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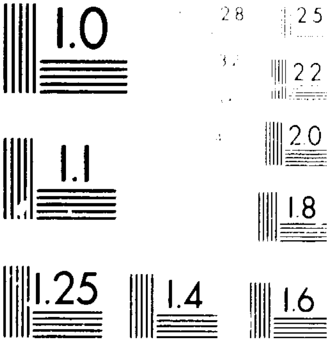

MICROCOPY RESOLUTION TEST CHART

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APPLICATION OF ALTERNATIVE FUELS FOR INTERNAL COMBUSTION ENGINES. IIP, DEHRA DUN.

DP/IND/82/001

INDIA

FINAL REPORT *

Prepared for the Government of India,
by the United Nations Industrial Development Organization,
acting as Executing Agency for the United Nations Development Programme

Based on the work of Prof. A. S. Katchian
Expert in Combustion Studies in C.I Engines
under the post 11-04

UNITED NATIONS INDUSTRIAL DEVELOPMENT ORGANIZATION

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from: H. Seidel

Mission time schedule

UNIDO Project "Application of Alternative Fuels for Internal Combustion Engines" (DP/IND/82/001) was started at Indian Institute of Petroleum, Dehra Dun in July 1982.

Expert's actual mission period was from 11.10.1984 to 20.3.1985. Period from 11.10.1984 to 22.10.1984 (8 working days) was spent for briefing at UNIDO Headquarters, Vienna, visits to Department for Internal Combustion Engines and Automotive Engineering, Technical University, Vienna, AVL, Graz, Austria, briefing at UNDP, New Delhi and also on travel Moscow-Vienna-New Delhi (via Zurich, Paris, Karachi without stop-over) - Dehra Dun. Work at duty-station started from 23.10.1984 and lasted till 14.03.1985 (inclusive). One day delay in the start of work at duty-station was due to 8 hours delay in Vayudoot flight from Delhi to Dehra Dun.

Two business trips were undertaken during the mission:

a) from 2.01.1985 to 7.01.1985 (4 working days) to Delhi.

b) from 6.01.1985 to 14.01.1985 (7 working days) - to Delhi, Bombay, Pune, Madras.

Work done during visits and business trips is dealt with in part 7 of the report.

One day sick leave was taken during the mission. At the end of December 1984 interim report was prepared by the expert. Copies of the report were made available to Project Director, Dr.I.B.Gulati, Project Coordinator Mr. S.Singhal, Project Leader, Dr.B.P.Pundir. The report was also handed over and discussed with UNIDO Chief Technical Field Adviser, Dr.M.Kamal Hussain during expert's visit to UNDP Delhi in the beginning of January 1985. Interim report is given in Appendix '4'.

Introduction

It follows from the above stated that expert's mission started after more than two years of the project progress. During this period the following activities were initiated at IIP in the lines of the expert's mission:

I. Diesel engine dual fuel operation with induction of fumigated methanol through inlet system along with air. This type of partial substitution of diesel fuel by alternative fuel is one of the simplest, practical and well investigated. In USSR, for instance, this appreach was extensively investigated with a number of alternative fuels (alcohol fuels inclusive) by Prof. A.D.Charomskiy in early 30-s. Since then great number of investigations and publications were accomplished. First publication in these lines was made by IIP Scientists in 1980 (4th Int. Symp. on Alcohol Fuels Application, Brasil). Control of fuel supply developed at IIP deserves positive appraisal. Within the scope of the UNIDO project work with methanol fumigation was continued on benches and also road tests were undertaken. Work is in progress to implement vast road trials on Ashok Leyland buses. Expert did not take part in these activities except discussions on the venue of road trials with Ashok Leyland executives during one day visit to Madras.

II. Glow plug ignited pure methanol approach was investigated by IIP scientists on Kirloskar Single Cylinder 5hp. diesel engine. Results of the investigations were presented at VI Int. Symp. on

Alcohol Fuels Application (Canada, 1984). This work deserves positive appraisal. Two locations of glow plug were tried. Also effects of fuel injection system parameters and level of electric power supply to glow plug on engine performance and stability of operation were studied at different operating conditions. Attempt has been made to measure glow plug temperature. Main findings of the work may be summurized in short as follows. Stable operation of the engine at full load conditions is possible with efficiency close to that of base diesel engine in case when electric power supply is 0.15 KW. Stability of operation and efficiency of engine is much poorer as compared to base fuel engine at light loads. Greater electric power supply is necessary for the same conditions to improve stability and efficiency. Work done had several shortcomings:

1. Dynamometer chosen was not quite suitable for 5 hp engine.
2. Fuel consumption measured on base diesel engine might be exagerated because of measurements carried out by volume, which is not advisable for diesel fuel consumption measurements.
3. No measurements of fuel injection rate and needle motion in case of methanol injection was attempted (except start of needle motion).
4. Fixing of thermocouple on glow plug was not the best possible.
5. Methanol was supplied to fuel injection pump by gravitation.

6. No optimization of swirl, local turbulence and position of spray with respect to glow plug was attempted. This last drawback is due to the fact that work was carried out without cooperation with industry. IIP workshop facilities are not suitable for design optimization work. At the same time in many cases research carried out without optimization of design may be only of academic interest.

Expert, however, is glad to stress that at prevailing conditions work done by Dr. B.P.Pundir and Mr. Dinesh Kumar should be complimented. Remarks made above are aimed to better future R&D work in direction under consideration.

During the mission expert took part in discussions of the results of experiments with glow plug ignited pure methanol engine and also in planning of future activities. Suggestions were made to change fuel feed system, dynamometer of the test bench, method of fuel consumption measurement. On request of project leader Dr. B.P.Pundir works aiming to increase charge temperature during compression, to attempt surface ignition of methanol, to decrease electric power supply to glow plug were incorporated into expert's programme (see points 9 and 12, appendix No.1). Corresponding information about expert's activities is given in part 5 of the report.

III. Research on fuel injection and atomization. Comparative studies of fuel injection and atomization with diesel fuel and methanol were undertaken. One of the IIP scientists -

Mr. A.K.Aigal was sent to England to study atomization research methods based on the use of Malvern instrument supplied in the course of previous UNESCO Project at IIP (1976-1981). Unfortunately no IIP report is available concerning the results of this injection and atomization studies. No publications were made. A lot of high level valuable information related to fuel injection and atomization is given in two comprehensive reports prepared by UNIDO Consultant Dr. Anton Cernej. However in these reports results of IIP research were not analysed, which may be considered as shortcoming in project organization. Because of the above stated, the expert had to analyse conditions of investigations and also primary materials obtained. As a result of such an analysis the following observations should be made.

1. Experiments were carried out with three out of four nozzle orifices clogged.

2. No attempt was made to model injection process with actual 4-hole nozzle in case when three out of four nozzle orifices were clogged. Hence injection process parameters with respect to time (camangle) in both the cases were quite different. Therefore results of fuel injection process investigation obtained may not be applied to actual Escort diesel engine fuel injection system or any other actual system.

3. Results of comparative fuel atomization investigation would be of interest if one could relate time and space resolved particle size measurements to instantaneous fuel injection rates. Thus generalized results may be then used to calculate particle sizes in case of any fuel injection process events. However derivation of quantitative relation between fuel injection rate and sizes of particles from the results of measurements carried out on the bench available in IIP Engine Laboratory is not possible as there is no reliable method to compute actual trajectories of the particles injected especially into cross-sectional gas stream which is the case on the bench used in the course of experiments.

4. In processing data on methanol injection, change in methanol cyclic delivery with speed was not taken into account.

5. Results of mean particle sizes measurements in case of diesel fuel do not coincide with results of direct measurements carried out by number of investigators.

Due to the above observations expert came to conclusion that comparative investigation of fuel injection process with diesel fuel and methanol had to be started anew. Later on based on these results atomization research may be attempted. In this latter research measurements with Malvern Instrument should be supplemented by direct particle size measurements. Actual expert activities hence were not limited to comparative analysis of fuel injection and atomization in case of diesel fuel and methanol on the basis of experiments carried out prior to expert's arrival to duty station, which was meant in point 11 of the programme (see Apendix 1). Scope of activities had to be enlarged. The results of the same are given in part 6 of the report. In general, after consideration of research and development work completed and in progress, at IIP, analysis of all possible ways to use methanol in diesel engines and a number of discussions with IIP scientists the programme of expert's work on the project was finalized (See Appendix 1). As already mentioned above, this programme has undergone some changes in the course of the mission, the changes being made to greater benefit of the project. As may be seen from the programme, mainly design work and training of IIP engineers were planned. In some cases expert tried to accomplish fabrication of designed parts and to aquire hardware so as to accelerate achievement of the aim of second project stage - tests on 4-stroke pure methanol engines, multy-cylinder engines inclusive. In expert's opinion it would had been possible to run this engine before completion of the mission provided better successiveness in different experts activities was ensured. However, still it is quite possible to achieve this goal in time specified by the project schedule provided cooperation with Ashok Leyland and Kirloskar Oil Engine Ltd. initiated by expert would materialise soon. Also number of full time working engineers in the lines of diesel engines activities should be increased. Prior to arrival of the expert only two full time engineers were associated with diesel research. During the mission number of such engineers varied from 2 to 4. Also project leader Dr. B.P.Pundir took part in the work. Number of discussions were held with Project Coordinator Mr. S. Singhal and Project Director Dr. I.B.Gulati.

Expert wishes to acknowledge fruitful cooperation with his colleagues - Mr. H.C.Wolff from Federal Republic of Germany and Dr. S.Radzimirski from Polish People Republic, both Experts on the UNIDO project - DP/IND/82/001.

Part 1. <u>Current and future Research facilities in the lines of expert's mission. Methods of diesel engine tests</u>.

By the time of expert's arrival to duty-station, no engine bench research was going on at Engine Laboratory related to the use of methanol in diesel engines. No single cylinder research diesel engine was available either. Ricardo single-cylinder research diesel engine test bench purchased from UNDP/UNIDO funds was supplied to IIP, Dehradun on 10 th December of 1984 . No commissioning of the bench took place till the end of expert's mission.

The main reasons of the delay in commissioning of the bench are as follows:

1. The bench was supplied by the firm (Cussons Ltd., England) without manuals. The manuals were received at IIP only in mid-January. Only after getting the manuals assembling of the bench might be completed.

2. The most important version of Ricardo diesel engine designed for methanol research and provided by two fuel injection pumps - that is direct injection unsupercharged version, could not been started even on diesel fuel as it had low compression ratio equal to 12.5. Two other versions - spark ignition version and indirect injection diesel engine version are not quite suitable for methanol research. In spark ignition version compression ratio is equal to 9, which is below the value at which high anti-knock ability of methanol may be

utilized. Application of indirect injection diesel approach to small size high speed engines operating on methanol is not advisable as with high anti-knock ability of methanol optimized spark ignition version may have higher efficiency than optimized indirect injection diesel version in rather wide range of operating conditions. In addition to the main reasons due to which commissioning of Ricardo bench was delayed the following observations are of importance:

1. Dimensions of Ricardo engine: diameter D=80,26 mm, stroke S = 88, 9 mm are rather far from dimensions of truck, bus and tractor type diesel engines, which are major consumers of diesel fuel in India. Dimensions and some other parameters of Tata, Ashok Leyland, Escort and Eicher diesel engines are given in Table 1.

Table 1

Model	Application (main)	D, mm	S, mm	ε	n_r rpm	Type of combustion chamber
Tata	trucks and buses	92	120	17	2800	open
Ashok-Leyland	trucks and buses	103.38	120,65	16	2400	open
Escort	tractors	110	120	17	2000	open
Eicher	tractors	115	150	17.7	1650	open

Basic design of Ricardo engine is not suitable for the modifications aiming on the change of cylinder bore and especially piston stroke.

At the same time, it is well known that results of experiments on single cylinder engines may be applied with certain reservations (and provided certain preconditions were set up and followed) to multi cylinder engines only in case when major engine sizes are the same in both the cases. Therefore Ricardo engine can not be used for R&D work on engines, which are major consumers of diesel fuel in India. At the same time, there is considerable scope of performance improvement in diesel engines, especially those of truck and bus models.

2. Spark ignition version of Ricardo engine can not be considered as research engine as no alternative parts are provided to carry out R&D work. According to performance curves obtained from the firm, indicated efficiency of gasoline version of the engine at mean engine speed is sufficiently high (40%). However at high speed range ($n > 3600$ r.p.m.) efficiency undergoes sharp decrease which should not be the case if performance is optimized for the whole speed range, test bench is properly designed and assembled and measurements are accurate.

3. Indirect injection diesel engine also can not be considered as research engine due to the same reasons as given above in p.2. Brake efficiency of indirect injection diesel engine version is lower than that of gasoline version in the whole speed range and full load conditions which should not be the case if both the engines are equally well optimized.

4. Performance of unsupercharged direct injection diesel engine version is not given as it could not be run at all.

5. Performance of supercharged diesel engine working on diesel fuel shows sharp deterioration with increasing speed. Supercharging pressure is too high for such a small size engine (sizes being close to those of car diesel engines).It is evident that no proper efforts were made by the supplier to optimize working process of direct injection supercharged diesel engine working on diesel fuel. Even if such a work is done, it will be difficult to find any reasonable application for such an R&D work as no small size diesel engine is having pressure of supercharging equal to 3 bars. From the comments made above it is clear that rather well-known consulting firm Ricardo and Co. supplied to IIP engine which can not be used for research on the UNIDO Project. In expert's opinion strong actions should be taken against the firm which was not willing to cooperate in correcting the mistake done. Meanwhile to utilise at least part of the equipment supplied the engine should be changed. There are several options.

1. At IIT, Bombay single-cylinder research diesel engine was designed and partially fabricated. This engine is in many respects more advanced than the engines available in the world market - such as Ricardo and AVL engines. Therefore it is advisable to undertake actions for its fabrication. This engine may be used for research on application of alcohol fuels to Indian make direct injection diesel engines. The features of the engine designed at IIT, Bombay which make it more advanced than Ricardo and AVL diesel engines are:

a) Option to use unit-injector or conventional fuel injection system.

b) Possibility to change independently timings of: inlet valves, exhaust valves, injection start.

As compared to Ricardo engine additional advantage is, possibility to adapt the research engine to different bore and stroke in the range of Indian make truck, bus and tractor type diesel engine sizes. Fabrication of the engine after certain revions of its design may be undertaken with the help of Prototype Development & Training Centre at Rajkot or any other proper enterprise. The expert initiated activities in this direction before the end of mission.

2. Ricardo test bench may be used to carry out research on Indian make single cylinder diesel engines - such as Kirloskar AV-57. Kirloskar Oil Engine Ltd. agreed to supply this advanced diesel engine to IIP free of cost during the expert's visit to Pune.

3. As a result of visit to AVL Graz expert came to conclusion that specification of Universal Single-cylinder Experimental Diesel Engine should be prepared by IIP and AVL should be approached to find out the cost of such an unit and period required for the supply. The engine should be supplied with the parts necessary to simulate operation of Indian make truck and bus diesel engines and also to

certain extent future requirements of the country's transportation system and agriculture. Similar recommendation was made by expert at the end of his mission as UNESCO expert in 1981-1982 and also in interim report on the current mission. Choice between options 1 . for the use on UNIDO project should depend on cost and time factors. However, expert is of the opinion that option 1 should be materialized in any case as this may benefit not only current UNIDO project at IIP but also future projects and in general diesel engine research activies in India, which are of a importance. Also hard currency may be saved.

Option 2 should be used immediately as Ricardo test bench was supplied to IIP more than 3 months back and is not still used to fulfill the project aims. Due to drawbacks in supply stressed above, the results of work done according to p.9 of the programme (Appendix 1) may not be directly used to the benefit of the project aims. Though design developed at IIP during expert's mission (see part 5 of the report) can not be tested on Ricardo diesel engine, experience gained in designing may be of use for tests on the other engines - such as Kirloskar AV-57 or similar. Regarding methods of diesel engine tests used in Engine Laboratory the following observations were made by expert:

1. Performance parameters obtained on the same model of engine in different tests vary from each other by more than 10%. Even character of change of engine parameters - such as for instance specific fuel consumption is different in different test runs.

Parameters obtained at IIP Engine Laboratory are in general worse than those claimed by manufacturers and also worse than parameters which one could expect to get on engines of particular type. For instance in tests on Tata (model 692DI) open type combustion chamber diesel engines fuel consumption measured was in the range from 208 to 227 gms/h.p.hr., whereas minimum fuel consumption at full load was in the range from 195 gms/h.p.hr. (n = 2400 r.p.m.) to 202 (n = 2000 r.p.m.). In some other run minimum fuel consumption was observed at 1000 rpm. The same magnitude of variation was obtained on Ashok-Leyland (model ALU-370) open type combustion chamber diesel engine. Very low excess of air (down to 19%), high exhaust temperatures (up to 780°C) and high soot emission (up to 8.5 Bosch units) were measured on direct injection truck diesel engines. In no case power output claimed by manufacturers of truck and bus diesel engines was obtained at IIP after correction of parameters according to Indian standard conditions.

Below the reasons for the stated above are discussed. There is no doubt that most of Indian make truck and bus diesel engines are overfueled with a result of poor efficiency, high smoke and high thermal loading. This may be one of the reasons of fast deterioration in engine parameters, for instance, as a result of nozzle orifice cocking, increased blow-by etc.

In the region of low air excess diesel engines are generally very sensitive to any change in engine conditions and also in atmospheric conditions. Dehra Dun is located at the altitude of ~600 m. Atmospheric temperature in summer time may be higher than 40°C. Under these conditions to get accurate results in performance tests is rather difficult. At the same time accurate performance test is precondition to any other tests, tests on application of alcohol fuels, inclusive. The most reliable way to find out if engine assembly, conditions of all engine parts, timings are proper is to carry out accurate performance tests and then to compare results of these tests with standard performances of particular engines. Unfortunately standard performances of most of truck and bus diesel engines are not available at IIP Engine Laboratory. Expert is of opinion that Indian make diesel engine standard performances should be made available to IIP Scientists. Before start of any performance test special procedure should be followed to ensure proper conditions of the engine. This procedure should include:

1. Check of all the nozzles, check of effective area of nozzle orifices, inclusive. For this purpose constant flow (pressure) bench should be designed and fabricated.

2. Calibration of fuel injection pump with the injectors of the particular engine. Taking into account above said about low excess of air at full load conditions, sharp and unequal

rate of change in indicated efficiency with air-fuel ratio in different engines, expert came to conclusion that diesel engine tests at IIP Engines laboratory should be carried out at conditions $\alpha = \alpha_s$. Here α_s is relative air fuel ratio (excess air coefficient) at standard conditions. Hence cyclic delivery of the pump should be properly adjusted. Then equations (8) and (9) from the lecture on correction of diesel engine parameters for standard conditions delivered in 1981 and made available to IIP Scientists in writing should be used to correct measured power output and efficiency for unsupercharged diesel engines. Mechanical efficiency necessary to calculate correction factors may be determined either by motoring or by William's line method. There is correction procedure for supercharged diesel engines in the same lecture.

3. Check of conditions and proper setting of valves, its seats and timings.
4. Check of nozzle tip reach.
5. Check of friction losses.
6. Check of rate of blow-by and maximum compression pressure.

Instruments used for measurements should ensure inaccuracy not more than: 0.5% for torque and fuel consumption, ± 1 rpm for speed, $\pm 5^{\circ}$ for exhaust temperature measurement, $\pm 1\%$ for air consumption measurement. Change in temperature of cooling water and lubricating oil during tests should not be more than $\pm 2^{\circ}$. Exhaust back pressure for diesel engines on test bench should not be more than 0.03 bar. Periodic calibration of all instruments used should be made with the help of specially kept sophisticated equipment available from instrumentation section of the Laboratory. Only after getting performance parameters close to the standard ones research on alcohol fuel application may be attempted. During the tests it is advisable to return periodically to certain control conditions (say rated condition and low speed high load condition), to make measurements of parameters and to compare the parameters with those obtained at the start of corresponding research programme. This may help in checking if considerable change in engine conditions took place in the course of research programme. If there is no requirements specified by manufacturer or following from the aims of the test, it is advisable to run diesel engine on bench with disconnected fan without air cleaner and exhaust muffler. These conditions should be stressed in any technical report along with atmospheric conditions. In conclusion it is necessary to stress that experimental research work on diesel engines has meaning only in cases when proper conditions of engines are ensured during test programme and if necessary accuracy and repeatability of measurements are obtained. In many cases it is advisably to go for multi-factorial approach and mathematical planning of experiments. In special cases even on single-factorial experiments statistical approach should be used to get reliable test data.

Two more suggestions should help to increase diesel research standard at IIP Engine Laboratory.

1. To organize short term course in test and calibration procedures and methods. It is worth noting that number of text books and monographies in these lines are available in Russian. Therefore now days it is essential to have at least few engineers who know technical Russian language.

2. To send two-three engineers for a period of one to two years to AVL, Graz, Austria or another well established engine R&D firm or laboratory for detailed training. On 25th February, 1985 meeting of leading Engine Laboratory Scientists was organized by project coordinator, Mr. S.Singhal, on request of the expert to discuss some of the matters related to the further increase in standard of research activities at IIP Engine Laboratory. Except the expert and the project coordinator Dipl.Eng. H.C.Wolff and Dr S.Radzimirski, both UNIDO Experts on the project, took part and contributed to the discussions. Expert hopes that the discussions held would be of help in future research activities.

Part 2. Promoting ideas related to main aims of the Project and Training Activities:

Main aim of the current UNIDO project is to save diesel fuel by partial or full substitution of it with methanol. In expert's opinion any other activities to improve diesel engine efficiency and other technical parameters also serve to achieve the project aims. Workshop "Perspective of alcohol fuel utilization in IC Engines" organized by IIP at the end of October 1984 with participation of Government Officials, scientists from a number of Universities, colleges, institutes and some specialists from industry played positive role in promoting ideas of alcohol utilization in piston engines. Expert took part in the workshop, made presentation "Utilization of methanol in diesel engines currently under production ", chaired one of the workshop sessions and made suggestions regarding most practical ways to use pure methanol in diesel engines at concluding session of the workshop. Later on presentation was elaborated and prepared in written form (see Appendix 2). In the presentation comparative analysis of properties of diesel fuel and methanol was given, especially of those affecting engine performance parameters. Classification of all methods to use methanol in diesel engines was also presented. Greatest attention was paid to methods to use pure methanol in diesel engines currently under production. As a result of detailed analysis of these methods the expert came to conclusion that the following methods seem to be most practical ones at the current stage:

1. Conversion of diesel engine into external mixing spark ignition engine.

2. Charge stratification inside combustion chamber and spark ignition of the charge.

Use of methanol with ignition improver is attractive in the sense that no major changes in diesel engine design are required. Unfortunately cost of fuel with ignition improver still is too high. Therefore work should be continued by experts in chemistry and corresponding branch of chemical engineering to increase efficiency and decrease cost of the ignition improvers. Two other methods - use of glow plug to ignite methanol injected into combustion chamber and use of hot surfaces to ignite methanol should be further investigated as their practicality is not yet clear. More research and development work is needed to find out if these methods may compete with those recommended above. It was stressed that in application of the methods in which heated surfaces are used to ignite methanol it is necessary to avoid high consumption of electrical energy and do not interfere in optimized mode of air-fuel mixture formation (which is the case in the method suggested at IIT, Madras - reference 22 in Appendix 2). On 11th February 1985 2-hour lecture "Utilization of methanol in diesel engines" was delivered for staff members and research scholars at IIT, Bombay. In this lecture all the methods to use methanol in diesel engines (partial substitution methods inclusive) were analysed.

In general review of publications made, reports of UNIDO experts (A. Kowalewicz, A. Cernej, P.Eyzat), report of Ad-hoc Expert Group Meeting on Modification of Internal Combustion Engines for Utilization of Synthetic Fuels, discussions with Prof. H.P.Lenz during visit to Department for Internal Combustion Engines and Automotive Engineering, Technical University, Vienna and also discussions held with IIP Engine Laboratory scientists helped the expert to finalize programme of activity on the project. While doing so it was taken into account that during final stage of the project pure methanol 4-stroke engine should be tested.

Taking into account necessity to work on saving diesel fuel, expert prepared lecture "Modern trends in diesel engine development" (Appendix 3). This 2-hour lecture was delivered to staff members and research scholars at IIT, Bombay on 8th of February 1985. Special emphasis was made on those ways to improve diesel engines parameters which are not still utilized in vast scale in India. In addition to training through presentation at workshop and lecturing, training of IIP Engineers and Scientists (B.P.Pundir, Dinesh Kumar, A.K. Aigal, S.K.Singhal, S.Das) was accomplished by individual and group discussions in the lines of the expert's activities (see Appendix 1) and also on engine cycle modelling, ways to improve diesel engine parameters, ways to increase standard of research activities at IIP Engine Laboratory and also in the lines of development and performance improvement of diesel engines. Some suggestions regarding detailed training of Engine Laboratory personnel were also made in the previous part of the report. It is necessary to stress in addition that study tours and short-term fellowship (mostly at UK Universities) provided to IIP scientists according to the project documents, though of substantial use, cannot serve the same purpose as one to two years detailed training in well established R&D laboratory or firm. This is true because of educational system prevailing in India. According to expert's knowledge obtained during expert's tour standard of R&D work at some of Indian firms is sufficiently high. Therefore if it is difficult to organize detailed one-two years training of IIP engineers abroad this may be possible within the country itself. This kind of training may fill gap between engineering educational system and requirements of high level R&D activity. Also certain kind of technical books with detailed information about design, engineering methods of stress analysis, methods to assess efficiency of all diesel engine systems, test methods may be of great use for training purpose. This kind of books are not always available in English as the firms in USA, UK and also in some other countries are not prepared to share their R&D experience with outsiders. At the same time, the type of technical literature under discussion is available in Russian. This again proves necessity to have at least few engineers in command of Russian language.

At the end of previous expert's mission at IIP, Dehradun a book on development of truck diesel engines was translated by Dr. H.C.Dhariwal from IIT Bombay and edited by the expert who happened to be one of the authors of the book. Unfortunately the book is not yet made available to specialists in the field of diesel engines research and development, IIP scientists inclusive. During the mission actions were taken to make the book available to few concerned specialists.

Part 3: Comparative analysis of engine cycles and parameters with diesel fuel and methanol

As stressed above, two ways to use pure methanol are most practical for diesel engines currently under production. Charge stratification inside combustion chamber and spark ignition of the charge was successfully used in application of methanol to diesel engines by MAN (FRG)-FM process (details are given in Appendix 2). In this case fuel is injected on combustion chamber walls (so called "wall wetting" principle). It is supposed that in this case for heating and evaporating of fuel at least partially heat otherwise wasted to cooling medium is used. This may be very essential in case of methanol because latent heat of methanol as applied to the same amount of heat released during combustion is 10 times more than that of diesel fuel. At the same time there are other methods - such as dual fuel injection into combustion chamber volume or glow-plug ignition of fuel injected into combustion chamber volume in which most of the heat spent to heat and evaporate methanol is taken from air charge itself. It is of substantial interest to find out effect of these differences in source of fuel evaporation heat on cycle and engine parameters.

Also according to comparison of other properties in case of diesel fuel and methanol (Appendix 2) there are other substantial differences - such as differences in extent of change in number of mole due to combustion and also in composition and specific heat of combustion products.

The second practical way to use pure methanol in diesel engines currently under production is, as mentioned above, conversion of diesel engines into external mixing spark ignition engines. In this case complete evaporation of methanol may be achieved in special evaporator (Mercedes-Benz approach). Introduction of vapourized methanol through inlet system decreases amount of air charge sucked into cylinder. Change in number of mols due to combustion of methanol is also quite different in case of external mixing as compared with internal mixing.
Also compression ratio of engine should be decreased in case of external mixing and spark ignition as compared to all modes in diesel operation. Content of methanol vapour in fresh charge affects comparession work.
To get the same power output as in diesel operation air fuel ratio in spark ignition external mixing version of engine should be decreased. Hence specific heat and adiabatic exponent will change accordingly. This will also affect cycle and engine parameters.

To find out the effects outlined above set of calculations were carried out by the expert.

Aims of the calculations:

1. To find out effects on cycle parameters of differences in the properties of methanol and diesel fuel-such as, ratio of number of mols prior, in the progress and at the end of combustion, fraction of three-atom gases in the charge and hence its specific heat, evaporation heat.

2. To assess the change in diesel engine parameters as a result of its conversion into spark-ignition external mixing engine. Conditions and details of study are given in Appendix 5. Four cycles were considered : one with diesel fuels and three with methanol. Three of the cycles considered are related to internal mixing, one to external mixing. Two methanol internal mixing cycles differed by the accounting for heat taken from charge to heat and evaporate methanol. Main findings may be summarized as follows:

1. In case of internal mixing and fuel evaporating due to heat supplied from the charge methanol cycle is inferior as compared to diesel fuel cycle. Inferiority is much higher in terms of efficiency than in terms of mean cycle pressure because as applied to internal mixing mode calorific value of air-methanol mixture is higher than that of air-diesel fuel mixture. The results obtained show that negative effects of greater specific heat and latent heat in case of methanol prevail over positive effect of greater change in number of mols due to combustion.

The fact that in reality even in engines in which fuel is atomized in combustion chamber volume (and hence is evaporated mostly due to heat taken from the charge itself) when compared with results of calculations proves convincingly that more efficient air-fuel mixing and combustion may be obtained with methanol than with diesel fuel.

2. In case of internal mixing and fuel evaporation due to otherwise wasted heat methanol cycle is slightly superior as compared to diesel fuel cycle in terms of efficiency and

considerably superior in terms of mean cycle pressure. Therefore modes of air fuel mixing in which methanol is directed to combustion chamber walls should be prefered (such as MAN-FM process).

3. One of the promising ways to use pure methanol in diesel engines currently under production is conversion of the same into spark ignition engines with external mixture formation and evaporation of methanol in special evaporator utilizing part of the heat rejected into cooling medium (Daimler-Benz approach) or with exhaust gases.

Cycle of such an engine is more efficient than that of with internal mixing and fuel evaporation due to heat taken from the charge in spite of lower values of compression ratio, excess of air and also mols ratio.

In addition to the effect of regeneration (use of otherwise wasted heat to evaporate methanol) decrease in compression work not only due to lower compression ratio but also owing to presence of methanol vapours having high specific heat plays positive role in getting high cycle parameters.

4. To get mean cycle pressure in spark ignition external fuel air mixing methanol engine the same as in base diesel engine it is sufficient to decrease excess of air by 15%. The main reasons due to which excess of air should be decreased to get the same power output are decrease in efficiency and in amount of air charge inducted into cylinder per cycle, effect of the latter being greater. Taking into account that air fuel ratios of Tata and Ashok Leyland diesel engines

at rated conditions are rather low, one comes to conclusion that methanol versions of these engines at rated conditions should be supplied by mixture close to stoichiometric one.

More detailed analysis of different effects is given in Appendix 5.

Results of calculations and analyais made cover points 5 and 6 of the programme (Appendix 1). In addition to this expert carried out complete derivation of set of equations for computer modelling on assumption of charge consisting of two parts: a) mixture of air with residual gases; b) equilibrium combustion product for stoichiometric air-fuel ratio (two zone model).

While performing calculations based on this model it is possible to increase accuracy of cycle parameters comparison due to accounting for heat exchange and also that of dissociation. According to expert's experience no change in main findings of calculations considered in short above may be anticipated. Still modelling of this kind, that is consideration of certain extreme cycles (say with ideal air fuel mixing, equilibrium combustion products, with or without heat exchange, etc) are of great interest as they make it possible to differenciate effects of different factors and hence to understand deeply experimental results. Also calculations of the kind outlined above help in finding out the best practical solutions of certain problems and proper planning of experimental research activities. This may be proved by the results of calculations described in this part of the report. On the basis of calculations it was possible to come to conclusions that wall wetting in case of internal mixing and also conversion of diesel engine into high compression ratio (ε =12) external mixing engine with complete evaporation of methanol in evaporator working on otherwise wasted heat may be benefitial.

In direct contrast to the result of consideration of extreme cycles, experience gained during last 20 years shows that attempts to model actual air fuel mixing and combustion so far proved to be practically useless. In spite of the fact that dozens of models (some of them very complicated and computer time consuming) were developed in different countries, no model could stand check by comparing results of calculations with basic experimental findings. While comparing results of cycle calculations with few experimental indicator diagrams all the investigators claim good accuracy, though their approaches and initial assumptions are quite different,sometime opposite. This "accuracy" is obtained with the help of adjusting coefficients. The main reasons of uselessness of the models aiming to simulate actual air-fuel mixing and combustion in diesel engines suggested so far are:

1. Lack in proper understanding of governing physical and chemical factors, especially lack in understanding of basic laws of turbulent heterogeneous combustion.
2. Extremely complicated nature of air-fuel mixing and combustion processes in diesel engines.

3. Great variety of design versions of diesel engines which is impossible to describe by some universal (inevitably simplified) approach.

Modelling is fruitful as applied to some of the processes and diesel engine systems for which basic physical understanding is available:

1. Fuel injection process
2. Gas exchange process
3. Processes in turbocharging systems
4. Governing of engine
5. Stress and strength analysis taking into account mechanical and thermal leading.

Expert hopes that the comments made may help Engines Laboratory scientists to find most fruitful lines in devoting their time and energy. There is great scope of improvement of diesel engine parameters in close cooperation with industry. Not all R&D centres in country are still sufficiently experienced. IIP Engine Laboratory with its good experimental facilities may benefit R&D work in diesel engine industry to a great extent.

Part 4. : Designing of Methanol Version of diesel truck and bus engines

As truck and bus engines are one of major consumers of diesel fuel in India, expert felt that at final stage of project more efforts should be made to substitute diesel fuel for methanol in these engines. Almost all previous efforts at IIP Dehra Dun, IIT Delhi, IIT Madras, IIT Bombay and other institutions were dedicated to small size, small power output diesel engines. This opinion was also expressed by the expert in interim report (Appendix 4).

Out of two options mentioned above the expert choose conversion of diesel engine into external mixing spark ignition engine. In this work experience obtained by the expert while converting diesel engines into spark ignition gas engines could be utilized- References 16,17, Appendix 2. Other reasons for this choice are given below.

Three options of converted diesel engine into external mixing spark ignition engine may be considered:

1. Engine with carburated methanol.
2. Engine with central body methanol injection having mechanical or microprocessor control.
3. Engine with special evaporator and gas-air mixer.

In cases 1 and 2 spiral type evaporator may be installed between carburator (central body fuel injection device)and engine inlet.

In spite of personal involvement in the matter of acquiring necessary carburator and central body methanol injection system, no hardware to implement two first options could be obtained (Bosh Co. at FRG, Mico Bosh office at Delhi and other firms at Delhi were approached). At the same time the equipment required (at least in gasoline version) is definitely available in the market. Therefore this equipment could had been purchased in the Austria through UNIDO, Vienna. Also spark ignition system suitable for the use of high compression ratios could had been obtained. Then the equipment could had been modified for methanol application. The main reason why stratified charge internal mixing forced ignition approach (MAN-FM process) was not chosen for experimental investigation was related to facilities available at IIP. Work on methanol application to diesel engines was going on at IIP without much of actual interaction with industry. At the same time MAN-FM process application requires major changes in engine design, which cannot be accomplished with the help of IIP workshop.

During second business tour the expert came to conclusion that it was quite possible to organise cooperation with big and experienced firms- such as Kirloskar Oil Ltd and Ashok Leyland. It is worthwhile to mention that only TELCO R&D Director K.G.K. Rao declined to meet the expert during his visit to Pune and discuss the matter of converting Tata truck diesel engine into methanol fueled engine. Strange as it is, he happens to be the only representative of industry in advisory committee of the UNIDO project on methanol application. (See Report on results of business trip to Delhi, Bombay, Pune, Madras- Appendix 6). This should be taken into account in further work on the project. To implement two first options the expert made necessary calculations and found main dimensions of several types of evaporators extracting heat from exhaust gases and from cooling water, spiral type evaporator working on heat of cooling water inclusive. Sketch of the last evaporator was prepared by the expert. Then drawings of evaporator were finalized by the drawing section. In fig. 1 overall views of the evaporator are given. Unfortunately while implementing the design electric heating element was not incorporated into warming jacket of evaporator as suggested by the expert. This heating element may be used during engine start and warm up. After obtaining carburator and central body methanol injection system tests should be carried out with and without evaporator to find out performance parameters and also to investigate lubrication and wear in both the cases. It may turn out beneficial to redesign inlet manifold while implementing two first options.

