T H E U N I V E R S I T Y O F M I C H I G A N

COLLEGE OF ENGINEERING
Department of Mechanical Engineering

Progress Report

# AIR-BOOSTED CYCLE ANALYSES

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# INTRODUCTION

To increase the mean effective pressure of an engine to secure responsiveness, there is a method of approach in addition to that of increasing the degree of supercharge, viz., that of supplying the additional air and fuel necessary during the expansion stroke, maintaining the peak pressure approximatly constant. The engine indicator diagram will then vary as shown in Fig. 1, where 12345 represents the indicator diagram at what is called normal rated load and maximum rpm. As the vehicle resistance increases and the engine slows down, an increase in mean pressure is required to maintain maximum hp; this is secured by supplying additional air and fuel to the cylinder at such a rate that the maximum pressure is maintained constant and the diagram will change to 12367, 12389, etc., as the need arises.

The advantages of such a procedure are:

1. The engine behaves and is rated as a normal engine of present-day design, without additional loads, etc., so long as normal operating conditions are applied.

2. The engine is capable of being operated at increased mean pressures for temporary overloads (period of overload can be long if suitably designed).

3. Increased load achieved without increase in maximum cylinder pressure and thus constant engine stress.

4. Increase of load achieved without increase (probably a decrease) in maximum combustion temperature.

5. Greater energy available for recovery by compounding.

6. Equipment to achieve this purpose only operates when required.

7. The increase in weight, space and costs involved can be kept small by designing the engine for its normal output requirements, say 300 hp, and employ the proposed principle to meet the emergency needs of, say, 500-600 hp only when needed; by this means little change in weight, space, etc., is seen. A typical design would need to be worked out to establish the requirements accurately.

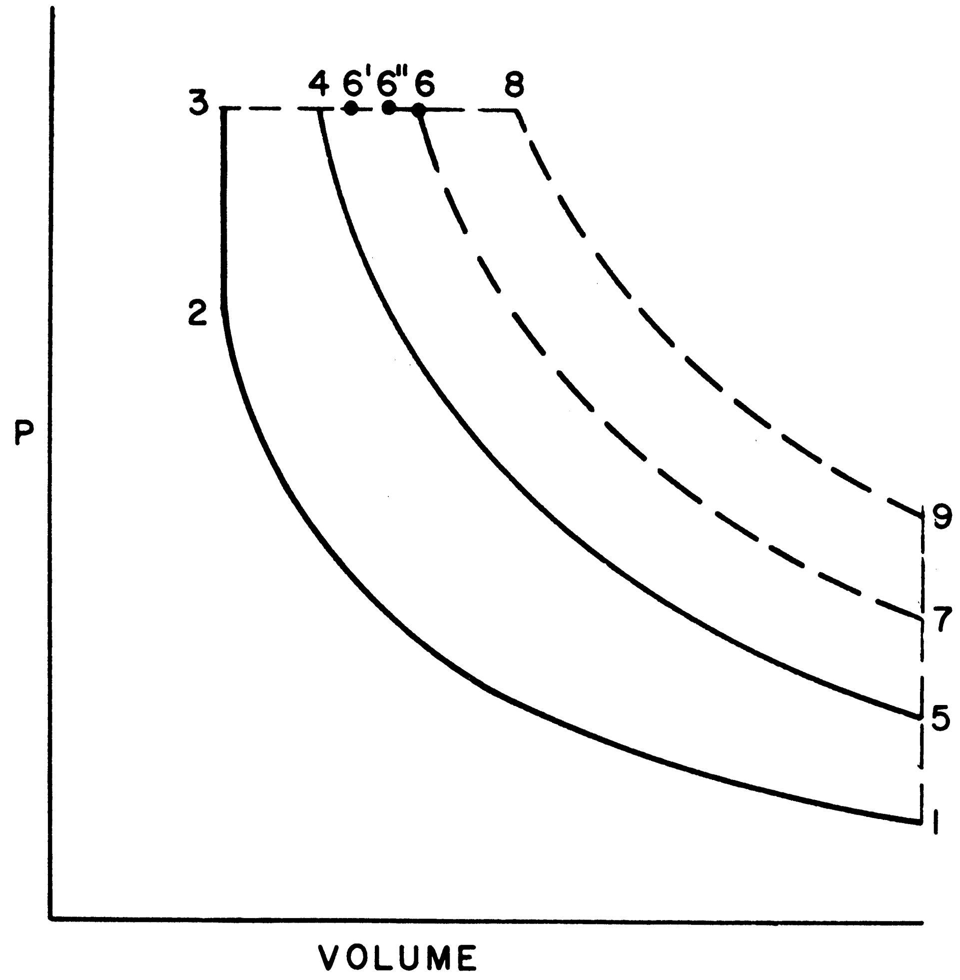



Fig. 1. Engine indicator diagram.

## ENGINE ARRANGEMENT

The proposed cycle of operation would be achieved by providing an air compressor capable of delivering air at, say, 1600 psi to a small receiver, to damp out oscillations and provide a small reserve. This receiver is connected to an engine supply manifold, and flow to the cylinders is provided by a mechanically controlled valve in the cylinder head capable of being varied as regards period or lift or both.

When need arises for the use of responsiveness—as the engine stalls on high load—the compressor is coupled to the engine and the air supply is fed to the receiver and then to the cylinder as in Fig. 2. The appropriate increase in fuel is also added to the air supply. If high load is required at maximum speed for any reason the air supply is also available for this purpose. Thus a typical design could be as follows:

| | |
|---|---|
| Normal power at max rpm | 400 hp |
| Maximum emergency power at max rpm | 600 hp |
| Normal power at min rpm | 200 hp |
| Maximum power at min rpm | 400 hp |

Engine responsiveness from $rpm_{max}$ to $rpm_{max}/2$ (constant 400 hp).

Other combinations of hp and speed could also be achieved. In the above example, a 400-hp engine can produce 600 when required, and also be responsive at 400 hp over a 2:1 speed range. Alternatively, the full 600-hp normal engine could be designed and made responsive at 600 hp down to, for example, half maximum rpm; this unit would of course be larger than the former. A typical arrangement would be as shown in Fig. 2; the turbo-charger would be geared to the engine if compounding was to be employed.

## PERFORMANCE

The performance estimate for this type of power plant was obtained in the manner given below. The method is admittedly an approximate one, but is believed to be within a reasonably small percentage of what can be expected, and sufficiently close to judge if the system has merit. First the basic engine cycle is analyzed, including heat losses by the method described in Progress Report No. 1. The details of the cycle employed are:

| | |
|---|---|
| Compression ratio | 15:1 |
| Supercharge ratio | 2:1 |
| F/A ratio | 0.05 |

