REPORT No. 190

CORRECTING HORSEPOWER MEASUREMENTS TO A STANDARD TEMPERATURE

By STANWOOD W. SPARROW

Bureau of Standards

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SUMMARY.

This report was prepared for publication by the National Advisory Committee for Aeronautics. The relation between the temperature of the air at the entrance to the carburetor and the power developed by the engine is discussed. Its scope is limited to a consideration of the range of temperatures likely to result from changes of season, locality, or altitude, since its primary aim is the finding of a satisfactory basis for correcting power measurements to a standard temperature.

The tests upon which this report is based were made upon aviation engines in the altitude laboratory of the Bureau of Standards. From the results of over 1,600 tests it is concluded that if calculations be based on the assumption that the indicated horsepower of an engine varies inversely as the square root of the absolute temperature of the carburetor air the values obtained will check closely experimental measurements. The extent to which this relationship would be expected from theoretical considerations is discussed and some suggestions are given relative to the use of this relationship in correcting horsepower measurements.

INTRODUCTION.

Knowledge as to the change of engine power resulting from a given change of carburetor air temperature is essential to the proper interpretation of engine acceptance and standard flight tests. In such tests it is not always feasible to maintain a constant carburetor air temperature but power measurements should be corrected to the same temperature before they are compared. Previous work on this subject has been confined chiefly to experimental determinations of the effect of changes in carburetor air temperature upon the brake horsepower of particular engines. In most early discussions of this subject the volume of charge received by the engine in unit time is assumed to be independent of charge temperature. Were this assumption true the power developed by an engine would be directly proportional to the density of the air at the entrance to the carburetor. That such is not the case was shown in Technical Report No. 45 of the National Advisory Committee for Aeronautics. That the volume of charge received by the engine in unit time does change with change in temperature was noted in tests made at the Bureau of Standards in 1919. Reference to this is made in a paper entitled "Compression Ratio and Thermal Efficiency of Airplane Engines," published in the Journal of the Society of Automotive Engineers for March, 1921. This condition is discussed also by Gage in Report No. 108 of the National Advisory Committee for Aeronautics, entitled "Some Factors of Airplane Engine Performance."

In "The Armour Engineer" for March, 1921, White advocates correcting engine power by considering it to vary inversely as the square root of the absolute temperature. He shows theoretically that the weight of air flowing through a given orifice follows this relation so long as the head producing flow remains constant. As evidence that this head does remain constant he gives a plot of manifold suction which shows no change over a wide range of entrance air temperatures.

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The report which follows contains the results of tests made upon an aviation engine operating under a wide range of conditions of engine speed, throttle opening and entrance air pressure. Actual comparisons are made with results obtained with other engines and the factors influencing air flow are discussed.

The major portion of the experimental data discussed in this report was obtained with an eight-cylinder water cooled aviation engine having a bore of 4.72 inches and a stroke of 5.12 inches. This engine was supplied with four sets of pistons making it possible to obtain compression ratios of 5.3, 6.3, 7.3, and 8.3. The engine as mounted in the altitude chamber received its air through a pipe from a chamber containing refrigerating coils. In going to the engine the air first passed over these coils and then over electric heating grids which made it possible to obtain the desired entrance air temperature and to maintain it constant during a series of runs. Throughout these tests the fuel used conformed to Government specifications for domestic aviation gasoline.

Runs were made at air pressures corresponding to altitudes of 5,000, 15,000, and 25,000 feet and at approximately full, three-quarters, and one-half load, except when prevented by preignition or detonation. The purpose of operating at reduced throttle was to obtain some idea as to the results to be expected with engines whose induction systems offer considerably more resistance to flow than did that of the engine under test.

SOURCES OF ERROR.

That work of this sort requires accurate and sensitive measuring apparatus and means for maintaining conditions constant while measurements are being taken is generally appreciated. There are some sources of error, however, which are often overlooked with disastrous results. One such source is the difficulty of adjusting the air-fuel ratio to give maximum power. In some instances tests have been made in which only one run was taken at each temperature. The objection to this procedure is that a slight error in the adjustment of the air-fuel ratio may cause a greater change in power than a difference in entrance air temperature of 15 or 20° C. An even worse situation arises when the adjustment of mixture ratio is left unchanged with change in temperature. In this case the change in carburetor air temperature changes the power development by changing the air-fuel ratio as well as the weight of charge received by the engine in unit time. Difficulties arising from an incorrect adjustment of mixture ratio have been avoided in the present experiments by taking measurements with several mixture ratios at each temperature. Plots of power versus air-fuel ratio show whether or not the correct adjustment for maximum power was made and the maximum power value.

Another annoying condition which appeared in some of the first runs of this investigation was the formation in the induction system of sufficient quantities of frost to actually restrict the air flow. To bring about such a condition (1) the air entering the carburetor must contain a considerable amount of moisture, and (2) the temperature of this air must drop to the freezing point of water. Dangers arising from the formation of large quantities of snow in the induction system are discussed in Technical Note No. 55 of the National Advisory Committee for Aeronautics. As far as experimental work is concerned, this trouble is more serious, because less easily recognized, when the formation takes place rather slowly. In such cases the change in flow due to the difference in thickness of frost coating, combines with the change strictly chargeable to a difference in entering air temperature and so long as temperature changes are made in the same direction, a smooth curve results. If, instead of making each run at a colder temperature than the preceding or in the reverse order, the runs are scattered, then this frost formation makes the results erratic and its presence is recognized. Fortunately this condition was recognized in the early runs of this investigation and avoided thereafter by cooling the air to a temperature so low that its moisture content was extremely small. This comparatively dry air was then heated to the desired temperature. In this connection it is of interest that, barring this snow trouble, no measureable changes in power have been observed from changes in the humidity of the entering air.

EXPERIMENTAL RESULTS.

Figure 1 will serve as an illustration of the general method of plotting the results. In the center of the figure is shown a group of curves of indicated mean effective pressure plotted against fuel consumption in pounds per hour. This is essentially a plot against air-fuel ratio, since a change in air-fuel ratio does not change appreciably the rate of air flow. From the maximum points shown by these curves the curve of I.M.E.P. versus temperature was plotted. Only the curves of this last-mentioned type are shown in Figures 2 to 10. All of these curves, however, were obtained in the same manner as that in Figure 1, namely, from plots of I.M.E.P. versus pounds of fuel per hour.