Third option could be materialized with the facilities available at IIP and hardware available in India (except high compression ratio spark ignition system and

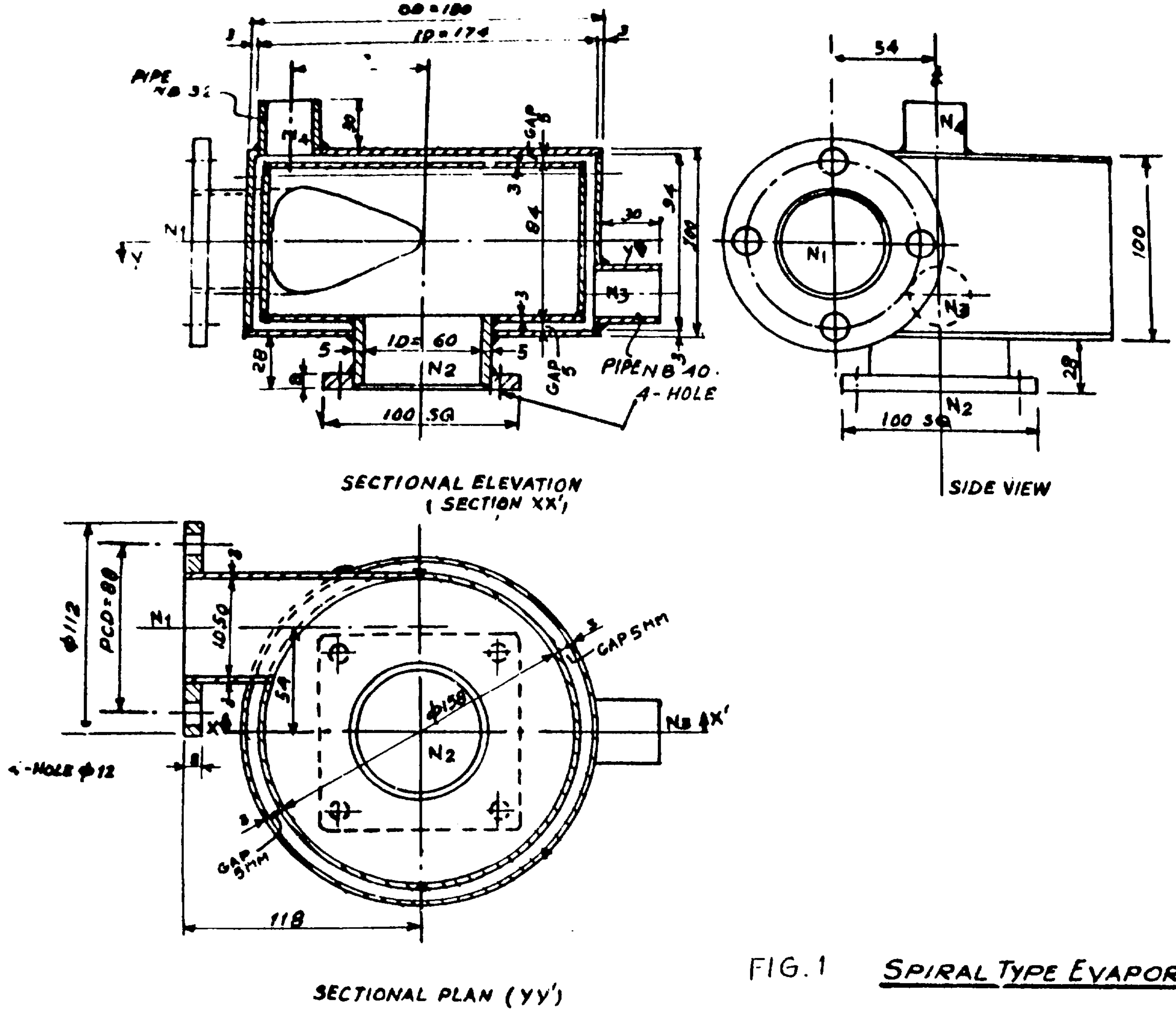

FIG.1 SPIRAL TYPE EVAPORATOR

precision gas pressure regulator). During first business trip to Delhi expert visited different firms to find out hardware. As a result of the trip materials for manufacturing tube type evaporator were procured and also quotations for ignition system of 6-cylinder in-line engine were obtained. Later on the system was procured.

As this system is not suitable for high compression ratio operation it will be necessary in the course of first trials to decrease compression ratio of methanol version truck engine down to 8.5 - 9 to avoid pre-ignition by putting additional gaskets between cylinder block and cylinder head. On acquiring necessary ignition system gaskets may be removed to make greatest use of high octane rating of methanol. On acquiring materials it became possible to finalize dimentions of the multi-tube evaporator. Calculations of heat transfer and rate of methanol evaporation were performed by the expert during first month of the mission. These calculations made it possible to find out necessary surface of heat exchange and hence to design the evaporator. Also simple strength analysis was performed. All the dimensions and sketch of the evaporator were then handed over to drawing section to finalize the drawings. In fig. 2, overall view of the evaporator is shown. After fabrication it is essential to check if evaporator is leak-proof before its test. Fig. 3 illustrates lay out of the methanol version of Tata and Ashok Leyland truck engines. Engine with minor modifications

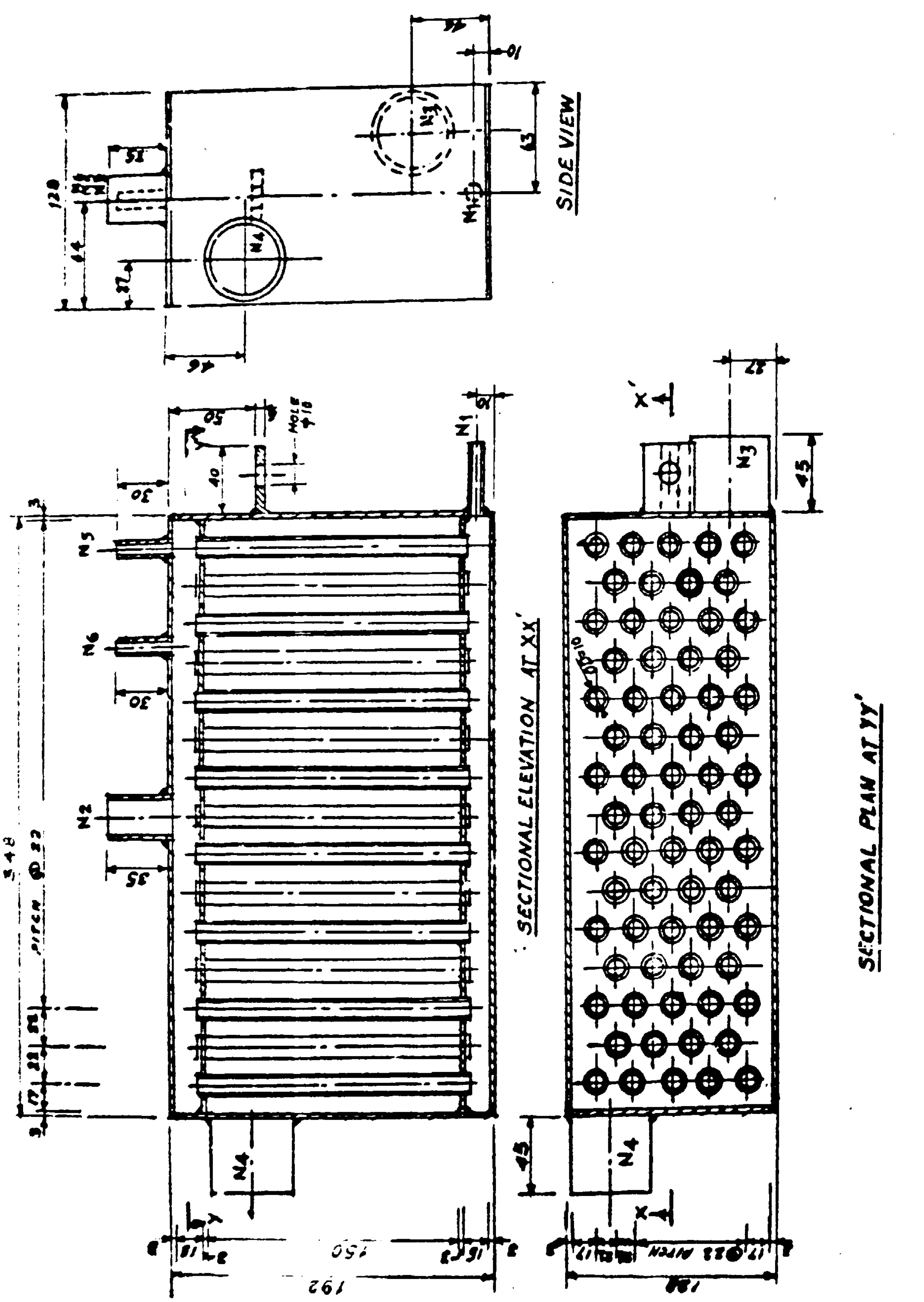

FIG 2. MULTI TUBE EXCHANGER TYPE EVAPORATOR

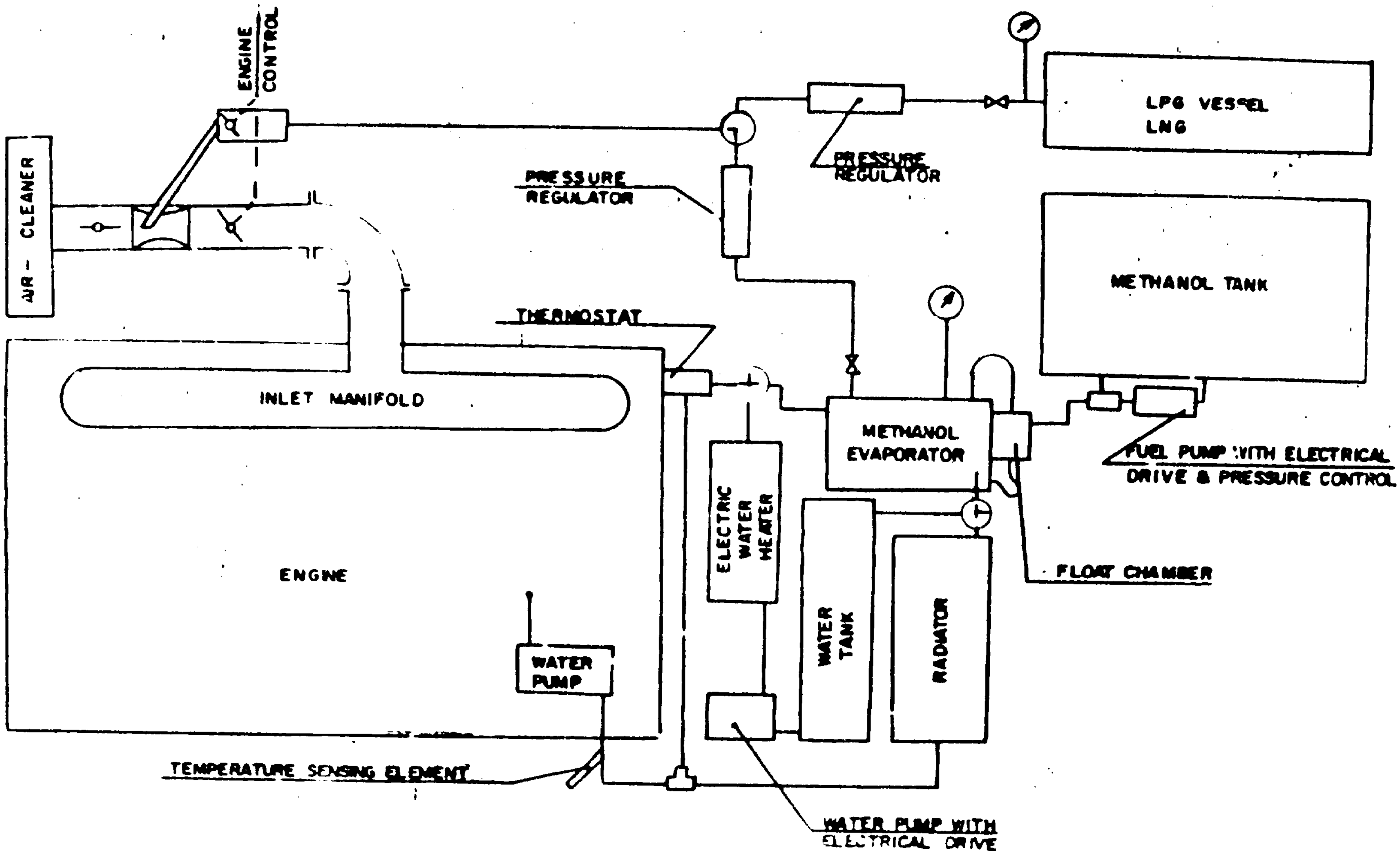

FIG 3 LAY-OUT OF METHANOL - LPG - LNG TEST ENGINE

may be fueled by M 100, LPG, LNG or CNG. The most complicated design is needed in case of M 100 as substantial amount of heat supply should be ensured to completely vaporize fuel. Maximum heat power output of evaporator in this case should be equal approximately to 14 KW. After engine is heated necessary heat supply rate may be ensured by water circulating through engine cooling system. The evaporator is connected in sequence to engine cooling system. For starting and warming up special circuit with electrically driven water pump and water heater is provided. For this purpose due to evident reasons power of heater may be considerably lower than 14 KW. Also heat of exhaust gases may be used for methanol evaporation during warm up. This additional exhaust gas heat exchanger used during warm up is not shown in Fig.3. During bench tests control of the engine may be ensured by changing resistance in gas supply line and by mixture throttling. Thus optimum change in mixture strength at all operating conditions may be determined during bench tests. Then automatic control system should be designed. According to expert's experience it would be possible to get change in mixture strength with operating conditions close to optimum one by change in position of throttle if special precision gas pressure regulator is used, which keeps outlet pressure of gaseous fuel in narrow limits. By changing spring forces and valve areas of the regulator it will be possible to adjust nature of change in mixture strength to the desired one. Throttling device should be used to adjust maximum gas flow to particular engine. Also approach used by Mercedes-Benz may be tried -Ref.18, Appendix 2.

Changes in dimensions of combustion chamber to decrease compression ratio down to 12 were found by the expert.

Estimated "dead volume" in Tata diesel engine is equal to $Vd = 8.58\ cm^3$, volume of combustion chamber to $41,28\ cm^3$ and total clearance volume to $48,86\ cm^3$. Estimations were made on the basis of measurements of dimensions of combustion chamber for compression ratio equal to 17, as no drawingsof the engine were available.

In fig. 4 sketch of combustion chambers of diesel engine and methanol engine is shown. Cylindrical combustion chamber with diameter Dc.ch = 63 mm and depth Hc.ch = 20 mm will ensure compression ratio $\varepsilon = 11,8$.

New dead volume Vd' may be found as

$$Vd' = Vd - \frac{\pi}{4}(Dc.ch\ d^2 - Dc.ch\ m)\,\delta_{cl} = 8.58 - \frac{3,14}{4}(6.3^2 - 6^2)0,1 = 8.29\ cm^3$$

Above "d" for diesel, "m" for methanol engines, δ_{cl}-clearance between piston top and surface of cylinder head assumed to be equal to 1 mm. Volume of methanol engine combustion chamber

$$Vc.ch = \frac{\pi Dc.ch.m^2}{2}(Hc.ch + \delta cl) = \frac{3.14 \cdot 6.3^2}{4} 2,1 = 65.46\ cm^3$$

Then total clearance volume $Vc = Vc.ch + Vd' = 73,75\ cm^3$

Compression ratio

$$\varepsilon = \frac{Vh + Vc}{Vc}\ ;\ Vh = \frac{\pi\ Dcyl^2}{4}.\ Sp = \frac{3.14 \cdot 9.2^2}{4}.12 = 797,71\ cm^3$$

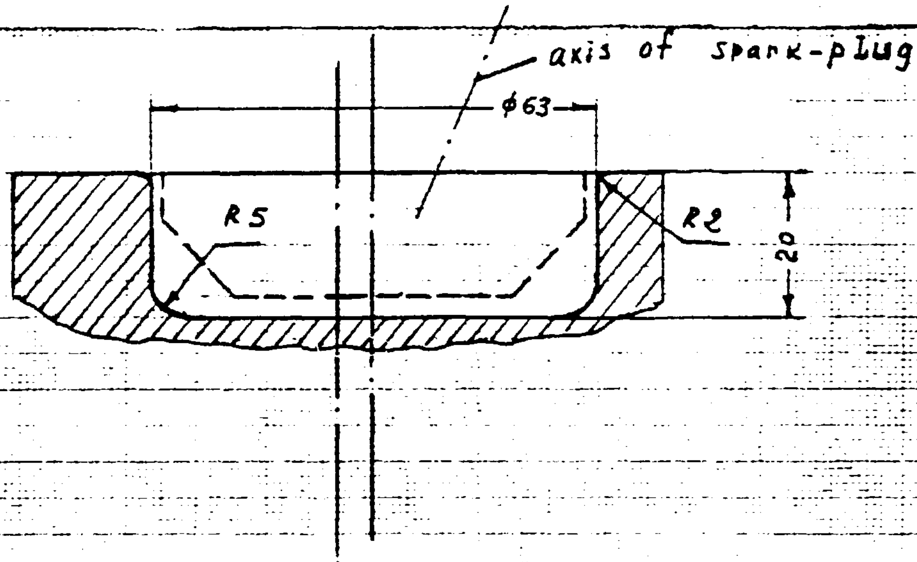

Fig. 4 "Combustion Chamber"

Above: Vh is swept volume, Dcyl- cylinder diameter, Sp - piston stroke $\varepsilon = \frac{73,75 + 797,71}{73,75} = 11,8$

In case if Hc.ch = 19 mm $\varepsilon = 12,3$.

In case if thickness of the gasket is equal to 11 mm three additional gaskets will decrease compression ratio from 11.8 down to 9.17. Of course with the gaskets shape of combustion chamber will be not favourably, also turbulence intensity will decrease. So one cannot expect to obtain good performance parameters. Still in expert's opinion start of work should not be further delayed. For first trials this approach may be justified. As it was impossible to purchase special electrically driven feed pump for the supply of M 100 from tank to evaporator under pressure greater than 2 bars, as temporary solution plunger feed pump installed on high pressure diesel engine pump may be used. With plungers removed to drive the cam shaft and feed pump small high speed electrical motor may be used for the purpose.

Tests of Tata diesel engine at conditions $\alpha = \alpha_s$ were performed before changing over to methanol version under expert's supervision. Correction of power and specific fuel consumption by equations suggested in the expert's lecture delivered in 1981 and available at IIP was applied. In table 2 results of different tests of Tata model 6-- diesel engine carried out at IIP Engines Laboratory are presented along with last test results. It is evident that in the last tests better fuel consumption was obtained. Reasons:-

Table 2

Source of information	BSFC at rated condition gms/h.p. hr.	Minimum BSFC at full load conditions, gms/h.p.hr.	Speed corresponding to minimum BSFC, r.p.m.
Report of UNIDO Expert A.Cernej, Nov. 1984, p.14.	208	195	2400
Report of UNIDO Expert A.Cernej, Nov.1984, p.163	216	195	
Engine Laboratory experiments with different diesel fuels carried out in 1984.	221-227	196-202	1000-2000
Engine Laboratory experiments carried out in Feb-March 1985 under expert's supervision	195	179·5	1600

1. Thorough check of nozzles.
2. Precise calibration of the pump with working injectors.
3. Sufficiently accurate measurements of fuel consumption and other parameters.
4. Application of scientifically justified correction factors for conditions $\alpha = \alpha_s$

By the end of expert's mission methanol version of Tata model 682 truck engine was almost ready for trials. It would have been possible to perform the tests during the mission provided engineer associated with the project worked on it during the whole mission period. Unfortunately he could spare less than one month on the project work, as he was engaged in other research activities. Further work should be conducted with the help of Dr. S. Radzimirski, UNIDO expert, who is essentially specialist in spark ignition engines. In conducting the work it is necessary to take into account that ignition system procured by far is not the best suitable for the purpose. But that was the only ignition system available in the market. Much colder spark plugs are needed. According to expert's expertency, Bosh-320 or Bosh-340 instead of spark plug corresponding to Bosh-225 should be used with compression ratio 12. Also gas pressure regulators available were designed for LPG. TWO Holl and was approached to get gas pressure regulator suitable for methanol operation. Hence further efforts are required to obtain necessary hardware:

1. Carburator of SU or Zenit-Stromberg type.

2. Central body methanol injection with mechanical and microprocessor controls.

3. Transistorized ignition system suitable to operate at high compression ratios.

4. Precision gas pressure regulator.

Further work should be conducted in close cooperation with Ashok Leyland. Representatives of the firm expressed keen interest in development of methanol- LPG- LNG- CNG versions of their truck and bus engines (see Appendix 6).

Corresponding agreement was drafted at IIP during visit to IIP of Chief Engineer, Research Mr. A.S. Subramanian. (Appendix 8).

Part. 5 : Activities directed to research in other modes of methanol application to diesel engines

In part 1 of the report glow plug methanol diesel engine work at IIP was discussed. To promote this work further some design work was carried out by Mr. Dinesh Kumar under expert's guidance. In fig. 5 lay out of small pre-chamber with glow plug is shown as applied to Ricardo research single cylinder direct injection engine. Work was done before the manuals were obtained at IIP and it turned out that direct injection version of the engine cannot be run. Volume of prechamber constitute ~ 10% of total clearance volume. Shape and dimensions of prechamber were chosen in such a way as to locate it in one of Ricardo research cylinder head holes provided for location of injectors (two injector holes were provided to enable investigation of the dual injection method). Because of space limitations shape of prechamber is not the best one. Also it is not possible to locate standard glow plug. Therefore, special glow plug was designed for the purpose.

Pilot injection should be done into prechamber (10-15 % of total methanol amount, with the help of small plunger diameter pump and single hole nozzle. Main injection into main combustion chamber should be accomplished with the help of bigger injection pump and nozzle having the same number and position of nozzle orifices, as in base diesel engine. Plunger diameter and nozzle orifices area should

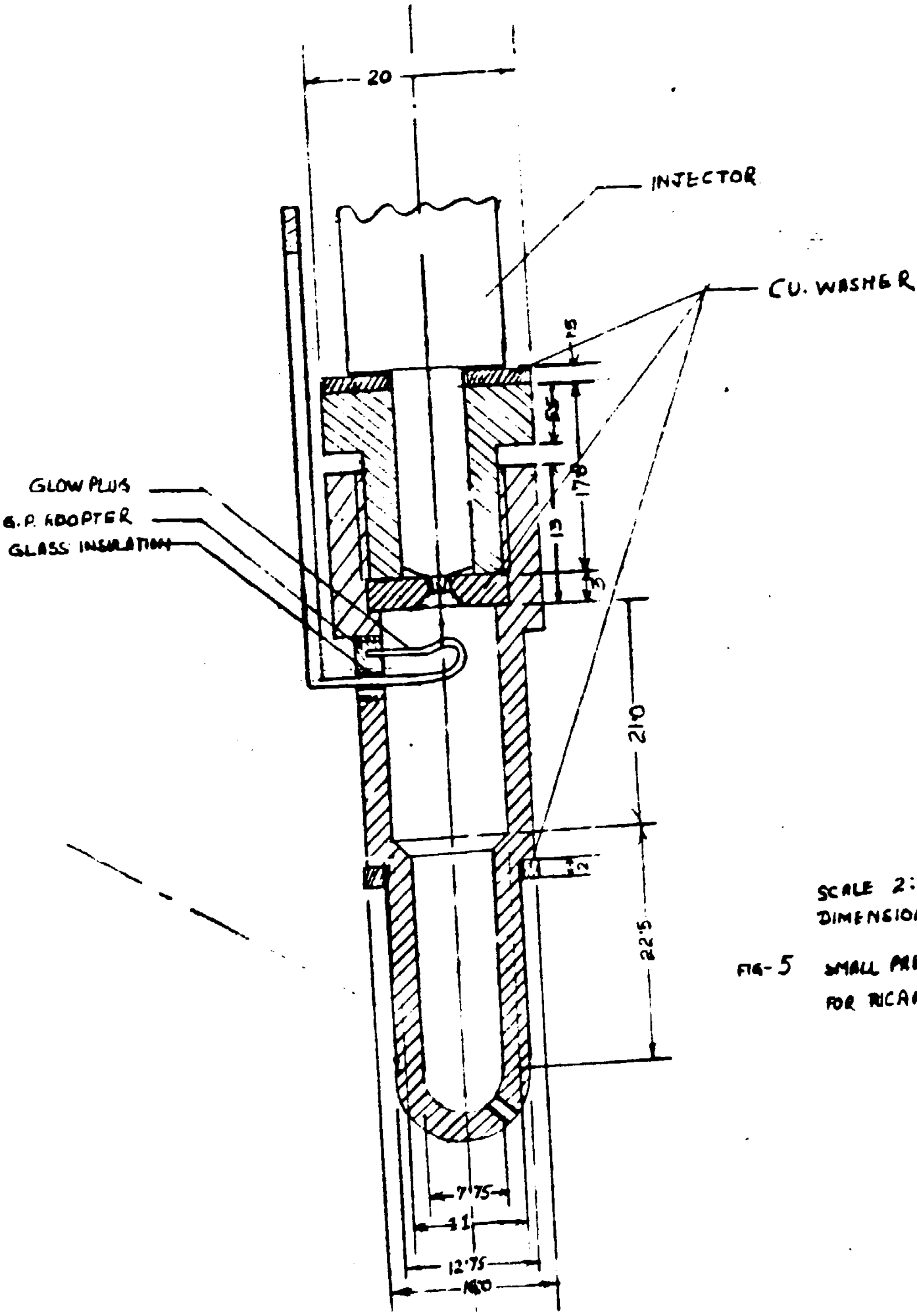

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FIG- 5 SMALL PRECHAMBER FOR RICARDO HYDRA be increased as compared to diesel version to obtain duration of injection not more than 24-26° of crank angle. Also pressure of injection should be kept equal or slightly greater than in diesel fuel injection to ensure necessary fuel sprays propagation. It is expected that starting will be obtained with the help of glow plug, ignition of fuel-air mixture in prechamber and ignition of mixture in main chamber by hot gas flow from prechamber into main chamber. After warm up glow plug may be switched off if design ensures high temperature of prechamber throat. Efforts should be directed to getting low temperature of nozzle and high temperature of pre-chamber throat. Also position of throat holes directing burning gases into main chamber may be of some importance.

In the method described above mode of air fuel mixing in main chamber where most of the fuel is injected, does not undergo considerable change which is the case when IIT, Madras hot surface methanol ignition system is implemented. (Ref. 22 Appendix 2). As compared to ignition of methanol by glow plug installed in the main chamber (Ref. 19, Appendix 2) in the approach just described considerable electric power supply saving is obtained. The method under consideration was suggested more than 20 years back by Soviet Scientist, Dr. V.N. Svobodov and successfully used for a fuel having cetane number equal to 3. However, no research has been carried out with methanol.

It is suggested to conduct multi-factorial experiments using at each speed following governing factors:

1. Amount of fuel injected into pre-chamber.
2. Injection advance for pilot fuel.
3. Amount of main fuel.
4. Injection advance of main fuel.

Maximum amount of heat supply should correspond to that introduced with diesel fuel. Variation of pilot fuel should be substantially less than that of main fuel. Minimum amount of main fuel should represent light load conditions. It is clear that this experiment may be conducted only after successful development work which should ensure stable operation of the warmed up engine with glow plug switched off.

Experiments may be conducted on AV-57 Kirloskar single cylinder diesel engine. In fig. 6,7,8 versions of Kirloskar AV-1 single cylinder pistons with nimonic insert (fig.6), ceramic insert into aluminium alloy piston (fig.7) and ceramic insert into cast iron piston (fig.8) are shown. Design work has been carried out by Mr. Dinesh Kumar and drawing section under expert's guidance. Experience of IIT Mardas studied during business trip to Madras was used while designing nimonic insert piston with air gap. Some recent publications show that very soon the kinds of pistons shown in Fig. 6-8 may be marketed. Restriction of heat rejection into cooling medium may help to increase compression temperature and promote methanol ignition. Hence it will be possible to

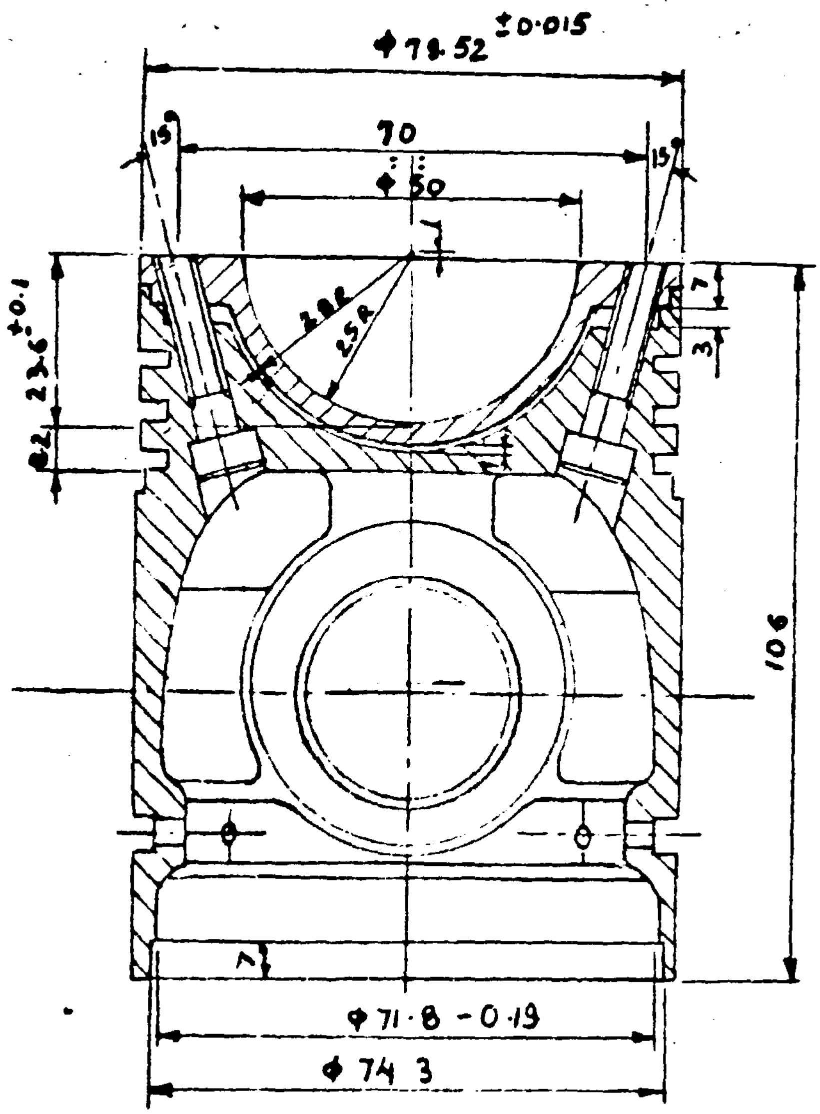

FIG 6

NOTE: ALL DIMENSIONS ARE IN MM UNLESS OTHERWISE STATED

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2							REV. 0 1 2 3 4
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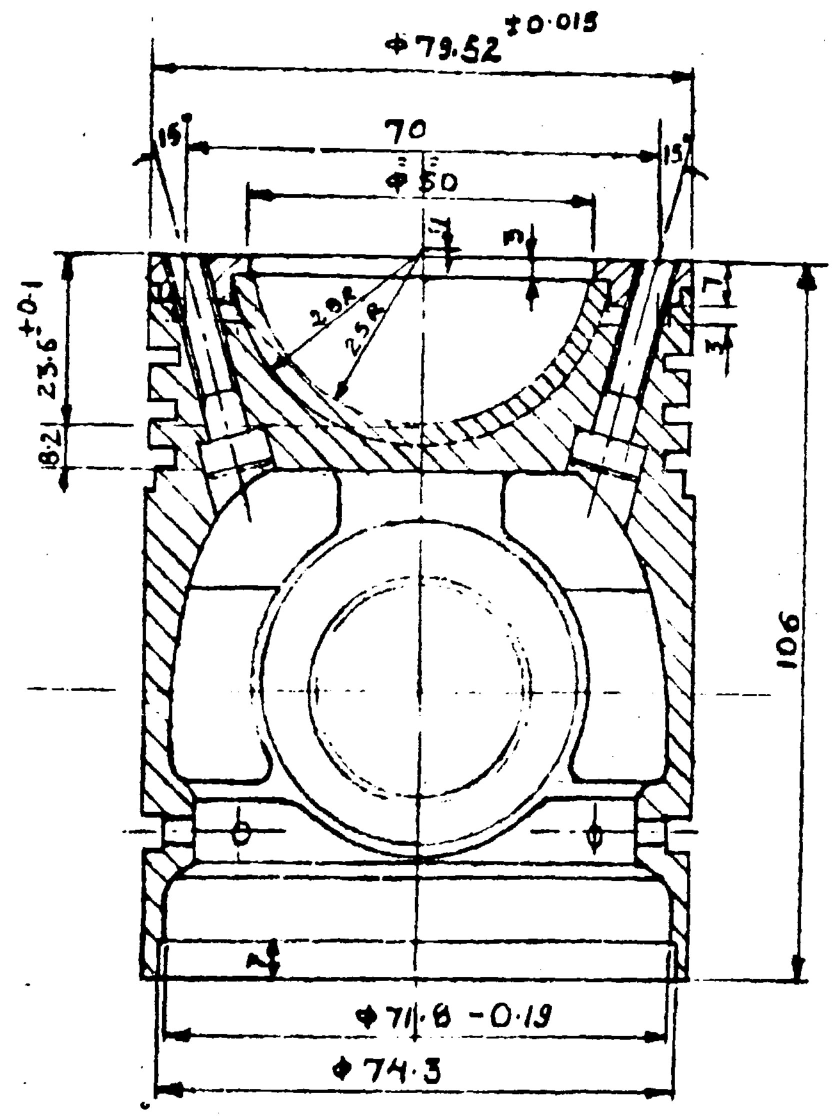

FIG. 7

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SCIENTIST	DINESH KUMAR
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TITLE

ALUMINIUM ALLOY PISTON WITH CERAMIC INSERT

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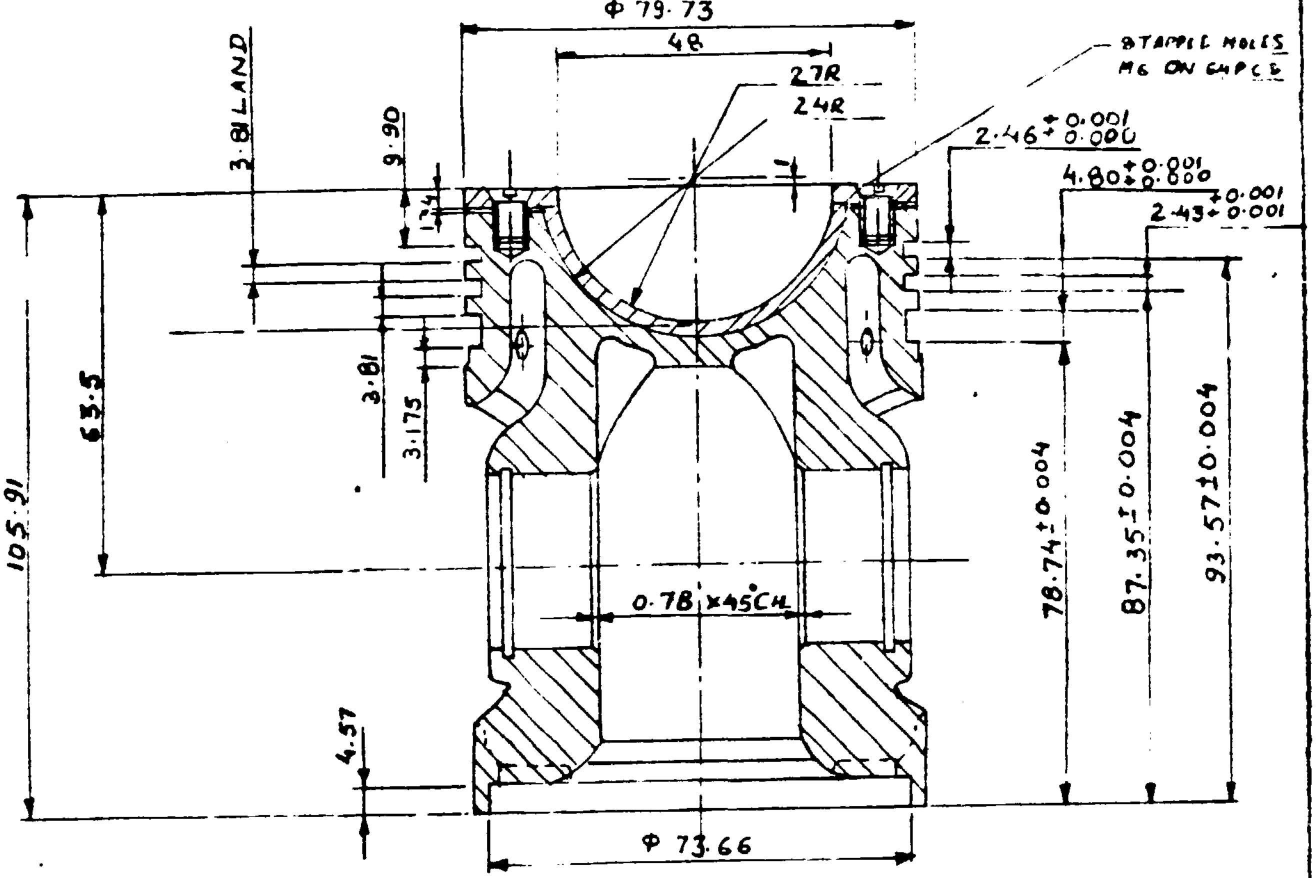

FIG 8

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SCIENTIST: D KUMAR
CHD. BY
DRN. BY: P S LALL
DT. OF START: 28/12/85
DT. OF FINISH: 4/3/85

TITLE: CAST IRON PISTON WITH CERAMIC INSERT

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INDIAN INSTITUTE OF PETROLEUM DEHRADUN (U P) ensure better stability of engine operation at light loads and to decrease electric power supply to glow plug. Also piston surface ignition of methanol may be attempted at least at high loads. To find out effect of hotter surface of combustion chamber on methanol ignition expert during visit to Bombay with the assistance of IIT Bombay obtained as a gift RSZ plasma coated Kirloskar AV-1 diesel piston (thickness of coating 0.4 -0.5 mm). This piston may be tested at IIP Kirloskar glow plug AV-1 diesel engine. For the same purpose 2 cast iron piston assembles were obtained as a gift from Kirloskar Oil Ltd, Pune. It is well known that due to lower heat conductivity cast iron may ensure higher combustion chamber surface temperature. Also RSZ plazma coating of cast iron pistons should be attempted as temperature expansion coefficient of RSZ is much closer to that of cast iron than to aluminium alloy. Hence one may expect greater reliability of coating in case of cast iron provided proper technology of coating is applied.

Detailed information about ceramic industry and ways to get and to use nimonic metal was obtained during business tour. The information was passed over to IIP Engine Laboratory scientists.

While performing tests with versions of piston characterized by higher surface temperature, it is necessary to take into account the following :

1. While coating piston to avoid high rate of wear, it is advisable do not coat outer piston head surface located

between piston top and first ring.

