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EFFECT OF EXHAUST BACK PRESSURE ON ENGINE POWER

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NACA

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

CONFIDENTIAL BULLETIN

EFFECT OF EXHAUST BACK PRESSURE ON ENGINE POWER

By Benjamin Pinkel

SUMMARY

This bulletin presents data on the effect of exhaust back pressure on engine power for 14 aircraft engines. The data show that the effect of exhaust back pressure on engine power varies with engine speed, range of back pressure and manifold pressure involved, and engine type. The use of a single factor for the variation of power with back pressure was found to be very inaccurate. The only method of accurately determining the effect of back pressure on engine power that can be recommended at present is to test the specific engine type at the engine speed for which the data are desired. A convenient method of presenting these data is described.

INTRODUCTION

This bulletin was prepared at the request of the Subcommittee on Exhaust Gas Turbines of the National Advisory Committee for Aeronautics, and presents data obtained from members of this Subcommittee and from other sources showing the effect of exhaust back pressure on engine power for 14 aircraft engines. The results are presented in terms of nondimensional factors whereby data taken at one inlet manifold pressure may be used to determine the effect at other inlet manifold pressures.

ANALYSIS

A simplified picture of the engine cycle provides some insight into factors that influence the effect of exhaust back pressure on engine power. The equations obtained from the simplified picture will be used as a guide in determining the proper factors for plotting data on the effect of exhaust back pressure on engine power. The four strokes of the engine cycle are shown in figure 1. Although the cylinder pressures are known to vary appreciably during the inlet and exhaust strokes, they are represented in the simplified diagram as constant-pressure lines having pressures equal to the inlet manifold pressure and the exhaust manifold pressure, respectively.

The work W_1 is given by

$$W_1 = 778\ h\eta_t m_e \text{ foot-pounds} \tag{1}$$

where

m_e mass of combustible mixture in engine, slugs
h heat released by combustion per slug of combustible mixture, Btu/slug
η_t thermal efficiency of cycle

The value of $h\eta_t$ varies slightly with engine speed and fuel-air ratio in the range of fuel-air ratios between 0.065 and 0.10.

The work W_2 is given by

$$W_2 = v\left(p_m - p_e\right) \tag{2}$$

where

v displacement volume of engine, cubic feet
p_m inlet manifold pressure, pounds per square foot
p_e exhaust manifold pressure, pounds per square foot

The work W_s done by a geared supercharger in compressing the mixture is

$$W_s = Km_eU^2 \tag{3}$$

where

U supercharger impeller tip speed, feet per second
K coefficient that varies slightly with the volume of air entering supercharger and supercharger speed

If W_f is the work against the mechanical friction of the engine without supercharger, the total net shaft work W per cycle is

$$W = m_e \left(778h\eta_t - KU^2\right) + v\left(p_m - p_e\right) - W_f \qquad (4)$$

But W/v is equal to the net brake mean effective pressure of the engine and supercharger and W_f/v is equal to the friction mean effective pressure of the engine alone. Therefore

$$\frac{\text{bmep}}{p_m} = \frac{m_e}{p_m v}\left(778h\eta_t - KU^2\right) + 1 - \frac{p_e}{p_m} - \frac{\text{fmep}}{p_m} \qquad (5)$$

If it is assumed, for the sake of simplicity, that the clearance volume is completely filled with exhaust gas at a pressure p_e at the end of the exhaust stroke, and if it is also assumed that this exhaust gas is adiabatically compressed from the pressure p_e to the inlet pressure p_m by the incoming charge, then the volume occupied by the residual exhaust gas reduces from v_c to

$$v_c\left(\frac{p_e}{p_m}\right)^{\frac{1}{\gamma_e}}$$

where

v_c clearance volume

γ_e ratio of specific heats for exhaust gas

The volume filled by fresh charge is then

$$v + v_c - v_c\left(\frac{p_e}{p_m}\right)^{\frac{1}{\gamma_e}} = v\left(\frac{r}{r-1}\right)\left[1 - \frac{1}{r}\left(\frac{p_e}{p_m}\right)^{\frac{1}{\gamma_e}}\right]$$

where

r compression ratio of engine

By application of the gas law the mass m_e of fresh charge in this volume is

$$m_e = \frac{p_m}{RT_m}\eta\, v\left(\frac{r}{r-1}\right)\left[1 - \frac{1}{r}\left(\frac{p_e}{p_m}\right)^{\frac{1}{\gamma_e}}\right] \qquad (6)$$

where

η constant of engine taking account of the fact that temperature in cylinder at end of intake stroke differs from inlet manifold temperature
T_m temperature in inlet manifold, $^\circ F$ absolute
r compression ratio of engine
R gas constant of the charge, foot-pounds per slug per $^\circ F$

When equation (6) is substituted into equation (5), there results

$$\frac{\text{bmep}}{p_m} = \frac{\eta}{RT_m}\left(\frac{r}{r-1}\right)\left[1 - \frac{1}{r}\left(\frac{p_e}{p_m}\right)^{\frac{1}{\gamma_e}}\right]\left(778h\eta_t - KU^2\right) + 1 - \frac{p_e}{p_m} - \frac{\text{fmep}}{p_m} \qquad (7)$$

Equation (7) may be written

$$\frac{\text{bmep}}{p_m} = \frac{778h\eta_t - KU^2}{RT_m} f\left(\frac{p_e}{p_m}\right) + 1 - \frac{p_e}{p_m} - \frac{\text{fmep}}{p_m} \qquad (8)$$

In the derivation of equation (6) the assumption was made that sufficient time was available for the pressure in the cylinder to attain inlet manifold pressure at the end of the intake stroke. Although this condition may prevail at low engine speed, it does not necessarily hold at high speed. An additional important variable is obviously the velocity of the gas through the intake port which is roughly proportional to $v_d N/A$,

where

N engine speed, revolutions per second
A flow area at the intake valve, square feet

The expression $v_d N/A$ can be placed in nondimensional form by dividing by a factor proportional to the velocity of sound $\sqrt{RT_m}$.

Equation (8) may be written

$$\frac{\text{bmep}}{p_m} = \frac{778h\eta_t - KU^2}{RT_m} f\left(\frac{p_e}{p_m}, \frac{v_d N}{A\sqrt{RT_m}}\right) + 1 - \frac{p_e}{p_m} - \frac{\text{fmep}}{p_m} \qquad (9)$$

It may be necessary to introduce a parameter for the exhaust valve similar to that presented for the inlet valve.

As in the case of W_f, the quantity friction mean effective pressure in equation (9) represents only the friction of the engine moving parts and differs from the quantity conventionally called friction mean effective pressure. The quantity conventionally called friction is the power required to motor the engine at the same inlet manifold pressure, exhaust back pressure, and engine speed as prevailed in the power run. This quantity is approximately equal to $W_s - W_2 + W_f$. If the friction mean effective pressure corresponding to the conventional friction is written with the subscript c, then

$$\frac{(\text{fmep})_c}{p_m} = \frac{KU^2}{RT_m} f\left(\frac{p_e}{p_m}, \frac{v_d N}{A\sqrt{RT_m}}\right) - \left(1 - \frac{p_e}{p_m}\right) + \frac{\text{fmep}}{p_m}$$

A number of definitions of the indicated mean effective pressure are possible. If the indicated mean effective pressure is defined as $\text{bmep} + (\text{fmep})_c$, then

$$\frac{(\text{imep})_c}{p_m} = \frac{778h\eta_t}{RT_m} f\left(\frac{p_e}{p_m}, \frac{v_d N}{A\sqrt{RT_m}}\right) \qquad (10)$$

This indicated mean effective pressure includes only the indicated work of the compression and expansion strokes W_1. Another indicated mean effective pressure can be defined to include the indicated work of the intake and exhaust strokes W_2 as follows:

$$\frac{\text{imep}}{p_m} = \frac{778h\eta_t}{RT_m} f\left(\frac{p_e}{p_m}, \frac{v_d N}{A\sqrt{RT_m}}\right) + \left(1 - \frac{p_e}{p_m}\right) \qquad (11)$$

It is important that in each case it be realized which of these definitions for indicated mean effective pressure is being used.

It is evident from equations (9) to (11) for a given engine operated at a given engine speed and inlet manifold temperature that $(\text{bmep} + \text{fmep})/p_m$, $(\text{imep})_c/p_m$ and imep/p_m are functions principally of p_e/p_m and, in a plot of these functions, the same curves will be obtained for a given engine speed regardless of whether the back pressure or inlet manifold pressure is varied. A different curve will, however, be obtained for each engine speed. In the usual case the value of friction mean effective pressure is unknown but little dispersion of the data is introduced by plotting bmep/p_m against p_e/p_m when the variation in p_m is small. Different curves for bmep/p_m against p_e/p_m will be obtained for different impeller-gear ratios and when large variations in p_m occur.

The curves for different engines may be expected to be different because of differences in the values of

(1) $v_d N/A$
(2) Ratio of exhaust-valve area to inlet-valve area
(3) Valve timing
(4) Ratio of the tip speed of the geared supercharger to the engine speed
(5) Thermal efficiency
(6) Volumetric efficiency

When a curve of bmep/p_m or imep/p_m is plotted against p_e/p_m it is important that the data be corrected to the same inlet manifold temperature. Equation (10) indicates that the engine power varies inversely as the absolute inlet manifold temperature. This analysis, however, is approximate because it does not take into consideration phenomena that occur in the intake process. Unfortunately, there is no agreement in the literature on the variation of engine power with inlet manifold temperature; various organizations use different correction factors. Further work with a view to standardizing this correction is desirable.

RESULTS AND DISCUSSION

Data on the effect of the exhaust back pressure on engine power have been received from various sources for the following engines:

Wright 1820-G105A
Pratt & Whitney R-1830-64
Allison V-1710-39
Allison XV-1710-6
Daimler-Benz 601-A
Navy XV-715-2
Pratt & Whitney XR-2800-4
Bristol Mercury VI
Bristol Pegasus XVIII
Rolls-Royce Merlin 46
Rolls Royce Merlin II
Bristol Perseus VIII (sleeve valve)
Bristol Perseus XII (sleeve valve)
Junkers Jumo JU-211D

These data are shown in figures plotted as $imep/p_m$ or $bmep/p_m$ against p_e/p_m. The range through which p_e and p_m vary is given in the figures in inches of Hg absolute.