Dotted curves in Figures 1 to 10 show I.M.E.P.'s computed from the values at 15° C., as taken from the best curve through the experimental points, assuming the power to vary inversely as the square root of the absolute temperature.
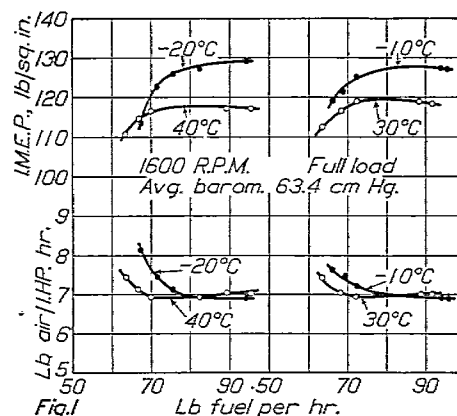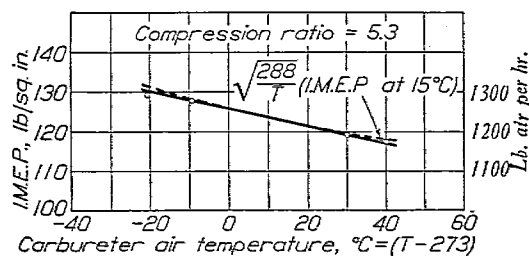
It will be noted that the maximum difference between the computed and experimental curves is less than 2 per cent. Less than half of the tests made in this investigation are plotted in this report. While the remainder of the tests often contain erratic points, they show the same general relation as Figures 1 to 10.

In Figure 1 it will be noted that the minimum value of pounds of air per indicated horsepower hour occurs when the power developed is a maximum. This minimum value is practically the same for all carburetor air temperatures. If the energy obtained per pound of air at the mixture ratio giving maximum power is not affected by the carburetor air temperature, then the power developed by the engine must be directly proportional to the weight of air received by the engine in unit time.
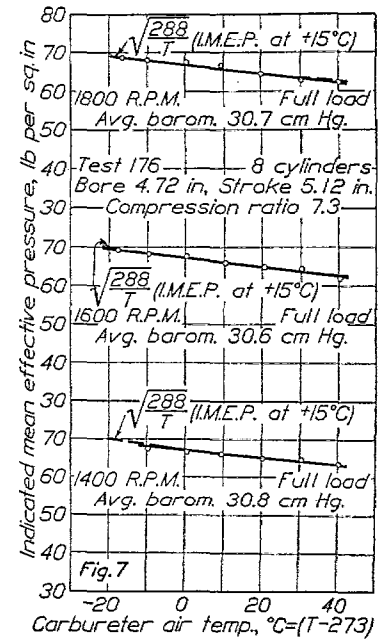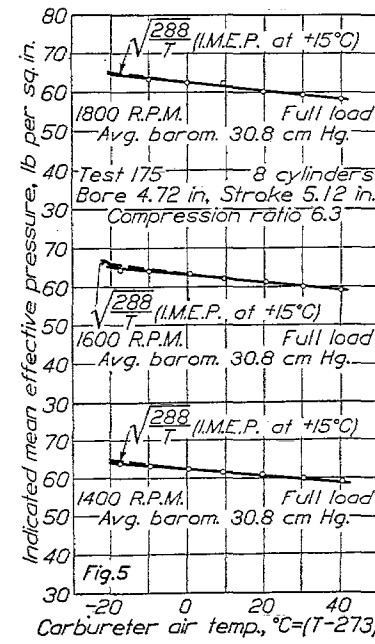

Fig.1 Lb fuel per hr.

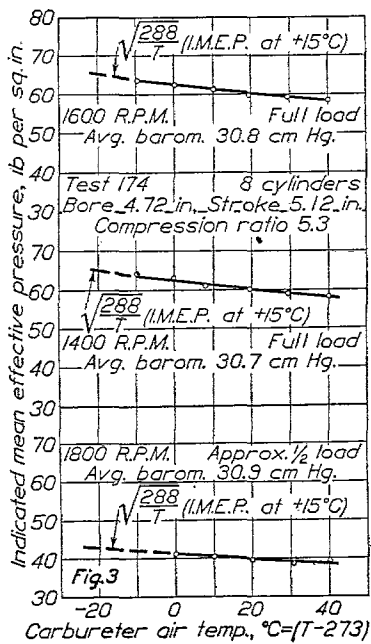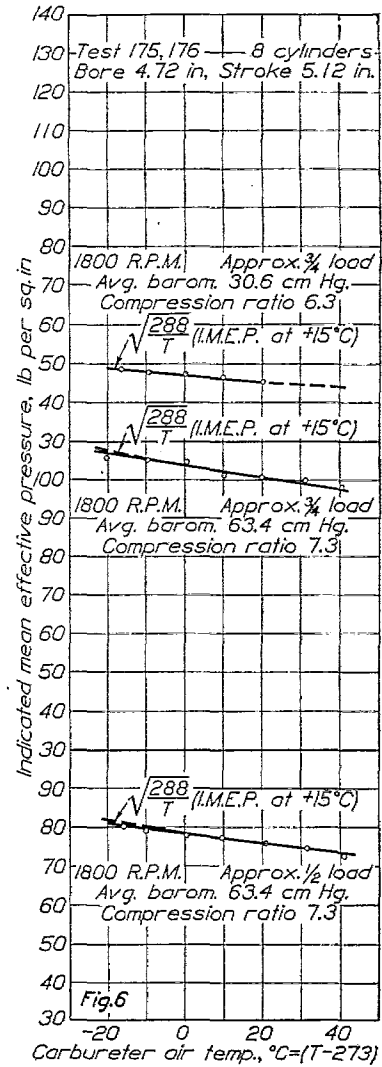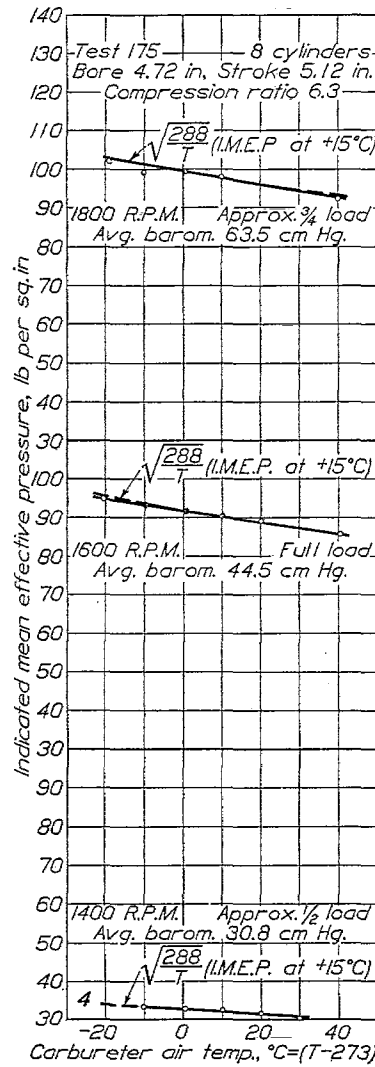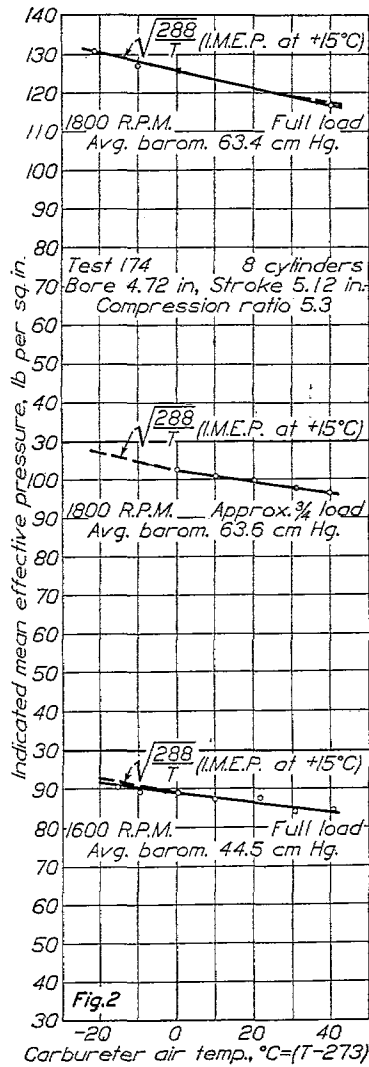Test 174, 8-cylinder engine, bore 4.72 inches, stroke 5.12 inches.