2. Decrease in volumetric efficiency and increase in compression work may take place. Net effect of all the factors should be studied experimentally and also by calculations. For the last purpose single and two zone cycle models may be applied. The first was successfully used during work of expert at IIP on UNESCO project in 1981-1982. In further work, it is advisable to make calculations for the whole 4-stroke cycle as with semi-adiabatic approach, it is essential to find out effect of increased heating of the charge during inlet stroke. Also increase in enthalpy of exhaust gases may be of interest for further application of supercharging and turbocompounding. To find out effect of increased surface temperature on cycle parameters, calculations should be performed at different values of surface parts temperature. In all other respects method of calculation developed earlier and described at IIP Report No. EL 0582, January 1982 may be applied. Prior to computer analysis IIP scientists are advised to read part 6 of the lecture "Modern trends in diesel engine development" (Appendix 3). Use of two-zone model for analysis of semi-adiabatic engine performance is justified by increase in charge temperature and hence greater effect of dissociation on cycle parameters. Once again, expert wants to warn that calculations are aimed to better understanding and planning of the experimental work rather than to prognosis of actual engine cycle.

Some design work was carried out in the lines of application of dual injection-unstabilized emulsion approach. Some of the methods are given in short in Appendix 3.

In fig. 9 cross-section of injector body is shown designed by Mr. Dinesh Kumar and drawing section on suggestion and under supervision of the expert. In this case mixing of the fuels may take place in sac volume itself. Therefore, possible separation of two fuels (in particular case, that of methanol and diesel fuel) is almost entirely eliminated. One-way valve is provided to ensure proper operation and flexibility. Separation may take place in spring room of the injector body where leaking fuel mixture may be accumulated. It will be necessary to find optimum solution for leaking fuel connection. Closed type injector body may be tried. In this case after some time of operation combined loading of the needle will take place. Computer simulation and experiments may help to modify injection system for new conditions of operation. As a first version two pump system may be recommended for this dual fuel operation with provision to change independently injection pressures and timings of supply of both the fuels. It will be of interest to find out effects of ;

1. Advance of diesel fuel supply with respect to methanol supply.

2. Retardation of end of methanol supply with respect to the end of diesel fuel supply etc.

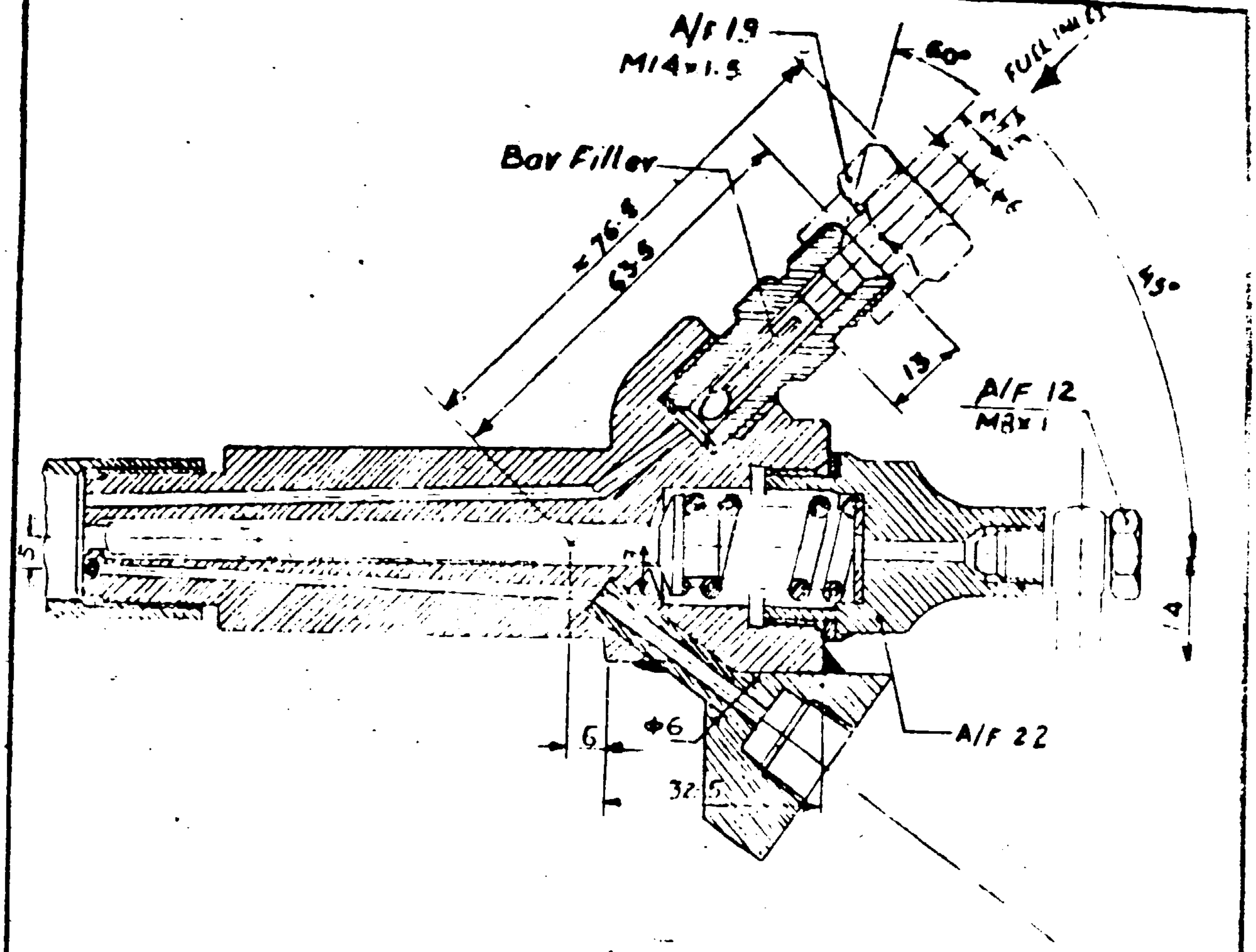

FIG.9

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TITLE

NOZZLE HOLDER For dual injection

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Also other fuels combinations may be investigated with this design. K.S Diesels from Rajcot may be helpful in fabricating the injector. This was revealed as a result of preliminary communication. Unfortunately the expert could not visit K.S Diesels during his work on mission and finalize the design with the manufacturers.

Part -6: Comparative analysis of fuel injection and atomization in case of diesel fuel and methanol

Short analysis of results obtained prior to the start of expert's mission was given in the Introduction to the report. It is necessary to add that as in technical literature there is no information related to atomization of both the fuels, in spite of the shortcomings of the experiments carried out it is advisable to finalize report after application of corrections to particle sizes and methanol injection rates. Though results may not give quantitative relation of particle sizes to rate of injection, some useful comparative analysis may be made under supervision of Dr. A. Cernej during his third mission on the project.

Below short information is given about comparative analysis of fuel injection with both the fuels. Work carried out during the mission in these lines consisted from two parts.

In first part expert developed an engineering method based on theory of similarity for approximate choice of main elements of fuel injection system to feed diesel engine by methanol instead of diesel fuel (Appendix 7).

As a result of this work most important changes in fuel injection system were outlined. These are: increase in plunger diameter and speed, increase in effective area of nozzle orifices and increase in retraction volume of delivery valve. Quantitative recommendations were given as regards to these changes aimed at obtaining the same injection duration and pressures with methanol as in case of diesel fuel and getting at the same time necessary increase in volume of methanol injected (by 2.25 times) to ensure the same heat supply into cylinder.

It has been stressed that use of theory of similarity helps to narrow limits of variation of main fuel injection system elements in computer modeling and experimentation. Hence time and expenditure are saved.

Also necessity in wear resistance and reliability tests were stressed at the end of the report. The quantitative changes in fuel injection system dimensions obtained by the method developed may be considered as extreme ones, as in reality one may expect that certain increase in duration of injection will not be too harmful when changing over from diesel fuel to methanol. This may be expected from results of analysis given in part 3 of the report and Appendix 5. Therefore, in second part of the work on fuel injection, experiments were carried out by IIP scientist Mr. A.K. Aigal under supervision of the expert. These experiments were aimed to ensure normal development of injection process for necessary methanol supply.

As a basis of comparison fuel injection system with cyclic delivery of diesel fuel equal to 44 mm^3/cycle at rated camshaft speed equal to 1000 rpm was chosen. Main dimensions of the systems are given in Table 1.

Fuel	Plunger diameter dpl mm	Number of nozzle orifices	Diameter of nozzle orifices, mm	Delivery valve retraction volume mm^3	Cyclic delivery at rated conditions mm^3/cycle
Diesel	7	4	0.23	37.6	44
Methanol	9	4	0.32	68.4	100

Cam profile (hence maximum plunger speed Cpl max) and high pressure pipe were kept the same for both the cases. However, as delivery period was longer in case of methanol injection mean plunger speed during delivery period was greater for alternative fuel.

Comparison of dimensions chosen shows that product dpl. Cpl is increased by ~ 1.84 times, whereas it should have been increased by 2.25 times to get the same duration and pressure of injection. Nozzle orifices area is increased by ~ 1.94 times, whereas it should have been increased by 2.18 times. Delivery valve retraction volume is increased by 1.82 times instead of 2.25 times. Also volumes in delivery valve chamber, pipe and injector were not changed. Naturally injection pressure characteristics are different with both the systems (both the fuels).

Two sets of experiments were carried out: at constant rack and at constant delivery.

The main findings of research carried out may be summarized as follows:

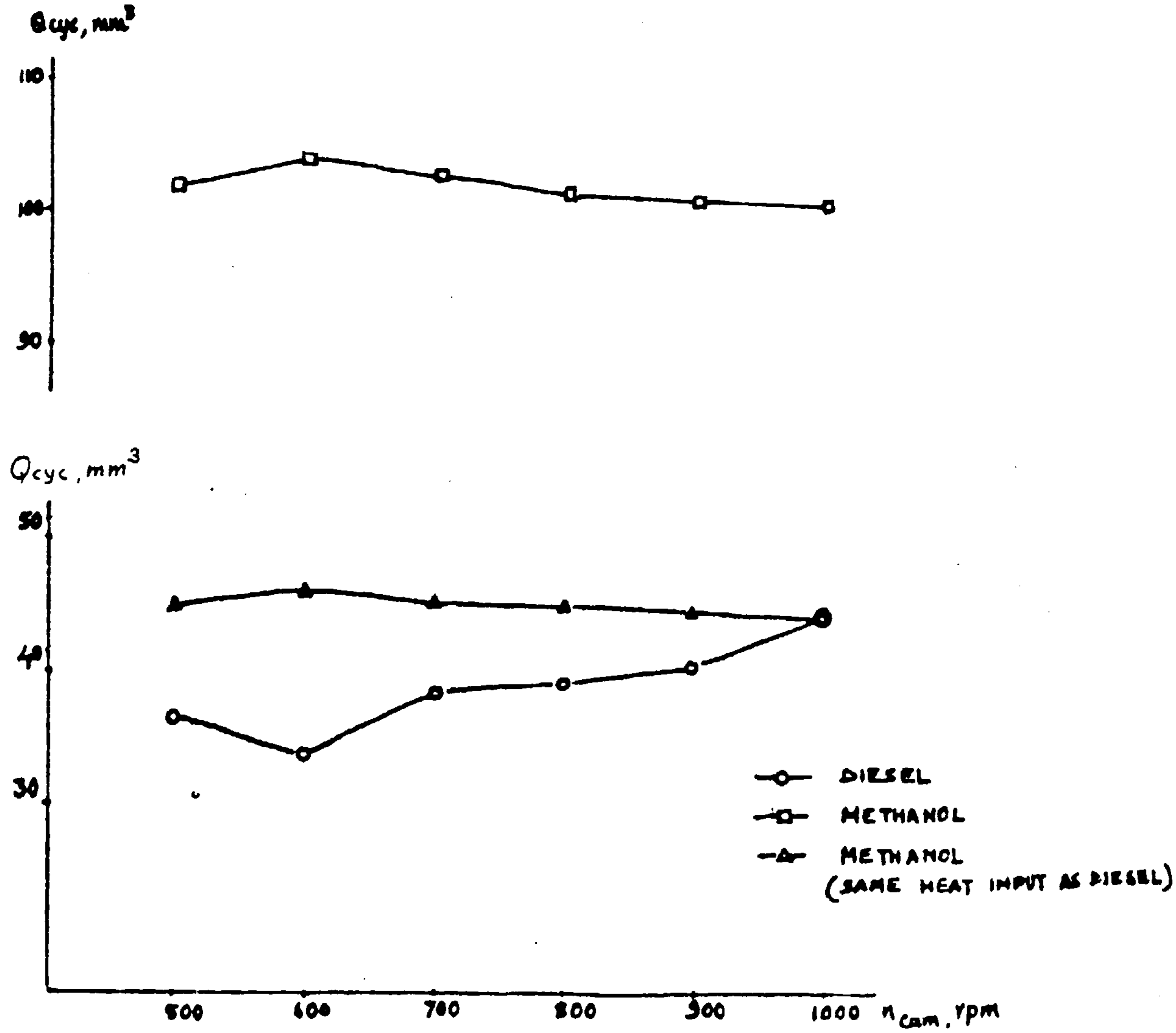

Fig. 10 DEPENDENCE OF Qcyc ON CAM SHAFT SPEED.

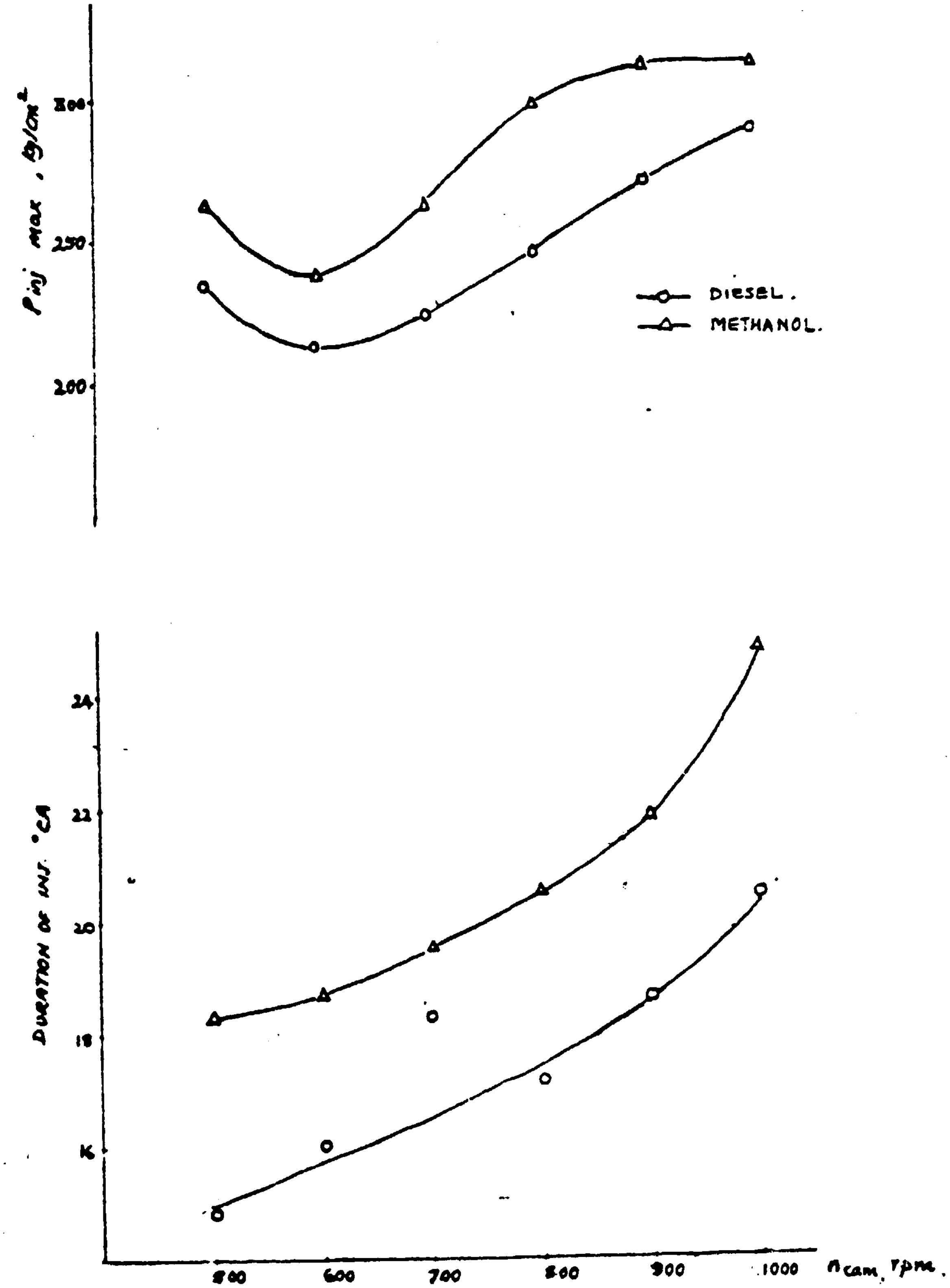

Fig. 11,9 DEPENDENCE OF DURATION OF INJ. AND Pinj (MAXIMUM) ON CAM SHAFT SPEED AT CONSTANT RACK POSITION.

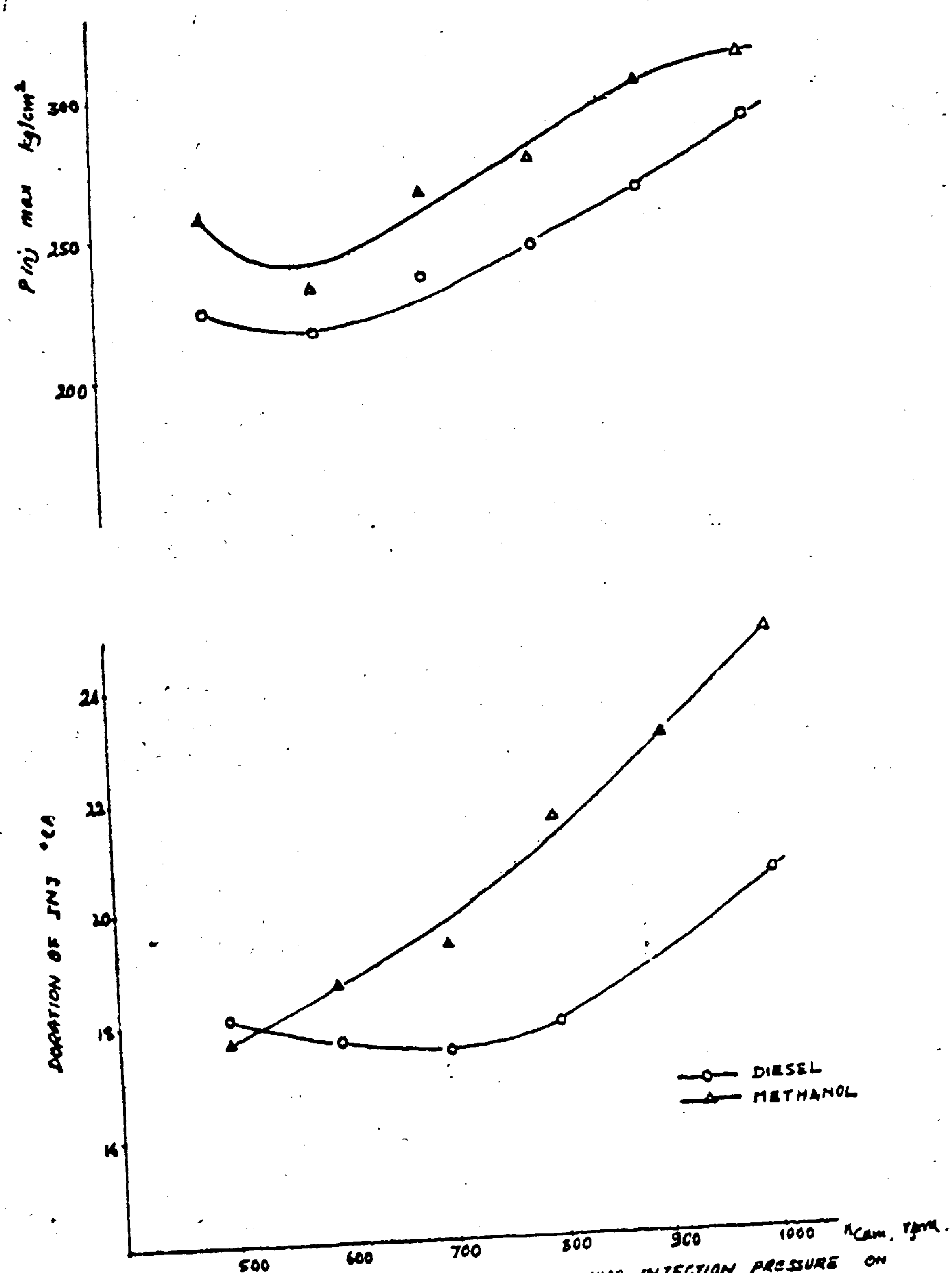

Fig 11,b DEPENDENCE OF INJECTION DURATION AND MAXIMUM INJECTION PRESSURE ON CAM SHAFT SPEED AT CONSTANT CYCLIC DELIVERY.

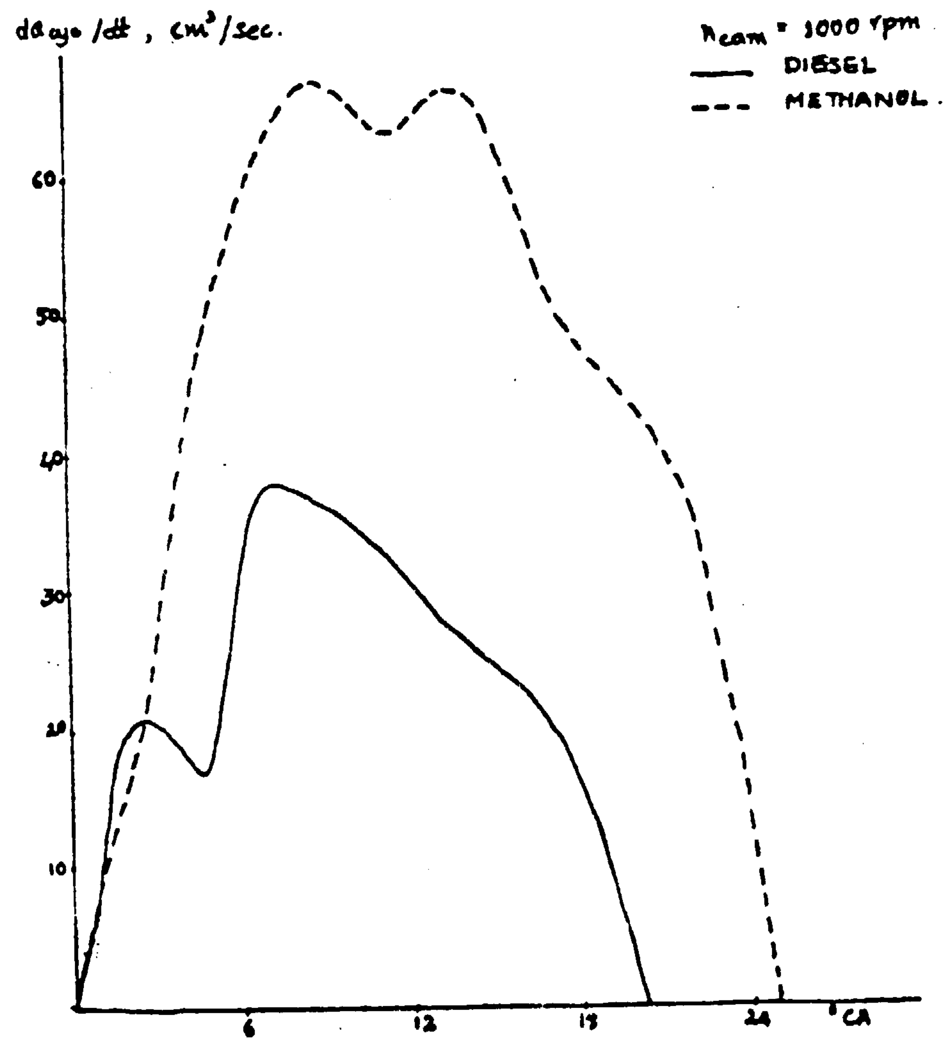

Fig. 12a INJECTION CHARACTERISTICS FOR DIESEL AND METHANOL

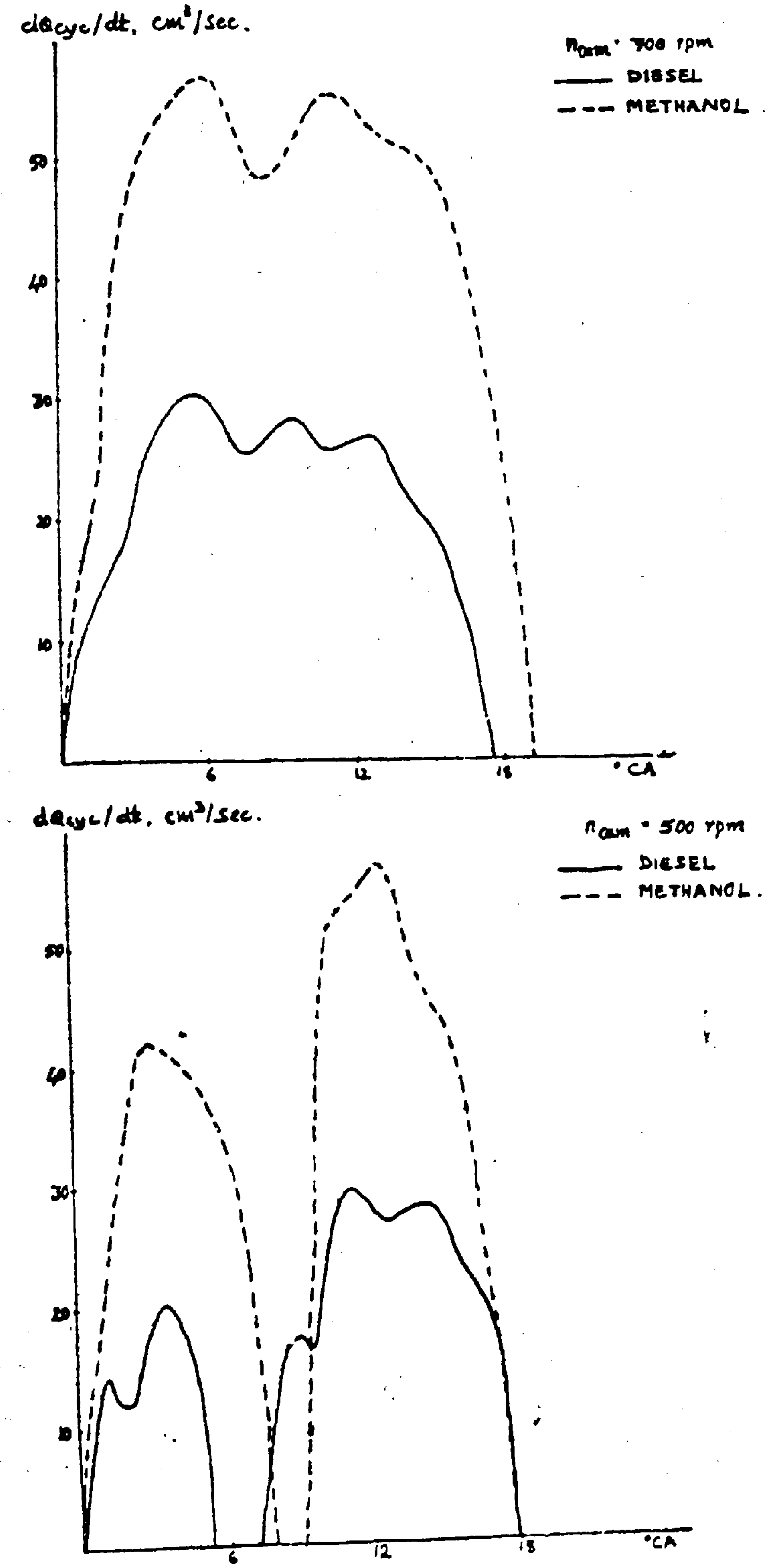

Fig 12.b INJECTION CHARACTERISTICS FOR DIESEL AND METHANOL

a) Nature of change in cyclic delivery φ_{cycl} with cam speed is quite different with both the fuels. In case of diesel fuel φ_{cycl} essentially increases with speed. Hence correcting device is needed to get necessary engine torque increase with decrease in diesel engine speed. In case of methanol cyclic delivery change with speed is more favourable. This may be seen in fig.10. In analysing this result one should take into account greater methanol compressibility.

b) Fuel injection system chosen for methanol ensures greater rate of injection and injection pressure at all the speeds. At the same time almost in the whole speed range injection duration is longer with methanol (fig. 11,12). When compared with results of analysis based on theory of similarity one may explain phenomena outlined above by the following main facts:

Volume in delivery valve chamber was not increased in case of methanol as follows from theory of similarity. Also nozzle orifices area was increased to a lesser extent than it is necessary to keep pressure of injection the same as in case of diesel fuel injection.

These two factors alongwith greater duration of delivery period and higher mean plunger speed ensured higher injection pressure.

Longer delivery period and higher maximum pressure of injection are reasons for longer injection duration. Deviations from recommendations which follow from theory of similarity were quite intentional as with too great increase in plunger diameter and speed contact pressure between cam and roller follower may exceed permissible value. At the same time slight increase in duration of methanol injection as compared to that of diesel fuel injection cannot be harmful.

More detailed information on the work will be given in IIP report which is now being prepared with expert's participation.

Future work:

1. Investigation of fuel atomization with both the systems.

2. Choice of pump lubrication system and injection system durability test. Special measures should be developed to ensure high durability of nozzle and pump.

Part -7: Work done during visits to different institutions and firms

Information about work done during visits is given above and in Appendixes 4 and 6. Here it is however proper to summarize results of work done:

a. It has been found that AVL research diesel engine may be considered as one of two major options for IIP Engine Laboratory.

b. Cooperation between IIT Bombay, IIT Madras and IIP Dehradun in the lines of UNIDO project was initiated.

c. Cooperation with Ashok Leyland in the lines of UNIDO project was initiated.

d. Part of hardware necessary to implement project aims was acquired (some of it as gifts) at Delhi, Bombay, Pune.

e. Work to finalize design and fabricate Indian make single-cylinder research diesel engine was initiated.

f. Know-how and information in the lines of UNIDO project were learnt and communicated to IIP scientist.

Part -8: Concluding remarks and recommendations

1. Vast and detailed information related to the use of alcohol fuels in piston engines has been accumulated and analysed with high competence by UNIDO experts and IIP scientists prior to the start and in the course of implementing the project.

Attempts were made to inform Government of India and relevant scientific, industrial and educational organisations and corresponding specialists about World activities in the field of production and application of alcohol fuels and different technical and technological aspects of the use of alcohols in piston engines.

Special publication under title,"Alcohol Fuels Engine Application" has been started, which is of a great use.

It is however expert's conviction that greater effort should be made to come to definite decisions at Government level regarding perspective of use of methanol in India. In case of positive decisions are taken, industry should be encouraged to start serious R&D in the field concerned. At the same time policy in obtaining methanol at economical price and in sufficiently large quantity should be finalised.

2. UNIDO project at IIP in the most important direction of methanol application to diesel engines have been going on without actual cooperation with industry. At the same time, it is evident that only on the basis of close cooperation with

industry serious R & D work may be carried out and most of the technical problems concerning the use of methanol in diesel engines in India may be solved. Therefore, during the mission efforts were made to initiate cooperation with industry. It has been found that some big and experience firms- such as Kirloskar Oil Engine Ltd and Ashok Leyland, are willing to cooperate.

Vice-president of Kirloskar Oil Engine Ltd, M.S.Chandorkar expressed interest in application of methanol to one of the last advanced designs- AV-57 single cylinder diesel engine which undergoes tests before marketing. As heat rejection to coolant is restricted in this engine it may be recommended to apply to this engine glow plug approach,small prechamber glow plug approach inclusive. Representatives of Ashok Leyland (Executive Director,Product Development & Quality and Chief Engineer-Research) expressed keen interest in development of pure methanol external mixing spark ignition engine on the basis of Ashok Leyland truck and bus diesel engine according to expert's suggestions. As has been planned in the course of mission this engine with minor modifications may be fueled by different gaseous fuels: methanol, LPG, LNG, CNG. Representatives of Ashok Leyland agreed that this multi-fuel ability of basic design concept is very important at the current stage of alternative fuel application to engines under production. Draft of agreement to cooperate was prepared and agreed upon. Initiation of cooperation with two above mentioned firms is one of the achievements of the mission.

3. Work in alcohol application to diesel engine is going on in number of other than IIP institutions. Work at IIT, Madras and IIT Bombay may be of interest for UNIDO project. Therefore, during the mission cooperation between three institutes was initiated. Agreement to this effect, if properly implemented, may be to the benefit of the project. Initiation of the cooperation with both the institutes may be also considered as one of the achievements of the mission.

4. Prior to expert's arrival two directions of application of methanol to diesel engines were tested at IIP Engines Laboratory.

a) Methanol fumigation in inlet system of light duty diesel engine.

b) Glow plug ignition of methanol in single cylinder 5 h.p. AV-1 Kirloskar Diesel Engine.

Further R & D is required in both directions. Work in first direction is advisable to continue in close cooperation with Ashok Leyland with corresponding field trials on Ashok Leyland buses at Madras. Work in second direction should be continued in cooperation with Kirloskar Oil Engine Limited, IIT Bombay and IIT Madras. The following bench trials should be made:

a) Glow plug ignition of injected methanol in AV-1 diesel engine equipped by cast-iron piston instead of aluminium alloy piston.

b) Glow plug ignition of injected methanol in AV-1 diesel engine equipped by PSZ plasma coated piston.

c) Glow plug ignition of injected methanol in AV-1 diesel engine equipped by nimonic insert-air gap piston.

d) Glow plug ignition of injected methanol in AV-57 coolant heat restricted diesel engine.

e) Small prechamber pilot injection surface ignition of methanol in AV-57 engine. In the course of the mission most of the hardware necessary to implement these trials were either acquired or acquiring of hardware was negotiated and agreed upon. Also design work was carried out.

It will be of interest to optimize swirl and position of sprays with respect to glow plug. Before any trial is accomplished test bench should be modified according to expert's suggestions to improve accuracy and repeatability of measurements. On AV-57 diesel engine if installed on Ricardo bench rate of injection may be determined and optimized in the course of tests with the help of Tsnita chamber.

Work outlined above with regard to glow plug aims to increase stability of operation at light loads, to decrease electric power supply, to increase efficiency and to decrease hydrocarbon and CO emission.

Only after these trials it may be possible to assess if multi-cylinder diesel engine glow plug approach should be investigated.

5. In the course of mission test bench version of pure methanol external mixing spark ignition engine was designed and fabricated on the basis of Tata truck diesel engine. In this engine methanol is evaporated completely in multi-tube type evaporator prior to its supply to gas-air mixer through gas pressure regulator. With minor modifications this engine may be fed by LPG, LNG,CNG. In the same lines Ashok Leyland engine version should be fabricated in close cooperation with the firm.

Also the following versions of external mixing spark ignition engines should be tested:

a) With carburator and spiral type evaporator.

b) With central body fuel injection system and spiral type evaporator.

Spiral type evaporator was designed and fabricated during the mission.

Hardware (carburators, central body fuel injection system with mechanical and microprocessor control, transistorized ignition system suitable for high compression ratio operation, precision gas pressure regulator) should be imported as it is impossible to acquire these in India. Thorough comparison of three versions of external mixing spark ignition engines carried out on Ashok Leyland engine in cooperation with the firm should be made. On the basis of this comparison optimum version should be chosen.

6. In the course of mission dual fuel approach with single injector was developed and corresponding design work carried out. The injector as applied to Escort tractor type diesel engine may be fabricated by K.S Diesels at Rajkot. Then test should be carried out in cooperation with Escort firm.

7. In the course of mission engineering method of choosing fuel injection system elements was developed. Experimental comparative analysis of injection process with diesel fuel and methanol helped to recommend extent of increase in plunger cross-section area , nozzle orifices area and retraction volume of deliver valve (by 1.82-1.94 times).

Work should be continued to compare atomization with both the fuels (both the fuel injection systems) and ensure proper pump lubrication system and durability of pump and nozzle.

8. Direct injection version of Ricardo single cylinder research engine cannot be used for work on the project. Therefore, it is recommended:

a) To carry out short tests with evaporated and injected methanol in spark ignition version of the engine. Unfortunately results of these tests may not reveal best methanol engine parameters as Ricardo firm supplied engine with compression ratio = 9, which is too low for methanol application.

b) To install on the bench AV-57 Kirloskar engine and to carry out experiments as outlined above.

c) To take strong actions in order to get in short period properly designed and optimized diesel versions of Ricardo engine suitable for methanol research.

9. Standard of diesel engine performance test and research should be further increased at IIP Engines Laboratory in the lines outlined above. One-two year training of two-three Engines laboratory engineers in well established firm or laboratory may be of great use to achieve this aim.

10. Number of engineers and scientists working in the lines of diesel methanol research should be increased to achieve best implementation of corresponding project aims before July 1986.

11. It is necessary to improve successiveness in different international experts activities to the benefit of final project aims.

Appendix 1

PROGRAMME

of work of UNIDO Expert A.S.Khatchian on project DP/IND/82/001/ (IIP, Dehra Dun, India).

Period of assignment: from 11 October 1984 to 10 April 1985.

