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RELATION OF PREIGNITION AND KNOCK TO ALLOWABLE ENGINE TEMPERATURES

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ADVANCE RESTRICTED REPORT

RELATION OF PREIGNITION AND KNOCK TO ALLOWABLE ENGINE TEMPERATURES

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SUMMARY

The results are given of an investigation of some of the limitations that now prevent increases in the temperature level of engine cylinder heads, and a review of previous work in the field is included to supplement these results. Attention was given, in particular, to the effects of fuel knock and surface ignition on cylinder temperatures and the effects of cylinder temperatures on performance. Data were obtained from a Wright C9GC air-cooled cylinder and from a Lycoming O-1230 liquid-cooled cylinder.

The analysis suggests that the relationship between the curve of ignition temperature of a fuel-air mixture plotted against density and the curve of hot-spot temperature at the time of ignition plotted against density may indicate whether dangerous preignition characteristics exist. The conclusion was reached that the general temperature level of the cylinder head may possibly be raised, even though surface ignition is present, provided that the fuels used have safe preignition characteristics. The data show that for short durations moderate audible knocking probably has little effect in creating hot spots or in causing surface ignition.

INTRODUCTION

Comprehensive investigations have been made to improve the cooling of engine cylinders in order to permit increases in engine power without exceeding fairly well-established temperature limits. Of equal importance is the fact that certain cylinder areas should not be unnecessarily cooled because the cost of cooling in high-speed aircraft may be considerable. The practice of cooling high-temperature regions by overcooling adjacent areas obviously involves a waste of power. Only by operation with all cooling surfaces raised to the maximum practical temperature can maximum cooling effectiveness be achieved.

The advantages of raising the average cylinder temperatures lie chiefly in the lowering of the power required for cylinder cooling and a possible lowering of engine friction with increased cylinder temperatures. The lowered cooling requirements reflect on the weight and drag of radiators and air scoops in the case of liquid-cooled engines and on the fin weight, cooling-air pressure drop, and cowling drag in the case of air-cooled engines. Reference 1 has shown that for a constant cylinder temperature the heat loss from an engine cylinder is approximately proportional to the 0.64 power of the indicated horsepower. With higher cylinder temperatures the amount of heat to be dissipated is reduced because the difference between the gas temperature and the cylinder temperature is reduced. Also, because the fin temperatures are higher with higher cylinder temperatures, less cooling air is required for a given heat dissipation.

A study was made of cylinder temperatures and of the effect of cylinder temperatures on engine operation for the purpose of finding the physical limitations that now prevent raising the general temperature level of the low-temperature areas of the cylinder head.

A wide variety of opinions exists as to what is the chief danger of high cylinder temperatures. Final cylinder failure obviously results from high temperatures, high pressures, or both. The initial causes of failure in question are fuel knock, preignition, and the failure of sealing devices. The failure of sealing devices refers particularly to sticking piston rings and other parts such as valves that may overheat when leakage of gas occurs. There seems to be some question as to the cycle of events leading to cylinder failure following a rise in cylinder temperatures. The particular cycle of events obviously depends upon the conditions of operation, the initial temperature of the various parts, the design features, and the fuel. Several possible sequences of events following a rise in cylinder temperatures that may lead to cylinder failure are the following:

Fuel knock --->destructive pressures or temperatures, or both

Fuel knock ⟶high temperatures ⟶preignition ⟶destructive temperatures

Fuel knock ⟶failure of gas seals (stuck piston rings)
destructive piston temperatures

Fuel knock ⟶failure of gas seals (stuck piston rings)
⟶ high piston temperatures ⟶preignition
[illegible]

Preignition --> destructive temperatures

Preignition --> high temperatures --> fuel knock --> destructive temperatures and pressures

Preignition --> high temperatures --> failure of gas seals (stuck piston rings or leaking exhaust valves) --> destructive temperatures

Failure of gas seals (stuck piston rings or leaking exhaust valves) --> high piston or valve temperatures --> fuel knock --> destructive temperatures and pressures

Failure of gas seals (stuck piston rings or leaking exhaust valves) --> high piston or valve temperatures --> preignition --> destructive temperatures

Failure of gas seals (stuck piston rings or leaking exhaust valves) --> high piston or valve temperatures --> fuel knock --> high temperatures --> preignition --> destructive temperatures

No attempt is herein made to furnish a complete answer to the questions arising from analyses of cylinder failure, but additional data are presented that may serve to clarify some of the doubtful points. Preignition is analyzed as a phenomenon independent of knock, and some of the factors that control preignition are presented. Fuel knock is discussed in its relation to engine temperatures. The relation between engine temperatures, preignition, and engine operation is then discussed to show some of the physical factors that limit the raising of engine temperatures.

Data from tests of both a liquid-cooled and an air-cooled engine cylinder are presented. The work was carried out at the Langley Memorial Aeronautical Laboratory of the National Advisory Committee for Aeronautics.

APPARATUS AND METHODS

The Lycoming O-1230 single-cylinder test setup used in these tests has been fully described in reference 2. Temperature measurements were made with thermocouples installed in the cylinder as shown in figure 1. These thermocouples were installed within 1/8 inch of the combustion-chamber surface and were peened in place.

Bendix 41C spark plugs of the bottom-seating type were used during these tests. A thermocouple was installed in the tip of the center electrode by drilling a small hole axially through the electrode. This spark plug was installed in the hole nearest the exhaust valve.

Additional piston cooling and cylinder lubrication were supplied by oil sprays from two holes, 0.040 inch in diameter, drilled in the piston end of the rifle-drilled connecting rod. These holes were arranged to direct an oil spray on the lower side of the piston crown.

The engine was cooled with ethylene glycol at a temperature of 250° F except where otherwise specified. The amount of coolant circulated was maintained constant at 84 pounds per minute.

The setup of the Wright C9GC cylinder used in some of these tests was similar to the setup of the Lycoming O-1230 cylinder. BG LS465 spark plugs were used in this cylinder. The thermocouples in the cylinder head, for measuring the combustion-chamber surface temperatures, were installed within 1/8 inch of the inside of the combustion-chamber wall. Figure 2 shows the approximate locations of these thermocouples.

PREIGNITION

Effects of Preignition

The occurrence of ignition before the spark, as may be caused by ignition from a hot spot in the cylinder, may lead to an extremely early time of ignition with certain fuels and cause high pressures and destructive temperature conditions. This effect is shown by the results reported by Spencer in reference 3, in which the duration in the cycle of peak pressures and temperatures was prolonged several times the normal value after the onset of heavy preignition. Figure 3 shows the effect of early ignition, obtained by advancing the spark, on the cylinder temperatures of a Wright C9GC air-cooled cylinder. This curve was obtained for operation without knock. The spark was advanced until the running became very rough. The data indicate that the rate of temperature rise of the cylinder head increases with advance angle.

In cases of severe preignition, the rate of heat application to the combustion-chamber walls may reach several times the normal rate of heat application. As conventional aircraft-engine pistons are more or less indirectly cooled and, furthermore, as the heat capacity is small, failure of the pistons usually occurs soon after the onset of well advanced preignition and usually before cylinder-head failure.

Figure 3 indicates also some of the effects on engine temperatures of changing the compression ratio. Although most of the head temperatures increased as the spark was advanced, the barrel temperatures first decreased, reached a minimum, and then increased. As the spark is advanced from some low value to the optimum value, the effect is the same as the effect of increasing the expansion ratio or, normally, the compression ratio. In this case, as the spark was advanced from 10° to 20° B.T.C., which was the optimum spark advance, the cycle efficiency increased. This increase resulted in higher peak temperatures but in lower average gas temperatures. In this range of spark advance, the peak temperatures were more effective in controlling most of the cylinder-head temperatures and, as a result, these cylinder-head temperatures increased. On the other hand, the barrel temperatures were more influenced by the average gas temperature during the expansion stroke and therefore decreased. The exhaust-valve-guide temperature was influenced mainly by the exhaust-valve temperature, which in turn was influenced mainly by the expanded-gas temperature. The exhaust-valve-guide temperature therefore behaved more like the lower barrel temperatures. As the spark was advanced beyond the optimum position from 20° to 25° B.T.C., all engine temperatures increased because combustion was taking place too early in the cycle. There was therefore no further increase in the effective expansion, and the sum of the heat losses during the combustion process and expansion stroke became successively greater. The conclusion drawn from these data is that an increase in the compression ratio of the engine may increase all the cylinder-head temperatures except those influenced mainly by the exhaust-gas temperature and will decrease the barrel temperatures.

Preignition and Knock

Preignition is sometimes accompanied by fuel knock and is often the forerunner of fuel knock. As will be shown later, however, there is considerable evidence to show that the occurrence of moderate audible fuel knock for short durations may have only a slight effect in causing surface ignition. In this respect, it has been pointed out in references 4 and 5 that preignition and fuel knock are separate phenomena and should be treated as such. Apparently, much misinformation exists as to the relative effects of fuel knock and preignition on piston and cylinder failures.

In a great number of cases of engine failure, the primary cause of failure has undoubtedly been erroneously attributed to fuel knock rather than to preignition. This conclusion has, in a number of cases, followed from the fact that the addition of tetraethyl lead (TEL) to the fuel prevented a recurrence of the difficulty. This conclusion is justified, provided that the use of tetraethyl lead affects only the knocking properties of the fuel. In reference 6, Heron states as follows: "Some evidence obtained in multicylinder aircraft engines suggests that tetraethyl lead has an effect in suppressing overheating of the cylinder unit that is out of proportion with its properties of suppressing audible knock and controlling the rate of pressure rise. Tetraethyl lead in quite small concentrations has a remarkable effect in suppressing preignition and autoignition. This possibly suggests, in view of the pronounced effects of lead in allowing increase of output in engines of high cylinder-wall temperature, that the tendency to preignition is one of the major causes limiting the usefulness of a fuel in such engines."

This effect of lead as described by Heron is further substantiated by the data of reference 7, in which it is shown that lead increased the radiant energy of combustion. The results reported in reference 8 showed that increasing the octane number of a fuel by adding tetraethyl lead or by adding benzol also increased the preignition limit. Additional data on the effect of lead on cylinder temperatures are shown in figure 4. These data were obtained under closely controlled conditions for the purpose of showing the effect of lead on temperatures. It will be noted that, although the temperatures obtained with the leaded fuel are generally lower, the effect is relatively small. The addition of lead had little effect on the temperature of the spark-plug electrode and the average barrel temperature.

A consideration of the theory, which states that the occurrence of knock depends upon the density and temperature of the last portion of the charge to burn, indicates that preignition which causes the end gas to burn nearer top center also tends to foster fuel knock whereas preignition which causes the end gas to burn well ahead of top center may lessen the possibility of fuel knock. This theory therefore furnishes one explanation of why preignition is often accompanied by knock and why, in other cases, in which preignition may be severe enough to cause immediate engine failure, fuel knock may not be present. A specific instance of this nature is referred to in reference 8. The type of fuel obviously has a marked effect on the preignition characteristics.

Effect of Fuels on Preignition Characteristics

The use of certain fuels is known to cause higher surface temperatures for the same power output than others. This characteristic is generally found in the aromatic fuels. In this group benzene causes particularly high temperatures. Additional data on this subject are shown in figure 5, which compares the surface temperatures obtained with S-1 and benzene. Although the heat content of benzene is lower than that of S-1, the combustion temperatures are higher. (See reference 9.) Studies made of the indicator cards for S-1 and benzene show no essential difference in the pressures obtained (reference 3).

Run-Away Preignition

Engines operating with certain fuels such as benzene have a tendency to "run away" once preignition is encountered; whereas with other fuels, such as isooctane, operation is in general quite stable during self-igniting operation. The use of such fuels as benzene is dangerous because with these fuels preignition may become severe enough to wreck the cylinder in such a short time that no appreciable rise in temperature or lowering of the engine output is noticed by the operator. As previously noted, the reason for burned pistons, rings, and cylinder parts can be understood from a consideration of the long duration of the peak pressures and temperatures that results when ignition occurs early on the compression stroke. The inertia of the rotating parts of high-speed single-cylinder engines and the power of the remaining cylinders that are functioning normally in multicylinder engines tend to minimize the immediate effect of power drop caused by extreme ignition advance of one cylinder.

