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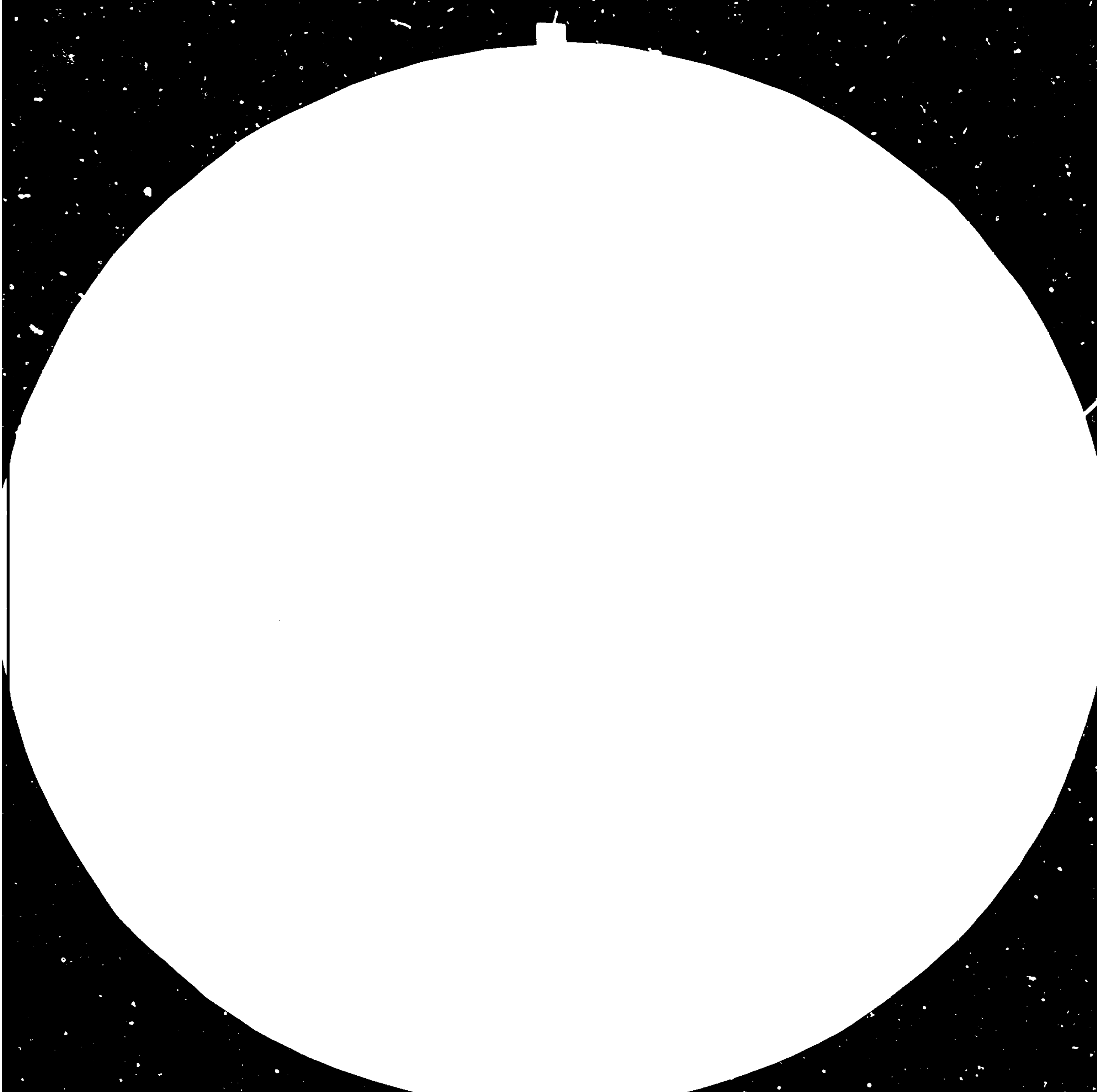

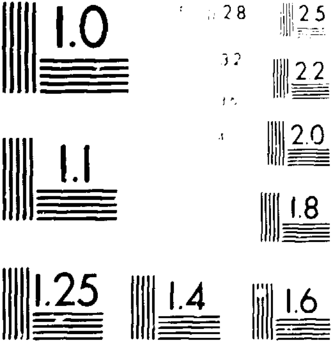

MICROCOPY RESOLUTION TEST CHART

RESTRICTED

DP/ID/SER.A/489
17 January 1984
ENGLISH

13255

APPLICATION OF ALTERNATIVE FUELS FOR
INTERNATIONAL COMBUSTION ENGINES, IIP, DEHRA DUN

DP/IND/82/001

INDIA

Technical Report: Methanol 4-stroke high speed diesel engines *

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 Anton Cernej
UNIDO Expert

United Nations Industrial Development Organization
Vienna

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PART I CONCERNING THE INJECTION AND COMBUSTION OF METHANOL

I. INTRODUCTION

With the rapidly rising costs of crude oil uncertainity of supply,rational economic crises and the doubts surrounding the quantity of crude recoverable at a resonable cost significant changes relevant to world fuel supply will take place in future years.In some areas the fuels will change from the more conventional hydrocarbon fuels to alternatives such as alcohols and this change will in them create in significant changes as regards detailed fuel specifications(Cetane number,volatility,aromatics, calorific value,etc.).

High speed truck and light duty diesel engines are not fuel tolerant,and this is particularly the case concerning alcohols. To burn the methanol in a DI diesel engine a dual fuel or a glow plug approach is required to overcome the poor ignition quality,assisted ignition has to initiate combustion.The ignition quality,fuel injection,combustion,engine outputs and service lives are mainly influenced by those below cited differences between the properties diesel oil and methanol fuels:

1. Methanol is monocomponent HCO fuel and quickly evaporates at a single defined temperature.
   Diesel oil is a multicomponent HC fuel and evaporates within defined temperature range.
2. The calorific value of methanol is 2,5x lower than that of diesel oil D2.
3. Methanol evaporates at a higher temperature than diesel fuel.
4. The evaporation temperature of methanol is much lower in respect of 95% dest. temperature of diesel oil.
5. Methanol needs a higher temperature for ignition
6. The cetane number ratio Diesel oil D2/methanol is nearly 10.
7. The density of methanol is lower than that of D2.
8. The viscosity " " " "
9. The compressibility " higher " "
10. The pressure wave propagation velocity " lower " "
11. The lubrication quality of methanol is poor and the lub quality of diesel oil is sufficient concerning fuel injection system.

12. Corrosion attack is more likely when with methanol than with diesel fuel.

As a result of the above cited differences the following must be considered:

a) The energy required for an ignitable air-fuel mixture when operating with methanol is higher by factor 8 when compared with the same "point" of diesel fuel-air mixture.

b) For the use of methanol fuel injection equipment changes or at least new settings will be required to:
   - the increased fuelling
   - and to compensate for the higher fuel compressibility.

c) Reduced ignition quality of the fuel will normally result in worse THC (or/and THCO) emissions in practical operationes even though steady state hot tests have shown little change. This is because of cold start, misfire and cold operationes. Cold operation may have an increasingly severe effect on delay period, misfire and late combustion.

Misfire and spontaneous ignition or knock have to be examined under:
   - conditions of cold operation
   - low partial loads
   - transients

"Clinical" hot steady state bench verifications in laboratory conditions are insufficient. Under automotive legislative test cycles THC emissions are frequently increased by 25-45%.

d) The "equivalent" diesel fuel consumption in service operation may be similar to that of diesel fuel alone if misfiring and late combustion are avoided.

e) Poor injection, cold operation, misfiring and long time low rated speed operation with methanol may initiate excessive wear.

f) The lower calorific value, higher compressibility and lower Pressure wave propagation velocity of methanol when comparing with that of diesel oil cause:
   - prolongated injection
   - different residual pressure
   - altered injection timing
   - small differences in mean integrated injection pressure
   - reduced max. injection pressure

-spray penetration and fuel atomisation may be greatly influenced

-more leakage overflows because of lower density.

II. The fuel injection system under consideration:

The IIF lab., to the methanol fuel 1% of the lub. oil was doped. If separation and excess of wear are really avoided this approach may be acceptable.

Without drawings on disposal adhock measurements depict the system used:

1. plunger dia ø 7 mm
2. max. stroke of plunger ~9mm
3. stroke for theoretical discharging 6 mm with tangential cam
4. Common inlet and spill part of 3 mm dia.
5. dead volume of TDC plunger position

$$V_{KT} = 0.5 \cdot \frac{\pi \cdot 6^2}{4} + 180 \ [mm^3]$$

6. retraction volume 30 mm^3 (Atlas -ø6mm)
7. dead volume of the relief valve 800 mm^3 cap.
8. high pressure tube ø2x600 mm with volume of 1885 mm^3
9. injector dead volume ~900 mm^3
10. total dead volume ~3500 mm^3
11. ratio (ret.vol:tot.vol) 100=0,86
12. nozzle holes 3x0,26 mm.
13. hydraulic needle ratio ø3/ø6
14. presetting of the follower ~ 2,8-3 mm.
15. fuelling 75 mm^3/stroke at 750 min^{-1} (camshaft) measured in engine operation.

III. General approach:

a. The start of injection is mainly influenced by:

1-opening injector pressure
2-residual p_o.
3-operational speed
4-injection system used
5-cam-follower presetting
6-compressibility of the fuel used
7-pressure wave propagation velocity

With conversion to methanol usually the residual pressure increases but because of the points 6 and 7 the start of injection is often delayed.Timing compensation is easy attainable. Some problems here may be expected with the widened speed range when an advancer has to be used.If the rated speed is less than (or equel to) 2000 rpm the timing device can be normally avoided.It should be mentioned that using advancer injection equipment is more expensive and some uncertainties concerning timing still remain.Moreover,with high in-cylinder peak pressures at higher speeds it is advisable to avoid an advancer and to accept higher fuel consumption.

Retarded timing may be of benefit for better ignition and to increase pressure if,because of prolongated injection late combustion does not occur.If so,the best procedure is to shorten the injection period while simultaneously maintaining the same--cylinder peak pressure and the same pressure rise (or less), as those for diesel fuels.

Injection period

Injection period depends on:
a. Fuelling or injection quantity
b. System used (plunger dia,nozzle holes,etc.)
c. Operational speed.

Converting to methanol as fuel injection period is normally very high influenced because of lower calorific value.Operating with methanol for the same power output the engine has to be 2,5x more fuelled or with fuel 2,5x more volumetrically metered than that which is case for diesel oil;this means that the period of injection is prolongated However,the long injection period causes late combustion and higher thermal loading on the combustion chamber walls.This may not be the problem with tuned down engines.Moreover,if the max. mean effective pressure is relatively low an injection period correction and a rematching may be unnecessary.

However,when operating with diesel fuel at reasonable levels of mean pressure and converting to methanol the fuel injection system has to be rematched.With fuelling increased care must be taken for the cam and the follower.

The loaded part of the cam increases and so the possibility for very high wearing at the contact surfaces as Fig.1 shows:

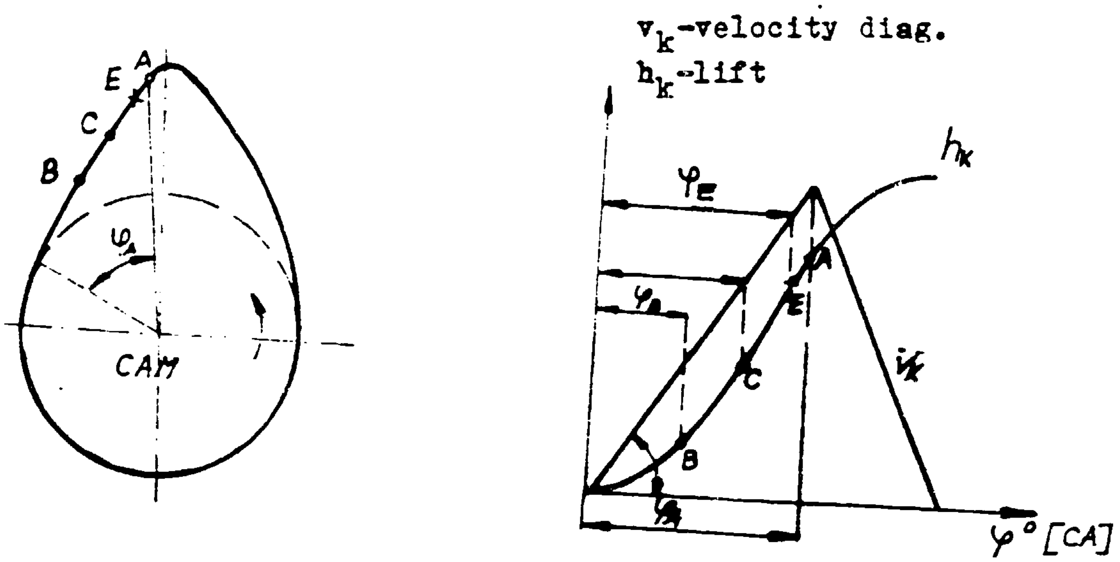
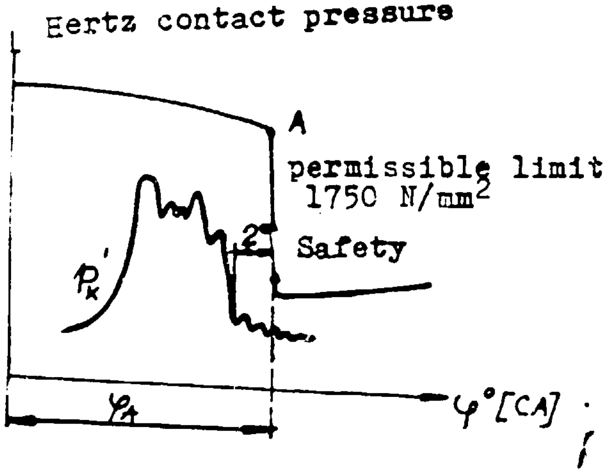


Fig.1 Relations cam-follower and fuelling



p_k-Fuel pressure diagram above the plunger.

In Fig.1 point B depicts the start of geometrically defined delivery with diesel oil and point C shows the end. With methanol the end of discharging must be removed to point E. Between the point E and A safe distance must be kept, according to Fig.1.
Removing the point B on the left the average discharge velocity decreases and the duration of injection additionaly increases.

If B-E shows too long period of injection or/and the safe limit A is reached is only to do:

1-increase the plunger dia.If it is still possible to use in the injection pump.

2-increase the full plunger lift if the new defined camshaft can used.Thus C-E distance is lenghtened

3-change the cam form with the larger angle φ (see Fig.1)

4-reduce the retraction volume if possible

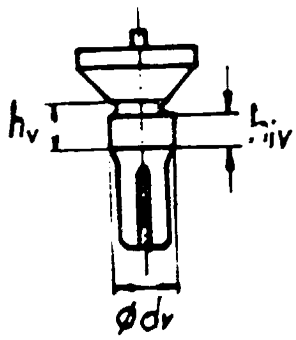
5-increase the total "flow cross-sectional area of the injector. This is particularly important for the nozzle holes,when possible,because of the spray development.

It seeme that the most suitable way of reducing the injection period is to use possibilities of points 1 and 2.
Retraction volume can be practically reduced only if:

-after injection is avoided

-the injector needle closing velocity is still acceptable.This means that the average pressure reduction rate befor the nozzle during the spilling,should be kept at about 150 bar/1°CA.

-existed value has not the ratio h_v/dv too small (Fig.2)



The volume retracted =

$$= V_R = \frac{\pi \cdot d^2}{4} \cdot h_v = \text{constant}$$

Fig.2 The Piston of the relief valve typ ATLAS

If After injection takes place a snubber valve may be used but it should he matched to the specific system (Fig.3).Dia ϕd has to be defined according to the law of reflection.In this way backward pressure wave intensity can be reduced.The dead volume of the snubber valve has to be kept as small as possible especially when methanol is to be used.

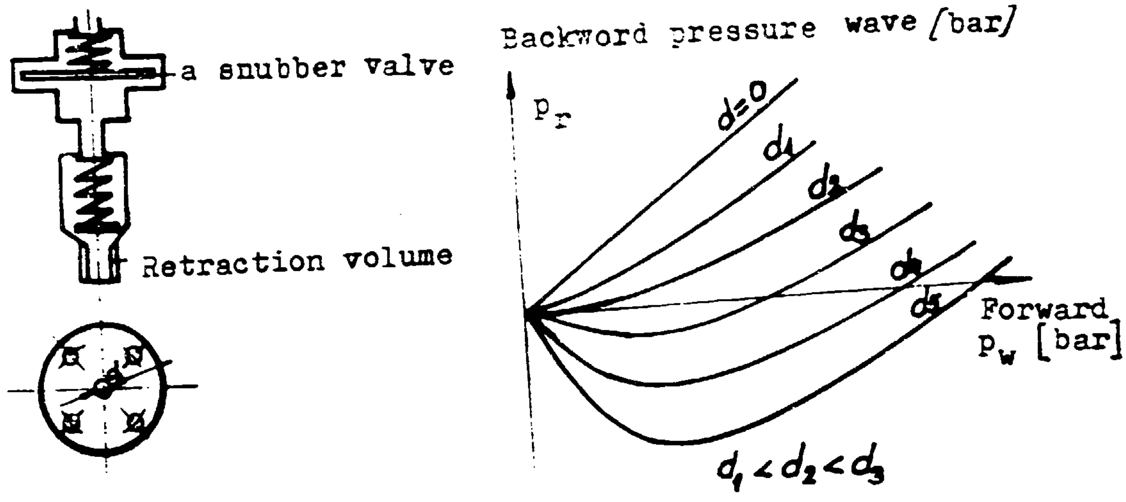


Fig.3 Snubber valve
Producers:Bosch,F&M,CAV,VTS-Maribor)

Lately (See "Steyr" publications).
Snubber valve was used as to increase the injection pressure. However,it is of benefit only in a relatively narrow region of speeds.Moreover,with methanol as fuel,the so called "reflection charging", may be poorer than that of diesel oil.

If should to be noticed that since methanol is a more compressible fluid attention must be paid to reduce the dead volumes. The possibilities are as follows:
-in the cap of relief valve
-in the high pressure tube
-in the injector holder
There is a basic rule:"Reduce dead volumes in the high pressure system, as much as possible, but do not increase the flow resistances and do not convert the potential energy into the kinetic mode, before the nozzle holes".The latter has to be given seriousl consideration when working on injection design.
The so called correction with the module of compressibility

$$\alpha \cdot V \cdot \frac{dp}{dt} \longrightarrow \text{"compressible flow"}$$

can only be reduced by V (dead volume).The coefficient of compressibility of methanol:

$\alpha_{METHANOL} > \alpha_{DIESEL\ OIL}$ (1,33 10^{-9} m^2/N 0,59 10^{-9} m^2/N)

is the main reason why operating with methanol the above modul is getting important. Moreover, because of the lower calorific value (2,5x less then that of diesel oil) fuelling must be drastically increased. To avoid late combustion, efforts must be made to reduce the injection period and this may be achieved with increased pressure rise or with dp/dt in the above modul. Thus, with methanol both factors V and dp/dt, in addition to the coefficient of compressibility, cause the unwanted prolongation of injection.

Again when, operating with methanol the theoretically defined modul of discharging

$$\int_{\varphi_b}^{\varphi_e} A_k \cdot v_k \cdot d\varphi \qquad \text{(see also Fig.1.)}$$

A_k-plunger cross-sectional area

v_k-instantaneous plunger velocity which depends on lift presse-ting, cam form and speed is greater than those for diesel oil

there is no use in increasing only A_k or/and v_k, as the increase of the discharging rate does not always means a growth of injection rate. Moreover, it may be that only the duration of injection increases and then in turn the thermal loading of the combustion chamber and the specific fuel consumption. In such a case the peak in-cylinder pressure is relatively low and the pressure diagram is extended. This may present a specific problem if the engine is getting higher up rated.

In order to reduce the injection period the flow capacity of the injector has to be investigated, both theoretically and experimentally. At a start the following rule seems to be valid:

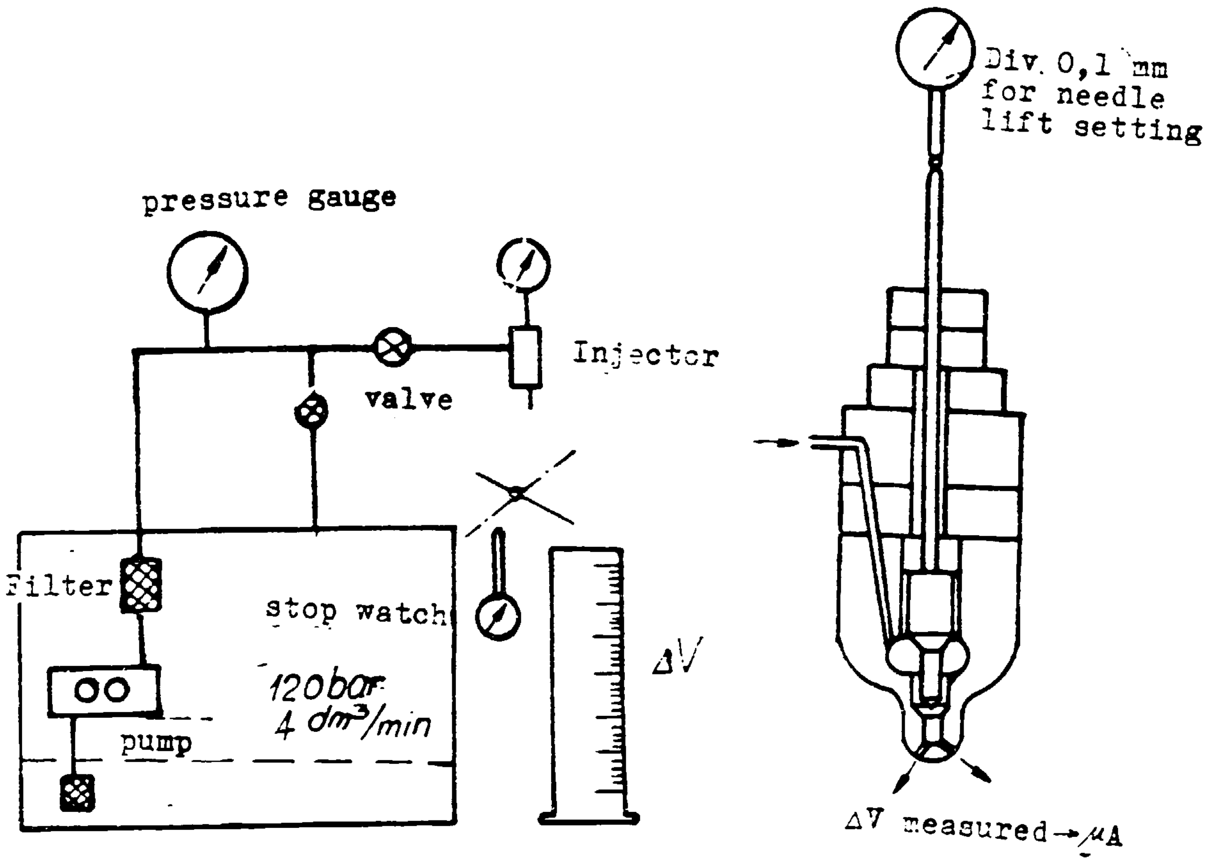
$$\frac{V_{c_2}}{V_{c_1}} = x \longrightarrow \frac{\mu A_2}{\mu A_1} = \frac{x}{3} \div \frac{x}{5}$$

V_{c1} and V_{c2}: Start fuelling and increased resp.

μA_1 and μA_2: The same for the nozzle holes cross-sec. flow areas.

The given rule considers the same fuel and reasonably high load. On the contrary, if the fuel is to be changed, the changes in the mixture formation and in the combustion rate must be appreciated.

Besides that, with low ratings there is no reason for the nozzle flow area to increase at all.
Injector adjustments can be satisfactorlly accomplished only, with the accompaying flow rate measurements.The test bench measurements show Fig.4.



p=const. Δt=const.=30 sec, Δp=const=100 bar
Fig.4 Flow measurements of the nozzle holes

Dealing with diesel engines steady state test bench flow measurements may not be ommitted.For the nozzle alone,injector holder,tubes,intake and spilling ports etc. the flow capacities are very important.For the nozzle the necessary diagram Fig.5 is given.

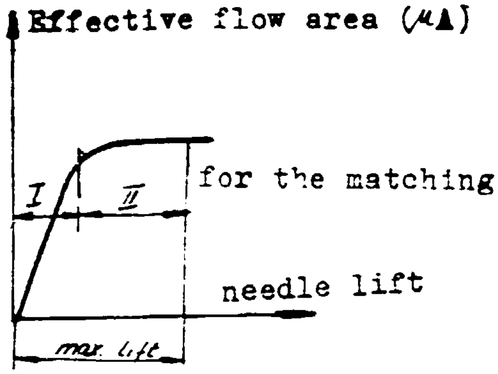


Fig.5. Eff. flow area of the nozzle vs. needle lift.

In region I the control flow area is on the needle seat and in region II, the nozzle holes control the flow. Here it should be mentioned that the geometry increased flow area does not mean an increase of eff. flow area. This is especially so, if the small nozzle holes are observed. The inlet hole resistance and the hole roughness may have a large influence upon the flow capacity. Improvements can be made by lapping as was explained in the IIP Lab.

With increased hole flow areas the cross-sectional area around the needle tip has to be checked (Fig.6), at the full lift.

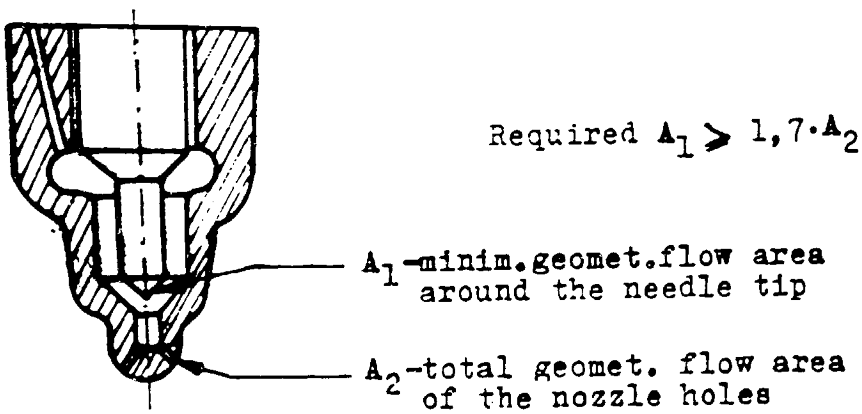


Fig.6 Relation of the injection flow areas.

If the required ratio (Fig.6) is not achieved the fuel flow will be smothered at A_1 and the some of potential energy will be lost. Small correction here may produce good results Fig.7.

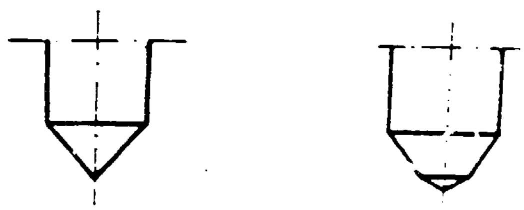


Fig.7. The small correction of the needle tip.

IV. Proposals for the system considered (Matching techniques)

With regard to the measured data collected, the following findings may be of interest.

1.Operating with methanol the peak in-cylinder pressure at rated power is lower by 20-25% compared with that when diesel oil was burnt.

2.Three nozzle holes and a centrally located combustion chamber, by means of an intake swirl port, form the air-fuel mixture ignited by glow plug.

3.The output is relatively low.

4.No overheating was observed

5.The in-cylinder pressure diagram is rather stretched to be similar to that for late combustion

6.Excess of leakage fuel injector overflow was observed in methanol operation.

7.The measurements of glow plug surface temperatures are in progress.

8.The test bench droplet distribution measurement demonstrate:
 -with methanol Sauter mean dia. is less than with diesel oil, spray developed under ambient pressure but was ventilated using a simulated air swirl.

9.Although the pressure diagram streched and the peak pressure low and the energy delivered into the glow plug was incalculated;the specific diesel equivalent consumption fuel rate operating with methanol was 2% better than that, for diesel fuel.This result,if correctly measured and evaluted, is encouraging.However,the in service operation determined fuel consumption is of more importance

10.No glow plug burning out was observed.

11. The engine compression ratio 16 seems to be too low for partial load range and for acceleration from zero to full load because of misfiring. Because of the manual start, it seems impossible to increase it from 16 to 18. However, an electrical starting device can be introduced.
12. As yet detailed attention, to the cold-starting and warming-up characteristics has not been given; misfire or knocking were not observed (at full load).
13. No experiments were performed when using small quantities of the vegetable oil mixed with methanol.
14. After runing the engine for about 200 hrs. the piston chamber, cylinder head and cylinder liner showed no cracking or other demage.

Taking into account the above findings and the data collected in Section II the following proposals may be made:

1. Concerning the nozzle holes a rough calculation showed:

$$\frac{\Delta V}{\Delta t} = \frac{\pi \cdot d_i^2}{4} \cdot 3 \cdot \mu \cdot \sqrt{\frac{2}{\rho} \cdot \Delta p} =$$

$$= \frac{\pi \cdot 0{,}26^2}{4} \times 3 \times 0{,}7 \times \sqrt{\frac{2}{780} \cdot 200 \cdot 10^5} \times 10^3 \quad mm^3/s$$

$$\Delta V_{injected} = 75 \ mm^3/cycle$$

$$\Delta t \rightarrow \varphi_{750} = 13{,}5 \div 14 \ °CA \quad (27 \div 28° \text{ on engine})$$

Although the value of 13,5°CA for 750 rpm. seems to be high it may concluded, that with such injection time the engine may operate satisfactorily. Again, it demonstrates the low engine power output

However it may be of benefit to increase diameters of the holes to 0,28 mm. The increase proposed, ranges about 15-16% in eff. flow area but it has to be confirmed with the measurements described in Fig. 4.

2. With regard to the calculated plunger lift for a delivery of 2,8 mm it can be seen that the ratio: 6:2,8 is extremelly high.

Besides that, 2mm of the max. lift must be reduced because of the limited Hertz permissible pressure (See Fig. 1).

It seems reasonable to adopt a plunger dia of ø9mm.

3.The sum of the dead volumes of the high pressure system amounts to 3500 mm^3 and this is increadibly large for the engine considered.This is especially so, because of the compressibility of methanol.

Reducing he tube length (if possible),inserting the relief valve spring and reducing the interior diameters of the retraction valve cap, is likely to reduce, whole dead volume for 25% or to 2800 mm^3.If the high pressure tube can not be shortened it is reasonable to reduce it inner dia from ø2 mm to 1.5 mm. because of the low fuelling level.In this way the tube dead volume will be reduced by 825 mm^3 or by 78%.

However,with reduced inner dia. of the tube the volume retracted may not be changed (V_R=const).

In any way check the tube acc.Fig.8.

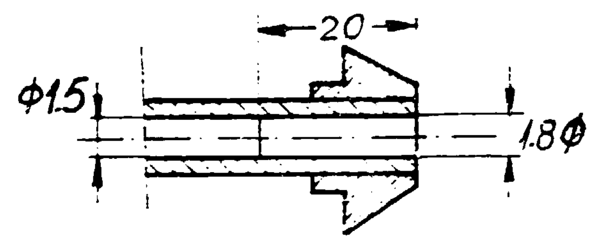


Fig.8 End of the tube.

4.In respect to the fuelling of methanol and the system dead volume of only 30 mm^3 is unsufficient.If after injection in engine operation was really avoided (because of low rate of spilling), the residual pressure,although not measured,seems to be too high.Actually,the sufficient nozzle leakage overflow observed is a good signal of the excessive residual pressure.In this case the fuel pressure diagram has been shaped acording Fig.9.

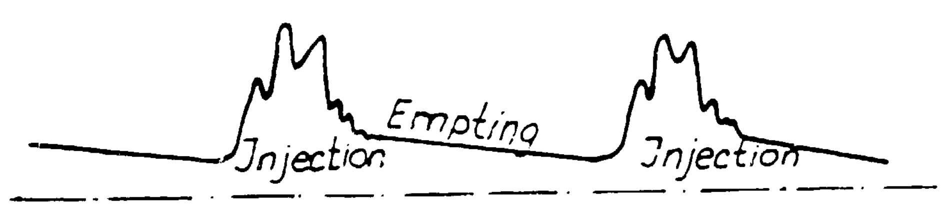


Fig.9 Residual pressure in excess and undefined .

It may be recommended, with the dead volume of the high pressure system unchanged, to increase the retraction volume by 60%. However, if the dead volume can be decreased (if's preferable) than the relief valve piston may be unchanged.

5. Needle of the injector should be redesigned according to Fig.10.

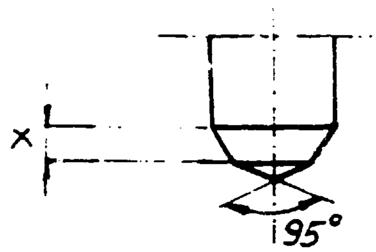


Fig.10 Needle tip proposal

The size X has to be defined in agreement with the injector produced.

The movable parts of injector have a mass in exess, the transmitting stick can be omitted (Fig.11).

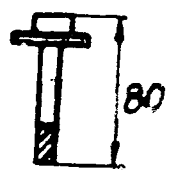


Fig.11 The transmiting stick

This proposal is given not only because of the needle seat service life, but to "Corona" formation and to overheating and holes coking can be produced.

6. With increased plunger dia. as well as with increased nozzle flow area timing may be retarded and removed foward TDC. In this way the faster burning velocity of methanol fuel may adopted and at the same time late combustion avoided.

7. Because of the lower evaporation temp. of methanol as compared with that of diesel oil, the pressure in the pump gallery has to be increased and the more efficient gallery space cleaning undertaken. During experiments in engine operation the pressure and temperature measurements in the gallery are to be recommended, as well as

a looking glass, to supervise fuel vapour.

8. In matching the injection to the specific engine demands is no use to begin immediately the running engine.

The procedure is as follows:

- collect the data and organize the bank (drawings, fuel properties, separate data of some test benches-A nozzle etc.).
- using computer techniqiues, calculate the injection parameters of interest for the possible combinations
- evaluate the calculated data and measure the injection parameters of the system chosen, on the pump test bench.
- evaluate again but with a running engine
- if fuel changes, long term testing on the separate bench have to be included.

(About 500 h, see SAD-truck producer praxis).

All the items cited above have to be improved in IIP.

Laboratory equipment was explained together with the measuring procedure. Three books were presented

- injection and carburation of fuel
- data bank and calculations
- droplet burning

The diagrams relating to the matter considered were given and the discussions organized.

With dias, drawings and interim reports the training was supported.

9. Spray-time penetration and its arrangement in the combustion chamber are mostly influenced by injection and by combustion itself. However, when converting to methanol, spray-time penetration will be higher and the fuel-comb. chamber contact may increase. The latter may not be a problem with relatively low loads. However, with upratings the heat release rate may be poor, especially in cold operations and with full loads.

Atomization of methanol may be coarser than in the case of diesel oil. This finding may be unexpected but there is an explanation for this (cougulation).

The heat of evaporation of methanol is 1109 kJ/kg and that of diesel 251 kJ/kg or by factor 4,4 more energy needed for methanol fuel. Further, the calorific value of methanol amounts 19.665 kJ/kg and that of diesel oil 43.100 kJ/kg or by factor 2,19 more fuelling

considering the metering by weight.The result of the aforementioned is a poor methanol ignition quality.
Moreover,the time when the methanol will be widely applied in service operations "the ceramic engines" may be already in the series productions.It means not the adiabacity, but the isolation of the combustion chamber components and the exhaust ports.
The combustion chamber of Di-diesel engine isolated with ceramic offers the possibility for better fuel ignition quality.In our experiments we isolated only the piston combustion chamber according Fig.12.

