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A RATIONALE FOR ADVANCES IN THE TECHNOLOGY OF I.C. ENGINES

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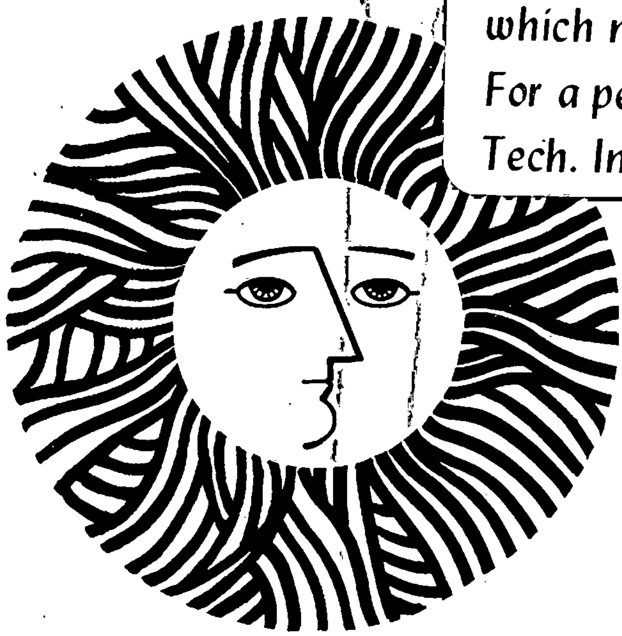

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ABSTRACT

A perspective view of the evolution of I.C. engines reveals that by the end of the 1960's they attained a peak in their technology for the purpose they served up to then. At that time they became confronted with two severe constraints stemming from the demands for, first, pollution control due to the concern over the environmental impact, and, then, fuel economy due to the energy crisis. So far these demands have been met primarily by peripheral engine system improvements, such as catalytic converters and electronic controls. This paper is based upon the premise that it should be advantageous to respond to the new constraints by further advances in engine technology, designed to (1) minimize pollutant emissions, (2) maximize engine efficiency, and (3) optimize tolerance to a wider variety of fuels. A sense of direction for such advances is derived from a critical assessment of the fundamental advantages of reciprocating I.C. engines as prime movers for automobiles, and the review of their recent development. On this basis it is shown that major impact in this respect could be made by <u>controlled combustion</u>, a concept that has not yet been given the attention it deserves. Intrinsically, controlled combustion is based upon a proper treatment of active radicals, the essential elements of the combustion reaction. Practically, this can be achieved by a variety of means, such as charge stratification, exhaust gas recirculation, homogeneous lean burn, combined with enhanced ignition and enhanced auto-catalysis. Thus, the most desirable advances in engine technology would transfer a good deal of the functions served by catalytic converters and electronic controls into the chemistry and mechanics of combustion processes taking place in the engine cylinder. While such advances can be, and are indeed, made by following most of the trends already established, it is hoped that these developments could be enforced by the realization of their fundamental significance and the consequent acquisition of a firm sense of direction, the necessary prerequisite for success in a technological progress.

INTRODUCTION

In the 1960's the technology of internal combustion engines was at the peak of performance in response to the service they were providing at that time for one of the most highly demanded commodity: the automobile. This was manifested by such features as:

- compactness in on-board energy as well as specific power
- controllability and consequent flexibility in satisfying a wide variety of operating demands
- reliability combined with a most impressive durability
- relatively low cost due to superb engineering of manufacturing processes

As a consequence, no particular need was felt for further progress in engine technology except for the challenge of normal competition between engine manufacturers.

In the 1970's the industry became confronted with two major constraints:

- the requirement for control of pollutant emissions due to the concern of the society over the environmental impact of engines,

and

- the demand for fuel economy due to the energy crisis

The response of automotive technology to these constraints was primarily in the form of peripheral improvements of the propulsion system. These were principally associated with the exploitation of two practical means:

- catalytic converters

and

- electronic controls

The major argument put forth here is that a more fruitful approach to meeting this challenge can be derived from the application of fundamental principles of combustion to processes occurring in the cylinder of reciprocating internal combustion engines, whereby resorting to energy consuming peripheral devices becomes unnecessary. In the following we discuss the advantages of the basic I.C. engine as the prime mover for the automobile and the most prominent fundamental aspects of its combustion processes. Principles of controlled combustion, that is one that would maximize fuel economy and minimize the formation of pollutants, are then exposed and examples presented of recent engine developments where they are in part incorporated. Our objective in doing so is to provide thereby a firm sense of direction in automotive engine research that would hopefully lead to a significant technological progress.

INTRINSIC ADVANTAGES OF INTERNAL COMBUSTION ENGINES

In providing service as prime movers for automobiles, the reciprocating piston internal combustion engine has a distinct advantage over any alternative propulsion system one can envisage. This is due to a number of factors which, in order to develop our theme, are listed here, in spite of their often most elementary character. The need to do so at this time is enforced by the impressive number of very costly, unsuccessful developments which have been undertaken in recent years, evidently without due regard to their elementary, fundamental flaws. Thus the following features must be taken into account.

1. Technological superiority of the 1960's as specified in the Introduction. The compactness in energy and power is depicted in Fig. 1, a graphical representation of the performance regimes of various prime movers on the plane of specific power vs. specific energy, estimated some time ago for a vehicle of one ton curb weight (1,2).

2. Relatively high energy conversion efficiency (3), which, in contrast to the classical Carnot and Stirling cycles, is essentially independent of the maximum temperature of the working substance and can be improved significantly by the simple expedient of charge dilution. This is demonstrated in Fig. 2, presenting the thermal efficiencies as a function of the equivalence ratio, with compression ratios as parameters, for the Otto cycle (Fig. 2a) and Diesel cycle (Fig. 2b). In the computations, the compression cycle was assumed to be at constant composition, while thermodynamic equilibrium was attained at the end of the combustion process (the difference between results obtained with the expansion process assumed to be at either constant composition or at shifting equilibrium is negligible).

As it appears here, a reduction of the equivalence ratio from 1.0 to 0.5, is for an Otto cycle, equivalent to raising the compression ratio from 8.5 to 12. Moreover, since dilution inhibits the tendency for knock, diluting the charge can be combined with an increase in compression ratio. The effect of dilution in the case of Diesel cycle is even more impressive, contributing a fundamental reason for the remarkably high part load efficiency of Diesel engines. However, whereas lean operation of these engines is practically very easy to achieve, for Otto cycle spark ignition engines it is quite difficult.

It should be noted that the Otto and Diesel cycle efficiencies plotted in Fig. 2 compare favorably with the thermodynamic efficiency of the Stirling cycle whose major claim to fame is the relatively high value of this parameter. In contrast to I.C. engines the latter is, however, limited by the temperature of the heat transfer wall which, for practical reasons, cannot exceed the level of $1000^{o}K$. Moreover, the cycle efficiency is restricted by the fact that the heat transfer rate is finite. As a consequence, the maximum cycle efficiency is $\eta_{effective} = 1 - (T_c/T_h)^{1/2}$, rather than $\eta_{theoretical} = 1 - T_c/T_h$, where T_c and T_h are the temperatures of the cold and hot reservoirs, respectively (viz. (4)). For $T_c = 300^{o}K$, this yields the value of $\eta_{effective} = 0.45$, well within bounds of the numbers displayed in Fig. 2. At the same time, the argument that the external combustion engines operating on such cycles can have relatively clean exhaust gases, should be considered fundamentally incorrect. The conditions existing in the cylinder of an internal combustion engine are, as it will be made here clear later, much more suitable in this respect in spite of the relatively short time available there for this purpose, especially if adequate methods for enhanced ignition (5) are properly exploited.

3. Remarkably low thermal distortion due to cyclic temperature variation in the cylinder that effectively cuts by half the temperature drop across its walls, combined with uniform temperature distribution along them -- a feature that makes reciprocating piston engines definitely superior to rotary engines.

4. A virtual batch-type operation,whereby the working substance is processed by samples taken into the cylinder at each cycle, making the efficiency of internal combustion engines essentially independent of the low Reynolds number limitation that restricts the applicability of gas turbines for automobiles.

5. Above all, the remarkable potentiality, so far relatively unexplored, for significant improvements in:

- the minimization of pollutant emissions
- the maximization of engine efficiency
- the optimization of tolerance to various fuels combined with virtual elimination of the tendency to knock

that is achievable by controlled combustion.

In order to gain an appreciation of the last point, a rudimentary understanding of the essential features of the combustion process is necessary. This is provided in the next section.

THE COMBUSTION PROCESS

In engines,the combustion process is an exothermic oxidation reaction taking place in a gaseous phase.Its most natural form is a flame. Hence, most combustion systems are based on the use of flames. There are essentially three types of flames: diffusion flames, laminar flames in premixed gases, and turbulent flames in premixed gases. Salient properties of each are presented here in turn.

Diffusion Flames

Such flames occur whenever fuel is not thoroughly premixed with the oxidizer, as is the case in direct injection engines, notably the Diesel engines. The structure of such a flame, evaluated for a typical self-similar jet (6), is depicted in Fig. 3. Before ignition the fuel and the oxidizer diffuse into each other as shown in Fig. 3a. After ignition, the flame is established at the stoichiometric contour, that is when the ratio of O to F attains stoichiometric proportion, and acquires the structure shown in Fig. 3b. In the diagrams of Fig. 3, all the parameters have been normalized with respect to their maximum values. As shown in Fig. 3b, both the fuel and the oxygen are completely depleted at the flame front, while the products and the temperature, expressed in terms of its normalized value, τ, attain their peak levels. As it is here apparent, the fuel approaches the temperature peak in the absence of oxygen, a condition causing pyrolysis and the concomitant formation of particulates. Thus, as it has been indeed pointed out on a number of occasions (7), diffusion flames have the dubious attribute of automatically maximizing the production of pollutants.

Laminar Flames in Premixed Gases

The structure of such flames provide a smooth transition from the unburned mixture to the burned gases and is associated with a distinct normal burning speed, a characteristic parameter for a given fuel-air mixture. Today this phenomenon is well understood, as manifested by a remarkable agreement between the results of numerical computations and experimental observations. For illustration, given here in Fig. 4 are the results obtained by Westbrook (8) for a methane-air mixture. Figure 4a shows the flame structure in terms of major species concentrations and the temperature profile, while Fig. 4b provides information on the concomitant radical concentration profiles. The normal burning speed, S_u, is a sensitive function of temperature, T_u, which for the methane-air mixture can be expressed as follows (9):

$$S_u = 10 + 3.71 \times 10^{-4} (T_u/K)^2 \quad \text{cm/sec} \tag{9}$$

Thus, it is well established that laminar flames in premixed gases are governed entirely by molecular diffusion, thermal conductivity and chemical kinetics. As a consequence, such flames are distinguished by a remarkable controllability that can be attained simply by proper adjustments in mixture composition and initial temperature.