Hence if the I.M.E.P. shown in Figure 1 varies nearly inversely as the square root of the absolute temperature, it follows that the weight of air received by the engine in unit time must vary in the same manner. Such is the case and the I.M.E.P. curve is also a curve of pounds of air per hour as shown by the scale at the right.

That other engines show similar characteristics is shown in Figures 11, 12, and 13. In these figures are plotted ratios of pounds of air per hour at various temperatures to pounds of air per hour at a carburetor air temperature of +15° C. These curves are comparable to the I.M.E.P. curves of Figures 1 to 10 for with these engines also no appreciable change in the power developed per pound of air per hour resulted from a change in the carburetor air temperature. As before, the dotted lines in the figure show values based on the assumption that the weight of air received by the engine in unit time varies inversely as the square root of the absolute temperature. The greatest difference between the calculated and experimental results occurred with the 6-cylinder engine and is shown in Figure 11. Even this is less than 3 per cent. For the other engines, a 12-cylinder and an S, the experimental and calculated results agreed within 2 per cent.

Considering the wide range of conditions under which these measurements have been made and the difference between the engines, the experimental evidence seems to justify the conclusion that with aviation engines the power varies very nearly inversely as the square root of the absolute temperature over the range of temperatures covered in these tests. Figure 14 gives an idea of the magnitude of changes in some of the quantities discussed thus far.

Fig.2 — Indicated mean effective pressure, lb per sq. in. vs Carbureter air temp., °C=(T-273)

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1800 R.P.M.   Full load
Avg. barom. 63.4 cm Hg.

Test 174   8 cylinders
Bore 4.72 in, Stroke 5.12 in.
Compression ratio 5.3

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1800 R.P.M.   Approx. ¾ load
Avg. barom. 63.6 cm Hg.

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1600 R.P.M.   Full load
Avg. barom. 44.5 cm Hg.

Fig.4

Test 175   8 cylinders
Bore 4.72 in, Stroke 5.12 in.
Compression ratio 6.3

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1800 R.P.M.   Approx. ¾ load
Avg. barom. 63.5 cm Hg.

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1600 R.P.M.   Full load
Avg. barom. 44.5 cm Hg.

1400 R.P.M.   Approx. ½ load
Avg. barom. 30.8 cm Hg.

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

Fig.6

Test 175, 176   8 cylinders
Bore 4.72 in, Stroke 5.12 in.

1800 R.P.M.   Approx. ¾ load
Avg. barom. 30.6 cm Hg.
Compression ratio 6.3

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1800 R.P.M.   Approx. ¾ load
Avg. barom. 63.4 cm Hg.
Compression ratio 7.3

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1800 R.P.M.   Approx. ½ load
Avg. barom. 63.4 cm Hg.
Compression ratio 7.3

Fig.3

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1600 R.P.M.   Full load
Avg. barom. 30.8 cm Hg.

Test 174   8 cylinders
Bore 4.72 in, Stroke 5.12 in.
Compression ratio 5.3

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1400 R.P.M.   Full load
Avg. barom. 30.7 cm Hg.

1800 R.P.M.   Approx. ½ load
Avg. barom. 30.9 cm Hg.

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

Fig.5

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1800 R.P.M.   Full load
Avg. barom. 30.8 cm Hg.

Test 175   8 cylinders
Bore 4.72 in, Stroke 5.12 in.
Compression ratio 6.3

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1600 R.P.M.   Full load
Avg. barom. 30.8 cm Hg.

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1400 R.P.M.   Full load
Avg. barom. 30.8 cm Hg.

Fig.7

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1800 R.P.M.   Full load
Avg. barom. 30.7 cm Hg.

Test 176   8 cylinders
Bore 4.72 in, Stroke 5.12 in.
Compression ratio 7.3

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1600 R.P.M.   Full load
Avg. barom. 30.6 cm Hg.

$\sqrt{\dfrac{288}{T}}$ (I.M.E.P. at +15°C)

1400 R.P.M.   Full load
Avg. barom. 30.8 cm Hg.

Carbureter air temp., °C=(T-273)

Indicated mean effective pressure, lb per sq. in.

Fig. 8

Indicated mean effective pressure, lb per sq. in.

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1800 R.P.M. Approx. ¾ load
Avg. barom. 63.5 cm Hg.

Test 177 8 cylinders
Bore 4.72 in, Stroke 5.12 in.
Compression ratio 8.3

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1400 R.P.M. Approx. ½ load
Avg. barom. 63.2 cm Hg.

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1800 R.P.M. Approx. ½ load
Avg. barom. 63.5 cm Hg.

Carbureter air temp., °C=(T-273)

Fig. 9

Indicated mean effective pressure, lb per sq. in.

Test 177 8 cylinders
Bore 4.72 in, Stroke 5.12 in.
Compression ratio 8.3

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1800 R.P.M. Approx. ¾ load
Avg. barom. 30.8 cm Hg.

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1600 R.P.M. Full load
Avg. barom. 30.5 cm Hg.

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1800 R.P.M. Full load
Avg. barom. 30.5 cm Hg.

Carbureter air temp., °C=(T-273)

Fig. 10

Indicated mean effective pressure, lb per sq. in.

