Chapter III

EXPERIMENTAL RESULTS AND DISCUSSION

3.1 Presentation of experimental results

The characteristics of engine performance and exhaust gas emissions as straight diesel operation and diesel in combination with gas (Commercial Butane) were investigated. Results are presented graphically in Fig.10 to 30. For simplification, these graphs are separated into two parts following the results obtained from two engines. The corresponding tabulated results and sample of calculation are presented in Appendix I.

3.2 Discussion of section I

In this section, experimental results from Kubota engine are presented. The experiments were successively carried out at a constant speed of 1200,1400,1800 and 2000 rpm. For each speed the engine was run with both straight diesel and various combinations of gaseous fuel and diesel fuel. The test was commenced by varying brake load from about 2 lb_f to 10 lb_f with an increment of about 2 lb_f. At the maximum brake load of each speed, the gaseous fuel was increased from a minimum 0.0053 lb_m/min. until the engine started knocking. The fuel mixture ratio hence represented the maximum of gaseous fuel that can be admitted to displace diesel fuel without damaging the engine. More details of this operating method have been described in Part 1 of experimental procedure in the preceding chapter. The heating value of diesel fuel was obtained directly from Diesel Engineering Handbook[1] whereas for gaseous fuel it was obtained from Esso gas specification.[2]

3.2.1 Brake thermal efficiency

The engine's performance as straight diesel and as diesel in combination with gas are given in Fig.10 to 13 which show the variation of brake thermal efficiency with respect to brake mean effective pressure (Bmep) at speed 1200, 1400,1800 and 2000 rpm respectively. The curves in each figure show the influence of gas addition on brake thermal efficiency as compared with straight diesel.

The efficiency curves are almost identical with respect to shape and thus obviously show that the brake thermal efficiency of gas-diesel combination under low load conditions are considerably inferior to straight diesel. This inferiority can be explained as under low load conditions requiring small quantity of diesel oil,any addition of gaseous fuel will

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[1]Karl W. Stinson, Diesel Engineering Handbook. (11th ed., Stanfort: Conn. Diesel Publications, 1969), p. 38.

[2]Esso Standard Thailand Limited, Physical Properties of Commercial Propane and Butane (June 1967).

automatically decrease the quantity of diesel fuel, resulting in poor injection characteristics and consequently poor air/fuel mixture distribution. Combustion, therefore, does not take place properly and some parts of fuel passed through the combustion chamber without taking part in the combustion process. Moore and Lewis[3] confirmed that at very weak mixture most probably only that region of the charge directly in contact with injected spray was fully reacted. As the percentage of the gaseous fuel in the mixture was increased, the brake thermal efficiency became less for same value of Bmep. This is because the increased percentage of gas in air makes it more difficult for fuels to find enough oxygen to ignite efficiently.

At high load conditions, over 80 % of maximum Bmep, a slight improvement in brake thermal efficiency over straight diesel was obtained at 1200 and 1400 rpm. This improvement is due to increased amount of diesel fuel injected, which act as tiny "spark plugs" among the gas/air mixture, thus promoting more efficient combustion. Fig.18-19 show hydrocarbon and carbon monoxide, the amounts of which are much lower than straight diesel emissions.

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[3] N.P.W. Moore and J.D. Lewis, "An Investigation of Combustion in Dual Fuel Engines" IV Congre's International du Chaufflage Industriel (10/3/1952).

At higher speeds of 1800 and 2000 rpm, the brake thermal efficiency at high load conditions for gas diesel running are lower than straight diesel. As for the exhaust emissions, the hydrocarbon and carbon monoxide, they are also lower than that of straight diesel (see Fig.20-21). However, the lower amount of these quantities usually indicate an improvement in combustion.

The results obtained agree with other previous works on dual fuel experiments,[4,5] i.e., thermal efficiency is greatly reduced at low load conditions. However, this is not the case for high load conditions, where the thermal efficiency has an edge over straight diesel, particularly at low speed.

3.2.2 Exhaust smoke

Fig.14 to 17 represent exhaust smoke level which was measured in Bosch smoke number and plotted against Bmep, for straight diesel and dual fuel runnings at speed 1200,1400,1800 and 2000 rpm respectively. It clearly shows that smoke emission level steeply increases as Bmep is increased over 60 psi. Under the same Bmep, for straight diesel, exhaust smoke level at low

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[4] D. Lyon, A.H. Howland and W.L. Lom, "Controlling Exhaust Emissions from a Diesel Engine by LPG Dual Fuelling" Air Pollution Control in Transport Engines (London, I. Mech. E., 1972), pp. 45-46.

[5] N.P.W. Moore and R.M.S. Mitchell, "Combustion in Dual Fuel Engines" Proc. Joint Conference on Combustion (London, I. Mech. E., 1972), pp. 300-301.

speed is higher than at high speed, as can be seen in Fig.14-17 at maximum Bmep, the Bosch smoke number are 6.5,5.6,4.75 and 4.5 as speed increases from 1200 to 2000 rpm. Comparing with the smoke limit provided in the previous chapter, where the maximum acceptance smoke number is 4.0, the smoke obtained at maximum Bmep are above acceptance level.

Under running in combination with gaseous fuel, the curves show a reduction in smoke level which are more dramatic at high load or Bmep and also with high percentage of gas. At high speed the effect of reduction in smoke level is more than at low speed. Fig.14 shows about 40 % reduction at maximum Bmep, whereas Fig.17 shows about 50 % reduction for the same 1.85 % gas in air mixture. This reduction is due to the effect of premixed flame of the gaseous fuel which is mixed with air long before ignition occurs, while in normal diesel, the fuel injected mainly burn as a diffusion flame, and as a result is pyrolysed to form smoke where oxygen can not mix with fuel in time.

3.2.3 Exhaust gas emissions

In Fig.18-21, under part load conditions (Bmep lower than 60 psi), the emissions curves for dual fuel show larger amounts of hydrocarbon and carbon monoxide than that obtained from straight diesel for all speed range, whereas carbon dioxide is less. This indicates that poor combustion efficiency take place in the combustion chamber. A considerable increase in both hydrocarbon and carbon monoxide was obtained when more gas was added, with carbon dioxide behaving the opposite. Consequently, serious drop in brake thermal efficiency as already mentioned in subsection 3.2.1 is resulted.

As the load is increased, hydrocarbon decreases to meet straight diesel emission level at maximum Bmep, with carbon monoxide nearly constant for all loads. However at maximum Bmep, carbon monoxide for dual fuel is lower than that obtained from straight diesel. The amount of decreases in emissions also depend on speed, Fig.22 shows the effect of speed on emissions at maximum Bmep. The hydrocarbon is more reduced at high speed especially at 1800 rpm, whereas carbon monoxide is more reduced at low speed. From these results, it may be said that the gaseous fuel is effective in reducing only one component of emissions, namely hydrocarbon at high speed and carbon monoxide at low speed.

3.2.4 The influence of gas concentration on engine performance and exhaust emissions at maximum load

The previous results indicated that the admission of gaseous fuel are most effective only at high load. The variatinn of gas concentration is an important to the engine performance and exhaust emissions. In order to find out optimum amount of gas that can give the best results in both performance and emissions and also to find the maximum amount of gas that can be added without knocking, the test was revised by keeping load and speed constant and varying the gas from minimum to maximum until the engine seriously knocks.

Fig.23 shows the variation of engine performance and emissions with gas concentration at 1200 rpm. The engine started knocking as the gas concentration reached 41 % of total fuel by weight (1.75 % gas in air). Brake thermal efficiency was greatly improved over normal diesel run for every concentration of gas. As for the exhaust gas emissions, carbon monoxide was dramatically reduced from normal diesel run while hydrocarbon was increasing, however at 37 % gas in total fuel a lowest hydrocarbon level was obtained. The exhaust gas temperature and smoke were also reduced, and in case of smoke the reduction was nearly half at maximum gas concentration. It appears that the most satisfactory results concerning performance and emissions is obtained around 37 % gas in total fuel. This results agreed well with that obtained in Ford 2500 engine[6], whose the most effective concentration obtained was 28.3 %.

At 1400 rpm the exhaust emissions behave the same as obtained at 1200 rpm, hydrocarbon and carbon monoxide tend to increase as concentration of gas was over 30 %, as a result of reduction in engine thermal efficiency. As seen in Fig.24, there is no improvement in brake thermal efficiency and the tendency is such that it is going below that of the normal diesel run as gas concentration became more than 30 % of total

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[6]"Diesel Exhaust Emissions Reduction by L.P.G. Supplementary Fuelling" Clean Air. (Vol.3, No. 10, Summer 1973), p. 45.

fuel. Like at 1200 rpm, the engine knocked at about 40 % gas (1.75 % gas in air), and again the exhaust smoke was reduced by half.

For high speed 1800 and 2000 rpm, results were identically obtained as shown in Fig.25 and 26. Brake thermal efficiency varies slightly with gas concentration and close to straight diesel, and therefore no significant effect due to gaseous fuel can be drawn. However, the exhaust emissions have shown worse combustion efficiency than normal diesel run as hydrocarbon was observed to increase dramatically. Only the exhaust temperature and smoke were reduced. The engine also knocked at 40 % gas in total fuel (1.85 % gas in air) at 2000 rpm, but with somewhat lower for 1800 rpm, i.e., at 36 % gas in total fuel (1.53 % gas in air).

The results obtained so far, indicates that the additional gas should not be more than 40 % of total fuel, which corresponds to about 1.75 % gas in air. Comparison of inflamability limit of butane gas from the other work[7] confirms that the engine knock occurs as the air-gas mixture reach the lower inflamability limit. It is also shown that the pronounced advantage of gas addition is obtained at low speed, hence it seems that the optimum concentration of gas for best emissions may depend on speed.

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[7] Lyon et al., op.cit., p. 44.

