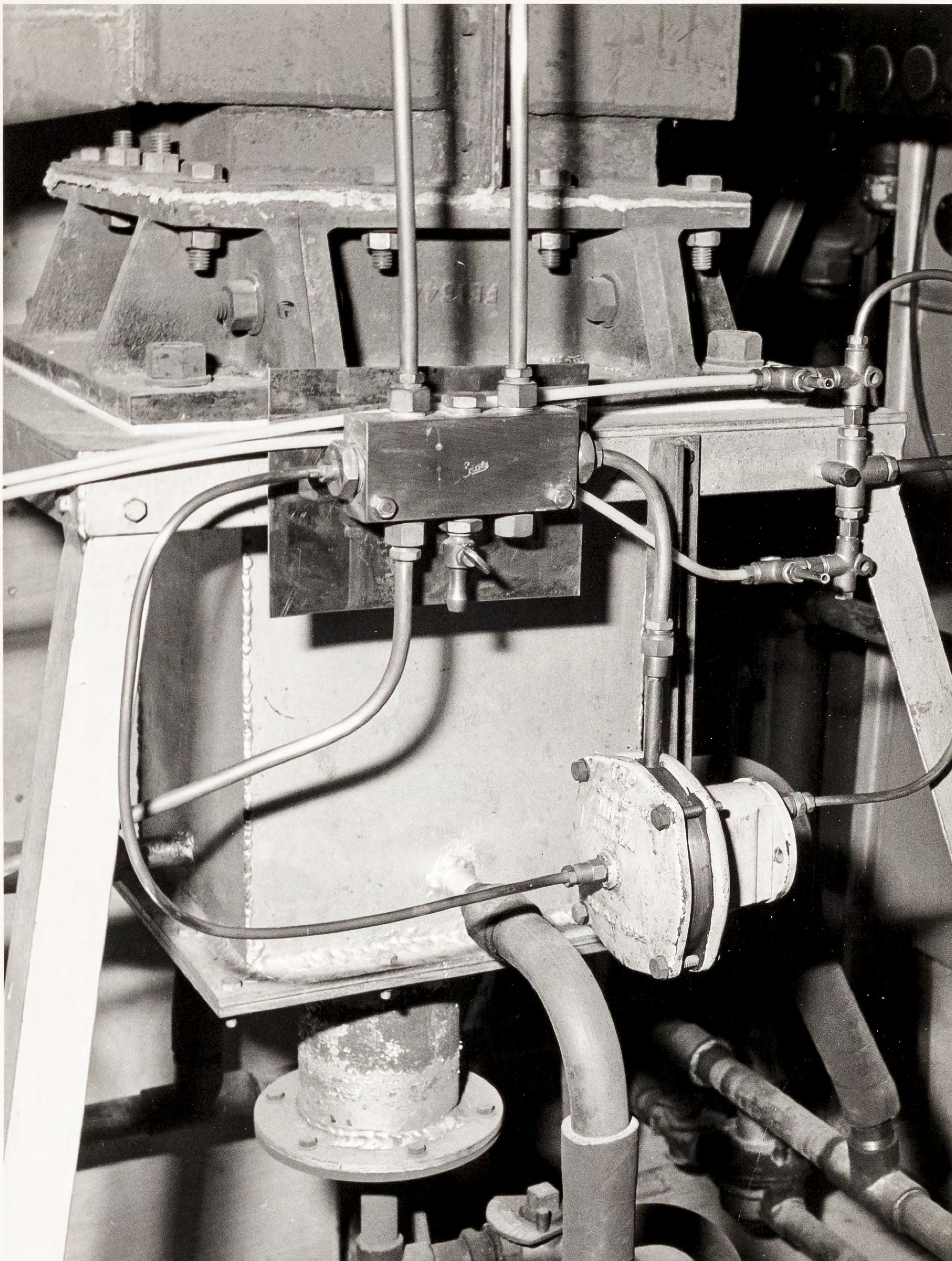


FIG 7-22



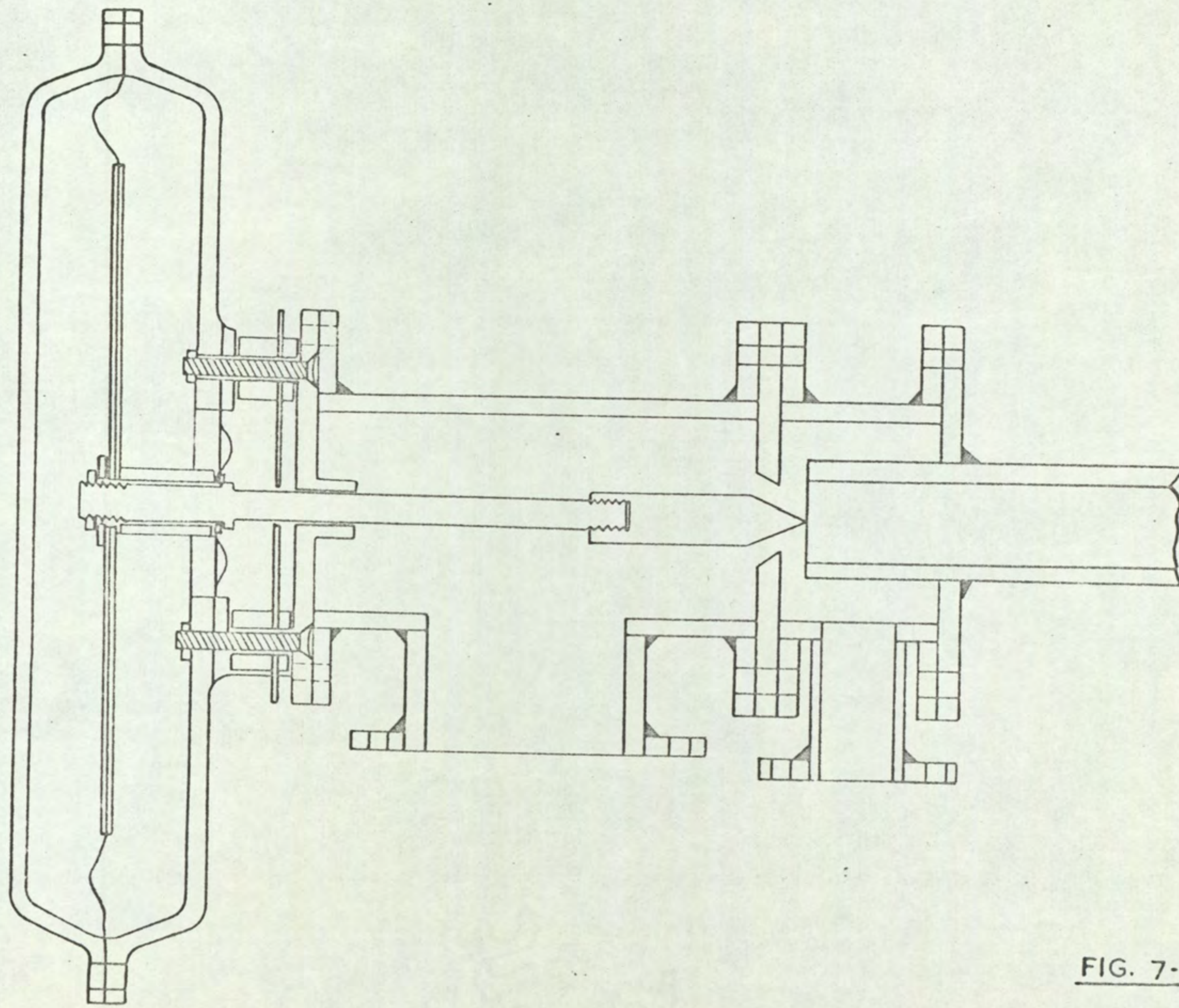


FIG. 7-23

a 1/4 inch thick disc inserted into the inlets to the motor diaphragm chambers, hunting of the system was eliminated.

The flow characteristic of the damping orifices is shown in Fig. 7-24 and it is obvious from this, and from the fact that the swept volume of the motor diaphragm was about 50 cu.in., that full stroke operation of the injector plug would take several minutes. This was confirmed by an experiment to determine the transient response of the system. The burner was lit with the plug in the fully retracted position and the variation of air/gas ratio with time was noted. The results are presented in Fig.7-25 which shows that, after one overshoot the time to reach the set air/gas ratio of 4:1 was six minutes.

It was felt that a transient response time of this order was not unreasonable since in practice the time taken for the combustion air temperature to change by several hundred degrees centigrade would be appreciably longer.

Experiments were performed to determine the variation of air/gas ratio with throughput, the combustion air being at room temperature, and the results of these experiments are shown in Fig.7-26. These results, which can be compared with those shown in Fig.7-17 for the Mk.VI injector, show that at throughputs above about 780 ft^3/hr of air the air/gas ratio was maintained within 3.8% of the set value of 4.15:1 and that at lower throughputs to within an accuracy of +4.6% and -3.8%. Although these errors are larger than occurred with the Mk.VI injector they are within the $\pm$ 5% limit acceptable for industrial use.

Experiments were therefore continued on the Mk.VII injector to

FLOW CHARACTERISTIC OF 14 THOU. DIAMETER DAMPING ORIFICE.

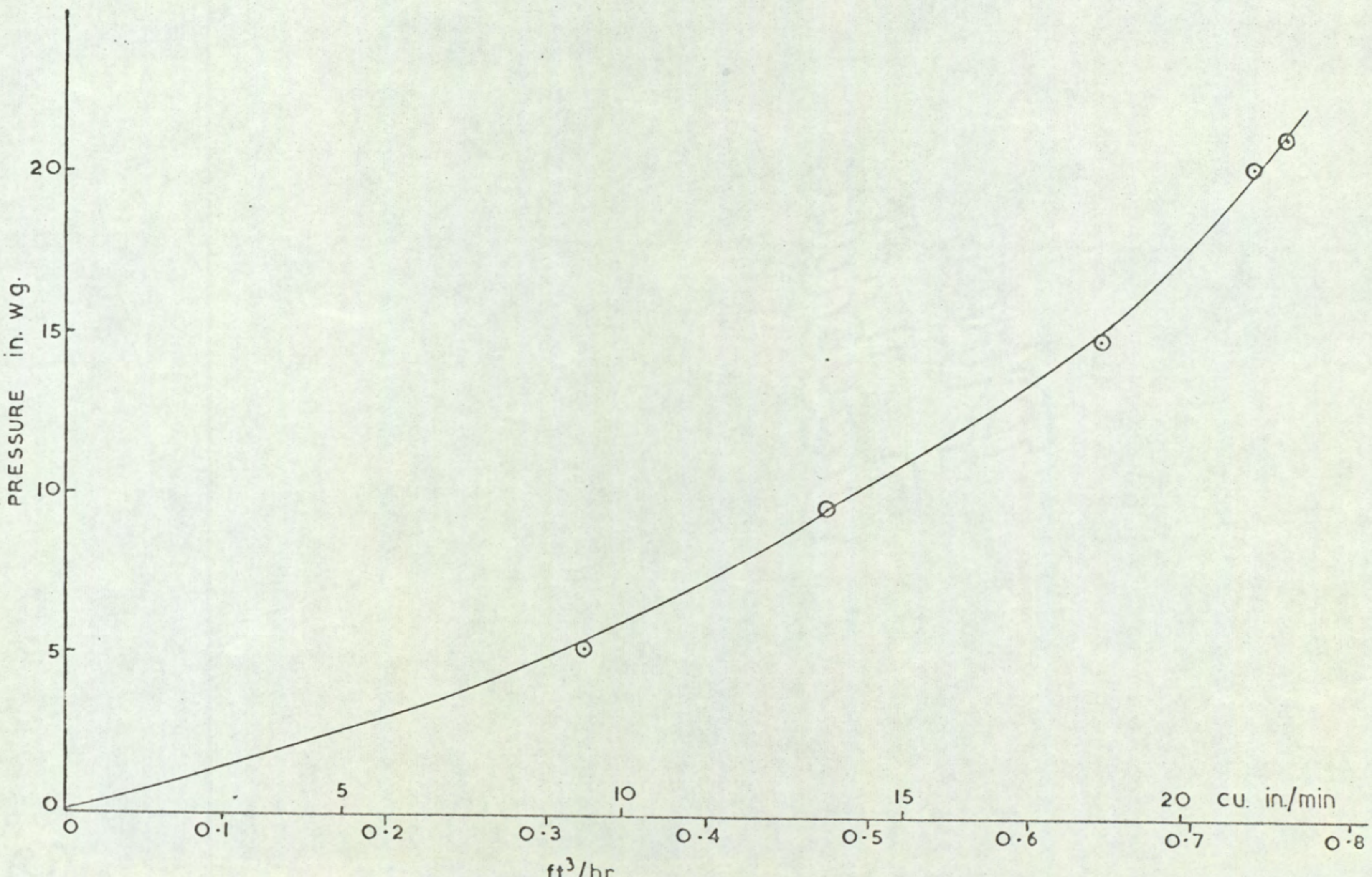
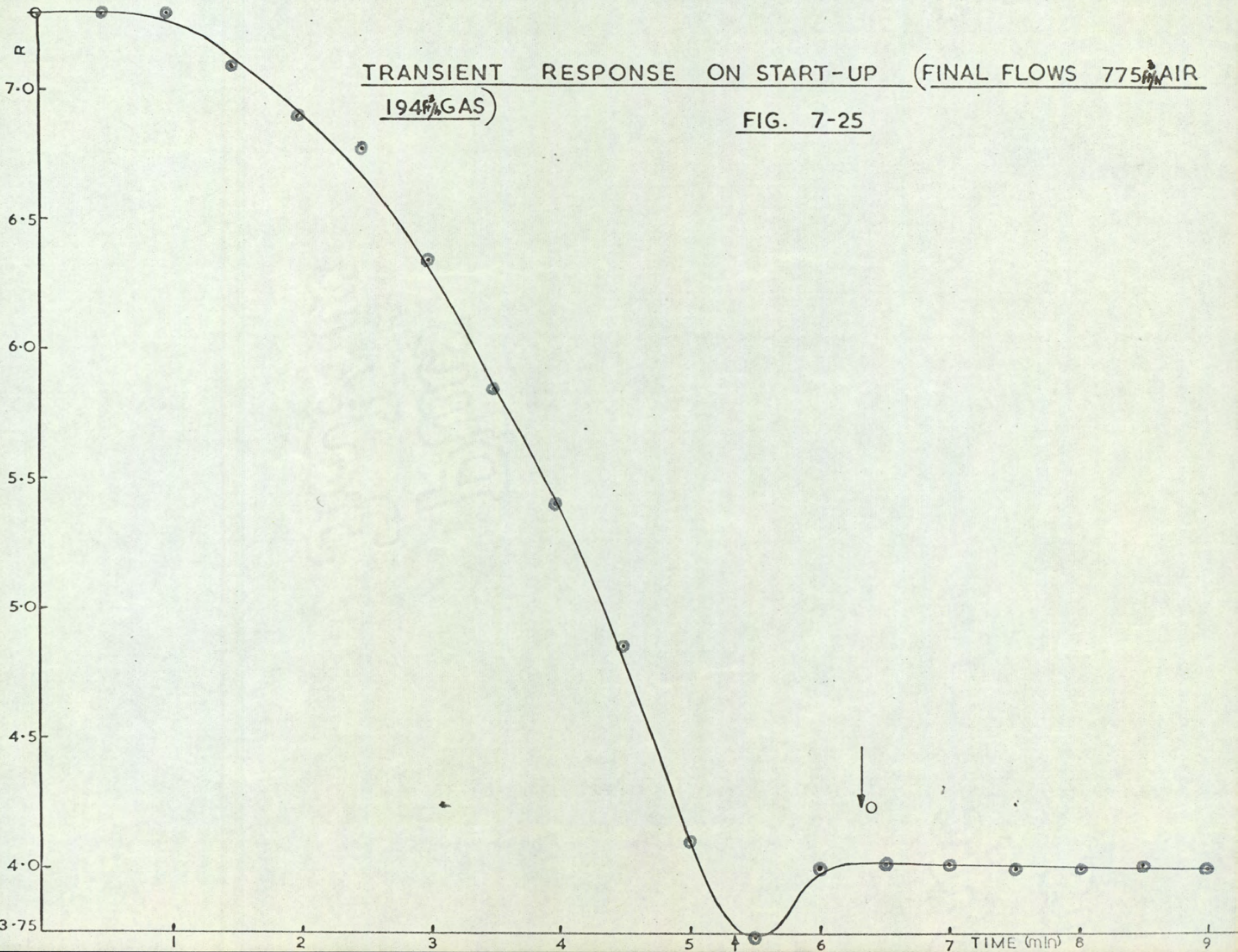


FIG. 7-24



TRANSIENT RESPONSE ON START-UP (FINAL FLOWS 775 ft^3/h AIR 194 ft^3/h GAS)

FIG. 7-25

VARIATION OF RATIO WITH THROUGHPUT.