Factors Controlling Stability of Preignition

The question of whether surface ignition will result in a stable ignition advance is apparently dependent on the fuel and the temperature characteristics of the surface causing ignition. Instability of ignition advance with certain fuels occurs with rising surface temperatures. Reference 10 has shown that, once ignition is started in a given locality, the surface of that locality will show an immediate and marked rise in temperature; that is, when a hot spot becomes hot enough to cause ignition, its temperature is further increased. Under certain conditions it is this increase in temperature of the hot spot that leads to a rapid advance of the preignition and sudden engine failure with certain types of fuel. Inasmuch as spark plugs are already sources of ignition, their initial rise in temperature, caused by their becoming sources of ignition, has already taken place. Further increases in spark-plug temperature must come from changes in operating conditions, such as power increases, or from an advance in the time of ignition in the cycle.

The variation of ignition lag and ignition temperature with pressure change possibly determines whether the ignition characteristics of a fuel are dangerous. The relationship of the curve showing the variation of fuel-ignition temperature (or hot-spot temperature required for ignition) with density, for constant ignition lag (illustrated in fig. 6(a)), with respect to the curve showing the variation of the temperature of the igniting surface (hot spot) with the gas density at the time of ignition (illustrated in fig. 6(b)) possibly determines the time of occurrence of surface ignition in the cycle. The ignition curve of positive slope with the ignition zone at the lower rather than the higher temperatures is justified by data presented in a subsequent figure. The hot-spot temperature curves are drawn with different slopes to show a series of possibilities without discussing the reasons for the particular slopes.

Inasmuch as these curves (figs. 6(a) and (b)) may be plotted to the same scale, they may be superimposed as shown in figure 6(c). For the first example, slope a is positive and slope b is negative. Suppose the cylinder is operating with a hot-spot temperature at 1. This hot-spot temperature causes ignition at the density represented by 2. Ignition at this point causes the hot-spot temperature to rise to 3, which causes ignition at 4. The hot-spot temperature then drops to 5 and causes ignition at 6. This cycle repeats, and a condition of equilibrium is reached at the point of intersection of the two curves. This point represents a density corresponding to an ignition before the normal spark ignition. The degree to which this preignition has advanced depends upon the conditions of operation and the fuel. In some cases it may stabilize a very short distance ahead of the spark and cause no bad effects, while in other cases it may not stabilize until it has advanced a great distance ahead of the normal spark, causing damage to the engine.

For the purpose of this analysis, it must be assumed that the hot-spot-temperature curve represents the temperature after the hot spot has become a source of ignition; that is, after its initial rise in temperature due to its becoming a source of ignition has taken place. This consideration, of course, applies only to hot spots other than the spark plugs.

In actual engine operation the path of the hot-spot temperature would not be shown in figure 6(c) because the temperature change from cycle to cycle would be very small. Instead, the ignition would progress up the ignition curve in very small steps until the intersection was reached, as shown in figure 6(d). A mathematical treatment of these phenomena made by Dr. David T. Williams of the thermodynamics division appears in the appendix. Figure 6(c) illustrates the point to better advantage and represents the final condition of stability, in which the steps shown are infinitesimally small. In figures 6(e) to 6(i) this method of illustration will be used.

Figure 6(e) shows a case in which slope a is negative and slope b is positive. As the hot-spot temperature is increased from 1 to 2, the point of ignition is moved to 3. This movement causes a hot-spot temperature at 4, which causes ignition at 5, and so on. A condition of equilibrium is reached at the point of intersection of the two curves. Here again, the degree of preignition encountered before equilibrium is attained depends on the conditions of operation and the fuel.

If slopes a and b are both negative, and if slope b is greater than a, an unstable condition exists. In figure 6(f), a rise in temperature of the hot spot from 1 to 2 causes ignition to occur at 3. This earlier ignition causes the hot-spot temperature to rise to 4, which causes more advanced ignition in the direction of 5. In this way, the ignition occurs earlier and earlier in each succesive cycle and run-away preignition takes place, causing probable damage to the engine. If preignition starts by a rise in hot-spot temperature from 1' to 2', however, the point of ignition is unstable in the other direction; that is, the preignition retards toward top center until it no longer exists. The direction in which preignition progresses, then, depends on which side of the point of intersection it is started.

Figure 6(g) illustrates a case similar to figure 6(f) except that slope a is greater than slope b. From a line of reasoning similar to that used for figure 6(f), it is seen that preignition in this case will become stable at the point of intersection of the two curves, regardless of the side on which it is started.

If slopes a and b are both positive and if slope b is greater than slope a, an unstable condition exists somewhat similar to that of figure 6(f). The illustration of figure 6(h) shows this condition. The direction of progress of the preignition depends on which side of the intersection it begins.

In figure 6(i) is shown the case in which slopes a and b are both positive, and slope a is greater than slope b. It is readily apparent that the preignition becomes stable at the point of intersection of the two curves, regardless of the side on which it is started.

The foregoing analysis shows that the time of occurrence of surface ignition will depend upon the following conditions:

Condition (1). - If the slope of the curve of fuel-ignition temperature against density is opposite in sign to the slope of the curve of surface temperature against density at the time of ignition, the time of ignition will reach some stable point and will neither advance nor retard from this point.

Condition (2). - If the slopes of the curve of fuel-ignition temperature against density and the curve of surface temperature against density at the time of ignition are both negative or both positive, and if the slope of the curve of fuel-ignition temperature against density is the smaller, the time of ignition will either advance or retard in an unstable manner. When both curves are negative and the slope of the curve of fuel-ignition temperature against density is the smaller, the direction in which this unstable ignition time moves will depend upon whether the hot-spot temperature is greater or less than the temperature at the point of equal hot-spot and fuel-ignition temperature. If the hot-spot temperature is greater than the temperature at this point of equal temperatures, the time of ignition will advance in the cycle.

Condition (3). - If the slopes of the curve of fuel-ignition temperature against density and the curve of surface temperature against density at the time of ignition are both negative or both positive, and if the slope of the curve of fuel-ignition temperature against density is the greater, the time of ignition will reach some stable point and will neither advance nor retard from this point.

Ignition-temperature curves for several fuels are shown in figure 7, as reproduced from reference 11. Figure 8 shows ignition-temperature curves for isooctane and benzene taken from figure 7. The ignition lag was held constant for the curve for isooctane at a value of 1 second. Ignition-temperature curves for lags corresponding with those encountered in high-speed engine operation would probably lie to the right at higher pressures and upward at higher temperatures. It is apparent that, for any given hot spot, the stability of preignition largely depends upon the position of the ignition with reference to the ignition-temperature curve. The continuously changing slope of the ignition-temperature curve changes the relationship between itself and the hot-spot-temperature curve as the temperatures or densities are changed. For this reason operation in an unstable part of the ignition-temperature curve soon advances or retards the preignition to a stable part of the curve unless the operation is on either end of the curve, in which case it may or may not become stable, depending on the direction of initial instability.

Unlike the ignition curve for isooctane, the ignition curve for benzene extends upward to the left, as shown in figure 8. It will be noted that near one end of the curve the ignition lag is 10 seconds and near the other end it is 30 seconds. A curve for constant ignition lag would probably have a small slope. A curve for very short ignition lags, such as are encountered in engine operation, would probably lie to the right at higher pressures and upward at higher temperatures. Because the slope of the ignition curve for benzene is small, it is probable that a curve of hot-spot temperature with a negative slope would have a greater slope than the ignition curve, and run-away preignition would take place in most cases.

The foregoing discussion may explain why engines have suddenly failed while operating with benzene when surface ignition has taken place but have remained intact while operating under identical or higher temperature conditions with isooctane. As the ignition-temperature curve for benzene is also characteristic of that for methane, acetone, propylene, These data show that the temperatures of the combustion-chamber surfaces change very appreciably with power and mixture even though the temperature criterion is maintained constant. This trend is particularly noticeable for the areas having some degree of thermal isolation, such as the spark-plug electrodes.

The most valuable criterion of temperature control, when the danger of preignition is considered, is obviously the highest temperature in the combustion chamber. This point, however, may not remain the highest during all conditions of operation. For example, the exhaust valve may have a higher temperature than the spark-plug electrodes only during very lean-mixture operation when the over-all burning period is greater. Unfortunately, the hottest areas in the combustion chamber present the greatest difficulties in taking temperature measurements. The same obstacles that make these surfaces difficult to cool also make measurement of temperature difficult.

The abnormally high temperatures in the combustion chamber that result from deterioration may not be reflected to any great extent by the temperature criterion. For example, the overheating of exhaust valves, as caused by excessive wear of the guides, may not be indicated to any appreciable extent by the temperature of the rear spark-plug boss. In experimental test work, the temperature of the exhaust-valve guide may prove to be a valuable supplement to the temperature of the spark-plug boss or of the coolant.

Spark-plug temperatures. - In present-day engines, the spark plug is probably the most vulnerable part of the combustion chamber as concerns preignition. The problem of spark-plug cooling is one of maintaining sufficiently high insulator and electrode temperatures to avoid fouling at low outputs and of preventing insulators and electrodes from reaching preignition temperatures at high outputs, and the solution for higher outputs lies in obtaining adequate temperature control or in obtaining spark plugs resistant to fouling at lower temperatures.

Test results in figure 10 show the variation of electrode temperature with indicated mean effective pressure. If a temperature of 500° F is assumed as the lowest safe nonfouling temperature, the lowest permissible indicated mean effective pressure is 50 pounds per square inch. Likewise, if 1200° F is assumed as the maximum safe electrode temperature, the highest allowable indicated mean effective pressure will be 264 pounds per square inch. The lowest allowable temperature is obviously a function of the duration of operation and the amount of oil and fuel being consumed and, likewise, the upper temperature limit is a function of durability of the spark plug at the elevated temperature as well as a function of pre-ignition. From the curves of figure 10, it is apparent that a small increase in the allowable minimum indicated mean effective pressure, which would permit the use of a colder operating spark plug, would greatly increase the allowable maximum indicated mean effective pressure. Likewise, any improvement in the plug insulator that will permit a temperature increase to approach the surface ignition limit (1300° to 1600° F) will greatly extend the indicated-mean-effective-pressure range. The practical temperature limit of mica plugs is approximately 1000° F, as the mica tends to dehydrate above this temperature. This dehydration causes it to expand and to become brittle with consequent leakage (reference 13). Tests were recently made at Langley Memorial Aeronautical Laboratory to ascertain the effect of small controlled gas leaks through the mica on the electrode-tip temperature (reference 14). These results showed a considerable temperature rise for small amounts of gas leakage.

Other factors remaining constant, the temperature of any area in the combustion chamber is an indication of the resistance to heat flow from that area in the combustion chamber to the cooling means. A high temperature at the spark-plug electrode indicates poor heat flow from the electrode to the cylinder wall. The coolant temperature would therefore be expected to have little effect on the temperature of the spark-plug electrode. This effect is indicated in figure 11(a), in which the temperature of the spark-plug electrode tip changes approximately 0.34° F at a coolant temperature of 250° F for 1° F change of the coolant temperature.

Highly heated areas, which are subject to high rates of heat application caused by the flow of hot gases but which are not thermally isolated, will obviously respond more to change in jacket temperatures than will the spark plugs. This condition is probably true of the exhaust valves. The small effect of jacket temperature on spark-plug temperature illustrates the futility and cost of attempting to cool hot spots in the combustion chamber by lowering the temperature level of the entire cylinder head. From this line of reasoning, it would appear that, in order to cool hot spots, every effort should first be made to improve the thermal conductivity from the hot spot to the cooling medium before increased cooling for the entire head is attempted.

In reference 10 it was shown that little change in electrode temperature was effected by water-cooling the external portion of the spark plug. This result indicates that little extension of the operating range of spark plugs can be expected by external cooling unless the temperature gradient between the electrode tip and the external portions is reduced.

Temperature reductions in spark-plug electrodes of as much as 400° F have been obtained (reference 10) by cutting out the ignition from the spark plug on which the measurements were being taken. These data suggest a method of reducing spark-plug temperatures that would obviate the fouling difficulties. At high outputs two pairs of spark plugs might be used alternately, one pair for one cycle, the other pair for the following cycle, etc. At idling speeds, a single plug could be fired during each cycle. If such an arrangement were practical, it would extend the upper power limit by an appreciable amount.