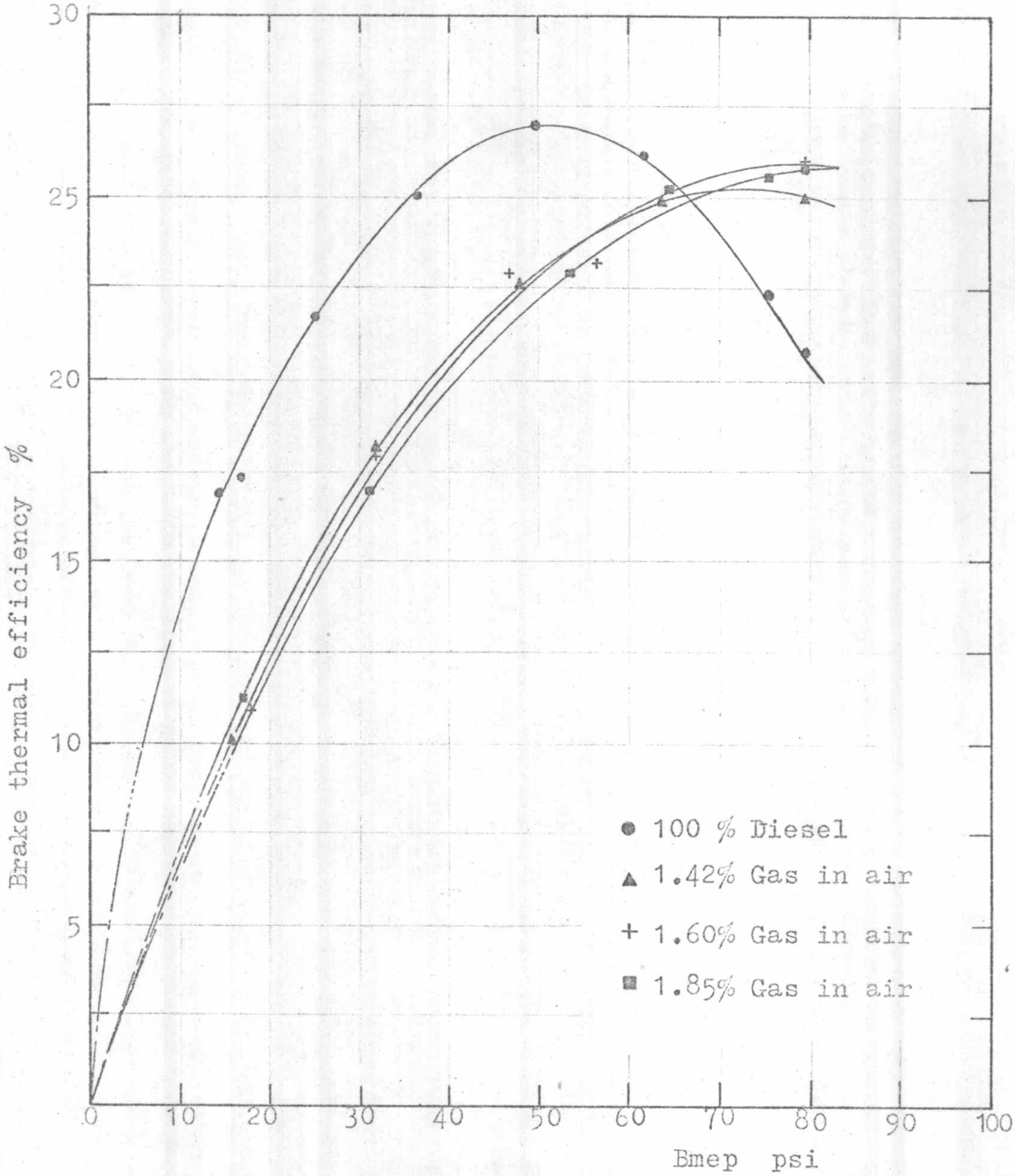

Fig.10 Variation of brake thermal efficiency with power output for different gas admission in Kubota KND 3 engine at 1200 rpm.

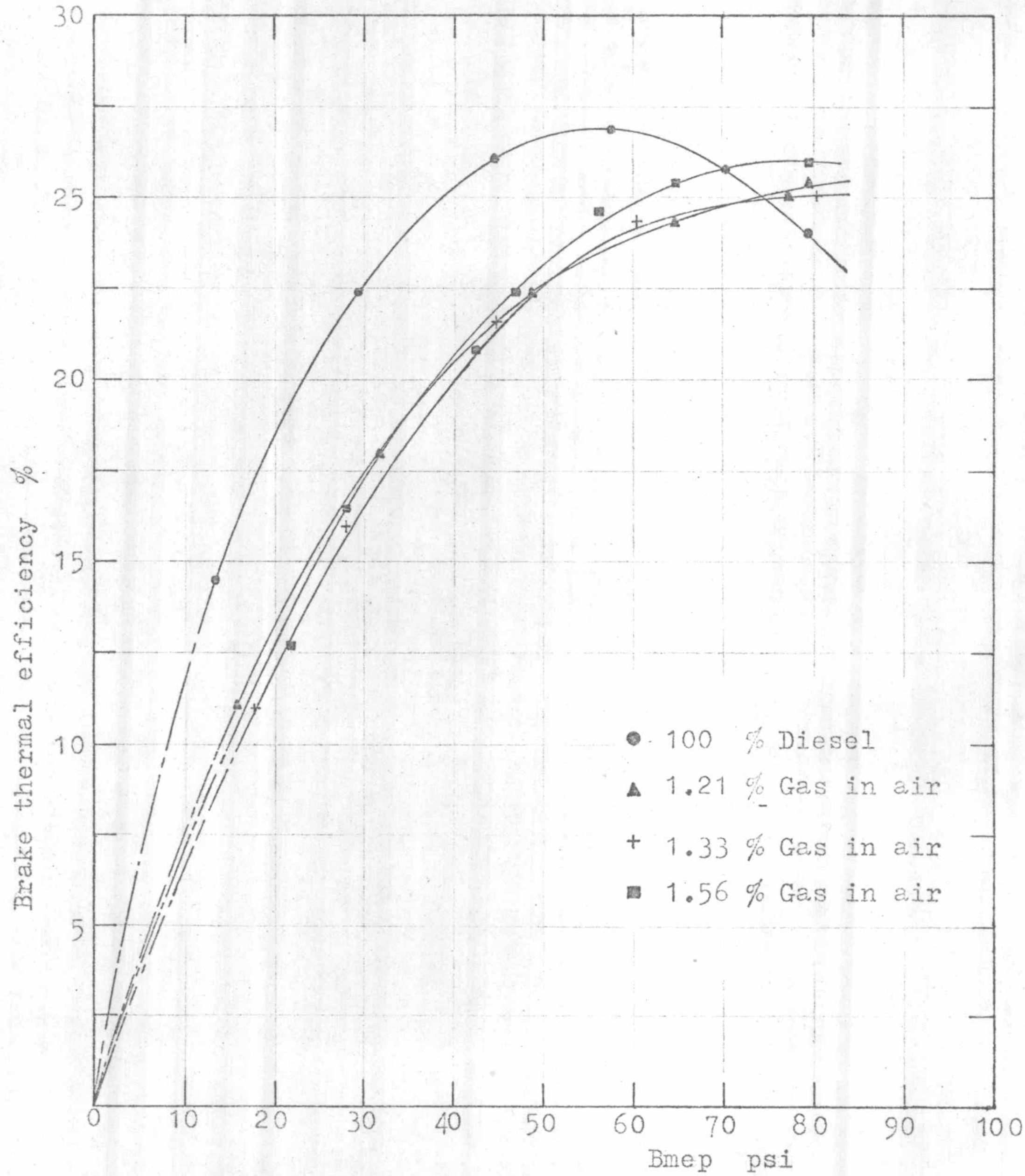

Fig. 11 Variation of brake thermal efficiency with power output for different gas admission in Kubota KND 3 engine at 1400 rpm.

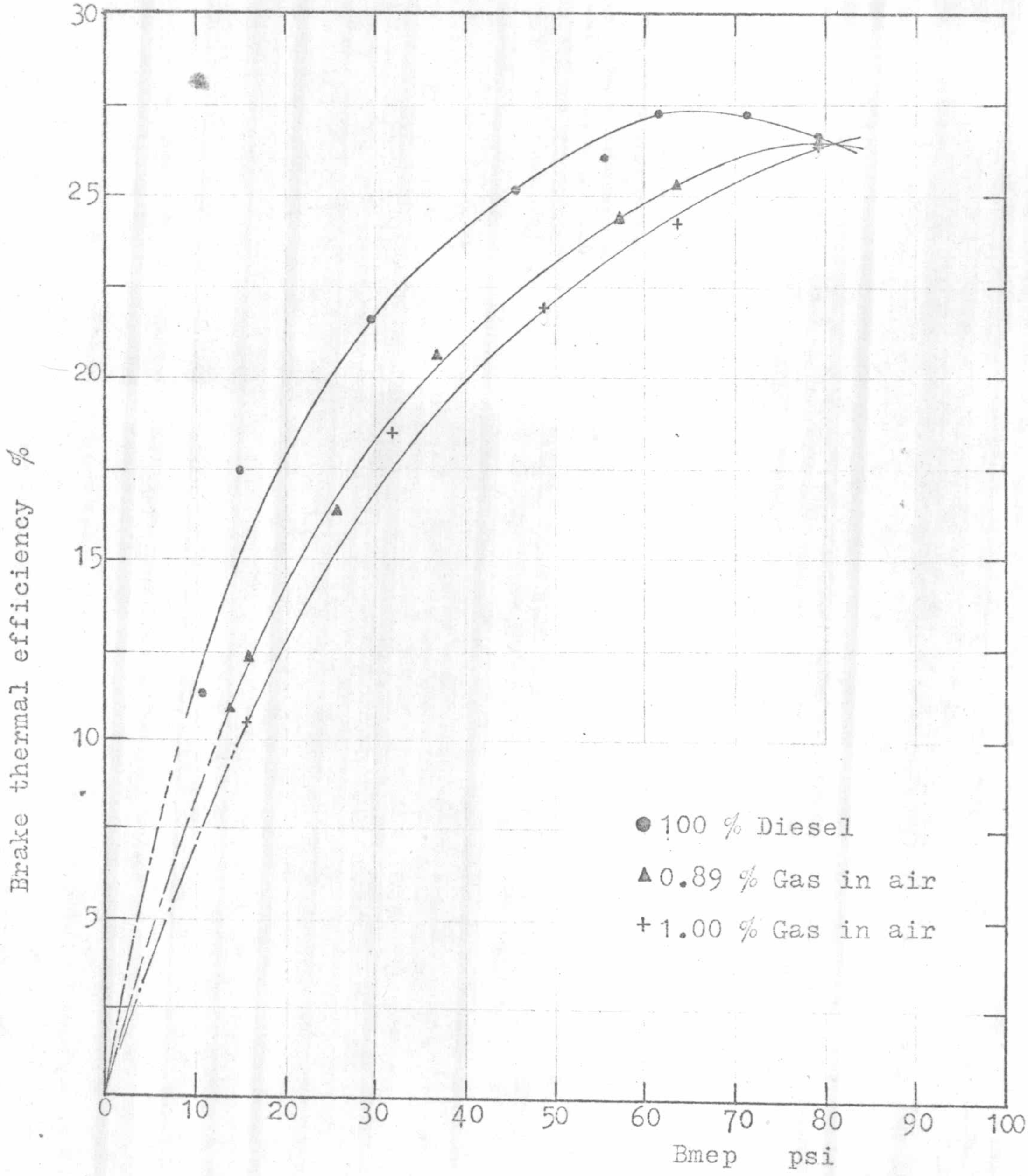

Fig. 12 Variation of brake thermal efficiency with power output for different gas admission in Kubota KND 3 engine at 1800 rpm.