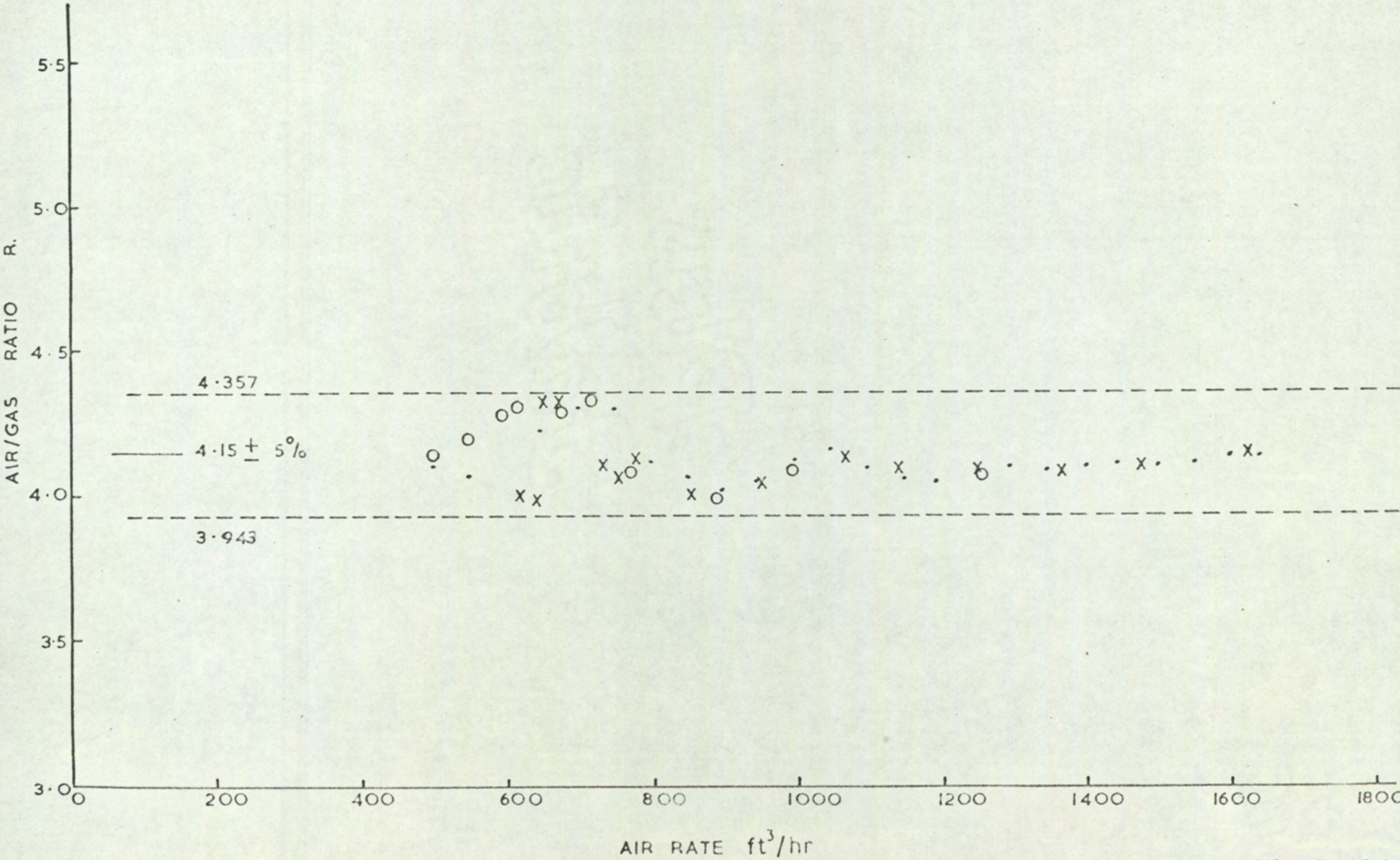


FIG. 7 - 26

examine the effect of preheated combustion air on its performance.

It was found that the system maintained the air/gas ratio constant as the air temperature varied up to an air temperature of 300°C. At this temperature the plug was in the fully retracted position and above this temperature the system did not control.

The experimental results, which are presented Fig.7-27, show no significant variation of air/gas ratio with air temperature at temperatures below 300°C. The results obtained with an air temperature of 350°C on the other hand, i.e. 50°C outside the control range of the system, show an appreciable deviation of the ratio from the desired value.

Two basic conclusions are drawn from the results of these experiments.

Firstly, the prototype bridge control system maintains a constant air/gas ratio regardless of changes in air temperature and, over the working range of throughputs (i.e. 600 ft^3/hr - 1700 ft^3/hr), maintains the ratio within the desired tolerance of $\pm$ 5%.

Secondly, the prototype injector performs satisfactorily; except that it restricts the maximum control temperature to 300°C. At this temperature the plug is in the fully retracted position and consequently the air nozzle area cannot increase further as the temperature rises. It would appear from this that the effective discharge coefficient of the air nozzle is only 0.66 compared with the required value of at least 0.765, discussed above on page VII-17, this factor must be taken into account in designing any future injector of this type.

In the light of the above results it was decided that the main

VARIATION OF AIR/GAS RATIO WITH THROUGHPUT AT A SERIES OF TEMPERATURES.

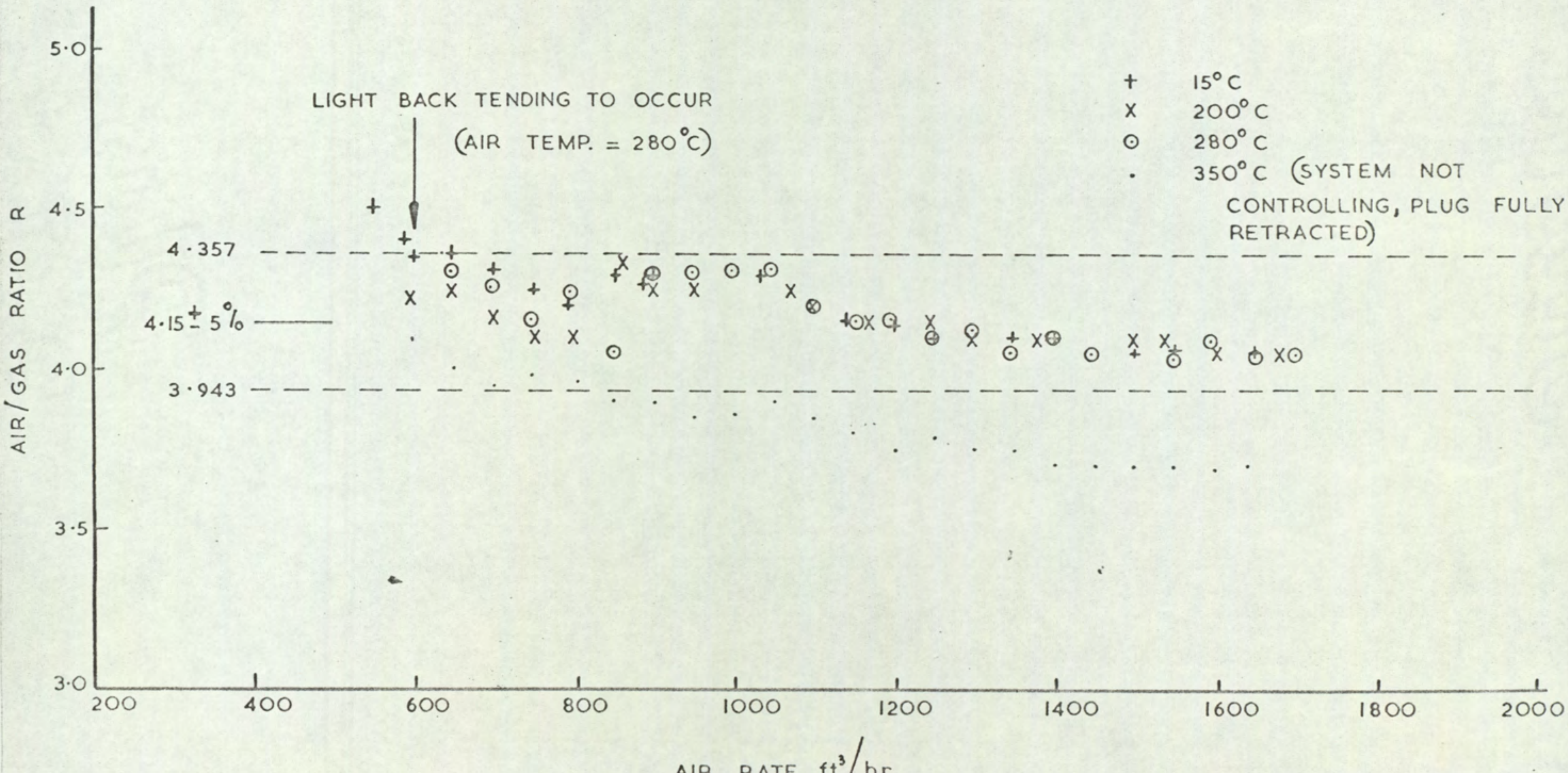


FIG. 7-27

objective of the work had been accomplished and the experimental work was concluded.

VIII

CONCLUSIONS

The value of recuperation by air preheat as a means of waste heat recovery has been illustrated, and the fuel saving that can be attained by using recuperators of various efficiencies has been calculated for town gas furnaces operating at various temperatures.

The recuperator used in this work was chosen as a commercially available unit, which, from the manufacturers specification, would be an economically attractive addition to an industrial furnace. Fitted to a furnace operating at 1000°C the anticipated fuel saving was 21.5% which, on a furnace operating for 50 weeks per year at 40 hours per week, represents a cash saving of £73.5 p.a. (based on a price for gas of 18d per therm). This compares with a capital cost for the unit of £250 which would thus be cleared in 3.4 years.

In practice the unit did not meet the manufacturers specification; giving a maximum preheat temperature of about 400°C compared with an anticipated 500°C when supplied with products of combustion at about 1000°C. The resulting fuel saving would therefore be about 17.5%, representing a cash saving of £61.3 p.a. on the above basis. A period of 5.7 years would therefore be required to clear the capital cost of the unit.

Whilst the inefficiency of the unit was due to high wall loss, lagging the outside of the unit resulted in the mild steel casework rising to excessively high temperatures.

In addition to problems associated with heat transfer in the unit, considerable difficulty was encountered due to air leakage from the packed joints between the heat exchanger tube and the case work, at one time this amounted to 20% of the air entering the recuperator. Whilst the use of packing and sliding joints may be necessary to allow for differential expansion, it is obvious that such joints should be incorporated in the flue gas side of a recuperator, which is generally at about atmospheric pressure, rather than on the air side which can be subject to pressures of 1 - 2 P.S.I.G.

It was felt that a leak tight recuperator could be designed having a performance comparable to that anticipated from the unit used in this work, and which could be marketed at a considerably lower cost. It is the author's view that such a unit would attract considerable interest from the gas industry.

The general need to control air/gas ratios to the stoichiometric value has been discussed, and the percentage of fuel wasted due to deviations from this ratio for a variety of furnace operating temperatures has been calculated.

The effect of preheating the combustion air upon the air/gas ratio produced by air blast burners has been examined in detail and the mathematics, originally developed by Francis and Jackson[2], for this type of burner has been extended to include the effect of the gas restrictor and gas supply pressure, and their influence on variations in air/gas ratio due to air preheat changes.

A technique for controlling the air/gas ratio to an air blast burner to within $\pm$ 5% of the stoichiometric value over a range of throughput and preheat temperatures has been developed and a prototype system for industrial use has been produced.

During the course of the work a detailed examination was made of the forces acting on flexible diaphragms and the results provide an explanation of the performance characteristics of gas governors.

In general the outlet pressure of an appliance governor tends to vary with throughput by several tenths of an inch w.g. over the working range. On the other hand 'zero' governors are usually accurate to within a few hundredths of an inch w.g. The results of the experiments described in Section VI indicate that the difference could well be due to the fact that a 'zero' governor is a null balance device and the changing effective area of the diaphragm has no effect on the magnitude of the outlet pressure. An appliance governor, on the other hand is a force balance device and the outlet pressure will vary with throughput due to the changing effective area of the diaphragm as it moves through its stroke.

It follows that if accurate pressure control is required, it may be well worth constructing a null balance appliance governor. This could be done by pressure loading a 'zero' governor from a small pilot governor; the outlet pressure of which could be virtually constant since its throughput could be constant.

The pneumatic bridge technique that has been developed also has potential value outside the field of air/gas proportioning. The technique could have possible application where flow ratio control of any two fluids is required providing some contamination of one fluid by the other is permissible. As developed in this work, the bridge output is fed to a sensitive differential pressure amplifier. However, in order to achieve adequate sensitivity, the design of the amplifier necessitates a leak to atmosphere from one diaphragm chamber and the need to compensate for this leak complicates the bridge system. If a differential pressure amplifier having adequate sensitivity, could be designed, which has no leakage of the input signal pressure, for example a diaphragm sensing device having leak tight diaphragm chambers, then the design of the bridge could be greatly simplified.

An amplification technique worthy of consideration would be the use of a pressure switch to sense the bridge output and the use of an electrical actuator as the final control means. Unfortunately however, no suitable pressure switches, i.e. switches having an adequate sensitivity, ruggedness, and reasonable price, are at present commercially available.

The development of either an improved differential pressure amplifier or suitable pressure switch would simplify the pneumatic bridge technique and enable it to be used for air/gas ratio control on burners other than of the air blast premix type and as an alternative to conventional flow ratio controllers for more general flow ratio control duty.

The air/gas ratio controller that has been developed relies on metering the air supply to the burner upstream of the recuperator. Its application is therefore restricted to installations using air tight recuperators and where each burner has an individual recuperator. It is felt that any future work on this topic should be directed at overcoming these restrictions and in particular at providing accurate air/gas ratio control on installations where many burners are supplied with hot air from a single recuperator.

Experience during this current work would indicate that the most promising line of attack would be to control the throughput of each burner by means of an individual gas valve linked to a variable area air nozzle: the linkage being designed so that the ratio of air nozzle area to gas valve area remains constant over the required throughput range. By supplying gas at a constant pressure from an appliance governor and air at a controlled pressure dependent upon air temperature, it should be possible to maintain a constant air/gas ratio at various throughputs and air preheat temperatures.

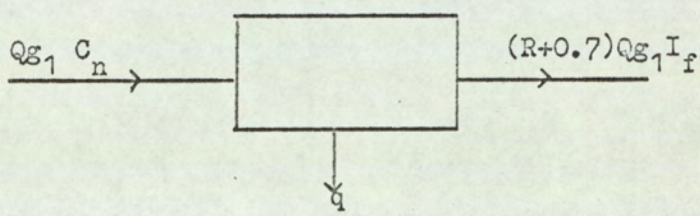
If the temperature of the air leaving the recuperator is measured and its pressure accordingly controlled, (the control valve could of course be upstream of the recuperator so controlling the cold air supply), then many burners could be supplied from a single recuperator and any air leakage from the recuperator would not affect the air/gas ratio.

APPENDIX I

Calculation of Fuel Saving achieved when using Recuperation

The percentage of fuel saved by applying recuperation by air preheat to a furnace can be calculated from an energy balance.

Consider a furnace without recuperation where the waste gases leave the furnace at a temperature T_f having an enthalpy I_f; in which the heat rate to the load together with wall losses is q and where the gas input in ft.^{3}/hr. is Q_{g1}. The energy flow can thus be represented as:-



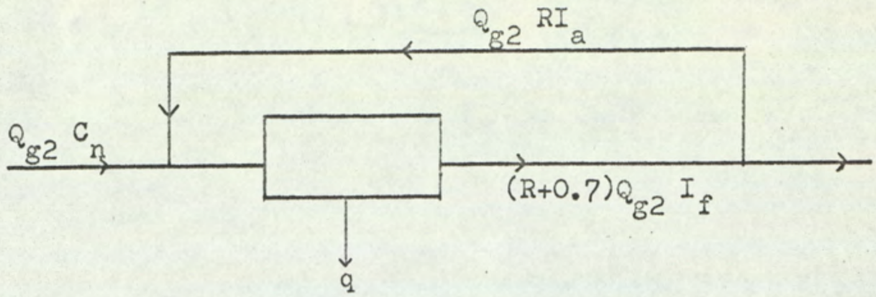
i.e. $Q_{g1}\, C_n \;=\; q + (R + 0.7)\, Q_{g1}\, I_f$ ....(1)

where C_n is the net calorific value of the gas

R is the air/gas ratio

(R +0.7) is the S.T.P. products/gas volume ratio.