The fuel-air ratio has a marked effect upon the temperature of combustion-chamber surfaces, especially spark-plug electrodes and other areas that have any degree of thermal isolation. This effect is so great, in fact, that the temperature of spark-plug electrodes has been used as an indication of the value of the fuel-air ratio. The curves of figure 4(a) show this effect for the Wright C9GC cylinder. From these curves it is apparent that the problem of spark-plug cooling is greatest near the stoichiometric mixture.

Exhaust valve and seat temperatures. - Few data are available concerning the temperature of sodium-cooled exhaust valves at high outputs. Although no data were taken of exhaust-valve temperatures during these tests, information was obtained on the temperature of the exhaust-valve guide and of the metal between the valves adjacent to the valve seat. As the heat flow from the valve is conducted through these areas, the temperatures of these areas furnish a fairly good index of the exhaust-valve temperature.

The temperatures of sodium-cooled hollow-head exhaust valves have been presented in reference 15. These measurements show that, in normal operation, the temperature of the center of a large conventional valve of the type used in an air-cooled engine may reach a value of 1300° F with a temperature of 1100° F at the seat and that the valve stem at a location near the guide, when closed, may reach a temperature of slightly less than 900° F. As exhaust valves are scrubbed with the hot gases on all sides, the rate of heat application is very high, and the temperature attained may not be an indication of the degree of thermal isolation of the valve from the cylinder head.

In reference 16, changing from a hollow-head valve without sodium to one with sodium raised the valve-guide temperature approximately 110° F. As the flow of heat from the exhaust valve to the cylinder head is by conduction, a change in valve-guide temperature or valve-seat temperature will effect a somewhat smaller change in the valve temperatures under constant power conditions. Furthermore, figure 11(b) shows that an appreciably large change in coolant temperature is required to cause a given change in the temperature of the head adjacent to the valve seat. The change in the temperature of the center of the head amounts to approximately 0.61° per 1° change in coolant temperature at 250° F coolant temperature.

The rate of rise of temperature of exhaust valves with increase in power should be determined. The slope n of the temperature curve of the exhaust-valve guide as shown in figure 9 indicates that the exhaust-valve-guide temperature rises much faster than the temperature of the head between the valves. The exhaust valve probably behaves similarly.

Very little information is available to show how closely present exhaust valves are operating with respect to preignition limits. Exhaust-valve temperatures are at present probably too close to the maximum allowable limit to permit any increase in the cylinder-head temperature level. This condition particularly applies to the leaner mixtures, as is indicated by the rapid rise in exhaust temperatures shown in figure 12(a) and in the temperature of the guide shown in figure 12(b). A more direct method of cooling the exhaust valve would permit an increase in the cylinder-head temperature level.

In reference 17 the statement is made that a limitation to exhaust-valve temperatures is the rapid oxidation of valve steels and stellites in the presence of lead oxide at temperatures above 1380° F. Below this temperature, lead attack will not occur. A mixture of lead oxide and iron oxide was found to form an insulating layer that tends to foster preignition. These difficulties apparently limit the practical temperature of the exhaust valve to around 1350° F for satisfactory operation when available materials are used.

Little is apparently known concerning the effect of gas leakage between the exhaust-valve stem and the guide. Although leakage at the seat may cause erosion and warping, quantitative data are lacking. Worn valve guides are known to cause higher valve temperatures. Any appreciable flow of hot exhaust gases through the guide, as might result when a large difference in pressure exists between the exhaust port and the rocker-arm housing, will probably result in a very appreciable rise in exhaust-valve temperature. Such a condition would be obtained with the use of an exhaust turbine.

Effect of sharp edges and carbon deposits on preignition. - Sharp edges, such as exposed spark-plug threads, are commonly believed to cause surface igntion under certain conditions. This statement applies to sharp edges of materials having melting points above that of the ignition temperature of the fuel. In this respect little difficulty has been experienced at LMAL with exposed threads of aluminum alloy, owing to the fact that these threads soon melt away when overheating occurs. The best remedy for exposed threads is obviously their elimination. This statement applies particularly to steel threads.

Information concerning the effect of carbon deposits on preignition is meager. Tests at LMAL have shown that graphite tends to burn away at a temperature of 900° F. Carbon deposits will most likely burn away at about this temperature. If so, for operation at a constant load, carbon deposits are not likely to be a direct source of ignition for any length of time. Reference 18 has shown that carbon deposits vary in thickness inversely as the severity of operating conditions. The thickness of the deposits increases until the decrease in thermal conductance through the carbon layer is sufficient to bring the outer surface to the ignition temperature of the carbon. As the thickness of carbon deposits increases with decrease in temperature, extended periods of operation at reduced loads and low cylinder temperatures tend to build up relatively heavy deposits. A sudden increase in power may then cause a momentary period of preignition during the time required to burn away the carbon.

If carbon deposits are a source of trouble as caused by preignition, the solution is evidently not one of better cooling. Increased cooling only tends to increase the thickness of the carbon layer and thus effectively raises the compression ratio. The solution, however, may be one of changing the character of the carbon deposits so that ignition of the carbon will take place at a lower temperature. A better solution, however, would be to provide a fuel of such characteristics that surface ignition can be tolerated without fear of damage to the engine.

FUEL KNOCK

Effect of Fuel Knock on Cylinder Temperatures

The rise in cylinder-head temperatures with the onset of fuel knock is often mentioned in the literature, but few recorded data demonstrating this fact have been presented. Figures 13 and 14 show the engine performance and cylinder temperatures as a function of power output when engine operation, originally well below the audible knocking condition, was increased to give audible knock.

In these figures and in the discussions presented herein the following definitions hold true: "Audible pinking" refers to the very first knock that can be detected by ear. "Audible knock" refers to a knock level obtained by increasing the inlet pressure by 3 or 4 percent above that for audible pinking. "Violent knock" refers to the knock level obtained by increasing the inlet pressure by approximately 20 percent above that for audible pinking.

The three fuels used in the tests of figures 13 and 14 were: a 100-octane reference fuel (S-1), a 92-octane reference fuel (90 percent S-1 plus 10 percent M-2), and a 92-octane reference fuel plus 1 ml tetraethyl lead. The addition of lead to the 92-octane reference fuel raised the octane number to approximately 100.

Special care was exercised during these tests to hold all of the engine conditions constant except those varied experimentally. The fuel-air ratio was held constant at a value giving peak temperatures. In this part of the mixture curve, slight errors in the mixture have little effect on cylinder temperatures. The fuels and the cooling conditions were chosen to eliminate any possibility of afterfiring or preignition. Although it was impossible to maintain cooling conditions exactly constant, the necessary cylinder temperature corrections were approximately 1° or 2° F.

In these experiments the conditions of test were held constant for not more than 5 minutes after the temperatures leveled off and in most cases under violent knocking conditions this time was less than 3 minutes. This period of time was greater in every case than would be experienced during the normal take-off period of most airplanes. In this respect these data do not show the temperature trends that would occur over a period of hours. Tests conducted elsewhere in which knocking conditions were maintained for hours have shown very gradual temperature-rise effects. There is evidence to show, as will be referred to later in the report, that this gradual temperature rise may be caused by the effect of knocking combustion upon the lubricating-oil film which normally seals the piston rings. In this case the gradual deterioration of piston-ring sealing would cause a gradual increase in piston-ring blow-by and consequently cylinder temperatures.

The engine performance is shown in figure 13. The differences in power output and volumetric efficiency obtained with the three fuels are difficult to explain from the characteristics of the fuels alone. This variation in power is not of particular interest, however, inasmuch as these tests were made to determine the engine performance and the engine temperatures as affected by different degrees of knock. The curves of figure 13 are particularly significant in several respects. The fact that the power did not fall off an appreciable amount, even though violent audible knock was encountered, is contrary to published information (references 8 and 19). This result possibly indicates that marked reductions in power, which may be accompanied by knock, may have been caused, in some cases, by preignition rather than fuel knock. In figure 13 the small drop in power is fully accounted for by the change in the rate of rise of the volumetric efficiency that accompanies violent knock. Whether the lowered volumetric efficiency can be explained by the rise in combustion-chamber temperature is uncertain.

In MacCoull's tests (reference 8) the power was shown to decrease when knock was observed. The knocking condition was reached by raising the compression ratio. In this case, any decrease resulting from knock in the weight of air inducted will cause a more pronounced loss of power when plotted against compression ratio than against induction pressure. The curve of compression ratio plotted against power is generally rather flat at the higher ratios and little reduction in power is required to cause the curve to peak and fall off.

Figure 14 shows the cylinder temperatures corresponding to the performance of figure 13. Of particular interest is the small rise in temperatures above the normal temperature curve up to the point at which the indicated mean effective pressure has been increased about 12 percent above the first indication of audible knock. In general, the temperatures rose rather rapidly when the indicated mean effective pressure was increased from a value of 12 percent to a value of 20 percent above the audible pinking level. This general behavior of temperatures has also been observed on the liquid-cooled Lycoming O-1230 cylinder. In no case did the temperatures of the combustion chamber of the Lycoming cylinder rise above the normal temperature curve when changing from the nonknocking to the audible knocking region.

The curves of figure 14 show, in general, a small drop in temperature at or just preceding audible knock. This drop in temperature has also been observed on the Lycoming cylinder. Unless temperatures are taken at small increments of inlet pressure, its presence may be overlooked. The temperature curves are fairly smooth up to the knock region and from there on are variable. These curves also demonstrate that the temperature rise above the normal curve is not a good indication of fuel knock unless severe knocking is being experienced. This fact is particularly true for the loaded fuel.

Figure 14 shows that the temperature of the spark-plug electrode increases about 15 percent from audible pinking to violent knocking, as compared with 9 percent for the exhaust end-gas zone. This evidence verifies to some extent the statement, as has been pointed out in reference 20 and elsewhere, that the maximum temperatures in knocking combustion are found not in that part of the mixture to knock but in the first part of the mixture to be inflamed.

Rassweiler and Withrow (reference 20) show that the maximum temperature rise of the gases in the firing end of the combustion chamber is only about 5 percent greater during knocking combustion than during normal combustion and is about the same during knocking combustion as during normal combustion from 30° A.T.C. to the end of the power stroke. At the center of the combustion chamber the temperature rise of the gases is only about $2\frac{1}{2}$ percent greater during knocking combustion than during normal combustion and, from 20° to 120° A.T.C., the flame temperatures are somewhat lower during knocking combustion than during normal combustion. In the end zone, it is shown that, from 30° to 120° A.T.C., the flame temperatures are also somewhat lower during knocking combustion than during normal combustion. Whether the lowered flame temperatures following knocking combustion are caused by excessive heat loss during the period of maximum temperatures or by other reasons is not clear. Midgley and McCarty (reference 7) and others have shown that radiant heat increases when the combustion changes from the normal to the knocking phase. As concerns the cylinder-cooling problem, the information of reference 7 on flame temperature during and following knocking combustion substantiates the small effect on cylinder temperatures observed in the present tests.

It is generally accepted that the shock waves during knocking combustion may cause very high rates of heat transfer from the gases to the combustion-chamber walls during the short period of knock. These high rates of heat application may be sufficient under certain operating conditions to momentarily soften the surface of aluminum alloys. Although the thermal conduction and cooling afforded these surfaces throughout the rest of each cycle is apparently sufficient to cause early solidification of the surface metal, it is possible that the high inertia forces acting on the piston surfaces during knock may tear away small particles of the softened surface and thus cause the eroded appearance often present after prolonged periods of knocking combustion. Cylinder-wall temperatures measured near the combustion-chamber surface indicate only the average temperature of the engine cycle and thus provide little indication of the instantaneous surface temperatures during knocking combustion.

Spark-plug overheating during knocking operation has been attributed to high rates of heat flow caused by shock waves. The present results show an inappreciable rise in spark-plug temperature with knocking operation, regardless of whether the spark plug is located in the end zone or not. At LMAL, experience has shown that passages which connect recesses with the combustion chamber sometimes overheat. This overheating is attributed to the flow of gases through the passages as caused by pressure changes throughout the cycle rather than by the oscillations resulting from knock.

