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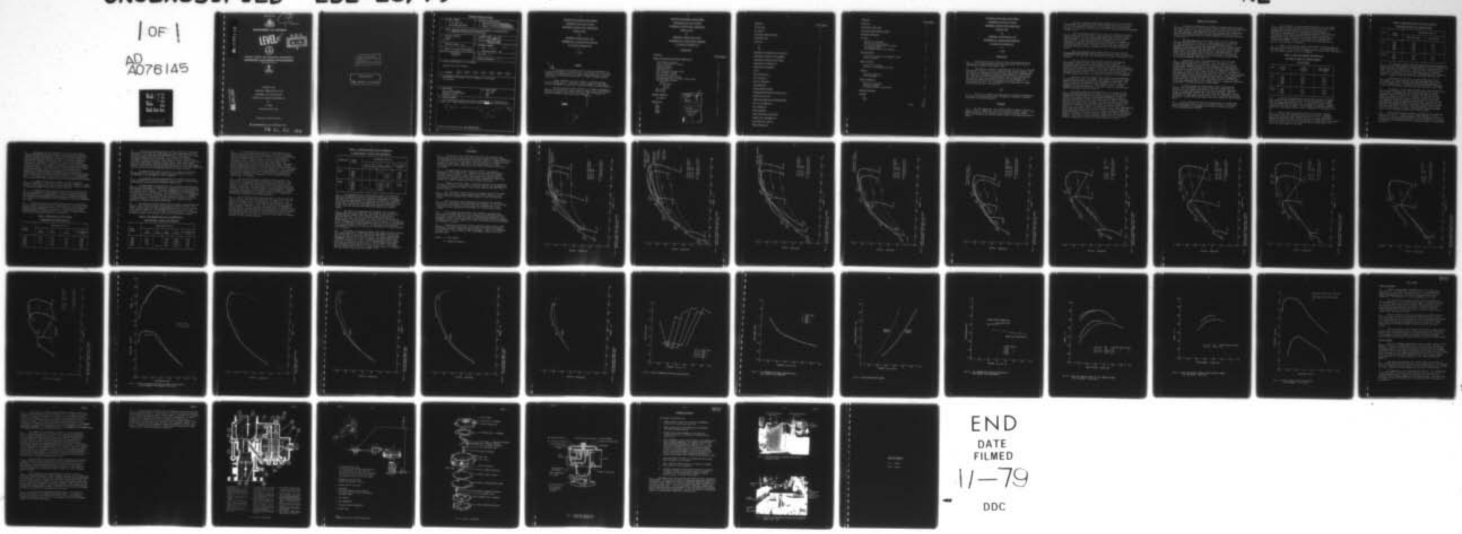

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TECHNICAL NOTE

LABORATORY INVESTIGATION OF THE PERFORMANCE OF A HOLDEN ENGINE OPERATING ON LIQUIFIED PETROLEUM GAS

BY

N. WEBB

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16. ABSTRACT/SUMMARY

This Technical Note details the relative performances of a Holden L6 six cylinder 2.85 ℓ engine operating on both liquified [redacted] gas (LPG) and petrol.

(See summary on page 2)

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TECHNICAL NOTE

ON

LABORATORY INVESTIGATION OF THE PERFORMANCE OF A HOLDEN ENGINE OPERATING ON LIQUIFIED PETROLEUM GAS

BY

N. WEBB

SUMMARY

A laboratory investigation into the relative performances of an engine when operated on both liquified petroleum gas (LPG) and petrol showed that the engine operated at higher thermal efficiency on LPG and also that it would operate satisfactorily at leaner air-fuel mixtures on this fuel.

Engine performance was less affected by retarded ignition for LPG than for petrol. Furthermore a large increase in dwell angle from the recommended setting had no significant effect on LPG performance.

The LPG carburettor when installed in its normal configuration maintained an essentially constant mixture strength with no part throttle leaning of mixtures to give better efficiency nor corresponding full throttle enrichment to give best engine torque.

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LABORATORY INVESTIGATION OF THE PERFORMANCE OF A HOLDEN ENGINE OPERATING ON LIQUIFIED PETROLEUM GAS

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N. WEBB

INTRODUCTION

1. This technical note is one of a series being published during 1978 and 1979 by the Aeronautical Research Laboratories (ARL) and the Engineering Development Establishment (EDE).

2. These notes all relate to investigations into the use of liquified petroleum gas (LPG) as a fuel for automotive spark ignition engines. This note relates to laboratory testing of a Holden engine and the measurement and comparison of performance parameters when operating on petrol and on LPG. The engine was from a Holden Belmont utility truck which had been converted to a dual fuel (petrol/LPG) vehicle by the Department of Administrative Services and had completed 11 000 km of operation. An ARL technical note will cover the measurement and comparisons of exhaust emissions carried out at the same time.

AIM

3. The aim is to compare the performance of the engine for operation on LPG with its performance on petrol including the effects of incorrect adjustment of ignition parameters.

PROGRAMME

4. The test engine was a six cylinder Holden L6 type of 2.85 litre capacity and 9.4:1 compression ratio with no exhaust emission controls. The engine was fitted with an open positive crankcase ventilation system. Details of the petrol and LPG fuel systems fitted to the engine are given in Annex A.

5. The fuels used were premium grade commercial petrol of 98 Research Octane Number, 43.73 megajoules per kilogram (MJ/kg) lower calorific value and 14.50 stoichiometric air/fuel ratio, and commercial LPG containing 98 per cent (%) propane of 46.57 MJ/kg lower calorific value and 15.68 stoichiometric air/fuel ratio.

6. The test procedure adopted was to carry out an analysis of torque and efficiency for both fuels over a broad spectrum of equivalence ratios, speeds and throttle settings under minimum spark advance for best torque (MBT) conditions. This was followed by an examination of the mixture characteristics of both carburettors and an examination of the effects on performance of ignition timing and dwell angle variations at limited fuel, speed and throttle settings. The effects of these and other parameters on exhaust emissions were also studied and will be the subject of an ARL report.

7. The term equivalence ratio herein refers to the ratio of the stoichiometric mass air/fuel ratio to the actual mass air/fuel ratio and is consequently less than unity for lean mixtures and greater than unity for rich mixtures.

8. With the exception of full throttle, throttle settings were established in terms of engine intake airflow and each series of measurements were carried out under constant airflow conditions. The selected airflows were 85, 60 and 30% relative to the full throttle airflow measured at the prevailing ambient dry air density during a set of preliminary runs carried out at selected speeds of 1500, 2000, 2500, 3000 and 3500 revolution per minute (r/min). At 3500 r/min a relative airflow of 45% was employed instead of 30% due to the low load limitations of the dynamometer.

9. During all runs the standard vehicle exhaust system discharging to atmosphere was fitted to the engine. The exhaust heat crossover valve remained fitted and operated during all measurements on both fuels. The engine cooling fan also remained fitted throughout the runs as did the alternator which was connected to a fully charged battery. Coolant temperature was maintained between 78 and 86^{o}C. Details of the principal parameters measured are given in Annex B.

10. The intake system was non-standard due to the presence of the airflow measuring apparatus and the absence of the air cleaner which was removed to facilitate connection of the airflow apparatus. Because of instability in intake airflow at some speeds when operating on petrol with the LPG carburettor by-pass open (refer Annex A), most of the petrol runs were carried out with the by-pass closed and the intake air directed through the LPG carburettor. The added restriction and/or change in the intake system geometry produced by this means eradicated the airflow instability. As a result however, the conditions were no longer representative of an actual operational system and full throttle airflows and torques for petrol operation were reduced by 1 or 2%. Ambient air density differences between one series of runs and another also affect full throttle airflows and torques. The result is that full throttle torques cannot be satisfactorily compared. Part throttle torque comparisons however are still valid and efficiency comparisons at both full and part throttle are valid.

RESULTS AND DISCUSSION

11. The effects on MBT torque and effiency of varying equivalence ratio are shown in Fig 1 to 10. These show full and part throttle curves of efficiency against MBT torque for varying equivalence ratio for both fuels at each of the speeds employed. Equivalence ratio contours constructed by interpolation from the known equivalence ratios for each torque/effiency data point and torque versus efficiency curves for fixed carburettor settings are also shown in Fig 1 to 10.

12. The torque/efficiency curves mostly cover all of the throttle and speed settings cited in para 8. In some figures certain curves have not been plotted because all or most of the corresponding results obtained were faulty; in these cases measurements were not repeated because adequate overall information had been obtained. In the case of the petrol measurements faulty results were due mostly to spark plug problems. Other faulty results, both petrol and LPG, were due to instrument errors. The spark plug problems appeared due to excessive build-up of lead deposits on the centre electrode insulation at very rich mixtures, causing leakage of current and imperfect fuel ignition.

13. Studying first of all the torque/efficiency curves for high relative airflows (85% and full throttle), it is seen that at rich mixtures where combustion is air-limited, torque is relatively constant and efficiency increases markedly as the stoichiometric air/fuel mixture is approached. As the mixture moves into the lean region and combustion becomes fuel-limited,torque drops in line with the reduced fuel supply and efficiency, initially at least, remains relatively constant. The effect is more pronounced with LPG for which the curves also show a more abrupt transition from air-limited to fuel-limited combustion than petrol. This is indicative of better mixture distribution between cylinders and a more homogeneous charge within each cylinder for LPG. The flatter nature of the LPG curves in the lean region,combined with the sharper downturn as combustion efficiency deteriorates,is further indication of these effects.

14. At lower airflows the LPG and petrol curves correspond more closely. For rich mixtures the torque is relatively constant for the same reason as with higher airflows.whilst for lean mixtures efficiency as well as torque drops markedly due to the more significant effects of thermal and mechanical losses at the lower levels of energy input involved. Nevertheless, at 60% airflow efficiency falls off less rapidly for LPG than for petrol. At 30% airflow the forms of the curves appear the same for both fuels.

15. In all cases for both fuels, the equivalence ratios applicable to the extreme data points of each curve have been noted. This has been done for the purpose of supplementing the frame of reference provided by the equivalence ratio contours.

16. The data points at which misfiring occurred in the lean region in one or more cylinders are indicated and the equivalence ratios noted. Although none of the data points was chosen during the tests as representing the actual onset of misfiring or a given frequency of misfiring, a comparison of the corresponding equivalence ratios for each fuel provides still further indication of the better lean burning capabilities of LPG. For LPG at full throttle and 85 and 60% airflow, misfiring has commenced around 0.7 equivalence ratio compared with around 0.8 for petrol. At 30 and 45% airflow the relevant figures are 0.8 to 0.9 for LPG and 0.9 for petrol.

17. Table 1 shows best full throttle efficiencies and efficiencies at best full throttle torque extracted from Fig 1 to 10. At the same speed LPG best efficiency is up to 1.8 per cent higher than for petrol whilst efficiency at best full throttle torque is up to 3.6% higher.

TABLE 1 - BEST FULL THROTTLE EFFICIENCY AND

EFFICIENCY AT BEST FULL THROTTLE TORQUE -

LPG AND PETROL

FUEL	SPEED (r/min)	BEST EFFICIENCY (%)	EFFICIENCY AT BEST TORQUE (%)
LPG	1500	33.2	30.5
"	2000	33.4	30.9
"	2500	32.9	30.4
"	3000	31.7	29.0
"	3500	30.2	27.8
Petrol	2000	32.0	27.8
"	2500	31.1	26.8
"	3500	29.4	25.8

18. A study of the equivalence ratio contours in Fig 1 to 10 shows that, except in some instances at 30% airflow, LPG efficiency is higher under all engine operating conditions. In other words, for any torque at a given speed and equivalence ratio the engine will operate at higher efficiency on LPG than on petrol. The difference in efficiency, which is greater for lean mixtures than for rich, is generally of the order of 1 to 2%.

19. Table 2 shows the equivalence ratios at best full throttle efficiency and best full throttle torque for the two fuels. Although these values could have been obtained from Fig 1 to 10 by interpolation between the equivalence ratio contours, for better accuracy they were obtained from plots of full throttle efficiency and torque against equivalence ratio, such as are shown in Fig 11 for both fuels at 2000 r/min. The curves for other speeds are not shown.

TABLE 2 - EQUIVALENCE RATIO AT BEST FULL THROTTLE

EFFICIENCY AND BEST FULL THROTTLE TORQUE

FUEL	SPEED (r/min)	EQUIVALENCE RATIO AT	
		BEST EFFICIENCY	BEST TORQUE
LPG	1500	0.88	1.09
"	2000	0.85	1.10
"	2500	0.87	1.09
"	3000	0.90	1.10
"	3500	0.89	1.10
Petrol	2000	0.90	1.17
"	2500	0.92	1.18
"	3500	0.93	1.17

20. For best full throttle torque LPG requires mixtures approximately 10% rich at all speeds compared with approximately 17% rich for petrol. The equivalence ratios required for best efficiency show some variation with speed. For LPG this would be due, in part at least, to the flat nature of the curves in the vicinity of best efficiency and the consequent difficulty in determining the exact peak. Notwithstanding this, appears to require a slightly richer mixture for best efficiency than does LPG.

21. Referring to Fig 12 which is a part reproduction of Fig 2 (as example) it is seen that part curve AB drawn tangentially to the torque/efficiency curves represents the peak efficiency obtainable at any given engine torque. A carburettor delivering mixtures represented by this curve would provide for peak efficiency operation at all times.

22. The point 'B' on the curve represents the torque and efficiency that would be obtained at full throttle valve opening. The torque at this point is substantially less than best full throttle torque indicated by point 'D' and also less than that at best full throttle efficiency indicated by point 'C'. To obtain best efficiency or best torque after attaining full throttle position it would be necessary to subsequently enrich the mixture by fuel control. Fig 13 to 15 show peak efficiency curves for both fuels at 2000, 2500 and 3500 r/min and illustrate the potentially higher efficiency obtainable from the engine when operating on LPG. Point B is shown also on each of these curves. It will be seen that for LPG there is much greater potential for fuel control of the carburettor beyond point B than for petrol.

23. Various factors mitigate against attaining the ideal operating curves in Fig 13 to 15. Factors such as exhaust emissions, vehicle driveability, ignition system performance, mixture combustibility and the practicalities of carburettor design must all be considered in arriving at the optimized mixture requirements for a particular engine in a particular application.

24. Nevertheless, petrol carburettor design has traditionally aimed at approximating the form of these curves, inasmuch as provision is generally made for operation on relatively lean mixtures under part throttle conditions combined with some form of mixture enrichment at wide throttle openings. Mixture enrichment in the petrol carburettor fitted to the test engine is achieved, in a manner typical of many carburettors, by means of a 'power' jet subsidiary to the main metering jet (refer Annex A). The 'power' jet comes into operation when manifold vacuum falls below a level determined by a control spring on the 'power' jet operating piston. This level is set at the manifold vacuum produced at some predetermined wide opening of the throttle valve. Employing this technique mixture enrichment commences before full opening of the throttle valve is attained, not subsequent to it as postulated in para 22. The method does result, however, in control of the carburettor at all times by the simple expedient of controlling throttle valve position.

25. In common with many types available, the LPG carburettor employed (refer Annex A) contains no mixture enrichment capability. Later versions of the carburettor/converter system however, are claimed to provide for mixture enrichment at wide throttle valve openings.

26. In studying the actual mixture characteristics of both the LPG and petrol carburettors the equivalence ratios of the mixtures delivered by the carburettors were determined over a range of speeds and airflows. For convenience the measurements were made with the carburettors mounted on the engine in their normal dual fuel configuration, not on a carburettor flow bench.

27. Dealing firstly with the petrol carburettor this was tested in the 'as received' condition with its standard main jet of 1.52 mm orifice diameter fitted. Referring to Table 3 it is seen that at any speed the equivalence ratio decreases as the throttle is opened from 30 to 60% airflow followed by a marked increase as the throttle is opened further to 85% airflow, remaining high up to full throttle.

TABLE 3 - EQUIVALENCE RATIO FOR PETROL

CARBURETTOR WITH STANDARD MAIN JET

SPEED (r/min)	EQUIVALENCE RATIO AT			
	30% Airflow	60% Airflow	85% Airflow	Full Throttle Airflow
1500	1.375	1.095	1.445	1.425
2000	1.14	1.07	1.49	1.525
2500	1.14	1.085	1.55	1.56
3000	1.15	1.115	1.515	1.50
3500		1.115	1.56	1.505

28. The values of equivalence ratio in Table 3 have been interpolated onto Fig 6 to 10 to show the torque/efficiency curves obtainable with the carburettor. The curves have been segmented due to the uncertainty of their exact form between 60 and 85% airflow. It is evident that mixture enrichment takes place well before full throttle position. Moreover the enrichment is excessive and at all speeds results in full throttle equivalence ratios well beyond those required for best torque. Additionally engine efficiency would benefit from lower equivalence ratios at 30 and 60% airflow. Finally at 1500 r/min 30% airflow the equivalence ratio is quite high suggesting that the idle jet is still discharging fuel at this engine setting.

29. In appraising the curves in Fig 6 to 10 it should be noted that an improvement of 1% in efficiency above an existing efficiency of say 25% represents a 4% improvement in fuel consumption.

