PRESSURES GENERATED IN COMBUSTION CHAMBERS BY THE IGNITION OF AIR - GAS MIXTURES

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SYNOPSIS

A relationship between the energy content of stoichiometric pockets of natural gas-air mixture and the pressure rise resulting from their ignition has been derived from experiment and is reported in this paper. It is shown that the pressure rise is proportional to the energy available up to approximately 5 Btu/ft^3 (185 kJ/m^3) and that the increase in pressure measured in pounds force per square inch is numerically equal to the energy release measured in British thermal units per cubic foot. Above about 5 Btu/ft^3 energy release, the pressure rise increases more rapidly than is predicted by the simple linear relationship that holds at low energy releases.

The way in which this relationship is modified when the chamber is fitted with vents or flues has been investigated. It is shown that the venting of combustion products through circular orifices can significantly lower the pressure rise produced until the vent is reduced to a critical size, at which combustion products cannot escape rapidly enough and the full explosion pressure is obtained as if the chamber were not vented. It is demonstrated that flues cannot in general be considered to be effective explosion reliefs.

During burner start-up, unignited gas may be released into the combustion chamber where it mixes with the atmosphere. In practice only a small proportion of this gas, that is, the gas near the burner nozzles, is within the limits of flammability. However, it is shown that a weak mixture of gas and air having proportions below those of the lower flammability limit can contribute to the pressure rise, the effect becoming more marked as the temperature of the chamber is increased. The effect of flame speed on the relationships derived for natural gas-air mixtures was determined by repeating most of the experiments with town gas and propane.

The implications of the experimental work on industrial appliance design are discussed as well as the application of the results to other enclosures that could contain flammable gases.

Introduction

The use of gas-fired plant in industry is now increasing rapidly and it is necessary to ensure that all this plant will operate with a high standard of safety. The most important requirement for ensuring safety when using any fuel is that of the prevention of explosions and fires. Appropriate control systems and good burner designs normally preclude the accumulation of sufficient fuel in a plant to cause a hazardous pressure rise in the event of ignition.

Whilst every effort is made to make such systems "fail safe", in the event of a multiple fault situation, involving perhaps mal-operation of the plant, an explosion hazard could arise.

The consequences of an explosion depend on the rise of pressure and the structural strength of the equipment. The pressure developed is a function of the heat released in the explosion and the degree of venting. As combustion takes place the products formed escape through openings in the combustion chamber, thereby relieving the pressure. The pressure developed in a vented chamber depends on the rate of pressure rise which varies with the flame speed of the gas-air mixture. As the flame speed increases, so the rate at which combustion products are formed increases, and the pressure drop across the orifice through which the gases are being vented becomes greater. In a completely closed vessel the maximum pressure developed will not depend on the flame speed unless the vessel is extremely large.

The experimental work reported in this paper indicates the order of pressure rise to be expected in closed and vented enclosures arising from the explosive combustion of stoichiometric pockets of the mixture of air and gas. The remainder of the enclosure contained either air or an air-gas mixture having a composition below the lower explosive limit (L.E.L.).

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These results together with a knowledge of the strength of appliances and the effectiveness of venting by flue-ways, *etc.* enable a maximum safe rate of energy release for an appliance to be calculated and, from this, the maximum throughput at which the burner can be fired during the ignition cycle can be determined. The effects of this on burner design and control are outside the scope of this paper but have been adequately discussed elsewhere.[1,2]

An isolated pocket of a stoichiometric mixture in a typical gas-fired installation is an idealized situation. It was chosen as the basis of the experimental work because of its reproducibility. In practice, under "fault" conditions the sequence of events at burner start-up is more likely to produce a pocket of flammable mixture, not necessarily stoichiometric, surrounded by a mixture of air and gas having a composition below the L.E.L. Experiments performed to simulate this condition, together with investigations of the mixing of air and gas at a burner nozzle,[1] suggest that the calculated energy releases derived from pressure rises produced in the idealized experimental system of an isolated stoichiometric pocket surrounded by air are well on the safe side.

Apparatus and Experimental Technique

The experimental work involved the measurement of the pressure generated in a chamber by the ignition of stoichiometric pockets of gas-air mixture, confined in polythene bags, under various conditions of venting. The chamber and associated equipment is fully described elsewhere[1] and is shown in outline in Fig. 1. The pressure vessel itself, of volume 0·136 m^3 (4·8 ft^3), had a design working pressure of 21 MN/m^2 (3000 lbf/in^2) and was chosen because it would not be moved sufficiently by the explosions to affect the pressure transducers which were acceleration sensitive. Because of its mass, it also had a large thermal capacity and consequently its temperature did not greatly exceed the

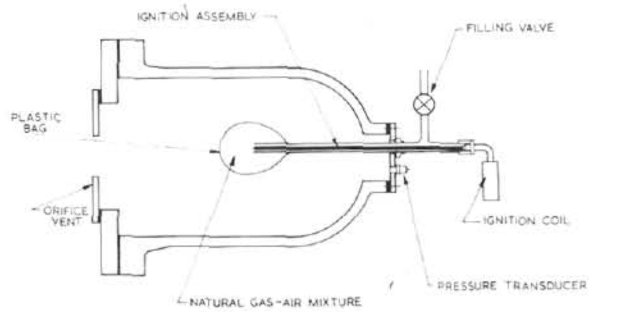


Fig. 1.—*Experimental vessel used for pressure rise measurements*

ambient temperature after a series of experiments. The flat end of the vessel could be fitted with a blank for use as a closed vessel, with sharp edged holes for the experiments on orifice vents or with a screwed flange to take flue pipes.