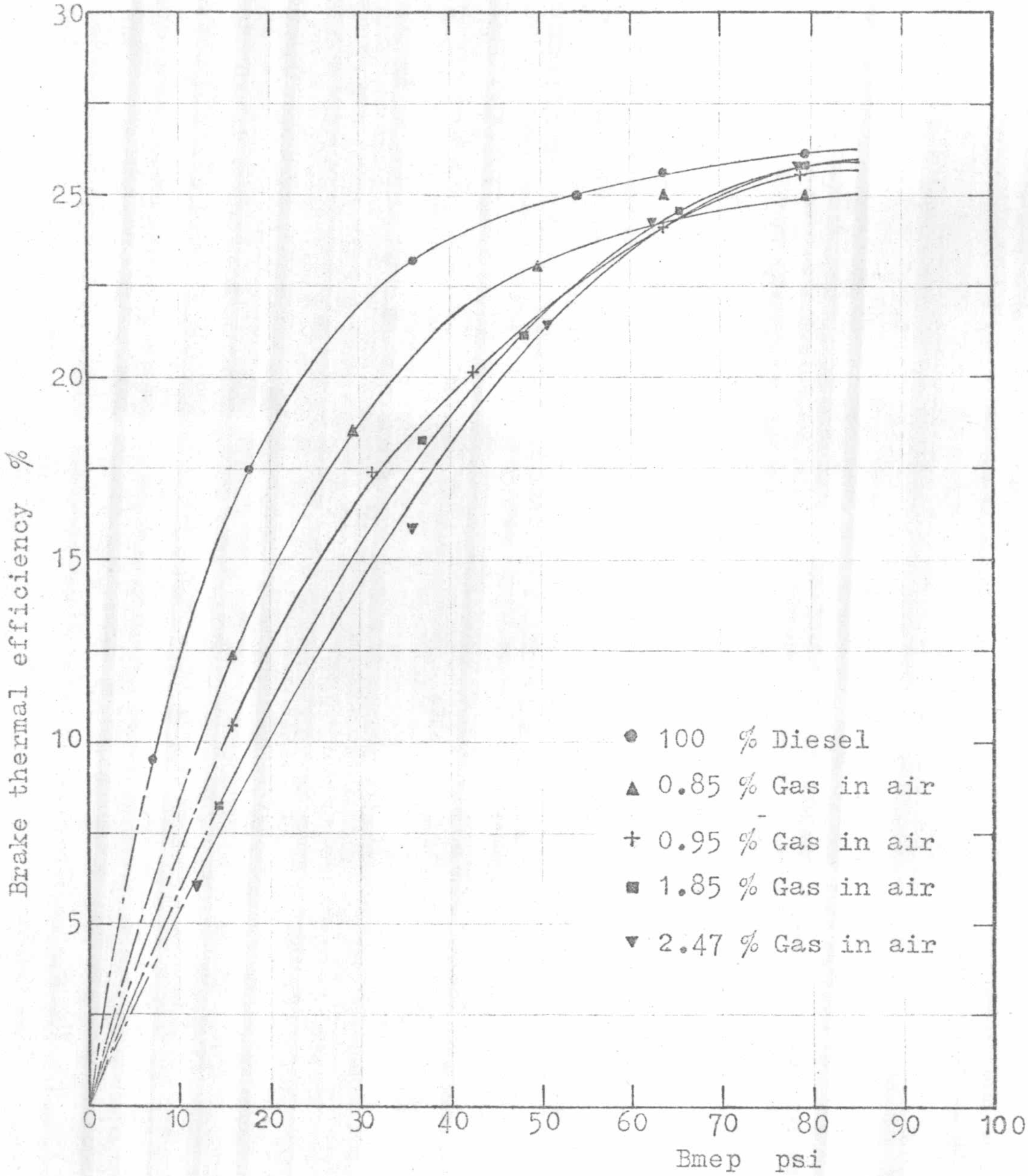

Fig.13 Variation of brake thermal efficiency with power output for different gas admission in Kubota KND 3 engine at 2000 rpm.

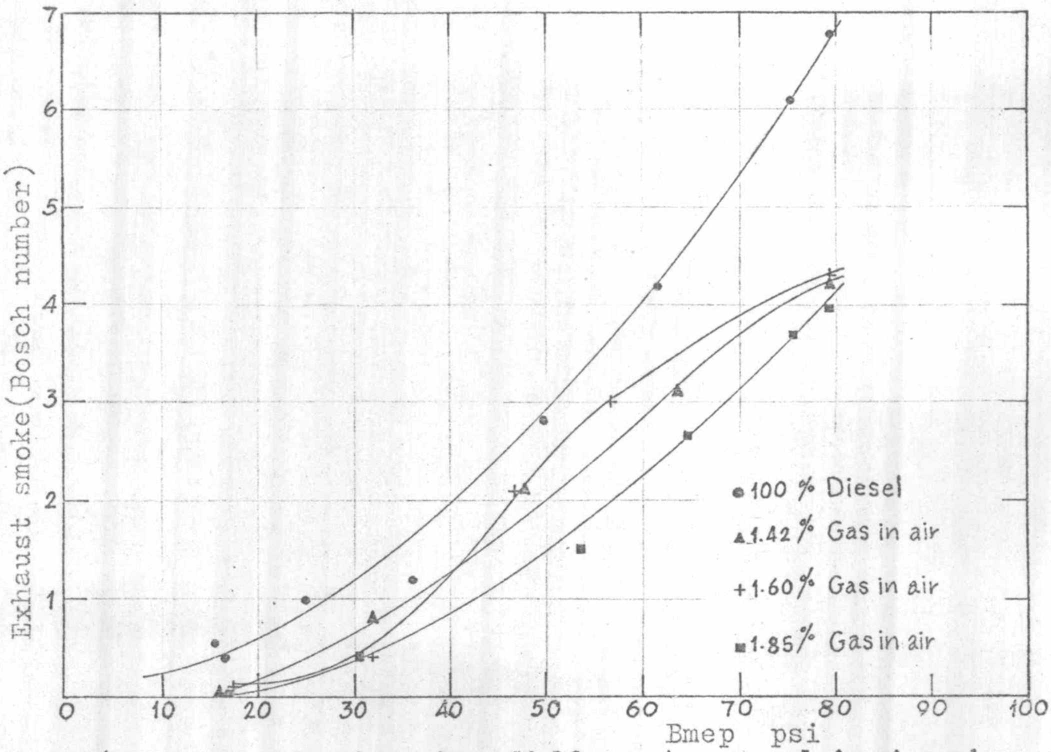

Fig.14 Smoke emission for different gas admission in Kubota KND 3 engine at 1200 rpm.

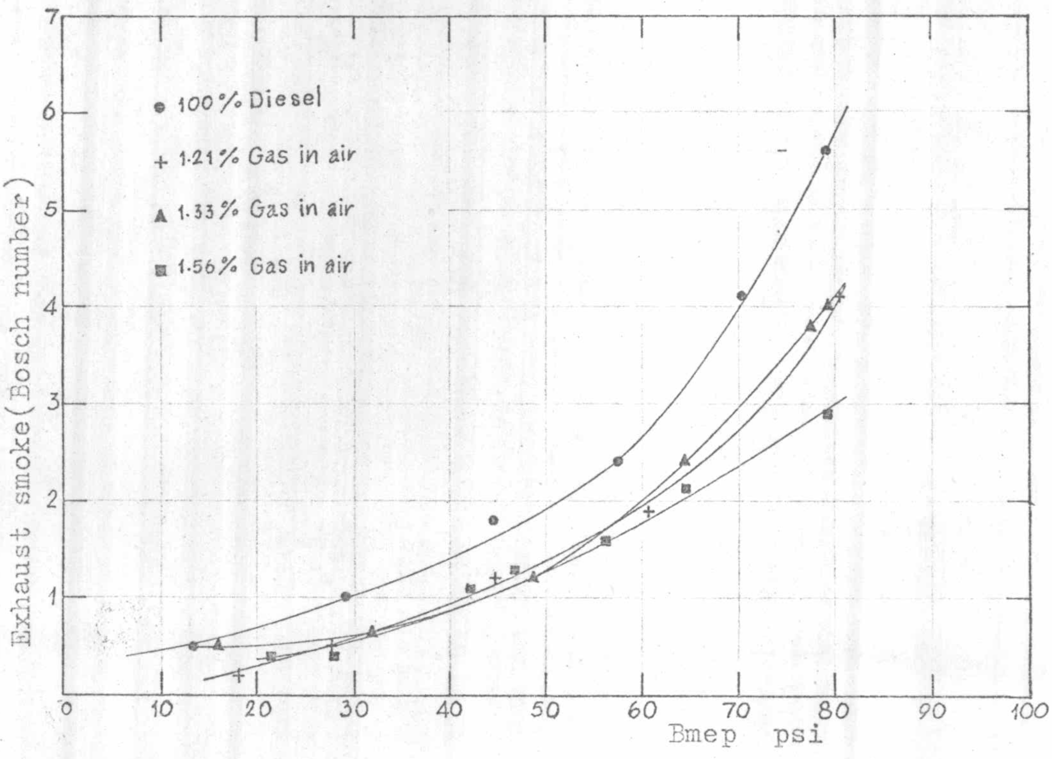

Fig.15 Smoke emission for different gas admission in Kubota KND 3 engine at 1400 rpm.

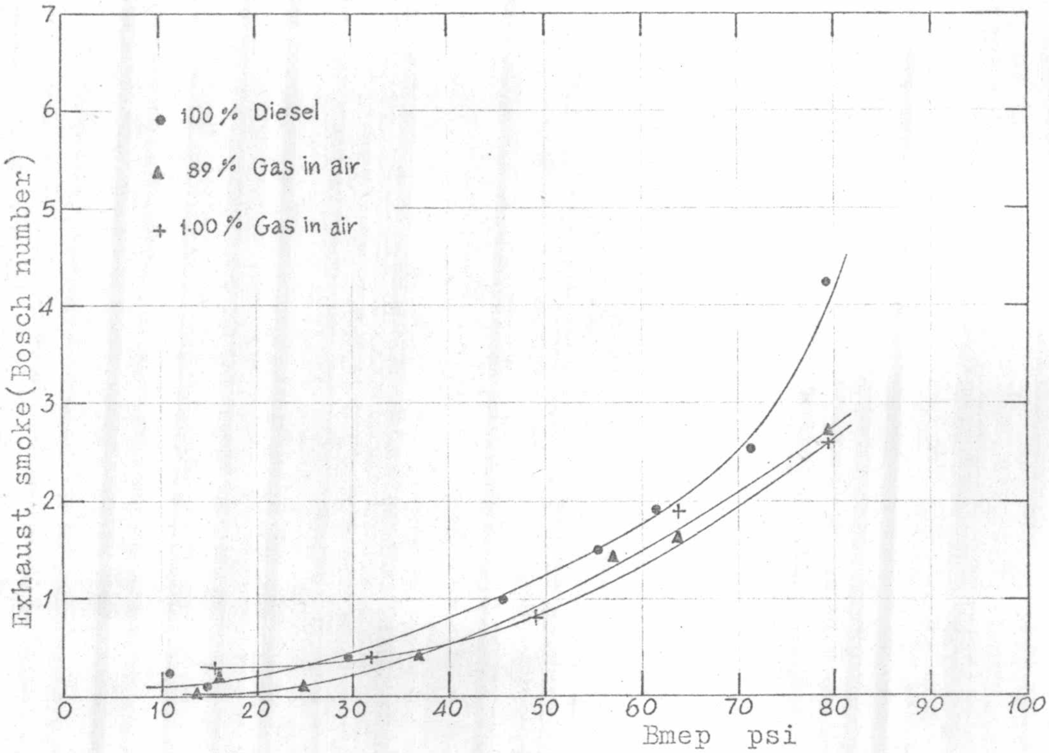

Fig. 16 Smoke emission for different gas admission in Kubota KND 3 engine at 1800 rpm.