If with a similar furnace operating at the same temperature the combustion air is preheated to a temperature T_a (and enthalpy I_a) by recuperation, then the gas input rate Q_{g2} will be lower and the heat flow can be represented as :-



The energy balance is thus

$$Q_{g2}\,C_n + Q_{g2}\,RI_a = q + (R+0.7)\,Q_{g2}\,I_f \qquad \text{..........(2)}$$

From equation (1) $Q_{g1} = \dfrac{q}{(C_n - (R+0.7)I_f)}$ ....(3)

From equation (2) $Q_{g2} = \dfrac{q}{(C_n + RI_a - (R+0.7)I_f)}$ ....(4)

Now the percentage fuel saving achieved by fitting recuperation is:-

$$\frac{Q_{g1} - Q_{g2}}{Q_{g1}} \times 100$$

which can be developed from (3) and (4) as

$$\frac{Q_{g1} - Q_{g2}}{Q_{g1}} = \frac{R\,I_a}{C_n + RI_a - (R+0.7)I_f}$$

Thus the percentage fuel saving on a furnace with a furnace gas temperature T_f, achieved by preheating the combustion air to temperature T_a, can be calculated. This has been done for a series of furnace gas temperatures and air/preheat temperatures and the relevant data tabulated below. The calculations are based on a town gas of net calorific value 437 Btu/ft^3 having a stoichiometric air/gas ratio 4.15:1. Values of enthalpy for air and combustion products were found from the literature[6].

Furnace Gas Temperature °C	Air Preheat Temperature °C	I_f Btu/ft^3.	I_a Btu/ft^3.	Percentage Fuel Saving
200	200	6.6	6.1	5.9
	150		4.5	4.41
	100		2.9	2.9
	50		1.3	1.3
400	400	13.4	12.8	12.5
	300		9.4	9.5
	200		6.1	6.4
	100		2.9	3.1
600	600	21.5	19.9	19.9
	500		16.3	16.9
	400		12.8	13.8
	300		9.4	10.5
	200		6.1	7.1
	100		2.9	3.5
800	800	19.5	27.2	27.7
	700		23.5	24.9
	600		19.9	21.9
	500		16.3	18.7
	400		12.8	15.3
	300		9.4	11.7
	200		6.1	7.9
	100		2.9	3.9
1000	1000	38.0	34.7	36.3
	800		27.2	30.9
	700		23.5	27.8
	600		19.9	24.6
	500		16.3	21.1
	400		12.8	17.4
	300		9.4	13.3
	250		7.8	11.34
	200		6.1	9.1
	100		2.9	4.6

The Enthalpy datum temperature is 60°F.

Cont.......

Furnace Gas Temperature °C	Air Preheat Temperature °C	I_f Btu/ft^3.	I_a Btu/ft^3.	Percentage Fuel Saving
1200	1200	46.7	42.6	45.5
	1000		34.7	40.4
	800		27.2	34.8
	700		23.5	31.5
	600		19.9	28.1
	500		16.3	24.2
	400		12.8	20.1
	300		9.7	15.5
	200		6.1	10.6
	100		2.9	5.5
1400	1400	55.7	50.8	55.8
	1200		42.6	51.4
	1000		34.7	46.3
	800		27.2	40.3
	700		23.5	36.9
	600		19.9	33.1
	500		16.3	28.8
	400		12.8	24.1
	300		9.4	19.0
	200		6.1	13.2
	100		2.9	6.7
1600	1600	65.5	59.0	67.2
	1400		50.8	63.9
	1200		42.6	59.7
	1000		34.7	54.7
	800		27.2	48.6
	700		23.5	45.0
	600		19.9	40.9
	500		16.3	36.2
	400		12.8	30.8
	300		9.4	24.6
	200		6.1	17.5
	100		2.9	9.2

The Enthalpy datum temperature is 60°F

APPENDIX II.

Calculation of Fuel Wasted Due to Deviations From the Stoichiometric Air/Gas Ratio

The percentage of fuel wasted due to deviations from the stoichiometric air/gas ratio can be calculated from knowledge of the heat available from a given volume of gas burned under various conditions.

If one cubic foot of gas is completely burned and the resulting water vapour remains in the vapour phase (as occurs in all but very low temperature applications) then the heat released is the net calorific value (C_n) of the gas. Since in general the combustion products leave the furnace or appliance chamber at an elevated temperature, sensible heat is lost from the furnace and the heat available for doing work on the load is the difference between the heat released and the flue loss; i.e. the difference between the net calorific value and the enthalpy of the flue gases relative to room temperature. Where stoichiometric combustion occurs the available heat H_s per cubic foot of gas burnt can be expressed as

$$H_s = C_n - Q_P I_P$$

Where Q_P is the stp. volume of products produced per ft^3 of gas burnt and I_P is the enthalpy of stoichiometric combustion products at the furnace gas temperature.

If combustion occurs with excess air then the additional air must also be raised to furnace gas temperature so reducing the amount of available heat to a lower value H_a where:-

$$H_a = C_n - Q_p I_p - q_a I_a$$

where q_a is the s.t.p. volume of excess air per ft^3 of gas burnt and I_a is the enthalpy (Btu/ft^3) of air at the Furnace gas temperature.

If insufficient air is present for complete combustion then the available heat H_a per ft^3 of input gas is also limited by the potential heat which remains unreleased. An elementary method of evaluating H_a is to consider stoichiometric combustion as taking place until all the available oxygen is consumed, leaving stoichiometric combustion products and excess town gas. On this assumption:-

$$H_a = C_n - q_g C_n - Q_p I_p - q_g I_g$$

where q_g is the s.t.p. volume of residual town gas per ft^3 of input gas.

Q_p is the s.t.p. volume of stoichiometric combustion products per ft^3 of input gas.

I_p is the enthalpy (Btu/ft^3) of stoichiometric combustion products at the furnace gas temperature.

I_g is the enthalpy (Btu/ft^3) of town gas at the furnace gas temperature.

This method is of course an approximation since in practice the combustion reaction goes to an equilibrium condition which is determined by the final temperature of the combustion products. The residual combustibles are generally H_2 and CO and for a particular town gas their concentration can be determined either by analysis of the combustion products or by calculating the equilibrium composition of the combustion products from a detailed analysis of the particular town gas.

In practice detailed analyses are rarely available and it is felt that the elementary method will generally suffice for practical calculations.

Having calculated H_s and H_a at the particular furnace gas temperature and air/gas ratio the percentage of fuel wasted by not burning at the stoichiometric ratio is given by

$$W = \frac{(H_s - H_a)\,100}{H_s}$$

Figure 2, Section 1, shows a graph of percentage fuel wasted for a gas of Gross C.V. 485 Btu/ft^3 (Net C.V. 437 Btu/ft^3) having a stoichiometric ratio of 4.15:1. These results have been calculated using the elementary method for rich combustion and the actual calculations are given in Table II - 1.

In order to compare the results obtained by calculating the percentage of fuel wasted due to gas rich combustion by the two methods, calculations were performed on a chromatograph analysis of town gas supplied at Solihull on the 17th October, 1963.

The town gas composition (mole fractions) was as follows :-

Table II - 1

lculation of Percentage Fuel Wasted due to Deviations from the Stoichiometric Ratio With a Gas of Gross C.V. 485 Btu/ft^3 (Cn 437 Btu/ft^3) and Stoichiometric Air/Gas Ratio 4.15 : 1

				Furnace Temp. 500°C						Furnace Temp. 100°C						Furnace Temp. 1500°C					
R	Q_p	q_a	q_g	Q_pI_p	q_gC_n	q_gI_g	q_aI_a	H_a	$\frac{H_s - H_a}{H_s}$	Q_pI_f	q_gC_n	q_gI_g	q_aI_a	H_a	$\frac{H_s H_a}{H_s}$	Q_pI_f	q_gC_n	q_gI_g	q_aI_a	H_a	$\frac{H_s - H_a}{H_s}$
3.32	3.88		0.2	68.3	87.4	3.25		278.05	0.209	147.2	87.4	6.92		195.48	0.226	234.2	87.4	10.96		104.44	0.280
3.528	4.125		0.15	72.5	65.5	2.44		296.56	0.157	156.8	65.5	5.19		209.51	0.171	249.0	65.5	8.2		114.3	0.212
3.735	4.36		0.10	76.8	43.7	1.625		314.87	0.105	166	43.7	3.46		223.84	0.114	264	43.2	5.48		123.82	0.146
3.941	4.61		0.05	81.2	21.82	0.81		333.17	0.052	175	21.82	1.73		238.45	0.056	278.2	21.82	2.74		134.24	0.074
4.15	4.85			85.4				351.6	0	184.4				252.6	0	292				145	0
4.5	4.85	0.35		85.4			5.69	345.91	0.016	184.4			12.1	240.5	0.048	292			19.2	125.8	0.132
5.0	4.85	0.85		85.4			13.8	335.8	0.045	184.4			29.4	223.8	0.114	292			46.5	98.5	0.321

H_2	0.5608
O_2	0.0025
N_2	0.1385
CO	0.0472
CO_2	0.0843
CH_4	0.1103
C_2H_4	-
C_3H_8	0.0029
C_3H_6	-
I C_4H_{10}	0.0175
N C_4H_{10}	0.0297

This analysis gives a stoichiometric air/gas ratio of 4.133 and a gross C.V. of 473.4 Btu/ft^3 (Net C.V. 425-4 Btu/ft^3).

Table II - 2 shows the calculation of the percentage fuel wasted due to deviations on the rich side of stoichiometric at four furnace gas temperatures. The results shown in column A are the results obtained from the equilibrium compositions and those in column B are obtained by the elementary method. The differences between the two percentages are shown in column C from which it can be seen that in general the simple approach gives a lower percentage of

Table II - 2

Calculation of Fuel wasted due to deviations on the rich side of stoichiometric by the "equilibrium" method and by the "residual gas" method. Town gas of Gross c.v. 437.4 Btu/ft^3 (C_n = 425.4 Btu/ft^3) Stoichiometric air/gas ratio 4.133:1 .

R	Temp.°K.	N_2	H_2O ~~N_2O~~	CO_2	CO	H_2	Q_P	x $Q_P \times CO \times 318$	y $Q_P \times H_2 \times 270$	Residual Heat (x + y)	Temp.	Q_PI_P	$Q_PI_P + x + y$	Ha	H_s	$\frac{H_s - H_a}{H_s} \times 100$ (A)	Q_P	Q_PI_F	I_g	$q_g = (1 - \frac{R}{4.133})$	I_gq_g	q_gC_n	$Q_pI_f + I_gq_g + q_gC_n$	Ha	$\frac{H_s - H_a}{H_s} \times 100$ (B)	A - B	$\frac{A - B}{A}$
4	1200	0.68744	0.21179	0.09016	0.00401	0.00659	4.7	5.993	8.363	14.356	927°C	167	181.4	244.0		5.24	4.69	163.1		0.03218	1.0233	13.689	177.812	247.588	3.55	1.69	0.323
3.75		0.67401	0.20648	0.08562	0.01260	0.02129	4.45	17.830	25.580	43.410	1700°F	155	198.4	227.0	257.5	11.84	4.39	152.9	31.8	0.09267	2.9469	39.422	195.269	230.131	10.63	1.21.	0.1022
3.50		0.65939	0.20040	0.08096	0.02166	0.03758	4.20	28.929	42.616	71.545	I_f 34.8	146	217.5	207.9		19.26	4.10	142.9		0.15316	4.8705	65.154	212.925	212.475	17.49	1.77	0.0919
3.25		0.64340	0.19343	0.07616	0.03128	0.05572	3.95	39.291	59.425	98.716		137.3	236.0	189.4		26.45	3.80	132.2		0.21365	6.7941	90.887	229.881	195.519	24.07	2.38	0.0900
4	1400	0.68744	0.21292	0.08903	0.00514	0.00546		7.682	6.929	14.611	1127°C	204.2	218.8	206.6		4.13	4.69	204			1.2743	13.689	218.963	206.437	4.21	-0.08	-0.0194
3.75		0.67401	0.20977	0.08233	0.01589	0.01800		22.456	21.627	44.113	2060°F	193.7	237.8	187.6	215.5	12.95	4.39	190.9	39.6		3.6697	39.422	233.992	191.408	11.18	1.77	0.137
3.50		0.65939	0.20559	0.07577	0.02685	0.03239		35.861	36.730	72.591	I_f 43.5	183	255.6	169.8		21.21	4.10	178.2			6.0651	65.154	249.419	175.981	18.34	2.87	0.135
3.25		0.64340	0.20024	0.06935	0.03849	0.04891		47.845	52.163	100.008		172	272	153.4		28.82	3.80	165.2			8.4605	90.887	264.548	160.852	25.36	3.46	0.120
4	1600	0.68744	0.21372	0.08823	0.00594	0.00466		8.878	5.914	14.792	1327°C	245.6	260.4	165.0		4.90	4.69	244.7			1.5318	13.689	259.921	165.479	4.62	0.28	0.0571
3.75		0.67401	0.21217	0.07998	0.01824	0.01565		25.811	18.803	44.614	2420°F	232.1	276.7	148.7	173.5	14.29	4.39	229.4	47.6		4.4111	39.422	273.233	152.167	12.30	1.99	0.139
3.50		0.65939	0.20928	0.07208	0.03054	0.02870		40.789	32.546	73.335	I_f 52.2	219.3	292.6	132.8		23.46	4.10	214			7.2904	65.154	286.444	138.956	19.91	3.55	0.151
3.25		0.64340	0.20505	0.06454	0.04290	0.04410		53.887	47.033	100.920		206.2	307.1.	118.3		31.82	3.80	198.5			10.1697	90.887	299.557	125.843	27.47	4.35	0.137
4	1800	0.68744	0.21430	0.08765	0.00652	0.00408		9.745	5.178	14.923	1527°C	290.5	305.4	120.0		5.81	4.69	289.9			1.7956	13.689	305.385	120.015	5.80	0.01	0.0017
3.75		0.67401	0.21382	0.07828	0.01994	0.01395		28.217	16.761	44.978	2780°F	275	320.0	105.4	127.4	17.27	4.39	271.2	55.8		5.1710	39.422	315.793	109.607	13.97	3.30	0.191
3.50		0.65939	0.21197	0.06939	0.03323	0.02601		44.382	29.495	73.877	I_f 61.8	260	333.9	91.5		28.18	4.10	253.2			8.5463	65.154	326.900	98.500	22.68	5.50	0.195
3.25		0.64340	0.20853	0.06106	0.04638	0.04062		58.258	43.321	101.579		244.1	345.7	79.7		37.44	3.80	234.5			11.9218	90.887	337.308	88.091	30.85	6.59	0.176

fuel wasted than the equilibrium approach. However for any assessment of the efficiency of an industrial process the differences are probably insignificant and the elementary method will generally give an adequately accurate value.

APPENDIX III

Calculation of the Relationship between Gas Restrictor Area and Air Preheat Temperature required to Achieve a Constant Air/Gas Ratio with Varying Preheat.

If an air blast injector is designed to produce a stoichiometric ratio when supplied with air at room temperature, the proportion of gas entrained will increase as the air temperature is raised. This effect can be overcome if the gas restricting valve is progressively closed as the air temperature rises. In order to evaluate the possibility of continuously adjusting the gas restrictor automatically, according to the air preheat temperature, the relationship between restrictor area and preheat temperature necessary to maintain a constant air/gas ratio, has been calculated for Midlands Research Station burners type A and C.

The following symbols are used.

A	Cross-section area
g	Acceleration due to gravity
K	Constant; involving the discharge coefficient expansion factors etc. at an orifice.
$P_{1,2,\text{ etc.}}$	Pressure
Q	Volume flow rate
R	Air/gas ratio by volume at s.t.p.
S	Pressure loss in velocity heads due to friction. [This term includes both friction factor and dimensional ratio. The numerical values of S used are those quoted by Frances and Jackson[2] appropriate to the type of burner considered.]

T	Absolute temperature
ρ	Density
σ	Specific gravity of gas (Air=1)
P	Products/gas ratio by volume at s.t.p.

Subscripts

0	s.t.p. conditions
1	Air nozzle
2	Downstream of gas restrictor but before mixing
3	Upstream of gas restrictor
4	Mixture tube after entrainment and enlargement
5	Mixture nozzle
6	Combustion chamber after combustion zone
7	Exit nozzle
a	Air supply
g	Gas supply

Considering the burner shown in Fig.III-1 and writing down the various changes in static pressure through the burner gave the following series of equations. It can be assumed that the gas supply enters the air jet at right angles and so adds nothing to the axial momentum.

Pressure drop in tunnel due to friction

$$P_6 - P_7 = S_{67} \frac{\rho_7 Q_7^2}{2gA_7^2} \qquad \text{.......... (1)}$$

PARALLEL INJECTOR PARALLEL TUNNEL BURNER.

