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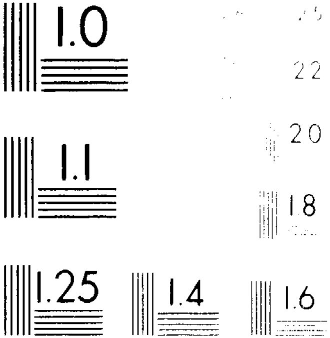
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APPLICATION OF ALTERNATIVE FUELS FOR INTERNAL COMBUSTION ENGINES, IIP, DEHRA DUN .

DP/IND/82/001

INDIA

14245

TECHNICAL REPORT *

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

Based on the work of Anton Cernej
Expert in fuel introduction in internal combustion engines
under the post 11-02

FROM
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TOPICS REQUIRED

1. Proposals concerning methyl fuel use in vehicular 4-stroke engines. Power outputs range 75 ÷ 110 kW NA-DI version only. Piston dia!s range Ø95 ÷ 105 mm with open bowls:

   - ω typ
   - cylindrical DB bowl typ

   The most interesting aspects:
   - ignition
   - performancies
   - outputs

2. Dual fuel operations with mixed D2-CH_3OH composition. Here the mixing procedure is required as well as proposals for approach.

3. HP injection related to methyl-fuel diesel engine operation. The benefits and potential improvements using DE-IHM-KHD-IAM new results.

4. Piston engine mechanics and balance. Approach to calculations.

5. Approach to piston bowl design for DI-diesel engine taking into account: injection performancies swirl, squis as well as geometrical relations.

6. Needle lift measurments and practical approach to fuel injection calculations.

7. Experimental approach to fuel atomization in bench test experiments. Screening effects and differencies between Sauter mean droplet dia's defined by calculation and by practical bench test experiments.

8. Comment and proposal pertinent IIP (7 Pages) Report worked out by D.Kumar and P.P.Pundir.

9. NOx, THC-THCC measurments.

Topics point 2 and Point 6 were as more urgent selected by IIP for the begining of the common work.

Topic 2

Up to 30 % neat CH_3OH may be added to diesel oil without addition of a combustion improver (Hexanolnitrat, Amylnitrat). However, because of water presented, starting, warm-up period, idling, long term low-load operation in city-drive application, stops intervals and transients the above cited percentage may be reduced on 20 %. Thus for vehicular engines, combining other fuel saving techniques with 20 ÷ 25 % methyl fuel doping, 20 ÷ 25 % diesel oil in-field consumption rate may be reduced.

Practically for vehicular application, without: glow plug assistance combined with ceramic comb. chamber isolation (ATI) or spark ignition assistance, only 20 ÷ 25 % of methyl fuel may be considered as a max. amount, which may be mixed with D2 without ignition improver. Glow plug-ceramic isolation -- and spark plug techniques will be discuss in TOPIC 1. Here, dual fuel injection technique was omited because of a high first cost. MAN pressure distributor technique will be once again touched in TOPIC 1 (See the first report, Černej A., 1983.). One new approach with glow plug ignition and methyl fuel doping control in dependence of the HP pump control rod position will be shortly explained in TOPIC 1, as well.

According to the forementioned in the TOPIC 2, the attention will be paid to dual mixing technique only. Thus, up to 25 % by volume methyl fuel mixed with diesel oil is considered.

1. The system for supplying methyl fuel/gasoil

Fig.1 shows a diagrammatic sketch of fuel mixing system proposed.

Potential problems:

1.1 - relatively modest decrease of diesel oil consumption
1.2 - first cost increase
1.3 - because of methyl fuel tank, dead box increase and two different fuel to tank in.
1.4.- unknown reaction of specified engine (CI, swirl, bowl ..)
1.5 - unknown methyl fuel water content
1.6 - vapour formation (cavities) in fuel pump (pump gallery) and in low pressure system may produce injection irregularities
1.7 - starting (se pos.5+6 or/and JX) and warm-up periods
1.8 - service life of HP pump
1.9 - fuelling increase for the power output given
1.10- turbo-charged version is omited
1.11- THC and THCO exhaust emission increase
1.12- Vapour lock between fuel tank and LPP (Fig.1)

Because of very short time on desposal only, it's the best way to start with the problem denoted under 1.4.

Problem 1.4 - unknown reaction of specified engine (related to points 1.2, 1.5, 1.6, 1.7, 1.9, 1.11 also)

For the engine selected, when burning methyl fuel - diesel oil blends, its reactions have to be investigated at first. It is not because of obtaining ideas about the best blending rations only. Morever, in order to decrease the lab- and the engine first cost (what must be changed ?) as well as to save the research time, preliminary organized engine test bench experiments may not be omited.

For the sake of forementioned we need:

a - simple prepared fuel blends
b - on test bench experimental programme
c - selected instrumentation on desposal
d - evaluation criterions

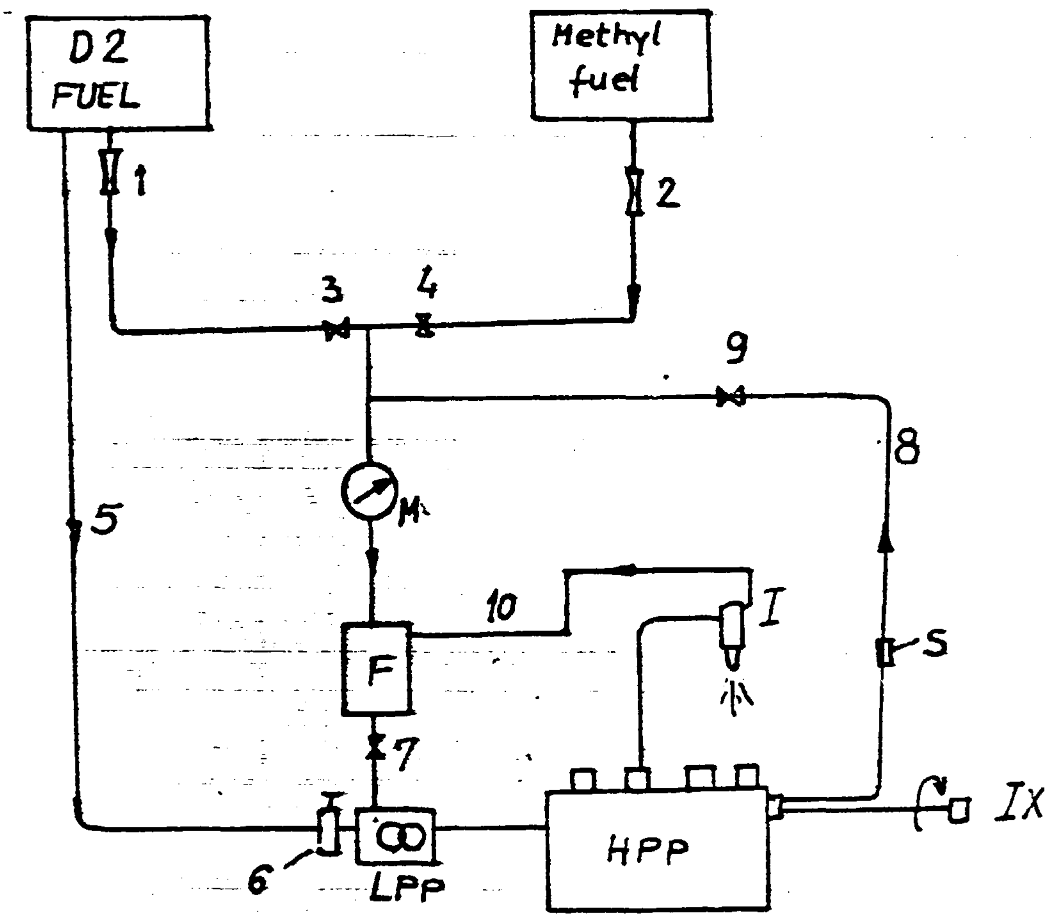


Fig.I Diagrammatic skech of dual fuel
mixing and injection system

1,2 - flow control
3,4,7,9 - single direction flow valve
5 - D2 fuel supply line for starting (alternatively)
6 - existed small hand pump
8 - mixing line
10 - injector leakage overflow line
M - pressure gauge (during experiments needed only)
S - sample checking (during experiments needed only)
F - fuel filter
LPP - existed low pressure pump
HPP - high pressure injection pump
I - injector
IX - mixer (alternative for Pos.5)

Point 1.4-a. Provisional blends preparation

In a simple fuel container (see Fig.2) selected components to be blended, have to be tanked. The amounts of the single "fuels" mixed is recommended as follows:

	CH_3OH % by vol	DIESEL OIL % by vol.	WATER %by vol
Group A	20	75	5
	20	70	10
	20	65	15
Group B	30	65	5
	30	60	10
	30	55	15

Unfortunately, it is very well established fact that CH_3OH will not go into solution with diesel fuel, especially not in the presence of water. However, an on-board emulsifier can produce unstabilized methyl fuel - diesel oil blend (Fig.2).

Emulsifier (Pos.3, Fig.2) use normally chemists in their lab's and it may be borrowed for short time. The fuel in the tank (Pos.4, Fig.2) must be kept at reasonable temperature, what can be done with water cooling flow (Pos.1, Fig.2). Just a narrow plastic tube (Pos.5, Fig.2) serves as a vent pipe stretching out of the lab. To prevent (vapour) cavities the distance between low pressure pump (Pos.9, Fig 2) if existed, and fuel tank (Pos.4) has to be short.

Fuel temperature in the pump sump can be high as 65 °C and beccause of known phenomenon during spilling and fuel flowing into the barrel (Fig.3), the pressure flucuations may produce cavities, thus injection irregularities and the measurements in error. For the information Fig 4 shows CH_3OH vapour saturation pressure vs. temperature as well as for gasoline. Only to menttion, Fig 4 demonstrate also clearly, the cold start problem when convert to methyl fuel.

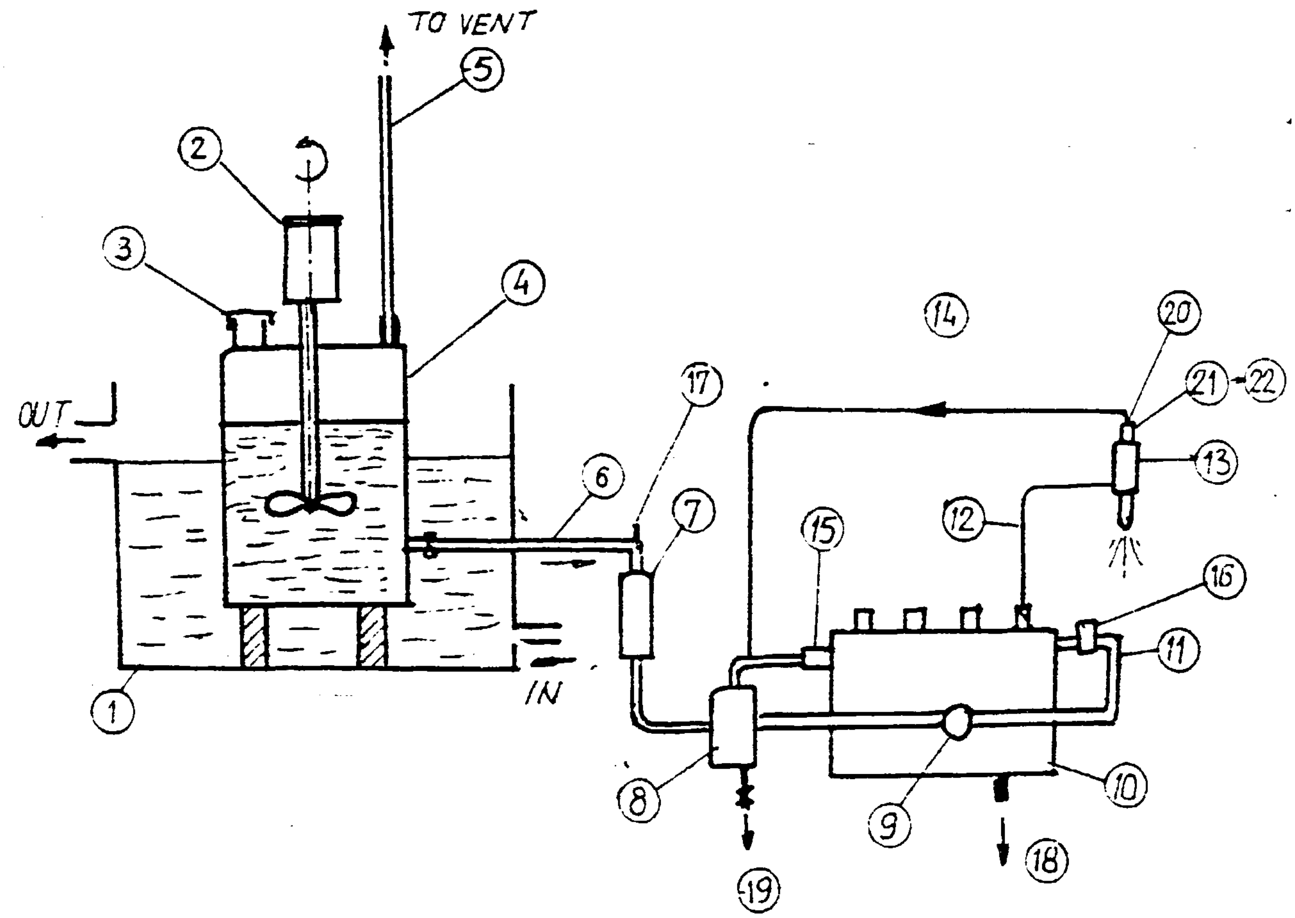


Fig 2 Diagrammatic skech of on-board emulsifier

1 - water tank
2 - emulsifier
3 - sealling cap
4 - fuel tank
5 - plastic tube Ø8
6 - tube
7 - fuel consumption measur. device
8 - fuel filter
9 - low pressure pump
10 - high pressure pump
11 - tube
12 - HP tube
13 - injector
14 - overflow leakage tube
15 - pressure control valve
16 - pressure gauge
17 - thermometer
18 - oil sample
19 - fuel sample
20 - inductive needle lift measurement
21 - induct. bridge
22 - registration

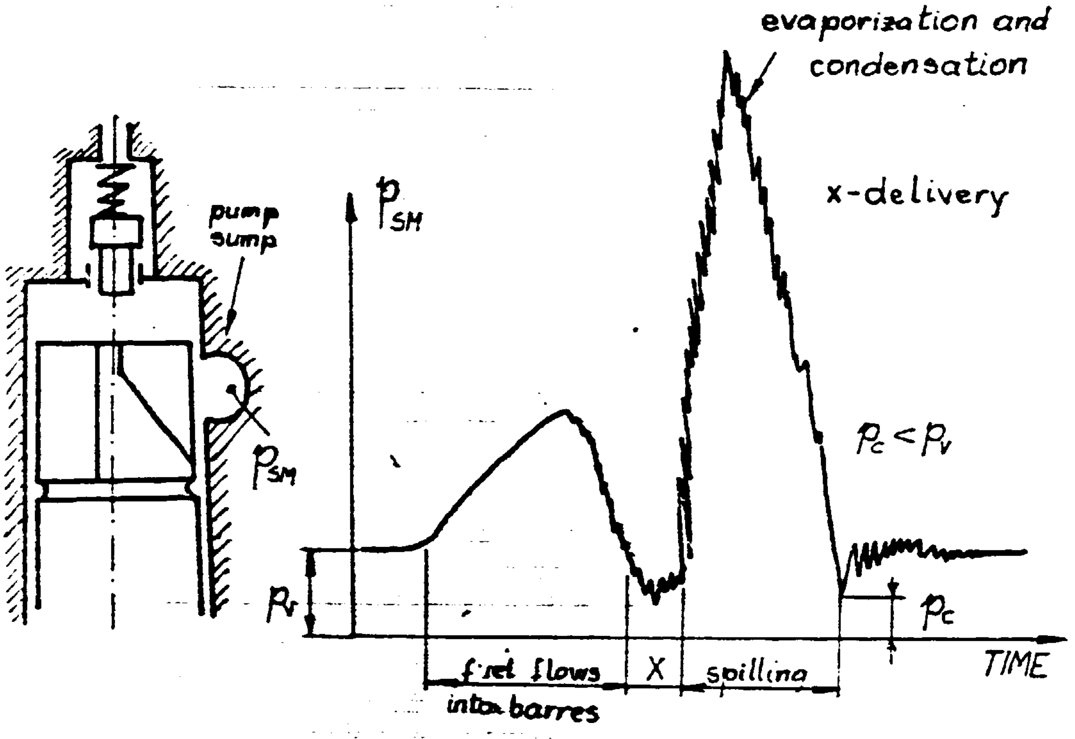


Fig 3 Effect of spilling and filling on cavities formations in the sump, vs. time

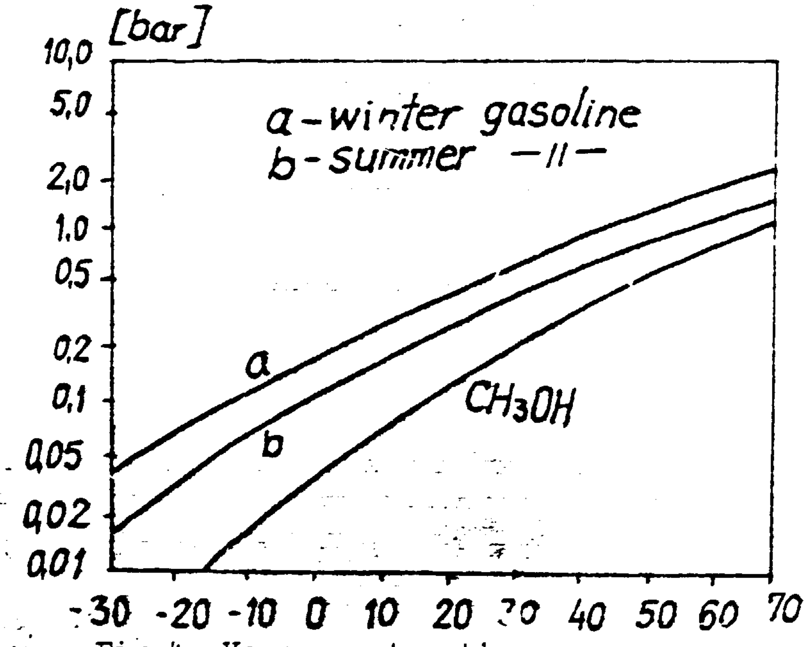


Fig 4 Vapour saturation pressure vs. temperature

Fig 4 explaines very clearly, that the valve (Pos.15, Fig 2) may not be omited, but its opening pressure has to be adjusted at p_V=0,15 - 0,7 bar above the ambient one. On the contrary, if the low pressure pump does not existe the supply pressure valve (Pos.15) could not be installed and provisionary measure must be undertaken to cool the sump body from outside or to press fuel air. In the later case is the best way to compress the fuel tank (Pos.4, Fig2) taking into account the fuel measurement device.

To the HP pump (Pos.10, Fig 2) and to the injector (Pos.13) special attention must be paid. At first to be sure, that fuel mixture tested corresponds exactly to the wanted one, therefore the fuel sample checking (Pos.19). To have idea about leakage (in order to avoid expensive redesign) sample (Point 18, Fig 2) is very useful. Fuel consumption measuring device (Pos.2) must be of high accuracy with error of repeatibility less than 1 % (under all testing conditions).

The HP pump and injector itself, must have surplas of capacity in order to ensure a resonable duration of injection and satisfied fuel atomization for the all fuel mixture tested. The later can be done by readjustings or, if need, by changing the components of injection equipement.

With the first fuel chosen the injection parameters have to be investigated but with the fuelling changed. It can be done on the engine in operation under next conditions:

- needle lift registered
- TDC marking
- pressure before injector
- accurate measur. of fuel consumption rate
- pump rod position measur. for loads and rotational speeds given.

The later way is recommended. Namely, the afore cited measurements accomplished with in-cylinder pressure diagrammes can enable the later unavoidable analysis on the whole. Besides that, we have the posibility to avoid the missmeasurements, for example: afterinjection, if happend, we change retracting delivery valve at once.

It is also very useful some precalculations, to obtain the idea about the fuelling and other injection parameters.

For precalculations approach we need:

- supposing the same engine efficiency we calculate the fuelling for our mixtures chosen. Here, for outputs given D2 fuel consumption rate must be known
- D2 calorific value must be known, per example in the Europe D2-O. ranges 42750 - 42500 kJ/kg.
- neat methanol has lower CV than D2

	kJ/kg	kJ/m³	kg air stoichiometric / kg fuel
CH_3OH	19600	15490	6,46
D2	42600	35600	14,3

If we are going to add 15 % water to methanol we decrease the calorific value. Per expample, adding 15 % water to CH_3OH we have:

	kJ/kg	kJ/m³
CH_3OH + 15% H_2O (methyl fuel)	16510	13640

(It is assumed in Germany that the water content in methyl fuel in dependence of the production, may reach (till) 15 % by volume).

Thus we can now calculate the fuelling for every mixture given.

- Using short programmes for fuel injection calculations (Programme I and II given IIP) we may traced the potential errors in before hand.

Point 1.4 - b. On test bench experiments programme

It may be suggested the bellow programme presented:

Starting

Calorific value per volume of diesel oil is 2,2 times per volume higher than the value of CH_3OH. Relative to diesel fuel one liter of neat methanol requires 4,4 times more heat of vaporisation. Since the large fuel flow and high vaporization heat figures, methanol needs 9,8 times more heat than diesel fuel to evaporate. Besides that in the presence of water ignition temperature increases onward. Thus, neat methanol (latest data) approaches Cet.No 3 but adding water 10 % by vol. to CH_3OH Cet.No ranges about 1,5 ÷ 2. Ignition temperature increases with water content increased:

CH_3OH	ignition temperature (stoichiometric)	478 °C
CH_3OH + 5% water by vol.	"	480 °C
CH_3OH + 10% water by volume	"	485 °C
CH_3OH + 20% water by volume	"	490 °C
CH_3OH + 30% water by volume	"	510 °C

All above mentioned effects drasticly the starting process in CR engine. One of the important goal, dealing with CH_3OH, in FRD noted:

increase of self ignition property.

Thus in our experiments we may not neglected this fact. It means we have to compare the ignition quality for the specific engine, when fuelled with D2 and with our mixture selected.

One of the approaches to study starting is more time consuming but enables very deep insight into the whole process. Morever for the friction analysis such approach may not be omited (not our subject). For IIP as a scientific institute is to be recommended (perhaps for the next future). Forementioned is the reason to show the whole procedure shortly.

To study starting. Approach 1.

Instantaneous effective torque may be written as follows:

$$m_e = m' - (m'' - m''' - m'''')$$

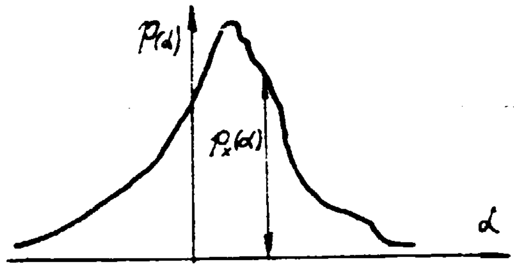
where:

$\mathfrak{M}'$ - instanteneous gas pressure torque

$$\mathfrak{M}' = \frac{\pi \cdot D^2}{4} \cdot p_x(\alpha) \cdot \frac{\dot{x}}{\omega}$$

$$\dot{x} = r \cdot \omega \cdot \left[\sin\alpha + \frac{\lambda}{2} \sin 2\alpha \right]$$

$$\lambda = \frac{r}{l}$$



ω - instanteneous angular velotity

During start period engine is disconnected, it means $\mathfrak{M}_e = 0$

$\mathfrak{M}'' \approx 0$ instant. potantial energy torque

Thus:

$$\mathfrak{M}''' = \mathfrak{M}' - \mathfrak{M}'' \qquad \text{inst. frictional torque}$$

$$\mathfrak{M}'' = \frac{1}{2}\omega^2 \cdot \bar{R}' \cdot \sin 2\alpha + \dot{\omega} \cdot \left[\bar{R} + \frac{1}{2} \cdot \bar{R}' \cdot (1 - \cos 2\alpha) \right]$$

$$\bar{R} = k^2 \cdot m + r^2 \cdot \left(1 - \frac{s'}{l}\right) \cdot m' \quad ; \quad \bar{R}' = r^2 \cdot \left(\frac{s'}{l} \cdot m' + m'' \right)$$

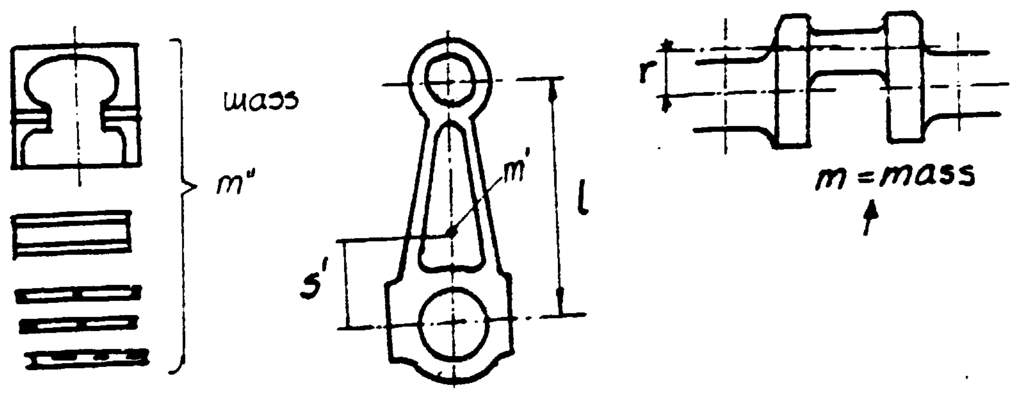


Fig 5 Notation

Storing $p_x(\alpha)$, ω and $\dot{\omega}$ for every cycle during starting we may:

- follow the ignition stability
- follow firing fouling
- to adjust properly advancing

- follow effect of switch-on diesel fuel starting
- calculate the frictional instant. losses

(For much more information see TOPIC 4).

To study starting. Approach 2.

Comparable starting period measurements with D2 fuel and mixture selected under the same ambient - and adjusting conditions. When reached rated speed the start period is accomplished. The best way for such measurements is the pen recorder trace registration of engine rotational speeds.
However, for different fuels may not be avoided:

- the advance readjusting
- start fuelling change

to find out the shortes start period. In the case of too high $dp_x(\alpha)/d\alpha$ or/and p_{max} in-cylinder the selected advance of injection may be kept during starting and warm-up periods only. The solution for changing the beginning of injection in engine operation may not be difficult. However, the first price increases.

To study warm-up period

During warm-up period may be recommended:

- to observe in-cylinder pressure diagramme to follow p_{max} or irregularities
- to measure THC exhaust emission (when possible equivalent C_3H_8 diluted by air during calibration). Here to be mentioned: dealing with CH_3OH in warming-up period the next correction is valid:
  every C-atom in THCO produces error in reading by factor 2/3

Again comparable measurements are needed.

The best way is to find out, by means of startings and warming-up periods, the max possible CH_3OH content or/and to analyse for potential improvements. Per example: swirl ratio decrease may help to improve starting, change in nozzle orifices distribution or/and dia's may control the rate of evaporization and with this, the rate of temperature decrease before ignition, here is no doubt that the compression ratio (CR) increase may help to greater extent for better startings. Morever, with CR increase we only compensate the "lost" because of CH_3OH dopei. CR = 16 may not be applied with CH_3OH content of 30 % by volume. Probably CR=18 - 19 with swirl decreased is more promising way for higher CH_3OH + water contents and engine may benefit from high compression, assuming reasonable max in-cylinder pressures. (Before ignition neat methanol/air stoichiometric mixture drops in-cylinder temperature for 122 ^{o}C. It means that we have to increase CR up to 25 which produces a high in-cylinder pressure).

Max. in-cylinder pressure may be diminished by prolongated injection and with this decreases the temperature drop of combust. mixture as well.

Combining:

-increases CR

- decreased SR (swirl ratio)

- prolongated injection period

we may increase methyl fuel content without first cost increase.

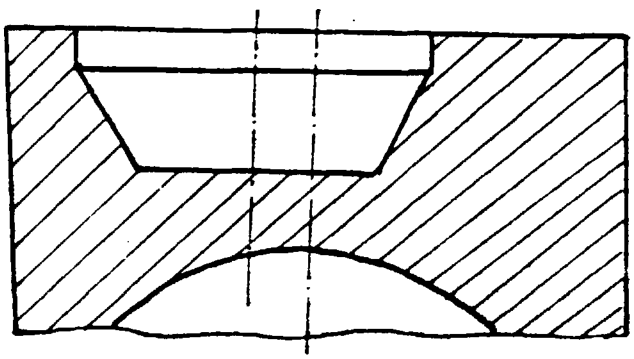
Starting and warm-up period are turning points for ahead investigations. However, in the case of low possible content of methyl fuel because of starting (warm-up satisfactory), some other means are also on desposal, per example, diesel oil start, but it complicate the operation via switching-on technique.

To study engine performancies

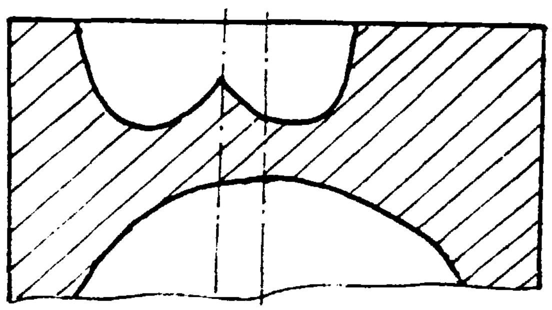
Programme related performancies is stringent dependent on engine application. Thus, the same programme could not be applied for vehicular engine and for engine coupled with water pump.

IIP intends to apply CH_3OH + water + D2 mixture on the next two vehicular engines:

Make	A	B
Combustion chamber	open	open
Bore x stroke [mm]	Ø92x120	Ø105,38x120,65
Swept volume [dm^3]	4,78	6,075
In row	6 cyl.	6 cyl.
Compression ratio	1 : 17	1 : 16
Rated power [kW]	82,35	80,88
Typ	NA	NA
Rated speed [rpm]	2800	2400
Mean eff. pressure at rated power [bar]	7,38	6,66
Minimum fuel consumption rate [g/kWh]	265	279
Minimum fuel consumption rate at [rpm]	2400	2400
Fuel consumption rate (calculated) [g/kWh]	283	298
Lowest loading (calculated) [rpm]	1400	1200



Comb. chamber A
(old version Daimler-benz)



Comb. chamber B
(old version Lyland)

Fig 6

Injection pump MICO
(BOSCH - Typ A)
Injector 4 holes MICO
(BOSCH DLLA 150S 187)

Injector 4 holes

Looking at the data collected it could not be avoided to menttion, engines selected are remained behind in developement. Morever, the fuel consumption figures look very disappointing. It seems to be resonable at first to reduce the fuel consumption rate in diesel oil operation. Per example:

engine A has 4 piston rings

engine B has even 5 piston rings

Piston may be also changed related to its height. Applying 3 piston rings, reducing the piston height and changing the skirt ovality, the both, first cost and fuel consumption rate may be decreased. It is especially truth at higher piston velocities (than 9 m/s), having in mind that 75% of the heat friction losses is related to piston group.

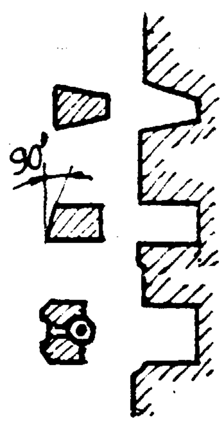


Fig 7

Fig 7 shows the ring set which may be recommended.

Here is only one very small subject touched but certainly must be more than one. Just comparing with other up-to-date engines may be concluded that fuel consumption rate is to be reduced up to 15 %, turbocharging and speed decrease out of consideration.

However,

- rotational speed decrease with piston stroke increased
- turbo-charging
- swirl and squish decrease with injection pressure increase and less dependent of rotational speed

are characteristics of the modern vehicular engines. Morever, dealing with vehicular engines, better matching to the vehicle demands may save still more fuel.

With above cited potential fuel reduction in mind, may not be exaggerated to put the question about the course of investigation. To save 10 ÷ 15 % D2 fuel with mixing technique with first cost increase or at first to reduce the high diesel oil consumption rate and to obtain more modern competitive engine?

In any case to obtain performancies - for decision - the next programme may be proposed:

1. Full load characteristic
2. 75 % of full load characteristic
3. 50 % of full load "
4. Idling stability

Besides standard data measured, THC and soot measurements have to be included.

Evaluation criterions. Point 1.4-d

The most important criterion is to estimate the benefit of the quantity of diesel oil saved via blending and compare it with:

- first cost increase
- complications with dual fuel system in practical service
- potential diesel oil using other techniques
- possibility and troubles related to methyl fuel tank location to existed vehicles
- reasonable fair engine characteristics matched to vehicle operation demands (startability, polution, fuel consumption, transients, torque back up, stability, oil degradation and change interval of the oil, service life, reliability in operation).

For the information:

1. Investigations of DFG (IHM), BMVi (se AIF - No5074) showed that up to 10 - 30 % of CH_3OH (engine dependent) may be added to diesel oil, the larger quantity may produce starting, missfiring and knock problems. (See also MTZ 45, 1984).
2. Ricardo News "World wide Engine and Fuel Relationship" reported:
   "Up to, say, 30% could be used by blending with diesel fuel or in a dual fuel engine where the alcohol was carburated into the cylinder and the mixture was ignited by an injection of diesel fuel. Alternatively, ignition improvers additives, such as isopropyl nitrate may be used, but the quantities required, up to say 20 % make the process prohibitively expensive. The best solution would seem to be a multi -fuel

engine such as MAN FM which, on an energy basis, when operating on pure alcohol fuel, would give a thermal efficiency equivalent to that of an IDI diesel engine"

3. "Instituto di Macchine e Technologie Meccaniche" Trieste (Italy) reported:
   "Test have been made on a serial truck engine running on a gas-oil/methanol mixture containing up to 30 % methanol , with and without the addition of a combustion improver. Max. ranged 35 % CH_3OH by volume..

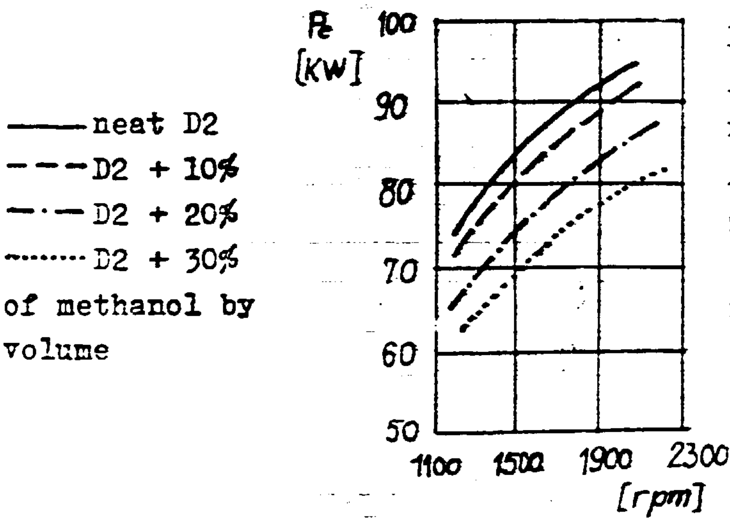


Fig 8 Performance of the four-cylinder engine running on different gas-oil/ /CH_3OH mixtures (truck engine in regular production). Data obtained without fuelling increase.

Conclusions of "Instituto di M. e T.M.":

- without fuelling increase, addition of CH_3OH produced power output drop and specific thermal consumption increase
- adding ignition improver and more than 35% of CH_3OH by volume, the results obtained showed that besides of fuel cost increase, the startability was not improved because of a very high CH_3OH cooling effect. Thus, ignition improver effect was canceled by cooling
- additional reason for higher specific thermal consumption rate when mixing methanol was it cooling effect, which does not increase the volumetric efficiency in CI-NA engine.

  (See also MTZ 44 (1983) 1)
  (Using turbo-charged version CH_3OH may be used for air cooling and in this way to increase the volumetric efficiency. Morever, charge-air cooler may be omited which produces: the first cost decrease and also box volume may be reduced).

Topic 1

It was mentioned earlier that CH_3OH need, for the same diesel engine power output, nearly by factor 10 more energy for mixture formation than that in diesel fuel operation.

However, is more useful to consider the mixture formation history for stable ignition, as well as for low load operation. When comparing diesel fuel and CH_3OH related to CI engine operation problems. The coarse comparison results in:

- diesel fuel may be ignited at CR=15, without any problem, supposing a reasonable low SR ratio
- neat methanol, for the same combustion chamber, asks CR=25, what is too high because of high in-cylinder pressure. Again the difference between diesel oil operation and neat methanol approaches 10 but now in CR ratio. It would not be difficulty to explain this evidence.

In order to ignite the neat methanol in diesel engine is only to combine:

1. To compensate the temperature before the ignition related to temperature drop because of high heat of vaporisation.
2. To decrease the heat tranfer in the preignition period
3. CH_3OH vaporization control in preignition period
4. To start injection in more fovourable ambient for ignition
5. To conserve more heat in the combustion chamber via isolation
6. Ignition pulse from outside

However, combining 1-2-3-4-5 A
or 1-2-3-4-6 B

may produce successful ignition in neat methanol operation. Morever, method B is much more within reach than method A because of the next reasons:

- to conserve more heat, ceramic isolation is unavoidable, but we have not the heat on desposal before the ignition. However, heat may be accumulated during starting.
- the self ignition with very sensitive fuel is not easy to control under all operation conditiones. Per example knock operational problem in heavy load at high ambient temperatures.

- ceramic components are still our future

However, we must confess, that the method A is less expensive related to the first cost and may be more releable in the services.

Forementioned depicts that the both A and B method have to be considered. B for the first neat methanol use in diesel engine application and A method as a following one.

Is to be mentioned, the suggestions given in the TOPIC 1 are restricted related to:

- IIP considers NA version only
- only single fuel, neat CH_3OH has to be tanked
- the attention will be paid to the aforecited method B

Point 1

The net heat of evaporization of methanol approach 1110 kJ/kg and that of diesel fuel 250 kJ/kg only. In order to ignite neat methanol in diesel DI engine, our experimental results showed, that CR ratio must be at the least 25 supposing cold-starting ability at reasonable low ambient temperatures also. We apply z combustion chamber and CR=17 in diesel fuel operation with NA version.
It means that up to 8 units the compression ratio has to be increased. To apply CR=25-26 the main drawbacks is a high mechanical loading.

However, to support the ignition some compensation of temperature drop is still desirable. Morever, the net compression pressure will not be increased although the geometrical CR ratio becomes higher. As was jet recommended in the first report (1983) CR may be enhanced for up to 2 units without mech. loading increase.

Point 2

To decrease the heat tranfer in the preignition period, the next may be suggested:
- swirl ratio (SR) decrease (it was also jet recommended in the first report)
- light designed cast-iron piston

It is no doubt, that converting to neat methanol, SR ratio has to be decreased. Doing that, results in:

- reduced air motion save more heat of compression into combustion chamber and with this the temperature before ignition increases
- the ignition stability will be improved
- with SR reduced volumetric efficiency increase in NA version, more power output and higher compression temperature because of more air introduced
- less thermal loading of the parts formed the combustion chamber
- better control via injection

However, the rate of SR decrease must be considered with POINT 3 together. It is not reasonable to decrease to much SR ratio with spark plug supported, but still in this case related to diesel fuel operation, SR ratio may be reduced for at the least 15 ÷ 20 %.

It is well established fact that cast iron posses less thermal conductivity than that of Alu alloy, and with this before the ignition more heat may be saved. Light modern design of cast iron piston with cut-nonthrust skirt sides and with three piston rings set (1st keystone-and 2nd tapered 90' compression rings, spring loaded elastic oil ring) may not be much heavier than of Alu-alloy one.

Point 3

Methanol prevaporization control means to control the fuel quantity-injected into combustion chamber or/and evaporated before ignition.

It may be done by means of fuel film dispersion on combustion chamber wall, discontinued needle lift (still in developement), and modified cam shape of HPP camshaft.

Point 3.1 and Point 6

Fuel film deposition on combustion chamber wall is well known method initiated by MAN with M-process. Further developement of M-process and its modifications are well known till

now also, thus no need for a background.
The fuel deposition on in-piston bowl wall is connected with spark-plug igntion. This is the reason, that in this point the spark-plug ignition procedure may not be omited.

Fig 9 shows well known MAN combustion chamber of FM engine.

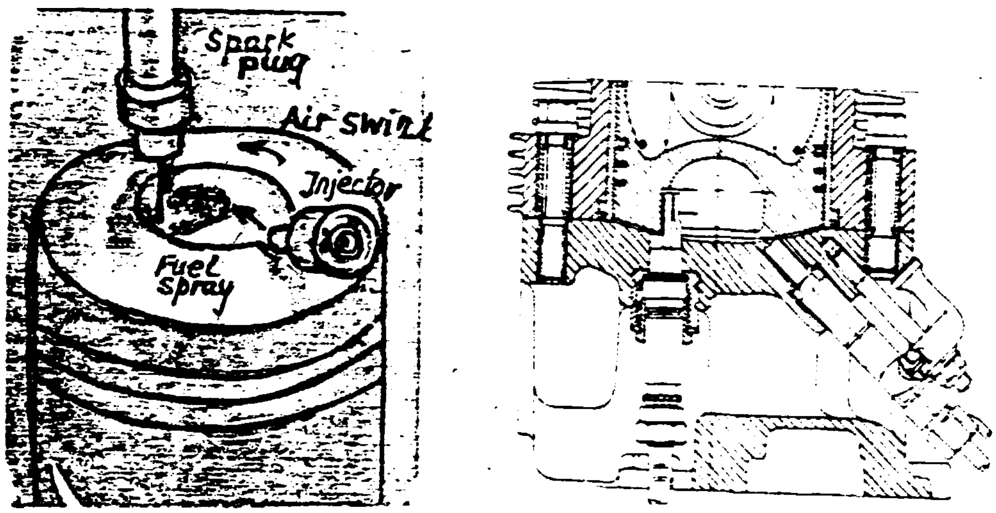


Fig 9 Combustion chamber of engine L 9204 FM

MAN-FM combustion system has been derived from the diesel engine and is excellently suited for operation on methanol or other alcohols, because this hybrid system benefits from high diesel compression, direct injection and nonthrotling output control while ignition of the mixture is ensured by spark-plug as in the otto-cycles Fig 9 (Method B in this report).

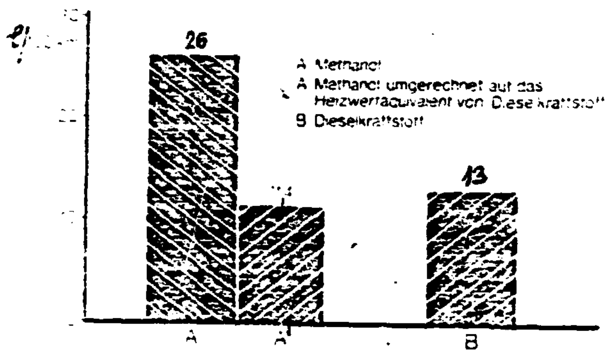
The control mechanism in the proposed method B is projected on separation between injection- and mixture formation functions. To do this (during injection) fuel was deposite in the wall of the spherical chamber. The spark-plug is situated on the opposite side of the injector and its electrodes reach the fuel film zone.

The most important features of method B may be the follows:

- CR ratio as for diesel engine
- direct injection with single hole injector
- wall fuel deposition

- spark plug ignition
- range of piston dia. Ø85 ÷ Ø130

One part of the heat of evaporization is taken from combustion chamber wall in engine operation. It means that one part of the heat transfered to the wall comes back in the process increaseing its efficiency. Thus, converting to methanol, broke th rmal efficiency must be better than in diesel fuel operation. To support this statement the Fig's 10 and 11 are shown.



A - methanol
A' " equivalent (termal)
B - diesel fuel

Fig 10 Fuel consumption of 2,1 tons vehicle with methanol engine L9204 FM compared with diesel engine L9204 M, constant speed 60 km/h. Methanol thermal equivalent fuel consumption decreased for 14 %.