30. The carburettor mixture characteristics are further illustrated in Fig 16 wherein the values of equivalence ratios in Table 3 have also been plotted against absolute airflow in gram per second (g/s).

31. The measurements of equivalence ratio on the LPG carburettor were made both with and without the balance line connected between the carburettor and converter (refer Annex A). The latter configuration, in conjunction with a low restriction air cleaner, is the one recommended by the manufacturer and normally used in LPG conversions. Use of the balance line is recommended for cases where a desired air cleaner has a high restriction. It performs a similar function in isolating air cleaner restriction from any influence on mixture strength as does the venting of the float chamber into the carburettor air horn on the petrol carburettor.

32. The equivalence ratios obtained with the balance line connected are shown in Table 4 and are plotted in Fig 17 against absolute airflow. The values in Table 4 have also been interpolated into Fig 1 to 5 as in the case of the petrol carburettor. For the measurements the power mixture valve (refer Annex A) was set in its fully open, ie richest, position. No attempt was made to optimize the setting of the valve for efficiency or torque.

TABLE 4 - EQUIVALENCE RATIO FOR LPG CARBURETTOR AT

FIXED SETTINGS - BALANCE LINE CONNECTED

SPEED (r/min)	EQUIVALENCE RATIO AT				
	30% Airflow	45% Airflow	60% Airflow	85% Airflow	Full Throttle Airflow
1500	1.19		1.065	1.005	1.00
2000	1.115		1.02	0.985	0.97
2500	1.11		1.00	0.945	0.93
3000	1.055		0.97	0.925	0.91
3500		0.99	0.95	0.915	0.89

33. It is seen from Table 4 and Fig 17 that equivalence ratio decreases with increasing airflow from 1.19 to 0.89 and is dependent solely on absolute airflow, having no response to the throttle position. From the carburettor curves in Fig 1 to 5 it is seen that engine efficiency is mostly good particularly at higher speeds where the equivalence ratio is lower. Except at 1500 r/min however, the available full throttle torque is well below best torque. The margin becomes progressively greater with increasing speed being 10 Newton metre (Nm) below best torque at 2500 r/min and 19 N m below at 3500 r/min. The percentage reductions are 5.5 and 12% respectively. Since the power mixture valve was fully open during the measurements no improvement in full throttle torque was possible. Therefore, with the balance line connected, the carburettor is inadequate as far as maximum available engine torque is concerned.

34. Two sets of measurements were made with the balance line disconnected, one with a low inlet restriction, the other with a high restriction. As mentioned earlier no air cleaner was fitted to the engine; inlet restriction was provided by the airflow measuring nozzle selected in each case. The same setting of the power mixture valve was employed in both cases so that the effects could be gauged of a change in air cleaner restriction, due for example to dust accumulation, without compensatory readjustment of the power mixture valve. As before the setting was not optimized for engine performance.

35. The nozzle restriction curves applicable to the two cases are shown in Fig 18. The restriction applied by the large nozzle is of the order to be expected from a low restriction air cleaner whilst that applied by the small nozzle is what might be expected due to a moderate degree of dust accumulation. Table 5 and Fig 19 show the equivalence ratios obtained. For each case, the equivalence ratio is substantially constant but with a marked difference of approximately 0.1 between the two. The measurements were carried out at limited speed and throttle settings influenced by the practical range of airflows over which each nozzle could be employed. As with Fig 17, Fig 19 shows that there is no mixture enrichment at wide throttle openings.

TABLE 5 - EQUIVALENCE RATIO FOR LPG CARBURETTOR

AT FIXED SETTING - BALANCE LINE DISCONNECTED

RESTRICTION	SPEED (r/min)	EQUIVALENCE RATIO AT				
		30% Airflow	45% Airflow	60% Airflow	85% Airflow	Full Throttle Airflow
Lower	2000			1.04		1.035
	2500				1.015	1.015
	3000				1.00	1.025
	3500				1.00	1.00
Higher	1500			1.105	1.105	1.125
	2000			1.105	1.135	1.135
	2500			1.125	1.125	
	3000	1.11		1.125		
	2500		1.11	1.115		

36. The equivalence ratio values shown in Table 5 for 2000 r/min have been interpolated onto Fig 2 to show the efficiencies obtainable. With the lower inlet restriction, efficiency is approximately 1% higher at 60% airflow and approximately 2.5% higher at full throttle. Comparison of the curve for the lower inlet restriction with that obtained with the balance line connected shows that whereas part throttle efficiencies at 60% airflow are similar, the former gives greater full throttle torque.

37. The effects of varying ignition parameters were studied at limited speed and torque conditions. Fig 20 and 21 show the effect of retarded ignition timing on full throttle torque for both fuels at 1500 and 3500 r/min. The loss of torque with retarded ignition is mostly less for LPG than for petrol. At 1500 r/min for 10^{o} retardation at approximately 1.05 equivalence ratio the loss in torque is 7 N m for both fuels. At approximately 0.83 equivalence ratio the loss for 10^{o} retardation, is 5 N m for LPG and 8 N m for petrol. At 3500 r/min at approximately stoichiometric conditions the loss for 10^{o} retardation is 4 N m for LPG and 9 N m for petrol.

38. The effects of varying distributor dwell angle are shown in Fig 22. This figure shows plots of efficiency against equivalence ratio for LPG only at 2000 r/min for both full throttle and 30% airflow. Measurements were made at both 28 and 44^{o} cam dwell angle. The engine manufacturer's recommended setting is 25 to 28^{o}. A significantly smaller dwell angle could not be reliably set since at 28^{o} the corresponding contact breaker points gap of 0.64 mm was already close to the distributor cam height. The corresponding points gap for 44^{o} dwell was 0.13 mm. It can be seen from the figures that at both full and part throttle there is no effect on efficiency from increasing the dwell angle to 44^{o}.

CONCLUSIONS

39. The results of the investigation show that LPG has better lean burning characteristics than petrol thus providing greater inherent potential for lean mixture, and therefore higher efficiency operation. Higher efficiencies are obtainable with LPG at any mixture setting, being up to 2% higher for lean mixtures at high torque. Additionally, best torque occurs at a lower equivalence ratio for LPG resulting in best torque efficiencies being up to 3.6% higher.

40. The Impco Model CA 300A Carburettor fitted to the engine in combination with the Impco Model J Converter does not provide for mixture enrichment at full throttle. This limits its ability to be adjusted for lean part throttle mixtures without significantly reducing full throttle torque output. However, later versions of Impco carburettor converter fuel systems are claimed to improve part throttle engine efficiency.

41. Further to the above there is definite scope for an LPG carburettor to combine both throttle valve control of mixture quantity and subsequent fuel valve control of mixture strength in order to fully realise the fuel's lean burning potential.

42. With the balance line fitted and with maximum setting of the power mixture valve, the mixture becomes progressively leaner right up to full throttle such that available full throttle torque is up to 12% below best torque.

43. With the balance line between the LPG carburettor and converter disconnected, significant mixture enrichment and consequent loss in efficiency could occur as the engine air cleaner restriction increases due to dust accumulation unless compensatory readjustment of the power mixture valve is made.

44. At the speeds and equivalence ratios examined, the engine when operating with retarded ignition at full throttle on LPG suffers no greater, and in most cases, less torque loss than for retarded petrol operation. It is reasonable to expect that for any speed, load and equivalence ratio the engine's performance on LPG in this regard will at least be similar to, if not better than, its performance on petrol.

45. At 2000 r/min at both full throttle and 30% airflow on LPG a large increase in distributor dwell angle such as might be caused by distributor cam follower wear has no effect on engine performance. Again it is reasonable to expect that at least no drastic loss in performance would occur at any speed and load.