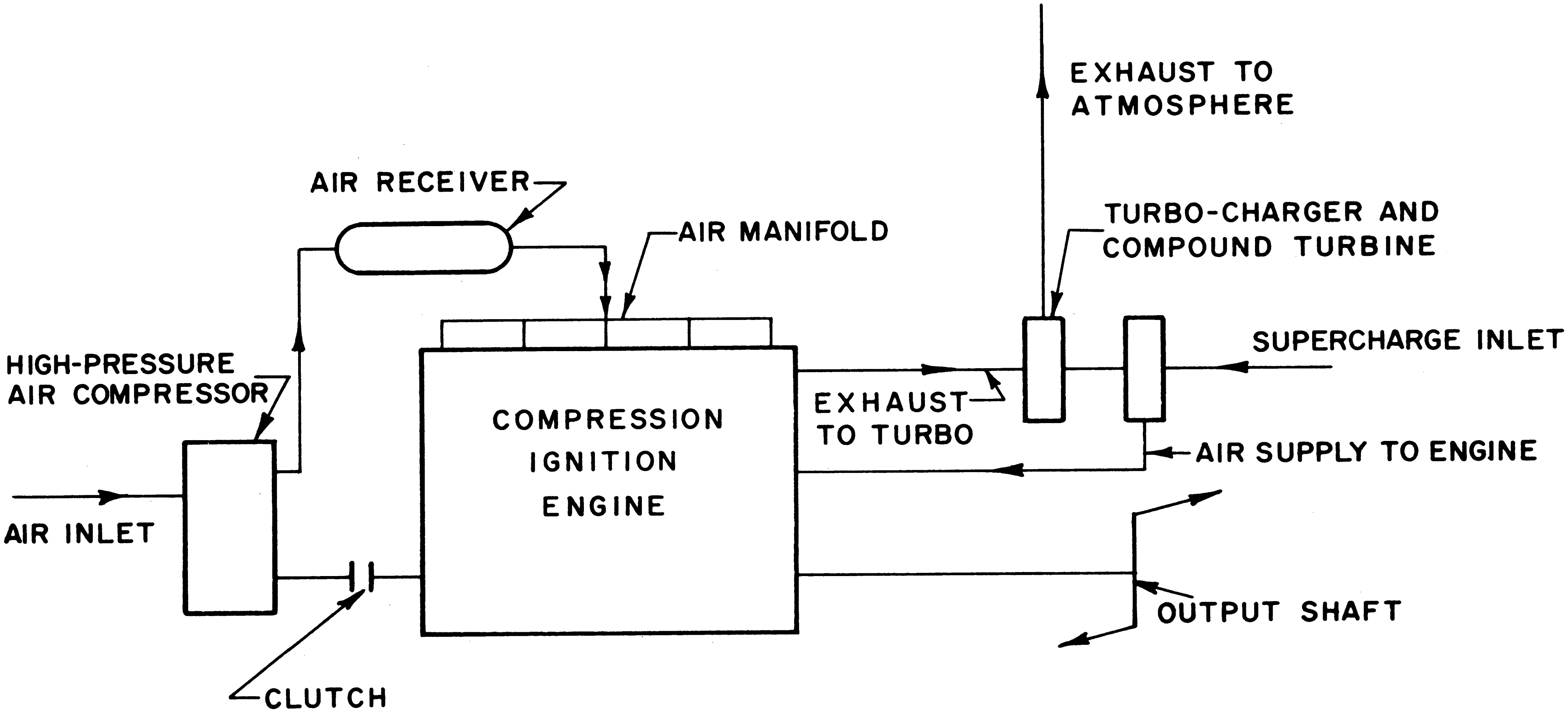



Fig. 2. Air-supply flow diagram.

| | |
|---|---|
| Calorific value of fuel | 18500 Btu/lb |
| Turbo-compressor efficiency | 0.70 |
| Turbine efficiency | 0.86 |
| Ambient air | $P_a$ = 14.7 psi and $T_a$ = 125°F |
| Total heat loss | 21% |
| Heat-loss compression | 0.4% |
| Heat-loss combustion | 1.7% |
| Incomplete combustion | 2.0% |
| Heat-loss expansion | 5.0% |
| Heat-loss exhaust | 5.5% |
| Heat-loss to oil | 1.8% |
| Miscellaneous | 4.6% |

## MANIFOLD CONDITIONS

With the above efficiency values, the engine manifold conditions become

$$P_m = 29.4 \text{ psia} \qquad T_m = 753° \text{ abs}$$

If an aftercooler of 60% effectiveness is employed, the manifold temperature is reduced to 612° abs.

## INITIAL CONDITIONS IN CYLINDER

For the above manifold conditions, the state in the cylinder at the beginning of compression would be approximately as follows for 1 lb of air trapped.

It is assumed that with turbo changing at the high pressures proposed and with valve overlap, no exhaust gas is present in the cylinder and the volume of the 1 lb of air is the cylinder volume; this is considered a reasonable assumption for a first approach to the problem.

$P_1$ = 29.0 psia
$T_1$ = 765° abs, no aftercooling
= 625° abs with 60% aftercooling
$V_1$ = 9.75 cu ft/lb or 7.96 cu ft/lb with aftercooling.

## END OF COMPRESSION

Using the charts for cycle determination with the specified heat losses, the following states represent the end of compression.

| | No Aftercooling | 60% Effective Aftercooling |
|---|---|---|
| Index of compression n | 1.343 | 1.353 |
| Pressure $P_2$ psia | 1080 | 1110 |
| Temperature $T_2^{\circ}$ abs | 1912° | 1622° |
| Work of compression Btu | 230.2 | 192.2 |
| $V_2$ cu ft/lb | 0.65 | 0.531 |

## CONSTANT-VOLUME COMBUSTION

Assuming an arbitrary limit of 1500 psi for the peak cylinder pressure, a certain percentage of the fuel can be burnt approaching constant-volume combustion from point 2 to point 3 of the cycle. This will be the same for all the cycles; the only difference is that of temperature when using an aftercooler. Using the heat-loss factor for this process, the following gives the conditions at point 3.

| | No Aftercooling | 60% Aftercooling |
|---|---|---|
| $P_3$ (assumed) psia | 1500 | 1500 |
| Temperature increase | | |
| $\Delta T$ | 560 | 490 |
| Btu's required | 118 | 100 |
| Btu's used including loss | 135 | 117 |
| $T_3^{\circ}$ abs | 2472 | 2112 |
| $V_3 = V_2$ | 0.65 | 0.531 |

## CONSTANT-PRESSURE COMBUSTION

The balance of the heat available in the fuel is now added at constant pressure to the 1 lb of air asperated into the cylinder. Then:

Total Btu of 0.05 lb of fuel = 0.05 x 18500
= 925 Btu

Heat loss and incomplete combustion = 20 Btu

Heat available = total Btu supplied - heat used at constant volume - heat loss

Heat available = 925 - (135+20)
= 770 Btu (no aftercooling)

Heat available = 925 - (117+20)
= 788 (60% aftercooling).

The charts give the following for the ratio of $V_4/V_3$ for the above heat supply:

$V_4/V_3$ = 2.06 no aftercooling
= 2.25 with aftercooling.

It follows that the temperatures can be calculated from

$$T_4 = T_3 \frac{V_4}{V_3}$$

assuming that R, the gas constant, remains at a constant value. Thus

$T_4$ = 1500° (no aftercooling) or 4760° (with aftercooling)

and the contents of the cylinder at point 4 of the cycle becomes:

| | No Aftercooling | 60% Aftercooling |
|---|---|---|
| Air, lb | 1.0 | 1.0 |
| Fuel, lb | 0.05 | 0.05 |
| Gas constant for mixture | 54.0 | 54.0 |
| $P_4 = P_3$ psia | 1500 | 1500 |
| $V_4$ cu ft | 1.34 | 1.196 |
| $T_4$ | 5100° | 4760° |
| Work of expansion $144P_3(V_4-V_3)/778$ Btu | 191.9 | 184.3 |

The above represents the changes in state for the normal cycle combustion process. At point 4 when responsiveness is desired, additional fuel and air are supplied to the cylinder and, if the F/A is maintained at 0.05, then, for each lb of air supplied, 925 additional Btu will be added or, say, 910 Btu when heat loss is assumed.

If the additional supply is assumed to be at a temperature of 700°R when admitted and is at 1500 psia when in the cylinder, and if this supply is considered to enter as a separate entity, is supplied with the necessary fuel, then burnt at constant pressure, still in its separate container, followed by mixing with the charge that was normally asperated at constant pressure, then the approximate state at the end of this phase of the cycle can be obtained as given below.

In actual conditions it is possible that the assumed 700° for the compressed air is low, since the compressor will deliver the air at a high temperature. Such increase in inlet temperature will result in improved performance to that calculated and thus the figures below can be considered somewhat pessimistic in an over-all evaluation.