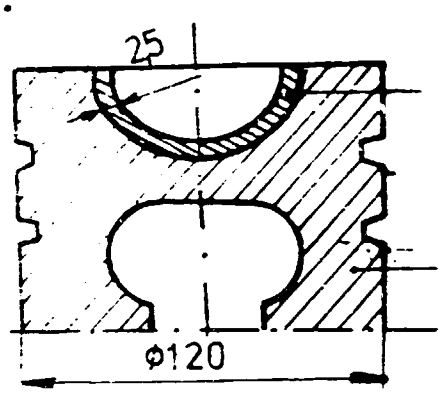


Fig.12 Isolated piston comb.chamber
(VTŠ-Maribor - TAM)

KHD patented the ceramic piston for truck engines shown in Fig.13 ,using toughened partially stabilized zirconia for isolation.
With the piston presented in Fig.12 we concluded that the fuel cetane number could be reduced by factor 2-2,5 for the DI diesel engine applications.Thus,in this way energy may be saved and the ignition quality improved.Mixing the ignition improver (kerobrosol) with methanol fuel and isolating the combustion chamber the ignition plug may omitted.Further experiments with isolated piston bowl are in progress.It may be recommended that IIP also takes the ceramic into consideration.
Recent developments in electronic techniques give the possibility of measuring ignition delay without engine modifications and it should be possible to use these as the basis for a new system.It may be of interest in methanol operation for larger diesel engines.
Regarding the type of diesel engine a modern injection system has

to meet the following requirements:

- High injection pressures to achieve short injection periods, effective atomization and effective fuel dispertion.
- Triangular-shaped law of injection with high maximum rates and short end of injection.
- Constant commencement of injection, constant injection period, and constant delivery volume as a function of speed at a constant effective plunger stroke.
- Steady delivery volume characteristics without instabilities as a function of pump delivery volume and speed.
- Avoidance of excessive retractions and after injections during the whole operating range of the injection system.

The most important measures taken to improve combustion of DI-diesel engines are as follows:

- late commencement of injection
- high compression ratio
- high volumetric efficiency
- accelerated combustion
- more effective air utilization.

The late commencement of injection and the high compression ratio result in small ignition lags so the fuel is still injected at the commencement of comubstion and during combustion. This leads to a smoother combustion with lower pressure rise.

The engine requirements for late start of injection imply an extremely short injection period, to prevent combustion from being shifted too far into the expansion stroke (late combustion). The fuel spray now have enough energy to penetrate the air mass even at low speeds, so larger diameter bowls are possible.

Several parameters of the basic combustion system can be designed to provide a short combustion duration with control of the initial characteristics, when high pressure injection is available. These include:

- minimization of the ignition delay period by increasing compression ratio (especially on NA engines) control of in-cylinder air motion and control of fuel spray properties.
- Optimising the fuel-air mixing process through the use of optimized air motion, a larger number of spray plumes and higher

air velocities.

-reducion of the tail-off of heat release rate at the end of combustion by minimizing the amount of wall impingement fuel, by matching fuel penetration to combustion bowl size and providing sharp cut-off at the end of injection (See Dr.Herzog-F&M, Parker-John Deere).

PSZ-toughened partially stabilized zirconia-Nilsen Sintered Products (Australia)

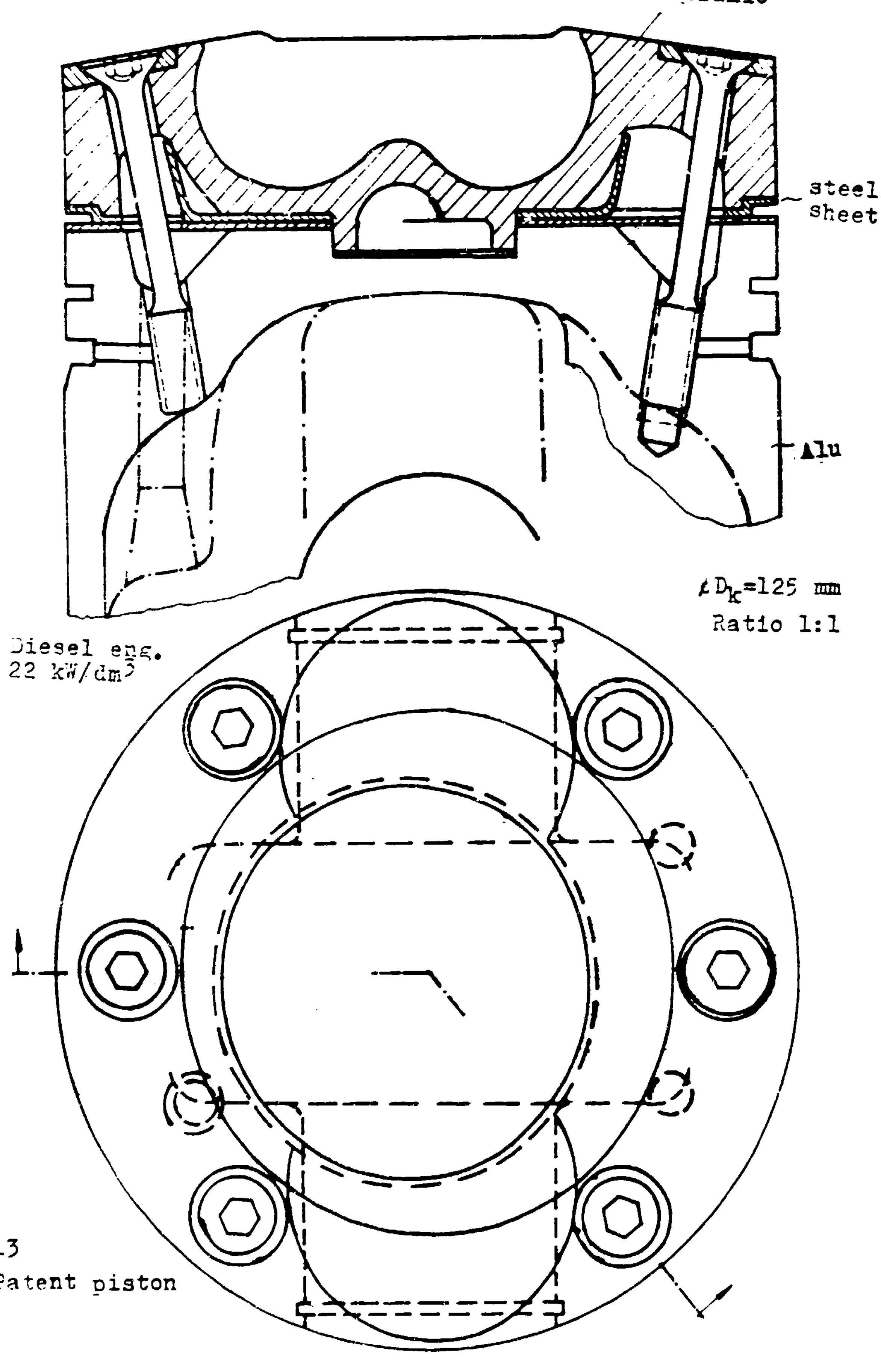


Fig.13
KHD-Patent piston

Prepared for IIP

PART II

"The impact of the crude oil resources upon the developement of 4-stroke high speed DIESEL ENGINES"

(Prolongation - 9 workdays)

Černej Anton

Dehra Dun, Sept. 1983

1. Introduction (Ono Syassen)

Concerning with small engines attention is given to the problem of adapting these engines:

- to burn modified fuel qualities
- to use alternative fuels
- to reduce fuel consumption

2. Alcohol

When using alcohols (Ethanol or Methanol) we meet eight main problem:

ENGINE
- poor ignition quality
- poor lubrication ability

FUEL INJEC-TION SYSTEM
- considerable tendency to cavitation and corrosion
- aggression toward elastomers
- considerable increase of fuelling metered by volume
- increased compressibility of the alcohol fuel in the high pressure dead volumes
- low evaporation temperature (cavities, gallery enviroment, overflow at nozzle)
- low density and increased leakage

The seven last points are mainly related to injection equipment but the following new developements my also be considered:
- oil lubricated in pump driving
- measures related to cavitation and corrosion
- resistante elastomers
- compact pumps

The main problem is the poor ignition ability of the engines. In order to overcome this we have 5 possibilites at our disposal:

- pilot ignition
- doping of ignition improver
- glow plug
- spark plug
- FM-spark plug - process

In the possibilities above cited, there are still some likely variations.

For the porpose of completition it should be mentioned, that "heat isolated engines" with combustion chamber wall temperature of about 700 °C, are "automatically" disposed to accept alternative fuels.

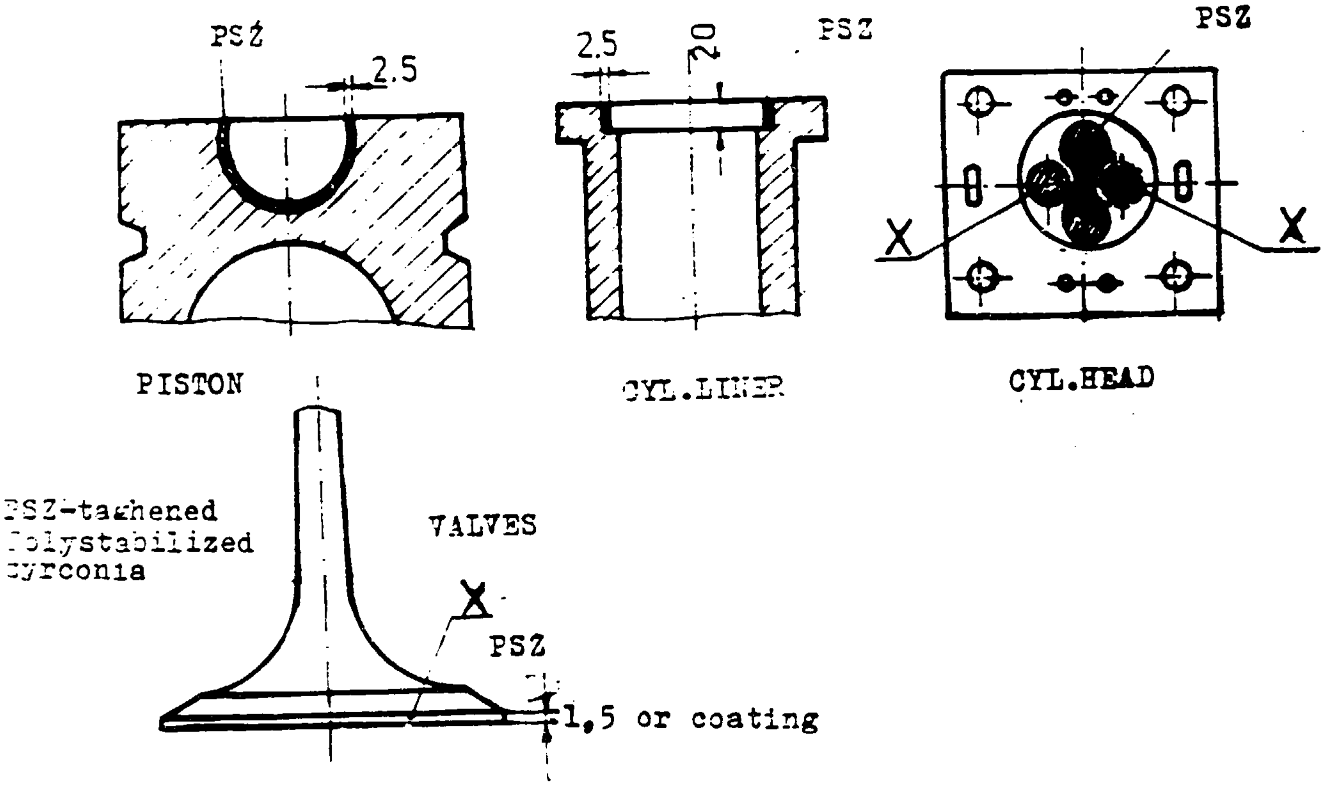


Fig.1 Sufficient isolation with PSZ ceramic to maintain 700 °C of the comb. chamber walls

Altough ceramic isolation may be applied, low rated speed, startings and cold operation problems still remain. However, fuel consumption will be reduced and glow plug service life will be improved.

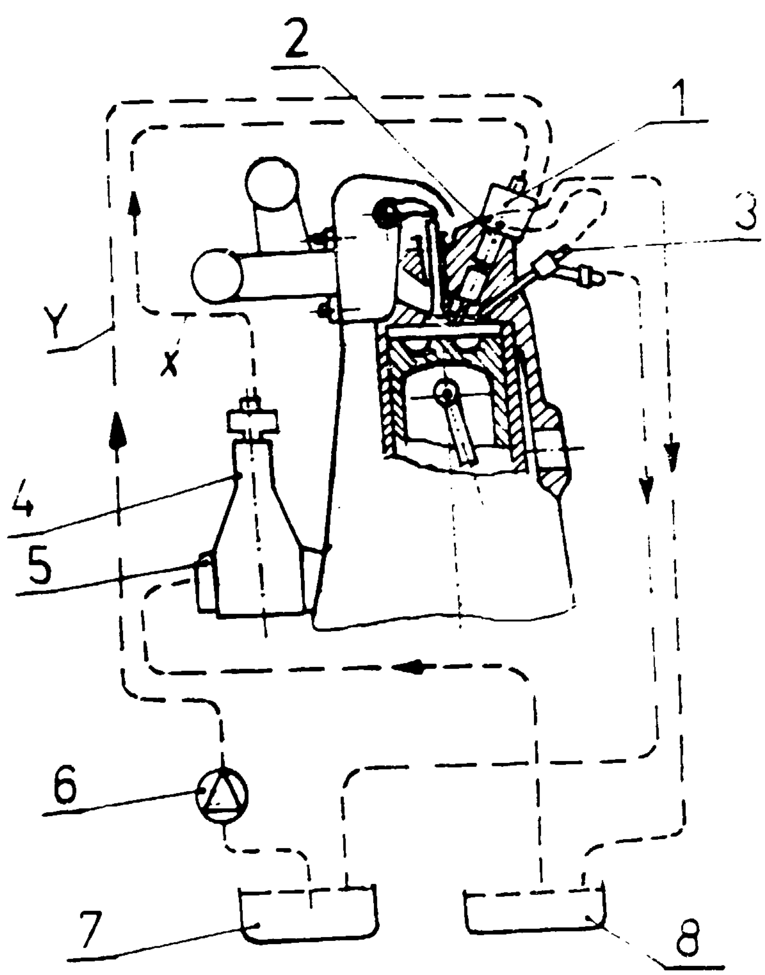
All the forementioned methods, for alcohol conversion, have their advantages as well as their disadvantages. At this time it is not completely clear which conversion will be the best. Unfortunately it seems, that independent national solutions may come ("island solution") in dependence of recources, fuel consumption and of the rafineries on desposal. For engine producers and carriers this is no joyful prospect.

However, for very large countries such as: Brasil, India, China etc. a resonable compromise is possible.

One of the main disadvantages of the pilot ignition system was, that it had two separate injection pumps. For small engine systems it is too expensive, pilot fuel metering control is difficult and the pilot injector becomes overheated. MWM solved this problem with "pressure distributor" and installed only one single pump for both, methanol and diesel oil injection.

MWM - pilot injection system with one single pump and pressure distributor is shown in fig.2. The low pressure pump 6 (in fig.2) may be ommited, if diesel oil tank (small one) is elevated.

MWM solution (1980)



1-pressure distributor -P
2-standard nozzle holder--methanol injection
3-nozzle holder for pilot ignition
4-high pressure pump for methanol
5-low pressure pump for methanol
6-low pressure pump for diesel fuel
7-diesel fuel tank
8-methanol fuel tank

Fig.2 Only one HP pump serves, by means of P distributor, the main and the pilot injection

The pressure distributor may be small and directly connected with the injector inlet - 2. Instead of a glow plug, a pilot injector was installed. The pump 4, as before operates with methanol and its "impuls", by means of distributor 1, serves both injectors.

The injector 1 as "pencil" nozzle may replace glow plug. Note: because of the cooling problem the injector 3 has to be removed as far as possible, almost to the combustion chamber edge.

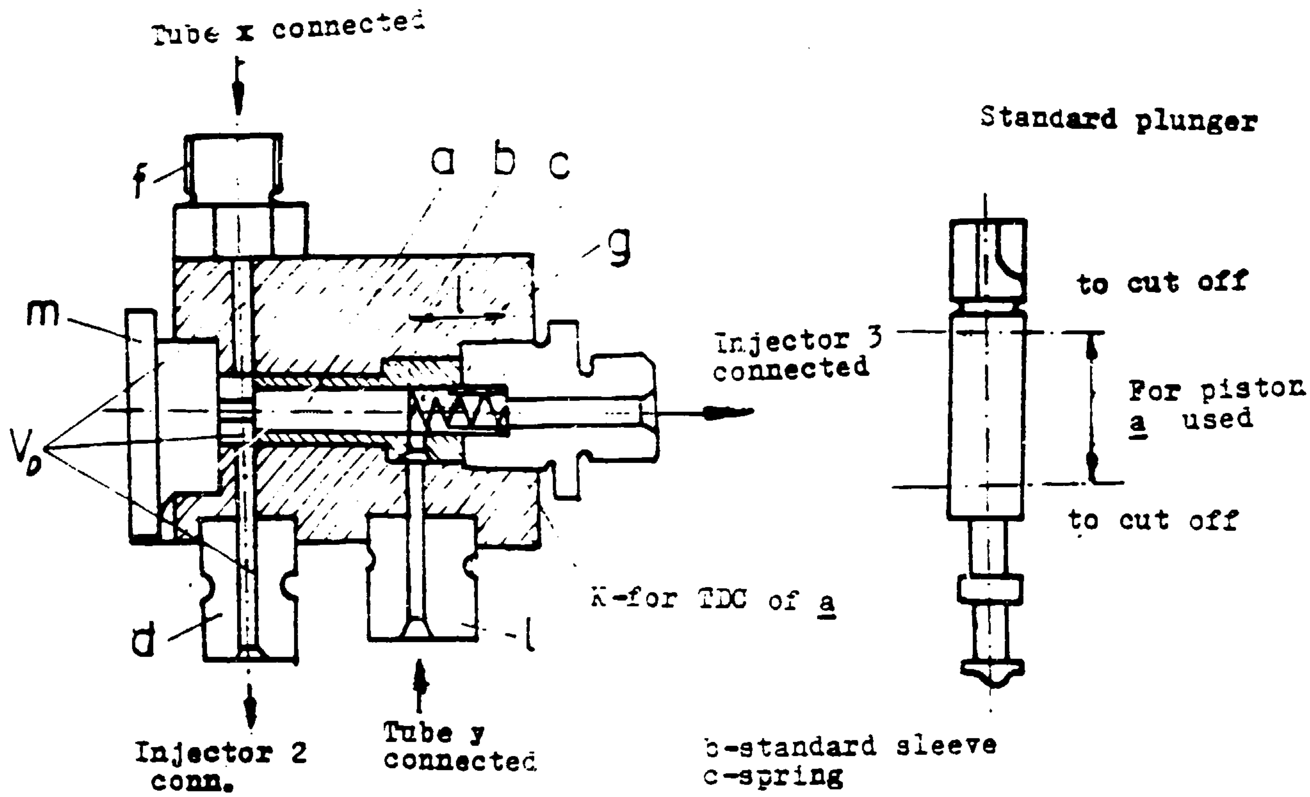


Fig.3 Pressure distributor, see Fig.2

In Fig.3 the pressure distributor is only sketched and a more accurate corresponding drawing has to be made. In the design the following should be considered:

- the distributor may be smaller than that shown in Fig. 2
- the connection "d", "e" and "f" may be mutually (along the circle) dislocated
- the connection "d" may be omited and the body "g" can be twisted directly on injector 2

- the dia. of the piston "a" has to be small, its stroke short and the whole dead volume V_D, as small as possible
- the role of the spring "c" is in moving piston "a" at BDC, the diesel oil injector start is controled by the spring in injector 3
- the opening pressure of injector 3 has to be 10 ÷ 20 % smaller than that of injector 2
- the sealings were not considered in Fig.3.
- deaeratetion and first fillings have to be solved
- the insert K may be easily changed

Having the pressure distributor and the appropriate pencil nozzle we may also have the possibility for comparable dual experiments, with glow plug and with pilot injector. Here, there is no need to change the cylinder head.

If the engine has some ceramic, see Fig.1, then is possible to cut off diesel "fuelling" in an appropriate way in warm operation. Thus, the pilot serves only for starting, cold operation and low loads. This may be done by blocking the piston "a" with "m". For the stationary applications this could be very promising.

3. General approach - combustion

Some significant features of methanol:

1. Compared with vegetable oils the combustion of methanol is "cleaner" but a small quantity of vegetable oil may be doped to methanol for the purpose of lubrication.

2. Methanol may be produced on a large scale and from renewable resources; it may be stored for a long time and handling techniques in service and distribution nets remain conventional.

3. Being monocomponent liquid fuel methanol quickly evaporates at 65 °C. Diesel fuel possesses a boiling region of 170 ÷ 370 °C.

4. Compared with diesel oil methanol has:
   - 2,3 X lower stichiometric ratio (kg air/kg fuel burnt)

- 10 X lower cetane number
- 2,5 X lower calorific value
- 4,4 % lower density
- 4,42 X higher heat of evaporation
- 5,54 X higher cooling of stoichiometric mixture during evaporation than that of gasoline
- 5 % volume increase by combustion comparing with gasoline
- H/C high and is "doped" with "O" (oxygen)

Our conventional diesel figure of heterogenous mixture formation and figure of fuel spray significance may not be applied to methanol. With diesel fuel the first HC components to evaporate, initiate ignition. With methanol a glow plug or pilot fuel initiates the beginning of combustion. The quick evaporation or rapid "gasification" of methanol forms a more homogenous mixture.

The ignition delay may not be highly influenced because of the high heat of evaporation. A higher air-fuel ratio in methanol operation or a higher inert mass presented does not allow a high decrease of temperature. This statement in our experiments has already been demonstrated.

Oxygen "doped", with high H/C ratio, methanol fuel reveals better conditions for oxidation and thus, for faster sootless combustion. Just having a significantly lower boiling point than that of diesel fuel, the evaporation is rapid and vapour distribution in the combustion chamber is very fast. The latter means, that here more air-gas mixture has to be considered together with the more homogenous "fuel--oxygen" time contact, than that, found with heterogenous diesel oil-air mixture history.

From the consequences of the above there is one very important point, to be observed; that the more homogenous mixture accelerates the flame propagation or combustion. This "disadvantage" we have to use for our benefit. It is useful to remember, that for years we've learnt about diesel oil, and now we must learn how to live with methanol fuels.

However, some experiences have alredy been collected and those of interest may be reported as follows:

- with methanol the form of the piston bowl as well as the intensity of the air swirl, are not of special importance. Morever, the swirl intensity may be reduced, so the heat transfers to the combustion chamber walls.
- spray macro distribution is not as important as that for diesel fuel. Thus, converting to methanol and increasing the fuelling, one or two nozzle holes may be added without any special difficulties in operation. This means that the nozzle holes may be increased (supposing that ignition is not disturbed).
- with a faster injection rate the optimum timing has to be retarded but when operating with methanol the optimum position for the injection end does not depand on fuelling. Burning alcohols in a DI diesel engine may not demand change of timing with the engine load.
- using the combustion chamber isolation with ceramic and high injection rate of methanol with a shorter injection period and with retarded timing, the properties of methanol offer the following:
  a) our dream may becomes truth in accelerating combustion by the piston moving down
  b) light engine design of interest especially for passanger car applications
  c) sootless exhaust and thus the catalytic THC removing, it means less pollution including noise emission
  d) good combustion efficiency
  e) less energy for the glow plug and increased glow plug service life
  f) upratings with turbocharging without air cooling
  g) less soot emmitted by torque back up matching, for truck applications. The CO - exhaust limit at low speeds and higher loads may be also increased in methanol operation.

4. Experimental approach

1. The measured methanol specific consumption rates have to be reduced as follows:

$$be_{Eq} = \dot{m}_A \frac{H_{MF}}{H_{DF}} \cdot \frac{1}{P_e} \cdot 10^3 = \frac{\dot{m}_A \cdot 0{,}4 \cdot 10^3}{P_e} = 400 \frac{m_A}{P_e} \left[\frac{g}{kWh}\right] \cdots (1)$$

$\dot{m}_A$ - [kg/h] - methanol consumption rate as measured

$\frac{H_{MF}}{H_{DF}}$ [-] - $\frac{\text{methanol waterless}}{\text{diesel oil}}$ calorific value ratio

Pe [kW] - power output as measured

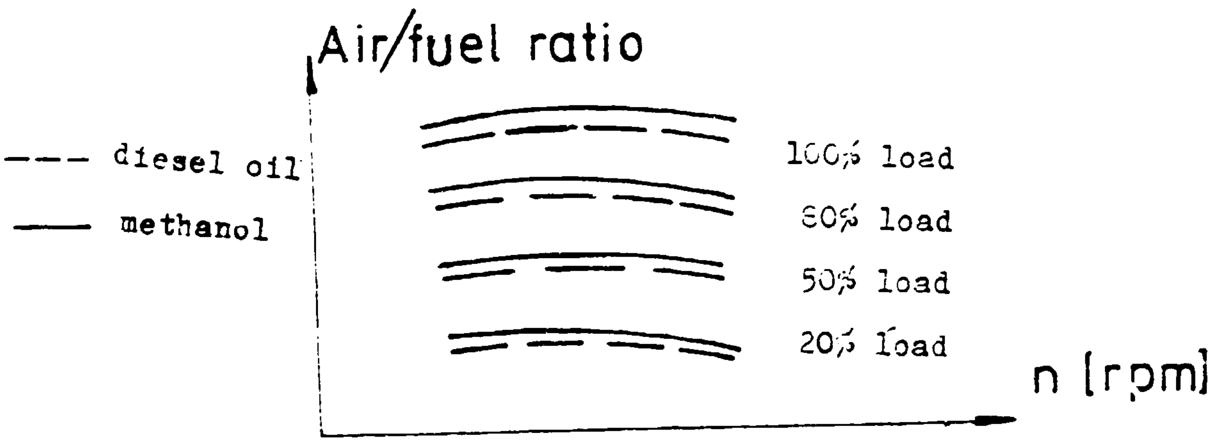
2. For the porpouse of comparison with other engines developed, the calculation of the equivalent per cycle burnt specific fuel mass is useful:

$$b = m_A \frac{H_{MF}}{H_{DF}} \cdot \frac{1}{V_H} \left[\frac{mg}{dm^3 \cdot cycle}\right] \qquad (2)$$

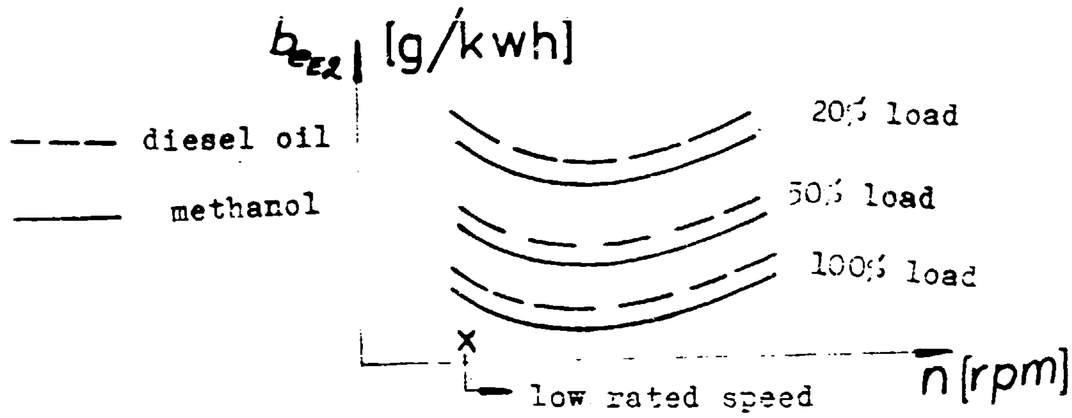
m_A [mg/cycle] - per cycle burnt fuel mass

V_H . [dm^3] - swept volume

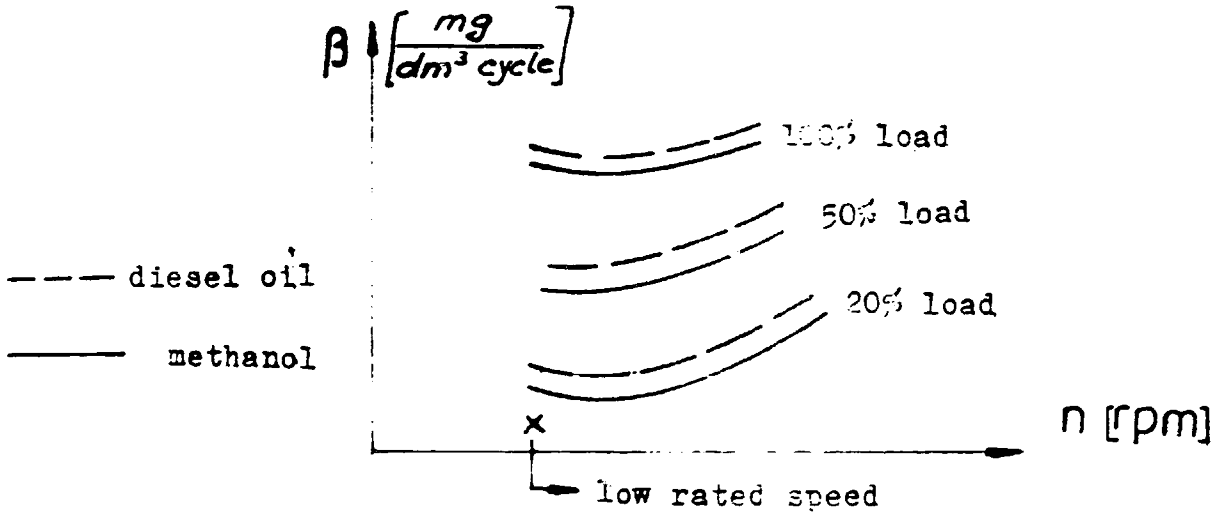
3. For the both diesel oil and methanol operation the next diagram has to be evaluated:



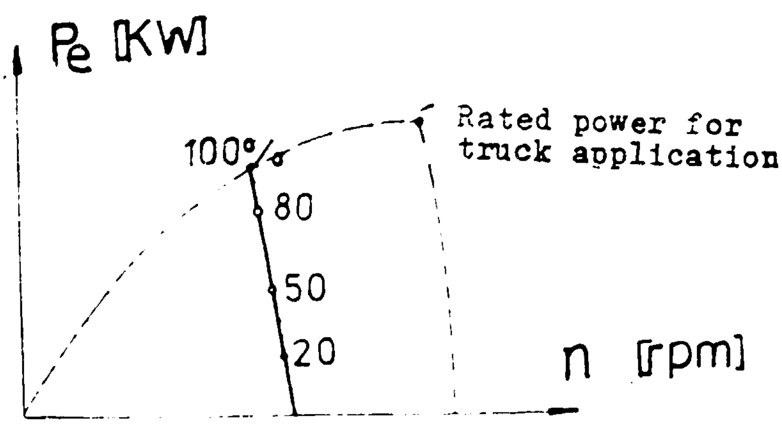
4. Specific consumption rate (eq.1)



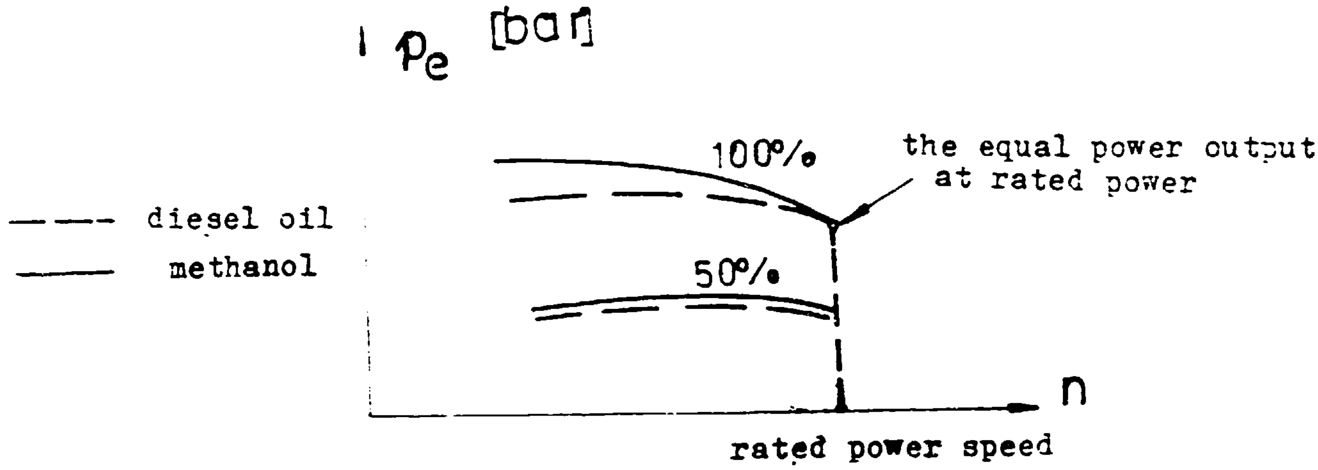
5. Specific consumption rate per cycle (eq.2)



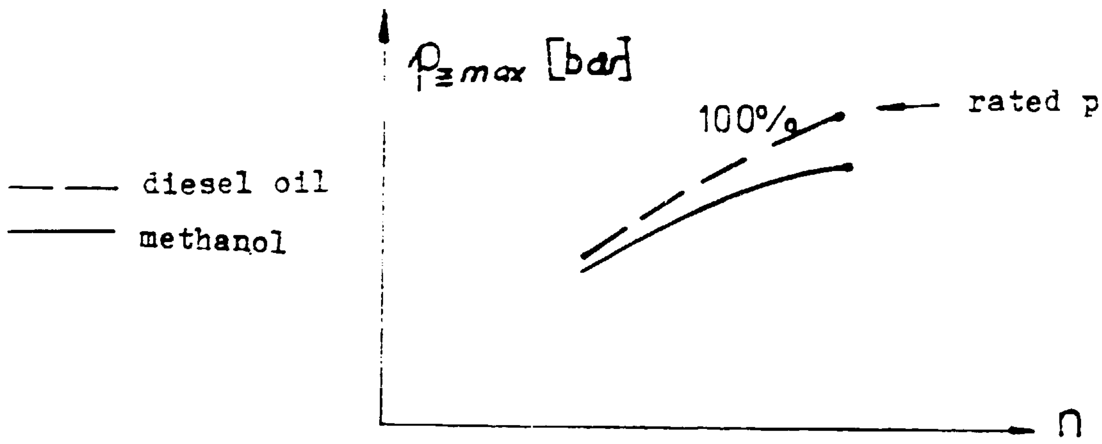
6. In the generator, water pump etc. applications the same has to be done along the rated line as follows:



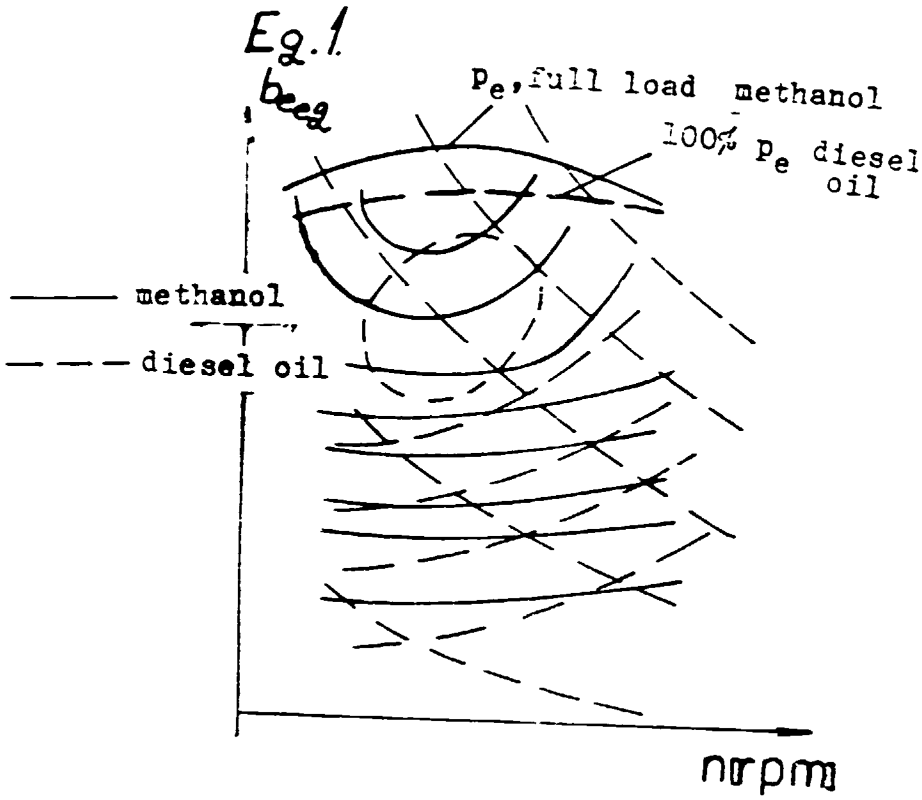
7. A comparison of the mean effective pressure between diesel oil and methanol operations:

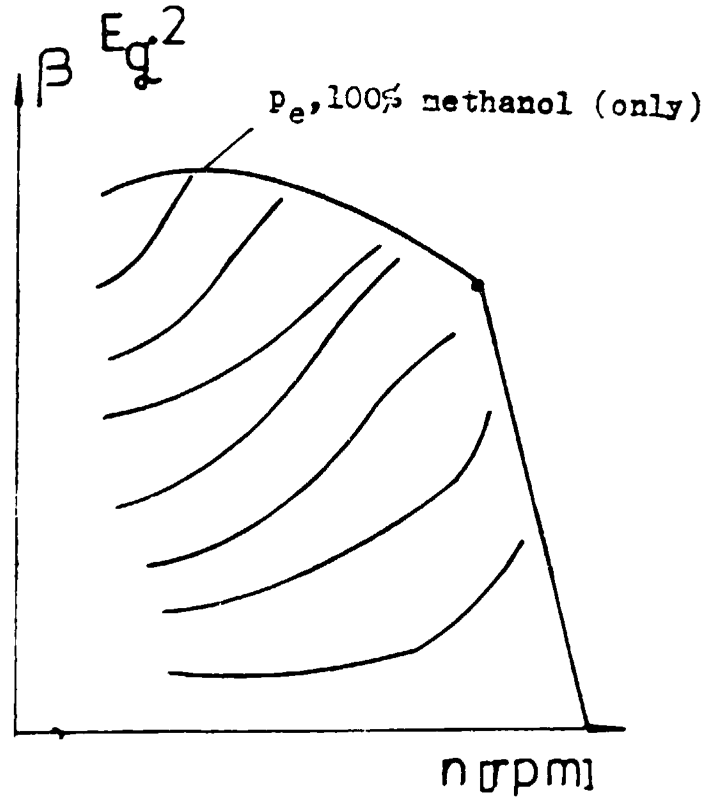


8. In cylinder peak pressure:

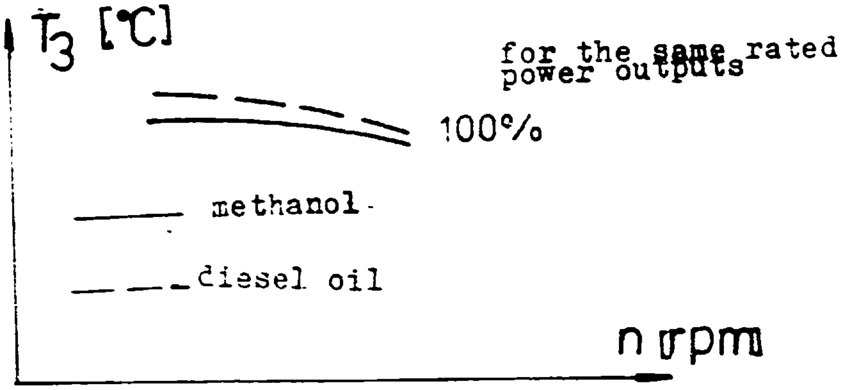


9. The following diagrammes are useful:

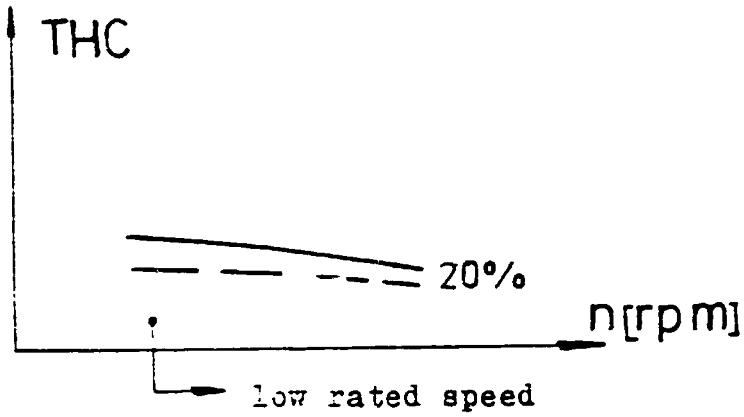

Eg.1
b_{eeq}
p_e, full load methanol
100% p_e diesel oil
methanol
diesel oil
n[rpm]


Eg 2
β
p_e, 100% methanol (only)
n [rpm]

10. **Exhaust gas temperature:**



11.

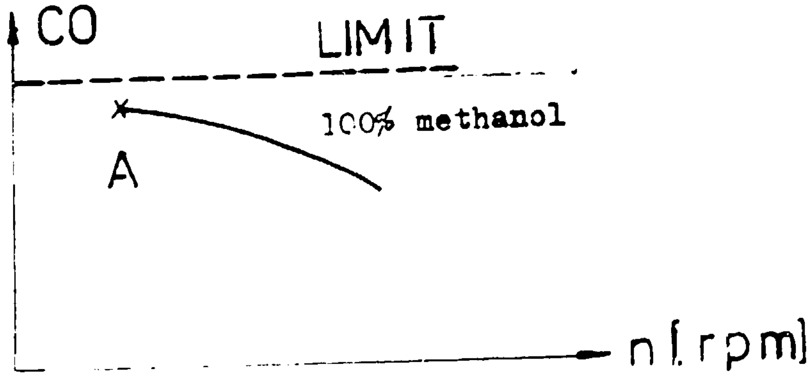


Note: because of C_3H_8/N_2 calibration procedure and FID response considering TCHC components do take:

$$\underset{\text{(FID output)}}{THC} \approx 2/3 \underset{\text{(measured)}}{THCO}$$

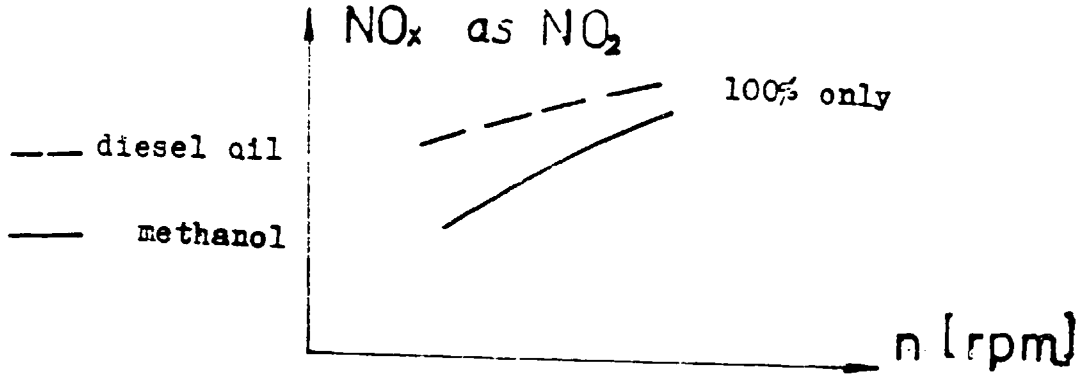
With THCO components FID was'nt calibrated.

12.

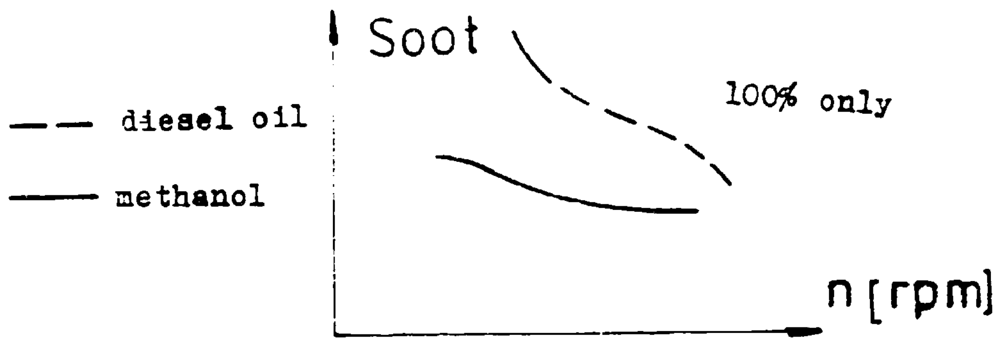


With increasing torque back up in methanol operation check point A - lowest full load speed

13.



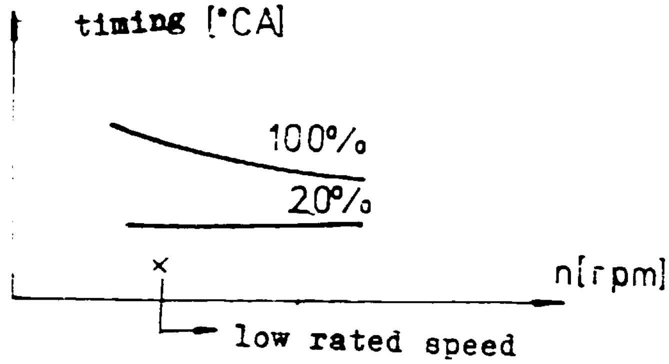
14.



15. The values in diagramms 11, 12 and 13 are not "as measured". Using the data from diagram 3 the measured concentration in ppm has to be recalculated as stoichiometric. Without such recalculations the presentation of only the measured data, has no significance and can not be compared with the other coresponding collected data.

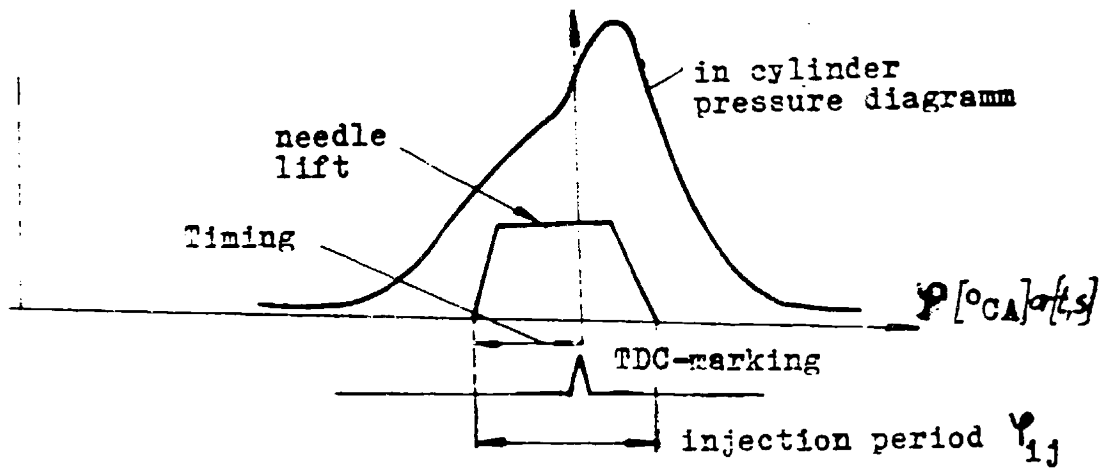
16. Timing and injection period

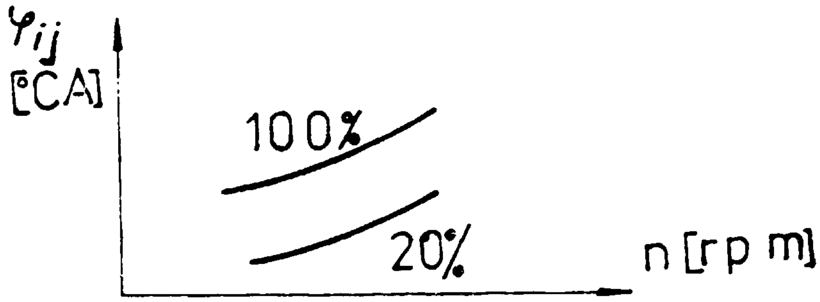
For the one static timing set, we needed for all the experiments



This can be done with measurments:

- of needle lift
- and with accurate TDC position marking

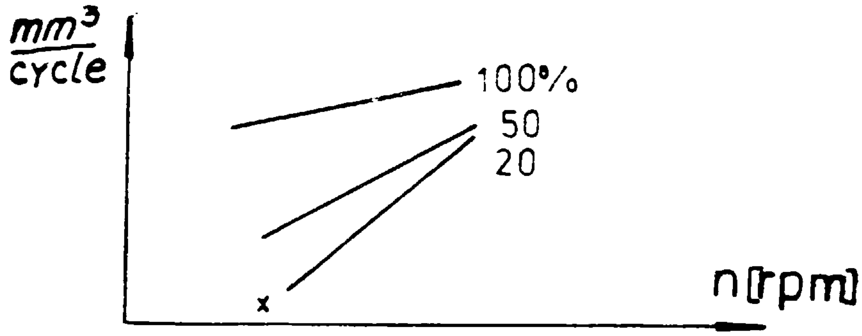




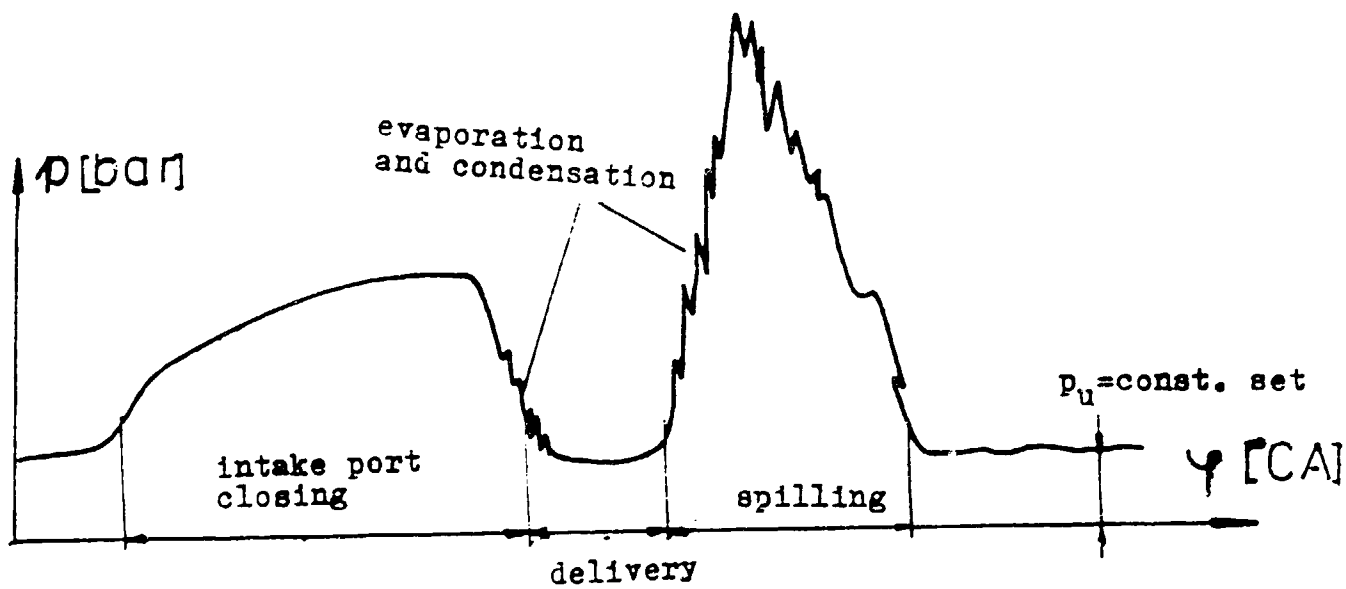
Needle lift is to be constantly observed while the engine is in operation, because of:

- certain possible irregularities
- after injection problems
- oscillations of the needle at low speeds and with low loads

17. Fuelling in engine operations



With measured temperature and pressure of the methanol fuel in the injection pump supply gallery. Having a low evaporation temperature in warm operation (especially with single cylinder engine) fuel vapour formed may cause settings error. This is especially the case, when both spilling and intake sleeve port have a common gallery. In gallery then we may register the next diagram as measured with low pressure quartz tranducer:



As regards vaporisation, it may be, that our rack position was in error.

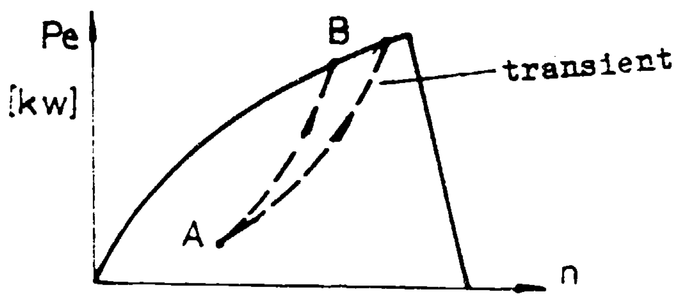
18. While the engine is in operation is advaitsable to check the oil in the injector pump looking for methanol; leakless element sets have to be used.

19. In order to prevent the evaporation at the injector leakage overflow, the overflow tube has to be connected with the filter fuel outflow.

20. The temperature of the glow plug must be observed and the glow plug energy consuption measured. In the case of an external glow plug power supply its energy must be incalcu- leted in relation to the fuel consumption rate (as was done in IIP).

21. In cold operation THC has to be measured as shown in point 11.

In order to check for misfiering two techniques may be used. Observing the in-cylinder pressure diagram or with THC measurment. In transient operation misfiering offen occurs.



The rack position from A to B may be changed in 0,1 s but the engine may responsed by misfiering.

5. Fuel injection matching

Converting to a methanol fuel system will be influenced by:

- 2,5 X fuelling
- the different physical fuel properties

The best approach may be as follows:

A - calculation
B - test bench, out of engine operation, matching
C - in engine operation approval
D - fixation of data matched

5.1 Matching

Having no other input data and converting one diesel engine to methanol operation, the best initial approach may be:

$$\varphi_{\substack{\text{injection}\\\text{period of}\\\text{methanol}}} = \varphi_{\substack{\text{injection}\\\text{period of}\\\text{diesel fuel}}} \qquad (3)$$

for the same rated power outputs.

However, to achieve the start condition (3) and to avoid mismatchings, we have first to consider all the injection events by calculation. Here we have 2,5 X more fuelling by volume, higher fuel compressibility and some lower pressure wave propagation velocity.

The tools in hand are:

a - plunger dia.

b - ratraction volume

c - cam

d - cam-follower prelift for delivery

e - dead volumes

f - nozzle holes cross-sectional effective flow area

The plunger dia. must be increased by about factor 1,5. Injection system operating with methanol is very sensitive to residual pressure, esspecialy at low speeds. The results may be unwanted poor torque back up.

It seems resonable to keep up the residual pressure level at about 1 ÷ 3 bar over atmospheric pressure. Accordingly, the retraction volume has to be matched.

The cam shape and especially the max. stroke can not be easily changed, if is not provided beforehand.

Cam-follower prelift for geomet. defined commencement of delivery, has to be increased. The limit here is the permissible Hertz contact pressure of the cam tip (as explained in Part I).

Dead volumes has to be decreased as much as possible. The possibilities:

- smaller inner tube dia. (seefor resistance)
- smaller dead volume in the relief valve holder by inserting

the spring.

The second most effective tool, besides plunger dia., are nozzle holes. They may be increased in dia. - as first attempt - by factor 1,2 ÷ 1,25. Here, depending of the geometry proportions and combustion chamber, may be of benefit to increase also the hole number (instead of 3 take 5 without increased dia. of the hole).because of fuel dispersion.

It may be suggested at first: to increase the plunger dia. and nozzle holes, to correct the retractin volume as well as to reduce relief holder dead volume ($\sim V_R = 50 \div 70$ mm^3)

By calculation, all the influencing parameters have to be checked and the appropriate correctons introduced. All the input data calculation procedure are in the two books presented.

For the system and presetings chosen, the pump test bench experiments have to show:

p_{II} - before the nozzle
p_o - in the system
h_i - needle lift
φ_{ub} - injection period
$\Delta\varphi_z$ - change in the retardation
V_c - fuelling
$\dot{V}_{ct}$ - injection low
d_{32} - Sauter mean dia.
l_y - spray penetration

Check for after injection and needle lift oscilations at low speeds. Check for injection irregulaties of very low loads.

Note: the system means that the tubes and injector must be the same as for the engine.

The above diagramms have to be made for full load at:

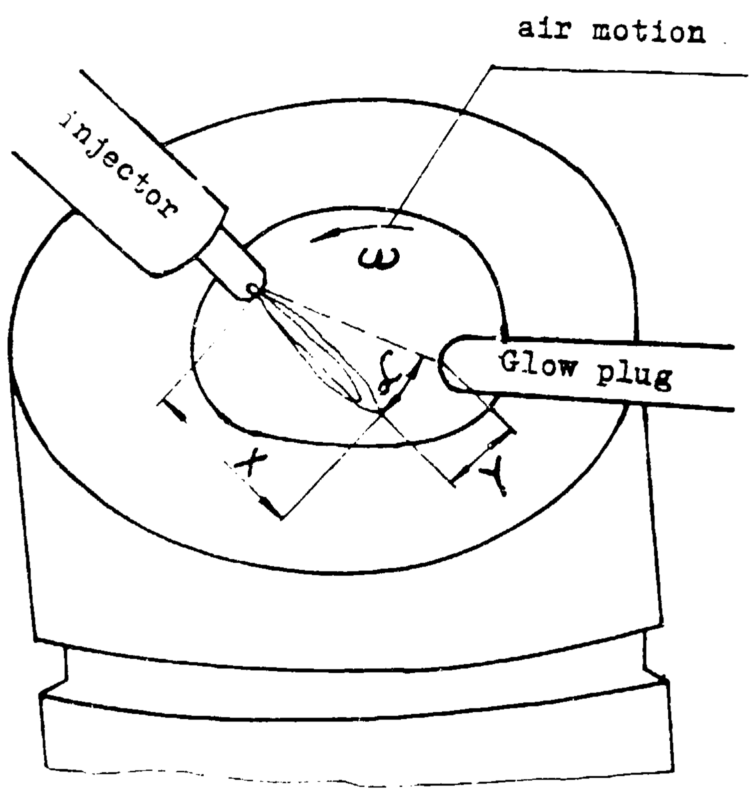
power rated speed
max torque speed
lowest full loaded speed

Separate has to be evaluated:

- mean p_{II}

- max p_{II}
- ratio $\frac{\max p_{II}}{\text{mean } p_{II}}$ - as less as better
- V_C/φ_{ub} $\left[mm^3/^oCA \right]$ - average injection velocity of fuel

Attention must be given to the nozzle holes, their number, space distribution and Φ dia. depend not only of fuelling and of injection period wanted. Morever, with methanol the spray-air motion history at various speeds and at all possible loads, must be considered. This is not because of combustion, but because of ignition. Spray penetration and dispersion depend to a large extent on the operentional regime. Thus, spray penetration increases with speed increased, as does dispersion. Swirl intensity also increases with increased speed, so we have enough energy for combustable mixture formation at higher speeds. Now the position of glow plug or injector pilot spray have to be adapted (the same happens with the changed fuelling).

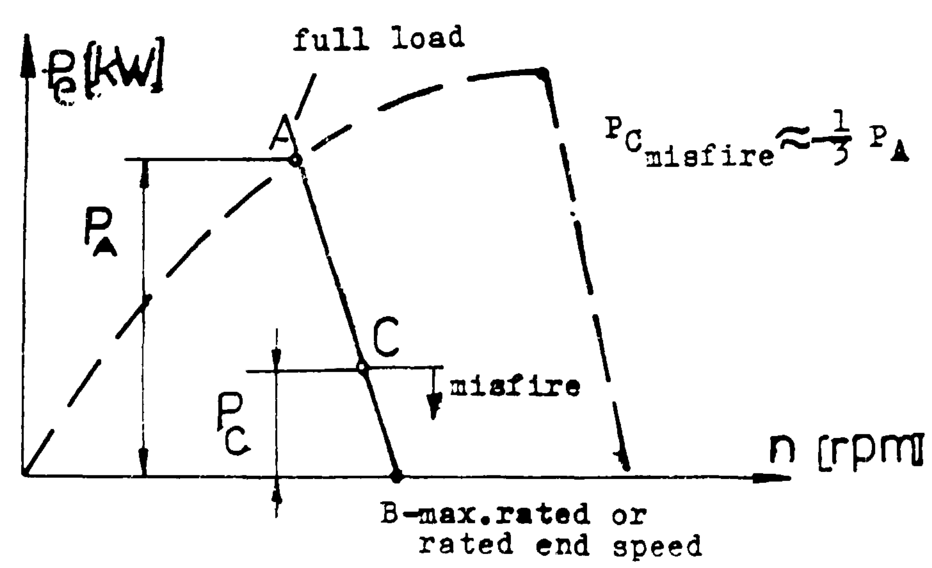


At low loads and/or at low speeds, penetration X decreases as well as air swirl intensity ω, thus, at the instant appropriate for ignition we dont have sufficient methanol concentration in the vicinity of the hot glow plug. The consequence of this is misfiring. This assumes a glow plug surface temperature $T_{gp} \geqslant 700\ ^{o}C$. Thus, air motion test rig experiments and spray deviation observations have to be complieted. Cnly in this way can we make any decisions concerning:

- the distribution of the nozzle holes
- the dia. of the nozzle holes
- the number of the nozzle holes
- the air swirl intensity need
- and for the positions of glow plug and of the injector given, to follow the ignition events with regard to operational regimes.

Only the total effective flow area of the nozzle holes gives us the computation, as well, as distance X in dependence of time.

However, we must also consider the application in service. The most difficult matching may be for truck applications and the most simple for the socalled single speed control engine, as in power units for generators, water pumps etc.



The P=grade of the line A - B which ranged about 10 ÷ 12 % is not importante to speed shifting. Thus, our discussion concerning the influence of the speed variation on spray and on swirl developement, may be ommited. Thus, we only have to consider the influence of loading at constant swirl intensity. This means, that we may now increase the injection rate and avoid the low spray penetration at low loads. The swirl intensity may be decreased, if needed. At higher loads now (P_A), we may expect more fuel surface evaporation, but this may be of little importance.

This discussion directes us once again towards the question of faster injection.

After negatiations beetwen IIP and fuel injection equipement producers (after all the considerations mentioned) two or three sample nozzles may be purchased. Here the providing time is rather long.

In engine test bench experiments we have to investigate timings observing fuel consumption rate, in cylinder peak pressure, exhaust temperature and combustion irregularities. Cold start and cold low loads operationhave to be checked. Here a hand advancer, is a very useful tool making possible the timing alternations as a running engine. With one speed control in methanol operation the optimam timing for all loads may be easily defined.

The fixation of data matched means, that we are obliged to repeat the fuelling pump test bench measurments $\left[V_C=V_C(n)\right]$ but now, with standard equipement (test bench nozzle, tubes etc.). It is also very useful in any contact with injection equipement producer.

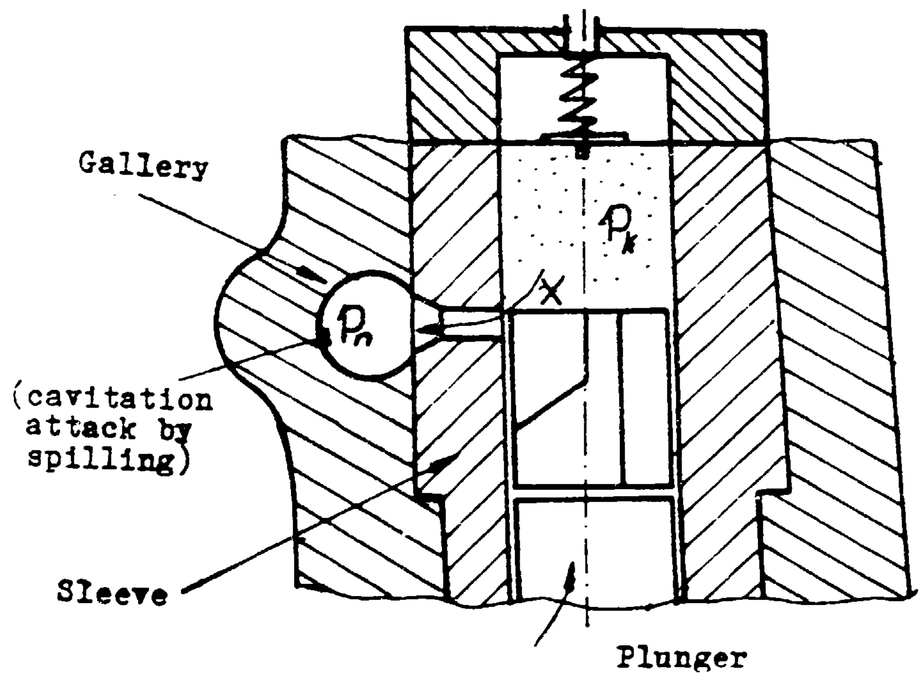
NOTE: because of the short time at our disposal, the time needed for data collection and for the experiments, this report is broken.

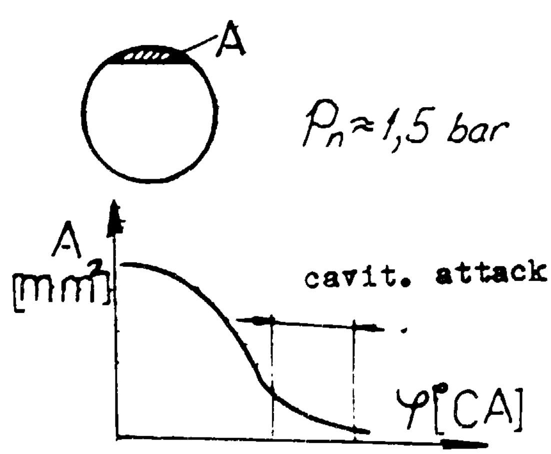
6. Some aspects pertinet cavitation attack

Operating with a methanol fuel injection system one frequently suffers from cavitation. Two types of cavitation may be encountered, the first is caused by the high out flowings and the second one is influenced by the high reflected pressure waves.

<u>The first type</u> usually attacks: spill and inlet ports of the pump sleeve, plunger helix and the top edge, gallery walls opposite the spill ports, and nozzle holes.

Just before the commencement of geometry defined delivery, the outflow intake port area is small and still decreases with increased plunger velocity. This results in abrupt evaporation in the high velocity outflow, accompanied by condensation. This process can be seen in the next figure.

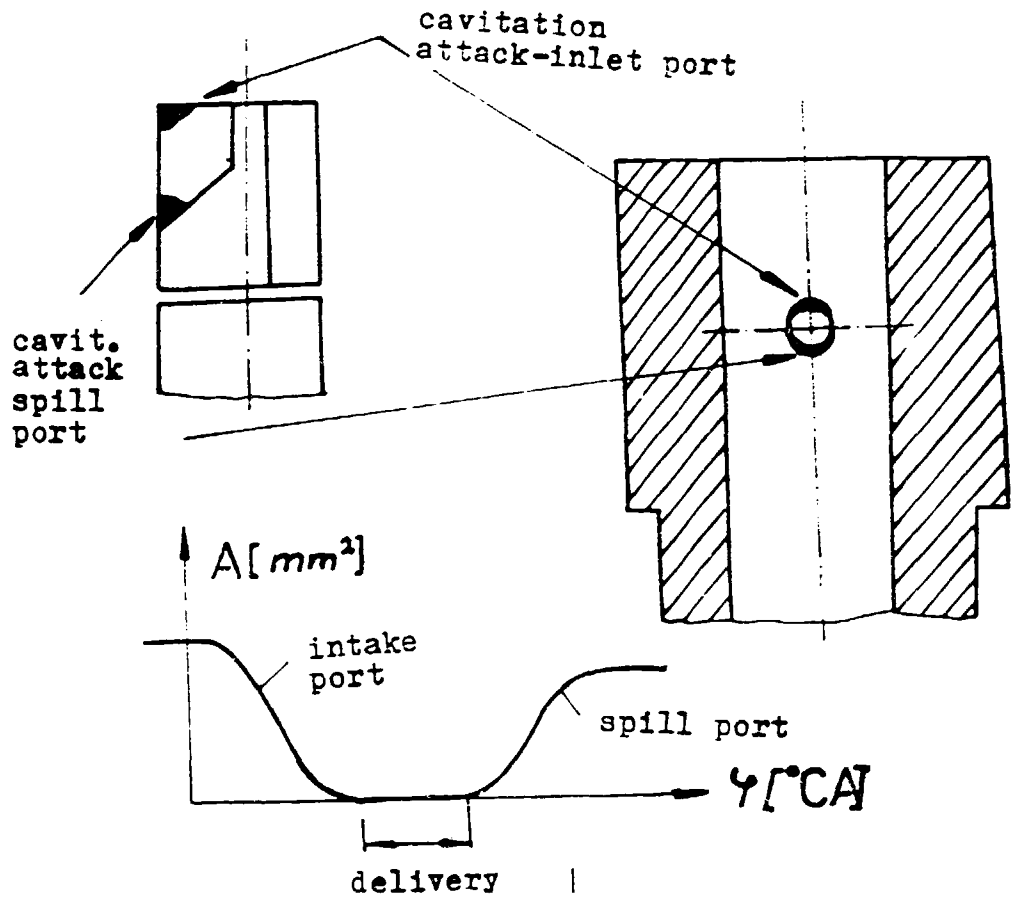




At the position shown $p_k \gg p_n$

$$\text{Velocity} = \sqrt{\frac{2}{\varrho}(p_k - p_n)}$$

Methanol has a low evaporation temperature.
As a consequence:



The same happens by spilling, but with much higher intensity, thus, the gallery may be also damaged by cavitation (as shown).