In reference 21, the statement is made that experience has shown the effect of fuel knock on the heat losses to be small. Additional data are presented in figure 15 showing the effect of knock on the heat flow to the jacket. These data show no marked break in the curve when knock is encountered.

Some evidence exists to indicate that fuel knock may affect cylinder temperatures through the sticking of piston rings, which results in excessive blow-by. Experience at

LMAL has shown that piston rings have a much greater tendency to stick, even at low outputs, during knock testing than during normal operation. The oil consumption of engines is known to increase during knocking operation. One explanation for this increased consumption is that more of the oil on the upper cylinder walls and ring belt is burned away and leaves residues which accelerate ring sticking.

The data indicate that moderate audible knocking probably has little immediate effect in creating hot spots or in causing surface ignition. In general these tests show that, if a rapid rise in cylinder temperatures precedes cylinder failure, preignition rather than knock may be the cause of failure. In many cases, the simultaneous occurrence of surface ignition and fuel knock has undoubtedly confused the analyses of engine failures.

The test data of figures 13 and 14 have been included to illustrate independently of surface ignition the effects of violent knocking on cylinder temperatures. More extensive tests under more severe knocking conditions would bring out more clearly the effects of violent knocking on the rise in cylinder temperatures and would also indicate the strength limitations caused by the pressures involved. The importance of knowing the effect of extremely violent knocking on both the cylinder temperatures and the cylinder stresses arises from the use of very rich fuel mixtures to obviate fuel knock and overheating at high outputs. If rich mixtures are employed to the fullest extent, a momentary reduction in the fuel supplied causes violent knocking and possible failure. Whether failure from knock will occur as a result of mechanical limitations, of high temperatures and preignition, or of both, depends upon the conditions of operation. A great deal of experience has apparently been gained in this field but few test data are available. Owing to the number of conflicting variables, little reliable information can be obtained without systematic test work.

Effect of Cylinder Temperatures on Fuel Knock

Considerable data are available showing the effect of coolant temperature on fuel knock. Some of these data are summarized in reference 4, in which the conclusion is drawn that the coolant temperature has about the same effect on fuel knock as has the inlet-air temperature.

Cylinder temperatures affect knock mainly by varying the end-gas temperature. This temperature will vary with any factor that affects the heat exchange between the cylinder and the charge. The degree of turbulence, the surface-volume ratio of the combustion chamber, and the location and shape of the end zone are some of the important factors involved. Additional data on this subject are shown in figure 16. This information indicates a relatively small effect of combustion-chamber surface temperature on fuel knock, contrary to the conclusion drawn in reference 4.

The actual temperatures of the surfaces in the end zone probably have a much greater effect on knock than the average surface temperature of the combustion chamber. The heat abstracted from the cylinder walls during the intake and compression strokes is probably small because, in the usual case, the charge temperature during intake and compression is little below that of the average for the cylinder surfaces. During combustion and during the resulting compression of the end gas, heat is lost to the combustion-chamber surfaces. As the gases in the end zone are at a comparatively high temperature just before combustion is completed, it is apparent that, if the end-zone wall surfaces are cool, the heat flow to the walls will be rapid and an effect on knock will result. If a very hot surface, such as an exhaust valve, lies in the end zone, however, little end-gas cooling will result and the tendency for fuel knock will be increased.

These facts are well known, and designers have endeavored by various means to locate the end-gas zone in the coolest areas of the combustion chamber by suitably placing the spark plugs to direct the flame front. When two spark plugs are used, the location of the end zone can be controlled, to some extent, by advancing the ignition of one spark plug with respect to the other. The location of the end-gas zone is also affected by the gas movement in the cylinder. One scheme for cooling the end gases that has been frequently employed in automobile engines is to increase the surface-volume ratio of the end-gas zone. This method incurs a greater heat loss to the coolant than otherwise and, for that reason, is not of interest to aircraft-engine designers.

In the tests reported in reference 16, the temperature of an exhaust valve in which sodium was used for cooling was reduced from approximately 1300° to 200° F by removing the sodium and circulating water through the valve. This temperature reduction permitted an increase in indicated mean effective pressure in the lean-mixture range of approximately 10 percent for the same intensity of knock. The increase at rich mixtures was much smaller.

CONCLUSIONS

The analysis of the data of this report and the accumulated data of other sources may allow the following general conclusions:

1. The relationship between the curve of ignition temperature of a fuel-air mixture plotted against density and the curve of hot-spot temperature plotted against density at the time of ignition may indicate whether dangerous preignition characteristics may be expected. If the fuel used provides for stable preignition characteristics, it may be possible safely to raise the general temperature level of the cylinder head, or to increase the power, even though surface ignition is present. If the fuel provides for unstable preignition characteristics, the remedy may lie in changing the fuel rather than in eliminating the sources of surface ignition.

2. Fuel knock is accompanied by a slight drop in cylinder temperatures at or just preceding audible knock. For short intervals of time no appreciable rise in combustion-chamber temperatures above the normal rise caused by increasing the inlet pressure and the power is caused by fuel knock until the power is increased about 12 percent above that obtained at the first indication of audible knock. For short durations it is therefore unlikely that moderate audible knocking, in itself, has much effect in creating hot spots or in causing surface ignition. If a rapid rise in cylinder temperatures precedes cylinder failure, preignition rather than knock may be the cause of failure.

3. The drop in power output or the increase in specific fuel consumption as caused by knock alone is inappreciable up to violent audible knock.

4. A more direct means of cooling the overheated regions of the cylinder would permit raising the temperatures of the adjacent areas, which may now be overcooled.

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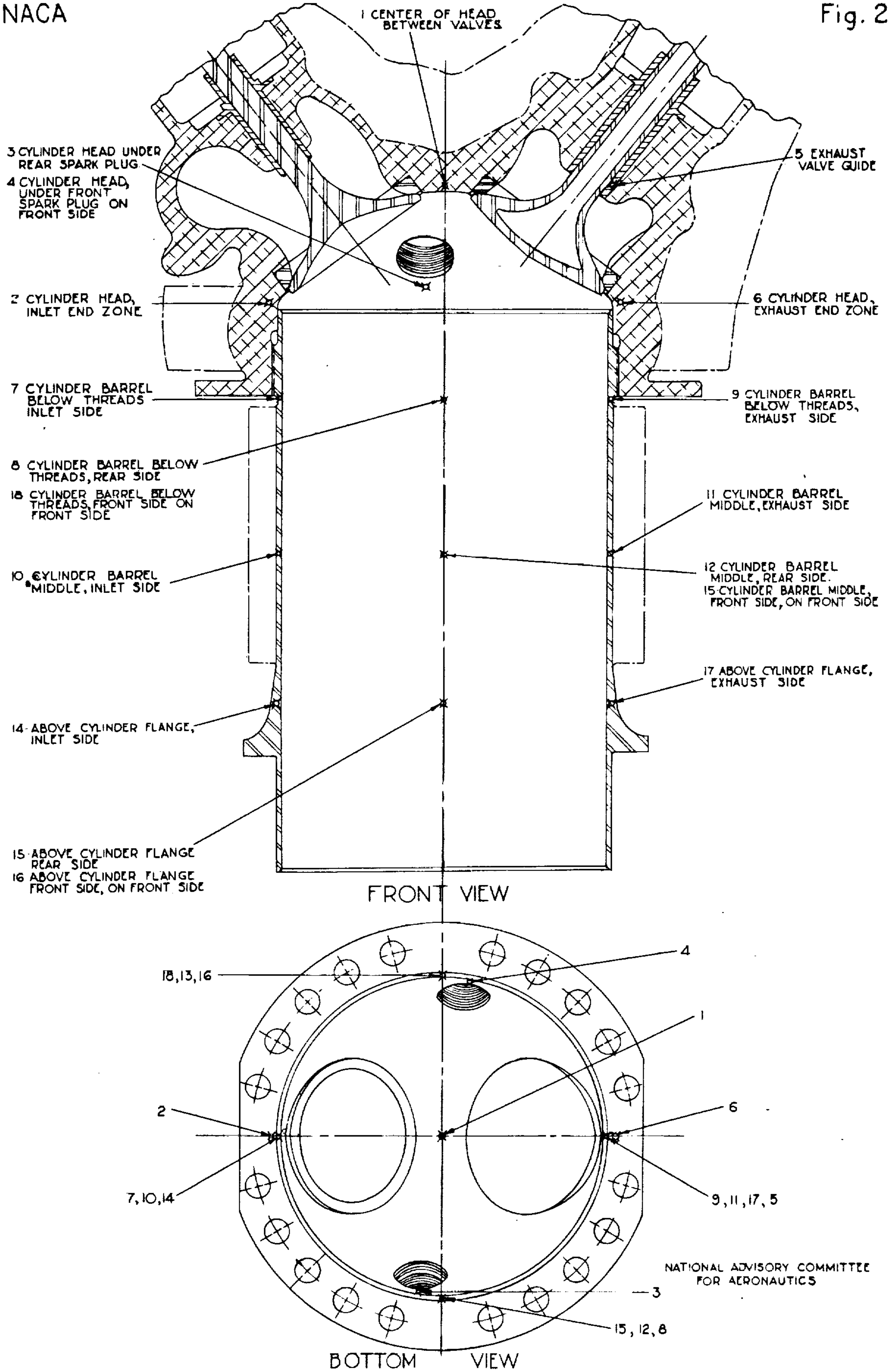

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FIGURE 2.- LOCATION OF THERMOCOUPLES ON THE WRIGHT C9GC CYLINDER.

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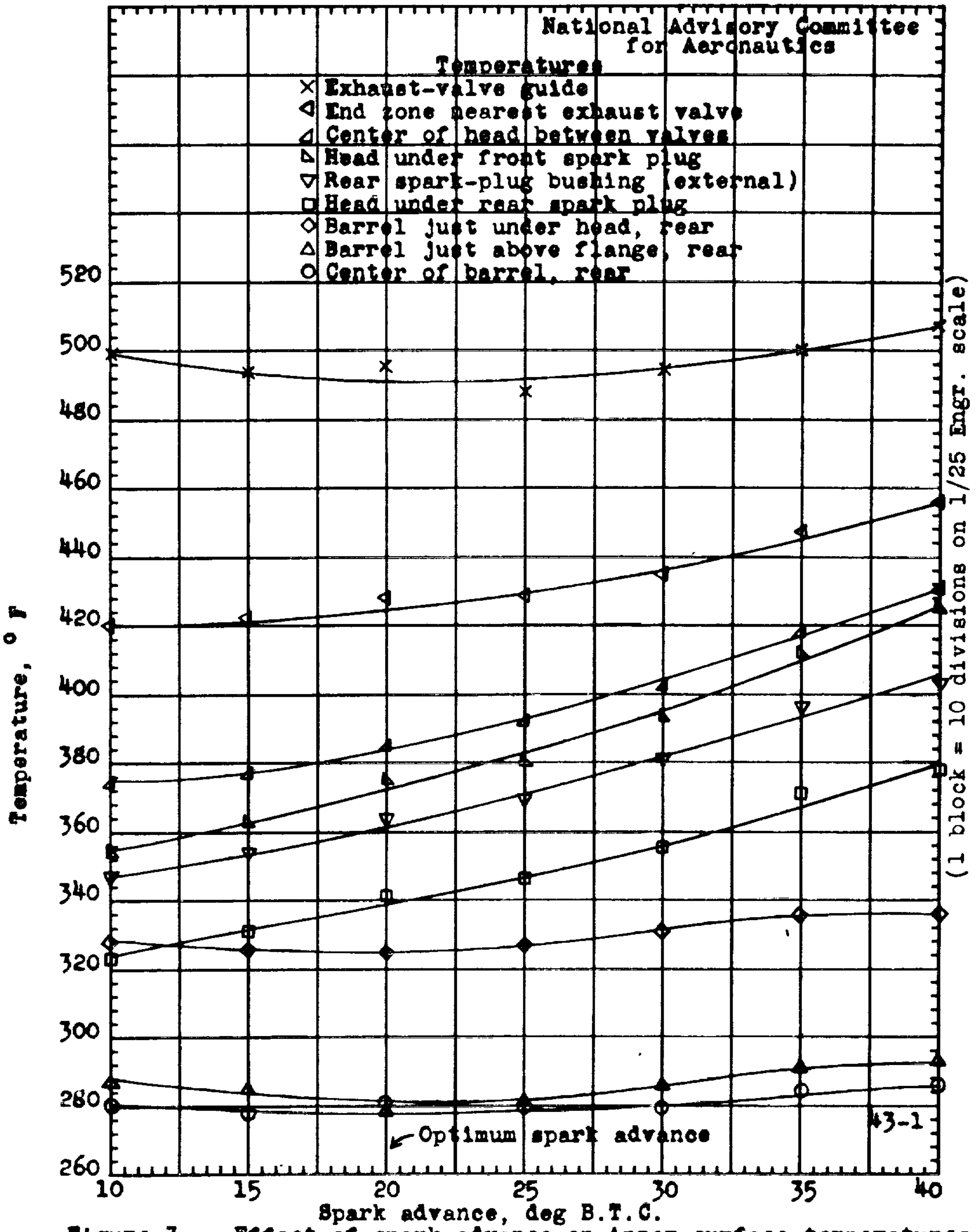

Figure 3. - Effect of spark advance on inner-surface temperatures. Wright C9GC cylinder; engine speed, 2000 rpm; compression ratio, 7.0; inlet-air temperature, 150° F; cooling-air pressure drop, 6.7 inches of water; inlet-air pressure, 29.7 inches Hg absolute; fuel-air ratio, 0.078.