The first operation, admitting air at 1500 psi and 700°F, can be determined as follows:

$$\Delta v = V_{6'} - V_4 \text{ (Fig. 1)}$$

$$V_{6'} = \text{volume of normal change plus vol of } \Delta w \text{ lb of air}$$

$$\Delta v = \frac{\Delta w\ RT}{P}$$

$$R = 53.34 \text{ for air}$$

$$T = 700°$$

$$P = 1500 \text{ psi}$$

$$\Delta v = 0.1728\ \Delta w \text{ cu ft.}$$

Assume that the volume of the liquid fuel is negligible and determine the conditions when $\Delta w$ is gradually increased from 0 to 1.0 lb of air. Then:

| | | | | | | |
|---|---|---|---|---|---|---|
| $\Delta w$ lb | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| $\Delta v$ cu ft | 0.0 | 0.0345 | 0.0691 | 0.1037 | 0.1381 | 0.1728 |
| $V_{6'}$ cu ft No aftercooling | 1.34 | 1.375 | 1.409 | 1.444 | 1.478 | 1.513 |
| $V_{6'}$ cu ft 60% aftercooling | 1.196 | 1.23 | 1.265 | 1.300 | 1.334 | 1.369 |

On adding and burning the fuel at constant pressure, we have

$$Q_A = \frac{C_p}{R} (V_6'' - V_6')P_4 \tag{1}$$

where

$Q_A$ = Btu added

$C_p$ = mean specific heat for the temperature range

$R$ = $J(C_p - V_c)$ for temperature range

$V_6''$ = Volume of products after combustion.

The process must be a trial-and-error one since the range of temperature is not known until the final temperature is available. Thus assume this temperature, proceed with the calculation, and see if the average calculated temperature is equal to the assumed average value used to determine $C_p$. By this means a correct assumed average temperature is obtained from which the values of $C_p$ and $C_v$ control the temperature:

From graph for $T_{avg}$ $\quad C_p = C_{p_{avg}}$

$C_v = C_{v_{avg}}$

$R = J(C_{p_{avg}} - C_{v_{avg}})$

and substitution in Eq. (1) will give

$Q_A = K\Delta V$

$K = \text{constant} = \frac{C_p}{R} \times 1500 \times 144$

Assume $T_{avg}$ = 2200° then $C_p = 0.305$ and $C_v = 0.237$

$R = 778 (.305-.237) = 52.9$

$Q_A = 1245\ \Delta V$

or

$$\Delta V = \frac{910}{1245} = 0.731 \text{ cu ft/lb}$$

| | | | | | | |
|---|---|---|---|---|---|---|
| $\Delta w$ lb = | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| AV = | 0.0 | 0.146 | 0.292 | 0.438 | 0.585 | 0.731 |
| $V_6''$ = No aftercooling | 1.34 | 1.525 | 1.701 | 1.882 | 2.063 | 2.244 |
| $V_6''$ = 60% aftercooling | 1.196 | 1.376 | 1.557 | 1.738 | 1.919 | 2.099 |

The temperature at the end of combustion of the additional air and fuel is given by:

$$Q_A = \Delta w(1+f)C_p(T_{6''}-700)$$

but

$$Q_A = \Delta w \times 910$$

Thus

$$T_{6''} = \frac{910}{C_p x1.05} + 700$$

$$= \frac{910}{0.305x1.05} + 700$$

$$= 2840 + 700$$

$$= 3540° \text{ abs.}$$

Thus the average temperature of the process is

$$T_{avg} = \frac{1}{2}(3540+700)$$

$$= 2120°$$

against the assumed average of 2200° abs. This is close enough since $C_p$ changes somewhat slowly at these temperatures.

The cylinder contents now consist of

(a) 1.05 lb of combustion products from the asperated air occupying $V_4$ cu ft at $P_4$ = 1500 psi and temp $T_4^\circ$ abs.

(b) $\Delta w(1+f)$ lb of combustion products supplied at TDC occupying $(V_{6''}-V_4)$ cu ft at $P_4$ = 1500 psi and temp $T_{6''}$ = 3540° abs.

These two quantities are now mixed at constant pressure with the result that:

Heat lost by hot gas—heat gained by colder gas

$$(1+f)C_p(T_4-T_m) = \Delta w(1+f)C_p(T_m-T_{6''})$$

$$T_4 - T_m = \Delta w T_m - \Delta w T_{6''}$$

$$T_m = \frac{T_4+\Delta w T_{6''}}{1+\Delta w}$$

With no aftercooling, $T_4$ = 5100°

With aftercooling, $\epsilon$ = 0.60 $T_4$ = 4760.

Thus $T_m = \frac{5100+\Delta w \times 3540}{1+\Delta w}$ or $\frac{4760+\Delta w \times 3540}{1+\Delta w}$ which yields the following when $T_g$ = 5100 for no aftercooling:

| $\Delta w$ | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|
| $T_m = T_6$ | 5100 | 4840 | 4650 | 4510 | 4405 | 4320 |

when $T_g$ = 4760 and $\epsilon$ = .60 aftercooling:

| $\Delta w$ | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|
| $T_m = T_6$ | 4760 | 4560 | 4415 | 4300 | 4220 | 4160 |

The actual state at point 6 is now available for all values of $\Delta w$ up to 1.0 lb of air.

No Aftercooling

| $\Delta w$ | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|
| $P_6$ | 1500 | 1500 | 1500 | 1500 | 1500 | 1500 |
| $V_6$ | 1.34 | 1.525 | 1.701 | 1.882 | 2.063 | 2.244 |
| $T_6$ | 5100 | 4840 | 4650 | 4510 | 4405 | 4320 |

With 60% Aftercooling

| $\Delta w$ | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|
| $P_6$ | 1500 | 1500 | 1500 | 1500 | 1500 | 1500 |
| $V_6$ | 1.196 | 1.576 | 1.557 | 1.738 | 1.919 | 2.099 |
| $T_6$ | 4760 | 4560 | 4415 | 4300 | 4220 | 4160 |

An additional advantage of the cycle being examined is a reduction of maximum gas temperature as additional air is added.

## EXPANSION STROKE

The thermodynamic cycle can now be completed by expanding the gas, corrected for heat losses, back to the original volume of the 1 lb of air. This operation was carried out on the charts provided for the purpose with the following results.

| $\Delta w$ = | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|
| Expansion ratio $\epsilon = 0$ | 7.26 | 6.4 | 5.72 | 5.18 | 4.72 | 4.34 |
| $P_7$ | 114.0 | 134 | 155 | 181 | 193 | 216 |
| $T_7$ | 2800 | 2660 | 2650 | 2730 | 2740 | 2750 |
| Expansion ratio $\epsilon = 0.60$ | 6.65 | 5.78 | 5.09 | 4.56 | 4.13 | 3.77 |
| $P_7$ | 127 | 167 | 180 | 206 | 240 | 263 |
| $T_7$ | 2670 | 2680 | 2675 | 2755 | 2740 | 2770 |

The work of the expansion is given by

$$W_e = \frac{P_6V_6 - P_7V_7}{(n-1)J}$$

$$= \frac{144\ P_6}{0.3 \times 778}\left(V_6 - \frac{V_1}{P_6/P_7}\right)$$

With no aftercooling $V_1 = 9.75$

With 60% aftercooling $V_1 = 7.96$

No Aftercooling

| $\Delta w$ = | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|
| Work of expansion Btu | 556 | 603 | 650 | 663 | 760 | 790 |

60% Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| Work of expansion Btu | 482 | 524 | 558 | 604 | 639 | 659 |

EXHAUST MANIFOLD TEMPERATURE

The next step is to obtain the expected exhaust gas temperature. To do this we have the following equation

$$\left\{\begin{matrix}\text{Heat in gases at}\\ \text{end of expansion}\end{matrix}\right\} + \left\{\begin{matrix}\text{Work done by piston}\\ \text{in exhausting gas}\end{matrix}\right\} = \left\{\begin{matrix}\text{Energy of}\\ \text{gas in manifold}\end{matrix}\right\} + \left\{\text{Heat losses in process}\right\}$$

$$(1+\Delta w)(1+f)C_vT_7 + \frac{144\ P_e(V_7-V_2)}{778} = (1+\Delta w)(1+f)C_pT_e + Q_L$$

where

$P_e$ = pressure in exhaust manifold

$T_e$ = temperature in manifold

$Q_L$ = heat losses of exhaust

= 50 Btu

$V_7$ and $V_2$ = volume of gas, cu ft

$$T_e = \frac{C_v}{C_p}T_7 + \frac{0.185\ P_e(V_7-V_2)-50}{(1+\Delta w)(1+f)C_p}\ .$$

Since the variation of $T_e$ is of minor magnitude due to $T_7$ having but a small variation, the values of $C_v$ and $C_p$ can be considered constant and equal at all values of $\Delta w$. The average values used are $C_v = 0.23$ and $C_p = 0.30$ for the gases and temperatures involved. Calculation gives the following values for the state in the exhaust pipe.