Where large variations in inlet manifold pressure p_m occur in any given set of test data, the curves of $bmep/p_m$ against p_e/p_m do not give an accurate indication of the variation of brake mean effective pressure with exhaust pressure p_e because the term $fmep/p_m$, equation (9), is not a function of p_e/p_m. The curves of $bmep/p_m$ against p_e/p_m are omitted for these cases. This criticism does not apply to the quantity $imep/p_m$; and all the data for which the friction horsepowers were known were therefore plotted on the basis of indicated mean effective pressure regardless of whether p_e or p_m was varied. Good correlation of the data on this basis is noted. The indicated mean effective pressure in these figures includes the contributions of the four strokes of the cycle.

For the purpose of facilitating comparison of the various engines listed, table I shows the change in brake mean effective pressure and indicated mean effective pressure per unit change in back pressure for several representative conditions.

The values were obtained by measuring the slopes of tangents to the curves of figures 2 to 15 at the three values of p_e/p_m given in table I.

In general, the higher the engine speed and the lower the values of p_e/p_m the smaller is the effect of back pressure on engine power. The larger the ratio of impeller tip speed of the geared supercharger to engine speed the smaller is the variation of $bmep/p_m$ with p_e/p_m. The curve of $imep/p_m$ against p_e/p_m has the property that it is unaffected by the impeller tip speed of the geared supercharger (figs. 2(a), 8, 9, 10, and 11) provided that the power is corrected to a constant value of T_m. These conclusions are in accordance with equations (9) and (10).

Large differences in the effect of exhaust back pressure on engine power is noted among the engines listed. These differences are possibly the result of the difference in valve overlap between the engines. Further work is required to provide an insight into the cause of this difference.

CONCLUSIONS

On the basis of the analyses presented, the following conclusions may be made:

1. A single factor for the effect of exhaust back pressure on engine power is very inaccurate.

2. Accurate values for the effect of back pressure on power for an engine of a given designation can be obtained only by testing an engine of the same designation at the engine speed for which the information is desired.

3. The variation of engine power with exhaust back pressure decreases as engine speed and inlet manifold pressure increase and as exhaust back pressure decreases.

4. From a plot of imep/p_m against p_e/p_m data on the effect of exhaust back pressure on engine power obtained at one inlet manifold pressure can be used to calculate the effect of back pressure at other inlet manifold pressures.

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TABLE I. - CHANGE IN BRAKE MEAN EFFECTIVE PRESSURE AND INDICATED MEAN EFFECTIVE PRESSURE PER UNIT CHANGE IN EXHAUST BACK PRESSURE

Engine designation	Engine speed (rpm)	Impeller tip speed / Engine speed for main stage (ft/rev)	$d(bmep)/dp_e$			$d(imep)/dp_e$		
			p_e/p_m			p_e/p_m		
			0.6	1.0	1.4	0.6	1.0	1.4
Wright 1820-G105A	2300	20.6	-2.77	-3.83	-4.78	-2.74	-3.83	-4.20
		28.8	-2.13	-3.13	-3.70			
Pratt & Whitney R-1830-64	1750	26.2	-1.92	-2.49	-2.77			
	2150	26.2	-1.53	-1.92	-2.17			
	2450	26.2	-1.11	-1.63	-1.92			
Allison V-1710-39	2000	21.9	-1.43	-3.83	------			
	2600	21.9	-1.57	-1.57	------			
	3000	21.9	-1.72	------	------			
Allison XV-1710-6	2000	21.9	-2.66	-5.75	------			
	2600	21.9	-1.72	-3.90	-11.50			
	3000	21.9	-1.25	------	------			
Daimler-Benz 601-A	2000	Hydraulic drive	-1.34	-3.00	------			
	2400		-1.06	-2.07	------			
Navy XV-715-2	1500	14.2	------	-7.30	-7.85			
	1500	16.4	------	-6.65	-7.30			
	2000	14.2	------	-6.75	-7.41			
	2000	16.4	------	-5.95	------			
	2500	14.2	------	-6.63	-8.16			
	2500	16.4	------	-5.96	------			
	3100	14.2	------	-6.63	------			
	3100	16.4	------	-5.55	------			
	3500	14.2	------	-5.65	------			
	3500	16.4	-3.00	-5.09	------			
Pratt & Whitney XR-2800-4	1600	21.44				-1.96	-2.50	-2.78
	1800	21.44				-2.33	-2.86	-3.13
	2000	21.44				-2.44	-3.03	-3.23
	2200	21.44				-1.59	-2.27	------
	2400	21.44				-1.18	-1.75	-1.96
Bristol Mercury VI	1800	24.19	-1.81	-3.50	------			
	2200	24.19	------	-3.20	------			
	2400	24.19	------	-2.87	------	------	-2.77	------
Bristol Pegasus XVIII	2250	20.50	-1.68	-3.64	------	-1.95	-3.90	------
	2250	29.42	-1.53	------	------	-1.95	-3.90	------
Rolls Royce Merlin 46	2200	25.82				-1.59	-3.45	------
	2400	25.82				-1.85	-3.13	------
	2600	25.82				-1.64	-2.78	------
	2800	25.82				-1.59	-2.70	------
	2950	25.82				-1.33	-2.56	------
Rolls Royce Merlin II	2400	23.05	-1.96	-3.45	------	-1.96	-3.70	------
	2600	23.05	-2.08	-3.23	------	-2.04	-3.45	------
	2800	23.05	-2.13	-2.94	------	-2.08	-3.03	------
Bristol Perseus VIII	1600	------------------	-1.45	-3.00	-3.00			
	2000	------------------	-1.56	-2.86	-3.12			
	2200	------------------	-1.61	-3.12	-3.12			
	2400	------------------	-1.71	-2.93	-3.33			
	2600	------------------	-1.49	-2.78	-3.23			
	2800	------------------	-1.45	-2.56	-3.23			
Bristol Perseus XII	2200	------------------	-1.67	-3.00	-3.70			
Junkers Jumo JU-211D	1950	19.42	-2.42	-4.12	-5.27			

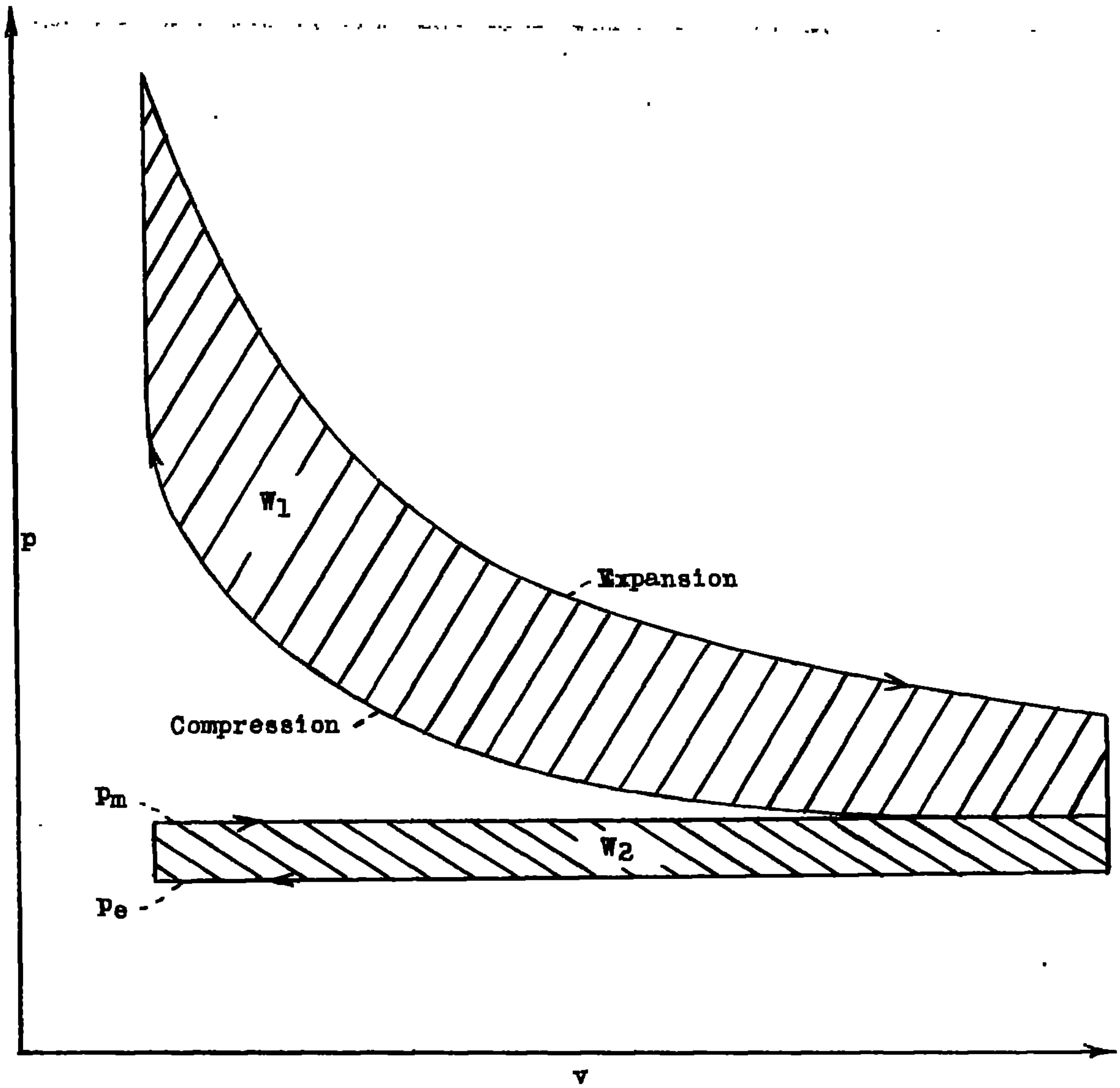
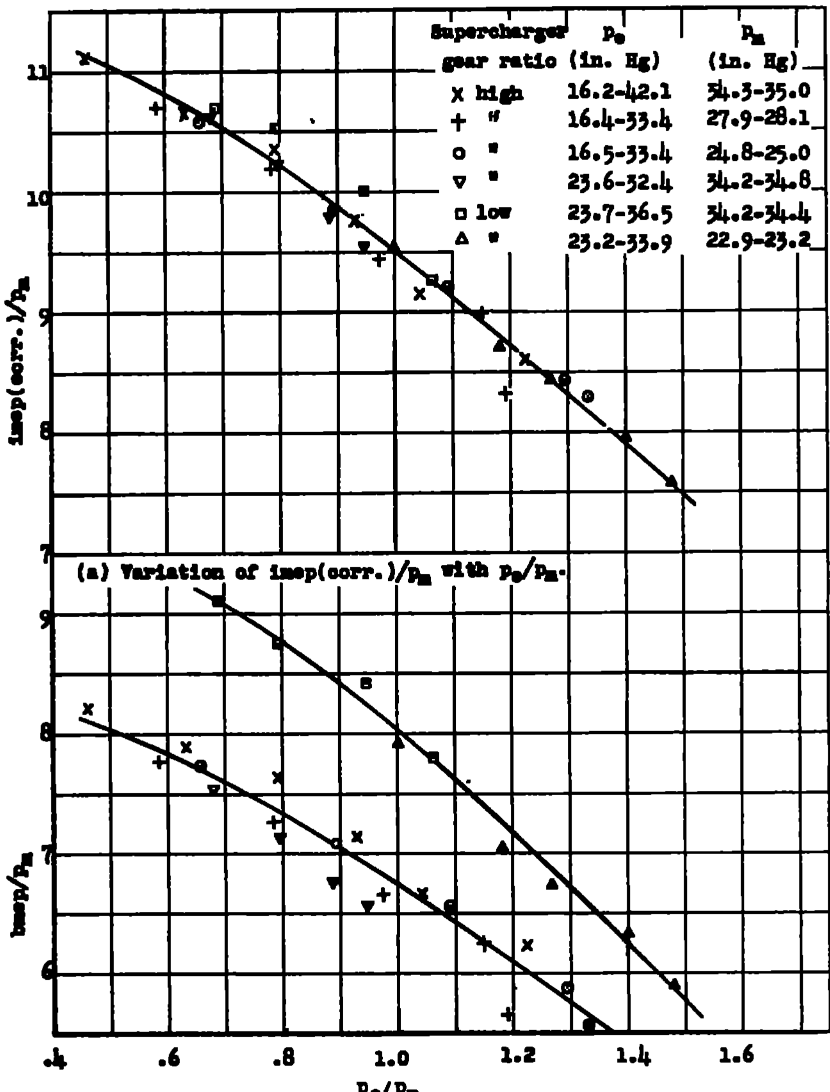