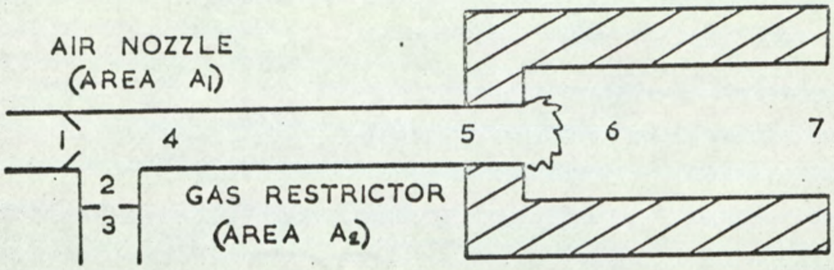


FIG. III-1

PARALLEL INJECTOR, CONVERGING TUNNEL BURNER.

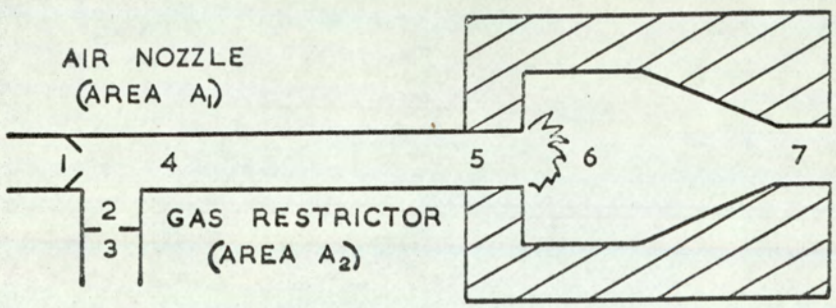


FIG. III-2

Change in momentum between points 5 and 6.(i.e. mixture nozzle and tunnel).

Combustion is assumed to occur in this section.

$$(P_5 - P_6)A_6 = \frac{\rho_6 Q_6^2}{g\ A_6} - \frac{\rho_5 Q_5^2}{gA_5}$$

$$\therefore \quad P_5 - P_6 = \frac{\rho_6 Q_6^2}{gA_6^2} - \frac{\rho_5 Q_5^2}{gA_5 A_6} \qquad \text{.......... (2)}$$

Pressure drop in mixture tube due to friction:

$$P_4 - P_5 = S_{45}\ \frac{\rho_5 Q_5^2}{2gA_5^2} \qquad \text{.......... (3)}$$

Change in momentum on mixing

$$(P_2 - P_4)A_4 = \frac{\rho_4 Q_4^2}{g\ A_4} - \frac{\rho_2 Q_2^2}{g\ A_1}$$

$$\therefore \quad P_2 - P_4 = \frac{\rho_4 Q_4^2}{g\ A_4^2} - \frac{\rho_2 Q_2^2}{gA_1 A_4} \qquad \text{.......... (4)}$$

Since the total mass flow between sections 4 and 7 is constant

$$\rho_4 Q_4 = \rho_5 Q_5 = \rho_6 Q_6 = \rho_7 Q_7 \qquad \text{.......... (5)}$$

From the definition of air/gas ratio and products/gas ratio

$$Q_g = \frac{Q_1}{R}\ \frac{T_o}{T_1} = \frac{Q_4}{R + 1}\ \frac{T_o}{T_4} = \frac{Q_6}{P}\ \frac{T_o}{T_6} \qquad \text{.......... (6)}$$

Also since $\rho_4 Q_4 = \rho_1 Q_1 + \rho_g Q_g$

$$\frac{\rho_1 Q_1}{\rho_4 Q_4} = \frac{R}{R + \sigma} \qquad \text{..........} \; (7)$$

Summing equations (1), (2), (3) and (4) to eliminate intermediate pressures gives:

$$P_2 - P_7 = \frac{\rho_4 Q_4^2}{g A_4^2} - \frac{\rho_2 Q_2^2}{g A_1 A_4} + \frac{S_{45} \rho_5 Q_5^2}{2g A_5^2} + \frac{\rho_6 Q_6^2}{g A_5^2} - \frac{\rho_5 Q_5^2}{g A_5 A_6} + \frac{S_{67} \rho_7 Q_7^2}{2g A_7^2} \qquad ..(8)$$

Now the flow of gas through the gas restrictor is given by:

$$Q_g = K_2 A_2 \sqrt{\frac{(P_3 - P_2)\, g}{\rho_g}}$$

$$\therefore \qquad P_3 - P_2 = \frac{Q_g^2 \rho_g}{K_2^2 A_2^2 g}$$

In practice P_3 is governed to atmospheric pressure or P_7

$$\therefore \qquad P_2 - P_7 = P_2 - P_3 = -\frac{Q_g^2 \rho_g}{K_2^2 A_2^2 g} \qquad \text{..........} \; (9)$$

Making appropriate substitutions from (5),(6),(7) and (9) in each term of equation (8) so as to express all Q's in terms of Q_g and ρ's in terms of ρ_1 gives:

$$\frac{\rho_1 (R + \sigma)(R + 1)\, Q_g^2 T_1 T_4}{g A_4^2 \; T_0^2} - \frac{\rho_1 \; R^2 Q_g^2 T_1^2}{g A_1 A_4 T_0^2} + \frac{S_{45} \rho_1 \; (R + \sigma)(R + 1) \; Q_g^2 T_1 T_4}{2g A_5^2 T_0^2}$$

$$+ \frac{\rho_1 (R + \sigma) P \; Q_g^2 T_1 T_6}{g A_6^2 \; T_0^2} - \frac{\rho_1 \; (R + \sigma)(R + \;) \; Q_g^2 T_1 T_4}{g A_5 \; A_6 \; T_0^2} + \frac{S_{67} \; \rho_1 \; (R + \sigma) \; P \; Q_g^2 T_1 T}{2g A_7^2 \; T_0^2}$$

$$= - \frac{Q_g^2 \rho_g}{K_2^2 A_2^2 g} \qquad \text{..........} \quad (10)$$

Dividing equation (10) throughout by $Q_g^2 \rho_1 T_1$, multiplying through by T_o and making the substitutions $\rho_1 = \rho_a \dfrac{T_o}{T_1}$ and $\sigma = \dfrac{\rho_g}{\rho_a}$

where ρ_a is the S.T.P. density of air

and finally regrouping gives:

$$\left[\frac{(R + \sigma)(R + 1)}{A_4^2} + \frac{S_{45}\,(R + \sigma)(R + 1)}{2\,A_5^2} - \frac{(R + \sigma)(R + 1)}{A_5 A_6} \right] T_4 - \frac{R^2}{A_1 A_4}\, T_1$$

$$+ \left[\frac{(R + \sigma)\; P}{A_6^2} + \frac{S_{67}\,(R + \sigma)\; P}{2\,A_7^2} \right] T_6 = \frac{-\sigma\, T_o}{K_2^2\, A_2^2} \qquad \text{..........} \quad (11)$$

Neglecting any dissociation of products, the dependence of both T_4 and T_6 on T_1 has been calculated from published data[6] assuming $R = 3.8$, $P = 4.5$, Gas of gross CV = 450 Btu/ft^3, and found to be :-

$$T_6 = 0.331 \;\; T_1 + 3932.7$$

$$T_4 = 0.79 \;\; T_1 + 110$$

Substituting these values of T_4 and T_6 in equation (11) gives:

$$T_1 \Bigg\{ \left[\frac{(R + \sigma)(R + 1)}{A_4^2} + \frac{S_{46}(R + \sigma)(R + 1)}{2\,A_5^2} - \frac{(R + \sigma)(R + 1)}{A_5\, A_6} \right] 0.79 - \frac{R^2}{A_1 A_4}$$

$$+ \left[\frac{(R+\sigma)P^2}{A_6^2} + \frac{S_{67}(R+\sigma)P^2}{2\,A_7^2} \right] 0.331 \Big\} + \Big\{ \left[\frac{(R+\sigma)(R+1)^2}{A_4^2} + \frac{S_{45}(R+\sigma)(R+1)^2}{2\,A_6^2} \right.$$

$$\left. - \frac{(R+\sigma)(R+1)}{A_5 A_6} \right] 110 + \left[\frac{(R+\sigma)P^2}{A_5^2} + \frac{S_{67}(R+\sigma)P^2}{2\,A_7^2} \right] 3932.7 \Big\} = \frac{-\sigma T_o^2}{K_2^2 A_2^2}$$

The above equation is general to any parallel tunnel, parallel injector burner.

Therefore consider a burner of this type designed for maximum pressure efficiency, (Midlands Research Station Type A,) and substitute the appropriate values in the above equation.

$R = 3.8$

$P = 4.52$

$\sigma = 0.475$

$S_{45} = S_{67} = 0.3$

$A_1 = 0.10521$ sq. in.

$A_4 = A_5 = 0.29224$ sq.in.

$A_6 = A_7 = 0.78540$ sq.in.

$T_o = 520^oR.$

This gives $K_2A_2 = \sqrt{\dfrac{1}{1.255\ T_1 - 656.8}}$ .... (12)

This equation shows that for:

$T_1 = 523^oR$ K_2A_2 is infinite

$T_1 < 523^oR$ K_2A_2 is imaginary

$T_1 < 523^oR$ K_2A_2 is real and gets smaller with increasing T_1

The application of a similar mathematical argument to a burner having a parallel injector, converging tunnel as shown in Fig. 2 leads to the following general equation.

$$\begin{aligned}
T_1\Bigg\{ &- \left[\frac{(R+\sigma)\,P}{2\,A_6^2} + \frac{(1+S_7)(R+\sigma)\,P}{2\,A_7^2}\right] 0.331 + \frac{R^2}{A_1A_4} + \left[\frac{(R+\sigma)(R+1)}{A_5A_6}\right. \\
&\left. - \frac{S_{45}\,(R+\sigma)(R+1)}{2\,A_4^2} - \frac{(R+\sigma)(R+1)}{A_4^2}\right] 0.79\Bigg\} + \Bigg\{- \left[\frac{(R+\sigma)\,P}{2\,A_6^2}\right. \\
&\left. + \frac{(1+S_7)(R+\sigma)\,P}{2\,A_7^2}\right] 3932.7 + \left[\frac{(R+\sigma)(R+1)}{A_5A_6} - \frac{S_{45}\,(R+\sigma)(R+1)}{2\,A_4^2}\right. \\
&\left. - \frac{(R+\sigma)(R+1)}{A_4^2}\right] 110\Bigg\} = \frac{\sigma\,T_o}{K_2^2A_2^2} \qquad \text{..........} \quad (13)
\end{aligned}$$

Considering the M.R.S. type C burner we can substitute the appropriate values as follows:

$S_{45} = 0.3$

$S_7 = 0.12$

$R = 3.8$

$\sigma = 0.475$

$P = 4.52$

$A_1 = 0.12946$ sq.in.

$A_4 = A_5 = 0.40152$ sq.in.

$A_7 = 0.78540$ sq.in.

$A_6 = 2.92553$ sq.in.

$T_o = 520^\circ R.$

This gives $K_2A_2 = \sqrt{\dfrac{1}{0.6874\ T_1 - 354.7}}$ ....(14)

Thus for $T = 516^{o}R$ K_2A_2 is infinite

$T < 516^{o}R$ K_2A_2 is imaginary

$T > 516^{o}R$ K_2A_2 is real and gets smaller with increasing T_1

The foregoing calculations have been based on burners that are optimized for maximum pressure efficiency and the final expressions for K_2A_2 indicate that with combustion air near room temperature K_2A_2 is very sensitive to air temperature. In view of this the calculation was repeated for two non-optimized burners namely 1 inch type A burners having under-sized air nozzles; the air nozzles having diameters of 0.360 in. and 0.300 in. compared with the optimized burner nozzle of 0.366 in.

Substituting the appropriate dimensions in the general equation for a parallel tunnel, parallel injector burner the following expressions were obtained:-

1 in. Type A burner having a 0.360 in. dia. air nozzle

$$K_2A_2 = \sqrt{\frac{1}{1.31916\ T_1 - 656.815}}$$

1 in. Type A burner having a 0.300 in. dia. air nozzle

$$K_2A_2 = \sqrt{\frac{1}{2.1838\ T_1 - 656.815}}$$

In order to illustrate the above results graphically, in terms of a plot of restrictor area A_2 versus air preheat temperature T_1, the factor K was assumed to have a value of 0.8484, entry coefficients were neglected and a discharge coefficient for the restrictor of 0.6 was assumed.

The expressions for A_2 thus become:-

1 in. Type A with 0.366 in. dia. air nozzle $A_2 = \sqrt{\dfrac{1}{0.90360\ T_1 - 472.9068}}$

1 in. Type A with 0.360 in. dia. air nozzle $A_2 = \sqrt{\dfrac{1}{0.94980\ T_1 - 472.9068}}$

1 in. Type A with 0.300 in. dia. air nozzle $A_2 = \sqrt{\dfrac{1}{1.57234\ T_1 - 472.9068}}$

1 in. Type C $A_2 = \sqrt{\dfrac{1}{0.49493\ T_1 - 255.384}}$

Air Preheat		Gas Restrictor Area (in^2)			
		A Type Burner			C Type Burner
^oC	^oR	Air Nozzle 0.366 in.	Air Nozzle 0.360 in.	Air Nozzle 0.300 in.	
0	460	-	-	0.06320	-
10	510	-	0.29499	0.05513	-
20	528	0.48830	0.18703	0.05290	0.41034
25	537	0.28482	0.16409	0.05189	0.31018
50	582	0.13737	0.11188	0.04755	0.17496
100	672	0.08629	0.07777	0.04139	0.11380
200	852	0.05803	0.05453	0. 03397	0.07755
300	1032	0.04665	0.04440	0.02949	0.06258
400	1272	0.04009	0.03840	0.02642	0.05388
500	1392	0.03569	0.03432	0.02414	0.04803
600	1572	0.03249	0.03131	0.02237	0.04374

APPENDIX IV

Variation of Air/Gas Ratio with Air Temperature for an A type Burner having an Undersize Air Nozzle.

In view of the relatively small restrictor adjustment required to maintain a constant air/gas ratio with varying air temperature for a non-optimized burner, as shown in Appendix III, the **air/gas** ratio with cold air has been calculated for a 1 in. Type A burner having an air nozzle 0.300 in. dia. with the gas restrictor adjusted to give a stoichiometric ratio at an air temperature of $500^{\circ}C$.

Consider equation (11) for a type A burner derived in Appendix III

$$\left[\frac{(R+\sigma)(R+1)}{A_4^2} + \frac{S_{45}(R+\sigma)(R+1)}{2A_5^2} - \frac{(R+\sigma)(R+1)}{A_5 A_6} \right] T_4 - \frac{R^2}{A_1 A_4} T_1$$

$$+ \left[\frac{(R+\sigma)\ P}{A_6^2} + \frac{S_{67}(R+\sigma)\ P}{2A_7^2} \right] T_6 = - \frac{\sigma\ T_o}{K_2^2 A_2^2} \quad \text{..........} \ (1)$$

Substitute the appropriate values

$$\sigma = 0.475$$

$$S_{45} = S_{67} = 0.3$$

$$A_1 = \frac{\pi}{4}(0.300)^2 \ in^2$$

$$A_4 = A_5 = \frac{\pi}{4}(0.610)^2 \ in^2$$

$$A_6 = A_7 = \frac{\pi}{4} \ in^2$$

$A_2 = 0.02414$ (As determined in Appendix III)

$$K_2 = 0.8484$$

$$T_1 = T_4 = T_o = 520^{\circ}R$$

As a first approximation $P = 4.52$

$$T_6 = 4100^oR$$

Equation (1) becomes $R^2 = 1.5915R - 29.5158 = 0$

giving $R = 6.287$

Combustion of 1 ft^3 of gas (Net CV 403 Btu/ft^3 ; Stoichiometric Air/Gas Ratio 3.8:1) with 6.287 ft^3 of air produces 4.5 ft^3 of stoichiometric products and 2.487 ft^3 of air.

It can be shown that at a temperature of 2705°F : -

4.5 ft^3 stoichiometric products contains 268.2 Btu

2.487 ft^3 of air contains 134.9 Btu

$\therefore$ Combustion products when $R = 6.287$ contain 403.1 Btu

$$\therefore \quad T_6 = 2705^oF = 3165^oR$$

$$P = 4.5 + 2.487 = 6.987$$

Substituting the new values for T_6 and P in equation (1)

gives:- $R^2 - 1.8330R - 29.6305 = 0$

giving $R = 6.436$

By similar argument it can be shown that when $R = 6.436$

$$T_6 = 2660^oF = 3120^oR$$

$$\therefore \quad P = 7.136$$

Substituting these values in equation (1) for a third approximation

$$R^2 - 1.8432R - 29.6353 = 0$$

giving $R = 6.4429$.

Therefore the air/gas ratio with air at room temperature (15°C) is 6.44:1

APPENDIX V

Calculation of Air Nozzle Variation required to maintain a Constant Air/Gas Ratio with varying Preheat Temperature.