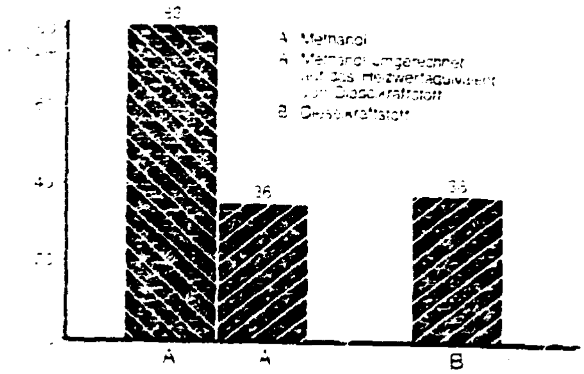


Fig 11 Fuel consumption of MAN city Bus with methanol engine D2566FMUH compared with diesel engine D2566MUH, urlem cycle. Methanol thermal equivalent fuel consumption decreased for [illegible],5 [illegible].

The above Figures depicte:

1. At higher loading more fuel saving with methanol. It conclusion supported the heat come-back in the cycle.
2. Converting to methanol fuel, saving may reach up to 10% in general application.

The heat drop cause by methanol vaporization may be used for mean effective pressure increase or/and for better matching to the vehicle demands.
Fig 12 support this statement.

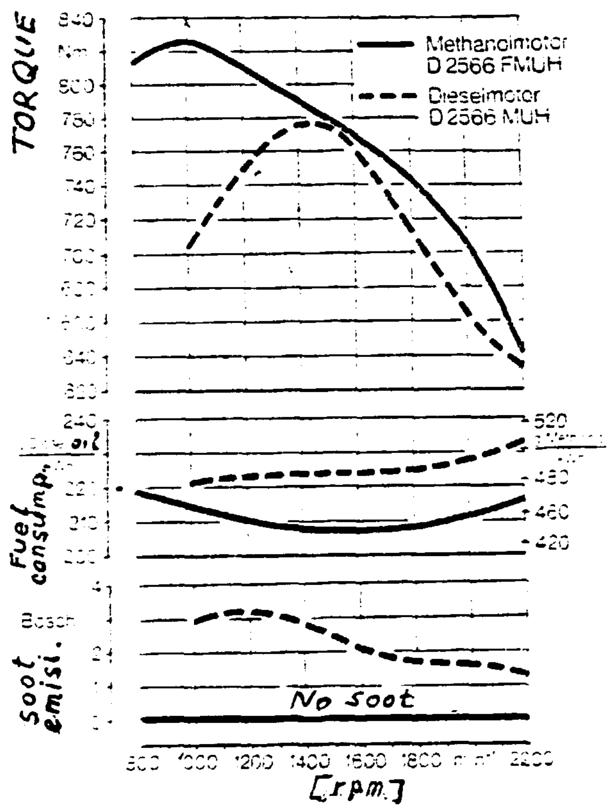


Fig 12 Full-load characteristics of methanol engine D 2566 FMUH compared with diesel engine D 2566 MUH

Fig 12 depicts:

1. Converting to methanol max. torque increases up to 6,5 % but its speed position is much more improved by 42 %.

2. Torque back-up in methanol operation reaches 30 % and in diesel fuel operation 22 %.
3. Sootless exhaust in methanol operation, still one support more, for torque back up increase.

Morever, converting to methanol the parts forming combustion chamber are less loaded, thus the service life of spark plug electrodes will be extended. Spark-plug is of specific design but all other ignition system is normal transistored high--tension ignition system, used in automobiles.

The bellow cited may be recommended:

1. Combustion chamber is spherical with small gaps for injector and spark-plug. For given piston dia. and CR=18 all other proportions are very simple to obtain.
2. Ignition system is normal as for modern engines used in automobiles.
   The exeption is the spark-plug which maybe purchased from BOSCH.
3. Single hole injector
4. Using the programme sent last year (1983, se the first report) fuel injection may be very accurately calculated. Morever, fuel spray - combustion chamber wall contact in the programme enables to follow the fuel deposition.

Point 3.2

In order to decrease the fuel quantity in the combustion chamber before the ignition needle lift event may be modified, Fig 12.
During injection at high loads, using system in Fig 12, the nozzle discharge areas may be changed, Fig 13.

In the preignition period the discharge area a and time b may be accomodated to the specific demands. In our case we may control the injection quantity before the ignition changing a and b, Fig 13. It means we control the temperature drop before the ignition also.

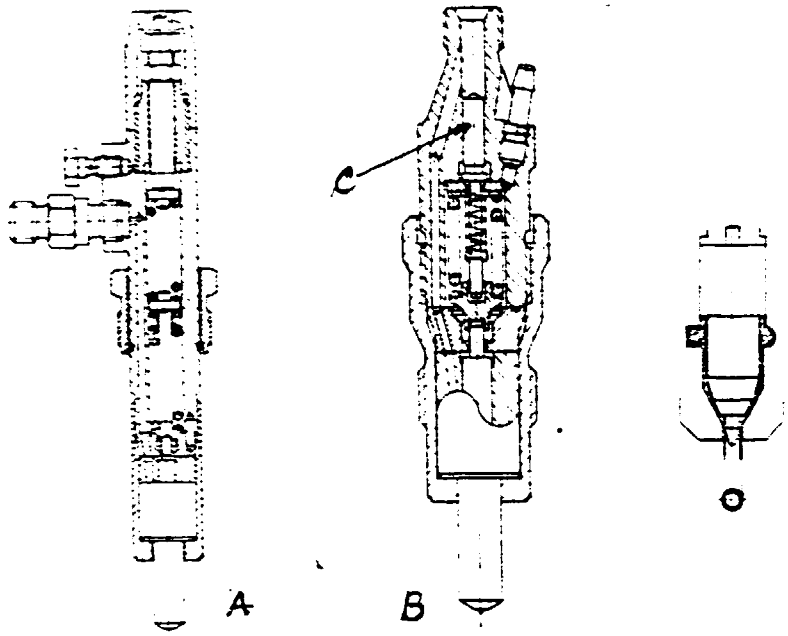


Fig 12 A - Two-spring injector, B - Bosch
injector with control plunger ,
C - pintle nozzle

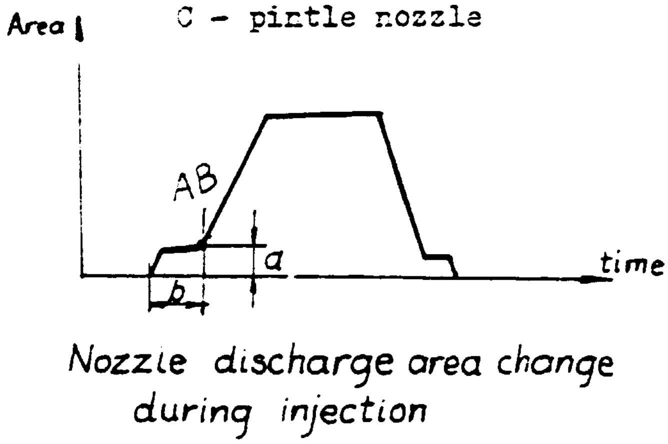


Fig 13 Nozzle discharge area change
during injection

In the system A, Fig 12 the bottom spring control the opening pressure, at AB point, Fig 13 the upper spring starts with compression. This system was developed by MAN. Bosch used other approach, B Fig 12, here a small piston C controls the point AB in Fig 13. The diameter of piston <u>C</u> is smaller than needle dia, thus the fuel acting on the both, controls with pressure the needle lift.

Morever, the system shown is very useful in low-load operation. Over-fuelling or inertia-supported fuelling is prevented. Because of low pressure levels nozzle discharge area remains small and the injection period becomes prolongated. At low-loads and low speeds the system shown in Fig 12 may be solution to avoid misfiring.

The same events may be followed by single nozzle.

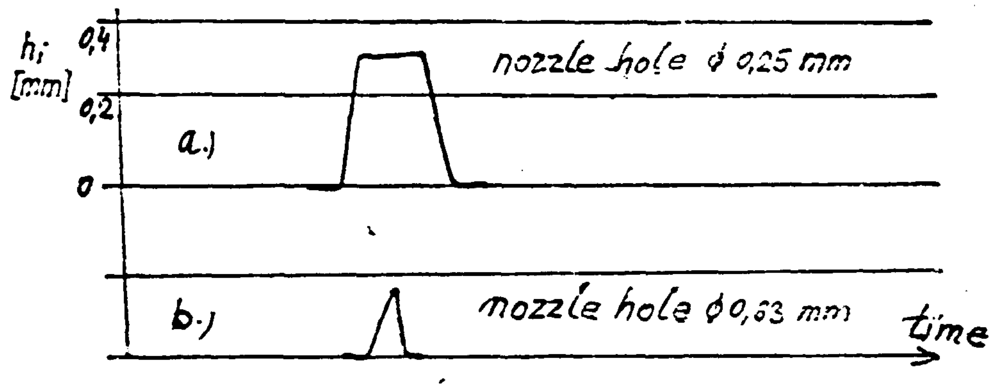


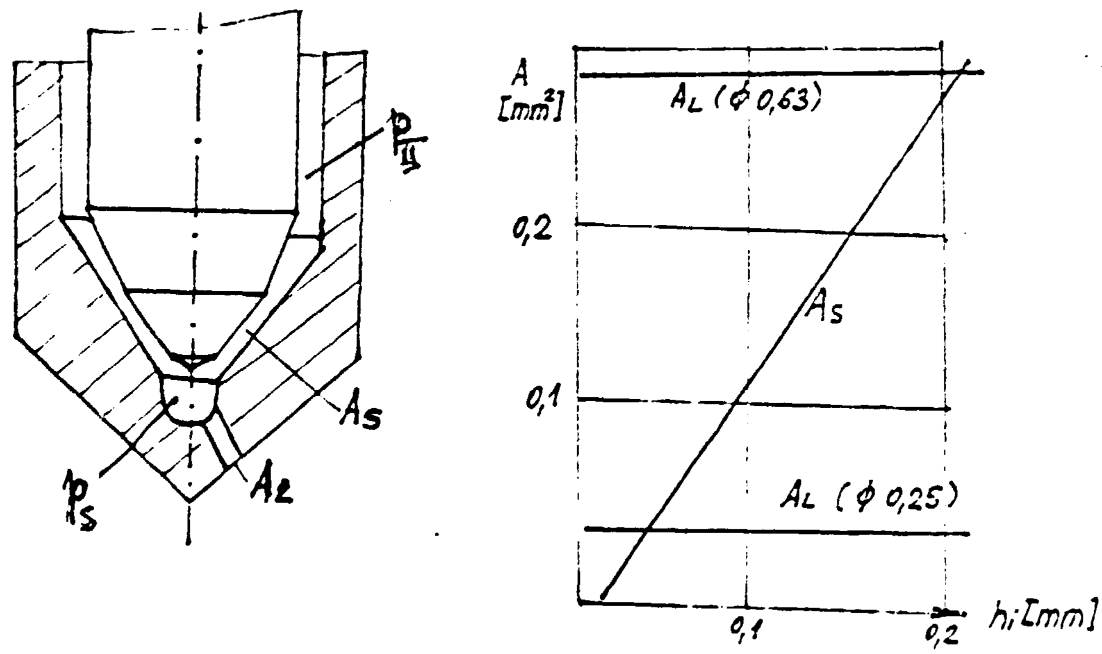
Fig. 14 Idling speed, fuelling 15 mm^3/cycle

a) simple-hole nozzle Ø0,25 mm

b) simple-hole nozzle Ø0,63 mm

h_i - needle lift

The reason for the needle lift change may be explained in Fig 15.



Taking into account Fig 15 and Fig 16 the self control HP pump - HP line - injector may be clearly observed.

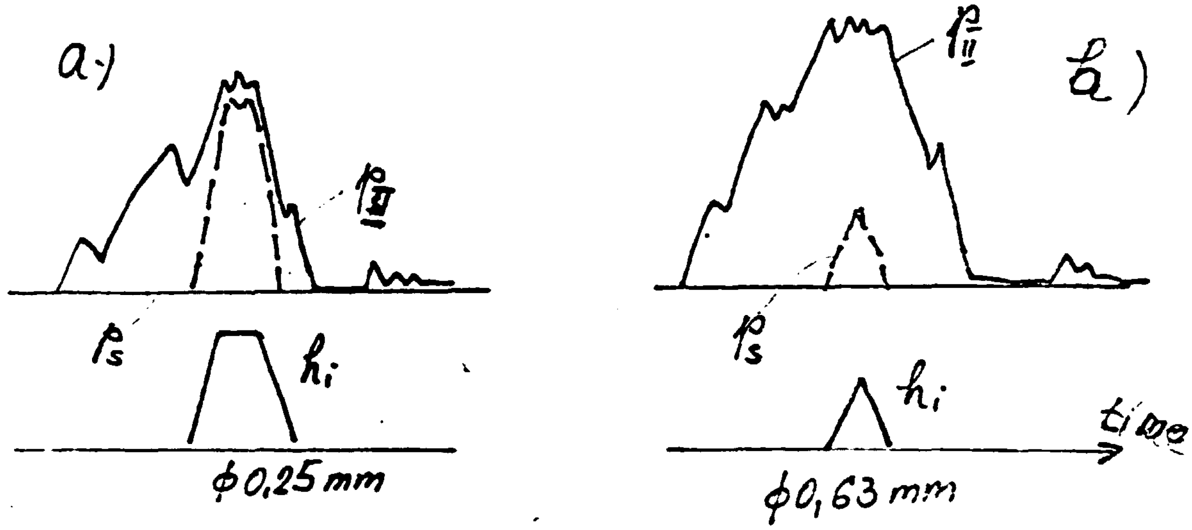


Fig 16 Self-control between p_{II}, p_s and h_i (se Fig 15). Note p_{IIb} p_{IIa}, thus i · p_{II} events a - b could not be compared related to amplitudes in Fig 16

The next expression shows:

$$p_{AS} + \frac{\rho \cdot W_{AS}^2}{2} = p_s + \frac{\rho \cdot W_s^2}{2}$$

where:

p_{AS}, p_S - are instanteneous pressures in the cross-sectional area AS and before the nozzle hole respectively

W_{AS}, W_S - are instanteneous velocities respectively

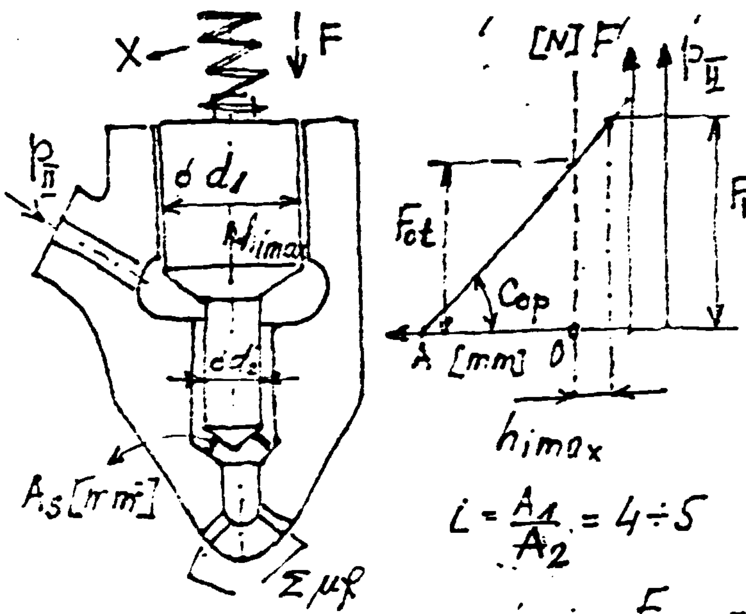
ρ - fuel density

Following the expression given the philosophy about needle lift control and fuel deposition may be explained.
Dealing with spray tip - comb chamber wall contact control via injector, the injector springs have to be adapted (Fig 12) to our demands:

X1 - injection period
X2 - area of fuel film deposition
X3 - history of fuel film deposition
X4 - preignition fuelling
X5 - injection pressure

(Points X1 ÷ X5 may be calculated only using Programme 1, sent IIF, see the 1st Report)

In the first Programme sent, relations related to spring are omited. Therefore the additional informations are given bellow (se also the book Cernej-Dobovisek).



$$p_{\bar{I}ot}(A_1 - A_2) = F_{ot} \quad \ldots(1)$$

$$A_s = 1.9 \Sigma \mu f \quad \ldots(2)$$

$$h_{imax}$$

$$C_{op} = \frac{p_{\bar{I}ot}}{h_{imax}} \frac{d_1^2}{4} \left(1 - \frac{1}{i}\right) \ldots$$

$$F_{max} = F_{ot} + C_{op} \cdot h_{imax} \ldots$$

Fig 17 F - spring force, F_{ot} - spring opening force
- effective flow-cross-sectional nozzle holes area

p_{ot} - injection oppening pressure
C_{op} - spring rate, A - area
i - hydraulic injector ratio

$$A_1 = \frac{\pi \cdot d_1^2}{4} \quad ; \quad A_2 = \frac{\pi \cdot d_2^2}{4}$$

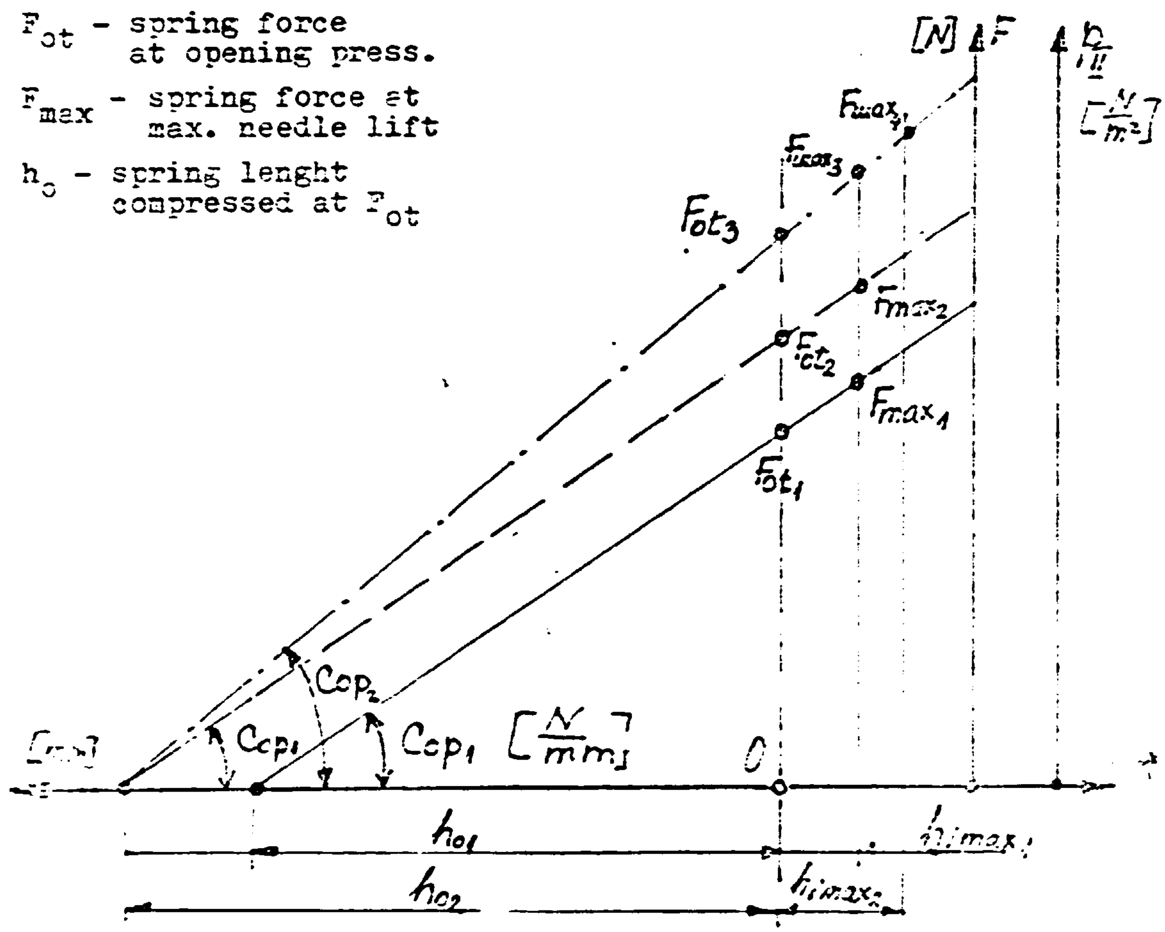


Fig 18 Spring force accomodation

Fig 18 shows:

- in order to change the opening pressure we change the spring precompresion

$$F_{ot2} - F_{ot1}, \quad F_{max2} - F_{max1}$$

- in order to change the both, opening pressure and spring rate increasing the needle lift also we have:

$$F_{ot3} - F_{ot1}, \quad F_{max4} - F_{max1}$$

Injector spring design.

Fig 19 shows Smith-diagramme for the spring wire steel of middle quality. Today max. quality approaches:

$$\tau_{max} = 65000 \ N/cm^2 \quad \text{and} \quad \tau_w = 35000 \ N/cm^2$$

Using the max.steel quality but middle quality in production we calculate:

$$\tau_{max} = 50000 \ N/cm^2 \quad \text{and} \quad \tau_w = \frac{\tau_{max}}{2}$$

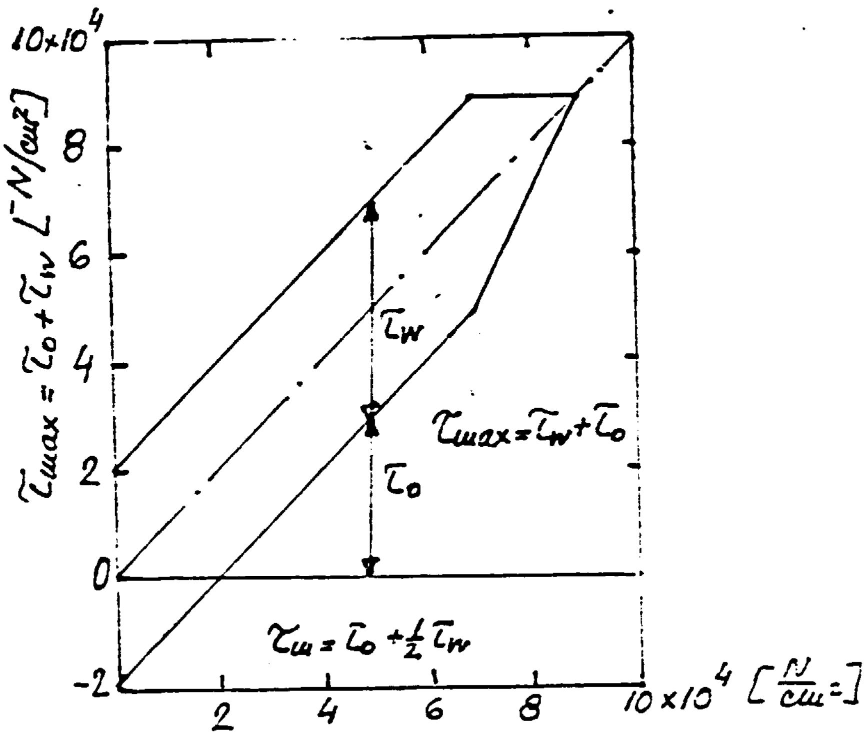


Fig 19 Middle spring steel quality

Defining h_o and $h_{i,max}$ we have the next expressions:

$$h_o + h_{i,max} = h_{i,max} \frac{\tau_{max}}{\tau_w} \tag{1}$$

$$d = \sqrt[3]{\frac{8 \cdot F_{max} \cdot D \cdot \varphi}{\pi \cdot \tau_{max}}} \tag{2}$$

d - spring wire dia.

D = see Fig 20

$$\psi = 1 + \frac{5}{4}\left(\frac{d}{D}\right) + \frac{7}{8}\left(\frac{d}{D}\right)^2 \qquad (3)$$

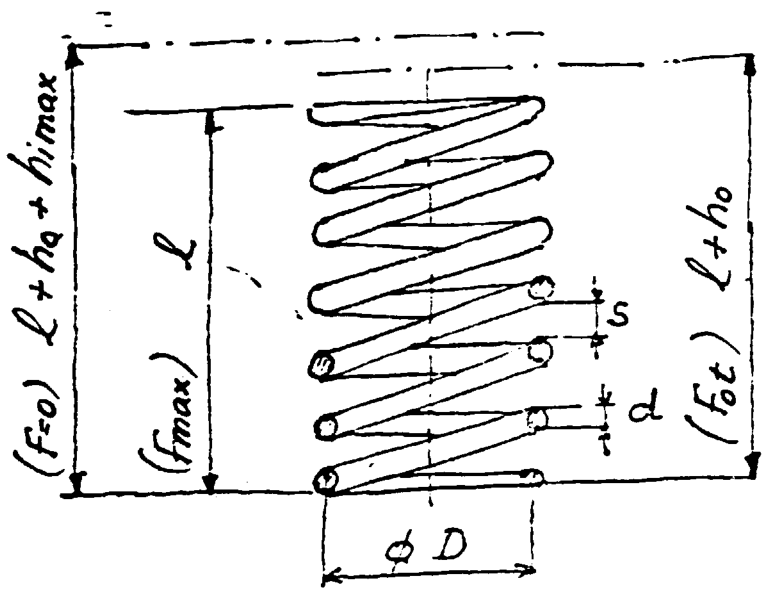


Fig 20 Injector spring proportions

The active coil number may be calculated as follows:

$$h_o + h_{i,max} = \frac{8 \cdot z \cdot D^5 \cdot F_{max}}{G \cdot d^4} \qquad (4)$$

where:

$G = 8{,}5\ 10^6\ \ N/cm^2$

$$\psi_1 = 1 - \frac{3}{16}\left(\frac{d}{D}\right)^2 \qquad (5)$$

The whole coil number z_u as:

$$z_u = z + 1{,}5 \qquad (6)$$

The spring lenght at max load:

$$l = (z_u + 1) \cdot d + z_u \cdot s \qquad (7)$$

where: $s = (0{,}15 \div 0{,}3)\, d$ (8)

The first eigen frequency as:

$$f_e = 0{,}8178 \cdot \frac{\tau_w}{\psi \cdot h_{i\,max}} \qquad (9)$$

$\tau_w \left[\frac{N}{cm^2}\right]$, $h_{i\,max}\,[cm]$

Taking into account oscillations of the spring results in:

$$\tau_{wD} = \tau_w + 4{,}5 \cdot 10^6 \cdot \psi \cdot \xi_x \qquad (10)$$

where:

$\xi_x\,[cm]$ - the amplitude of the first harminic

The penetration of the spray tip is directly dependent on Δp (pressure drop at nozzle hole) and indirectly on p_{II} (measured pressure before injector).

$$\Delta V_c = \mu \cdot A \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{\Delta p} \cdot \Delta t$$

The pressure drop Δp may be calculated using one of three programmes given to IIP. However, to get idea about relations between Δp and p_{II} Fig 21 is presented (se W.Dall-Ricardo)

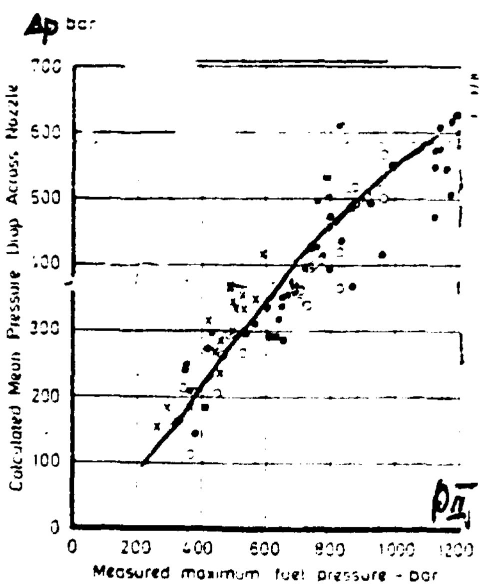


Fig 21 The relation between Δp calculated and p_{II} measured

Note: Before the needle lift control according Fig 12 we may analyse the pump-injector self-control.

$A_k v_k$ delivery pulse of HP pump influenced by connecting chanel (HP tube) reach the injector area deformed. Two Eq's control the fuelling events at injector:

$$\dot{V} = \mu \cdot A \cdot \sqrt{(p_{II} - p_2) \cdot \frac{2}{\rho}}$$

$$m \cdot \ddot{h}_i + F_o + c \cdot h_i + F_{tr} = p_{II} \cdot A_x$$

(se book Černej-Dobovišek) but only are at the pump:

$$A_k \cdot V_k = \frac{\pi \cdot d_k^2}{4} \cdot (V_o + \beta \cdot \varphi) \cdot n = C_1 \cdot d_k^2 \cdot (V_o + \beta \cdot \varphi)$$

d_k - plunger dia.
V_o - defined velocity at geometricaly defined start of delivery (prelift setting)
β - defined cam shape
φ - angle

It means according to above Eq's that injector possesses a very high influence on the whole process being represented by two Eq's. It was the reason for needle lift control application

Point 3.3

Modified cam shape of HPP camshaft

In the past as well as still nawdays the people are trying to control the preignition fuelling by means of modified cam shape.

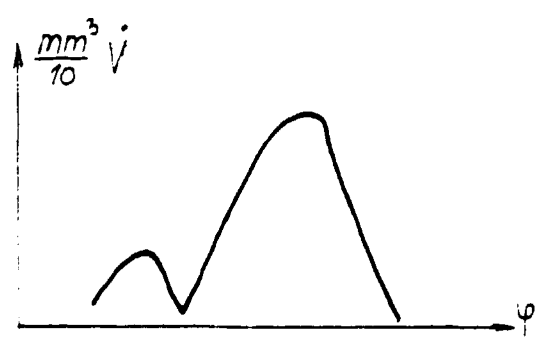


Fig 22 devided fuelling

To do this cam shape has to be modified according to Fig 23.

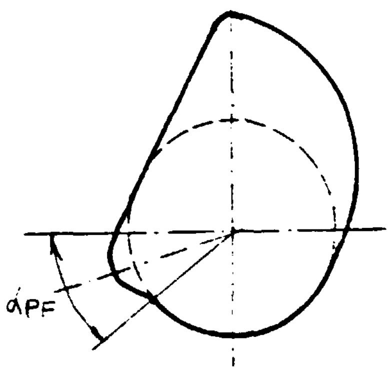


Fig 23 modified cam for devided fuelling

The shape in the region α_{PF} defines the prefuelling. However, still today no practical applications af the cam shown in Fig 23, thus this way may not be recommended.

Point 4

To start injection in more favourable ambient for ignition

Retarding the timing fuel starts to penetrate in combustion chamber later, when the temperature of compressed air becomes higher and therefore the in-combustion chamber ambient more favourable for ignition. It means that the ignition delay becomes shorter as well as fuelling injected till TDC smaller.
The aforementioned said supposes:

- late combustion and knocking are avoided under all operational condition
- product

$$\int \dot{x} \cdot p(\alpha)$$

  during expansion stroke reached reasonable high level in full load operation
- conventional in-cylinder peak pressures
- reasonable high pressure rate increase
- exhaust gaseous emission figure better or the same as for diesel fuel operation

- fair fuel consumption figure in low load operation also.

The above cited in neat methanol operation is possibly to reach only applying:

- spherical combustion chamber
- single hole nozzle or pintle nozzle
- spark plug ignition control
- matched SR
- mat_ched CR
- matched injection events

To improve engine performances is to be completed with:

- investigations concerning injector controled prefuelling
- retarded timing
- optimization of fuel deposition

Still one benefit from the proposal given may be a modest demand related to fuel system power capacity. Thus, the both, fuel consumption and first cost may be reduced. However, the both have to be investigated, low pressure and high pressure injection. It can be done by calculation at first and then optimised solution can be applied.

Topic 6

One inductive tranducer of a high sensitivity 50 kHz for needle lift measurements was given to IIP as well as the sketch for appropriate bridge.

Unfortunately IIP not disposes of an appropriate bridge which as well as various tranducers may be purchased by us.

The inductive tranducer given was suitable for injector selected. Min. tranducer which may be produced in the lab. conditions is Ø4 mm outside dia. It should be taken into account that 600 coils in one direction and 600 coils in another one have to be put in, as well as outside and inner isolation.

Last year we trained pressure measurements before the injection and this the both. pressure and needle lift. In this way the main parameters of injection may be recorded. In the first report rig test bench for (μ A) of injection was suggested, and completing that all calculations can be made.

Morever, the needle lift measurements are unavoidable tool for accurate calculations.

Based on Woschni results Dr Ivan Filipović developed before 7 years two practical programmes for simple fuel injection calculation.

MODEL I

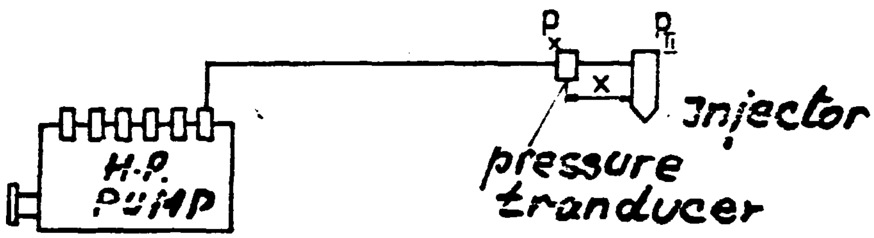


Fig.1 : Sketch of HP system. p_x at distance x.

As for input data, the total transient pressure p_x (Fig 1) at defined distance x must be registered (or stored). But also for the system given we collected the next input data:

- effective cross-sec. injector flow area vs. needle lift ($\mu_r A_b$)
- fuel properties
- in-injector dead volume V_b
- forces and geometrical proportions of injector (see nomenclature.

The whole calculation is based on the well known d'Alambert solution for the pressure wave transport:

$$p = p_o + p_v + p_r \qquad (1)$$

It means, the total transient pressure measured at distance x may be written as sum of: forward directed pressure wave p_{vx}, receding pressure wave p_{rx} and residual pressure p_o.

$$p_x = p_o + p_{xv} + p_{xr} \tag{2}$$

pressure at the injector inlet:

$$p_{II} = p_o + p_{IIv} + p_{IIr} \tag{3}$$

The same may be drown as shown in Fig 2.

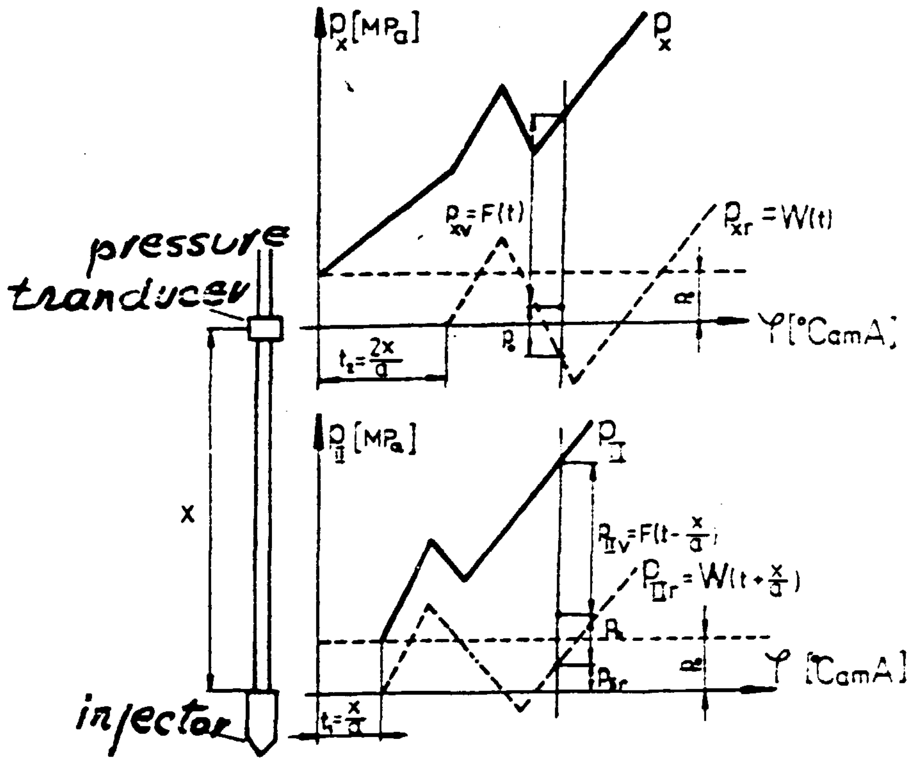


Fig 2 The sketch of pressure events at measuring distance x and at injector inlet

Based upon the Fig 2 and afore cited the next eqs. for calculation of injection events, may be written:

- mass continuity equation describes the injection function:

$$\frac{dp_{II}}{dt} = \left[A_c \cdot w_{II} - (\mu_b A_b)\cdot\sqrt{\frac{2}{\rho}(p_{II} - p_z)} - A_2 \cdot v_i\right]\cdot\frac{1}{\alpha V_b} \qquad (4)$$

- based upon the equation of needle motion (inertia due to mass of the injector moving parts (m_i), forces due to control spring (F_{ob}, h, C_{ob}) and fluid pressures are included), we may write the next two eqs.:

$$\frac{dv_i}{dt} = \left[(A_2 - A_x)\cdot p_{II} - F_{ob} + A_x \cdot p_B - h_i \cdot C_{ob}\right]\cdot\frac{1}{m_i} \qquad (5)$$

$$\frac{dh_i}{dt} = v_i \qquad (6)$$

- where p_B may be calculated as follws (Fig 3)

$$p_B = \frac{(\mu_b A_b)^2}{(\mu A)_B^2}\cdot(p_{II} - p_z) + p_z \qquad (7)$$

- total transient velocity at II - II (see Fig 3)

$$w_{II} = \frac{1}{a \cdot \rho}\cdot\left[p_o - p_{II} + 2 \cdot F(t - \frac{x}{a})\right] \qquad (8)$$

- receding pressure wave

$$W(t + \frac{x}{a}) = p_o - p_{II} + F(t - \frac{x}{a}) \qquad (9)$$

where is:

$$F(t - \frac{x}{a})_t = F(t)_{t-\frac{x}{a}} \qquad (10)$$

- forward directed pressure wave measured at distance <u>x</u> may be written as follows:

$$p_{vx} = F(t) = p_x - p_o + W(t) \qquad (11)$$

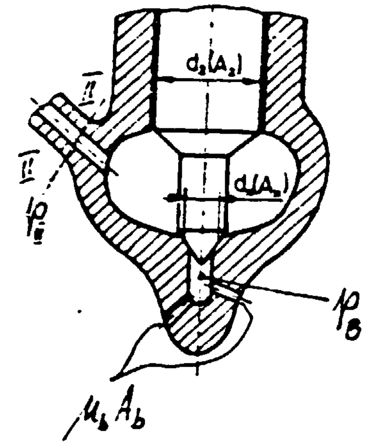


Fig 3 Sketch of injector

where is:

$$W(t)_t = W\left(t + \frac{x}{a}\right)_{t-\frac{x}{a}} \qquad (12)$$

- law of injection as:

$$\dot{q}_c = \mu_b A_b \cdot \frac{1}{6 \cdot n} \cdot \sqrt{\frac{2}{\rho}(p_{II} - p_z)} \quad \left[\frac{mm^3}{1^\circ}\right] \qquad (13)$$

- integrated law of injection as:

$$q_c = \int_{\varphi_p}^{\varphi_k} \dot{q}_c \, d\varphi \qquad (14)$$

The unknown values in eqs. 4–14 of model I are:

p_{II}, V_i, h_i, p_B, w_{II}, $W(t)$, $W(t + x/a)$, $F(t)$, $F(t - x/a)$, q_c, q_c. Solution of above system of eqs. $y' = f(x,y)$ may be found using Runge Kutta approximation (4–step) and variable time step.

MODEL II

For the system where the residual pressure tends to drop bellow the fuel vapor pressure Model I could not give satisfactory results. Here, the Model II was developed in which the both: the total transient pressure p_x at distance $\underline{X}$ and needle lift h_i were registered and used as input data .

- needle velocity may be written as follows:

$$v_i = \frac{h_{i+1} - h_{i-1}}{2\Delta\varphi}\cdot 6n \qquad (15)$$

- mass continuity equation in injector

$$\frac{dp_{II}}{dt} = \left[A_c \cdot w_{II} - (\mu_b \cdot A_b)\cdot\sqrt{\frac{2}{\rho}\cdot(p_{II} - p_z)} - A_z \cdot v_i\right]\cdot\frac{1}{\alpha \cdot V_b} \qquad (16)$$

- the total transient injector inflow velocity is:

$$w_{II} = \frac{1}{a\cdot\rho}\left[p_o - p_{II} + 2\cdot F\left(t - \frac{x}{a}\right)\right] \qquad (17)$$

- receding pressure wave may be written as follows:

$$W\left(t + \frac{x}{a}\right) = p_o - p_{II} + F\left(t - \frac{x}{a}\right) \qquad (18)$$

- where is:

$$F\left(t - \frac{x}{a}\right)_t = F(t)_{t - \frac{x}{a}} \qquad (19)$$

- the ralation for forward directed pressure wave at distance $\underline{x}$ may be written as:

$$F(t) = p_x - p_o + W(t) \qquad (20)$$

- where is:

$$W(t)_t = W\left(t + \frac{x}{a}\right)_{t - \frac{x}{a}} \qquad (21)$$

- law of injection:

$$\dot{g}_c = \mu_b \cdot A_b \qquad (22)$$

- fuel quantity injected per cycle:

$$g_c = \int_{\varphi_p}^{\varphi_k} \dot{g}_c \, d\varphi \qquad (23)$$

The unknown values in eqs. 15 ÷ 23 of model II are:

v_i, p_{II}, w_{II}, $W(t)$, $W(t+x/a)$, $F(t)$, $F(t-x/a)$, q_c, q_c.

For numerical solution of above system of eqs. a 4-step Runge-Kutta approximation may be applied using variable time step (se listings II appended).