To avoid inconsistency in the results because of variations in the shape of the stoichiometric pocket, a polythene bag was chosen which was just filled by the mixture being used. The air and gas were metered into the bag by displacing the required volumes with water from a 500 ml burette. As the size of the bag corresponded to the amount of mixture required for any particular experiment, no pressure corrections were made because the pressure in the bag was about atmospheric. Samples of the fuel gases used were analysed and their calorific values were measured by Boys' calorimeter and used to calculate the actual energy input in the experiments.

The pressure transducers were dynamically calibrated in the rig shown diagramatically in Fig. 2. In this apparatus air was passed into the system through valve V_1 and, with valves V_2 and V_3 shut, pressure was built up to the desired level as shown by the pressure gauge, P. (This was calibrated against a dead weight tester.) Valve V_1 was then shut and V_3 (a fast-opening valve) was operated. This action generated a pressure pulse on the sensitive face of the transducer of the same order of rise-time as that generated by the ignition of the mixture of gas and air. If the distance between V_3 and the transducer T is kept small the volume of pipe between them is only a small proportion of the total system volume and the pressure drop on opening V_3 is negligible. Before calibration

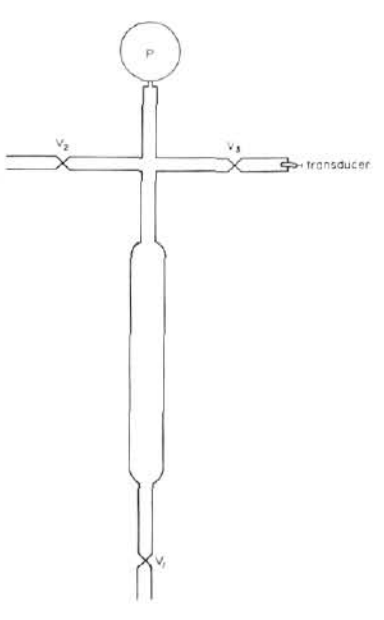


Fig. 2.—*Diagram of transducer calibration assembly*

at a different pressure, valve V_2 was opened to evacuate the system to atmospheric pressure. The calibration error was judged to be less than $\pm 2{\cdot}0\%$ at full-scale deflection on the U-V recorder chart.

At least two, and whenever possible three, transducers were used to measure the pressure rise in each experiment in order that spurious results (due perhaps to thermal shock) could be detected. The results presented are the average of at least six pressure recordings. Occasional inconsistent results that could be attributed to zero drift in the pressure recording system or loss of transducer signal were neglected. With nominally identical experiments, results agreed within approximately 5% in a closed chamber and about 10% in a vented chamber.

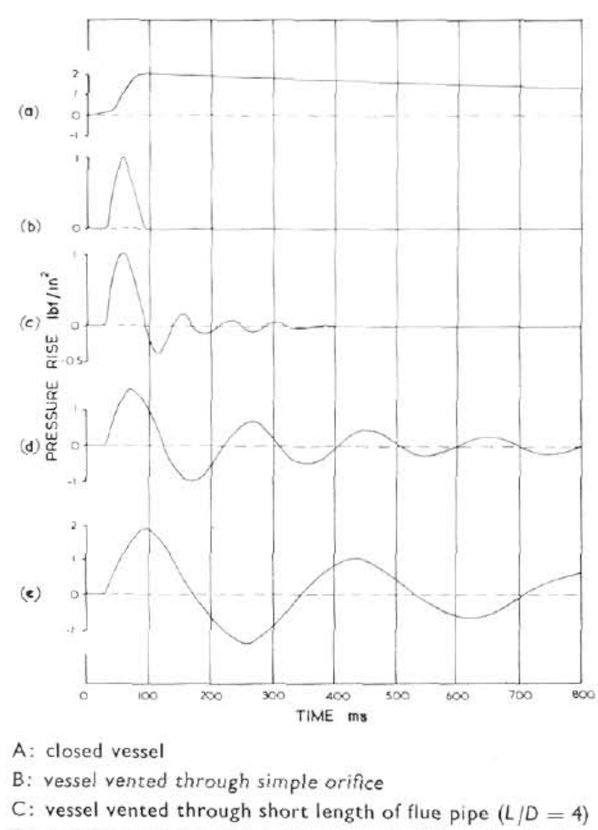
Discussion of Results

Typical pressure-time traces for different degrees of venting of the pressure vessel are shown in Fig. 3.

Experiments with Natural Gas

Pressure rise in a closed chamber

A typical pressure-time trace for combustion of a stoichiometric pocket in a closed chamber is shown in Fig. 3A. The pressure rises to its maximum value in approximately 40 ms and then slowly decays as the chamber atmosphere cools.