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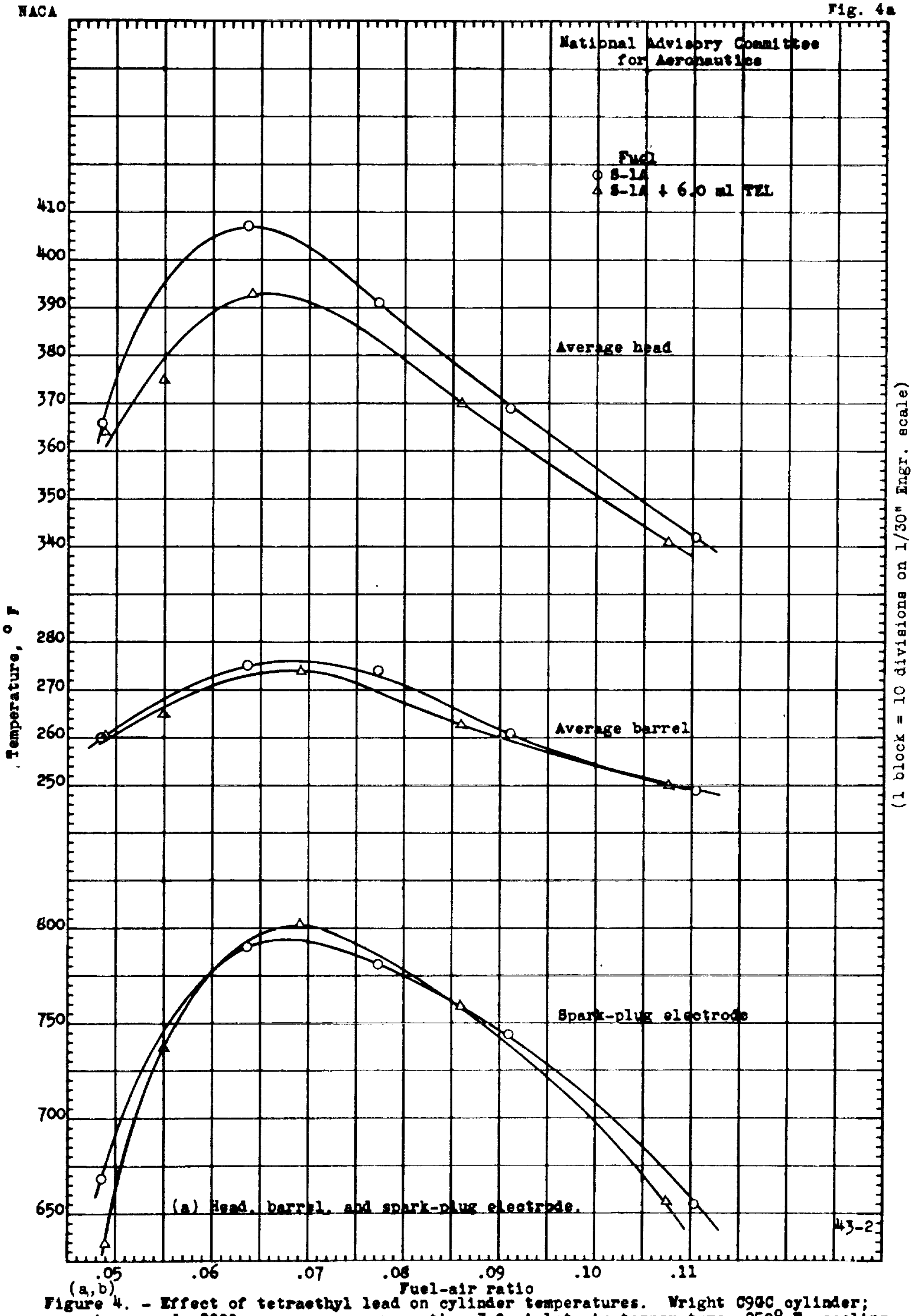

Figure 4. - Effect of tetraethyl lead on cylinder temperatures. Wright C9GC cylinder; engine speed, 2000 rpm; compressure ratio, 7.0; inlet-air temperature, 250° F; cooling-air pressure drop, 2.7 inches of water; indicated mean effective pressure, 93 pounds per square inch.

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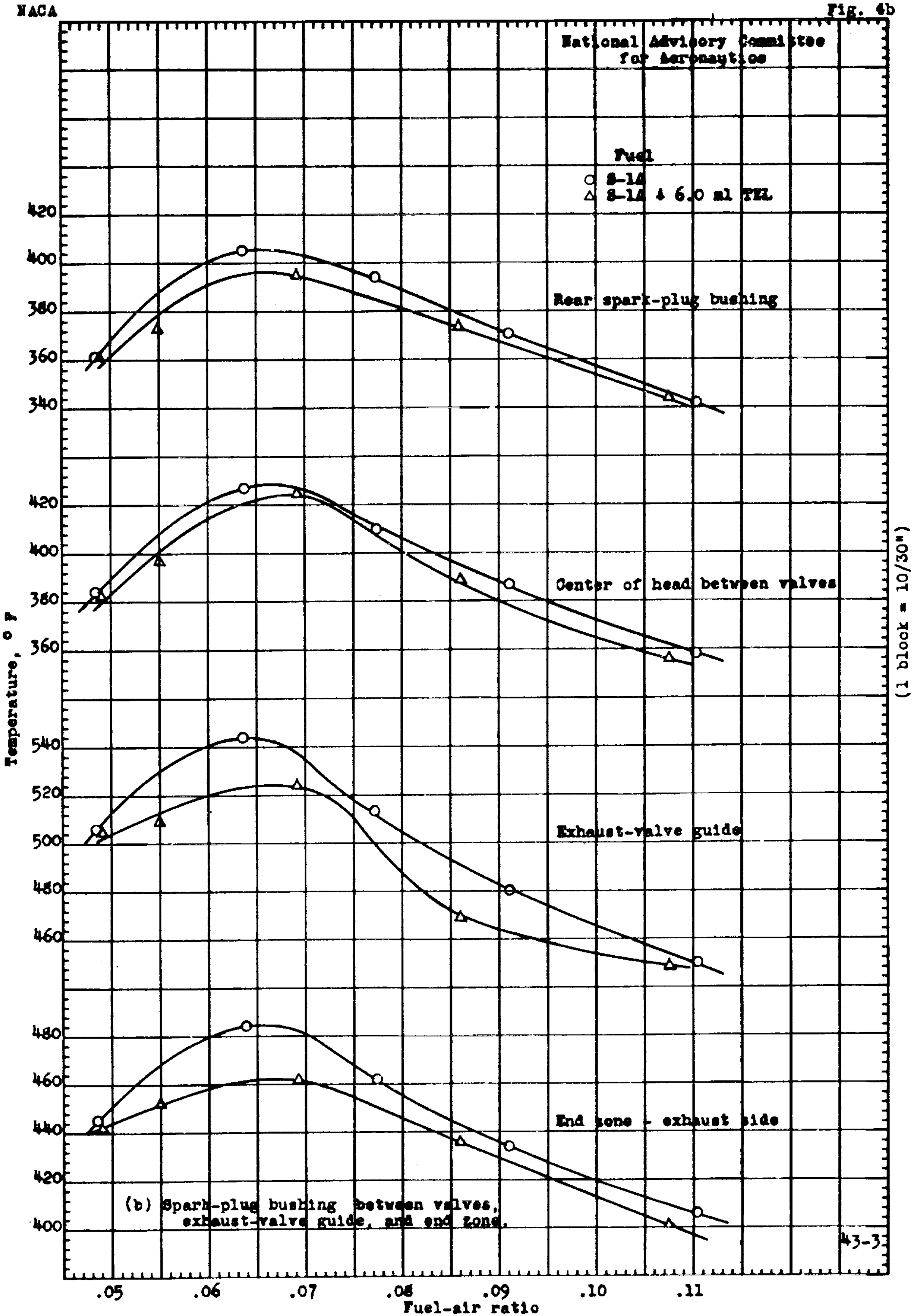

(b) Spark-plug bushing between valves, exhaust-valve guide, and end zone.

Figure 4. - Concluded.

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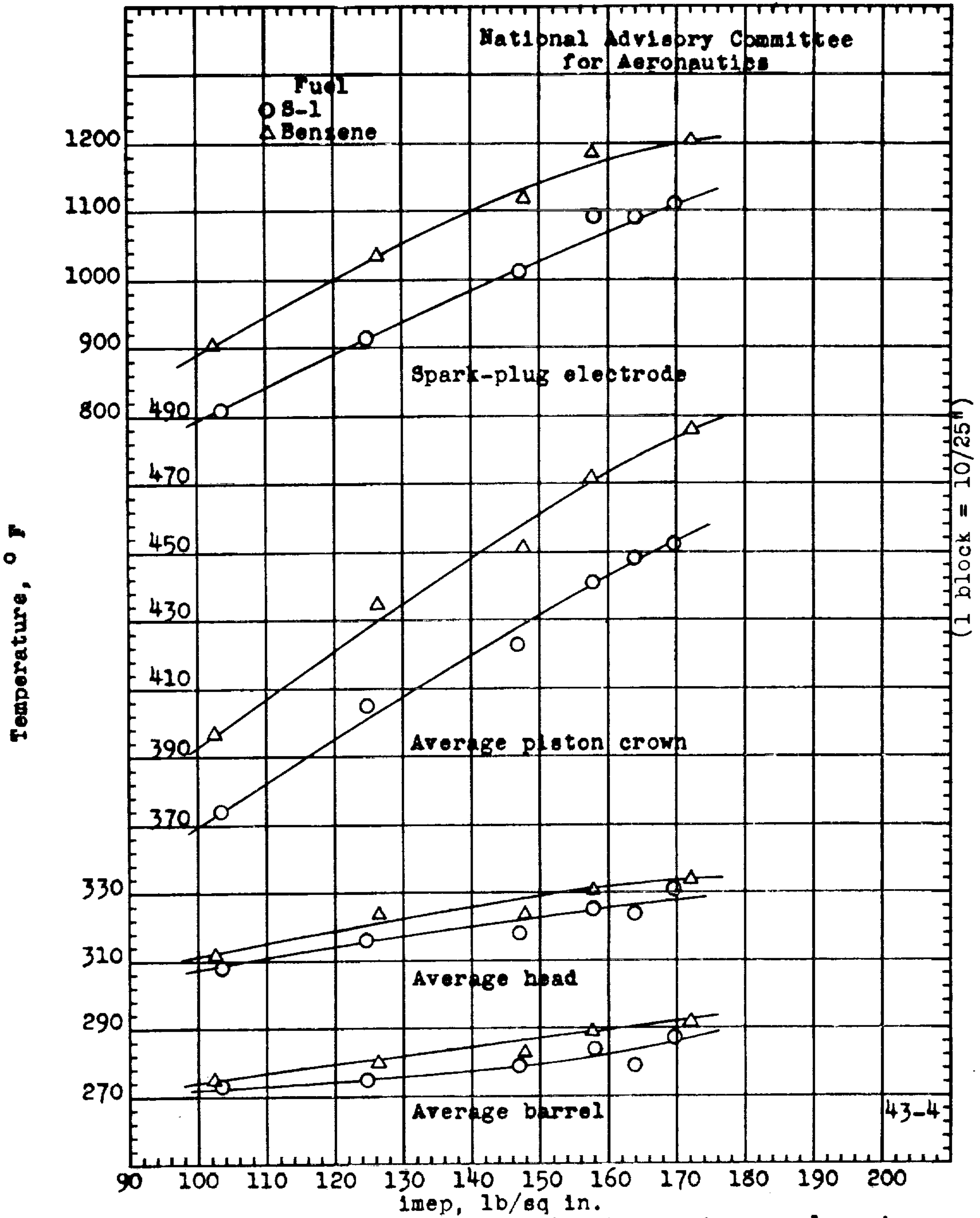

Figure 5. - Effect of fuel on engine temperatures. Lycoming O-1230 cylinder; engine speed, 2000 rpm; compression ratio, 8.17; inlet-air temperature, 86° F; coolant-outlet temperature, 249° F; maximum temperature mixture.