Nomenclature

a (A)	- velocity of sound
A_c (AC)	- high pressure tube cross-sect. flow area
A_x (AX)	- needle cross-sect area (see Fig 3)
A_2 (A2)	- " " " "
C_{of} (COF)	- spring rate
F_{of} (FOF)	- opening force of the injector spring
h_i (Y(3),HIX)	- needle lift
m_i (AMI)	- mass of the injector moving parts
n (AN)	- pump rotational speed
p	- pressure
p_E (PE)	- pressure in the injector sack volume
p_o (PO)	- residual pressure
p_r	- receding pressure wave
p_v	- forward directed pressure wave
p_x (PX)	- pressure at measuring distance x
p_z (PZ)	- in-cylinder pressure
p_{xr}, $W(t)$ (WT,PRQ)	- receding press-wave at measuring distance x
p_{xv}, $F(t)$ (FT,PVX)	- forward directed pressure wave at dist. x
p_{II} (Y(1))	- total transient pressure at II - II (at injector inlet)
p_{IIr}, $W(t+x/a)$ (WTL,PDR)	- receding pressure wave at II - II
p_{IIv}, $F(t-x/a)$ (FTL,PDV)	- forward directed pressure wave at II - II
q_c (QC)	- low of injection
t, t_1, t_2	- time
v_i (Y(2),V)	- needle velocity
V_b (VB)	- injector dead volume
w_{II} (WII)	- total injector inflow fuel velocity

x - distance between press. tr. and inject.(see Fig 1)

E=1/ (EMOD) - bulk modul of elasticity

$_bA_b$ (AMBAB) - effective injector flow area

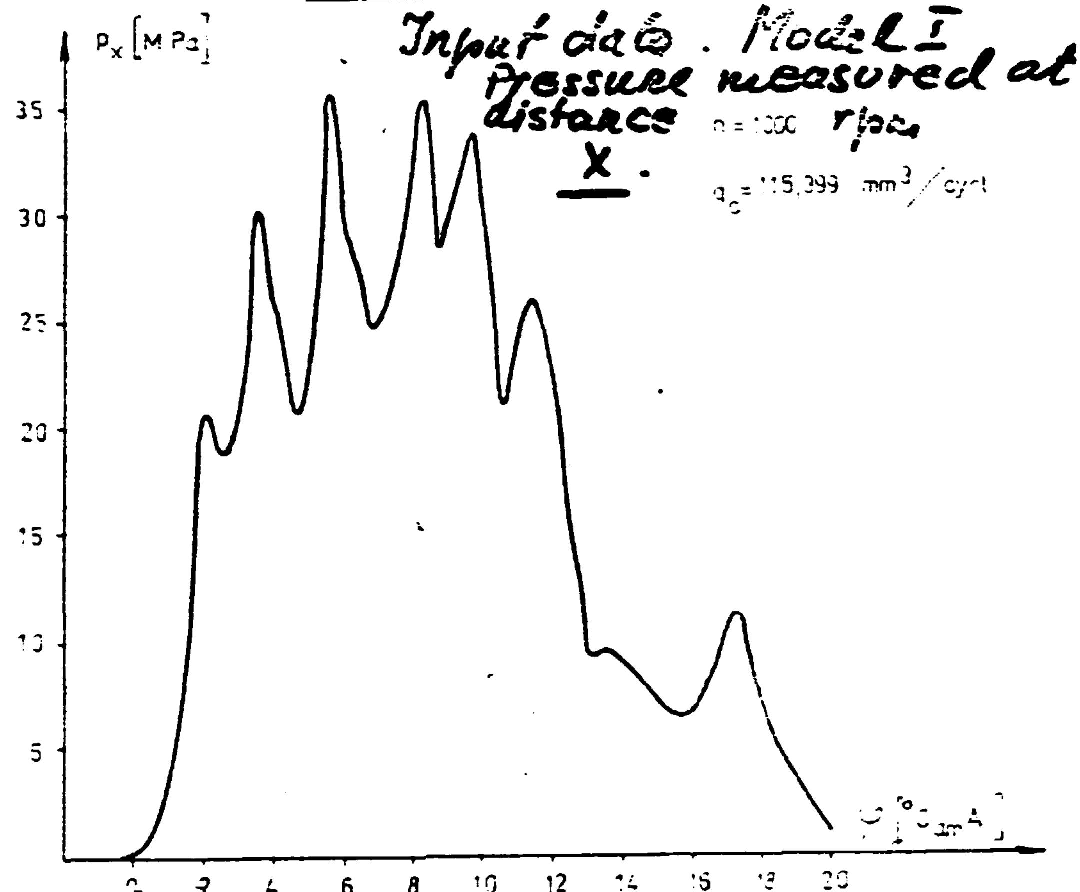
$A)_B = {}_bA_b)max$

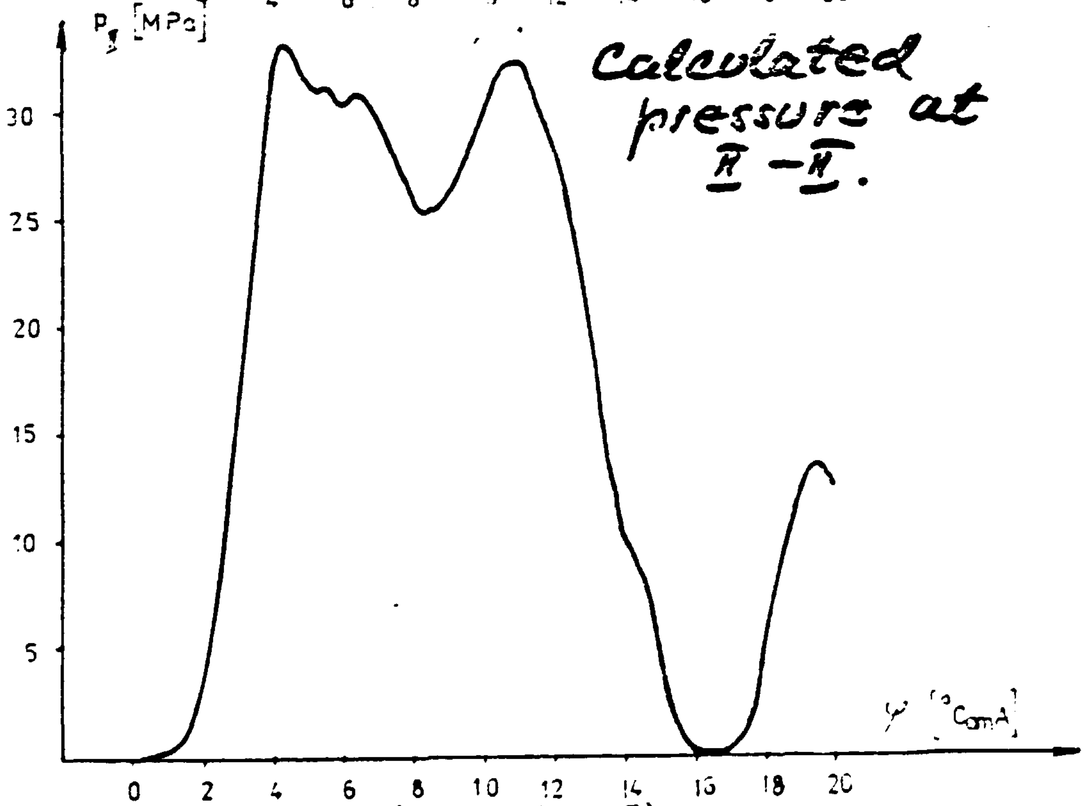
(AMAB) - max. effective injector flow area

(RC) - fuel density

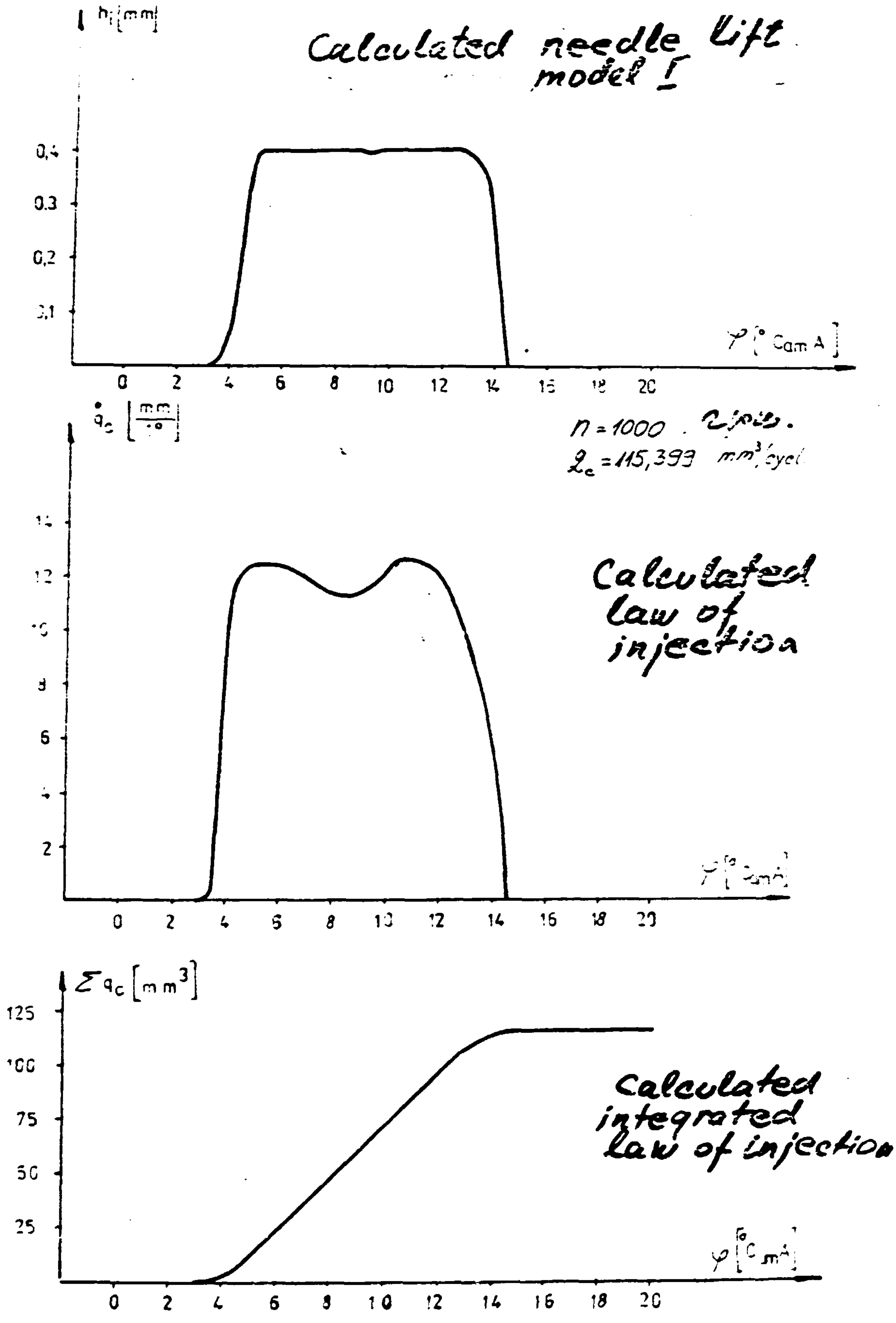
(X) - instantaneous value

RESULTS

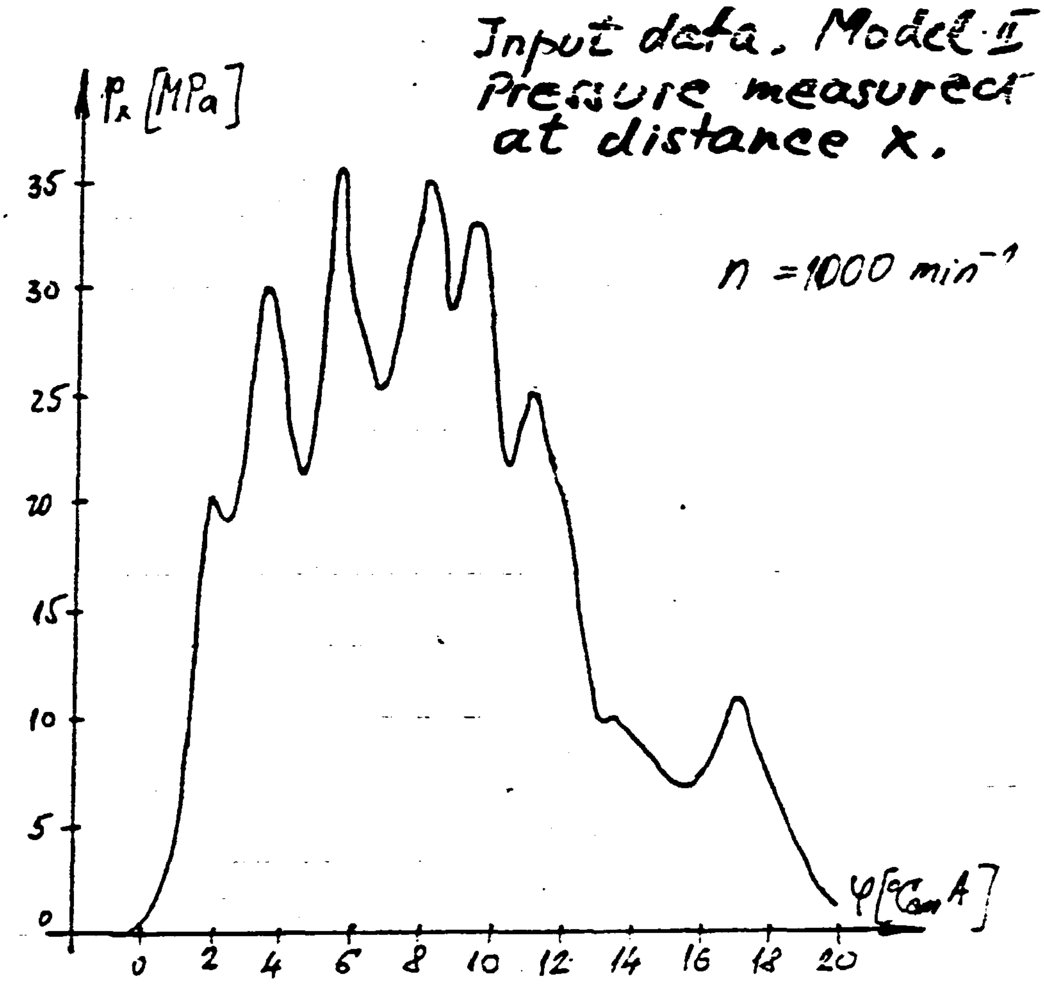


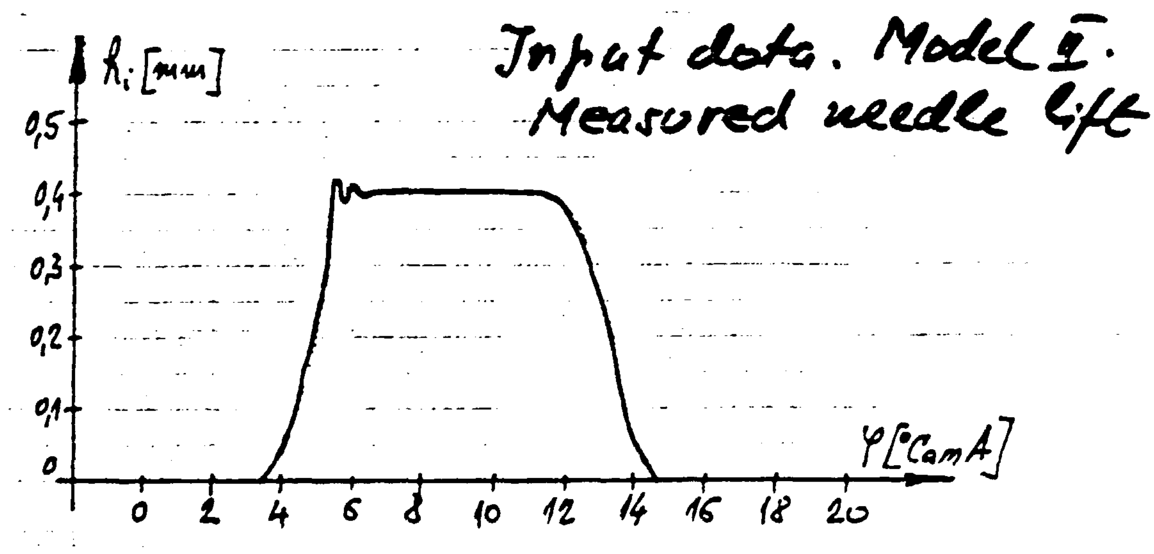


(se listings I)



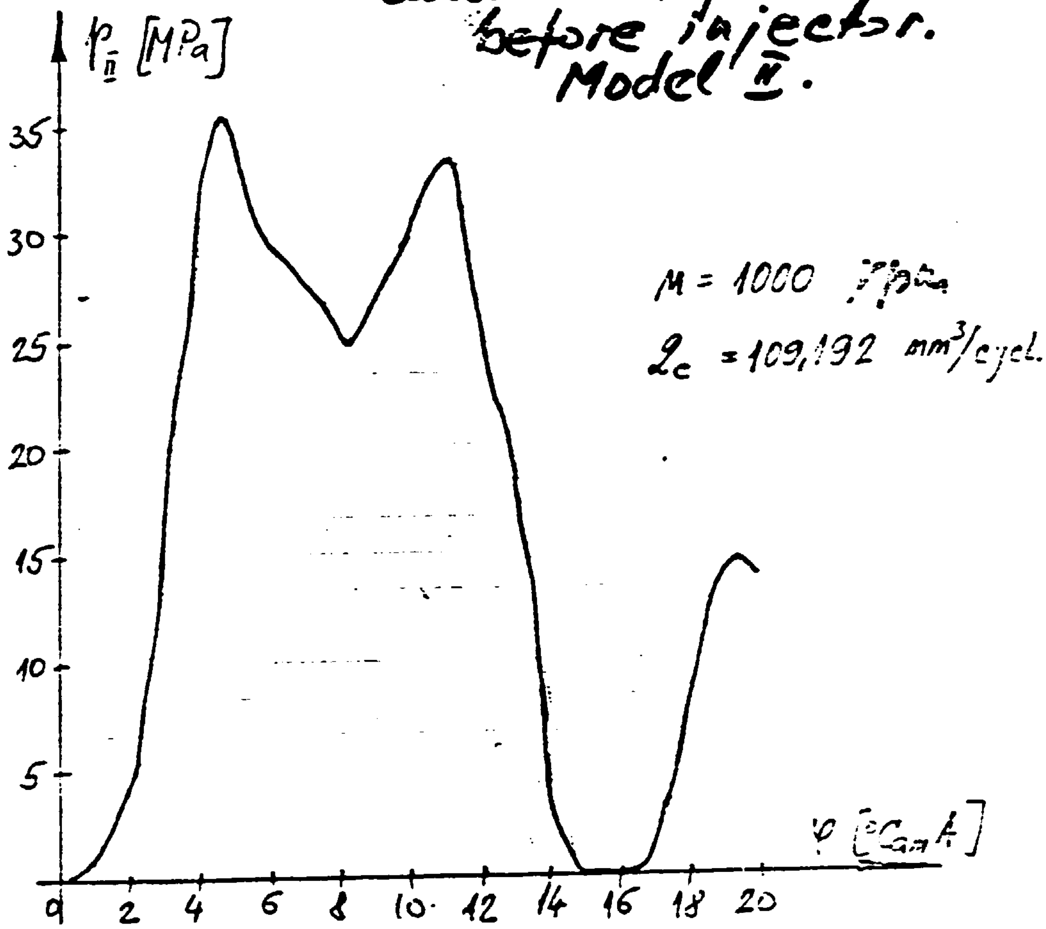
(se listings I)

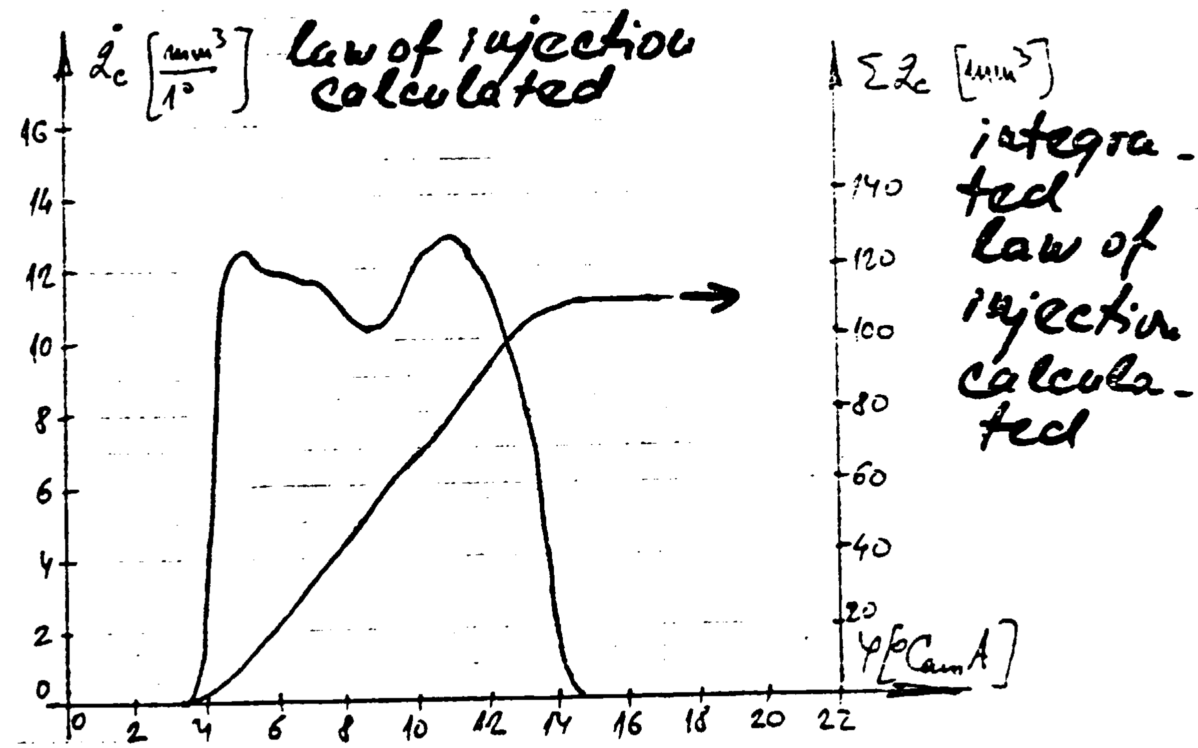




(see listings II)

Calculated pressure before injector. Model II.





(se listings II)

Topic 7

Laser doppler effect (LDE) may be applied only by accomplishing:

- the correct selection of optical depth
- the calibration

According to the principle LDE enables to define:

- the range of droplet velocities in the spray
- the degree of turbulance
- velocity change of the spray investigated
- spray penetration time -
- density of droplets per unit volume
- droplet mean dia
- droplet distribution
- droplet shape

For the nozzle hole d_B, in-cylinder pressure p_z and pressure before the injector p_{II} given we define the next dimensionless parameters:

- the velocity at the nozzle hole exit

$$V = \sqrt{\frac{2}{\rho_g}(p_{II} - p_z)}$$

- Weber number

$$We = \frac{\rho_g \cdot V^2 \cdot d_B}{\sigma_g}$$

- Laplace number

$$L_p = \frac{\rho_g \cdot d_B \cdot \sigma_g}{\mu_g^2}$$

- density ratio

$$M = \frac{\rho_z}{\rho_g}$$

where:

ρ_g - fuel density
ρ_z - in-cylinder air density
μ_g - fuel viscosity $[Ns/m^2]$
σ_g - fuel surface tension $[N/m]$

To estimate the spray cross-sectional area at the distance of the nozzle hole x: the cone angle of the spray may be written as follows:

$$tg\frac{\alpha}{2} = D_1 \cdot W^{0,32} \cdot L_p^{0,07} \cdot M^{m_1} \qquad \text{.........} (1)$$

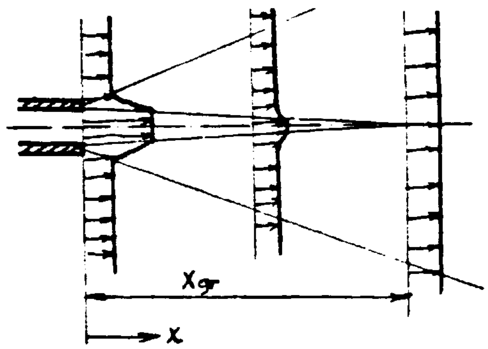
where:

$D_1=0,0112 \quad m_1=0,5 \quad$ for $\quad p_z \geqslant 0,7$ MPa

$D_1=0,00364 \quad m_1=0,25 \quad$ for $\quad p_z < 0,7$ MPa

The end of the potential core of the spray may be expressed as:

$$X_{gr} = C_1 \cdot d_g \cdot W_e^{0,25} \cdot L_p^{-0,4} \cdot M^{-g_1} \qquad \text{........}(2)$$



where:

$C_1=8,85; \; q_1=0,6 \quad$ for $\quad p_z \geqslant 0,7$ MPa

$C_1=46,9; \; q_1=0,24 \quad$ for $\quad p_z < 0,7$ MPa

Mean volume droplet dia. $d_{30}=d_k$ may be expressed as:

$$d_k = d_g \cdot 2,21 \cdot (M \cdot W_e)^{-0,266} \cdot L_p^{-0,073} \qquad \text{......} (3)$$

and its volume as $V_k = d_k^3 \cdot \pi / 6$

The expressions for the mean specifis fuel flow at distance x may be given as:

For $\quad x < x_{gr}$

$$\frac{g_m}{g_o} = N_1 \cdot \left(\frac{d_g}{x}\right)^{1,38} \cdot L_p^{-0,14} \cdot W^{-0,55} \cdot M^{-m_1} \qquad \text{..........} (4)$$

where:

$m_1 = 1,2 \quad$ for $\quad p_z \geqslant 0,7$ MPa

$m_1 = 0,6 \quad$ for $\quad p_z < 0,7$ MPa

For $x > x_{gr}$

$$\frac{g_m}{g_o} = 0{,}125 \cdot F_1^{-2{,}0} \cdot \left(\frac{d_b}{x}\right)^{2{,}0} \cdot L_p^{-0{,}2} \cdot We^{-0{,}6} \cdot M^{-2 \cdot n_1} \qquad \ldots\ldots\ldots (5)$$

where: $F_1 = 0{,}019$; $n_1 = 0{,}8$ for $p_z \geq 0{,}7$ MPa

$F_1 = 0{,}003$; $n_1 = 0{,}4$ for $p_z < 0{,}7$ MPa

The coefficient N_1 in Eq (4) may be calculated as follows:

For $p_z < 0{,}7$ MPa

$$N_1 = 138{,}88 \left(\frac{d_B}{X_{gr}}\right)^{0{,}62} \cdot L_p^{-0{,}06} \cdot We^{-0{,}05} \cdot M^{-0{,}2} \qquad \ldots\ldots\ldots (6)$$

For $p_z \geq 0{,}7$ MPa

$$N_1 = 347 \left(\frac{d_B}{X_{gr}}\right)^{0{,}62} \cdot L_p^{-0{,}06} \cdot We^{-0{,}05} \cdot M^{-0{,}4} \qquad \ldots\ldots\ldots (7)$$

The mean droplet velocity at distance x may be approximately expressed as follows:

For $x < x_{gr}$

$$V_m = 0{,}7 \cdot A_i \cdot v \cdot \left(\frac{d_B}{x}\right)^{0{,}43} \cdot We^{0{,}24} \cdot L_p^{-0{,}114} \cdot M^{-0{,}43 \cdot n_1} \qquad \ldots\ldots (8)$$

where: $n_1 = 0{,}5$ for $p_z \geq 0{,}7$ MPa

$n_1 = 0{,}225$ for $p_z < 0{,}7$ MPa

For $x > x_{gr}$

$$V_m = \frac{1}{2\sqrt{2} \cdot D_1} \cdot v \cdot \left(\frac{d_B}{x}\right) \cdot We^{0{,}21} \cdot L_p^{0{,}16} \cdot M^{-m_1} \qquad \ldots\ldots\ldots (9)$$

where: $D_1 = 3$; $m_1 = 1$ for $p_z \geq 0{,}7$ MPa

$D_1 = 0{,}22$; $m_1 = 0{,}45$ for $p_z < 0{,}7$ MPa

Coefficient A_i in the Eq (8) may be calculated as:

$$A_i = K_1 \cdot \left(\frac{d_B}{X_{gr}}\right)^{0{,}57} \cdot We^{-0{,}03} \cdot L_p^{-0{,}046} \cdot M^{-b_1} \qquad \ldots\ldots\ldots (10)$$

where: $K_1 = 0{,}17$; $b_1 = 0{,}285$ for $p_z \geq 0{,}7$ MPa

$K_1 = 2{,}295$; $b_1 = 0{,}13$ for $p_z < 0{,}7$ MPa

Fuel concentration may be expressed as:

$C_m = g_m / V_m$, g_m /kg/m^2s/ ; C_m /kg/m^3/

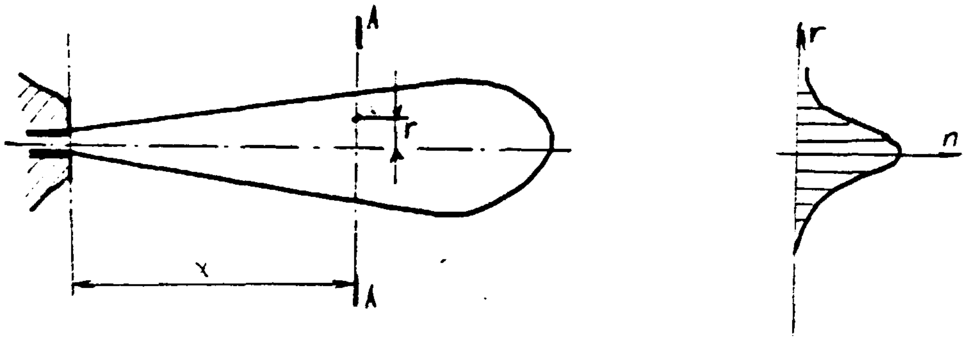
and the mass of one droplet:

$P_k = \varrho_g \cdot V_k$; $V_k = d_k^3 \cdot \pi / 6$

and finally:

$m_m = C_m / P_k$ / $\frac{\text{number of droplet}}{m^3}$ /

The concentration expressed in number of droplet per unit volume may be obtained at any distance r of the spray axis also.



$$n = n_m \cdot exp\left[-\frac{1}{6 \cdot a_c^2}\left(\frac{r}{x}\right)^2\right] \qquad ....(11)$$

where:

$$a_c = F_i \cdot We^{0,3} \cdot L_p^{0,1} \cdot M^{n_1} \qquad .... (12)$$

where:

$n_1 = 0,8$ for $p_z \geqslant 0,7$ MPa

$n_1 = 0,4$ for $p_z < 0,7$ MPa

The possibility of application of LDE in connected with poly-dispersed spray.At the some time in the optical direction we have abotement and dissipation of emission.Therefore,the degree of influence of heterogeneous dispersed spray on the measurements has to be estimated at first.For purpose of estimation of spray influence the optical depth may be used,expressed as follows:

$$\tau = \frac{a^2 \cdot L}{l^3} \qquad .... (13)$$

where: a=radius of droplet

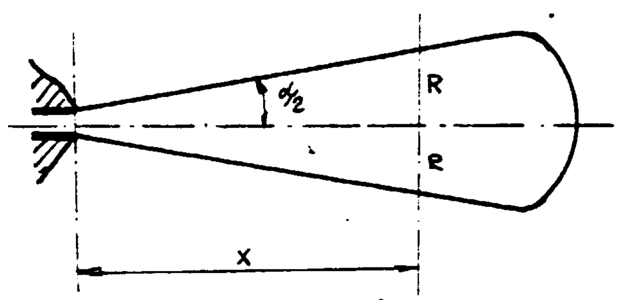
L=lenght of optical depth

l=mean distance between droplets

Dissipation is too large for $\tau >> 1$,thus LDE method could'nt be applied or the large error of the results of measurement has to be tolerated.

The prerequisite related to a small error of measurements supposes:

$$\tau \leq 1 \qquad \text{..........(14)}$$



$L=2R=2x \cdot tg\,\alpha/2$ - lenght of optical depth.
Using the Fig. shown and Eq (13) results in:

$$\tau = 1{,}5 \cdot \frac{C_m \cdot x \cdot tg\frac{\alpha}{2}}{\rho_g \cdot \pi \cdot d_{32}} \qquad \text{......... (15)}$$

where:

Sauter mean dia. $d_{32}=1{,}48d_k$

Fig.3 shows an example of optical depth obtained by experiments.

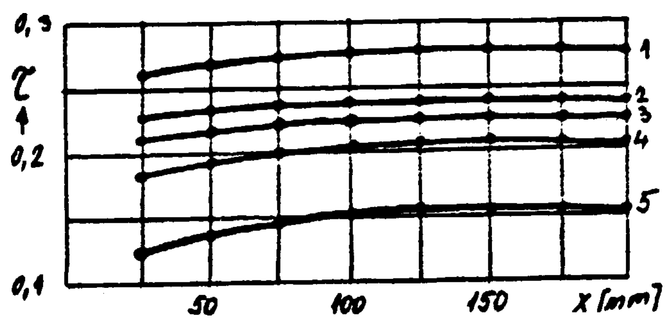


Fig.3 Optical depth by measurement vs. distance x for various nozzle holes d_B and pressures

1-p_{II}=5 MPa ,p_z=0,5 MPa ,d_B=0,3 mm

2-p_{II}=20 MPa , p_z=2 MPa , d_B=0,2 mm

3-p_{II}=20 MPa , p_z=2 MPa , d_B=0,3 mm

4-p_{II}=20 MPa , p_z=0,5 MPa ,d_B=0,3 mm

5-p_{II}=80 MPa , p_z=0,5 MPa ,d_B=0,3 mm

Approaching to τ=1 the informations of the movement of some droplets may be lost because of a large optical depth.Optical depth shown in Fig 3 was calculated bat also checked indirectly.

Spray tip velocity along x axis in shown in Fig 4. Max. velocity equals fuel out-flow one at the nozzle hole exit.At large distancies x and low pressure p_{II},droplet velocity approaches zero.

It's well established fact,that our attention must be paid to the spray characteristics in the period of ignition delay.At the end of ignition delay period the tip velocity or "velocity of mean Sauter droplet" may be reduced till 20 m/s.Therefore,the application of LDE method is important for low spray velocities also.

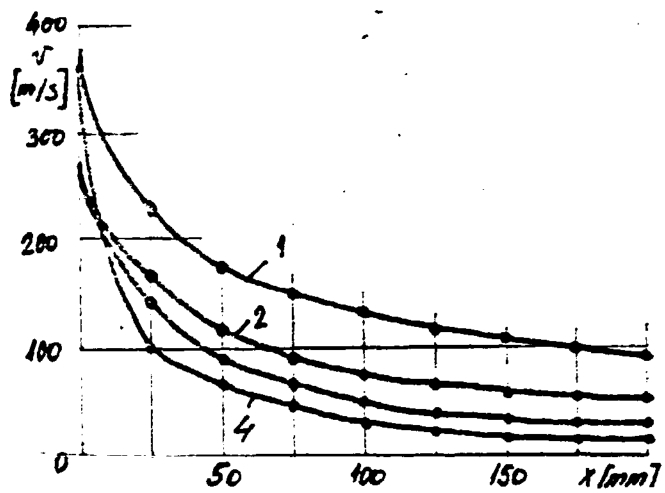


Fig.4 Spray tip velocity vs. distance x

1-p_{II}=80 MPa ,p_z=0,5 MPa ,d_B=0,5 mm
2-p_{II}=40 MPa ,p_z=0,5 MPa ,d_B=0,5 mm
3-p_{II}=40 MPa ,p_z=0,5 MPa ,d_B=0,3 mm
4-p_{II}=80 MPa ,p_z=2,0 MPa ,d_B=0,5 mm

Droplet concentrations characteristics-n vs. lenght of spray-x are shown in Fig 5. It may be seen that the concentration changs in a quite large proportion.However,LDE method is effective for relatively small droplet number per unit volume.

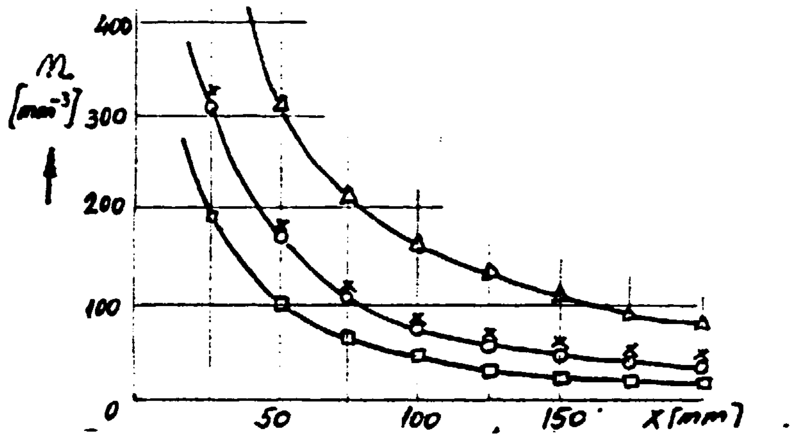


Fig.5 Droplet concentration vs. distance x

Δ-p_{II}=60 MPa,p_z=2,0 MPa ,d_k=20 μm ,d_B=0,3 mm
x-p_{II}=60 MPa,p_z=2,0 MPa ,d_B=0,3 mm,d_k=28 μm
o-p_{II}=20 MPa,p_z=2,0 MPa ,d_B=0,3 mm,d_k=40 μm
□-p_{II}=60 MPa,p_z=0,5 MPa ,d_B=0,3 mm,d_k=40 μm

The min. volume for analysis approaches 0,002-0,0025 mm^3 and the max. concentration ranges 400-500 droplets/mm^3.The later said supposes rig test experiments under pressurized ambient. Fig 5 shows,that the above requirement related to concentration was reached at:

$$x \geqslant 35 \div 40 \text{ mm}$$

TAM approach for x<40 mm may be explained as follows:

-measurements at x=50 mm and at x=70 mm
-analysis
-using the data measured,calculation may be applied for shorter x distancies.

The volume mean dia. of droplet for diesel oil D2 ranges 20÷100 μm. Therefore,the range of 5-500 μm of dispersed droplets is of interest.For the purpose of calibration helps disperser producing monodispersed spray of about 5μm droplets dia.

Fuel spray applied in diesel engine is unsteady with duration of about 2÷5 ms,but the pressure range increase (dp_{II}/dt) may reach 60÷70 MPa/1 ms effecting very drasticaly all characteristics of the spray.

LDE method may be applied in rig test experiments for fuel spray investigations,supposing the bellow cited characteristics of it instrumentation :

- -velocity range 20÷400 m/s
- -max. droplet concentrations of about 400 droplets/mm^3
- -droplet dia.'s of 5÷500 μm
- -spherical droplet shape
- -period of measurements 10÷15 ms
- -max. velocity of changing of parameters investigated 300%/1ms.

LDE may not be applied for spray analysis at x < 40 mm.

Examples of calculation

1. Diesel oil D2

Fuel density ρ_g=840 kg/m^3

" viscosity μ_g=2,1·10^{-3} Ns/m^2

" surface tension σ_g=0,028 N/m

Nozzle hole dia. d_B=0,4 mm

Pressure before nozzle hole p_{II}=600 bar

"In-cylinder" pressure p_z=40 bar

"In-cylinder" temperature T_z=1100 K

Air "in-cylinder" density ρ_z=12,67 kg/m^3

Spray angle:

$$tg\frac{\alpha}{2} = D_1 \cdot We^{0,32} \cdot Lp^{0,07} \cdot M^{m_1}$$

Weber number:

$$We = \frac{\rho_g V_g^2 \cdot d_B}{\sigma_g} = \frac{840 \cdot \left(\sqrt{\frac{2}{840} \cdot (600-40)}\right)^2 \cdot 0,4 \cdot 10^{-3}}{0,028} = 1,6 \cdot 10^6$$

Laplace number:

$$Lp = \frac{\rho_g \cdot d_B \cdot \sigma_g}{\mu_g^2} = \frac{840 \cdot 0,4 \cdot 10^{-3} \cdot 0,028}{(2,1 \cdot 10^{-3})^2} = 2133,3$$

Density ratio:

$$M = \frac{\rho_z}{\rho_g} = \frac{12,67}{840} = 0,0155$$

Thus:

$$tg\frac{\alpha}{2} = 0,0112 \cdot (1,6 \cdot 10^6)^{0,32} \cdot (2133,3)^{0,07} \cdot 0,0155^{0,5} = 0,2305 \rightarrow \alpha = 25,96°$$

Potential core end: $X_{gr} = C_1 \cdot d_B \cdot We^{0,25} \cdot Lp^{-0,4} \cdot M^{-g_i} =$

$$= 8,85 \cdot 0,4 \cdot 10^{-3} \cdot (1,6 \cdot 10^6)^{0,25} \cdot (2133,3)^{-0,4} \cdot 0,0155^{-0,6} = 71,487 \text{ mm}$$

Volume mean droplet dia.:

$$d_k = 2{,}21 \cdot d_B \cdot (M \cdot We)^{-0{,}266} \cdot L_p^{-0{,}073} =$$

$$= 2{,}21 \cdot 0{,}4 \cdot 10^{-3} \cdot (0{,}0155 \cdot 1{,}6 \cdot 10^{6})^{-0{,}266} \cdot 2133{,}3^{-0{,}073} =$$

$$= 34\,\mu m$$

Sauter mean dia.:

$$d_{32} = 1{,}48 \cdot d_k = 50{,}32\,\mu m$$

Specific dimensionless flow at distance x= 90 mm $(x > x_{gr})$ may be calculated as follows:

$$\frac{g_m}{g_o} = 0{,}125 \cdot F_1^{-2} \left(\frac{d_B}{x}\right)^2 \cdot L_p^{-0{,}2} \cdot We^{-0{,}6} \cdot M^{-2n_1} =$$

$$= 0{,}125 \cdot 0{,}019^{-2} \cdot \left(\frac{0{,}4 \cdot 10^{-3}}{0{,}09}\right)^2 \cdot 2133{,}3^{-0{,}2} \cdot (1{,}6 \cdot 10^{6})^{-0{,}6} \cdot 0{,}0155^{-2 \cdot 0{,}8} =$$

$$= 2{,}212 \cdot 10^{-4}$$

Spray tip velocity at distance x=90 mm is calculated as:

$$V_m = \frac{1}{2\sqrt{2} \cdot D_1} \sqrt{\frac{2}{\rho}(p_I - p_2)} \cdot \left(\frac{d_B}{x}\right) \cdot We^{0{,}21} \cdot L_p^{-0{,}16} \cdot M^{-n_1} =$$

$$= \frac{1}{2 \cdot 1{,}41 \cdot 3} \cdot \sqrt{\frac{2}{840}(600-40) \cdot 10^{5}} \cdot \left(\frac{0{,}4 \cdot 10^{-3}}{0{,}09}\right) \cdot (1{,}6 \cdot 10^{6})^{0{,}21} \cdot 2133{,}3^{-0{,}16} \cdot 0{,}0155^{-1} =$$

$$= 72{,}03 \text{ m/s}$$

2.<u>Methanol</u>

CH_3OH density ρ_g=774 kg/m^3
" viscosity μ_g=0,4 · 10^{-3} Ns/m^2
" surface tension σ_g=0,021 N/m

All other date as for the aforegiven example with diesel oil

Weber number:

$$We = \frac{774\left(\sqrt{\frac{2}{774}(600-40) \cdot 10^{5}}\right)^2 \cdot 0{,}4 \cdot 10^{-3}}{0{,}021} = 2{,}13 \cdot 10^{6}$$

Laplace number:

$$L_p = \frac{774 \cdot 0{,}4 \cdot 10^{-3} \cdot 0{,}021}{(0{,}4 \cdot 10^{-3})^2} = 4{,}06 \cdot 10^{4}$$

Density ratio:

$$M = \frac{12{,}67}{774} = 0{,}0164$$

Spray angle:

$$tg\,\frac{\alpha}{2} = D_1 \cdot W^{0{,}32} \cdot L_p^{0{,}07} \cdot M^{m_1} =$$

$$= 0{,}0112 \cdot (2{,}13 \cdot 10^{6})^{0{,}32} \cdot (4{,}06 \cdot 10^{4})^{0{,}07} \cdot 0{,}0164^{0{,}5} = 0{,}319 \quad \rightarrow \quad \alpha = 35{,}4^\circ$$

The end of potential core:

$$X_{gr} = C_1 \cdot d_B \cdot We^{0{,}25} \cdot L_p^{-0{,}4} \cdot M^{-g_1} =$$

$$= 8{,}85 \cdot 0{,}4 \cdot 10^{-3} \cdot (2{,}13 \cdot 10^{6})^{0{,}25} \cdot (4{,}06 \cdot 10^{4})^{-0{,}4} \cdot (0{,}0164)^{-0{,}6} =$$

$$= 22{,}8 \text{ mm}$$

Volume mean droplet dia.:

$$d_k = 2{,}21 \cdot d_B \cdot (M \cdot We)^{-0{,}266} \cdot L_p^{-0{,}073} =$$
$$= 2{,}21 \cdot 0{,}4 \cdot 10^{-3} \cdot (0{,}0164 \cdot 2{,}13 \cdot 10^{6})^{-0{,}266} \cdot (4{,}06 \cdot 10^{4})^{-0{,}073} =$$
$$= 25\ \mu m$$

Sauter mean dia.:

$$d_{32} = 1{,}48 \cdot d_k = 37\ \mu m$$

Specific dimensionless flow at distance x=90 mm ($x > x_{gr}$) may be calculated as follows:

$$\frac{g_m}{g_o} = 0{,}125 \cdot F_1^{-2} \left(\frac{d_B}{x}\right)^2 \cdot L_p^{-0{,}2} \cdot We^{-0{,}6} \cdot M^{-2n_1} =$$
$$= 0{,}125 \cdot 0{,}019^{-2} \cdot \left(\frac{0{,}4 \cdot 10^{-3}}{0{,}09}\right)^2 \cdot (4{,}06 \cdot 10^{4})^{-0{,}2} \cdot (2{,}13 \cdot 10^{6})^{-0{,}6} \cdot 0{,}0164^{-2 \cdot 0{,}8} =$$
$$= 9{,}45 \cdot 10^{-5}$$

Spray tip velocity at distance x=90 mm is calculated as:

$$V_m = \frac{1}{2\sqrt{2} \cdot D_1} \sqrt{\frac{2}{\rho}(p_I - p_2)} \cdot \left(\frac{d_B}{x}\right) \cdot We^{0{,}21} \cdot L_p^{-0{,}16} \cdot M^{-n_1} =$$
$$= \frac{1}{2 \cdot \sqrt{2} \cdot 3} \sqrt{\frac{2}{774} \cdot (600-60) \cdot 10^{5}} \left(\frac{0{,}4 \cdot 10^{-3}}{0{,}09}\right) \cdot (2{,}13 \cdot 10^{6})^{0{,}21} (4{,}06 \cdot 10^{4})^{-0{,}16} \cdot 0{,}0164^{-1} =$$
$$= 47{,}54\ m/s$$

<u>Note</u>: All expressions derived serve for the both, experimental arrangement and avoidance of measurement in error.