The burners considered are 1 in. Type A, 1 in. Type A with undersized air nozzle, and a 1 in. Type C.

Considering first the general equation for a parallel tunnel/ parallel mixture tube burner derived in Appendix III.

$$T_1 \left\{ \left[\frac{(R+\sigma)(R+1)}{A_4^2} + \frac{S_{45}(R+\sigma)(R+1)}{2\,A_5^2} - \frac{(R+\sigma)(R+1)}{A_5 A_6} \right] 0.79 - \frac{R^2}{A_1 A_4} + \left[\frac{(R+\sigma)\,P}{A_6^2} + \frac{S_{67}(R+\sigma)\,P}{2\,A_7^2} \right] 0.331 \right\} + \left\{ \left[\frac{(R+\sigma)(R+1)}{A_4^2} + \frac{S_{45}(R+\sigma)(R+1)}{2\,A_5^2} - \frac{(R+\sigma)(R+1)}{A_5 A_6} \right] 110 + \left[\frac{(R+\sigma)P}{A_6^2} + \frac{S_{67}(R+\sigma)\,P}{2\,A_7^2} \right] 3932.7 \right\} = \frac{-\sigma\,T_o}{K_2^2 A_2^2} \qquad \text{.......... (1)}$$

Substituting the appropriate values for an optimized type A burner

$T_o = 520^oR$

$R = 3.8$

$P = 4.52$

$\sigma = 0.475$

$S_{45} = S_{67} = 0.3$

$A_4 = A_5 = 0.29224$ in²

$A_6 = A_7 = 0.78540$ in²

$A_2 = \infty$

Equation (1) becomes

$$159.624\ T_1 - \frac{T_1}{A_1}\ 49.411 + 162237.51 = 0 \qquad \text{...........} \ (2)$$

$$\therefore \quad A_1 = \frac{T_1 \quad 49.411}{159.624\ T_1 + 162237.51} \qquad \text{...........} \ (3)$$

A non-optimized type A burner having an undersized air nozzle requires the gas restrictor to be partly closed in order to produce stoichiometric combustion when supplied with air at room temperature.

Consider therefore a burner having an air nozzle 0.300 in. diameter. The calculations in Appendix III showed that with air at 520°R the gas restrictor area A_2 must be 0.05386 sq.in. in order to obtain an air/gas ratio of 38:1 (assuming K_2 to have a value of 0.8484)

Inserting the appropriate values in equation (1)

$$A_2 = 0.05386\ in^2$$

$$K_2 = 0.8484$$

Equation (1) becomes:-

$$159.624\ T_1 - \frac{T_1}{A_1}\ 49.411 + 280556.55 = 0 \qquad \text{...........} \ (4)$$

$$\therefore \quad A_1 = \frac{T_1 \quad 49.411}{159.624\ T_1 + 280556.55} \qquad \text{...........} \ (5)$$

Consider now the general equation (13) for a converging tunnel/parallel injector burner, derived in Appendix III.

$$T_1 \left\{ - \left[\frac{(R+\sigma)P}{2\,A_6^2} + \frac{(1+S_7)(R+\sigma)P}{2\,A_7^2} \right] 0.331 + \frac{R^2}{A_1 A_4} + \left[\frac{(R+\sigma)(R+1)}{A_5 A_6} - \frac{S_{45}(R+\sigma)(R+1)}{2\,A_4^2} \right.\right.$$

$$\left.\left. - \frac{(R+\sigma)(R+1)}{A_4^2} \right] 0.79 \right\} + \left\{ - \left[\frac{(R+\sigma)P}{2A_6^2} + \frac{(1+S_7)(R+\sigma)P}{2\,A_7^2} \right] 3932.7 \right.$$

$$\left. + \left[\frac{(R+\sigma)(R+1)}{A_5 A_6} - \frac{S_{45}(R+\sigma)(R+1)}{2\,A_4^2} - \frac{(R+\sigma)(R+1)}{A_4^2} \right] 110 \right\} = \frac{\sigma\, T_o}{K_2^2 A_2^2} \qquad \text{........(6)}$$

Substituting the appropriate values for a 1 in. Type C burner

$$T_o = 520^oR$$

$$S_{45} = 0.3$$

$$S_7 = 0.12$$

$$R = 3.8$$

$$\sigma = 0.475$$

$$P = 4.52$$

$$A_4 = A_5 = 0.40152 \ in^2$$

$$A_7 = 0.78540 \ in^2$$

$$A_6 = 2.92553 \ in^2$$

$$A_2 = \infty$$

Equation (6) becomes:

$$- 108.0134\ T_1 + 35.9633 \frac{T_1}{A_1} - 87606.57035 = 0 \qquad \text{.......... (7)}$$

$$\therefore\ A_1 = \frac{35.9633\ T_1}{108.0131\ T_1 + 87606.57} \qquad \text{.......... (8)}$$

Substituting for T_1 in equations (3) (5) a d (8) we get the following results:

Air Preheat Temperature		Air nozzle Area A_1 sq. in.		
T_1 °C	T_1 °R	1 in. Type A	1 in. Type A with 0.300 in nozzle	1 in. Type C.
15.6	520	0.10476	0.07067	0.13007
93	660	0.1219	0.08450	0.14938
204	860	0.1419	0.10170	0.17135
316	1060	0.1580	0.11645	0.18862
427	1260	0.1713	0.12925	0.20256
538	1460	0.1825	0.14046	0.21404

These results are presented graphically in Fig. 5, Section 2 of the paper

APPENDIX VI

The effect on air/gas ratio of small errors in air nozzle area

In order to assess the required accuracy of a variable area air nozzle, the effect on air/gas ratio of small deviations in air nozzle area from that required to produce a stoichiometric ratio has been calculated. The calculations being carried out on a 1 in. Type A burner, having air nozzles 1, 2 and 5% oversize and undersized in area.

Consider Equation (11) from Appendix III

$$\left[\frac{(R+\sigma)(R+1)}{A_4^2} + \frac{S_{45}(R+\sigma)(R+1)}{2\,A_5^2} - \frac{(R+\sigma)(R+1)}{A_5 A_6}\right] T_4 - \frac{R^2}{A_1 A_4} T_1$$

$$+\left[\frac{(R+\sigma)\,P}{A_6^2} + \frac{S_{67}(R+\sigma)\,P}{2\,A_7^2}\right] T_6 = -\frac{\sigma\,T_o}{K_2^2 A_2^2} \qquad \text{.......... (1)}$$

Substitute the appropriate values:-

$$\sigma = 0.475$$

$$S_{45} = S_{67} = 0.3$$

$$A_2 = \infty$$

$$A_4 = A_5 = 0.29225 \text{ in}^2$$

$$A_6 = A_7 = 0.78540 \text{ in}^2$$

$$T_1 = T_4 = T_o = 520^oR$$

Six values of $A_1 = 0.10521 \pm 1\% \pm 2\% \pm 5\%$.

The method of calculation was one of successive approximations as follows:-

Estimate a likely value of R

Determine T_6 from published data[6]

Determine I_f (heat content ft^3 of combustion products at temp. T_6) from published data.[6]

Calculate P **from the equation**

$$I_f = \frac{C_n}{P}$$

where C_n is the net calorific value of the gas.

Substitute the above values in equation (1) and check that the equation is satisfied and repeating with improved values of R until R is accurate to two decimal places.

This procedure gave the following results.

% Error in A_1	R	P	T_6 (^oR)	% Error in R
+ 1	3.85	4.51	4090	+ 1.32
- 1	3.75	4.41	4105	-1.32
+ 2	3.895	4.518	4080	+2.50
- 2	3.70	4.488	4105	-2.63
+ 5	4.09	4.675	4010	+7.63
- 5	3.55	4.488	4095	-6.58

These results are plotted in Fig. VI - 1 from which it can be seen that in order to maintain R within ± 5% of the stoichiometric value, A_1 must be accurate to ± 3.8%.

VARIATION OF ERROR IN RATIO (R) WITH ERROR IN AIR NOZZLE AREA (A_1)

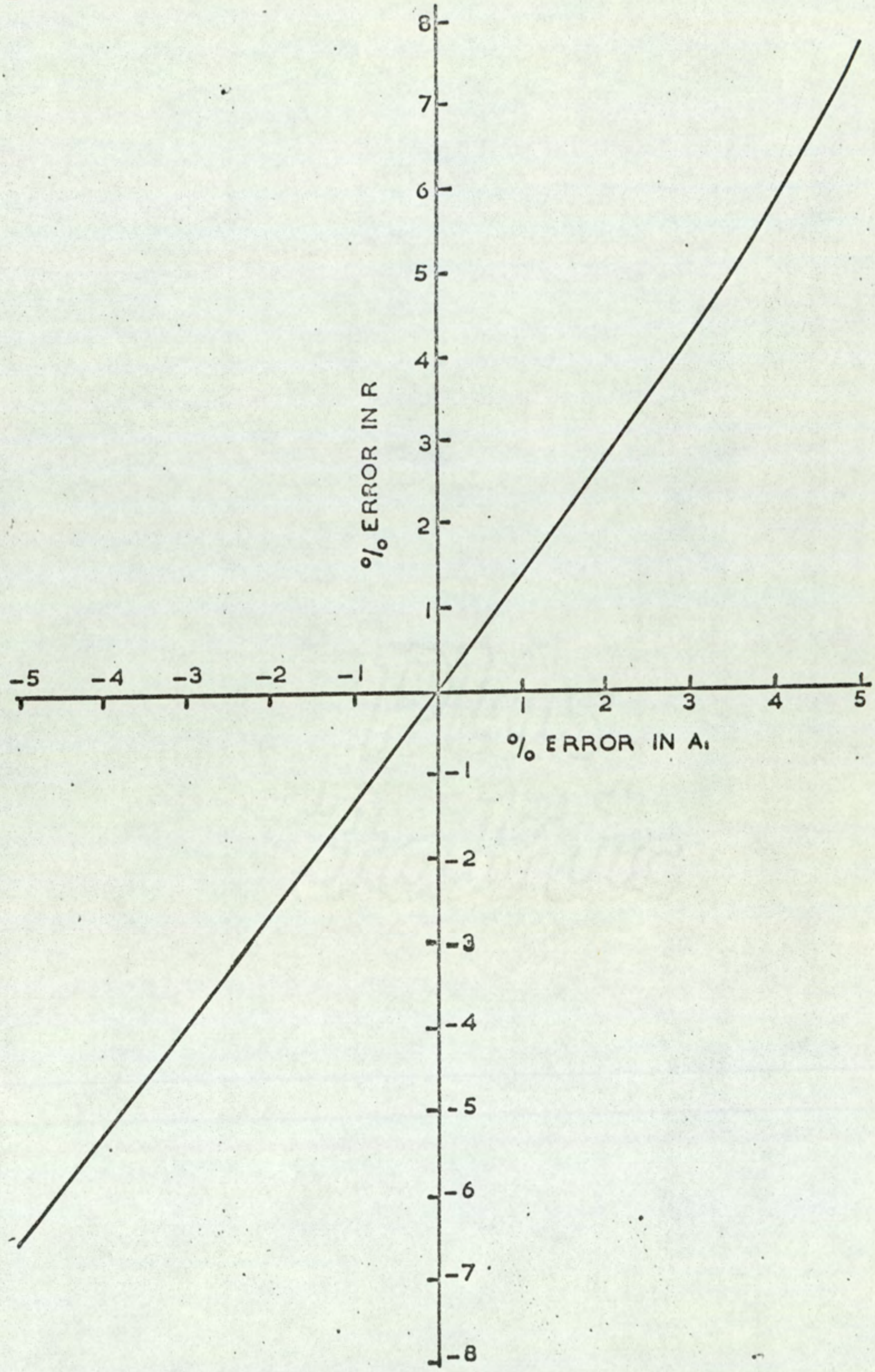


FIG. VI-1

APPENDIX VII

Maintenance of Constant Air/Gas Ratio with varying preheat temperature by Gas Pressure Adjustment

Consider the operation of an air blast tunnel burner with the air nozzle optimized for preheated air. If the gas is supplied at atmospheric pressure the air/gas ratio will increase as the air preheat temperature is reduced. However, a possible technique for maintaining a constant air/gas ratio, is to progressively increase the gas supply pressure above atmospheric as the air temperature falls, in order to compensate for the reduced injection effect.

In order to determine the actual gas pressure required to achieve a stoichiometric ratio, and how this varies with throughput and air temperature, the mathematics developed in Appendix III has been developed further.

Referring to Appendix III, consider equation 8 (page III - 4) for a parallel tunnel , parallel **injector burner.**

The left hand side of this equation is $P_2 - P_7$

where P_2 is the gas pressure downstream of the gas restrictor

and P_7 is the atmospheric pressure

Now the flow through the gas restrictor is

$$Q_g = K_2 A_2 \sqrt{\frac{(P_3 - P_2) g}{\rho_g}}$$

where P_3 is the gas pressure upstream of the gas restrictor.

$$\therefore \quad P_3 - P_2 = \frac{Q_g^2 \rho_2}{K_2^2 A_2^2 \, g}$$

$$\text{or} \quad P_2 = P_3 - \frac{Q_g^2 \rho_g}{K_2^2 A_2^2 \, g}$$

Substituting this in the left hand side of equation 8 gives:-

$$P_3 - P_7 - \frac{Q_g^2 \rho_g}{K_2^2 A_2^2 \, g}$$

$$\text{or} \quad \psi - \frac{Q_g^2 \rho_g}{K_2^2 A_2^2 \, g}$$

Where ψ is the gauge pressure upstream of the gas restrictor i.e. the gauge outlet pressure of the zero governor.

Proceeding with the mathematics as in Appendix III we obtain the general flow equation for any parallel tunnel parallel injector burner.

$$\frac{\psi_g T_o}{Q_g^2 \rho_a} - \frac{\sigma T_o}{K_2^2 A_2^2} = T_1 \left\{ 0.79 \, (R + \sigma)(R + 1) \left[\frac{1}{A_4^2} + \frac{S_{45}}{2A_5^2} - \frac{1}{A_5 A_6} \right] - \frac{R^2}{A_1 A_4} \right.$$

$$\left. + \, 0.331 \, (R + \sigma) \; P \left[\frac{1}{A_6^2} + \frac{S_{67}}{2A_7^2} \right] \right\} + \left\{ 110 \, (R+\sigma)(R+1) \left[\frac{1}{A_4^2} + \frac{S_{45}}{2A_5^2} - \frac{1}{A_5 A_6} \right] \right.$$

$$\left. + \, 3932.7 \, (R + \sigma) \; P \left[\frac{1}{A_6^2} + \frac{S_{67}}{2A_7^2} \right] \right\}$$

Substitute the appropriate values for a 1 in. Midland Research Station Type A burner with the air nozzle optimized for an air temperature of 500°C.

$$R = 3.8$$

$$P = 4.52$$

$$\sigma = 0.475$$

$$S_{45} = S_{67} = 0.3$$

$$T_o = 520^{o}R$$

$$\rho_a = 0.076 \text{ lbs/ft}^3$$

$$g = 4.17 \times 10^8 \text{ ft/hr}^2$$

$$A_1 = 0.00125 \text{ ft}^2$$

$$A_4 = A_5 = 0.00203 \text{ ft}^2$$

$$A_6 = A_7 = 0.00545 \text{ ft}^2$$

$$A_2 = \infty \quad \text{(burner optimized)}$$

This gives:-

$$\frac{\psi}{Q_g^{2}} = 1.179 \times 10^{-3} - 8.407 \times 10^{-7} T_1$$

Thus if $T_1 = 60^{o}F$ and $Q_g = 200$ ft^3/h

then $\psi = 5.70$ in.w.g.

In order to achieve the corresponding air flow of 760 ft^3/h through the burner the differential pressure across the air nozzle must be 7.5 in.w.g., thus the air supply pressure must be 13.2 in.w.g. (since ψ is the gauge pressure at the exit of the air nozzle).

$\therefore$ At room temperature the gas pressure must be 43.1% of the air supply pressure.