A: closed vessel

B: *vessel vented through simple orifice*

C: vessel vented through short length of flue pipe ($L/D = 4$)

D: vessel vented through medium length of flue pipe ($L/D = 28$)

E: vessel vented through long length of flue pipe ($L/D = 160$)

Fig. 3.—*Typical pressure-time traces resulting from the explosion of small pockets of stoichiometric air—gas mixture within a 4·8 ft³ vessel*

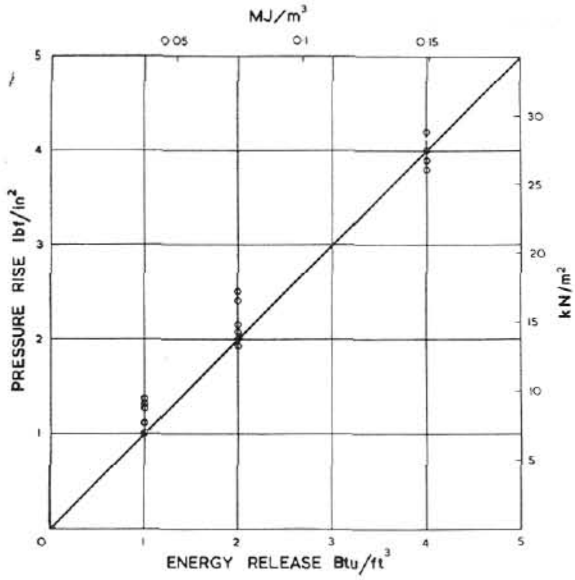


Fig. 4.—*Variation of pressure rise with energy release due to the explosion of stoichiometric pockets of natural gas—air mixtures within a closed steel vessel of volume 0·136 m³ (4·8 ft³) for the pressure range 0–5 lbf/in³*

The results of the measurements of pressure rises in closed vessels are presented in Figs 4 and 5. It can be seen from Fig. 4 that the curve is approximately linear up to energy releases of 5 Btu/ft³ (0·2 MJ/m³) and thus it is clear that the pressure rise in pounds per square inch is numerically equal to the energy release in British thermal units per cubic foot of enclosure volume for small energy releases. However, as the energy release is increased the relationship departs from linearity as shown in Fig. 5, an energy release of 20 Btu/ft³ (0·75 MJ/m³) for example producing a pressure rise of approximately 27 lbf/in² (189 kN/m²).

The curve drawn in Fig. 5 is a best fit second order polynominal having the form:

$$P = 1{\cdot}013E + 0{\cdot}016E^2 \quad . \quad . \quad (1)$$

where the pressure P is measured in pounds per square inch and energy release, E, in British thermal units per cubic foot of chamber volume.

Pressure rise in an orifice vented chamber

When the chamber is vented the simple relationship for closed chambers is modified because pressure rise is now a function of vent area as well as energy release. A typical pressure pulse obtained with orifice vented chambers is shown in Fig. 3B. The duration of the pulse is approximately 50 ms. The results are shown in Fig. 6 as a set of curves relating the pressure rise to vent area coefficient, for a number of values of energy release. The vent area coefficient, K, is defined as the ratio of the cross sectional area of the vessel to the area of the vent; for an open-ended vessel, therefore, this coefficient is unity. The definition of K was derived from work performed with cubical vessels[3] but can give useful correlations when applied to vessels in which the ratio of maximum to minimum side length does not exceed three.

The important point to note from Fig. 6 is that for the larger sizes of vents (that is, small values of K) some venting is achieved but as the vent becomes smaller the pressure developed inside the chamber gradually increases, until at some value of K (about 50) the vent ceases to have any influence and the closed-chamber pressure is developed. One explanation for this is that with values of K above 50, the resistance to flow across the orifice vent, because of the inertia of the air, is too great to allow any venting until a significant time after combustion of the stoichiometric pocket has reached completion.

Pressure rise in a flue-vented chamber

All appliances have built-in vents of one form or another. The two main vents on most installations are the flue and the air inlets. In most cases a flue pipe is installed, its length being determined by the installation details of the appliance and its diameter related to the throughput of the appliance, according to the relevant code of practice.[4] For a particular size of installation therefore the flue diameter will be defined but the flue length will depend upon local building considerations. The effect of flue venting of a chamber was therefore investigated for pipe diameters of 75 mm, 100 mm, and 150 mm (3 in, 4 in, and 6 in) and various lengths up to approximately 20 m.

A representative set of results giving the relationship between pressure generated in the vessel and flue length for three values of energy release is shown in Fig. 7. The curves start at a pressure corresponding to the venting due to an orifice. As flue length is increased, the pressure rise increases until it attains the value corresponding to that measured in an unvented chamber.

Pipe length increments of approximately 2 m (6 ft) were used to obtain the results presented in Fig. 7. On using pipe length increments of 0·3 m (1 ft) in order to increase the accuracy of this series of experiments, a new phenomenon became apparent. The results for 75 mm pipes are presented in Fig. 8, where it can be seen that instead of smoothly increasing with flue length, as before, the pressure rise shows distinct maxima and minima at particular lengths. The pressures recorded at these maxima are slightly greater than those which would be expected in a closed vessel. These effects were found with all pipe diameters.