No Aftercooling $V_7 = 9.75$, $V_2 = 0.65$ cu ft

| | | | | | | |
|---|---|---|---|---|---|---|
| $\Delta w$ = | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| $T_7$ | 2800 | 2660 | 2650 | 2730 | 2740 | 2750 |
| Gas lb | 1.05 | 1.26 | 1.47 | 1.68 | 1.89 | 2.10 |
| $P_e$ (assumed) | 25.5 | 25.5 | 25.5 | 25.5 | 25.5 | 25.5 |
| $T_8$ | 2127 | 2021 | 2013 | 2075 | 2087 | 2098 |

60% Aftercooling $V_7 = 7.96$, $V_2 = 0.531$ cu ft

| | | | | | | |
|---|---|---|---|---|---|---|
| $\Delta w$ = | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| $T_7$ | 2676 | 2680 | 2675 | 2755 | 2740 | 2770 |
| Gas lb | 1.05 | 1.26 | 1.47 | 1.68 | 1.89 | 2.10 |
| $P_e$ (assumed) | 25.5 | 25.5 | 25.5 | 25.5 | 25.5 | 25.5 |
| $T_8$ | 2002 | 2015 | 2016 | 2080 | 2073 | 2096 |

It is seen that the exhaust temperature does not vary over a wide range. This is reasonable since the same F/A ratio is being employed in all cases and the mass of gas varied. The values obtained vary from a low of 2000° abs to a high of 2127° abs or 1540° to 1667°F, a not unreasonable value for an F/A ratio of 0.05, possibly 10% high at the most, but heat losses for a length of exhaust pipe to the point of temperature measurement can lower these values by 200°F or so quite easily.

## WORK OF CYCLE

It is now possible to evaluate the net work of the cycle and the fuel quantity needed for this output.

The work done by the gas during the constant pressure expansion process is as follows:

$$W = \frac{144\ P_3(V_6 - V_3)}{778}\ \text{Btu}$$

| | | | | | | |
|---|---|---|---|---|---|---|
| $\Delta w$ | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| With no Aftercooling | | | | | | |
| W | 191.8 | 243.2 | 292.5 | 342.5 | 393.0 | 443.5 |
| 60% Aftercooling | | | | | | |
| W | 184.8 | 234.8 | 284.8 | 335.5 | 386.0 | 435.5 |

There is a small pressure difference between inlet and exhaust pressures. Thus there is a net positive work performed here, which is as follows:

$$\text{Suction loop work} = \frac{(P_m - P_e)(V_1 - V_2)}{J}$$

Since $P_m$ and $P_e$ were maintained constant at 29.4 and 25.5 psia, respectively, this becomes

$$\text{Suction loop work} = 0.722\,(V_1 - V_2)$$

For the case with no aftercooling, $V_1-V_2 = 9.75-0.65 = 9.10$ cu ft, while with aftercooling, it becomes $V_1-V_2 = 7.96-.531 = 7.429$ cu ft.

Suction loop (no aftercooling) = 6.58 Btu
Suction loop (with aftercooling) = 5.36 Btu.

WORK SUMMARY

| Item | No Aftercooling | | | | | | Aftercooling, $\epsilon$ = 0.60 | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| Suction loop (positive Btu) | 6.58 | 6.58 | 6.58 | 6.58 | 6.58 | 6.58 | 5.36 | 5.36 | 5.36 | 5.36 | 5.36 | 5.36 |
| Compression work (negative Btu) | -230.2 | | | | | | -192.2 | | | | | |
| Const. press. exp. Btu | 191.9 | 243.2 | 292.5 | 342.5 | 393.0 | 443.5 | 184.3 | 234.8 | 284.8 | 335.5 | 386.0 | 435.5 |
| Expansion Btu | 556 | 603 | 650 | 663 | 760 | 790 | 482 | 524 | 558 | 604 | 639 | 659 |
| Net work of cycle Btu | 524.3 | 622.6 | 718.9 | 781.9 | 929.4 | 1009.9 | 479.5 | 572 | 656.0 | 752.7 | 838.2 | 907.7 |
| I.H.P./lb of basic air/sec | 742 | 881 | 1018 | 1107 | 1313 | 1442 | 679 | 809 | 928 | 1066 | 1187 | 1283 |
| Total air flow, lb/sec | 1.0 | 1.20 | 1.40 | 1.60 | 1.80 | 2.00 | 1.0 | 1.20 | 1.40 | 1.60 | 1.80 | 2.00 |
| Fuel flow, lb/sec | .05 | .06 | .07 | .08 | .09 | 0.10 | .05 | .06 | .07 | .08 | .09 | 0.10 |
| Total gas flow/lb of basic air/sec | 1.05 | 1.26 | 1.47 | 1.68 | 1.89 | 2.10 | 1.05 | 1.26 | 1.47 | 1.68 | 1.89 | 2.10 |
| Fuel per hr, lb/lb of basic air | 180 | 216 | 252 | 288 | 324 | 360 | 180 | 216 | 252 | 288 | 324 | 360 |
| Fuel lb per I.H.P./hr | .243 | .245 | .248 | .260 | .247 | .250 | .265 | .267 | .271 | .270 | .273 | 0.280 |
| I.M.E.P. psi | 312 | 370 | 426 | 465 | 552 | 599 | 344 | 416 | 477 | 547 | 610 | 660 |

Considering these results and plotting the I.M.E.P. on a diagram such as Fig. 1 of Progress Report No. 1, it is seen that if the engine with no aftercooler was rated at 312 I.M.E.P. at 2800 rpm, responsiveness would exist down to a speed of 1475 rpm for an additional air flow equal to that of the basic engine air flow: a not unattractive performance, particularly since the S.F.C. changes by about 10%, as a maximum, in the process on an indicated bases. Since the engine speed would decrease with responsiveness, the mechanical efficiency will generally improve under such conditions, with the result that the change in S.F.C. can generally be less than the 10% calculated.

## COMPOUNDING

The above calculations have been based upon turbo-charging, the assumption being that the exhaust gases are capable of driving the compressor only.

In the cycle under consideration, it is apparent that as the auxiliary air flow ($\Delta w$) increases, the energy available at the end of the expansion stroke is also increased, e.g., the pressure $P_7$ changes from 114 psi at $\Delta w = 0$ to 216 psi when $\Delta w = 1.0$; greater blow-down energy is thus available. To examine what improvement can be obtained from such conditions by means of compounding, the following cases were investigated.

<u>Case 1</u>. Where it is assumed that conventional present-day turbocharger efficiencies are employed.

<u>Case 2</u>. Where a reasonable increase in turbine and compressor efficiencies are employed, increases that can be easily justified with the best of modern design; however, such design will probably add something to their cost, unless produced in large numbers.

### CASE 1

Assumed isentropic compressor efficiency = 0.70
Assumed isentropic turbine efficiency = 0.86
Air manifold pressure = 29.4 psi
Exhaust manifold pressure = 25.5 psi
Atmospheric pressure = 14.7
Air flow (basic) = 1.0 lb/sec.

<u>Compressor</u>.—The compressor performance is given by the following:

$$W = \frac{wC_pT_a}{\eta_c}\left(R^{\frac{k-1}{k}} - 1\right)$$

where

$W$ = Btu of work for compression
$w$ = air flow, lb
$C_p$ = 0.241 Btu/lb/°F
$T_a$ = atmospheric temperature, °R
$R$ = pressure ratio of compressor
= $P_m/P_a$
$P_a$ = atmospheric pressure
$k$ = 1.4
$\eta_c$ = Isentropic compressor efficiency.