Turbulent Flames in Premixed Gases

These flames are prevalent under most practical circumstances, especially those existing in internal combustion engines. As illustrated in Fig. 5, a schlieren photograph of a turbulent flame stabilized in a combustion tunnel (10), such flames are predominantly influenced by the large scale vortex structure of turbulent flow and their fronts are established at the contours between the fresh charge and the recirculating combustion products. As a consequence, the laminar flame structure, with its normal burning speed, becomes deeply imbedded in the convolutions and striations of the turbulent combustion front, and a good deal of controllability is lost.

The obvious question to be posed in view of the characteristic properties of flames described above, is how a combustion process can be controlled. This question is addressed in the next section.

HOMOGENEOUS COMBUSTION PROCESS

In a combustion process of this kind the chemical reaction occurs coherently throughout the whole charge. This is illustrated schematically in Fig. 6, where Fig. 6a displays the principal feature of a flame, the combustion process occurring in such a way that the full amount of exothermic energy, E, is released throughout a small fraction of the mass of the charge, M, and the front of this process sweeps through the mass. In contrast to this, in the case of homogeneous combustion process the exothermic energy is released gradually throughout the mass of the charge and the process is completed when the full amount of this energy is released, as shown in Fig. 6b. It should be noted that in all the diagrams of Fig. 6 areas cross-hatched at 135^{o} denote products of complete combustion, while those cross-hatched at 45^{o}, refer to intermediate constituents of incomplete combustion. These are then the two extremes. In a practical situation, mixed and intermediate types of processes can occur. Figure 6c represents the essential feature of knock, the flame front where the full amount of exothermic energy is released sweeping through the charge, while a small fraction of it undergoes a homogeneous combustion process at a high rate, generating an explosion. On the other hand, one may have a homogeneous combustion process occurring over a limited fraction of the mass, giving rise to a flame front as depicted in Fig. 6d. The two processes do not have to be, of course, concurrent, that of the homogeneous combustion having been completed before the flame front. A situation like this occurs, for example, in the case of jet ignition, where a cloud of ignition sources is burned throughout its extent, giving rise to a spheroidal flame front (11,12).

Figure 6e describes schematically a situation which is essentially the same as that of Fig. 6a; however as a consequence of the fact that the flame front is distributed throughout the mass, practically it is equivalent to 6b. Figure 6f represents again a situation which is essentially the same as that of Fig. 6d, but in practice it produces an equivalent effect to that of Fig. 6b.

Thus, it should be noted that the effects of highly distributed flames is practically the same as that of a homogeneous combustion, although the two processes are essentially different from each other.

Is a homogeneous combustion process a completely new concept? From the fundamental point of view, certainly not, but insofar as practical applications are concerned, it is relatively new. In fundamental studies, the process of homogeneous combustion has been thoroughly explored in detonation research (viz. e.g.(13)), in particular, in the investigation of combustion processes behind shock waves (viz. e.g. (14)), and, on the other side of the scale, in combustion research involving stirred (15) and plug flow reactors (16,17). As an example, Fig. 7 shows a cinematographic sequence of interferograms of combustion behind a reflected shock wave in a stoichiometric mixture of hydrogen and oxygen, diluted with argon (18). The relatively equal amplitude of the fringe waves throughout the width of the reaction zone, as their intensity increases with time, demonstrates the characteristic property of the uniform distribution of the homogeneous combustion process.

Examples of practical applications of the homogeneous combustion process are provided by the LAG (Avalanche Activated Combustion) engine of Gussak illustrated in Fig. 7, copied from the 1966 U.S. Patent 3,230,939; the Toyota-Soken (TS) combustion (19) and the so-called Active Thermal-Atmosphere Combustion (ATAC) process of Onishi (20), the salient features of which are displayed in Fig. 9. The latter two are based on ignition provided by "prompt exhaust gas recirculation" described here later.

As demonstrated by the latter applications, the controllability of a homogeneous combustion process is indeed superb, cycle-to-cycle oscillations recorded by a pressure transducer disappearing entirely when the engine is shifted to this mode of operation.

As it becomes clear from the above, a homogeneous combustion process can be attained by proper mixture preparation,combined with some form of a distributed, enhanced ignition system. The essential prerequisite for this purpose is a thorough premixing and essentially lean composition, produced by dilution with either excess air or the recirculated exhaust gas,or both. The principal means of control are provided by active radicals whose proper composition is maintained in radical pools. This is illustrated in Fig. 10, taken from a recent paper on the thermochemistry of ignition (21). Here the results of thermochemical computations of the ignition process in a methane-air mixture, subject to heat losses expressed in terms of a thermal relaxation time of 10^{-2}sec., are displayed in four representative cases. Case 1 is that of thermal ignition; case 2 is the corresponding thermal extinction, the difference between the two being due to just 20^{o} in initial temperature. case 3 shows ignition produced by the introduction of radicals; case 4 is the corresponding extinction following the introduction of radicals at an initial temperature only 20^{o} lower. One should note that the time scale in these diagrams is logarithmic. Thus the plateaus in the composition and temperature profiles are, in effect, significantly longer than they appear there by sight. These plateaus represent the conditions of radical pools that lead eventually to a relatively rapid ignition that is referred to for this reason as "thermal explosion" (22,23).

The primary control systems in homogeneous combustion are:

- enhanced ignition -- a primarily physical process,

or

- enhanced auto-catalysis -- a primarily chemical process

The major problem in the realization of homogeneous combustion is the relatively narrow band of controllability between explosion and extinction. As exemplified by the LAG, TS, and ATAC engines, this problem is certainly treatable. In effect, controlled combustion is equivalent to controlled explosion, and the background knowledge we have acquired in detonation research over the last 20 years (13,14) should be in this respect particularly helpful. One should note that by the same token the phenomenon of knock is in the case of controlled homogeneous combustion essentially eliminated. Controlled explosion is obviously tantamount to controlled knock.

OVERVIEW OF PROGRESS IN ENGINE DEVELOPMENT

As pointed out in the previous section, from the point of view of combustion fundamentals it is highly desirable that advances in engine technology should be concerned primarily with the development of engines operating with a homogeneous, premixed charge. The relative position of such engines with respect to others has been described by Uyehara, et al. (24). A comprehensive assessment of combustion processes in a wide variety of stratified charge engines, including a discussion of the advantages and disadvantages of lean charge engines, was presented by Newhall (25). A comparison of potential efficiencies of various prime movers for automobiles has been compiled by Lauck, et al. (3) who provided fundamental arguments for the intrinsic superiority of I.C. engines over alternative automotive propulsion systems.

In an earlier paper (5), we (JDD and AKO) presented the concept of enhanced ignition with particular relevance to engines using premixed charge. This concept is based on the principle of controlling the processes of initiation and propagation of combustion by not only providing ignition sources but also distributing them throughout the combustible mixture.

In order to establish an interrelationship between the various methods developed in this connection, their respective positions were expressed graphically on a plane of relative energy and relative volume, as shown here in an expanded form on Fig. 11. The relative energy, , is defined for this purpose as the ratio of the expended electrical energy or the exothermic energy content (heating value) of the medium used to ignite the main charge in the cylinder to the exothermic energy content of the intake gas; the relative volume, , is taken as the ratio of the volume of the prechamber or preparation chamber to the unswept volume in the cylinder.

The standard spark ignition, including the recently developed high ignition energy systems, are at the lowest level on the energy scale and zero relative volume. Open chamber stratified charge (SC) engines are also located in this region, because combustion is initiated there by the use of standard spark ignition systems. Plasma jet ignition engines are one or two orders of magnitude higher in relative energy and at a finite, albeit small, value of relative volume. Divided chamber stratified charge and torch cell engines are one more order of magnitude higher in relative energy, but since it is chemical in nature rather than electrical, the actual cost in terms of energy expenditure is in this case much lower. The relative volumes of typical systems of this kind can easily attain values of 0.25 as indicated in Fig. 11. The May Fireball engine (26) can be considered as an extreme case of a torch cell system and we placed it at a location corresponding to a relative energy and relative volume both of an order of 10^2. The prompt EGR ignition engine (19, 20), a system based on the use of a homogeneous combustion process described here in some detail later, has about the same relative energy as the divided chamber SC and torch cell engines, but is distinguished by a significantly higher relative volume that is, in effect, equal to the engine compression ratio. It should be noted that in normal operation these engines do not use any electrical energy for ignition.

Displayed also on the diagram of Fig. 11 are the direct injection (DI) and the indirect injection (IDI) Diesel engines. In the case of normal Diesel engines, the relative energy is actually infinite,while for the dual-fuel Diesel engines it is at a level of 1 to 10. Typical examples of such engines are power plants using natural gas as the main fuel in the aspirated charge and Diesel fuel to provide a pilot flame for ignition, situated at the lower level, and the alcohol Diesel engines, situated at the higher level within this range. The relative volume for the DI Diesel is practically zero, while for the IDI Diesel it is, of course, finite.

One should note that the techniques used to achieve clean and efficient combustion rely primarily on the use of charge stratification and/or exhaust gas recirculation, combined with homogeneous lean charge, enhanced ignition, and, possibly, enhanced auto-catalysis. In the diagram of Fig. 11 these systems are expressed in terms of compartments in space. However, it should be noted that the same objective could be also attained by proper staging of the preparatory processes in time, so that, in effect, systems of zero relative volume can be made in principle equivalent to those of finite relative volumes. This can be accomplished by exploiting the ignition delay for mixing, in order to attain proper conditions for controlled combustion.

As it appears thus from this overview, present day I.C. engines have reached a stage in their development where several alternative methods are, in effect, used for combustion control. The conventional S.I. engine, with a three-way catalytic converter and a micro-processor controller for monitoring the ignition timing as well as the EGR and A/F mixture composition, attained a remarkably high level of performance in meeting the present day emission and economy standards, but, for fundamental reasons, they are quite restricted in fuel adaptability and in efficiency, the latter as a consequence of the high temperature required for proper operation of catalytic reactor in the exhaust system. The divided chamber SC and torch cell engines are similarly limited in fuel adaptability, while the latter has, on top of it, an inadequate effectiveness in emission control.

POLLUTANT EMISSION CONTROL

All Diesel engines, with the exception of rarely used spark assisted systems, rely on auto ignition of a stratified charge produced by the fuel injector. As pointed out here earlier, the combustion process is then essentially uncontrolled, although a small amount of control can be achieved by proper mixing produced by the jet of fuel, associated with proper injection timing and adjustment of its rate. As a result of stratification, these engines have the advantage of reduced HC levels, compared to homogeneous charge engines, because fuel can be kept away from the cylinder walls and crevices at the obvious sacrifice, of course, in specific power output. However, as a consequence of the intrinsic lack of control over the combustion process, these engines are severely handicapped by relatively high emissions of all the other pollutants.