The pressure-time traces are illustrated in Fig. 3C–E. These show that for short pipes the pressure becomes a heavily damped sinusoidal oscillation, the damping becoming

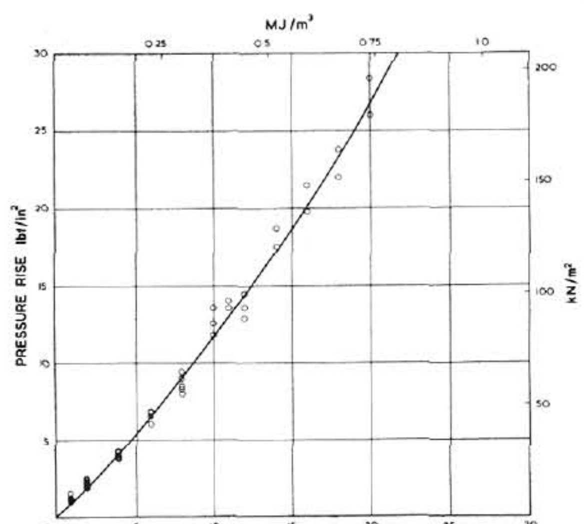


Fig. 5.—*Variation of pressure rise with energy release due to the explosion of stoichiometric pockets of natural gas—air mixtures within a closed steel vessel of volume 0·136 m³ (4·8 ft³) for the pressure range 0–30 lbf/in²*

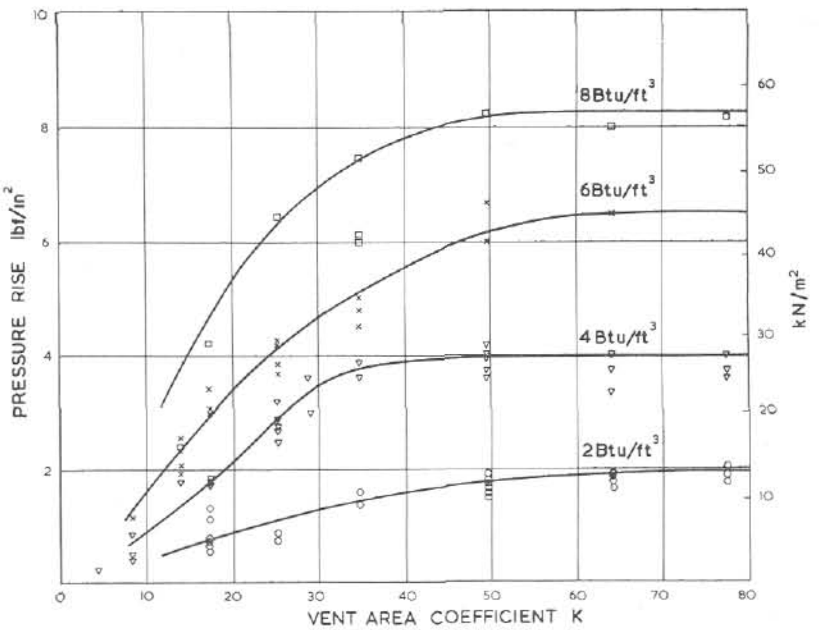


Fig. 6.—*Variation of pressure rise with vent area coefficient of a sharp edged orifice vent for various values of energy release*

less and less as the length of flue is increased. It can be shown that the frequency of the pressure oscillation is not the "organ-pipe" frequency usually associated with pipes but that of a Helmholtz resonator system the chamber of which is the explosion chamber itself and the "neck" the flue pipe, even when this "neck" is 20 m long. A possible explanation of the observed maxima and minima is that standing pressure waves were somehow produced in the flue by interaction of the Helmholtz oscillation and normal organ-pipe oscillations. The results show that, in general, the relief of pressure through a flue pipe is negligible, even short pipes increase the pressure generated in a chamber.

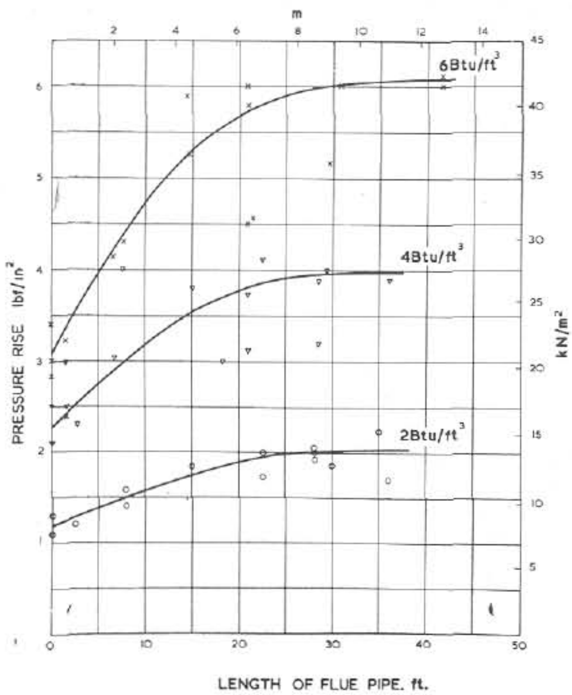


Fig. 7.—*Effect of venting through different lengths of 100 mm (4 in) diameter flue pipe on the pressure rise for various values of energy release*

The contribution to the pressure rise due to weak gas-air atmospheres

The experiments discussed so far were all performed in a vessel containing an air atmosphere. However, in practice an ignitable pocket will almost certainly be surrounded by an atmosphere containing gas. A study was made of the effect of such atmospheres on pressure rise by modifying the vessel so that it could be filled with premixed air-gas mixtures of known composition. A small stoichiometric pocket corresponding to 1 Btu/ft³ (37 kJ/m³) of chamber volume was used as the ignition source. Atmospheres ranging from pure air to approximately 3·5% v/v natural gas in air were investigated. Results for natural gas are shown in Fig. 9 (15°C line) where it can be seen that mixtures above about 2% and below the L.E.L. contribute significantly to the pressure generated. In the experiments reported above, the atmosphere in the vessel and the pockets contained in the plastic bags were at ambient temperature. This simulates the conditions prevailing when a burner is first lit in a plant starting from cold. In practice, burners are often relit when the plant is at an elevated temperature. In order to simulate this condition the pressure vessel was fitted with a heating jacket by which the chamber atmosphere temperature could be raised to approximately 120°C maximum. The experiments at ambient temperatures were then repeated at temperatures of 50°, 70°, and 105°C.