For methanol let's calculate the parameters related to LDE experimental application.

g_o (see Eq 4) may be calculated as:

$$g_o = \rho_g \cdot V$$
$$V = \sqrt{\frac{2}{\rho}(p_I - p_2)} = \sqrt{\frac{2}{774}(600-60) \cdot 10^{5}} = 373\ \frac{m}{s}$$
$$\rho = 774\ kg/m^3$$
$$g_o = 373 \cdot 774 = 2{,}88702 \cdot 10^{5}\ \frac{kg}{m^2 s}$$
$$g_m = 9{,}45 \cdot 10^{-5} \cdot 2{,}88702 \cdot 10^{5} = 27{,}28 \cdot \frac{kg}{m^2 s}$$
$$C_m = \frac{g_m}{V_m} = \frac{27{,}28}{47{,}54} = 0{,}57388\ \frac{kg}{m^3}$$
$$n_m = \frac{C_m}{P_k}$$
$$P_k = \rho \cdot V_k = 774 \cdot \frac{1}{6} \cdot 25^3 \cdot \pi \cdot 10^{-18} = 6{,}33 \cdot 10^{-12}\ kg$$
$$n_m = \frac{C_m}{P_k} = \frac{0{,}57388}{6{,}33 \cdot 10^{-12}} = 90{,}6\ \frac{droplets}{mm^3}$$

(Note $m^3 = 10^9\ mm^3$)

It means that we have less than 400 droplets/mm^3, thus related to $\underline{n_m}$ LDE for the distance 90 mm selected, may be applied.

Optical depth

$$\tau = \frac{d_k^2 L}{l^3}$$

$$L = 2 \cdot x \cdot tg\frac{\alpha}{2} = 2 \cdot 90 \cdot 0{,}319 = 57{,}42 \text{ mm}$$

To define l we calculate:

$$90 \cdot \left(d_k + \frac{l}{2}\right)^2 \cdot \frac{\pi}{4} = 1 mm^2 = 10^6 \mu m^2$$

$$\left(25 + \frac{l}{2}\right)^2 \cdot 90 \cdot \frac{\pi}{4} = 10^6 \rightarrow l = 188 \mu m$$

$$\tau = \frac{25^2 \cdot 57{,}44 \cdot 10^3}{188^3} = 5{,}36$$

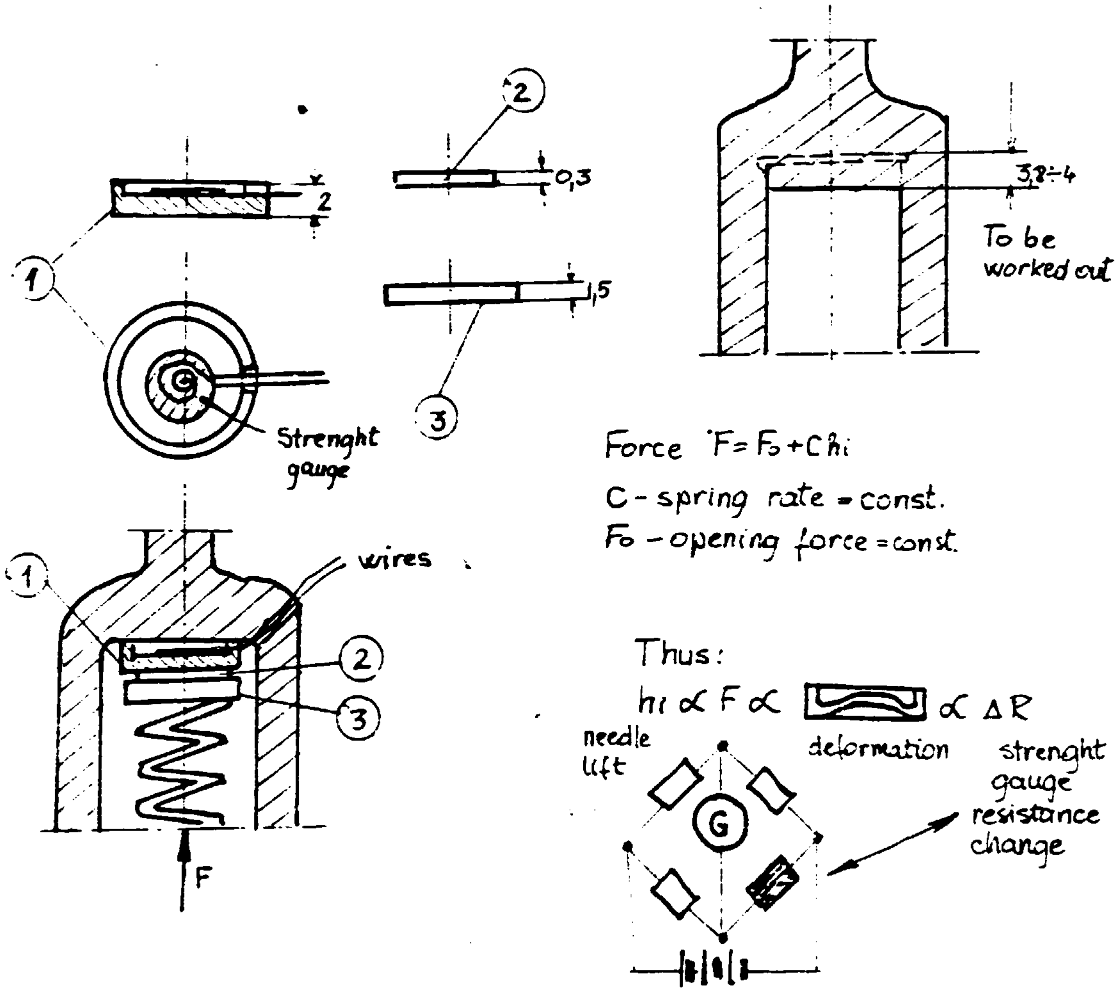
Some disturbancies caused by screening effect may be expected. However, because of low $\underline{C_m}$ some corrections may be applied. In order to define the optical depth we have to equal τ=1 This calculation gives L=10,7 mm

Auxiliary needle lift measurement technique

Needle lift measurements may be performed using strenght gauge technique also. If one appropriate bridge and a small membrane strenght gauge exist the other arrangements are very simple.

However, the needle lift diagrames recorder suffer from:

- some nelinearity (what is also not inportant)
- some other oscillations may be noticed in needle lift diagrames at the end



The Fig's shown explain the arrangement

<u>How to calculate injector holes</u> ?

Fuel injected per one cycle or fuelling may be expressed as follows:

$$G = b_e \frac{P_e}{3600 \cdot n_M \cdot z \cdot i_z} \left[\frac{g}{\text{cycle cylinder}} \right] \qquad ....(1)$$

where:

b_e /g/kWh/ -fuel consumption rate

P_e /kW/ -power output

n_M /s^{-1}/ -rotational engine speed

z =0,5 - 4 stroke eng. ,=1 -2 stroke engine iz - cylinder number

Substituing Eq (1) into expression of the injection law results in:

$$\mu_D F_D = \frac{G}{4472 \cdot \rho_K \cdot \tau_E} \cdot \sqrt{\frac{\rho_K}{p_{EVD} - p_G}} \qquad [cm^2] \qquad \text{......(2)}$$

where:

ρ_K /g/cm^3/ -specific fuel mass

τ_E/s/ -injection time

$p_{EVD}=p_{II}$ /MPa/ -fuel pressure before injector

p_G /MPa/ -in-cylinder pressure

F_D /cm^2/ -geometrical cross-sectional flow area of the injector holes

μ_D /-/ -coefficient of discharge (ranges 0,6÷0,8).

Distance of the spray tip vs. time (or angle) is a factor for fuel system matching to the combustion chamber and piston movement (see x in "Dynamic").

Morever,depending on the comb. process in developement and fuel used the importance of ignition delay may be very high. Thus, quantity of the fuel injected and spray tip history could'nt be neglected.

At first we calculate the angle of ignition delay (α_{zv}); per example for D2 fuel acc. Sitkei as follows:

$$\alpha_{zv} = 0{,}36 \; n_M \left[0{,}5 + \left| \frac{0{,}0266}{p_c^{0,7}} - \frac{0{,}0735}{p_c^{1,6}} \right| e^{\frac{3930}{T_c}} \right] [°KW] \qquad \text{......(3)}$$

or by time:

$$\tau_{zv} = \frac{\alpha_{zv}}{360 \cdot n_M} \qquad [s] \qquad \text{.....(3')}$$

Eq (3) was derived for diesel fuel but the same influencing factors with have with other fuels also. With CH_3OH,acc. to Eq (3),the both,the pressure-and the temperature of compression are droping down,because of excess cooling. The above results in:knocking or misfiring. Therefore,retarded injection as well as controled injection quantity in τ_{zv}-time ,may help significantly.

Siegfried and Ahmed developed the formule for spray tip penetration as:

$$\Delta S_{st} = 0{,}027 \frac{d_d}{(\Sigma\Delta\tau)^{0,85}} \cdot H_o^{0,48} \cdot R_e^{0,3} \cdot \left(\frac{\rho_K}{\rho_L}\right)^{0,15} \quad [cm] \qquad ......(4)$$

where: $\Delta\tau$-increment of time /s/

$$H_o = \frac{w_o \cdot \tau}{d_d} \; ; \qquad R_e = \frac{w_o \cdot d_d}{\nu_K} \; ; \; \rho_L\text{-air density } [g/cm^3]$$

Eq (4) was derived recently,where only in-cylinder pressure but without air motion was respected.

From Eq (4) nozzle hole dia. may be obtained:

$$d_d = \left| \frac{20827 \cdot 10^{5} \cdot S_{st} \cdot \nu_K^{0,3} \cdot \Delta\tau^{0,37}}{w_o^{0,78}\left(1+\frac{1}{2^{0,85}}+\frac{1}{3^{0,85}}+\dots+\frac{1}{j^{0,85}}\right)} \left(\frac{\rho_L}{\rho_K}\right)^{0,15} \right|^{1,22} \quad [cm] \qquad .....(5)$$

The fuel velocity at the exit of the nozzle hole may be calculated by:

$$w_o = 4{,}4721 \cdot 10^{3} \cdot \mu_D \sqrt{\frac{p_{rvo} - p_c}{\rho_K}} \quad \left[\frac{cm}{s}\right] \qquad ......(6)$$

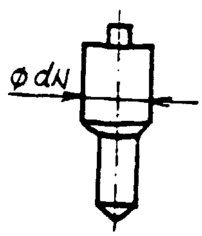
From Eq.'s (2) and (5) we obtain the number of the nozzle holes as:

$$i = \frac{\mu_D F_D}{\mu_D \frac{\pi d_D^2}{4}} \qquad ....(7)$$

Dia ø d_{sa} (see Fig 1) may be calculated by:

$$d_{sa} \gtrapprox \frac{2}{\pi} \cdot d_d \cdot i \quad [cm] \qquad ...(8)$$

The simple approach shown,may be useful by CH_3OH application also.



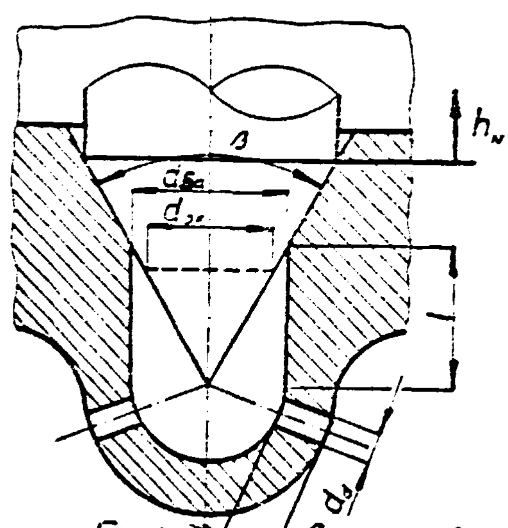


Fig 1.

After aforeshown calculation the choise related to needle seat dia (d_{si}) or needle opening pressure ($P\ddot{o}$) has to be done.

When the bellow cited data are known:

- d_N needle dia (Fig 1)
- p_z max in-cylinder pressure
- d_{SA} sack hole dia (Fig 1)
- $p_{\ddot{o}}$ needle opening pressure
- p_s needle closing pressure

needle seat dia. (d_{si}) may be calculated as follows:

$$d_{si} = \sqrt{\frac{d_N^2 (p_{\ddot{o}} - p_s) + 0{,}5 \cdot d_{sa}^2 (p_s - p_z)}{p_{\ddot{o}} + 0{,}5 (p_z - p_s)}} \quad [cm] \qquad \ldots(9)$$

Sometimes is more convinient to use an accustomed hydroulic ratio (see the book Černej-Dobovišek,"Injection"). Doing that, seat dia. ϕd_{si} known and the needle closing pressure may be defined as:

$$p_s = \frac{(d_N^2 - d_{si}^2)\, p_{\ddot{o}} - (d_{si}^2 + d_{sa}^2)\, 0{,}5 \cdot p_z}{d_N^2 - 0{,}5 (d_{si}^2 + d_{sa}^2)} \quad [MPa] \qquad \ldots\ldots(10)$$

However, injector hole dia. (d_a) must be examined related to cavitation also.

Thus minim. (d_{dmin}) nozzle hole dia. to prevent cavitation may be defined as:

For $l_d/d_d > 3$ (see Fig 1)

$$d'_{dmin} = 0{,}0053 \sqrt{\frac{b_e \cdot P_e \cdot K_{\ddot{u}}}{\rho_K \cdot i_z \frac{\alpha_E}{2} \cdot i} \sqrt{\frac{\rho_K}{\mu'_d} \left(\frac{2 - 2\mu'_d - \mu_d \cdot \lambda \cdot l_d/d_d}{p_s - p_o} \right)}} \quad [cm] \ldots(11)$$

For $l_d/d_d \leq 3$

$$d_{dmin} = 0{,}00634 \sqrt{\frac{b_e \cdot P_e \cdot K_{\ddot{u}}}{\rho_K \cdot i_z \frac{\alpha_E}{2} \cdot i} \sqrt{\frac{\rho_K}{\mu'_d} \left(\frac{1 - \mu'_d}{p_o - p_o} \right)}} \quad [cm] \ldots\ldots(12)$$

where:

$K_{\ddot{u}}$ /-/ -coefficient of overloading (1,1÷1,12)

α_E /°CA/ -duration of injection

Supposing the avoidance of cavitation phenomenon, the max. velocity at the nozzle hole may be calculated as:

$$w_{\partial max} = 3162{,}4\sqrt{\frac{p_s - p_o}{\rho_k\left(\frac{1}{\mu_d} - 1 - \frac{\lambda}{2}\cdot\frac{\ell_d}{d_d}\right)}} \quad \left[\frac{cm}{s}\right] \qquad ......(13)$$

where:

λ =coefficient of friction in the nozzle hole (λ=0,02)

To obtain the effective cross-sectional nozzle hole area in dependence of needle lift two locations must be considered:

-flow area at needle seat

-flow area at nozzle holes

The both types of needle tip will be considered:

Type 1

Type 2

Geometrically defined cross-sectional flow area of the nozzle holes:

$$F_d = i\,\frac{d_d^2\cdot\pi}{4} \quad [cm^2] \qquad ......(14)$$

For the cross-sectional flow area of the needle seat (see Fig 1) the next expression may be derived:

$$F_{si} = h_n\,\pi\left(d_{sa} - \frac{h_n}{2}\sin\beta\right)\sin\frac{\beta}{2} \quad [cm^2] \qquad ....(15)$$

The expression (15) may be applied for the both needles, Type 1 an Type 2 under condition:

$$d_{of} \leqq d_{sa} - \frac{h_n}{2}\sin\beta \qquad h_n \leqq \frac{2(d_{sa} - d_{of})}{\sin\beta}$$

respectively

$$F_{si} = h_n\,\pi\left(d_{of} + \frac{h_n}{2}\sin\beta\right)\sin\frac{\beta}{2} \quad [cm^2] \qquad ....(16)$$

The expression (16) may be applied for needle Type 2 under condition:

$$d_{of} > d_{sa} - \frac{h}{2}\sin\beta$$

respectively

$$h_n > \frac{2(d_{sa} - d_{of})}{\sin\beta}$$

Discharge coefficient of the needle cone-seat area (Fig 1) depends on quality of surfaces and ranges 0,77-0,9. Discharge coefficient of nozzle hole may be calculated, as shown bellow.
Ratio of geometrical flow areas of nozzle holes and sack hole (see Fig 1-ϕd_{sa}) may be expressed as:

$$m = \frac{F_d}{F_{sa}} \qquad \text{.........} (17)$$

and pressure ratio as:

$$x = \frac{p_{sa} - p_G}{p_G} \qquad \text{.......}(18)$$

where: p_{sa}-instanteneous fuel pressure in sack hole
p_G-instanteneous in-cylinder pressure
$\underline{x}$ value is limited related to cavitation, thus ($x_{max}=x_{grenz}$):

$$x_{Grenz} = \frac{1-\Psi^2 m^2 + 2\Psi^2 - 2\Psi}{-2\Psi^2 + 2\Psi} \quad [-] \qquad \text{...........}(19)$$

Coefficient Ψ may be found by means of Tabeles in dependence of $\underline{m}$ and rims.
Furthermore discharge coefficient of nozzle hole may be calculated as:

$$\mu_d = \sqrt{\frac{1-\Psi^2 m^2}{1-\Psi^2 m^2 + 2\Psi^2 - 2\Psi}} \quad [-] \quad \text{for } x \leq x_{Grenz} \qquad \text{........}(21)$$

or in other case:

$$\mu_d = \Psi' \sqrt{1 + \frac{1}{x}} \quad [-] \quad \text{for } x > x_{Grenz} \qquad \text{...........}(22)$$

where:

$$\Psi' = \frac{\Psi}{\sqrt{1-\Psi^2 m}} \qquad \text{.......}(20)$$

The pressure in the sack hole may be expressed as follows:

at first the mean flow rate to be defined:

$$\dot{V} = \frac{G}{\gamma_E \cdot \rho_k} \quad \left[\frac{cm^3}{s}\right] \qquad ....(23)$$

The pressure in the sack hole:

$$p_{sa} = \frac{50 \cdot \dot{V} \cdot \rho_k}{10^9 \cdot (\mu_d F_d)^2} + \frac{p_z + p_c}{2} \qquad ....(24)$$

And effective flow area of the injector holes:

$$\mu_D F_d = \sqrt{\frac{(\mu_d F_d)^2 \cdot (\mu_{si} \cdot F_{si})^2}{(\mu_d \cdot F_d)^2 + (\mu_{si} \cdot F_{si})^2}} \quad [cm^2] \qquad ....(25)$$

Sack volume of injector may be calculated as:

$$V_{sa} = \frac{d_{sa}^2 \pi}{4} \cdot \left(l - \frac{d_{sa}}{6\, tg\frac{\beta}{2}} + \frac{d_{sa}}{3}\right) + \frac{1}{6\, tg\frac{\beta}{2}} \cdot \left(\frac{d_{DF}^2 \pi}{4} d_{DF}\right) \; [cm^3] \qquad ....(26)$$

Expression (26) is valid for:

- $d_{DF} > d_{sa}$
- for nozzle Type 1 and 2 (for Type 1 $\rightarrow$ d_{DF}=C) (see Fig 1)

7.1 Controlled singl-hole nozzle (CSHN)

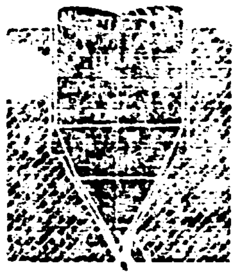
The function of controlled nozzle as improvement related to ignition was jet explained in the section-neat methanol use in diesel engine. Hewever, pintle nozzle suggested may not be used because of cylinder head. Existed cylinder head to operate with CH_3OH can not be replaced so easily. This is the reason to show one different aproach for the use in lab. experiments.

Controlled singl hole nozzle has not the aforementioned drawbeck, namely, fuel spray may be directed fairly independent of nozzle, holder axis. Thus, existed cylinder head may be used and fuel spray deposition area on the wall of piston bowl may be optimized.

In experiments with CSHN, MAN tryed four goals to reach:

- cylinder head of methanol diesel engine must be the same as for diesel oil operation
- controlled effective flow area
- at low partial loads more fuel dispersed in air
- at higher loads more fuel deposited on the wall

Fig 2 shows the idea.
Difference between one conventional nozzle and CHSN may be explained by:



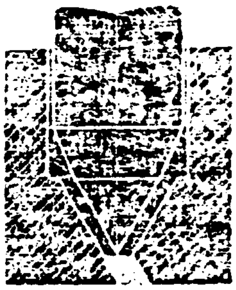


Fig 2 - CSHI

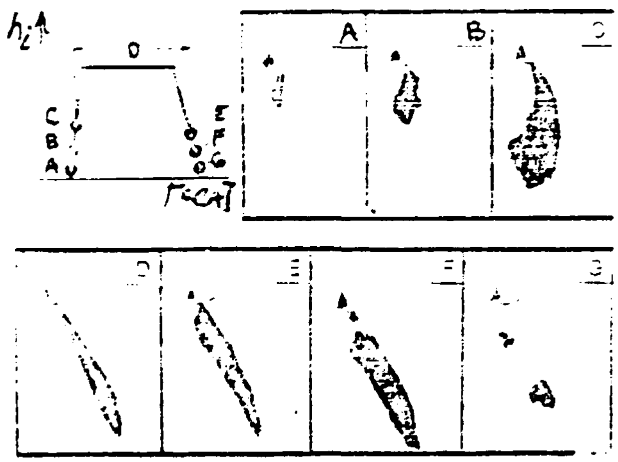


Fig 3 Fuel jet patterns of a controlled single hole nozzle at different positions of nozzle needle (h_i)

- no sack hole
- short toward needle axis directed single hole emerging in nozzle seat surface
- needle tip immersed into nozzle hole

With (CSHN) small needle lifts,seat flow area becomes smaller than that of nozzle hole,thus,fuel spray obtaining a high velocity in the seat area unhampered penetrates more toward the center of the piston bowl,being dispersed more in air.

With fully open injector,hole flow area becomes smaller than seat area,thus,directed dense fuel spray follows hole angle designed.

By means of spray patterns in Fig 3 MAN demonstrated the abovecited . With CSHN some decrease in fuel consumption figure as well as better startability of engine were observed.

Defining hole and seat area conversion of more potential energy into kinetic form,before seat area,may produce unwanted pressure losses as well as uncontrolled fuel spray direction. To have idea about above said,by black collored surface demonstrates Fig 4 the relation of kinetic energy converted.

black collored
part of kinetic
energy

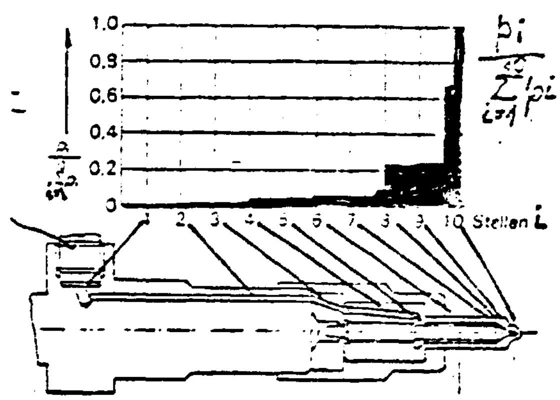
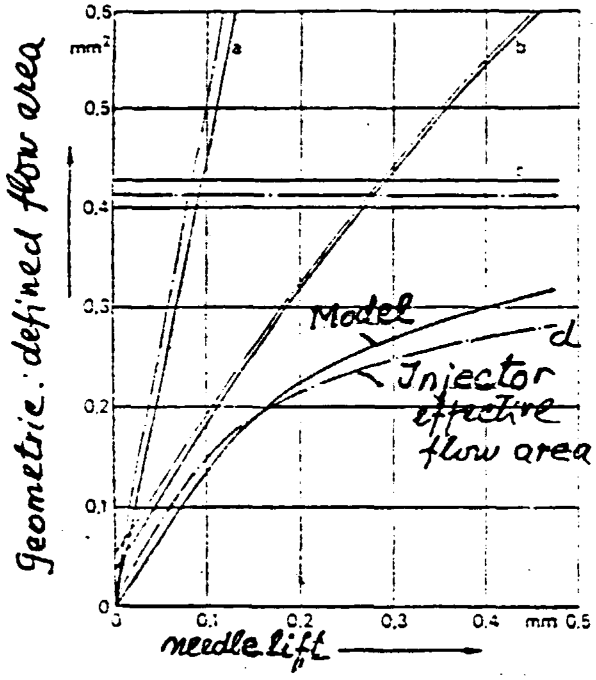


Fig 4

Is to be mentioned,that at position x (see Fig 4) some fillers (fuel filters) converte to much potential energy into kinetic one.It may happend converting to methanol because of fuelling increased.In any way, before seat area only small pressure drop may be permited.

To obtain more informations about ratios of flow areas,Fig 5 is presented.

When before the needle seat slope of any free area vs. needle lift becomes smaller than that of line a (see Fig 5),injector holder must be corrected.



Injector
DLLA 150 S 186

Fig 5 Flow areas vs. needle lift
a - needle seat flow area
b - entrance flow area of sack hole
c - nozzle holes flow area
a,b,c - geometrically defined
d - effective flow area of assambled injector

Coefficient of discharge μ_D vs. Reynolds number is presented in Fig 6.As parameter fuel pressure and relative needle lift h/h_{max} were used.

h/h_{max} means: measurements were performed without nozzle,thus this data are valid for nozzle holder.

For seat area given coefficient of discharge decreases with nozzle hole dia. increased.The later said has to be considered charging the fuel also.Fig 7 demonstrates the aforementimed.Fig 7 demonstrates also, that for test bench measurements of injector effective flow area is no use the increase the pressure drop over 100 bar (as well jet recommended in the 1st report).

With controlled single hole nozzle seat area must be seriously considered.To give more information about it Fig 8 is presented. Fig 8 demonstrates a high conversion of potential energy into kinetic form of the needle seat area,when low needle lift was set.In other words,lowering the needle lift fuel velocity incre-

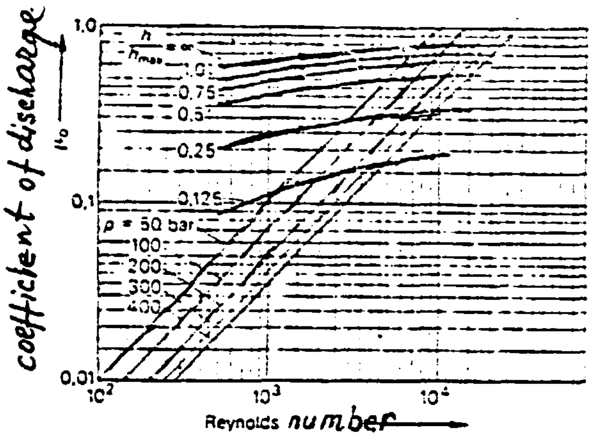


Fig 6 Coefficien of disharge μ_D vs. Reynolds number
DLLA 150 S 186 - injector

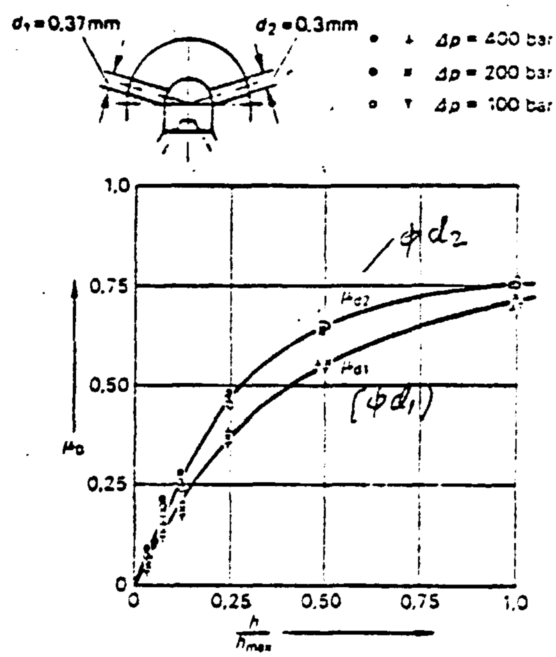


Fig 7 Coefficient of discharge μ_D vs. relative needle lift with nozzle hole dia. and pressure drop (Δp) at nozzle hole as parameters

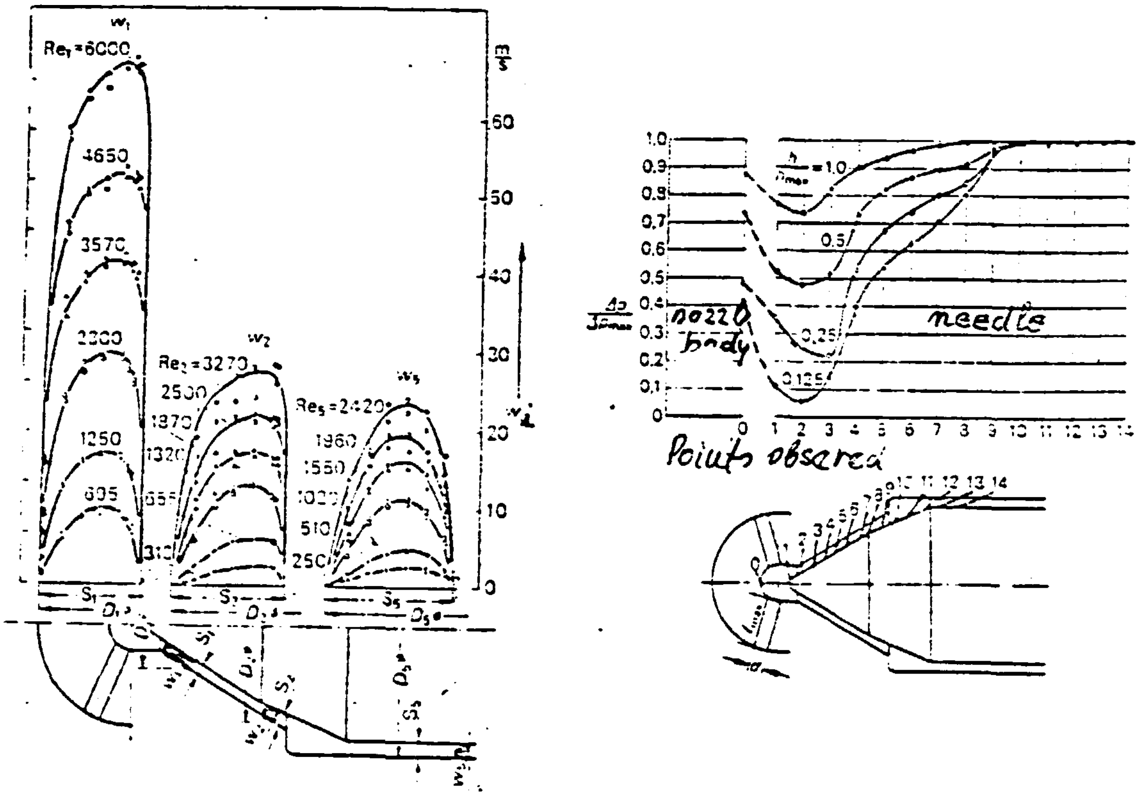


Fig 8 Velocity change along seat area for variuos pressure drops (on the left) and pressure change at various points in dependence of relative needle lift (on the right).The later measurements mentioned are valid for : nozzle hole dia. Ø 0,365 mm, Δp_{max}=400 bar, h_{max}=0,4 mm, l=1,5 mm.

ases drastically,but in seat flow area.This is the reason that controlled single hole injector was suggested by MAN.

For most practical cases:

$$2,5 \leqslant l/d \leqslant 4,5$$

$$\mu_D \approx 0,8 \quad \text{if} \quad \text{seat flow area} \geqslant 1,7 \cdot \text{nozzle flow area and } R_e \geqslant 10^4$$

Under above conditions the penetration of fuel spray can be obtain·d from the following equation:

$$S = 13,6 \left[\left(\frac{\Delta p}{\rho_g}\right)^{1/2} \cdot t \cdot d \right]^{1/2} \cdot \left(\frac{530}{T_g}\right)^{1/4}$$

where:

s - penetration distance of jet /in/

Δp - mean effective pressure drop across injector/lbf/sq.in/

ρ_g - density of gas in bomb or engine cylinder /lbm/cu.in/

t - injection time /sec/

d - injector orifice diameter /in/

T_g - mean temperature of gas in bomb or engine cylinder /R/

The above Eq. was derived for diesel fuel injected in quescent compressed air.However,calculating the fuel velocity and replacing the fuel next expression may be used:

$$S = \left[3 \cdot v_{jet} \cdot t \cdot d \cdot \left(\frac{F}{\rho_g}\right)^{1/2} \right]^{1/2}.$$

The both expression given for spray penetration may be considered as very approximative ones.The reason to show then was, first coarse estimation of spray-piston bowl relations.

With:

- book translated,explained and given
- training : theory and praxis
- practical measurements
- first report
- additional material sent as appendices
- three programmes for calculation given
- needle lift measurement completed
- second report

may be considered the engineers being dealing with fuel introduction are fearly well informed related to diesel fuel as well as methanol use.

7.2 Injector wear

Although injector wear was not included in the topics,during subsequent discussion related to lubrication and wear, some recommendations about injector wear were wanted.Thus HP-pump and - tube out of cosiderations.

Locations of injector wear (see Fig 9)

1. Needle seat
2. Nozzle holes
3. Needle-body sliding area of nozzle

4. Injector spring-seat area
5. Sealing surfaces on injector holes-nozzle body
6. Sack hole
7. Needle lift upper seat
8. Stem (b)-needle contact surface

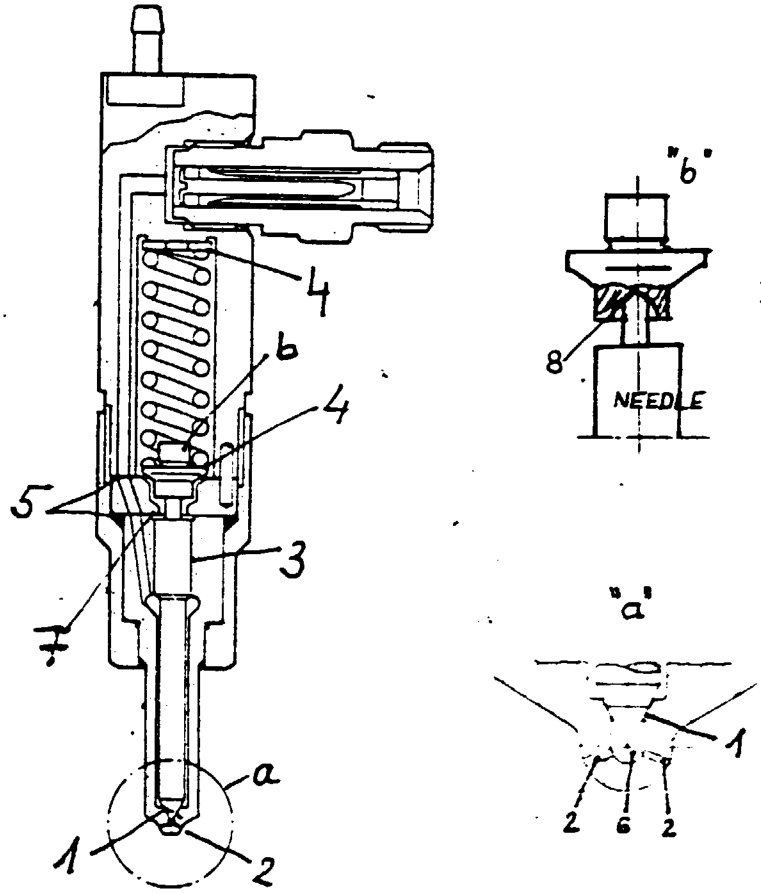


Fig 9 Locations of injector wear

<u>1. Needle seat</u>

A. Hammering of surface layer accompanied by wear
B. Contact corrosion
C. Erosion
D. Cavitation (in specific case only)
E. Chemical corrosion

<u>2. Nozzle hole</u>

A. Cavitation
B. Erosion
C. Chemical corrosion (Coking out of consideration)

<u>3. Nozzle needle-body sliding area</u>

A. Wear cause by sliding friction
B. Corrosion

4. Injector spring seat area

A. Contact corrosion

B. Chemical corrosion

5. Sealing surfaces on injector holder-nozzle body

A. Chemical corrosion

6. Sack hole

A. Cavitacion

7. Needle lift upper seat

A. Hammering of surface layer accompanied by wear

8. Stem needle contact surface

A. Contact corrosion

1A Hammering of surface layer accompanied

1B by wear and contact corrosion

a) Forces effecting needle movement may be balanced as :

$$\underbrace{m \cdot \ddot{h}_i}_{\text{inertia force}} + \underbrace{F_o + c \cdot h_i}_{\text{spring force}} \pm \underbrace{F_{tr}}_{\text{friction}} = \underbrace{p_{II} \frac{\pi d^2}{4}}_{\text{fuel pressure}}$$

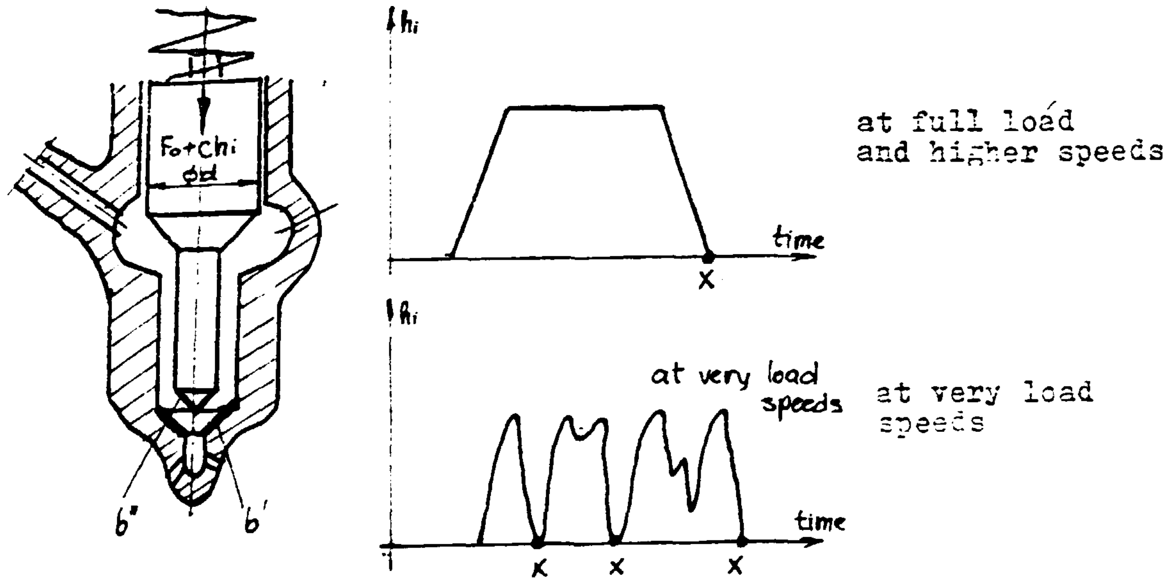


Fig 10

The both surfaces b′ and b′′ may be hommersed.

At closing (point x, Fig 10) masses of needle, stem and one half of spring have been accelerating toward the seat. Needle velocity becomes largest at point x (see Fig 10). Although, max. needle lift with high speed engines amounts 0,25 ÷ 0,5 mm, the hit force is quite high. The impact of the both surfaces, depending on thermal treatmen of the steels used,caused some small dislocation of the surface layer.At the same time contact corrosion attended as a small high cycle relative movements between two metalic surfaces.The later said is the main reason of seat wear.

With greater masses,higher speeds and lower stiffness of nozzle body seat area,contact corrosion may produce a very short service life of injector,esspecially when dirtyness in oil and bad surface texture in production are attended (see Fig 11).

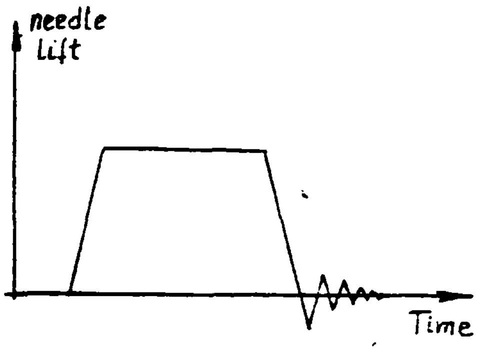


Fig 11 High cycle oscillation at seat area

1C,D,E Erosion,Cavitation and Chemical Corrosion

Fig 12 explains,cavitation wear in seat area may appear very rarely.

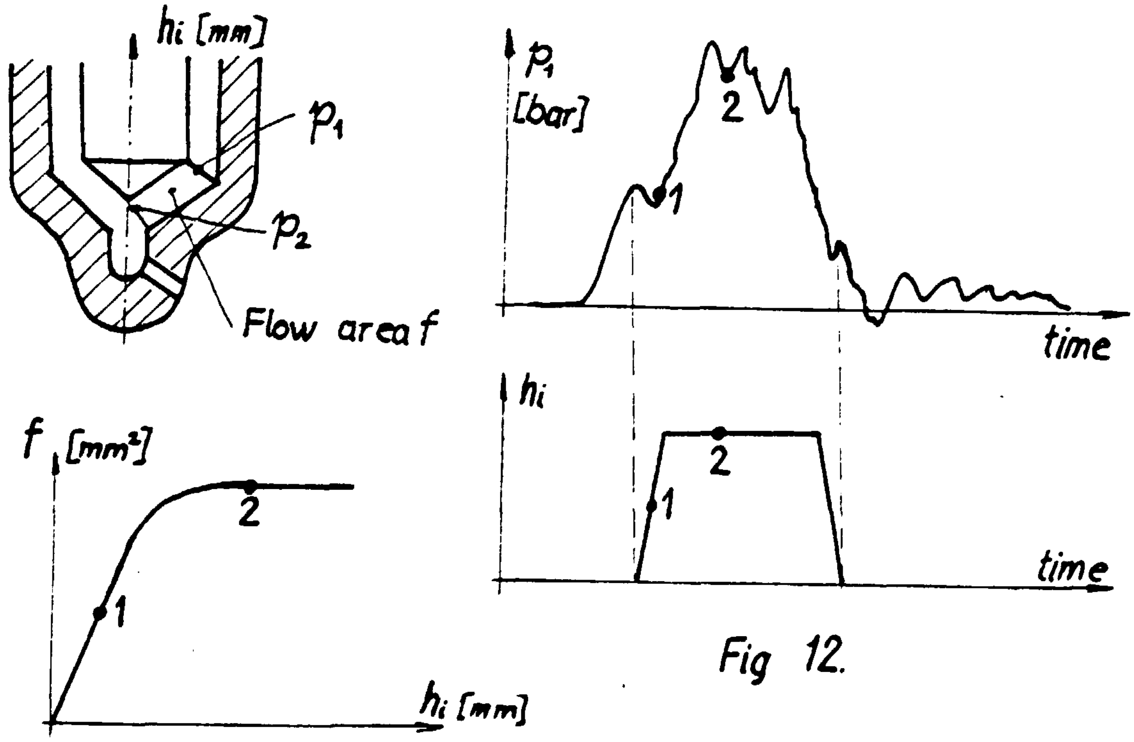


Fig 12.