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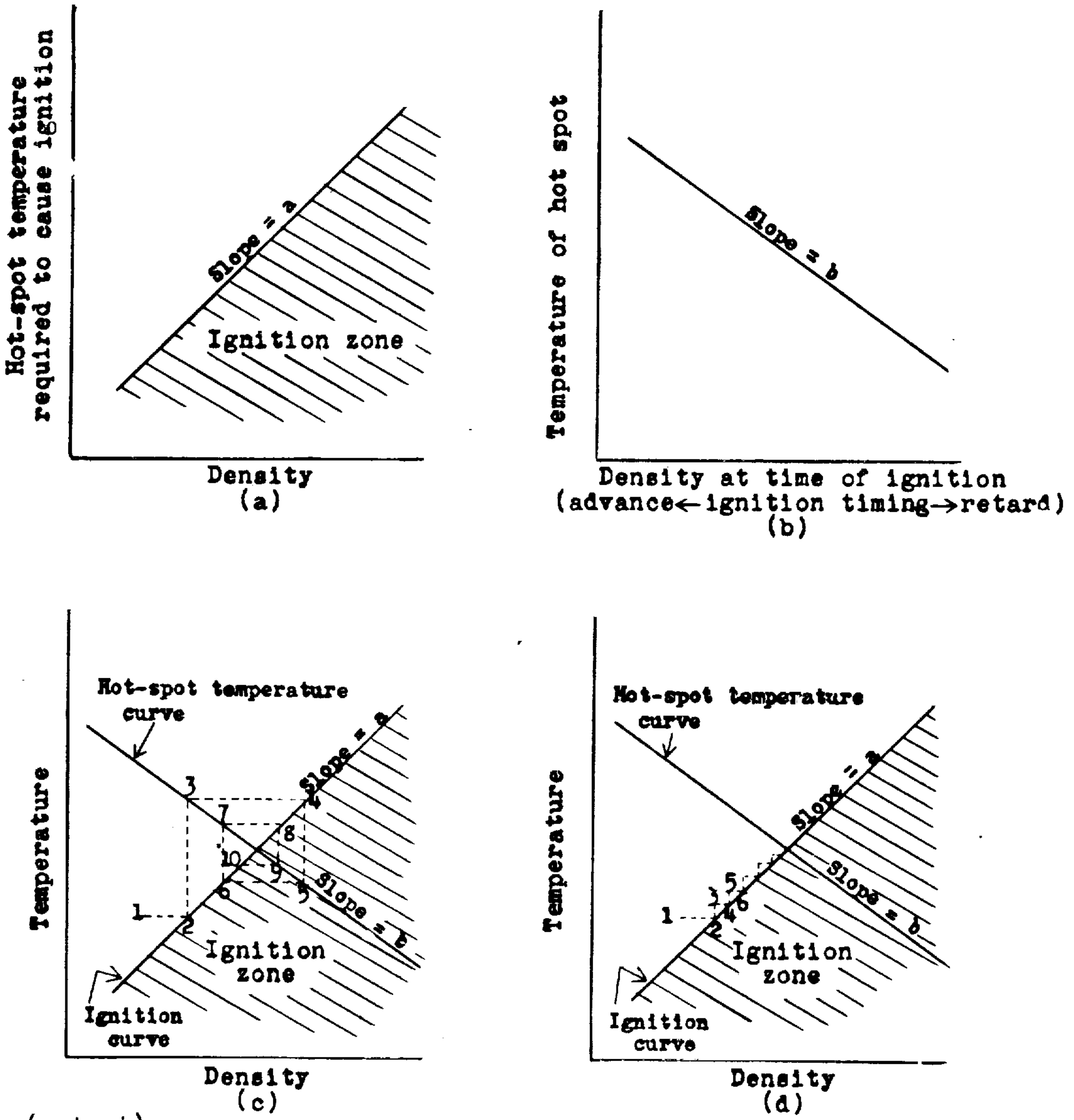

(a to i)
Figure 6. - Curves illustrating conditions controlling stability of preignition.

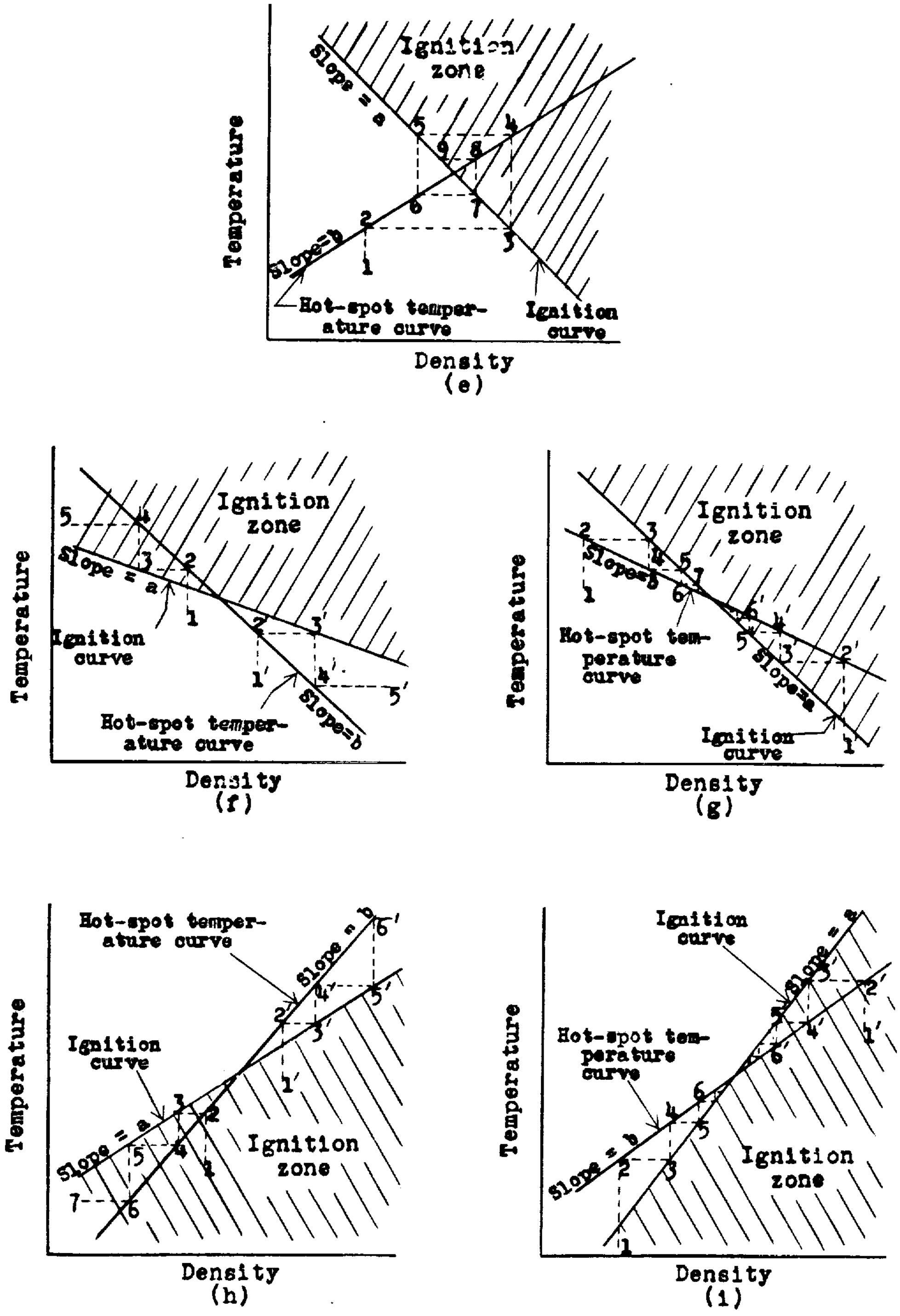

Figure 6. - Concluded.

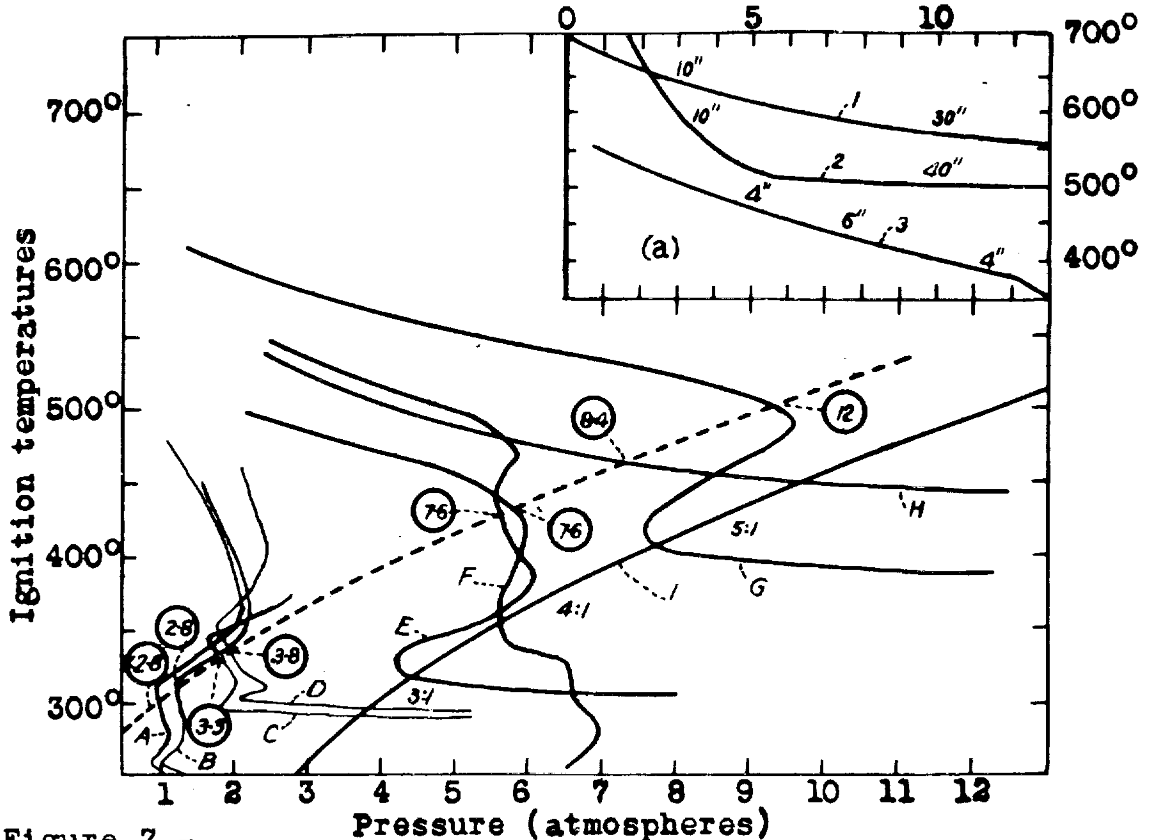

Figure 7.-
One-second lag ignition-point curves: theoretical mixtures with air
A = Octane, B = heptane, C = hexane, D = pentane, E = iso-octane, F = diisopropyl ether, G = propane, H = propylene.
Values in circles denote known knock ratings.
Inset (a). Ignition-point curves: 1 = Benzene, 2 = methane, 3 = acetone.