Premixed charge engines, on the other hand, offer numerous possibilities for in situ control of pollutant formation. These are, for each of the pollutants in turn, as follows:

HC

Insofar as the HC emission problem is concerned, initial research (27) identified both the wall quenching and the effect of crevices as their main sources. Subsequently (28) the roll-up vortex was discovered and recorded by schlieren cinematography (29) while more recently (30, 31) the wall quenching concept has been discredited, both on theoretical as well as experimental grounds. Thus, the roll-up vortex is today considered to be rich in HC, principally as a consequence of out-gasing from the space between the piston and the cylinder above the top ring, reenforced by an additional supply of HC deposited there by lubricating oil.

Ideally one would like to eliminate as much as possible all the sources contributing towards the formation of HC in the combustion chamber. One means would be to enhance the diffusion of the roll-up vortex throughout the charge and promote thereby mixing of HC issuing from the crevice at the edge of the piston with hot combustion products, containing free oxygen due to the use of lean mixtures, to enforce its oxidation. Another way would be to enhance the oxidation of HC by means of catalytic surfaces at the walls in the immediate vicinity of the roll-up vortex.

CO

The major causes of CO in IC engines are: (a) the deficiency in oxygen required for complete oxidation of fuel carbon, (b) the high temperature dissociation of CO_2, and (c) the exceptionally slow oxidation kinetics that is principally restricted by the elementary step: CO + OH, the slowest bimolecular reaction in the chemical chain mechanism of hydrocarbon combustion. To prevent the formation of CO requires therefore an ample supply of oxygen and an adequate amount of residence time at the moderately high temperatures to assure the completion of the chain reaction process. In order to obtain sufficiently low CO levels it might be necessary to enhance the slow oxidation reaction by the use of radicals -- certainly a most worthwhile objective for an enhanced auto-catalytic technique, as yet to be developed.

NO_x

The kinetics of NO_x formation in engines have been a subject of a thorough study resulting in most satisfactory predictions of measured levels with and without EGR (32). Figure 12 presents predicted results for a fixed combustion rate with various gas/fuel ratios (G/F = A/F + EGR/F). The diagram illustrates that to attain a relatively low NO_x level, say 5 grams/kw-hr, a large amount of EGR must be used with a stoichiometric A/F mixture, while less EGR is required for higher A/F ratios, up to 21/1 when the EGR requirement is practically annihilated. Nonetheless, as described here later, the use of relatively large amounts of EGR can be of great benefit to the control of combustion in I. C. engines, besides its influence in curbing NO_x emissions.

HOMOGENEOUS LEAN CHARGE

As it has been already pointed out, in order to attain low levels of NO_x emission,one has a choice between a near stoichiometric mixture with a certain amount of EGR, combined with a three-way catalytic converter and micro-processor controller, and mixtures of relatively high A/F ratios combined with enough EGR to eliminate the need for chemical processing of exhaust gases. The latter route is more attractive since it should be associated with improved efficiency due to high A/F ratio and increased compression ratio capability due to lower knock tendencies of lean mixtures. So far the desirable end result has been difficult to attain primarily because of the onset of the lean missfire limit (LML). This can be attributed to the use of relatively weak, single point ignition systems that lead to cyclic missfire or to slow combustion rates, making the process subject to bulk quenching. This causes high HC levels which, together with the inherent low exhaust temperatures, prevent adequate reduction of pollutant emissions. In multi-cylinder engines the situation is aggravated by malformation of the mixture and its maldistribution.

However, single cylinder engine experiments (33) showed that stable and fast combustion rates can be attained at A/F ratios of up to 23/1 and that at such conditions the HC levels are not worse than those of normal burn rate engines operating at A/F of 16/1. While the more rapid burn rate yielded higher NO_x levels, the EGR was quite effective in reducing them without any loss in efficiency under the WOT, MBT, constant speed conditions, at which these tests were run.

One should bear in mind that a successful development of a homogeneous,lean burn engine depends very much upon the use of an appropriate enhanced ignition system. This might be attained either in the form of a LAG type prechamber, an advanced version of a divided chamber stratified charge engine described in the next section (34,35,36) or in the form of plasma jet igniters (11,12,37,38,39,40). The demonstrated capability of these enhanced ignition systems to promote fast combustion rates in gasoline engines indicate their potentiality in reducing the octane requirement when they are properly used.

The concept of enhanced auto-catalysis implies supplying the combustible mixture with materials which could reduce the activation energy required for reaction. This may be done in the form of active radicals provided by the igniter or derived from catalytic surfaces around the combustion chamber. It should be noted that in either case the full potential of such devices can be developed only in combustion systems where the reactants are thoroughly premixed.

CHARGE STRATIFICATION

In principle, there are two types of charge stratification: the open chamber SC engines where liquid fuel is injected directly into the cylinder forming a cloud of combustible mixture which is either allowed to or prevented from filling the entire unswept volume, and the divided chamber SC engines where a prechamber is fed with fuel-rich mixture, while the intake gas is fuel lean.

The direct injection open chamber SC engines received a lot of attention, principally because their evolution required the least drastic changes of the conventional engine system. Here the control of the combustion process relies on proper mixing of the injected fuel with air before ignition takes place. In order to match adequately the fuel spray with the turbulent flow field in the cylinder, sophisticated fuel injectors, combined with careful control techniques, are required. Nonetheless, these engines have shown many advantages (41), such as:

(1) multi-fuel capability,

(2) relatively high efficiency at low and medium loads since they can operate then satisfactorily with little or no throttling,

(3) capability for high compression ratio, attained as a consequence of the Diesel-like stratification,

(4) potentially low HC emissions, and

(5) high overall A/F as well as EGR tolerance leading to low NO_x emissions

Basically, however, the operating conditions in such engines are the same as in direct injection diesels. The combustion process is accomplished by diffusion flame with all its disadvantages associated with high temperature peaks, producing high NO_x levels and excessive particulate emissions. One can offset some of these disadvantages by allowing for the formation of a more homogeneous mixture before ignition, but this tends to increase knock tendency of the system. Moreover, the high swirl rates necessary for mixture formation and control result in higher heat losses and lower volumetric efficiencies decreasing thus the specific power output.

Most of the divided chamber SC engines, such as the Honda CVCC (42), rely on the turbulent torch formed by combustion in the prechamber to initiate combustion in the main charge. This enhances the combustion of the charge, but does not eliminate knock nor prevents bulk quenching, because for complete combustion the charge must still rely on the relatively slow propagation of a flame across its entire mass. Placing the prechamber at a central location reduces the required flame travel distance and hence the combustion time, as exemplified by the recent version of the engine showing improved fuel economy without increasing pollutant emissions.

Engines of this kind are natural candidates for the application of an enhanced ignition system, such as the LAG process of Gussak (34,35,36) where, instead of a turbulent torch issuing from the prechamber, the flame is intentionally extinguished by the use of a jet with a sufficiently strong turbulent shear to break it apart. Thus the main charge in the cylinder is provided only with combustion products, seeding it with active radicals that are capable of initiating combustion simultaneously at many points. The best operating A/F range for such systems is claimed to be at the 22/1 to 30/1 level, yielding significant fuel economy gains in comparison with conventional engines.

The major attractiveness of these techniques lies in the possibility of complete elimination of knock, as well as inhibiting bulk quenching as a consequence of the significantly more rapid consumption of the mixture in the cylinder following its multi-point ignition.

PROMPT EGR IGNITION

Recently a number of studies have been reported on successful uses of combustion products for ignition and combustion control in flowing combustible mixtures (35,43), combustion cells (44,45), and, in the form of promptly recirculated exhaust gases, for the operation of two-stroke gasoline engines (19,20). It has been also speculated that in a similar way prompt EGR could be used to enhance ignition in two and four-stroke Diesel engines, reducing significantly idle knock (46).

The application of this principle was extensively studied, mainly in two-stroke gasoline engines. Under certain operating conditions, at idle to medium loads, it has been demonstrated that, with adequate mixing of a proper amount of prompt EGR (a substance readily available in a two-stroke engine by using the crankcase to compress the exhaust gases and admitting them into the cylinder through ports opened by the piston as it approached the BDC), a multitude of ignition sites are distributed throughout the compressed charge, resulting in an essentially homogeneous combustion process (viz. Fig. 9). The spark ignition system used to start the engine can be turned off and the combustion process becomes remarkable uniform, devoid of any noticeable cycle-to-cycle variation in cylinder pressure it generates. Significant reductions in (a) the duration of combustion, (b) fuel consumption, (c) noise, and (d) hydrocarbon emissions were observed and the lean missfire limit was significantly extended, typically 17/1 up to 22/1 in A/F (20).

Measurements (made by the use of multi-channel emission spectroscopy) of precombustion and combustion reactions under conventional spark ignition and homogeneous combustion process operation indicated the attainment of higher concentrations of $CH\cdot$, $C_2\cdot$, $CHO\cdot$, $HO_2\cdot$, $O\cdot$ radicals before combustion and $OH\cdot$ radicals after combustion in the latter case (19).

To sum up, the advantages of the prompt EGR ignition system, as revealed from single cylinder two-stroke engine tests reported so far, are the remarkable ability to be switched on and off, excellent part-load performance, negligible cycle-to-cycle variations, ease of operation with high excess air, virtual absence of knock, good fuel economy, and relatively low HC, CO and NO_x emission levels, albeit the latter not much below the characteristic value for two-stroke engines. The major disadvantages are relatively narrow operating range and the unconventional and unproven technology. As a consequence, future potential of this concept is quite uncertain. Most probably the best service such a system may eventually furnish is in part-load operation, providing thereby a significant assistance in improving the overall performance of the homogeneous charge engines. For us, it provides an interesting example of engineering implementation of the principle of a controlled, homoegeneous combustion system.

SUMMARY AND CONCLUSIONS

In summary the following points are noteworthy:

(1) The technology of I.C. engines has been advanced to such high levels, with all the concomitant attributes of outstanding engineering and highly developed production facilities, that one should expect its progress to continue over a long period of time, irrespectively of potential competition from alternative power plants, at least as far as automobile propulsion is concerned.

(2) The most prominent demands upon the progress of I.C. engine technology stem from the concern over the environmental impact and from the shortage of fuel (in this order!). On a long range, the tasks imposed by these demands cannot be accomplished by engine improvements derived from better engineering designs without due consideration given to all the scientific avenues available for this purpose.