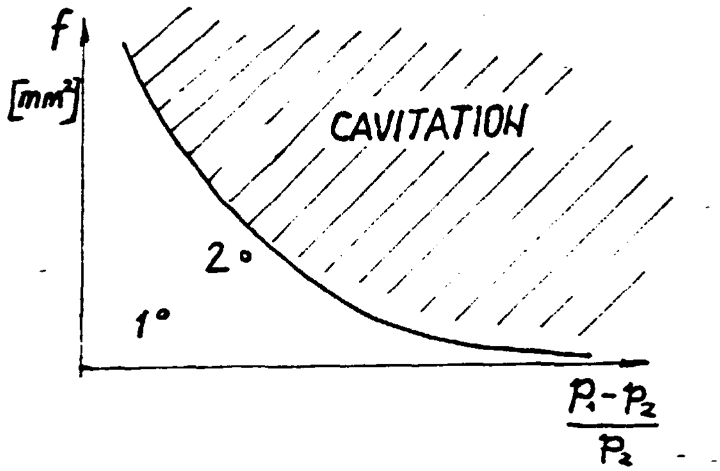


Fig 12

The reason for cavitation is very well known, thus we are not going into details. Erosion becomes higher with dirtyness in oil and is getting higher trough time. Chemical attack with diesel oil is mostly promoted if water presented.

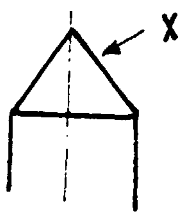
Shortcomings

The main drawback is lost of sealling at needle seat. Some increased max. needle lift is not important but accompanied decrease of nozzle opening pressure may be of influence.
Lost of sealing may produce :

- increased soot emission
- increased THC emission
- increased fuel consumption
- increased thermal loading
- increased cylinder-piston-rings wear
- decreased power output
- nozzle hole cocking

Inspection

1. Comparing the surface x of injector tested with new one



2. Increasing opening pressure of injector - p_o for 10 bar at first, than with constant oil

pressure - no checking if any droplet or leakage may be noticed at nozzle hole exit.

The same procedure should be done with increased p_o till 350 bar. Now at 300 bar checking, very small weting on the tip of nozzle body may be allowed.

2. Nozzle hole wear

2 A Cavitation

Cavitation wear will be considered for the both,nozzle holes and sack hole.Fig 13 depicts the cavitacion wear of

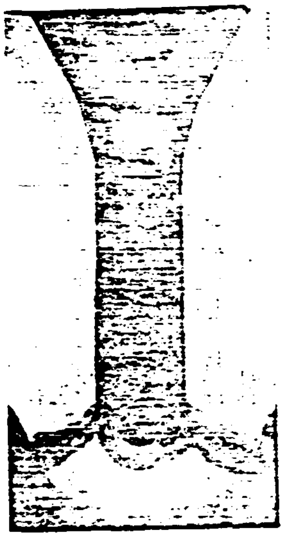
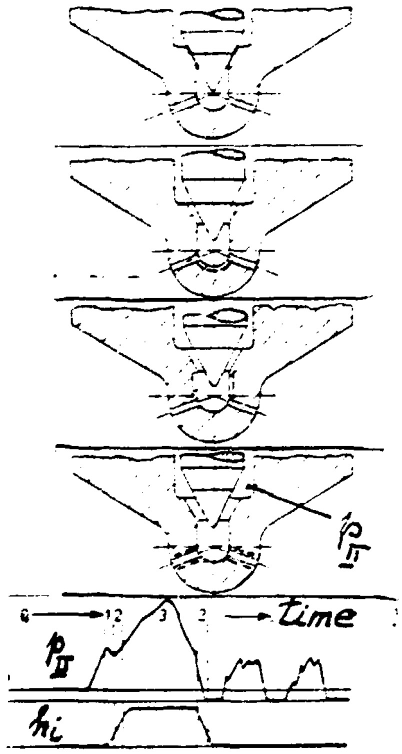


Fig 13 Cavitation wear in nozzle holes after 1000 $\underline{\underline{h}}$ of operation.



Phase 0
Before injection
Sack volume filled with gas

Phase 1
First opening phase of needle
Wear at the bottom of sack
---- ZONE ATTACKED

Phase 2
Flow cavitation narrowst
cross-sec. seat area
Cavitation attack at sack
circumference

Phase 3
Total pressure drop in
nozzle holes.Flow cavitation
in nozzle holes

p_{II}-fuel pressure

h_i -needle lift

Fig 14

nozzle holes after 1000 $\stackrel{h}{=}$ operation.Measured effective cross--sectional flow area of nozzle holes increased by factor 1,67. Unfortunately cavitation wear of nozzle holes may be experienced with some diesel engines.Converting to methanol cavitation wear may increase noticeably.

To give more insight in the cavitation wear of nozzle holes, Fig 14 is presented.Fig 14 cleary demonstrates the locations of potential wear of cavitation.Related to injection wear in nozzle holes is much more important then in sack hole.

Erosion in the nozzle hole increases with impurities content (in fuel) increase.

Concering chemical corrosion must be taken into account the temperature also as well as gas attacking.

Shortecomings

The main drawbacks:

- decrease of fuel injection pressure
- decrease of atomization intensity
- decrease of injection period
- decrease residual pressure
- increase in fuelling
- change in fuel spray penetration history .

The above cited results in:

- increased soot emission
- increased THC emission
- increased CO emission
- increased fuel consumption
- decreased power output
- piston demages
- late combustion and thermal overloading
- increase of noise.

Inspection

1.Before testings, if any change,the new injector has to be checked related to cavitation in nozzle hole.The same must be done converting to methanol.

Fuel pressure in injector has to be determined.It may be done by calculation or by measurements.By measurements or by calculation needle lift diagrame has to be accomplished also.

Having the both :pressure-and needle lift diagramme it's possible to determine three positions accor ding to Fig 14.

On rig test bench (see drawing given in the first report) $Q=f(h_i)$ measurements may be performed. However, here pressure drop must be accomodated to corresponding pressure diagrame.

$$\Delta Q = \mu f \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_1 - p_2} \cdot \Delta t$$

measured usually $2 \div 4\ dm^3$ — TO CALCULATE — measured /s/

ρ -fuel density

$p_1 - p_2 = \Delta p$ -pressure drop

μf -may be determined for any pressure drop Δp wanted

Doing that, results in diagrame shown in Fig 15. Marked points A' indicate the beginning of decrease in effective nozzle hole flow area (μf).

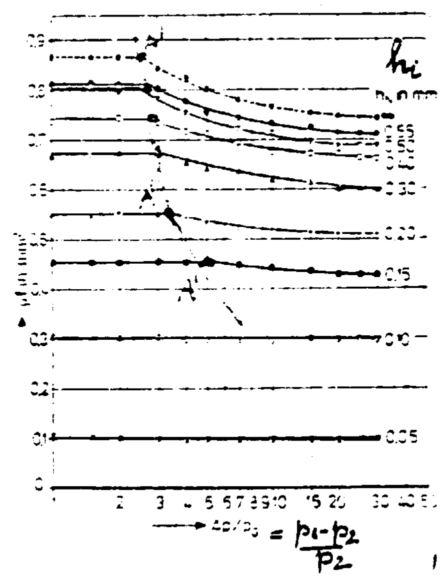


Fig 15 Marked points A' depicte start of cavities formation in the nozzle hole

p_1 - pressure in injector

p_2 - simulated in-cylinder pressure (for measurements only p_2=ambient)

μf -eff. cross-sectional flow area of nozzle holes

In order to avoid some uncertanities related to cavities formation in seat area, the additional measurements have to be performed.

Cutting the bottom part of nozzle body (by grinding only) μf area of needle seat can be measured, acc. to Fig 16. For the diagrame Fig 16 shown, μf seat area is not critical related to cavitation up to $h_i \approx 1$ mm. Comparing the both, Fig 15 and Fig 16, may be concluded, that nozzle holes are much more inclined to cavitation wear then needle seat area.

2. With injector tested the best inspection may be measurements at h_{imax} with Δp =100 bar=const., supposing that

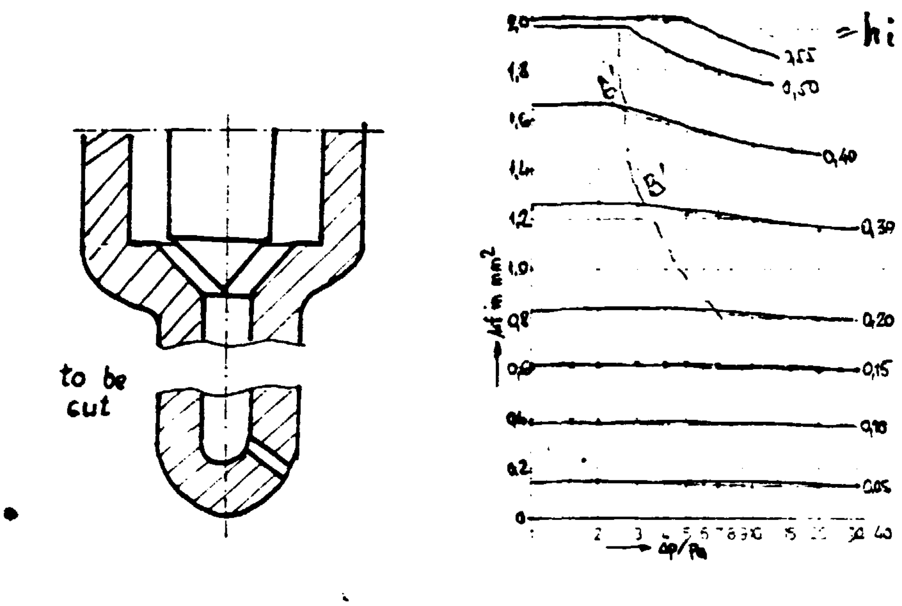
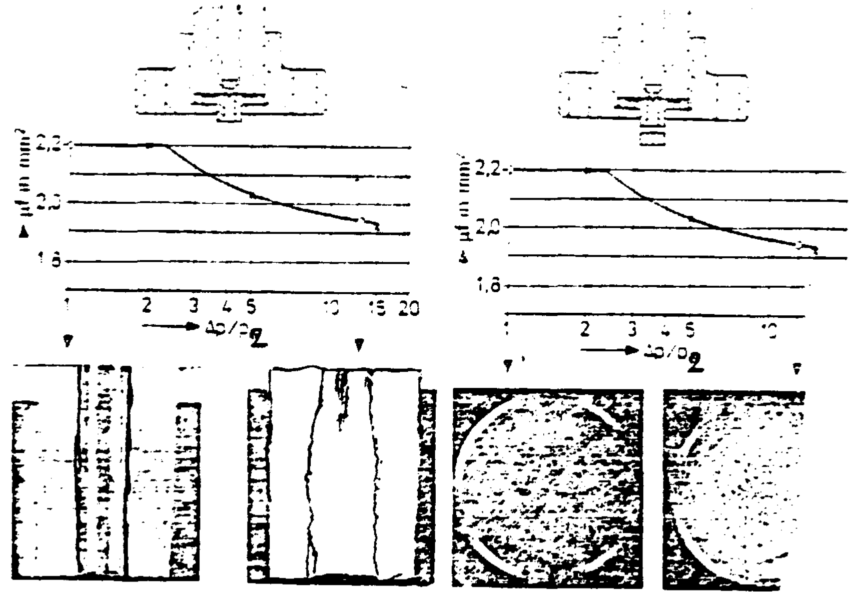


Fig 16

such data before testing existed also.By comparison wear of cavitation may be estimated.It's useful during testings,via μf measurements to follow cavitation wear vs. time (if existed). μf measurements mentioned are yet explained in Report 1.



p_1=200 bar	p_1=280 bar	p_1=200 bar	p_1=280 bar
p_2=100 bar	p_2=20 bar	p_2=100 bar	p_2=20 bar
wear=0	wear 17 mg	wear=0	wear 15 mg

to study cavitation at sack circumference

to study cavitation wear at sack bottom

Fig 17

Fig 17 explaines the procedure how to study cavitation at sack circumference and sack bottom.

<u>3. Needle - nozzle body sliding area</u>

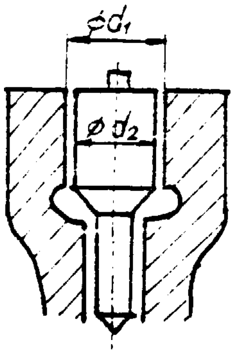
Here wear of sliding friction will be considered only. However, some negotiations must be made at first:

1. It's impossibile to measure the clearance between needle and nozzle hole (sliding hole) at various positions in order to calculate mean effective one. Morever, so defined says nothing about the leakage and reapitibility of. measurements is very poor.

To solve this problem the bellow shown method may be suggested.

Again
$$Q = \mu f \sqrt{\frac{2}{\rho}} \sqrt{\Delta p}\, \Delta t$$

substituing μ=1 results in
$$f = \frac{Q}{\sqrt{\frac{2}{\rho}}\sqrt{\Delta p}\,\Delta t} = \Delta d\,(2d_1 - \Delta d)\frac{\pi}{4}$$

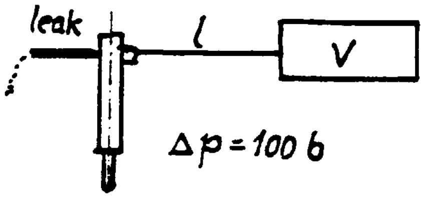


d_1 - producers nominal value let say d_1=6 mm.

This Δd may be defined more effective and easily measured as follows:

Q - overflow leakage quantity in time Δt at defined pressure drop Δp ($\Delta p \approx p_1$ because of p_2=1).

2. For every type of injector leakage time must be accomplished.



t_{min} = ... [s]

t_{max} = ... [s]

Volume V, line l as well as p must be the some as for engine producer. Time observation of leakage is not only because of to much overflow. Morever, the danger comes from to low leakage. Leakage overflow, although unwanted related to injection, is anavoidable related to cooling and to lubrication of sliding ports. Many stickings and seizures of injector needle are known becouse of to small overflow leakage.

3. One must be sure (or to check)that heat treatment of hardening was completed with very deep cooling ($-55^{\circ}C \div 5\ h$). This is important treatment for the both: needle and nozzle body.The reasion for that is rest-austenit which by increased heating may be transformed into martensit.This process is accompanied (per example) by increase of needle dia.,which may produce total loss of gap (Δ d=0).If happend,extreme high friction produces overheating and seiure.

A high needle wear had occured in one type of vehicular diesel engine with new injector supplier.The inspection had performed acc. to Fig 18.Measurements gave evidence about dobtfull heat treatment of needle (see Fig 18,b) although the diameter of nozzle body was "stable" during heating up period (Fig 18,a).Thus needle - nozzle body gap may decrease in operation (see Fig 18,d) what in the praxis at random occured. The reason for above investigations were:

- two needle seizure in servis operation
- unstable idling with about fifty engines accompanied with increased fuel consumption rate in low load operation.

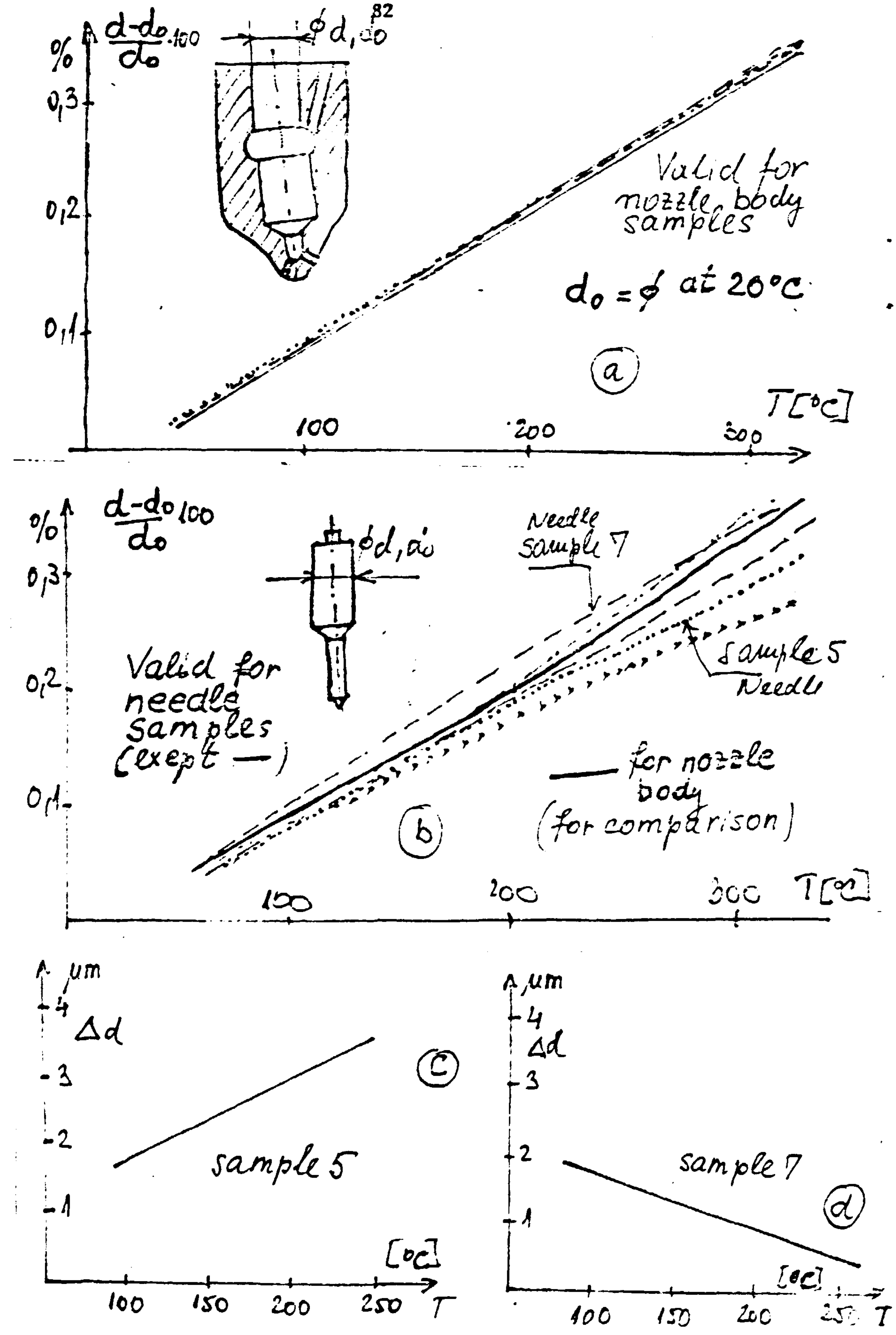
4. Any kind of deformation of-nozzle body or needle increases friction accompanied with wear inseased.

a) The cap x in Fig 19 must be tighten carefully with order torque.The same may be recommended when fasten injector - cylinder head.A high friction may be noticed with many injectors acc. Fig 20.

The reason for that, is to high torque by fastening in cylinder head or by injector assembling.

b) Any overheating may produce increased friction and inversly increased friction produces higher heating in sliding area.

Poor cooling of cylinder head,etc.,may produce overheating but with increased wear of sliding part especially at low speed and low load operation, overheating may occur because of hot gas penetration in the space y (see Fig 21).When instanteneons fuel pressures drop under in-cylinder gas pressures (see Fig 21) for any time (or angle α_K),hot gases penetrate into space y accompanied by: overheating,needle "colouring" and in the end,by nozzle hole cocking.(Till now 17 reasons are known related to nozzle hole cocking).



% $\frac{d-d_o}{d_o}\cdot 100$
$\phi d, d_o$
0,3
0,2
0,1
Valid for nozzle body samples
$d_o = \phi$ at 20°C
a
T [°C]
100
200
300
% $\frac{d-d_o}{d_o}\cdot 100$
$\phi d, d_o$
0,3
0,2
0,1
Needle sample 7
sample 5 Needle
Valid for needle samples (exept —)
for nozzle body (for comparison)
b
100
200
300
T [°C]
µm
4
Δd
3
2
1
C
sample 5
[°C]
100
150
200
250
T
µm
4
Δd
3
2
1
sample 7
d
[°C]
100
150
200
250
T

Fig 18

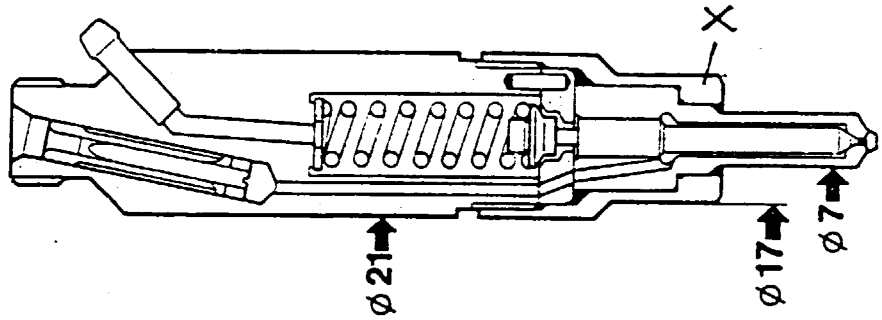


Fig 19

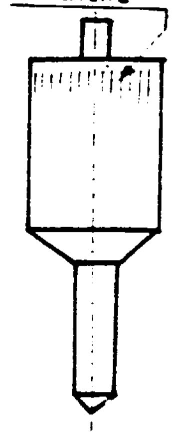


Fig 20

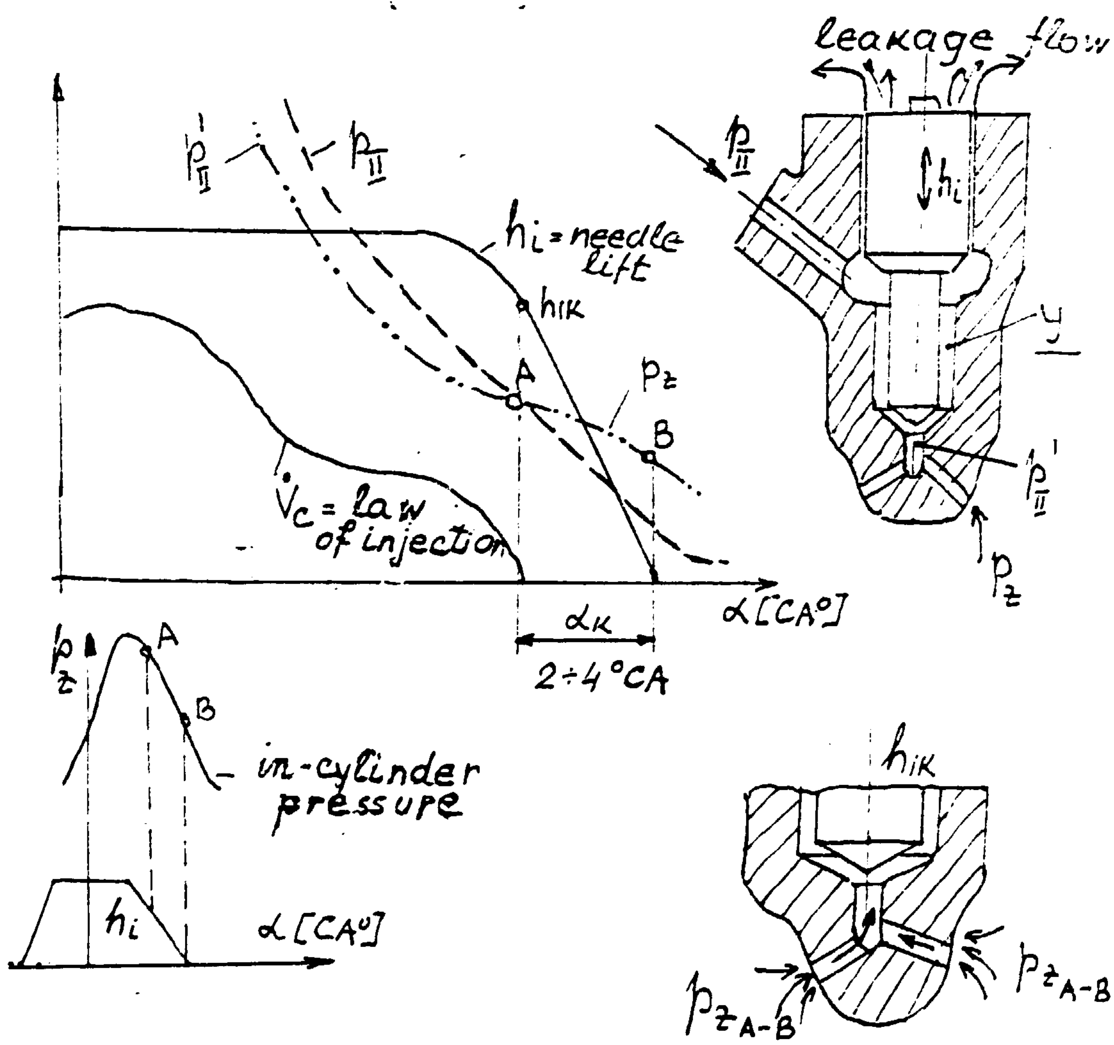


Fig 21

5. According to Fig 22, injection fuel system is coupled hydraulically. During injection, leakage flow although very small, is

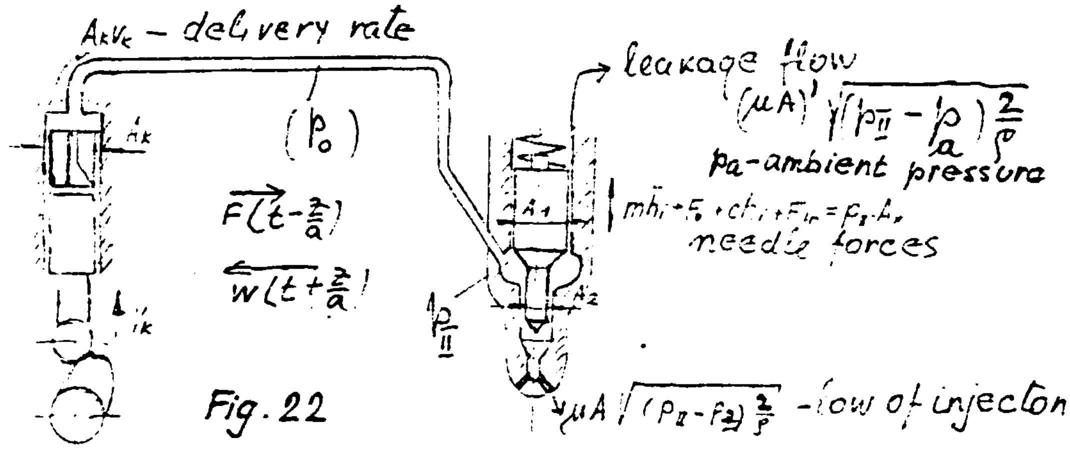


Fig 22

only one connection of HP system with ambient. When leakage flow increases:

$$\int (\mu A)'\sqrt{(p_{II}-p_a)\frac{2}{\rho}}$$

fuelling decreases $\longrightarrow$ $V_c = \int \mu A\sqrt{(p_{II}-p_a)\frac{2}{\rho}}$

Although, leakage flow rate must be higher at higher speeds related to higher pressures p_{II}, the fuelling corrected will be lower beacuse of time (see Fig 23).

Residual pressure p_o will be influenced also and unwanted corrected. The time between two cycles is a long period and the closed HP system has still connection with ambient via leakage flow. It may have a high influence in methanol application as was (in the first Report) yet explained. A high wear and lower fuel density, accompanied with CH_3OH evaporization in the upper part of sliding area, may produce leakage flow, at low speed--low load operations, more then 20 % of corresponding fueling.

To avoid the abovecited leakless injector may be the solution (still in developement). Fig 24 shows diag. sketch of leakless injector. The first results have been encouraging (soot emission decreased, torque back up increased, injector is getting smaller, higher injector closing pressure, less expensive system). However, retraction volume of the retraction valve must be matched to the demands of the closed system as well as the volume V_A (see Fig 24). When matching was poor (Fig 24, a and b dated lines) resulted in after injection and nedlle blocking.

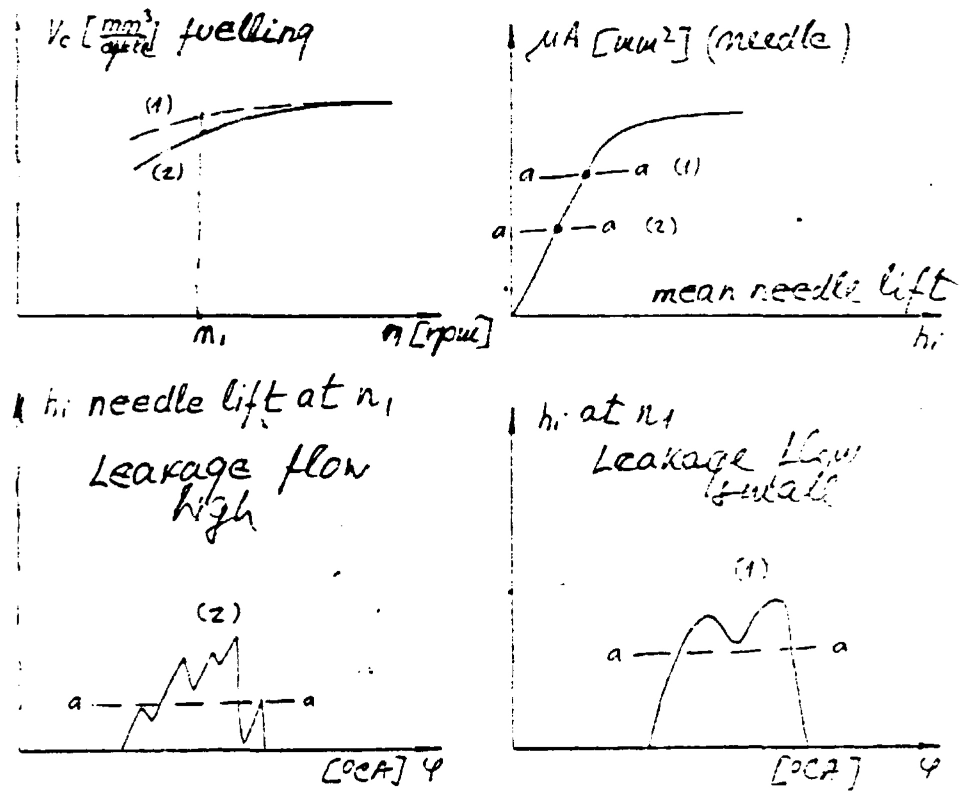


Fig 23

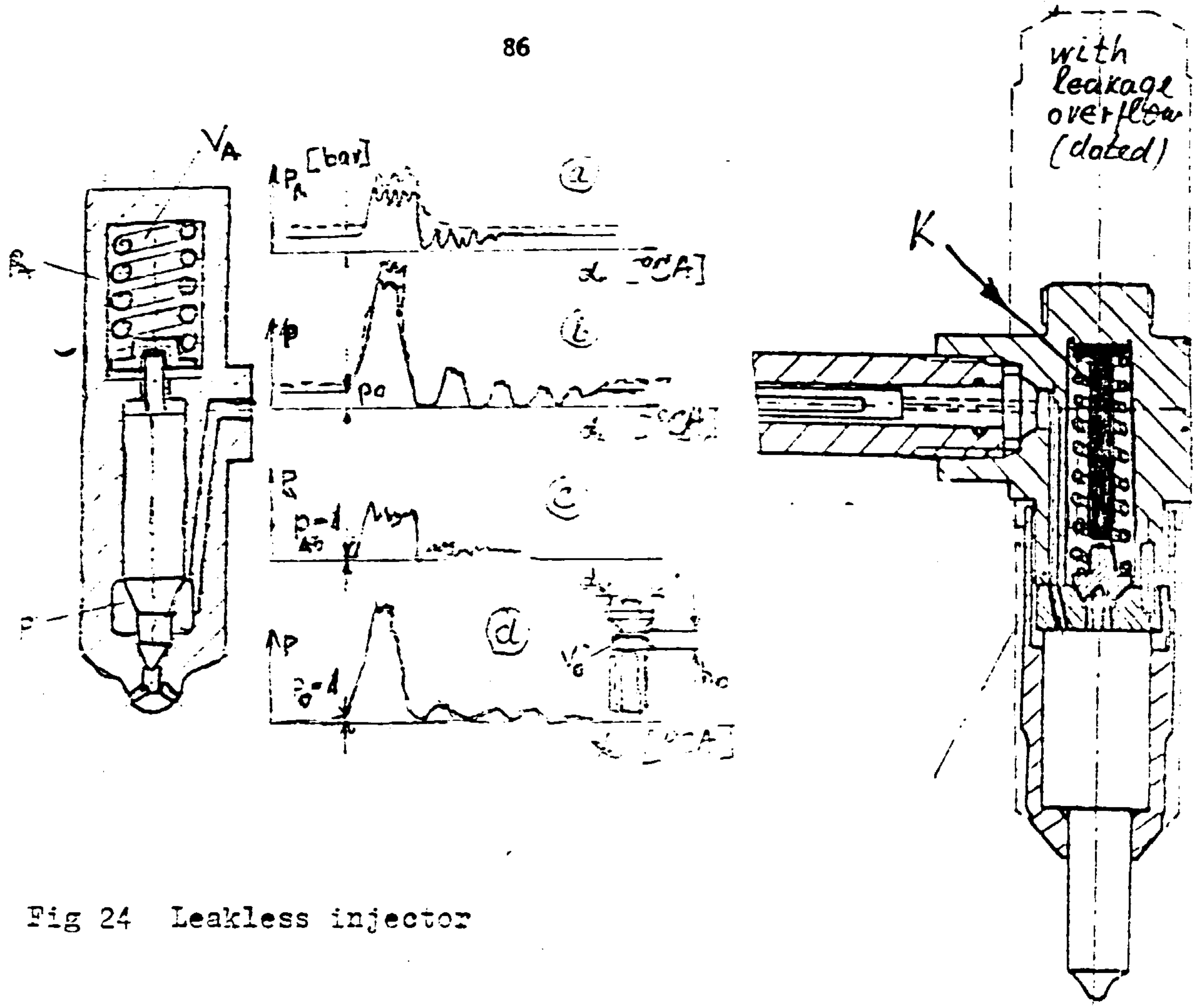


Fig 24 Leakless injector

With good matching (see c and d, Fig 24) a promissing results are achieved.To decrease V_A as much as possible filler K was used.

6. Using experiments to define eff. leakage flow area μA vs. pressure p_{II} the calculation can be made for the all injection parameters wanted. It may be a small modul added to the over-all injection fuel programme.It's to be suggested to use the 1st programme sent as much accurate one.

Is to be mentioned that wear increase with higher content of very fine dispersed dirtyness in diesel oil.

Shortcomings

The main drawbacs:

- change in fuelling
- unwanted change in residual pressure
- change in injection pressure
- unstable injection parameters at low speed-low load operation
- load dispersion at low speeds

The above cited results in :

- soot emission increase at low speeds
- poor torque back up
- nozzle hole dockings
- increase of fuel consumption rate
- unstable idling operation
- increase in CO emission at low speeds
- some decrease in power output

Inspection

The points mention in negotiotions 1 - 5 said all about inspection.The most important are leakage flow measurements and pressure drop in time.Doing that Fig's 25 and 26 may be obtained.

Fig 25 shows the time elapsed in /s/ vs. diameter clearence Δd for pressure drop 350-300=50 bar.This diagrame may be used for μA determination.

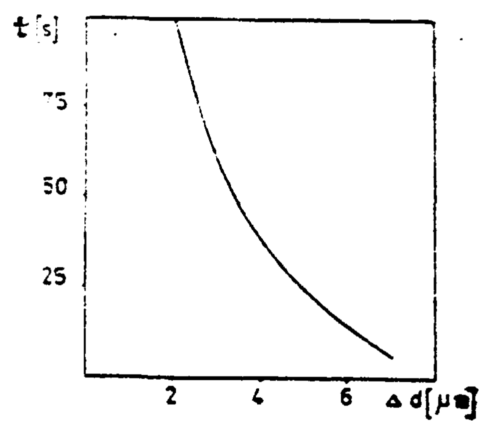


Fig 25 Time t elapsed for 50 bar pressure drop vs. diametrical clearance needle-nozzle body

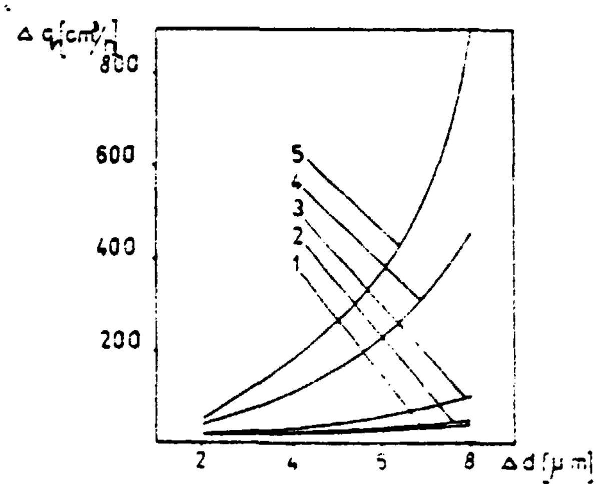


Fig 26 Relationship of fuel leakage in needle-nozzle assembly, against diametral clearance for various rotating speeds
5 - 1100 rpm HP pump
1 - 700 rpm HP pump

In the end,related to other types of injector wear,may be said; they are much more concentrated foward change of injector opening as well as closing pressure.If may effects the whole injection events but also the timing.

However, related to engine performances, the largest impact may have decreased closing pressure of injector. With decreased pressure difference at the end of injection, fuel is poor dispersed and introduced in the very favourable ambient for cracking having no time to complete the combustion. Nozzle hole cocking and a high soot emission are the results (see Fig 27).

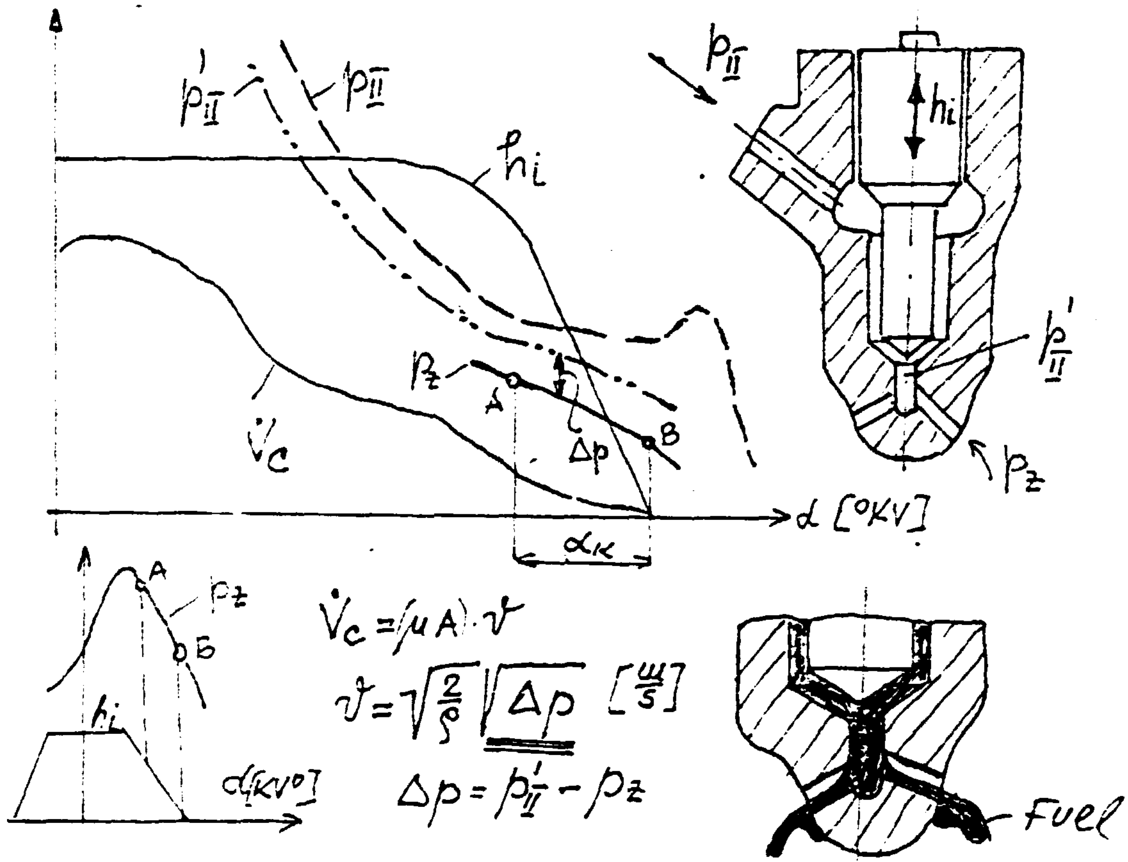


Fig 27 Poor dispersion of the fuel because of a low pressure difference at the end of injection (low closing pressure).

Is to be mentioned, that the decrease of injector opening (as well as closing) pressure may be effects by spring itself but the reasion is often the wear on the spring on needle contact surfaces. The inspection is very easy to performe by checking the injector opening pressure change.

A short overview - system testings

In endurance tests the service life of fuel injection system under fuel load conditions may be determined depending upon the main influencing factors of peak pressure and (constant) rotational speed.

However, in practice, it is the service life expressed in hours run (h) or in distance driven (km or miles) which is of interest.The prerequisite for determining the service life is the knowledge of data given the percentage in distribu_tion of load during operation in field.A simple classification unit suitable for field-use,was taken to determine this load distribution.Using this method,a two-dimensional time distribution can be obtained for the control-rod travel/rotational-speed performance characteristics for all the possibile applications of the pump.

Using load-collectives,and the theory of the linear failure accumulation according to Palmgren-Miner,a procedure was developed for determining the service life from the full-load life.

The relationship of service life to full-load life represents a characterization of the collective.Multipication with the mean driven speed results in the so-colled "fuel-load distance"

$$VZ\ \frac{\text{service /h/}}{\text{full load/h/}}\quad V_m\ \frac{\text{(km)service}}{\text{service /h/}} = VS\ \frac{\text{(km) service}}{\text{full load /h/}}$$

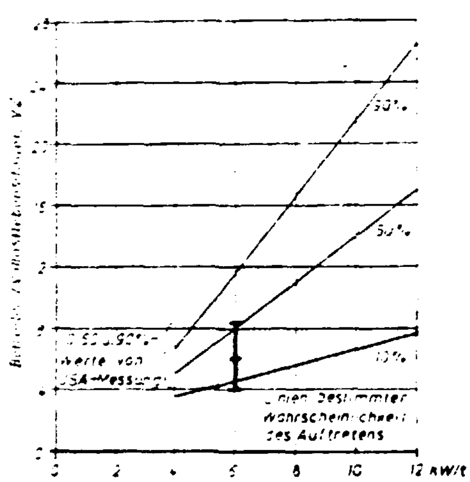


Fig 28 Relationship of service to full-load life VZ versus rated output total weight(statistical evoluation of 96 results with trucks).

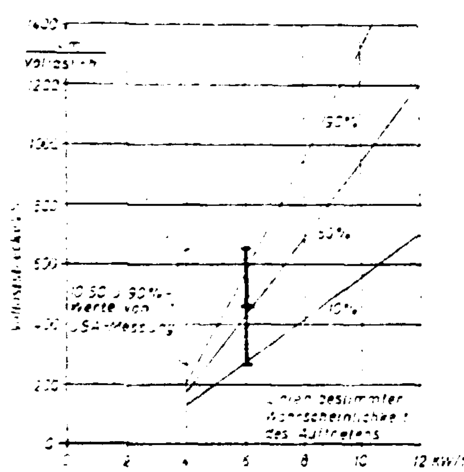


Fig 29 Full-load distance VS versus rated output/total weight

<u>Example</u> Truck 6 kW/t at 10 % VS_{10}=270

Service life 500.000 km, testing for injection system

$\frac{500.000}{270} = 1850$ h

VZ_{10}=4,5 The method shown is more applicable for the pump. For the whole system we are using the next shown procedure 1000 h.

1) 35% $0,85\ M_{max}$ $n = 2/3\ n_{M_{max}} + 1/3\ n_{P_{emax}}$

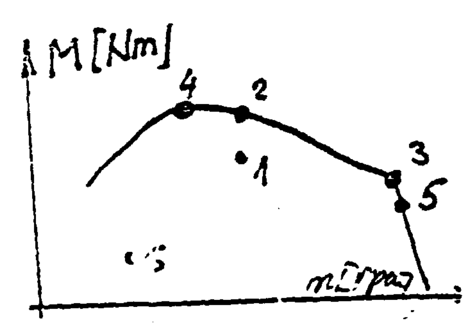
2) 15% full load " " "

3) 15% P_{emax} $n_{P_{emax}}$

4) 20% M_{max} $n_{M_{max}}$

5) 5% $0,8\ P_{emax}$ $1,03\ n_{P_{emax}}$

6) 10% idling n_{min}



TOPIC 8

Proposlas related to: "Glow plug supported combustion in neat methanol operation"

1. CR- Compression ratio

In-cylinder peak pressure measured demonstrates very low values related to DI diesel engine operation; peak pressure reached hardly 50 bar at rated power. It means, besides of relatively low rated mean effective pressure, that in methanol operation compensation of CR drop was not applied.