Figure 1.- Engine indicator diagram.



(a) Variation of imep(corr.)/p_m with p_e/p_m.

(b) Variation of bmep/p_e with p_e/p_m

Figure 2. - Effect of exhaust back pressure on engine power for Wright 1820-G105A engine. Engine speed, 2300 rpm; ratio of impeller tip speed to engine speed for high blower, 28.8 feet; for low blower, 20.6 feet; compression ratio, 6.3.

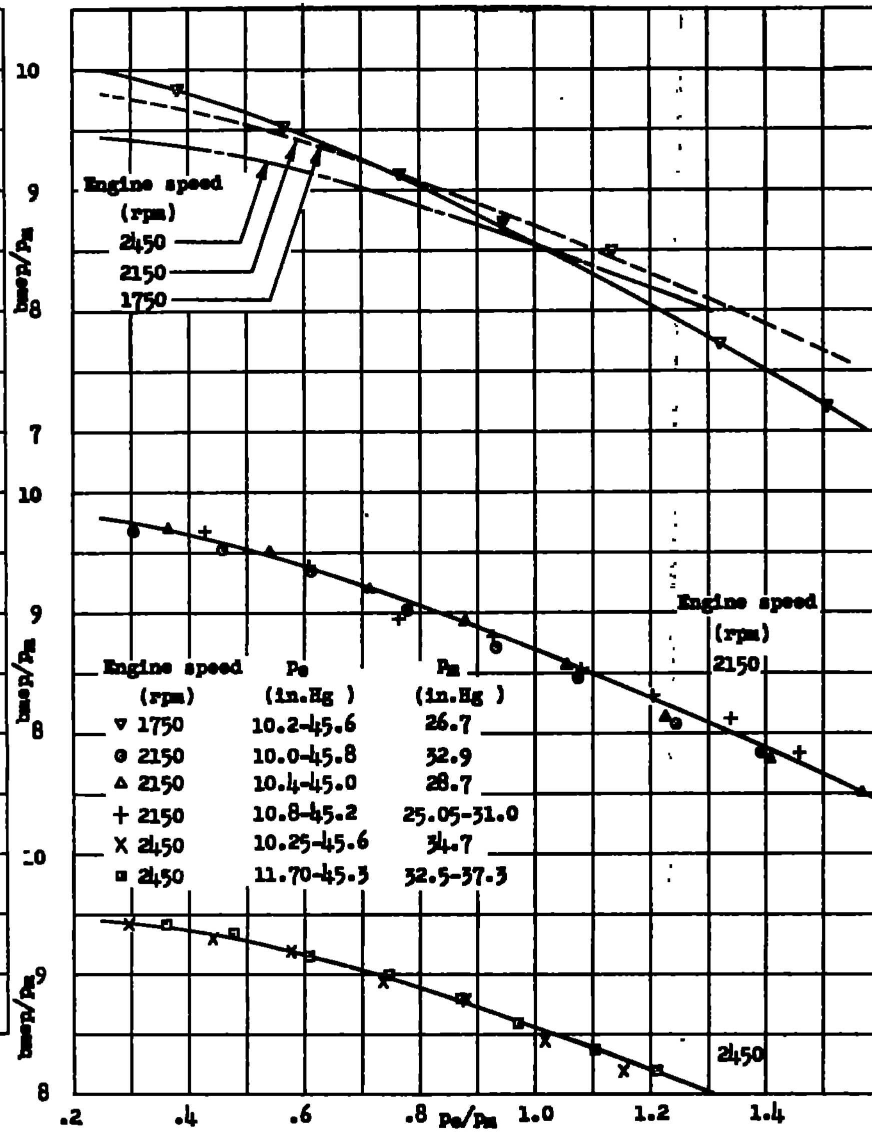


Figure 3. - Effect of exhaust back pressure on engine power for Pratt & Whitney R-1830-64 engine. Ratio of impeller tip speed to engine speed, 26.2 feet; compression ratio, 6.5.

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Figs. 2,3

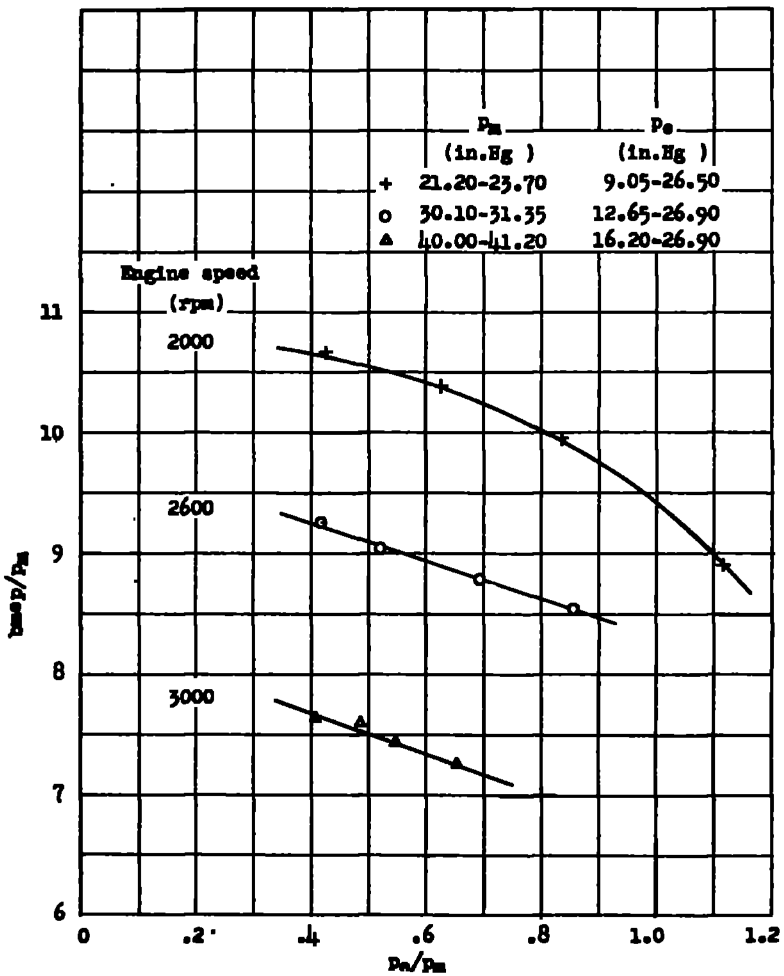


Figure 4. - Effect of exhaust back pressure on engine power for Allison V-1710-39 engine. Ratio of impeller tip speed to engine speed, 21.9 feet; compression ratio, 6.65.

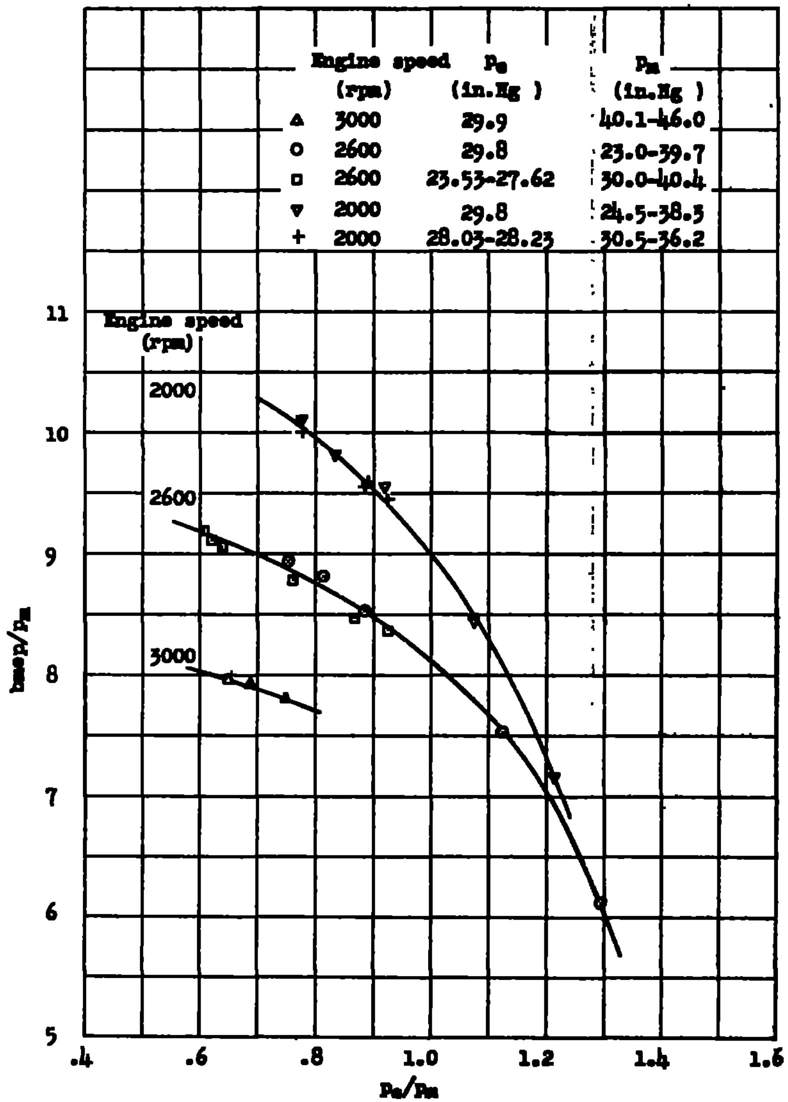


Figure 5. - Effect of exhaust back pressure on engine power for Allison XV-1710-6 engine. Ratio of impeller tip speed to engine speed, 21.9 feet; compression ratio, 6.65.