1. Visits to Vienna Technical University and AVL, Graz to study the research work in progress at corresponding laboratories and to discuss problems pertaining to the fulfilment of the project.
2. Participation in workshop - "Perspective of alcohol fuel utilization in IC Engines" at IIP, Dehradun.
3. Study of research and development work completed and in progress at IIP, Dehradun, UNIDO expert's reports inclusive.
4. Development of the programme of activity in cooperation with colleagues from IIP Engines Laboratory.
5. Analysis of engine cycles as applied to the fueling with diesel fuel and methanol.
6. Theoretical analysis of the change in engine parameters as a result of diesel engine conversion into pure methanol engine with external mixing and spark ignition.
7. Designing of methanol evaporator for application to engine as in 6 above.
8. Designing of engine parts necessary to convert a 6-cylinder truck diesel engine to work on pure methanol with external mixing and spark ignition.
9. Designing of engine parts necessary to convert Ricardo research engine into pure methanol engine with small pre-chamber and pilot methanol injection.
10. Designing of injector to be fed by two fuels.
11. Comparative analysis of fuel injection and atomization in case of diesel fuel and methanol.
12. Redesigning of engine parts to fix ceramic inserts to facilitate methanol ignition and combustion.
13. Visits to Indian Institutions and firms to discuss and solve problems relevant to the programme.
14. Training of Engine Laboratory personnel through individual and group discussions related to the programme.
15. Preparation of the reports on expert's activities.

UNIDO Expert. A.S.KHATCHIAN.

Approved by: Project Director, Dr.I.B.Gulati
Project Coordinator, S.Singhal.

Appendix 2

Utilization of methanol in diesel engines currently under production

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Properties of methanol differ to a great extent from those of diesel fuel Table-1 (1) - (9). Considering mass composition of the fuels one may observe lower quantity of carbon and higher content of oxygen in methanol. Due to this both calorific value per mass unit of fuel to burn mass unit of fuel and stoichiometric amount of air are considerably lower in case of alcohol fuel. Consequently calorific value of stoichiometric air fuel mixture is almost the same in both the cases. (slightly lower for methanol as applied to conditions of external air fuel mixture preparation and slightly higher for the same fuel at internal mixing conditions). Therefore one may expect at least preservation of power output in case if pure methanol is being fed to diesel engine instead of diesel fuel. Greater amount of oxygen contained in the fuel is one of the reasons due to which carbon and particulate content are negligible in exhaust of pure methanol engines.

Turning over to problems of fuel injection and atomization one should stress lower calorific value per unit volume viscosity, surface tension, and higher methanol compressibility.

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* Currently UNIDO Expert at IIP D.Dun, India

TABLE ·1 Properties of Methanol and Diesel Fuels

Parameters		Methanol	Diesel
Mass fractions of :	C	0.375	0.87
	H	0.125	0.13
	O	0.5	-
Ratio of number of mols before and after combustion for stoichiometric mixture		1.21	1.065
Fraction of three-atom gases in stoichiometric combustion products		0.35	0.26
Density at 20°C kg/m^3		791	840
Lower Calorific value in	MJ /m^3	15780	35700
	MJ / kg	19.95	42.5
Calorific value of stoichiometric air-fuel mixture (P=1 Bar, t=20°C)			
- as applied to internal mixing in kJ /M^3		3638	3461
- as applied to external mixing in kJ /M^3		3175	3475
Stoichiometric air/fuel ratio in kgs of air/kg of fuel		6.55	14.55
in kmols of air/kg of fuel		0.2242	0.496
Heat of evaporation (1.013 bar) in kJ /kg		1104	250
Boiling point/range (1.013 bar), °C		64.7	170-360
Vapour pressure (at 37.8°C) in bars		0.37	
Octane number :	Research method	110	-
	Motor method	92	-
Cetane number		3	45-55
Viscosity (20°C), CP		0.6	3.3
Molecular weight		32.04	180-200
Relative volumetric fuel consumption for the same heat input		2.28	1
Electrical conductivity at 20°C, m^{-1}		4,4.10^{-5}	10^{-3}
Miscibility with :	water	good	poor
	[illegible] hydrocarbon fuels	poor	good
Ignition temperature, K		743	473 -495
Flash temperature, K		284	348
Inflamability limits, vol %		6.7-36	1,58-8,2

To ensure the same heat supply volumetric quantity of fuel delivered per cycle should be 2.28 times greater in case of methanol. In case of sufficient capacity of fuel injection system this may be achieved by the increase in plunger active lift (delivery period). Due to this maximum injection pressure may be even higher for methanol injection in spite of greater fluid compressibility (10). To overcome considerable increase in injection duration when methanol is fed to diesel engine increase in plunger diameter and velocity may be recommended. Both the measures will be more effective in reducing injection period if they are accompanied by decrease in pump and volumes of the system and increase in nozzle orifices area. To choose main dimensions of fuel injection system for methanol application theory of similarity may be utilized (11). In case of methanol injection adjustment of retraction volume of delivery valve may be necessary. Also retardation of injection start with load decrease may be benefitial to improve ignition conditions at light loads. * Lower viscosity and surface tension tend to ensure finer atomization of methanol as compared to diesel fuel. There are no reasons to expect considerable changes in spray penetration in case of methanol injection. Specific heat of fuel vapour is lower and thermal conductivity of both liquid fuel and fuel vapour is higher with methanol. Much higher are saturation pressure and diffusivity. Due to the differences stressed above one may expect higher rates of heating, evaporation and airfuel mixture form-ation in case when

* Problems of lubrication, durability and safety are out of the scope of this paper.

methanol is being injected into diesel combustion chamber. However much higher latent heat in case of methanol tends to hinder methanol evaporation, especially when rate of heat supply from the charge or combustion chamber walls is not sufficiently high. Applied to equal heat delivery with fuel, latent heat is almost 10 times higher in case of methanol as compared to diesel fuel. Decrease of temperature of stoichiometric air-alcohol fuel mixture due to heat of evaporation according to different investigators is equal to 122-200°C, whereas for conventional fuels it is of the order 17-30°C (12), (13), (14). In case of injection into combustion chamber, local decrease of charge temperature may be quite high and as evaporation rate depends on temperature conditions in fuel sprays this may affect rate of evaporation and air fuel mixture formation considerably. The effects discussed above may be responsible for greater increase in ignition delay when alcohol diesel fuel emulsion is being injected into cylinder as compared with partial substitution of diesel fuel by alcohol fumigation in induction system Fig-1 (2).

The most important negative property of methanol is its low cetane number. According to different sources CN of methanol is equal to 3-4. It is felt however that standard cetane rating engines and corresponding methods of rating are not suited for alternative fuels with properties differing from those of diesel fuels to a great extent. Due to considerable differences in properties of fuels at equal ignition delay amount of ignitable mixture formed during delay may be quite different in case of different alternative fuels. Therefore CN of alternative fuels determined by standard diesel fuel procedure is to a great extent conditional. It is necessary to look for a more universal method of assessing ignitability of fuels widely differing from each other by properties. One of the promising approaches may be assessment of ignitability by amount of heat released during kinetic phase of heat release (15). Discussing ignition problems it is important to stress that methanol being resistant to self ignition due to compression is rather easily ignited by hot surfaces.

Regarding combustion and its effects on cycle parameters it is of interest to stress that coefficient of change in number of moles is considerably greater in case of methanol as compared to diesel fuel and this tends to ensure greater cycle work. Unfortunately fraction of three atom gases is also greater with alcohol fuels. As stressed above much greater is heat spent to evaporate fuel. To find out overall effect of these difference on cycle parameters calculations were performed assuming Pa, Ta, ε, n_1, P_z, α = idem.
Here : Pa, Ta-pressure and temperature of charge at the start of compression ; ε - compression ratio ; n_1 - compression exponent ; P_z - maximum cycle pressure ; α - relative air fuel ratio. According to calculations efficiency of cycle in case of methanol is lower by 7.5%. Difference in mean cycle is less (4%) as calorific value of methanol air mixture at conditions of internal mixture formation is higher than with diesel fuel. With diesel fuel maximum temperature of cycle is higher.

One should take into account that differences in actual cycle parameters may not coincide with the above findings due to the effects inserted by dissociation, cooling losses, actual air fuel mixture formation and combustion processes. It is reasonable to expect that the above mentioned processes may be better organized in case of single component fuel-such as methanol, than in case of multy component diesel fuels, espcially at high load conditions.

Properties of methanol discussed above where taken into account while developing different methods to use it as diesel fuel- Fig-2. Here all methods are divided into two major groups. Only second group of the methods will be discussed below, namely methods of use of pure methanol or methanol with ignition improver. In these methods either 100% of methanol is used or methanol constitute major portion of fuel. Also important common features of the methods is use of only one fuel feed system and one fuel tank.

<u>Conversion of diesel engine into spark ignition engine with external air fuel mixture formation</u>:

This method is best suited to open combustion chamber diesel engines. To implement this method it is necessary :

1. To decrease compression ratio down to 10-13 depending on engine size and speed by machining of bowl in piston.
2. To remachine head of cylinder in order to fix spark plugs instead of injectors.
3. To ensure fixing and driving of distributor instead of fuel pumps.
4. To design, fabricate and install fuel evaporator.

5. To install carburettor, fuel injection system or gas air mixer on induction system.

6. To solve problems of engine starting, heating up and control with modified air fuel mixing and combustion systems.

In application of this method experience gained while converting diesel engines to operate on LPG or LNG may be utilized (16) (17). This is especially true in case when methanol is supplied into induction system after its evaporation Fig-3 (18). In any case complete evaporation of methanol before branching of induction system is essential as in diesel engines the latter is not designed to ensure uniform distribution of air fuel mixture containing liquid fuel film among engine cylinders. Optimization of air fuel mixture strength and ignition advance is essential to get best efficiency at all operating conditions. As engine can run on rather thin mixtures when fed by methanol it is possible to use quality control in rather wide load range and apply quantity control only in the range of loads close to idle conditions (18). More complete and timely combustion in comparison with diesel engine ensures rather high efficiency in spite of decrease in compression ratio, use of richer mixtures and throttling at light loads. Smokeless exhaust, low noise and low emission are advantages of methanol version of the engine as compared with base dieselengine.

When engine is designed to drive electric generator or to be installed on tractors and load construction machinery, precision and all speed governing is essential. To meet requirements of such a governing gas air mixer with special type throttle valve may be recommended Fig-4 (16) (17). Essential features of this design is splitting of mixture into two equal streams and directing these streams to two flat slotted throttles. To slotts may be shaped in such a way as to get linear relations between engine torque and speed at conditions when governor controls amount of air fuel mixture supply to the engine. Axial force acting on throttle shaft is almost zero because of symmetric design and also torque developed as result of action of aerodynamic gas forces on throttle is negligible. Due to both conditions base diesel engine direct action mechanical governor may be used successfully to implement quantitative control.

Use of methanol with ignition improver :

No changes in engine design are required to get best performance in this case. However with commercially available ignition improvers amount of the improver to be added into diesel fuel to obtain necessary CN is quite high Fig-5 (2). In spite of considerable content of nitrogen in the additives no increase in NO_x emission was observed. By proper choide of additive amount it is possible to avoid increase in ignition delay duration and rate of pressure rise. There is however critical quantity of ignition improver below which a sharp increase of CO and CH emission is observed (6). Content of soot and particulate emission is negligible when methanol with ignition improver is used in diesel engine. It is essential that ignition improver ensures the same rate of ignition delay increase with decrease in load as diesel fuel Fig-6. In case of faster increase in ignition delay, operation of engine at light loads and idle conditions may be unstable.

Decrease in quantity of additive necessary to ensure stable ignition may be obtained by retarding start of injection without fear of impermissible soot concentration in exhaust gases (15). Addition of ignition improver should not increase cost of fuel more than by 10-15 %. Still this target has not been achieved. Therefore development of sufficiently cheap and effective ignition improver is one of important directions of reasearch work to make utilization of methanol in diesel engines practical.

Use of glow plug to ignite methanol injected into combustion chamber :

The method may be applied to direct and indirect injection diesel engines. At IIP successful attempt has been made to apply this method to direct injection single cylinder Kirloskar engine (19). It turned out necessary to supply at least 0.18 Kw power to glow plug at all operating conditions. Increase of efficiency was recorded at high loads as compared to that of base diesel engine. At light loads efficiency suffered because of difficulty to ignite poor air fuel mixture. Hence stability of consequent cycles was not sufficiently high. In case of indirect injection engines it is possible to switch-off glow plug at full load conditions. However well known low efficiency of indirect injection engines is the reason why this method will not find wide use in practice.

Use of hot surfaces to ignite methanol :

This method was developed in USSR by Dr. V.N. Svobodov (20). Its principle is illustrated in Fig-7. Pilot injection of low

cetane number fuel is done into small pre-chamber (3-7 % of clarance volume depending on engine size.). Major part of the same fuel is being injected into main chamber located in piston bowl. Starting is made possible due to the use of glow plug installed in pre-chamber. After short period of engine operation throat of pre-chamber gets heated due to flow of combustion products at high velocity from pre-chamber into main chamber. Glow plug may be swithced off soon after engine start. At all running conditions ignition in pre-chamber is accomplished by hot surface of throat. Ignition of major portion of fuel injected into piston bowl is ensured by hot combustion gas jets flowing from pre-chamber. It is essential to stress that all precautions are taken to the least disturbance of mode of air fuel mixture formation process in main chamber. This and small volume of pre-chamber are main reasons why efficiency is almost as high as that in base diesel engine fed by high cetane number fuel.
At IIT Madras modification of the method was suggested (21), (22). In this case electrical heating of the pre-chamber wall is utilized. Whole fuel is injected through pre-chamber.
Charge stratification inside combustion chamber and spark ignition:
The best results in application of this method were obtained on MAN diesel engines with so called FM-process Fig-8 (7),(14).

In spite of high compression ratio detonation does not occur as ignitable mixture is being formed first close to spark plug and after ignition-close to flame front. Heat to evaporate methanol is partly taken from spark plug electrodes and combustion chamber walls. For methanol medel of FM diesel engine spark plug with three side electrodes was developed.

It made possible to fix spark plug in conventional way, that is, by threads. It turned out possible to decrease length of electrodes by 10 mm as compared with base FM-MAN engine fed by diesel fuel. Shorter electrodes have lower temperature and are more durable.

As may be seen in Fig-9 and 10 power output, efficiency and smoke emission are quite favourable in case of FM-methanol diesel engine. 1984 Kalifornian diesel engine emission standard can be satisfied with the use of catalyst, durability of the latter being not affected by the content of soot, sulfur and lead in exhaust gases (7), (14).

Conclusions :

First and last of the methods considered above seem to be most practical at the current stage. Use of ignition improver needs development of sufficiently cheap and effective additives. Practicality of two other methods is not yet clear. More research and development work is needed to find out if these methods may compete with those recommended. In application of the methods in which heated surfaces are used to ignite methanol it is necessary to avoid high consumption of electrical energy.

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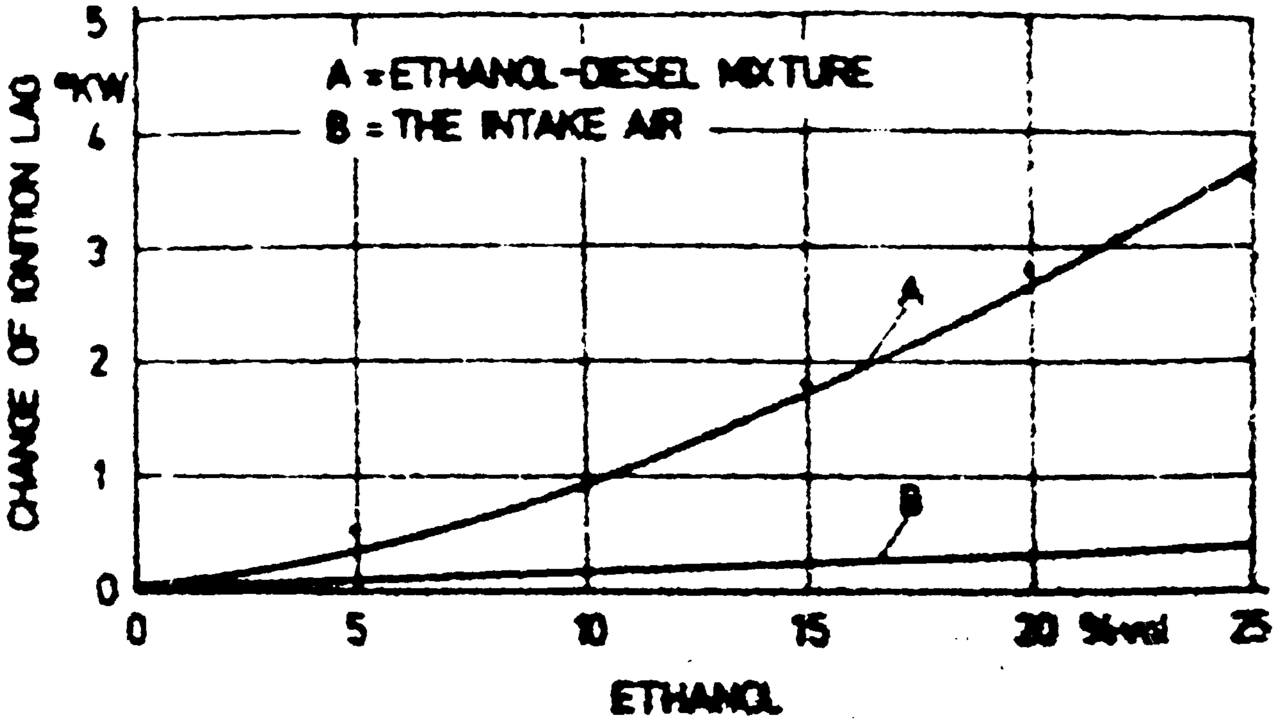

Fig-1 Effect of ethanol addition on ignition lag in case of external and internal supply of the alcohol.

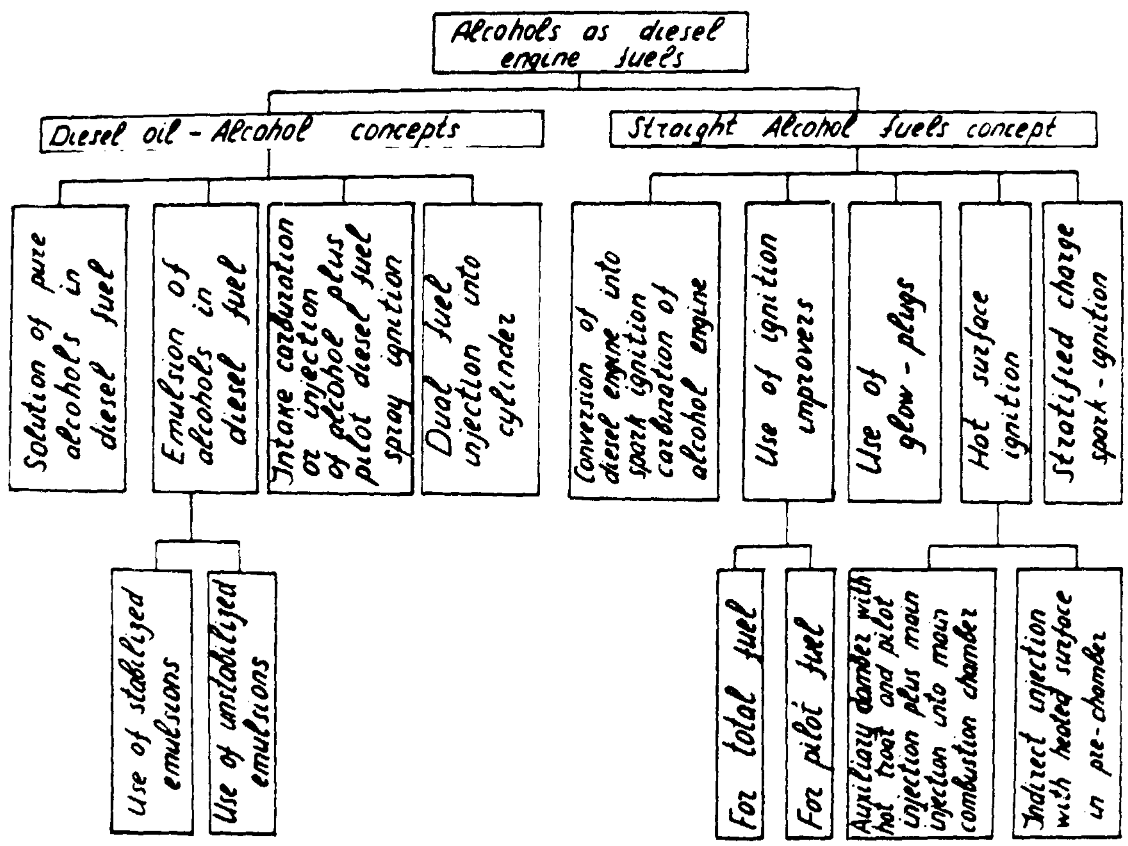

Fig-2 Classification of the methods to use alcohols in diesel engines under production.

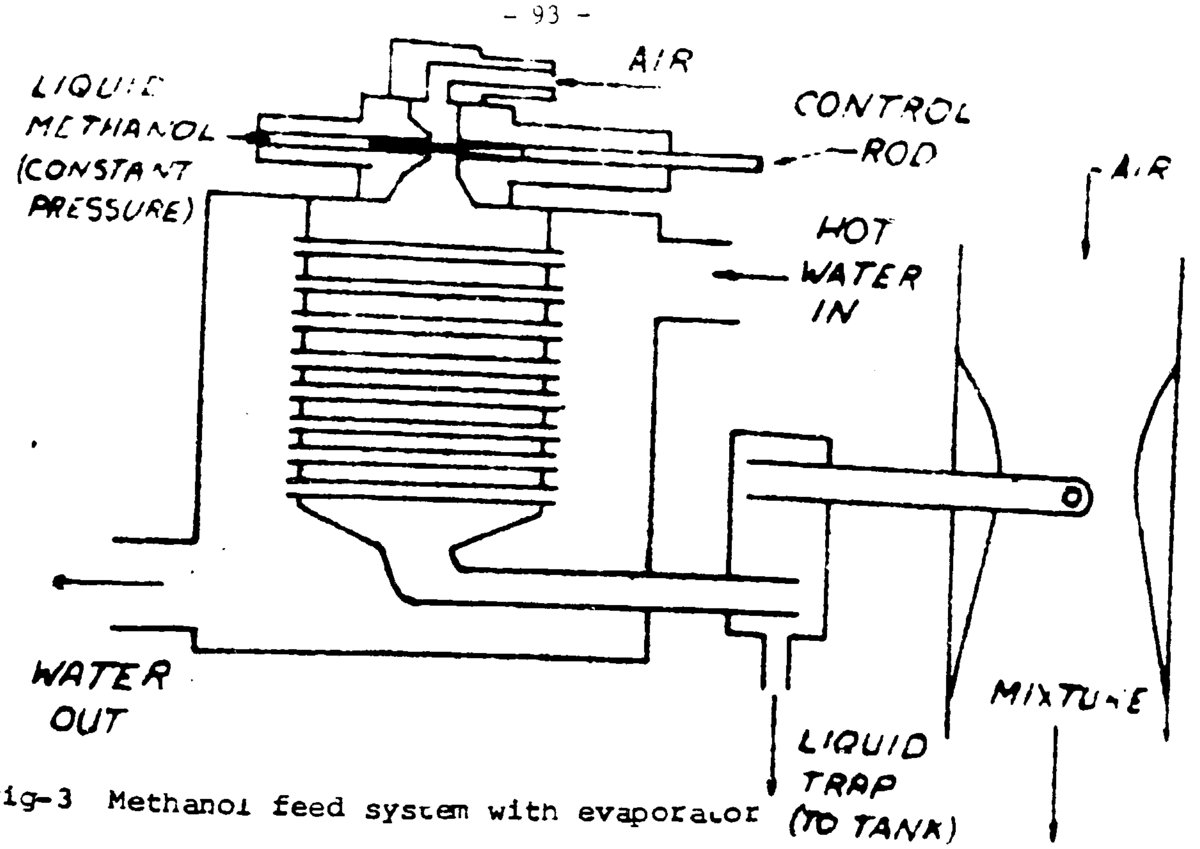

Fig-3 Methanol feed system with evaporator

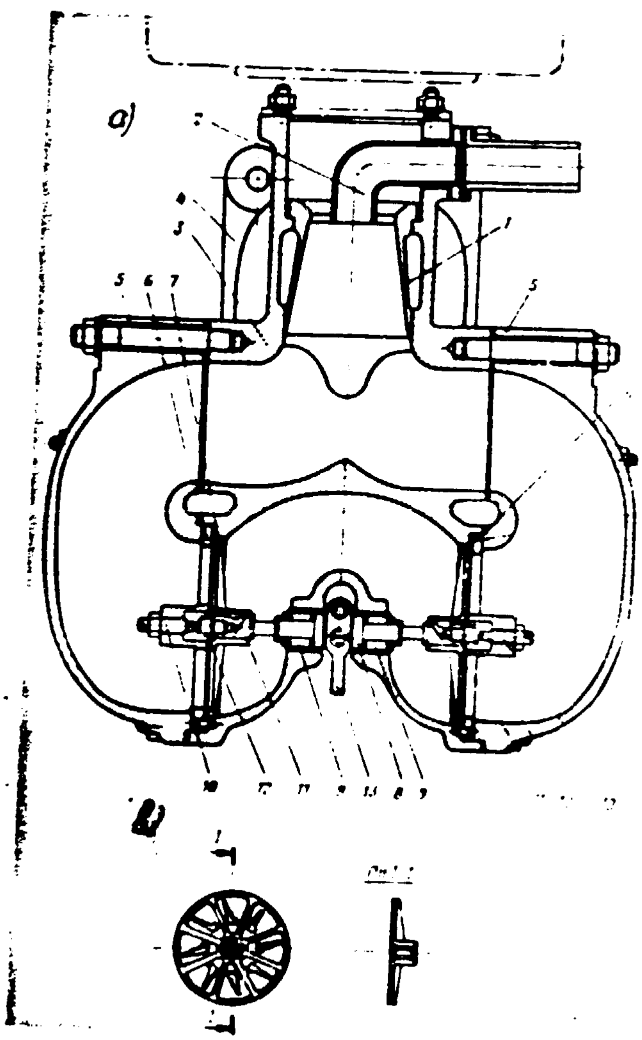

Fig-4 Gas-air mixer with two gas-air streams and flat slotted throttle.

a- cross-view of the mixer ;
b- moving part of the throttle
1- diffusor, 2- gas supply tube,
3- main body, 4- branch to connect the mixer to engine,
5- side caps, 6- fixed parts of slotted throttles,
7- moving parts of slotted throttle,
8- shaft of the throttles,
9- roller bearings,
10- clearance adjustment screws,
11- key, 12- stoppers, 13- link

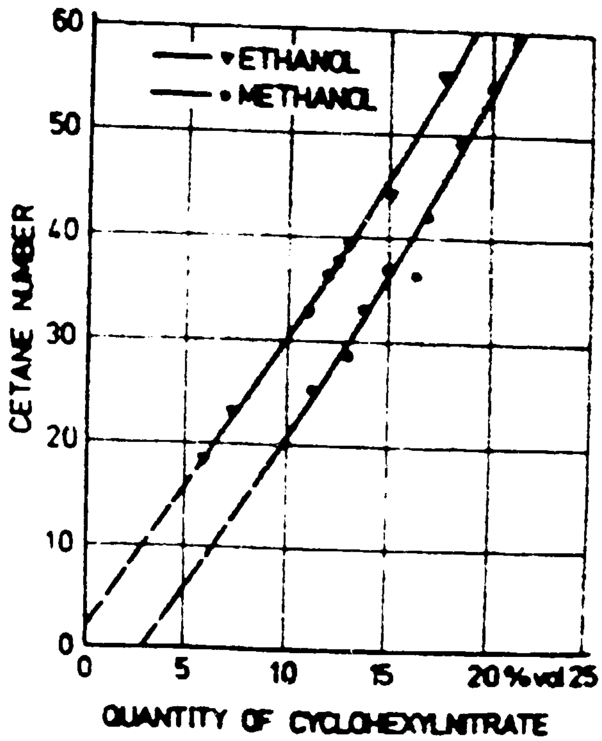

Fig-5 Improvement of cetane number through cetane number improver

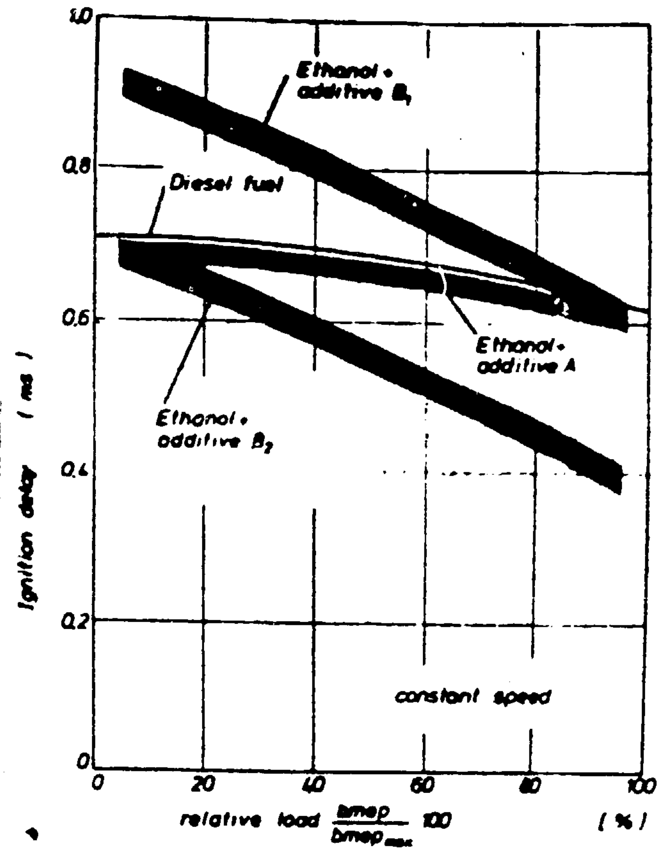

Fig-6 - Ignition delay of ethanol fuels in comparison with diesel fuel

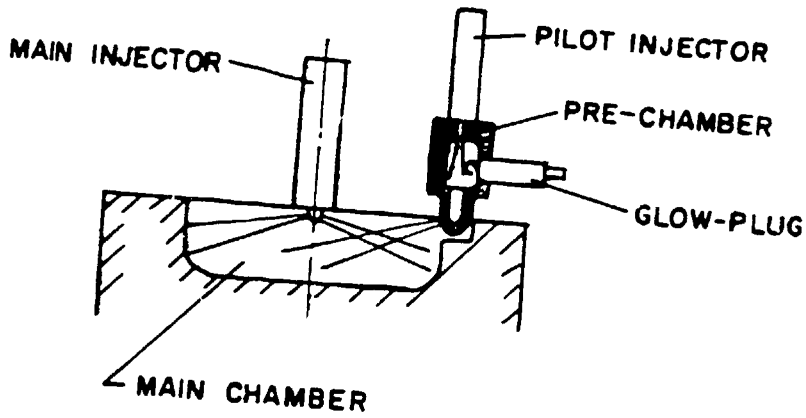

FIG. 7 - Schematic presentation of small pre-chamber used to ignite methanol in main chamber.

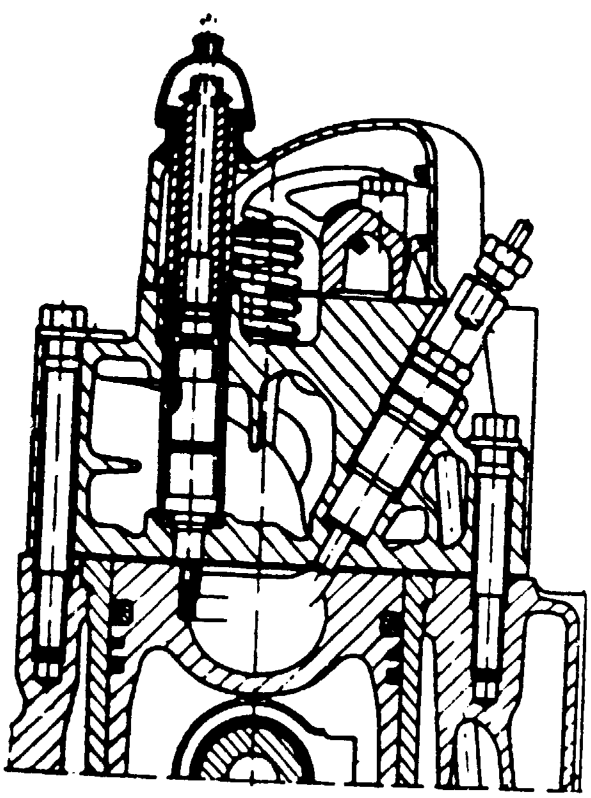

Fig. 8 - Combustion Chamber of D 2566 FMUH Methanol Engine

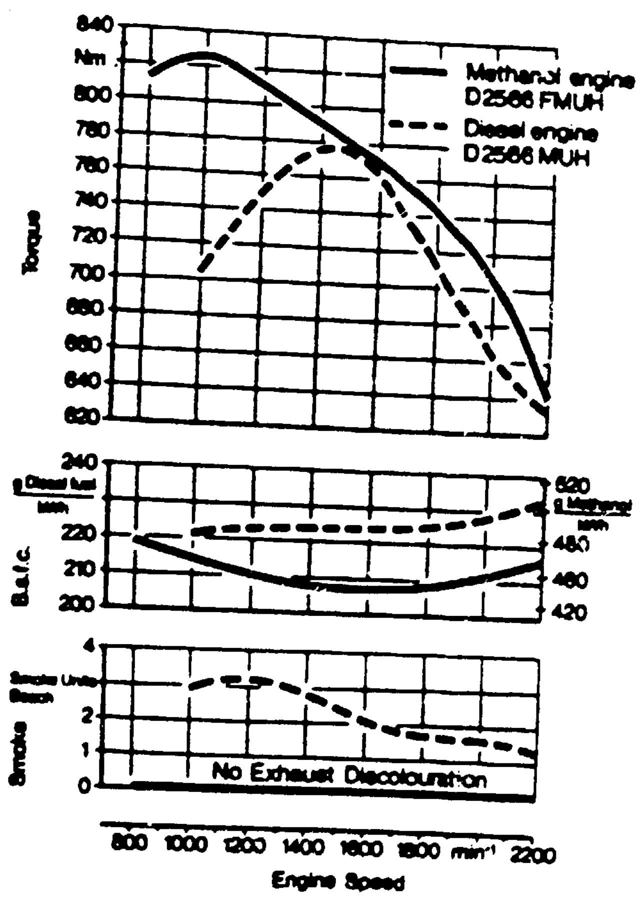

Fig-9 - Full-Load Characteristics of Methanol Engine D 2566 FMUH Compared with Diesel Engine D 2566 MUH

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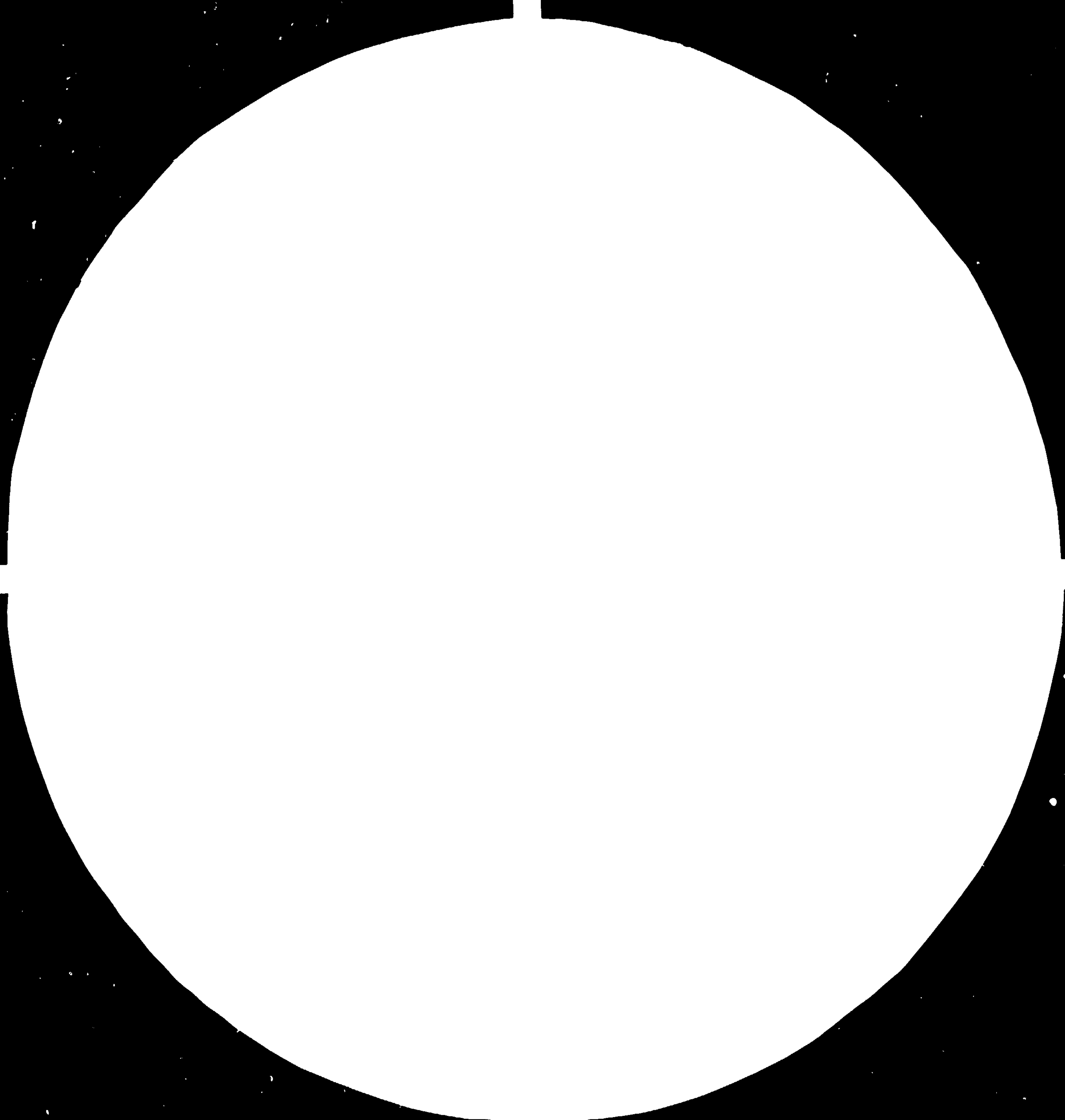

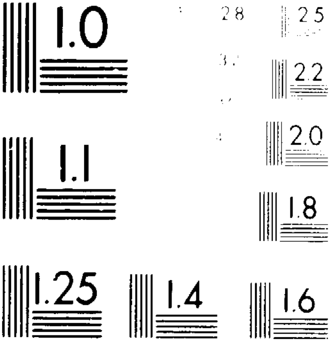

MICROCOPY RESOLUTION TEST CHART

FM Engine

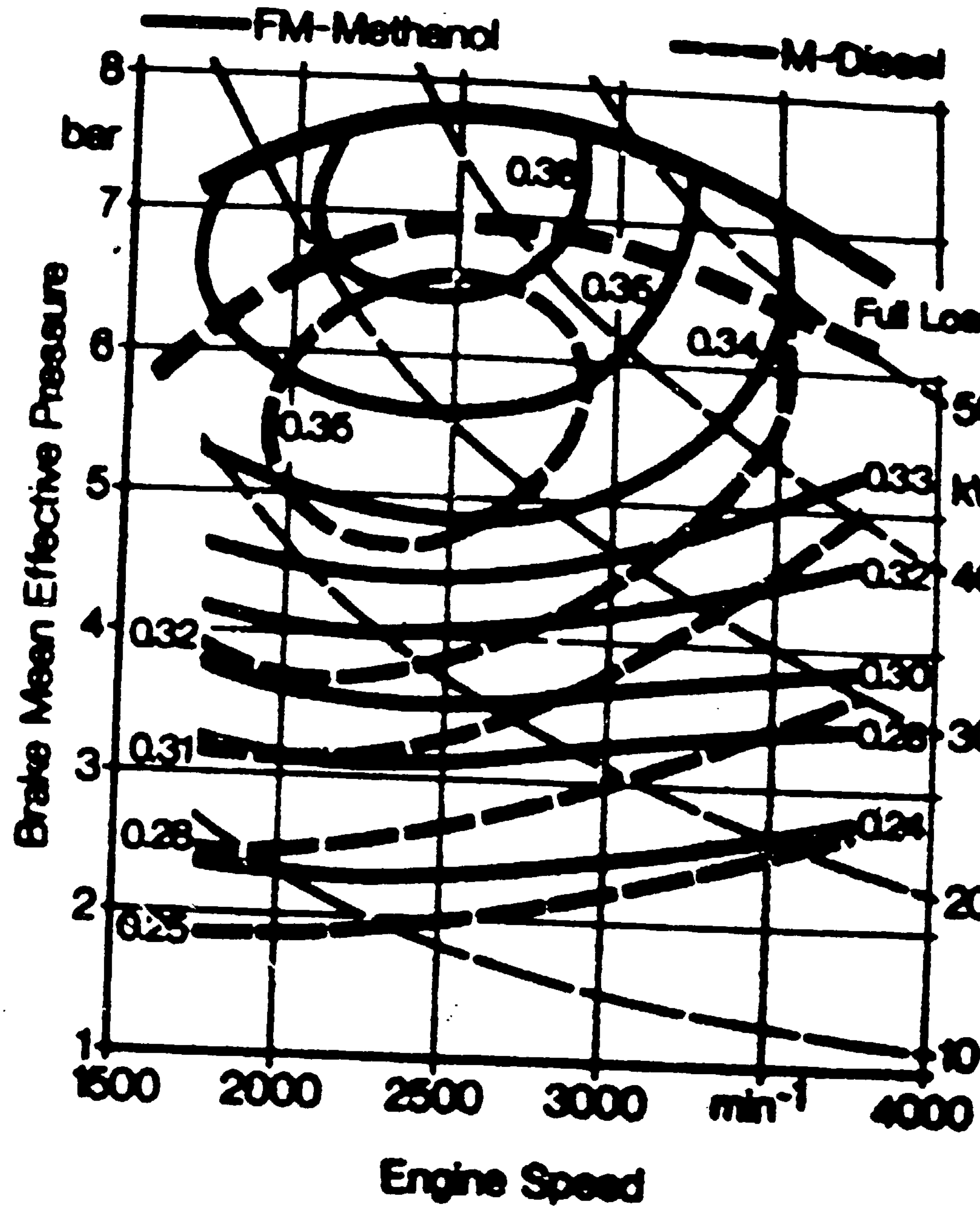

Fig-10 - Brake Thermal Efficiency of L 9204 FM Methanol Engine and L 9204 M Diesel Engine

Appendix 3

Modern trends in diesel engine development

1. Tendencies of development of truck and car diesel engines established in 70-S, according to prognosis, would continue in 80-S. The main priorities are: to save fuel and metal, to decrease emission and noise with accompanying increase in durability and reliability. Though the directions of parameters improvement seem to contradict each other new scientific and technical ideas proved that it would be possible to arrive to reasonable compromises giving definite priorities to different parameters in particular cases depending on areas of application and service conditions.