Peak of compression pressure for CR=16,5 is quite enought high for diesel oil operation. However, converting to methanol, because of 10 times more energy needed for mixture formation than that in diesel oil operation, effective peak compression pressure will be considerably lower. This fact askes for CR compesation when CH_3OH is to be used as fuel. The more CH_3OH doped and mixed with diesel oil the larger CR compensation requirement. In the case of neat methanol CR compensation requirement reachs the max value (Fig 1).

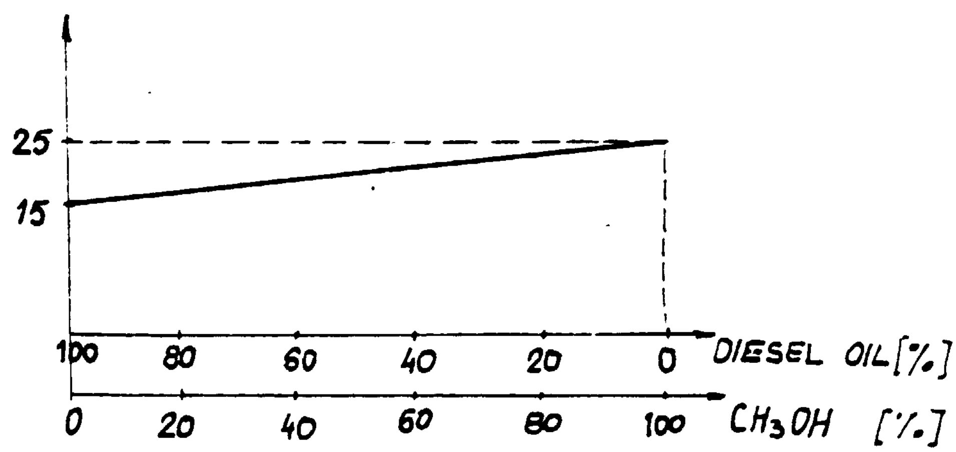


Fig 1 CR sufficient for stable ignition at starting when diesel oil-CH_3OH blends are used in diesel engine (without ignition improver)

Some disproportions may be noticed by peak in-cylinder pressure and neat peak compression pressure ratios.

A simple calculation may show that neat peak compression pressure amounts of about 40 bar at rated power. It means that oforementioned ratio amounts 1,25, what is pritty low for DI engine. Morever, in-cylinder pressure history may be as shown in Fig 2.

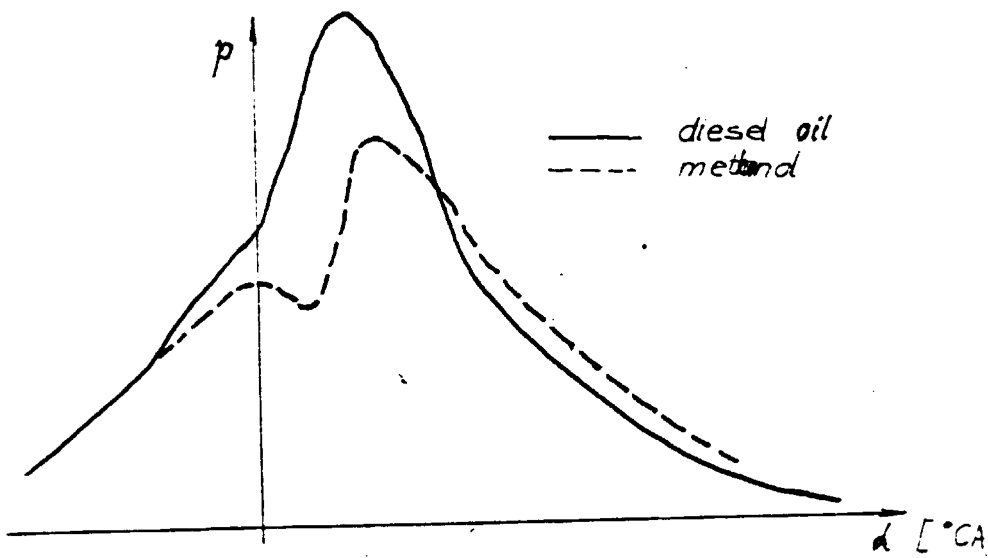


Fig 2 In-cylinder pressure diagrames:

——————— diesel oil

- - - - - - - - methanol

The in-cylinder pressure diagrame for methanol combustion shown in Fig results in:
- late combustion
- small amount of $\int p\dot{x}$
  (x - instanteneous piston velocity)
- unstable low load operation.

CR compensation is also required when using methanol in ST engines.

To demonstrate the later said, some VW experimental results are presented. VW used the both techniques with methanol: carburation and fuel injection. The results of both techniques mentioned as well as results for gasoline use are shown.

Piston temperature measurements were made with a 1,6 l VW gasoline engine. The engine was run on the test bed in different versions with regard to mixture formation according to the use of different fuels, such as gasoline (regular), CH_3OH, M 15 (15% CH_3OH + 85% gasoline) and ethanol.

It was found, that with M15 (only 15 % CH_3OH was mixed) piston temperatures were about 10°C lower as compared to engine operation on gasoline, provided the engine build up was kept practically unchanged.

The temperature rise found with the carburetor version, when this engine was operating on CH_3OH fuel, is due to the increased compression ratio 12,5 compared to 8,5. As the piston temperature depends on combustion temperature being lower with alcohol fuels than with gasoline fuel, the influence of increased compression ratio is partially compensated.

A further increase of the compression ratio towards 13,4 was achieved using fuel injection version (5 points compared with gasoline operating on CH_3OH with ϵ increased, the thermal efficiency was evident. Comparing with gasoline 4 % power output increase was measured.

Exhaust temperatures were 90 ÷ 120°C lower with CO emission increased.

The Fig's 3 and 4 show with gasoline engine also, that "last of compression" must be compesated. It's no use to fray CH_3OH application without CR compesation.

This appendix was presented only to demonstrate that the same happend with SI engine also related to CR lost by CH_3OH application in "existing engines".

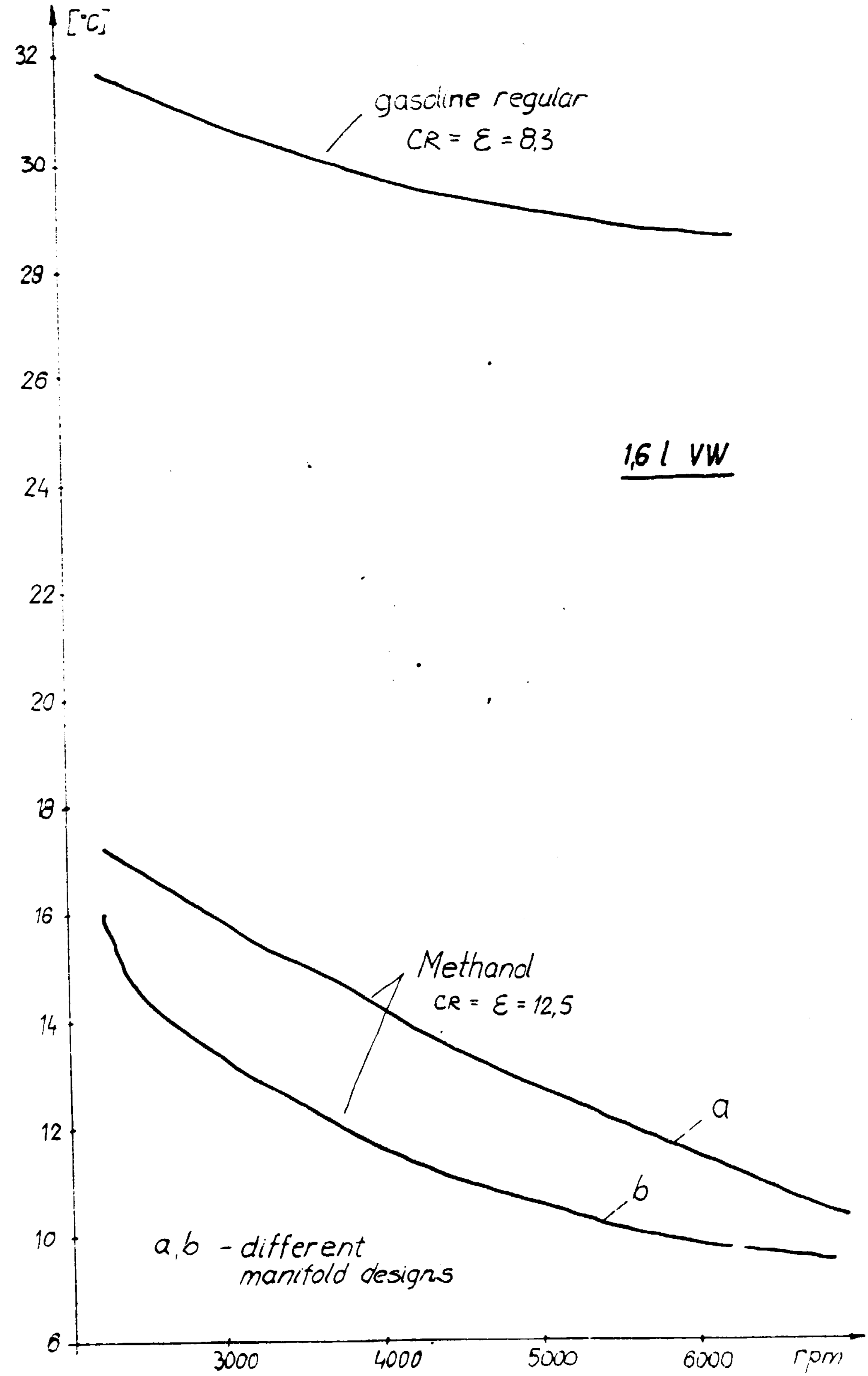
Algthouh Fig 4 presents piston temperatures for SI engine, CR was increased by factor 1,6 in CH_3OH operation comparing with gasoline. Only with this increase top piston temperatures were the same for the both fuels.

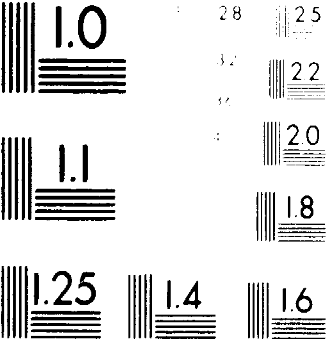
With diesel engine in CH_3OH application factor 1,6 results in CR=25,6 for the same top piston temperatures (t_1) as in diesel oil operation. This very approximate comparison demonstrated the need for same CR compensation in methanol operation.

Propolsal Converting to methanol CR ratio must be increased for 2 units if glow plug is to be used.

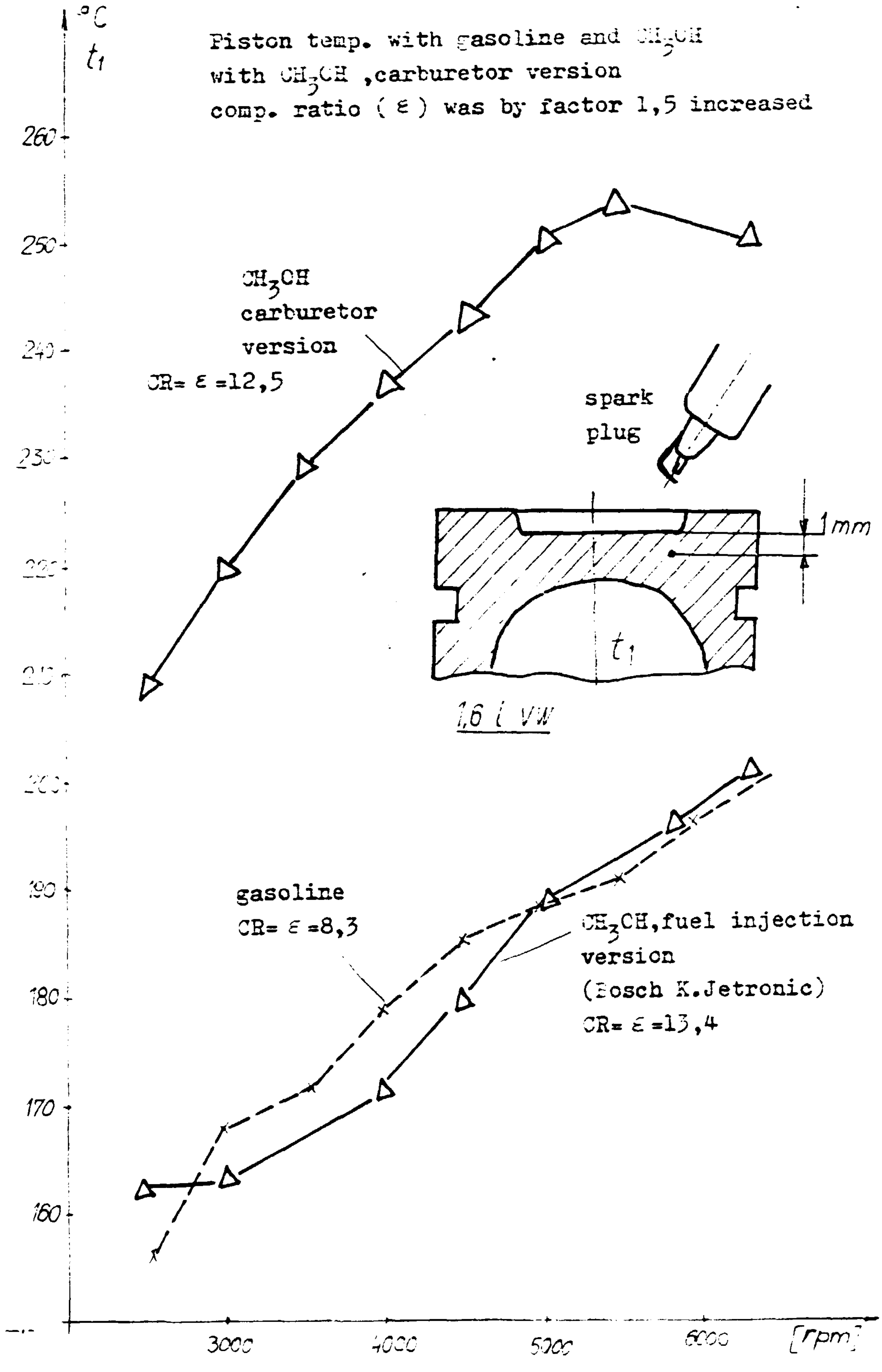
Increased CR ratio may help to:

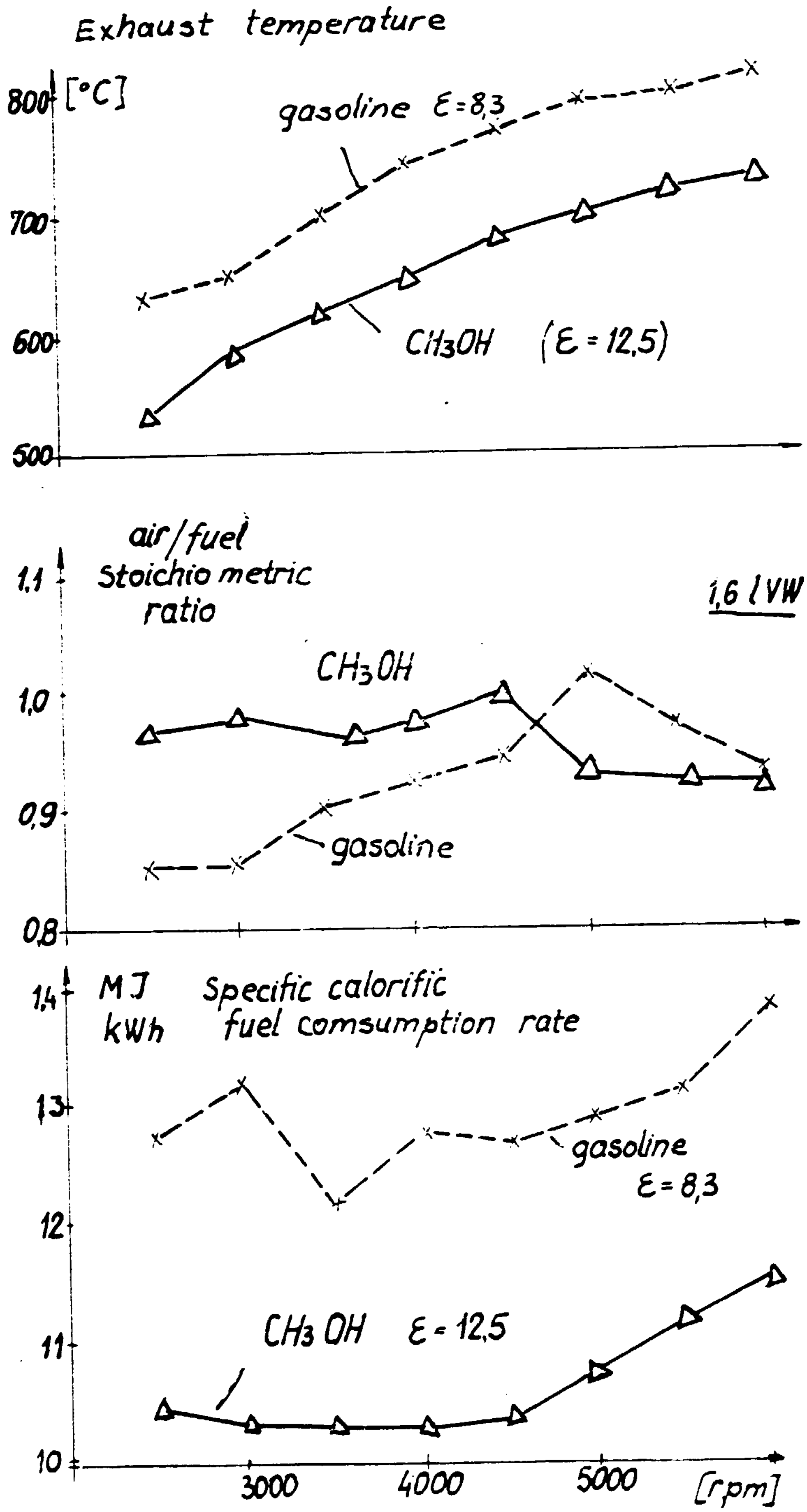
- improve startability
- decrease operation of missfiring
- decrease fuel consumption rate
- decrease energy consumption of glow plug


[°C]
32
30
29
26
24
22
20
18
16
14
12
10
6
gasoline regular
CR = ε = 8,3
1,6 l VW
Methanol
CR = ε = 12,5
a
b
a,b - different
manifold designs
3000
4000
5000
6000
rpm



MICROCOPY RESOLUTION TEST CHART


°C
t_1
Piston temp. with gasoline and CH_3OH
with CH_3OH ,carburetor version
comp. ratio (ε) was by factor 1,5 increased
260
250
240
230
220
210
200
190
180
170
160
CH_3OH
carburetor
version
CR= ε =12,5
spark
plug
1mm
t_1
1,6 l VW
gasoline
CR= ε =8,3
CH_3OH,fuel injection
version
(Bosch K.Jetronic)
CR= ε =13,4
3000
4000
5000
6000
[rpm]


Exhaust temperature
[°C]
800
700
600
500
gasoline ε=8,3
CH3OH (ε = 12,5)
air/fuel
Stoichio metric
ratio
1,1
1,0
0,9
0,8
1,6 l VW
CH3OH
gasoline
MJ
kWh
Specific calorific
fuel comsumption rate
1,4
13
12
11
10
gasoline
ε = 8,3
CH3 OH ε = 12,5
3000
4000
5000
[rpm]

2. Injection

More advanced injection to ignite methanol is a consequence of:

- high ignition delay of methanol
- large quantity of fuel injected in the combustion chamber before ignition
- start of injection in less convinient conditions for ignition

The solution may be controlled injection as was yet explained in this report.Controlled injection is not so difficulty to realize using two injector springs or the injector with hydraulically controlled needle lift.The both systems are shown as well as explained in this report.Using controlled injection,the fuelling in the pre-ignition period decrease same does in-cylinder temperature drop also. Thus ,better ignition and improved low--load operation may be expected.

In this report,the system with single hole injector for controlled injection is shown also.However,to apply single hole nozzle swirl ratio and combustion chamber bowl must be matched.Using existing engine some of controlled injection effect may be realized with multi-hole nozzle also.

Number and distribution of nozzle holes have a great influence on diesel engine operation.This influence increases with energy for mixture formation increased,as is the case in CH_3OH operation.It's of interest to remember,that in methanol operation the best results were obtained when the same injection system was used as for diesel fuel injection.It means nearly by factor 2 prolongated injection period.This fact results in late combustion only.Converting to methanol the ground rule may be:

$$\alpha_{\substack{\text{period of injec.}\\\text{diesel oil}}} \approx \alpha_{\substack{\text{period of injec.}\\\text{methanol}}} \qquad (1)$$

if no change in the mixture formation process.

In the most suceesful methanol diesel engine L9204FMV (see p.232 "Neuen Krafftstolten out der Spur")MAN by first matching, doubled plunger dia. and nozzle hole cross-sec. area comparing with diesel fuel operation (engine L9204FM). Ricardo institute in "Ricardo news" reported,that converting to methanol the capacity of fuel injection system must be increased..or the mixture process unchanged,our experiments showed,that the optimum may be reached nearly for the condition given by Eq 1.

Thus injection rate must be increased comparing with diesel oil injection.However,the whole injection system must be matched,acc. to increased fuelling and injection rate (retraction volume,dead volumes,plunger pre-lift setting etc).

3. Swirl and sqwish (SR and SQ)

With some combustion chambers high SR intensity produces a high heat transfer to the combustion chamber walls and with this unwanted cooling of compressed air.It decrease the startability as well as combustion stability in low load operations. Swirl intensity may be reduced in methanol operation because of higer velocity of-flame propagation and combustion(non-sooty flame).But decrease of swirl may help to reduce the appearance of missfiring and to increase the combustion stability with load decreased.

As a rule of thumb,it may be suggested:

- converting to methanol SR may be reduced in amount of 20%.

4. Combustion chamber isolation

Cast iron instead of alu-alloy piston may help to conserve more heat into combustion chamber.Doing this,the next characteristics will be improved:

- startability
- low load operation
- reduction of fuel consumption because of additional heat returned to the combustion process.

The above propolsal may not be of any problem, nearly "Kirloskor" reported,that just in the engine considered a new cast iron piston was applied (see Appendix sent).

About 3000 swiss franks costs ceramic overlay on the piston and on the cylinder head ("Plasma-Technik" Swiss),but is no use when any producer in India does'nt exist.However,it'll be of interest to collect information in India where plasma-technique has been practisized.If any, it's possibly to get much better isolation.To be mentioned,that some tool producers have been using ceramic plasma-coating to increase the service life of their tools.

5. Switching on technique

It's well estabilished fact that using methanol the next points must be considered:
- starting
- low load operation
- cold operation
- transients

when methanol used in diesel engine.

Using mixing technique, the ignition quality of diesel oil decreases in the presence of methanol (becouse of cooling effect).

If extra good isolation of combustion space does'nt exist, using glow plug, stable starting and low load operation without missfiring could'nt be realized.

For good starting switching on technique may be proposed , although not for vehiculor application. However, per example, for watering purpose it's still possibly to apply switching-on technique. Switching-on technique enables during starting to use diesel oil only. In low load operation with load increased quantity of methanol doped increses also. In full load operation methanol only is injected. A diagrammatic sketch of switching--on technique is shown in Fig 6.

The piston 1 (see Fig 6) is connected with control rack of HP pump. The engine considered has a single governed speed, thus rack position is load dependent so does the piston 1 also. The fuel volume "closed" between the inlet of the pump sump and fuel distribution FD, must be as small as possible. When sump of HP pump are good a very good mixing of fuels mixed may be obtained. This mixing is produced by filling and spilling process (Fig 7). Pressure diagrame shown in Fig 7 depictes that in the sump of HP pump a high fuel turbulence exists. However, the temperature in the sump must be low because of methanol presented. Missfiring may be produced by methanol cavities promotion in the sump also.

With the small single cylinder engine the low pressure pump was'nt use, but because of cavities formation, if high temperature, sump volume must be pressed. With low pressure pump and overflow valve, the fuel pressure in the pump sump may be kept over CH_3OH evaporization point. Overflow valve and short circut

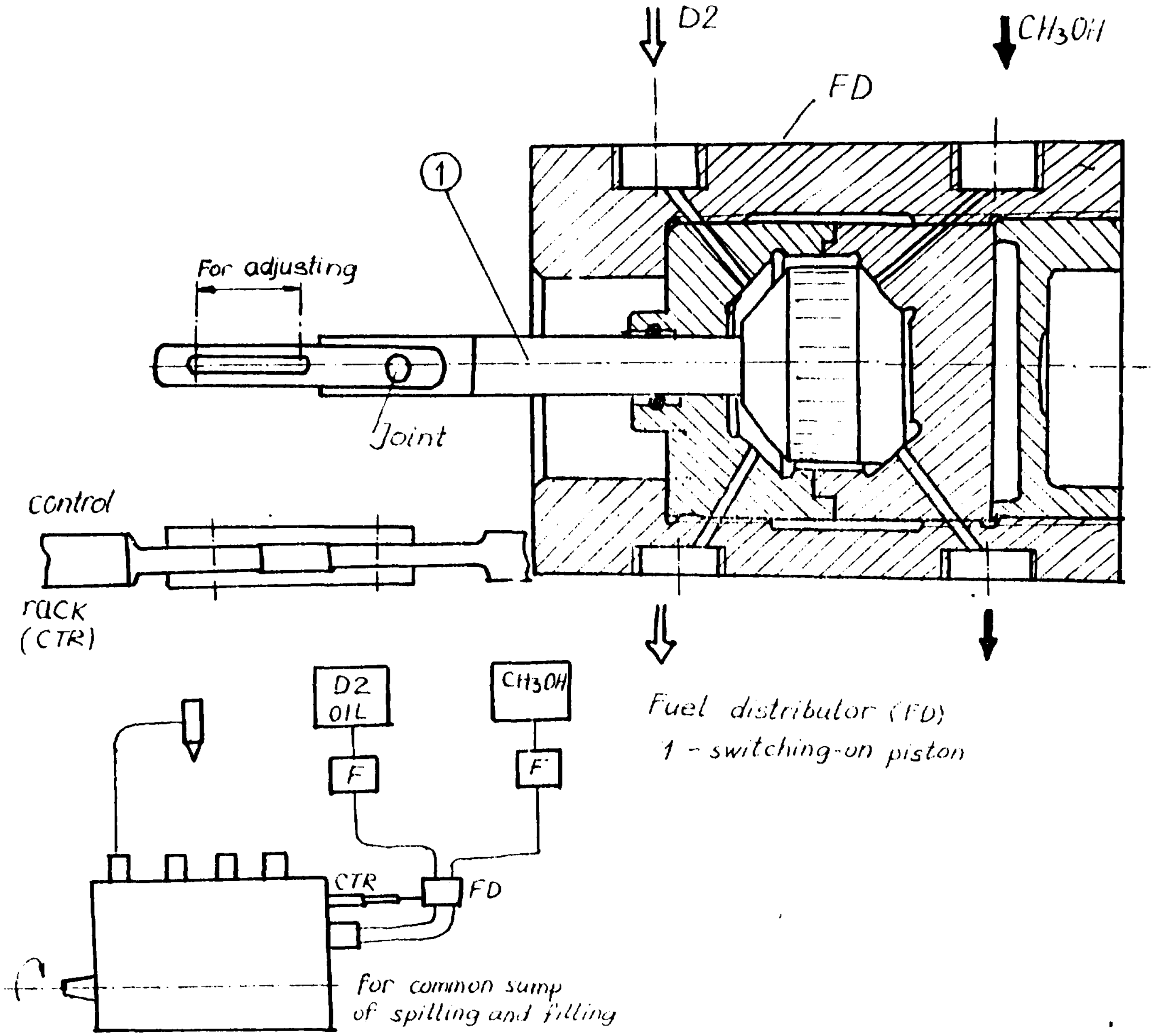


Fig 6 Switching-on technique

of low pressure system help in better mixing as well as in better colling.

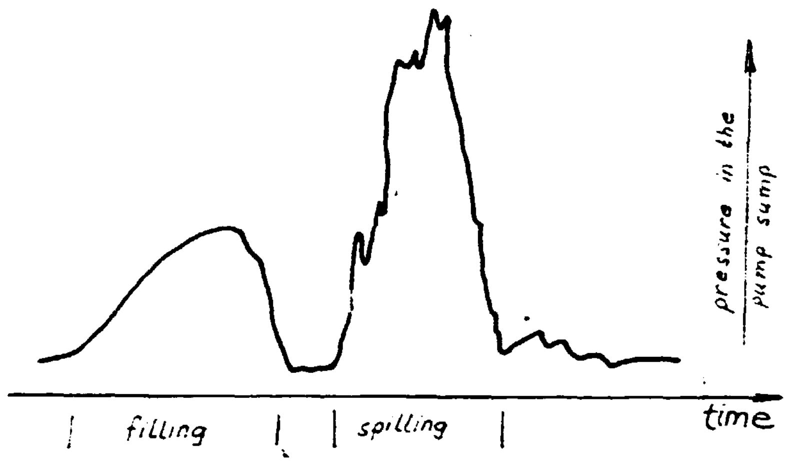


Fig 7

6. Valve timing

Timing fairly satisfactory for very slow speed, as for starting, would be theoretically that intake valve should open at top center and close at bottom center, whereas the exhaust valve should open at bottom center and close at top center. But with high speeds it is necessary to advance the opening of the exhaust valve and retard the closing for the intake valve in order to:

- reduce the work of exhaust
- induct more amount of air.

Overlapping of the closing of the exhaust and the opening of the intake valve makes possible:

- the scavenging of the clearance space
- the cooling of combustion space.

With single cylinder engine considered 1500 rpm is fairly low and timing optimized with overlopp decreased may produce :

- better starting
- more stable low load operation.

Is to be recommended to use calculat. programme for timing optimization at first.

7. Conclusions

1. Glow plug supported combustion,when burning methanol in diesel engine,could'nt be recommended for vehicular applications without ceramic isolated combustion chamber.

2. For diesel engines in watering application when fuelling with methanol,to obtain better startability and low load operations is to be recommended to:

- increase CR in to 2 ÷ 3 units
- apply switching-on technique
- apply cast iron piston
- reduce swirl ratio for obout 20%
- optimize injection with increased methanol injection rate,for obout 50%
- optimize nozzle hole distribution using calculation programme related to spray tip - piston bowl wall time contact
- estimate by calculation a potential benefit with overlapping decreased .

3. When good matched for diesel fuel operation,a small HP pump has no capacity to deliver two times more fuel or the pressure diagrames correspond to Fig 8

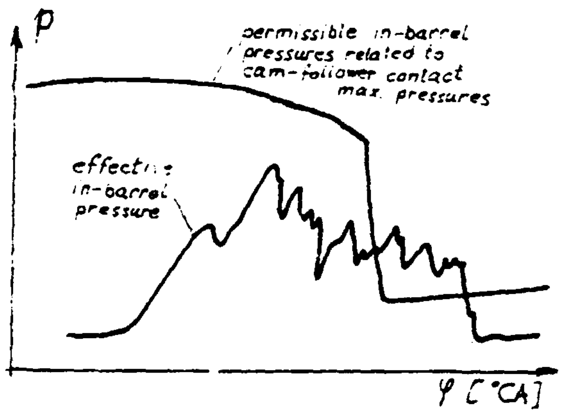


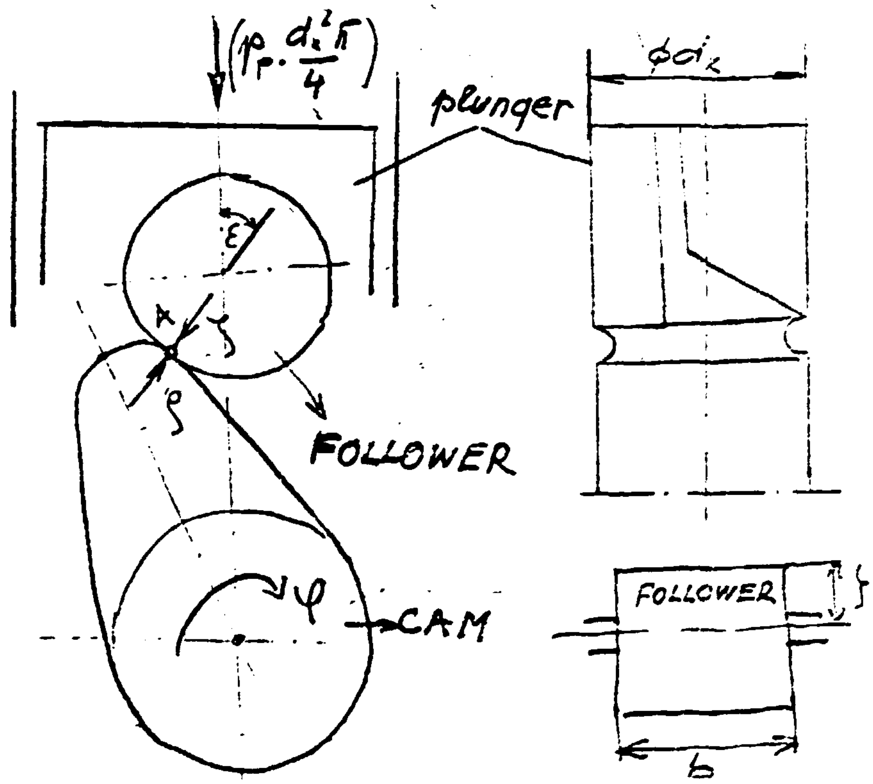
Fig 8 In barrel pressure diagrames

Thus,using separate modul in fuel injection calculation programme,with pre-lift and cam form given,the permissible in-barrel pressure diagrame must be compared with effective one or to suffer from very fast cam-follower wear in the HP pump.When decreasing pre-lift,the mean fuelling rate will be additionally reduced and period of injection still more prolongated.

4. For any application after-injection phenomens must be absolutly avoided. Again, calculation programme may solve this problem or needle lift measurements.

Permisible pressure contact

Acc. to Fig max. permissible pressure in the barrel of HP pump may be calculated as follows:



9

Fig 9

Expression (1) was derived from the work of Stipek T. (see MTZ No 1 1980, MTZ No 5 1978) but using contact permissible pressure of 1700 N/mm^2 in the every point A considered (see Fig 9).

When calculating by Eq 1 for various angles φ max. permissible in-barrel pressures may be obtained (Fig 10).

Using:

- effective in-barrel pressure diagrame p_p obtained by calculation or by measurements after refraction valve (because of simplicity)

- beginning of delivery known (pre-lift)
- period of delivery known.

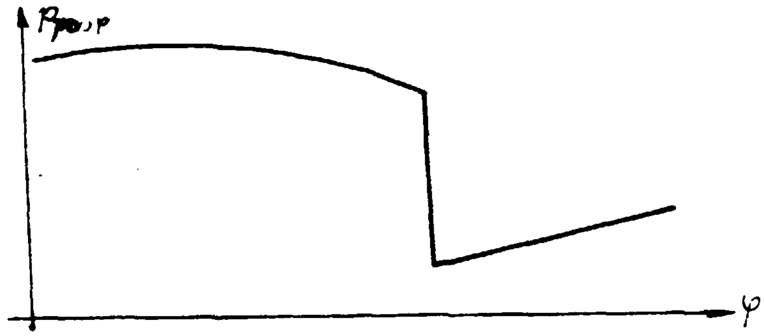


Fig 10 Max. permissible in-barrel pressure v.s. cam angle

The next Fig 11 diagrames should be complited

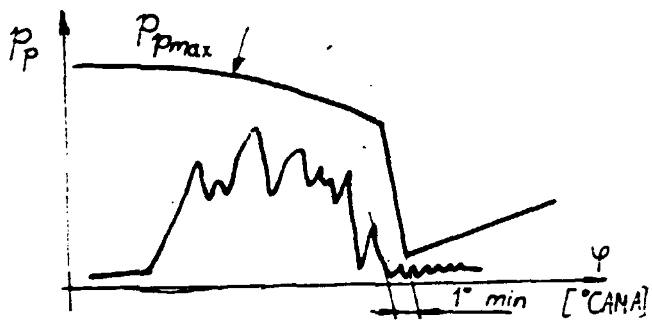


Fig 11

By injection matchings,particularly when fuelling increases for pre-lift given or/and pre-lift increases for fuelling given.

<u>How to increase CR ?</u>

CR by existing engine may be increased by:
- thiner cylinder head gasket,somtimes the problem because of small top clearence

or/and

- piston with noufinished crown.

To do that piston producer must be contacted.

In Fig 12 not complitely finished piston is shown,which has to be purcbased.With thick line in Fig 12 nonfinished surfaces

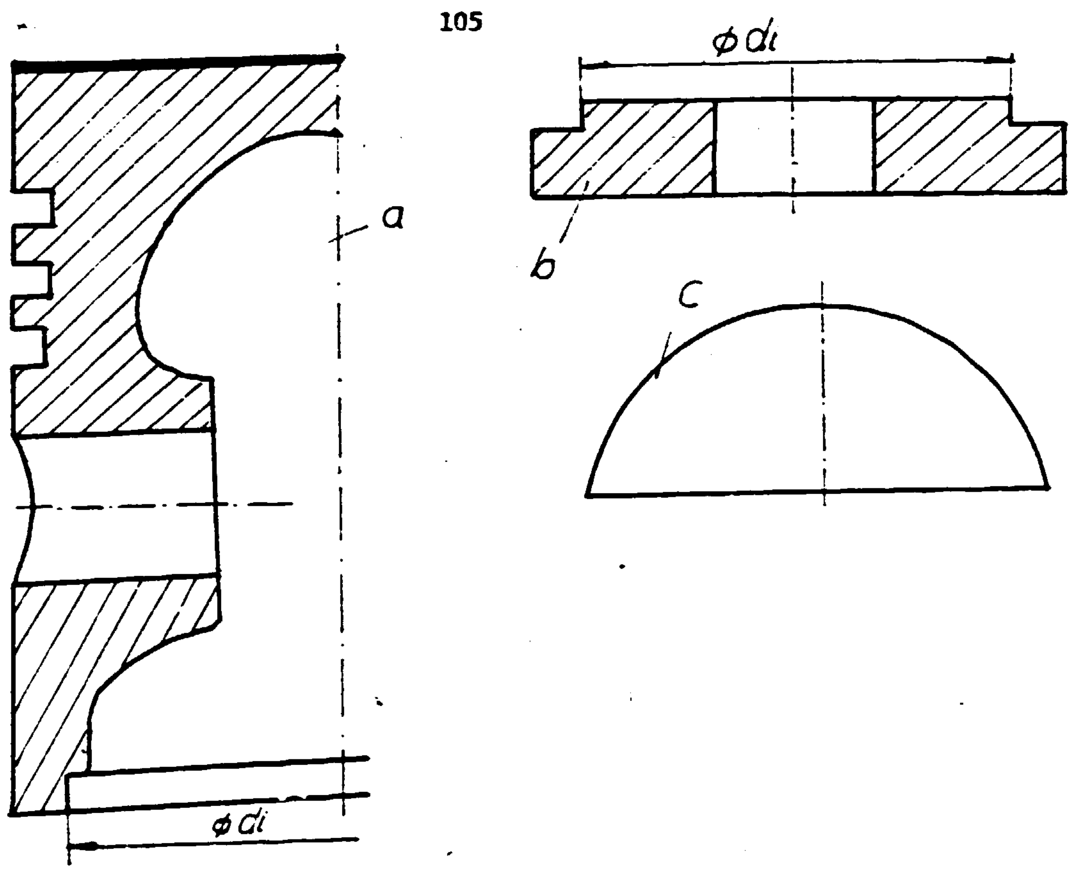


Fig 12

are shown ; Ø d_i serves for finishing in IIP workshop (see b and c in Fig 12).With primitive model c (thin sheet) evry simple combustion chamber may be produced.

During our experiments per example we used (1978) comb. chamber volumes : 98, 100, 102, 104 and 106 cm^3 all of them produced as explained above.

How to avoid after - injection

Although needle lift tranducer given operates,the needle lift events are hardy to follow,becouse of a bridge .This is specially the truth looking for after injection phenomenion.

After injection may after happend when:

- increasing fuelling without change of retraction valve
- increasing fuelling without increase of needle hloe dia.'s.

The simplest way is to calculate injection process or to follow experimentally needle lift diagrammes.Only at rated power the above experiments have to be performed.

After injection may be avoided by:

- increased retraction volume,but if decreases the capacity of

the HP pump
- or/and decreased dead volume (mainly decreasing HP tube inner dia.).It is to be recommended,inner dia to decrease on 1,5 mm (for the Kirlaskon engine).It may help to reduce a high influence of compressibility of methanol upon injection period as well as upon pressure drop at nozzle holes.

<u>Increased TCH emission with mean effective pressure increased</u>

Normally with p_e increased TCH exhaust emission decrease. The reason for that is higher temperature with p_e increased and with this better complitation of combustion.

Is to be considered that,when late combustion happend although without missfiring , TCH may increase with loding insreased. It may be still one evidence for late combustion.However,when THC concentracions measured are not reduced at stoichiometric ratio,the later said may not be truth because of a very high dilution at low loads.

<u>Generally about spray tip penetration,swirl intensity and loads--single controlled speed</u>

With single cylinder engine SR intensity becomes constant and independent of mean effective pressure .SR promotes:
- heat release control with better mixing
- more cooling of charged air.

For the more deposited on the wall of piston bowl a high swirl intensity may control heat release rate only when temperature of the bowl sustaines evaporization.With constant swirl intensity and fuelling decreased,wall temperatures also decrease becoming the lowest at starting conditions.When wall temperature drops under critical value for fuel used,deficit of fuel evaporized is evident and stable heat release is not more possible. More heat for evaporization need than released via combustion.

Generally more deposit of the fuel on the wall may be expected with:
- smaller cylinder dia.
- higher injection intensity } Fig 13
- low injection intensity }

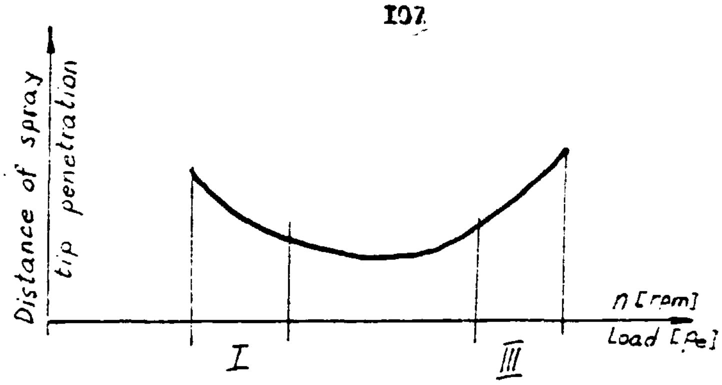


Fig 13

I because of dense core of the spray (bad dispersion), Sauter mean dia. becomes greater or mV product increases although velocity V is reduced.

III because of high intensity of injection, V becomes higher and although d_{32} decreases, product mV again increases.

The aforementioned was the reasion for controlled injection. Namely, with controlled injection at low loads and starting more fuel is despersed in air but at higher loads more fuel deposited on the wall of piston bowl.

Cavitation erosion on diesel injection pump

The wear by cavitation erosion is of importance for fuel injection systems, espetially when operating with methanol as fuel.

Cavitation attack is mostly pronaunced around spill port (plunger, barrel and wall of gallery). To prevent it the easiest way (although not always succeful) may be the use of safety screw, see Fig 14. Having, per example, radioactive isotope F-59 and operating with methanol at rated power fuelling, it's possible, on a single way, to study the tendency of wear, using just the safety screw as a most subjected to cavitation attack.

Relevant investigations were performed in Graz (Austria) using the test circuit shown in Fig 15.