Test 177 8 cylinders
Bore 4.72 in, Stroke 5.12 in.
Compression ratio 8.3

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1400 R.P.M. Approx. ¾ load
Avg. barom. 30.7 cm Hg.

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1400 R.P.M. Approx. ½ load
Avg. barom. 30.7 cm Hg.

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)
1800 R.P.M. Approx. ½ load
Avg. barom. 30.8 cm Hg.

$\sqrt{\frac{288}{T}}$ (I.M.E.P. at +15°C)

Carbureter air temp., °C=(T-273)

Fig. 11

Lb air per hr. ÷ lb air per hr at carbureter air temp., of 15°C

Test 183, 184 6 cylinders
Bore 6.625 in, Stroke 7.50 in.
1400 R.P.M. Full load

Avg. barom. 35.2 cm Hg.
Compression ratio 6.5
$\sqrt{288/T}$

Avg. barom. 35.2 cm Hg.
Compression ratio 5.5
$\sqrt{288/T}$

Avg. barom. 51.6 cm Hg.
Compression ratio 6.5
$\sqrt{288/T}$

Avg. barom. 51.6 cm Hg.
Compression ratio 5.5
$\sqrt{288/T}$

Avg. barom. 73.9 cm Hg.
Compression ratio 6.5
$\sqrt{288/T}$

Avg. barom. 73.9 cm Hg.
Compression ratio 5.5
$\sqrt{288/T}$

Carbureter air temp., °C=(T-273)

Fig. 12

Lb air per hr. ÷ lb air per hr. at carbureter air temp., of 15°C

Test 186, 187 8 cylinders
Bore 5.51 in, Stroke 5.91 in.
1800 R.P.M. Full load

Avg. barom. 35.2 cm Hg.
Compression ratio 6.5
$\sqrt{288/T}$

Avg. barom. 35.1 cm Hg.
Compression ratio 5.4
$\sqrt{288/T}$

Avg. barom. 51.5 cm Hg.
Compression ratio 6.5
$\sqrt{288/T}$

Avg. barom. 51.4 cm Hg.
Compression ratio 5.4
$\sqrt{288/T}$

Avg. barom. 73.0 cm Hg.
Compression ratio 6.5
$\sqrt{288/T}$

Avg. barom. 74.7 cm Hg.
Compression ratio 5.4
$\sqrt{288/T}$

Carbureter air temp., °C=(T-273)

Fig. 13

Lb air per hr. ÷ lb air per hr. at carbureter air temp., of 15°C

Test 154, 155 12 cylinders
Bore 5.00 in, Stroke 7.00 in.
1700 R.P.M. Full load

Avg. barom. 30.1 cm Hg.
$\sqrt{288/T}$

Avg. barom. 45.0 cm Hg.
$\sqrt{288/T}$

Carbureter air temp., °C=(T-273)

THEORETICAL CONSIDERATIONS.

The next step is to consider the manner in which changes in carburetor air temperature bring about changes in power and to see to what extent this experimentally determined method of correcting power measurements to a standard temperature can be justified theoretically. Variation in engine power must in all cases arise either from (1) change in power per unit weight of charge per unit time (degree to which unit weight of charge supplied at a given rate is utilized), or (2) variation in weight of charge received by engine per unit time.

The thermal efficiency of an engine, the ratio between the energy developed with and the calorific power of unit weight of charge depends almost entirely upon the expansion ratio as does the so-called air cycle efficiency with which it is usually compared. Air cycle efficiency is given by the expression

$$E = 1 - \frac{1}{r^{n-1}}$$

where

$E =$ air cycle efficiency.

$r =$ expansion ratio.

$n = 1.4$ for air.

It will be noted that air temperature does not enter into the above expression for thermal efficiency.

However, in actual operation there are at least two ways in which the energy derived from unit weight of charge might be affected by the temperature of the air entering the carburetor. In the first case the energy per unit weight of charge might be less with a high temperature than a low temperature because the proportion of spent gases to new charge is greater under such conditions. This is a direct consequence of the reduction in weight of charge with increase in entrance air temperature. The less the weight of fresh charge the greater will be its temperature when mixed with the spent gases in the clearance space. The higher the temperature of the gas at the end of the compression stroke, the lower will be the ratio of explosion pressure to compression pressure. Moreover, the greater the ratio of inert gas to fuel, the less will be the increase in temperature produced by combustion and this further decreases the ratio of explosion to compression pressure. Although in some cases the experimental results showed indications of this tendency, namely for the energy output per pound of air to decrease with increase in temperature in general the I.HP per pound of air per hour remained constant. This conclusion is strengthened by the fact that no great differences in efficiency are observed ordinarily between full and part throttle runs where the difference in charge weights is greatly in excess of that ever produced by changes in carburetor air temperature.

Another way in which the energy derived from unit weight of charge might be influenced by the entrance air temperature is dependent upon the influence of this temperature upon the vaporization of the fuel. A change in entrance air temperature may change the amount of fuel that remains to be vaporized during the compression stroke. Any influence of this sort would tend to decrease the power output per pound of air with decrease in carburetor air temperature. With fuels in common use, however, the loss in power that would be entailed even if all the heat of vaporization came from the heat of compression is extremely small. Furthermore, in present day engines a full load it is probable that the heat given the charge by the clearance gases and by the combustion chamber walls is sufficient to vaporize any fuel not vaporized be-

Weight of one cu ft of air at -20°C

Weight of one cu ft of air at +40°C

Volume received by engine when entrance air is -20°C

Volume received by engine when entrance air is +40°C

Weight received by engine when entrance air is -20°C   Fig.14   Weight received by engine when entrance air is +40°C

Effect of change in air temperature.
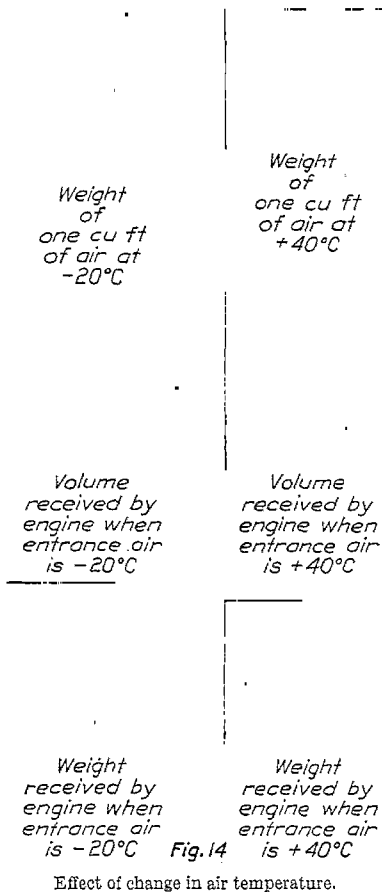
fore reaching the cylinder. It seems unlikely therefore that the power developed per pound of air per hour should change appreciably from this influence.