The results presented in Fig. 9 show that, as expected, a given pressure rise can be generated by weaker gas-air mixtures as the initial temperature of the reactants is raised. As well as decreasing the energy required to ignite the weak mixtures, an increase in ambient temperature will also increase the size of stoichiometric pocket of mixture which constitutes the ignition source. This combination of factors will tend to produce the effect presented in Fig. 9. The conclusion that must be drawn from these experiments is that weak gas-air atmospheres can contribute significantly to the pressure rise developed on ignition, even when the gas concentration is below the lower explosion limit.

Experiments with other Fuel Gases

The experiments reported so far have all been performed using Algerian natural gas as fuel. In order to determine the influence of fuel properties, notably burning velocity, on

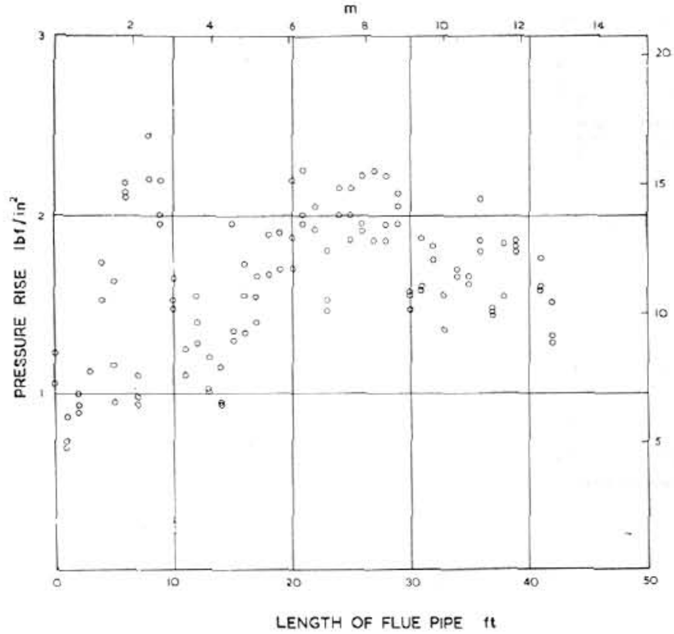


Fig. 8.—*Effect of venting through different lengths of 75 mm (3 in) diameter flue pipe on the pressure rise arising from the explosion of 75 kJ/m³ (2 Btu/ft³) pockets of stoichiometric mixtures*

the pressure rise developed following ignition of stoichiometric pockets of gas and air, some of the previous experiments were repeated using other fuel gases. Average compositions of these gases, which included town gas, propane, and methane, are given in Table I.

It was expected that burning velocity would not significantly affect the maximum pressure rise developed in a closed chamber. It can be seen that this assumption is valid from the results given in Fig. 10. This shows the variation of pressure rise with energy release for natural gas and town gas, which have burning velocities differing by a factor of approximately two.

With vented chambers, however, it is to be expected that pressure rise will be related to burning velocity, S_0. This is substantiated in Fig. 11 where, for low values of vent coefficient, the pressure rise is less for natural gas ($S_0 = 0{\cdot}36$ m/s) than for town gas ($S_0 \sim 0{\cdot}74$ m/s). The results given in Figs 10 and 11 are for vessels containing air as the atmosphere surrounding the stoichiometric pockets.

The effect to be expected when the surrounding atmosphere is a weak gas-air mixture of either below or just above L.E.L. proportions is not clear. With mixture compositions above L.E.L. the explosion pressure generated in a vented enclosure will depend, amongst other factors, on the flame speed, but since the flame speeds of near L.E.L. mixtures are all small and similar, any effect is likely to be immeasurable. In a closed vessel the pressure generated depends on the energy available. The results for sub-L.E.L. mixtures of the four

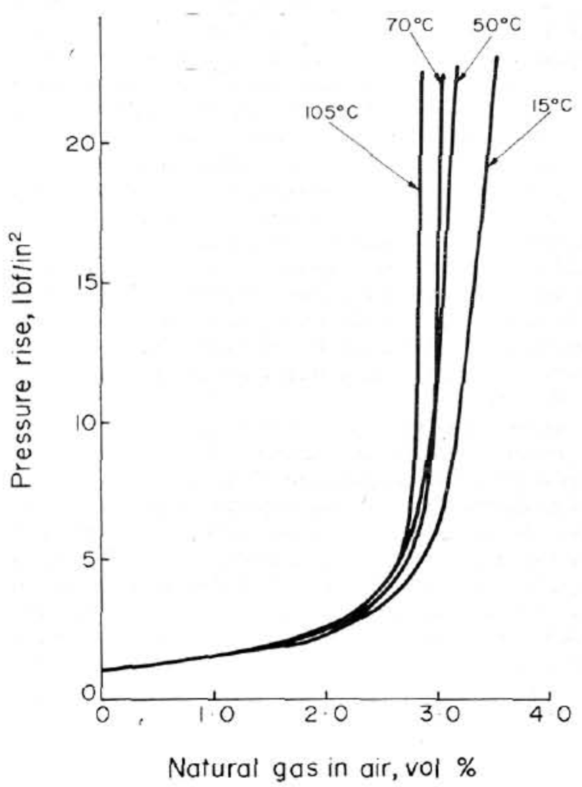


Fig. 9.—*Variation of the pressure rise with the concentration of gas in the atmosphere in a closed vessel for different initial temperatures; explosions initiated by 37 kJ/m³ (1 Btu/ft³) stoichiometric pockets*