A similar mathematical argument applied to a converging tunnel, parallel mixture tube burner, leads to the following general equation:-

$$\frac{\Psi g T_o}{Q_3^2 \rho_a} - \frac{\sigma T_o}{K_2^2 A_2^2 \rho_a} = T_1 \left\{ 0.331 (R + \sigma) P \left[\frac{1}{2A_6^2} + \frac{1 + S_7}{2A_7^2} \right] - \frac{R^2}{A_1 A_4} \right.$$

$$\left. + 0.79 (R + \sigma)(R + 1) \left[\frac{2 + S_{45}}{2A_4^2} - \frac{1}{A_5 A_6} \right] \right\} + 3932.7 (R + \sigma) P \left[\frac{1}{2A_6^2} + \frac{1+S_7}{2A_7^2} \right.$$

$$+ 110 (R + \sigma)(R + 1) \left[\frac{2 + S_{45}}{2A_4^2} - \frac{1}{A_5 A_6} \right]$$

Substituting the following values appropriate to a 1 in. Midlands Research Station type C burner with the air nozzle optimised for an air temperature of 500°C gives:-

$$\frac{\Psi}{Q^2 g} = 6.367 \times 10^{-4} - 4.537 \times 10^{-7} T_1$$

$R = 3.8$

$P = 4.52$

$\sigma = 0.475$

$S_{45} = 0.3$

$S_7 = 0.12$

$T_o = 520^{o}R$

$\rho_a = 0.076 \text{ lbs/ft}^3$

$g = 4.17 \times 10^8 \text{ ft/hr}^2$

$A_1 = 0.00146 \text{ ft}^2$

$A_4 = A_5 = 0.00279 \text{ ft}^2$

$A_7 = 0.00545 \text{ ft}^2$

$A_6 = 0.02032 \text{ ft}^2$

$A_2 = \infty$

Thus if $T = 60^{o}F$ and $Qg = 200\ ft^3/h$

then $\psi = 3.08$ in.w.g.

In order to achieve the corresponding air flow of $760\ ft^3/h$ through the burner the differential pressure across the air nozzle must be 5.5 in.w.g., thus the air supply pressure must be 8.58 in.w.g.

$\therefore$ At room temperature the gas pressure must be 35.9% of the air supply pressure.

APPENDIX VIII

Calculation of Error in Air/gas Ratio due to Forces acting on the Plug of the Mk.I. Compensating Injector.

In designing the Mk.I. injector it was assumed that the only significant cause of error in proportioning accuracy would be due to friction in the force balance mechanism. With this proviso it was assumed that movement of the plug would be restrained only by the forces acting on the two main diaphragms and that a force balance would be achieved when the differential pressures developed across these diaphragms were equal. This assumption is only valid if the force on the tapered portion of the plug, due to the accelerating air flow through the nozzle and suction developed by the injector, is negligible compared with the forces acting on the diaphragms. It was therefore thought desirable to calculate the forces acting on the tapered portion of the plug.

Consider therefore the forces acting on the plug, in its two extreme positions shown in Figs. *A* and *B* below.

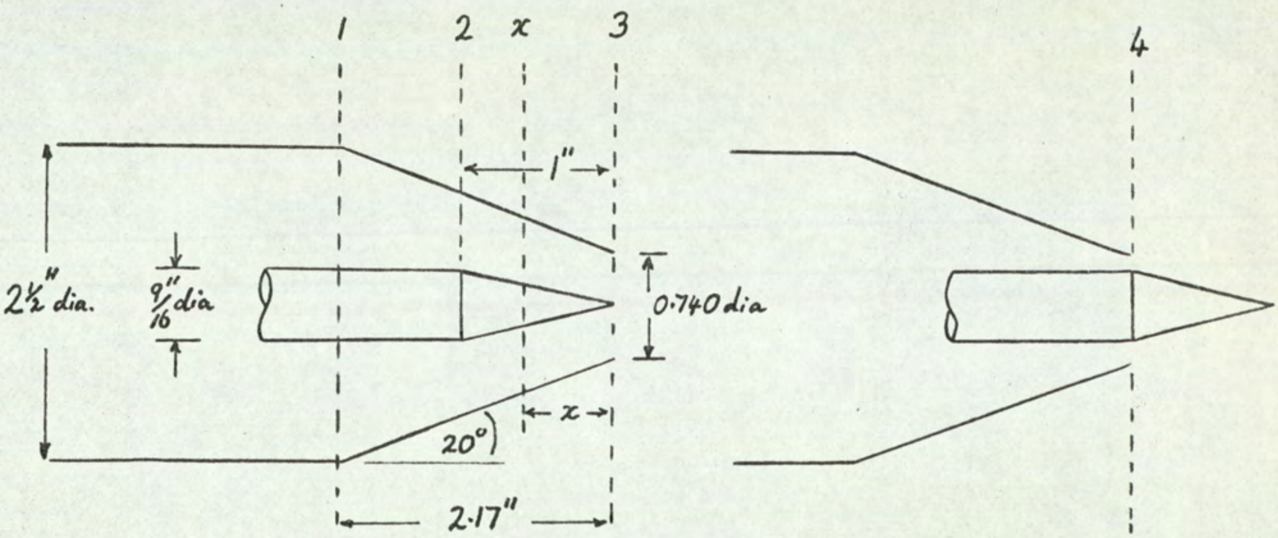


FIG. A FIG. B

In Fig. A there will be a net force acting to the left due to the acceleration of the air flow through the nozzle. This will tend to make the nozzle area greater than it should be and thus the air/gas mixture will tend to be gas weak.

In Fig. B there will be no resultant force on the plug due to the acceleration of the air stream, however the tapered portion of the plug will be exposed to the suction developed by the injector and this will produce a resultant force acting to the right. The air/gas mixture will thus tend to be gas rich.

The errors in air/gas ratio produced when the plug is in these two positions are obviously the extremes and with the plug in any other position the error will be less. It is therefore only necessary to calculate the error at these two positions.

Also the error in air/gas ratio is independent of flow and thus it is only necessary to calculate the error for one throughput.

Consider the first situation in Fig. A.

Let P_1, P_2, P_x, P_3, be the gauge pressure (lbs/in^2 at points 1, 2, x and 3.

a_1, a_2, a_x, a_3, be the flow area(in^2) at 1, 2, x and 3.

v_1, v_2, v_x, v_3, be the air velocity (in/sec) at 1, 2, x and 3.

Q = Volume flow rate ($in^3/sec.$)

H = Total Head (gauge) in the system.

g = Acceleration due to gravity (in/sec^2)

ρ = Density of air (lbs/in^3)

α = $\frac{1}{2}$ cone angle of **plug.**

Now at position x: $H = \dfrac{\rho\, v_x^2}{2g} + P_x$

The static pressure P_x acting on an elementary area of the plug, width δx, radius r_x, will produce a resultant force acting to the left.

$$= P_x \; 2\pi \; r_x \; \delta x \; \sin\alpha$$

Now, from the dimensions of the **plug.**

$$\frac{r_x}{x} = \frac{9}{32} \quad \text{and } \sin\alpha = 0.27074$$

$\therefore$ Force on elementary area $= P_x \, 2\pi . \frac{9}{32} . 0.27074 \; x\delta x$

$$= P_x \; 0.07614 . \; 2\pi . \; x\delta x$$

$$= \left(H - \frac{\rho v_x^2}{2\,g}\right) 0.07614 . \; 2\pi . \; x\delta x$$

Geometry of system gives $\quad a_x = \pi\,(0.37 + x \tan 20^{\circ})^2 - \pi\,(\frac{9}{32}\,x)^2$

Now $v_x = \dfrac{Q}{a_x} \quad \therefore \; v_x^2 = \dfrac{Q^2}{\pi^2 \left[(0.37 + x \tan 20^{\circ})^2 - (\frac{9}{32}\,x)^2 \right]^2}$

Now at positions 3: $\quad H = P_3 + \dfrac{\rho v_3^2}{2\,g}$

$$= P_3 + \frac{\rho Q^2}{2g\,(\pi . 0.37^2)^2}$$

$\therefore$ Force on elementary area

$$= \left(P_3 + \frac{\rho Q^2}{2g\,(\pi . 0.37^2)^2} - \frac{\rho Q^2}{2g\,\pi^2 \left[(0.37 + x\tan 20^o)^2 - (\frac{9}{32}\,x)^2 \right]^2} \right) 0.07614 . 2\pi . x\delta x$$

Substituting for tan $20^o = 0.36397$, factorizing, multiplying through by 2Π and integrating gives the total force as:

$$0.07614 \; \frac{\rho Q^2}{g} \int_0^1 \left[\frac{2\pi P_3 g}{Q^2 \rho} + \frac{1}{\pi\, 0.37^4} - \frac{1}{\pi \left[(0.37 + 0.36397\,x)^2 - (\frac{9}{32}\,x)^2 \right]^2} \right] xdx$$

$$= 0.07614 \; \frac{\rho Q^2}{g} \int_0^1 \left[\frac{2\pi P_3 g}{Q^2 \rho} + 16.986 - \frac{1}{\pi\,(0.1369 + 0.26934\,x - 0.14878x^2)^2} \right] xdx$$

$$= 0.07614 \; \frac{\rho Q^2}{g} \left\{ \left[\frac{\pi P_3 g}{Q^2 \rho}\, x^2 \right]_0^1 + \left[8.493\, x^2 \right]_0^1 - \frac{I}{\pi} \right\}$$

$$\text{Where} \quad I = \int_0^1 \frac{xdx}{(0.1369 + 0.26934x - 0.14878x^2)^2}$$

$$\text{Now} \int \frac{xdx}{Z^2} = - \frac{bx + 2a}{q\,Z} - \frac{b}{q} \int \frac{dx}{Z}$$

$$\text{and} \int \frac{dx}{Z} = \frac{1}{\sqrt{-q}} \log \frac{2cx + b - \sqrt{-q}}{2cx + b + \sqrt{-q}}$$

where $Z = a + bx + cx^2$ and $q = 4\,ac - b^2$

$$\therefore \; I = \left[- \frac{bx + 2a}{qZ} - \frac{b}{q} \, \frac{1}{\sqrt{-q}} \log \frac{2cx + b - \sqrt{-q}}{2cx + b + \sqrt{-q}} \right]_0^1$$

$$= - \frac{b + 2a}{q\,(a+b+c)} - \frac{b}{q} \, \frac{1}{\sqrt{-q}} \log \frac{2c + b - \sqrt{-q}}{2c + b + \sqrt{-q}} + \frac{2a}{qa} + \frac{b}{q} \, \frac{1}{\sqrt{-q}} \log \frac{b - \sqrt{-q}}{b + \sqrt{-q}}$$

Substitute appropriate values a = 0.1369

b = 0.26934

c = 0.14878

q = - 0.15401

$$\therefore I = 6.3540 - 12.98616 + 4.45632 \left[\log 0.18185 - \log(-0.18601) \right]$$

$$= 6.3540 - 12.98616 + 4.45632 \left[\frac{1}{10} \log 2.48443 - \frac{1}{10} \log 2.30258 \right]$$

$$= 6.3540 - 12.98616 + 0.44563 \log 1.07897$$

$$= 6.3540 - 12.98616 + 0.03396$$

$$= - 6.5982$$

$$\therefore \text{Force} = 0.07614 \ \frac{\rho Q^2}{g} \left[\frac{\pi P_3^2 g}{Q \rho} + 8.493 + \frac{6.5982}{\Pi} \right] \text{lbs.wt.}$$

The burner, which is designed to pass 500 s.ft^3/h of gas with air at room temperature and 1 lb/in^2 pressure, will pass 560 s.ft^3/h of gas if the air temperature is 1000°F (Ref. (3) P. 34).

Thus the air flow at 1000°F will be 2240 s.ft^3/h

$$\therefore Q = \frac{2240 \times 1460}{520} \ \text{ft}^3/\text{h}$$

$$= 3018.5 \ \text{in}^3/\text{sec}$$

$$\text{Also } \rho = \frac{0.076 \times 520}{1728 \times 1460}$$

$$= 1.5663 \times 10^{-5} \ \text{lb/in}^3$$

$$\therefore \frac{\rho Q^2}{g} = 0.36983 \text{ lb.in}$$

Now P_3 is designed to be - 2 in.w.g. (= - 0.0722 lb/in^2.)

at maximum flow.

$$\therefore \text{Total force} = 0.07614 \left[\pi P_3 + 8.493 \frac{\rho Q^2}{g} + \frac{6.5982 \rho Q^2}{\pi g} \right] \quad \ldots\ldots\ldots\ldots (1)$$

$$= 0.07614 \left[\pi (-0.0722) + 8.493.0.36983 + \frac{6.5982.0.36983}{\pi} \right]$$

$$= 0.2810 \text{ lbs.wt.}$$

Now the effective area of main diaphragm is 83.45 in^2, thus the application of a d.p. of 2 in.w.g. produces a force of 6.03 lbs.wt.

$\therefore$ In force balance, due to the acceleration of the air through the nozzle, the suction developed by the injector will be $\left(2 \times \frac{5.749}{6.03} \right)$ in.w.g. instead of 2 in.w.g.

$\therefore$ Calculate a more accurate value of total force by substituting the new value of $P_3 = - 0.0722 \times \frac{5.749}{6.03}$ lb/in^2. in equation (1) above.

$$\text{Total Force} = 0.07614 \left[\pi \left(- 0.0722 \times \frac{5.749}{6.03} \right) + \frac{8.493 \rho Q^2}{g} + \frac{6.5982 \rho Q^2}{\pi g} \right.$$

$$= 0.28183 \text{ lbs.wt.}$$

Thus the improved value of $P_3 = - 2 \times \frac{5.748}{6.03}$ in.w.g.

Thus the gas flow will be $560 \times \dfrac{\sqrt{2 \times \frac{5.748}{6.03}}}{\sqrt{2}} = 546.76$ s.ft^3/h.

instead of 560 s.ft^3/h and

the air/gas ratio = 4.0969:1 instead of 4.00:1

Consider now the situation in Figure B.

Total Force = Suction × cross sectional area of plug at position 4.

$$= 0.0722 \times \frac{\pi}{4}\left(\frac{9}{16}\right)^2 = 0.0179 \text{ lbs.wt.}$$

∴ The suction will be $\left(2 \times \frac{6.0479}{6.03}\right)$ in.w.g. instead of 2 in.w.g.

Substitute this value in the above equation to give a more accurate value for total force.

$$\text{Total Force} = 0.0722 \times \frac{6.0479}{6.03} \times \frac{\pi}{4}\left(\frac{9}{16}\right)^2 = 0.01796 \text{ lbs.wt.}$$

The suction will therefore be $\left(\frac{2 \times 6.048}{6.03}\right)$ in.w.g. instead of 2 in.w.g.

The gas flow will therefore be $500 \dfrac{\sqrt{2 \times \frac{6.048}{6.03}}}{\sqrt{2}} = 500.6$ s.ft^3/h.

instead of 500 s.ft^3/h. and

the air/gas ratio will be 3.99:1 instead of 4.00:1

Therefore, the assumption that the forces acting on the tapered portion of the plug could be neglected is valid, since the deviation of air/gas ratio from a required value of 4.00:1 due to this effect will be within the limits 3.99:1 and 4.10:1.

APPENDIX IX

"Linear" flow control valve.

In a conventional air blast burner installation the burner throughput is controlled by a valve in the air supply and, if the valve is correctly sized, there is little pressure loss across the valve at full flow, most of the available air pressure being dissipated across the burner air nozzle. Only at low flows is the pressure drop across the air valve large compared with that across the air nozzle. Thus only at very low flows is air flow linearly proportional to the port area of the air valve. At higher throughputs the air nozzle has an increasing influence on the flow rate and near full flow the change in port area required to achieve say a 10 ft^3/hr. change in flow is very much greater than that required to achieve a 10 ft^3/hr. change at low flow. If the valve port is suitably shaped then it is possible to arrange for flow to be linearly proportional to the angular setting of the valve. However, since the port shape would need to be adjusted for each size of air nozzle, this is not done in practice, and in general flow is not linearly proportional to valve setting.

In an installation having high/low throughput control this non linearity will be unimportant. However, in an installation fitted with proportional control the loop gain will vary with throughput and thus the controller can only be adjusted to give optimum performance at one throughput.

With this problem in mind, it was decided to arrange for the throughput of the recuperator heater to be linearly proportional to valve setting by using a valve having a linear characteristic (flow proportional to valve setting at constant differential pressure) and maintaining the differential pressure across the valve constant. It was intended to achieve this by installing an appliance governor, weight loaded to give 2.5 in.w.g. outlet pressure, upstream of the valve and pressure loaded from a pressure tapping immediately downstream of the valve.

The valve used was a 2in. Selas Standard Air Valve Type 225/2 and the governor was a weight loaded Jeavons 2in. Type J47 with a spring loaded top. The standard top of the weight loaded J47 is not suitable for pressure loading as it is only held down by two bolts and is not provided with a pressure tapping.

In order to compare the performance of the valve/governor system with the conventional arrangement, a quadrant cock and air rotameter were arranged downstream of the valve as shown in Fig. IX-1, the quadrant cock simulating the air nozzle of a burner.

The quadrant cock was adjusted to give a pressure drop of about 24 in.w.g. at an air flow of 2040 s.ft.^{3}/hr, the air flow rate being controlled with the air valve and measured with the rotameter. This particular flow and pressure was the designed maximum for the recuperator heater. After setting the cock, the air valve was opened fully and the port area adjusted to give 2040 s.ft.^{3}/hr.

APPARATUS FOR MEASURING THE PERFORMANCE OF THE SELAS AIR VALVE.