If the efficiency is unaffected by entrance air temperature as has just been shown for all practical purposes to be the case, then any change in power caused by a change in carburetor air temperature must be caused by and be directly proportional to a change in the weight of charge received by the engine in unit time.

AIR FLOW.

The next step is a consideration of how much the weight of air received by the engine in unit time is changed by a change in temperature. This weight is the product of the volume received in unit time and the density of the air, namely, its weight per unit volume. The volume occupied by the fuel is so small that no appreciable error is introduced by considering in this discussion the air alone rather than the air and fuel. The density of a gas varies inversely as its absolute temperature. This narrows the question to the effect of changes in carburetor air temperatures upon the volume flowing in unit time. Although equations for air flow are given in a great variety of forms, an examination will reveal that all show the velocity of flow to vary directly as the square root of the absolute temperature.[1] Hence the volume flowing in unit time at an absolute temperature of A degrees is $\sqrt{\dfrac{A}{B}}$ times the volume flowing in unit time at an absolute temperature of B degrees. It remains to investigate the factors determining the pressure difference which produces the flow.

The flow into the engine may be considered as the result of the following events, occurring in the sequence mentioned: (1) Motion of the piston causing an increase in the volume above the piston; (2) a decrease in the pressure of the gases above the piston caused by their expansion to fill the increased volume; (3) flow caused by the difference between the pressures within and without the cylinder. What is the effect of a change in entrance air temperature? Obviously the motion of the piston is not changed. Moreover, the percentage change in pressure resulting from adiabatic expansion of a gas depends solely upon the percentage change in volume and is independent of the temperature of the gas. Hence what might be termed the external conditions governing flow are uninfluenced by a change in entrance air temperature. It does not follow that the average pressure difference producing flow is unaffected by a change in entrance air temperature. The explanation follows. Flow into the cylinder takes place because the pressure within the cylinder is less than atmospheric. As a consequence of this flow the difference in pressure is reduced. A reduction in the pressure difference causes a reduction in flow.

It is apparent, therefore, that the pressure difference producing flow during any portion of the stroke depends not only upon the external conditions mentioned above, but also upon the flow that has occurred earlier in the stroke. An increase in the flow in any interval of time such as would result from an increase in entering air temperature would reduce the pressure difference causing flow and hence in so far as the flow is governed by the pressure head would reduce the flow in the next interval of time. An illustration of rather exaggerated conditions will emphasize the fact that some condition of this sort must exist. Suppose the areas of all the passages in the induction system of an engine now having a volumetric efficiency of 70 per cent to be doubled. No one would expect a volumetric efficiency of 140 per cent despite the fact that with a given pressure difference the amount of flow depends directly upon the area of opening.

From the foregoing it is evident that although with any pressure difference air flow (on a volume basis) varies directly as the square root of the absolute temperature, the influence of this effect in an actual engine is dependent upon the magnitude of the pressure difference which produces the flow. If this pressure difference be large it will be affected but little by a change

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[1] An analytical treatment of air flow into an engine cylinder is given by Mr. E. C. Kemble in National Advisory Committee for Aeronautics Report No. 50, entitled "Calculation of Low Pressure Indicator Diagrams."

in flow. Hence a change in temperature will cause a change in volume rate of flow more nearly in proportion to the square root of the absolute temperature than when the pressure difference is small, namely, when the volumetric efficiency is high.

In Figure 15 are shown pressures prevailing during the suction stroke of an engine. These values are based upon an actual i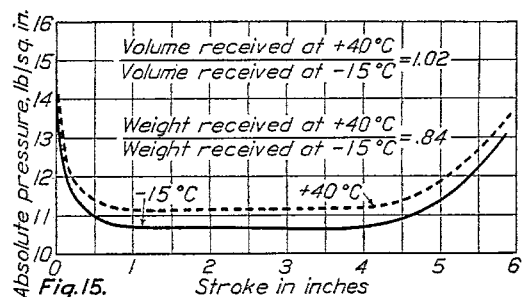ndicator.card but have been modified to eliminate changes in pressure due to heating the charge after it entered the cylinder or to "ramming" effects. The full line shows the pressure at −15° C. The dotted curve shows the pressures that would result if the entrance air temperature were +40° C. This curve was calculated from the full line curve according to the method described in Appendix 1.

Fig. 15.   Effect of changing entrance air temperature upon pressure during suction stroke.

On a volume basis the actual increase in flow with the higher temperature was only 2 per cent while the increase would have been 10 per cent if the pressure difference at each point in the stroke had been the same at +40° C. as at −15° C. It must be concluded therefore that the fact that the volume of air flow in engines does increase as the square root of the absolute temperature can only be accounted for partially by the fact that with any pressure difference air flow varies as the square root of the absolute temperature.

There must then be some other factor which affects the amount of charge received by the engine to an extent dependent upon the entering air temperature. This factor appears to be the heating of the charge in its passage to the engine. As the charge passes through the induction system it receives or loses heat and its velocity must be affected thereby.

The amount of heating that takes place even in aviation engines where every effort is made to make the induction system short and unrestricted is quite appreciable. What is of most importance as far as this report is concerned is that the amount of this heating is considerably influenced by the entering air temperature. In the upper part of Figure 16 is shown for one of the engines used in these tests the difference between the temperature of the charge when it enters the carburetor and when it leaves the manifold. These measurements were made with the engine driven by the dynamometer and with the fuel supply shut off. It will be observed that there was an increase in temperature of approximately 15° C. at an entrance temperature of −15° C. and of only 5° at an entrance temperature of +40° C.

In Figure 16 are also plotted temperature changes measured when the engine was operating under its own power. The temperature decrease,

Fig. 16.