Annexes: A - Fuel Systems

B - Parameters Measured

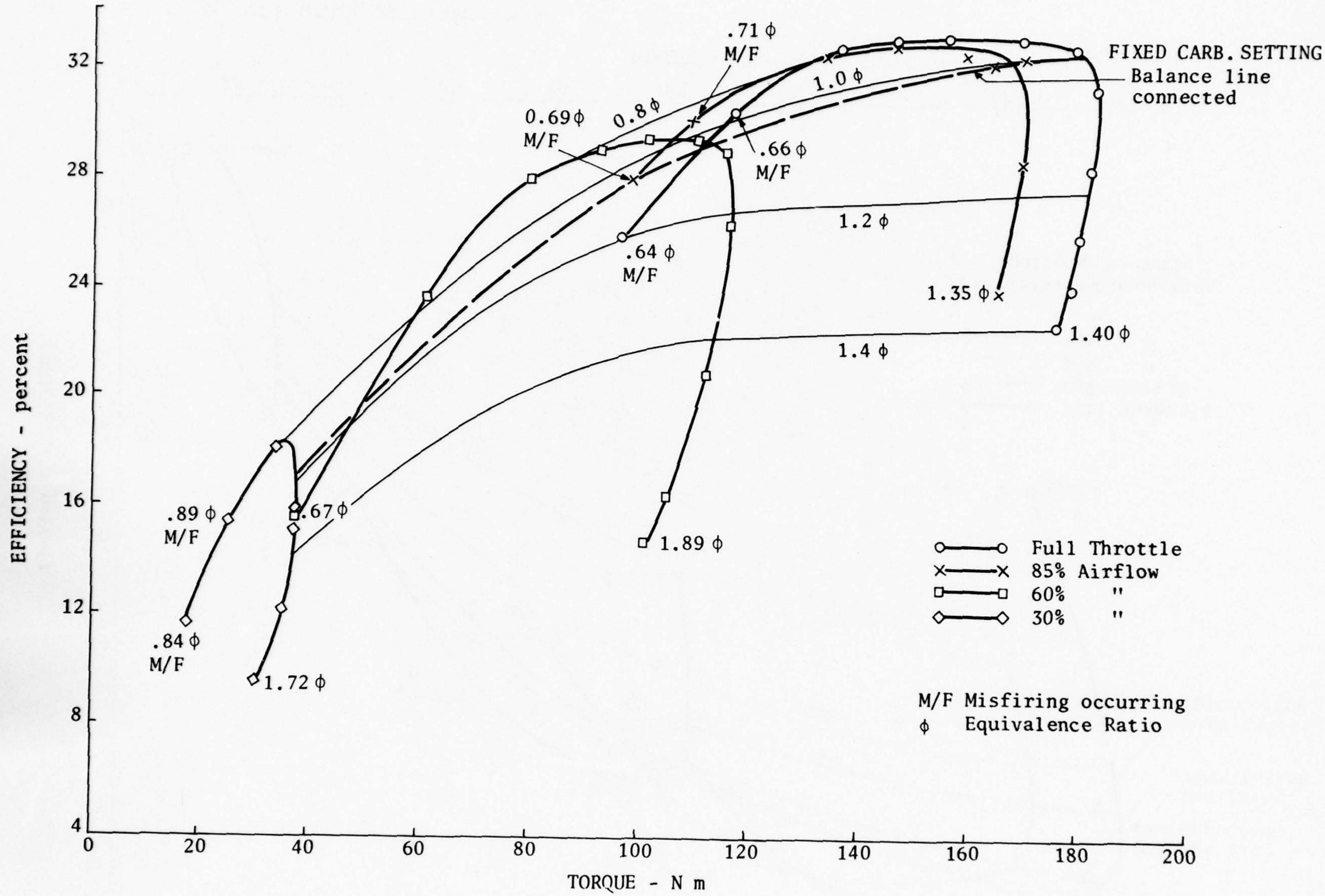


FIG 1 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - LPG 1500 r/min

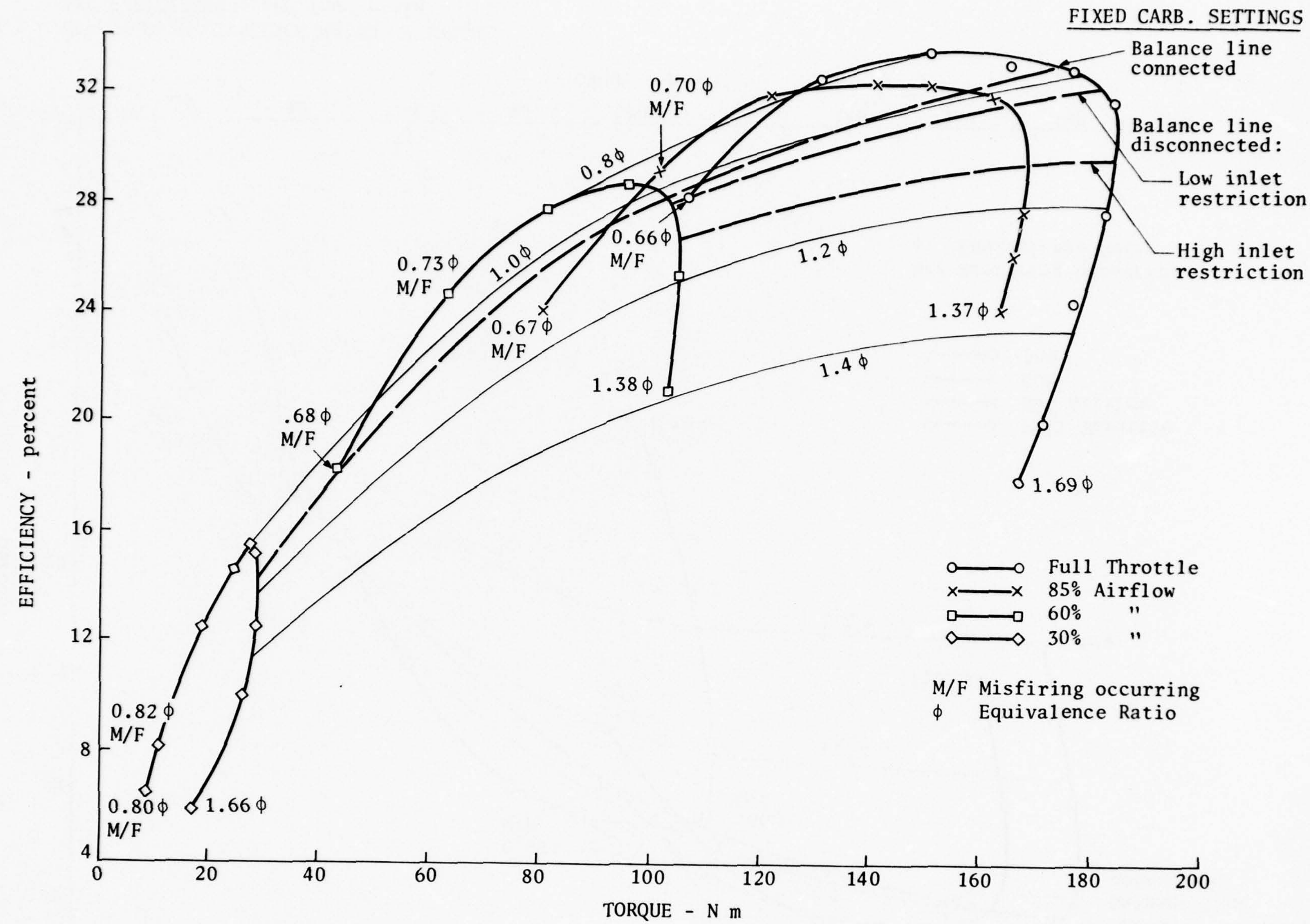


FIG 2 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - LPG 2000 r/min

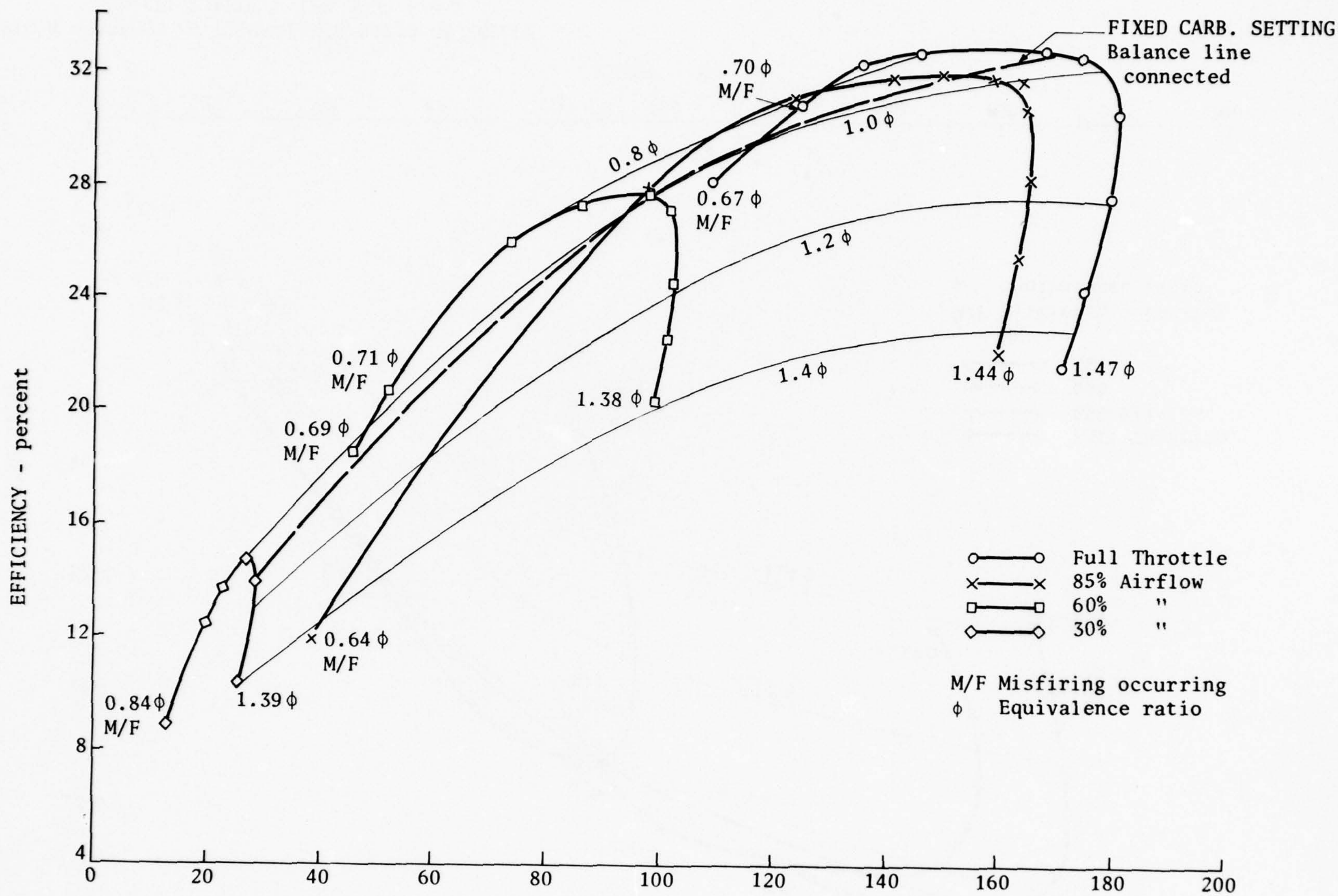


FIG 3 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - LPG 2500 r/min

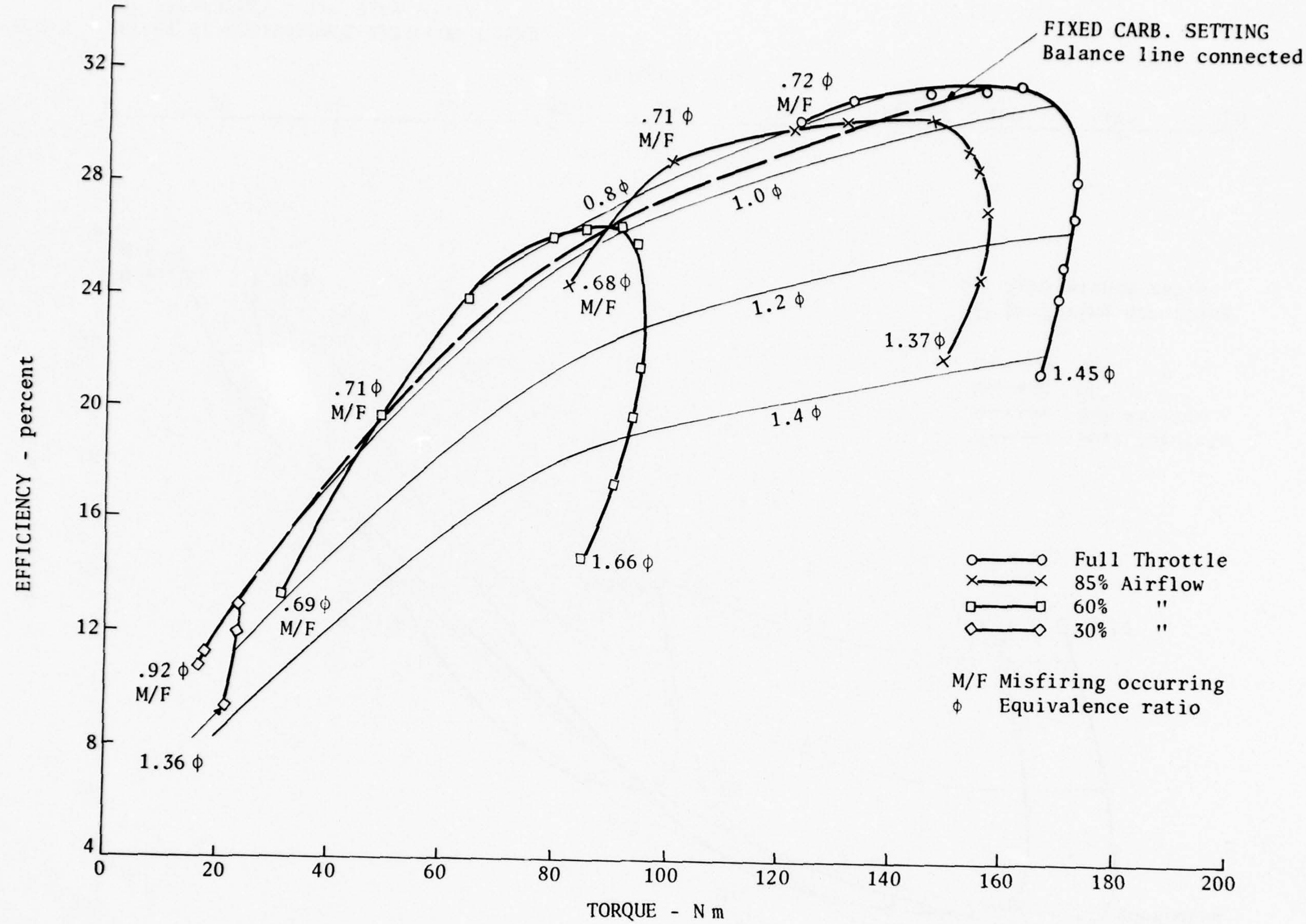


FIG 4 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - LPG 3000 r/min

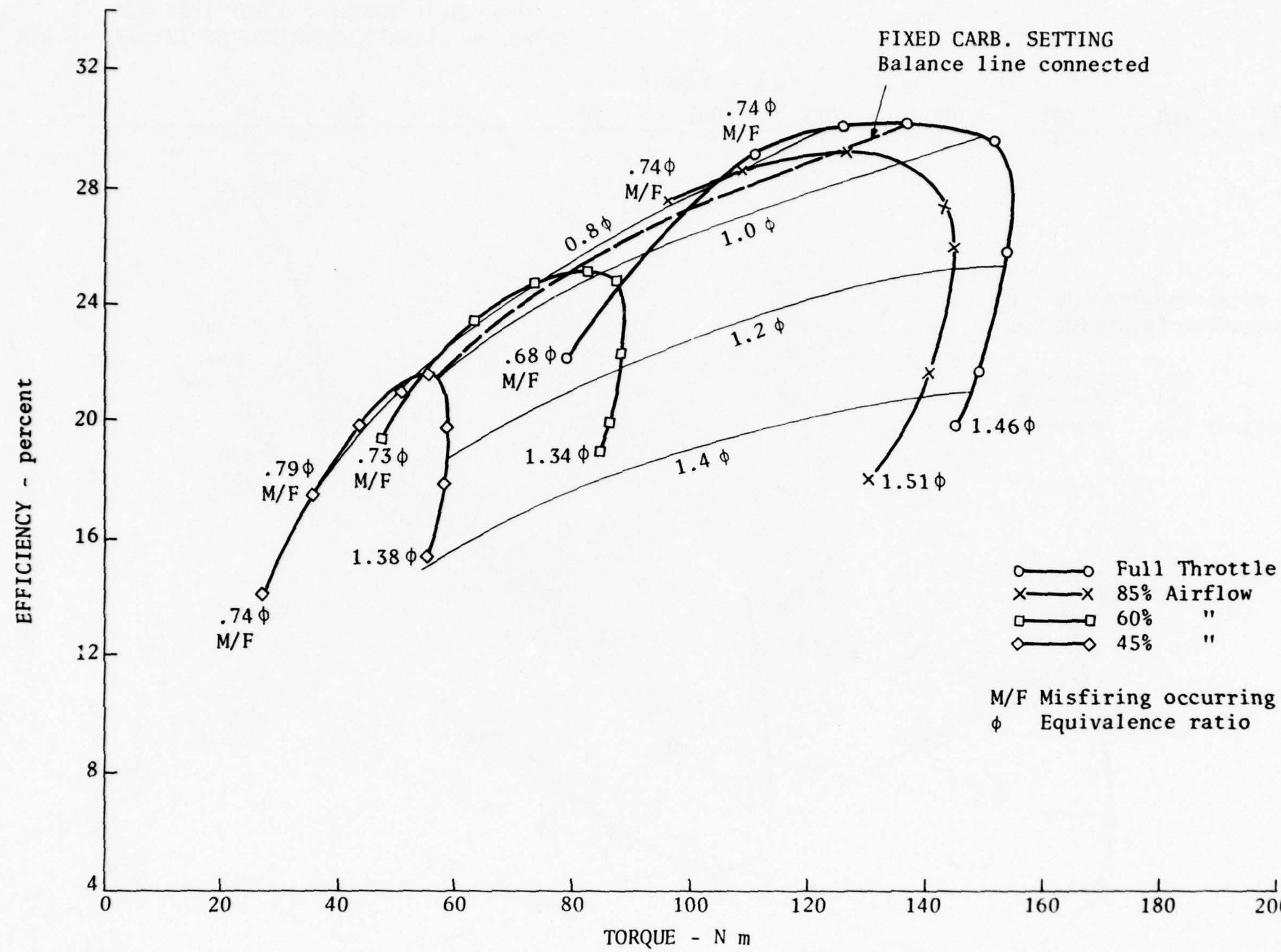


FIG 5 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - LPG 3500 r/min

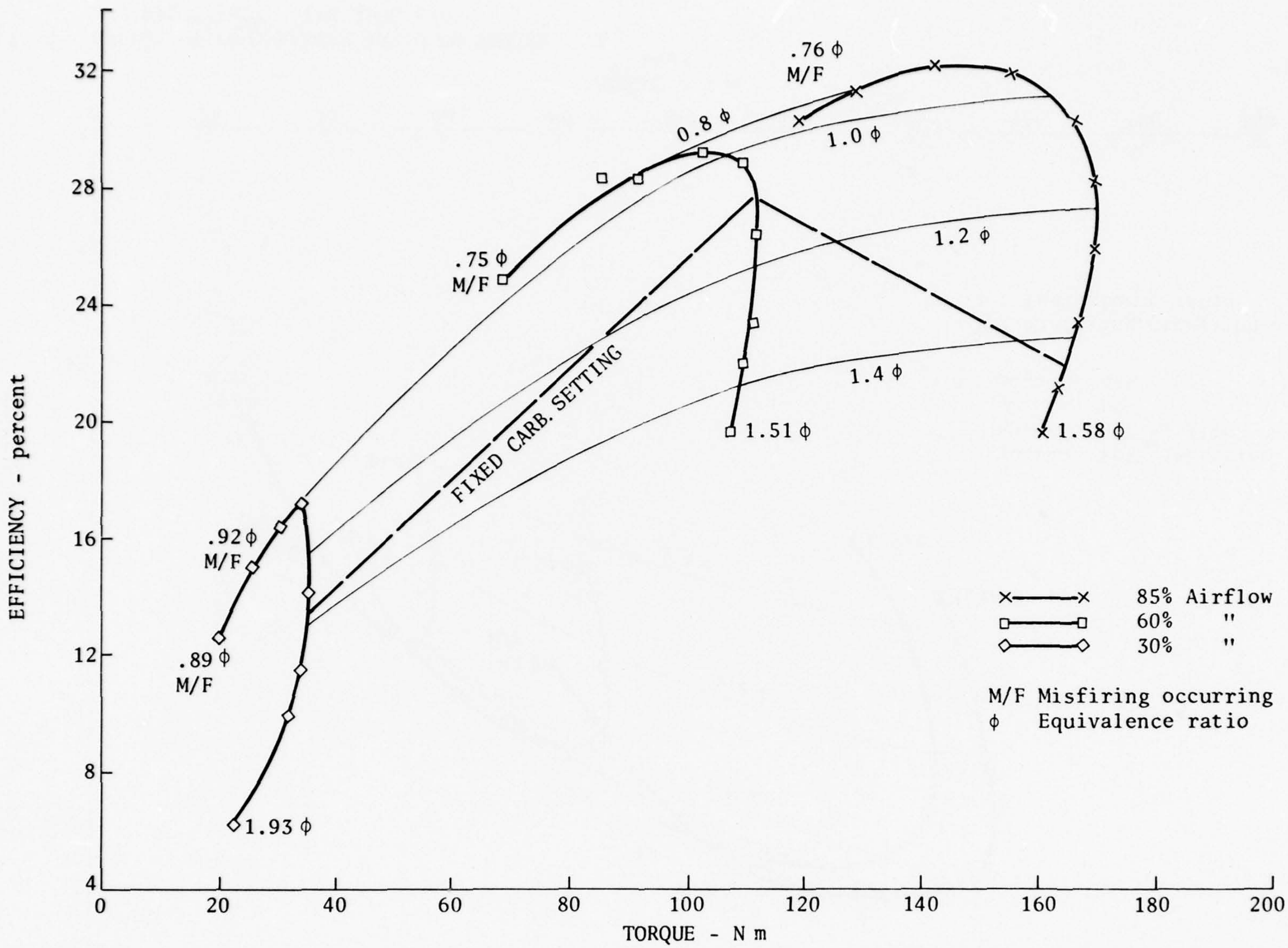


FIG 6 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - PETROL 1500 r/min

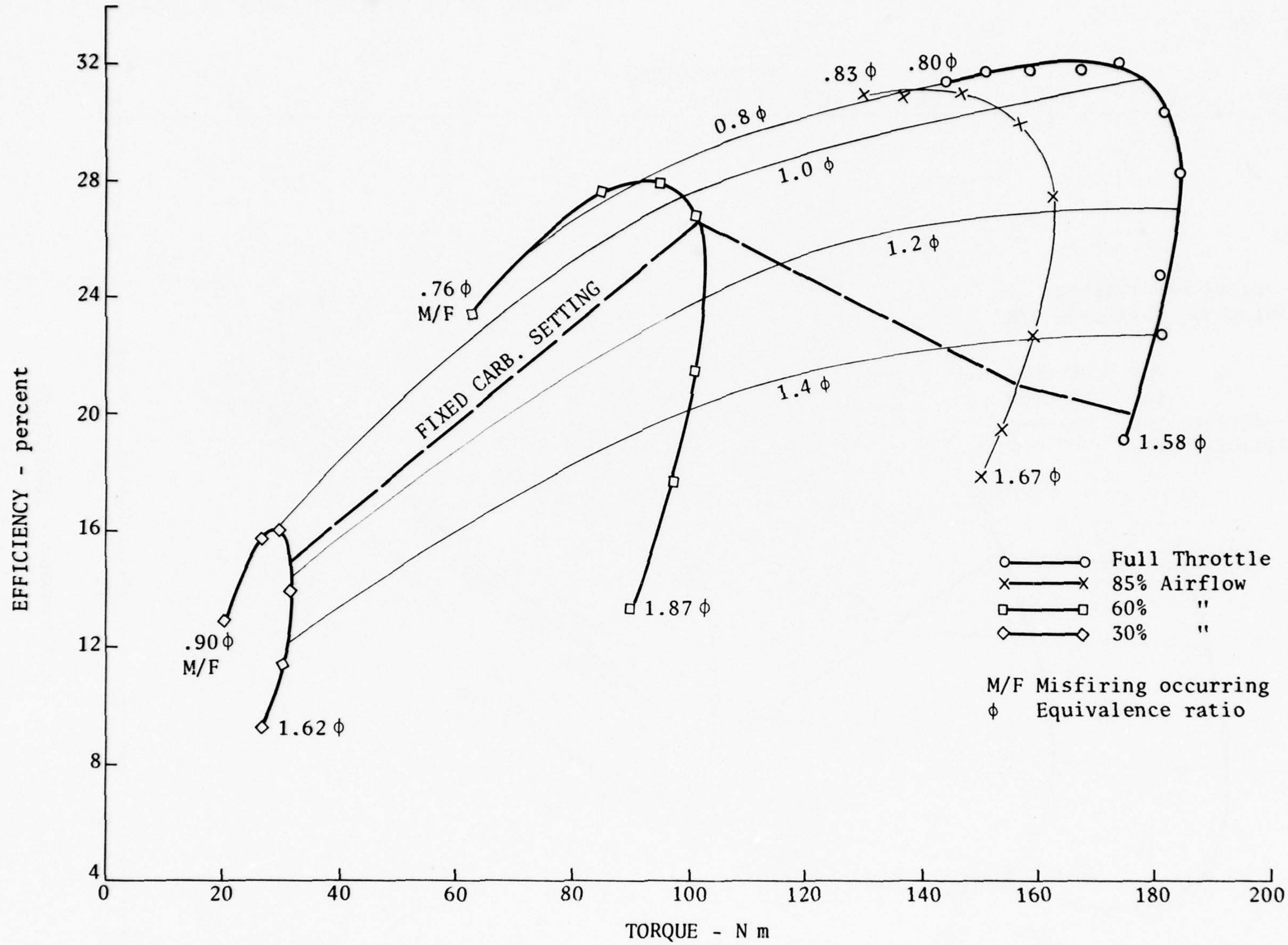


FIG 7 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - PETROL 2000 r/min

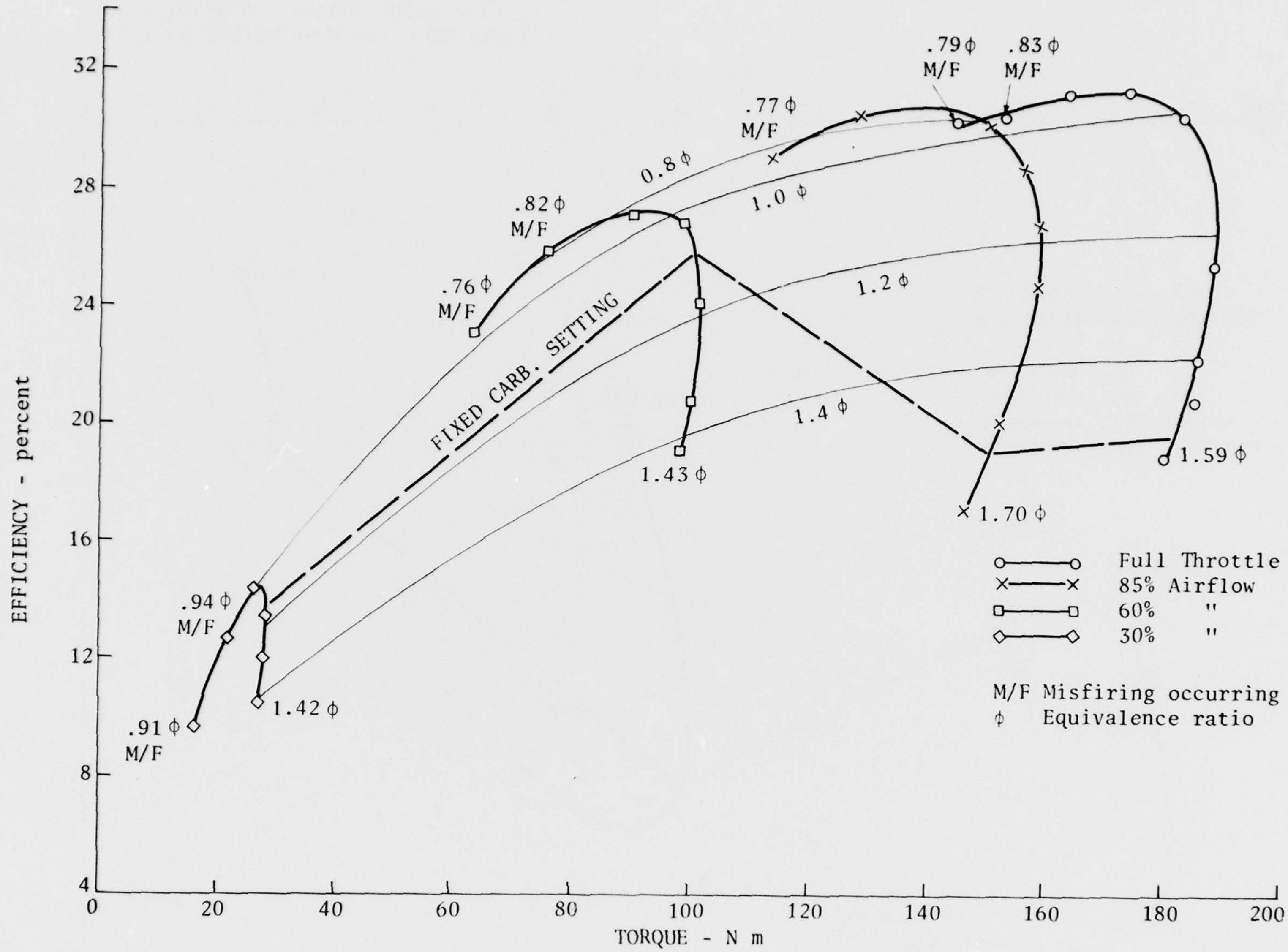


FIG 8 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - PETROL 2500 r/min

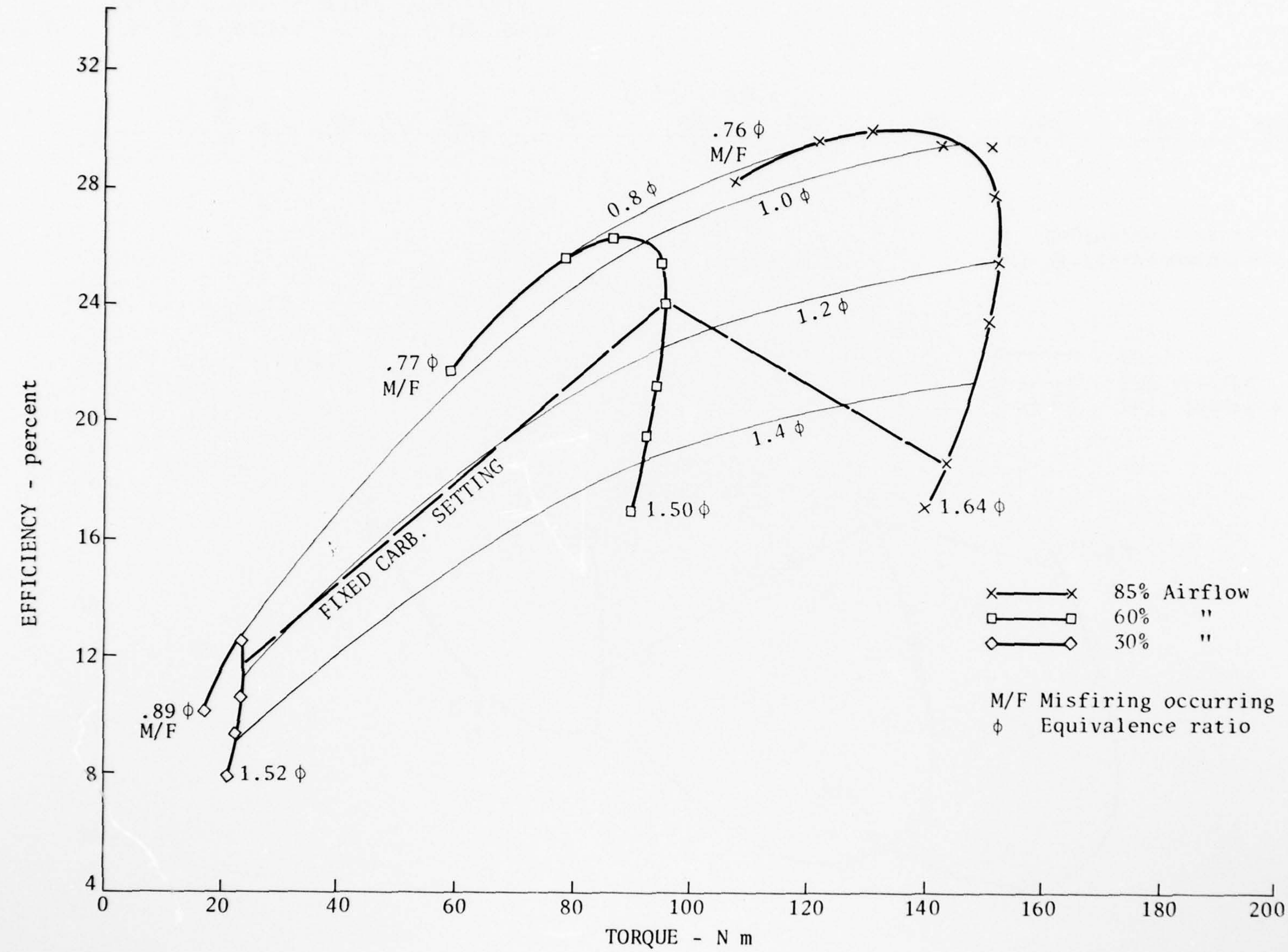


FIG 9 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - PETROL 3000 r/min

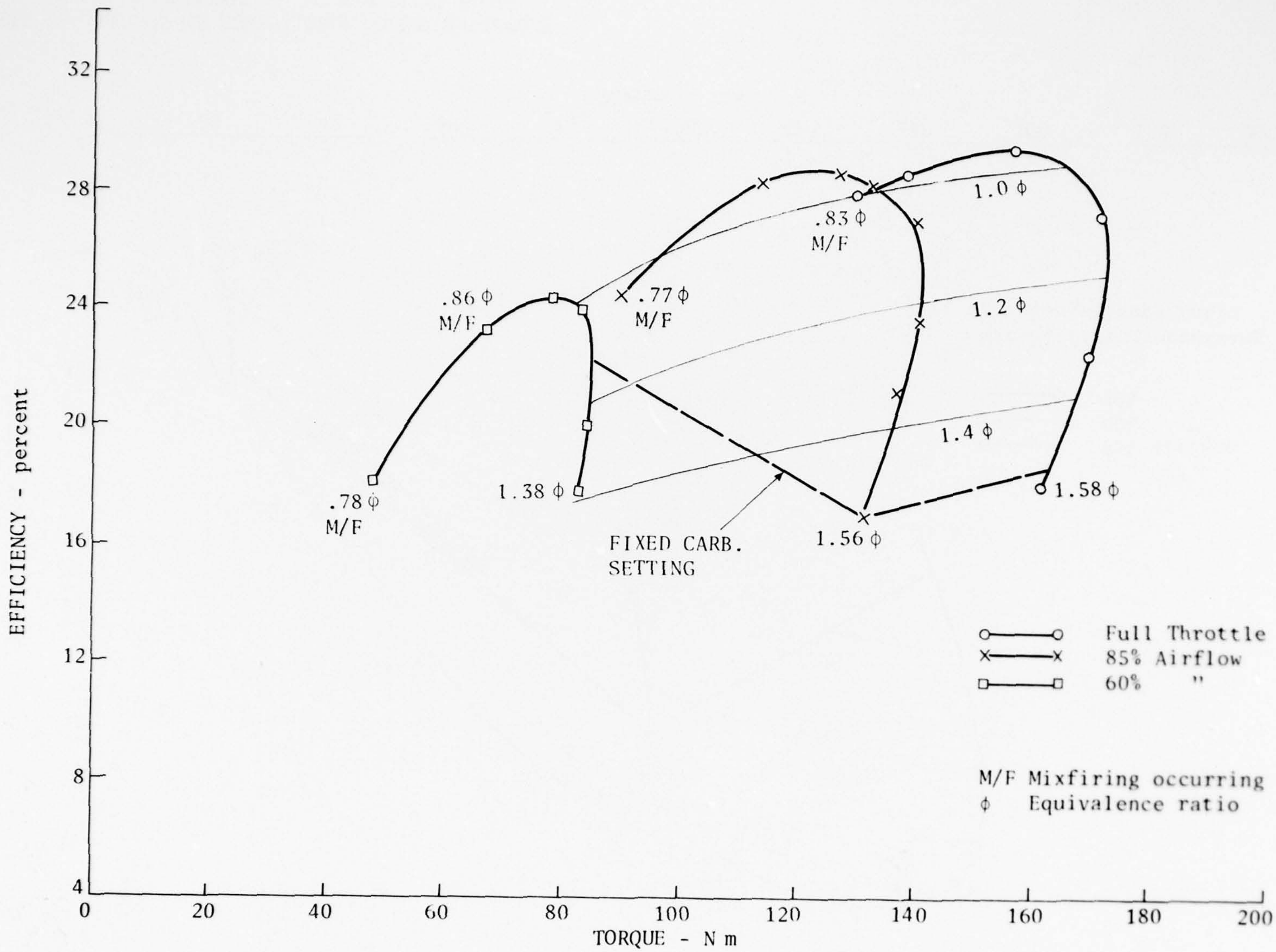


FIG 10 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - PETROL 3500 r/min

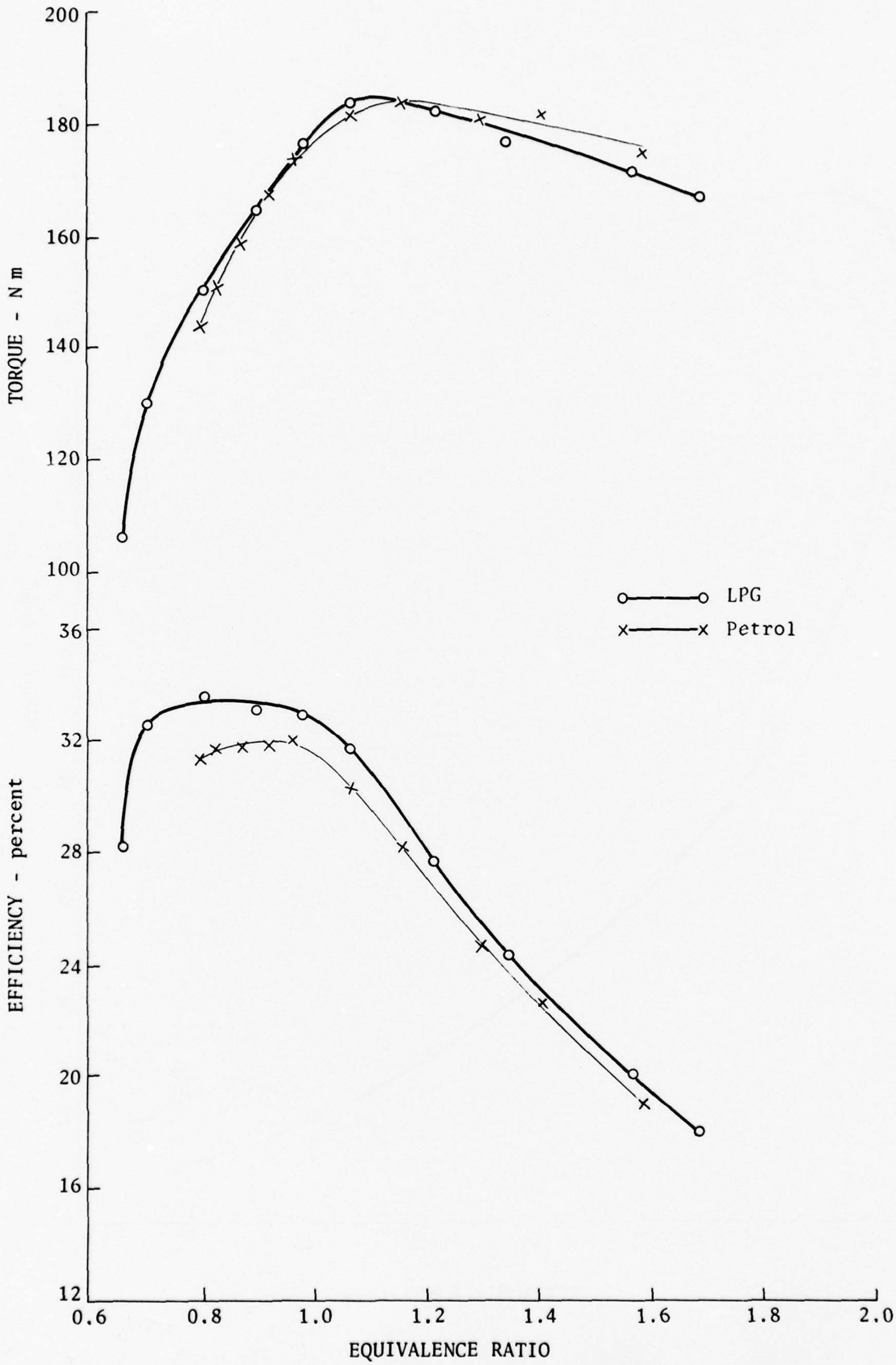


FIG 11 - EFFECT OF EQUIVALENCE RATIO ON TORQUE AND EFFICIENCY - LPG AND PETROL - FULL THROTTLE 2000 r/min

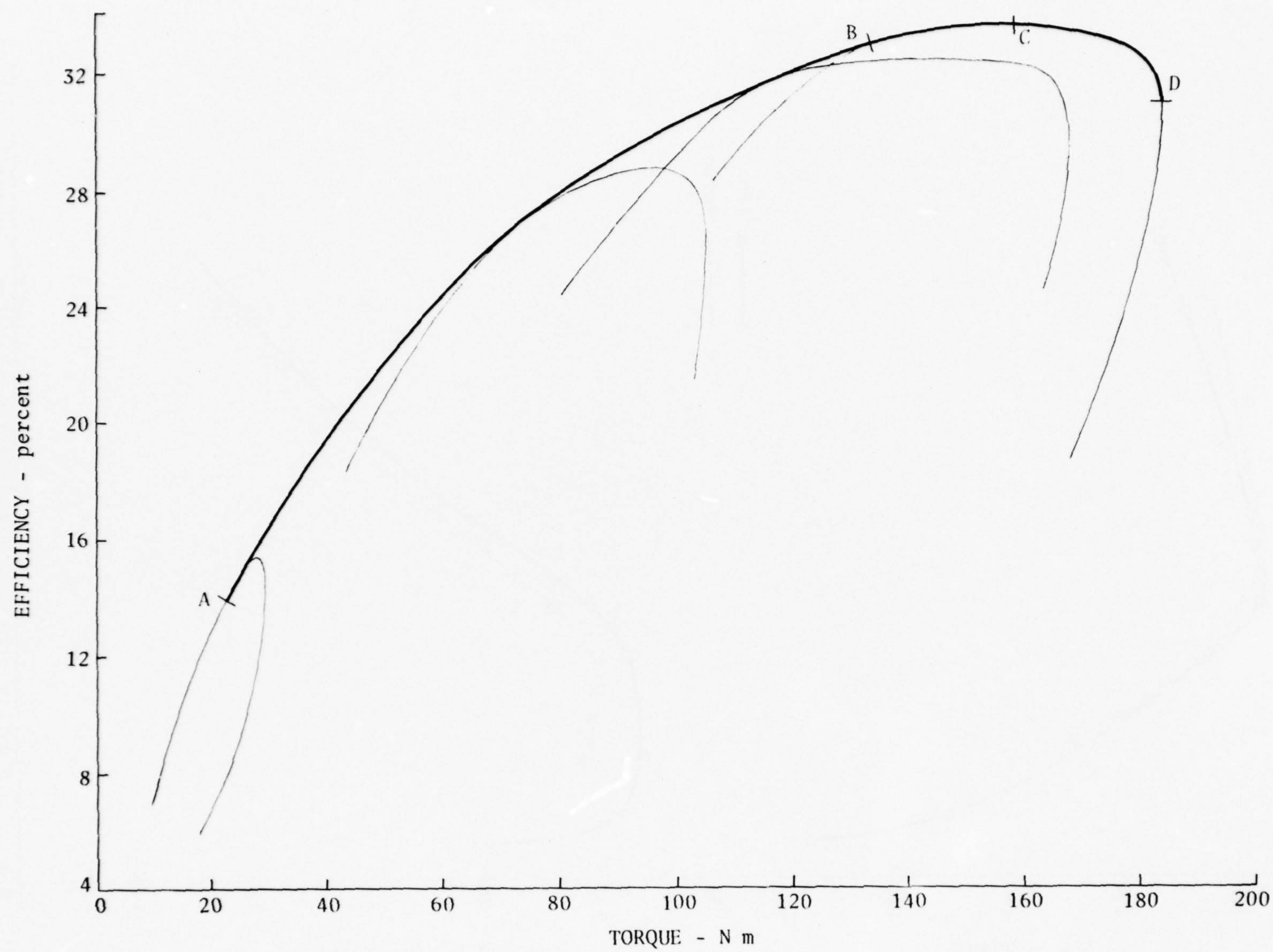


FIG 12 - PEAK EFFIENCY/TORQUE CURVE
- LPG 2000 r/min

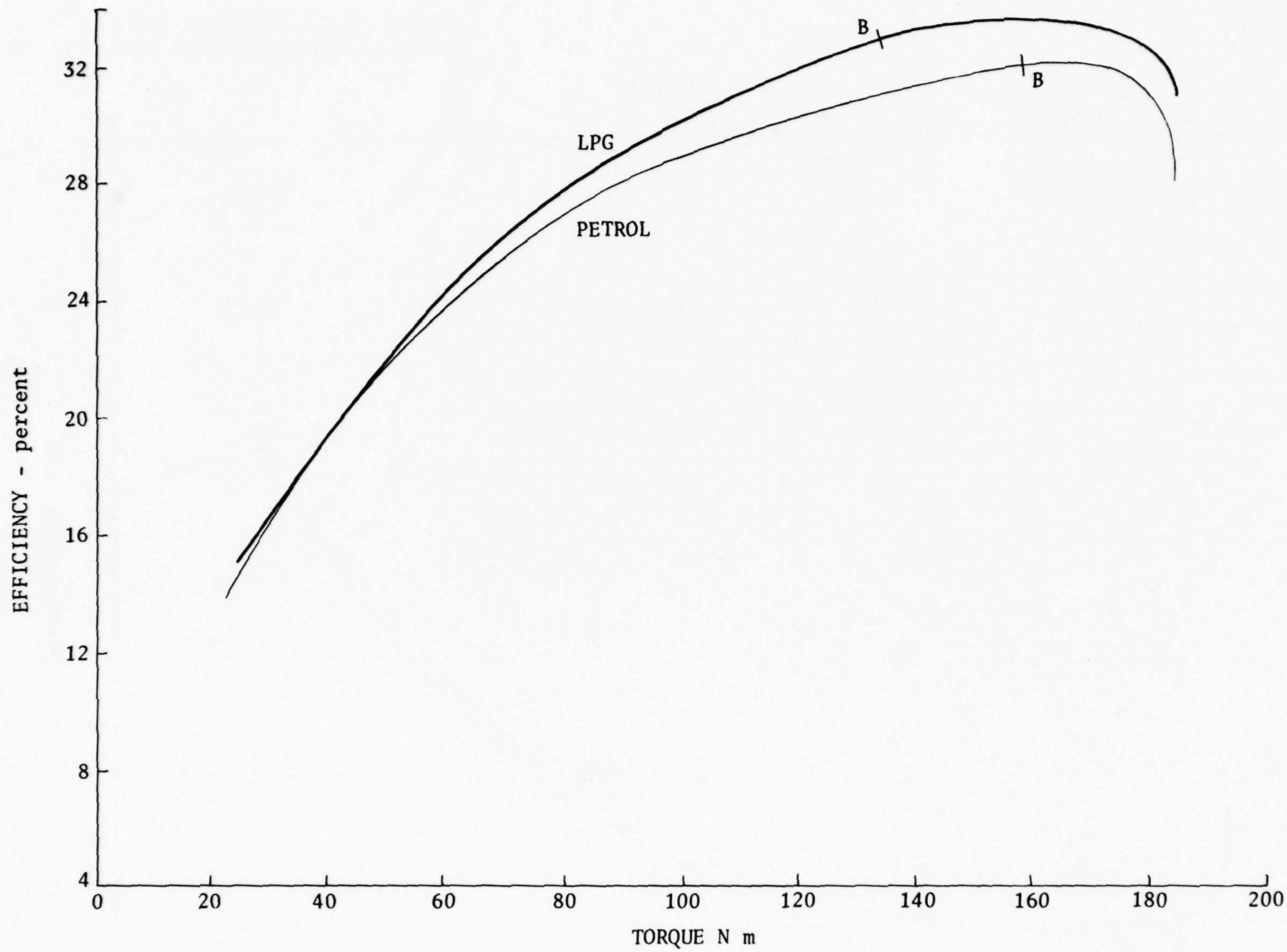


FIG 13 - PEAK EFFICIENCY/TORQUE CURVES
- LPG AND PETROL 2000 r/min

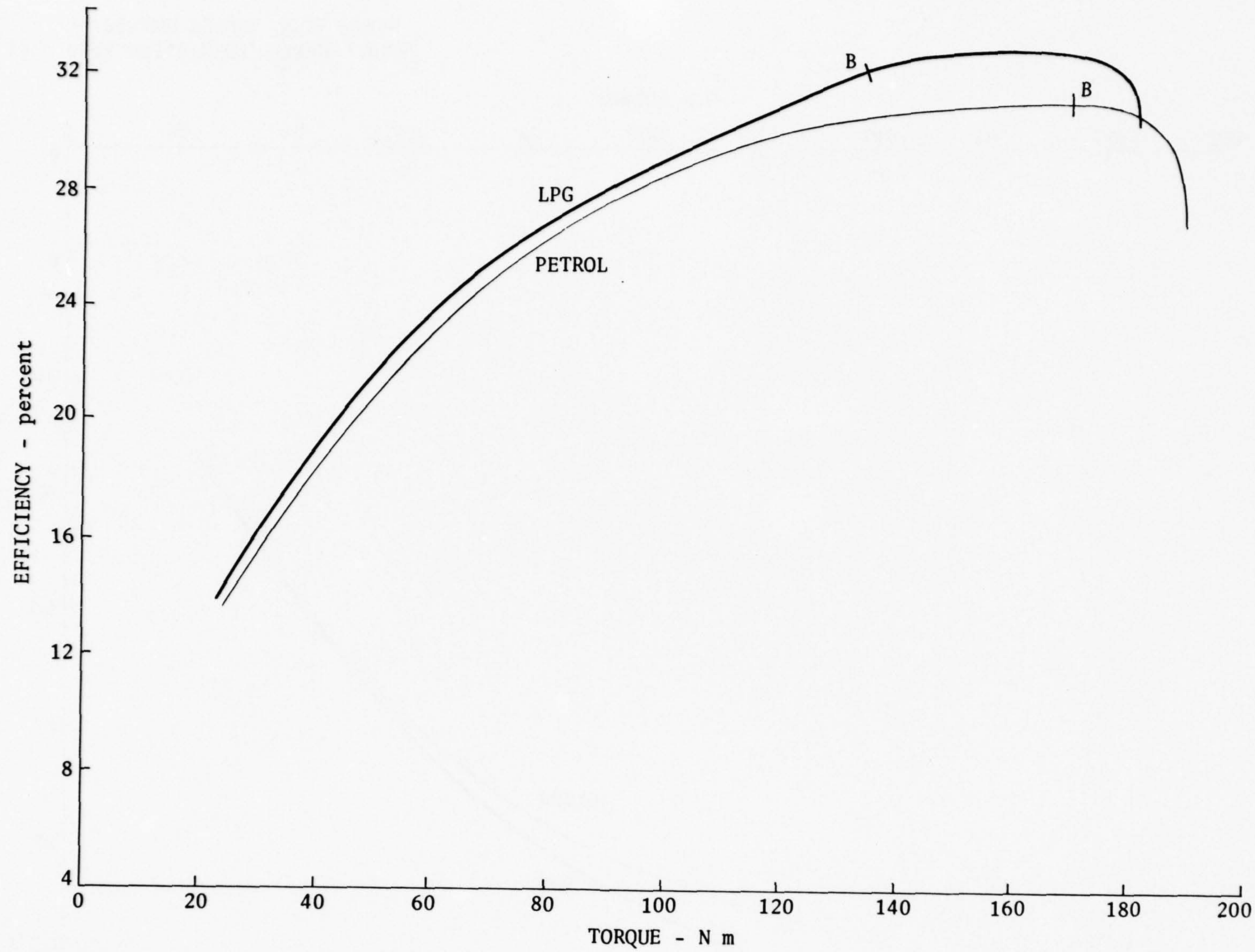


FIG 14 - PEAK EFFICIENCY/TORQUE CURVES
- LPG AND PETROL 2500 r/min

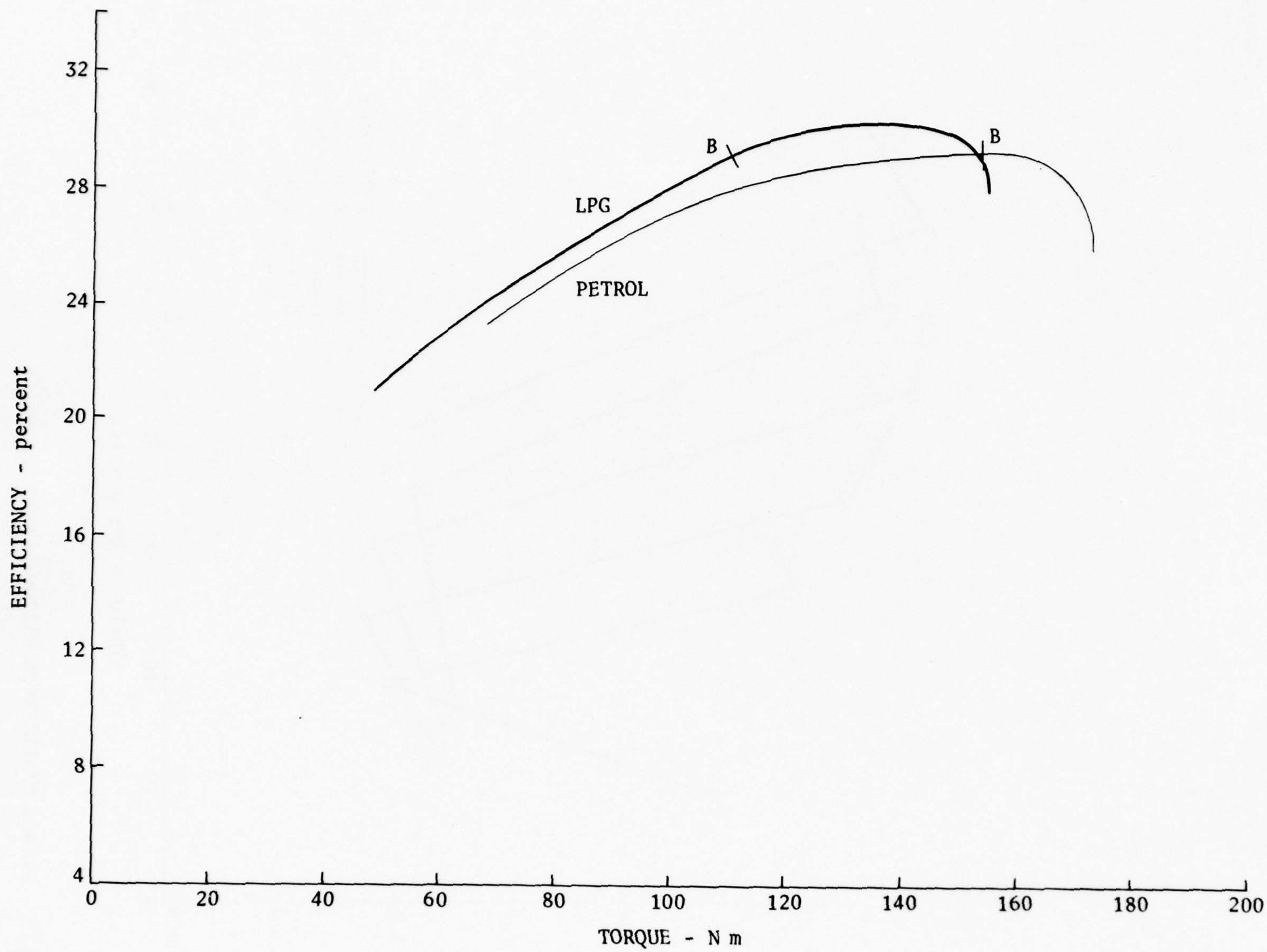


FIG 15 - PEAK EFFICIENCY/TORQUE CURVES
- LPG AND PETROL 3500 r/min

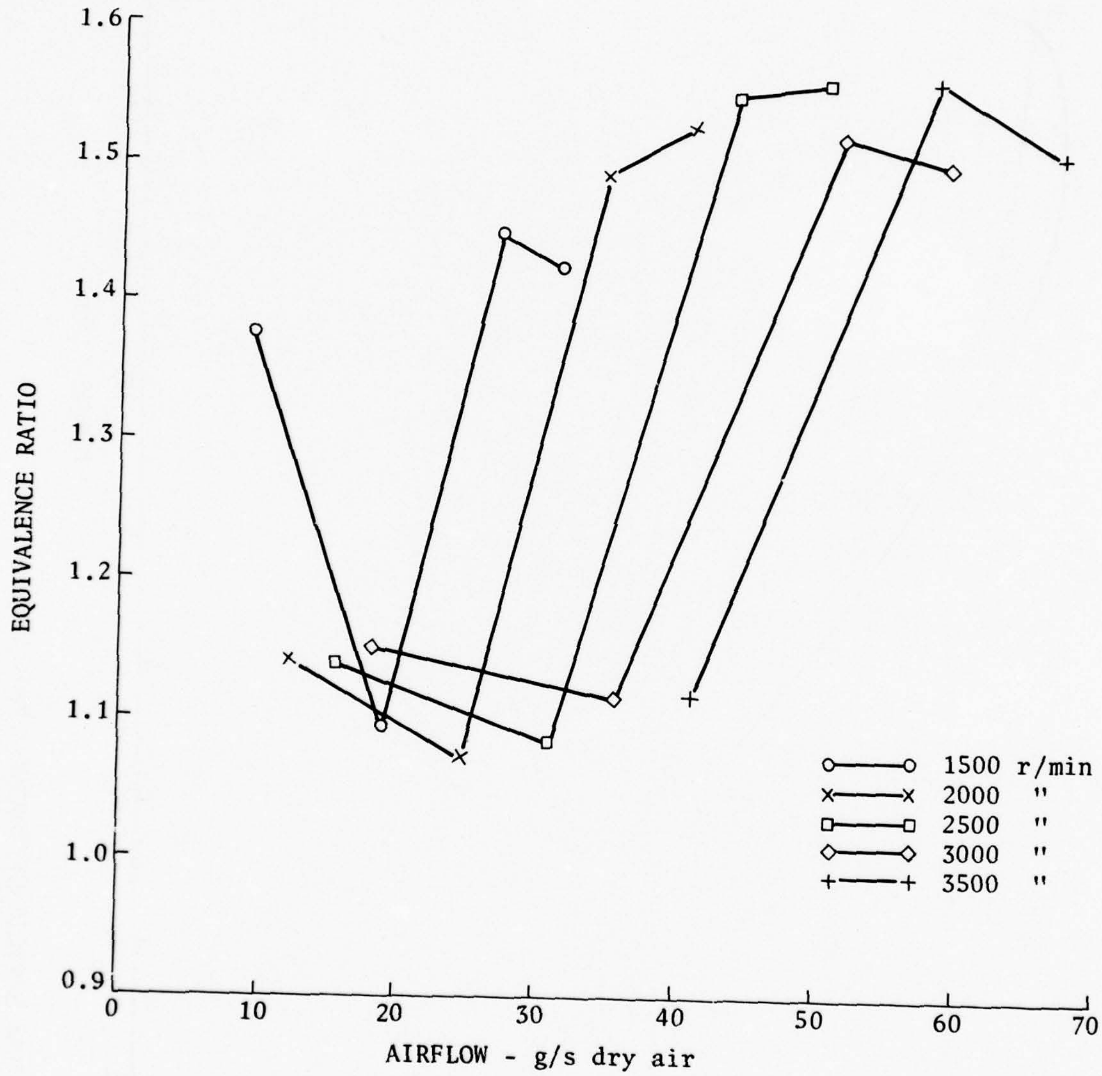