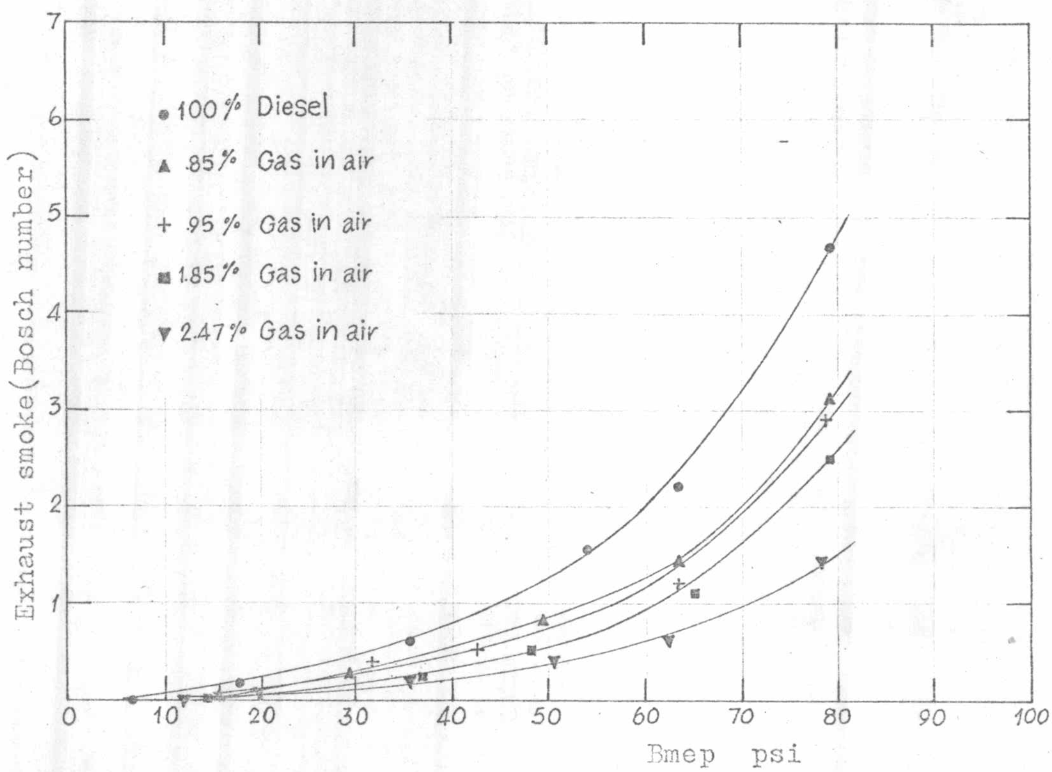

Fig. 17 Smoke emission for different gas admission in Kubota KND 3 engine at 2000 rpm.

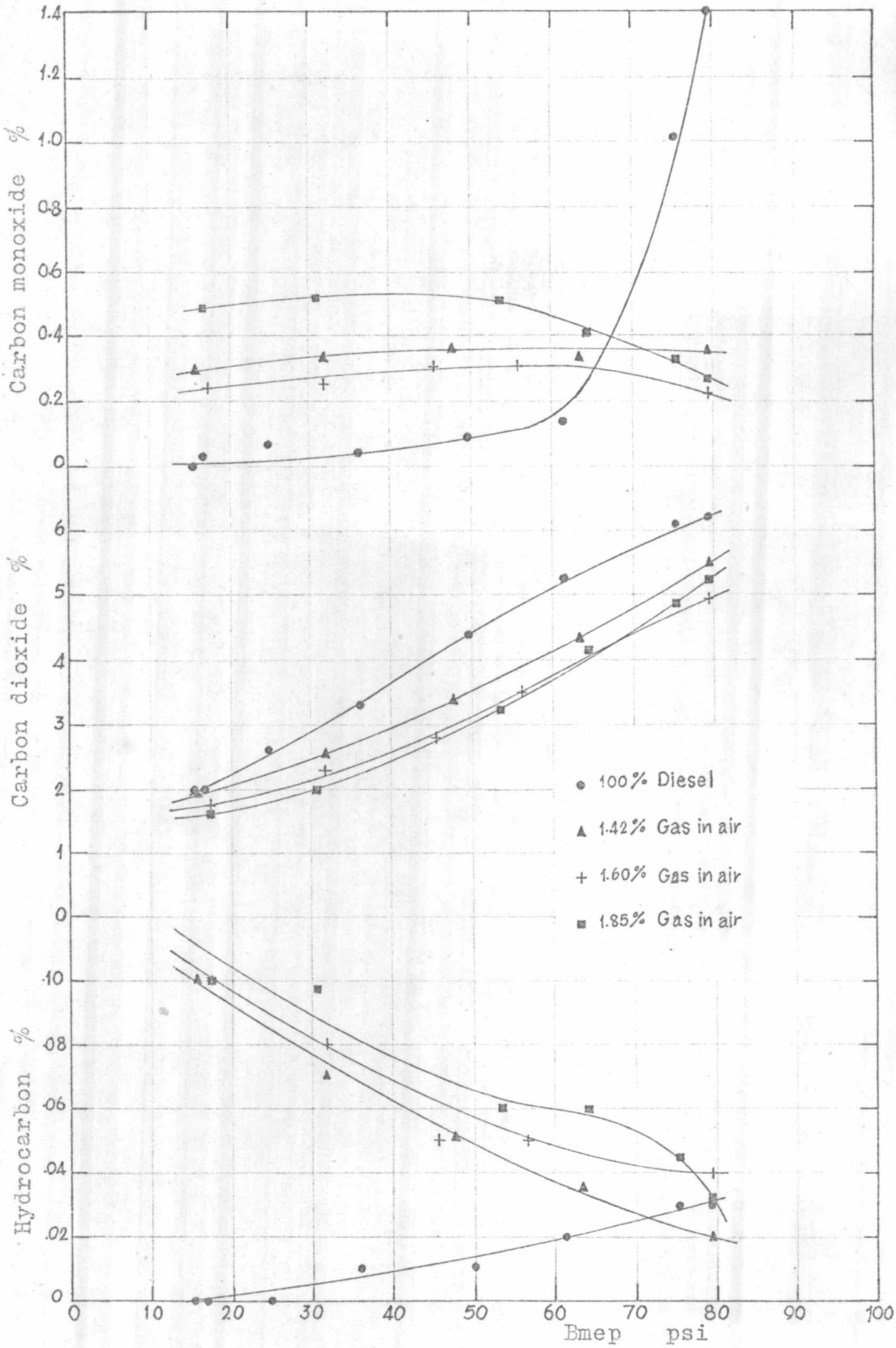

Fig. 18 Kubota KND 3 engine exhaust emissions for different gas admission at 1200 rpm.

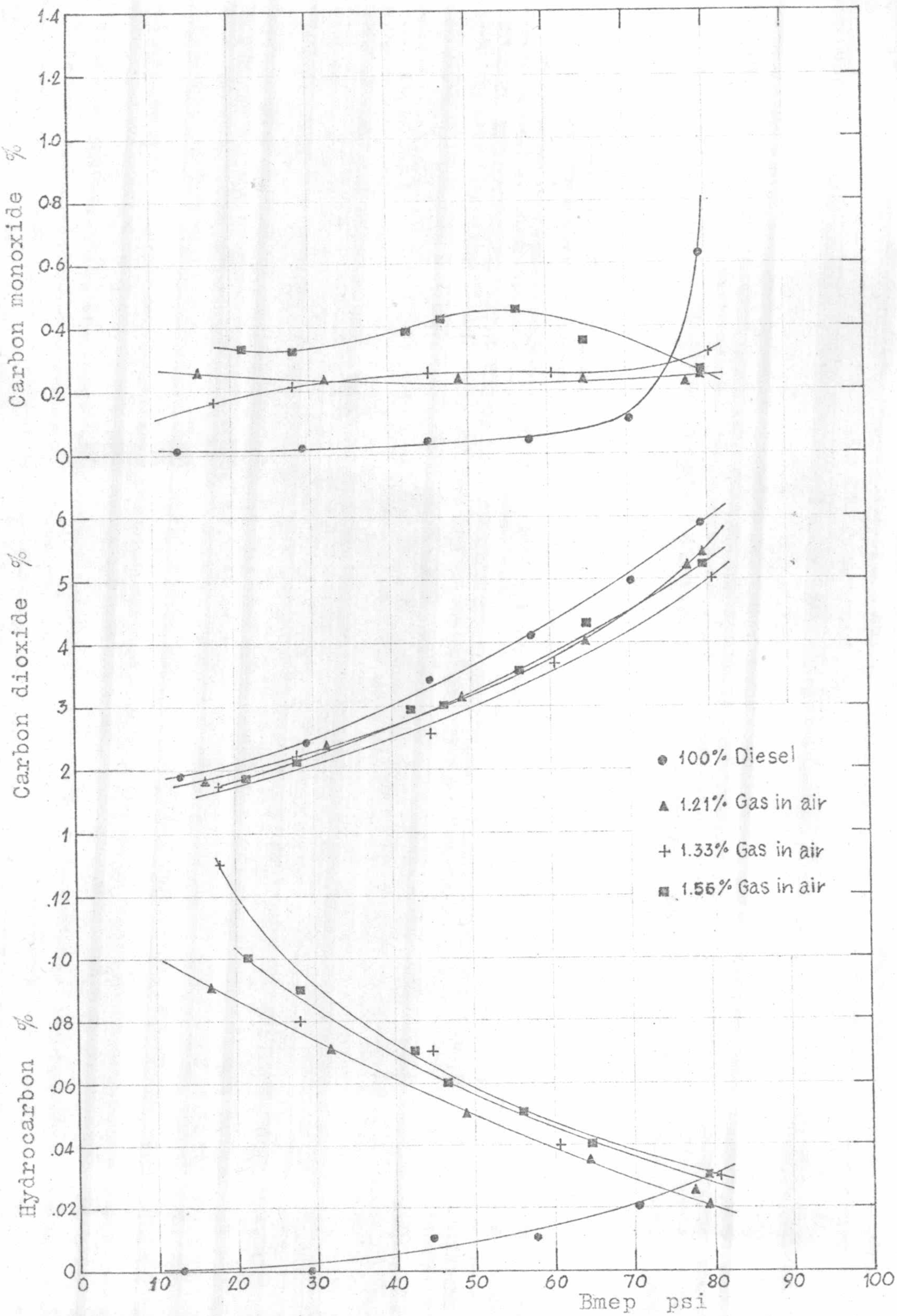

Fig. 19 Kubota KND 3 engine exhaust emissions for different gas admission at 1400 rpm.