(3) The extent and variety of possible engineering improvements are today so great that the technological progress can be achieved only on the basis of a well conceived sense of direction. The objective of this paper has been to provide a rationale for such a purpose. This has been derived solely from fundamental principles without much consideration given to any specific engineering, economic, sociological or political aspects, that one would have to take into account if one would wish to make a prognosis.

(4) The most promising direction in the progress of engine technology, one that could concomitantly minimize emissions, maximize efficiency, and optimize tolerance to a wide scope of fuels, is based on the concept of <u>controlled combustion</u>. According to it all the demands could be satisfied by processes occurring in the engine cylinder, without the use of a chemical processing plant in the exhaust system. The most attractive way to accomplish this is by a <u>homogeneous combustion process</u>.

(5) The target of controlled homogeneous combustion may be of immediate value by serving as an indicator of the desirable sense of direction in engine development. For, even if it is not fully attained, any progress in approaching it should be considered a technological advancement.

To sum up, the arguments presented here point out that, in response to modern demands put upon the I. C. engines, major advances in their technology should be associated with the attainment of controlled combustion, whereby fuel is thoroughly oxidized before the exhaust gases are let out from the cylinder, with proper chemical and physical means exploited then and there to inhibit concomitantly the formation of pollutants. For a pragmatically oriented reader, it should be noted that the advances we advocate can be realized either by the modification of spark-ignition engines or by the amelioration of compression-ignition engines. The engine system of the future that will incorporate the concepts we exposed can be thus thought about as a crossbreed between the Diesel and the Otto engine.

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FIGURE CAPTIONS

Fig. 1 Performance Regimes of Prime-Movers on the Plane of Specific Power and Specific Energy for a vehicle of 1 ton curb weight under steady driving conditions.

Fig. 2 Thermal Efficiencies of the Otto and the Diesel Cycles as a Function of Equivalence Ratios over a Range of Compression Ratios, r.

Fig. 3 Typical Temperature and Composition Profiles in a Diffusion Flame.
Symbols: F denotes fuel; O - oxygen; P - products, M - mass, $\tau \equiv (T - T_c)/(T_h - T_c)$, where T is the temperature, while subscripts c and h refer to the minimum (cold) and maximum (hot) limits, respectively.

Fig. 4 Composition and Temperature Profiles of a Laminar Flame in a Premixed Stoichiometric Methane-Air System.

Fig. 5 A Schlieren Record of a Turbulent Flame in Premixed Propane-Air Combustion System.

Fig. 6 Flames and the Homogeneous Combustion Process.

Fig. 7 Cinematographic Sequence of High Speed Laser Interferograms of a Homogeneous Combustion Process in an Argon Diluted Stoichiometric Hydrogen-Oxygen Mixture behind a Reflected Shock Wave (18).
Channel width: 3.8 cm
Time interval between frames: 2 microseconds

Fig. 8 Description of the LAG Process in U.S. Patent 3,230,939 (1966).

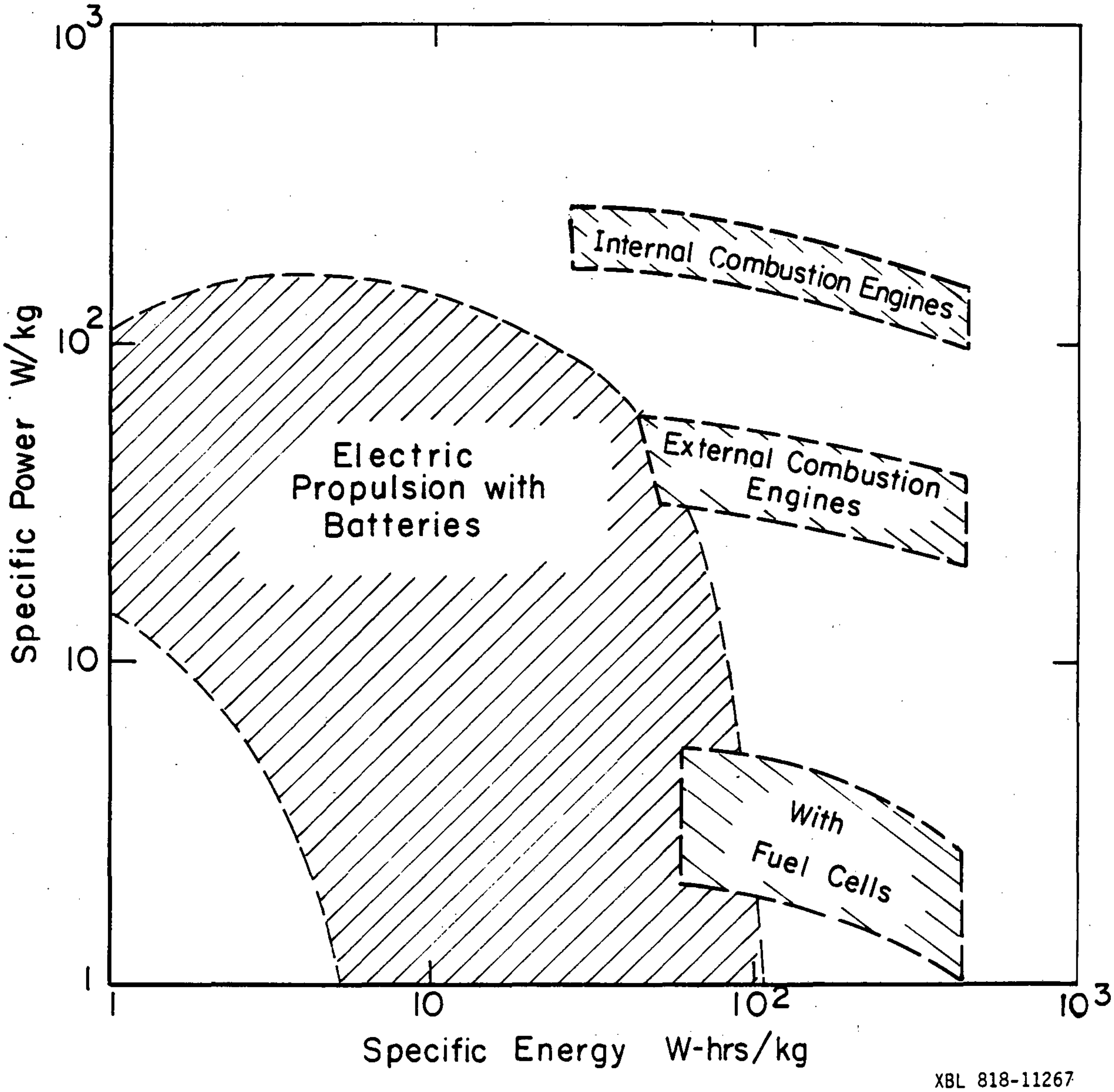
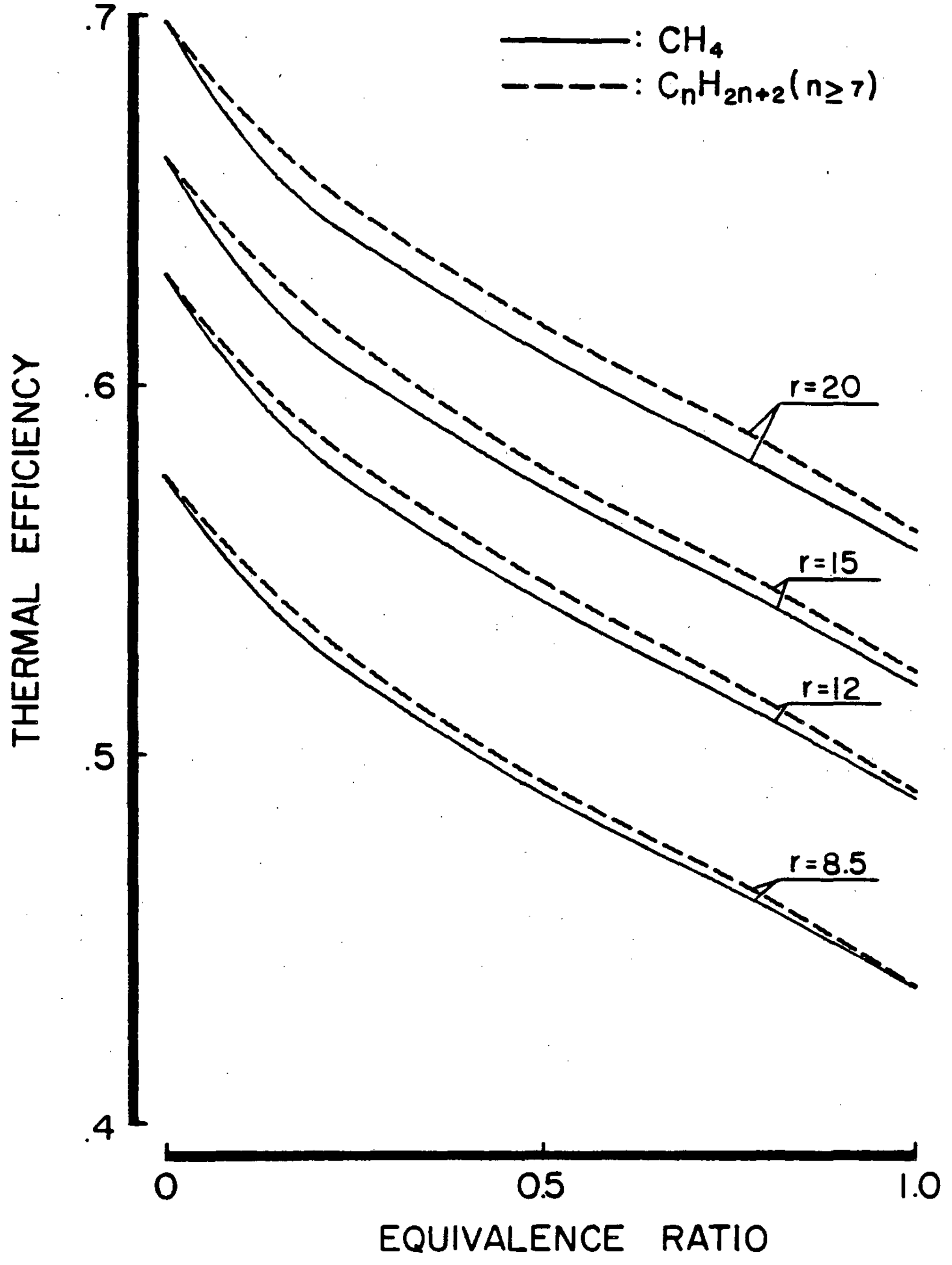

Internal Combustion Engines
External Combustion Engines
Electric Propulsion with Batteries
With Fuel Cells
Specific Power W/kg
Specific Energy W-hrs/kg
10^3
10^2
10
1
XBL 818-11267