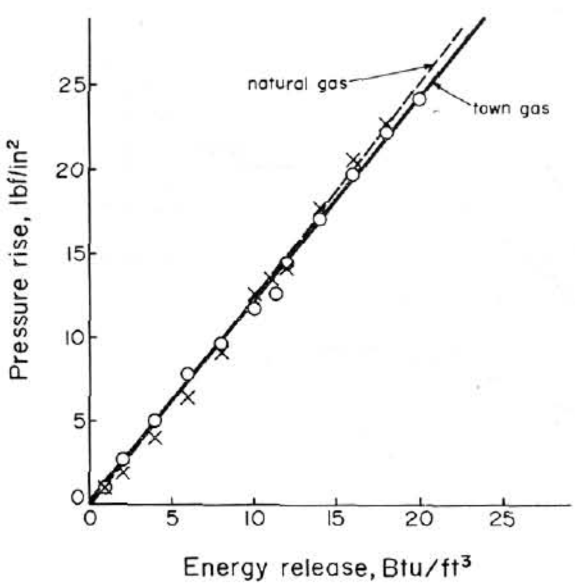


Fig. 10.—*Variation of pressure rise with energy release in a closed vessel for town gas and natural gas*

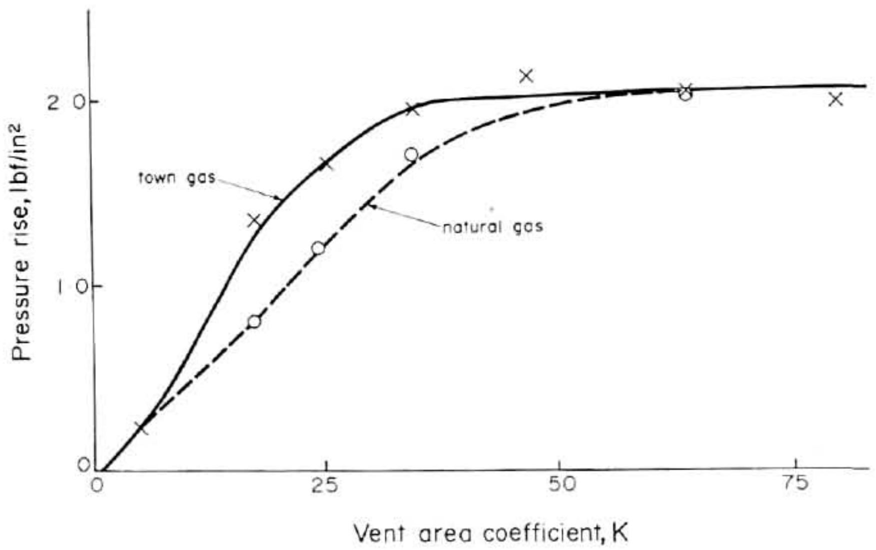


Fig. 11.—*Variation of pressure rise with vent area coefficient for 75 kJ/m³ (2 Btu/ft³) stoichiometric pockets of town gas and natural gas*

TABLE I.—*Average Composition of Fuel Gases*

	Natural gas	Town gas	Propane	Methane
CH_4	78·2	31·6	0·15	99·3
C_2H_4	—	—	1·85	—
C_2H_6	13·5	0·7	—	—
C_3H_8	4·6	—	91·45	—
C_4H_{10}	1·8	—	6·55	—
Higher hydrocarbons	0·2	—	—	—
H_2	—	51·6	—	—
CO	—	1·6	—	—
N_2	0·3	1·5	—	0·7
CO_2	—	12·7	—	—
O_2/A	1·4	0·3	—	—
Measured calorific value (Btu/ft³).	1192	498	2437	990

fuel gases are shown in Figs 12 and 13. In Fig. 12 the results appear to show a dependence of pressure rise on flame speed. This dependence is not so clear in Fig. 13 where the same results are plotted against the ratio N (% gas in mixture divided by % gas in stoichiometric mixture) and this could reflect the difficulty of determining accurately the stoichiometric air requirement of a fuel from its chemical analysis.

The source of ignition for all the experiments reported was a stoichiometric pocket containing the same gas as used in the atmosphere in sufficient quantity to give an energy release of 1 Btu/ft³. The size of the pocket must influence the pressure rise but since it has been shown[1] that the total energy content of the ignitable pocket of gas formed at a burner nozzle and the air—gas mixture below L.E.L. which surrounds the pocket is significantly less than 1 Btu/ft³ for a firing intensity of 100 000 Btu/h ft³ (2·93 MJ/h m³), the experimental procedure we have adopted will result in larger pressure rises than those likely to be encountered in practice if a delayed ignition occurs.

The effect of pockets formed in dead spaces, for example, in roof arches of furnaces when a lighter-than-air fuel is used, is a more intractable situation but a maximum pressure rise can be estimated by assuming that all the potential energy within the chamber is released.