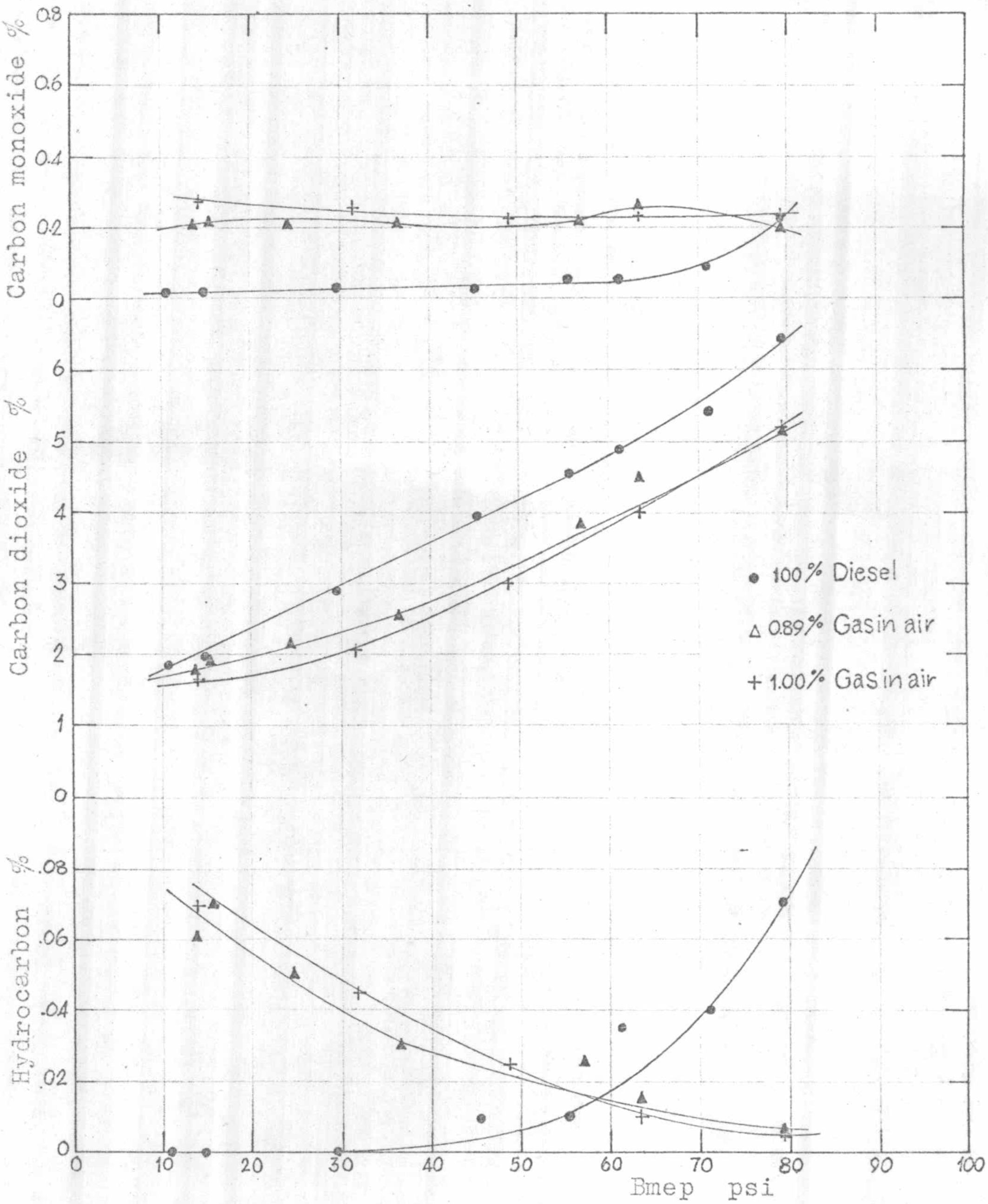

Fig. 20 Kubota KND 3 engine exhaust emissions for different gas admission at 1800 rpm.

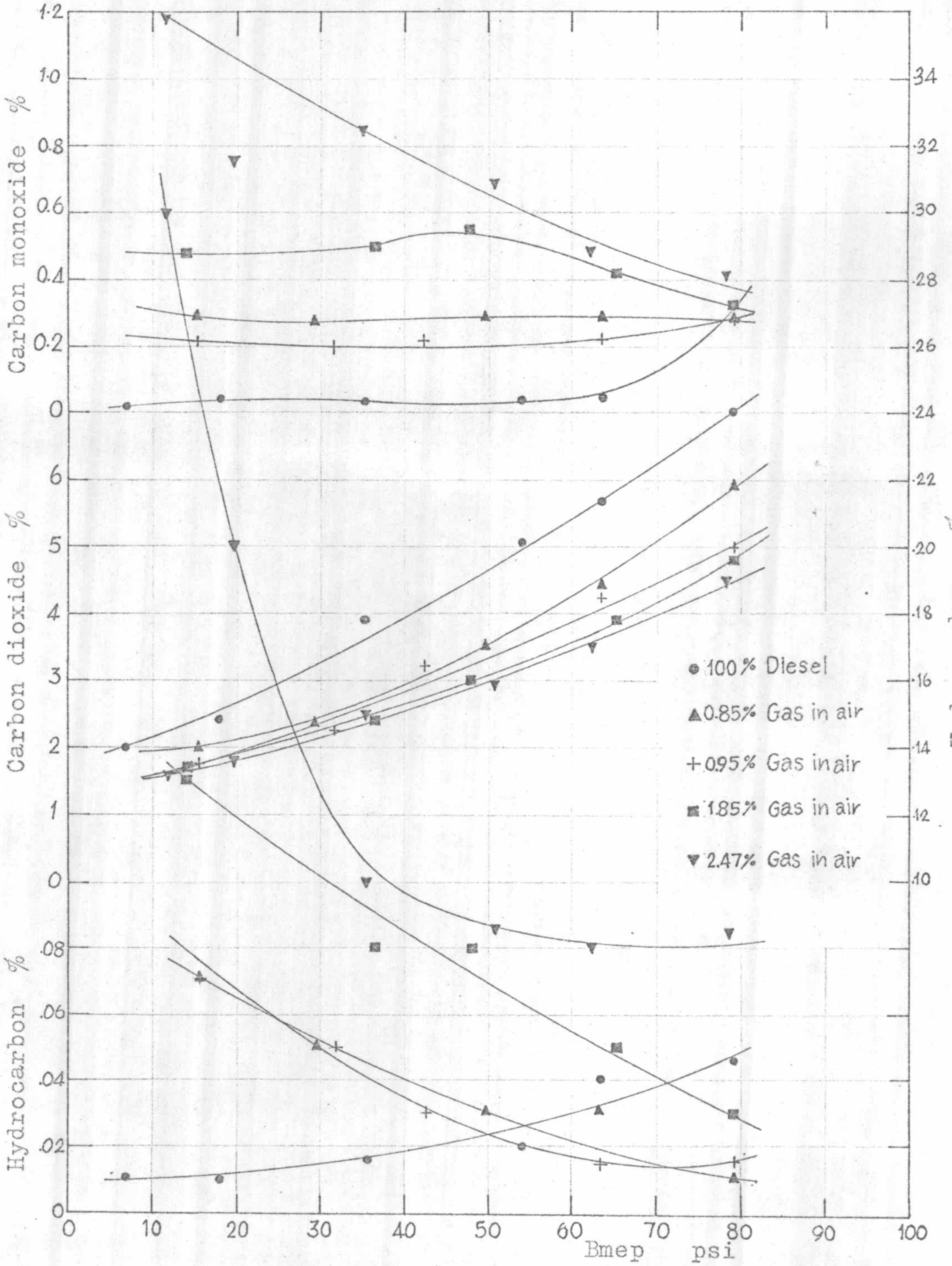

Fig. 21 Kubota KND 3 engine exhaust emissions for different gas admission at 2000 rpm.

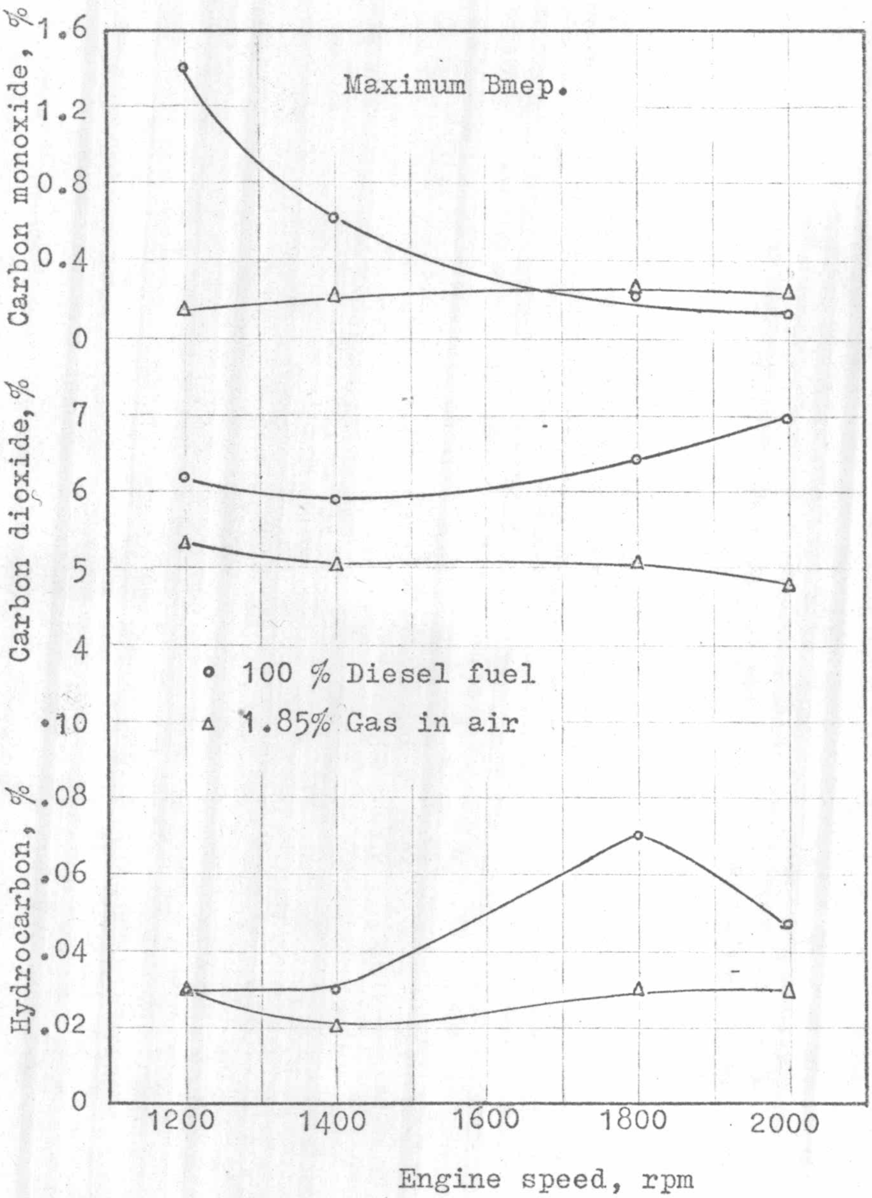

Fig.22 Effect of speed on engine exhaust emissions at maximum brake mean effective pressure in Kubota KND-3 engine