Fig. 1



OTTO CYCLE

XBL 785-8390

Fig. 2a

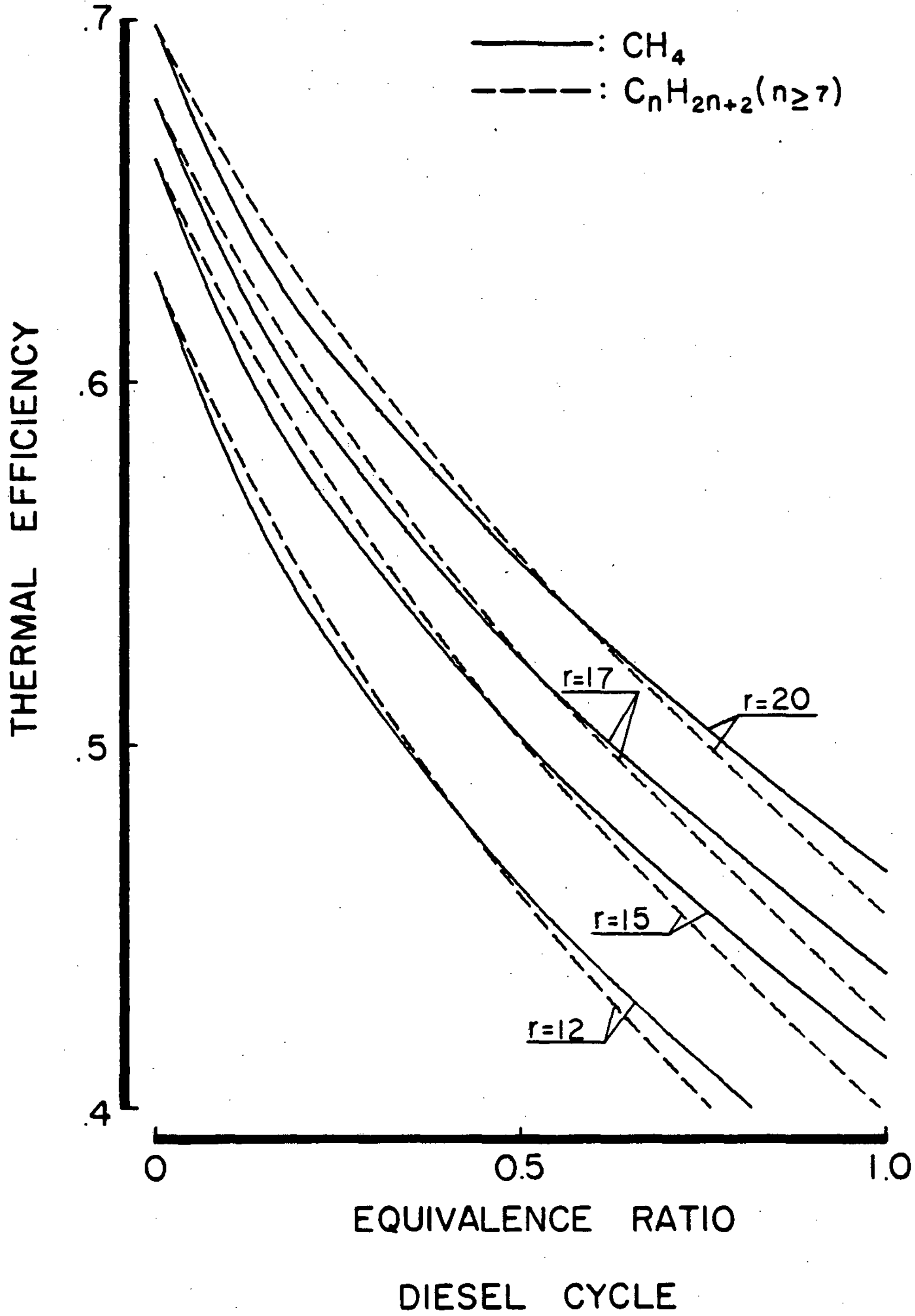


Fig. 2b

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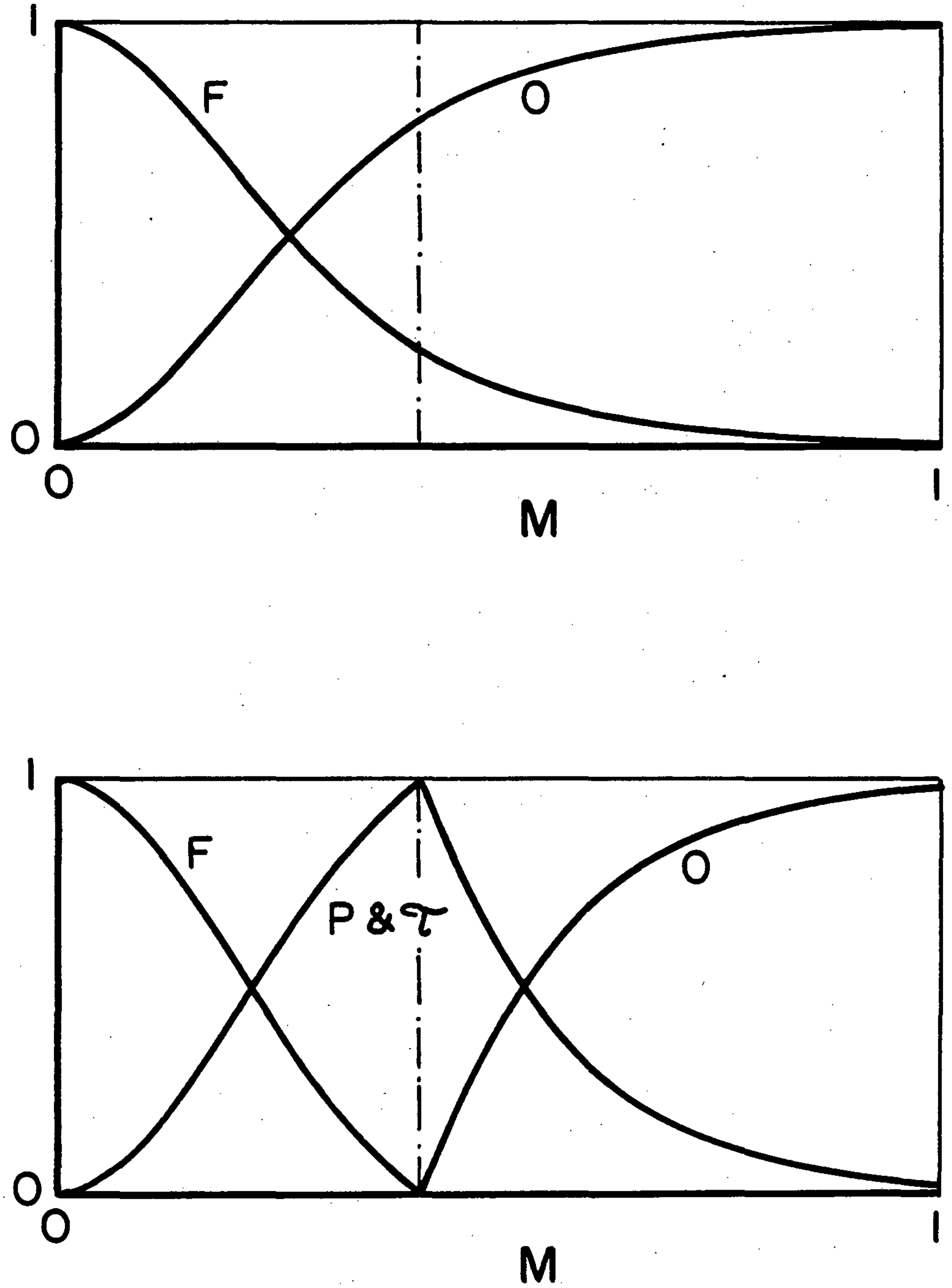