Applications of the Experimental Results to Appliance Design

Data on the magnitude of the pressure rise resulting from the explosive combustion of air—gas mixtures have been given in the foregoing section. In this section the application of this information to the design of appliances will be considered. It was shown that the pressure generated in a closed chamber by the explosive combustion of fuel-air mixtures is independent of the flame speed and can be taken as proportional to the energy release over the range 0–5 Btu/ft³

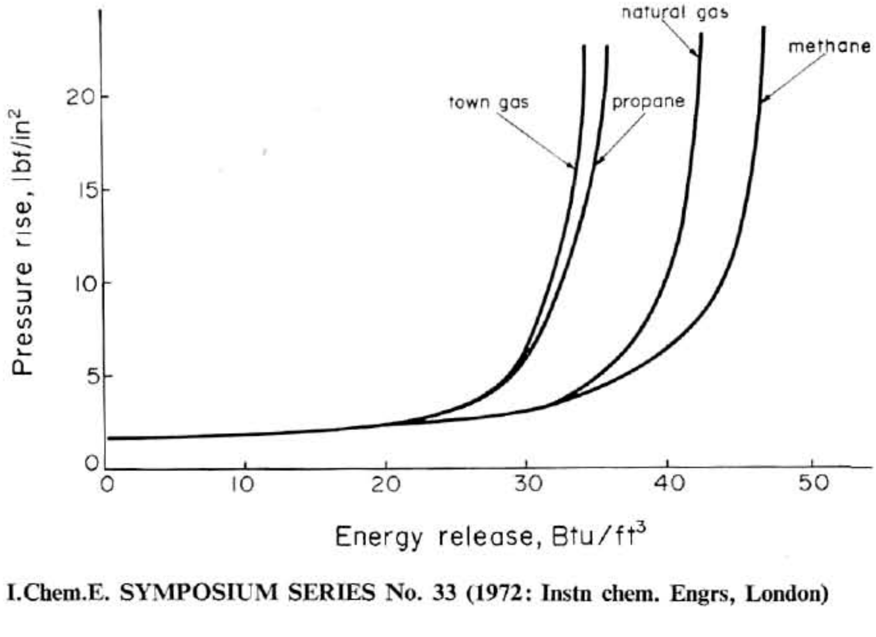


Fig. 12.—*Variation of pressure rise with the total energy available in both the pocket and the atmosphere, contained in a closed vessel of volume 0·136 m³ (4·8 ft³). The explosions were initiated by 37 kJ/m³ (1 Btu/ft³) stoichiometric pockets of the same gas*

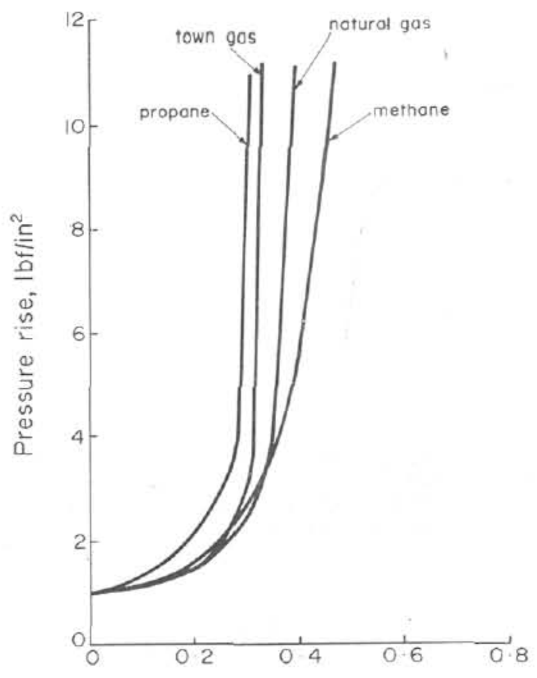


Fig. 13.—*Variation of pressure rise with N, the ratio of gas in the atmosphere to gas in the stoichiometric mixture*

(that is, up to 5 lbf/in^2) which is the pressure known to cause severe damage to structurally weak appliances. Thus if the strength of any unvented appliance is known, the maximum permissible energy that can be released can be calculated.

On starting any burner system there is a " trial for ignition period " during which an ignition source is present and fuel is allowed to flow. At the end of this period, if a properly established flame is not observed, either by the operator or by an automatic flame detector, then the fuel is shut off. If it is assumed that any fuel accumulated in the appliance could contribute to an explosion, then, if safety is to be ensured in the event of a delayed ignition, the maximum permitted energy release determined above defines the maximum fuel input during the trial for ignition period. Unfortunately there is little published information on the strength of gas-fired equipment. However, no practical appliance would withstand the pressure developed—of the order of 700 kN/m^2 (100 lbf/in^2)—by the ignition of a combustion chamber full of stoichiometric gas—air mixture. Even apparently robust equipment (certain types of boiler, for instance) may have weak fittings and attachments, such as inspection doors, which might fail, even in the event of only moderate pressure rises. Work on box ovens,[3] which are typical of structurally weak appliances, has shown that severe damage and risk of injury to personnel are avoided if pressure rises are not allowed to exceed approximately 14 kN/m^2 (2 lbf/in^2). In a closed chamber this implies an energy input of not more than 75 kJ/m^3 (2 Btu/ft^3) assuming all the gas admitted to the chamber is burnt. In a vented appliance the allowable energy release could be greater but this will depend largely on the degree of venting available.