For both cases, with and without aftercooling the air enters and leaves the charger under the same conditions; thus so long as $P_m$ remains constant, the compressor work per lb of air is constant. If the inlet air to charger is at 115°F then

$$W = \frac{1.0 x .241 x 565}{0.70} (2.0^{.286}-1)$$

$$= 194.5\ (1.219-1)$$

$$= 42.7 \text{ Btu/lb of air.}$$

This will remain constant as $\Delta w$ is varied since the auxiliary air does not flow through the charger.

Turbine Output.—The gas flow through the turbo will be the basic plus auxiliary air and fuel or $(1+\Delta w)(1+F/A)$, its pressure and temperature will be $P_8$ and $T_8$, and the relationship between these parameters and the work done is given to a first approximation by

$$W = \frac{(1+\Delta w)(1+f)C_pT_8}{\eta_T}\left[1 - \left(\frac{P_a}{P_8}\right)^{k-1/k}\right]$$

in which $= 0.04322\ (1+\Delta w)T_8$ Btu

$W$ = work output Btu

$\eta_T$ = turbine efficiency = 0.86

$P_a$ = atmospheric pressure, psia

$C_p$ = specific heat at an average temp of expansion of, say, 1800°R

= 0.29 when $F/A$ = 0.05

$k$ = 1.31

No Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| Δw | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| $T_8$ | 2127 | 2021 | 2013 | 2075 | 2087 | 2098 |
| W Btu | 91.9 | 105.0 | 122.0 | 143.5 | 162.3 | 182.0 |

60% Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| $T_8$ | 2002 | 2015 | 2016 | 2080 | 2073 | 2096 |
| W Btu | 87.2 | 104.7 | 122.2 | 143.9 | 161.5 | 181.9 |

## Output Compounded Engine

No Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| Δw | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| Net cycle work | 524.3 | 622.6 | 718.9 | 781.9 | 929.4 | 1009.4 |
| Turbine work | 91.9 | 105.0 | 122.0 | 143.5 | 162.3 | 182.0 |
| Total output | 616.2 | 727.6 | 840.9 | 925.4 | 1091.7 | 1191.4 |
| Input to comp. | 42.7 | 42.7 | 42.7 | 42.7 | 42.7 | 42.7 |
| Net output | 573.5 | 684.9 | 798.2 | 882.7 | 1049.0 | 1148.7 |
| I.H.P./lb of basic air/sec | 811 | 969 | 1130 | 1250 | 1486 | 1627 |
| Fuel flow lb/hr/lb basic air | 180 | 216 | 252 | 288 | 324 | 360 |
| S.F.C. lb/I.H.P./hr | .222 | .223 | .223 | .230 | .218 | .222 |
| I.M.E.P. psi | 341.0 | 406.0 | 474.0 | 525.0 | 624.0 | 682.0 |

60% Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| Net cycle work | 479.5 | 572.0 | 656.0 | 752.7 | 838.2 | 907.7 |
| Turbine work | 87.2 | 104.7 | 122.2 | 143.9 | 161.5 | 181.9 |
| Total output | 566.7 | 676.7 | 778.2 | 896.6 | 999.7 | 1089.6 |
| Input to comp. | 42.7 | 42.7 | 42.7 | 42.7 | 42.7 | 42.7 |
| Net output | 524.0 | 634.0 | 735.5 | 853.9 | 957.0 | 1046.9 |
| I.H.P./lb of basic air/sec | 741.0 | 896 | 1041 | 1208 | 1358 | 1480.0 |
| Fuel flows lb/hr/lb basic air | 180 | 216 | 252 | 288 | 324 | 360 |
| S.F.C. lb/I.H.P./hr | .243 | .241 | .242 | .239 | .239 | .243 |
| I.M.E.P. psi | 376 | 462 | 536 | 620 | 696 | 761 |

It is seen that, despite a change in output of roughly 2:1, the S.F.C. remains practically unchanged in each case, a distinct advantage as far as engine operating temperatures are concerned.

CASE 2

In view of the unit efficiencies obtained today in small gas-turbine power plants, it is believed that a substantial gain can be made in some of the units of a turbo charger, provided some additional cost can be justified in the process. It will be assumed that the characteristics of the turbo charger can be changed from those of Case 1 to the following:

Compressor efficiency = 0.75
Turbine efficiency = 0.88

The other items of the specification remain unchanged. The calculations, similar to Case 1, now give the following results:

$$\text{Work of compressor} = \frac{1.0 \times 0.241 \times 565}{0.75} \left( 2.0^{0.286} - 1 \right)$$

$$= 39.8 \text{ Btu/lb of air}$$

Turbine output = $0.0443\ (1+\Delta w) T_8$ Btu.

Thus the turbine output as $\Delta w$ varies will be as below:

No Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| $\Delta w$ | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| W Btu of turbine | 94.0 | 107.6 | 125.0 | 146.9 | 166.2 | 186.2 |

60% Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| W Btu of turbine | 89.3 | 107.1 | 125.2 | 147.1 | 166.2 | 186.0 |

It is assumed that the engine cycle remains unchanged. Actually, there will be a small change in the case with no aftercooling, since the engine manifold temperature will be reduced a few degrees. With the above figures, the output of a compounded unit then becomes:

Output Compounded Engine

No Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| $\Delta w$ | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| Net output Btu | 578.5 | 690.4 | 804.1 | 889.0 | 1065.8 | 1155.8 |
| I.H.P./lb basic air/sec | 819.0 | 978.0 | 1139.0 | 1259 | 1509 | 1658 |
| I.M.E.P. psi | 344 | 410 | 478 | 529 | 634 | 687.0 |
| S.F.C. lb/I.H.P./hr | 0.22 | 0.221 | 0.222 | 0.228 | 0.215 | 0.22 |

60% Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| $\Delta w$ | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| Net output Btu | 529.0 | 639.6 | 741.4 | 860 | 964.6 | 1053.9 |
| I.H.P./lb basic air/sec | 749.0 | 905 | 1049.0 | 1218.0 | 1364.0 | 1491.0 |
| I.M.E.P. psi | 380.0 | 466.0 | 541.0 | 625.0 | 699.0 | 768.0 |
| S.F.C. lb/I.H.P./hr | 0.241 | 0.239 | 0.240 | 0.237 | 0.237 | 0.241 |

Comparison of these results with those of Case 1 shows that little has been gained by a research effort to improve the turbo-charger efficiencies alone, other things remaining the same. However, it will be found that a reasonable gain in power is possible if the improved efficiencies are employed to increase the turbo-charger pressure ratio instead of holding it constant as here.

## HIGH-PRESSURE AIR COMPRESSOR

So far no allowance has been made for the power required to supply the high-pressure air employed. This power must be subtracted from the engine output and is calculated on the following basis.

It is assumed that a three-stage air compressor is employed which is to deliver at a pressure of 1600 psi for injection of the air into the cylinder where the pressure is to be maintained at 1500 psi. The previous calculations have assumed a maximum of 1 lb of compressed air to be supplied by the compressor per lb asperated into the engine. On this basis the power for the compression of 1 lb of air in a three-stage compressor can be obtained.

The conventional relationship for maximum efficiency of a multi-staged compressor will be employed.

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} \cdots = \frac{P_n}{P_{n-1}}$$

where

$P_1$ = initial pressure
$P_2$ = pressure in 2nd stage
$P_3$ = pressure in 3rd stage
$P_n$ = pressure in nth stage.

Then with no losses between stages, but with intercooling to the original temperature, it can be shown, for a three-stage machine, that

$$W = \frac{RT_1}{J} \frac{3n}{n-1} \left[ \left( \frac{P_4}{P_1} \right)^{\frac{n-1}{3n}} - 1 \right] \text{Btu/lb}$$

$$= \frac{53.34 \text{x} 585}{778} \frac{3 \text{x} 1.38}{0.38} \left[ \left( \frac{1600}{14.7} \right)^{\frac{0.38}{3 \text{x} 1.38}} - 1 \right]$$

$$= 234.5 \text{ Btu/lb of air.}$$

This assumes that the inlet temperature $T_1$ is 125°F and that n in $PV^n = C$ for the compressor is 1.38. In this case the efficiency of the process is given by

$$\eta = \frac{\log_e r}{\frac{3n}{n-1}\left[r^{\frac{n-1}{3n}} - 1\right]}$$

where

$r = P_4/P_1$ the pressure ratio.