   All these trends accompany considerable increase in number and share of diesel engines produced. Total nurber of diesel engines produced annually in the world is approaching 10 million and in USSR to 1 million, which constitutes about 10% of world production. Number of models increased considerably especially those designed for cars. In 70-S number of car diesel engine models produced in Europe increased from 3 to 17. In Japan 4 firms are producing car diesel engines. In USA first car diesel engine was introduced by GMC in 1977. USSR is about to join diesel car market with Lada diesel car. The same tendency is valid for India.

2. Qualitative developments is even more remarkable than that of quantitative ones. There are severel reasons for this. First is the effect of energy crisis and enormous increase in oil prices in 70-S. Due to these factors developed countries managed to decrease oil consumption considerably. In 1982-83 oil consumption was reduced by 20% as compared to 1979.

US reduced its consumption from about 925 million tonnes/ year in 1979 to 750 million tonnes in 1983 and is expected to reduce further by another 100 million tonnes/year by the year 2000. Among other measures the reduction in oil consumtion. In US was due to reduction in car sizes. US has 100 million cars and gasoline accounts for about 50% of tetal petroleum consumption. The number of 4-cylinder cars has increased from 10% to about 50%. Second reason was smoke,noise,emission regulations introduced in many countries. After few years of liberalization of these regulations because of energy rises during last few years there is considerable stringenting of the regulations especially in USA and Japan. In table-1 passenger car diesel emission requirements are presented for USA.

	Federal , g/mile				California, g/mile			
	HC	CO	NO_x	Particulate	HC	CO	NO_x	Particulate
1978	1,5	15	2	-	0,41	9	1,5	-
1980	0,41	7	2	-	0,41	9	1,5(2)	-
1984	0,41	3,4	1,5(4)	0.6(1)	0,41	7	1,9(2)	0.6
1985	0,41	3,4	1,0	0.6(1)(5)	0,41	7	1,0(2)	0,4

(1) All altitude requirement (2) 100000 mile NO_x durability requirement (3) Particulate standard of 0,2g/mile delayed for 2 years (4) NO_x waiver from 1,5 g/mile given for diesel or innovative technology.

The third reason is expansion in the successful use of computers in the field of diesel engine development. As Engels put it - necessity is more helpful in development of science then dozen of universities. So there was necessity and this helped to advance science.

Two examples of successful use of computer modeling may be sufficient to prove stated above. Use of finite element or finite differences methods in modeling of temperature, deformation and stress fields calculations in many diesel engines models ensured decrease in engine mass upto as high as 30% without any sacrifice in strength and reliability. The same methods helped to increase specific power out put of truck diesel engines by upto 80-100% with considerable decrease in specific engine mass and ensuring high durability andr reliability. Sesond area of successful use of computer modeling with respect to piston engines is that of gas and fuel flow modeling.

Development of experimental facilities at least kept along if not overspeeded development of computer modeling. Due to development of sophisticated instrumentation, data processing systems and engine test automation not only engine testing has been accelerated and made more precise but there came to life areas of experimentation which hardly were explored before. As an example studies of turbulence in diesel engine combustion chambers may be mentioned. These studies help in sophistication of inlet port and combustion chamber designs to improve efficiency and antismoke ability.

After these general comments let us consider some important directions of truck and car diesel engine development.

1. Turbocharging and combined supercharging may be considered as main direction of diesel engine development. If properly applied they save fuel and metal. Unfortunately this main direction of truck, tractor and car diesel engine development is not yet utilized in India. During number of years there was some unwillingness to use turbocharging in vast scale in the Soviet Union too. This was true only in truck, tractor and car diesel engines as in marine and railway service supercharging is being used for decades. During last 7-8 years use of supercharged models in truck and tractor diesels has increased to a very great extent. Now in many cases new base models are supercharged ones. This applies for instance to new families of tractor and truck diesel engines developed and put into production in Kharkov and Yaroslav diesel engine factories. Sometime engineers and scientists in India argue that supercharging is not perspective in this country because of specific road, traffic and maintenance conditions and hence much smaller power of truck diesel engines as compared to developed countries. In my opinion this is not correct as with efficient small size turbochargers available number of small size supercharged diesel engines are under production in the world which have efficiency better than that of Indian make truck diesel engines

by atleast 15-20%. Hence supercharging and other modifications may be used to save foreign exchange. According to different sources from 40 to 70% of foreign exchange in this country is spend to import oil and diesel engines are one of major consumers of the same. Advantages of supercharging are well known. Therefore let us consider some difficulties in application of supercharging to the engines operating in wide load and speed ranges. In Fig.1 points of truck diesel engine operation are plotted on compressor performance map. It is clear that wider is the operating range of an engine the greater compressor efficiency may deviate from the maximum value. The same is true as regards to turbine efficiency. With decrease in engine speed supercharging pressure drops considerably. If turbine is designes in such a way as to ensure high supercharging pressure at low engine speeds than with wide speed range operation exhaust work may get too high at high speed range and hence efficiency and power output may suffer considerably at corresponding conditions. Also smoke density will be very high at the same conditions. To overcome negative consequences following from difficulties in matching engine and turbocharger performances there are many ways. First and most promising way is to obtain compressor and turbine performances which can be better matched to diesel engine operating conditions. Also engine design may be matched to turbocharger performance. In fig.2 fuel cyclic delivery corrector is shown which decreases maximum fuel delivery when engine speed gets lower than maximum torque speed. Fig.3 shows how this effect is obtained. This type of diagram type actuator help to avoid excessive smoke at low speed range. Fig.4 shows combined effect of two correctors which help to get high torque back up and to avoid deterioration of process at low speed range.

Combined turbo and resonance supercharging may also solve the problem. Scheme of the system is shown in Fig.5. In Fig.6 volumetric efficiency and pumping losses are plotted against engine speed. Sacrifice in pumping losses at resonance speed is small as compared to the effect of better air charging of the engine at low speed range. Decrease in volumetric efficiency at high speed range is benefitial as supercharging pressure is high at these operating conditions. This helps to keep maximum combustion pressure at all the speeds almost the same.

In case of conventional supercharging systems increase in maximum cycle pressure limits engine boosting for a given engine structure. Fig.7 shows improved matching of turbocharger and engine in case of combined supercharging. In case of low boosting impulse type supercharging system may help to obtain better engine performance due to more efficient use of pulse energy at low speeds.

Considerable attention is now paid to Comprex supercharging system (Fig.8) as in many respects it may be more efficient than conventional system (Fig.9 and 10).
Two stage supercharging and Hyperbar systems are stillconsidered to be too expensive and complicated for the use on truck and car engines. To improve transient response of the engine several designs were suggested (Fig.11 and 12). Also Comprex system ensures good transient response (Fig.13).

During last few years it has been proved that in case of high boosting diesel engine,speed derating is benefitial. In our experiments on 8V truck diesel engine designed to develop 500 h.p. at n = 2200 r.p.m. (D/S = 140 /140) derating down to 1800 r.p.m. made it possible the increase of efficiency by 5-7% at full load conditions. Also maximum piston temperature decreased substantially. Improved turbocharger, and engine matching conditions enabled to obtain torque back up equal to 30%. Minimum fuel consumption at full load was equal to 152 gms/h.p. hr. which corresponds to brake efficiency 41%. It is worth noting that air to air intercooler has been used in this case as in almost all the modern supercharged engines. Use of air to air intercoolers is justified in supercharged engines even when supercharging pressure is rather low (1.5 bars and more). Intercooling ensures decrease in engine parts thermal loading, decrease in emission and efficiency improvement at a given engine rating.

Almost in all car diesel engines waste gate systems are being used to meet requirements of wide speed range operation.

Supercharged engines are provided by rather wide simple shape combustion chambers. It has been shown by our experiments that diameter of combustion chamber may play decisive role in limiting specific power output of engine. In case of small diameter combustion chambers located in piston thermal loading of cylinder head as well as that of piston may be too high.

2. In case of unsupercharged diesel engines choice of the combustion chamber shape and domenstous may affect diesel engine speed rating, efficiency, levels of particulate and other hazardous components emission. Also rate of pressure rise and diesel engine noise emission depend on the type of combustion chamber to a great extent.

According to experience gathered so far one of the most beneficial combustion chambers suggested so far is that of Doitz diesel engine. In fig. 14 and 15 corresponding mode of air fuel mixing is illustrated. Only two nozzle holes are used and high sword intensity is obtained due to the use of effective helical inlet port. According to our measurements charge velocity in combustion chamber may reach 60 $^{m}/s$. Inspite of rather low volumetric efficiency and high dead volumes of combustion chamber brake efficiency and mean pressures are at least not worse than those in Daimler-Benz (D-B) cylindrical combustion chamber, which has considerably higher value of combustion chamber to cylinder diameters ratio and higher volumetric efficiency. At the same time, softer combustion and lower noise emission is obtained with Doitz combustion chamber as compared to D-B combustion chamber. Starting ability of Doitz diesel engines is better than that of MAN engines. Unfortunately even medium supercharging in case of Doitz combustion chamber is accompanied by unadmissible level of cylinder head thermal loading. Great efforts are being made now-a-days to meet smoke and emission limitation by special combustion chamber design (Fig. 16,17,18 & 19). Number of combustion chambers shown are impractical as regards to production and local overheating. Some of them may found use. This is the case with combustion chamber shown in Fig.19 which ensures high local turbulence in prewall region and thus shortens after burning increases efficiency, decreases smoke density. Also efficient engine operation in wider speed range is made possible as compared to conventional cap shape combustion chamber. In general, while designing combustion chamber, it is necessary to decrease dead volumes as much as possible. Small overlap period helps to achieve this aim.

3. Great attention is being paid to optimising of injection process. It was shown in our experiments that increase in injection pressure and injection duration shortening is beneficial only in case if it is not accompanied by substantial increase in amount of fuel supplied with decreasing rate. Conditions of air fuel mixing and combustion for fuel portions supplied after ignition with decreasing rate were found to be unfavourable. This may be seen from Fig.20 and 21. Special design of injector shown in Fig.22 helps to increase the

pressure of nozzle closing and thus shortens final injection stage. Here $h < z$ and high pressure built in space 5 by the time spilling starts helps to increase needle closing pressure. To optimise injection process controlled hydraulic loading of the needle may be used. We developed injection system which made it possible to change injection pressure and shorten injection process to a considerable extent on running engine. Similar effect was employed by CAV England Fig.22a. According to experiments carried out for each particular engine there are optimum values of injection pressure and duration. In our investigations it has been found possible to decrease smoke density by several times and increase efficiency by 4-6% at rated conditions of supercharged diesel engine while keeping maximum cycle pressure unchanged by the increase in injection pressure. At low engine and swill speeds increase in injection pressure did not bring about improvement in efficiency and smoke density. Hence, flexible control of injection process and timing for truck and car diesel engines may be of importance especially in case of stringent pollution limitations.

The latter demand the modes of injection process and timing change with operating conditions which cannot be met by conventional injection systems.

4. Therefore, electronic control and electronic injection are now considered to be viable solution of the problem of flexible control of injection timing and injection process as a whole. In Fig.23 overall view of injection pump with

electronic gove or is presented. Fig.24 illustrates block diagram of electronic governor. With this kind of equipment it is possible to take into account precisely not only particular engine characteristics but also thermal conditions of engine operation, atmospheric conditions etc.

Fig.25 shows the same kind of system in case of unit-injectors which can easily ensure very high injection pressure. One of drawbacks of unit-injectors is difficulty to control injection advance with engine speed. Electronic governor and special design of helics on plunger (Fig.26) ensure proper control of injection advance with speed. This is essential in car open type combustion chamber diesel engines. Great attention is attributed to accumulator systems with electronic control. In Fig.27 design of electronically controlled injector is shown developed in USSR. To ensure fast response electromagnet operates light valve. When it opens hydraulic force is being defeloped which opens nozzle. Therefore, injector is known as electric-hydroulic type. Valve motion is controlled by microprocessor.

Microprocessor control of injection process improves engine efficiency at some operating conditions upto 5-7% , decreases emission, smoke density and also stresses acting on engine parts.

5. There is considerably increase in number of diesel cars being sold in the market, especially in Italy, W. Germany and France. By the year 1990 around 15% cars sold in the World

will be supplied by diesel engines, 30% of these amount being open type combustion chamber diesel engines. In Fig.28,29 & 30 types of open combustion chambers and injection systems are illustrated. According to Ricardo Co. analysis wall wetting type combustion chamber is beneficial if one strives to achieve reasonable compromise between engine parameters such as efficiency, noise, emission, mechanical loading of engine parts, cost of fuel injection system, starting ability etc.

Open type combustion chamber car diesel engines have following advantages as compared to engines with divided type combustion chambers.:

1) Lower fuel consumption- upto 10-20% in city operation conditions.
2) Lower heat rejection to cooling medium and hence radiator size and faster warm-up.
3) Lower thermal loading.
4) Better supercharging ability.

<u>Disadvantages</u>

1) Lower specific power output and maximum torque (by 5-10%) due to difficulty to optimize injection and swirl in wide engine speed range.
2) Lower emission (NOx emission is lower upto 50%).
3) Lower peak cylinder pressure and engine noise (upto 10-20 bars).
4) Lower fuel injection system cost.

However, almost all of these disadvantages may be overcome by the proper use of microprocessor control and controlled supercharging. Fig.31 stresses importance of inlet duct design in cases of car and light duty diesel engine operating in rather wide speed range. Fig.32 illustrates one of the recent developments in the field of car diesel engines- IVE CO supercharged diesel engine(power output 68 KW at 4200 RPM, maximum torque 216 nm)with plastic head and gear drive covers. In addition to the use of plastic covers to decrease noise cylinder block is made in two parts separated by resilient gaskets that damp high frequency vibrations. As may be seen from Fig.33 minimum fuel consumption obtained without fan, muffler and air filter is equal to 230 gm/kw.hr, which corresponds to about 38% brake efficiency. Increase in power output as compared to unsupercharged indirect injection version is 34% and increase in efficiency-18%. Ford of England claims increase in power output by 10%, in maximum torque by 8% and in fuel consumption upto 24% in direct injection 2.5-L diesel engine (model FSD 425, power output 50 KW at 4000 rpm, maximum torque 143 nm, compression ratio= 19) as compared to indirect injection version (ransit 2.36-L). Reasons of improvements obtained: lower heat losses to the cooling, higher mechanical efficiency (no pumping losses during compression- expansion, lower compression ratio), use of ram effect by proper design of inlet manifold, refined swirl port, 4.5 mm low inertia needle. Low noise emission is obtained by proper ribbing of cylinder block, expansion controlled pistons (reduction of clearance to 0.008 mm) selective assembly of pistons with four skirt sizes to match to the measured cylinder bores.

6. One of the most important directions of development which is considered by some of specialists as revolution in the field of diesel engine application is development of compound engines incorporating adiabatic diesel engine. First of all it is necessary to stress that diesel engine strictly speaking may not be adiabatic. Eve if we imagine engine parts fabricated from materials with negligible specific heat and very low heat conductivity, temperature of parts surface may not follow charge temperature and flame temperature at the same time. And, it is known that charge temperature is considerably lower than that of flame. If we assume surface temperature to be equal to so called heat exchange average temperature of the charge, then integral heat transfer may be equal to zero, but instantaneous heat transfer will not be equal to zero. There will be heat loss by charge during first period of combustion-expansion and heat gain from the walls during the following part of expansion when heat utilization approaches to zero. Hence, even if one achieves integral heat transfer equal to zero, this will not exclude effect of heat transfer on cycle efficiency. Also one should take into account decrease in volumetric efficiency and increase in compression and exhaust work while considering effect of reduction in heat transfer on engine parameters. Results of modeling do not coincide with experimental results due to the fact that while modeling investigators do not take

into account properly actual conditions of heat transfer and especially high radiant component of heat transfer in diesel engines (instantaneous share of radiant heat transfer may be as high as 0.7 -0.75).

Above stated does not mean that utilization of wasted heat cannot bring about considerable improvement in engine parameters. In W. Germany ElCO developed diesel engines with restricted heat transfer and obtained high diesel engine efficiency (upto 43%).
While assessing this result one should take into account that also decrease in friction losses was obtained in ElCO engines. Decrease in heat losses with coolant creates conditions for efficient use of exhaust heat in power turbine.

In fig.34 scheme of turbocompound engine is given. In addition to turbine which drives compressor to supercharge diesel engine, power turbine coupled through special drive with crank shaft is used to utilize part of heat taken away by exhaust gases. Fig.35 illustrates changes in heat balance due to compounding and heat transfer restriction. To restrict cooling losses ceramic parts,ceramic coating or ceramic inserts may be used. These materials, should have low heat conductivity, low specific heat, low density, high strength and high temperature expansion coefficients. The latter is important to facilitate matching with metal parts with which they are to be combined (Fig.36). In this respect partially stabilized Zirconia dioxide is considered as perspective ceramic. It is claimed that this material also possesses rather high strength and temperature shock resistance ability. Fig.37 illustrates Komatsu coolant heat restricted diesel engine. In this case it was found necessary to cool piston to keep temperature of friction surfaces and that of lubricant at reasonable level. Due to high lubricating oil evaporation rate tickness of synthetic oil film decreases considerably in case of high liner temperature (the latter being upto 400°C) (Fig.38). Firm however claims good wear resistance and sufficiently high reliability with proper choice of design, materials, technology of fabrication and oil. Practical decrease in coolant heat was 35%- Fig. 39. It has been found that turbine (0,82) and drive (0.89 to 0.93- drive ratio 30,6) efficiencies are of a great importance in getting high efficiency of turbocompound engine. Fig. 40 shows performance graphs of basic and compound engines. It may be observed that brake efficiency of the order of 50% was obtained in compound engine with restricted heat rejection. Considerable decrease in particulate emission due to high temperature combustion was also recorded on engines-Fig.41.

Trade-off curves plotted in Fig.42 show that greater decrease in NOx and HC emission may be obtained in engine under consideration by retardation of injection start with smaller sacrifice in efficiency as compared with conventional diesel engine. Also better multy-fuel ability may stress advantages of diesel engines with restricted coolant losses. Final aim of development is abolishing of both cooling and lubrication systems and reaching brake efficiency of the order 55%.

Cummins firm is very active in the field. It is claimed that programme is related to military application- Fig.43. In conclusion, it is necessary to stress that after 100 years of development the field of diesel engines is very dynamic and a lot of improvement may be expected in the years to come.

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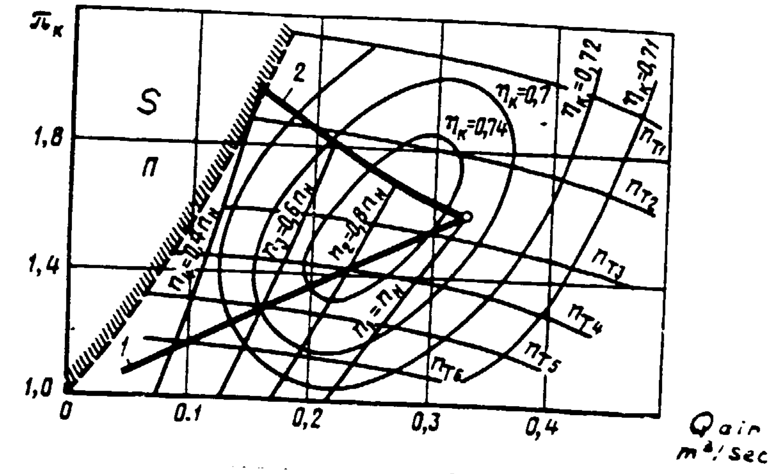

Fig.1. Compressor and diesel engine performance map

$n_1 \ldots n_4$ load performances of diesel engine at different engine speeds.

$nT_1 \ldots nT_6$ compressor performances at different turbocharger shaft speeds.

1 - full load speed performance with uncontrolled supercharging
2 - constant power output performance with controlled supercharging.

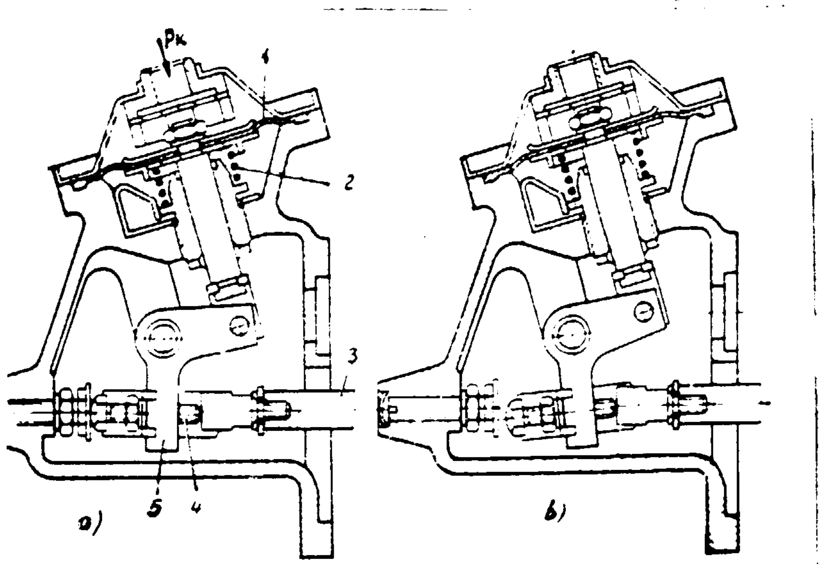

FIG.- 2 Cyclic fuel delivery corrector actuated by pressure of supercharging.

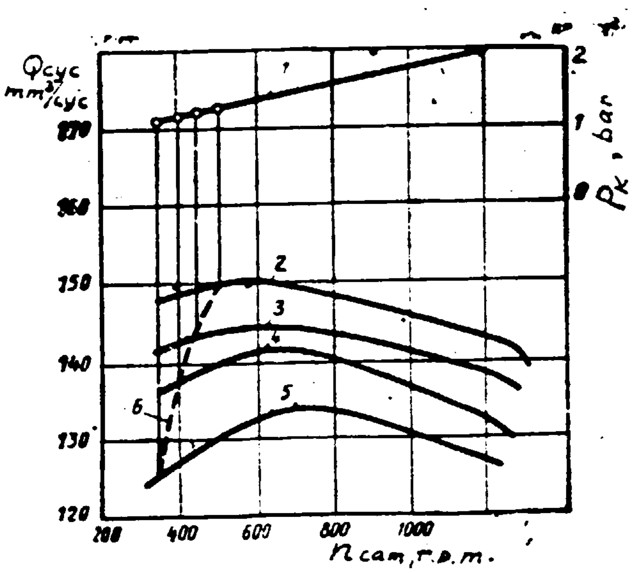

Fig.3. Correction of Cyclic delivery versus cam speed relation with the help of supercharging pressure
1- supercharging pressure
2..5-cyclic delivery for different rack positions.

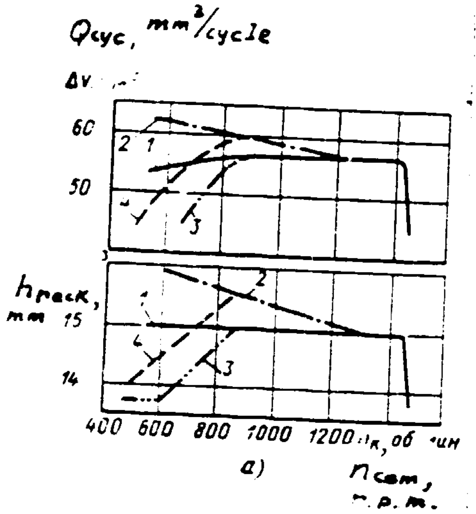

Fig.4. Combined correction of cyclic delivery versus cam speed relation.

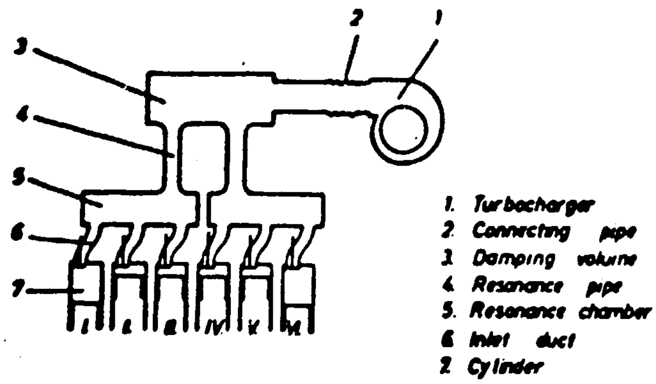

Fig.5 .
lay-out of combined resonance-turbocharging

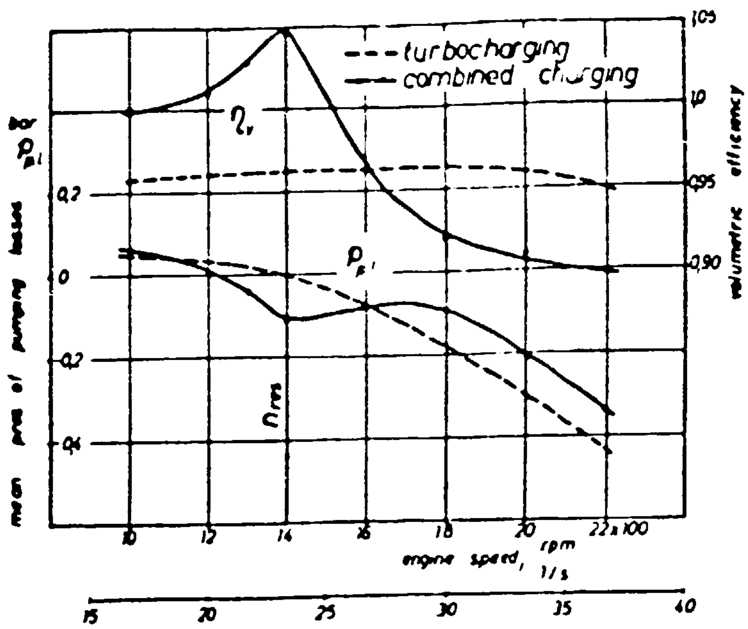

Fig. 6. Volumetric velocity and pumping losses versus engine speed.

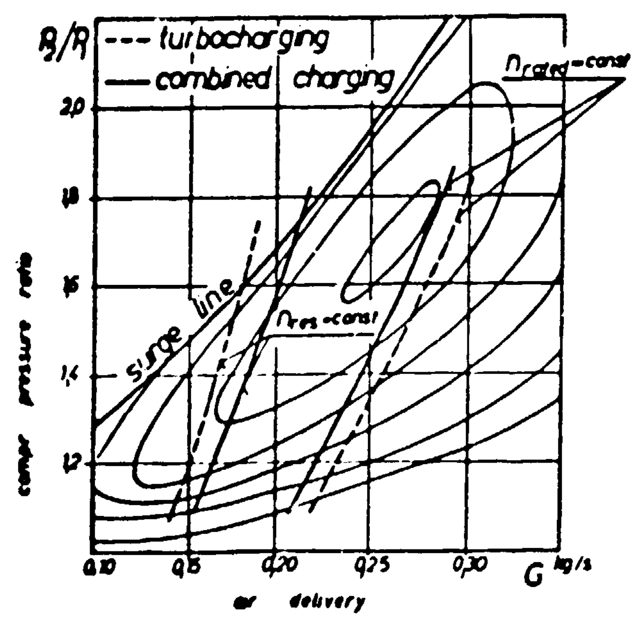

Fig. 7. Matching of engine and turbocharger.

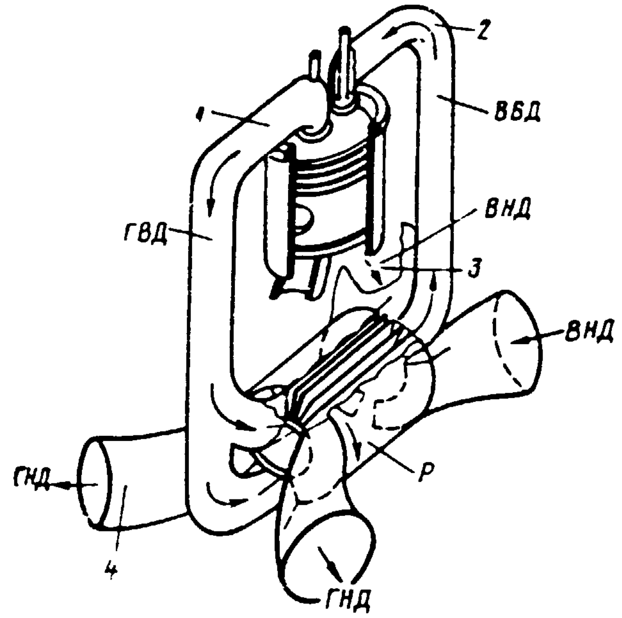

Fig.8. Comprex supercharging system

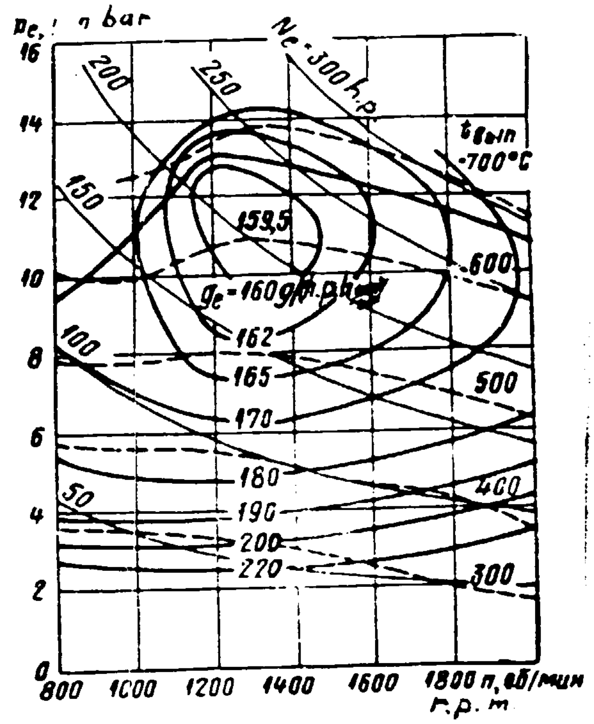

Fig. 9. Performance of turbocharged diesel engine

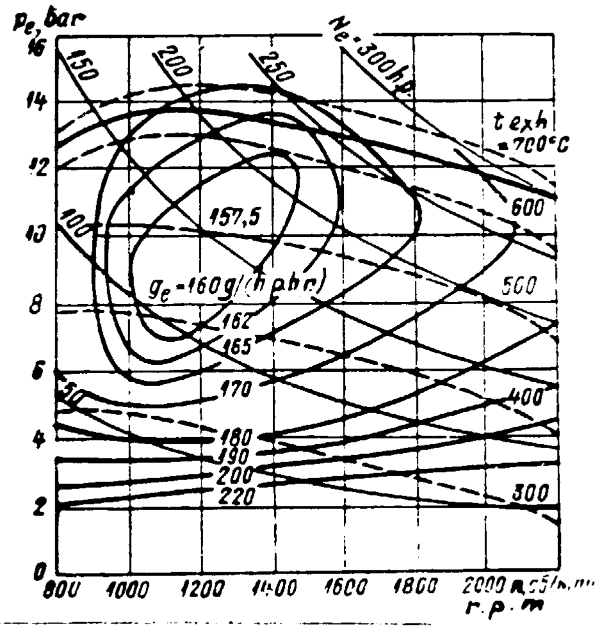

Fig.10. Performance of diesel engine with comprex system

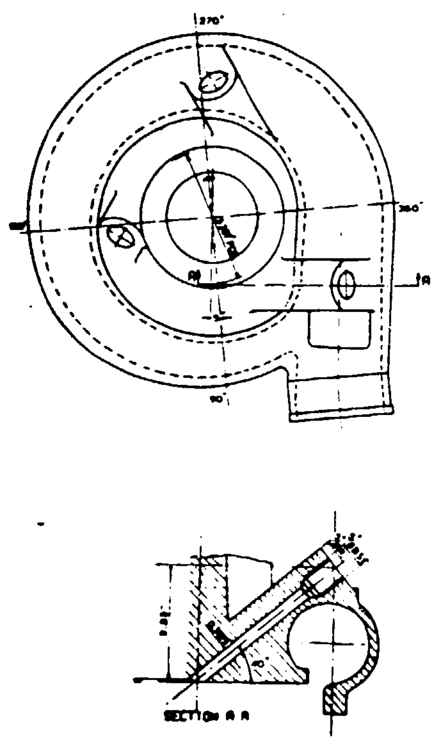

Fig.11 - Compressor casing modified for air injection

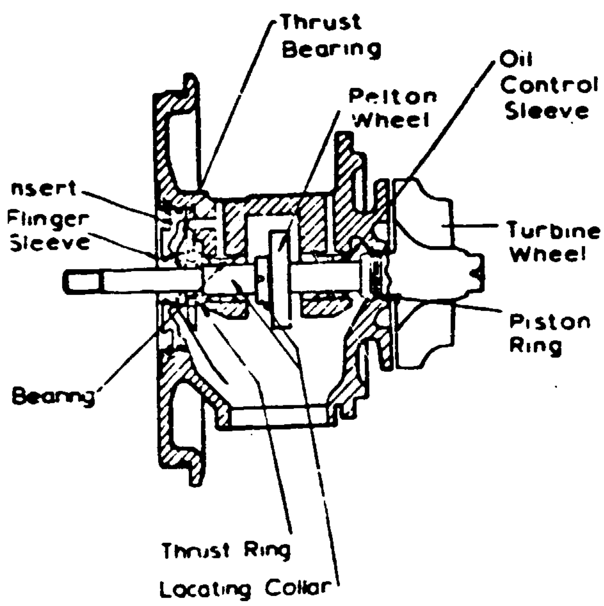

Fig.12 - Pelton wheel turbocharger

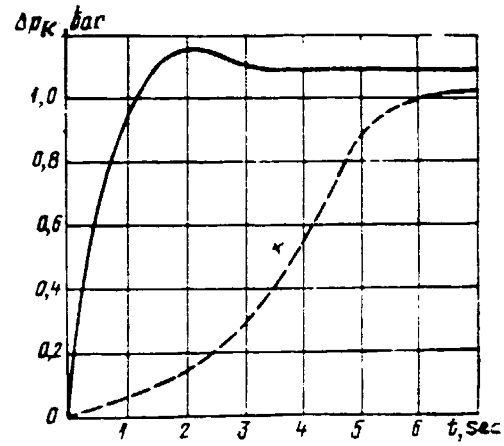

Fig.13. Transient responce of super-charged diesel engine
- engine with turbocharger
- engine with Comprex system

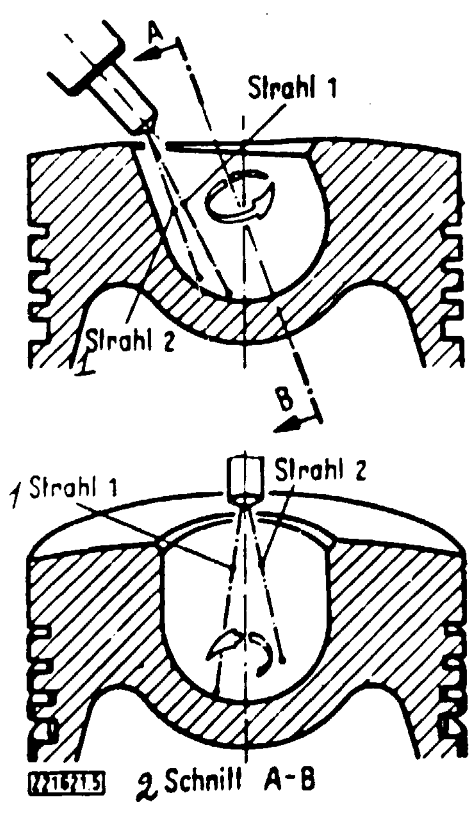

Fig.14. KHD combustion chamber

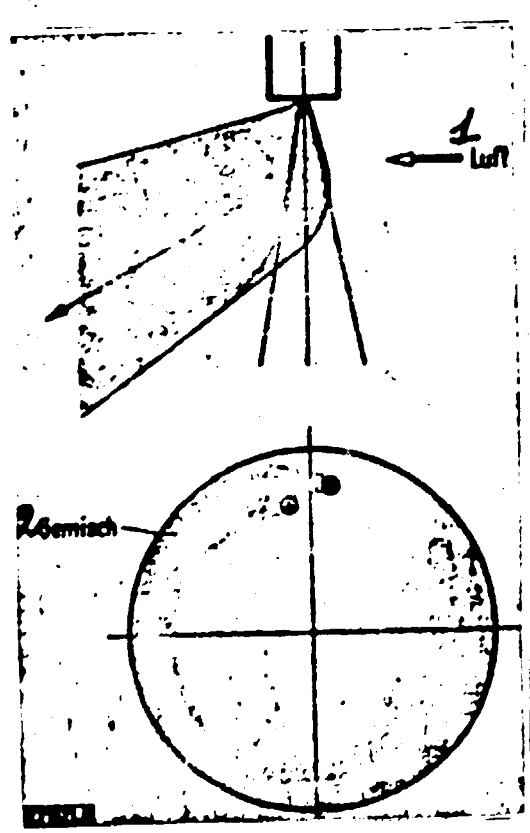

Fig.15. Scheme of air-fuel mixing in KHD combustion chamber

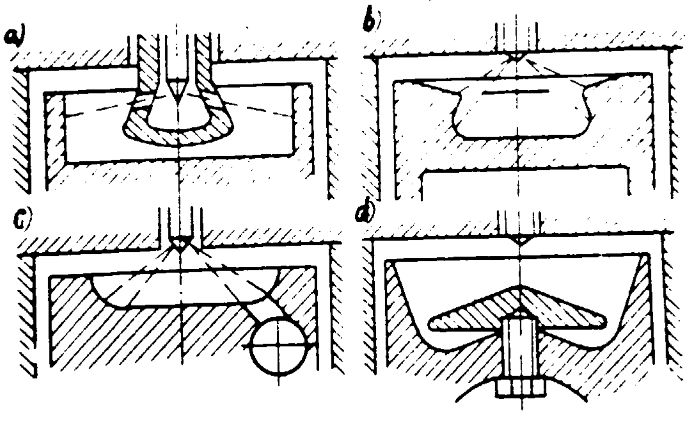

Fig. 16. Combustion chamber of first group.