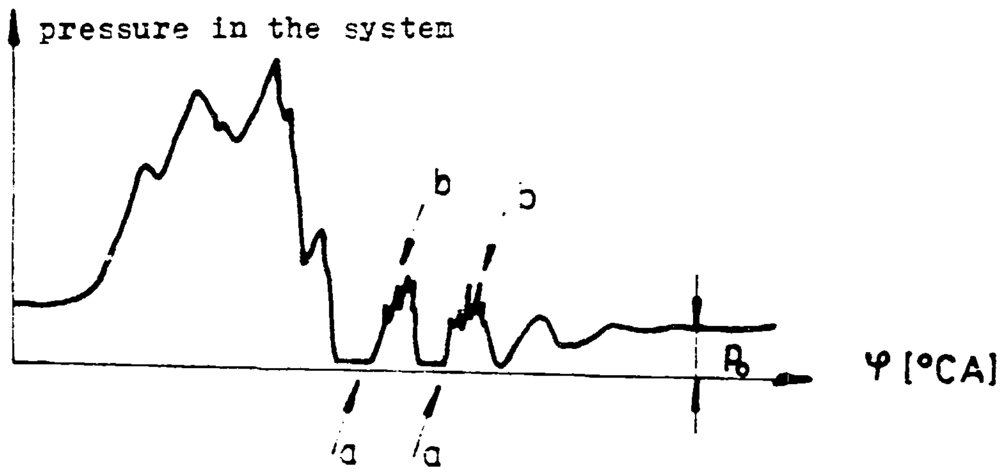
The main task in injector design is to keep the fuel energy (during injection) as much as possible in potential form and to transform into kinetic, by nozzle holes:

$$\frac{v^2 \varrho}{2} = p_{II} - p_z$$

Just because of the later again the cavitation, but now in the injector orifficies.

The second type of cavitation is closely connected with pressure waves after the injector needle closes. The reflected pressure wave developed after injection, if high enough, may produce cavitation at any position in the high pressure system.

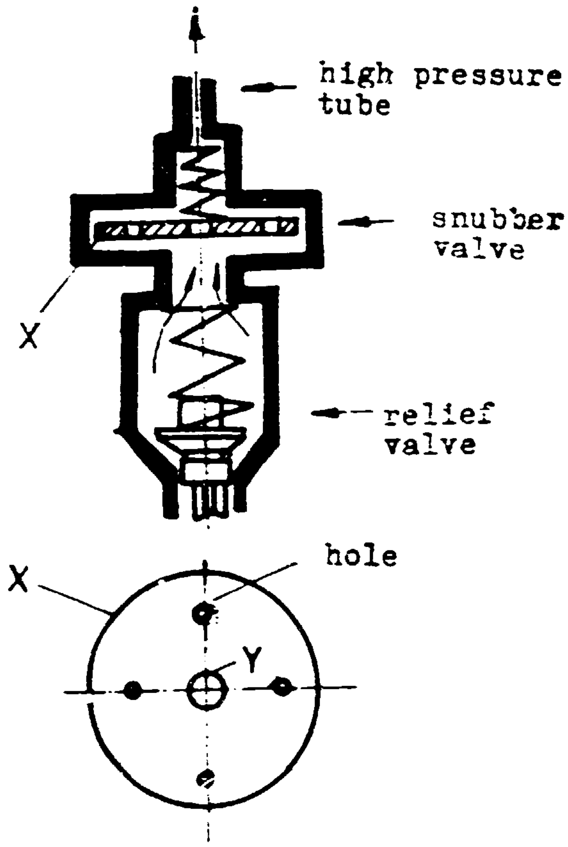
Pressure in the system

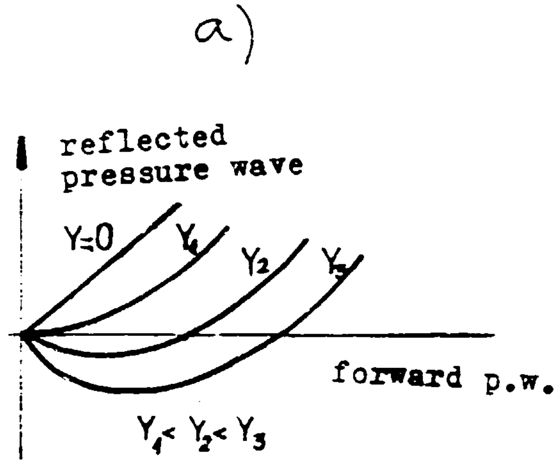


At cam angle "a" very fast evaporation accompanied by condensation "b". However, many systems in operation showing the same figures, did not suffer from cavitation. Only then, if the reflected wave are high, such a process is followed by cavitation.

To avoid the cavitation attack the very fast evaporation has to be avoided.

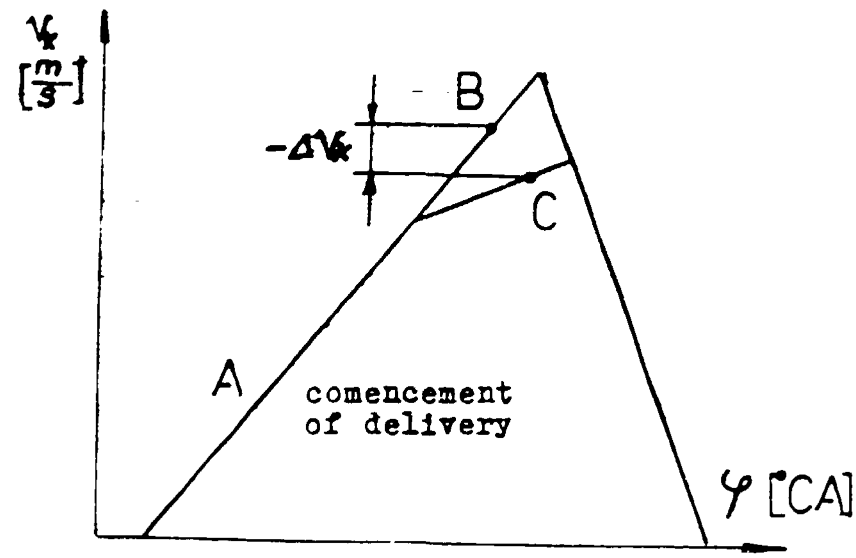
1. The high pressure reflected wave may be reduced by a snubber valve. During fuel delivery, as shown on the next figure, the fuel flowes through all holes. By spilling only the hole "y" is operativ and the reflection is smoothed.

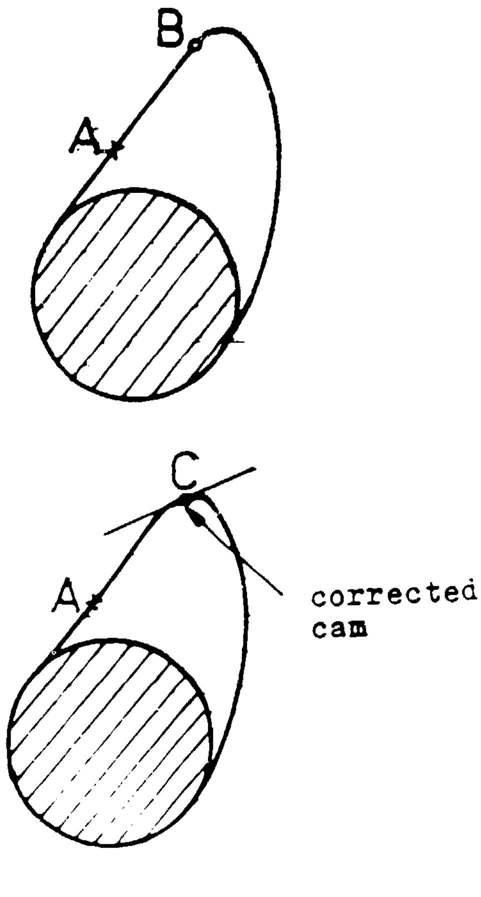




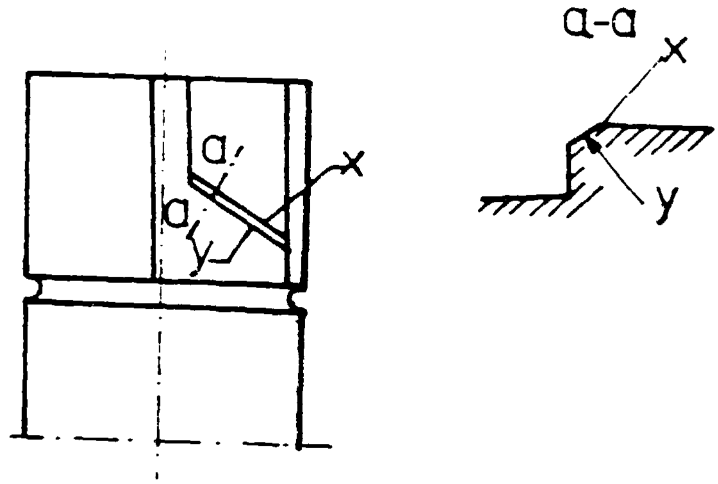
The hole dia. "y" has to be matched according to diagramm a), Φy usually ranges about Φ0.5 ÷ Φ 0,6 mm.

2. Some improvements may be achieved by lower plunger end velocity.



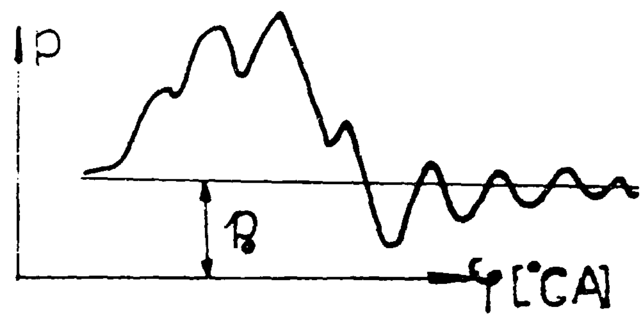


3. To prevent the high cavitation of spill ports and plunger helix edge a prespilling plunger may be used.



During prespilling the plunger shown decreases max. pressure waves.

4. By constant pressure relief valve, the residual pressure can be matched high enough and in this way cavitation may be avoided.



However, the latter method seems less promissing for small engines and for the methanol operation. With small engines the appropriate p_0=const relief valve is not so easy to develop and the high residual pressure produces a high leakage overflow between the injector needle and nozzle.

5. The best way to avoid cavitation may be by appropriate matching. Nozzle holes, relief valve, end of delivery etc. have to be adapted to each other.

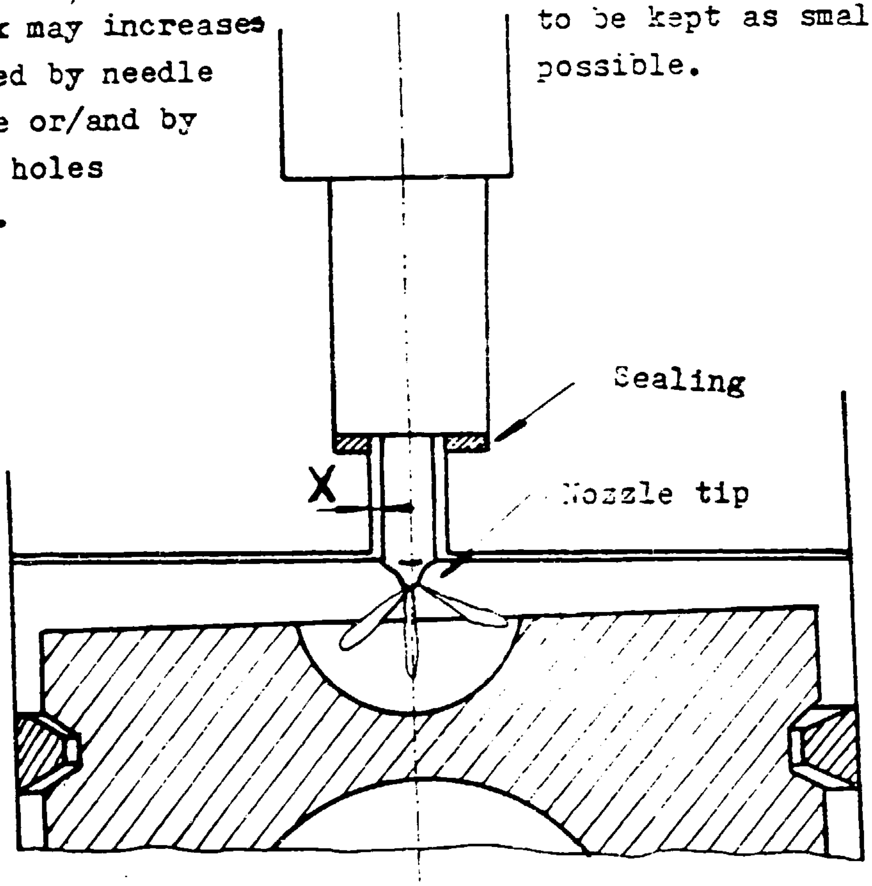
7. Injector - some aspects

Immediate contact with the combustion chamber has only injector.

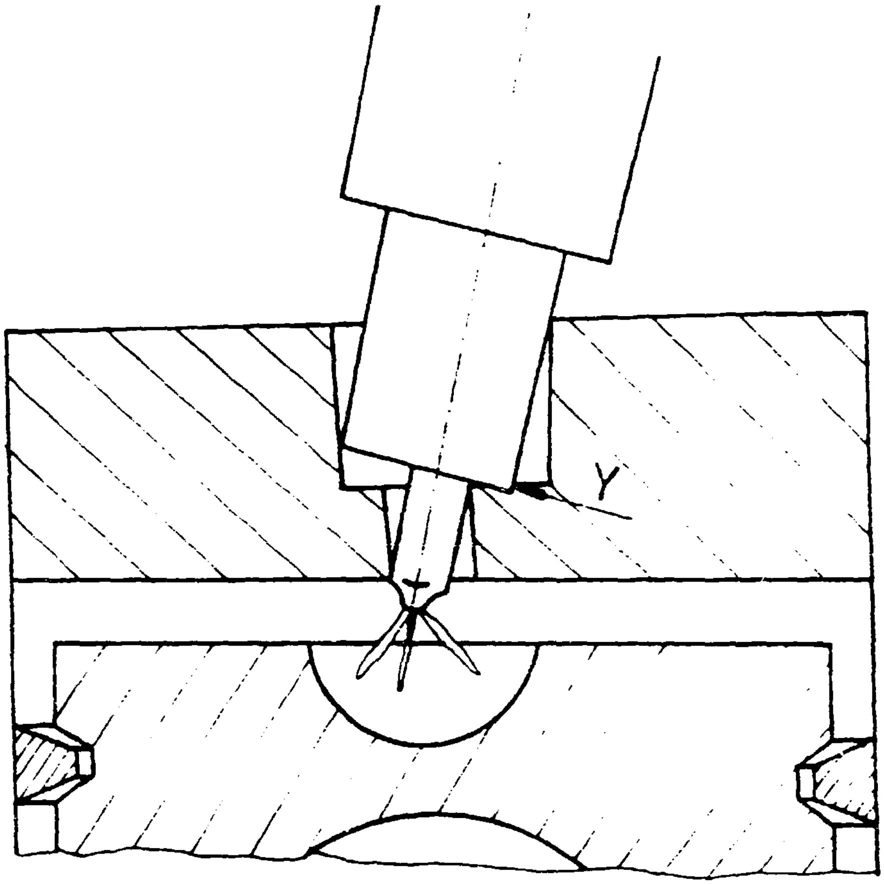
7.1

In cylinder head production, the clearance x may increases followed by needle seizure or/and by nozzle holes coking.

In order to avoid overheating the clearance x has to be kept as small as possible.



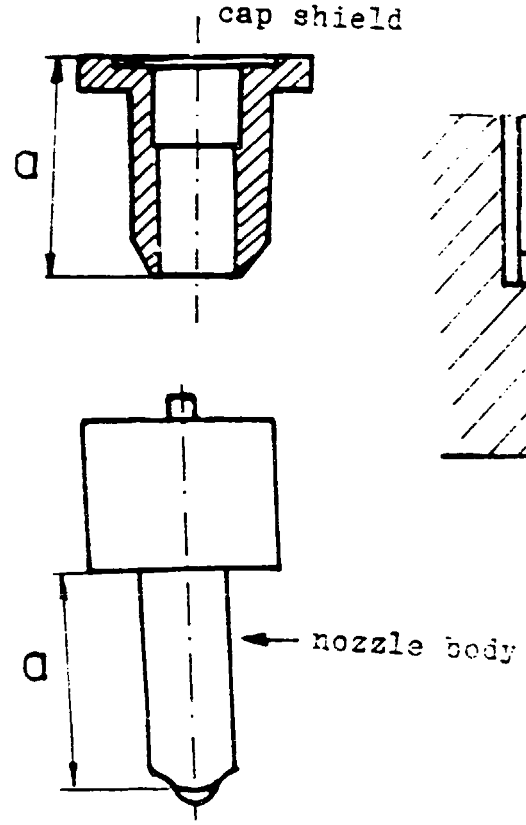
7.2. If the seat y is sloping at w high pressure contact takes place followed by the nozzle body twisting. A result may be irregular needle closing, often accompanied by piston crown melting. (see in the next Fig.)

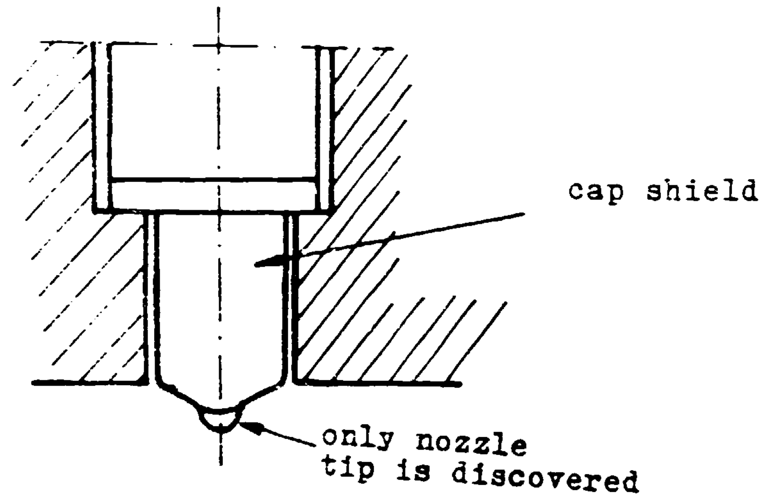


7.5. <u>The nozzle needle</u> fit has to be kept as small as possible. However, using methanol the lubrication in this region is poor, therefore the clearance between the needle outer dia. and the nozzle body hole must be kept constant.

Both needle and nozzle body are carbonized and hardened, but if rest austenit exists, during operation at high temperatures the transformation of austenit into martensit (at $> 150\ ^{o}C$) occurs, followed by the increase of the needle outer dia. The result may be the same as in the case of 2. Therefore the finished heat treatment may be proposed as follows: $5^{\underline{h}}$ at $-60\ ^{o}C$, deep cooling.

7.4. <u>If overheating</u> still takes place a nozzle cap shield may be unavoidable.





Cap shield is made from berylium bronze or from stainless steel.

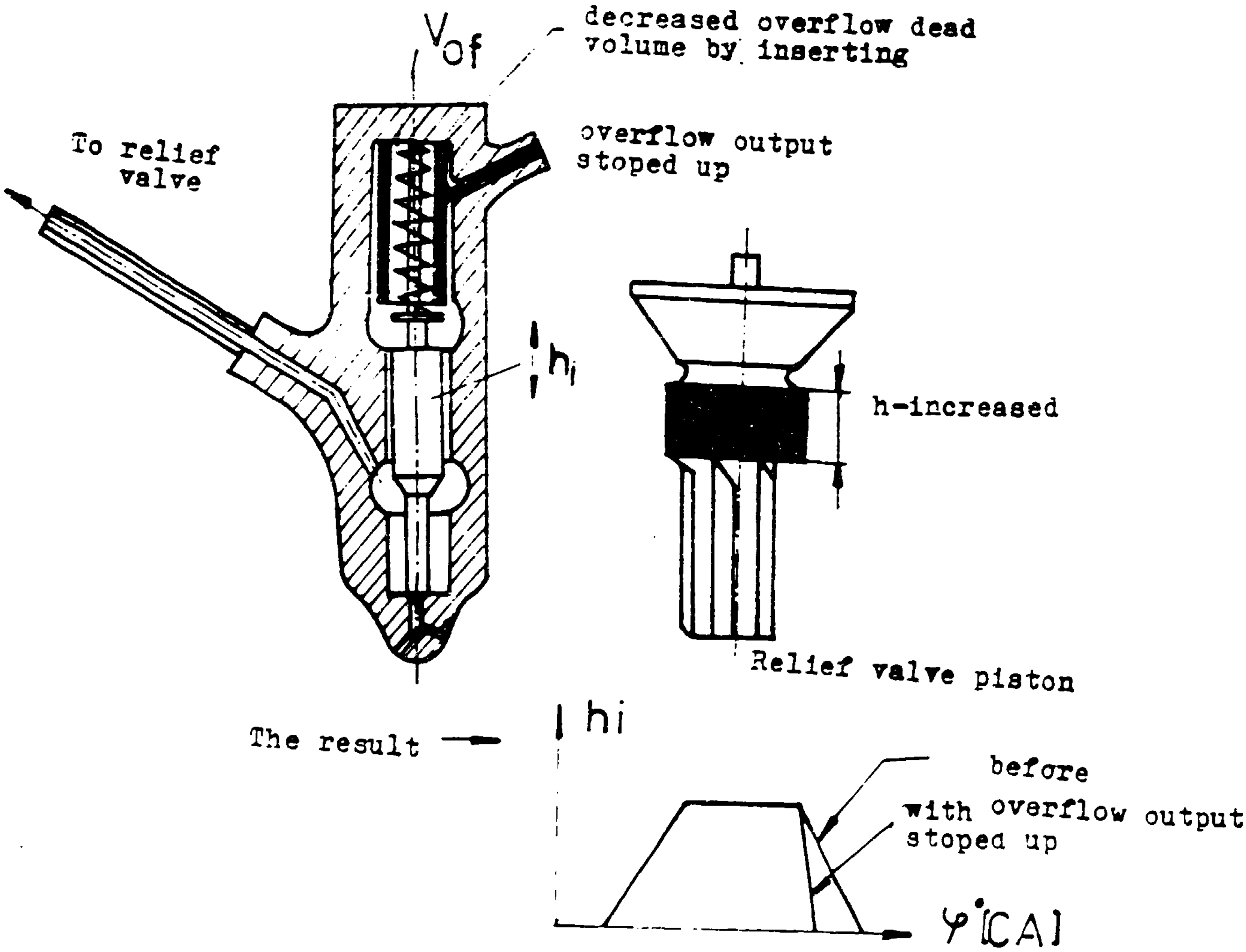
7.5. <u>Injector</u> leakage overflow is needed for cooling and lubrication. With a constant volume retraction system this leakage also controls the residual pressure in the high pressure dead volumes. The forementioned residual pressure correction depends on:

a) operational clearance, needle - nozzle body hole
b) operational density of the fuel used
c) retraction volume and dead volumes
d) operational speed and load

With methanol and with small retraction volume but larger dead volume, the injector leakage overflow, at low speeds and full loads, may amount to 1,5 % of the fuelling, but at low loads may amount to 20 %

Now we must take into account the low boiling temperature of methanol and prevent fuel vapour formation in the overflow. To save the fuel and to prevent boiling, the overflow output has to be connected to a low pressure system, possible after the fine fuel filter.

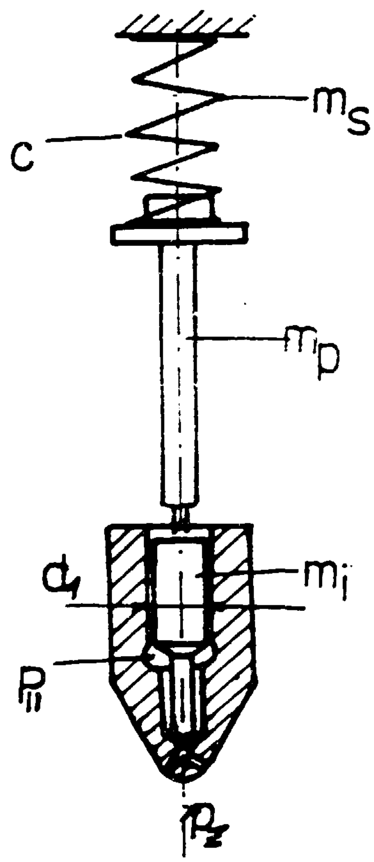
The new injector design, prof. Indra invention, claims faster needle closing by means of overflow communication. The same was reported some years ago by American Bosch but with a residual pressure constant system (p_o = const.). Indra developed his injector for the residual pressure variable system or with the constant retraction volume (V_R=const.), as was predominantly used.



<u>Prerequisites</u>: - small mass of injector moving parts
- the retraction height "h" must be precisely calculated
- dead volume V_{of} must be small

7.6. <u>The mass</u> of the injector moving parts influences:

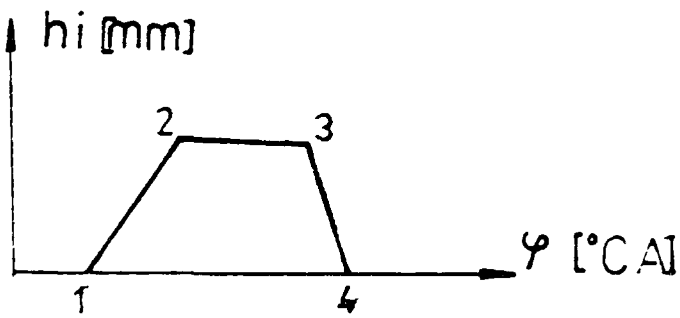
- the life of the needle seat
- needle oscillations
- seats high frequency oscillationes and "corona" phenomena
- needle closing and opening velocities



The moving mass amounts:

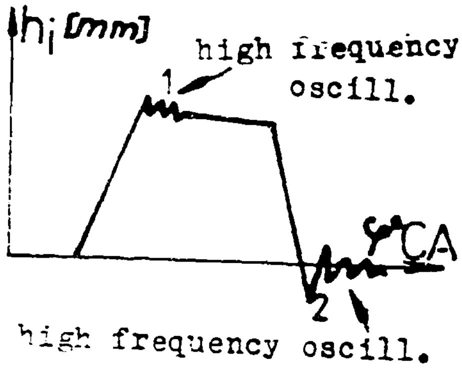
$$m_i + m_p + \frac{1}{2} m_s = m_{ij}$$

$$m_{ij} \ddot{h}_i + F_s + c h_i = \frac{d_1^2 \pi}{4} (p_{II} - p_z)$$



The later eq. is valid 1 - 2 and 3 - 4, thus the mass m_{ij} controls the needle lift.

With high m_{ij} the needle lift appears as shown in the next Fig.:



In this case may come to the "corona" formation of the event 2. The corona is the annular instanteneus fuel droplet mist.

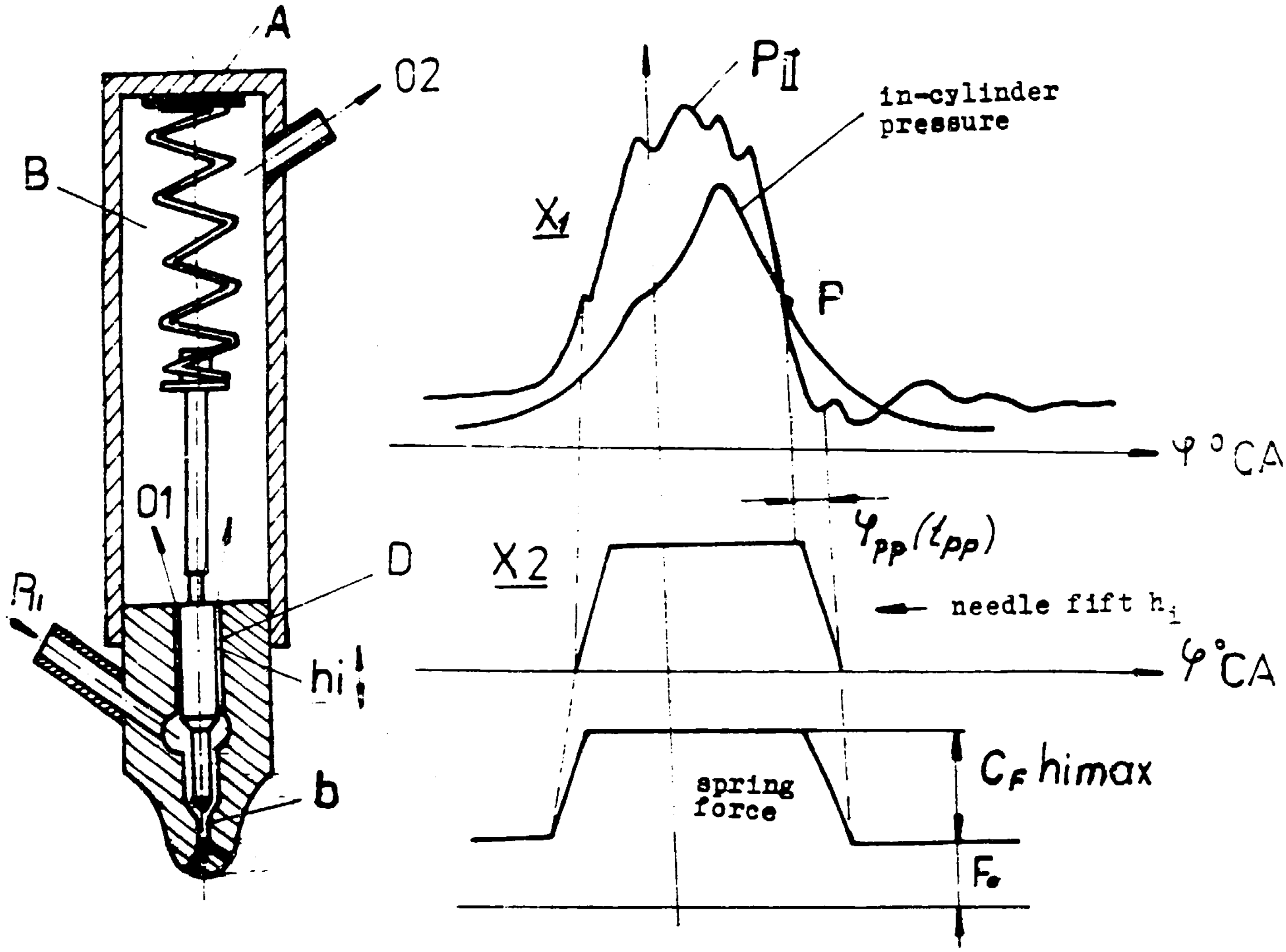
7.7 Opening and closing pressures

The significance of the fuel atomization in diesel engines, as mentioned before, when operating with diesel oil is much more outstanding than that found with methanol. Thus, the opening and closing pressure are also unimportant. As an argument for, see the recent "Volvo" findings.

After 1000 hours testing the new methanol diesel engine, the inspection showed a drastic decrease of the injector opening pressure. Before the long term test, the opening pressure was adjusted at 147 bar, but after testing it ranged from only between 41 and 118 bar. This means that the closing pressure dropped to 35 and 100 bar respectively. With such a pressure decrease and with diesel oil the engine could not operate at all, and exhaust soot emission would be extremly high. Thus, the "Volvo" findings was still more evidence, to show how intense the methanol combustion might be. Spray dispersion was not of a great importance, once, ignition was completed.

However, the methanol spray time events have a large significance, for the ignition and the stable first step of combustion without quenchings, and that in any way under all operational conditions. In the experiments mentioned, the "Volvo" engine had a pilot ignition or stable diesel oil ignition as support.

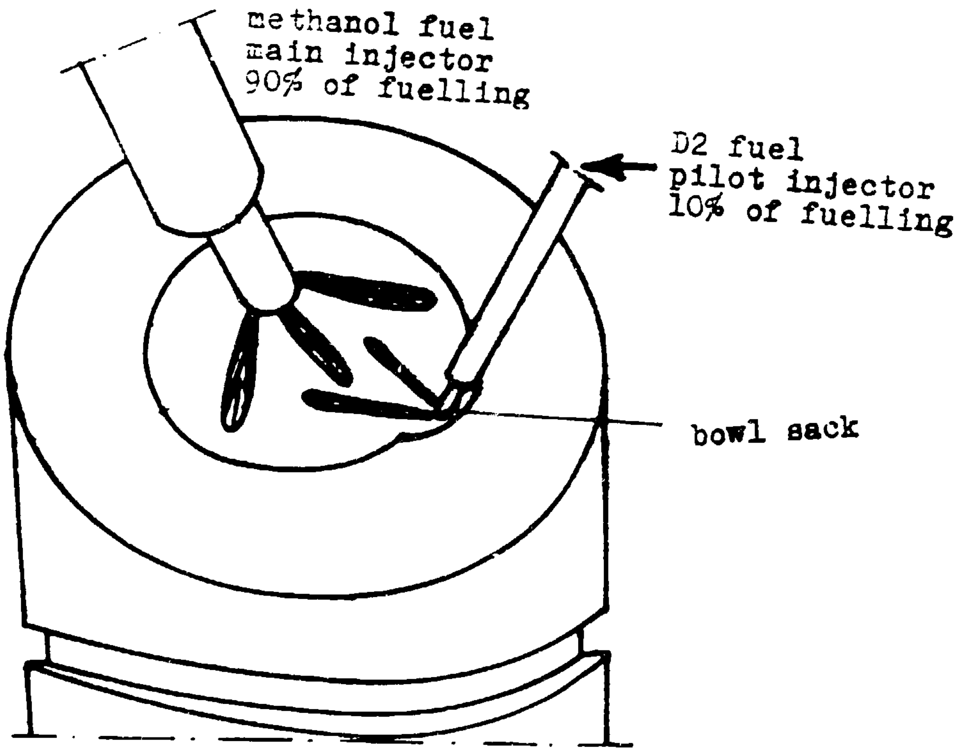
Undoubtedly we have to think over the reasons for such an enormus pressure decrease in the space of 1000 hs. operation. Morever, it should be mentioned that the opening pressure decrease in the methanol operation was also reported elswhere



Because of the fluctuating spring force in the presence of the methanol vapour - mist and contact friction, considerable wear at A may be experienced. The leakage overflow 01 is very often accompanied by evaporation. With decreasing spring force the point P may be reached from where p_{II} is lower than the in-cylinder pressure. If the needle is still open, the hot gases burst into space "b" heating up the needle and nozzle body. The consequence is still more evaporation and an increase of O2 leakage overflow. In this case the pressure increase in the volume B might not be of any help. See Figs. X1 and X2.