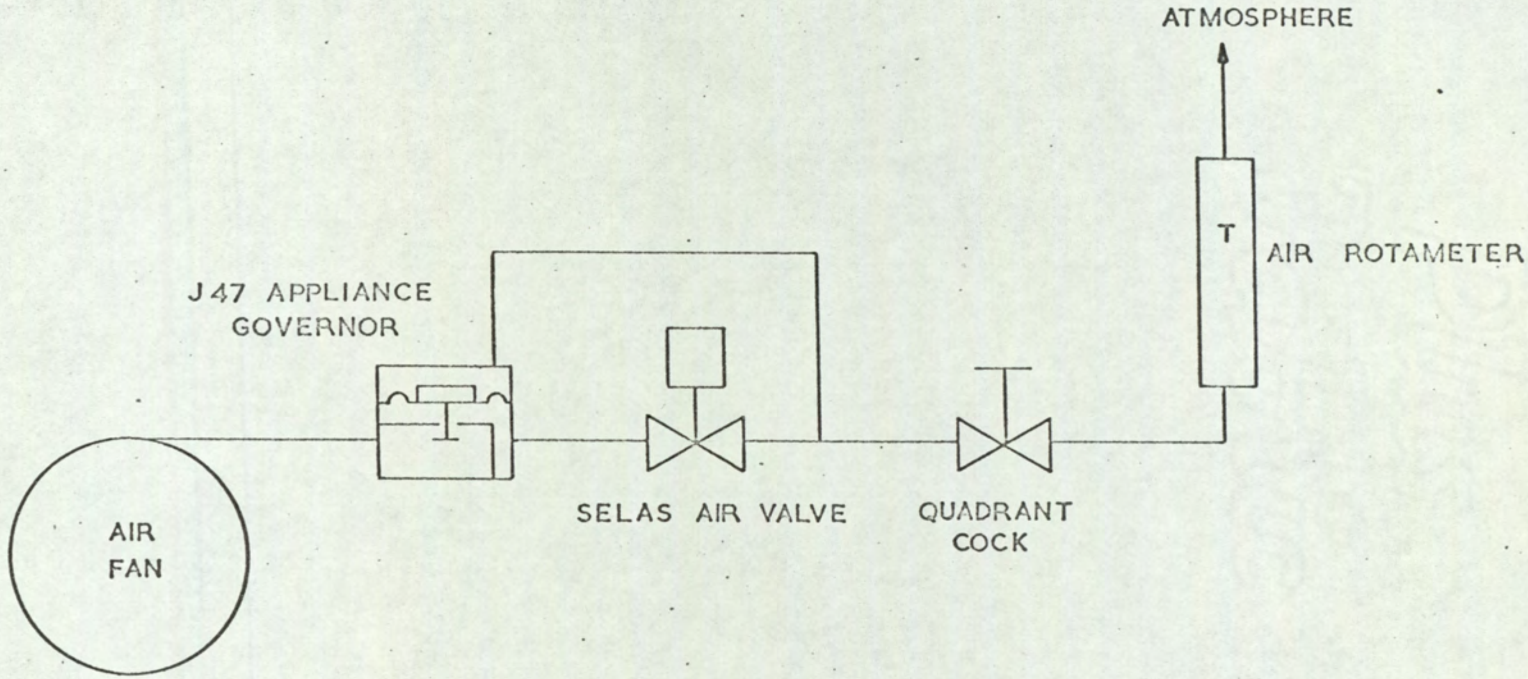


FIG. IX-1

The flow rate was then measured for a number of valve settings, the valve setting being indicated by a scale and pointer on the valve shaft.

In order to compare this system with the conventional one, the governor was jammed open by loading the top with air pressure from a tapping upstream of the governor, and the valve port was readjusted slightly to give a flow of 2040 s.ft.^{3}/hr. when fully open. The value of flow rate was again obtained for a number of valve settings. The results of these experiments are shown graphically in Fig. IX-2.

It can be seen that the curve of flow versus valve setting is much more linear with the combined valve and governor arrangement, than with the conventional arrangement. The deviation from linearity that remains cannot be accounted for by variations in the governor differential pressure and is therefore probably a characteristic of the valve.

It is felt that the results obtained justified the use of the valve and governor system and it was decided not to pursue the matter further in order to obtain perfect linearity.

VARIATION OF FLOW WITH VALVE SETTING.

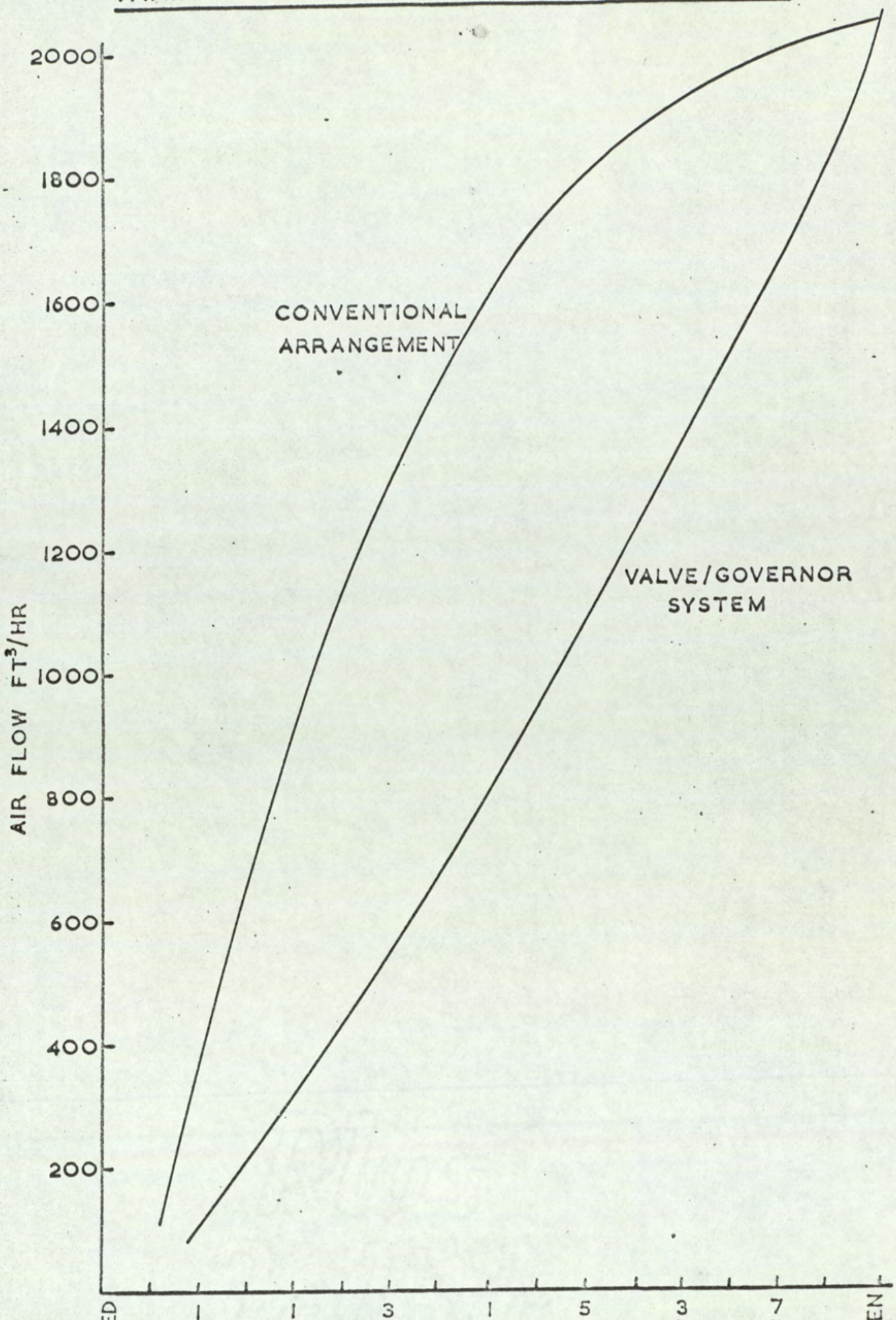


FIG. IX-2

APPENDIX X.

Heat Balance Calculations on the Recuperator

In order to assess the performance of the recuperator, heat balances were calculated based on the results of experiments in Section 5 of the paper. Values of heat contents of air and town gas combustion products were found from published data [6] and wall loss calculations were based on data included in the Industrial Gas Handbook [7].

Calculations based on the results quoted in Table 5 - 1.

Air flow 250 ft^3/hr.

Gas input 196 ft^3/hr. (Net CV. 403 Btu/ft^3.)

∴ Net heat input = 79,000 Btu/hr.

Stoichiometric combustion products reduced in temperature to 970°C by diluent air addition prior to entering recuperator.

Now:- Enthalpy of air at 970°C = 33.4 Btu/ft^3.

Enthalpy of stoichiometric combustion products at 970°C = 36.6 Btu/ft^3.

∴ Enthalpy of diluted combustion products at 970°C = 35 Btu/ft^3.

∴ Assuming negligible heat loss prior to dilution, flow rate of diluted combustion products = 2260 s.ft^3/hr.

Enthalpy of diluted combustion product leaving the recuperator at 576.5°C = 19.65 Btu/s.ft^3.

∴ Total heat leaving flue = 44,400 Btu/hr.

∴ Heat given up by combustion products = 34,600 Btu/hr.

Enthalpy of outlet air at 258°C = 7.9 Btu/ft^3.

$\therefore$ Heat gained by air = 1976 Btu/hr.

Wall Losses

The wall losses have been calculated from published data[7].

The basis for these calculations are that heat lost by natural convection can be expressed as:-

$H_c = 0.29\theta^{1.25}L^{-0.25}$ where the air flow is laminar and, where the air flow is turbulent as:-

$H_c = 0.22\theta^{1.33}$ for horizontal plane surfaces facing upwards

$H_c = 0.19\theta^{1.33}$ for cylindrical and vertical plane surfaces.

where H_c = heat lost by natural convection Btu/ft^{2}hr.

t_s = temperature of surface $^\circ$F.

t_a = ambient temperature $^\circ$F.

$\theta = (t_s - t_a)$ $^\circ$F.

L = characteristic dimension (ft) of surface representing the size, shape and orientation and determined as follows:-

Type of Surface	Characteristic Dimension (Ft.)
Rectangular vertical plane surface	Height
Rectangular horizontal plane surface	Mean of two sides
Cylindrical surface, diameter less than 2in.	Diameter
Vertical cylindrical surface, diameter more than 2in.	Height
Horizontal cylindrical surface, diameter more than 2in.	Mean length and diameter

The air flow is assumed to be laminar if the product of Grashof and Prandtl Numbers is less than 10^9

The Heat lost by radiation is based on the equation

$$H = \theta\epsilon\,(T_1^4 - T_2^4)$$

where H = heat loss by radiation (Btu/ft^{2}hr.)

ϵ = emmisivity of hot surface

θ = Stefan-Boltzmann constant (0.173×10^{-8} Btu/ft^{2}hr. $^\circ R^4$).

T_1 = Temperature of hot surface $^\circ$R

T_2 = Temperature of surrounding $^\circ$R

For the purpose of the calculations the surroundings are assumed to have an emmisivity of unity and be at a temperature of 60°F.

<u>Recuperator casework</u>

Mean casework temperature 169°C.

Casework area = 10.4 sq. ft.

L^3 = 17.6 ft^3.

∴ Air flow over the casework is turbulent.

∴ Convective loss = 310 Btu/ft^2 hr.

Total convective loss = 3220 Btu/hr.

Radiant loss with (ϵ = 1) = 570 Btu/ft^2 hr.

Emmisivity (ϵ) = 0.9

Giving total radiant loss = 5340 Btu/hr.

<u>Outlet Duct</u>

Duct temperature = 180°C

Duct area = 0.67 ft^2

L^3 = 0.0506 ft^3.

$\therefore$ Air flow over duct is laminar

Convective heat loss = 470 Btu/ft.^{2} hr.

Total convective heat loss = 315 Btu/hr.

Radiant heat loss with ($\epsilon = 1$) = 610 Btu/ft^2.

$\epsilon = 0.9$

Giving total radiant heat loss = 368 Btu/hr.

Air Pipe

Air pipe temperature = 120°C.

Pipe area = 2.25 ft^2.

$L^3 = 2.57$ ft^3.

$\therefore$ Air flow over pipe is laminar.

$\therefore$ Convective heat loss = 160 Btu/ft.^{2} hr.

Total convective heat loss = 360 Btu/hr.

Radiant heat loss with ($\epsilon = 1$) = 295 Btu/ft.^{2} hr.

$\epsilon = 0.5$

giving total radiant heat loss = 332 Btu/hr.

$\therefore$ Total wall loss = 9,935 Btu/hr.

Calculations based on the results quoted in Table 5 - 2

Air flow = 1750 ft^3/hr.

Gas flow = 196 ft^3/hr.

$\therefore$ Net heat input 79,000 Btu/hr.

Stoichiometric combustion products reduced in temperature to 975°C by diluent air addition prior to entering recuperator. Now:-

Enthalpy of air at 975°C = 33.7 Btu/ft^3.

Enthalpy of stoichiometric combustion products at 975°C = 36.9 Btu/ft^3.

$\therefore$ Enthalpy of diluted combustion products at 975°C = 35.3 Btu/ft^3.

$\therefore$ Assuming negligible heat loss prior to dilution, flow rate of diluted combustion products = 2240 ft^3/hr.

Enthalpy of diluted combustion products leaving the recuperator at 510.6°C = 17.3 Btu/ft^3.

$\therefore$ Total heat leaving flue = 38,750 Btu/hr.

$\therefore$ Heat given up by combustion products = 40,250 Btu/hr.

Enthalpy of outlet air at 385°C = 12.3 Btu/ft^3.

$\therefore$ Heat gained by air = 21,500 Btu/hr.

<u>Wall Losses</u>

<u>Recuperator Casework</u>

Mean casework temperature = 161 °C.

Casework area = 10.4 ft^2.

L^3 = 17.6 ft^3.

$\therefore$ Air flow over casework is turbulent.

$\therefore$ Convective heat loss = 300 $Btu/ft^2.\ hr$.

Total convective loss = 3120 Btu/hr.

$\therefore$ Radiant loss with (ϵ = 1) = 510 $Btu/ft^2.\ hr$.

Emmisivity (ϵ) = 0.9

giving total radiant loss = 4770 Btu/hr.

<u>Outlet Duct</u>

Duct temperature = 190°C

Duct area = 0.68 ft^2.

L^3 = 0 0506 ft^3.

$\therefore$ Air flow over duct is laminar

Convective heat loss = 505 $Btu/ft^2.\ hr$.

Total convective loss = 338 Btu/hr.

Radiant heat loss with ($\epsilon = 1$) = 710 Btu/ft^2. hr.

Emissivity (ϵ) = 0.9

giving total radiant loss = 429 Btu/hr.

Air Pipe

Air pipe temperature = 130°C

Pipe area = 2.25 ft^2.

L^3 = 2.57 ft^3.

$\therefore$ Air flow over pipe is laminar.

$\therefore$ Convective heat loss = 200 Btu/ft^2. hr.

Total convective loss = 450 Btu/hr.

Radiant heat loss with ($\epsilon = 1$) = 320 Btu/ft^2. hr.

Emmisivity (ϵ) = 0.5

giving total radiant loss = 360 Btu/hr.

$\therefore$ Total wall loss = 9467 Btu/hr.

Calculations based on the results quoted in Table 5 - 3

Wall losses

Part	Mean Temperature ($^\circ$C)	Area (ft^2)	Characteristic Dimension (ft.)	Emmisivity	Convective Heat loss (Btu/hr.)	Radiant Heat loss (Btu/hr.)
Panels A	203.7	7.05	2.42	0 .9	2,820	5,060
Panel B	170	1.61	2.42	0.9	484	810
Panel C	135	1.61	2.42	0.9	338	550
Outlet	180	0.695	0.415	0.9	292	375
Lower case	250	0.486	0.166	0.9	340	497
Channelling	230	4.28	0.915	0.9	1,800	3,400
Burner box	97.5	3.89	1.17	0.4	428	374
Box flange	200	0.88	1.0	0.4	128	264
Burner can	273.3	0.69	0.58	0.4	455	418
Burner flange	240	0.27	0.5	0.4	67	113

Heat lost before diluted products entered recuperator = 2247 Btu/hr.

Heat lost over recuperator = 16766 Btu/hr.

Enthalpy of air at 1201°C = 42.2 Btu/ft^3

Enthalpy of stoichiometric combustion products at 1201 $^\circ$C = 46.3 Btu/ft^3

$\therefore$ Enthalpy of diluted combustion products entering recuperator at 1201 $^\circ$C = 44.2 Btu/ft^3.

Gas input to burner = 183 ft^3/hr.

$\therefore$ Net heat input to burner = 73,800 Btu/hr.

Heat entering recuperator = 71,553 Btu/hr.