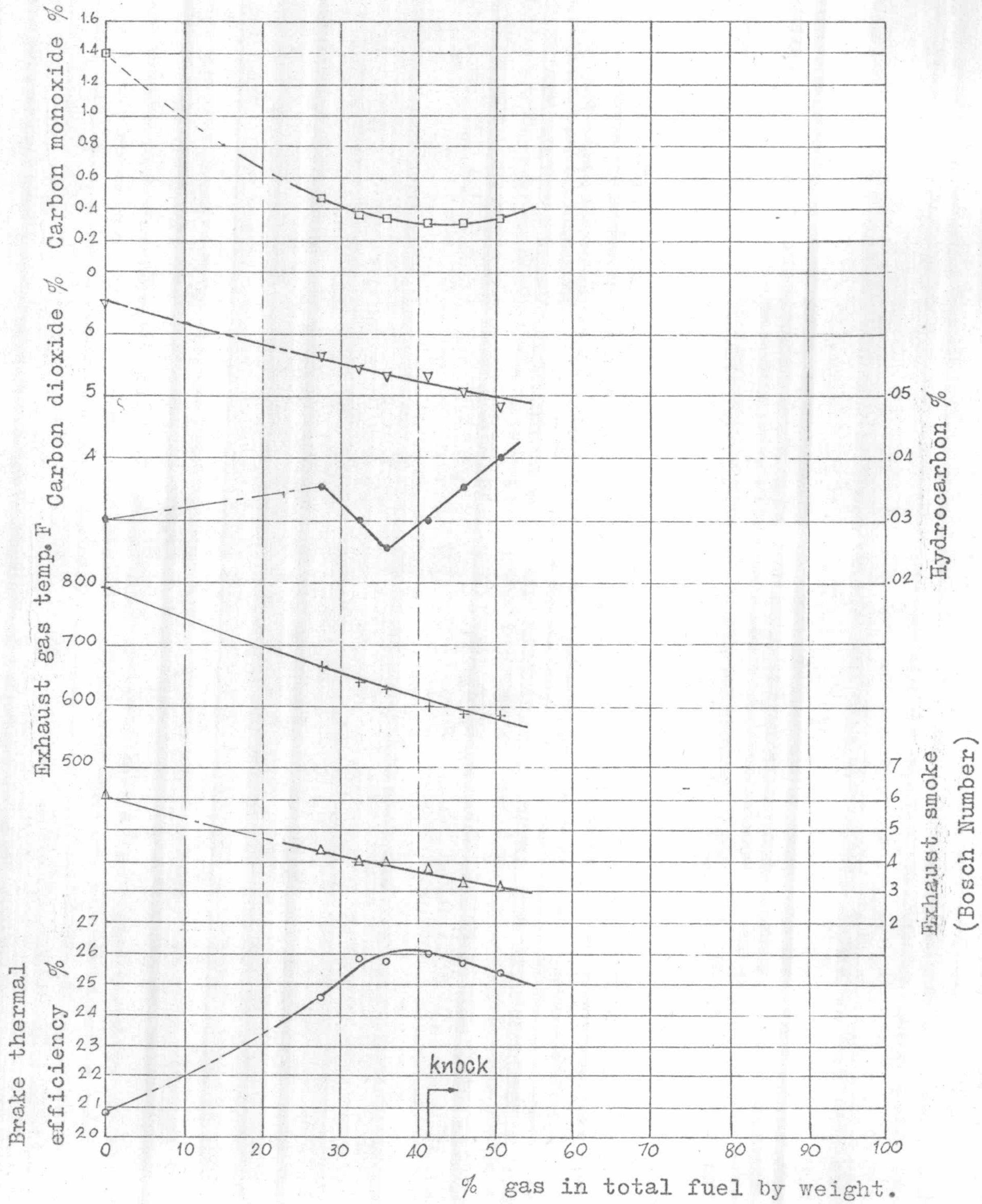

Fig.23 The influence of gas admission on brake thermal efficiency and exhaust emissions at maximum bmep. in Kubata KND 3 engine at 1200 rpm.

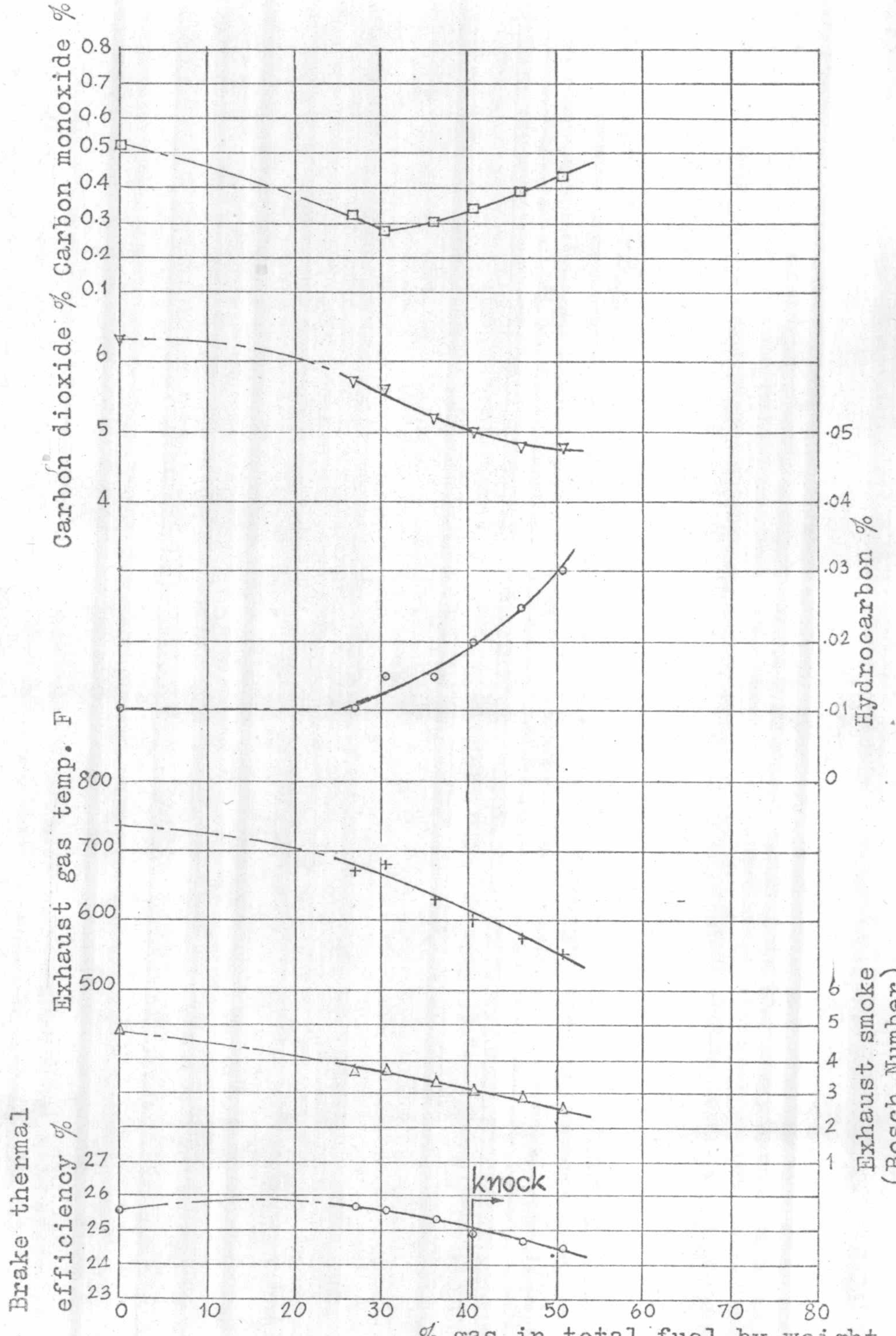

Fig.24 The influence of gas admission on brake thermal efficiency and exhaust emissions at maximum bmep. in Kubota KND 3 engine at 1400 rpm.

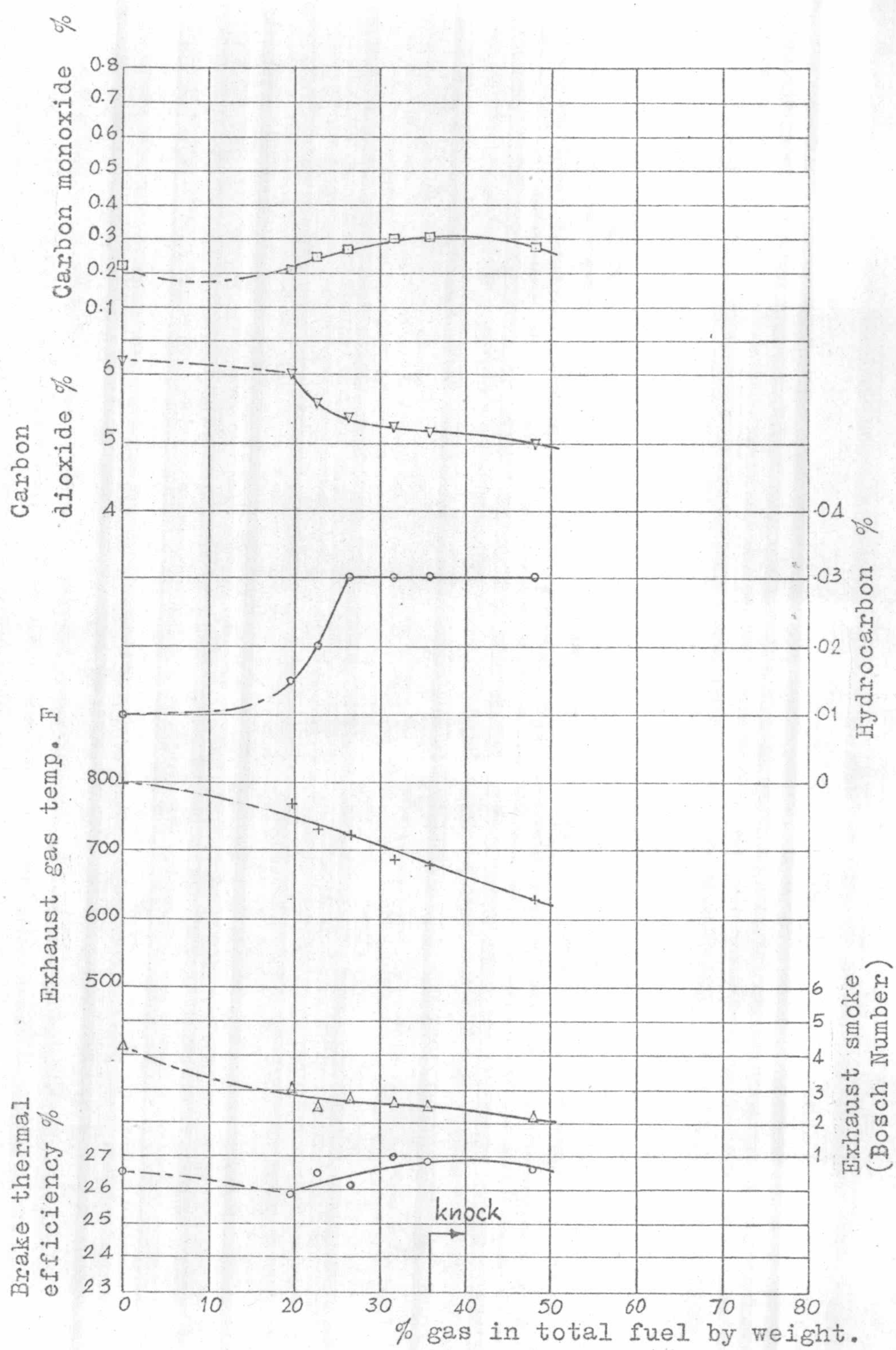

Fig.25 The influence of gas admission on brake thermal efficiency and exhaust emissions at maximum bmep. in Kubota KND 3 engine at 1800 rpm.