It is hoped that these findings initiate the idea, that the whole process must be accompanied by nozzle coking. However, this problem could not be expected with methanol as fuel. Only in the case, where methanol - vegetable oil mixture was used, experiments showed coking of the nozzle holes. On the contrary , with pilot ignition and with small oriffices in the pilot injector, mostly because of the overheating, diesel fuel coking often occured. To avoid it the following may be recommended:

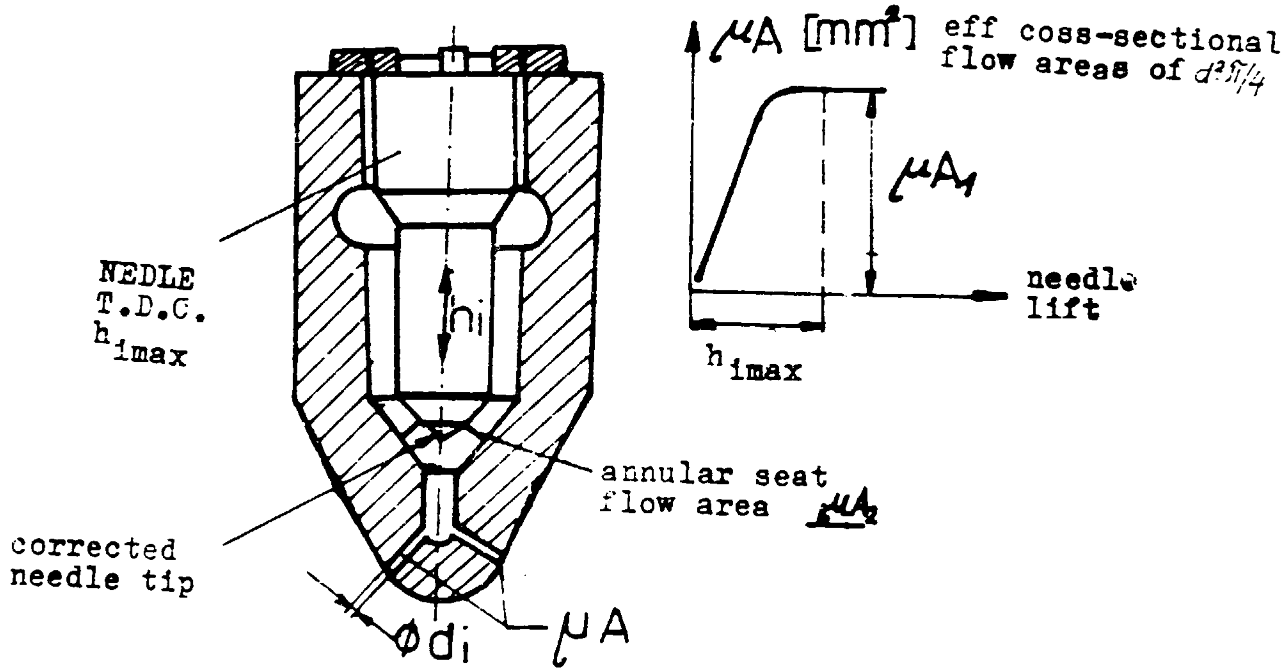
- the pilot injector is as near as possible to the combustion chamber edge or still better, use bowl sack, as shown.



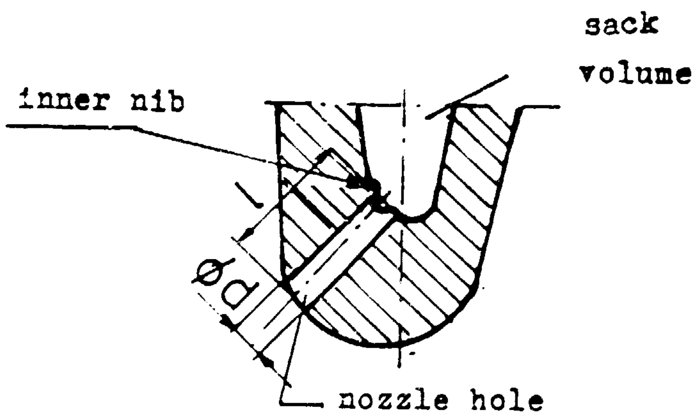
- to avoid large differences between opening and closing pressures
- to keep ratio "y" and to correcte the needle tip (95°)

For pilot injector:

$$\mu A_2 \approx 2 \mu A_1 \qquad \dots (y)$$



- the polished wall of the nozzle holes and without inner nib shown



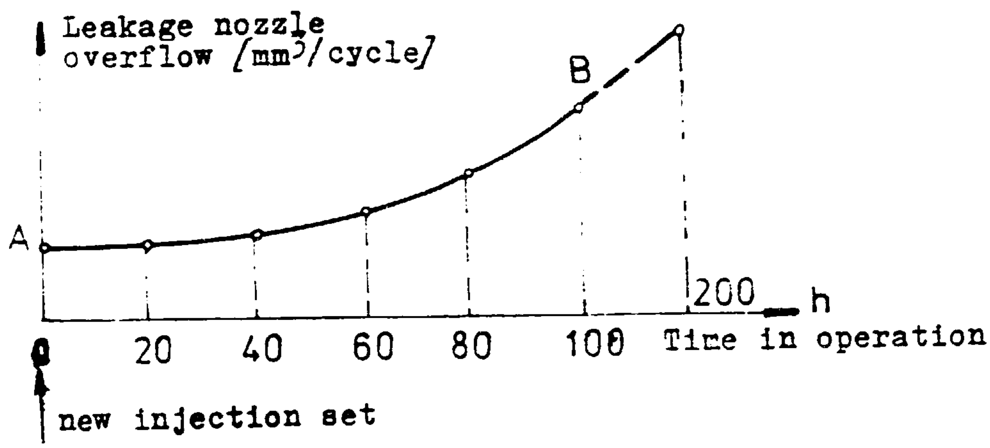
and not to large l/i ratio.

- to avoid nozzle dribbling and the phenomena shown in Figs.X1 and X2.

7.8 The injector wear

Wear and its consequences were discussed in a previous chapter. Now, in addition to the report of Pefley "Alcohol fuel corrosion and wear effects" (Ad-hoc expert group meeting) while dealing with injector problems, needle-nozzle hole wear might not be ommitted.

In the forementioned "Volvo" experiments, a high nozzle wear was noticed. In our own experiments we experienced a higher leakage overflow after ~50 hs. runnings. It is therefore advisable to measure the overflow and to get an idea of wear, as shown.

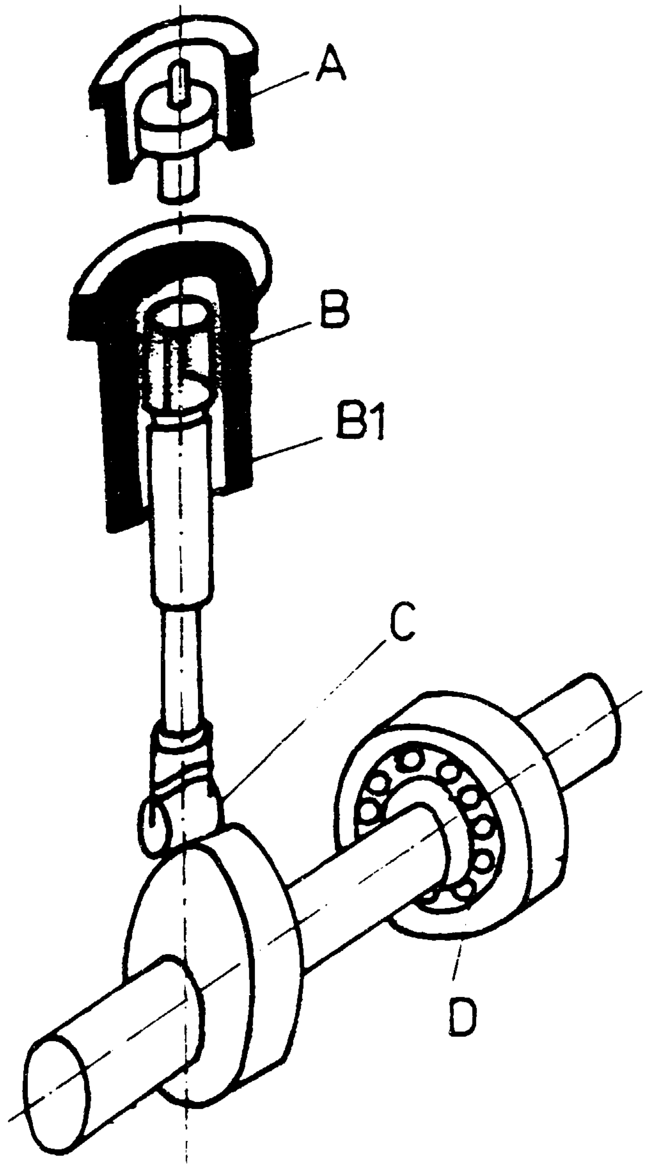


On the injector test bench (explained in part I) the above measurment have to be completed with two further checkings.

The pressure in the known volume should be increased to 10 bar less than the injector opening pressure. Because of the leakage te pressure decreases and at 50 bar the time should be registered (time for pressure decrease from (P_{op} - 10)bar to 50 bar). Such test bench measurments have to be performed after each 100 hs. of operation (see points A and B). The measurments on the test bench used calibration fluid at constant temperature (~ 35 °C). In our experiments at A we obtained about 5,3 minutes and after 50 hs., 4 minutes were registered.

Only a small improvement may be achieved with the pressure increase in the volume B (Fig. shown in the chapter 7.7) that the wear in the nozzle leading zone is not produced with the evaporation alone.

8. Injection pump wear



Cavitation wear was discussed in the chapter 6. Wear was noticed at sliding A, B, B1 and at rolling C, D surfaces. Consequences:

A - the retraction volume decreases, and the fuelling back up increases. With still higher wear after injection (may) takes place.

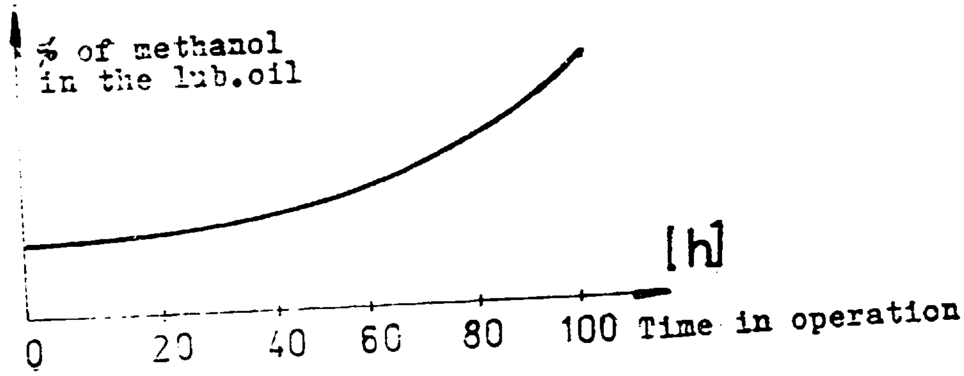
B - unstable low load and low speed injection. The fuelling onwards up increases, (bad torque back up) especially at partial loads being more erratic with lower speeds.

B1 - requires leakless plunger and very often the separate oil lubrication. The leakage at B1 produces a high wear at C and D.

C - the wear because of the methanol - oil mixture is especially high toward the delivery end.

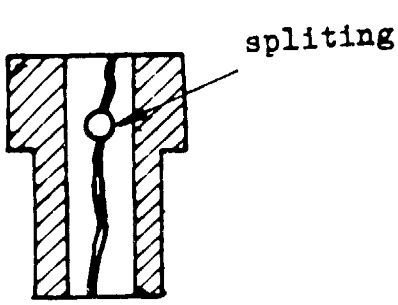
D - because of the methanol - lub oil mixture and wear hight changes of injection may result.

Similary as for injector, the same diagramm has to be performed (see chapter 7.8)



followed by test bench measurments. However, the time measured in relation to the calib. oil, pressure decrease from 300 bar to 100 bar.

In some experiments with methanol the plunger sleeve cracks or splits. The reason is still unknown and may be connected with some penetrations into the material of the sleeve?



9. The diesel engine used for the methanol test bench experiments

A DI "Kirloskar" TV1 - Φ 80 single cylinder engine was selected for the methanol application and had a single speed control (at 1500 rpm) by watering use.

Technical data

No. of cylinders	one
Bore (mm)	Φ 80
Stroke	110 mm
Swept volume	0,55292 dm^3
Rated speed	1500 rpm
Rated power	3,6765 kW
Specific power	6,65 kW/dm^3
Rated mean eff. pressure	0,532 MN/m^2
Rated end speed	~ 1650 rpm
Ratio r/l	0,23752

DI type with toroid chamber and intake port swirl. Three holes injector with Φ 0,26 dia. Vertical design, water cooled and with the starting handle and decompressor lever. Compression ratio about 16,5.

At first it must be recognized that concerning the engine selection IIP made a very good choise, because of the following:

- a single cylinder is practically the best prototype for experiments
- a single cylinder engine is inexpensive at first cost and in operation
- DI type has the best fuel consumption figure
- it is wide spread engine type
- all the engine components, including the injection equipement were produced in India

Engine testing has demonstrated (previous testing):

1. <u>With diesel oil</u>

   Timing 27 °BTDC
   Power output 3,676 kW at 1500 rpm, 21 °C
   D2 oil calorific value 41.907 kJ/kg
   D2 oil density at 20 °C 0,858 kg/dm^3
   Fuel consumption measured by volume:
   153,5 s for 50 cm^3

2. <u>With methanol</u>

   Timing 30 °BTDC
   Power output 3,676 kW at 1500 rpm
   Methanol calorific value 19.665 kJ/kg
   Methanol density at 20 °C 0,794 kg/dm^3
   Fuel consumption measured by volume:
   151 s for 100 cm^3

The above measurements indicate substantial gains in fuel efficiency, when operating with methanol. However, the results may be considered as still being of an essentially preliminary nature and this encourage more detailed measurements.

The calculation of the above measurements shows:

<u>Diesel oil</u>	
Engine test bench fuelling	26,1 mm^3/cycle
Specific consumption rate	274,35 g/kWh
<u>Methanol</u>	
Engine test bench fuelling (eq. glow plug energy out of consideration)	53 mm^3/cycle
Engine test bench fuelling including glow plug	56,5 mm^3/cycle
Specific consumption without glow plug	515 g/kWh
Diesel oil eq. specific consump. without glow plug	241,6 g/kWh
Diesel oil eq. specific consump. including glow plug	256,6 g/kWh

$$\frac{\text{Specific consump. of diesel oil}}{\text{Spesific eq. consump. of meth. (incl. glow plug)}} = \frac{274,35}{256,6} = 1,069$$

Ratio of the calorific values by volume = 2,3
Fuelling ratio x 1,069 = 2,314

The latter comparisons($2,3 \approx 2,34$) show that the measurments seem to be quite correct. Thus, with methanol the fuel consumption figure shows a 6,9 % improvement. However, it should be mentioned that the diesel fuel consumption seems to be unacceptable for DI type, ϕ 80 mm and 1500 rpm. With the avearage value the specific consumption must be about 14 ÷ 15 % lower.

The best approach seems to be as follows: to check the producer data concerning the fuel consumption, 5 engines measuring in the factory and comparing the results. If the consumption rate is still so high, it seems resonable to reduce diesel oil consumption. This may than be the first goal and the concept of the approach has to be changed. The following must also be considered:

- the engine considered is low uprated (p_{emax} = 5,3 bar)
- single speed operation (1500 rpm)
- scot emission by ~ 275 g/kWh of fuel consumption must be also high and this part, can be considered as "Air pollution abatement"
- it is no use to adopt the methanol combustion to a bad engine
- energy savings by conventional means could not be ommitted. The application of methanol indicates first of all the fuel alternation and must be complieted with the reduction of the liquid fuel consumption.

The repeatability of the fuel consumption measurments was elswhere estimated as follows:

	Rep. error
diesel oil - volumetric - stop watch	12 %
diesel oil Sepeler	5 - 6 %
diesel oil by weight (simple method)	2 %
diesel oil by weight aut. meas. device	0,5 - 0,7 %

Thus, it may be recommended, that fuel consumption measurements have to be improved. The estimated percentage of the fuel saved must be ~ by factor 3 higher than that of the repeatibility error.

In the above mentioned common measurements the glow plug temperatures were observed. The method: thin thermocouple wire - silica coated, to reduce the radiation influence.

At rated power:

- with diesel oil, glow plug surface temperature, without glow plug heating 960 °C
- with methanol, glow plug surface temperature with glow plug heating (150 - 210 W) 900 °C
- with methanol misfire checking showed (see Fig.),at 45 % load the start of misfire noticed by hearing,at 33 % load the remarable misfire observed,at 27 % heavy misfire.

It seems that the Ricardo findings are again demonstrated. In our case the misfiring probably started at 55 % load (we did not measure THC exhaust or/and in-cylinder pressure traces).

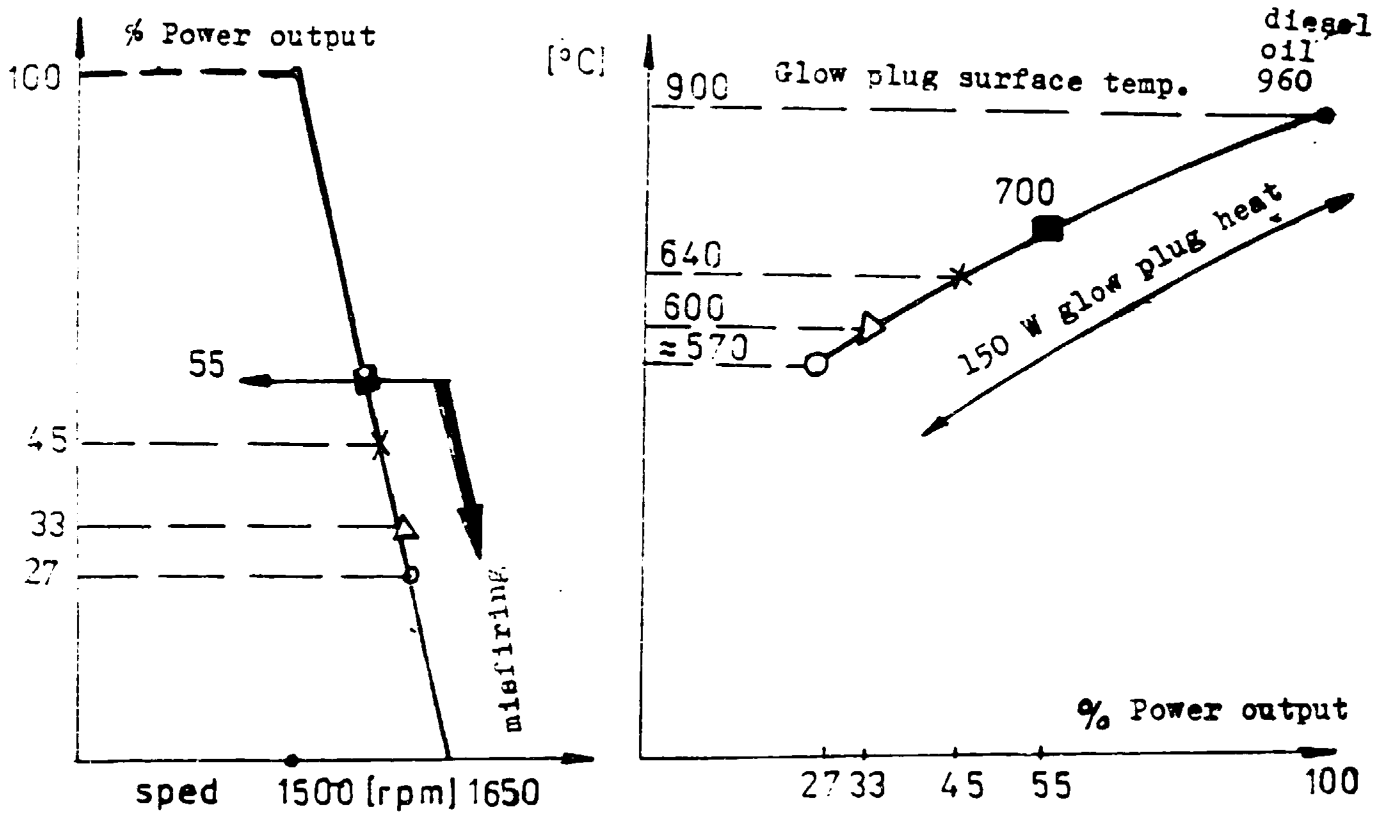
Here is should be mentioned that our calculation showed with ceramic isolation that the combustion chamber wall temperature (without glow plug heating) should be 700 °C with a 35 air-fuel ratio. This means that up to 50 % load is not need for glow plug heating at all.

The next whole figure is much clearer if we are going to consider the previous IFP measurments and that:

- the comparison between the in-cylinder pressure traces for the both fuel

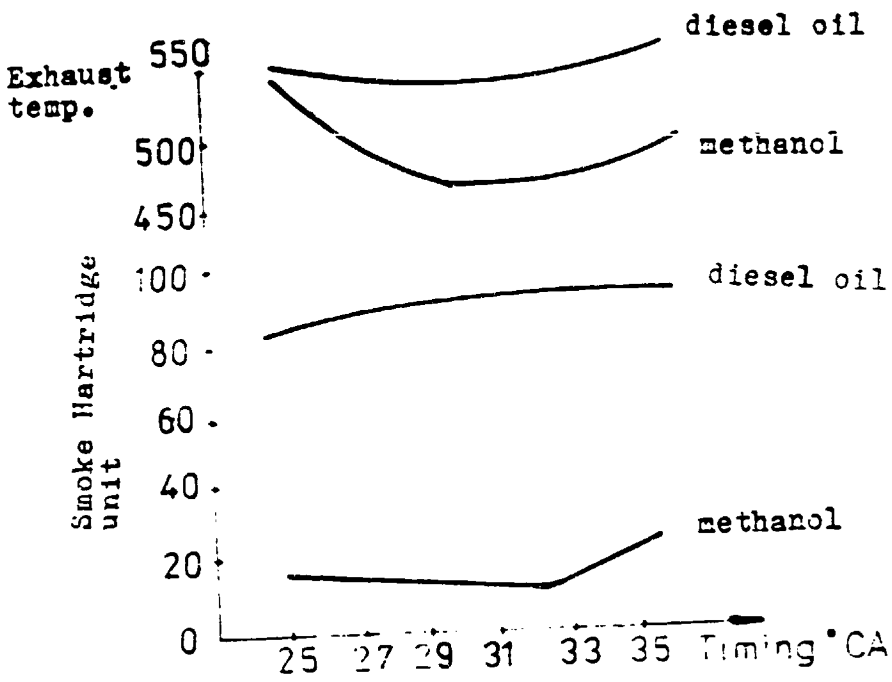
- the comparison between the exhaust temperatures
- and the comparison pertinent soot emission

  - marked points are Ricardo findings

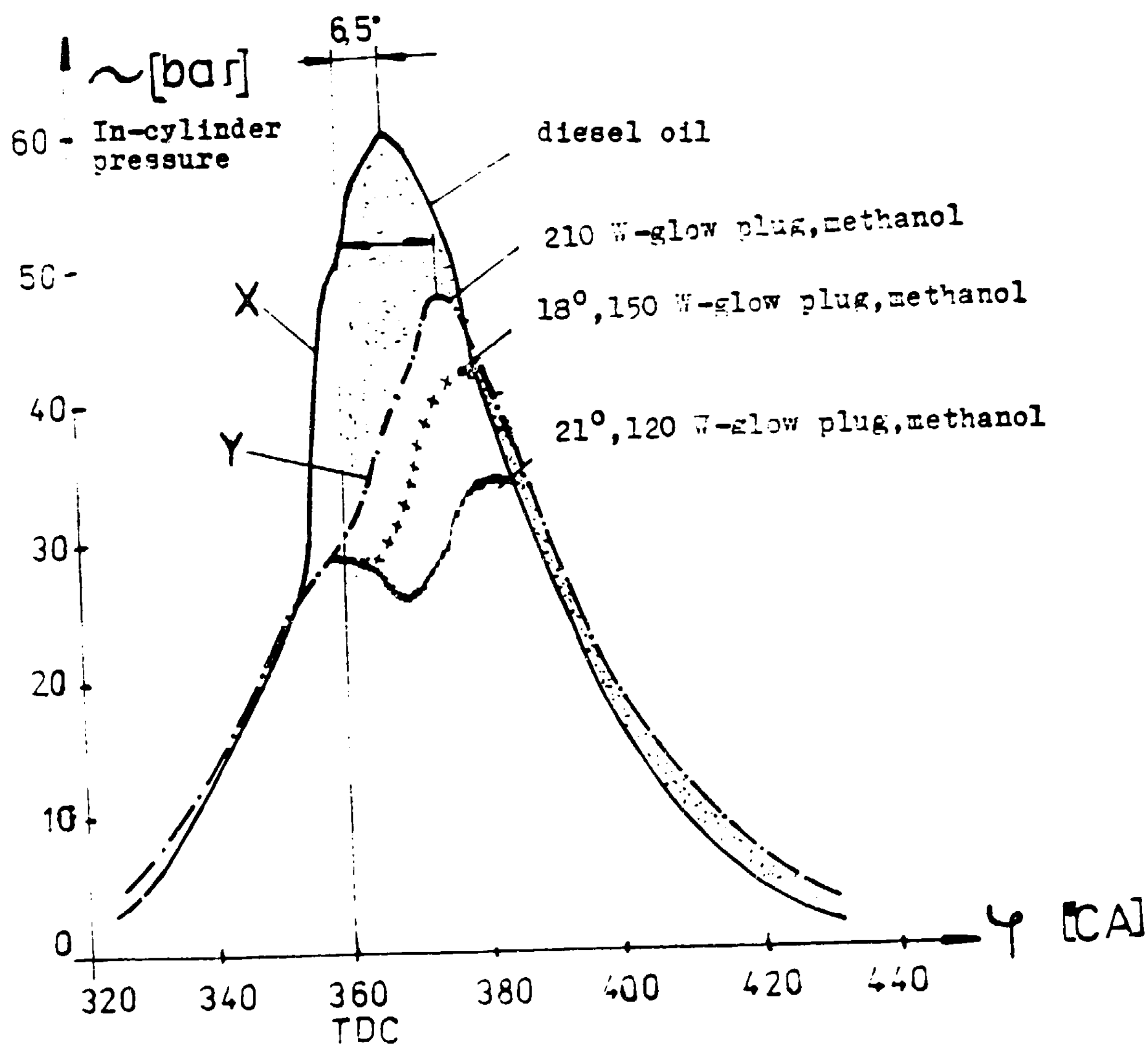


The appropriate diagramms are shown on the following page. Analysing the diagrams we may conclude:

- extremly high soot emission in diesel fuel operation and the soot emission insensibility toward timing. Bearing in mind the low rated mean effective pressure and such a large exhaust gas opacity, it seems that <u>after injection</u> takes place. Here, only the decrease of the dead volumes (as discussed before) or/and the increase of the retracted volume, may help. After injection in any case <u>must be avoided</u>.



With methanol energy saving at rated power 6,2 % estimated.



The Figs. are related to rated power

- very high exhaust temperature (in diesel oil operation) concerning single cylinder engine, low speed and low mean effective pressure. This finding demonstrates incorrect matching of fuel injection system. Better matching has to be done, but at first, by calculation (as discussed before)

- low peak in-cylinder pressure in diesel oil operation. Prolonged injection period and too high residual pressure may produce such a figure. Also $\varepsilon = 16{,}5$ as compression ratio is relatevely low for NA type.

- It seems that the retraction volume is to be too small related to the dead volumes. With the relief valve filler (se MICO INFORMATION VDT - 04CCC, 12, 1978.) and with the tube dia. 1,5 mm the figure must be better.

- for the nozzle and intake port used, 210 W has to be supply to the glow plug. This conclusion may be seen in in-cylinder diagrams. For the other relations between the nozzle holes injection intensity and air swirl, the glow plug energy demand may be lower.

If the one relatevely large part of the spray impacts directly the glow plug, follows, the high surface temperature decrease. By low loads the spray penetration is shorterand methanol vapour production may be very low with also low rate of evaporation.

- peak in-cylinder pressure in methanol operation is small and shifted more from the TDC position. It seems that the peak pressure of effective compression is significantly lower when compared with that of diesel fuel. The main reason may be a greater heat of evaporation in methanol operation; the reduction seems to be about 10 bar. The same amount can be observed as compared with the peak in-cylinder pressure by combustion.

The above discussion suggests the need for a higher compression ratio in methanol operation. It may be increased over $\varepsilon = 18$ without any problems for hand starting. In any way the injection period must be shortened and timing retarded.

- comparing the diagrams X and Y (page 44) it seems that with diesel oil the power output must be greater than with that of methanol. However, the measured outputs were the same. In order to check the p(α) diagrams the next procedure may be recommended. The whole calculation may be easily made using the existing computer.

The output rated torque was the same:

$$M_{e\,Pemax\,\text{Diesel oil}} \approx Me_{Pemax\,\text{Methanol}} \qquad (A)$$

or:

$$\int_{0\ \text{Diesel oil}}^{4\pi} \mathfrak{M}_e\, d\alpha = \int_{0\ \text{methanol}}^{4\pi} \mathfrak{M}_e\, d\alpha$$

The instanteneous torque $\mathfrak{M}_e$ maybe decomposed as follows:

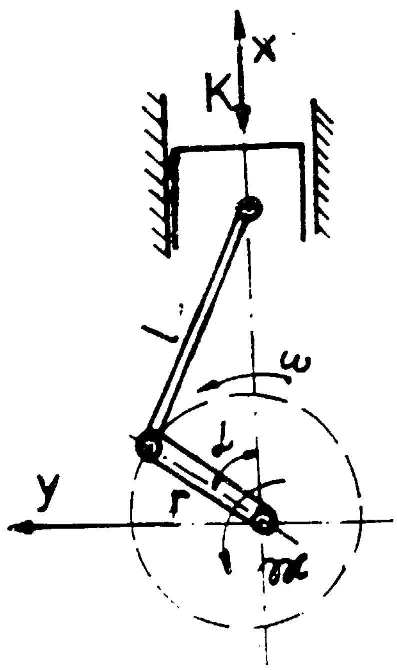
$$\mathfrak{M}_e = \mathfrak{M}' + \mathfrak{M}'' + \mathfrak{M}''' + \mathfrak{M}''''$$

instanteneous eff. torque = inst.torque produced by gas forces + inst.torque potential energie + inst.torque kinetic energie + inst.torq friction

(the last three terms: the same for the both fuel)

It means also:

$$\mathfrak{M}'_{meth.} = \mathfrak{M}'_{diesel\ oil}$$



Now is valid:

$$\mathfrak{M}' \cdot \omega = K \cdot \dot{x} = \frac{\pi D^2}{4} \cdot \frac{dx}{dt} \cdot p$$

or:

$$\mathfrak{M}' = \frac{p \cdot \dot{x}}{\omega} \cdot \frac{\pi D^2}{4} \ [Nm]$$

For our case

$$\mathfrak{M}' = const\ p \cdot \dot{x}$$

For the porpose of comparison if eq. (A) was satisfied.also must be valid:

$$\frac{\int_0^{4\pi} \dot{m}'_{methanol}\, d\alpha}{\int_0^{4\pi} \dot{m}'_{diesel\ oil}\, d\alpha} = \frac{\int_0^{4\pi} p(\alpha)\cdot \dot{x}(\alpha)_{meth.}}{\int_0^{4\pi} p(\alpha)\cdot \dot{x}(\alpha)_{diesel\ oil}} = 1$$

- rate of pressure rise [bar/ °CA] with methanol is by factor 3 ÷ 3,5 lower than with diesel oil. For diesel oil the value of 7 [bar/ °CA] is unexpectedly high, because of the swirl and warm operation. For methanol 2 [bar/ °CA] is relatively low and may increase till 5 [bar/ °CA] .

10. Recommendations for attending developement

The following recommentations are given:

1. All experimental work has to be complieted with standard measurments (pressure in the exhaust duct, pressure in the outlet manifold etc.). Air - fuel ratio has to be considered as standard and with methanol also THC exhaust.

2. Read out device on the engine test bench has to be with finer division.

3. Fuel consumption mesurements only by weight.

4. The all instruments used, as well as test bench, have to be methodically calibrated.

5. Is no use to make the matchings with diesel engine without needle lift observations.

6. Hand advancer may be very useful.

7. Protection must be improved (no rotating shaft streching in the working area)

8. Our goal must be 7,5 ÷ 13 % better equivalent fuel consumption figure as was by M.A.N. with engines L9204FM and D2566FMUH.

9. We must take into account that the air swirl means heat losses. Thus the air swirl intensity may be reduced. Now if so, we could not more operate the engine with diesel fuel.

10. The compression ratio must be increased nearly by factor 1,12. This can perhaps be done using a cylinder head gasket or with the reduction of combustion chamber.

11. Injection period has to be shortened and timing retarded. This can be done using a larger plunger dia.

12. Nozzle holes have to be increased and maybe also one (or two) hole more added. It depends of the whole spray-air swirl history but also of the glow plug position.

It is to be advised that much more than combustion chamber is the "evaporisation chamber" of interest.

13. By calculation followed with test rig experiments as well as test bench matchings the fuel system can only be adoped to the engine demands.

14. The some engine drawings as well as for the injection equipement are unavoidable. Thus, contact the engine and fuel equipement producer as soon as possible.

15. It is of interest to make a programme how to use the spray test bench data. Cnly complieted with other injection parameters and swirl characthistcs the spray data can be useful.

Here there is a need to divide the injection law into four parts and to define:

- spray distance or tip distance
- spray deviation
- d_{max} - max droplet dia.
- d_{32} - Sauter droplet dia.
- spray angle
- injection law per hole and the sum for comparison.
- influence of the swirl intensity and the air temperature upon the atomisation
- of interest are:
  1. First part in the time of ignition delay

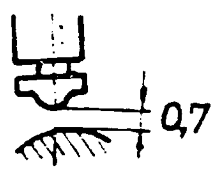
2. Sec. part in the time to max amplitude of injection law
3. Third part in the time of 3/4 of injection law
4. Forth part 1/4 of injection law (especially d_{max} and d_{32}).

All data collected are only useful for calculations and comparison. In the comparisons we can follow the influence of the injection parameters changed (dia. of nozzle holes, dia. of plunger, presseting follower-cam etc.).

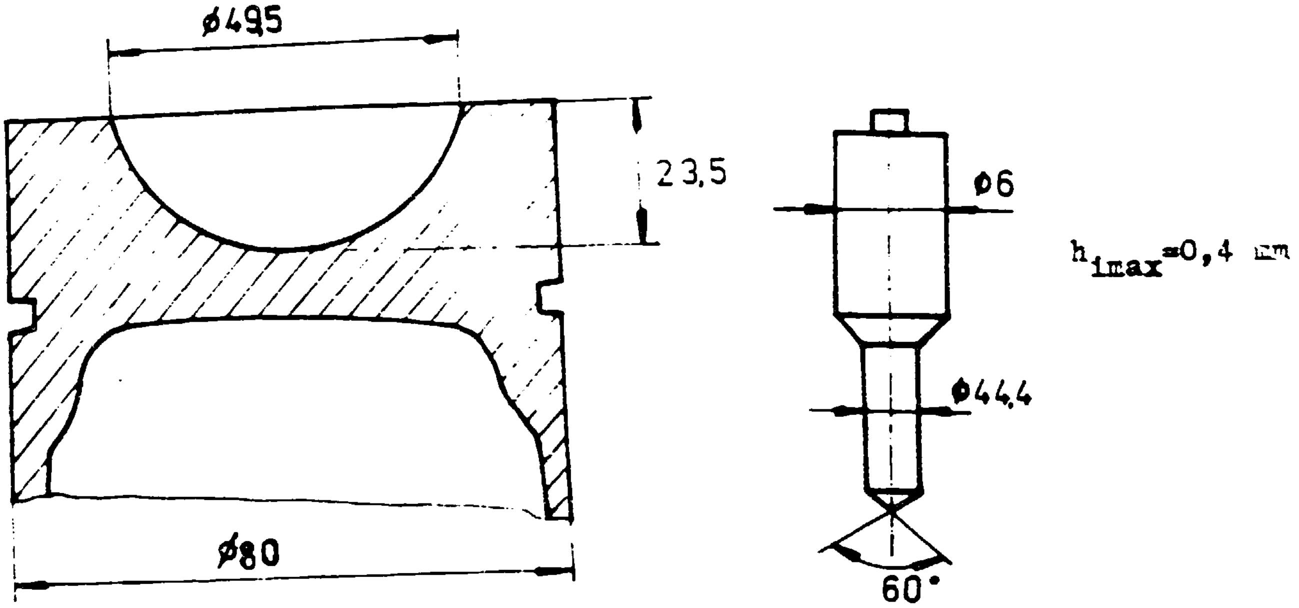
Having more data we would be able to predict the comb. seduencies and to adopt more accurate the Vibe function.