$\therefore$ Rate of flow of diluted combustion products = 1620 ft^3/hr.

Enthalpy of diluted combustion products leaving recuperator at 631.4°C = 21.75 Btu/ft^3.

$\therefore$ Total heat leaving flue = 35,200 Btu/hr.

Heat given up by combustion products = 36,353 Btu/hr.

Heat gained by air = 3120 Btu/hr.

Re-Calculation (using corrected surface temperatures) of heat balance based on the results quoted in table 5-3

Wall Losses.

art	Measured Mean Temp. (°C)	Corrected Mean Temp. (°C)	Area ft^2.	Characteristic Dimension (ft.)	Emmisivity	Convective Heat loss (Btu/hr)	Radiant Heat loss (Btu/hr)
nels A	203	280	7.05	2.42	0.9	4360	9630
nel B	170	234	1.61	2.42	0.9	795	1520
nel C	135	185	1.61	2.42	0.9	564	1010
tlet	180	250	0.695	0.415	0.9	522	782
wer Case	250	344	0.486	0.166	0.9	510	1090
annelling	230	317	4.28	0.915	0.9	3080	7680
rner box	98	133	3.89	1.17	0.4	779	514
x flange	200	275	0.88	1.0	0.4	485	464
rner can	273	375	0.69	0.58	0.4	746	845
rner flange	240	303	0.27	0.50	0.4	221	194

Heat lost before diluted products entered recuperator = 4248 Btu/hr.

Heat lost over recuperator = 31,543 Btu/hr.

Diluted combustion products entering recuperator at 1201°C contain 44.2 Btu/ft^3.

Net heat input to burner = 73,800 Btu/hr.

Heat entering recuperator= 69,552 Btu/hr.

$\therefore$ Rate of flow of diluted combustion products = 1,575 ft^3/hr.

Enthalpy of diluted combustion products leaving recuperator at 631.4 °C = 21.75 Btu/ft^3.

$\therefore$ Total heat leaving flue = 34,220 Btu/hr.

$\therefore$ Heat given up by combustion products = 35,332 Btu/hr.

Heat gained by air = 3120. Btu/hr.

Calculations based on the results quoted in Table 5-5

Wall Losses.

t	Mean Temp* °C	Area ft^2	Characteristic Dimension (ft.)	Emmisivity	Convective Heat loss (Btu/hr.)	Radiant Heat loss (Btu/hr.)
els A	269.5	7.05	2.42	0.9	4260	9000
el B	185	1.61	2.42	0.9	676	1248
el C	210	1.61	2.42	0.9	595	1015
let	230	0.695	0.415	0.9	424	626
er case	304	0.486	0.166	0.9	495	782
nnelling	280	4.28	0.915	0.9	2650	5550
ner box	140	3.89	1.17	0.4	1089	575
flange	184	0.88	1.0	0.4	299	236
ner can	327	0.69	0.58	0.4	621	608
ner flange	219	0.27	0.5	0.4	140	98

* The true temperature has been taken as the highest indicated temperature since any error with the surface pyrometer would tend to give a low result.

Heat lost before diluted products entered recuperator = 3,666 Btu/hr.

Heat lost over recuperator = 27,321 Btu/hr.

Enthalpy of air at 1150°C = 40.7 Btu/ft^3.

Enthalpy of stoichiometric combustion products at 1150°C = 44.2 Btu/ft^3.

Enthalpy of diluted combustion products entering recuperator at 1150°C = 42.45 Btu/ft^3.

Gas input to burner = 180.2 ft^3/hr.

.˙. Net heat input to burner = 72,600 Btu/hr.

Heat entering recuperator = 68,934 Btu/hr.

.˙. Rate of flow of diluted combustion products = 1622 ft^3/hr.

Enthalpy of diluted combustion products leaving recuperator at 701.1°C = 24.5 Btu/ft^3.

.˙. Total heat leaving the flue = 39,834 Btu/hr.

Heat given up by combustion products = 29,100 Btu/hr.

Enthalpy of ingoing air at 22.5°C = 0.4 Btu/ft^3.

Enthalpy of outgoing air at 403°C = 12.9 Btu/ft^3.

Air flow = 250 ft^3/hr.

∴ Heat gained by air = 3,122 Btu/hr.

<u>Calculations based on the results quoted in Table 5-6</u>

<u>Wall Losses</u>.

Part	Mean Temp.* (°C)	Area (ft^2)	Characteristic Dimension (ft.)	Emmisivity	Convective Heat loss (Btu/hr.)	Radiant Heat loss (Btu/hr.)
Panels A	179	7.05	2.42	0.9	2347	3940
Panel B	132	1.61	2.42	0.9	232	494
Panel C	87	1.61	2.42	0.9	110	232
Outlet	220	0.695	0.415	0.9	368	550
Lower case	218	0.486	0.166	0.9	331	416
Channelling	265	4.28	0.915	0.9	2442	5355
Burner box	145	3.89	1.17	0.4	934	606
Box flange	177	0.88	1.00	0.4	282	218
Burner can	325	0.69	0.58	0.4	625	618
Burner flange	222	0.27	0.50	0.4	143	105

* The true temperature has been taken as the highest indicated temperature since any error with the surface pyrometer would tend to give a low result.

Heat lost before diluted products entered recuperator = 3531 Btu/hr.

Heat lost over recuperator = 16817 Btu/hr.

Enthalpy of air at 1144.5°C = 40.2 Btu/ft^3.

Enthalpy of stoichiometric combustion products at 1144.5°C = 44.1 Btu/ft^3.

∴ Enthalpy of diluted combustion products entering recuperator at 1144.5°C = 42.15 Btu/ft^3.

Gas input to burner = 181 ft^3/hr.

.˙. Net heat input to burner = 72,950 Btu/hr.

Heat entering recuperator = 69,419 Btu/hr.

.˙. Rate of flow of diluted combustion products = 1646 ft^3/hr.

Enthalpy of diluted combustion products leaving recuperator at 574.8°C

= 19.65 Btu/ft^3

.˙. Total heat leaving flue = 32,319 Btu/hr.

Heat given up by combustion products = 37,100 Btu/hr.

Enthalpy of ingoing air at 22.5°C = 0.4 Btu/ft^3

Enthalpy of outgoing air at 395°C = 12.65 Btu/ft^3

Air flow = 1750 ft^3/hr.

Heat gained by air = 21420 Btu/hr.

<u>Calculations based on the results quoted in Table 5 - 9.</u>

<u>Wall Losses</u>

Area of asbestos lagging surface	=	16 ft^2
Characteristic dimension	=	3.17 ft.
Surface temperature	=	155°C
Ambient temperature	=	20°C

∴ Air flow over surface is turbulent

∴ Convective heat loss = 270 Btu/ft^2. hr.

∴ Total convective loss = 4320 Btu/hr.

Radiant loss with (ϵ = 1) = 487 Btu/ft^2. hr.

Emmisivity (ϵ) = 0.95

∴ Total wall loss = 11,720 Btu/hr.

Estimate of heat lost before diluted combustion products entered the recuperator (based on results given in Tables 5 - 5 and 5 - 6)

= 3600 Btu/hr.

Enthalpy of air at 1225°C = 43.6 Btu/ft^3

Enthalpy of stoichiometric combustion products at 1225°C = 47.6 Btu/ft^3

∴ Enthalpy of diluted combustion products entering recuperator at 1225°C

= 45.6 Btu/ft^3

Gas input to burner = 181 ft^3/hr.

∴ Net heat input to burner = 72,950 Btu/hr.

Heat entering recuperator = 69,350 Btu/hr.

∴ Rate of flow of diluted combustion products = 1540 ft^3/hr

Enthalpy of diluted combustion products leaving recuperator at 774°C

= 27.25 Btu/ft^3

∴ Total heat leaving flue = 42000 Btu/hr.

∴ Heat given up by combustion products = 37350 Btu/hr.

Enthalpy of ingoing air at 20°C = 0.2 Btu/ft^3

Enthalpy of outgoing air at 535°C = 17.5 Btu/ft^3

Air flow = 250 ft^3/hr.

Heat gained by air = 4330 Btu/hr.

<u>Calculations based on the results quoted in Table 5 - 10</u>.

<u>Wall Losses</u>

Area of asbestos lagging surface	= 16 ft^2
Characteristic dimension	= 3.17 ft.
Surface temperature	= 114°C
Ambient temperature	= 25°C
∴ Air flow over surface is turbulent	
∴ Convective heat loss	= 150 Btu/ft^2. hr.
∴ Total convective heat loss	= 2400 Btu/hr.
Radiant loss with (ϵ = 1)	= 224 Btu/hr.
Emmisivity (ϵ)	= 0.95
∴ Total radiant loss	= 3410 Btu/hr.

.˙. Total wall loss = 5810 Btu/hr.

Estimate of heat lost before diluted combustion products entered the recuperator (based on results given in Tables 5 - 5 and 5 - 6) =3600 Btu/hr.

Enthalpy of air at 1266°C = 45.2 Btu/ft^3

Enthalpy of stoichiometric combustion products at 1266°C = 49.8 Btu/ft^3

.˙. Enthalpy of diluted combustion products entering recuperator at 1266°C

= 47.5 Btu/ft^3

Gas input to burner = 185.3 ft^3/hr

.˙. Net heat input to burner = 74600 Btu/hr.

Heat entering recuperator = 71,000 Btu/hr.

.˙. Rate of flow of diluted combustion products = 1495 ft^3/hr

Enthalpy of diluted combustion products leaving recuperator at 625.6°C

= 21.6 Btu/ft^3

.˙. Total heat leaving flue = 32,300 Btu/hr.

.˙. Heat given up by combustion products = 38,700 Btu/hr.

Enthalpy of ingoing air at 25°C = 0.4 Btu/ft^3

Enthalpy of outgoing air at 472°C = 15.3 Btu/ft^3

Air flow = 1750 ft^3/hr

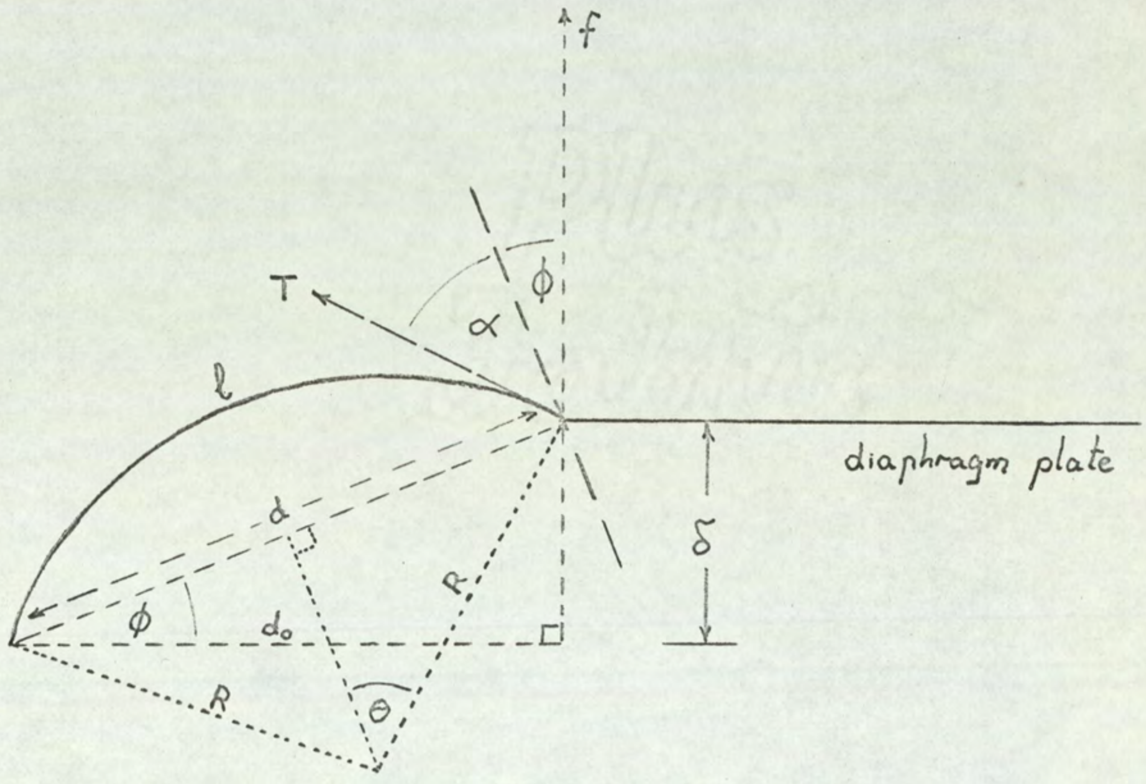
$\therefore$ Heat gained by air = 26,070 Btu/hr.

APPENDIX XI

Calculation of Forces on the Mk.I injector Diaphragm

The differential pressure that must be applied to one diaphragm of the injector in order to support the movement at various positions throughout its stroke, the injector being mounted vertically, can be calculated from knowledge of the weight involved and the geometry of the system.

Consider the geometry of a section of diaphragm in the diagram below:-



Let ΔP be differential pressure across diaphragm

F be force due to ΔP acting on diaphragm plate

f be force on diaphragm plate due to ΔP acting on annulus

T be tension in fabric of the annulus

W be the weight of the moving parts

δ be deflection of diaphragm from mid position

d_o be difference in radii of complete diaphragm and diaphragm plate

R_m be mean radius of annulus

A be area of diaphragm plate

Assume that the annulus has a circular section radius r, chord d, and arc length ℓ.

Also assume that the effective area of the annulus is the product of 2π, mean radius and chord, when resolving the force perpendicular to the chord.

Resolving force perpendicular to chord

$$2\,T \cos\alpha = d\,2\pi\,R_m\Delta P \qquad \text{...........(1)}$$

Now for positive values of ϕ, $f = T\cos(\alpha + \phi)$

$$= T(\cos\alpha\cos\phi - \sin\alpha\sin\phi)$$

But geometrically $\alpha = 90^\circ - \theta$

$\therefore$ $f = T(\sin\theta\cos\phi - \cos\theta\sin\phi)$

$$= T\sin(\theta - \phi) \qquad \text{...........(2)a}$$

If ϕ is negative $f = T \cos(\alpha - \phi)$

$$= T \sin(\theta + \phi) \qquad \text{..........(2)b}$$

From (1)

$$T = \frac{d\, 2\pi\, R_m\, \Delta P}{2 \cos \alpha}$$

$$= \frac{d\, \pi\, R_m\, \Delta P}{\sin \theta}$$

$$\therefore \quad f = d\, \pi\, R_m\, \Delta P \; \frac{\sin(\theta \mp \phi)}{\sin \theta} \qquad \text{..........(3)}$$

Now $\ell = 2\,\theta\, r$

and $\frac{d}{2} = r \sin \theta$

or $r = \frac{d}{2 \sin \theta}$

$$\therefore \quad \ell = \frac{2\,\theta\, d}{2 \sin \theta}$$

or $$d = \frac{\ell \sin \theta}{\theta} \qquad \text{..........(4)}$$

Also $\cos \phi = \frac{d_o}{d}$ ....(5)

and $\delta = d \sin \phi$

Now $W = f + F = f + A\Delta P$

$$= \left(d\, \pi\, R_m \frac{\sin(\theta \mp \phi)}{\sin \theta} + A \right) \Delta P$$

$$\therefore \quad \Delta P = \frac{W}{\left(d\, \pi\, R_m \dfrac{\sin(\theta \mp \phi)}{\sin \theta} + A \right)} \qquad \text{..........(6)}$$

This was evaluated as follows for a series of likely values of θ :-

1) A series of values of d were calculated from the equation $d = \frac{\ell \sin\theta}{\theta}$

2) A series of values of ϕ were calculated from the equation $\phi = \cos^{-1} \frac{d_o}{d}$

3) A series of values of δ were calculated from the equation $\delta = d \sin \phi$

4) A series of values were calculated for the expression $d \pi R_m \frac{\sin(\theta \pm \phi)}{\sin\theta}$

5) A series of values of ΔP were calculated from equation (6) above and plotted against δ

The appropriate values were :-

$\theta = 32^\circ \quad 36^\circ \quad 40^\circ \quad 44^\circ \quad 48^\circ \quad 52^\circ \quad 56^\circ$

$\ell = 1.3$ in.

$d_o = 1.125$ in.

$R_m = 4.6875$ in.

$A = 53.456$ in^2

$W = 3.84714$ lbs. wt.

which gave :-

δ (cm)	ΔP (in.w.g.)
- 1.28426	1.83267
- 1.17296	1.75545
- 1.03901	1.69483
- 0.86997	1.64390
- 0.64316	1.59739
- 0.23326	1.54397
0.00000	1.52792

δ (cm)	ΔP (in.w.g.)
0.23326	1.49802
0.64316	1.45125
0.86997	1.41526
1.03901	1.37928
1.17296	1.34136
1.28426	1.29956

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