Then

$\eta = 80\%$.

These values are close enough for present purpose (they may be a little optimistic). It follows that the net performance of an engine fitted with such a compressor will be as follows.

## CONTINUOUS PERFORMANCE AT MAXIMUM POWER

In this case, where the engine output can be maintained at all times equal to the maximum desired, Δw will vary from 0 to 1.0 for the complete range covered previously. It follows that:

### Net Output Compounded

No Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| Δw | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| Engine output Btu | 573.5 | 684.9 | 798.2 | 882.7 | 1049.0 | 1148.7 |
| Compressor input Btu | 0.0 | 46.9 | 93.9 | 140.8 | 187.6 | 234.5 |
| Net output engine and comp. Btu | 573.5 | 638.0 | 704.3 | 741.9 | 861.4 | 914.2 |
| Fuel lb/hr | 180 | 216 | 252 | 288 | 324 | 360 |
| S.F.C. | 0.222 | 0.237 | 0.253 | 0.274 | 0.266 | 0.279 |
| Over-all net I.M.E.P. | 341.0 | 379.0 | 418 | 441 | 512 | 543 |

60% Aftercooling

| | | | | | | |
|---|---|---|---|---|---|---|
| Δw | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
| Engine output Btu | 524.0 | 634.0 | 735.5 | 853.9 | 957.0 | 1046.9 |
| Compressor input Btu | 0.0 | 46.9 | 93.9 | 140.8 | 187.6 | 234.5 |
| Net output engine and comp. Btu | 524.0 | 587.1 | 641.6 | 713.1 | 770.4 | 812.4 |
| Fuel lb/hr | 180 | 216 | 252 | 288 | 324 | 360 |
| S.F.C. | 0.243 | 0.260 | 0.277 | 0.285 | 0.297 | 0.513 |
| Over-all net I.M.E.P. | 376 | 428 | 467 | 518 | 560 | 691 |

The above ideal results would be reduced to a brake horse power performance approximately as given below when corrected for normal losses.

B.H.P. Performance

| Δw | | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|---|
| B.H.P. | No cooling | 700 | 788 | 877 | 926 | 1036 | 1158 |
| | Aftercooling | 639 | 725 | 800 | 895 | 974 | 1028 |
| S.F.C. | No cooling | .257 | .274 | .287 | 0.31 | .322 | .347 |
| lb/B.H.P./hr | Aftercooling | .282 | .298 | .315 | 0.322 | .333 | .357 |

INTERMITTENT PERFORMANCE AT MAXIMUM POWER

The results given above represent the engine fitted with a compressor capable of maintaining the air supply indefinitely. At the opposite end of the scale, if the maximum output was maintained for a limited time only, then the responsive effect could be improved since the input to the compressor would be reduced proportionally.

Assume that the peak power is required for a period of 5 minutes per hour. There would be no change in the maximum power available except that resulting from the small change in compressor power, but the average compressor input would be reduced by 1:12 and the performance would be as shown below.

Net Output Compounded

| Δw for 5 min | | 0.0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 |
|---|---|---|---|---|---|---|---|
| Max. net output | No cooling | 700 | 831 | 963 | 1055 | 1208 | 1373 |
| B.H.P. | With cooling | 639 | 768 | 886 | 924 | 1046 | 1243 |
| S.F.C. | No cooling | 0.257 | 0.26 | 0.262 | 0.272 | 0.276 | 0.293 |
| lb/B.H.P./hr | With cooling | 0.282 | 0.282 | 0.284 | 0.312 | 0.310 | 0.295 |

The above figures are based upon a storage vessel capable of holding sufficient air for the five-minute burst of power, plus a compressor capable of re-charging the vessel in the remaining fifty-five minutes.

Two such bursts of power per hour could be obtained with little change in the above values. The longer the time required, the closer the results approach the continuous values.

## AIR COMPRESSOR

To visualize the final power plant, some idea of the compressor displacement necessary for the purpose is required.

Taking the case of continuous operation at the maximum value of $\Delta w$, i.e., 1.0 lb per lb of basic air, then for a 500-hp output engine with aftercooler:

$$\text{Basic air supply} = 1 \times \frac{500}{1028} \times 3600$$

$$= 1752 \text{ lb/hr}$$

with an ambient temperature of 115°F at 14.7 psia.

$$\text{Volume of air} = \frac{w\,RT}{P} = \frac{1752 \times 53.34 \times 575}{14.7 \times 144}$$

$$= 25500 \text{ cu ft/hr.}$$

This must be the capacity of the compressor since $\Delta w = 1.0$. Assuming a volumetric efficiency of 0.80 with a compressor (reciprocating type) operating at 4000 rpm and employing a double-acting L.P. compressor cylinder, then

$$\text{Cylinder displacement/rev.} = \frac{\pi D^2}{4} \times L \times 2 \text{ cu in./rev.}$$

$$\text{Air displacement} = \frac{\pi D^2}{4 \times 1728} \times L \times 2 \times 0.80 \times 4000 \times 60 \text{ cu ft/hr.}$$

where

$D$ = diameter of L.P. cylinder in in.

$L$ = stroke of L.P. cylinder in in.

Thus

$$25500 = \frac{\pi D^2}{4 \times 1728} \times L \times 2 \times 0.80 \times 4000 \times 60$$

$$D^2 L = \frac{25500 \times 4 \times 1728}{2 \times \pi \times 0.80 \times 4000 \times 60}$$

$$= 146.2 \text{ cu in.}$$

If $D/L = 1.0$, then $D^3 = 146.2$

$$D = 5.26 \text{ in. and } L = 5.26 \text{ in.}$$

This is a fairly large compressor, but not significantly out of proportion to a 500 B.H.P. Diesel engine when it is considered that due to its attachment to the engine a B.H.P. of 805 can be developed continuously.

Looking at the case for a 5-minute burst of power, then the maximum output of a basic 500-hp engine will become 974 B.H.P.; for this case, a compressor of the same characteristics as above is calculated as follows.

$$\text{Compressor delivery} = 25500 \times 5/60 \text{ cu ft/hr}$$

$$= 2130 \text{ cu ft/hr}$$

or

$$D^2L = \frac{2130 \times 4 \times 1728}{2 \times \pi \times 0.80 \times 4000 \times 60}$$

$$= 12.2 \text{ cu in./rev.}$$

If D = L, then

$$D = 2.3 \text{ and } L = 2.3.$$

This is a quite small compressor, compared with a 500-hp Diesel, to permit a maximum power of 974 B.H.P. for five minutes.

To the small compressor above, a receiver must be added to provide air storage.

$$\text{Compressed air/hr} = 1752 \times 5/60$$

$$= 146.1 \text{ lb/hr.}$$

Assume delivery air temperature of 700°R at a pressure of 1600 psi.

$$\text{Volume of air} = \frac{w\ RT}{P}$$

$$= \frac{146.1 \times 53.34 \times 700°}{1600 \times 144}$$

$$= 23.7 \text{ cu ft.}$$

Thus an air receiver of 24 cu ft volume will contain all the air for one boost without any allowance for the air delivered by the compressor during the period. This means a vessel of about 2 ft diam by 7.5 ft long, perhaps a not impossible size since the bulk of the engine, if the full 975 hp was required, will be reduced by a volume of about 50 cu ft if it is figured at an output of 10 hp per cut ft of engine volume. In other words, a 500-hp engine plus receiver and compressor will occupy 50+2+24 = 76.0 cu ft of space and develop a maximum of 975 hp. A normal engine of 975 hp would occupy 97.5 cu ft, or a net saving of 20 cu ft of engine space.