FIG 16 - PETROL CARBURETTOR MIXTURE CHARACTERISTICS

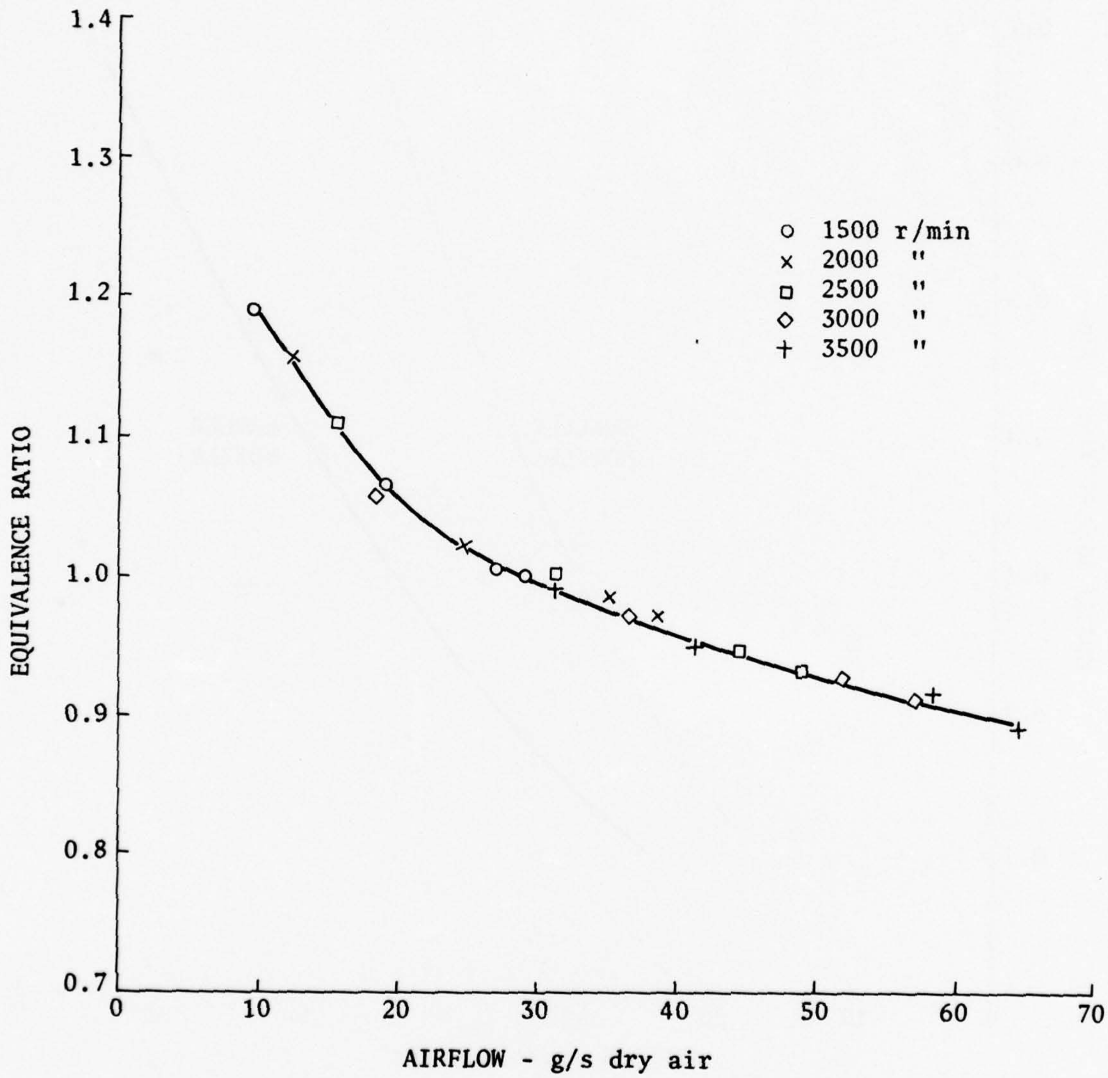


FIG 17 - LPG CARBURETTOR MIXTURE CHARACTERISTICS
WITH BALANCE LINE CONNECTED

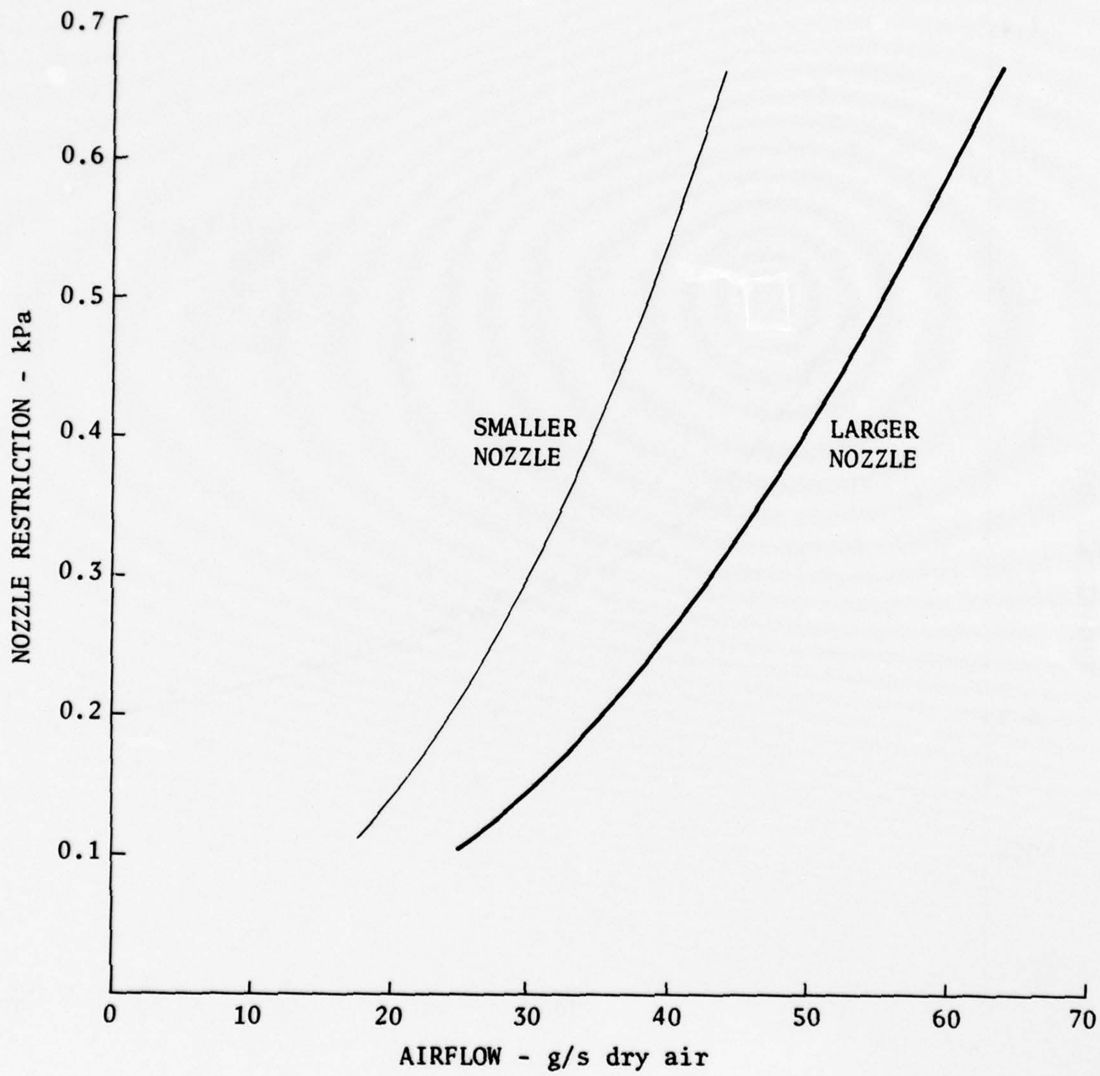


FIG 18 - NOZZLE RESTRICTION CURVES

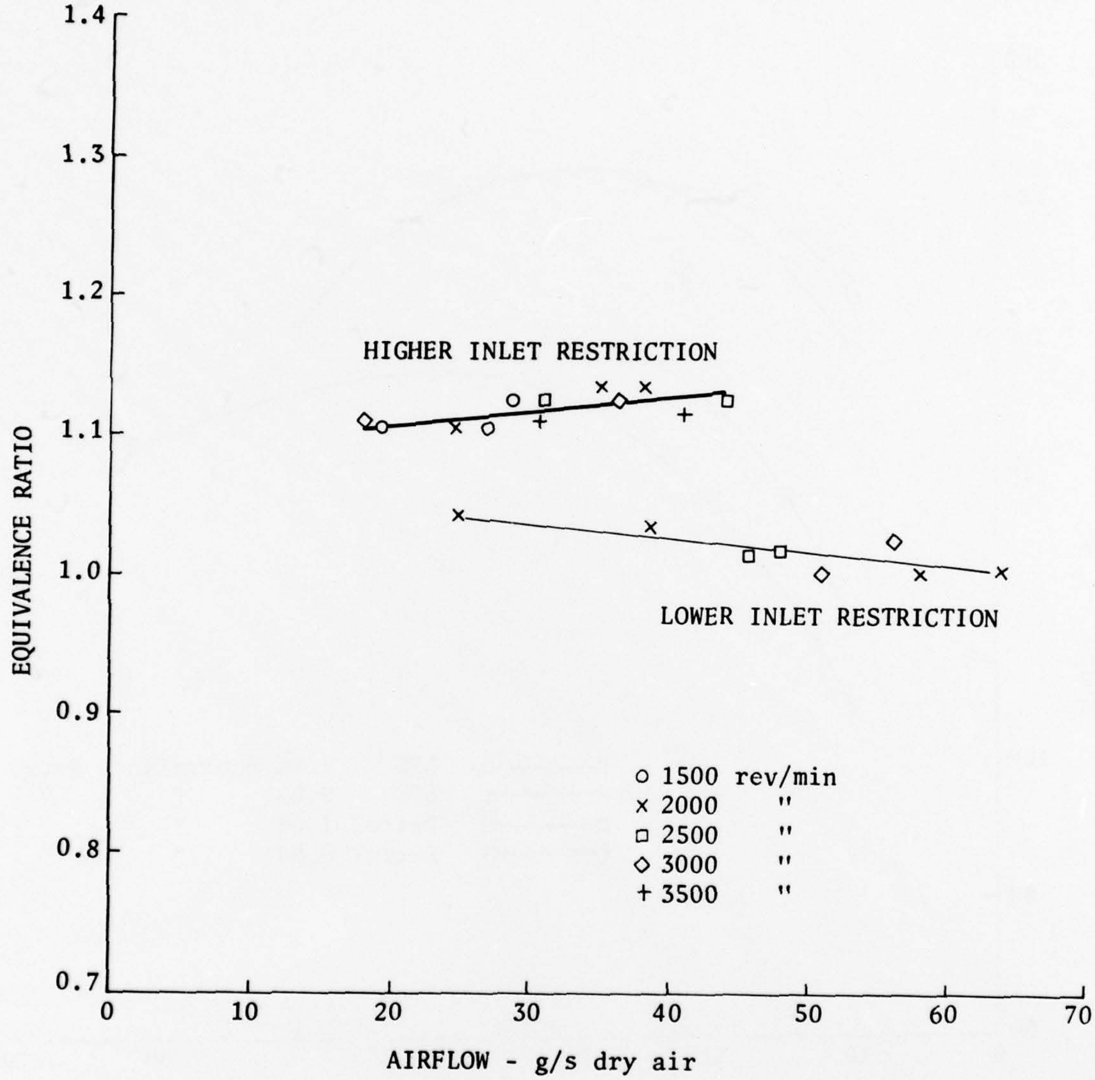


FIG 19 - LPG CARBURETTOR MIXTURE CHARACTERISTICS
WITH BALANCE LINE DISCONNECTED

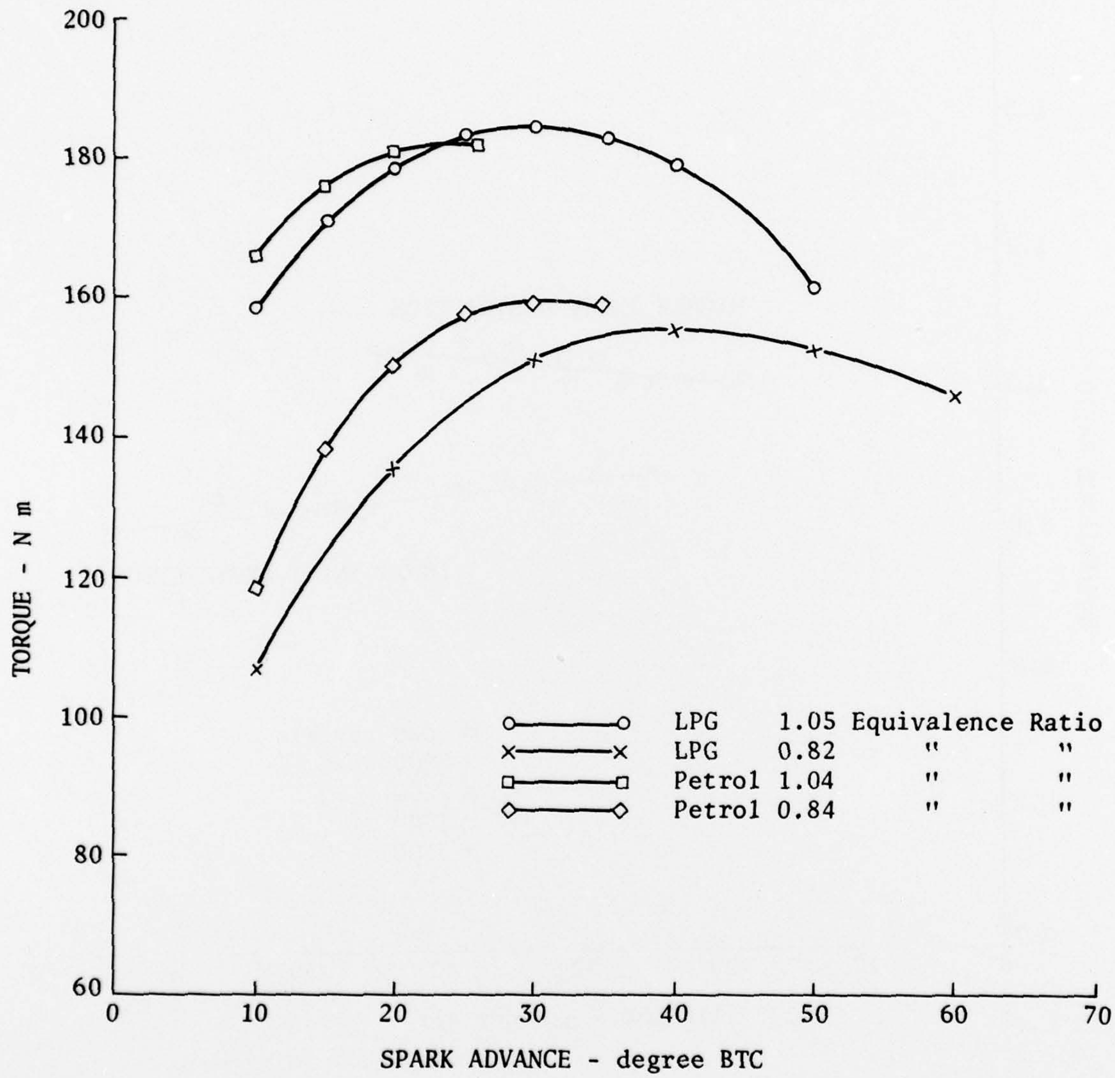


FIG 20 - EFFECT OF IGNITION TIMING ON FULL THROTTLE TORQUE
- LPG AND PETROL - 1500 r/min

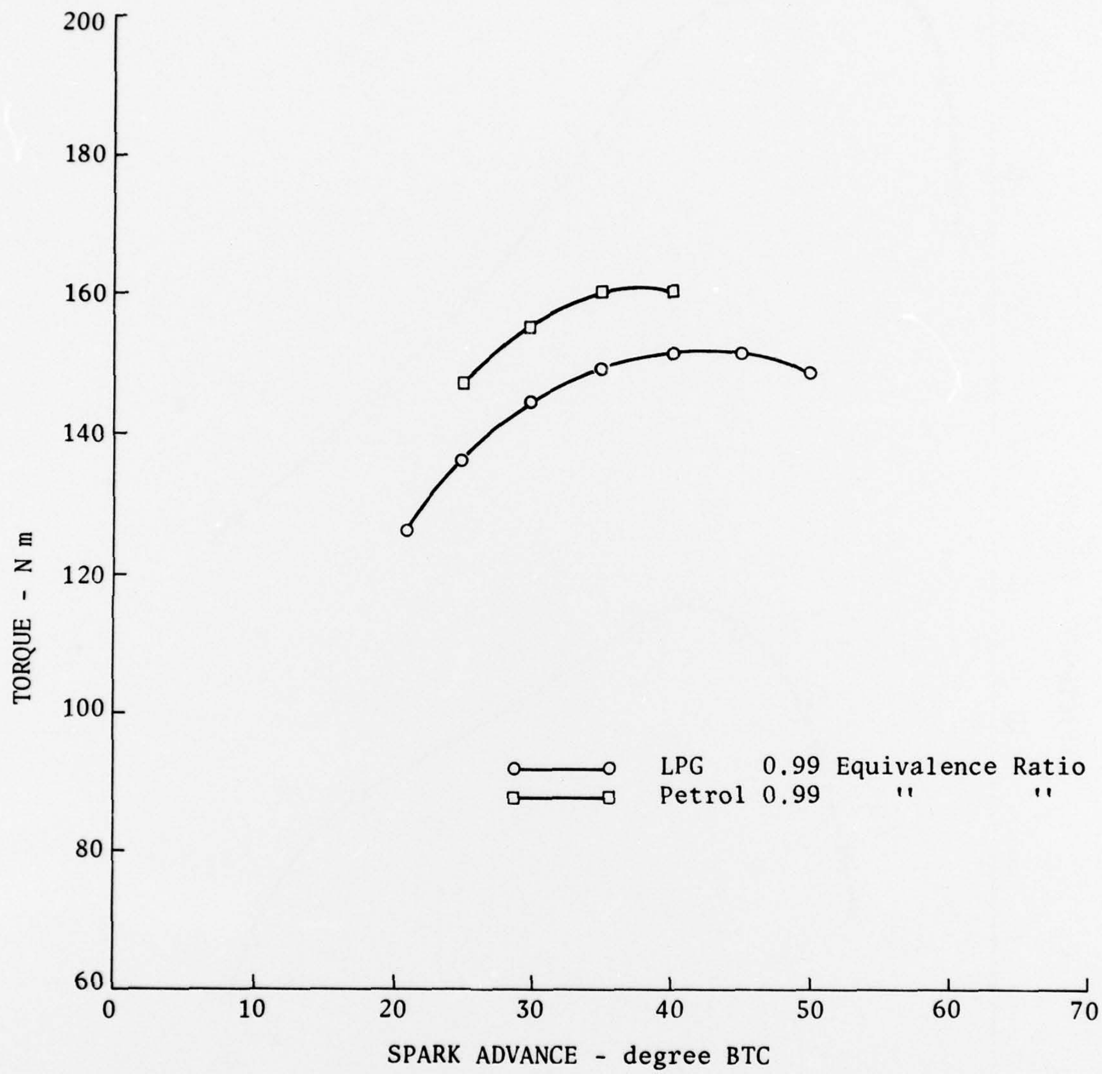


FIG 21 - EFFECT OF IGNITION TIMING ON FULL THROTTLE TORQUE
- LPG AND PETROL - 3500 r/min

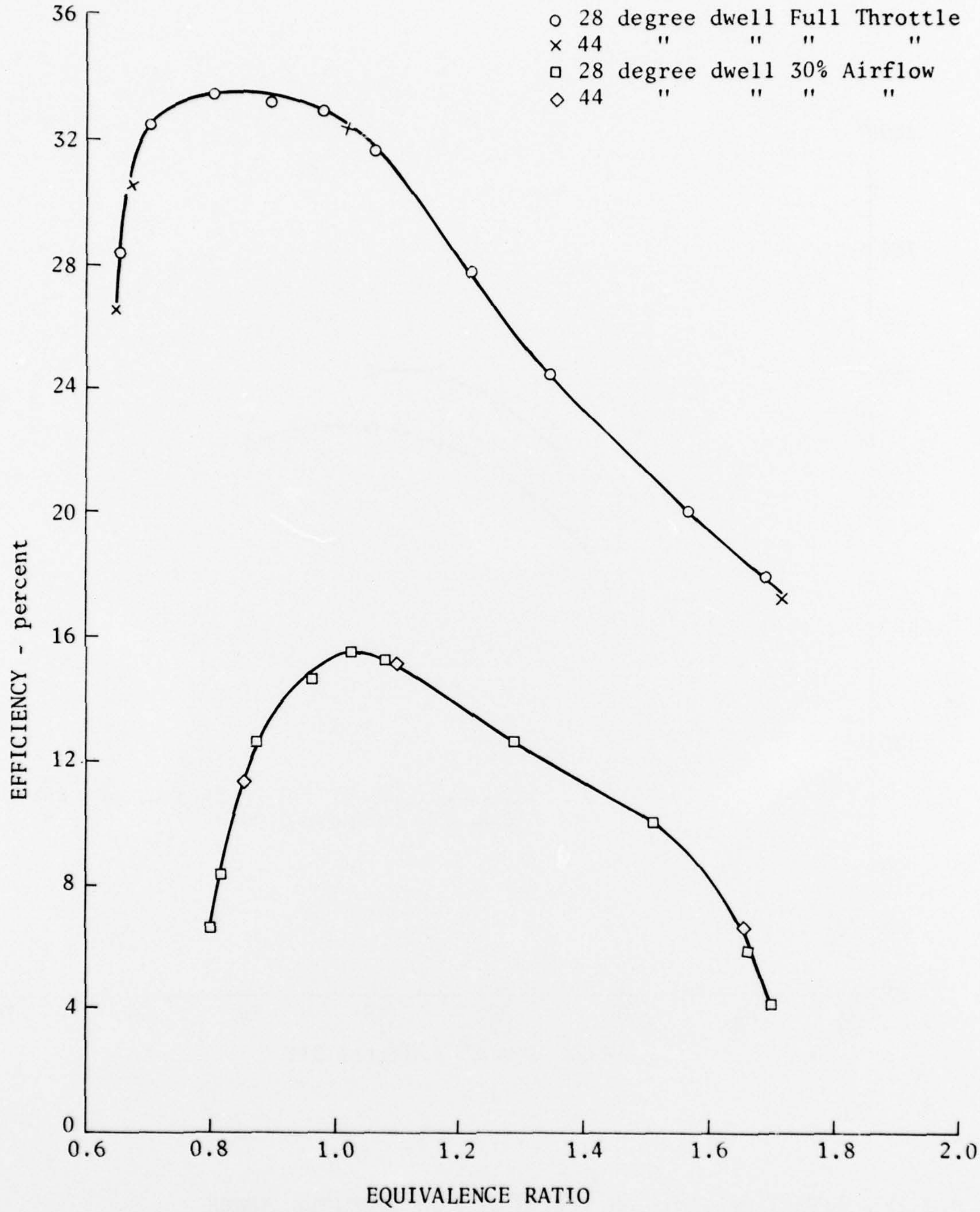


FIG 22 - EFFECT OF DWELL ANGLE ON EFFICIENCY
- LPG AND PETROL - 2000 r/min

FUEL SYSTEMS

Petrol Fuel System

1. Numbers in parentheses throughout para 2 to 4 refer to items of Fig 1. The petrol fuel system consists of a diaphragm pump, mechanically actuated from the engine camshaft and supplying fuel to Bendix Stromberg Model BXV2 single barrel, downdraft fixed venturi carburettor. The carburettor is shown diagrammatically in Fig 1.

2. The essential fuel metering elements of the carburettor are the main metering jet (15) controlling fuel supply to the main discharge jet (3) and a power by-pass jet (16) which supplies additional fuel to the discharge jet at wide throttle openings. The power by-pass jet is connected to an operating piston (8) which is acted on by manifold vacuum transmitted to it from a point below the throttle valve (12) via a duct within the carburettor body.

3. When the throttle valve is partly opened high manifold vacuum holds the piston up which in turn holds the by-pass valve jet closed. With the by-pass jet closed the quantity of fuel issuing from the discharge jet is controlled by the size of the orifice in the main metering jet, acted upon by the pressure differential across the venturi (1). Under these conditions the mixture remains relatively lean for good fuel economy.