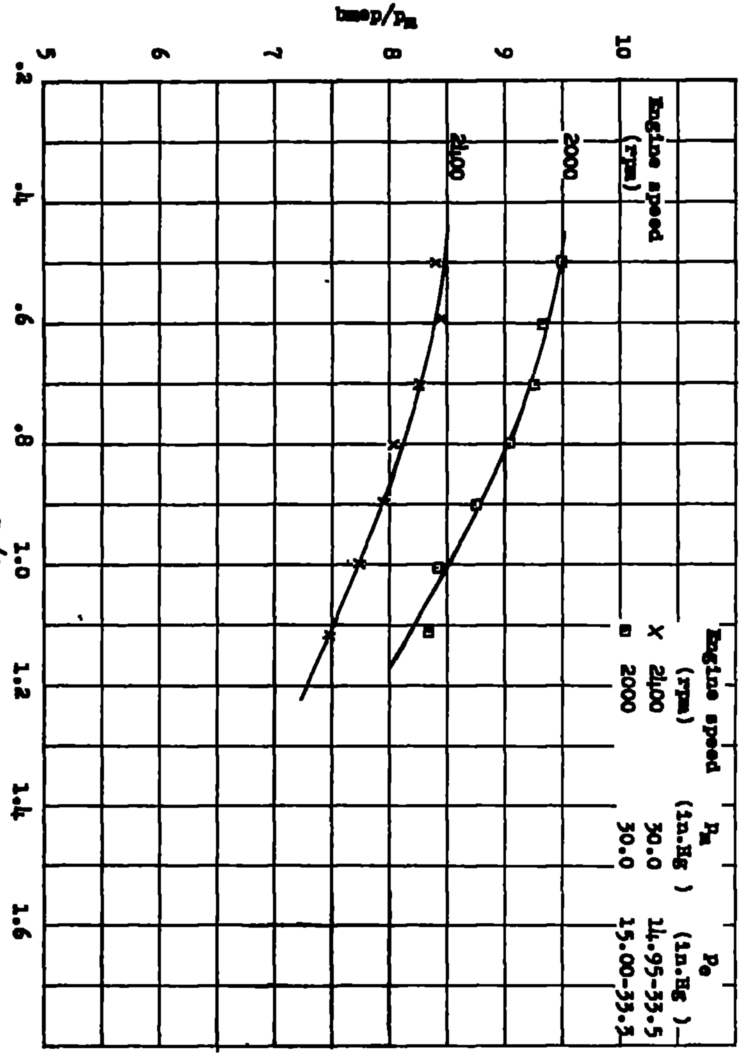
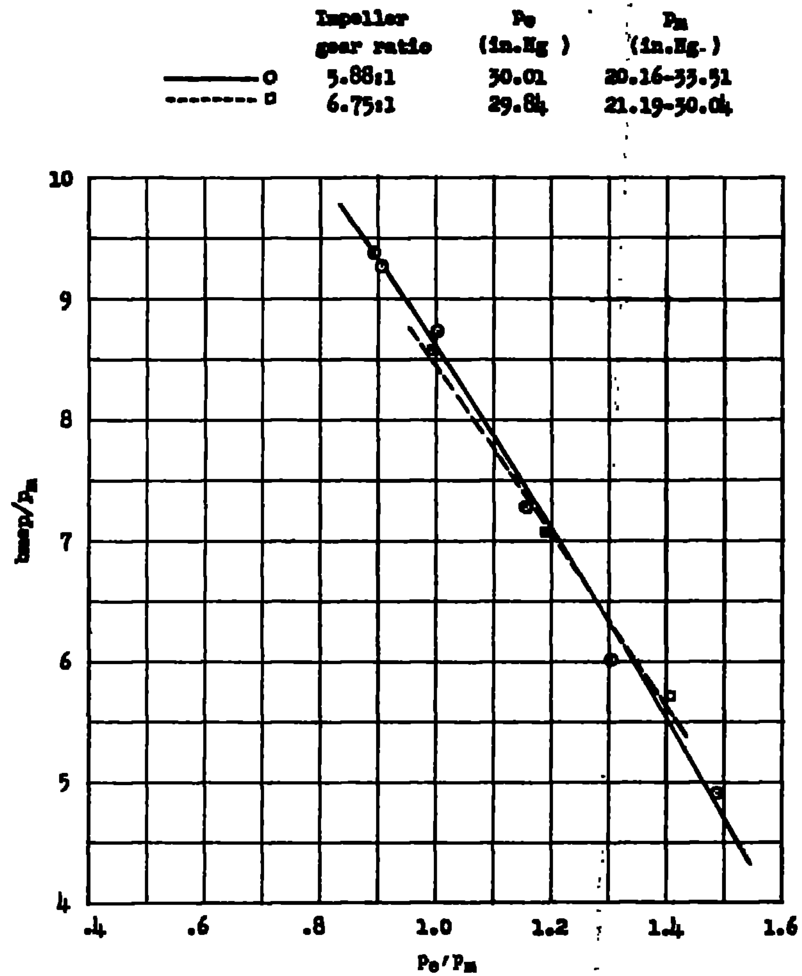
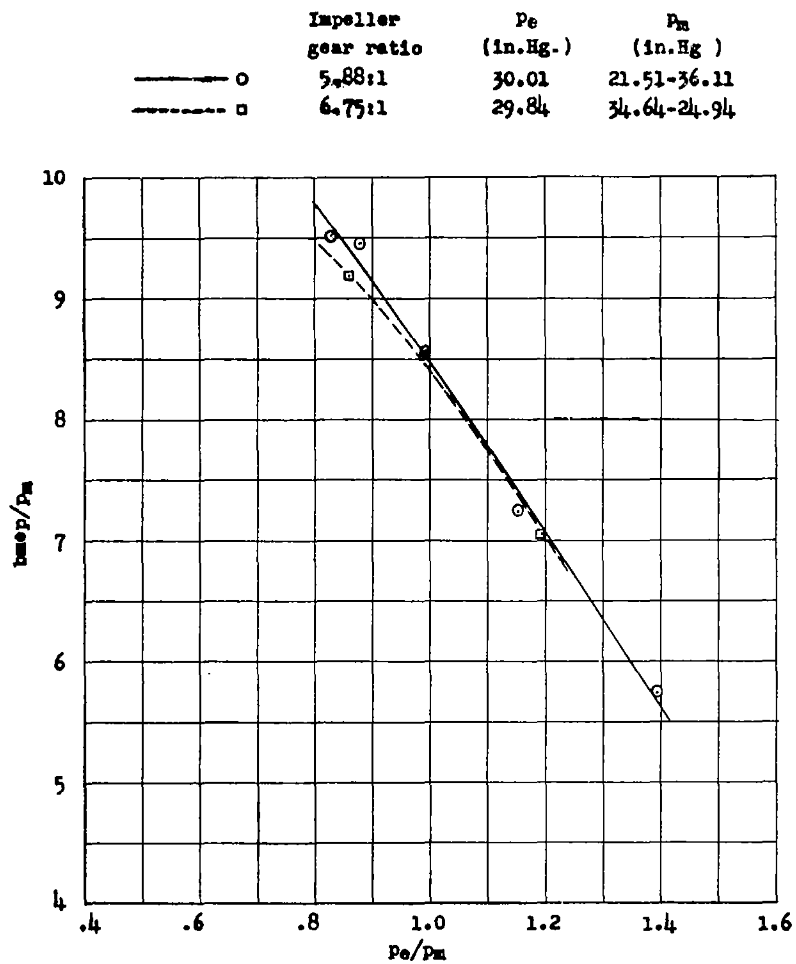


Figure 6. - Effect of exhaust back pressure on engine power for Daimler-Benz 601-A engine. Ratio of impeller tip speed to engine speed, variable; compression ratio, 6.9.



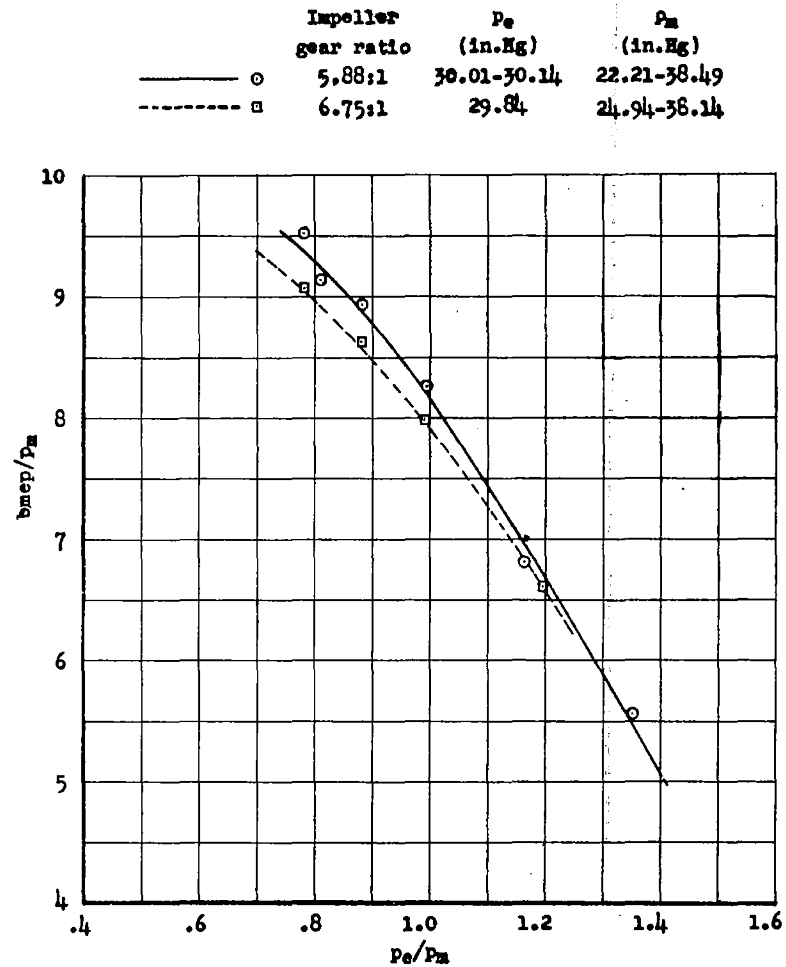
(a) Engine speed, 1500 rpm.