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Fig. 3

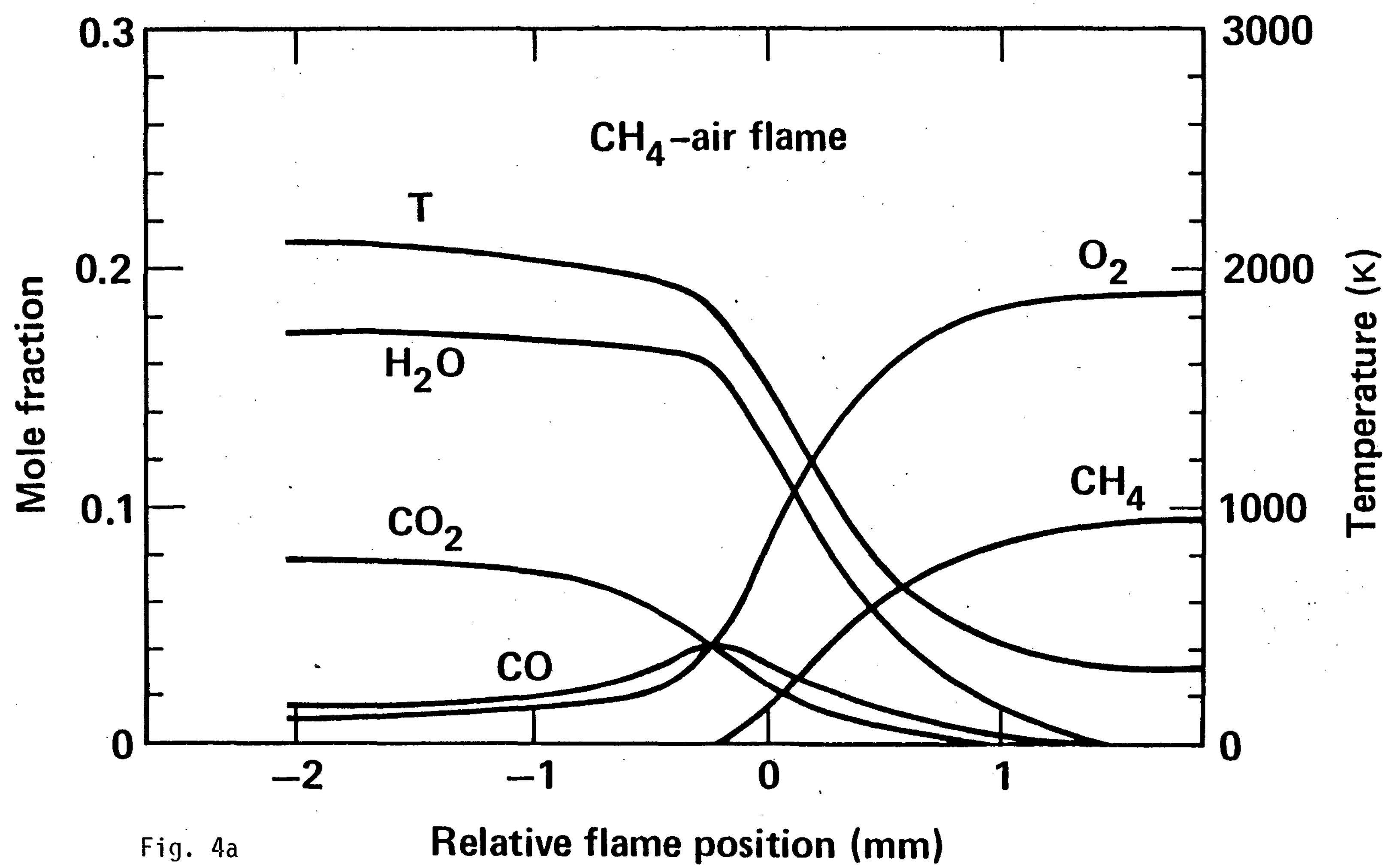

Fig. 4a

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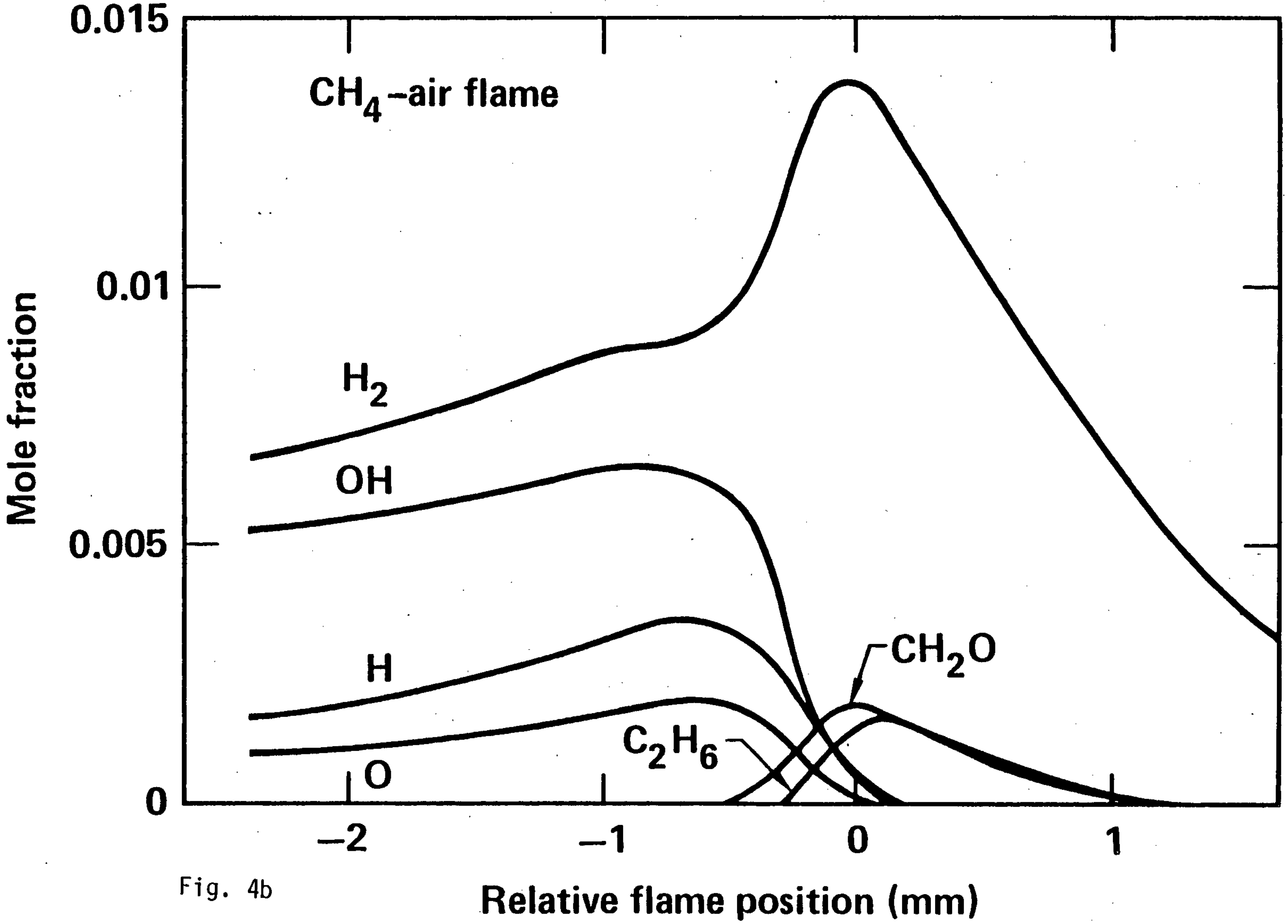