The experiments showed that, at the beginning the wear runs to mg quantites, however it was reduced by a factor of roughly 1000 after a run-in time, Fig 16. Fig 16 showes that, the wear reached 4 mg, in 1 ^{h} time interval after starting, approaching to only 2,5 μg/h after 25 hours of operation. In order to detecte such a small quantity, fuel was poured out, the whole system 6 times washed out and the new activeles fuel tanked in (point

2 and 4).

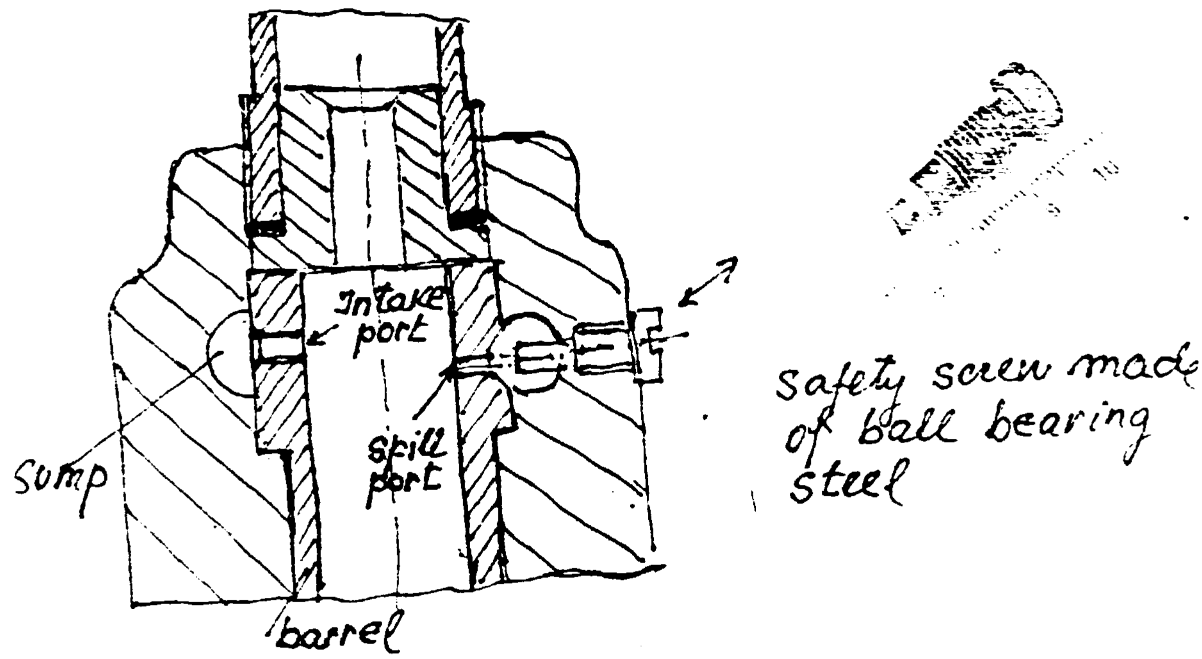


Fig 14 Safety screw apposite spill port

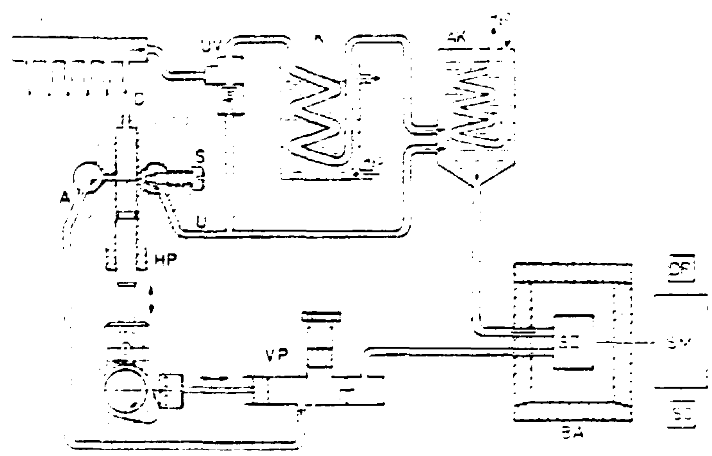


Fig 15 Test circuit for the diesel injection pump

VP - low pressure pump
HP - high pressure pump
A - inlet duct
D - pressure duct
U - overflow duct
S - radioactive screw
UV - overflow pressure valve
K - cooler

AK - balancing tank and cooler
SZ - szintillation meas. head
SM - measuring device
DR - printer
SC - strip chart recorder
BA - lead isolation

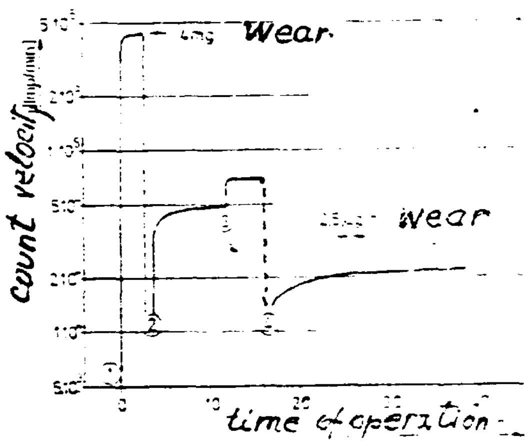


Fig 16 Time-dependent process of the wear by cavitacion erosion on a safety screw of the injection pumps.

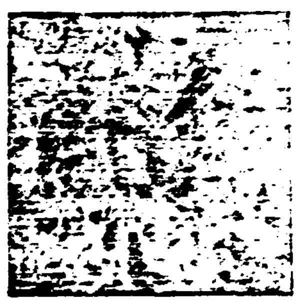


Fig 17

Fig 17 shows the section from the front of the polished safety screw after 44,5 h's of operation, which was subjected to cavitation. (For more information see the 1st Report)

Single cylinder engine in spark plug version and S/D ratio

The small engine considered has the ratio stroke/bore=1,375, mean piston velocity at rated power of 5,5 m/s and swept volume /surface cylinder ratio of 2 cm.

Simple calculation may show that,when using higher stroke max. in-cylinder pressure (for the some expansion pressure shape) must be increased (CR increased).CR increase,when spark plug is to be used,may not be necessary,but lower peak in-cylinder pressure in methanol operation results in poor efficiency asking for compensartion of CR loss.

The existed small engine may be modified to use spark plug instead of glow plug.For the first experiments it may be changed only.After successful test related to-ignition and low load operation finished the following improvements may be performed by matching:

a)injection period
b)spray penetration,distribution and atomization
c)nozzle holes-arrangement and hole dia.'s
d)CR ratio for better efficiency
e)SR ratio for better volumetric efficiency
f)cast iron piston application
g)combustion chamber modification
h)spark plug (position,spark duration)

Spark plug arrangement may not be more expensive than of glow plug offering bellow cited advantages:

- single fuel operation
- some engine for the both fuels,diesel oil and methanol as well as for their blends.
- smallest change when fuel charged
- relyable start under cold ambient conditions
- stable low load operations and transients with small TCH emissions.

In order to support abovecited APPENDIX XI is added (not in this report,for IIP only) related to "KOMATSU" experiments with spark ignition assisted diesel in multi-fuel operation (Automotive Engineering,November 1983).

TOPIC 5

"Approach to piston bowl desing for DI-Diesel engine"

Because of:

- many subjects considered
- relevant background not brought with
- anticipated long term visit of expert in combustion subject

all in TOPIC 5 considered, may be done in very short version and by heart only. However, when need, additional in formations related to the subject 5 may be sent thereafter.

1. Swirl ratio - SR

Average swirl ratio is defined as the rotational speed of air at TDC of compression divided by the engine speed. It may be measured and calculated by averaging the swirl ratios obtained from a number (N) of swirl profiles at varions axial locations.

$$\text{Swirl ratio} = SR = \frac{1}{R}\int_0^R \frac{\overline{w}}{r}\,dr \qquad (1)$$

$$\text{Average swirl ratio} = \overline{SR} = \frac{1}{N}\sum_{i=1}^{n}(SR) \qquad (2')$$

or average swirl ratio may be expressed as follows:

$$\overline{SR} = \frac{30\sum_{i=1}^{n}\left|\int_0^R \frac{\overline{w}}{r}\,dr\right|}{N\cdot R\cdot n\cdot \pi} \qquad (2)$$

where:

$\overline{w}$ - ensemble-averaged mean swirl velocity at radius r

R - bowl or cylinder radius

n - engine speed /rpm/

If it is assumed that air rotates like a solid body at TDC the equation above reduces to :

$$\text{Swirl ratio} = SR = \frac{\text{vortex speed}}{\text{engine speed}} = \frac{\overline{w}\ 30}{r\cdot n\cdot \pi} \qquad (3)$$

which has been used almost exclusively by many investigators due to the difficulty of obtaining detailed swirl velocity profiles.

When measuring speed of the wheel n_w (rpm) acc. to Fig 1

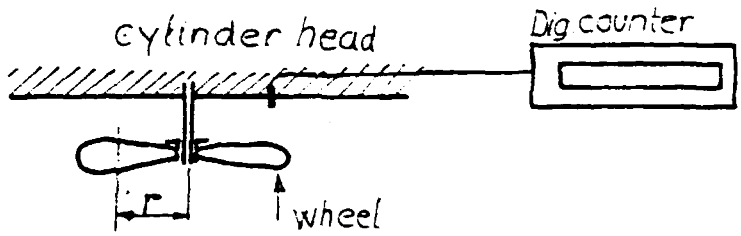


Fig 1

$$f \cdot \omega_{wheel} = \overline{w} \qquad (4)$$

Substituing Eq 4 into Eq 3 results in:

$$\overline{SR} = \frac{n_{WHEEL}}{n_{engine}} \qquad (5)$$

Thus, measuring n_w and dividing by n_e $\overline{SR}$ may be obtained.

2. Fuel spray accomodation

In the first report as well as in this report many approaches are given related to calculation of fuel tip penetration. We used one shown in the 1st report (see Appendix and example shown). The whole procedure was explained as well as calculation in the listings sent. The whole programme consist of:

- calculation of injection parameters
- fuel injection system matching
- data for calculation of dispersion
- data for calculation combustion
- data for calculation of spray penetration
- data for calculation fuel deposition.

As may be seen in the diagrams shown (see the outputs, 1st report), time or angle fuel contact history of spray tip - wall of piston bowl was presented. In this way fuel deposited on the wall may be calculated in dependance of crank angle. Morever, locations and areas of fuel deposition may be defined as well.

However, in the programme sent (1st report) swirl impact upon spray movement was not included, because of not existing $\overline{SR}$ data. But fuel tip deviation may be calculated using Eq's of momentum. Related to swirl moment of momentum may be calculated as:

Angular momentum or moment of momentum (per unit mass) : $AM = \frac{2}{R^2} \cdot \int_0^R \overline{w} \cdot r^2 \, dr$

Average angular momentum : $\overline{AM} = \frac{1}{n} \cdot \sum_{i=1}^{n} (AM)_i$

This calculation may be performed using one separate additional modul (was not sent in the 1st Report).

The calculation programme of fuel tip contact is most important one related to spray - bowl matching. It may be concluded that, many different comb. chambers are successfuly used but only with spray history matched. Spray history matched understands:

- speed and load dependent matching
- injection period, beginning of injection
- injector holes pressure drop
- low of injection
- atomization
- contact areas on piston bowl v.s. crank angle.

By means of above matched and followed calculation of engine performancies, is to be defined (for the combustion chamber chosen):

- nozzle hole distribution
- nozzle hole dia.'s
- position of injector (if not existed)
- injector open - and closing pressure
- retraction volume
- HP tube dia. and lenght
- plunger dia.
- effective plunger lift
- pre-lift setting
- swirl intensity, inlet (duct) port desing.

The above procedure described followed by some refinings by calculation is especially important when:

- piston bowl could'nt be properly designed because of existing stroke, l/r ratio and piston compresion height.
- capacity of existing pump is limited but larger HP pump could'nt be used because of space on desposal
- position of injector could'nt be changed because of existing cylinder head.

Only after abovecited procedure:
- rig test experiments
- on engine experiments
may be the following steps.

3. Combustion chamber shape

To change combustion chamber modifiging the bowl shape is normally more difficulty than to desing a bowl for the new engine.

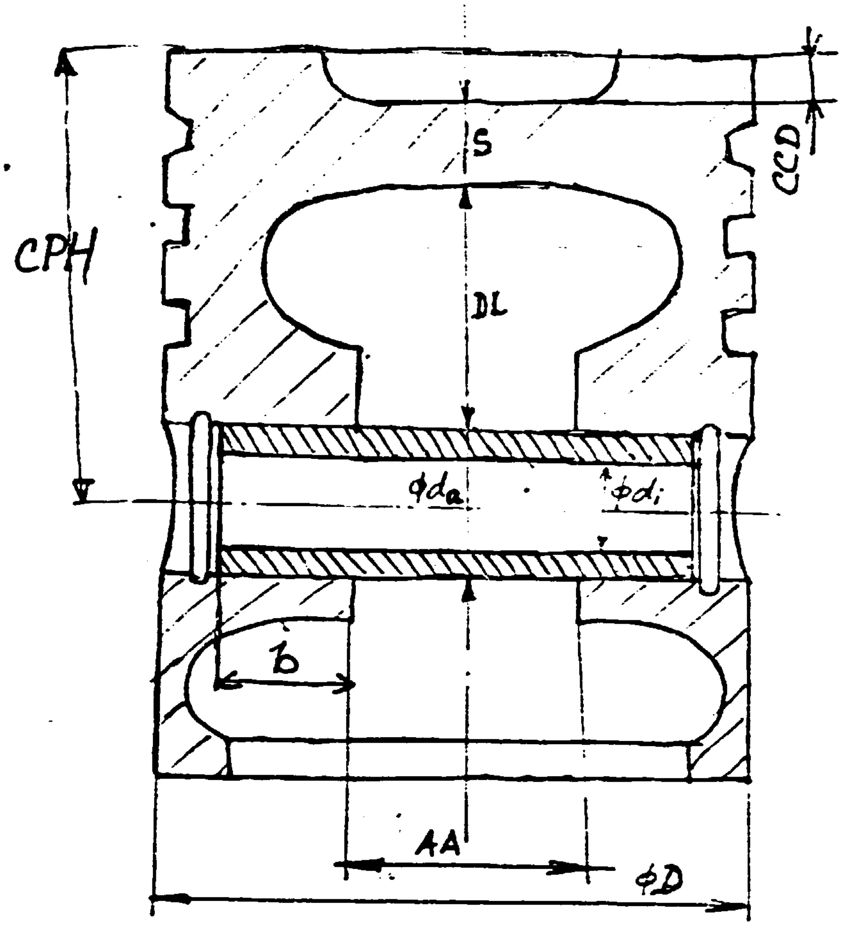
In the first case must be considered:
- existed CPH - see Fig 2
- existed r/l ratio see Fig 3,d
- accomodation height of piston (Fig 2 and Table 1)
- gudgeon pin outside dia. $\varnothing\ d_a$ (Fig 2)
- space for the balacing movement of conrad small end (Fif 3,d)
- piston crown thickness S (Fig 2)
- piston top clearance
- stroke and block of existing engine.

Thus,the depth of piston bowl CCD (Fig 2) for existing engine depends on many factors.

TABELE 1

Max.mean effective pressure /bar/	Max. specific piston loading $\frac{\text{max.power output}}{\pi \cdot D^2/4}$ /$\frac{kW}{cm^2}$/	Accomodation height DL=%D
< 8	< 0,185	< 20
8 ÷ 10	0,185 ÷ 0,22	20 ÷ 30
> 10	> 0,30	30 ÷ 35

The bowl shape acc. to Fig 2a was suggested for neat methanol combustion.This combustion chamber is typical representantation of combustion ÷ fuel deposion on the wall - single hole nozzle - high swirl.The bowl has a high height CCD (see Fig 2) and may not be applied for existing engine.In this case modification must be performed acc.to Fig 2b.However,only accurate measurement now may show if single nozzle hole may be used.(Very after three holes are needed)



$d_a = (0,32\ D$
$AA = (0,33\ D$
$DL_{min} = (0,12\ D$

$$p_b = \frac{p_{max} \frac{\pi D^2}{4}}{2\, b\, d_a}$$

p_{max} – max. in-cylind. press

$d_i = (0,42\ d_a$
$S \geq 0,10D$
$p_b < 60\ N/mm^2$

Fig 2 Piston proportions for diesel NA-DI engine

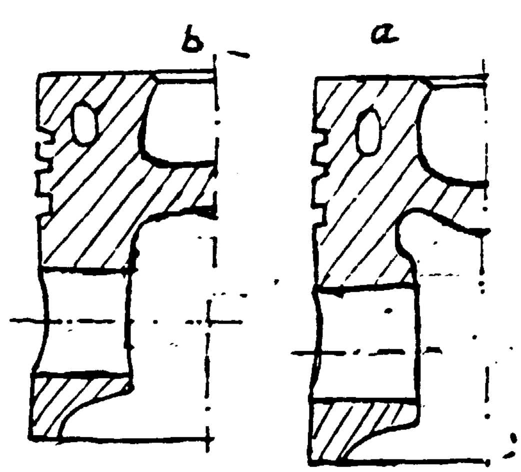


Fig 2a Spherical piston bowl

Fig 2b Semi spherical piston bowl

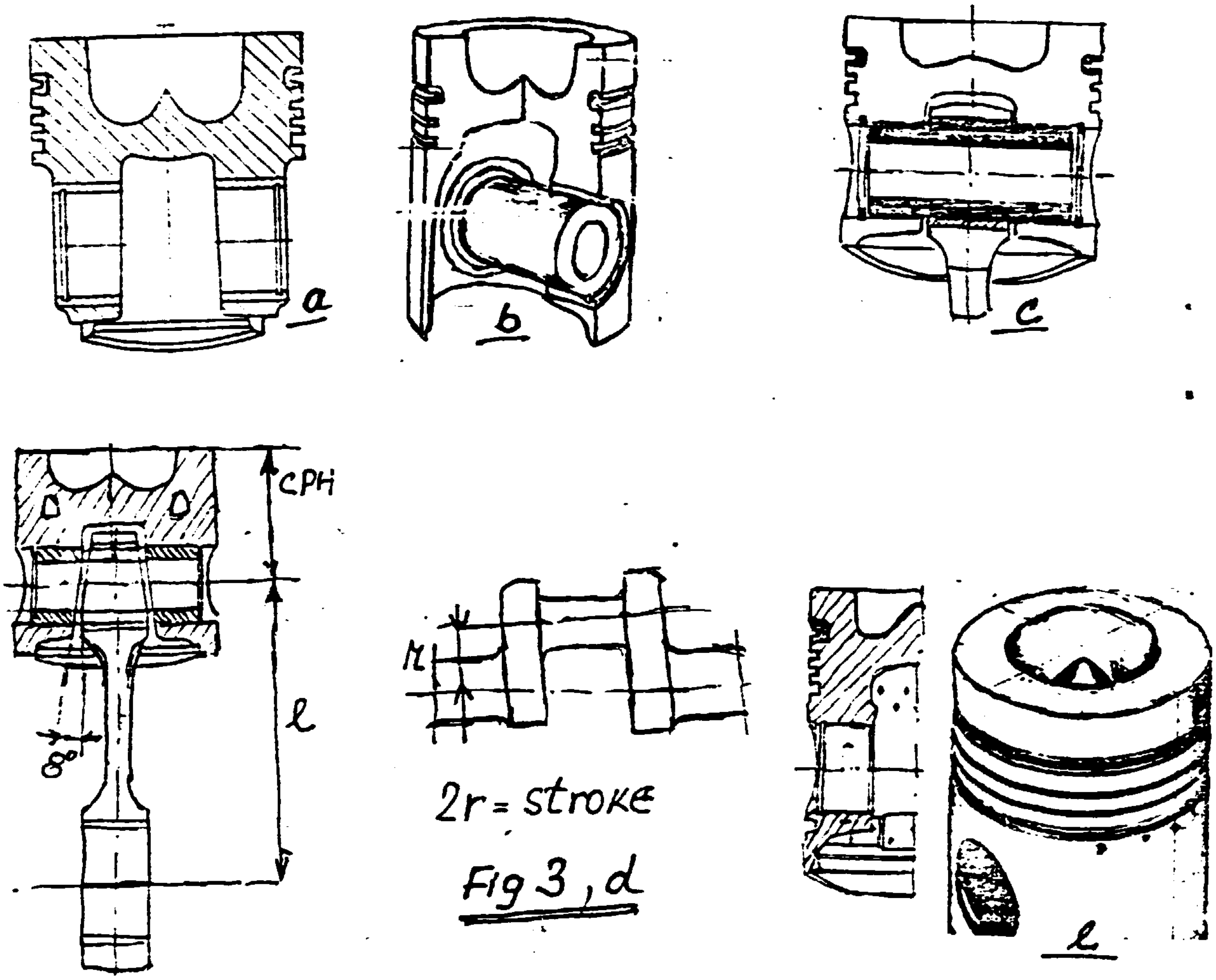


Fig 3

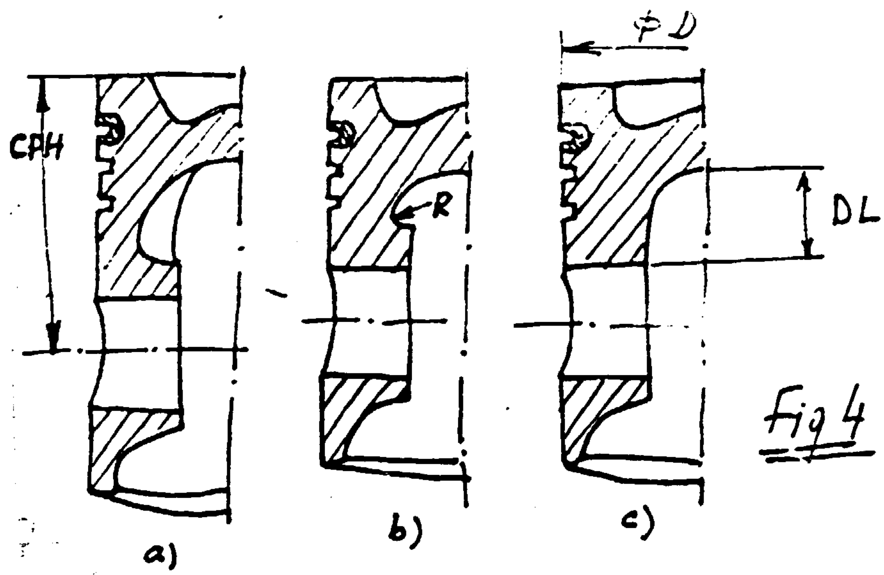


Fig 4

Generally is valid,the smaller piston dia. the deeper piston bowl,for the DI diesel engine.

Fig 3 shows the varions versions of omega shaped bowls.All of then have been successfuly applied but piston dia.,nozzle holes as well as swirl ratio separately matehed.For the bowls shown in Fig 3,b,c and d four nozzle simetrically distributed holes are used.Swirl of middle intensity.Chamber Fig 3,d asks for some higher swirl and more fuel deposition on the wall.Chamber type b,was yet applied in methanol combustion but with the pilot ignition.

Very after,dependent of CPH given(see Fig's 2 and 4) inner shape of piston must be modified.In Fig 4,c was the smallest hieght CPH on desposal.Somtimes,design d Fig 3 may help to decrease the outside dia. of gudgeon pin and to obtain more freedom for the design of piston bowl.

To folow the effect of circular motion of air swirl intensity has to be changed.For the purpose of experiments masked inlet valve may be used.Masked valve may be produced very easily acc. to Fig 5.

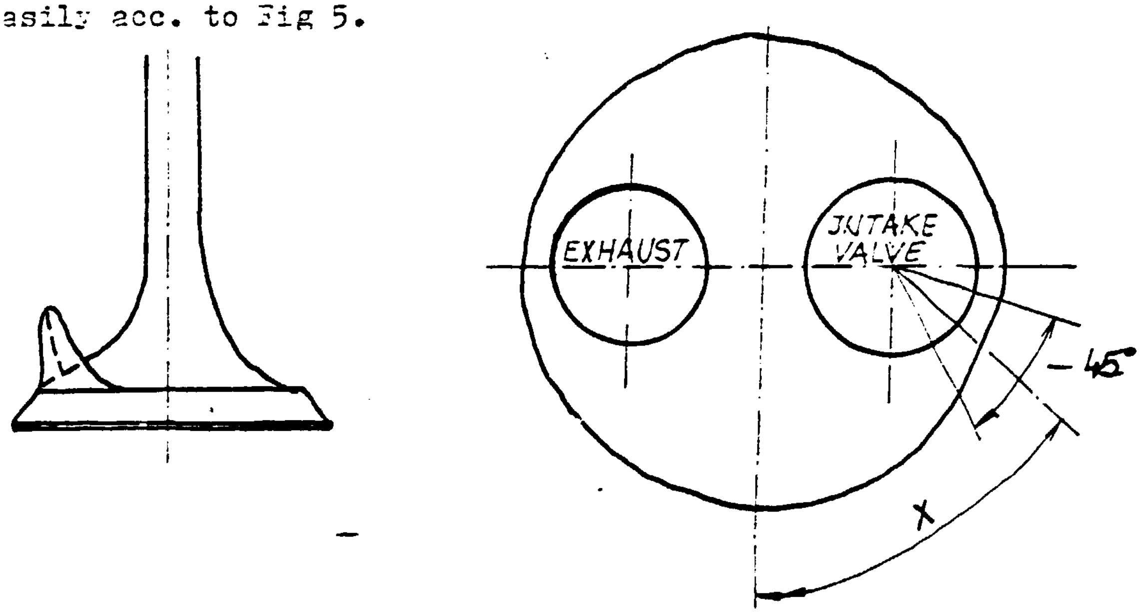


Fig 5 Masked inlet valve,angle x must be defined experimentally

The combustion of squish and swirl in the types of bowl Fig 2 and Fig 3,results in a toroidal air motion around the piston axis.Intensity of squish increases with the clearace of piston cyl. head decreased.

Fig 6 shows open DB new combustion chamber as a simplest one, but is more difficult to match the fuel injection system to it. Again one time, combustion chambers usually have a shape to conform to the spray form.

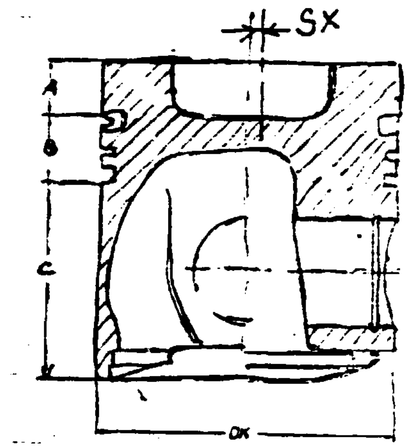


Fig 6

Till now open-combustion chambers were considered only. Open--chambers shown have the bowl dia.'s of about one half of piston outside dia.'s. The excentricity S_x (see Fig 6) must be experimentally defined related to swirl intensity desired and volumetric efficiency. Sometimes injector position existed and possibility of nozzle holes distribution effects the excentricity S_x also.

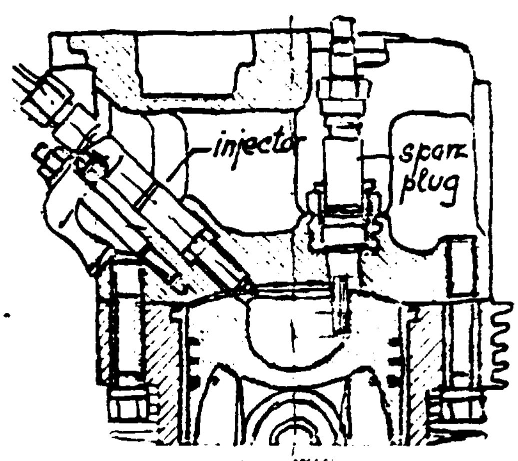


Fig 7 Comb. chamber of MAN engine L9204 FM

MAN combustion chamber shown in Fig 7 is of open type also. It has spherical bottom of the bowl other shown in Fig 3 toroidal bottom.

It may be interesting to compare till now shown and discussed with Komatsu combustion chamber (see Appendix XI,not in this report).The combustion chamber is of Mexican-hat type and Fig's 2 and 3 shown the whole arrangement for CH_3OH use in diesel engine.It may be still one time concluded that,many combustion chamber may be developed but every time,fuel spray history as well as swirl-squish interaction are decisive factors for the success.

It is of interest to mention that,the

Fig 3

both producers: MAN and KHD have the open combustion chamber but top of the piston is not flat.DB with cylindrical combustion chamber has a flat top of piston.

The newest developements of combustion chambers and pistons have been concentrated on saving the fuel and on increase of multi-fuel properties.The both mentioned means to save the heat or to reduce the heat loss.Ceramic components are still our future and investigations showed that,heat loss may be reduced with conventional materials also.

Per example:

MA-DI diesel engine,diesel fuel,swept volume $V_H = 1 \div 2$ dm^3,mean piston velocity $\dot{x}_{sr} = 8 \div 10$ m/s at $p_e = 7$ bar,specific fuel consumption rate may be correlated as:

for Alu-alloy piston:

$b_e = (210 + n \cdot 0{,}0158) - 12{,}3 \cdot V_H$ /g/kWh/

n - /rpm/ , V_H /dm^3/

for cast-iron piston:

$b_e = (205 + n \cdot 0{,}0158) - 12{,}4 \cdot V_H$ /g/kWh/

It's of interest to see the influence of heat loss reduction on combustion chamber and piston design.Morever,the strivings are directed to change the combustion process and to accomodate the injection,but now toward reduction of heat losses.

Fig 9 shows the participation of corresponding piston areas on heat transfer (loss) at TDC position.Fig 10 shows that in cold operation volume V_1 (top land clearance) approachy 13%.

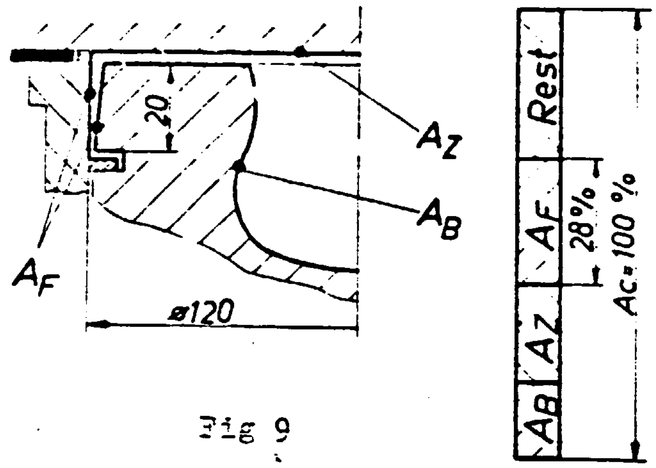


Fig 9

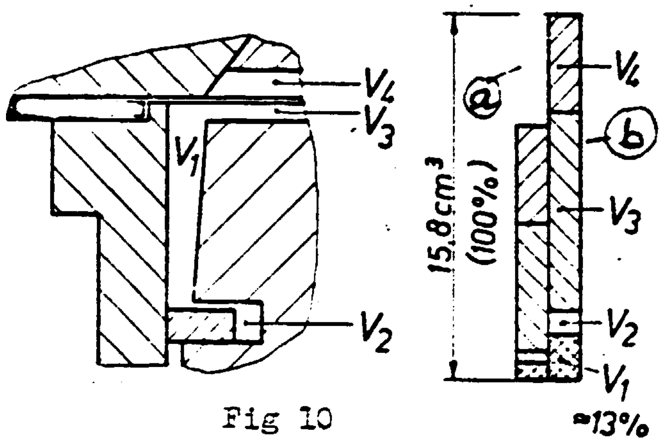


Fig 10

The investigations showed in Fig's 9 and 10 were evaluated related to fuel consumption and mean effective pressure.The results obtained were more then of interest,what can be seen in Fig 11.Fig 11 depictes that top land clearance dead volumen has a largest influence upon the both:

- increase of fuel consumption rate Δb_e
- decrease of mean effective pressure Δp_e.

Further on,it may be also concluded that the both,heat isolation and combustion process with less heat transfered to the walls have a positive effect on engine performance figures.Doing above cited,engine will be effectively prepared for the use of alternative fuels also.

In Fig 12 are two pistons shown.Combustion chambers are developed for the less heat transfered to the wall of the piston bowel.The piston shown in the left is IFA MM production and materials is cast iron.The right piston presented is modern Elsbett piston made from cast-iron or from steel.

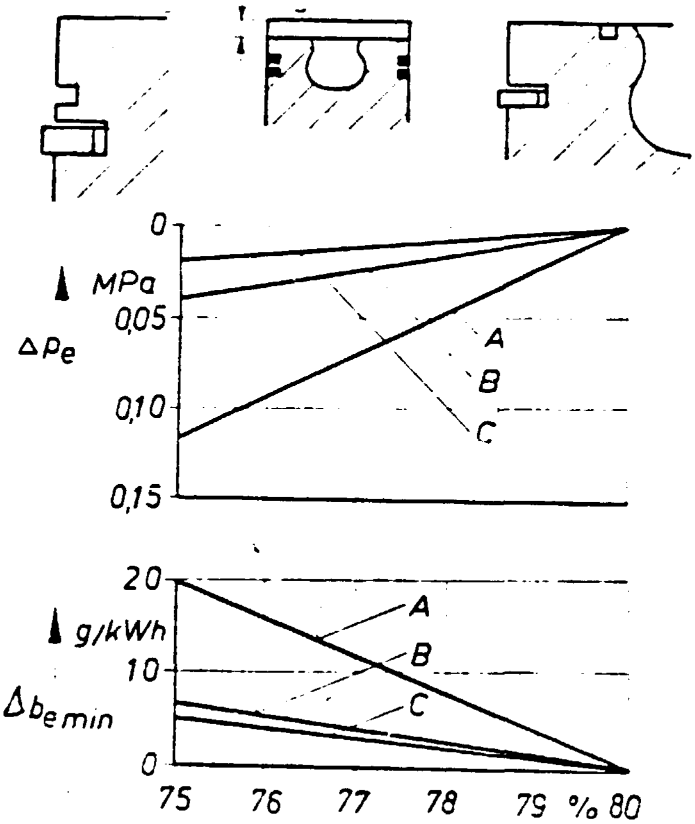


Fig 11

$$V_{\Sigma} = V_A + V_B + V_C$$

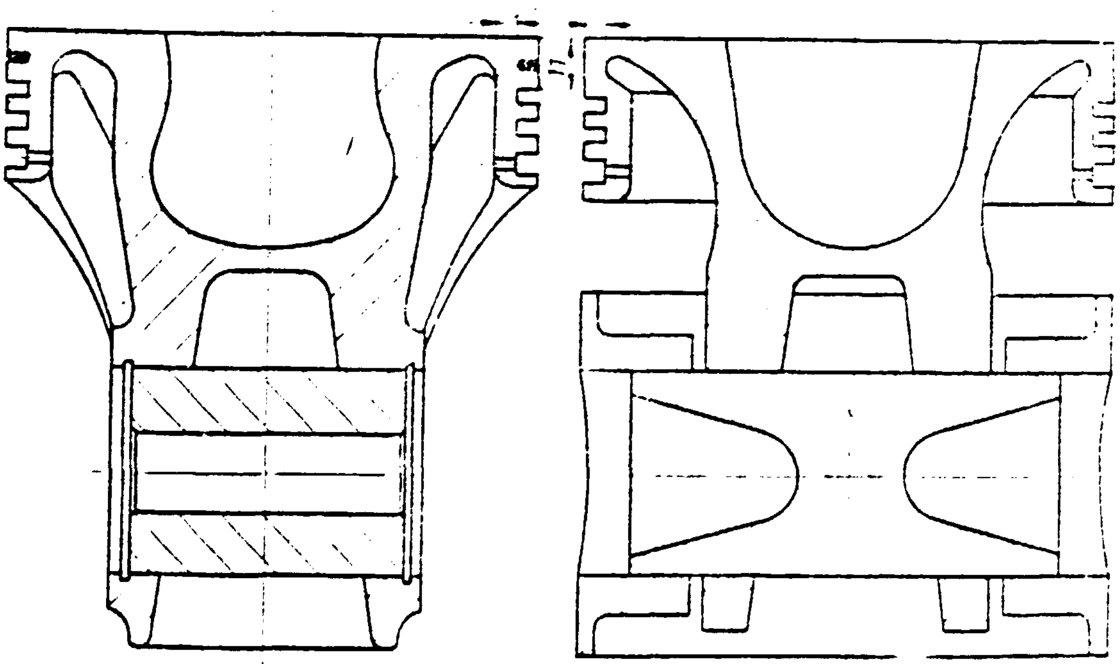


Fig 12 Cast iron IFA piston (left) and Elsbett steel piston (right)

Thus,we are new dealing with most modern combustion chamber of Elsbett.Fig 13 explains the combustion process of single jet combustion method for better heat isolation.

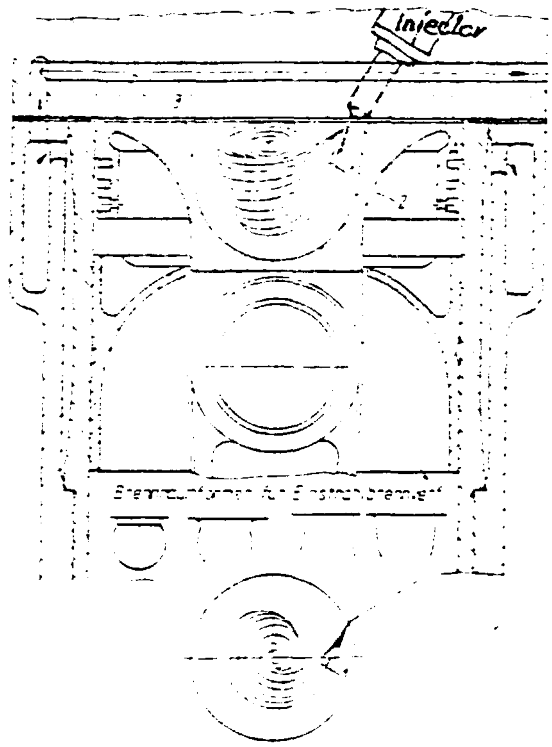


Fig 13 Single jet combustion method for better heat isolation (Elsbett)

1 - position of the jet
2 - piston (butts againts the liner at A)
3 - cylinder head without water pockets
4 - bore for coolant to the injection nozzle.

The combustion chamber is similar as for MAN but with Elsbett the spray exercises circular motion being introduced in the intense swirl (see pas 1,Fig 13).The gases with lower specific mass are more collected around the bowl center but the air,not immediately taking part in combustion,will be forced toward the wall being acting as a "heat transfer hindrauce". Air protection mantle around the bowl wall becomes thiner with fuelling increased.This is the reason for better fuel consumption figure at partial loads (comparing with other combustion process).It means that Elsbett process in the best at "cruise drive".

Fig 15 shows cross-section of oil-cooled Elsbett 3-cylinder car engine of 1,38 dm^3 swept volume.Piston dia.=Ø 80 mm , stroke = 90 mm.

Fig 16 shows the combustion Elsbett chambers for truck application.Cylinder design enables the increase of piston diameter without increase of stress on cylinder head and engine block.

The whole design procedure may be seen in Fig 17.

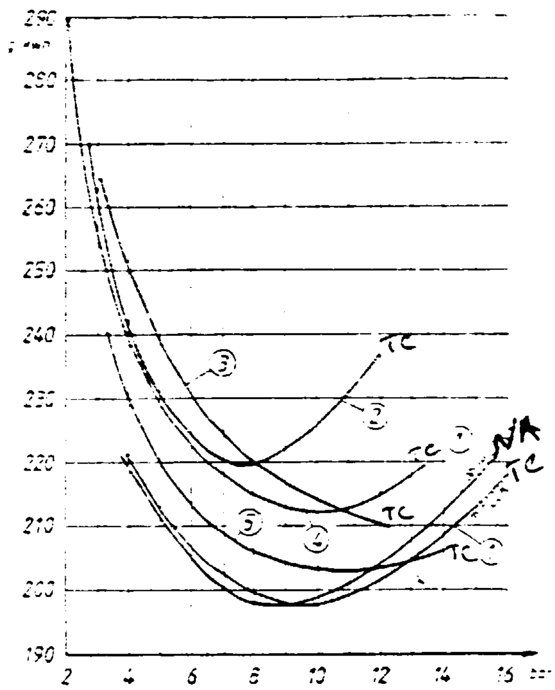


Fig 14 Part-load s.f.c. at 7,5 m/sec piston velocity

1 -heat isolated single jet DI engine 13,3 dm^3,6 cyl.
2 -heat isolated single jet DI engine 1,38 dm^3,3 cyl.
3 -single jet DI engine cooled charding air 11,4 dm^3,6 cyl.
4 -Four jet DI engine, without intercooling 9,6 dm^3,6 cyl.
5 -Four jet DI engine, without intercooling, with heat insulated engine components, 9,6 dm^3,6 cyl.

TC - turbocharged NA - natural aspirated

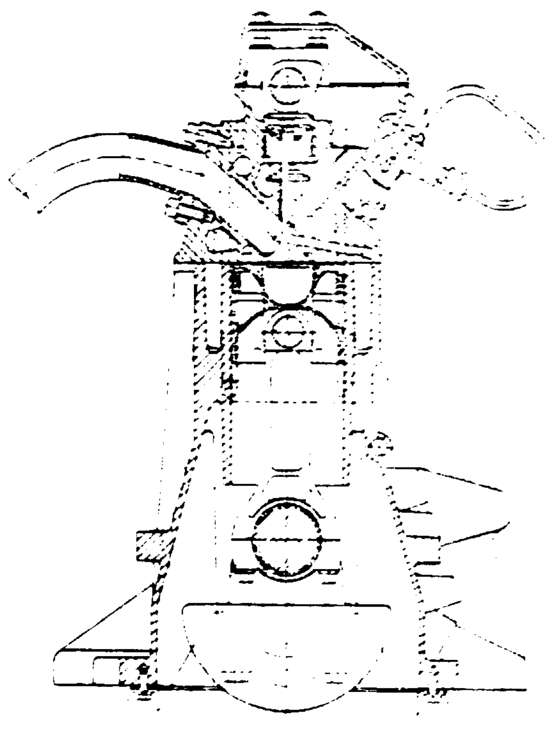
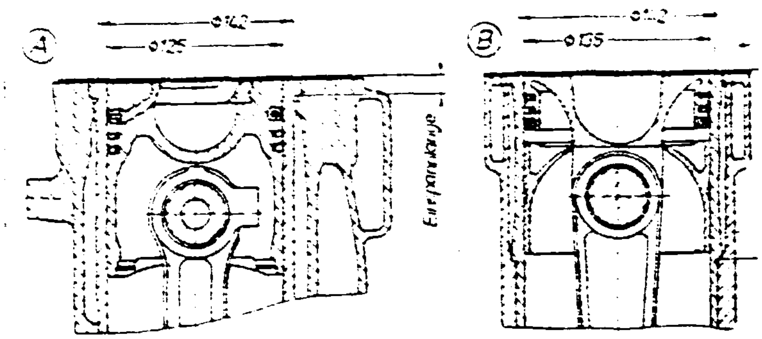


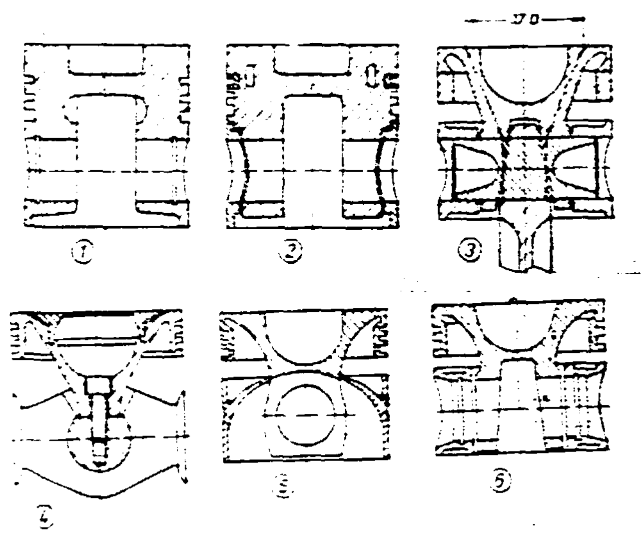
Fig 15 Cross-section of oil-cooled Elsbett 3-cylinder car engine



A - 125 ø mm ,142 mm gasket diameter

B - 135 ø mm with 142 mm gasket diameter and improved heat isolation

Fig 16 For truck application



1 -Aluminium piston
2 -Pendulating shaft piston
3 -Structural design of the new piston
4 -New piston as cross head piston
5 -as steel or cast iron articulated piston
6 -as articulated piston in ceramic

Fig 17

Heat saving achieved may be obtained by comparison of cooling requirements before and after conversion for improved heat isolation (Fig 18).