If the comparison is confined to the unit giving a continuous maximum output, when aftercooled, then for a normal output of 500 hp when $\Delta w = 0.0$, a maximum of 805 hp will be obtained for $\Delta w = 1.0$. In this case no receiver will be necessary and the engine bulk will still be 50 cu ft at 10 hp/cu ft; the compressor of 5-1/2 x 5-1/2 size could be expected to occupy not more than 3-5 cu ft, giving a total bulk of, say, 55 cu ft for an engine with a maximum capacity of 805 B.H.P. against about 80 cu ft for a normal turbo-charged unit of the same power.

It must be kept in mind that the arbitrary rating of a normal 500 B.H.P. was picked at random for purely comparative purposes. It is recognized that the 800 to 900 maximum horse power resulting is more than is necessary for the purpose in mind at the present time. The engine size, bulk, etc., for a moderate change, say to 300 hp normal and 480 to 550 hp maximum, would roughly result in volumes of 35 to 45 cu ft for continuous or intermittent operation, respectively.

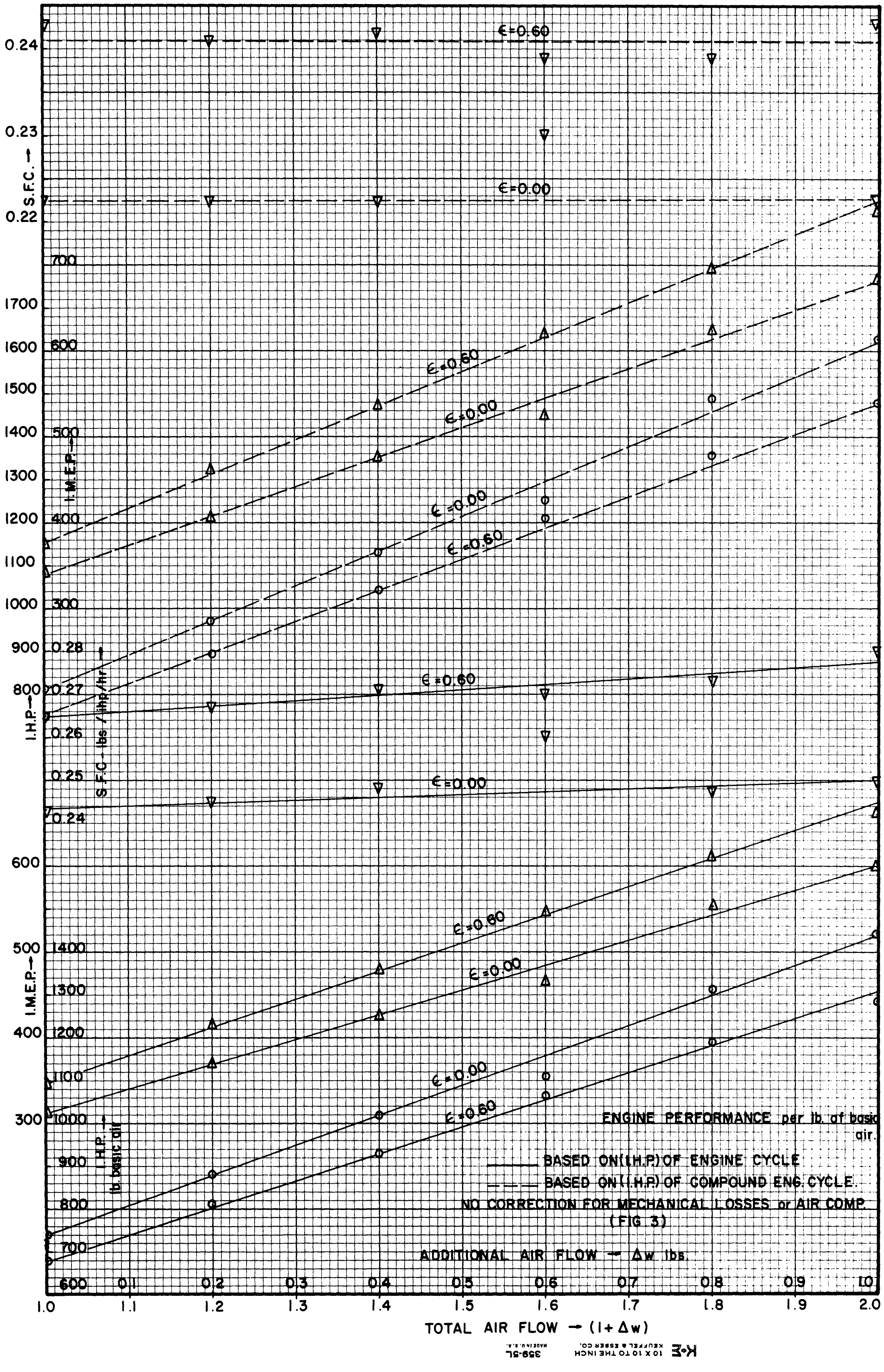
Figures 3, 4, and 5 show the performance at maximum output under the various conditions, on an engine I.H.P. and B.H.P. basis.

The specific fuel consumptions given represent the minimum attainable on the assumption of perfect combustion of the fuel with the air. In practice, particularly at the high F/A ratio of 0.05, some departure from these values is to be expected, since at such a ratio as the above some smoke is usual and a realistic value of S.F.C. would be some 15% greater than that shown. In addition, the B.H.P. given is that of the bare engine and must be reduced by the accessories driven, such as cooling fan, generator, etc. An allowance for these items cannot be made at the present time since the cooling fan power will depend not only on the engine but the efficiency of the transmission attached to it.

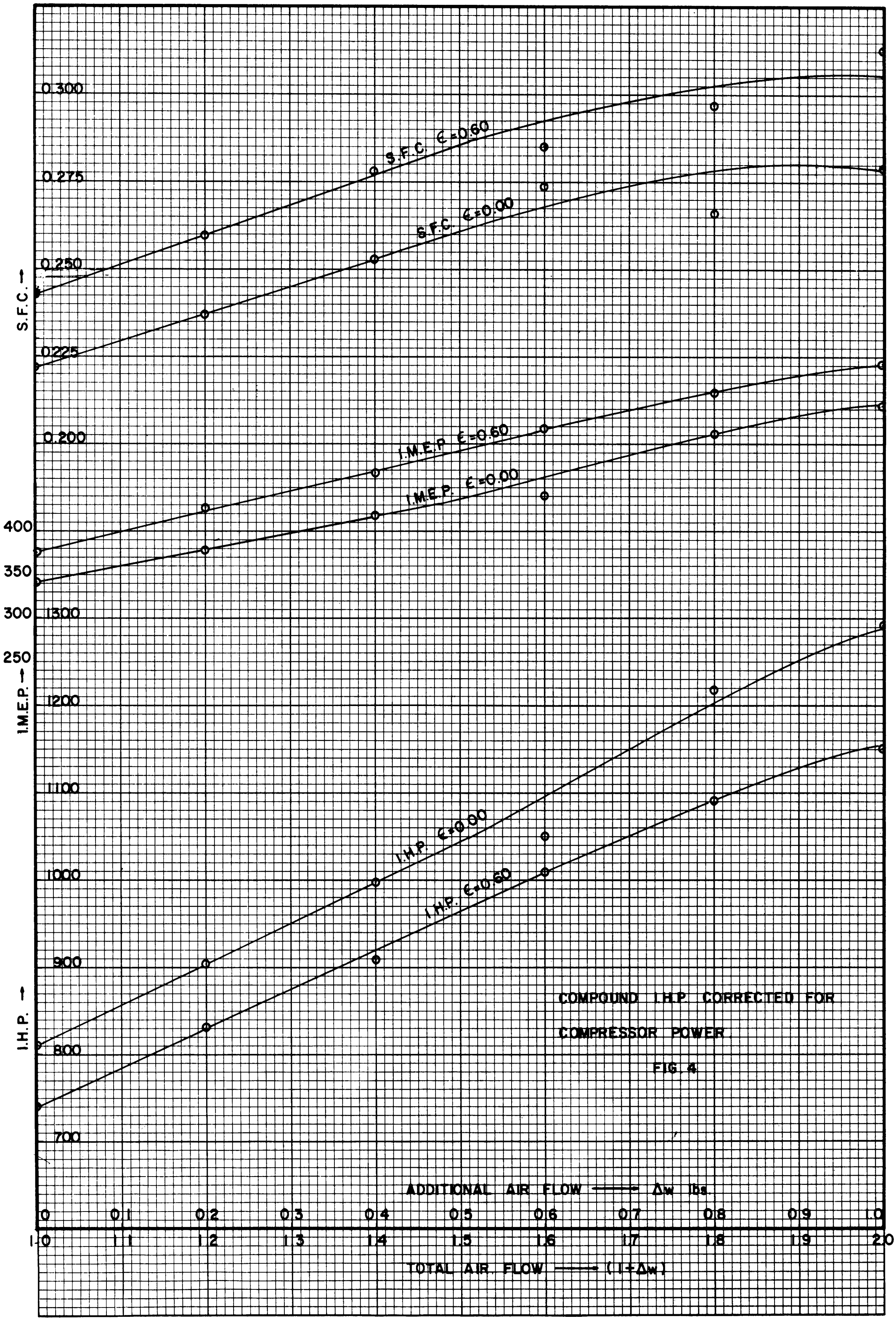
At a later date in the analysis, when the transmission is finalized and the required generator size is established, actual performance curves can be developed for the above conditions.