11. Data collected for the injection system

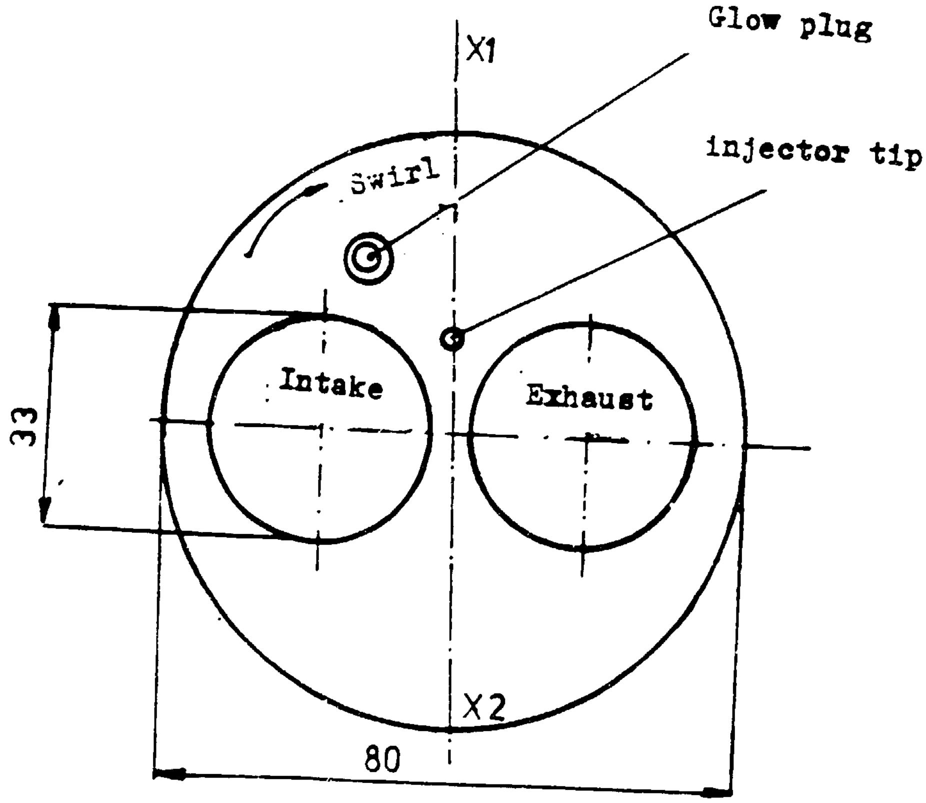
It was arranged that the data collected serves only to show the calculating procedure.

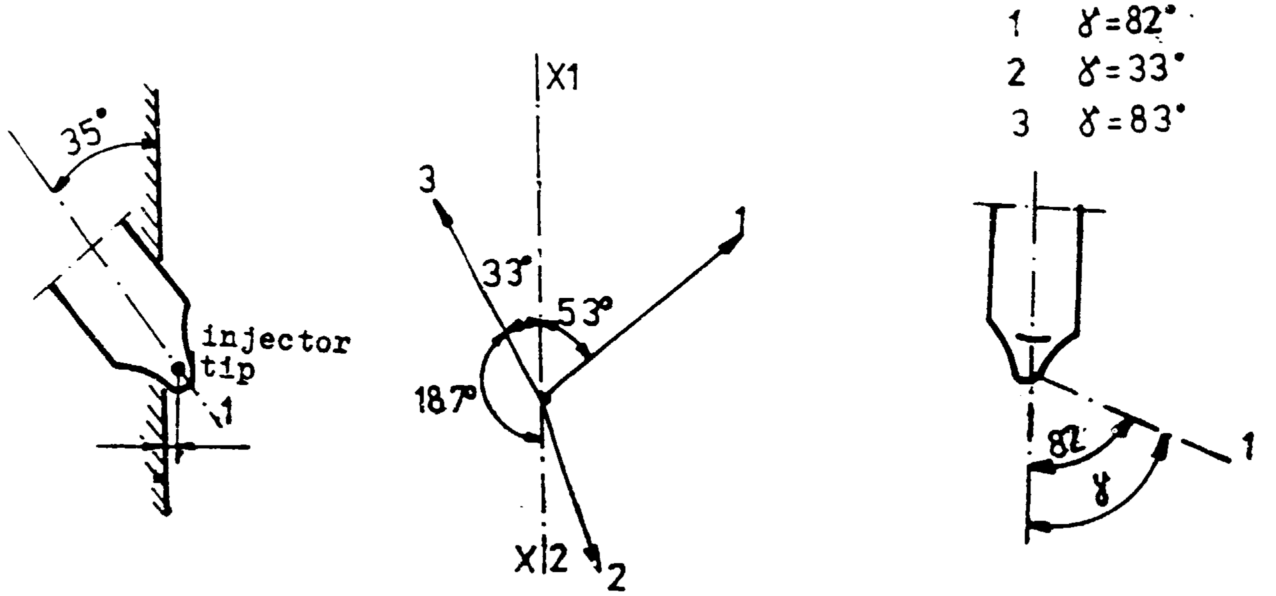


1. Plunger presetting 0,7 mm
2. Plunger dia. 6,988 mm
   as measured 7
3. Dead volume totl.
4. Retraction volume 30 mm^3 (ϕ 6)
5. Mass of retraction piston 3,05 g.
6. Dead volume of the relief valve holder 765 mm^3
7. High pressure tube ϕ 2 x 600 mm
8. Dead volume of injector
9. Rigidity of spring 110 kp/cm
10. Needle opening pressure 170 bar - methanol and 184 bar - - diesel oil
11. Angle of needle 61^o
12. Max lift of the needle 0,4 mm
13. Nozzle hole dia. 3 x ϕ 0,27 mm
14. Piston of the engine (showed on the next page)
15. Needle (showed on the next page)
16. Fuelling at 750 rpm, rated power
    diesel oil 26,1 ÷ 27 mm^3/stroke
    methanol 53 mm^3/stroke
    methanol with glow plug 60 mm^3/stroke
    Overflow leakage 1 ÷ 1,2 % of fuelling,
    for methanol (0,53 ÷ 0,6 mm^3/stroke)

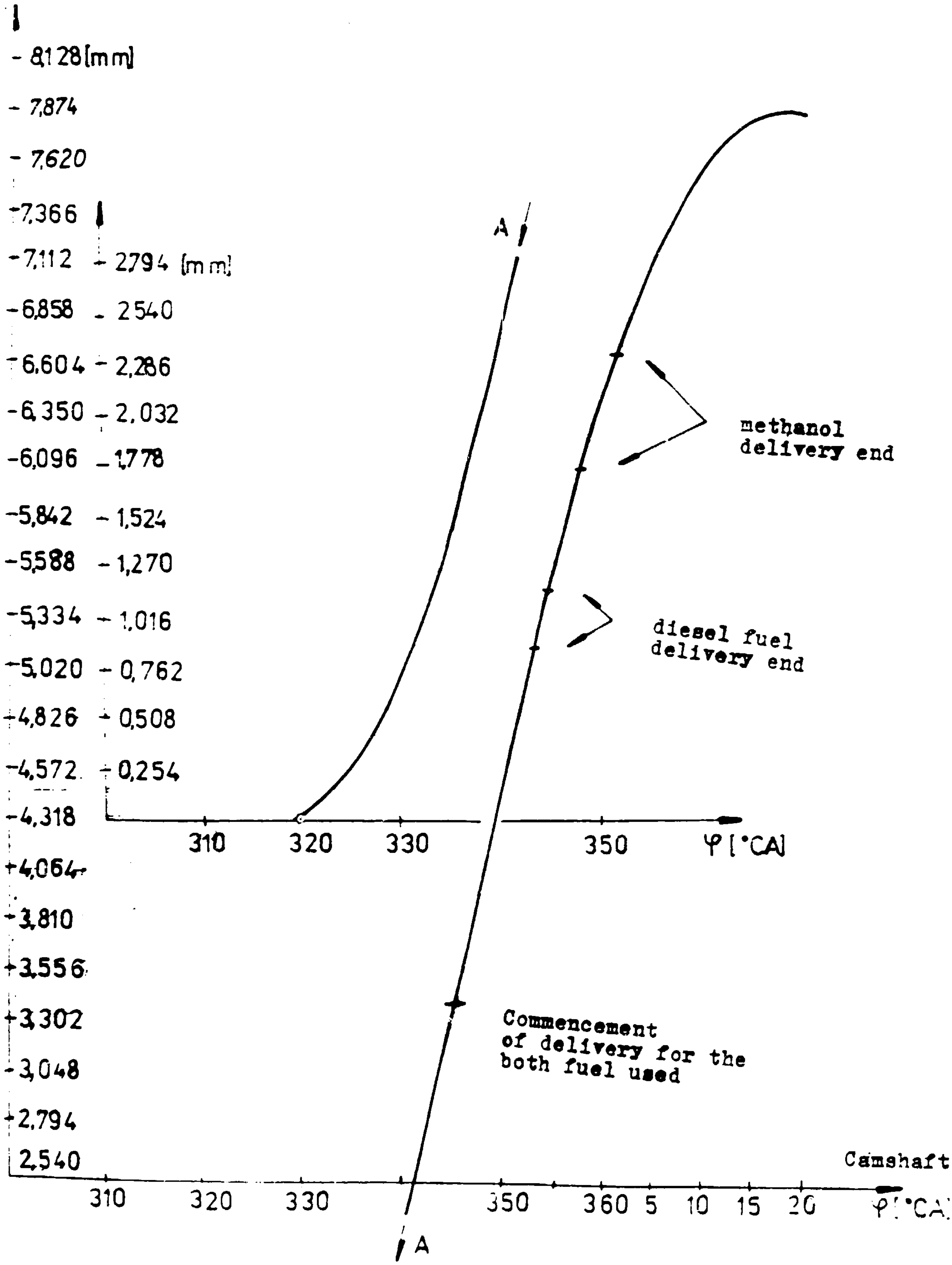


17. Nozzle holes arrangement and piston bowl




35°
injector tip
1
X1
3
1
33°
53°
187°
X
2
2
1 γ=82°
2 γ=33°
3 γ=83°
82
γ
1

18. Plunger lift diagram



12. Some additional suggestions

(A) - In the engine development we need also test rig experimentes performed for:

- the porpose of comparison
- to follow the intensity of parameters changed
- to collect the bank of data

Thus, we need:

- swirl intensity measurement
- air flow in dependences of the valve lift
- fuel flow rates measurements: nozzle holes, injector holder, relief valve, intake and spill ports of the sleeve etc.
- test bench for the pump equiped with all instrumentation required.
- carburator test bench (as explained)

(B) In changed the pump plunger we are obliged to correct the relief valve or better, every change in the fuel injection system requires the change of the retraction volume. We are also obliged to decrease the systems dead volumes as much as possible. Every 100 mm^3 decreased may help in matching.

(C) We must take care that the approach for the on - line exhaust gas analysis with diesel engines must be defferent than that of the gasoline engine. It often occured that in diesel operation our instrumentes (after mesurements) were out of order or recalibration showed the sensitivity decrease.

The main reason for the above is the very fine despersed soot particulates which "block" our sensors.

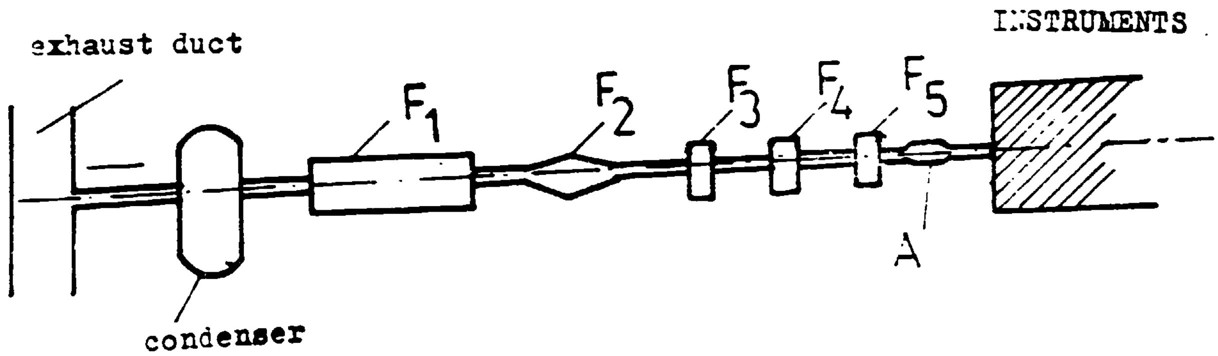
Is to be recommended:

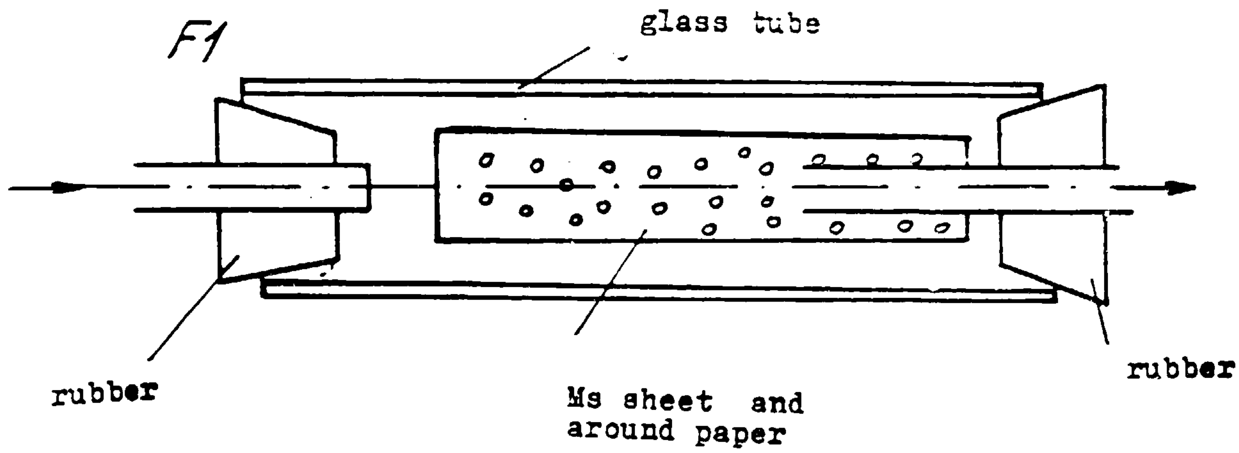
- in developement make the THC measurement divided from other gas analysis. In developement we know that the THC emissions are largest at low loads, but we have not soot emission in this region. Besides that we must heat up the sample line for THC measurements.

  For THC measurements we need O_2 oversensitivity correction. This problem does not arive for gasoline engines. At low loads we may have by factor 4 ÷ 5 more air than is needed for combustion, we changed the FID flame character and so the ionisation effect changes drastically the C-atom proporti-onallity.

For THC we need (being operating in sootless zone) only two till three glass filters No2 (as explained). But the filters have to be heated. The temp. is about 180 ÷ 210 °C.

- For CO, O_2, NO, CO_2 measurements at full loads the next system may be suggested:





F_3 - glass filter No 3

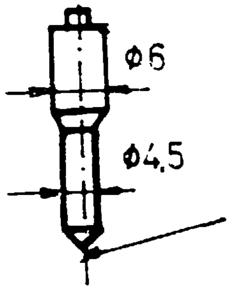
F_4, F_5 - glass filter No 2

A - some wool (no glass) for observation

F_2 - wool filter (as explained)

(D) In any way we must try decrease the ratio:

6/4,5 → and realize 6/3,5

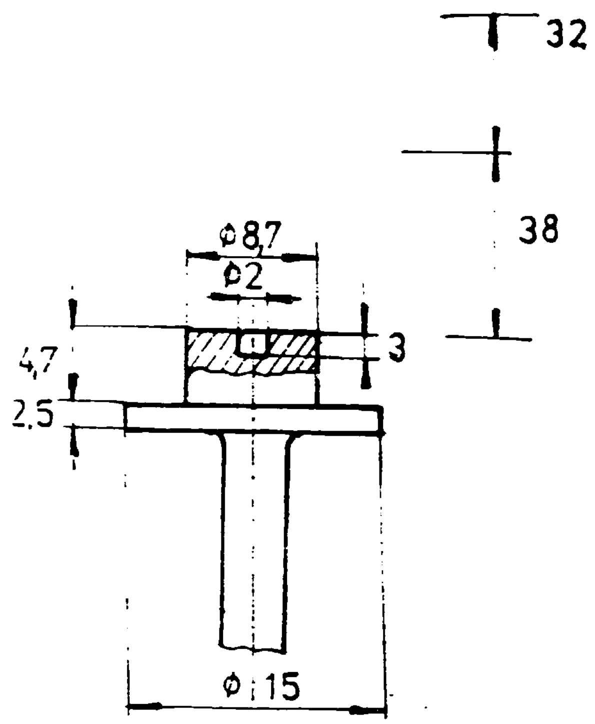


correct the tip according the min. cross-sect. flow area

13. Suggestion concerning instrumentation

- test bench for the injector flow rates measurements	1
- needle lift inductive tranducer	3
- bridge for inductive tranducer	2
- stroboskop with camera	1
- Oscillosscope storage	2
- THC instrument for diesel engines (see EPA prescrip.)	1
- thermocouples kit	2
- quartz tranducer calib. device	1
- precise pressure gauge	2
- adapter for low in-cylinder pressure measurements	2
- fuel consumpt. measurements device by weight	1
- flowmeters	3
- the book from Fristrom and Westenberg	1
- new compact pump	2
- hand advancer	1
- valve test rig	1
- swirl test rig	1
- piston temp. measurement device	1

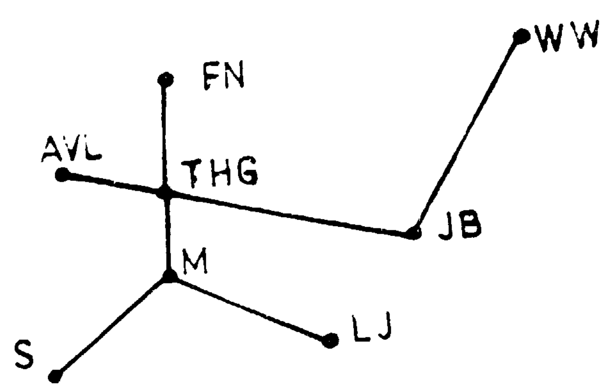
- for the needle lift inductive tranducer



- the nozzle taken for flow measurements
  3 x 0,27
  32 MICO DLLA 110 S639

14. Training recommendation

The three engeeners who have been working with me have to be given further training in other labs.



S, LJ, M - cities in Yugoslavia
AVL, THG - Graz, Austria
FM - Hallein, Austria
JB - Brno, Chehoslowakia
WW - Warshawa, Poland

1. Engineer S, M, AVL, JB, FM - 8 weeks
   Fuel injection calculations, bank of data, evaluations of exp. data, apparatus for fuel injection testings, matching.

2. Engineer S, M, AVL - 8 weeks
   Calculation of combustion processes, method of characteristics, heat transfer, evalutions of in-cylinder pressure diagrams, experimental approach, matchings

3. Engineer M, LJ, THG, WW - 5 weeks
   Combustion bomb set up, registration of events, control of processes, evaluations of the data collected.

In the case of agreement (IIP, UNIDO - UNDP) the proposers need 2 mounts in order to:
- make the detailed programmes
- arrange the visites in the institutes mentioned

15. Conclusion

IIP is a well organized institute with high educated men who are cooperative and have a solid knowledge of the fundamentals. The young engineers are quickly able to master a sophistaced approach to engine developement. There seems to be no any reason that the methanol project may be poor elaborated. Morever, the project can be supported only if the appropriate experts will be send in IPP.

IPP has many scientists and higly educated young engineers. However, engine developement is a new subject for IIP and therefore, the project requirests have to be supported. The need for support is pertinent to:

- developement approach
- measurement techniques
- collection of some exp. data performed
- engine design

However, support is also needed in relation to the programming of engine processes and to the collection of the data used in the engine developement. Here also only again proved practical programming.

Finally, it should be mentioned, that because of the language IIP needs also a good selection of the backgrounds of the German literature (MTZ, ATZ, Technische Berichte).

The profile for the following expert has to be an engineer with the good developement and lab experience, who also revied constantly the coresponding European technical journals and reports.

Appendix A

1. Sauter means diameter (without swirl impact)

$$d_{32} = 124{,}77\,\frac{d_b}{v_b}\left(\frac{\sigma_g}{\rho_v}\right)^{0{,}25}\left(1+\frac{3{,}31\,\mu_g}{\sqrt{\sigma_g\,\rho_g\,d_b}}\right)\cdot 10^{6}\ [\mu m]$$

Notation:

$d_b = 0{,}4\cdot 10^{-3}\ [m]$ -nozzle hole dia.

$$v_b = \sqrt{\frac{2(p_{II}-p_z)}{\rho_g}} = \sqrt{\frac{2(300-40)\cdot 10^5}{840}} = 248{,}8\ \left[\frac{m}{s}\right]$$

instantaneous fuel out-flow velocity

$p_{II} = 300\cdot 10^5\ [N/m^2]$ - instantaneous pressure before the nozzle hole

$p_z = 40\cdot 10^5\ [N/m^2]$ - instantaneous in-cylinder pressure

$\rho_g = 840\ [kg/m^3]$ - D2 fuel density

$\sigma_g = 0{,}0285\ [N/m]$ - D2 fuel surface tension

$\mu_g = 0{,}001742\ [Ns/m^2]$ - viscosity

$$\rho_v = \frac{p_z}{RT_z} = \frac{40\cdot 10^5}{287\cdot 500} =$$

$= 27{,}87\ [kg/m^3]$ - instantaneous in-cylinder air density

Calculation:

$$d_{32} = 124{,}77\cdot\frac{0{,}4\cdot 10^{-3}}{248{,}8}\left(\frac{0{,}0285}{27{,}87}\right)^{0{,}25}\left(1+\frac{3{,}31\cdot 0{,}001742}{\sqrt{0{,}0285\cdot 840\cdot 0{,}4\cdot 10^{-3}}}\right)\cdot 10^{6} =$$

$= 35{,}938\ \mu m$

2. **Spray penetration (without swirl impact)**

$$l_y = 2 d_b^{0,46}\, v_b^{0,54} \left(\frac{\varrho_g}{\varrho_v}\right)^{0,23} \Delta t^{0,54} \quad [mm]$$

$$\Delta t = \frac{\Delta\varphi}{360 n} = \frac{20}{360 \frac{2200}{60}} = 0,001575 \; [s]$$

$\Delta\varphi$ - [°CA] - crancshaft
n [min^{-1}] - rot. engine speed
d_b = 0,4 mm as in the point 1
v_b = 248,8·10^3 mm/s as in the point 1
ϱ_g = 840 kg/m^3 as in the point 1
ϱ_v = 27,87 kg/m^3 as in the point 1

Calculation:

$$l_y = 2 \cdot 0,4^{0,46} (248,8 \cdot 10^3)^{0,54} \cdot \left(\frac{840}{27,87}\right)^{0,23} 0,001515 =$$

$$= 70,6 \; mm$$

(In the book presented v_b dimension was in error)

3. **Fuel input data of interest for injection calculation**

	D2	Methanol
Sound velocity [m/s]	1380	1180
Modul of elasticity [N/m^2]	$1,68.10^9$	$0,75.10^9$
modul of compressibility [m^2/N]	$0,595 \cdot 10^{-9}$	$1,333 \cdot 10^{-9}$
surface tension [N/m]	0,0285 ÷ 0,0290	0,021 ÷ 0,0226

Note:

$$\frac{\text{compressibility of methanol}}{\text{compressibility of D2}} = \frac{1,333}{0,595} = 2,24$$

Compressibility of the methanol fuel is by factor 2 higher than that of diesel oil D2.

NCTATION

FORTRAN	MATHEMATIC	
A	a	-sound velocity
AC	A_c	- cross - sec. flow area of the high pressure tube
AK	A_k	- plunger cross-sec. area
AL	L	- lenght of the high pressure tube
ALFA	α	- coeficient of compressibility
ALS	l	- distance between the vapour cavities
AMAB	$(\mu A)_B$	- nozzle flow area
AMBABZ	$\mu_b A_b$	- injector flow area
AMI	m_i	- mass of the injector moving parts
AMPAPZ	$\mu_p A_p$	- instantaneous sleeve inlet flow area
AMV	m_v	- mass of the relief valve moving parts
AMVAV1Z	$\mu_v A_{v1}$	- relief valve flow area
AN	n	- pump rot. speed
AV	A_v	- cross-sec. area of the relief valve piston
AX	A_x	- needle seat area
A2	A_1	- cross-sec. needle area
BRCMAX	-	- max. cycle number
COP	C_{ob}	- the rigidity of the injector spring
CV	C_v	- the rigidity of the relief valve spring
DEFI	$D\varphi$	- Integ. step
EMOD	E	- modul of elasticity
FOP	F_{ob}	- preloading force of the injector spring

FT, FT2, FT3, FT4, FT5, FT6	$F(t)$	- cross-sec. I - I forward dirested wave
FTL, FTLJ, FTLJ1, FTLJ2, FTLO	$F(t-\frac{L}{a})$	- cross-sec. II - II forward directed pressure wave
FVO	F_{vo}	- preloading force of the relief valve spring
F1	m	- number of the fuel vapour cavities chosen in the HP tube
HI	h_i	- needle lift
HIMAX	h_{imax}	- max. needle lift
HK	h_k	- plunger lift
HKMAX	h_{kmax}	- max. plunger lift
HV	h_v	- relief valve piston lift
IK, IK1, JK, JK1, LK, LK1, LX1, MK, MK1	-	- integer variable
PB	p_B	- pressure in the sack volume
PBO	p_{BO}	- start pressure in the sack vol.
PISPAR	p_{ois}	- evaporation pressure
PN	p_n	- pressure in the gallery
PO, POI	p_o	- residual pressure
PZ	p_z	- in-cylinder gas pressure
QB, QB1	$\frac{dV_{p\check{z},bl}}{d\varphi}$	- the filling cavities change in the injector
QKL, QKL1	$\frac{dV_{p\check{z},kl1}}{d\varphi}$	- the same for relief valve holder
RO	ρ	- fuel density
SIGMAK	σ_k	- step functions
SIGMAO	σ_o	
SIGMA1	σ_1	
SIGMA3	σ_3	
VB	V_b	- injector volume
VCI	V_c	- HP tube volume
VC, VK	$\frac{v_k}{\omega}$	- relative plunger velocity
VKL	V_{kl}	- relief valve holder volume

VPC	$V_{p\check{c}}$	- volume of the cavities formed in the HP system
VPCB1	$V_{p\check{c},bl}$	- volume of the cavities in the injector
VPCC1	$V_{p\check{c},cl}$	- the same in the HP tube
VPCKL1	$V_{p\check{c},kll}$	- the same in the relief valve holder
VSR	V_s	- the whole HP volume
VVKZ, VVKZ1	V_u	- the in-pump volumeover the plunger
VVMIN	V_{kmin}	- the minimum in-pump volume over the plunger
WI	w_I	- fluid velocity at I - I
WIO	w_{Io}	- start fluid velocity at I - I
WII	w_{II}	- fluid velocity at II - II
WIIO	w_{IIo}	- start fluid velocity at II - II
WT, WTJ, WT2, WTJO, WTO	$W(t)$	- reflected pressure wave at I - I
WTL, WTLO, WTL2, WTL3, WTL1, WTLI	$W(t+\frac{L}{a})$	- reflected pressure wave at II - II
X	x	- the distance coordinate
XEND	φ_{kraj}	- end of the cycle
XGR	φ_{gr}	- the boundary for removing the effective delivery end
XP	φ_p	- start of the cycle
XI10, XI11, XI12, XI13, XI2, XI3, XI4, XI5, XI6, XI7, XI8, XI9, X1, X2	φ	- cranckshaft angle
Y	y	- unknown magnitudes to be calculated
Y10, Y20, .. Y70	y_o	- the start values of the unknown magnitudes
BI	F_i	- the impact force on the injector needle

GRE	Δ	- **the permissible method error**
GRPO	Δ_{po}	- **the residual pressure error**
I, IEND, INLF, IMOD, IMON, IMONKR, IMONKR1, IMON1, IMON2, IMON3, IREC, ISTEP, ITEST, IXA, IXSA, L, LV, M, MK1		- integer variables
NJED	n_{jed}	- the number of diff. equations
N1	n_1	- the div. number of the HP tube needed for residual pressure calc.
PTLI	p_{Tli}	- the instantaneous pressure amplitude at distance l_i
QCIKLUS	q_c	- fuel injected per cycle
QC	$q_c/\Delta\varphi$	- mean injection velocity
QK	$\dot{q}_c$	- law of injection
RV	F_v	- the force influenced on the relief valve
DERY	$dy/d\varphi$	- the first derivation of the variables

N O T A T I O N

A_c - cross-sec. flow area of HP tube
Ak - cross-sec. plunger area
A_v - Cross-sec. area of the relief valve piston
A_x - cross-sec. needle area of seat
A_1 - cross-sec. stem needle area
a - sound velocity
Cob - the rigidity of the injector spring
Cv - the rigidity of the relief valve spring
E - modul of elasticity
Fob - preloading force of the injector spring
Fvo - preloading force of the relief valve spring
$F(t)$, $F(t - \frac{L}{a})$ - pressure forward waves at I - I and at II - II, resp.
h_i - needle lift
h_{imax} - max. needle lift
h_k - plunger lift
h_v - lift of the relief valve piston
h_{v1} - position of the relief valve piston at the closing time of the needle
h_R - the retraction lift of the relief valve piston
L - lenght of the HP tube
l - cavities distance in the HP tube
$l_{pš}$ - lenght of the vapour cavity in the HP tube
m_i - mass of the injector moving parts
m_v - mass of the relief valve
n - pump rot. speed
p - pressure
p_k - in-pump pressure
p_n - pressure in the gallery
p_o - residual pressure
$p_{ɔis}$ - fuel evaporation pressure
p_z - in-cylinder gas pressure
p_I, p_{II} - pressure at I -I and at II - II, resp.
$\dot{Q}g$ - injector leakage overflow
q_c - injected quantity
$\dot{q}_c$ - law of injection

V_b - injection volume
V_c - HP tube volume
V_k - pumpe volume (over the plunger)
V_{kl} - relief valve volume
$V_{p\check{s}}$ - volume of the fuel vapour cavities
$V_{p\check{s},b}$ - volume of the fuel vapour cavities in the injector
$V_{p\check{s},c}$ - " " " " " in the HP tube
$V_{p\check{s},kl}$ - " " " " " in the relief valve holder

$V_{p\check{s},bl}$ - the change of the vapour cavities in the injector
$V_{p\check{s},cl}$ - " " " " " " in the HP tube
$V_{p\check{s},kll}$ - " " " " " " in the relief valve holder

V_s - volume of the HP system
v_i - needle velocity
v_k - plunger velocity
v_v - relief piston velocity
$W(t)$, $W(t + \frac{L}{a})$ - reflected pressure wave at I - I and at II - II, resp.
w_I, w_{II} - fluid velocity at I - I and at II - II, resp.
$\alpha = \frac{1}{E}$ - modul of compressibility
Δt - calculation time step
$(\mu A)_B$ - eff. flow area of the oriffices
$\mu_b A_b$ - eff. flow area of the injector
$\mu_p A_p$ - eff. flow area of the sleeve inlet and spill area
$\mu_v A_v$ - eff. flow area of the relief valve
- fuel density
σ_k, σ_o, σ_1, σ_2, σ_3 - step function
φ - cranckshaft angle
φ_{ku} - cranckshaft angle at the injection end
φ_{pu} - cranckshaft angle at the injection start
$\varphi_{vAvl=0}$ - cranckshaft angle at the relief valve closing

MATHEMATIC MODEL

1. Runge-Kuta method

1. The cont. eq. in the pump:

$$\frac{dp_K}{d\varphi} = \left[A_K v_K - \sigma_o (\mu_p A_p)\sqrt{\frac{2}{\varsigma}(p_K - p_n)} - \sigma_K (\mu_v A_{v_1})\sqrt{\frac{2}{\varsigma}(p_K - p_I)} - \right.$$
$$\left. - \sigma_1 A_v v_v \right] \cdot \frac{1}{360 n \alpha V_K} \quad \ldots \quad (1)$$

2. The cont. eq. in the reliefe valve holder:

$$\frac{dp_I}{d\varphi} = \left[\sigma_K (\mu_v A_{v_1})\sqrt{\frac{2}{\varsigma}(p_K - p_I)} + \sigma_1 A_v v_v - A_c w_I\right] \frac{1}{360 n \alpha V_{Kl}} \quad \ldots (2)$$

3. The force eqv. eq. at reliefe valve:

$$m_v \frac{d^2 h_v}{dt^2} + c_v h_v = A_v (p_K - p_I) - F_{vo}$$

which can be written as follows:

$$\frac{dv_v}{d\varphi} = \left[A_v (p_K - p_I) - F_{vo} - c_v h_v\right] \frac{\sigma_1}{360 n \alpha m_v} \quad \ldots\ldots (3)$$

and as:

$$\frac{dh_v}{d\varphi} = v_v \frac{\sigma_1}{360 n} \quad \ldots \quad (4)$$

4. The eqs. for velocity and pressure calculations at I - I:

$$w_I = \frac{1}{a \varsigma} \left[p_I - p_o + 2 W(t)\right] \quad \ldots \quad (5)$$

$$F(t) = p_I - p_o + W(t) \quad \ldots \quad (6)$$

5. **The cont. eq. in the injector volume:**

$$\frac{dp_{\bar{II}}}{d\varphi} = \left[A_e w_{\bar{II}} - (\mu_b A_b)\sqrt{\frac{2}{\varsigma}(p_{\bar{II}} - p_z)} - \sigma_3 A_1 v_1 - \dot{Q}_g\right]\frac{1}{360\, n\, \alpha\, V_b} \quad \ldots (7)$$

6. The force eqv. eq. of the injector:

$$m_i \frac{d^2 h_i}{dt^2} + C_{ob} h_i = (A_1 - A_x)p_{\bar{II}} + A_x \cdot p_B - F_{ob}$$

which can be written as:

$$\frac{dv_i}{d\varphi} = \left[(A_1 - A_x)p_{\bar{II}} + A_x p_B - F_{ob} - h_i C_{ob}\right]\frac{\sigma_3}{360\, n \cdot m_i} \quad \ldots (8)$$

and as:

$$\frac{dh_i}{d\varphi} = v_i \frac{\sigma_3}{360 \cdot n} \quad \ldots (9)$$

7. The flow through the orifficies and through the sack is the same, which gives:

$$p_B = \frac{(\mu_b A_b)^2}{(\mu A)_B^2} \cdot (p_{\bar{II}} - p_z) + p_z \quad \ldots (10)$$

8. The fluid velocity at II - II and reflected pressure wave:

$$w_{\bar{II}} = \frac{1}{a\,\varsigma}\left[p_o - p_{\bar{II}} + 2F\left(t - \frac{L}{a}\right)\right] \quad \ldots (11)$$

$$W\left(t + \frac{L}{a}\right) = p_o - p_{\bar{II}} + F\left(t - \frac{L}{a}\right) \quad \ldots (12)$$

9. The connection between the coresponding pressure waves at I - I and II - II:

$$F\left(t - \frac{L}{a}\right)_{\varphi} = F(t)_{\varphi - 360 \cdot n \cdot \frac{L}{a}} \quad \ldots (13)$$

$$W(t)_{\varphi} = W\left(t + \frac{L}{a}\right)_{\varphi - 360 \cdot n \cdot \frac{L}{a}} \quad \ldots (14)$$

10. **The injection law:**

$$\dot{Q}_c = (\mu_b A_b) \cdot \frac{1}{360n} \sqrt{\frac{2}{s}(p_{II} - p_z)} \qquad \ldots (15)$$

11. The integ. law of injection:

$$Q_c = \int_{\varphi_{pu}}^{\varphi} \frac{1}{360n} (\mu_b A_b) \sqrt{\frac{2}{s}(p_{II} - p_z)} \cdot d\varphi \qquad \ldots (16)$$

12. The residual pressure:

$$p_o = p_{s_{h_i=0}} - \frac{A_v h_{v_1}}{\alpha \cdot V_s} - \frac{Q_{v_1}}{\alpha V_s} \qquad \ldots (17)$$

where:

$$p_{s_{h_i=0}} = \frac{\sum_{i=1}^{n} p_{T_i} + p_I + p_{II}}{n+2} \qquad \ldots (18)$$

$$p_{T_i} = p_o + F\left(t - \frac{z_i}{a}\right) - W\left(t + \frac{L - z_i}{a}\right) \qquad \ldots (19)$$

$$Q_{v_1} = \sum_{h_i=0}^{\varphi_{\mu_v A_{v_1}=0}} \frac{\Delta\varphi}{360 \cdot n} (\mu_v A_{v_1}) \sqrt{\frac{2}{s} |p_K - p_I|} \qquad \ldots (20)$$

13. Emergency eq.:

$$\left.\begin{array}{l}
\sigma_0 = +1 \quad \text{for} \quad p_K \geq p_n \\
\sigma_0 = -1 \quad \text{for} \quad p_K < p_n \\
\sigma_K = +1 \quad \text{for} \quad p_K \geq p_I \\
\sigma_K = -1 \quad \text{for} \quad p_K < p_I \\
\sigma_1 = 0 \quad \text{for} \quad A_v(p_K - p_I) - h_v C_v - F_{vo} \leq 0 \text{ and } h_v \leq 0
\end{array}\right\} \qquad (21)$$

$$\left.\begin{array}{ll} \sigma_1 = 1 & \text{for all other cases} \\ \sigma_3 = 0 & \text{for } (A_1 - A_x)\cdot p_{II} + A_x p_B - F_{op} - C_{ob}\cdot h_i \leq 0 \\ & \qquad \text{and } h_i \leq 0 \\ \sigma_3 = +1 & \text{for all other cases} \end{array}\right\} \quad (21)$$

If the residual pressure was less than than the fuel evaporation pressure:

$$p_o < p_{ois} \qquad \ldots \quad (22)$$

the calculation method should be changed.
The quantity of the cavities may be defined as follows:

$$V_{p\check{c}} = -\alpha V_s \left(p_\bullet - p_{ois}\right) \qquad \ldots \quad (23)$$

and has to be distributed equaly to the coresponding volumes:

$$\left.\begin{array}{l} V_{p\check{c},kl} = \dfrac{V_{kl}}{V_s}\cdot V_{p\check{c}} \\ V_{p\check{c},c} = \dfrac{V_c}{V_s}\cdot V_{p\check{o}} \\ V_{p\check{o},b} = \dfrac{V_b}{V_s}\cdot V_{pc} \end{array}\right\} \qquad \ldots \quad (24)$$

In this case instead of eq. (2) the next eqs. should be used:

$$\left.\begin{array}{l} \dfrac{dV_{p\check{o},kl_1}}{d\varphi} = \left[\sigma_K (\mu_v A_{v_1})\sqrt{\dfrac{2}{\varsigma}|p_K - p_I|} + \sigma_1 A_v v_v\right]\dfrac{1}{360\,n} \\ \dfrac{dp_I}{d\varphi} = 0\,; \quad p_I = p_{ois} \\ \text{if was:} \quad V_{p\check{o},kl_1} \leq V_{p\check{c},kl} \end{array}\right\} \quad (2a)$$

However if $V_{p\check{z},kll} > V_{p\check{z},kl}$ we are going to use eq. (2).