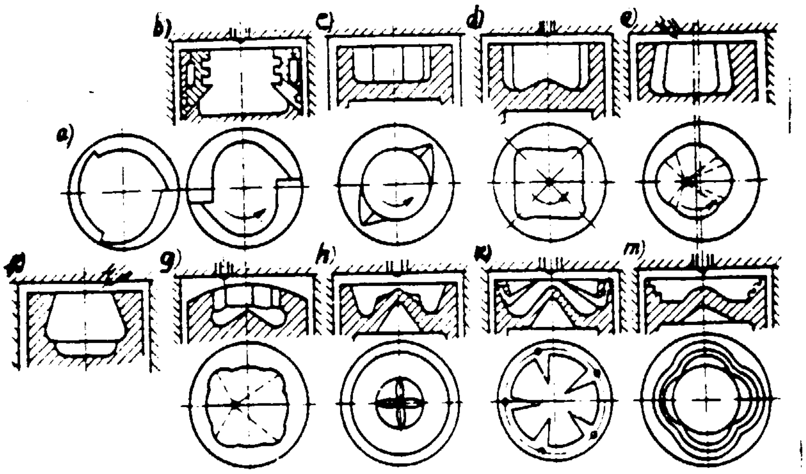

Fig.17. Combustion chamber of second group.

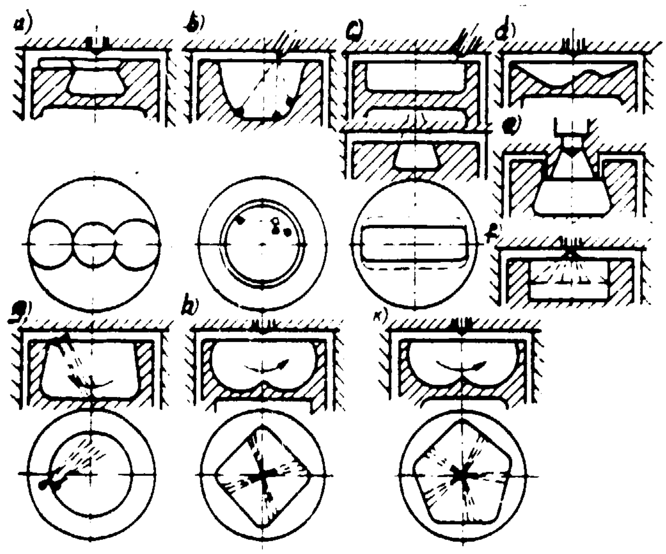

Fig.18. Combustion chamber of third group

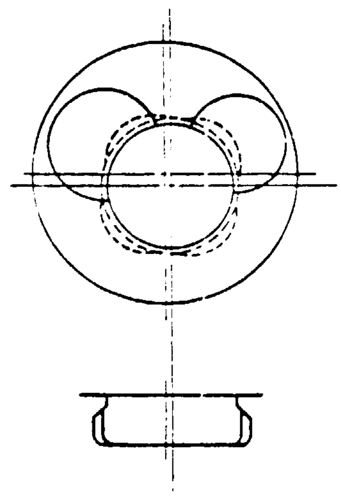

Fig.19. Hino Motors combustion chambers.

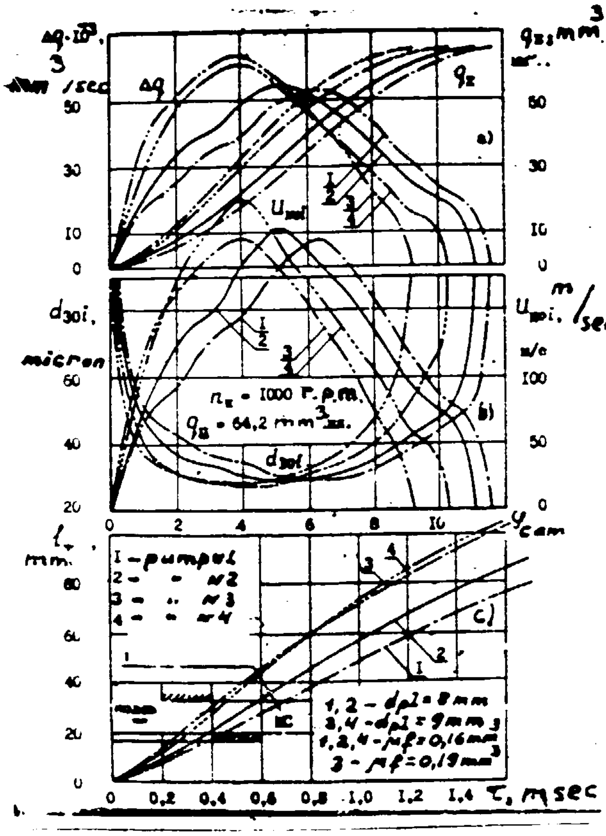

Fig.20. Injection and atomization characteristics for 4 injection systems.

a - relations of fuel injection rate (Δq) and amount of fuel injected (q_Σ) with respect to cam angle (ψ_{cam})

b - relation of linear injection velocity (u_{Koi}) and volume mean droplets diameter (d3oi) with respect to camangle.

c - propagation of sprays tip(l)with respect to time (τ).

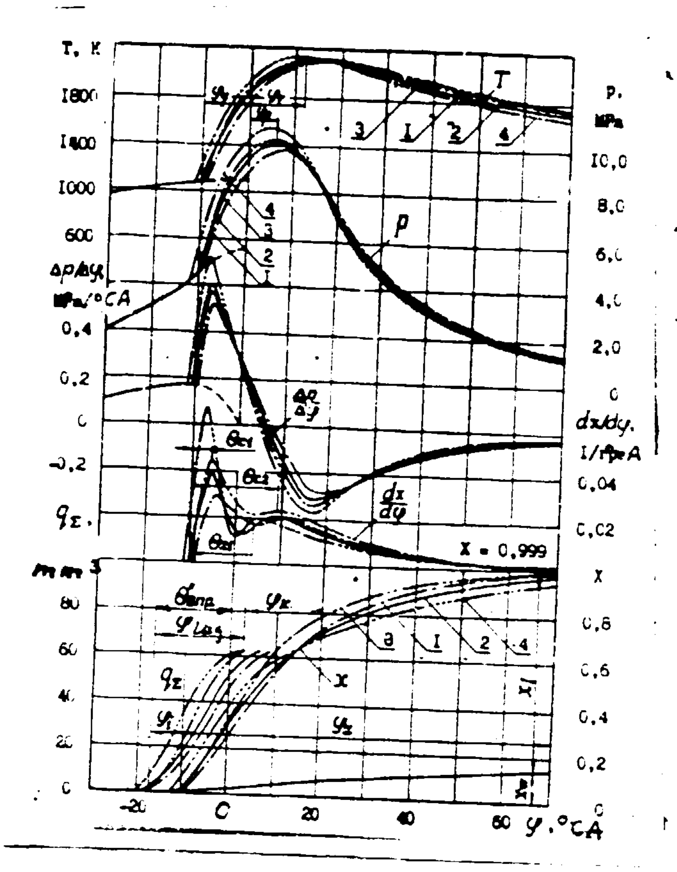

Fig. 21. Relations of charge temperatu (T), charge pressure (p), rate of pressure rise ($\Delta p/\Delta\varphi$), rate of heat release (dx/$d\varphi$), amount of fuel injected (qz) and relative amount of heat released (X) to crank angle for the same 4 injection systems as in Fig.20.

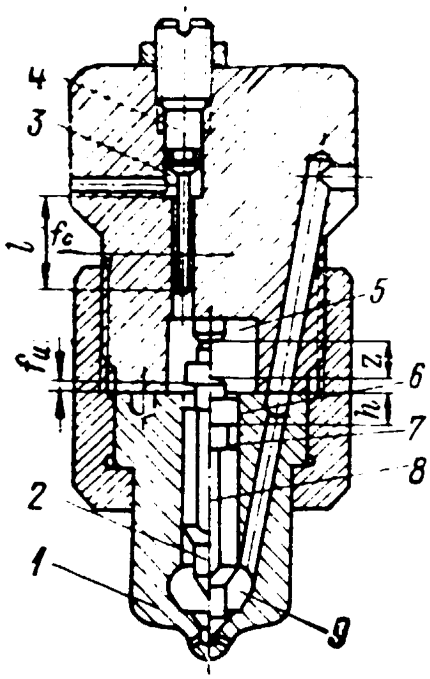

Fig.22. Nozzle with high needle closing pressure.

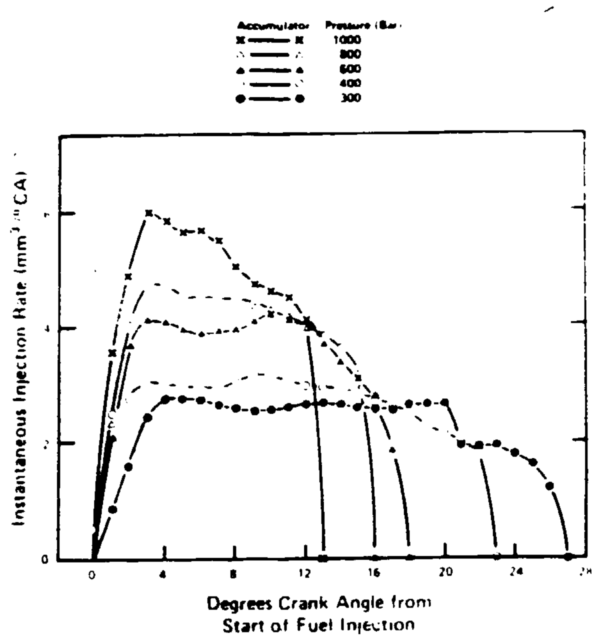

Fig.22,a. Rate of fuel injection versus crank angle in case of different accumulator pressure (CAV data).

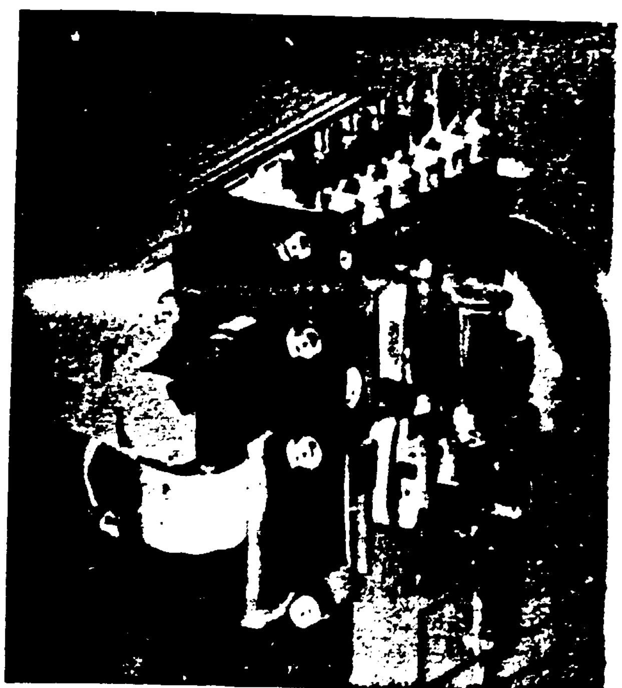

Fig.23.Pump with electronic governor.

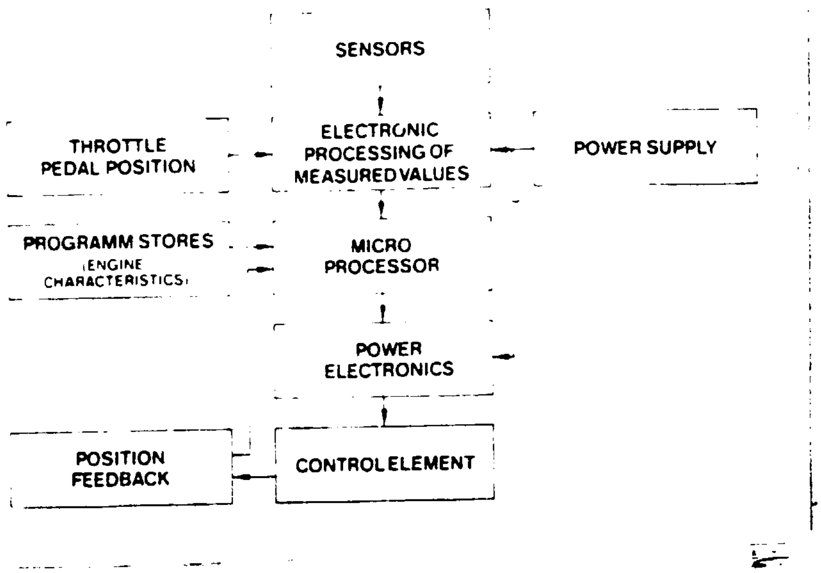

Fig. 24. Block diagram of the electronic governor

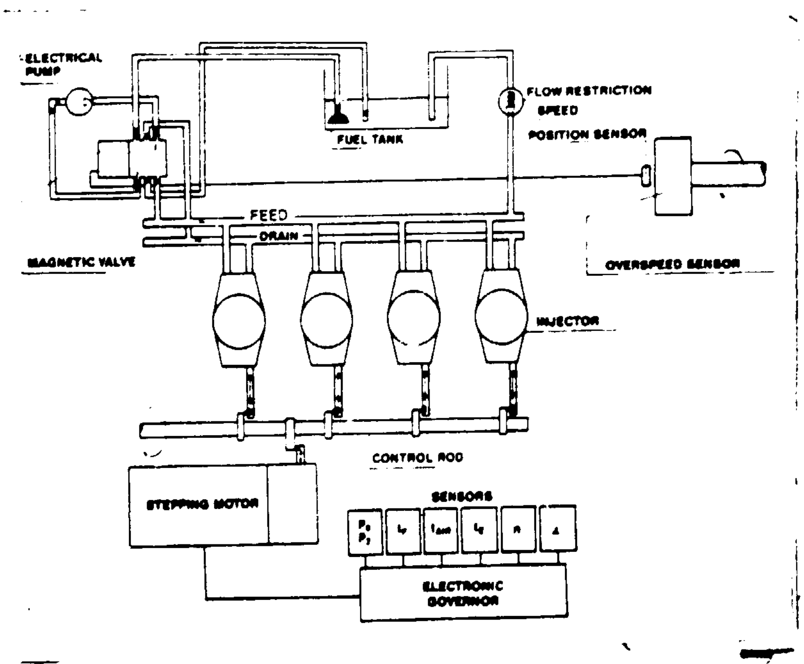

Fig.25. Unit-injectors with the electronic governor

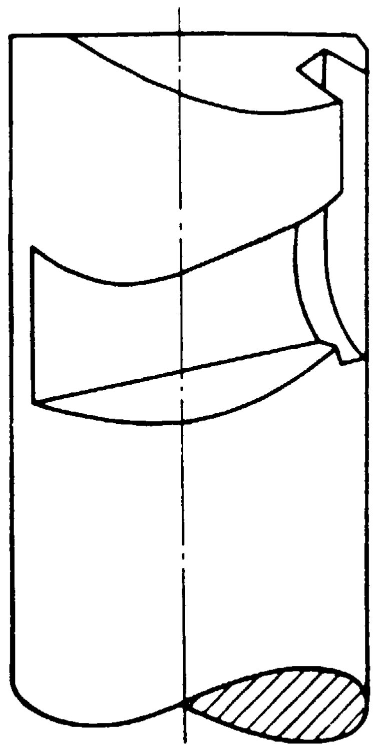

Fig. 26. Plunger design ensuring injection advance with speed increase.

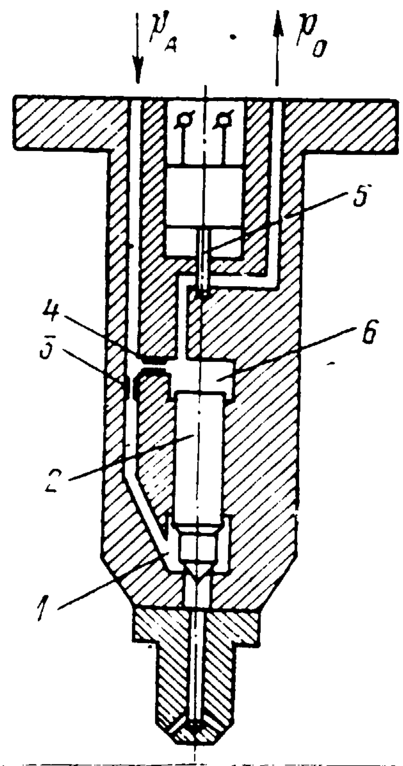

Fig. 27. Electric - hydraulic control nozzle.

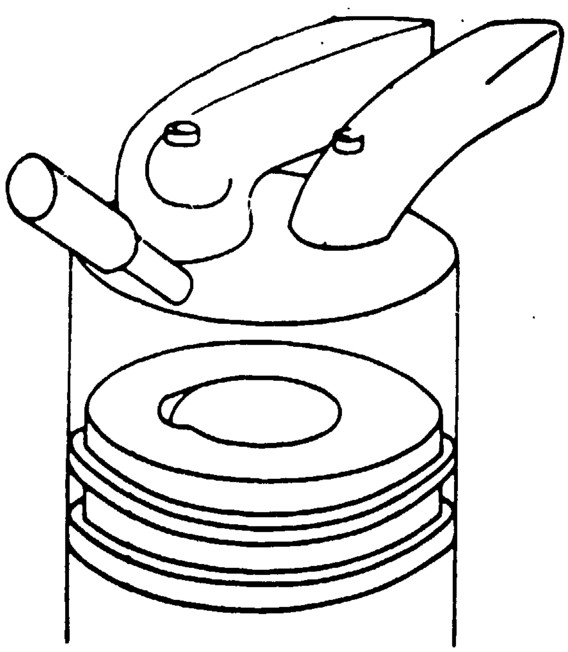

Fig 28 Wall-wetting

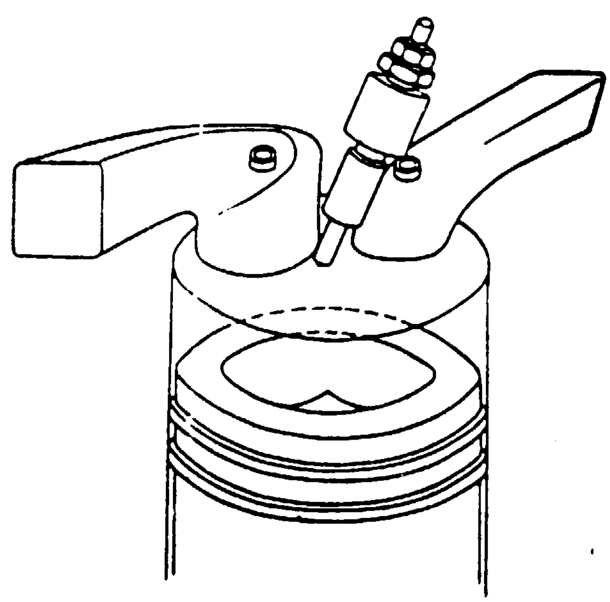

Fig 29 Conventional with pump-pipe-nozzle

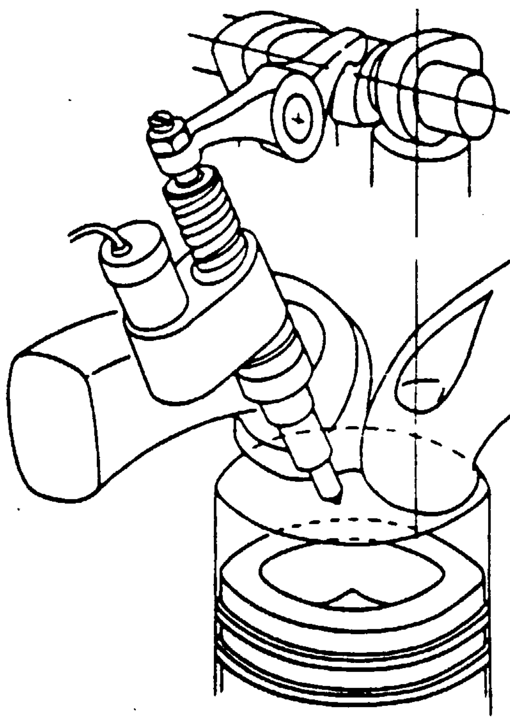

Fig 3 O Conventional with pump-injector

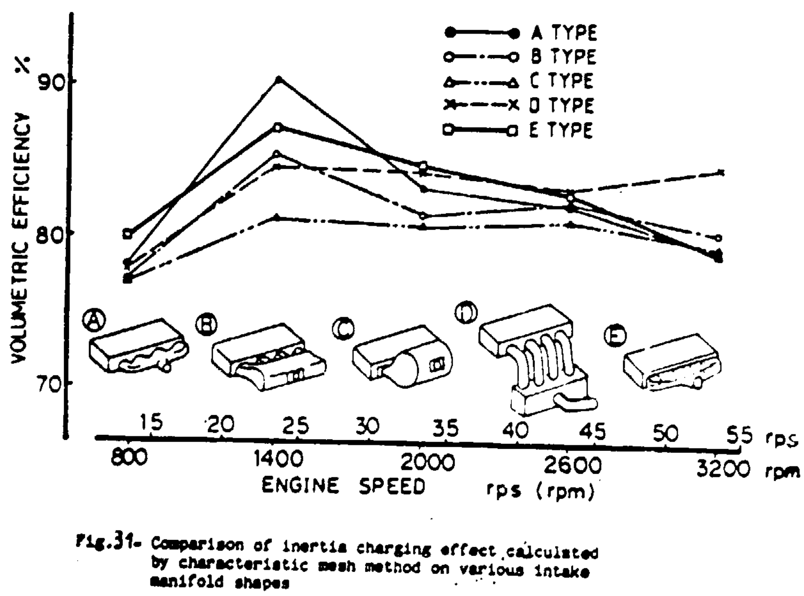

Fig.31- Comparison of inertia charging effect calculated by characteristic mesh method on various intake manifold shapes

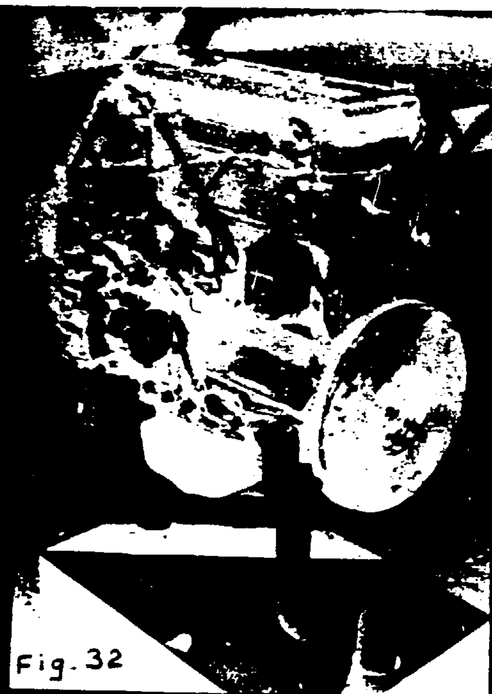

Fig. 32

Iveco turbo diesel cuts DI noise with two-section engine block, and large plastic top cover that encloses injectors as well as valve gear. The 2-L unit has 93-mm bore and 90-mm stroke.

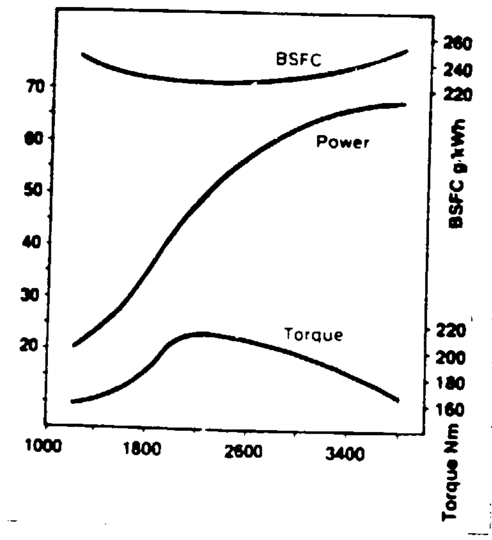

Fig. 33

Performance and fuel consumption curves of Iveco turbocharged diesel, showing gross ratings without fan, muffler, and air filter. Power is 68 kW (91 hp) at 3800 rpm, and peak torque 216 N·m (159 lb·ft) at 2200.

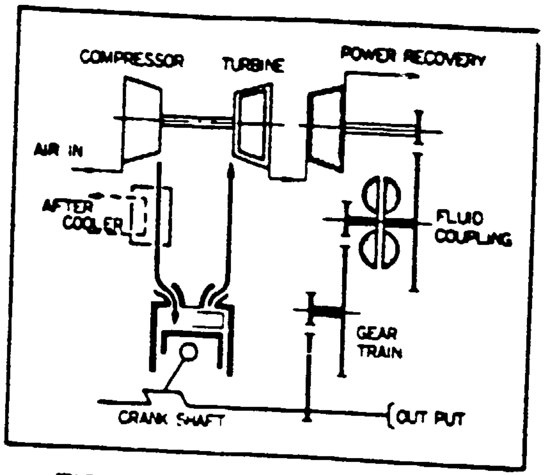

FIG.34 TURBOCOMPOUND ENGINE

(DIESEL ENGINE)

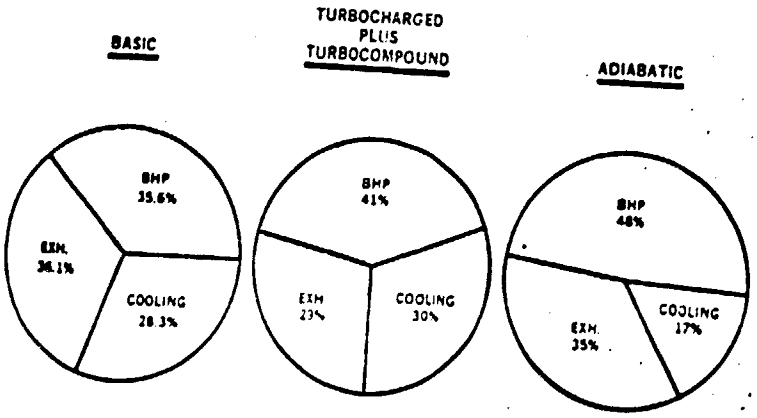

Figure 35. Energy Balance Comparison

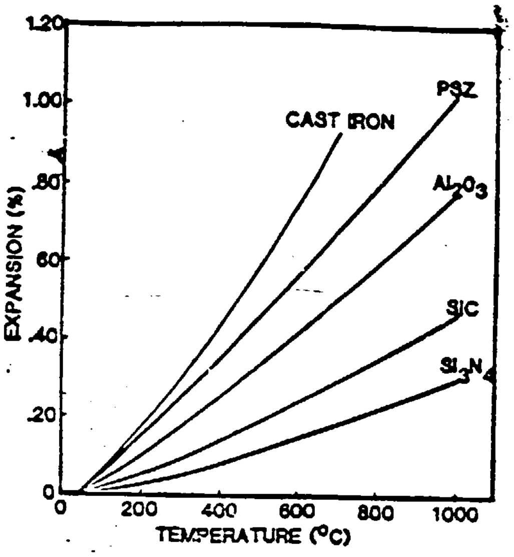

Figure 36 Thermal expansion of typical ceramics and cast iron

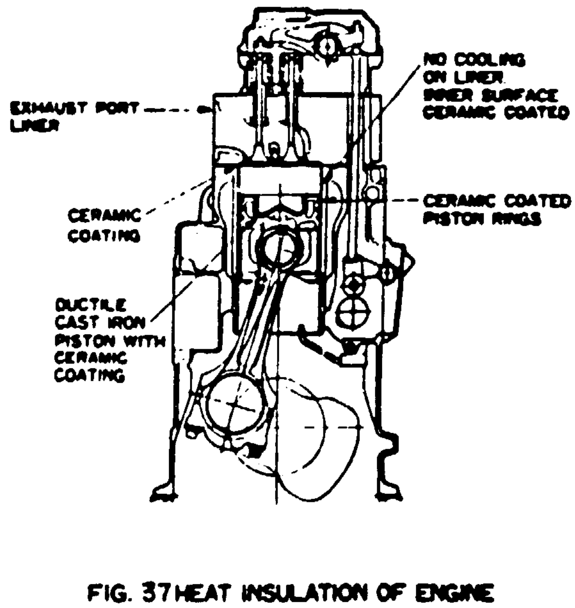

FIG. 37 HEAT INSULATION OF ENGINE

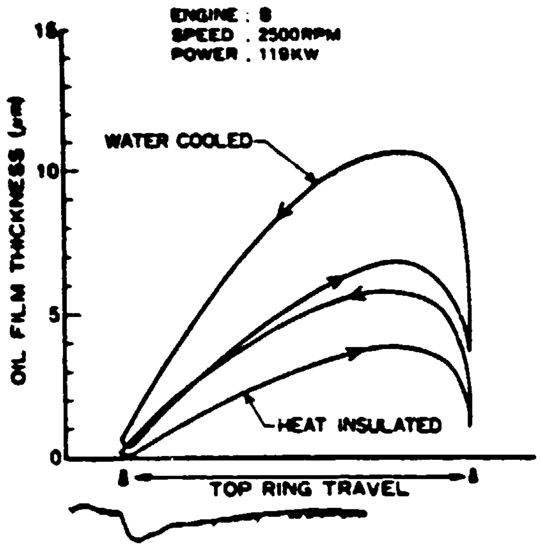

FIG. 38 CALCULATED OIL FILM THICKNESS OF TOP RING AND CYLINDER LINER WEAR PATTERN

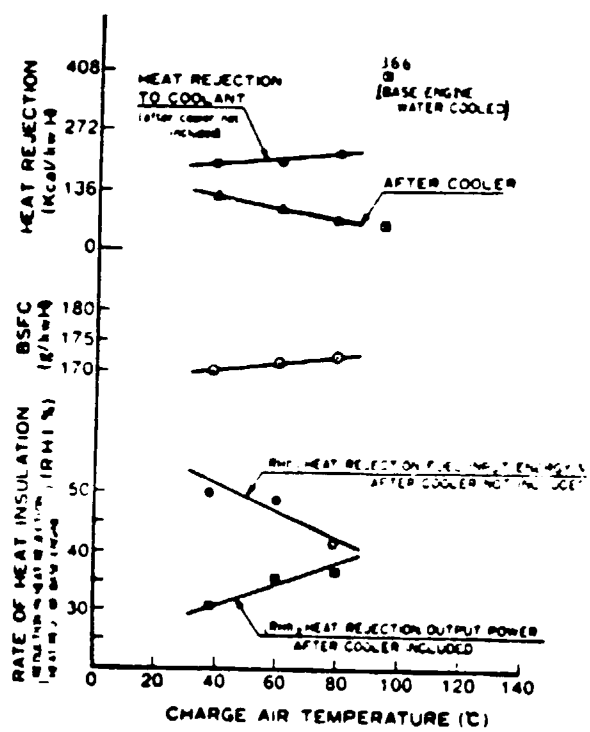

FIG. 39 HEAT REJECTION (ENGINE A)

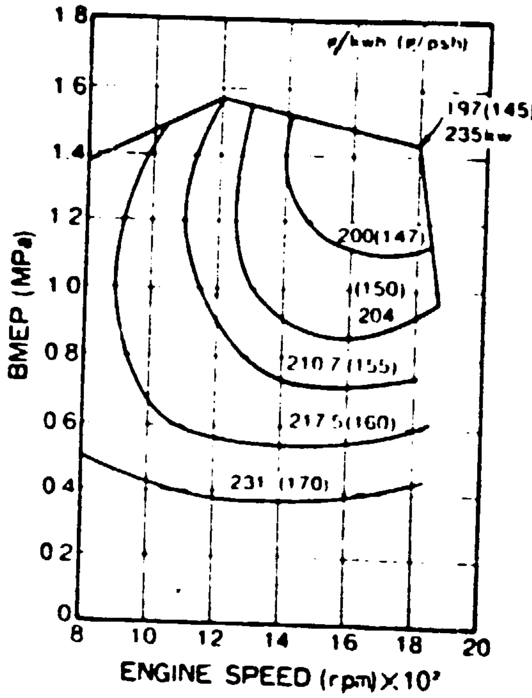

FIG. 40,a FUEL CONSUMPTION OF BASE ENGINE (WATER COOLED)

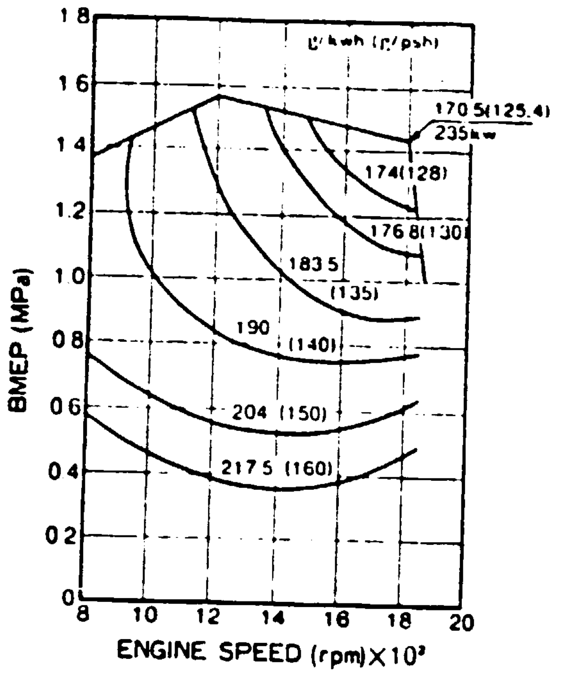

FIG. 40,b FUEL CONSUMPTION OF HEAT INSULATED TURBOCOMPOUND ENGINE

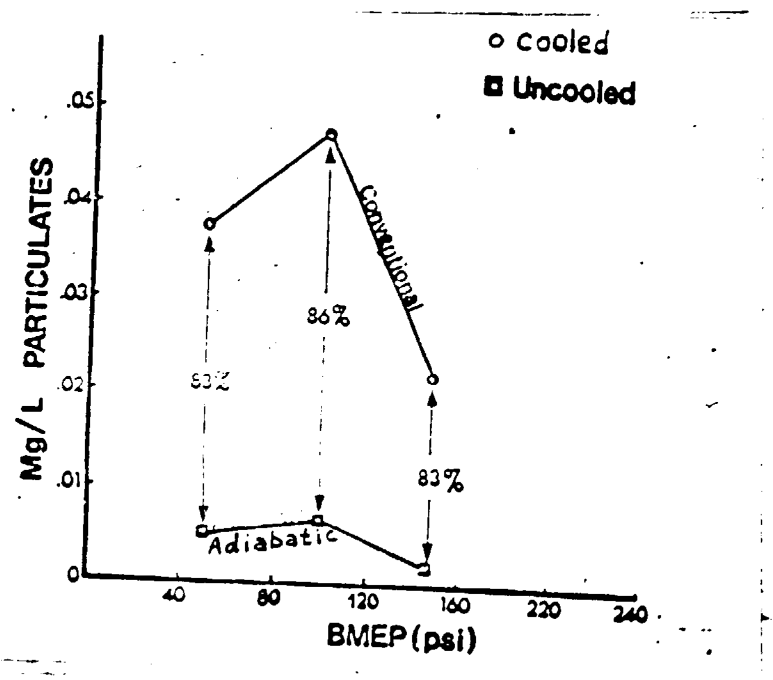

Fig. 41. Particulate emission

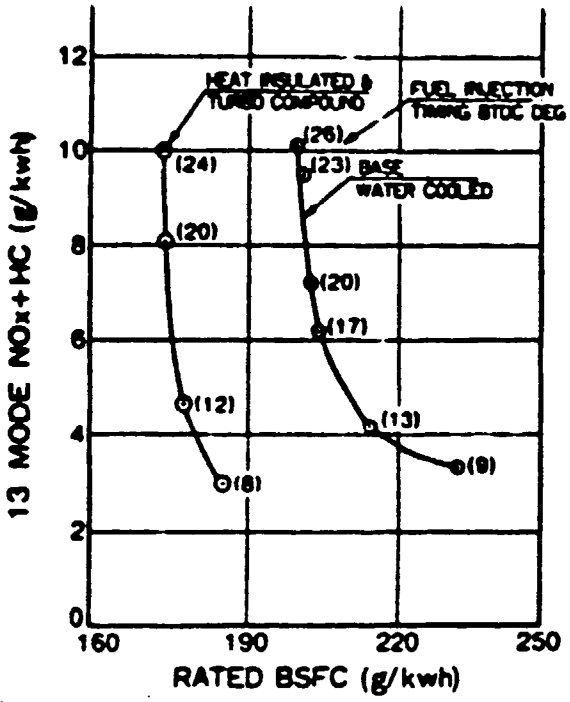

FIG. 42 NOx+HC vs. FUEL CONSUMPTION

ENGINE PROGRAMS

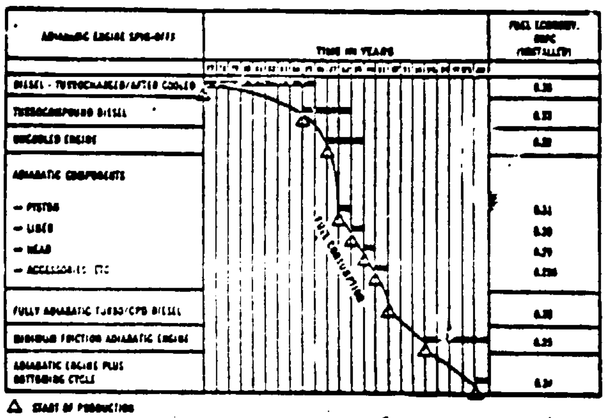

Figure 43 . Adiabatic Engine Spin-offs

Appendix 4.