Figure 7. - Effect of exhaust back pressure on engine power for Navy XV-1710-2 engine. Ratio of impeller tip speed to engine speed for low gear, 14.2 feet; for high gear, 16.4 feet; compression ratio, 7.5.



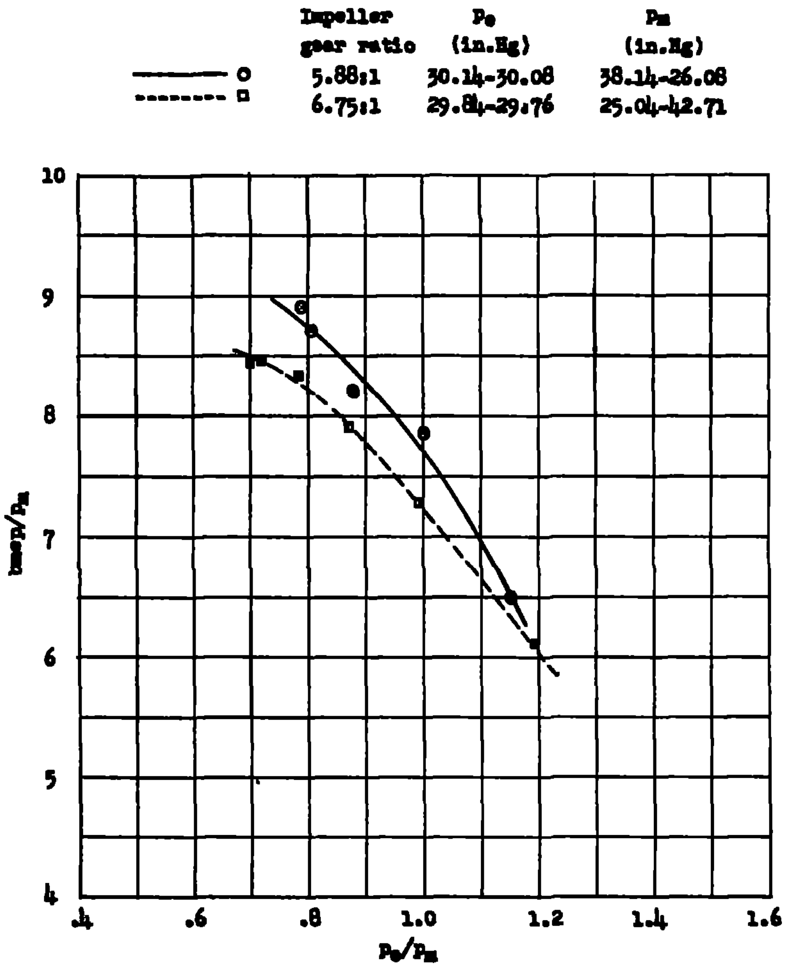
(b) Engine speed, 2000 rpm.

Figure 7. - Continued.

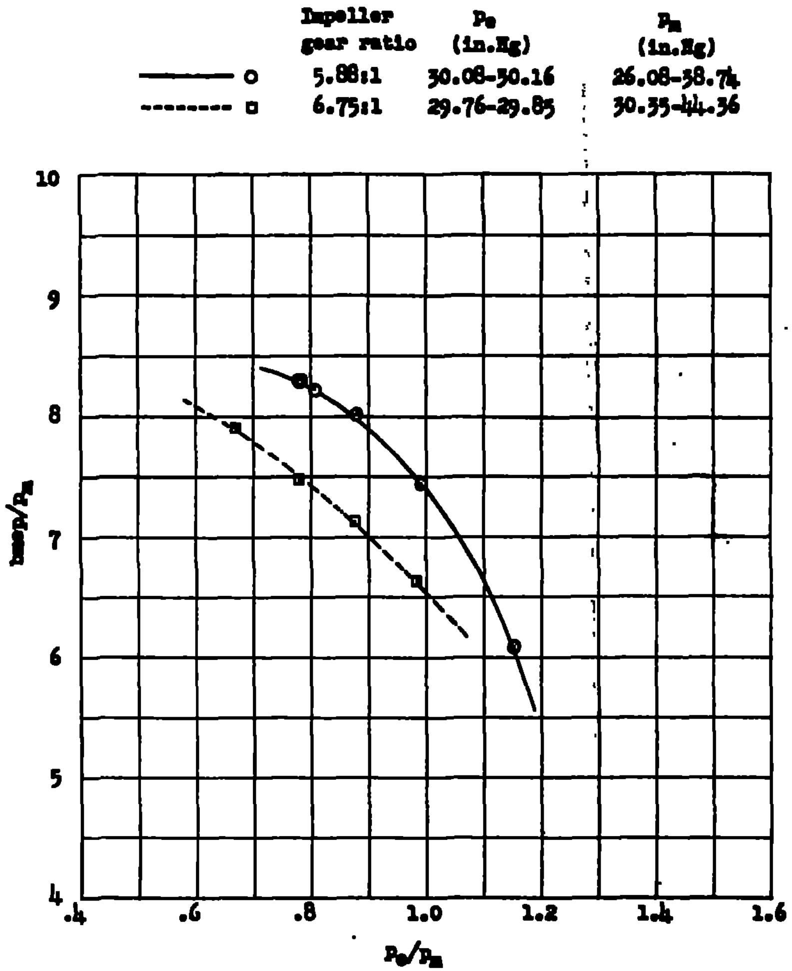


(c) Engine speed, 2500 rpm.

Figure 7. - Continued.



(d) Engine speed, 3100 rpm.
Figure 7. - Continued.



(e) Engine speed, 3500 rpm.
Figure 7. - Concluded.

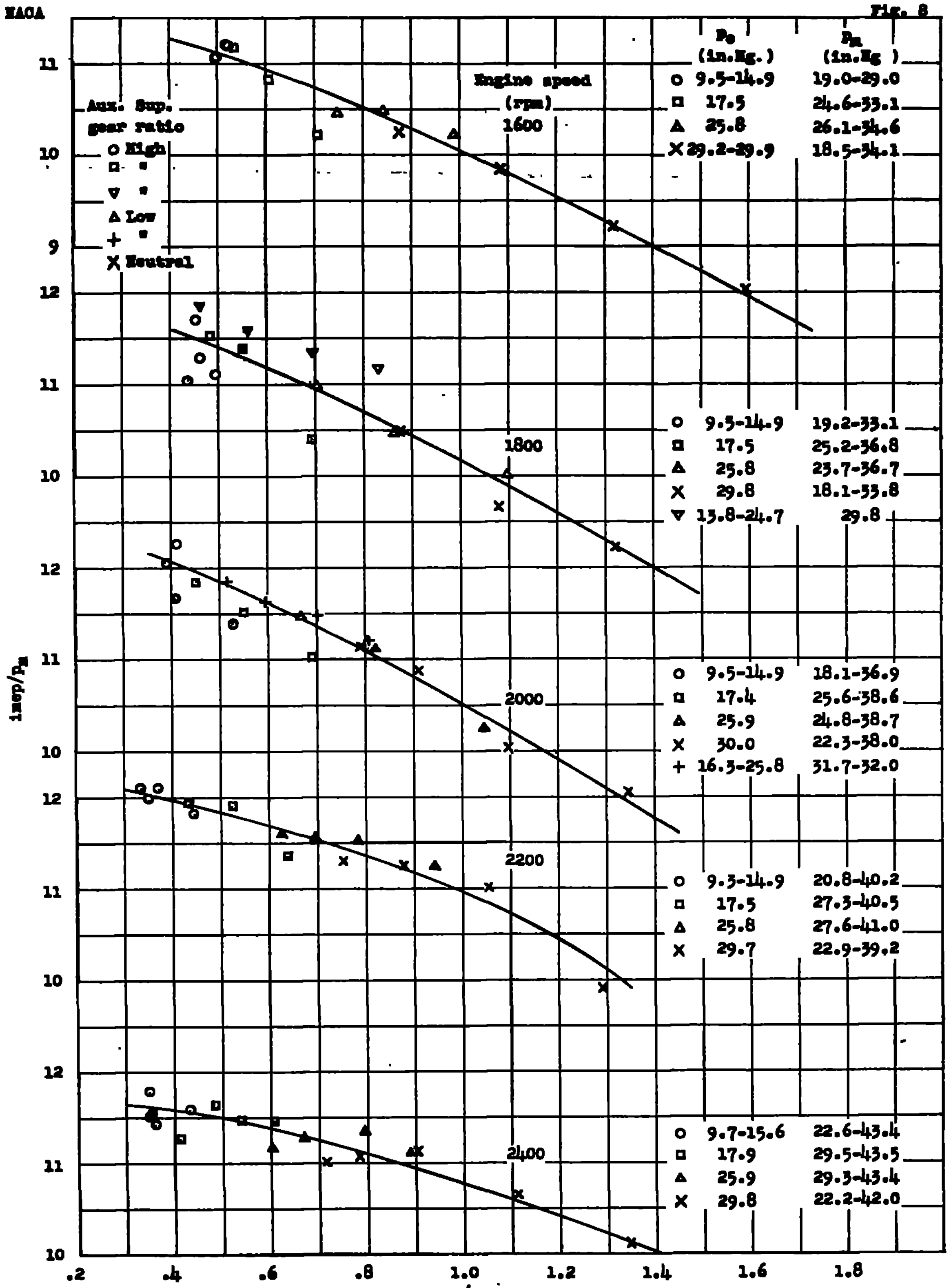


Figure 8. - Effect of exhaust back pressure on engine power for Pratt & Whitney XR-2800-4 engine. Ratio of impeller tip speed to engine speed for main stage, 21.44 feet; auxiliary stage in low gear, 20.29 feet; high gear, 27.14 feet; corrected carburetor-air temperature, 80° F.