## CONCLUSIONS

The method of additional air supply at the TDC of the compression stroke has advantages for the purpose of military vehicle propulsion where high

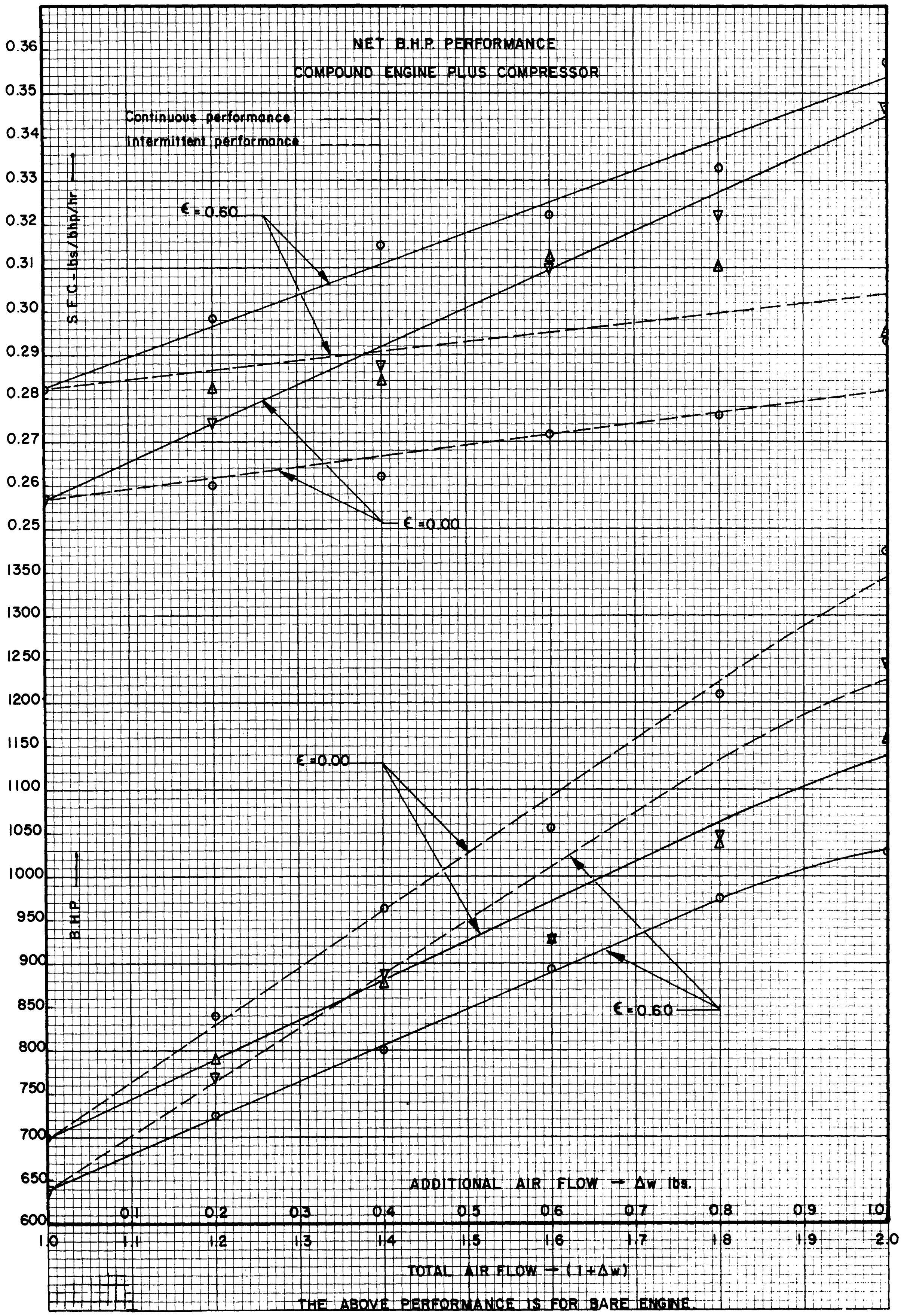


ENGINE PERFORMANCE per lb. of basic air.
BASED ON (I.H.P.) OF ENGINE CYCLE
BASED ON (I.H.P.) OF COMPOUND ENG. CYCLE.
NO CORRECTION FOR MECHANICAL LOSSES or AIR COMP.
(FIG 3)
ADDITIONAL AIR FLOW → Δw lbs.
TOTAL AIR FLOW → (1 + Δw)
S.F.C.
I.M.E.P.
I.H.P.
S.F.C. - lbs / ihp / hr
I.H.P. / lb. basic air
ε = 0.60
ε = 0.00



S.F.C.
0.300
0.275
0.250
0.225
0.200
S.F.C. ∈=0.60
S.F.C. ∈=0.00
I.M.E.P.
400
350
300
250
I.M.E.P. ∈=0.60
I.M.E.P. ∈=0.00
I.H.P.
1300
1200
1100
1000
900
800
700
I.H.P. ∈=0.00
I.H.P. ∈=0.60
COMPOUND I.H.P. CORRECTED FOR COMPRESSOR POWER.
ADDITIONAL AIR FLOW → Δw lbs.
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0
1.0 1.1 1.2 1.3 1.4 1.5 1.6 1.7 1.8 1.9 2.0
TOTAL AIR FLOW → (1+Δw)


FIG. 4



NET B.H.P. PERFORMANCE
COMPOUND ENGINE PLUS COMPRESSOR
Continuous performance
Intermittent performance
S.F.C. - lbs/bhp/hr
B.H.P.
€ = 0.60
€ = 0.00
€ = 0.00
€ = 0.60
ADDITIONAL AIR FLOW → Δw lbs.
TOTAL AIR FLOW → (1+Δw)
THE ABOVE PERFORMANCE IS FOR BARE ENGINE.

FIG .5

power is needed for a small fraction of the time. In addition, a degree of responsiveness can also be obtained at the same time by the use of this principle (roughly a 2:1 speed change at constant horse-power).

If the engine and compressor are properly proportioned, it appears that a normal 350-hp compression ignition engine could take the place of a 600-hp one, resulting in a reduction of engine bulk and possibly weight. There would be an increase in the fuel used at the maximum load condition, but a decrease at the 350 hp and less. Since the engine operates most of the time at part load, with 40% of the time at idle, it is believed that an actual reduction in the fuel rate in lb/hr will be achieved when this engine is coupled to the transmission and the operational performance finally established.

The penalty to be paid would be the additional complication of the air-supply system. However, the increased power is obtained without increase of stress levels.