4. When the throttle valve is opened sufficiently to reduce manifold vacuum below a pre-determined level a control spring acting on the piston forces it down to open the by-pass jet. Additional fuel then issues from the discharge jet thereby enriching the mixture. Mixture enrichment enables greater torque output to be obtained at full throttle valve opening.

LPG Fuel System

5. Fig 2 is a schematic diagram of a typical LPG fuel system, to which numbers and letters in parentheses in para 5, 6 and 9 refer. The fuel is stored as a liquid in the fuel tank (1) which is pressurised by the fuel's own vapour pressure. The tank is fitted with a filler valve (A), a vapour return valve (B) to return vapour displaced during filling to the bulk storage tank, an outage valve (C) opened during filling to vent fuel via lines (D and F) when the fuel reaches the permissible fill level, a vapour safety relief valve (E), a fuel supply valve (G), a fuel line (H) and a fuel gauge (K).

6. Fuel passes from the tank via a liquid safety relief valve (2) and a combined filter and lock-off valve (3) to a converter (4). The lock-off valve is a normally closed type which is operated either electrically through the vehicle ignition switch or pneumatically by engine intake vacuum.

7. The converter is a two stage pressure regulator. An exploded view of the Impco Model J converter fitted during the tests is given in Fig 3. A primary regulating valve reduces fuel pressure and also acts as an expansion valve to expand liquid fuel to the gaseous phase. Hot water from the engine cooling system provides the necessary energy for vaporization and expansion.