Application of Alternative Fuels for
Internal Combustion Engines - UNIDO
Project DP/IND/82/001
I. I. P., Dehra Dun, INDIA.

Interim Report

"Application of methanol to four-stroke diesel engines currently under production"*

Prepared for the Government of India by the
United Nations Industrial Development
Organisation, Acting as Executive Agency for
the United Nations Development Programme

Based on the work of A.S.Khatchian
UNIDO Expert

Period under review: 11-10-1984 to 31-12-1984

United Nations Industrial Development
Organization, Vienna.

* this document has been reporduced without formal editing.

Work done by the expert

1. Visit to Department for Internal Combustion Engines and Automotive Engineering, Technical University, Vienna, Austria to study research activities in progress and to discuss the problems related to the project.

2. Visit to AVL Graz, Austria to study research equipment being used and under production, research activities in progress and to discuss the problems related to the project.

   As one of the results of this visit, the expert came to the conclusion that specification of Universal Single-Cylinder Experimental Engine with Dynamometer should be prepared by IIP and AVL should be approached to find out the cost of such a unit and period required for the supply.

   The engine should be supplied with the parts necessary to simulate operation of Indian make truck and bus diesel engines and also to certain extent future requirements of the country's transportation system and agriculture.

   Results obtained on Ricardo Single Cylinder Research Engine though of interest, can not be used directly to predict performances of Indian make truck, bus and tractor diesel engines running on conventional diesel fuel and on methanol. Performing of all research and development work on multi-cylinder engines is too costly. That is why aquiring of suitable single cylinder research engine is essential.

3. Study of the Project documents and technical reports prepared by the experts associated with the project (A. Kowalewicz, A. Cernej, P.Eyzat) and the report of Ad-hoc Expert Group Meeting on Modification of Internal Combustion Engines for Utilization of Synthetic Fuels.

4. Preparing presentation "Analysis of the methods to use pure methanol in diesel engines currently under production". Text of the presentation has been finalized in writing. It may be attached to the final report.

5. Participation in workshop "Perspective of alcohol fuel utilisation in IC Engines" at IIP, Dehra Dun.

6. Study of Engine Laboratory facilities and research work done and in progress.

7. Individual and Group discussions on different aspects of fulfillment of the project aims with Engine Laboratory Scientists (Mr.Sudhir Singhal, Project Co-ordinator, Dr.B.P.Pundir, Mr.K.K.Gandhi, Mr.D.Kumar, Mr.A.K.Aigal, Mr. S.Dass, Mr. S.K.Singhal).

8. Preparing the programme of the expert's work on the project.

9. Analysis of the results of compar ative investigation of fuel injection and atomization in case of diesel fuel and methanol carried out in Engine Laboratory prior to the expert's arrival. Development of the suggestions related to the future work in the directions specified above.

10. Analysis of the methods to use methanol in diesel engines and choice of the most promising practical methods taking into account local conditions and facilities.

11. Development of the suggestions related to the conversion of truck and public bus diesel engines into external mixing spark ignition engines.

12. Preparing the lay-out schemes of Tata truck diesel engine converted into external mixture formation spark ignition engine.

13. Choice of the types of methanol evaporators to be analysed. Calculations of heat supply and evaporation rates for several types of evaporators. Choice of main dimensions and development of design lay-outs of two evaporators.
a) Swirl chamber type b) multi-tube type. Strength analysis for the latter. The materials developed were handed over to an engineer and a draughtsman to prepare detailed drawings under expert's supervision.

14. Preparation of list of equipment to be procured to make conversion of diesel engine into pure methanol engine possible. The list was handed over to an engineer for taking necessary action.

15. Development of the suggestions related to research activities on Ricardo Single Cylinder Engine test bench obtained by Engine Laboratory through UNI DO in December 1984. Designing the engine parts necessary to investigate on the bench the method (process) suggested by Dr. V.N.Svobodov for low cetane number fuels in case when diesel engine is fed by methanol is in progress.

16. Development of the suggestions for dual fuel diesel engine operation

17. Development of the method to choose main dimensions of fuel injection system in case of alternative fuels based on theory of similarity. Application of the method to find out necessary changes in dimensions of fuel injection system to use methanol instead of diesel fuel while keeping injection pressure and duration the same in both the cases.

18. Choice of fuel injection system elements to suit methanol injection requirements on the basis of experiments. Work is in progress to compare fuel injection process with disel fuel and methanol.

19. Calculation of engine cycles with methanol and diesel-fuel. In case of methanol, calculations were performed for conditions of internal and external mixture formulation. Results of the calculations made it possible to assess effect of certain differences in fuel properties on cycle and hence engine parameters. Technical report has been prepared on the basis of the calculations and corresponding analysis. It may be attached to the experts final report. In addition to detailed technical recommendations which are given in the written presentation at the workshop (to be published) and in the technical report mentioned above (more technical reports will be prepared by the end of the mission) it is necessary to stress several important comments concerning implementation of the project :

i) Vast and detailed information related to the use of alcohol fuels in piston engines has been accumulated and analysed with high competence by the experts and IIP Scientists in the course of implementing the project. Attempts were made to inform Government of India and relevant scientific, industrial and educational organisations about world's activities in the field of production and application of alcohol fuels and different technical aspects of the use of alcohols in piston engines. Special publication under title "Alcohol Fuels Engine Applications" has been started, which is of a great use.

It is however expert's conviction that greater efforts should be made to come to definite decisions at ministerial level regarding perspective of use of methanol in India. In case corresponding decisions are taken, industry should be encouraged to start serious R&D in the field concerned. At the same time the policy in obtaining methanol at economical price and in sufficiently large quantity should be finalized.

Only on the basis of close co-operation with industry most of the technical problems concerning the use of methanol in diesel engines in India may be solved successfully.

ii) Some quite interesting research has been carried out at IIP Engine Laboratory on substitution of diesel fuel by methanol.
This research was however confined to small diesel engines whereas major part of diesel fuel is consumed by truck, bus and tractor diesel engines. In expert's opinion greater efforts should be made at the following stage of the project implementation to aquire experience in utilization of methanol in above mentioned engines.

iii) There are substantial difficulties in aquiring hardware necessary to carry out R&D on utilization of methanol in diesel engines. This alongwith unwillingness of Indian diesel engine industry (reasons given above) to co-operate in fulfilment of certain part of project aims are the reasons why experience of international experts and IIP Scientists is not yet utilized in the best possible way.

iv) As the main aim of the project is to help India in solving some economical problems related fuel supply, I think it would be proper if international experts do not limit their activities to only use of methanol in piston engines. There is substantial scope in improving efficiency of truck, public bus, tractor type diesel engines. In some models efficiency may be increased by 15-20% which will help to solve the main aim of the project. Also activities aiming to introduce supercharged diesel engines into Indian economy should be undertaken with the assistance of international experts. This will help not only in decreasing specific fuel consumption but also to save other materials as well.

Report on results of comparative cycle calculations with diesel fuel and methanol.

Based on the work done at Indian Institute of Petroleum, Dehradun by UNIDO Expert Professor A.S.Khatchian from Moscow Automobile Maintenance and Highway Construction Technical University(USSR).

Results of cycle calculations with alcohol fuels as compared to those with conventional fuels reported so far in papers (1), (2),(3) are contradictory. Differences in properties of fuels under consideration (4) affect cycle parameters in opposite ways. Some of the differences, say between methanol and diesel fuel,tend to be benefitial for methanol cycle,whereas the others may deteriorate the parameters.

As reported in (4), one of the practical methods to design pure methanol engine is by converting diesel engine into spark ignition external mixture formation engine. Prognosis of the change in engine parameters as a result of such a conversion is of interest. As a first approximation this may be done on the basis of cycle calculations.

Aims of study

1. To find out effects on cycle parameters of differences in the properties of methanol and diesel fuel - such as: ratio of number of mols prior,in the progress and at the end of combustion; fraction of three-atom gases in the charge and hence its specific heat; evaporation heat.

2. To assess the change in diesel engine parameters as a result of its conversion to spark-ignition external mixing engine.

Conditions of the study.

Combustion is assumed to take place at constant volume and constant pressure. Maximum value of cycle pressure in all the cases is equal to 85 bars.

In calculations carried out cylinder diameter (D) and piston stroke(S) are assumed to be equal to 120mm. As heat exchange between charge and engine parts is not accounted for,all the conclusions made are valid for any engine provided compression ratios and air-fuel ratios are the same as those taken in calculations performed.

The table 1 conditions of cycle calculations are presented:

Table - 1

Designation of cycles	ε	pa, pa	Ta K	γ_r	α	Fuel	Mode of fuel-air mixing	Accounting for evaporation heat.
A	17	99348,9	330	0,03	1.6	Diesel	Internal	No
B	"	"	"	"	"	Methanol	"	No
C	"	"	"	"	"	"	"	Yes
D	12	98120	340	0.0424	1.35	"	External	No

In table 1: ε - compression ratio, pa, Ta - pressure and temperature of charge at compression start, γ_r - ratio of number of mols (Nr) of residual gases to number of mols (N1)of fresh charge, α- excess air ceffícient.

In cycles A,B,C mean adiabatic exponent was taken to be equal to 1.375. In cycle D mean adiabatic exponent was calculated as

$K1m = \frac{K_1^a + K_1^c}{2}$, where K_1^a and K_1^c are values of adiabatic exponent at the start and at the end of compression. The same approach was used in calculating mean adiabatic exponent for expansion process in case of all the cycles.

Cycle D was calculated on assumption that air fuel mixture is formed outside the cylinder and methanol is evaporated completely prior to mixing with air due to heat supply from engine cooling jacket. While choosing values of input parameters for cycle D inter-relations between the parameters known from theory of internal combustion engines (5),(6) were taken into account.

The parameters were chosen by iterations to get mean pressure of cycle not less than that in case of cycle A.

Set of simplified equations used is given below:

$$Pi = \frac{1}{R_\mu} \frac{Hu}{N1} \eta_v \eta_i \frac{Pk}{Tk} \qquad (1)$$ where

Pi-mean pressure of cycle, R_μ -universal gas constant equal to 8314 J/kmol.K, Hu-lower calorific value of fuel, $N1 = \alpha L_o$ in case of internal mixing and $N1 = \alpha L_o + 1/\mu_f$ for external mixing, L_o - stoichiometric amount of air in kmols required to burn 1kg of fuel, μ_f is molecular mass of fuel, η_v -volumetric efficiency, η_i -cycle efficiency, Pk,Tk -pressure and temperature of charge at the engine inlet

$$\eta_v = \frac{\varepsilon}{\varepsilon - 1} \frac{p_a}{p_k} \frac{T_k}{T_a(1+\gamma_r)} \qquad (2)$$

$$T_a = \frac{T_k + \Delta T + \varphi \gamma_r T_r}{1 + \gamma_r} \qquad (3)$$

In equation (3) ΔT is increase of charge temperature during its induction into cylinder, φ is ratio of specific heats of residual gases and fresh charge.

$$\gamma_r = \frac{1}{\varepsilon - 1} \frac{p_r}{p_k} \frac{T_k}{T_r} \cdot \frac{1}{\eta_v} \qquad (4)$$

In equation (4) pr,Tr are pressure and temperature of residual gases at the end of exhaust stroke.

$$p_r V_c = N_r R_\mu T_r \qquad (5)$$

where Vc is clearance volume.

$$T_r = T_b \left(p_r/p_b\right)^{K_{2b}} \qquad (6)$$

In equation (6) Pb,Tb are pressure and temperature of charge at the end of expansion, K2b is adiabatic exponent of charge at the end of expansion.

$$p_b = p_z/\delta^{K_{2m}} \qquad (7)$$

where $\delta = \varepsilon/\rho$, $\rho = V_z/V_c$, V_z is charge volume at the end of

Procedure of calculations

1. Number of mols of air:

$$N_{air} = \frac{p_a V_a}{(1+\gamma_r)\,8314\,T_a} \quad \text{in case of internal mixing}$$

$$N_{air} = \frac{p_a V_a}{(1+1/\alpha L_o \mu_f)(1+\gamma_r)\,8314\,T_a} \quad \text{in case of external mixing,}$$

where Va is charge volume at the start of compression

2. Fuel delivery per cycle.

$$G_{fc} = \frac{N_{air}}{\alpha L_o}$$

3. Number of mols of residual gases

$N_r = \gamma_r \cdot N_{air}$ in case of internal mixing

$N_r = \gamma_r\,(N_{air} + N_f)$ in case of external mixing, where

$$N_f = \frac{G_{fc}}{\mu_f}$$

4. Number of mols of combustion products per 1kg of fuel burnt ($\alpha > 1$):

$N_{CO_2} = c/12$, $N_{H_2O} = H/2$, $N_{N_2} = 0{,}79\,\alpha L_o$, $N_{O_2} = 0{,}21(\alpha-1)L_o$

5. Mol fractions of components in combustion products

$r_i = N_i \Big/ \sum_{i=1}^{i=4} N_i$, where $\sum_{i=1}^{i=4} N_i = N_2$

6. Number of mols of components in residual gases

$N_i^r = r_i \cdot N_r$

7. Number of mols of components in the charge prior to combustion

$N_{CO_2}^c = N_{CO_2}^r$, $N_{H_2O}^c = N_{H_2O}^r$ $N_{N_2}^c = N_{N_2}^r + 0{,}79\,N_{air}$

$N_{O_2}^c = N_{O_2}^r + 0{,}21\,N_{air}$, $N_f = N_f^c = G_{fc}/\mu_f$ in case of external mixing $N_f^c = 0$ in case of internal mixing.

8. Total number of mols in the charge prior to combustion

$Nc = \sum_{i=1}^{i=5} Ni$ in case of external mixing

$Nc = \sum_{i=1}^{i=4} Ni$ in case of internal mixing.

9. Mol fractions of components in the charge prior to the start of combustion.

$r_i^c = Ni^c / Nc$

10. Molar specific heat at the start of compression

$\mu C_v^a = \sum_{i=1}^{i=5} \mu C_{v_i}^a \cdot r_i^c$ in case of external mixing

$\mu C_v^a = \sum_{i=1}^{i=4} \mu C_{v_i}^a \cdot r_i^c$ in case of internal mixing

11. Adiabatic exponent at the start of compression

$K_1^a = 1 + 8314 / \mu C_v^a$

12. Determining mean adiabatic exponent of compression process K1m and values of molar specific heat μC_v^c and adiabatic exponent $K1^c$ at the end of compressions by iterations. $\mu C_v^c = \sum_{i=1}^{i=5} \mu C_{v_i}^c \cdot r_i^c$ in case of external mixing

$\mu C_v^c = \sum_{i=1}^{i=4} \mu C_{v_i}^c \cdot r_i^c$ in case of internal mixing

$K_1^c = 1 + 8314 / \mu C_v^c$

13. Pressure and temperature of charge at the end of compression

$P_c = P_a \, \varepsilon^{K_{1m}} \qquad T_c = T_a \, \varepsilon^{K_{1m} - 1}$

14. Internal energy of charge at the end of compression

$U_c = \mu C_v^c \cdot t_c \cdot N_c$

15. Ratio of pressure rise during constant volume combustion

$\lambda = P_z / P_c$

16. Number of mols of components in the course of combustion

$N_{CO_2}^x = N_{CO_2}^r + (C/12)\, G_{fc}\, X \;, \quad N_{H_2O}^x = N_{H_2O}^r + (H/2)\, G_{fc}\, X$

$N_{O_2}^x = N_{O_2}^r + 0{,}21\, \alpha\, L_o\, G_{fc} - 0{,}21\, L_o\, G_{fc}\, X \;, \quad N_{N_2}^x = N_{N_2}^r + 0{,}79\, \alpha\, L_o\, G_{fc}$

$N_f^x = N_f^c - N_f^c X$ in case of external mixing

$N_f^x = 0$ in case of internal mixing.

17. Total number of mols in the course of combustion $N_x = \sum_{i=1}^{i=j} N_i^x = A + BX$

where j is equal to 5 or 4 depending on mode of air fuel mixing

$A = N_{CO_2}^r + N_{O_2}^r + 0,21\alpha L_o G_{fc} + N_{N_2}^r + 0,79\alpha L_o G_{fc} + N_f^c$

$B = \frac{C}{12} G_{fc} + \frac{H}{2} G_{fc} - 0,21 L_o G_{fc} - N_f^c$

18. Total number of mols at the end of constant volume combustion.

$$N_{z'} = p_z V_c / 8314\, T_{z'} \qquad (9)$$

$N_{z'}$ is found for several values of $T_{z'}$ assumed in the range of temperatures in which value of $T_{z'}$ is expected to come true.

19. Fractions of fuel burnt at constant volume X_{v_1} are calculated by equation (8) assuming $N_x = N_{z'}$ and $X = X_{v_1}$.

20. Mol fractions of components in the charge $r_i^{z'} = N_i^{z'} / N_{z'}$ for each value of X_{v_1} is calculated using expressions given above and assuming $N_i^x = N_i^{z'}$ and $X = X_{v_1}$.

21. Molar specific heat of charge for each value of X_{v_1}

$$\mu C_v^{z'} = \mu C_{v_i}^{z'} \cdot r_i^{z'}$$

22. Internal energy of charge for each value of $T_{z'}$ and corresponding values of $N_{z'}$ and X_{v_1}

$$U_{z'} = \mu C_v^{z'} \cdot t_{z'} \cdot N_{z'}$$

23. Fractions of fuel burnt at the end of constant volume combustion according to energy balance equation for each value of $T_{z'}$ and corresponding values of $N_{z'}$ and X_{v_1}

$$X_{v_2} = \frac{U_{z'} - U_c}{G_{fc} \cdot H_u} \quad \text{for cycles } A, B, D$$

$$X_{v_2} = \frac{U_{z'} - U_c}{G_{fc}(H_u - \Delta H_u)} \quad \text{for cycle } C$$

24. Temperature of charge at the end of constant volume combustion is found graphically by plotting relations

$$X_{v_1} = f(t_{z'}) \quad \text{and} \quad X_{v_2} = f(t_{z'})$$

Equations given above may be also solved by iteration using computer programme.

25. Calculation of $N_{z'}$, $U_{z'}$, $\mu C_v^{z'}$, $r_i^{z'}$ and $\mu_{z'}$ using X_v and $t_{z'}$ found above. $\mu_{z'} = N_{z'}/N_c$

26. Number of mols of components at the end of combustion

$$N_i^z = N_i^x \quad if \quad x = 1$$

27. Total number of mols at the end of combustion

$$N_z = \sum_{i=1}^{i=j} N_i^z$$

28. Mol fractions of components in final combustion products

$r_i^z = r_i$ (see 5 above)

29. Ratio of number of mols after and prior to combustion

$$\mu_z = N_z / N_c$$

30. Temperature of charge at the end of combustion by energy balance equation.

$$(1-x) G_{fc} (H_u - \Delta H_u) + p_z V_c + U_{z'} = U_z + N_z \, 8314 \, T_z \qquad (10)$$

$U_z = u_z \cdot N_z$ where $u_z = \sum_{i=1}^{i=j} u_{zi} r_i$ is internal energy of 1kmol of combustion products. Values of u_{zi} for components are taken from tables given in (5) and (6).

Equation (10) is being solved graphically or by iterations. As a result of the solution value of T_z is determined.

31. Volume of charge at the end of combustion

$$V_z = N_z \, 8314 \, T_z / p_z$$

32. Ratio of volumes during constant pressure combustion

$$\rho = V_z / V_c$$

33. Molar specific heat μC_v^z at the end of combustion

$$\mu C_v^z = \sum_{i=1}^{i=j} \mu C_{v_i}^z \cdot r_i$$

Values of $\mu C_{v_i}^z$ are taken from table given in (5)

34. Adiabatic exponent at the start of expansion

$$K_2^a = 1 + 8314 / \mu C_v^z$$

35. Determining mean adiabatic exponent of expansion process K_{2m} and values of molar specific heat μC_v^b and adiabatic exponent K_2^b at the end of expansion.

$$\mu C_v^b = \sum_{i=1}^{i=j} \mu C_{v_i}^b \tau_i \quad , \quad K_2^b = 1 + 8314 / \mu C_v^b$$

36. Pressure and temperature at the end of expansion

$$p_b = p_z / \delta^{K_{2m}} \quad , \quad T_b = T_z / \delta^{K_{2m}-1}$$

37. Work done during compression

$$L_{comp} = \frac{p_c V_c}{K_{1m} - 1}\left(1 - \frac{1}{\varepsilon^{K_{1m}-1}}\right)$$

38. Work done during constant pressure combustion

$$L_{p=const} = p_z V_c (\rho - 1)$$

39. Work done during expansion

$$L_{exp} = \frac{p_z V_z}{K_{2m} - 1}\left(1 - \frac{1}{\delta^{K_{2m}-1}}\right)$$

40. Cycle work

$$L_{cycle} = L_{p=const} + L_{exp} - L_{comp}$$

41. Mean pressure of cycle

$$p_{cycle} = p_a \frac{\varepsilon^{K_{1m}}}{\varepsilon - 1}\left[\lambda(\rho - 1) + \frac{\lambda \rho}{K_{2m} - 1}\left(1 - \frac{1}{\delta^{K_{2m}-1}}\right) - \frac{1}{K_{1m} - 1}\left(1 - \frac{1}{\varepsilon^{K_{1m}-1}}\right)\right]$$

42. Cycle efficiency

$$\eta_{cycle} = \frac{L_{cycle}}{G_{fc} \cdot H_u}$$

Calculations of cycle parameters in case of constant volume and constant pressure combustion are most complicated as iterations are necessary to find out parameters at the end of constant volume or constant pressure combustion. In cases when mode of heat release is given in input data calculations are simplified.

Discussions of calculations results.

In table 2 conditions and results of calculations are given. The following observations seem to be of importance:

1. Cyclic heat delivery with methanol in case of internal mixing at conditions outlined above is greater because of higher calorific value of air fuel mixture.

This is one of the reasons why mean pressure of cycle with methanol is higher than with diesel fuel by 4.17% if heat of evapcration is not taken into account.

2. Greater increase in number of mols due to methanol combustion as compared to diesel fuel combustion affects cycle parameters positively. Due to greater value of $\mu_{z'}$ maximum combustion pressure is obtained in case of methanol with slightly lower heat input at constant volume conditions in spite of greater three-atom gases content in the charge and hence higher values of specific heat and lower value of temperature (compare data related to cycles A and B in table - 2). Increase in number of mols compensates greater specific heat values. Therefore the same pressure is reached at lower temperature and hence lower thermal loading of engine parts. From gas equations written for the start and end of constant volume combustion one gets

$$T_{z'} = T_c \frac{p_z}{p_c} \cdot \frac{1}{\mu_{z'}}$$

As for conditions of comparison $T_c \frac{p_z}{p_c}$ is the same in case of both the fuels

$$\frac{T_{z'}\ \text{methanol}}{T_{z'}\ \text{diesel fuel}} = \frac{\mu_{z'}\ \text{diesel fuel}}{\mu_{z'}\ \text{methanol}}.$$

Difference in $T_{z'}$ is about 3% (compare parameters of cycle A and B in table-2)

3. At conditions of constant pressure combustion greater increase in number of mols due to methanol combustion as compared to diesel fuel combustion is not fully compensating greater values of molar specific heat and greater number of mols (ρ increase by 2.28%, whereas heat input at constant pressure is greater by 6.7%). It is however clear that

greater value of μ_z helps in obtaining higher work done at ρ =const in spite of lower temperature at the end of combustion, which again helps in decreasing thermal loading of engine parts. It is worth noting in comparing combustion of diesel fuel and methanol that differences in $\mu_{z'}$ affect cycle parameters to a greater extent than differences in μ_z because during constant volume combustion differences in molar specific heat values and in total number of mols in case of both the fuels are yet not as great as they get at conditions of constant pressure combustion. That is why at conditions of V= const positive effect of greater $\mu_{z'}$ prevails as compared to the effects of higher molar specific heat values and greater number of mols.

4. Net effect of change in number of mols prevails upon two other effects considered above and therefore when evaporation heat is not taken into account efficiency of cycle is slightly higher in case of methanol combustion (compare cycles A and B in table-2) Comparison of mean cycle pressure for the same cases made above is also in favour of methanol, but increase in ρ cycle is much greater than increase in η cycle. It is worth noting that if heat exchange between charge and engine parts was taken into account advantages of the methanol cycle would had been substantially greater. This follows from the fact that temperature during expansion is considerably lower with methanol as compared to diesel fuel. Difference in T_z is equal to 137° and in Tb to 49°. Lower expansion ratio(δ) in case of methanol cycle is the reason why greater value of pressure at the end of expansion is obtained.

5. Evaporation of methanol plays significant part in cycle parameters obtained by calculations. Efficiency and mean pressure of cycle are lowered by 7.8% (compare data for cycles B and C in table-2). It is therefore advisable to use modes of air fuel mixing in which methanol is directed towards walls of combustion chamber. This will enable at least partial utilization of heat otherwise lost to cooling medium to heat and evaporate methanol. The observation made may be one of the reasons why high engine parameters were obtained on MAN engines with methanol injection and FM - process (8).
In case of heat exchange is taken into account decrease in cycle parameters due to evaporation will be slightly less. This may be expected from comparison of charge temperature during expansion in cycles B and C (See table-2).

6. In case of external fuel air mixture formation and complete evaporation of methanol in special evaporator (9) part of heat rejected into cooling medium is utilized. This helps to get high cycle parameters. In spite of decrease in compression (from 17 down to 12) and relative air fuel ratio (from 1.6. down to 1.35) efficiency of cycle is greater than in case of internal air fuel mixing by 1.6% (compare cycles C and D). Considerably lower compression work due to lower compression ratio and also due to lower adiabatic exponent helps to get high cycle parameters in case of external mixing. Though charge temperature is higher in case of D cycle one may expect admissible thermal loading of engine parts due to homogeneous combustion which ensures lower radiant heat transfer.

7. Mean pressure of cycle D is higher than in cycle A approximately by 2%. This proves that assessment of necessary excess of air

for cycle D to get the same ρ cycle as in cycle A was sufficiently precise. ($\alpha = 1{,}393$ differs from $\alpha = 1{,}35$ by 3%, hence inacuracy of the assessment is of the order of 1% as ρ cycle is proportional to $1/\alpha$). The main reasons due to which excess of air should be decreased in cycle D as compared to cycle A to get the same mean pressure of cycle are:

a) Decrease in efficiency due to decrease in compression ratio and necessity to use less lean mixture.

b) Decrease in air charge supplied per cycle due to induction of fuel vapour along with air and slight increase of induction system resistance as a result of incorporation of vapourized fuel-air mixer into induction system. Effect of second factor is greater than that of the first by 45%.

Following main conclusions may be made on the basis of analysis carried out:

1. In case of internal mixing and fuel evaporating due to heat supplied from the charge methanol cycle is inferior as compared to diesel fuel cycle. Efficiency is lower by 7,54% and mean pressure of cycle by 3,97%.

2. In case of internal mixing and fuel evaporation due to otherwise wasted heat methanol cycle is superior as compared to diesel fuel cycle. Efficiency is higher by 0.3% and mean cycle pressure is higher by 4,17%. Therefore modes of air fuel mixing in which methanol is directed to combustion chambers walls should be prefered. If heat losses to cooling medium are taken into account inferiority (First mode of comparison) would decrease and superiority (second mode of comparison) of methanol cycle

would increase as charge temperatures in methanol cycles are lower than those in diesel fuel cycle.

3. In getting higher efficiency at lower charge temperature greater increase in number of mols due to methanol combustion as compared to diesel fuel combustion plays positive part. Increase in mean cycle pressure is higher than that of efficiency due to higher claorific value of air fuel mixture in case of methanol when internal mixing is used.

4. The fact that in reality at high load conditions efficiency of diesel engine fed by methanol is in most cases higher than in base engine consuming diesel fuel even if fuel is atomized in combustion chamber volume (and hence is evaporated due to heat taken from charge) when compared with results of cycles analysis (see above conclusion 1) seems to prove that air fuel formation and combustion processes in case of methanol may be more efficient than in case of diesel fuel.

5. One of the promising ways to use pure methanol in diesel engines currently under production is conversion of the same into spark ignition engines with external air fuel mixture fomation and evaporation of methanol in special evaporator utilizing part of the heat rejected into cooling medium or with exhaust gases. Cycle of such an engine is more efficient than that of with internal mixing and fuel evaporation due to heat taken from the charge (compare cycles D and C) in spite of lower values of compression ratio and excess of air.

6. To get mean cycle pressure in spark ignition external fuel air mixing methanol engine the same as in base diesel engine it is sufficient to decrease excess of air by 13-15%. The main reasons

due to which excess of air should be decreased are decrease in efficiency and in amount of air charge inducted into cylinder per cycle, effect of the latter being greater.

Experience so far gained in practice (8), (9) seems to be in conformity with the results of cycles analysis given above.

Table - 2

D	C	B	A	Symbol of Cycle	
Methanol	Methanol	Methanol	Diesel	Fuel	
External	Internal	Internal	Internal	Fuel-Air Mixing Mode.	
12	17	17	17	Compression ratio ε	
1,35	1,6	1,6	1,6	Relative air fuel ratio α	
147,56147	141,32248	141,32248	63,88	Cyclic fuel Delivery, mg/cycle	
0	242,54	0	0	Heat spent to heat, evaporate and over heat fuel vapours J/Cycle	
2945,7623	2819,303	2819,303	2714,9	Total heat input Q1 J/cycle	
0,0424	0,03	0,03	0,03	Residual gas coefficient γ_r	
0,9812	0,993489	0,993489	0,993489	Pressure at the start of compression p_a, bars	
340	330	330	330	Temperature at the start of compression T_a, K	
51,391569	52,216054	52,216054	52,211	Number of Kmols prior to combustion Nc.10^6	
0,00363297		0,0022421	0,00256	$r^c_{CO_2}$	Fractions of components in charge prior to combustion (by volume)
0,00731114		0,0045146	0,00223	$r^c_{H_2O}$	
0,7148352		0,787332	0,7891	$r^c_{N_2}$	
0,1845437		0,2059113	0,2061	$r^c_{O_2}$	
0,0896709	0	0	0	r^c_{fuel}	
1,3353260	1,375	1,375	1,375	Compression mean adiabatic exponent, K1m	
27,090456	48,868649	48,868649	48,868649	Pressure at compression end p_c, bars	
782,27	954,8	954,8	954,8	Temperature at compression end T_c, K	

Table 8 (continued)

26,962704	22,407143	22,407143	22,3785	Molar specific heat of products at compression and, μC_v^c, KJ/Kmol
706,19	797,7146	797,7146	797,306	Internal energy of products at compression and U_c, J
53,493	54,8	54,578	52,37	Number of Kmols at the end of constant volume combustion $N_{z'} \cdot 10^6$
0,081	0,042	0,038	0,035	$r^{z'}_{CO_2}$ Fractions of components at the end of constant volume combustion (by volume)
0,1625	0,095	0,079	0,031	$r^{z'}_{H_2O}$
0,688	0,74	0,744	0,778	$r^{z'}_{N_2}$
0,06	0,133	0,139	0,156	$r^{z'}_{O_2}$
0,0085	0	0	0	$r^{z'}_{fuel}$
0,252	0,127	0,117	0,066	Fraction of three -atom & fuel molecules in charge at the end of constant volume combustion
1,0408346	1,0434658	1,0452341	1,0145371	Coefficient of change in number of mols at the end of constant volume combustion $\mu_{z'}$
2358	1582,5	1589,5	1637,75	Temperature at the end of constant volume combustion, $T_{z'}$, K
30,08	25,15	25,075	25,01	Molar specific heat of products at the end of constant volume combustion $\mu C_v^{z'}$ KJ/Kmol K
3354,3037	1805	1801,5	1808	Internal energy of products at the end of constant volume combustion, $U_{z'}$
0,898	0,39	0,355	0,3725	Fraction of heat released by the end of constant volume combustion Iv
26 .2945	1099,53	1000,85	1011,3	Amount of heat released by the end of constant volume combustion $Q_{z'}$, J/cycle
53,733559	58,868	58,868	54,215	Number of Kmols at the end of combustion $N_z \cdot 10^6$
0,0893163	0,0769793	0,0769793	0,0879	$r^{z}_{CO_2}$ Fractions of components at the end of combustion
0,1798305	0,1550419	0,1550419	0,0765	$r^{z}_{H_2O}$
0,6836763	0,6983633	0,6983633	0,76	$r^{z}_{N_2}$
0,0471169	0,0696153	0,0696153	0,0756	$r^{z}_{O_2}$

Table 2 (continued)

0,2592068	0,2320212	0,252212	0,1644	Fraction of three-atom products at the end of combustion
1,0455755	1,1274	1,1274	1,038	Coefficient of change in number of mols at end of combustion μ_z
2539	2127	2225	2365	Temperature at the end combustion T_z, K
29,33654	28,024357	28,245046	27,3039	Molar specific heat of products at the end of combustion μC_v^z, kJ/kmol K
3572,0608	3058	3252	3578,56	Internal energy of products at the end of combustion U_z, J
3,1376363	1,73935	1,73935	1,73935	Ratio of pressure change due to combustion λ
815787	1,44385	1,51241	1,4786	Ratio of volume change during **constant volume** combustion ρ
11,094895	11,774	11,24	11,49736	Ratio of volume change during expansion δ
25,746908	24,405594	24,578279	24,541682	Molar specific heat of products at expansion end μC_v^b, kJ/kmol K
1224,132	969,39	1038,17	1086,95	Temperature of charge at expansion end T_b
1,2833979	1,2966705	1,2945525	1,2978873	Adiabatic exponent at the start of expansion K_z^z
1,3229125	1,3406596	1,3368954	1,3327705	Adiabatic exponent at the end of expansion K_z^b
1,303155	1,3196605	1,3156239	1,3193299	Expansion mean adiabatic exponent K2r
3,693699	3,2905	3,52375	3,397851	Pressure of charge at expansion end p_b, bars
85,553267	320,01573	369,44748	345,07048	Work produced during constant pressure combustion $L_{p.const}$, J
1937,6371	1777,9318	1845,0352	1809,7737	Expansion work, L_{exp}, J
563,5546	723,35967	723,35967	723,35967	Compression work L_{comp}, J

Table 2 (continued)

Cycle work, L_{Cycle}, J	1459,6557	1374,5878	1491,123	1431,4345
Mean pressure of cycle p_{cycle}, bars	10,7551	10,128354	10,987	10,5475
Efficiency of cycle η_{cycle}	0,49545	0,487563	0,5289	0,5273
$\frac{\eta_{Cycle\ 1} - \eta_{Cycle A}}{\eta_{Cycle A}} \times 100$ %	- 6,04	-7,54	+ 0,3	0
$\frac{p_{Cycle\ 1} - p_{Cycle A}}{p_{Cycle A}} \times 100$ %	1,97	- 3,97	+4,17	0

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Appendix 6

Report

on results of business trip to Delhi, Bombay, Pune, Madras undertaken by UNIDO Expert, A.S.Khatchian from 6th February 1985 to 14th February, 1985 (inclusive).