Fig. 4b

XBL 818-11431

Fig. 5 XBB 818-7487

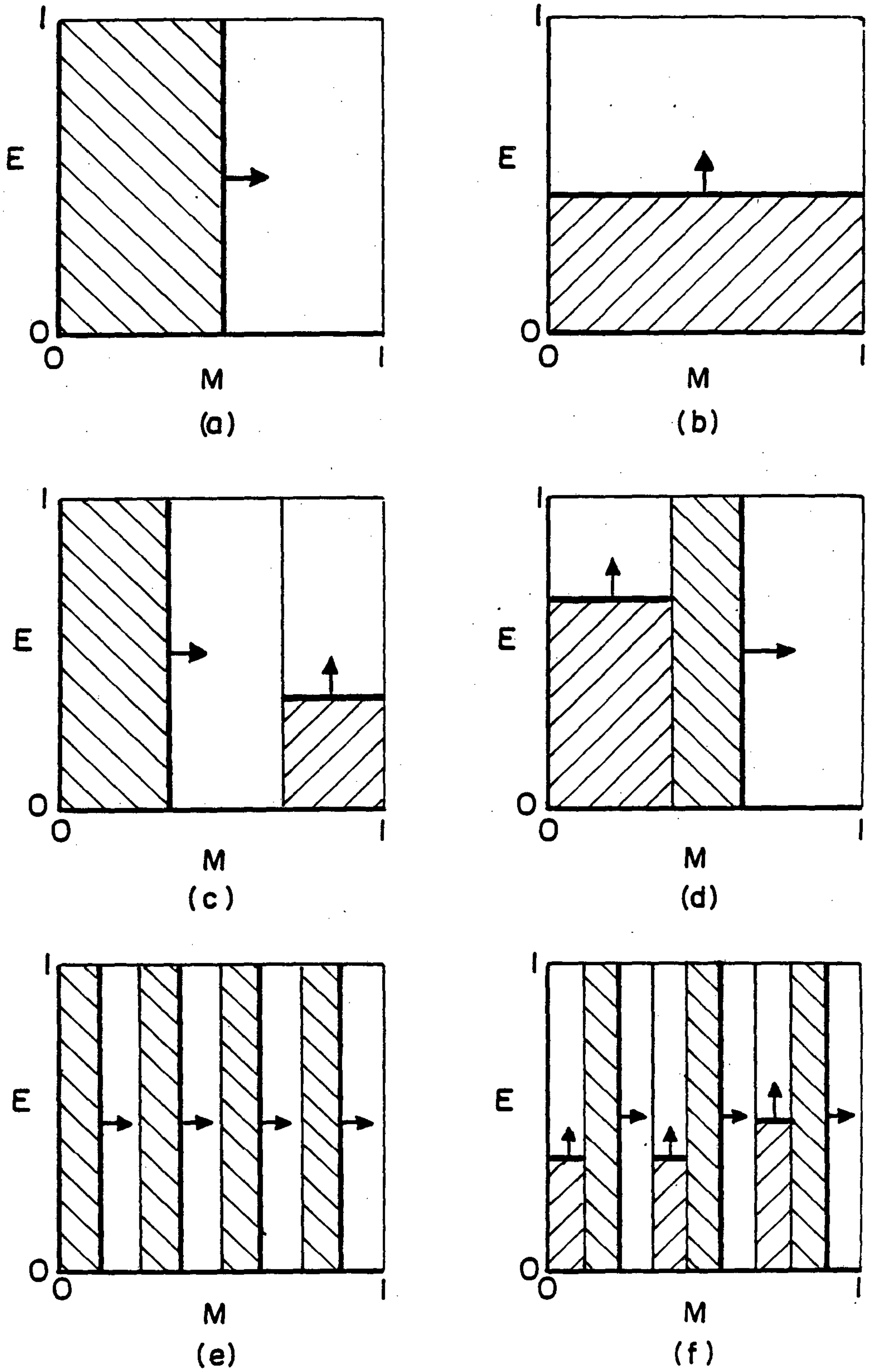


Fig. 6

XBL 818-11270

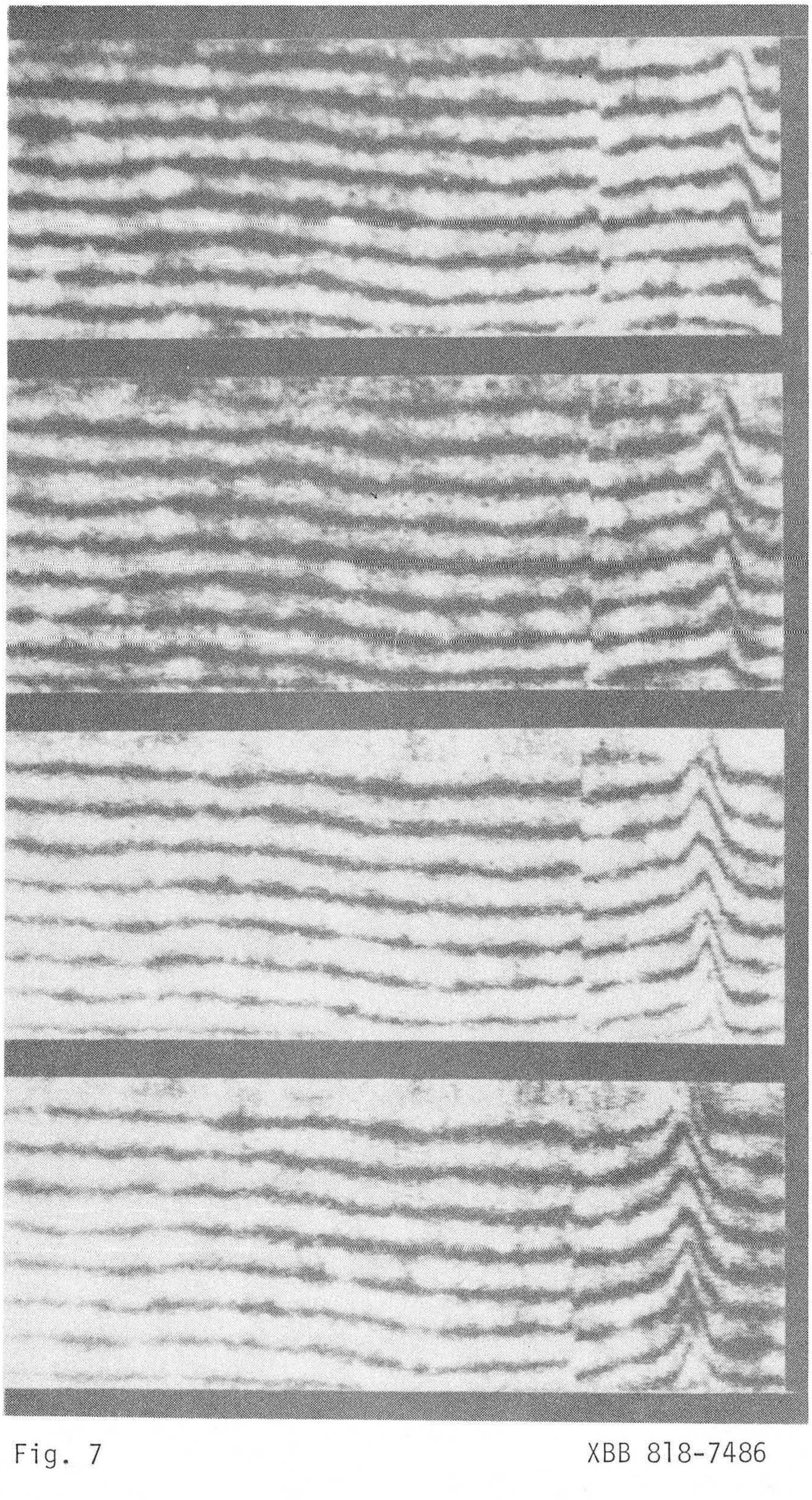
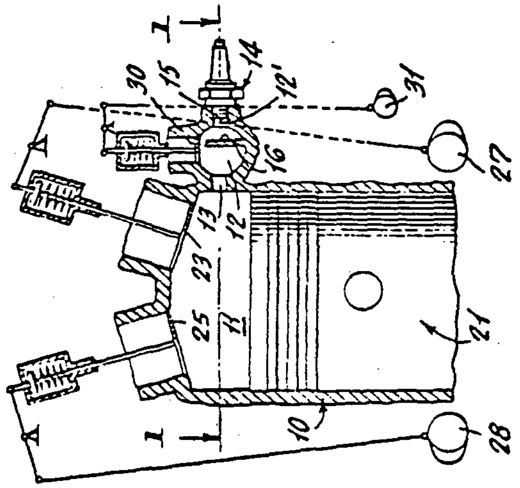


Fig. 7 XBB 818-7486

Jan. 25, 1966 L. A. GOOSSAK 3,230,939
METHOD OF PRECHAMBER-TORCH IGNITION IN
INTERNAL COMBUSTION ENGINES
Filed Feb. 4, 1963

Fig. 2.



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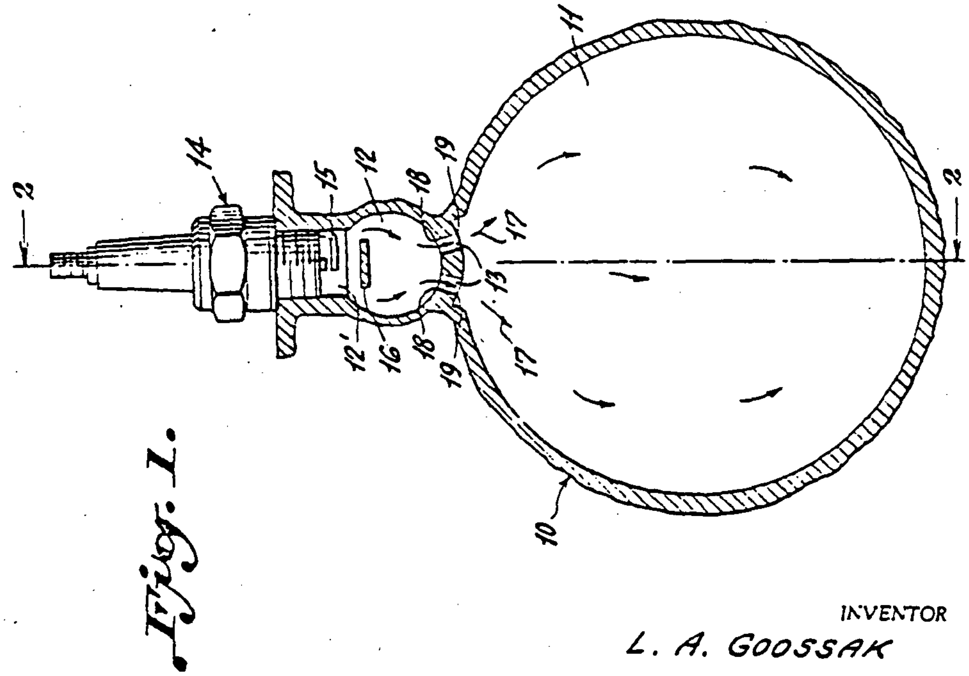