8. Fuel pressure is further reduced through a secondary regulating valve to a level acceptable to the carburettor. For the Impco Model CA300A carburettor fitted to the engine this pressure is -0.31 kilopascal (kPa) relative to atmospheric pressure. The slight negative pressure ensures that fuel within the converter is not delivered to the carburettor unless the engine is running. If desired the secondary regulating pressure can be referenced to carburettor inlet pressure, rather than atmospheric, by attachment of a balance line from the carburettor inlet to the atmospheric side of the regulating diaphragm. Use of the balance line is recommended by the manufacturer if a highly restrictive air cleaner is used.

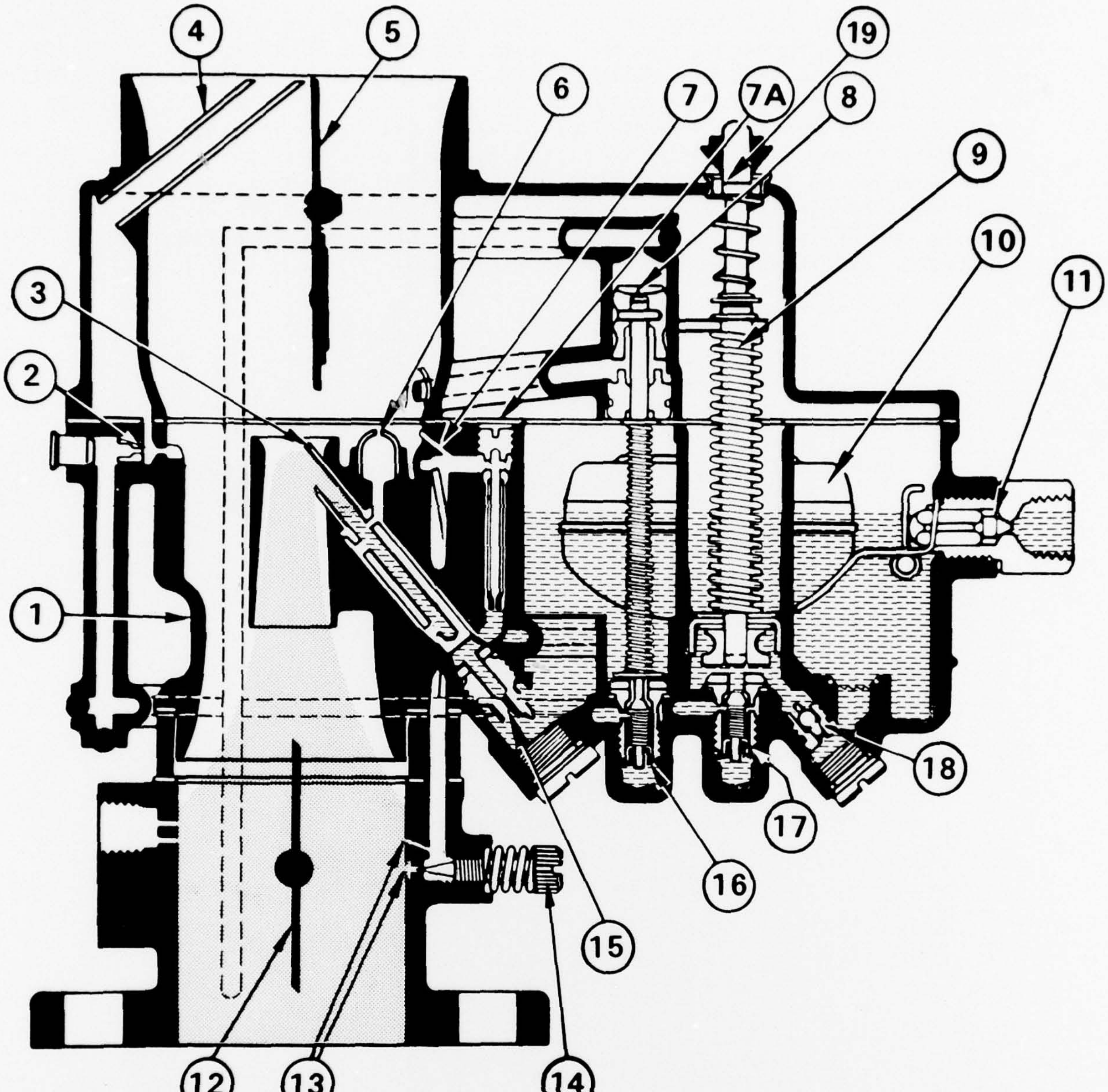
9. The carburettor (6) operates in effect as a variable venturi type. A carburettor similar in construction to the one used during the tests is shown diagrammatically in Fig 4. Air entering the carburettor passes through an air metering valve, creating a pressure differential. The lower pressure inside the carburettor airway is transmitted via a transfer tube to the upper side of a diaphragm connected to the air metering valve causing the diaphragm and valve to rise against the action of a control spring by an amount dependent upon the airflow. Also connected to the diaphragm is a gas metering needle valve shaped to admit the correct quantity of gas. The variable air valve sets up a strong metering force on the gas entering the carburettor at low airflows whilst offering minimum restriction at high airflows. An adjustable valve (power mixture valve) in the gas inlet to the carburettor is set after installation to obtain the desired mixture strength at maximum engine torque.

10. With the balance line connected between carburettor and converter a common reference pressure governs both the air and gas flows into the carburettor. Up to the point of mixing, airflow is subject only to air metering valve restriction, whereas gas flow is subject to both the gas metering valve and the power mixture valve restrictions. Hence both the air-gas metering valve system and the power mixture valve will influence mixture strength.