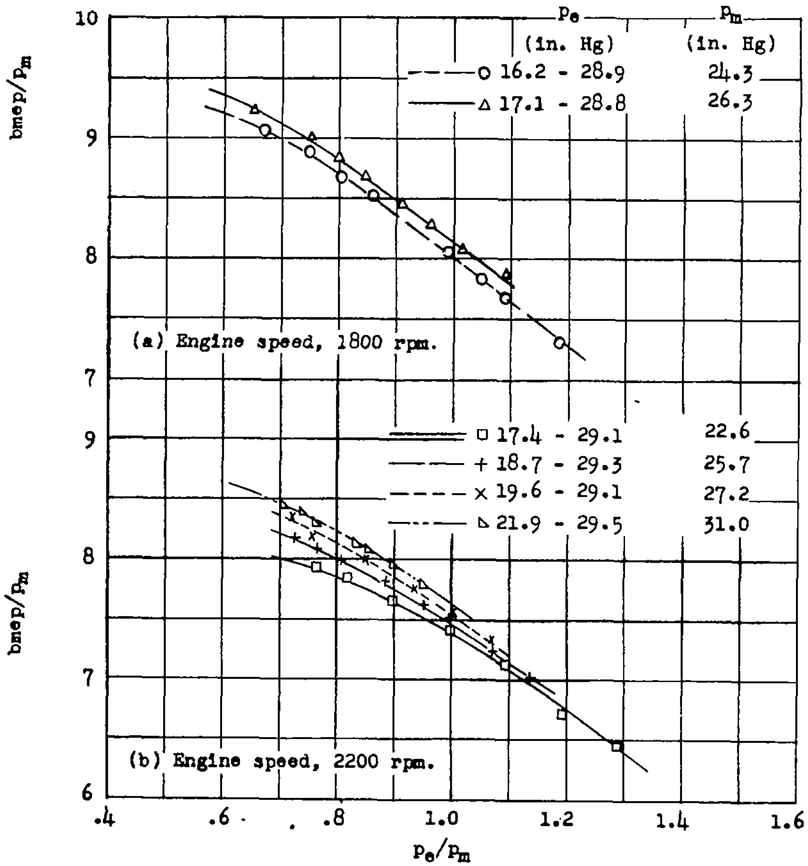


Figure 9. - Effect of exhaust back pressure on engine power for Bristol Mercury VI engine. Ratio of impeller tip speed to engine speed, 24.19 feet; intake air temperature, 80° F.

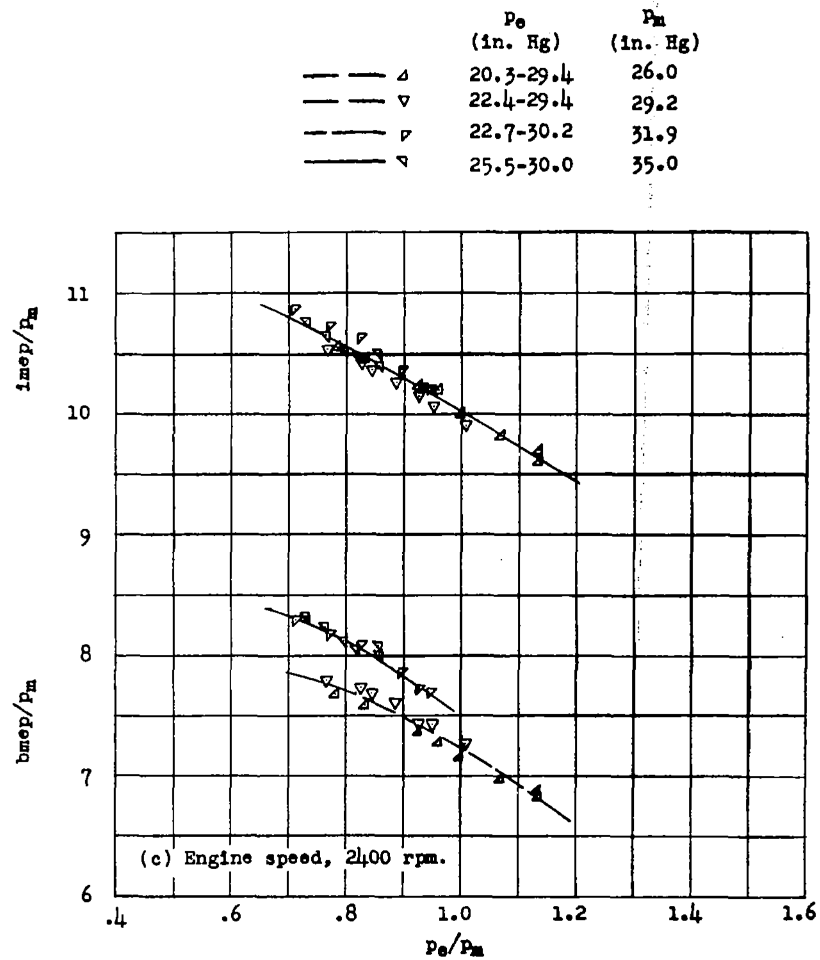


Figure 9. - Concluded.

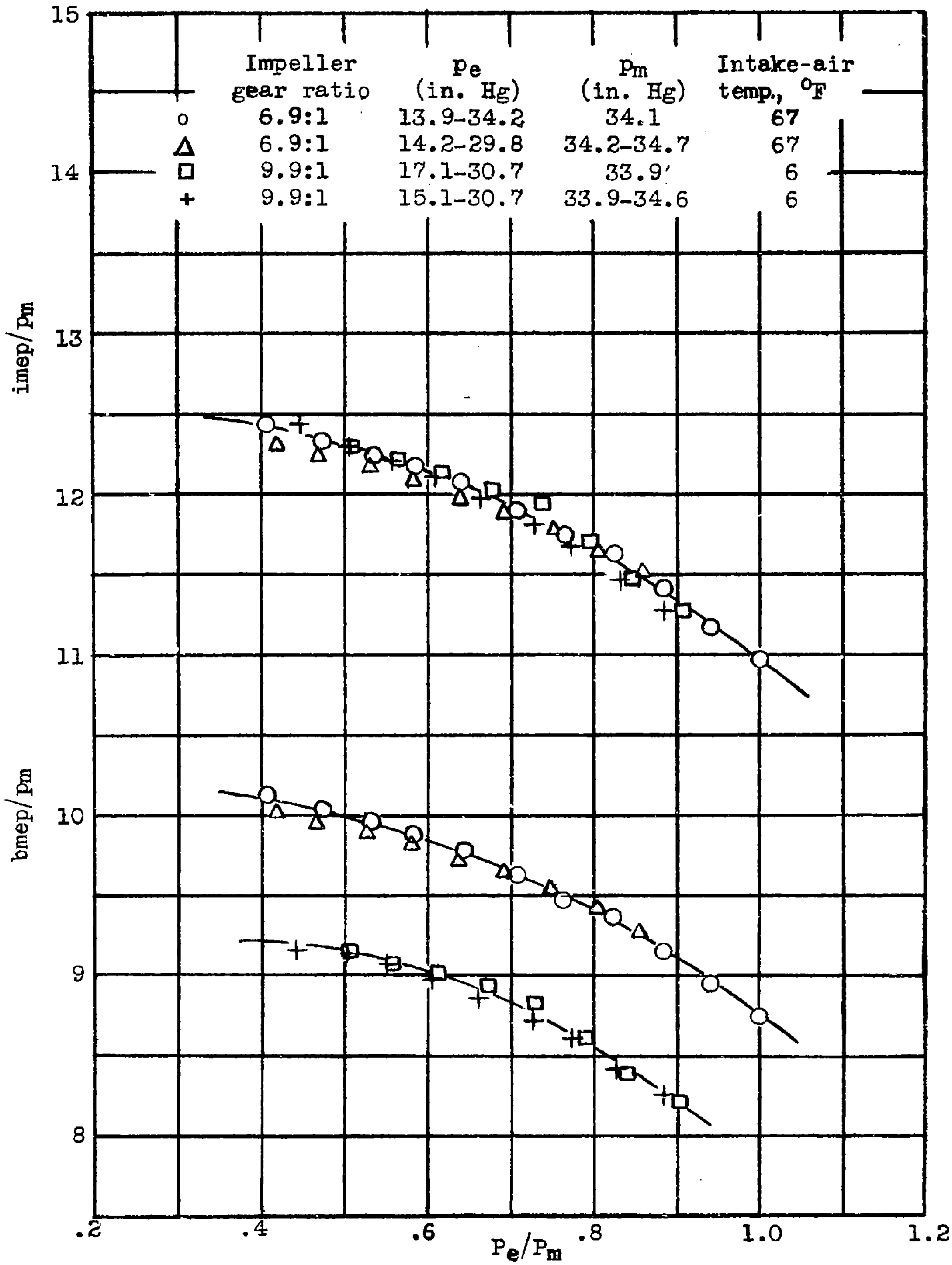


Figure 10. - Effect of exhaust back pressure on engine power for Bristol Pegasus XVIII engine. Ratio of impeller tip speed to engine speed for high gear, 29.42 feet; for low gear, 20.50 feet; engine speed, 2250 rpm.