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ATTORNEYS

Fig. 8

XBL 818-11269

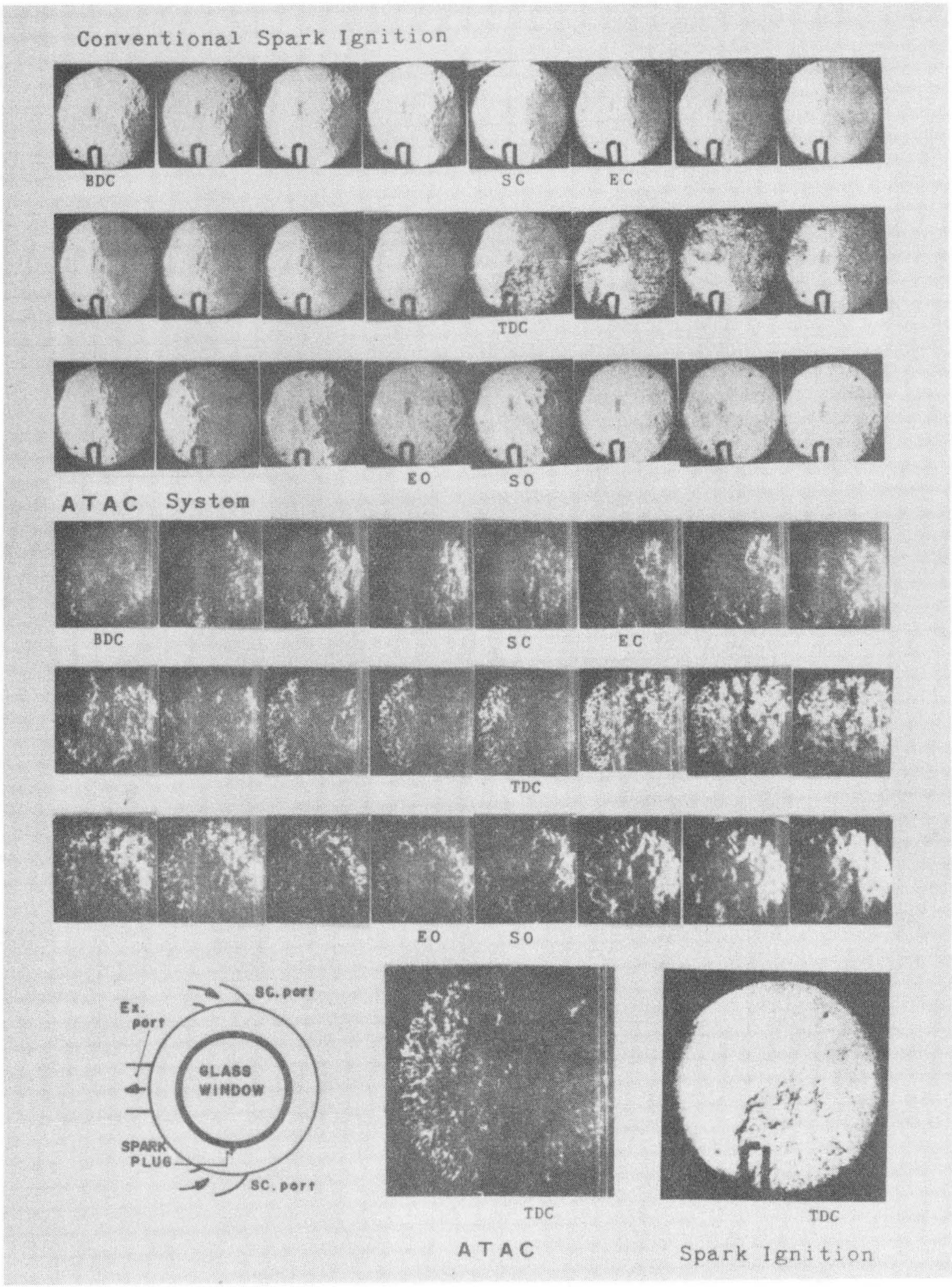


Fig. 9

XBB 818-7485

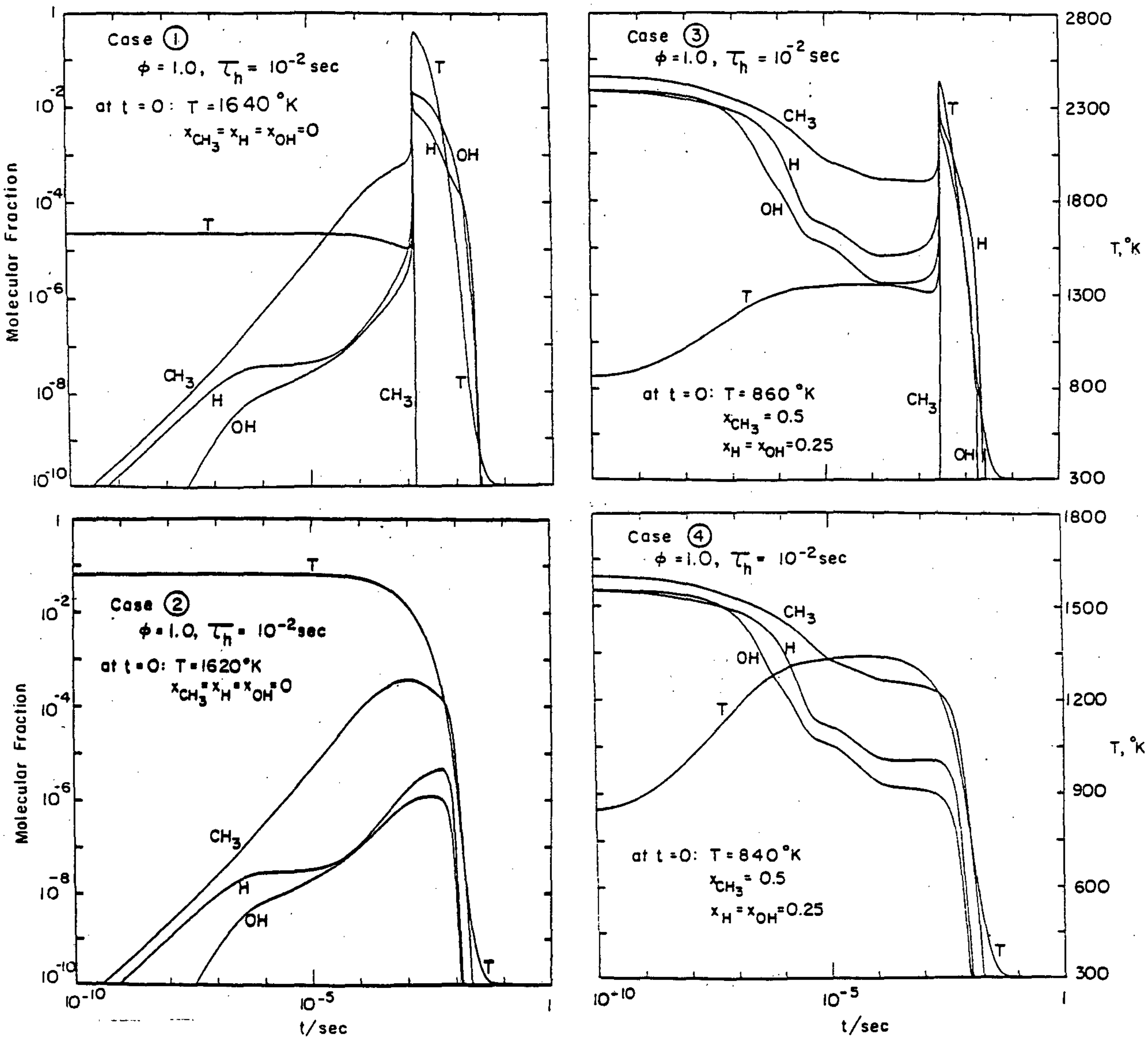
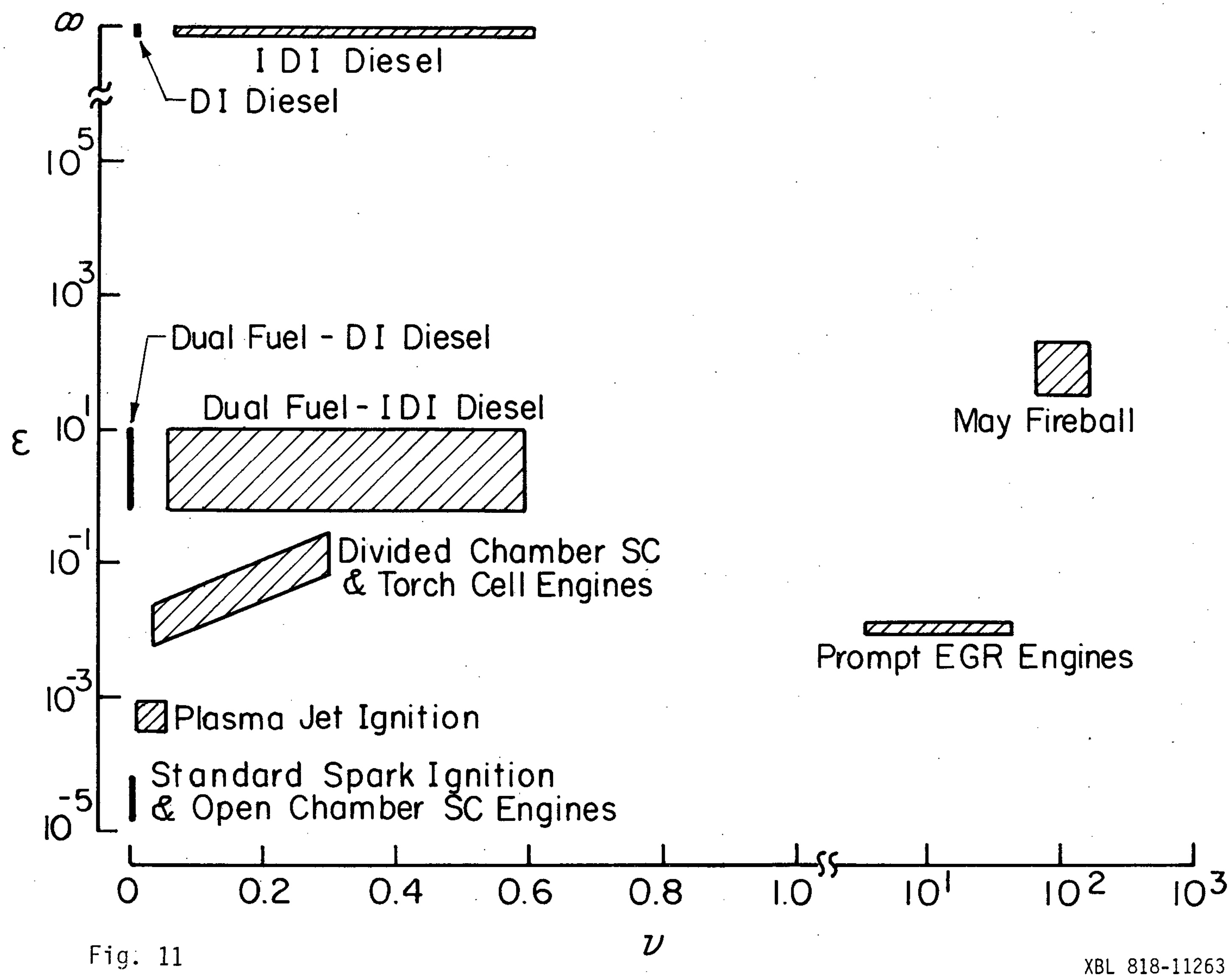


Fig. 10

XBL 7910-12572



∞
I D I Diesel
D I Diesel
10^5
10^3
Dual Fuel - D I Diesel
Dual Fuel - I D I Diesel
10^1
ε
10^{-1}
Divided Chamber SC
& Torch Cell Engines
10^{-3}
Plasma Jet Ignition
Standard Spark Ignition
& Open Chamber SC Engines
10^{-5}
May Fireball
Prompt EGR Engines
0
0.2
0.4
0.6
0.8
1.0
10^1
10^2
10^3
ν
XBL 818-11263

Fig. 11

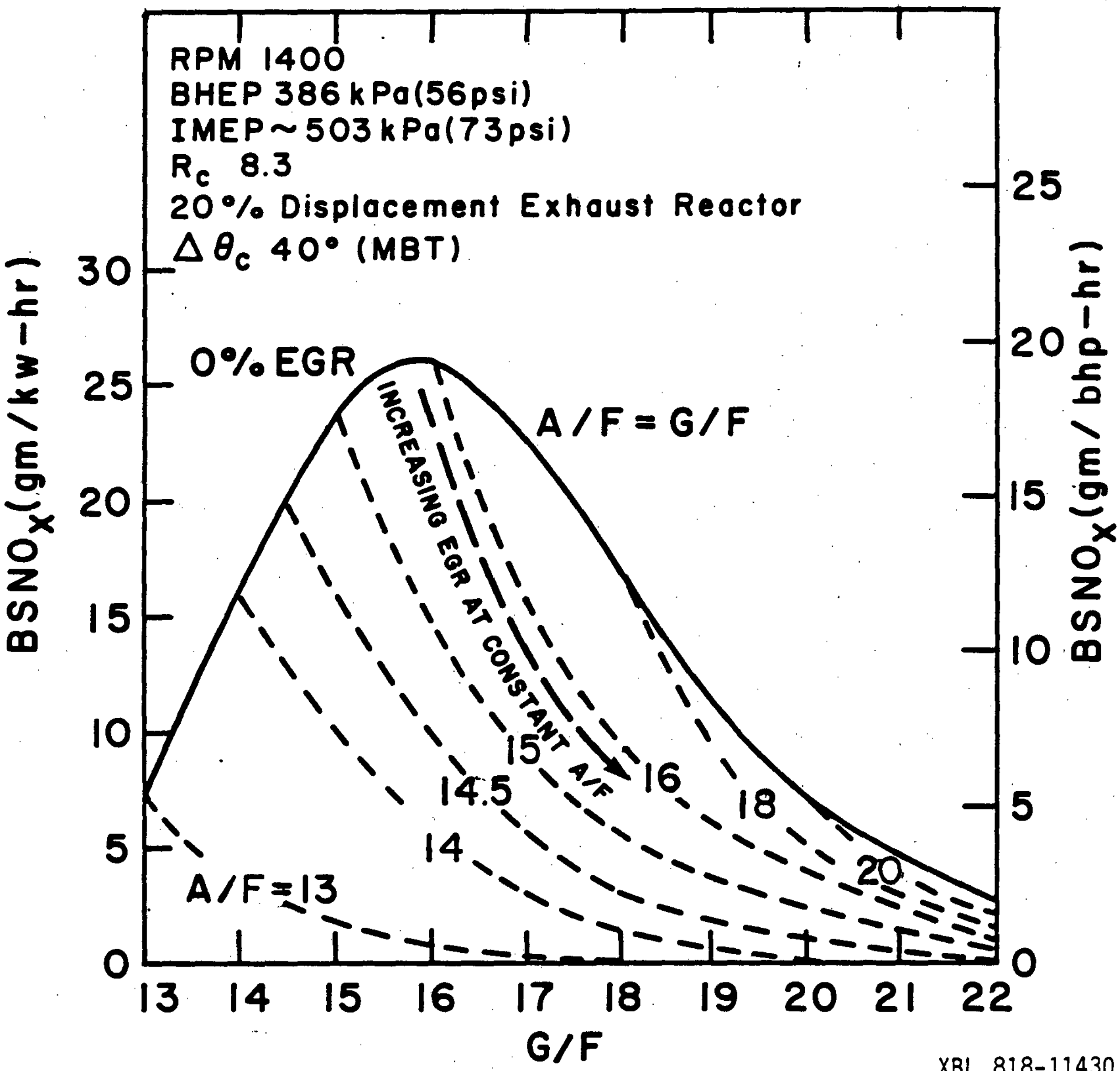


Fig. 12

XBL 818-11430

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