The vapour cavities in the HP tube are divided in m parts.

$$V_{p\check{c},c_1} = \frac{V_{p\check{c},c}}{m} \qquad \ldots (25)$$

The cavities distance amounts:

$$l = \frac{L}{m} - l_{p\check{c}} \qquad \ldots (26)$$

where:

$$l_{p\check{c}} = \frac{V_{p\check{c},c_1}}{A_c} = \frac{4\, V_{p\check{c},c_1}}{d_c^2 \cdot \pi} \qquad \ldots (27)$$

Because of the very low residual pressure the flow velocity at the distance z=l (till the first cavity) may be written as follows:

$$W_{c,l} = \frac{2}{a\varsigma} F\left(t - \frac{l}{a}\right) \qquad \ldots (28)$$

At the same time is valed:

$$W\left(t + \frac{l}{a}\right) = F\left(t - \frac{l}{a}\right) \qquad \ldots (29)$$

During the cavity filling the reflected wave will be generated and directed foward the I - I:

$$W\left(t + \frac{l}{a}\right)$$

The filling time of the first cavity ($l_{p\check{z}}$) will be defined as follows:

$$\int_{t_{p_1}}^{t_{k_1}} F\left(t - \frac{l}{a}\right) dt = \frac{a\varsigma}{2A_c} \cdot V_{p\check{c},c_1} \qquad \ldots (30)$$

as:

$$\Delta t_1 = t_{k_1} - t_{p_1} \qquad \ldots (31)$$

But calculating with the pressure propagation time together it will be:

$$t_1 = \frac{l}{a} + \Delta t_n \qquad \ldots (32)$$

The proceeure is the same for the all other cavities.

The total propagation time through the HI tube with cavities amounts:

$$t_{p\check{c}} = \frac{n \cdot l}{a} + \sum_{i=1}^{n} \Delta l_i \qquad \ldots (33)$$

The forementioned procedure was for two phase fluid (air and liquid) but after time $t_{p\check{c}}$ the following calc. procedure should be continued only with ane component phase (liquid).

For the case with the cavities in the injector instead of eq. (7) the next system should be used:

$$\left.\begin{aligned} &\frac{dV_{p\check{c},b_1}}{d\varphi} = \frac{w_{II}\, A_c}{360\, n} \\ &\frac{dp_{II}}{d\varphi} = 0\ ; \quad p_{II} = p_{o,s} \\ &\text{if is} \quad V_{p\check{c},b_1} \leq V_{p\check{c},b} \end{aligned}\right\} \qquad (7a)$$

However if $V_{p\check{c},bl} > V_{p\check{c},b}$ than the calculation will be continued in using the eq. (7).

The fiff. eqs. 1,2, 3, 4, 7, 8, 9, 2a, and 7a are of type:

$$y' = f(x,y)$$

and may be solved with RK - IV and with the changeable step.

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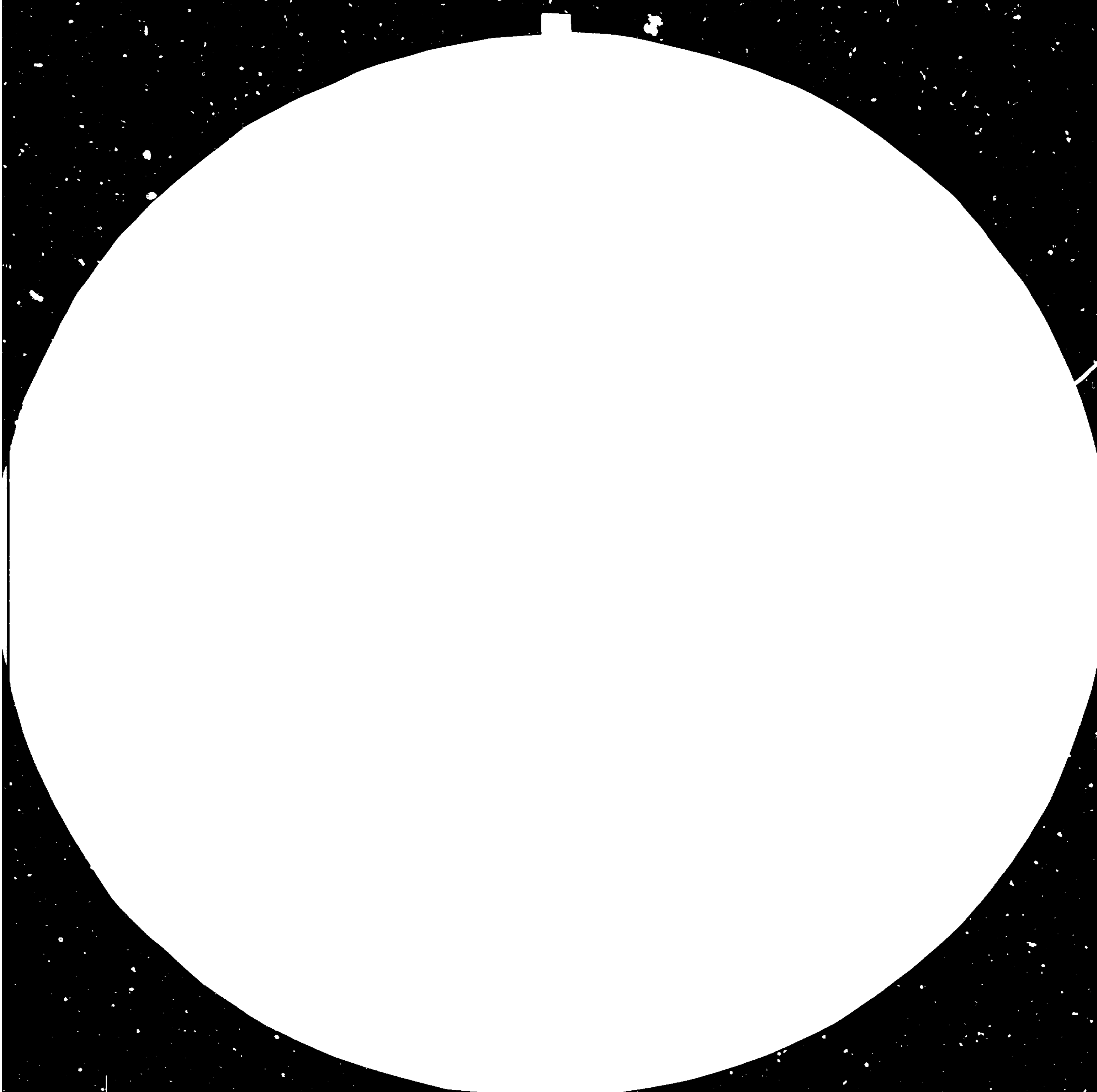
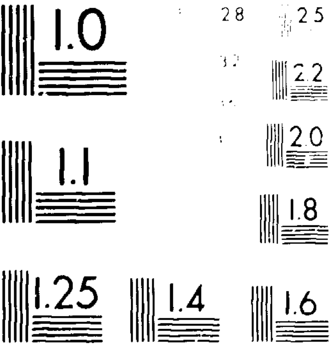




MICROCOPY RESOLUTION TEST CHART

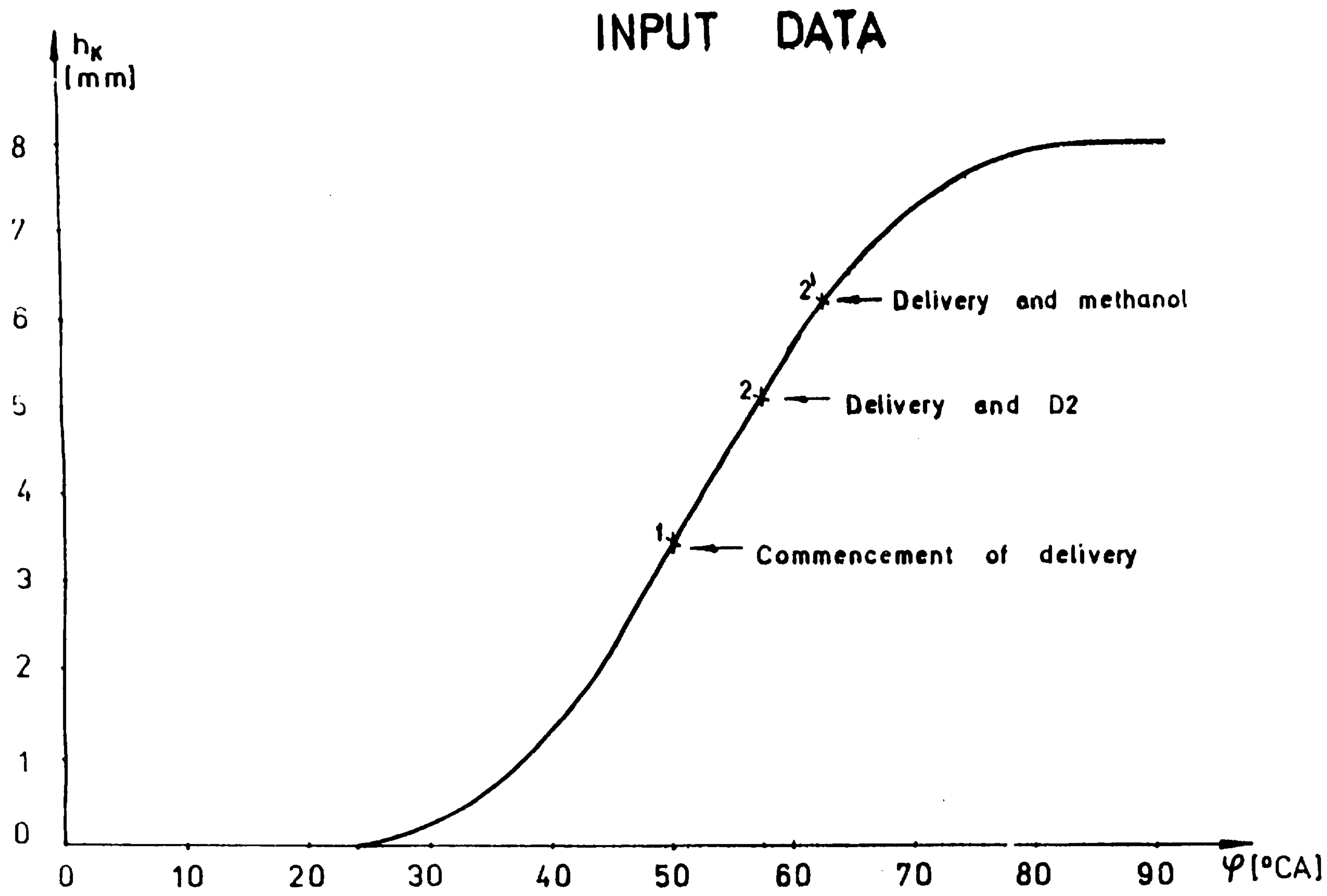


Fig: 1 PLUNGER LIFT

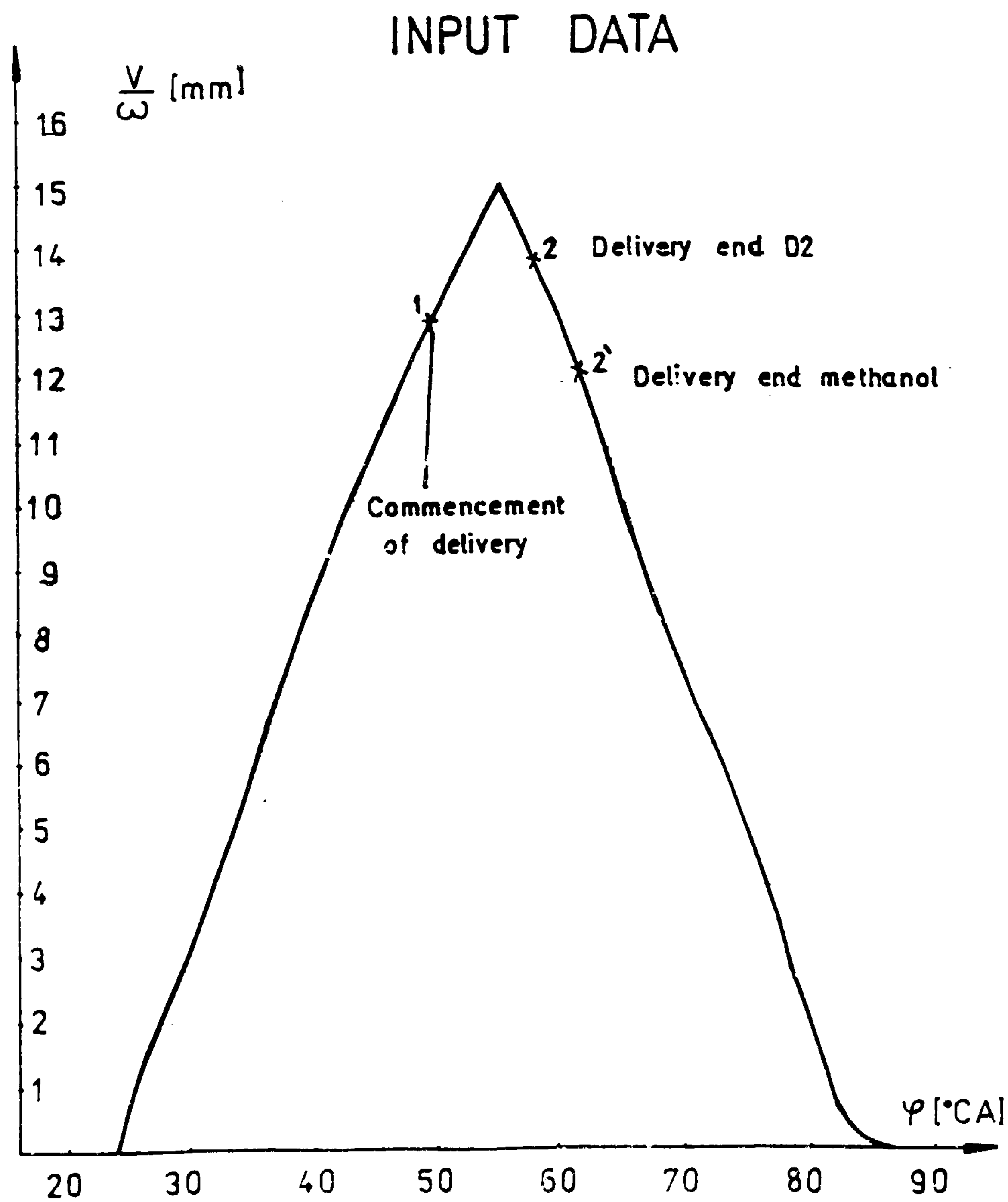


Fig. 2 Relative plunger velocity

INPUT DATA

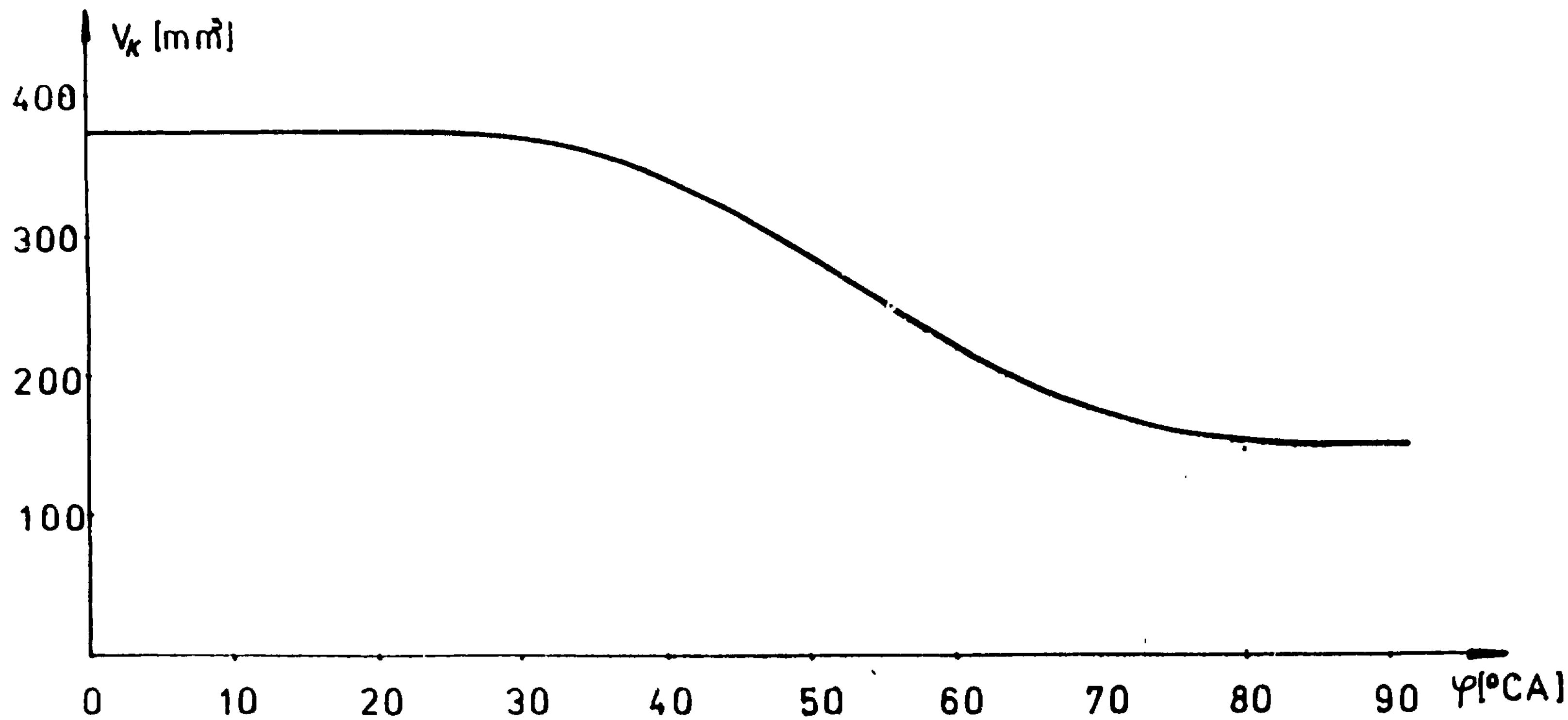


Fig 3 Instantaneous in-pump volume compressed by plunger

INPUT DATA

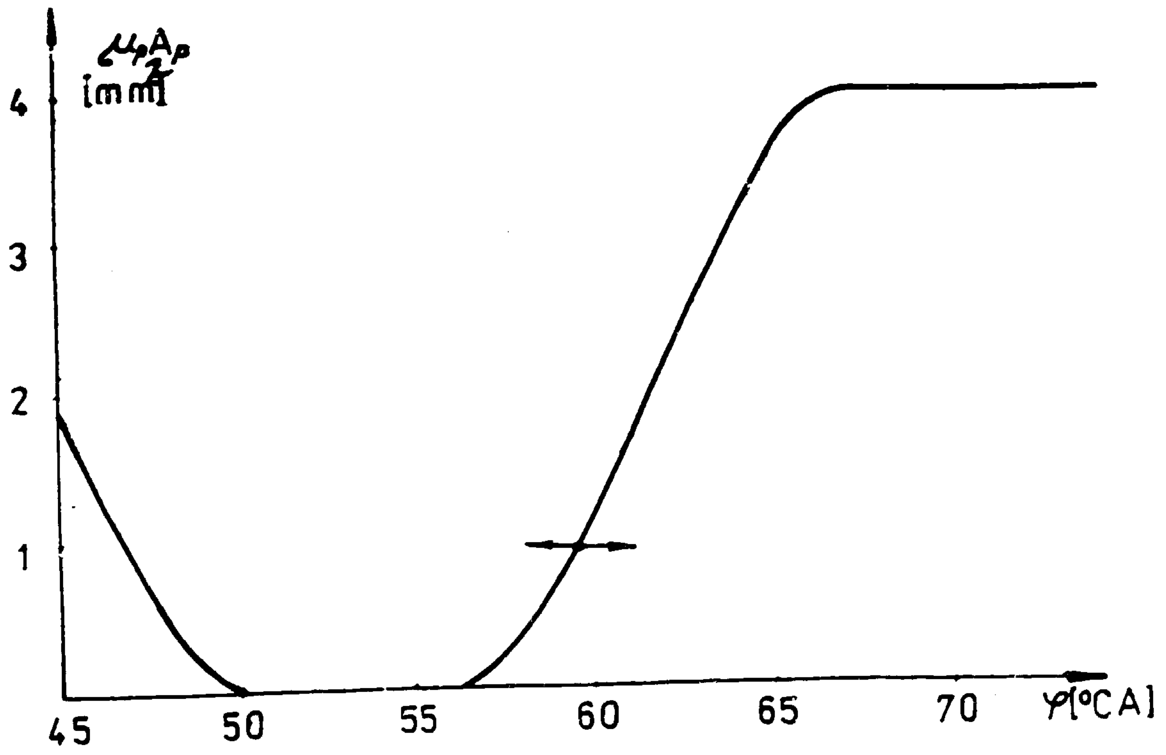


Fig. 4 Iristantanous cross-sect. flow areas comunication in-pump volume-gallery

INPUT DATA

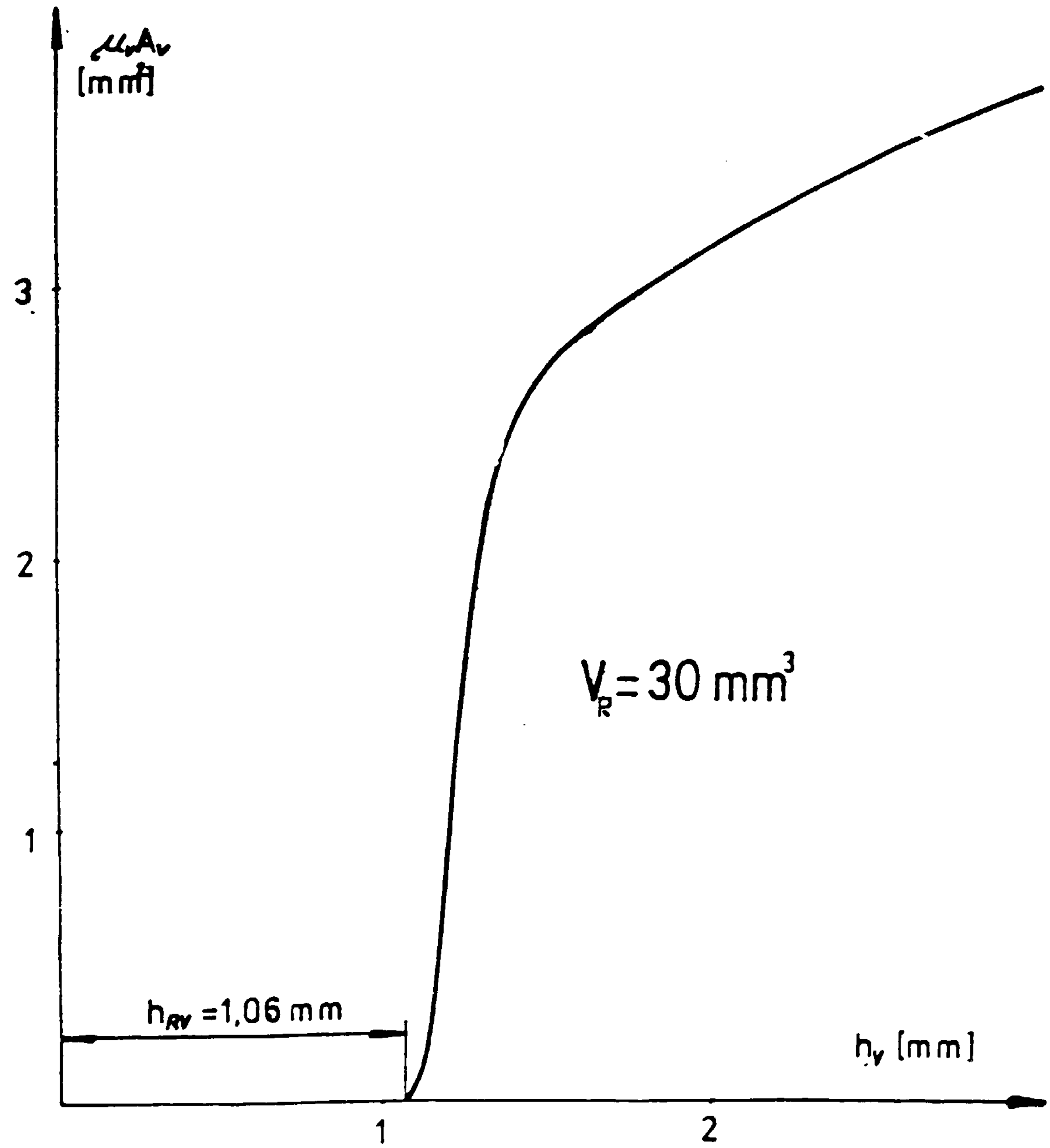


Fig. 5 Instantaneous cross-sect. relief valve flow area

INPUT DATA

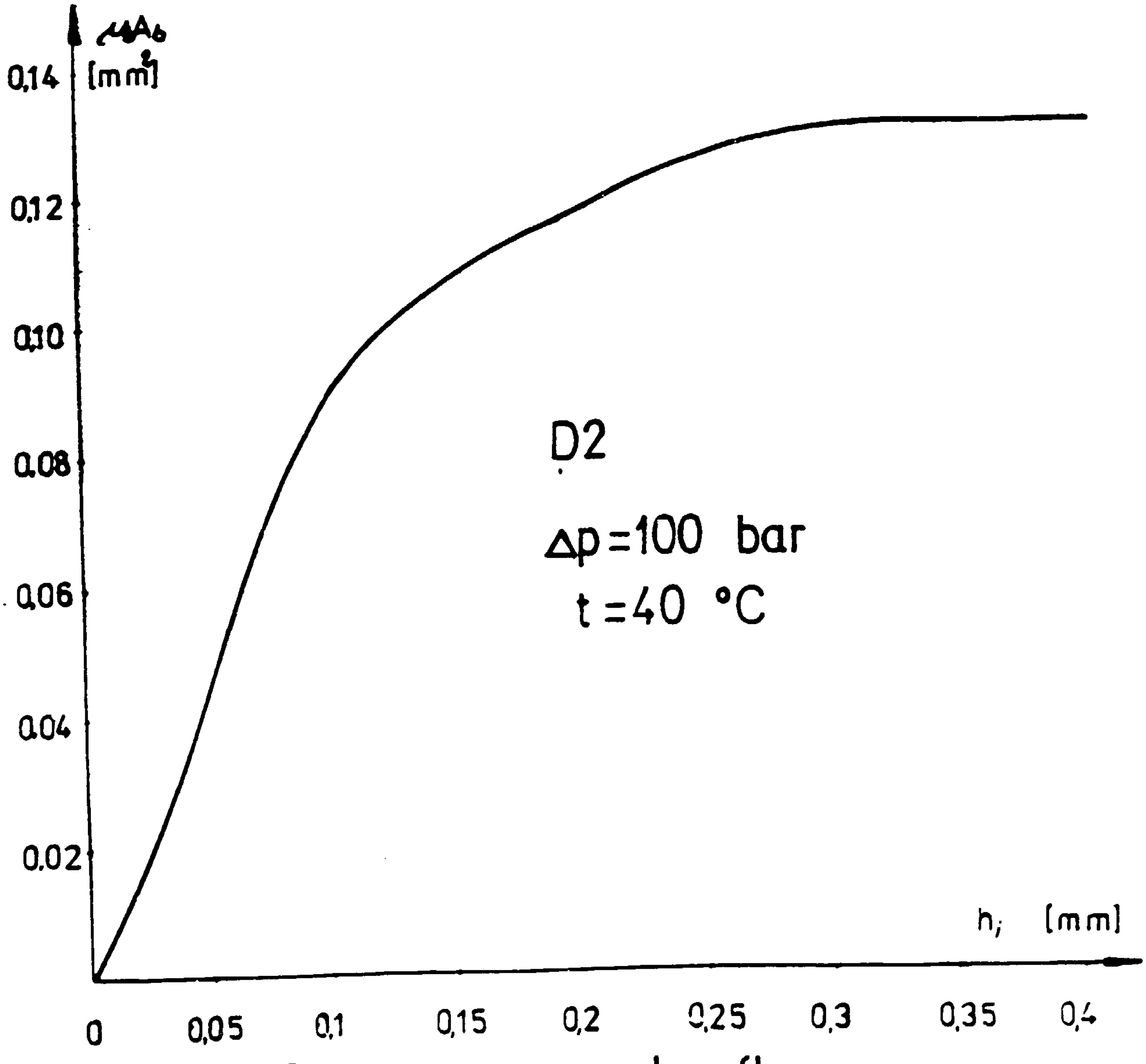
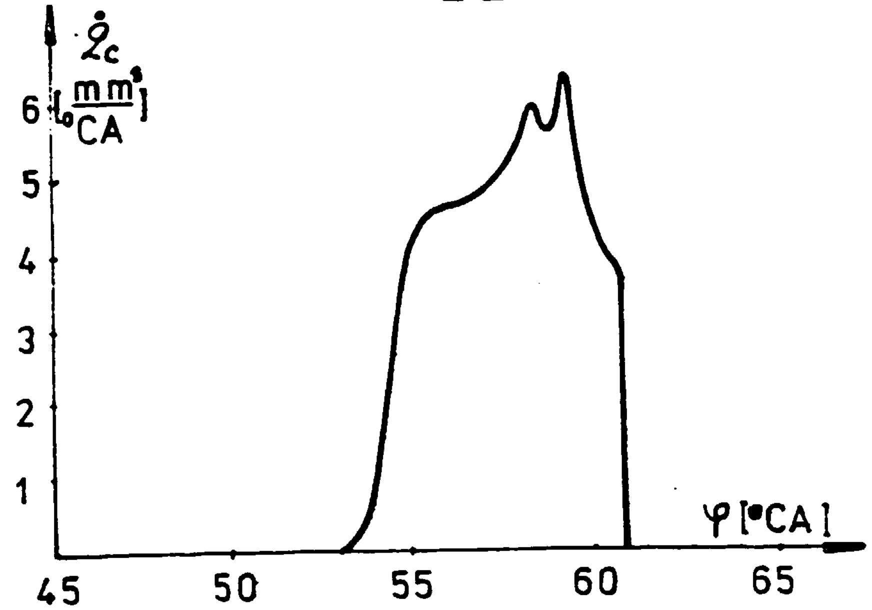


Fig. 6 Cross-sect. nozzle flow area (as measured with IIP sample)