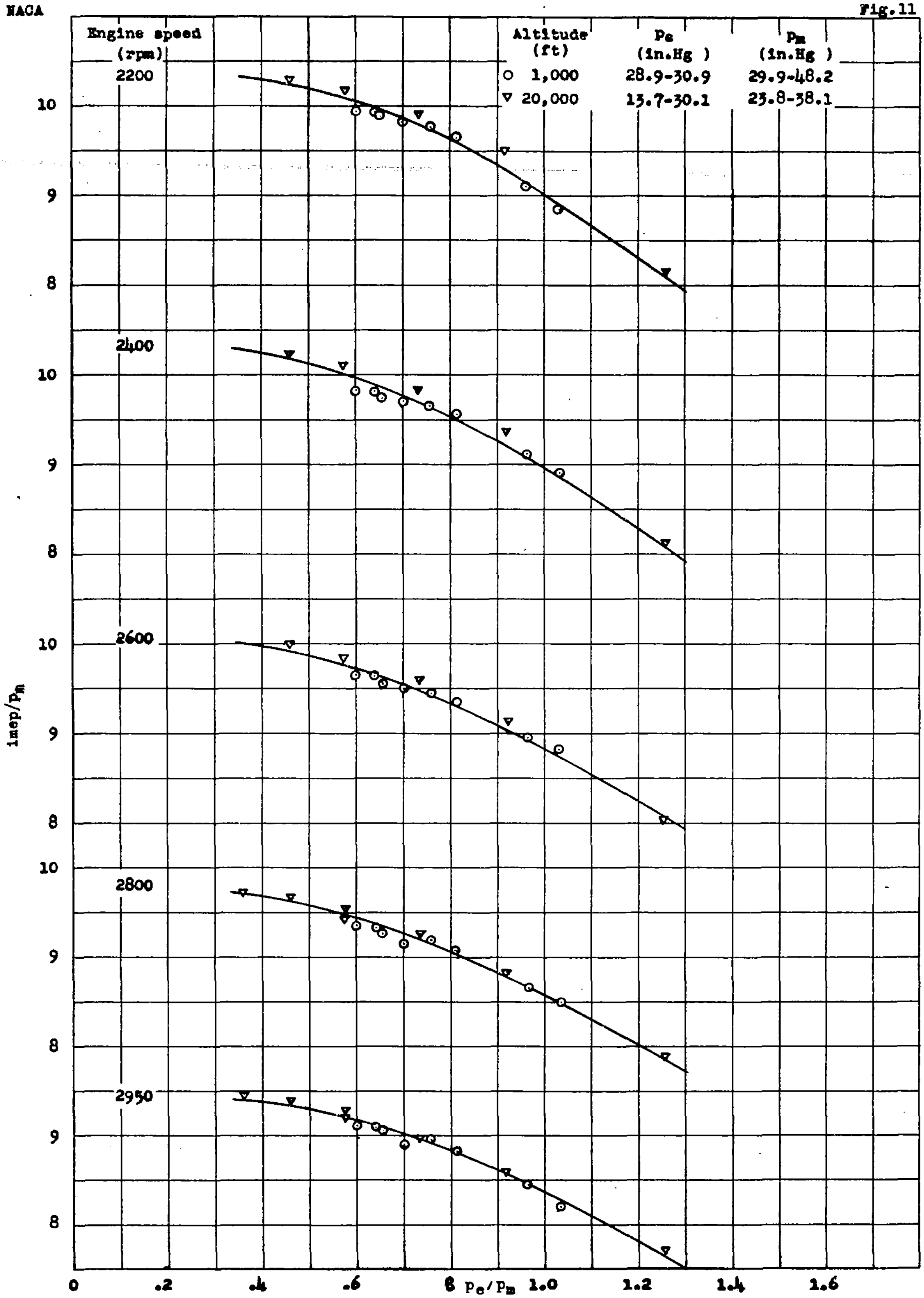


Figure 11. - Effect of exhaust back pressure on engine power for Rolls Royce Merlin 46 engine. Ratio of impeller tip speed to engine speed, 25.82 feet; corrected intake-air temperature, 80° F.

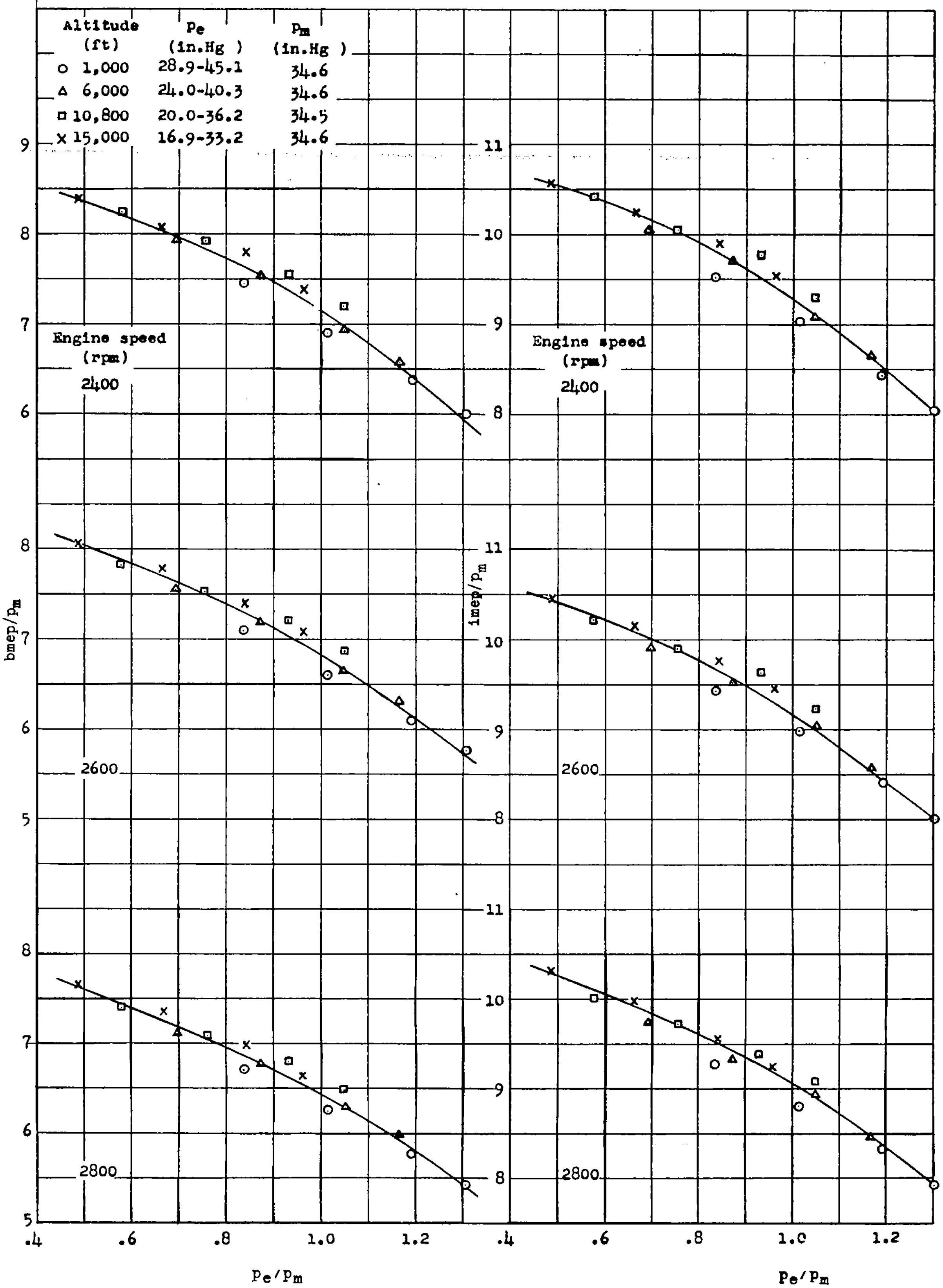


Figure 12. - Effect of exhaust back pressure on engine power for Rolls Royce Merlin II engine. Ratio of impeller tip speed to engine speed, 23.05 feet; corrected intake-air temperature, 80° F.

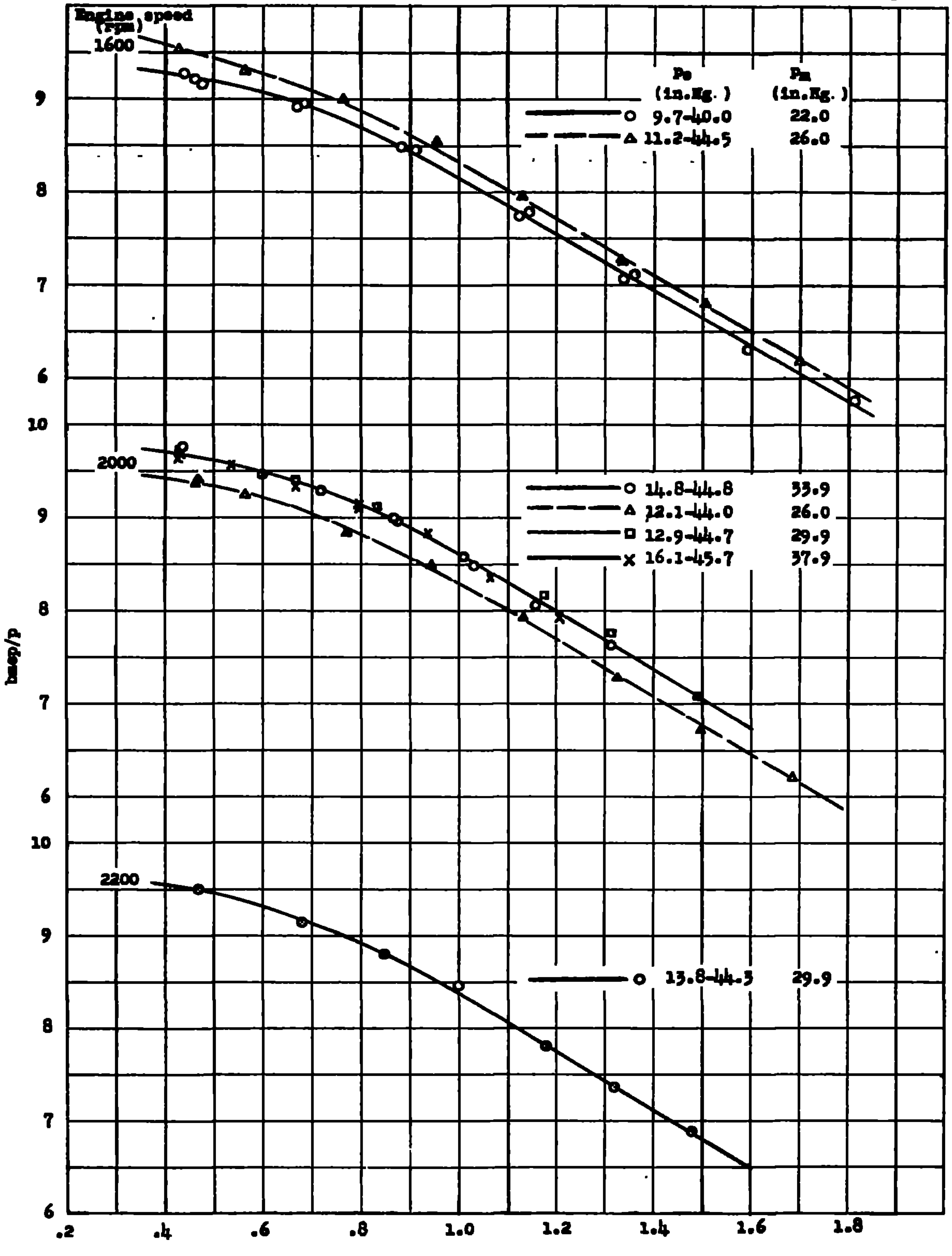


Figure 13. - Effect of exhaust back pressure on engine power for Bristol Perseus VIII engine. Corrected intake-air temperature, 80° F.