Most gas-fired equipment will have some degree of built in venting arising from air inlets and flue outlets. This may or may not be significant in terms of reducing any pressure rise occurring in the combustion chamber. If it is assumed that these openings can be treated as orifice vents, a vent area coefficient can be calculated and the maximum allowable energy release can be estimated from the results presented in Fig. 6.

Calculated vent area coefficients for some typical appliances are shown in Table II. For the sectional boilers listed it is assumed that the only possible vent is the flue, and the vent area coefficients quoted are based on the area of the roof of the main combustion chamber and the diameter of the flue spigot. The high temperature furnaces considered are fired by natural draught burners and as such employ secondary air inlets but these contain dampers as well as being small in comparison with the flue, and have been ignored. The type of air heater quoted has a main combustion chamber unvented except for a 150 mm (6 in) diameter relief tube. The low values of vent coefficients derived for sectional boilers and high temperature furnaces imply that appreciable venting would occur. For the air heaters listed the pressure relief that might be expected is relatively small.

TABLE II.—*Vent Coefficients of some Commercial Equipment*[1]

Chamber Size (in)			Area of Face Containing flue	Flue Area	Vent
Length	Width	Height	(in^2)	(in^2)	Coefficient
1. *Sectional boilers*					
10½	18	17	190	50	3·8
18	18	17	325	50	5·5
21	18	17	375	50	6·5
26	18	17	465	50	9·3
31	18	17	555	50	11·0
14	22	24	315	78	4·0
21	22	24	450	78	5·7
27	22	24	583	114	5·2
33	22	24	715	114	6·3
40	22	24	870	114	7·7
45	22	24	980	154	6·9
2. *Small high temperature furnaces*					
18	12	12	216	28·2	7·6
24	18	12	432	39	5·5
36	24	15	360	39	9·2
48	36	18	650	78	8·3
3. *Air heaters*					
38¼	23½	27½	642	28·2	22·8
44¼	26½	30	792	28·2	28·1
57¼	26½	31½	832	28·2	29·5
60	36	44½	1600	28·2	56·8

Experiments that simulate the usual method of exhausting combustion products, through a flue pipe, have shown that the addition of the pipe significantly reduces the venting afforded by the flue exit. Indeed, when the flue pipe is over 20 ft long it cancels out the venting provided by the exit itself so that once again such a combustion chamber must be regarded as closed and unvented. The effect of flue pipes less than 20 ft long is difficult to assess because of the superimposed resonance oscillations. The magnitude of these is likely to depend on the geometry of the flue and appliance but it appears that it is possible for pressures to be generated which exceed that predicted from the energy release in a closed chamber. The effect clearly needs more thorough investigation but at present it may be pointed out that its significance is that the positioning of flue breaks could be important.

In general, therefore, in order to ensure that the pressure rise does not exceed 14 kN/m^2 (2 lbf/in^2), the appliance must be treated as unvented and the maximum permissible energy release calculated on that basis. Where the appliance strength and or venting coefficient are known, larger values of energy release may be acceptable but without this knowledge an appliance should be treated as structurally weak and unvented.

Conclusions

All industrial appliances must be considered as unvented unless they have a permanently open orifice giving a vent area coefficient significantly less than 50; every flue reduces the venting effect of the orifice to which it is connected.

If it is assumed that all the energy of the fuel released during the trial for ignition period of any automatic burner is converted into thermal energy, then, to ensure safe start-up, this energy input must be restricted to less than two British thermal units for each cubic foot of the combustion chamber volume (that is, 75 kJ/m^3). Equally, this argument can be applied to enclosures other than the combustion chambers of fuel fired plant, for instance, many chemical plants contain inflammable gases which could be ignited.

The data obtained from experiments to determine the mixing of gas and air at burner nozzles,[1] together with the results presented in Fig. 13 for pressure rises developed in a chamber containing gas—air atmospheres below the lower flammability limit suggest that a less restrictive, but still safe, energy input criterion can be proposed. The gas mixing experiments reported in Ref. 1 show that the energy content of the combustible zone of gas—air mixture formed around the nozzle of certain types of burner can be relatively small. It is further shown in Fig. 13 that for atmospheres containing less than 15% of the stoichiometric gas requirement, no significant contribution to the pressure rise by the atmosphere is obtained, even with an ignition source equivalent to 1 Btu/ft^3 (37 kJ/m^3). These factors have enabled an energy input criterion for safe start-up, based on dilution, to be proposed[6] permitting burner start-up on full air flow and up to 10% of the maximum gas flow. It is acknowledged that the total energy input in this situation is greater than 2 Btu/ft^3 but, from experimental evidence to date, the energy release will not be greater than 2 Btu/ft^3 and hence pressure rises in excess of 2 lbf/in^2 (14 kN/m^2) will not be developed. This dilution criterion would apply to burner start-up from cold and re-starting in low temperature appliances. Further work is necessary to determine its applicability to re-starting a burner in a high-temperature appliance since experiments have shown that the effect of increasing the ambient temperature is to lower the gas concentration (in the chamber atmosphere) at which the contribution to the pressure rise becomes significant.

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Symbols Used

D = diameter of flue pipe (in or mm).
E = energy release (Btu/ft^3 or MJ/m^3).
K = vent area coefficient equal to cross sectional area of chamber/area of vent.
L = length of flue pipe (ft or m).
N = gas in mixture/gas in stoichiometric mixture.
P = pressure rise (gauge) (lbf/in^2 or kN/m^2).
S_0 = burning velocity (ft/s or m/s).

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