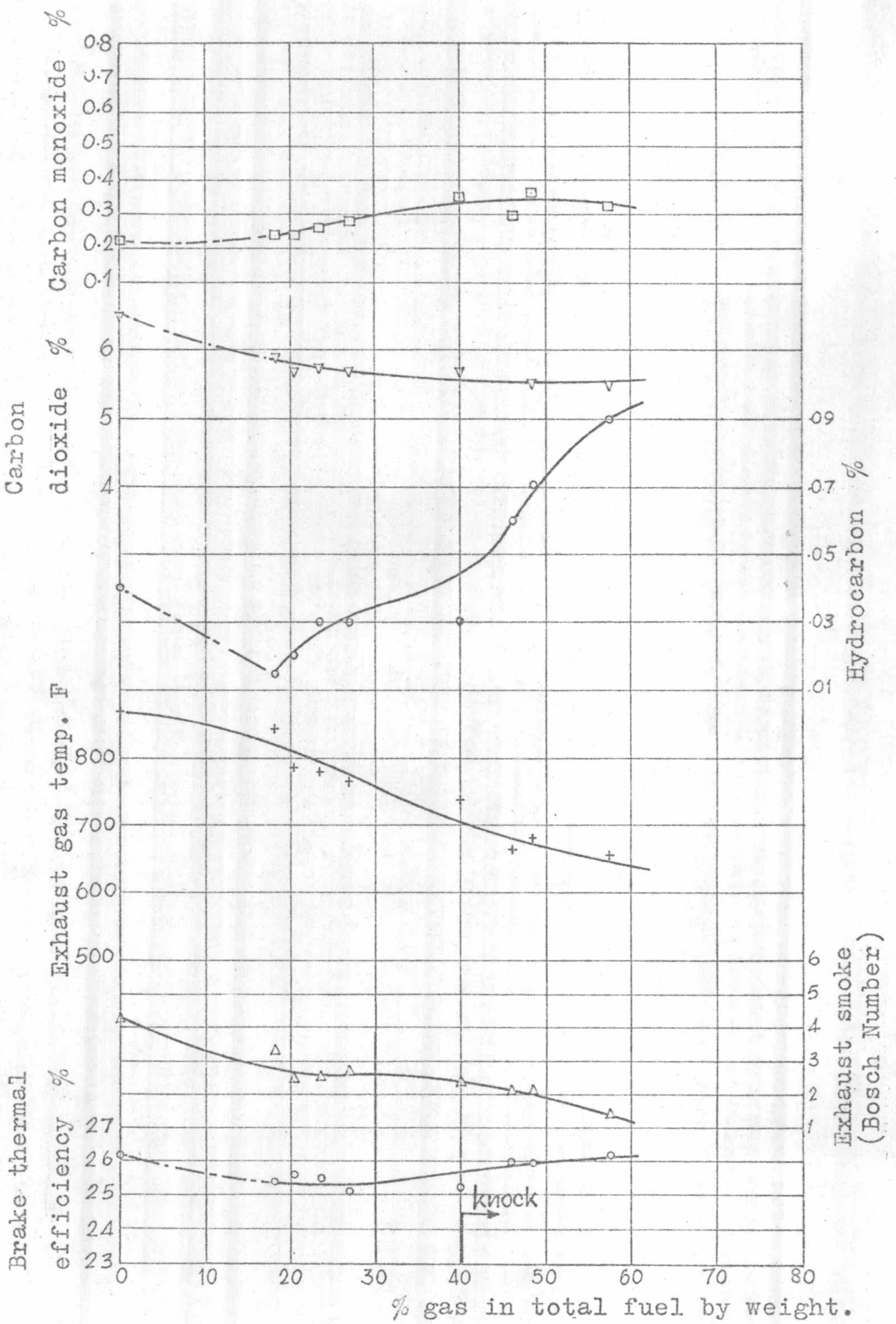

Fig.26 The influence of gas admission on brake thermal efficiency and exhaust emissions at maximum bmep. in Kubota KND 3 engine at 2000 rpm.

3.3 Discussion of section II

The experiment was conducted on the Petter AV-2 engine. The operating procedure can be briefed as follows:

The engine was tested only at rated speed of 1500 rpm, the cooling water was controlled to keep the outlet water temperature at 160 F. The engine was first run with pure diesel fuel and the brake load varied from minimum to maximum at 36 lb_f. Then, like in section I, gas was admitted to the intake air with varying concentrations. Again the engine was kept constant at full load and the gaseous fuel varied from minimum (0.0059 lb_m per minute) to maximum (0.0451 lb_m/min.) where the engine produced an unsteady noise and the speed became irregular.

3.3.1 Brake thermal efficiency

The brake thermal efficiency variation with Bmep was compared for straight diesel and diesel with gaseous fuel combination are given in Fig.27. With gas addition, the engine thermal efficiency starts to fall off, being more severe at light load as well as with increased gas rate, This agreed well with that obtained in Kubota engine. At high load, the thermal efficiency is still lower than the straight diesel but the difference is not as severe as in the case of light load. This is confirmed by Fig.29 which exhibits the effect of gas concentration on engine brake thermal efficiency at maximum load. This reduction in efficiency agrees well with the exhaust emissions shown in the same figure and will be discussed later. By comparing the present results with those obtained in Petter AV-1 engine, where the engine thermal efficiency was reportedly improved with increasing gas concentration reaching maximum value before declining, as shown in Fig. 30. The difference in results may be due to the different gaseous fuel being used. In this experiment commercial butane which is the mixture of mainly butane, propane and small amount of methane, whereas in the other work pure butane, propane and methane were used with methane showing lower thermal efficiency than straight diesel.

Moreover, it is of interest to note that for the Petter AV-2 engine in the present study, the maximum gas that can be added to the engine without rough running is about 50 % in total fuel by weight.

3.3.2 Exhaust smoke

The engine exhaust smoke are shown in Fig,28 which are plotted against the Bmep. It seems to be of similar trends to those obtained in the Kubota engine. At light load it is nearly the same as normal diesel, only at high load where reduction in smoke density is pronounced. As see in Fig.28, for pure diesel operation at maximum Bmep, the exhaust smoke number is 5.5 which is much over the acceptable level of 4.0, already mentioned in the preceding chapter. After running with 1.00 and 2.10 % gas in air, smoke reduces to 4.75 and 3.95, the latter being able to pass the limit. This point out that engine exhaust smoke reduction depend on the concentration of gas confirm in Fig.29. It should be note that, for normal diesel reduction in smoke indicates better combustion which is resulted in higher engine efficiency but Fig.29 does not show any rise in thermal efficiency despite the reduction in smoke. This point has already been discussed in subsection 3.2.2.

3.3.3 Exhaust gas emissions

The exhaust gas emissions from this engine are similar to that of the Kubota engine. For light load both hydrocarbon and carbon monoxide are considerably increased, as see in Fig.28. As the load increases hydrocarbon and carbon monoxide show a better combustion efficiency, identically to Kubota engine, with hydrocarbon reducing as already explained in subsection 3.2.3. This confirms that the combustion of gas-air mixture depend mainly on the amount of fuel oil. At high load carbon monoxide is lower than straight diesel whereas hydrocarbon is somewhat greater. Again the emissions show the tendency to increase as gas is added. Fig.29 shows the effect of variation gas concentration on engine efficiency and exhaust emissions at maximum load. Carbon monoxide decreases first and as gas is being added, starts to climb. This effect may be due to less fuel oil injected to ignite gas-air mixture efficiently as increasing gaseous fuel displace diesel fuel. As for hydrocarbon which increases as gas concentration is increased, showing good agreement with that obtained in Ford 2500 engine[8]

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[8] Clean Air, loc. cit.

Exhaust smoke density and temperature are similar in decreasing to about 20 % gas concentration and remain unchanged until the concentration of gas is over 40% where slight variation in smoke density is noticeable. By correlating the exhaust emissions and temperature mentioned above, it seems that the most effective concentration of gas for this engine is approximately 20 % in total fuel.

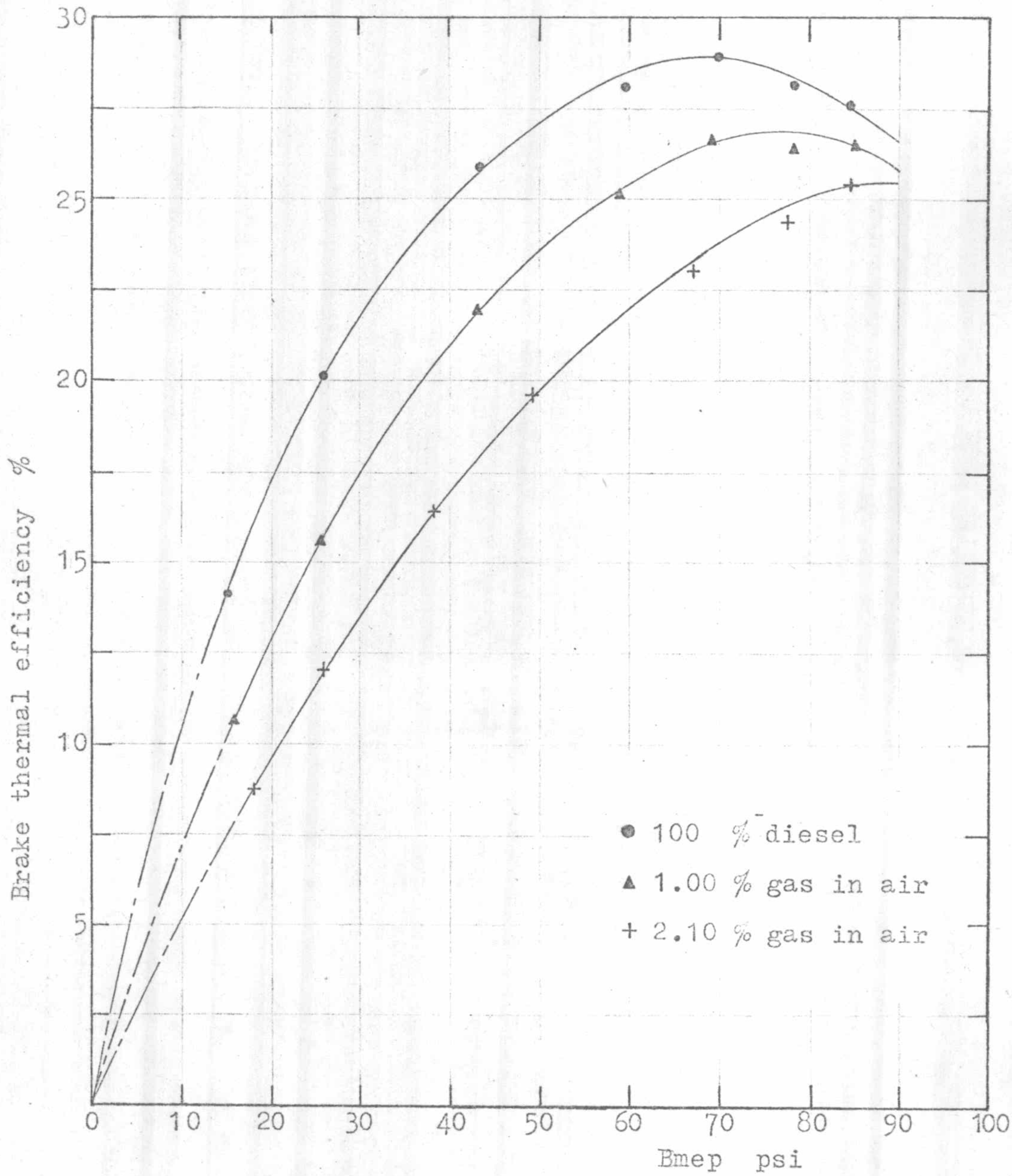

Fig.27 Variation of brake thermal efficiency with power output for different gas admission in Petter AV 2 engine at 1500 rpm.