(Reproduced from reference 11.)

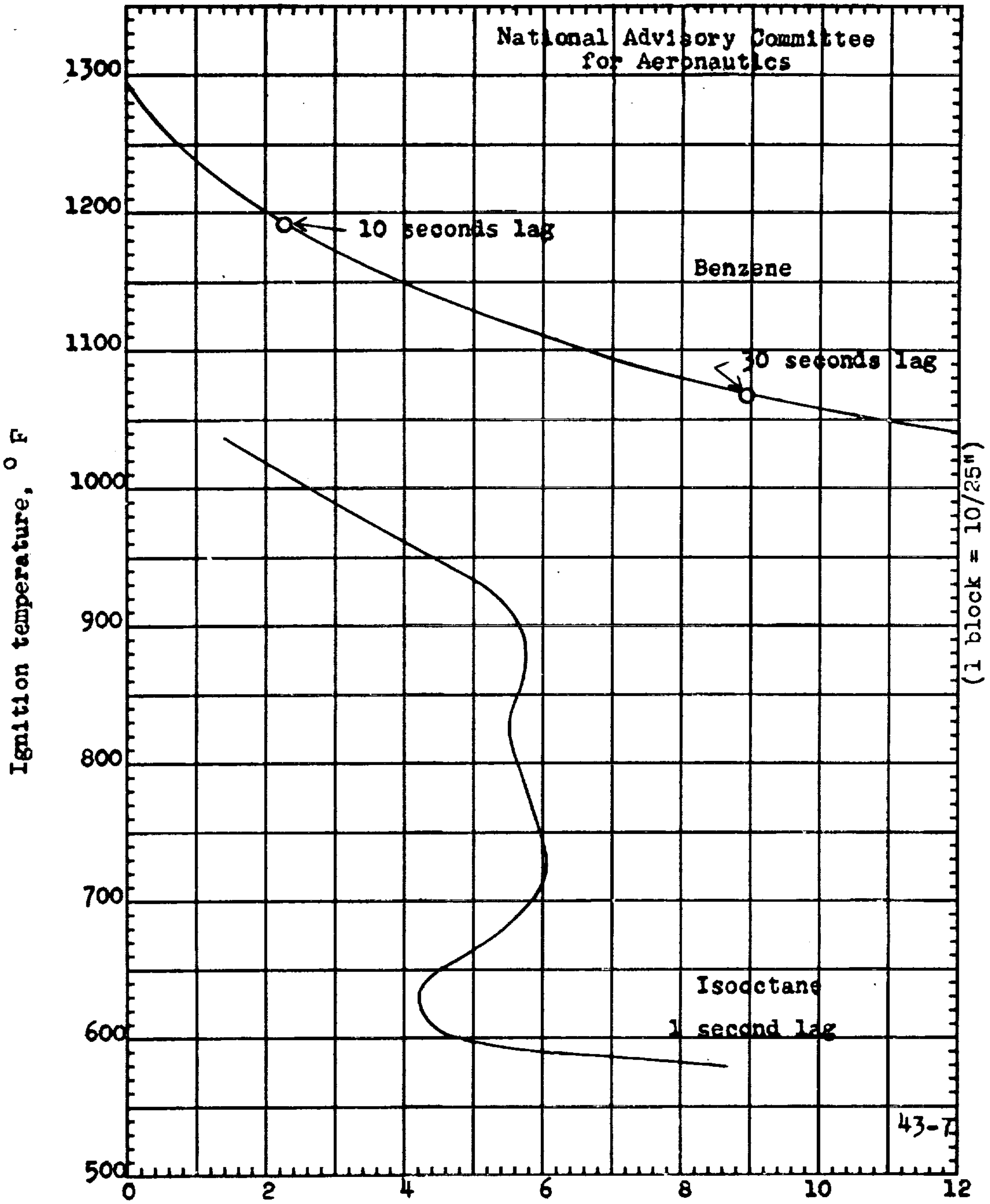

Figure 8. - Ignition-temperature curves for isooctane and benzene. Ignition lags as shown. Curves from figure 7.

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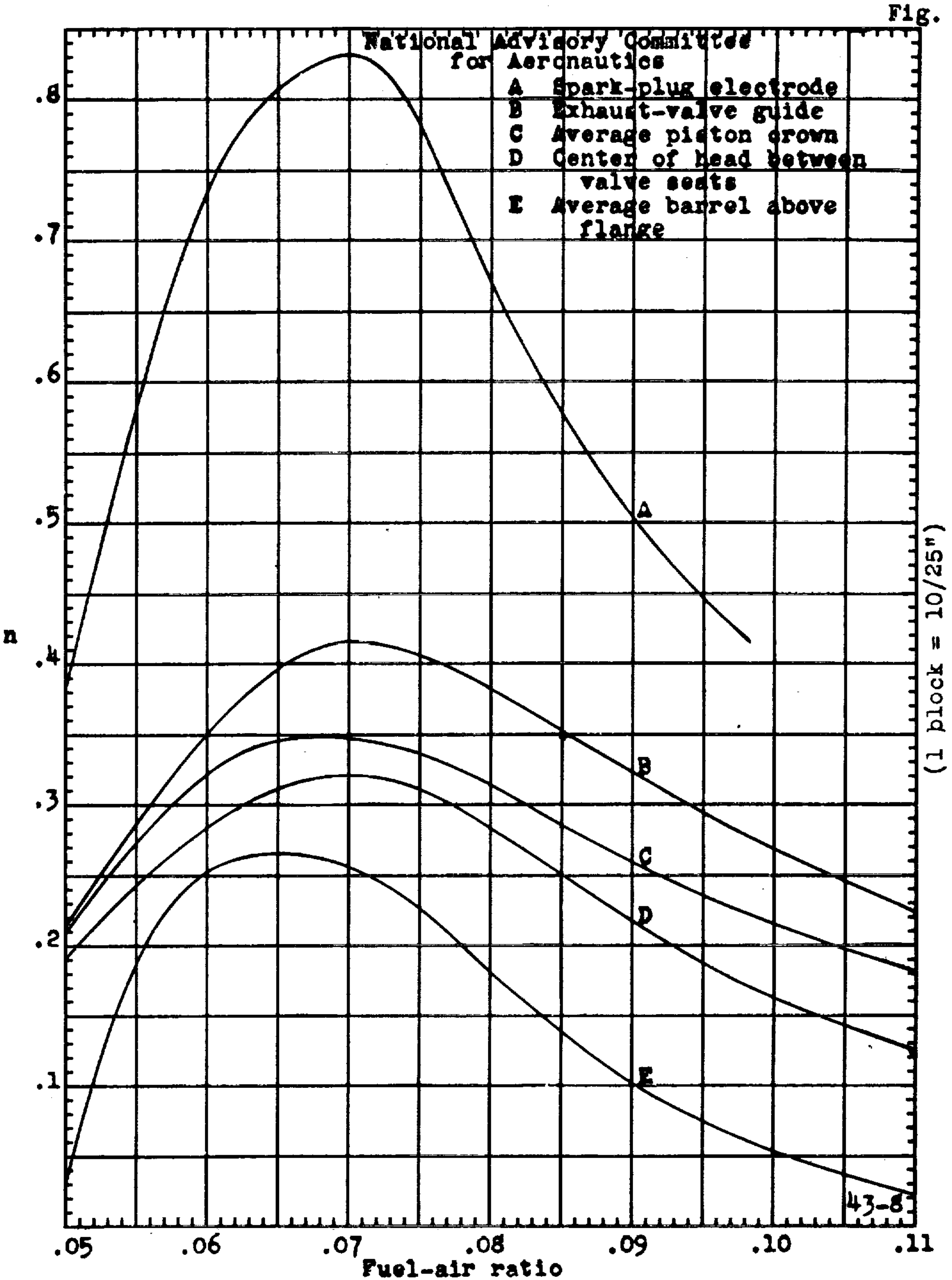

Figure 9. - Variation of the exponent n of the expression $T = cI^n$ with mixture ratio for an inlet-air temperature of 150° F. Lycoming 0-1230 cylinder; engine speed, 2000 rpm; compression ratio, 7.0; coolant-outlet temperature, 250° F.

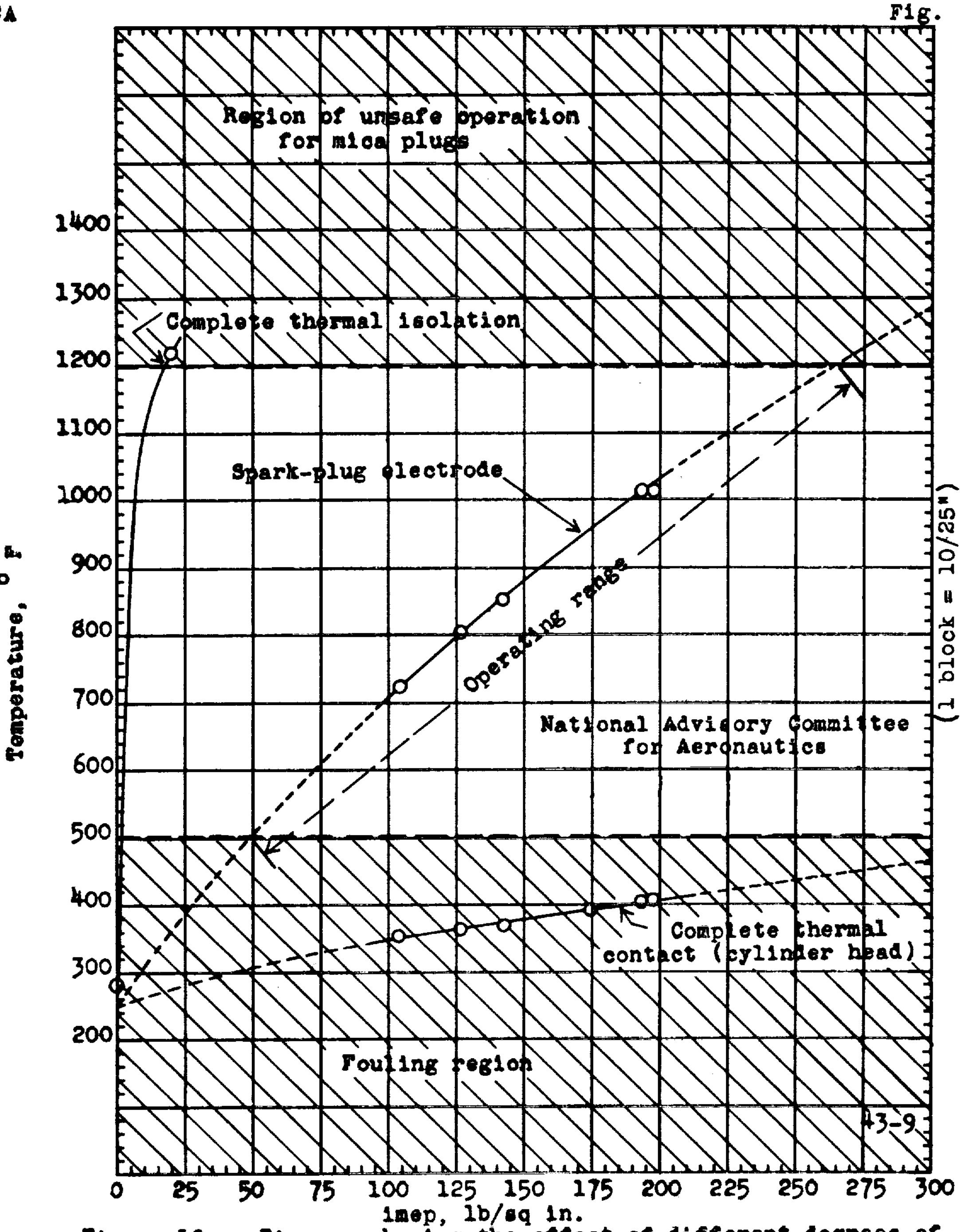

Figure 10. - Diagram showing the effect of different degrees of thermal isolation of the tip of the spark-plug center electrode with respect to the permissible operating range.