11. With no balance line connected, the gas reference pressure is constant at atmospheric, whereas the air reference pressure at the carburettor inlet varies with airflow and is lower. However, since the power mixture valve can be expected to have a 'square law' restriction characteristic with flow similar to that of the air cleaner, the gas pressure downstream of the valve will be similar to the carburettor inlet air pressure. With similar air and gas pressures being presented to the air-gas metering valve system, mixture strength will predominantly be under the control of this system and therefore governed by the form of the gas metering valve needle.

12. As in the petrol carburettor, mixture supply to the engine is controlled by means of a butterfly type throttle valve. No provision is made in the design of the carburettor/converter system to vary mixture strength with throttle valve position. Consequently no mixture enrichment occurs at wide throttle openings.

13. In dual fuel installations the LPG carburettor is mounted upstream of the petrol carburettor. In such cases, as in the CA 300A, the carburettor does not contain a throttle valve. Mixture supply to the engine is controlled at all times by the throttle valve in the petrol carburettor. In addition an electrically operated filter/fuel lock-off valve, rather than a pneumatically operated valve, is fitted in dual fuel systems together with a solenoid valve in the petrol fuel line and a fuel changeover switch. Also the CA 300A carburettor incorporates a manually operated air by-pass valve which is opened for petrol operation so that intake air by-passes the LPG carburettor and intake restriction is minimized.