Engine used in test, 8-cylinder, bore 4.72 inches, stroke 5.12 inches.

or more properly the absence of a temperature increase shown by these curves, may be attributed at least in part to the heat withdrawn from the charge in vaporizing the fuel. There is always some doubt as to the validity of measurements of manifold temperatures when liquid fuel is present. Readings may be influenced by liquid depositing on and subsequently vaporizing from the thermocouple. However, such measurements give at least qualitative evidence that a change in the amount of heat supplied the charge accompanies a change in entrance air temperature.
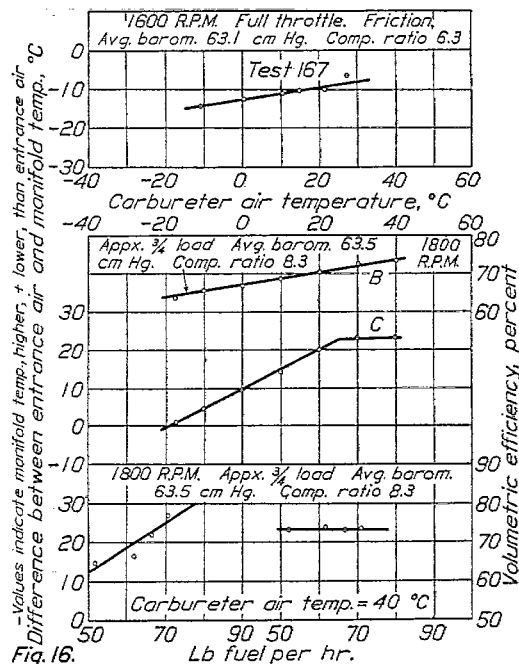
In general an improvement in volumetric efficiency would be expected from any increase in the amount of vaporization taking place before the closing of the inlet valve. The reason follows: With entrance air temperatures ranging from −20 to +40° C. and with a mixture ratio of 14 pounds of air per pound of aviation gasoline, complete vaporization of the fuel would reduce the charge temperature 20° C. provided the charge furnished all the heat for this vaporization. Because of the greater volume occupied by the fuel vapor than by the liquid, the pressure would tend to increase while a decrease in pressure would result from the decrease in temperature. The net result in the case cited would be a decrease in pressure of about 5 per cent. If, in the above case, all the heat of vaporization were supplied from external sources an increase in pressure of at least 2 per cent would result. In actual practice the heat of vaporization is usually supplied in part by the charge and in part externally. Hence increasing the amount of vaporization prior to the closing of the inlet valve would be expected to increase very slightly the volumetric efficiency.

Judged from an experimental standpoint the amount of vaporization would seem to have little or no effect on the volumetric efficiency. Curves B and C of Figure 16 are of interest in this connection. It will be noted that curve B shows an abrupt change of slope at a carburetor air temperature of about 26° C. Presumably at this temperature practically all of the fuel was vaporized before passing the point where the manifold temperature was measured. At lower temperatures an increase in carburetor air temperature was accompanied by an increase in the amount of fuel vaporized and the drop in temperature was thus increased. At higher temperatures since there could be no increase in the amount of fuel vaporized, no such change could occur. The chief item of interest is that there is no corresponding break in the curve of volumetric efficiencies, curve C. This comparison may not be as significant as at first appears for the reason that at all of the entrance temperatures vaporization may have been complete by the time the charge reached the cylinder. If this were the case the change in volumetric efficiency would not be due primarily to a difference in the location at which the heat interchange took place. The lower curves are not open to this objection. That at the left shows the difference in temperature drop that resulted from a change in air-fuel ratio. That at the right shows that no corresponding change in volumetric efficiency occurred. Although a difference in the amount of fuel vaporized must have resulted from each change in mixture ratio, there was evident no change in volumetric efficiency.

The discussion that has immediately preceded serves merely to explain qualitative results that have been already obtained experimentally. Its justification lies in the fact that even qualitative information as to the relative influence of the various factors that affect the change in power with change in entrance air temperature enables one to estimate the range of application of any factor experimentally determined. Thus in the case of the heat interchange that takes place as the charge passes from the carburetor to the cylinder, it is more important to know how much the amount of this heat interchange will differ with different engines than to know what the actual amount with any one engine is. Although more heat than that necessary to vaporize all the fuel might be supplied with a view to increasing the rate of vaporization, such a course is not probable, in aviation practice at least, because of the decrease in maximum power it would entail.

The foregoing analysis may be summarized as follows: With an increase in entrance air temperature the weight per cubic foot of charge decreases, varying inversely as the absolute temperature. The volume of charge received by the engine in unit time increases from two principal causes (1) because if the pressure difference were the same at two temperatures the velocity of flow would vary as the square root of the absolute temperature; and (2) because the amount of heat received by the charge (in its passage to the cylinder) is decreased or the amount of heat lost by the charge is increased with increase of entering air temperature. Both (1) and (2) are affected by engine design but changes which increase the former usually decrease the latter. Moreover, it is unlikely that engine designs will differ sufficiently to cause wide variations in either of these influences. Theoretical considerations thus support the experimental evidence that the effect upon volumetric efficiency of a change in carburetor air temperature does not differ greatly with different engines.

APPLYING THE CORRECTION FACTOR.

The foregoing indicates that corrections made according to the assumption that the power varies inversely on the square root of the absolute temperature will be extremely close in the majority of cases. A few words regarding the use of the correction factor are appropriate. First of all it should be noted that it applies strictly to indicated horsepower. There is no measurable change in friction horsepower with change in carburetor air temperature and this must be considered when correcting brake horsepower since brake power is the difference between I.HP and F.HP. Assume an engine operating at certain speed at which the friction horsepower is 40. Suppose it develops 400-brake horsepower at an entrance air temperature of $-20°$ C. and the brake horsepower it would develop at $+40°$ C. is desired. The first step is to determine the indicated horsepower which is the sum of the brake horsepower and the friction. In this case, I.HP = B.HP + F.HP = 400 + 40 = 440. Since the I.HP varies inversely as the square root of the absolute temperature, the I.HP at $+40°$ C. will be $\sqrt{\dfrac{273-20}{273+40}}$ times 440 which equals 396. Since the friction horsepower will still be the same, namely 40, the brake horsepower will be 396 − 40, which equals 356.

If a great deal of work of this sort were to be done, a chart such as that shown in Figure 17 might prove convenient. This chart is primarily for correcting brake horsepower measurements to a standard entrance air temperature of $+15°$ C. It, of course, requires knowledge of the friction horsepower but the brake horsepower correction factor can be read directly from the chart using the curve for the ratio of F.HP to B.HP corresponding to the case under consideration.

Assume as before that B.HP = 400; F.HP = 40 and carburetor air temperature = $-20°$ C. If it is desired to correct the B.HP to an air temperature of $15°$ C. the method is as follows: $\dfrac{F.HP}{B.HP} = \dfrac{40}{400} = \dfrac{1}{10}.$ From Figure 17 select the proper correction to $-20°$ C. and a .10 ratio of F.HP to B.HP. This factor is 0.932, which multiplied by 400 gives the desired value, namely 373. If the power at a temperature of $+40$ is desired as in the first example, another step is involved, namely, the determination of the factor giving the ratio between the power at $+15°$ C. and $+40°$ C. This is selected in the same way as before and is found to be 1.047. Evidently $\dfrac{B.HP \text{ at } 40° \text{ C.}}{B.HP \text{ at } -20° \text{ C.}} = \dfrac{B.HP \text{ at } +15}{B.HP \text{ at } -20} \div \dfrac{B.HP \text{ at } +15}{B.HP \text{ at } +40} = 0.932 \div 1.047 = 0.890.$ Multiplying 400×0.890 gives 356 as before.