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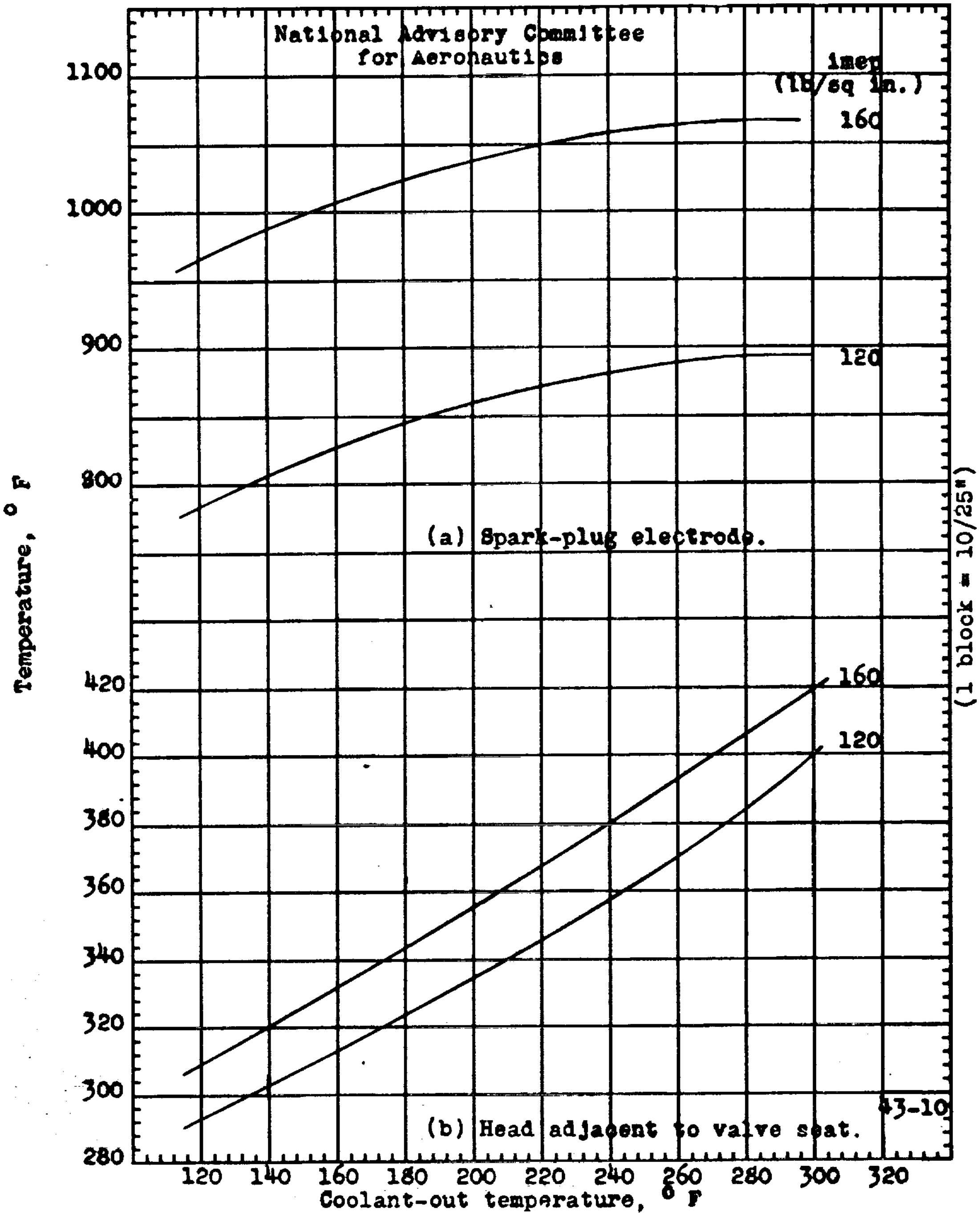

Figure 11. - Effect of coolant-outlet temperature on engine temperatures. Lycoming O-1230 cylinder; engine speed, 2000 rpm; compression ratio, 8.17; inlet-air temperature, 86° F; maximum temperature mixture.

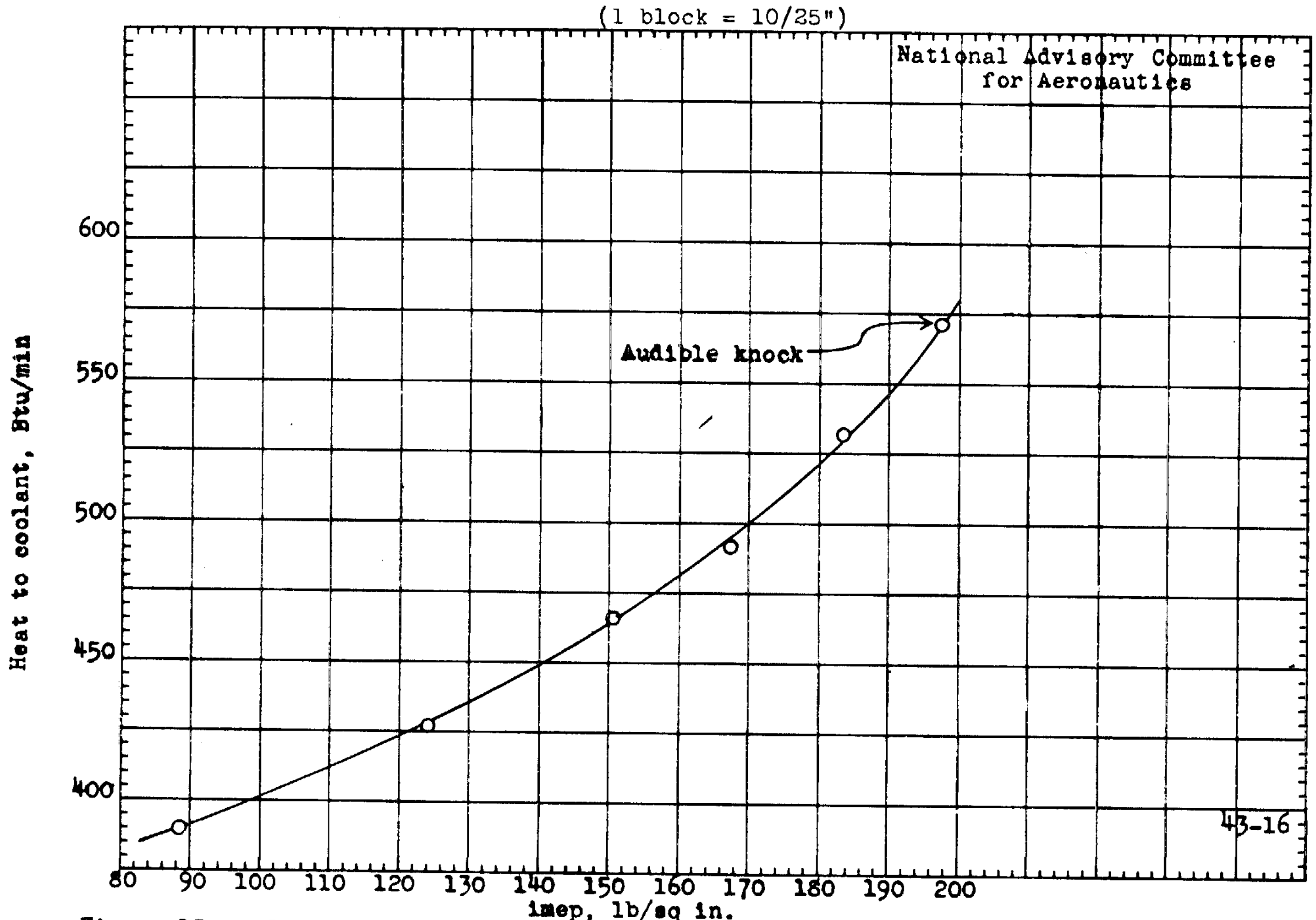

Figure 15. - Effect of knock on the heat flow to the coolant. Lycoming O-1230 cylinder; engine speed, 2000 rpm; compression ratio, 8.17; inlet-air temperature, 86° F; coolant-outlet temperature, 250° F; maximum temperature mixture.

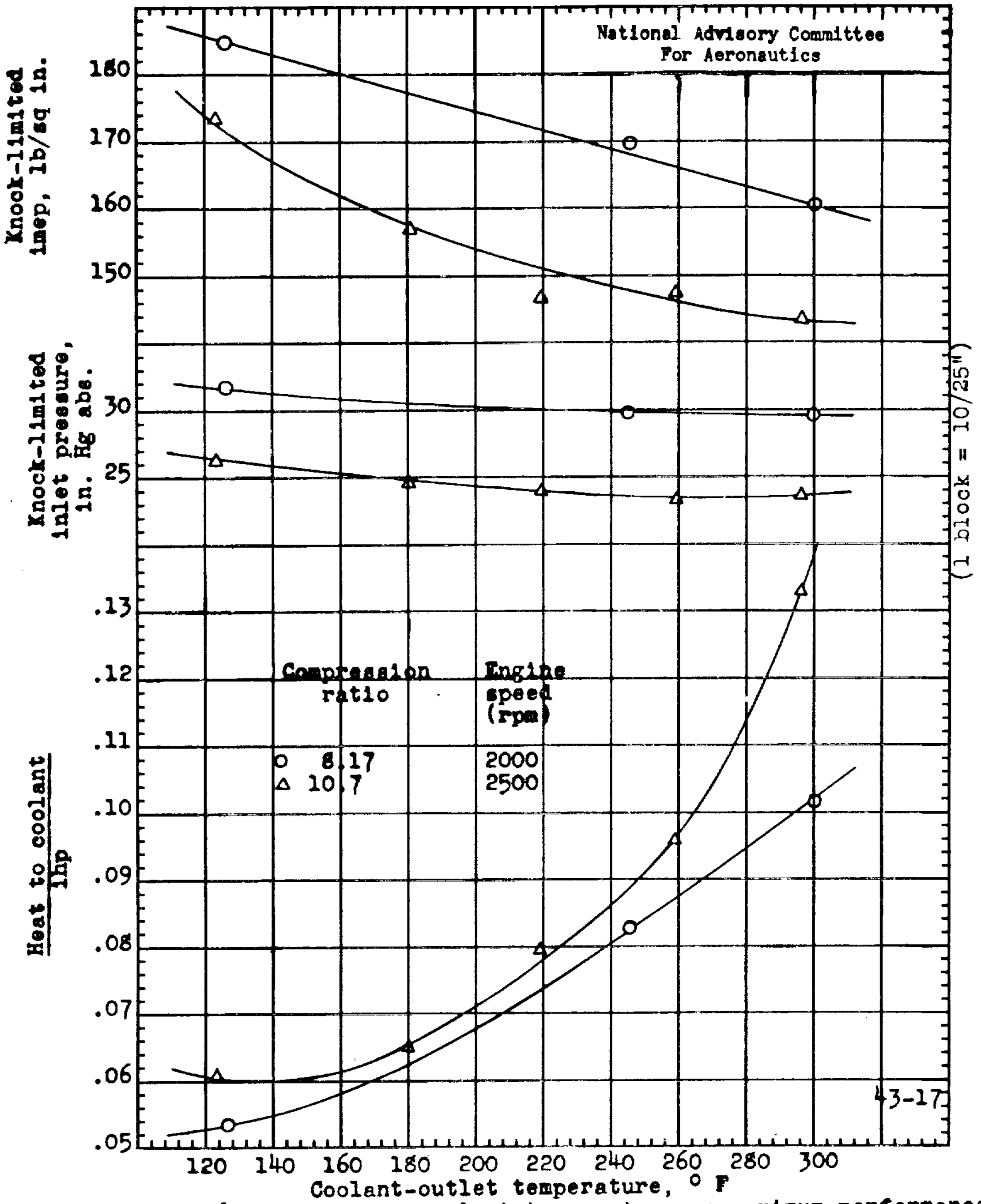

Figure 16. - Effect of coolant temperature on maximum performance as limited by knock. Lycoming O-1230 cylinder; inlet-air temperature, 86° F; maximum temperature mixture.

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