Aims of the trip as established after discussions with project coordinator Mr. S.Singhal.

1. To study research work going on in the lines of the project DP/IND/82/001.

2. To try to establish some new links of cooperation with different institutes and firms in the lines of the same project.

3. To collect technical data and to find out possible ways of aquiring some hardware necessary to implement aims of the project.

Work done during the trip at IIT, Bombay

1. Study of research activities going on in the lines of the UNIDO Project:

a) Use of methanol-diesel fuel-higher alcohols blends in diesel engine with PSZ plasma coated parts.

b) Measurements of pressure traces and flame propagation rates after spark ignition of methanol vapour - air mixtures having different strengths in constant volume apparatus. Expert took part in planning of the both research activities.

2. Discussions on possible ways of cooperation between IIT, Bombay and IIP, Dehradun in the lines of the UNIDO Project. First draft of corresponding agreements was prepared by expert.

3. Consideration of assembly and working drawings of single-cylinder research diesel engine and parts of the same engine already fabricated.

As a result of this work expert came to conclusion that the engine designed at IIT, Bombay in many respects is more advanced than the engines available on the world market - such as Ricardo and AVL engines. Therefore, it is advisable to undertake actions for fabricating the engine designed at IIT Bombay. This engine may be used for research on application of alcohol fuels to Indian make direct injection diesel engines. Unfortunately engine supplied to IIP by Ricardo Co. is not suitable for this purpose.

It is quite understandable that the activity suggested may benefit not only current UNIDO project at IIP but also further projects and in general diesel research activities in India, which are of a great importance. Also hard currency may be saved.

4. Delivering two lectures to IIT Staff members and research scholars:

a) Modern trends in diesel engine development.
b) Utilization of methanol in diesel engines.

Texts of both the lectures will be given in experts final report.

5. Consulting staff members and research scholars on methanol utilization in piston engines.

Assembly drawings of single cylinder research diesel engine and also psZ plasma coated Kirloskar AV-1 diesel piston were brought by the expert on his return from visit to Bombay. Taking into account that research activities at IIT Bombay in the lines of UNIDO Project are of a good standard it will be benefitial to sign an agreement about technical and scientific cooperation between two institutes.

<u>At ARAI and Kirloskar Oil Engine Ltd. R&D Centre</u>

1. Visits to different laboratories and general review of research activities going on at ARAI.

2. Discussions on modern trends in automotive diesel engine activities in India with ARAI director and his two deputies.

3. Study of the results of automotive diesel engine performance tests available at ARAI. Comparison of these results with those available at IIP (see for instance, results supplied by IIP Scientists to UNIDO Expert A. Cernei) revealed very substantial differences. Data available at ARAI are closer to reality. As a result of this comparison (and also taking into account further discussions at Ashok Leyland) expert came to following conclusions:

a) Review of accuracy of measurements of diesel engine parameters at IIP Engine Laboratory should be undertaken.

b) Indian make diesel engine standard performances should be made available to IIP Scientists.

c) Conditions of diesel engine tests and accuracy of all measurements should be made strictly corresponding to the Indian standards.

d) Any work on Application of methanol to engines should start only after getting at IIP benches performances of the base diesel engines close to the standard performances.

3. Study of R&D work going on at Kirloskar Oil Engine Ltd. especially R&D aimed at the increase of charge temperature by restricting coolant heat rejection as this may help to ignite methanol in diesel engine.

5. Discussion of possible ways of cooperation between Kirloskar Oil Engine Ltd. and IIP on the lines of UNIDO Project with Surendra Chandorkar, Vice President (Engineering,R&D) and his associates.

As a result of these discussion agreement has been reached about free of cost supply of AV-57 coolant heat restricted diesel engine with composite piston, special designs of cylinder and head of cylinder which undergoes different field tests before marketing. Vice-President of Kirloskar Oil Ltd expressed interest in results of methanol application research as applied to this hot running AV-57 diesel engine. On the request of expert some of drawings of two Kirloskar diesel engines and also the cast-iron piston assemblies were brought to IIP by expert to save time in supply.

At IIT, Madras and Ashok Leyland, Madras

1. Study of research work going on at IIT Madras in the lines of alternative fuel application to piston engines.

a) Use of adiabatic approach to fuel alcohols + diesel fuel) operating diesel engines.

b) Use of hydrogen as diesel fuel substitute.

c) Investigation of hot surface ignition as a means to develop multi-fuel engines.

d) Investigation of spark ignition alcohol engines at different compression ratios.

e) Application of methanol fumigation approach to supercharged engines.

2. Discussions with Prof. K.V.Gopalakrishnan, Prof. P.Srinivasa Rao on the possible ways of day to day cooperation in the lines of UNIDO project between IIP and IIT, Madras.

3. Consulting research scholars at IIT Madras in the lines of alternative fuels application to piston engines.

4. Study of the designs of comosite piston with nimonic insert and cylinder liner with air gap.

5. Dicussions on the possible ways of cooperation between IIP and Ashok Leyland in the lines of UNIDO project with Mr.R.Ramakrishnan (Executive Director, Product Development & Quality) and Mr.A.S.Subramania (Chief Engineer - Research).

As a result of discussions it has been found out that the firm is interested in R&D work on methanol application to diesel engines, especially in the development of pure methanol engines. Suggestion of expert to convert Ashok Leyland diesel engine into pure methanol external mixing spark ignition engine was approved by Ashok Leyland representatives. They expressed willingness to cooperate in corresponding R&D. Also cooperation in field trials of dual fuel Ashok Leyland diesel engines may be possible on buses run in Madras. Details of the cooperation may be worked out during visit to IIP of Mr. A.S.Subramanian. Also Standard performances of Ashok Leyland diesel engines were considered during the meeting.

At UNIDO, Delhi:

1. Discussions on project progress with Chief Technical Field Adviser Dr. M.Kamal Hussein.

2. Fixing the date of departure from duty-station on completing of the assignment, booking and getting tickets for return journey.

3. Organizing the dispatch of expert's unaccompanied baggage.

One extra day has been spent in New Delhi because of non-availability of air ticket.

As a result of business trip:

1. Cooperation between IIP, Dehradun, IIT Bombay, IIT, Madras, Kirloskar Oil Engine Ltd and Ashok Leyland was discussed and agreed upon.

2. Drawings of Kirloskar diesel engines and those of single-cylinder research diesel engine were aquired and brought to IIP, which may be of use for implementation of project aims.

3. Plasma coated and cast-iron machined Kirloskar diesel engine pistons were aquired and brought to IIP for implementing the project aims.

4. Lectures and consultations were held in the lines of UNIDO project.

5. Information about ceramic industry and ways to get and use nimonic metal was obtained for the use in implementing of UNIDO project aims.

Expert is of opinion that the tour would had been even more useful for implementing the UNIDO project aims if it was undertaken earlier. Organizing the tour took more than two months which is too long a period by any standard.

Actions to be taken immediately to make the greatest use of the tour results.

1. To send an engineer to Pune to get and dispatch Kirloskar AV-57 engine and also castings of cast-iron pistons to Dehradun.

2. To draft and sign cooperation agreements with IIT, Bombay, IIT, Madras, Kirloskar Oil Ltd. and Ashok Leyland in the lines of UNIDO Project.

3. To approach Hindustan Aeronautic Ltd. Sunabeda, Koraput, Orissa and Gas Turbine Research Establishment, Bangalore in order to get nimonic rods for fabricating piston inserts.

4. To make decision and to find partners in fabricating single cylinder research engine designed at IIT, Bombay.

A. S. Khatch 19.02.85

A.S.Khatchian
UNIDO Expert.

Report on development of the engineering method based on the theory of similarity for approximate choice of main elements of fuel injection system to feed diesel engine by methanol instead of diesel fuel.

Based on the work done at Indian Institute of Petroleum, Dehradun by UNIDO Expert Prof. A.S.Khatchian from Moscow Automobile Maintenance and Highway Construction Technical University (USSR).

Introduction

Methanol has calorific value per unit volume 2.25 times less than that of diesel fuel. Therefore to keep power output of diesel engine the same with both the fuels it is necessary to increase cyclic fuel delivery in volumetric units by 2.25 times provided efficiency is not changing due to conversion of diesel engine to be fueled by methanol instead of diesel fuel. In most of the publications it is reported that methanol fueled diesel engine efficiency at high loads may be the same or slightly higher than that of base engine if necessary changes in design are made to ensure proper methanol ignition (1), (2). As increase in efficiency is small one should find way to substantial increase in cyclic delivery while using methanol in diesel engine. There is no much of information as regards to optimum fuel injection characteristics in case when diesel engine runs on methanol. At the same time one way not anticipate necessity of increasing injection pressure and decreasing injection duration as compared to values obtained in diesel fuel injection (2). Therefore it is of interest to find out what changes in fuel injection system elements should be made when turning over to methanol injection instead of diesel fuel injection to get necessary increase in cyclic fuel delivery and to keep injection pressure and duration the same as in base engine. These changes may be considered as extreme ones, as in reality one may expect that certain increase in duration of injection

will not be too harmful

when changing over from diesel fuel to methanol at least in cases of pre-wall type air-fuel mixing modes.

Aim of development

To establish the engineering method to choose fuel injection system elements for methanol operation to get increase in cyclic fuel delivery by 2.25 (in volumetric units) while keeping injection pressure and injection duration the same as in diesel fuel injection.

Development of the method

In (3) theory of similarity was used successfully to find out necessary changes in fuel injection system elements while supercharging diesel engine or in case when fuel injection system elements are to be chosen for the new engine under development. In both the cases fuel properties were considered to be the same in base system and the system under development. So findings given in (3) cannot be used directly in methanol application. However general approach used in (3) may be of use in achieving the aim set above. Boundry conditions at pump end of the fuel injection system may be given in the following form:

$$-f_{pl} \cdot C_{pl} = 6n\alpha V_p \frac{dp_p}{d\varphi} + (\mu f)_o \sqrt{\frac{2}{\rho}}\sqrt{p_p - p_{in}} + \mu_{d.v.} f_{d.v}\sqrt{\frac{2}{\rho}}\sqrt{p_p - p_{dv}} + f_{d.v} \cdot C_{dv} \qquad (1)$$

$$\mu_{d.v.} f_{d.v.}\sqrt{\frac{2}{\rho}}\sqrt{p_p - p_{d.v.}} + f_{d.v.p.} C_{d.v.} = 6n\alpha V_{d.v.} \frac{dp_{d.v}}{d\varphi} + f_t C_t \qquad (2)$$

$$6nM\frac{dC_{d.v.}}{d\varphi} = f_{d.v.p}(p_p - p_{d.v.}) - f_{d.v.p.} p_{d.v.o} - \delta h_{d.v} \qquad (3)$$

$$C_{d.v.} = \frac{dh_{d.v.}}{d\varphi} 6n \qquad (4)$$

In equations (1) - (4):

fpl fd.v., fd.v.p ft are areas of: plunger cross section, passage between delivery valve and its seat, delivery valve piston cross section, passage in high pressure pipe, $(\mu f)_o$ is effective area of passages in plunger and plunger barrel; Cpl Cd.v. Ct are linear velocities of : Plunger, delivery valve, fuel at high pressure pipe inlet;

n is pump speed, r.p.m.

α is fuel compressibility;

Vp, Vd.v are volumes of: plunger chamber and delivery valve chamber ;

P_p, p_{in}, $p_{d.v.}$ are pressures at: plunger chamber, fuel feed system, delivery valve chamber;

φ is camshaft rotation angle;

μd.v. is discharge coefficient of fuel flow through passage between delivery valve and its seat;

ρ is fuel density;

M is mass of moving parts of delivery valve assembly;

Pd.v.o is delivery valve opening pressure;

δ is stiffness of delivery valve spring.

If we assume that change in pressures with respect to camangle and pump speed are the same for both the fuels than equation (1) may be written in case of methanol as follows:

$$f_{pl_m} C_{pl_m} = 6 n \alpha_m V_{pm} \frac{dp_p}{d\varphi} + (\mu f)_{om} \sqrt{\frac{2}{\rho_m}} \sqrt{P_p - P_{in}} + \mu_{d.v.m} f_{d.v.m} \times \sqrt{\frac{2}{\rho_m}} \sqrt{P_p - P_{d.v.}} + f_{d.v.p.m} C_{d.v.m.}$$

Let us introduce the following designations:

$$\frac{f_{pl_m}}{f_{pl}} = K_{f_{pl}};\ \frac{C_{pl_m}}{C_{pl}} = K_{C_{pl}};\ \frac{\alpha_m}{\alpha} = K_\alpha;\ \frac{V_{pm}}{V_p} = K_{V_p};\ \frac{(\mu f)_{om}}{(\mu f)_o} = K_{(\mu f)_o};$$

$$\frac{\rho_m}{\rho} = K_\rho;\ \frac{\mu_{d.v.m.} f_{d.v.m}}{\mu_{d.v.} f_{d.v}} = K_{\mu_{d.v.} f_{d.v}};\ \frac{f_{d.v.p.m}}{f_{d.v.p}} = K_{f_{d.v.p}};\ \frac{C_{d.v.m}}{C_{d.v}} = K_{C_{d.v}} \quad (6)$$

Substituting these expressions into (5) one gets

$$K_{f_{pl}} \cdot K_{C_{pl}} f_{pl} \cdot C_{pl} = K_{\alpha} K_{V_p} 6n \alpha V_p \frac{d\rho_p}{d\alpha} + K_{(\mu f)_0} K_{\rho}^{-\frac{1}{2}} (\mu f)_0 \sqrt{\frac{2}{\rho}} \sqrt{\rho_p - \rho_{dv}} +$$
$$K_{\mu_{dv} f_{dv}} K_{\rho}^{-\frac{1}{2}} \mu_{dv} f_{dv} \sqrt{\frac{2}{\rho}} \sqrt{\rho_p - \rho_{dv}} + K_{f_{dvp}} K_{C_{dv}} f_{dvp} \cdot C_{dv}$$

Let us devide last equation by $K_{f_{pl}} \cdot K_{C_{pl}}$

$$f_{pl} \cdot C_{pl} = \frac{K_{\alpha} K_{V_p}}{K_{f_{pl}} K_{C_{pl}}} 6n \alpha V_p \frac{d\rho_p}{d\varphi} + \frac{K_{(\mu f)_0} K_{\rho}^{-\frac{1}{2}}}{K_{f_{pl}} \cdot K_{C_{pl}}} (\mu f)_0 \sqrt{\frac{2}{\rho}} \sqrt{\rho_p - \rho_{dv}} +$$
$$+ \frac{K_{\mu_{dv} f_{dv}} K_{\rho}^{-\frac{1}{2}}}{K_{f_{pl}} \cdot K_{C_{pl}}} \mu_{dv} \cdot f_{dv} \sqrt{\frac{2}{\rho}} \sqrt{\rho_p - \rho_{dv}} + \frac{K_{f_{dvp}} K_{C_{dv}}}{K_{f_{pl}} \cdot K_{C_{pl}}} f_{dvp} \cdot C_{dv} \qquad (7)$$

Solutions of equation (1) and (7) will be the same if

$$\frac{K_{\alpha} K_{V_p}}{K_{f_{pl}} K_{C_{pl}}} = 1, \quad \frac{K_{(\mu f)_0} K_{\rho}^{-\frac{1}{2}}}{K_{f_{pl}} \ K_{C_{pl}}} = 1, \quad \frac{K_{\mu_{dv} f_{dv}} K_{\rho}^{-\frac{1}{2}}}{K_{f_{pl}} \ K_{C_{pl}}} = 1, \quad \frac{K_{f_{dvp}} K_{C_{dv}}}{K_{f_{pl}} \ K_{C_{pl}}} = 1$$

Taking into account designation (6) we can find:

$$\frac{\alpha_m V_{pm} f_{pl} C_{pl}}{\alpha V_p f_{plm} C_{plm}} = 1, \quad \frac{\alpha_m V_{pm}}{f_{plm} C_{plm}} = \frac{\alpha V_p}{f_{pl} C_{pl}} = idem$$
$$\frac{(\mu f)_{0m} \rho_m^{-0.5} f_{pl} C_{pl}}{(\mu f)_0 \rho^{-0.5} f_{plm} C_{plm}} = 1, \quad \frac{(\mu f)_{0m}}{f_{plm} C_{plm} \rho_m^{0.5}} = \frac{(\mu f)_0}{f_{pl} \ C_{pl} \cdot \rho^{0.5}} = idem$$
$$\frac{\mu_{dvm} \cdot f_{dvm} \rho_m^{-0.5} f_{pl} C_{pl}}{\mu_{dv} \cdot f_{dv} \cdot \rho^{-0.5} f_{plm} C_{plm}} = 1, \quad \frac{\mu_{dvm} f_{dvm}}{f_{plm} C_{plm} \rho_m^{0.5}} = \frac{\mu_{dv} f_{dv}}{f_{pl} \cdot C_{pl} \rho^{0.5}} = idem$$
$$\frac{f_{dvpm} C_{dvm} f_{pl} C_{pl}}{f_{dvp} \cdot C_{dv} \cdot f_{plm} \cdot C_{plm}} = 1, \quad \frac{f_{dvpm} \cdot C_{dvm}}{f_{plm} \cdot C_{plm}} = \frac{f_{dvp} \cdot C_{dv}}{f_{pl} \cdot C_{pl}} = idem$$

Equation (2) may be written in case of methanol as follows:

$$\mu_{dvm} f_{dvm} \sqrt{\frac{2}{\rho_m}} \sqrt{\rho_p - \rho_{dv}} + f_{dvpm} C_{dvm} = 6n \alpha_m V_{dvm} \frac{d\rho_{dv}}{d\varphi} + f_{tm} C_{tm} \qquad (8)$$

Let us introduce the following designations:

$$\frac{V_{dvm}}{V_{dv}} = K_{V_{dv}}, \quad \frac{f_{tm}}{f_t} = K_{f_t}, \quad \frac{C_{tm}}{C_t} = K_{C_t} \qquad (9)$$

Taking into account (6) and (9) it is possible to represent equation (8) as follows: $K_{\mu d.v.} f_{d.v} \cdot K_{\rho}^{-0.5} \sqrt{\frac{2}{\rho}} \sqrt{P_p - P_{d.v.}} + K_{f d.v.p} K_{C_{d.v.}}^{*} \times f_{d.v.p} \cdot C_{d.v} = K_{\alpha} K_{V_{d.v.}} \cdot 6 n \alpha V_{d.v} \frac{dp_{d.v}}{d\varphi} + K_{f_t} K_{C_t} f_t C_t$

Let us devide last equation by $K_{f_t} \cdot K_{C_t}$

$$\frac{K_{\mu d.v} f_{d.v} \cdot K_{\rho}^{-0.5}}{K_{f_t} \cdot K_{C_t}} \sqrt{\frac{2}{\rho}} \sqrt{P_p - P_{d.v.}} + \frac{K_{f d.v.p} K_{C d.v}}{K_{f_t} \cdot K_{C_t}} \cdot f_{d.v.p.} \cdot C_{d.v.} =$$

$$\frac{K_{\alpha} K_{V_{d.v}}}{K_{f_t} \cdot K_{C_t}} 6 n \alpha V_{d.v} \frac{dp_{d.v.}}{d\varphi} + f_t \cdot C_t \qquad (10)$$

Solutions of equations (8) and (10) will be the same if

$$\frac{K_{\mu d.v} f_{d.v} \cdot K_{\rho}^{-0.5}}{K_{f_t} \cdot K_{C_t}} = 1 \, ; \quad \frac{K_{f d.v.p} \cdot K_{C d.v.}}{K_{f_t} \cdot K_{C_t}} = 1 \, ; \quad \frac{K_{\alpha} \cdot K_{V_{d.v}}}{K_{f_t} \cdot K_{C_t}} = 1$$

Taking into account designations (6) and (9) we can find:

$$\frac{\mu_{d.v.m} f_{d.v.m} \rho^{0.5} f_t C_t}{\mu_{d.v.} f_{d.v} \rho_m^{0.5} f_{tm} C_{tm}} = 1 \, ; \quad \frac{\mu_{d.v.m.} f_{d.v.m}}{f_{tm} \cdot C_{tm} \rho_m^{0.5}} = \frac{\mu_{d.v} f_{d.v}}{f_t \cdot C_t \cdot \rho^{0.5}} = idem$$

$$\frac{f_{d.v.p.m} \cdot C_{d.v.m.} f_t \cdot C_t}{f_{d.v.p.} \cdot C_{d.v} \cdot f_{tm} \cdot C_{tm}} = 1 \, ; \quad \frac{f_{d.v.p.m} \cdot C_{d.v.m}}{f_{tm} \cdot C_{tm}} = \frac{f_{d.v.p.} \cdot C_{d.v}}{f_t \cdot C_t} = idem$$

$$\frac{\alpha_m \cdot V_{d.v.m} \cdot f_t \cdot C_t}{\alpha \cdot V_{d.v} \cdot f_{tm} \cdot C_{tm}} = 1 \, ; \quad \frac{\alpha_m V_{d.v.m}}{f_t \cdot C_{tm}} = \frac{\alpha V_{d.v}}{f_t \cdot C_t} = idem$$

Equation (3) may be written in case of methanol as follows:

$$6 n M_m \frac{dC_{d.v.m}}{d\varphi} = f_{d.v.p.m} (P_p - P_{d.v.}) - f_{d.v.p.m} P_{d.v.o} - \delta_m \cdot h_{d.v.m} \qquad (11)$$

Let us introduce the following designations:

$$K_M = \frac{M_m}{M} \; ; \quad K_{\delta} = \frac{\delta_m}{\delta} \; ; \quad K_{h_{d.v}} = \frac{h_{d.v.m}}{h_{d.v}} \qquad (12)$$

Taking into account (6) and (12) one may rewrite equation (11) as follows:

$$-K_M \cdot K_{C_{d.v}} \cdot 6n\, M \frac{dC_{d.v}}{d\varphi} = K_{d.v.p}\, f_{d.v.p}(P_p - P_{d.v}) - K_{d.v.p}\, f_{d.v.p}\, P_{d.v.o} -$$
$$- K_\delta\, K_{h_{d.v}} \cdot \delta \cdot h_{d.v}$$

Let us devide the last equation by $K_{f_{d.v.p}}$

$$-\frac{K_M \cdot K_{C_{d.v}}}{K_{d.v.p}} 6n\, M \frac{dC_{d.v}}{d\varphi} = f_{d.v.p}(P_p - P_{d.v}) - f_{d.v.p}\, P_{d.v.o} -$$
$$\frac{K_\delta \cdot K_{h_{d.v}}}{K_{f_{d.v.p}}} \delta \cdot h_{d.v} \qquad (13)$$

Solutions of equations (3) and (13) will be the same if:

$$\frac{K_M \cdot K_{C_{d.v}}}{K_{f_{d.v.p}}} = 1; \quad \frac{M_m\, C_{d.v.m}\, f_{d.v.p}}{M\, C_{d.v}\, f_{d.v.p.m}} = 1, \quad \frac{M_m\, C_{d.v.m}}{f_{d.v.p.m}} = \frac{M \cdot C_{d.v}}{f_{d.v.p}} = idem$$
$$\frac{K_\delta \cdot K_{h.d.v}}{K_{f_{d.v.p}}} = 1; \quad \frac{\delta_m \cdot h_{d.v.m} \cdot f_{d.v.p}}{\delta \cdot h_{d.v}\, f_{d.v.p.m}} = 1, \quad \frac{\delta_m \cdot h_{d.v.m}}{f_{d.v.p.m}} = \frac{\delta\, h_{d.v}}{f_{d.v.p}} = idem$$

Equation (4) for methanol may be written:

$$C_{d.v.m} = \frac{d h_{d.v.m}}{d\varphi} 6n$$

Using (6) and (12) the last equation may be rewritten as follows:

$$K_{C_{d.v}} \cdot C_{d.v.} = K_{h_{d.v}} \frac{dh_{d.v}}{d\varphi} 6n$$

Let us devide last equation by $K_{h_{d.v}}$

$$\frac{K_{C_{d.v}}}{K_{h_{d.v}}} C_{d.v.} = \frac{d h_{d.v}}{d\varphi} 6n \qquad (14)$$

Solutions of equations (4) and (14) will be the same if

$$\frac{K_{C_{d.v}}}{K_{h_{d.v.}}} = 1; \quad \frac{C_{d.v.m} \cdot h_{d.v}}{C_{d.v.} \cdot h_{d.v.m}} = 1$$

Now it is possible to write down all the criterions obtained from boundary conditions at pump and of the system:

$$\frac{\alpha V_p}{f_{pl} \cdot C_{pl}} = idem \, ; \frac{(Mf)_o}{f_{pl} \cdot C_{pl} \cdot \rho^{0,5}} = idem \, ; \frac{M_{d.v.} \cdot f_{d.v.}}{f_{pl} \cdot C_{pl} \cdot \rho^{0.5}} = idem$$

$$\frac{f_{d.v.p.} \cdot C_{d.v.}}{f_{pl} \cdot C_{pl}} = idem \, ; \frac{M_{d.v.} \cdot f_{d.v.}}{f_t \cdot C_t \cdot \rho^{0.5}} = idem \, ; \frac{f_{d.v.p} \cdot C_{d.v.}}{f_t \cdot C_t} = idem$$

$$\frac{\alpha V_{d.v}}{f_t \, C_t} = idem \, , \frac{M \cdot C_{d.v.}}{f_{d.v.p}} = idem \, ; \frac{\delta \cdot h_{d.v.}}{f_{d.v.p}} = idem \, ; \frac{C_{d.v.}}{h_{d.v}} = idem$$

Comparing and combining some of the criterions one may finally arrive to the following set of criterions, which are convenient to use for the choice of fuel injection system parameters

$$\frac{\alpha V_p}{f_{pl} \cdot C_{pl}} = idem \, , \frac{\alpha V_{d.v.}}{f_{pl} \cdot C_{pl}} = idem \, ; \frac{f_t C_t}{f_{pl} \cdot C_{pl}} = idem \, ; \frac{f_{d.v.p} \cdot C_{d.v}}{f_{pl} \cdot C_{pl}} = idem \, ;$$

$$\frac{(Mf)_o}{f_{pl} \cdot C_{pl} \cdot \rho^{0.5}} = idem \, , \frac{M_{d.v} \cdot f_{d.v}}{f_{pl} \cdot C_{pl} \cdot \rho^{0.5}} = idem \, , \frac{\delta \cdot h_{d.v}}{f_{d.v.p}} = idem \, ,$$

$$\frac{M \, C_{d.v.}}{f_{d.v.p}} = idem \, , \frac{C_{d.v}}{h_{d.v}} = idem$$

The last criterion is satisfied only in case if motion of delivery valve with respect to time (cam angle) is the same for both the fuels.

Linear velocity of fuel flow into high pressure fuel pipe may be given as (4):

$$C_t = \frac{1}{a\rho} \left[P_{d.v} - P_o + 2W(\tau) \right]$$

Here: po is residual pressure in pipe, $W(\tau)$ is reflected pressure wave magnitude at the moment, a is sonic velocity. As all pressure events are taken to be the same for both the fuel injection systems change in Ct is determined by $\frac{1}{a\rho}$. In (5) the following data is given for both the fuels (Table-1)

Table-1

Parameter	Fuel	
	Diesel,D2	Methanol
a, m/s	1380	1180
E, $\frac{N}{m^2}$	$1,68.10^9$	$0,75.10^9$
α, $\frac{m^2}{N}$	$0,595.10^{-9}$	$1,333.10^{-9}$

In table 1:E - modul of elasticity, α-coefficient of compressibility. Data on methanol need correction. Assuming sonic velocity to be properly measured one gets $E = a^2\rho = 1180^2 \cdot 790 = 1,1\ 10^9 \frac{N}{m^2}$ and $\alpha = 1/E = 0,91.10^{-9}\ m^2/N$. This value is closer to the relationship $\alpha = f(p)$ given in (4), than value given in table-1. In case of truck engine diesel fuel it is more correct to assume $p = 840\ kg/m^3$, $a = 1250\ m/s$ $\alpha = 0,762.10^{-9}\ \frac{m^2}{N}$.

Hence Ct in case of methanol is 1,126 times greater than in case of diesel fuel. To increase rate of fuel supply into high pressure pipe by 2.25 times ft should be increased by $\frac{2,25}{1,126} = 2$ times.

Naturally rate of fuel supply by plunger should also be increased by 2.25 times. From the criterions it follows that volumes in pump and delivery valve chambers are to be increased by $2.25 \frac{\alpha m}{\alpha} = 2.25 \cdot \frac{0.762.10^{-9}}{0.910.10^{-9}} = 1,88$ times.

With delivery valve speed being the same for both the fuels the delivery valve piston cross section should be also increased by 2.25 times.

Effective area of plunger barrel holes should be increased by $2,25\left(\frac{\rho m}{\rho}\right)^{0,5} = 2,18$ times. The same is true for effective area of passage between delivery valve and its seat. Extents in which fd.v.p. and μd.v.fd.v should change are close to each other. So there will be no much difficulties in satisfying both the conditions. Mass of delivery valve moving parts and stiffness of delivery valve spring should be increased by 2.25 times to ensure the same events of delivery valve motion at conditions outlined.

Boundary conditions at injector end of the fuel injection system may be given in the following form:

$$f_t C_t' = 6n\alpha V_{inj}\frac{dp_{inj}}{d\varphi} + (\mu f)_{inj}\sqrt{\frac{2}{\rho}}\sqrt{p_{inj} - p_{cyl}} + f_n C_n \qquad (15)$$

$$6nM'\frac{dC_n}{d\varphi} = (f_n - f_n')(p_{inj} - p_{injo}) + f_n' p_{inj}' + \delta' y \qquad (16)$$

$$6n\frac{dy}{d\varphi} = C_n \qquad (17)$$

$$p_{inj}' = \frac{(\mu f)_{inj}^2}{(\mu_{n.o} f_{n.o})^2}(p_{inj} - p_{cyl}) + p_{cyl} \qquad (18)$$

In equations (15) - (18):

Ct ,Cn are liner velocities of fuel flow from high pressure pipe into injector and needle;

Vinj is injector volume;

Pinj, Pcyl, Pinjo, Pinj$'$ are pressures: in injector, cylinder, at the start of nozzle opening, in sac volume;

μn.o is discharge coefficients of fuel flow through nozzle orifices;

fn, fn$'$ are areas of: cross section of needle guiding portion, needle at which fuel pressure is acting when needle is at the seat;

$(\mu f)_{inj}$ is effective area of nozzle;

Cn is needle velocity;

M' is mass of injector moving parts;

is stiffness of injector spring;

y is needle lift.

In the same manner as above it is possible to arrive to the following criterions $\frac{dV_{inj}}{f_t \cdot C_t'}$ = idem.

$$\frac{(\mu f)_{inj}}{f_t \cdot C_t \cdot \rho^{0.5}} = idem \ , \ \frac{(\mu f)_{inj}}{\mu_{n.o} \cdot f_{n.o}} = idem \ , \ \frac{f_n C_n}{f_t \cdot C_t'} = idem$$

$$\frac{M' C_n}{f_n} = idem \ , \ \frac{\delta' y}{f_n} = idem \ ; \ \frac{C_n}{y} = idem .$$

Linear velocity of fuel flow from high pressure pipe into injector may be given as (4):

$$Ct = \frac{1}{ap}\left[Po - Pin + 2F\left(\tau - \frac{L}{a}\right)\right],$$ where F is direct pressure wave and L is length of high pressure pipe.

From this equation it is clear that to get injection process events the same with both the fuels $\frac{L}{a}$ should be the same. Hence the length of pressure pipe should be decreased with methanol by 1,059 times.

With ft.Ct greater by 2.25 times in case of methanol effective areas of nozzle and nozzle orifices should be increased by 2.18 times. This follows from two first criterions obtained from boundary conditions at injector end of the system. To keep motion of the needle the same with both the fuels fn, fn' M', δ' should be increased by 2.25 times, whereas volume in injector is to be increased by 1.88 times. General conclusions following from above analysis: To keep pressure and parts movements events the same in case of methanol injection as compared to diesel fuel injection:

1. fpl·cpl, fd.v.p, M, δ, fn, fn', M', δ' should be increased by 2.25 times.
2. (μf)o, μd.v.·fdv, (μf)inj, μn.o·fn.c should be increased by 2.18 times.
3. Vp, Vd.v., Vinj should be increased by 1.88 times.
4. ft should be increased by 2 times.
5. L should be decreased by 1.06 times.

It is necessary however to stress that not all the changes outlined above are advisible to be implemented. Most important changes are: increase in fpl · Cpl, and μn.o.fn.c (and hence (μf)inj). Instead of increase in fd.v.p., increase in delivery valve retraction volume may be adopted. After choice of these parameters for methanol application it is advisable to perform fuel injection process calculations for different operating conditions by computer programme outlined in (4) and then to correct choice of parameters on the basis of computer simulation. Final check of viability of the choice of fuel injection system parameters should be done on the basis of experiments, durability tests inclusive. These last tests are of great importance as in case of methanol injection forces acting on pump parts may increase and wear resistance of the parts may decrease considerably. The latter may occur not only due to increase in forces acting but also as a result of the change in fuel parameters. To achieve necessary durability and reliability in case of methanol injection use of special additives in fuel itself and in pump casing lubricating oil seems to be unavoidable. Engineering method outlined above may help to decrease time and expenditure on computer modelling and on experiments by narrowing the limits of variation of main fuel injection system elements.

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Appendix 8.1

WORK PLAN FOR A JOINT STUDY BY IIP AND ASHOK LEYLAND ON CONVERSION OF ALU-370 ENGINE TO SPARK IGNITED METHANOL ENGINE.

1. Base Engine Installation — Action IIP (April,30,1985).

2. Base Engine performance at constant fuel delivery and constant excess air power — Action IIP
   sfc
   Smoke and gaseous emissions
   A/F ratio
   Exh Temp.
   Willan's Line - Friction losses

   Cylinder Pressure Data - may not be necessary

Discussion with AL — May, 15, 1985.

3. Design, Fabrication and Testing of Evaporator

   IIP designed methanol evaporator to be checked for application to Leyland Engine.

   Action - IIP

   If necessary new one to be designed. — Action - IIP

   Fabrication — Action - IIP

   Testing — Action - IIP

IIP data on heat transfer requirements of evaporator to be supplied to AL for design and fabrication of a compact Unit. — May,31, 1985.

4. Ignition System, including spark plugs.

   Procurement - Action IIP

AL to Liaison with MICO Bosch for selection of equipment and getting quotations sent to IIP

5. Central Body Fuel Injection System or/and Carburettor.

   As under item 4. — June, 30, 1985.

6. Pressure regulator, Fuel pumps etc. procurement (Indigenously or import)- Action IIP. June, 30, 1985.

7. Modification of Cylinder Head for installation of spark plugs and pressure pick-up. Action - AL (Pressure Pick-up installation details to be provided by IIP).

8. Modification of piston Cavity: **Action - AL**

CR to be 12:1 AL to check for making a compact cylinder-cal bowl centrally located vis a vis spark plug.

May, 31, 1985.

9. Assembly of prototype engine

At IIP

Action - IIP with AL's cooperation. July, 31, 1985.

10. Performance evaluation of Engine at IIP. Action - jointly by IIP and AL Engineers.

Power
Sfc
Emissions
Exhaust Temp.
P-t history etc.

Sept. 30, 1985.

11. Installation of the vehicle and Field Trials at Madras & Dehradun.

Action AL with Cooperation from IIP.

December, 1985.

Optimized engine would also be subjected to lubrication and wear performance and durability trials.

Appendix 8.3

AGREEMENT

On mutual cooperation between IIT, Bombay, IIT, Madras and IIP Dehradun for developing alcohol based piston IC Engines.

AIM To find various methods of utilising alcohol in diesel engines which may help in decreasing oil imports.

Work to be carried out at IIP Dehradun

1. Conversion of truck diesel engine into spark ignition external mixing alcohol engine.
2. Development of glow plug version of pure methanol engine.
3. Investigation of small pre-chamber pure methanol engine as suggested by V.N. Svobodov.
4. Investigation of diesel engine with ceramic insert in piston aimed to promote methanol ignition.

Conditions of Cooperation

1. All the Institutions to carry out research funded by their own resources.
2. All the Institutions to exchange technical reports on the above investigations. Each side communicates to others an analysis of the technical and scientific findings contained in the reports received from the others.

From I.I.P. Dehradun

Mr. S.Singhal

S. Singhal
Feb 4, 1985

C-820

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AD.86.07

ILL 5.5+I