Frequently the density (weight per unit volume) of the air at the entrance to the carburetor is specified instead of its pressure and temperature. A change of density may be brought about by a change in pressure, temperature, or both. If the temperature remains constant and the change in density is due to a change in pressure then the change in power will be directly proportional to the change in density. If, on the other hand, the pressure remains constant and the change in density is due to a change in temperature, then the change in power will be proportional to the square root of the density. There will be less likelihood of error if before making corrections the pressures and temperatures are specified and the corrections applied independently considering the I.HP to vary directly as the barometric pressure and inversely as the square root of the absolute temperature.

The above calculations have been based on the assumption that the adjustment was such as to give the same mixture ratio or that giving maximum power in each instance.[2] If this is not the case no general correction factor is applicable. It must be remembered that leaving carburetor adjustments unchanged when changing the entrance air temperature does not prevent but usually produces a change in air fuel ratio. For a given pressure difference less weight of air flows the higher its temperature. If now the air temperature be increased and the temperature of the fuel remains unchanged then at the same pressure as before the fuel flow will be the same and hence the mixture will become richer.

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[2] This assumes that the change in temperature does not cause large differences in distribution.

Figure 18 shows some measurements of fuel flow when only the temperature of the carburetor air was changed.   It will be observed that for a given position of the mixture ratio control lever the same weight of fuel flowed regardless of the carburetor air temperature.   An increase in carburetor air temperature is accompanied usually by an increase in temperature of the fuel. Provided the decrease in viscosity has a greater influence on the flow than the decrease in density the fuel flow, on a weight basis, will increase.   This is usually the case and the mixture becomes enriched from this source also.   The magnitude of the viscosity effect is governed largely by the design of the fuel nozzle.   This is discussed to some extent in Report No. 49 of the National Advisory Committee for Aeronautics in that section entitled "Discharge Characteristics of Fuel Metering Nozzles in Carburetors."   Some experiments were made more recently at this bureau in which a mixture of 40 per cent benzol and 60 per cent gasoline was flowed through a jet actually mounted in a carburetor.   A head of fuel was maintained practically equivalent to the pressure difference existing at the jet under full throttle operation.
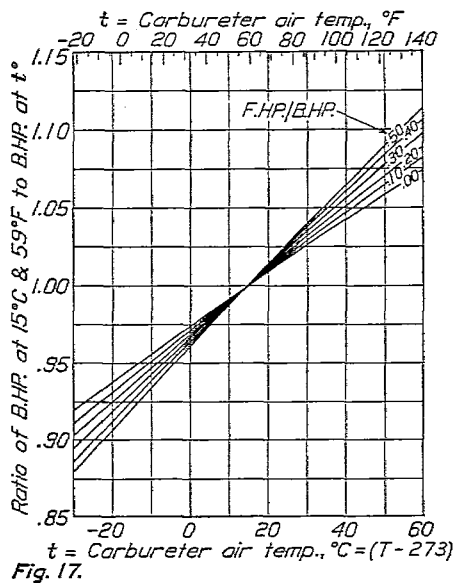
Fig. 17.

Temperature correction factor for B.HP. These curves are based on the assumption that I.HP varies inversely as the square root of the absolute temperature.
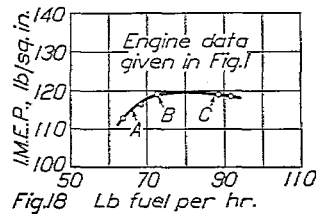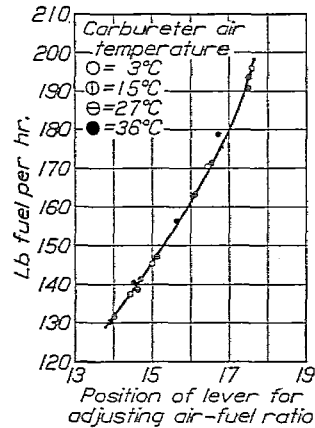
Fig. 18.

Test 187, 8-cylinder engine, bore 5.51 inches, stroke 5.91 inches, compression ratio 5.4.

The weight of fuel flow decreased about 9 per cent when the temperature of the fuel was reduced from 5° C. to −25° C.   The increase in air flow for such a change would be about 6 per cent.   Under such conditions the mixture ratio would be $\frac{1.06}{.91}=1.16$ or 16 per cent leaner than before.   The effect upon the power development of an engine of such a change on mixture ratio depends upon what mixture existed before the change took place.   Consider the figure in the lower part of Figure 18.   If the mixture before the change took place was that giving the I.M.E.P. shown at A, the enrichening would cause an appreciable increase in power.   If it were that shown at B there would be no considerable increase in power, but a marked decrease in economy.[3]

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[3] With the air port control type of carburetor, the square root correction factor is not directly applicable to power measurements even when the mixture is adjusted for maximum power. Carburetors of this type are designed so that more charge weight is supplied the engine as the charge is made leaner. Maximum power is obtained when the increase in charge weight produced by further movement of the adjusting lever would be offset by the decrease in power producing ability of the charge. Assume such a carburetor adjusted for maximum power. An increase in carburetor air temperature will increase the richness of the mixture and the power producing ability of unit weight of the charge. This will partially offset the decrease in charge weight resulting from the higher temperature. Moreover, if an adjustment is then made for maximum power the weight of charge will be increased still more. Hence one might actually get an increase in power with increase in temperature. The general principle of these carburetors is described in Technical Note No. 39 of the National Advisory Committee for Aeronautics entitled "High Thermal Efficiency in Airplane Service."

It is generally recognized that a change in entrance air temperature will change the rate of vaporization and for that reason will change the distribution. A natural question is "Why does not this affect the amount of power developed?" The curve in the lower part of Figure 18 throws light on this also. Consider this curve to represent the conditions in a single cylinder of the multi-cylinder engine since its shape is very similar to that of curves obtained in a similar manner with single-cylinder engines. With perfect distribution and the mixture adjusted for maximum power of the engine each cylinder would receive the same amount of fuel. For an 8 cylinder engine this would be one-eighth of the amount shown for point B. If now the distribution becomes so poor that some cylinders must receive the mixture shown at C in order that others may receive the mixture at B, the fuel consumption will be greatly increased but the power will not be changed to any extent.

CONCLUSIONS.