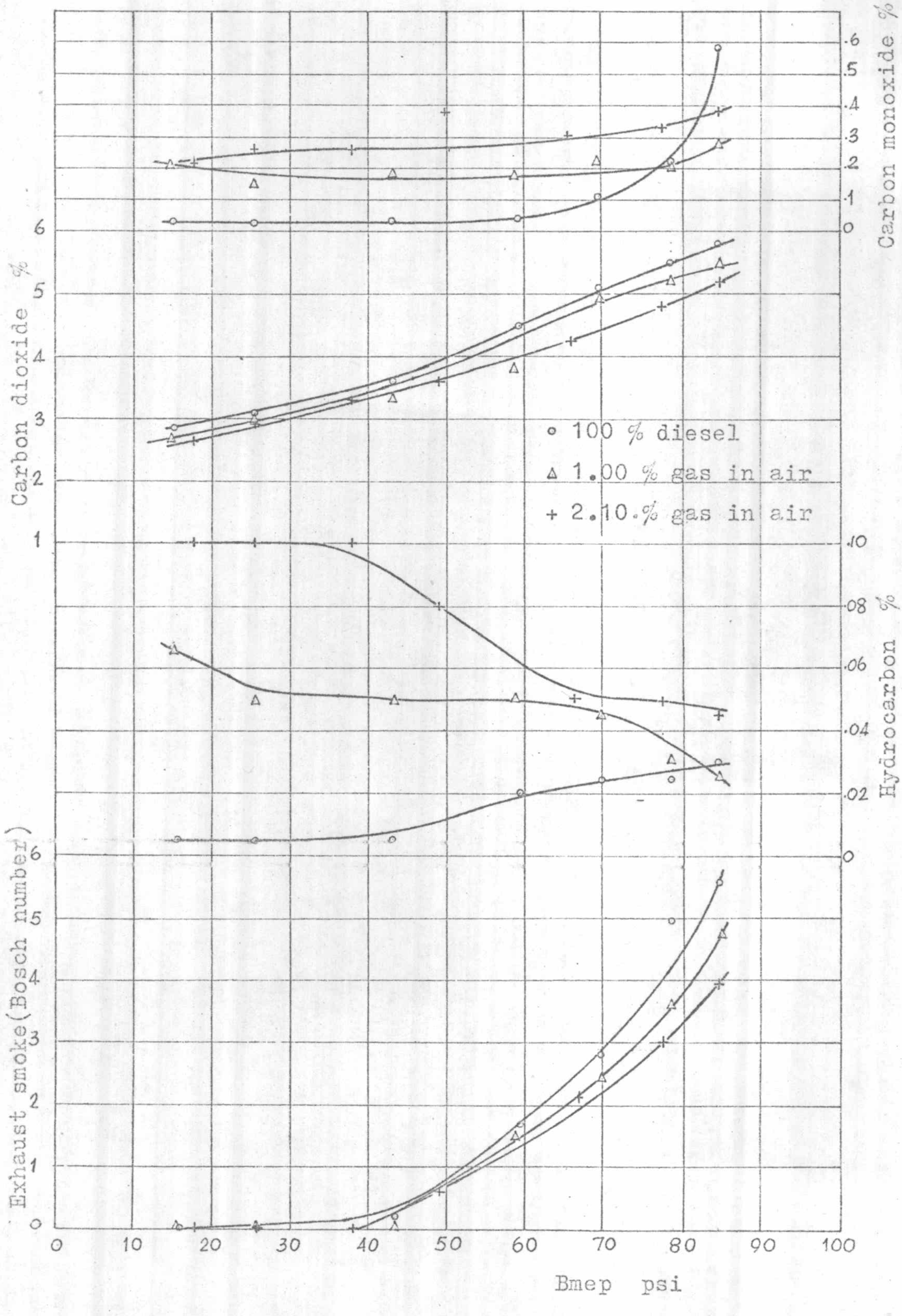

Fig.28 Exhaust emissions in Petter AV 2 engine for different gas admission at 1500 rpm.

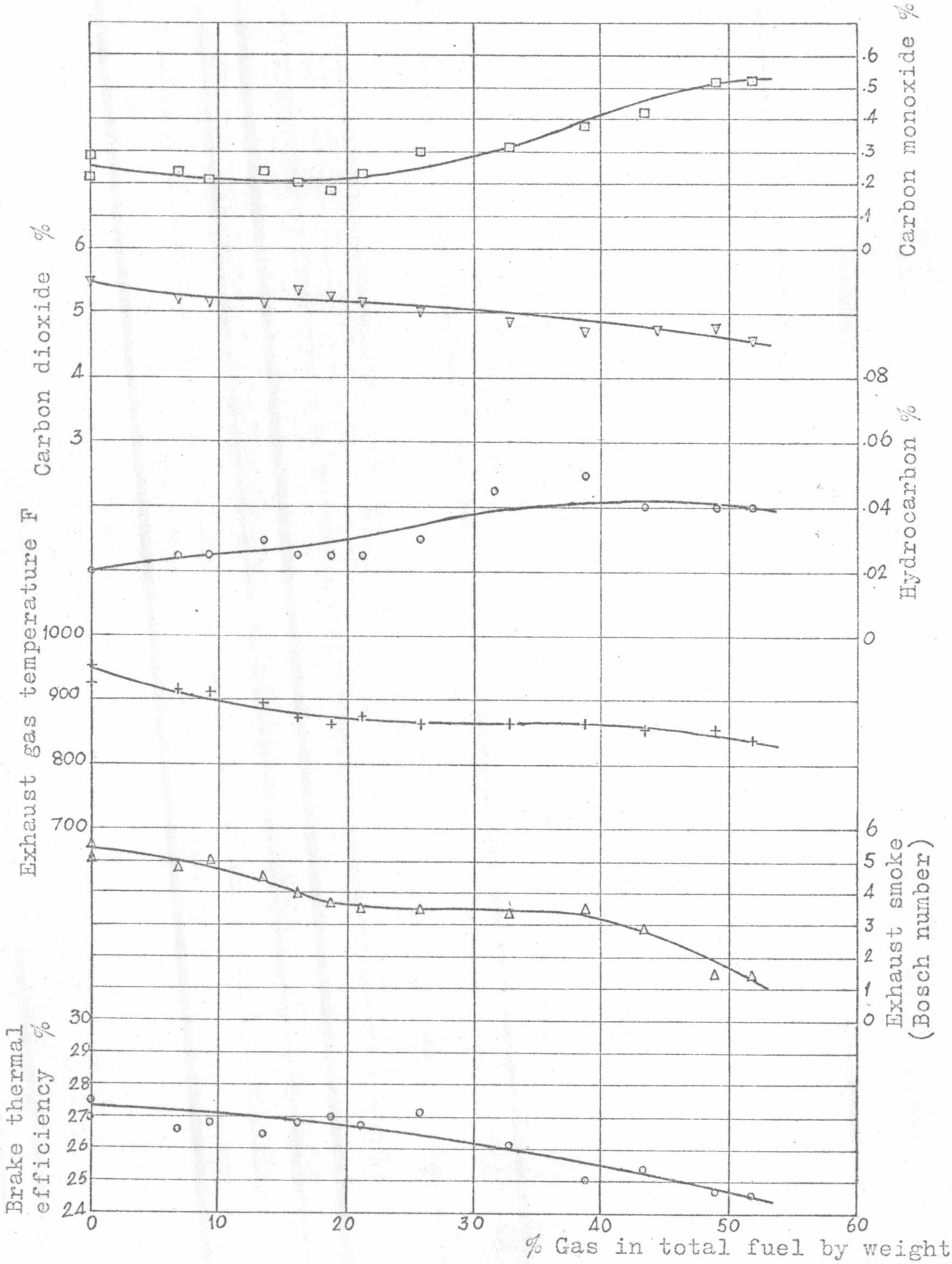

Fig.29 The influence of gas admission on brake thermal efficiency and exhaust emissions at maximum rated output in Petter AV 2 engine at 1500 rpm.

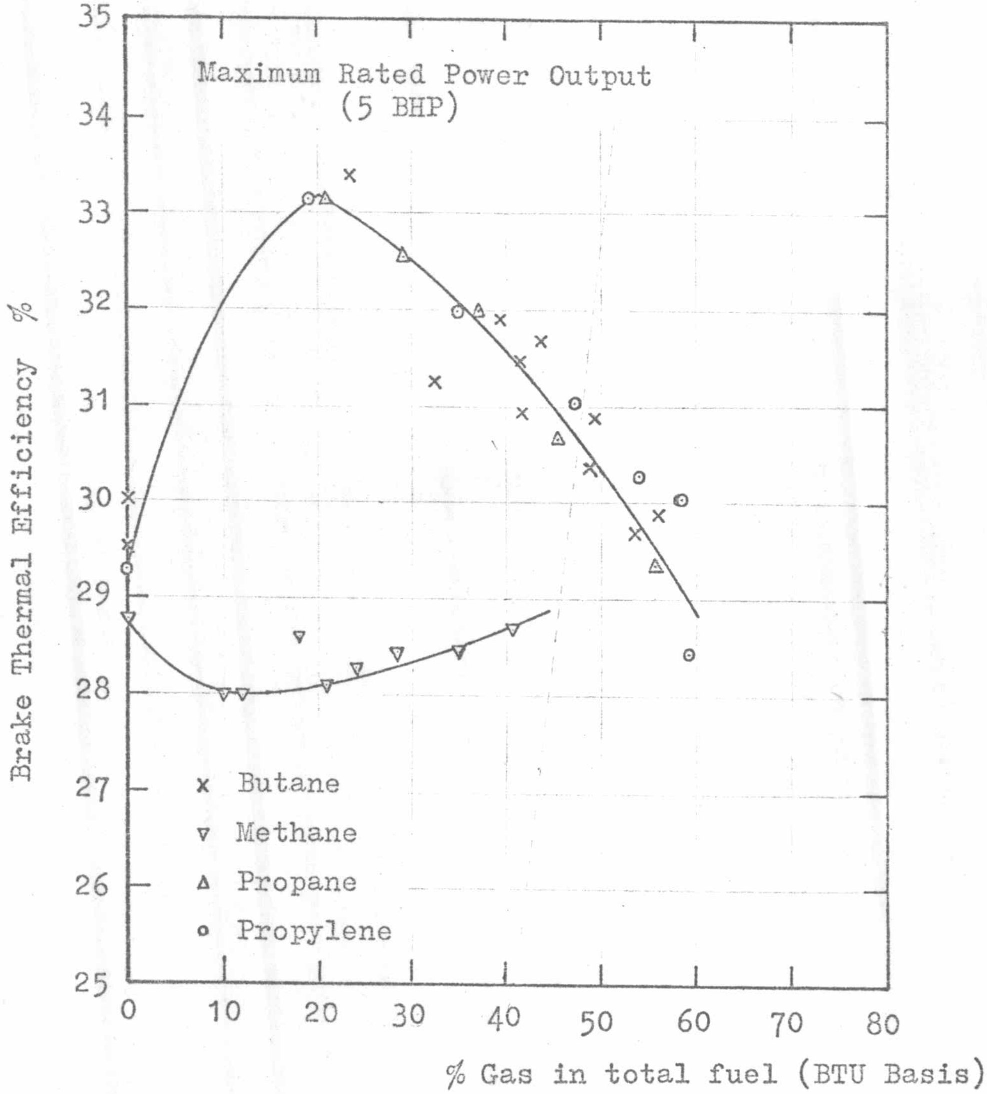

Fig.30 The influence of gas addition on thermal efficiency in Petter AV 1 (direct injection) engine at 1500 rpm.

Remark The curves in this figure have been converted from the original curves which plotted thermal input rate (Btu/hr) in place of Brake thermal efficiency. **From Air Pollution Control in Transport Engines. (London, I. Mech. E., 1972), p. 45.**