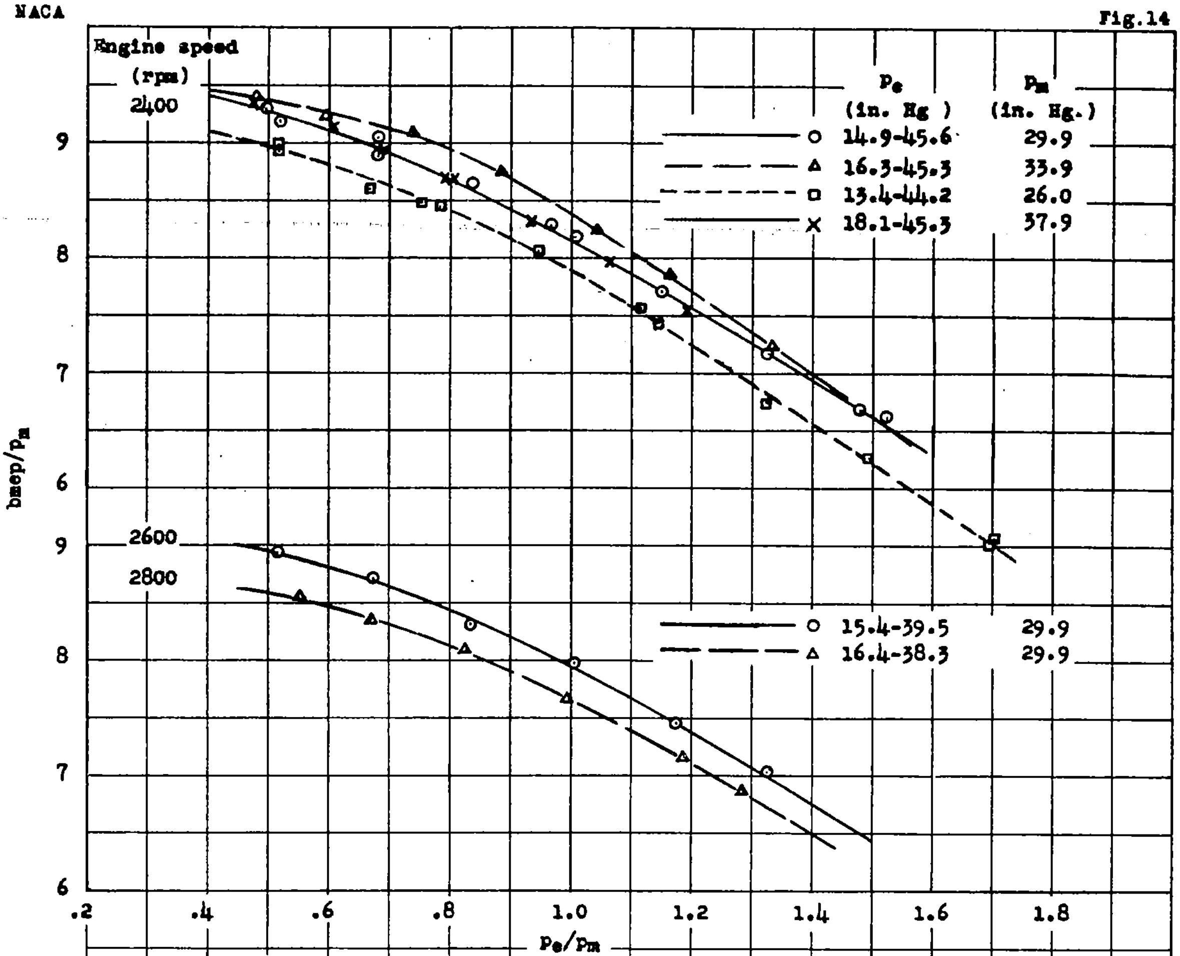


Figure 13. - Concluded.

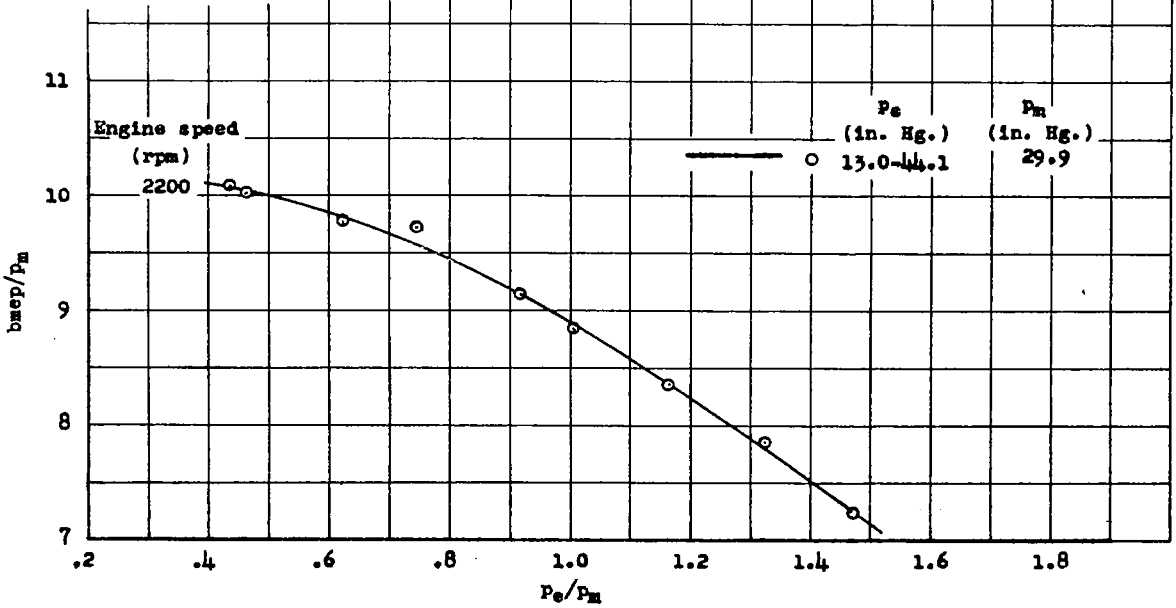


Figure 14. - Effect of exhaust back pressure on engine power for Bristol Perseus XII engine. Corrected intake-air temperature, 80° F.

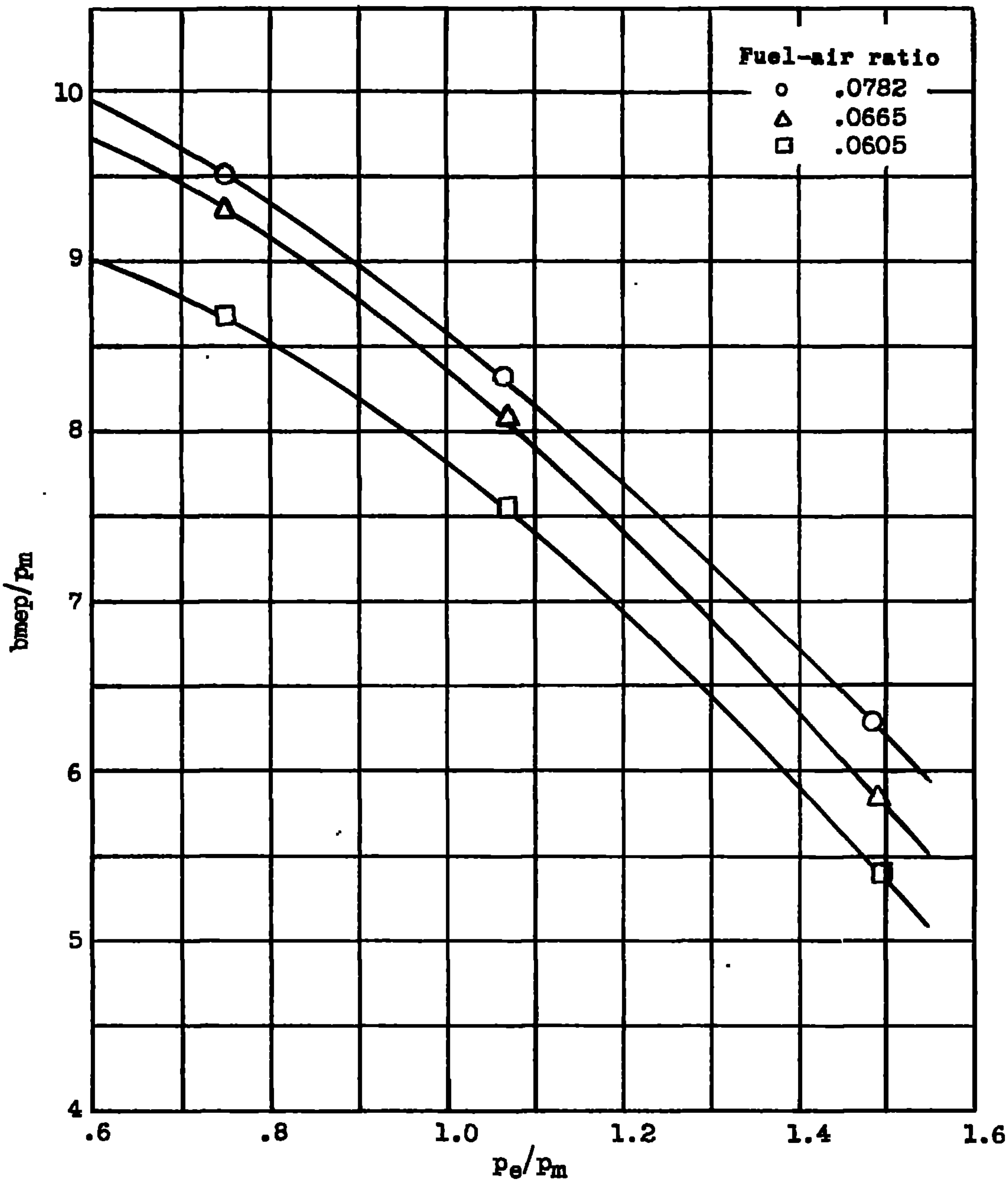


Figure 15.- Effect of exhaust back pressure on engine power for Junkers Jumo JU-211D engine. Ratio of impeller tip speed to engine speed for high blower, 28.14 feet; for low blower, 19.42 feet; engine speed, 1950 rpm; compression ratio, 6.7; corrected to carburetor-air temperature, 80°F; p_e, 17.2-35.0 in. Hg; p_m, 22.9-23.6 in. Hg; supercharger in low gear.