Fig. 7 Law of injection

D2



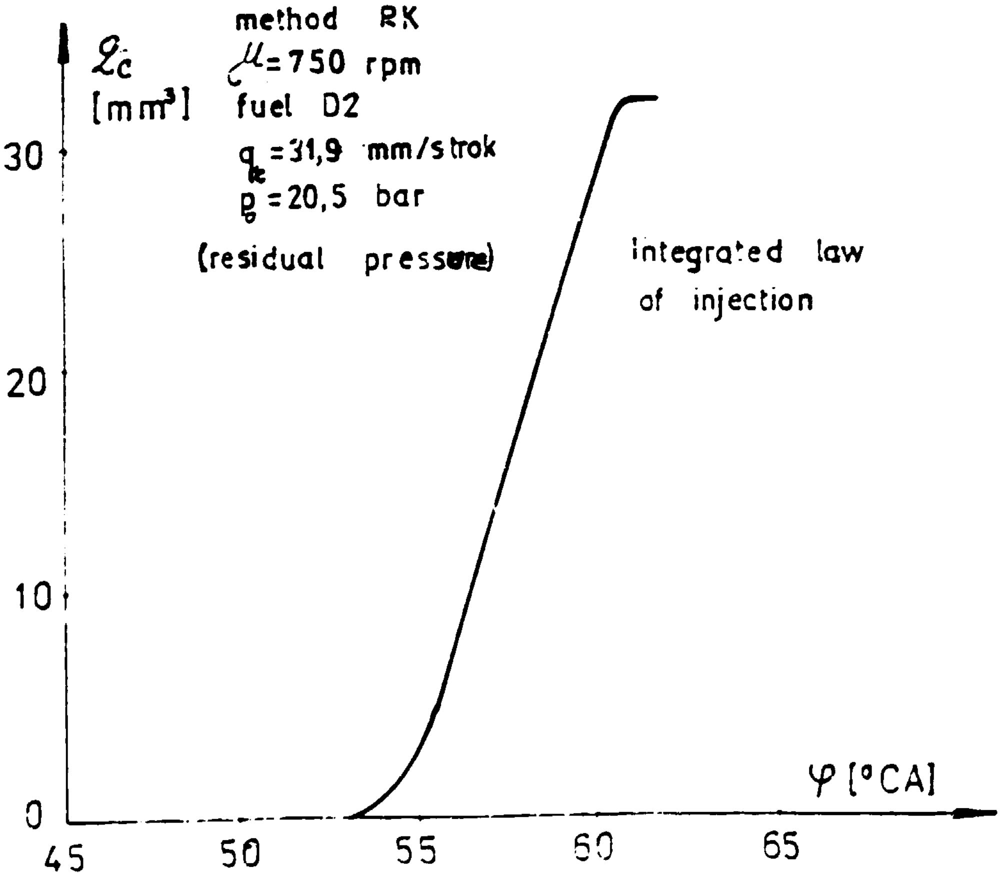


Fig. 8 Pressure before the nozzle holes

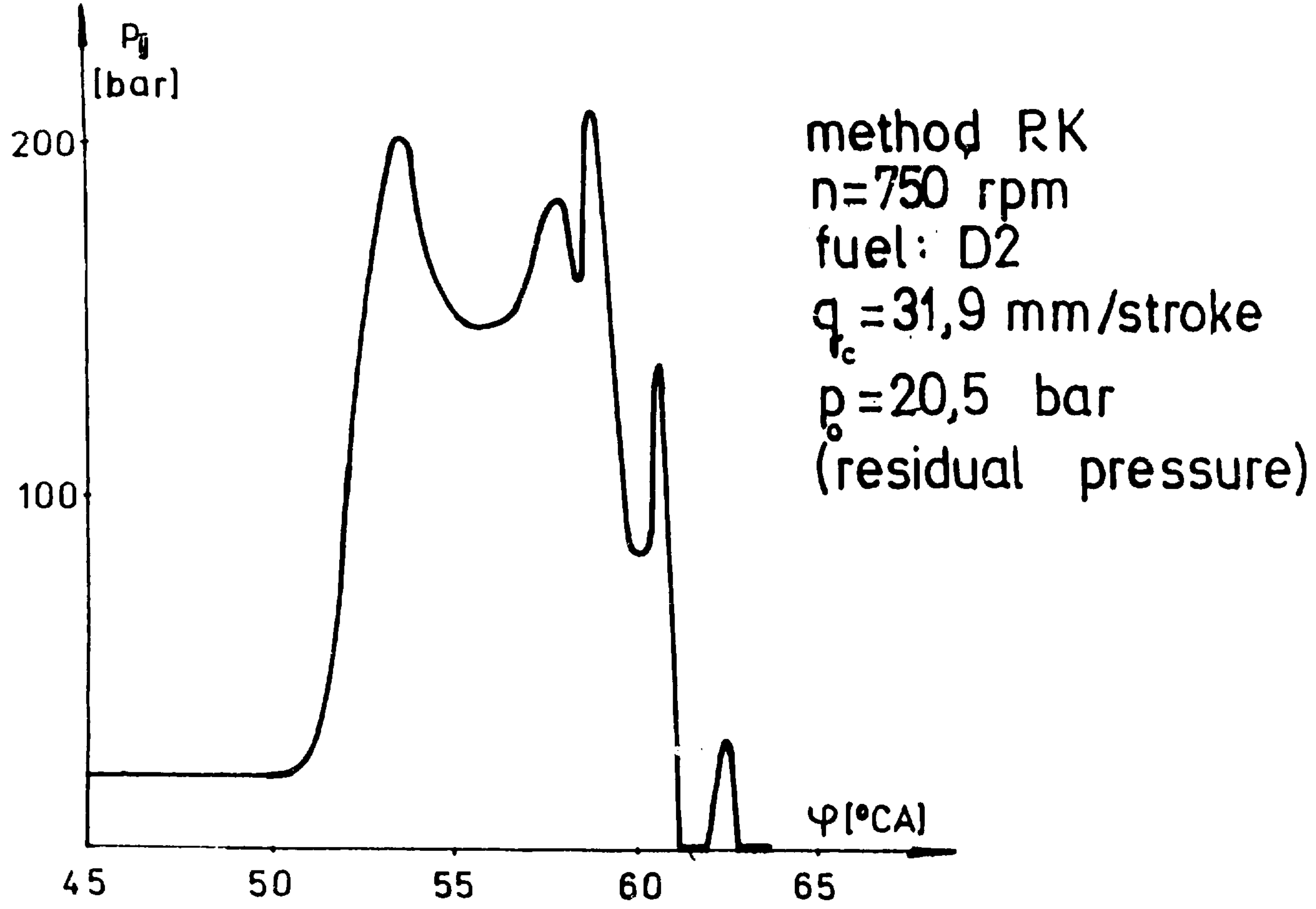
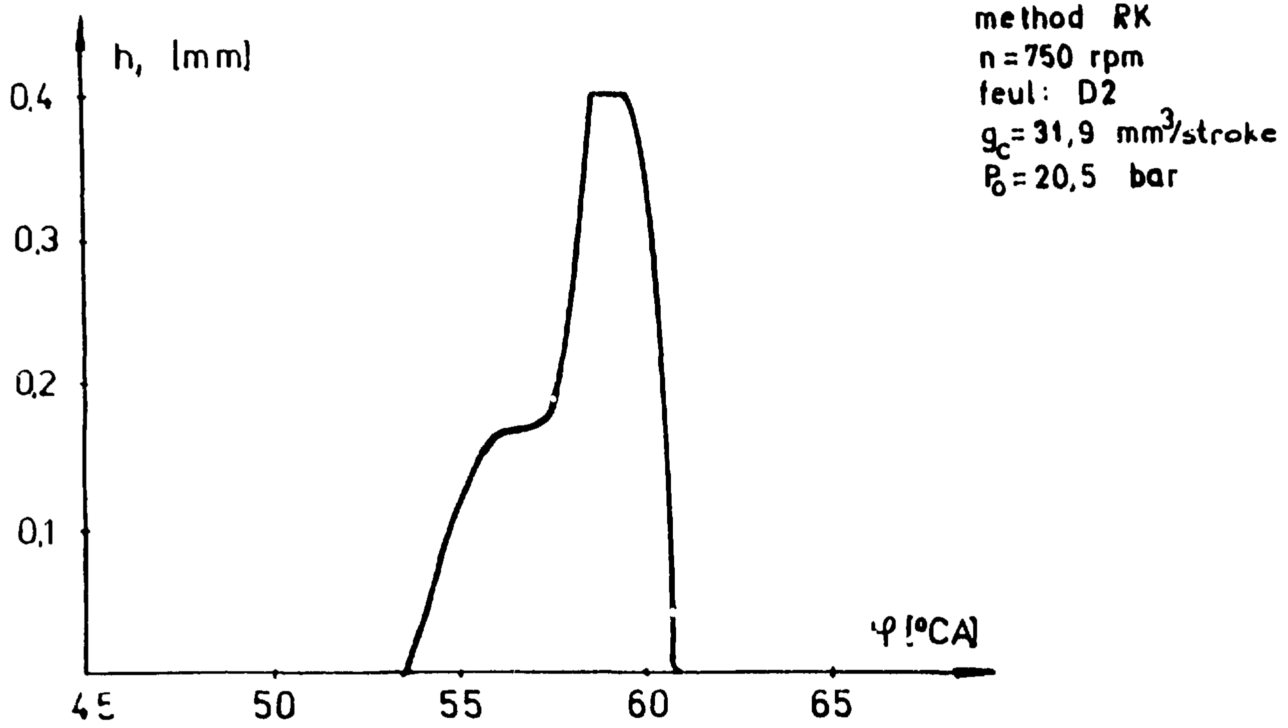


Fig. 9 Needle lift (D2)



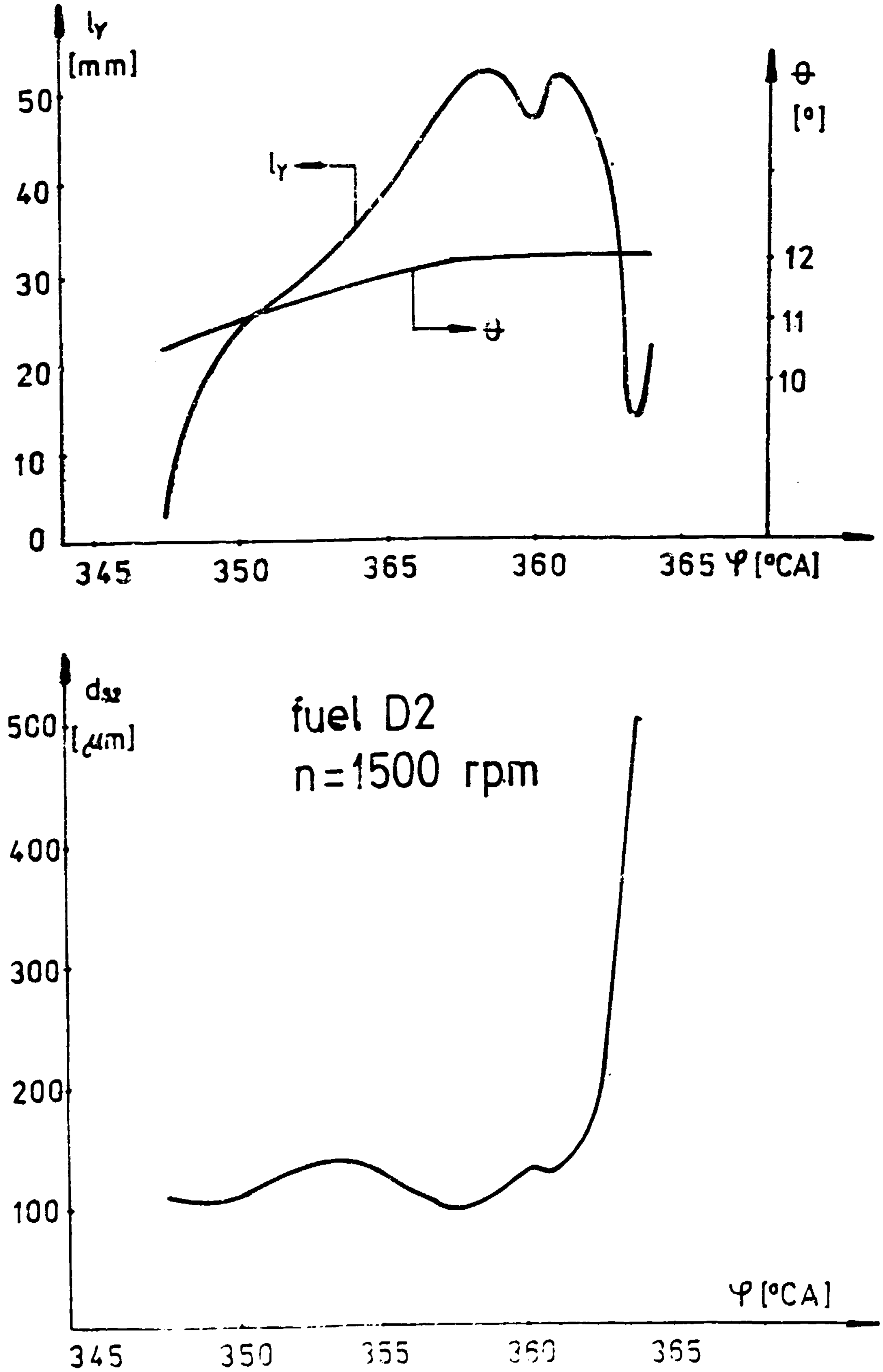


Fig.10 Sauter mean dia.(d_{32}), spray distance(l_y) and spray cone(ϑ) for D2

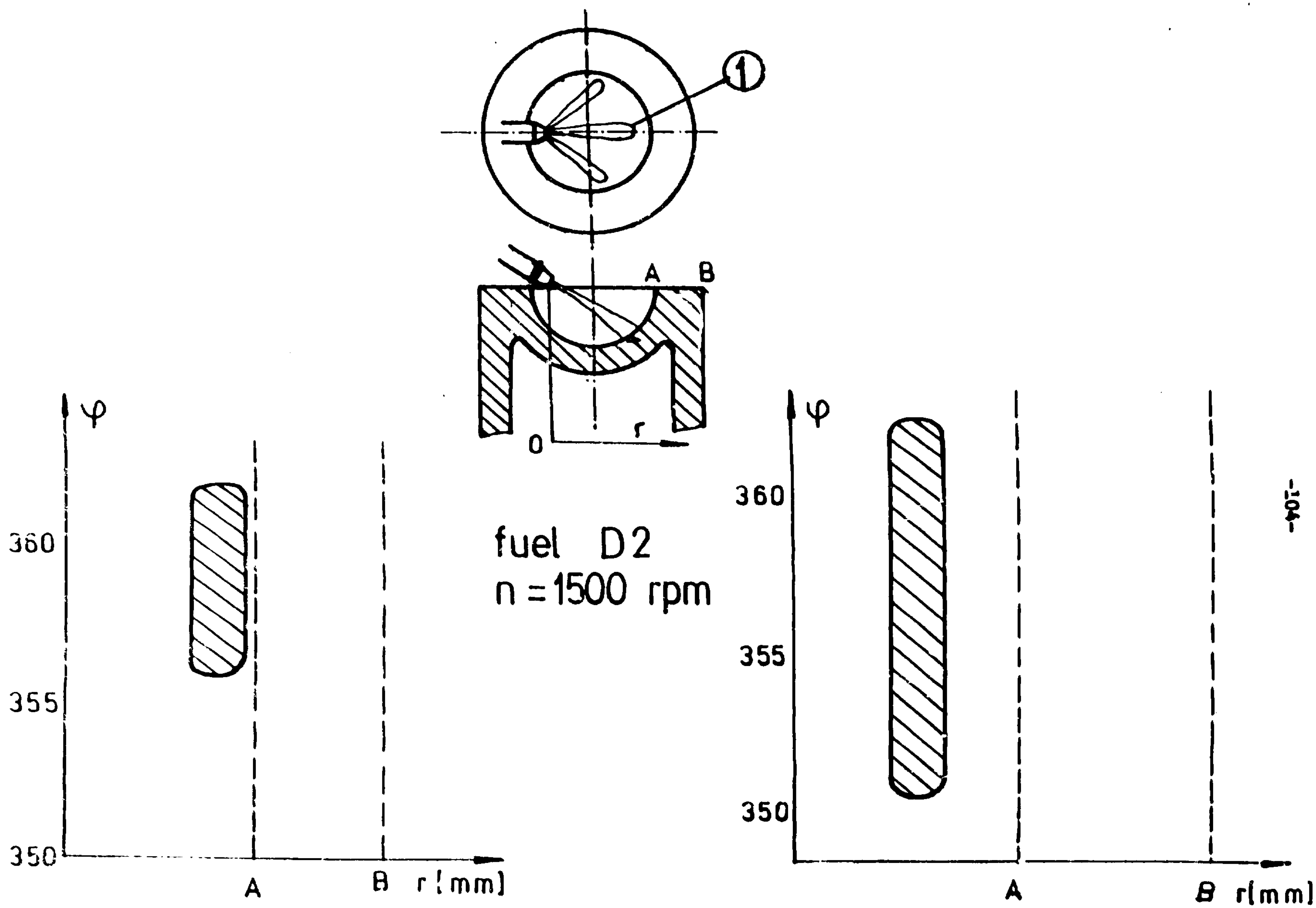


Fig.11 Fuel-comb. chamber wall time contact for D2, only spray 1 calculated

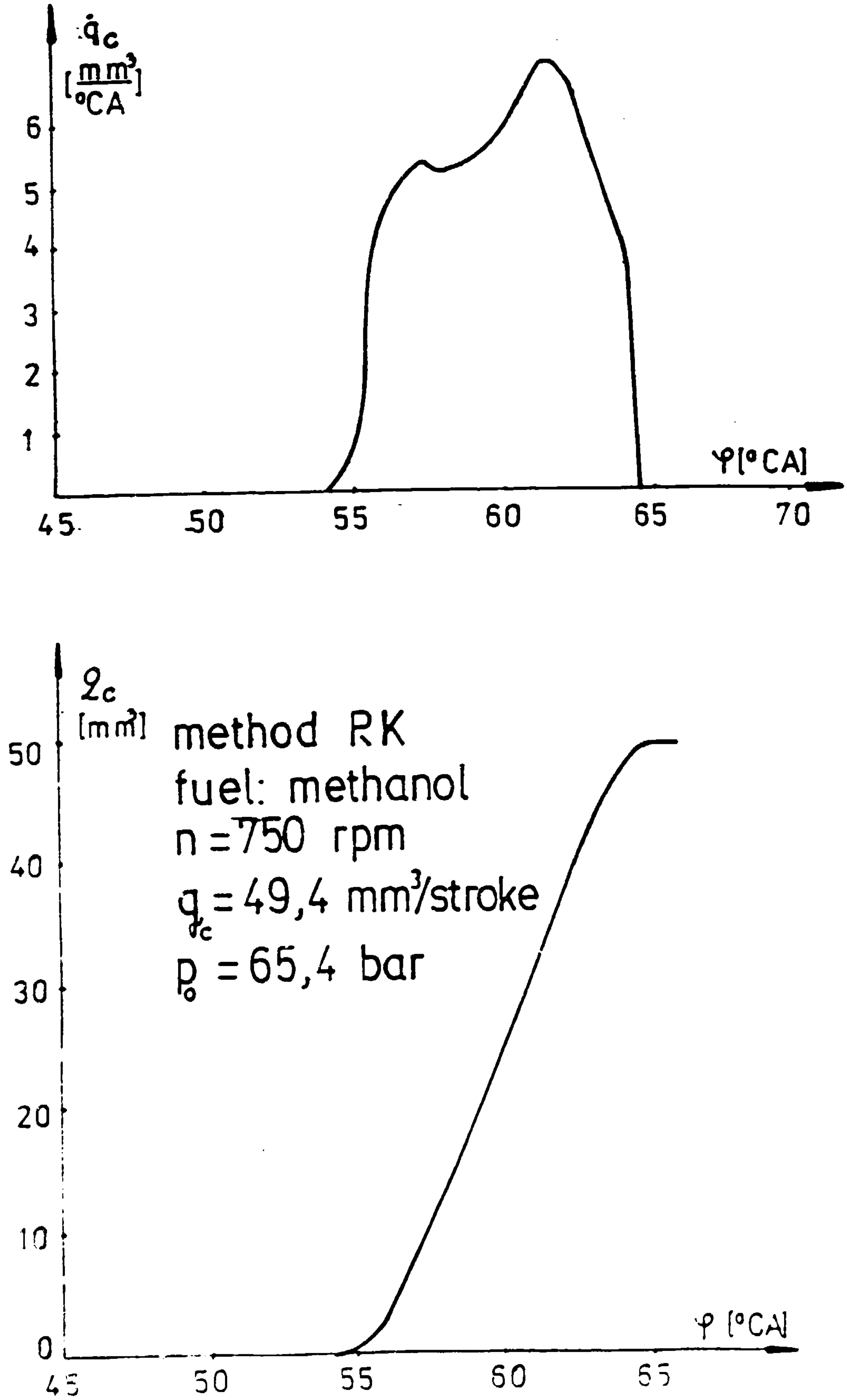


Fig.12 As for Fig.7, methanol

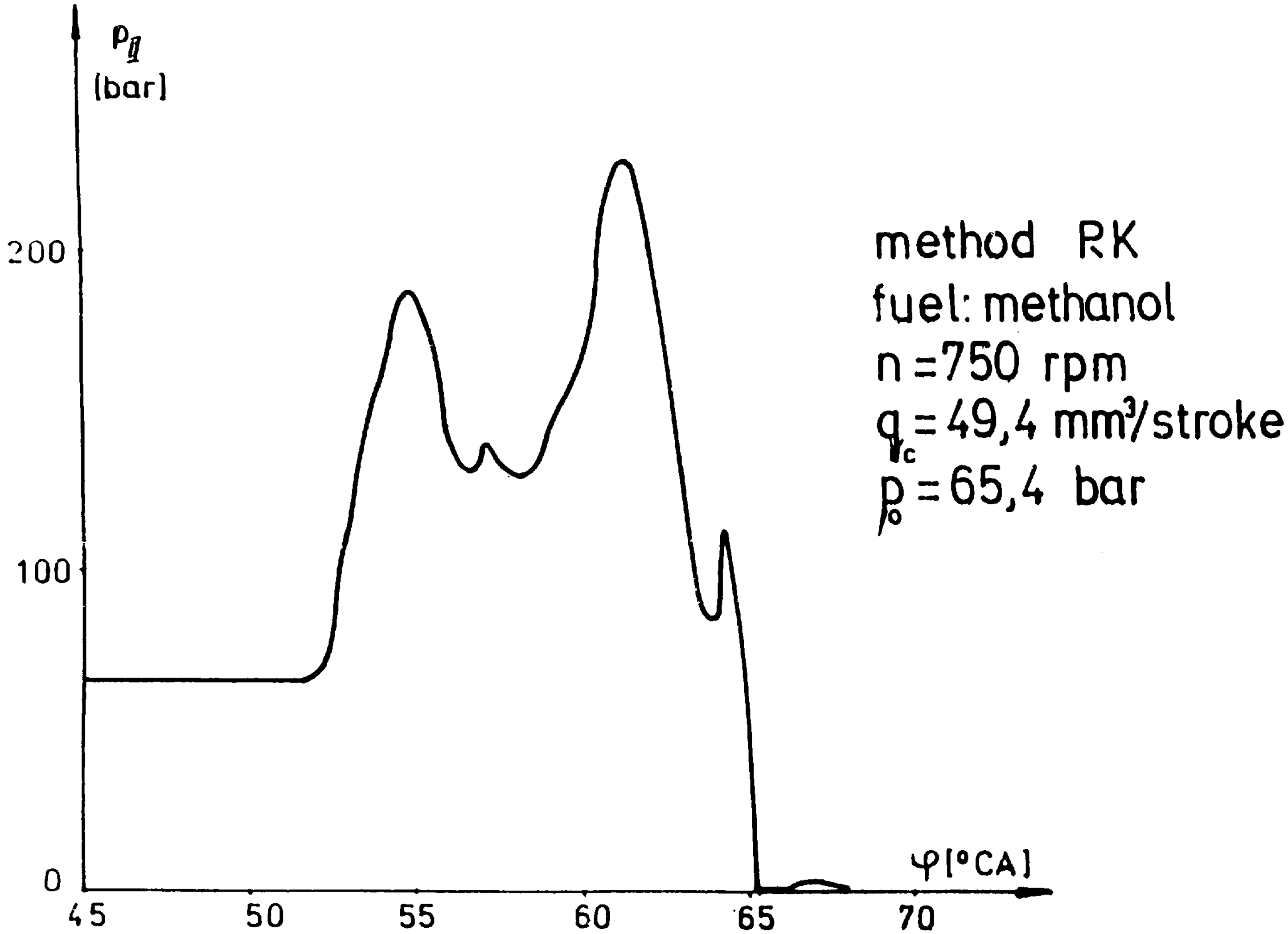


Fig.13 As for fig 8, methanol

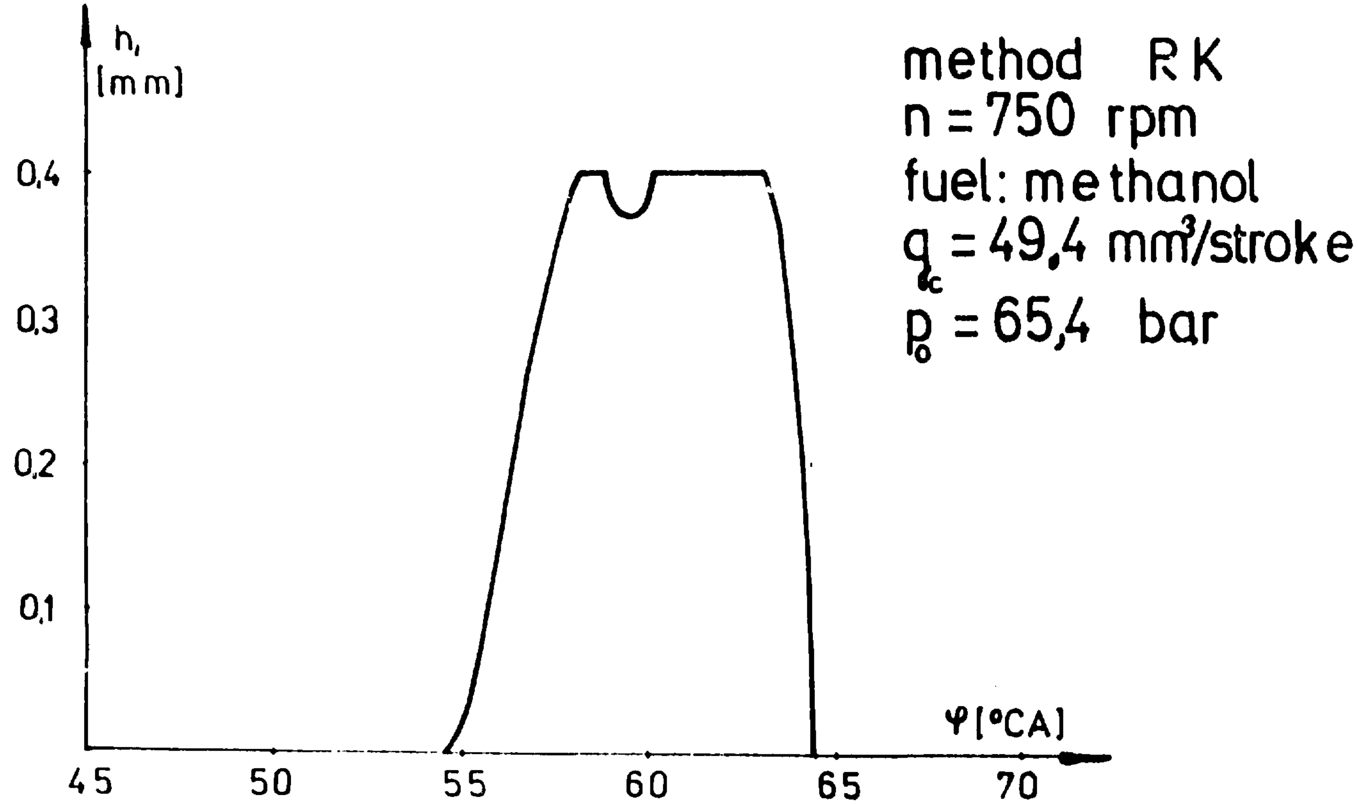


Fig.14 As for fig.9, methanol

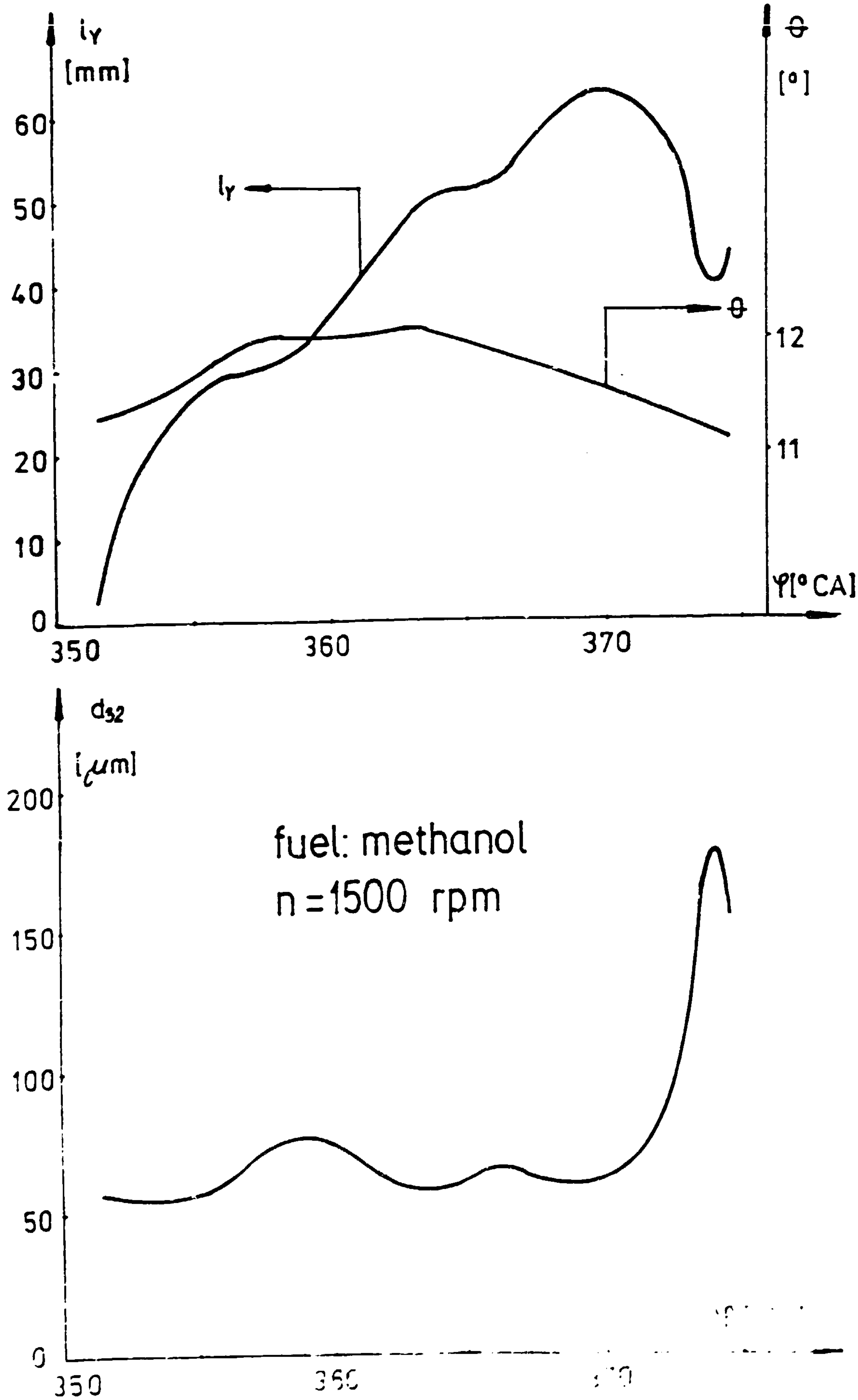


Fig.15 As for fig.10, methanol

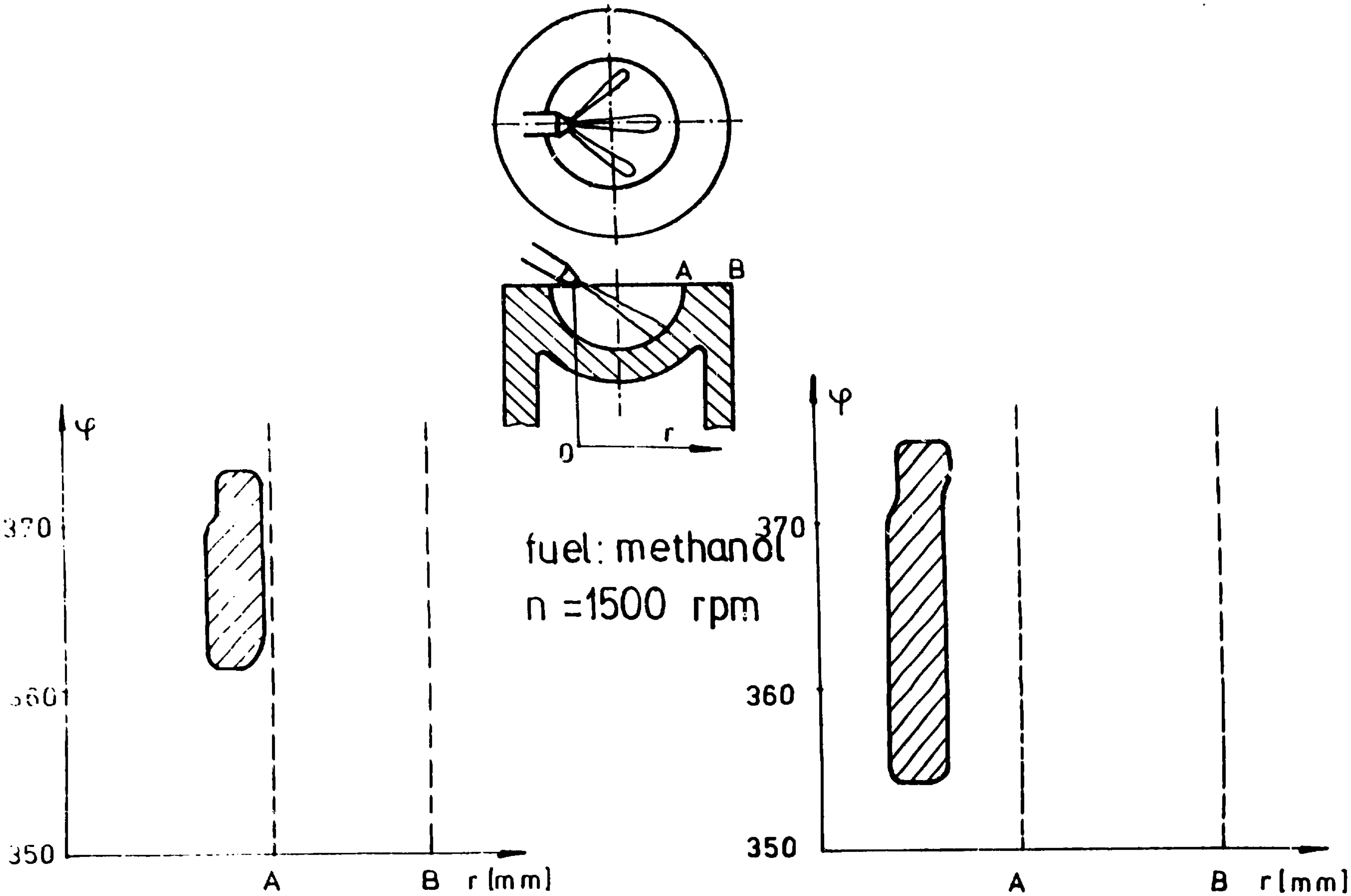


Fig.16 As for fig.11, methanol