It is concluded that the assumption that the indicated power of an engine varies inversely as the square root of the absolute temperature has sufficient justification to warrant its adoption as a basis for correcting horsepower measurements to a standard temperature. The factor is only applicable when the air-fuel ratio is adjusted to give the same mixture ratio or a maximum power mixture ratio at each temperature. Its range of application is believed to extend at least from entrance air temperature of $-40°$ C. to $+60°$ C. and there is no evidence that it would not be applicable at higher temperatures.

APPENDIX I.

CALCULATION OF CURVES SHOWN IN FIGURE 15.

As stated in the text, the full line curve of Figure 15 represents pressures existing during the suction stroke of an engine with, in this case, a carburetor air temperature of $-15°$ C. The curve is based upon an actual indicator card but the pressure values have been modified to eliminate any effect due to heating of the charge after it had entered the cylinder or to the "ramming" action of the gas. The dotted curve shows the pressures that would exist if the entrance air temperature were $40°$ C. other conditions remaining the same. It was calculated as follows: Piston positions were selected as indicated in column A of Table I and the flow was calculated for the intervals in which the piston passed from one position to the next. The first step was the calculation of the values given in columns A, B, C, D, E, F, and G of Table I, all of which are related to the curve for an entrance air temperature of $-15°$ C. Stroke and pressure, columns A and B, are read directly from the full line curve.

Assuming a piston of unit area makes the stroke volume column C the same as the stroke column A. The equivalent volume at an atmospheric pressure of 14.7 pounds per square inch is calculated on the assumption that the charge has expanded adiabatically from atmospheric pressure to the pressure in the cylinder according to the well-known relation $p_1 v_1^{1.4} = p_2 v_2^{1.4}$ in which p_1 is 14.7; p_2 the pressure given in column B; v_1 the volume to be determined for column D; and v_2 the volume given in column C. Column E gives the volume received during each interval and is also based upon a standard barometric pressure of 14.7. Values in column E are derived from column D merely by subtracting from the total volume at any stroke position the total volume at the stroke position immediately above it. Pressure differences given in column F are obtained by subtracting from 14.7 the pressure recorded in column B. Taking the square root of the values in column F gives the values of column G.

The values in the remaining columns result from calculating the flow when all conditions are the same except that the carburetor air temperature is $+40°$ C. instead of -15. For reasons fully explained in the text the volume of flow at any pressure difference varies directly as the square root of the absolute temperature. It is also explained that with a change in flow there will be a change in pressure difference. Moreover, it is known that the velocity of flow varies directly as the square root of the pressure difference. Hence if the pressure were known for any two piston positions the flow at an entrance air temperature of $+40°$ C. could be determined by multiplying the corresponding flow with an entrance air temperature of $-15°$ C. by the product of $\sqrt{\dfrac{273+40}{273-15}}$ and the ratio of the average of the square roots of the pressure heads at the two piston positions for the carburetor air temperature of $+40°$ C. to the similar average for $-15°$ C.

The scheme here employed was to guess the pressure head at each stroke position and then calculate the flow as mentioned above. Having obtained the flow the pressure head could be calculated and if it did not agree with the "guessed" value the process was repeated until satisfactory agreement was obtained. Referring again to Table I, column H shows the guessed pressure differences that proved to be correct. Column I shows the ratio between the flow at $+40°$ C. and $-15°$ C. For the flow while the piston was moving from 0.1 to 0.2 this ratio was as follows:

$$\sqrt{\frac{273+40}{273-15}} \frac{1.4+1.649}{2} \div \frac{1.516+1.76}{2}.$$

The flow as tabulated in column J is the product of the factor in column I and the flow tabulated in column E. Addition of the flow for each interval gives the total flow for column K.

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Column L is calculated according to the relation $p_1 v_1^{1.4} = p_2 v_2^{1.4}$ where $p_1 = 14.7$; p_2 the value desired; v_1 the value given in column K, and v_2 the value given in column C. Plotting the values of column L gives the dotted curve of Figure 15.

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TABLE I.

A	B	C	D	E	F	G	H	I	J	K	L
Stroke in inches.	Pressure in pounds per square inch.	Stroke volume.	Equivalent volume at pressure of 14.7.	Increase in volume during previous interval.	Pressure difference at air temperature of −15° C.	$\sqrt{}$	$\sqrt{}$		Volume flow at +40.	Total flow at +40.	Pressure at +40.
0.00	14.7	0.00	0.000	0.000	0.0	0.00	0.00	0	0.000	0.000	14.7
.10	12.4	.10	.089	.089	2.3	1.52	1.40	1.02	.090	.090	12.7
.20	11.6	.20	.169	.080	3.1	1.76	1.65	1.02	.082	.172	12.0
.30	11.3	.30	.248	.079	3.4	1.84	1.69	1.04	.082	.254	11.7
.40	11.1	.40	.327	.079	3.6	1.90	1.78	1.03	.081	.335	11.5
.50	11.0	.50	.405	.078	3.7	1.94	1.82	1.03	.080	.416	11.4
.75	10.8	.75	.602	.197	3.9	1.97	1.82	1.03	.203	.619	11.4
1.00	10.7	1.00	.797	.195	4.0	2.00	1.90	1.03	.200	.819	11.1
2.00	10.7	2.00	1.595	.798	4.0	2.00	1.88	1.04	.832	1.641	11.2
3.00	10.7	3.00	2.393	.798	4.0	2.00	1.87	1.03	.822	2.463	11.2
4.00	10.7	4.00	3.190	.797	4.0	2.00	1.87	1.03	.822	3.285	11.2
4.50	10.9	4.50	3.640	.450	3.8	1.95	1.82	1.03	.463	3.748	11.4
4.75	11.1	4.75	3.880	.240	3.6	1.90	1.78	1.03	.247	3.995	11.5
5.00	11.4	5.00	4.170	.290	3.3	1.82	1.67	1.03	.298	4.293	11.9
5.10	11.6	5.10	4.300	.130	3.1	1.78	1.61	1.01	.132	4.425	12.1
5.20	11.7	5.20	4.420	.120	3.0	1.73	1.58	1.01	.121	4.546	12.2
5.30	11.8	5.30	4.540	.120	2.9	1.69	1.54	1.01	.121	4.667	12.3
5.40	12.0	5.40	4.680	.140	2.7	1.63	1.48	1.00	.141	4.808	12.5
5.50	12.2	5.50	4.840	.160	2.5	1.56	1.41	1.00	.159	4.967	12.7
5.60	12.5	5.60	4.980	.140	2.2	1.40	1.31	.98	.138	5.105	13.0
5.70	12.7	5.70	5.120	.140	2.0	1.41	1.26	.98	.138	5.243	13.1
5.80	13.0	5.80	5.300	.180	1.7	1.32	1.19	.98	.177	5.420	13.3
5.90	13.2	5.90	5.470	.170	1.5	1.22	1.04	.97	.165	5.585	13.6