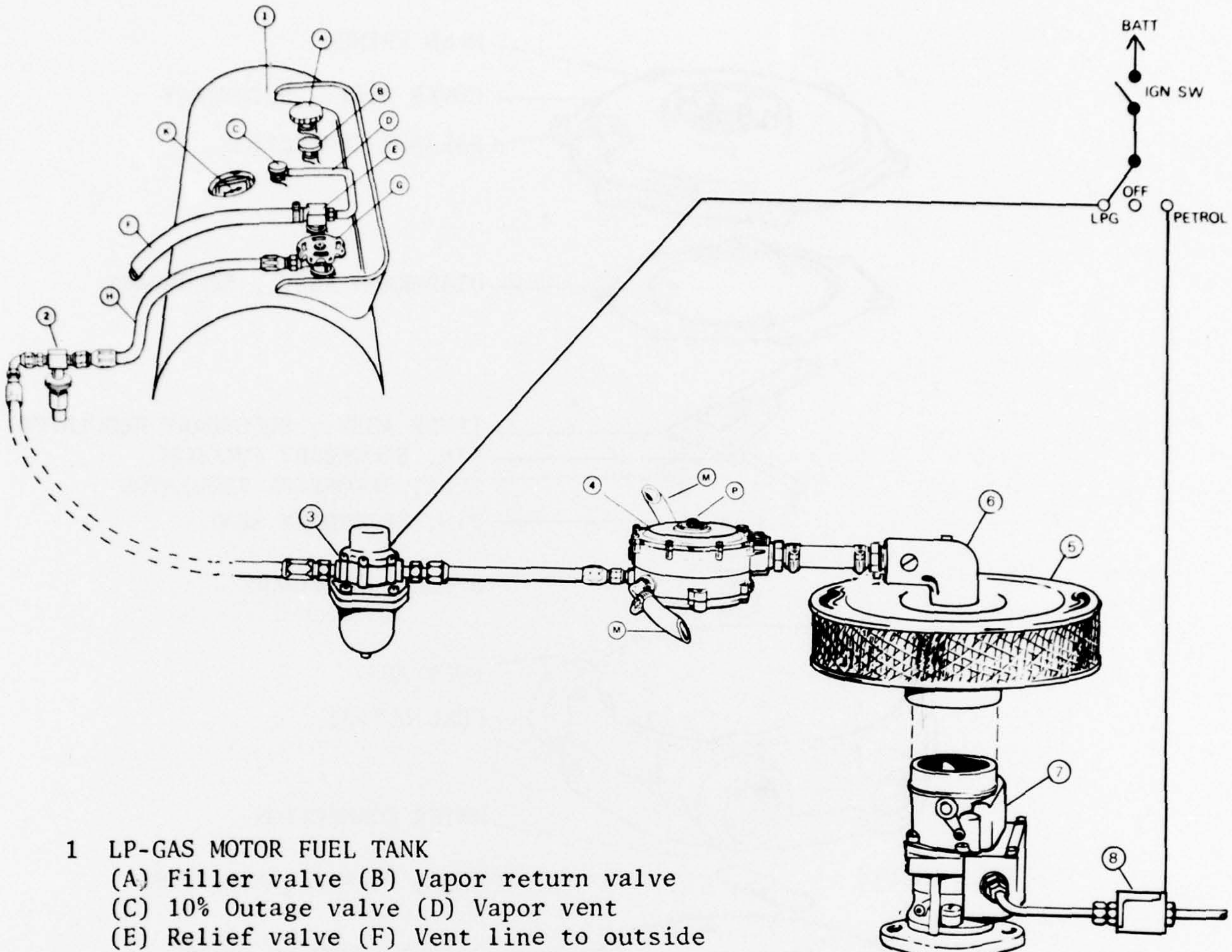


1. VENTURI: Increases air velocity in carburettor.
2. PUMP DISCHARGE NOZZLE: Meters fuel for pump discharge.
3. MAIN DISCHARGE JET: Mixes air and fuel and controls the quantity that is discharged into the air stream.
4. FLOAT CHAMBER VENT (INTERNAL): Vents chamber to atmosphere.
5. CHOKE VALVE: Restricts the air supply to enrich mixture for starting and during warm-up period.
6. HIGH SPEED AIR BLEEDER: Meters the air that is bled into main discharge system.
7. IDLE AIR BLEED: Meters air bled into the idle system.

7a. IDLE TUBE: Meters the fuel for the idle system.

8. VACUUM PISTON: Is controlled by the intake manifold vacuum and automatically operates the by-pass jet in accordance with speed or load of the engine.
9. ACCELERATING PUMP: Delivers additional fuel momentarily while accelerating to provide smooth and rapid acceleration.
10. FLOAT: Maintains the fuel in the float chamber at a definite level.
11. FLOAT NEEDLE VALVE AND SEAT: Controls the fuel that is admitted into the float chamber.
12. THROTTLE VALVE: Controls the quantity of mixture of fuel and air that is admitted into the intake manifold, and thereby governs the speed of the engine.
13. IDLE DISCHARGE HOLES: Discharge the fuel that is used during idle range.
14. IDLE NEEDLE VALVE: Controls the quantity of fuel that is discharged from the primary idle hole at closed throttle, also mixture ratio.
15. MAIN METERING JET: Meters all the fuel that is used in the range of normal speed operation.
16. POWER BY-PASS JET: Meters the fuel that is required for high speed running or pulling under load, in addition to fuel metered by metering jet.
17. ACCELERATING PUMP BY-PASS JET: Meters the fuel for the pump discharge.
18. ACCELERATING PUMP CHECK VALVE: Admits fuel into the pump cylinder.
19. VENT VALVE: Admits atmospheric pressure to float chamber when throttle valve is closed.

FIG 1-PETROL CARBURETTOR



1 LP-GAS MOTOR FUEL TANK
(A) Filler valve (B) Vapor return valve
(C) 10% Outage valve (D) Vapor vent
(E) Relief valve (F) Vent line to outside of vehicle (G) LP-Gas valve (H) LP-Gas high pressure hose line (K) Fuel gauge

2 HYDROSTATIC RELIEF VALVE
Vented to outside of Vehicle

3 FILTER AND LPG FUEL LOCK

4 CONVERTER
Two stage regulator and converter
(M) Water inlet or outlet from engine
(P) Hand Primer

5 AIR CLEANER

6 LPG CARBURETTOR

7 EXISTING PETROL CARBURETTOR

8 PETROL LOCK

FIG 2
ELEMENTS OF LPG FUEL SYSTEM (DUAL FUEL)

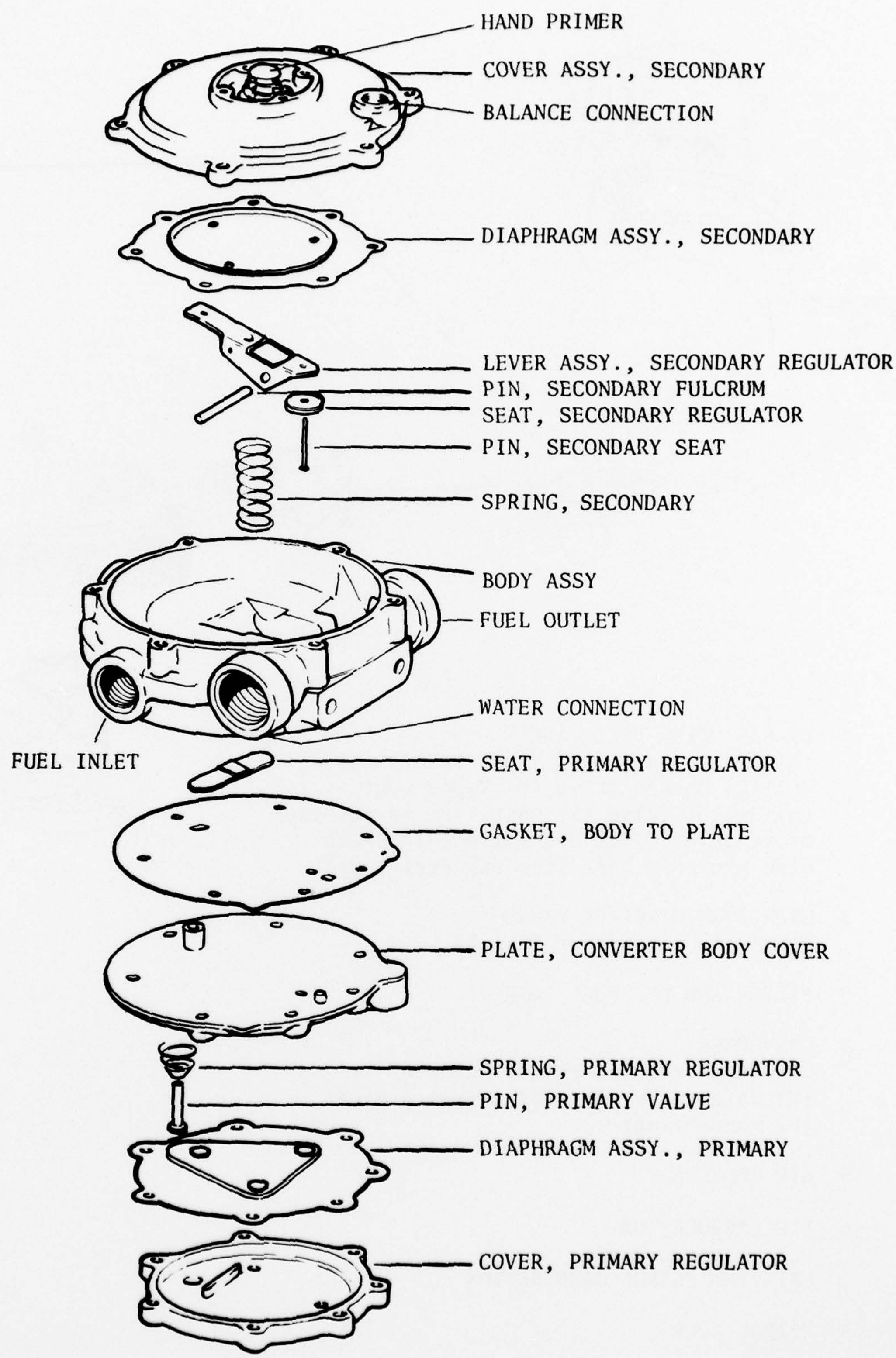


FIG 3 - MODEL J CONVERTER

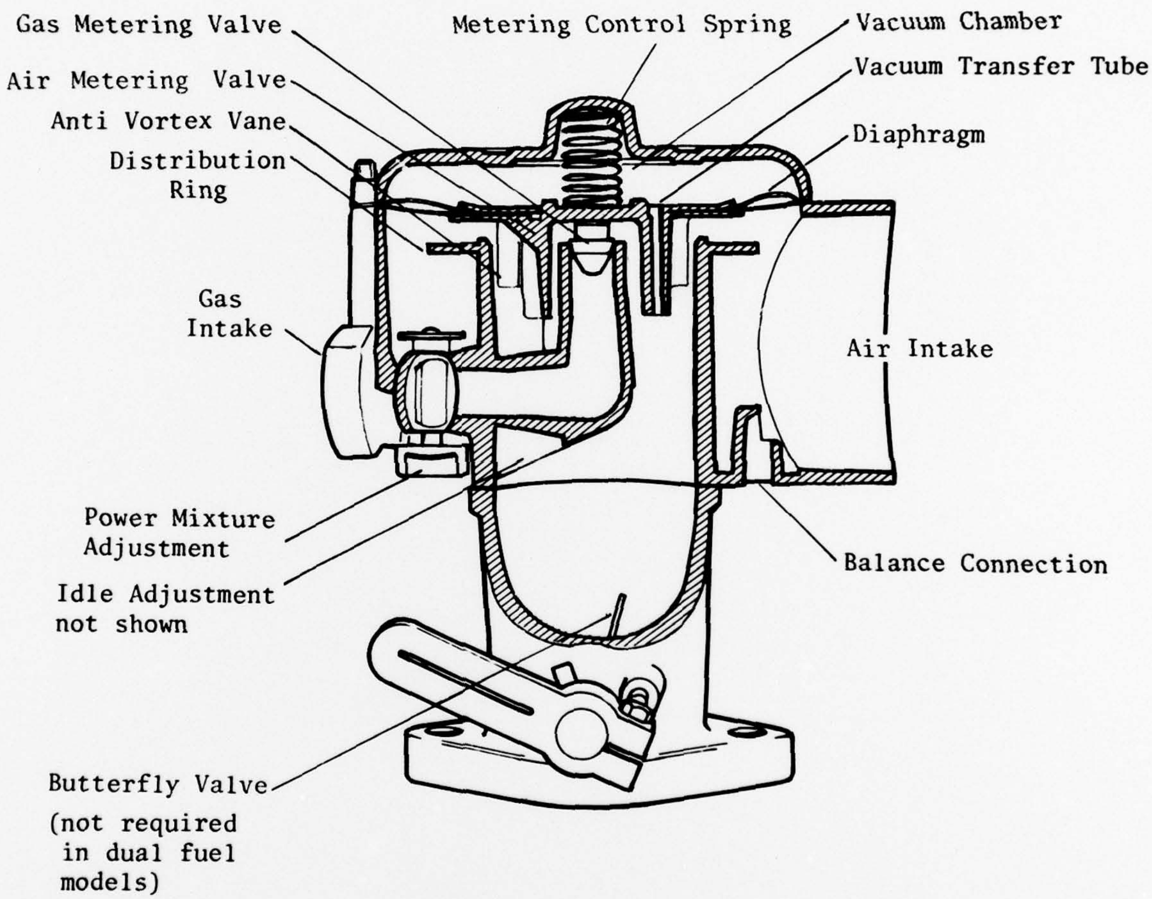


FIG 4 - DIAPHRAGM OPERATED AIR VALVE GAS CARBURETTOR

PARAMETERS MEASURED

1. The parameters measured were:

   a. Torque output by means of a hydraulic dynamometer fitted with an electrical load cell.

   b. Speed by means of a shaft mounted pulse generator and digital frequency meter.

   c. Intake airflow by measurement of the pressure differential across a selection of airflow nozzles (refer Fig 1).

   d. Fuel consumption in each case by means of an automatically timed weighing apparatus; the apparatus used for LPG fuel consumption measurement is shown in Fig 1. The converter was mounted on the weigh bottle so that fuel leaving the weighing scale was in the gaseous phase. If fuel in the liquid phase is fed from the weighing scale, uncontrolled spasmodic boiling inevitably takes place in the delivery hose causing displacement of liquid and consequent fuel consumption measurement errors.

   e. Ignition advance by means of a proprietary phase angle meter designed for the purpose.

   f. Dwell angle was measured and set by means of a degree plate fitted to the distributor.

   g. Cylinder misfiring was monitored by means of a pressure transducer mounted on the exhaust pipe and coupled to a cathode ray oscilloscope which displayed exhaust pressure pulses in engine firing order.

2. Equivalence ratio was derived from airflow and fuel consumption measurements. For petrol, the mixture strength was adjusted by means of a variable main jet comprising a specially contoured needle penetrating the jet orifice (refer Fig 2). For LPG the primary adjustment was made to the power mixture valve. For fine adjustment of mixture strength and to provide additional fuel for attainment of very rich mixtures a subsidiary needle valve was mounted downstream of the LPG carburettor on the ducting between it and the petrol carburettor (refer Fig 2).

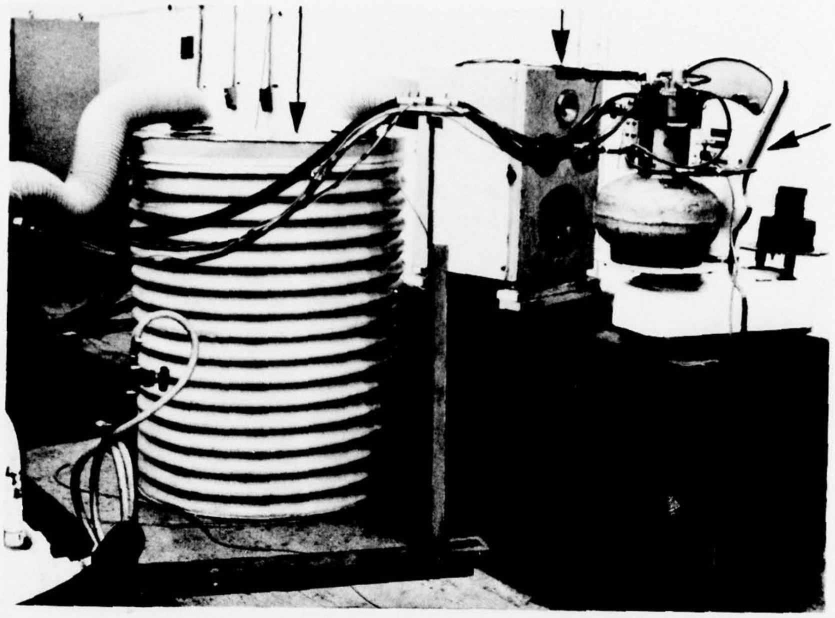


FIG 1 - AIRFLOW MEASURING APPARATUS AND LPG FUEL WEIGHING SYSTEM

FIG 2 - VARIABLE JET ADJUSTMENT (PETROL) AND SUBSIDIARY NEEDLE VALVE (LPG)

NEGATIVE NUMBERS

FIG 1. 6461E

FIG 2. 6461H