Thus:

1 -passenger car swirl chamber diesel engine
2 -passenger car DI engine after conversion
3 -truck engine before conversion
4 -truck engine after conversion, using water as coolant
5 -truck engine after conversion, using oil as coolant

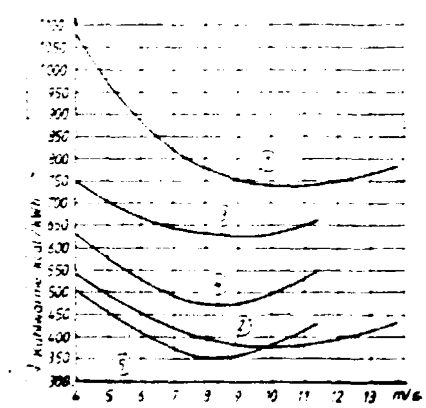
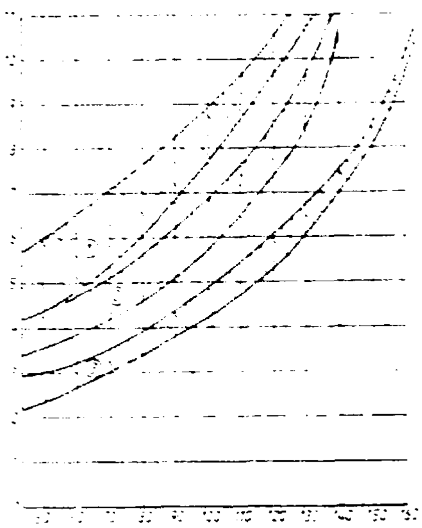


Fig 18 Cooling heat required /kcal/kW/ v.s. car velocity /$\frac{m}{s}$/



In the end it's interesting to see differencies in fuel consumption figure but now in field. Fig 19 depitces that passenger car with Elsbett 3-cyl. engine may have average fuel consumption of about 4 l/100 km.

1 -Gasoline engine
2 -IDI diesel engine
3 -single jet diesel engine

Fig 19 Fuel efficiency / l/100 km/ v.s. car velocity /km/h/ for passnger car of the 3 diferent combustion methods

As shown here developing the single jet combustion system further for improved heat insulation resulted not only in the best passenger diesel car engine concering noise, fuel efficiency and power; it also demonstrated that practically all improvements in the heat balance of the engine at the same time made the engine less expensive.

Re-entrant bowls claim for the inside turbulence. Namely, turbulence inside the bowl increases considerably to those generated early in the intake stroke. In the absence of swirl (see "Steyr"-Wien, "Famos"-Sarajevo) and with the re-entrant bowl configuration (acc. to Arcoumains-Bicen-Whitelaw), the squish induces a similar axial flow structure to that in the cylindrical bowl but of stronger nature. (See Perkins squish--lip). With er-entrant bowl squish in more prouanneed. The addition of swirl, however, results in the formation of two vortices rotating in oposite directions (see Fig 20). The swirling motion exhibited spiralling characteristic at the entry plane of the bowl and solid body type of rotation near the bowl base.

For the case of the cylindrical piston-bowl (open combustion chamber) and in the absence of swirl, a squish-induced toroidal vortex occupys the whole bowl space at TDC of compression. Interaction of swirl with compression-induced squish, however, results in the formation of a cownter-rotating vortex in the axial plane in addition to the near solid body rotation of the fluid in the tangential plane (see Elsbett combustion chamber). The squish, in the presence or absence of swirl,does not alter the overall turbulence levels which remain comperable to those obtained with the flat piston.This swirl profile at TDC of compression

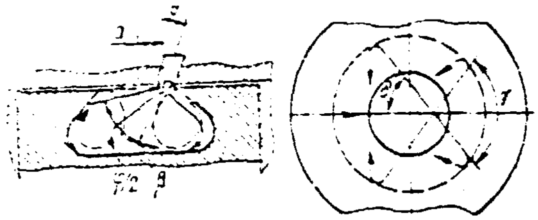


Fig 90 CHIDI-re-entrant bowl with new nozzle holes distribution (5 holes). a=10,7 mm, $\theta=24^0$, $\beta=15^0$, $\gamma=26$, $\varphi/2=75^0$

depends more on piston-bowl geometry rather than on the intake swirl field which is better reflected on the final angular momentum of the bowl contens (Ar.-Bic.-Whit.).

Practically considered re-entrant bowl is proue to cracks at the top rim of the bowl and in more difficulty to control the combustion proces.With this combustion chamber 4 or more holes are needed and fuel is macro-distributed by nozzle holes. With Elsbett and MAN as well as partly with D process (MWM) air motion is mixture controling.Re-entrant bowl produces more combuction noise with higher pressure rise.Peak in-cylinder pressure is normally higher than that with open combustion chamber.However,re-entrant bowl,it good matched,may show a good fuel consumption figure.

Comparing the both,open and re-entrant combustion chambers may be concluded that open piston bowl offers more benefit than re-entrant one.We may also conclude that,the swirling motion inside the cylindrical bowl is closed to solid body rotation while re-entrant bowl gives to complex flow patterns.Squish,in the presence or absence of swirl,does not augment the turbulent energy inside the cylindrical and open bowls contrary to re-entrant configuration where turbulence generation may be observed.

Having more complex flow patterny of intro_duced air, re-entrant bowl is less suitable related to control of:
- fuel spray patterns
- time-space history of mixture formation
- heat release rate
- heat transfered to the wall.

Open cylindrical combustion chamber with deep spherical bottom may be suggested for methanol combustion in diesel engine. Fuel spray pattern may be more like Elsbett or more like MAN. Otherwise, using controled injection at partial loads, MAN new engines, have more fuel distributed in air also (like Elsbett). At higher load is a large difference related to fuel spray patterns between MAN and Elsbett. Further on cast iron piston and Ck=18 may be suggested also (for CH_3OH combustion).

As yet mentioned, sometimes is not so easy to apply deep combustion chamber in existing engine. Three piston rings (1^{st} keystone, 2^{nd} tapered compression ring 90', 3^{nd} spring supported oil ring) is a standard today (in diesel engines). Top land may be only at 10% Dk. But above said does not help for deeper combustion chamber location because of:

-stroke given

-gudgeon pin dia.

-piston compression height given

- conecting rod small end

-min. bottom thickness of the bowl needed.

However, some compensation up to 3 mm may be obtained on a primitive way.

Controlling the temperature during forging the conn- rod may be manufactured with shorter lenght l (see Fig 21), let say up to 2 mm. Morever, very often happend in series production

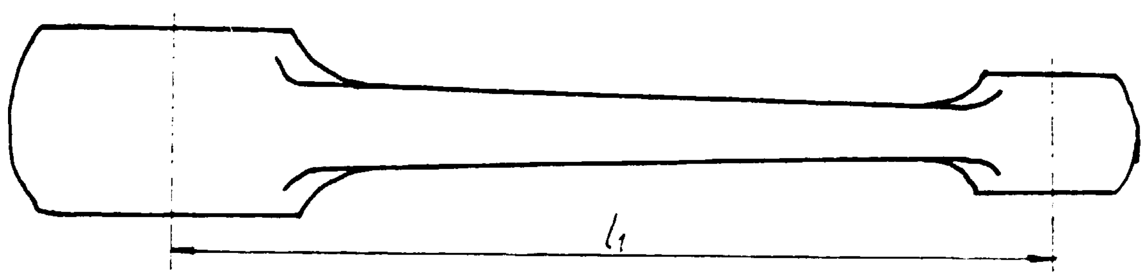


Fig 21

also that some of conn-rod forged become yet shorter and have to be refused. We've been using just refused conn-rods without special order.

The distance between the centers of conn-rod holes may be manufactured shorter also in the work-shop; let say up to 1 mm shorter (see Fig 22).

When counterbalance weights at BDC may come to contact with piston, it may be reworked-out acc. to Fig 23 or/and counter weights may be field out, Fig 24. Because of shorter distance l (see Fig 22) β_{max}=arc sin $\frac{r}{l}$ becomes greater and the conn-rod may come to contact with cylinder liner. In this case the windows have to be made in liner, see Fig 25.

After matching in engine experiments, new connecting-rod may be fabricated properly.

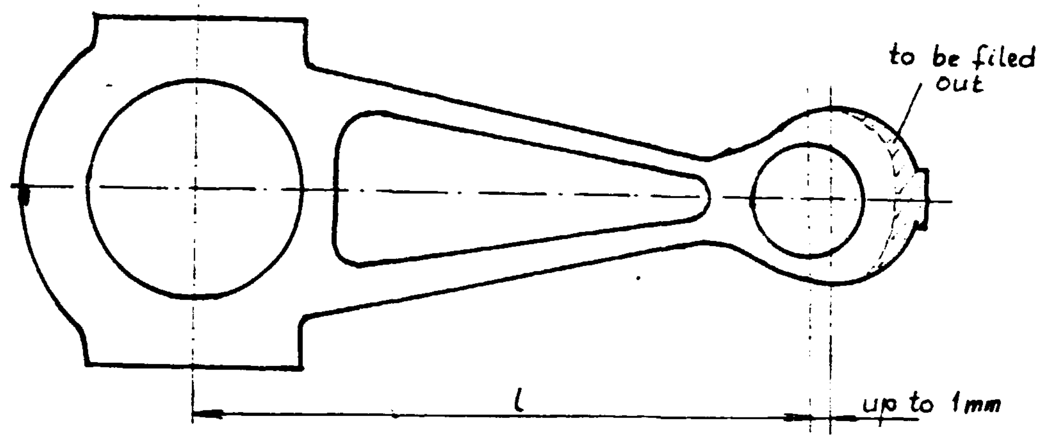


Fig 22

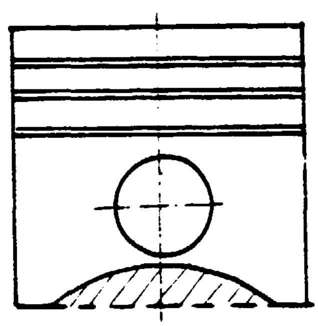


Fig 23

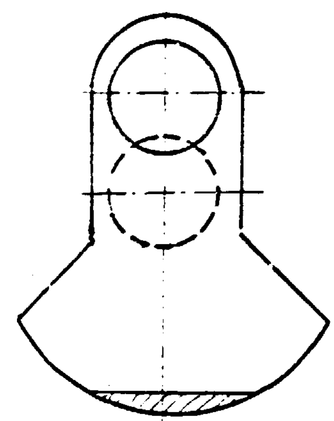


Fig 24

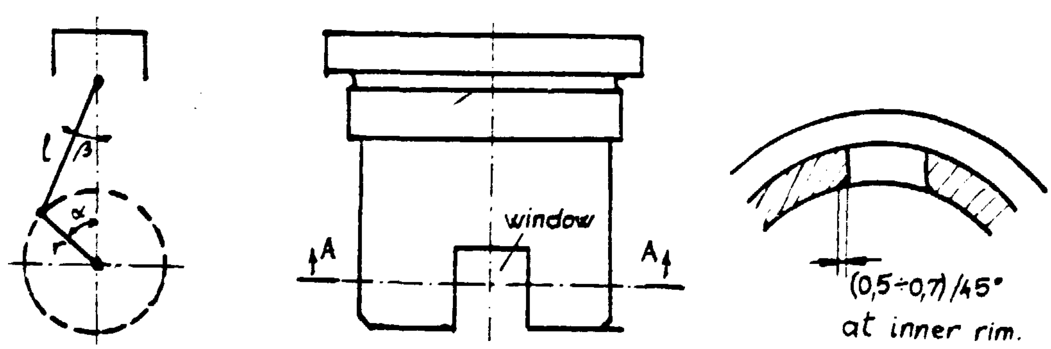


Fig 25

1-2 fuel tank
3 simple valve
4-5 fuel filter
6,7 supply tube
8-9 doping
10 single direct valve
11 conection
12 supply tube
13 LP pump
14 inlet tube
15 HP pump
16 common sump
17 realise screw
18 spring valve
19 mixing tube
81-91 instead of 8-9 in swithing-on application as was yet explained

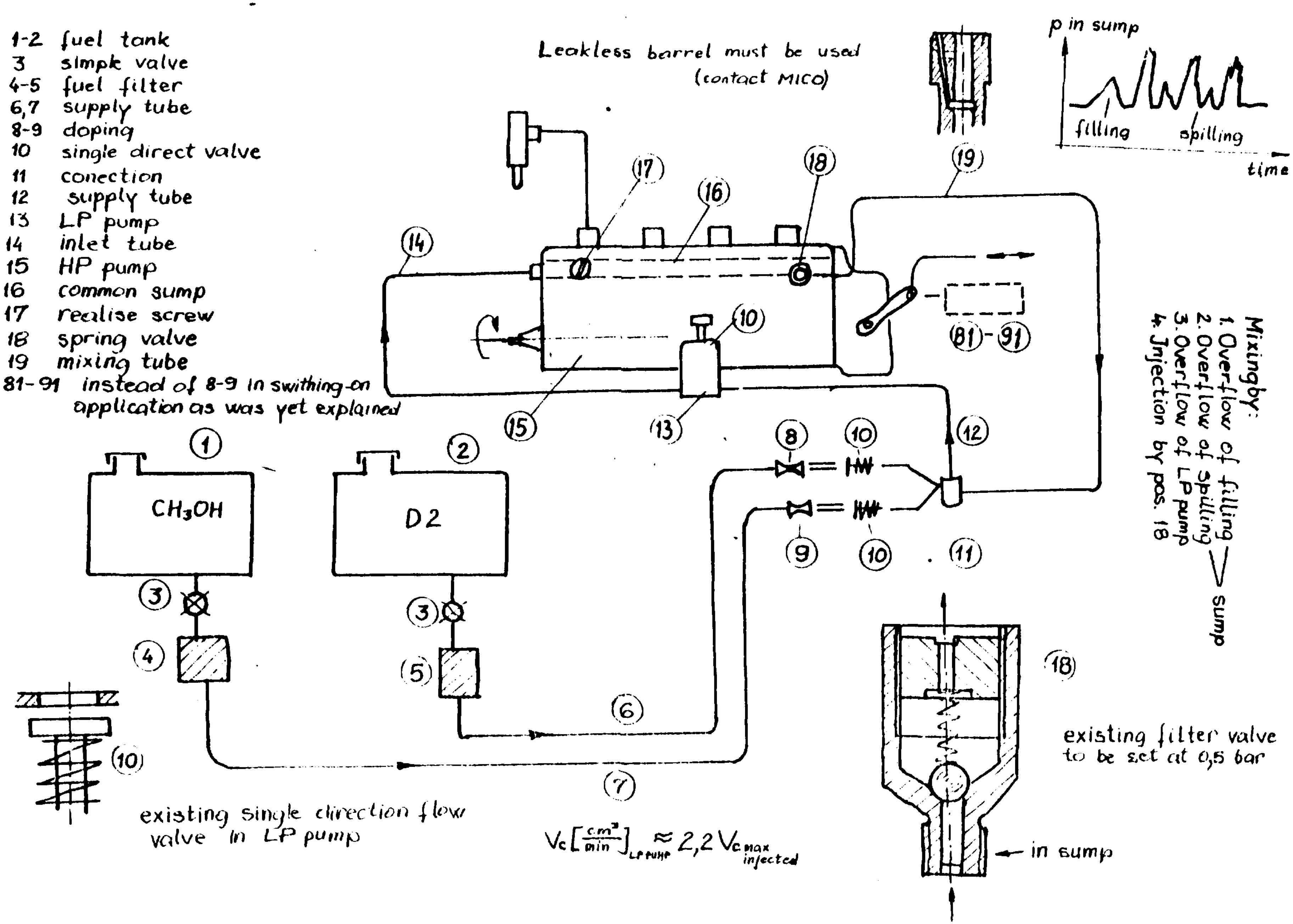


Fig 1

Appendix I 2 related to the TOPIC 2

In argument with suggestions given in the TOPIC 2 the relevance experiments were perfomed using D_2+CH_3OH blend as fuel. TATA in-line engine was used and A-type pumps ∅7x8 mm was applied. 20% by volume methanol was doped to diesel oil and with external constantly stired homogeneony unstable mixture was prepared. Additionally internal mixing was applied acc. to the Fig 1. Positions from 1 to 11 were replaced with stiring technique in this experiments. It was done purposely in order to check very fast:

- starting
- low load operation
- knocking
- full load operation and transient.

The experiments were performed on 27 August 1984. Cold start was not completed because of conditions on desposal.

Starting, under ambient lab conditions was without any problem as well as for low load operation. Fuel load was observed at 2800, 2000, 1500 and 1000 rpm. Exhaust temperatures measured were in error because of instrumentation on desposal. Transients observed were as usual, taking into account dinamometer-inertia and resistance coupled.

Completing successfully first provisional testing of engine reactions related to blend used, the bellow cited step of experiments may be suggested.

Using the arrangement shown in Fig 1 the engine perfomancies have to be observed, but now much more in detail. Doping quantity of methanol may be expected up to 30% by volume, it means, that for same power output, the fueling must be properly corvected.

Instead of very primitive flow doping pos. 8 and 9 (Fig 1) fuel distribution may be used (see pos. 81 - 91, Fig 1) as was yet explained in detail.

Doing above mentioned up to 12% of diesel fuel may be replaced by methanol.

Although the princip of mixing may be successful, it should be estimated the all change needed, to obtain as much of their benifits as is economically wortwhile.

ONLY TO MENTION :

The recent rapid application of the Diesel turbocharged engine to railroods has been brought about by the need for better perfomance and overall economy.

The main goods applying turbocharging were to:

- increase → $\frac{\text{KW output}}{\text{dm}^3 \text{ swept volume}}$
- decrease → $\frac{\text{m}^3 \text{ engine box volume}}{\text{kW output}}$
- decrease → $\frac{\$}{\text{kW}}$ in the first cost
- decrease → $\frac{\text{g}}{\text{kWh}}$, $\frac{\text{l}}{\text{100 km}}$ fuel consumption rate (let say obout 10%)
- decrease → engine exhaust emission
- decrease → engine noise emission
- simplify → engine production by reduction of engine types.

Although, these principles are still valid, NA engines should be modified also to obtain as much of their benifits as is economically wortwhile. Reducing, per example, the fuel consumption rate by NA version, is of benifit for later turbocharging also. Thus our attention must be paid to fuel saving by NA engine at first.

It may be meintioned, that combining turbocharging and CH_3OH as fuel further fuel saving may be obtained and thus <u>in a very simple way.</u>

TOPIC 9

Exhaust gas on-line analysis

1. THC - THCO

Dealing with methanol as fuel, especially at low load, in cold operations and transients some correction must be used in FID outputs, when as CH_3OH equivalent calibrated. It may be done by calibration itself or approximately by:

$$C_{THC}\ \text{measured} \cdot \frac{3}{2} = C_{THC}\ \text{evaluated}$$

In the case of calibration we are using the procedure acc. to Fig 1.

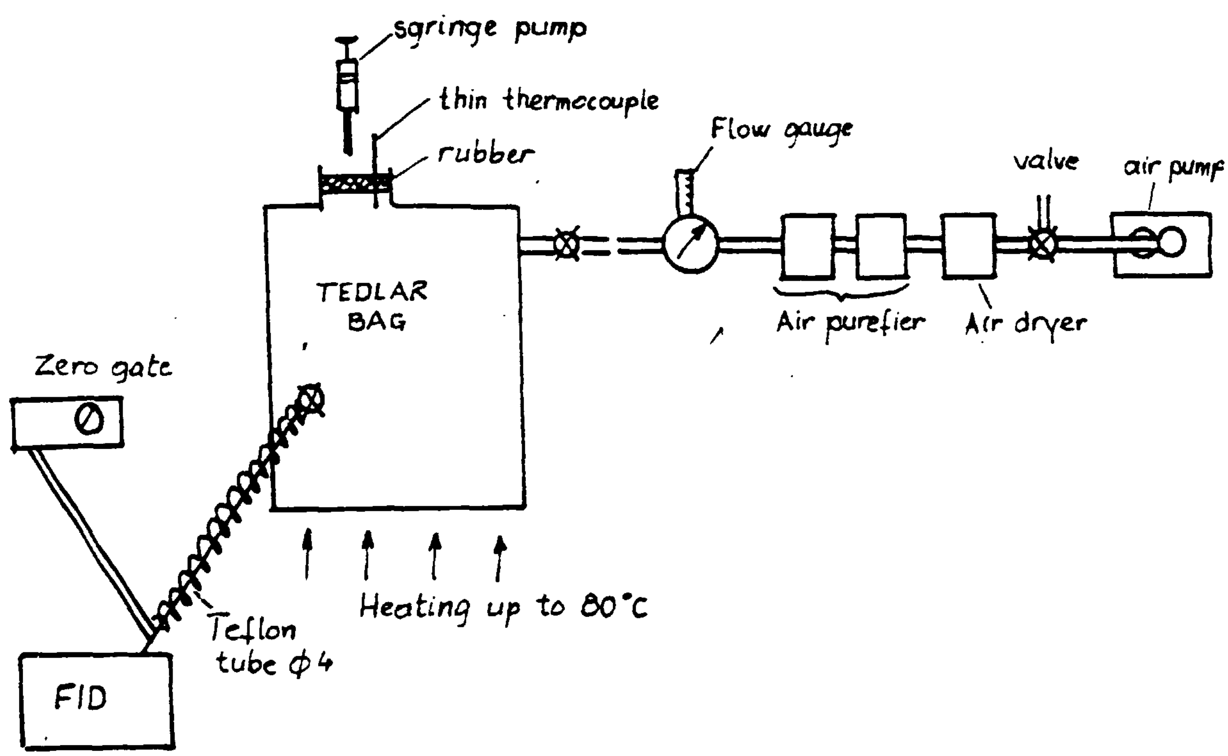


Fig 1 With pump CH_3OH of known volume should be injected

We are using air instead of N_2 when dealing with diesel engine. The reason for that:

- at low load operation exess air is very high, thus stoichmetric ratio may be exceeded six times
- just at low loads THC concentration are important

Having O_2 in excess our sample gas changes character of hydrogen envelope flame into more prexined one. Changing character of diffusion flame ionization effect will be changed and so does FID output also. To compensate that the best way is to use the purified air as dilution gas (see Fig 1). The same has to be used by C_3H_8 dilution. On the contrary dealing with SI engine weve been using N_2 as "carrier gas".

For THC equivalent C_3H_8 calibration and diesel exhaust analysis we use the procedure acc. to Fig 2.

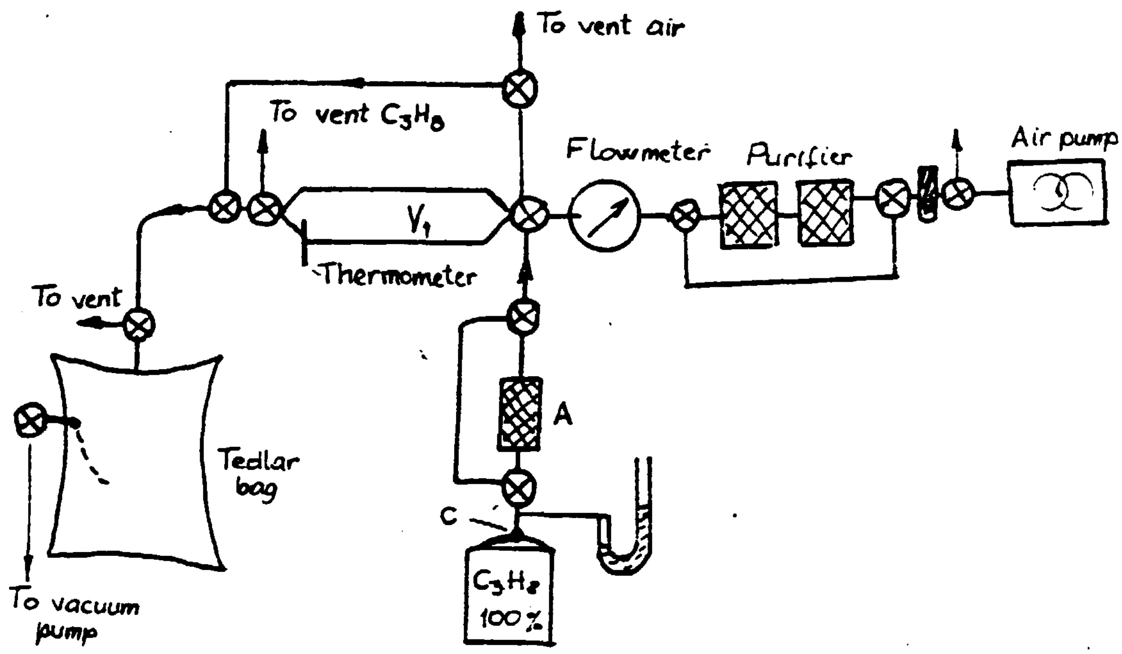


Fig 2 Preparation of C_3H_8 span gas

C_3H_8 % must be checked by gas chromatography. We use commercial C_3H_8 for industrial application. According to the impurities presented we choose the purifier - A (see Fig 2).

Volume V_1 is C_3H_8 trop, which must be precisely measured in before hand. Tube B is very long stretching out of lab. After purefying the volume V_1 with C_3H_8, valve C must be closed to wait about 30'for ambient pressure accomodation in V_1.

We use the our special very simple anregement to check the accuracy of flowmeters, what is very important for calibration gases.

CO_2 calibration gas we prepare in the same way using Kipp apparatus ($CaCO_3$+H_2SO_4). To remove the water one drying column may be used. However, because of very low pressure by CO_2 production, case must be taken related to resistances and time for trop spilling getting longer (NDIR-calibration).

CO is produced using anticacid+water. Very slowly only some water droplets are added and the temperature of acid is to be observed 40 ► 60°C. Again, because of low pressure core must be taken related to resistances (NDIR calibration).

Becouse of very high carbon particulates content in diesel exhaust gases, some filtration must be applied. However, THC emission is only of interest at very low load, where the soot emission is very low also. It means that, only two glass filters may be applied by heated up to 205°C. We use ∅ 4 mm teflon tube and heating technique the same as shown in Fig 1. In order to measure concretly combustion air is cleaned over two columns

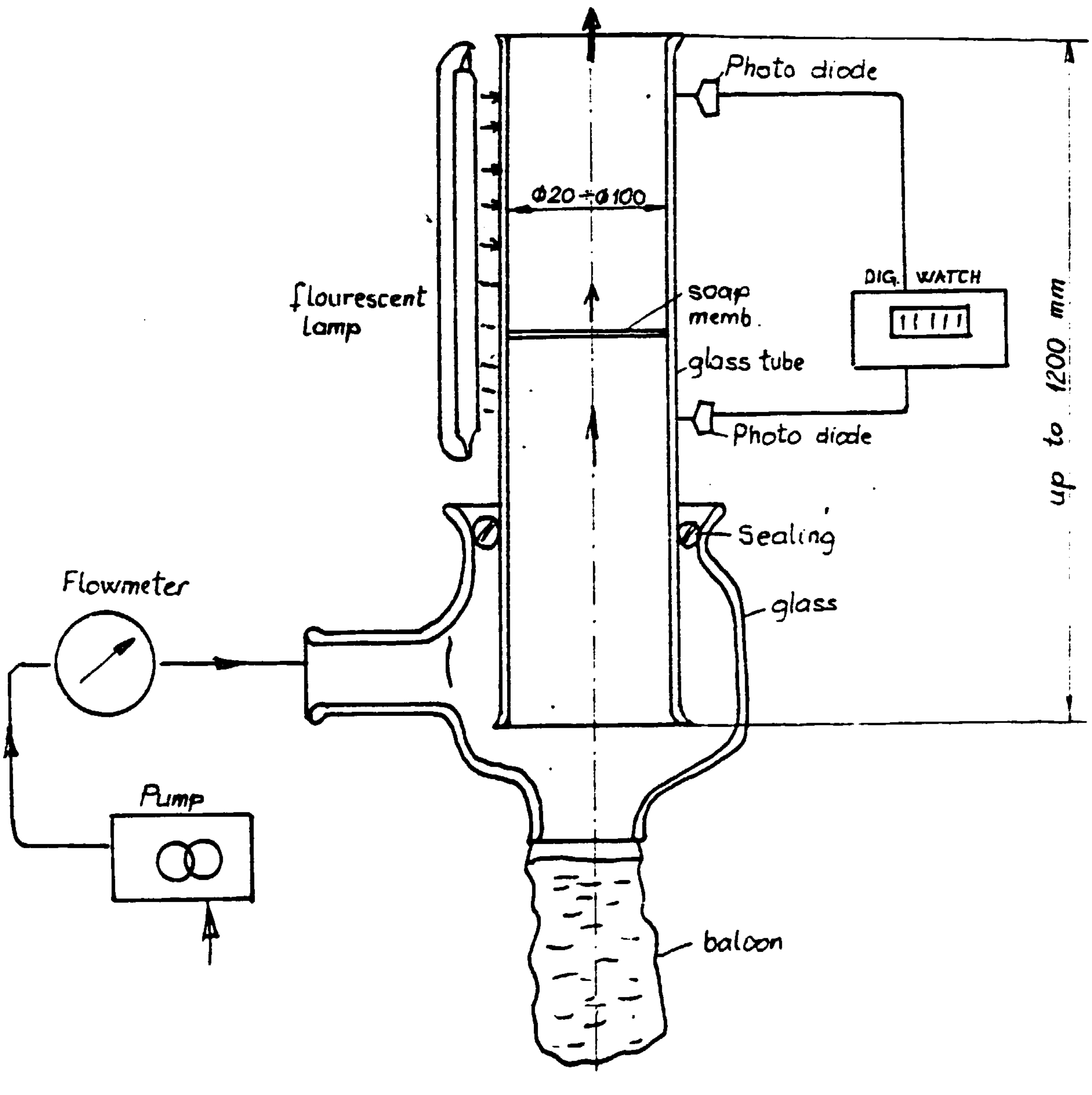


Fig 3

(separate air generation). To find plateaux increasing H_2 flow rate is practically impossible with some appatuses. Therefore is the best way to keep about 80% of max. value and calibrate of the same H_2 flow rate.

Background can be depressed only if clean air is to be used.

With H_2 low pressure setting oscillations of diffusion flame may be observed and variation in the date measuremed also.

Data measured must be reduced at stechiometric ratio a expresed as emission index

$$\frac{(g)\ TCO}{(g)\text{fuel burnt}} = \frac{(g)^{C_3H_8}}{(g)\text{fuel burnt}}$$

Only in this way analysis is possible.

2. NO_x measurements

One of the very difficult analysis is on-line NO_x measurements in diesel engine exhaust gas. The arguments may be mentioned:

1. NO_x is of interest at higest loads but the soot emission is also high
2. NO_x may be absorbed, destroyed and converted by: soot collected, high surface to volume ratio of sample line and material used.
3. Converter efficiency decreases with time
4. Partially closged cappilaries in the chemiluminesenct analyses
5. Optical filter of photo multiplyer soilling
6. High vacuum needed in reaction chamber
7. Span gas change with time (NO/NO_2)
8. Sample flow rate decrease because of soot collected on filters
9. NO_2 loss in water trap.

In our lab praxis we use the scheme shown in Fig 4.

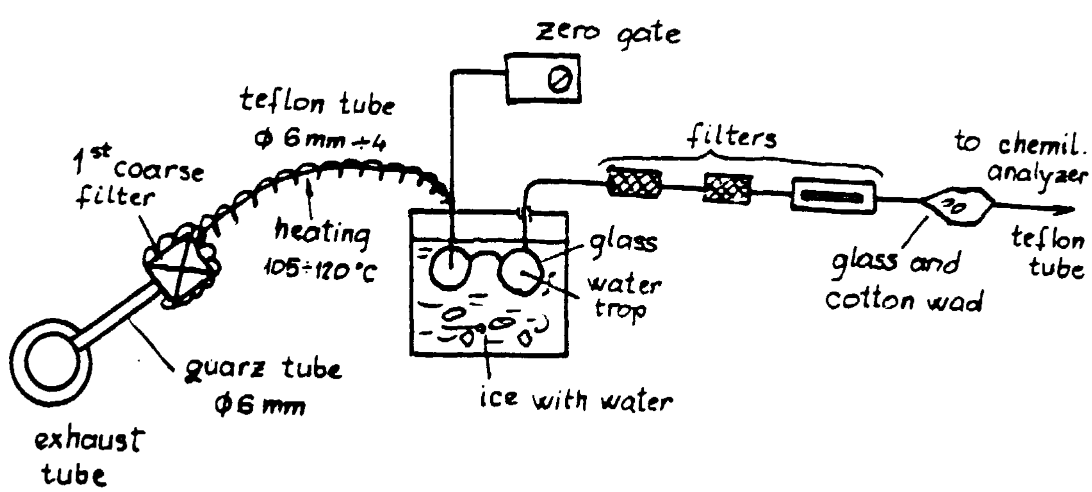


Fig 4 NO_x on-line measurements

Filters shown in Fig 4 are explained in the 1st Report already.

The first part of sample line is heated to avoid condensation of water in the teflon tube (surface/volume high).The lenght of heated tube approaches 1 ÷ 1,5 m only.Fast condensation of water but in smallest surface/volume ratio (two spherical glass traps).

At temperatures less than 100°C soot collected absorbes NO_x and at higer than 260°C starts NO_x destruction.At 900°C and with resoanable sample flow rate,soot collected in a column of about 300 mm lenght,destroys NO_x complitelly.

Therefore,care must be taken obout temperatures and omount of soot collected.However,in on-line measurements first sign is sample flow drop and decrease of analyzer outputs.Is to be noticed that,sample flow rate has to be kept the same as for calibration.

NO/N_2 high span calibration gas may be deluted by N_2 but,two "Oxisorb" column must be used to purity N_2 and to ovoied $NO \rightarrow NO_2$ conversion (calibration in error and bottle attack).

To check converter efficiency or calibration gas being long--lasting in the bottle,self made converter ($NO_2 \rightarrow NO$) may be used acc. to Fig 5.

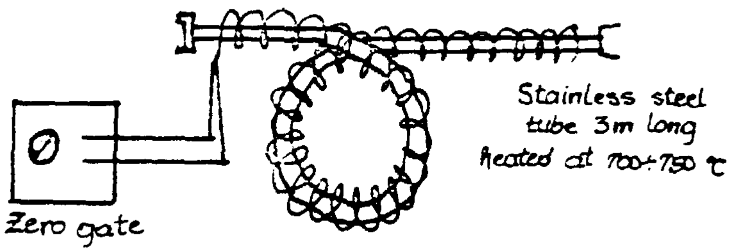


Fig 5 Converter $NO_2 \rightarrow NO$

Continuasly doping O_2 in NO/N_2 mixture unstable calib. gas NO_2/NO may be produced of known concetration.However,dynamically produced NO_2/N span gas must be in on-line technique continuarly used.

TOPIC

Dynamic

During mission two engineers have been trained related to piston-mechanics and balance. One booklet (66 pages) was presented and translated into english. The whole subject was considered in the way of computer application and calculation of:

$-x, \dot{x}, \ddot{x}, \beta, \dot{\beta}, \ddot{\beta} \rightarrow f(\alpha)$

$-X, Y, M_x, M_y, M_z$ instanteneous inertia loading

-Balancing

$-K, K', K'', \mu K'', K_1, K_2, K_3, K_4, P_1, P_2, P_3, P_4$ instanteneous whole loading

$-\mathfrak{M}_e = \mathfrak{M} - (\mathfrak{M}'' + \mathfrak{M}''' + \mathfrak{M}'''')$ instanteneous torques

$-\mathfrak{M}'''' = \mathfrak{M}_1'''' + \mathfrak{M}_2'''' + \mathfrak{M}_3'''' + \mathfrak{M}_4'''' + \mathfrak{M}_5''''$ instanteneous torques of friction

-fly whel calculation

-torsional vibration problem, calculation of eigen frequencies, resonant state only.

Some examples are given related to application. Per example using Eq. for $\mathfrak{M}'$ the whole starting process may be studed, of interest for CH_3OH use as fuel.

TOPIC 3

High - pressure injection

Efficiecy of diesel engine combustion depends mostly on fuel-
-air mixture formation.Mixture formation depends in:
-fuel pressure drop in nozzle holes
-fuel spray direction relative to air movement
-beginning of injection (timing)
-period of injection
-law of injection
-atomization of fuel injected
-stoichiometric ratio
-swirl and squish
-combustion chamber or piston bowl
-air temperature
-pressure of air.
The all above factors mentioned are location and time dependent and have to be optimised via compromise between:
-fuel consumption
-exhaust emission-combustion noise
-mechanical loading
-thermal loading.

As mentioned already,decisive factor is time-volume history of mixture formation in a very narrow space during a very short period of time.

With diesel engine the both,air and fuel take part in mixture formation.Thus,the energy E_R needed for combustible mixture formation may be expressed by:

$$E_R = E_q + E_z \qquad (1)$$

where: E_q - part of E_R given by fuel injection
E_z - part of E_R given by air motion.

A logical hypotesis may be : the higher E_z the lower E_q.
But higher E_z means:
-lower volumetric efficiency with NA engines
-higher thermal loading of the parts forming combustion chamber
-higher heat loss
-bad cold startings.

The drawbacks mentioned were the impetus to increase the part E_q or to apply high-pressure injection (HPI).However,it's not the whole philosophy of HPI, still one point must be considered. Namely,with conventional fuel injection system (CFI) using of the same time E_q as well as E_z for mixing,we experienced speed dependent dis portions related to the mixture quality (Fig 1).

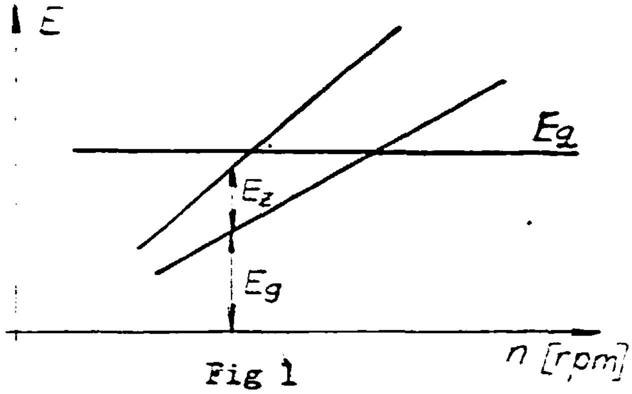


Fig 1

E_g (see Fig 1) may be considered as constant with vehicular diesel engine at full load operations because of nearly constant fuelling.Meanwhile, the both E_q as well as E_z are increasing with speed (n) increased (see Fig 1) ; thus at higher speeds we have surplass but at lower speeds deficit of energy needed for good mixture formation (soot,bad torque back up).

Using theoretically only E_z for mixture formation results in a high pressure drop Δp at nozzle holes.The question arises, what's then with spray tip penetration related to small piston dia.'s?Spray penetration getting longer and so doing fuel deposition on the wall when absence of swirl combustion may be poor.However,depending of nozzle hole geometry we may have a dense high penetrated spray or finer dispersed short penetrated,which supports faster transformation of liquid phase into vapourous one and better homogenisation of combustible mixture.

Thus,we are asking not for high penetration,on the contrary, the more intense spray must be better atomized only under all speeds.Here is the second point of HPI philosophy wich accents: <u>not only HP injection but independent of speed also.</u>

With much better atomization HPI clams for better combustion history which results in increase of thermal efficiecy and reduction of soot emission.Morever,when swirl intensity reduced engine volumetric efficiency increases and "unnecessary" heat transfered to walls will be reduced also.

However, it must be noticed that fast combustion can not be applied when accompanied with a high in-cylinder pressure rise (Fig 2) $dp_z/d\alpha$.

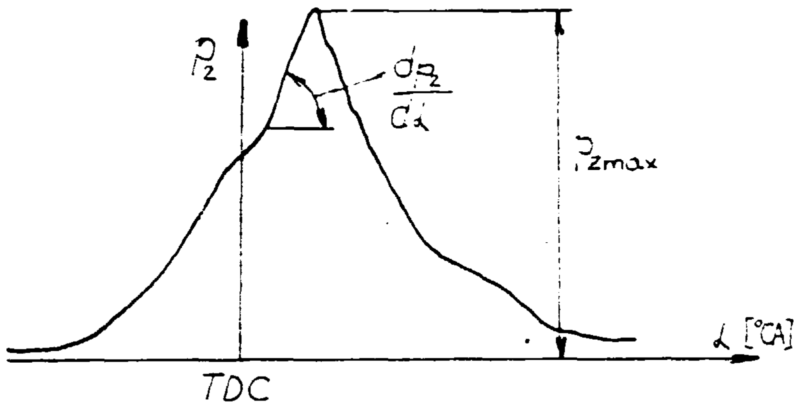


Fig 2

Otherwise, a faster combustion may be of benefit when timing retarded. Namely, retarded timing and fast combustion may result in $dp_z/d\alpha$ decrease as well as p_{zmax} reduced ; <u>but only if engine reacts.</u>

Years long (acc. to Parker) we connected HPI with shorter injection period (α_{ub}).

a) Converting some engines to HPI, shortening α_{ub} without change of timing we experienced

- $dp_z/d\alpha$ increase
- combustion noise increase
- p_{zmax} increase
- NO_x increase
- soot emission decrease
- specific fuel consumption decrease
- exhaust temperature decrease

b) HPI, shorten injection period (α_{ub}) and retarded timing resulted in

- specific fuel consumption increase
- max. in-cylinder pressure decrease
- $dp_z/d\alpha$ decrease
- exhaust temperature increase
- soot emission increase
- noise emission decrease.

<u>a)</u> and <u>b)</u> above are the general tendecies. The case <u>b)</u> speculates:

1. Timing decreased → better conditions for ignition
2. Faster ignition → faster combustion.

However, case b) gave very often disappointing results because of engine unexpected reactions. Thus we compromise in-between: fuel consumption figure, emission, mechanical and thermal loadings. Here is the question: <u>why many engines have not been reacted according to case b) speculations ?</u>

Some dilemmas about noncorresponding engine reaction may be discussed as follows ;

$\underline{\underline{x}}$ - HPI application increases the pressure in the whole HP system (retractive valve cap, HP tube, injection holder) but HPI means a high pressure drop (Δp) at nozzle holes only. When, per example, any where too high elasticity existes or/and too much resistances a high pressure produced by pump may be considerably lost and then it's not HPI at all.

$\underline{\underline{y}}$ - the more intense pressure drop at nozzle holes may be used for:

(1) - better fuel atomization
(2) - longer spray penetration
(3) - the both (1) and (2)

In the case (2) we expirienced too much fuel deposited on the wall and in the absence of intense swirl homogenization of mixture deteriorates.

In case $\underline{\underline{x}}$ or/and $\underline{\underline{y}}$ we increased fuel consumption only. More energy used for injection and because of p_{zmax} timing was retardet , thus the both directed to fuel consumption increase.

Coming back to the points $\underline{a}$ and $\underline{b}$ let show the typical diagrame we know (Fig 3).

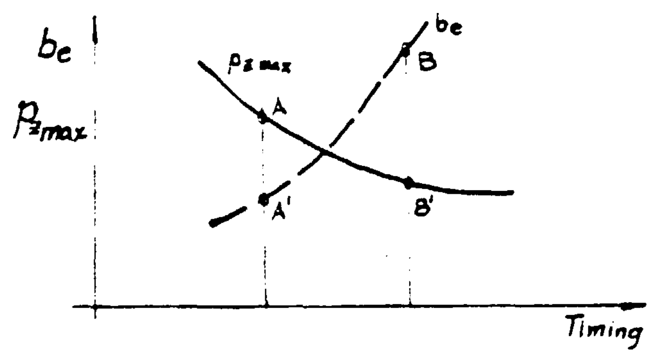


Fig 3

Fig 3 is a typical representation of HPI of of higher Δp application but accompanied by shorter period of injection.It was the essence of our HPI understanding because we applied HPI with shorter injection period only (acc. to Parker).May be not too short ? It was practically only geometrical redistribution acc. to Fig 4.

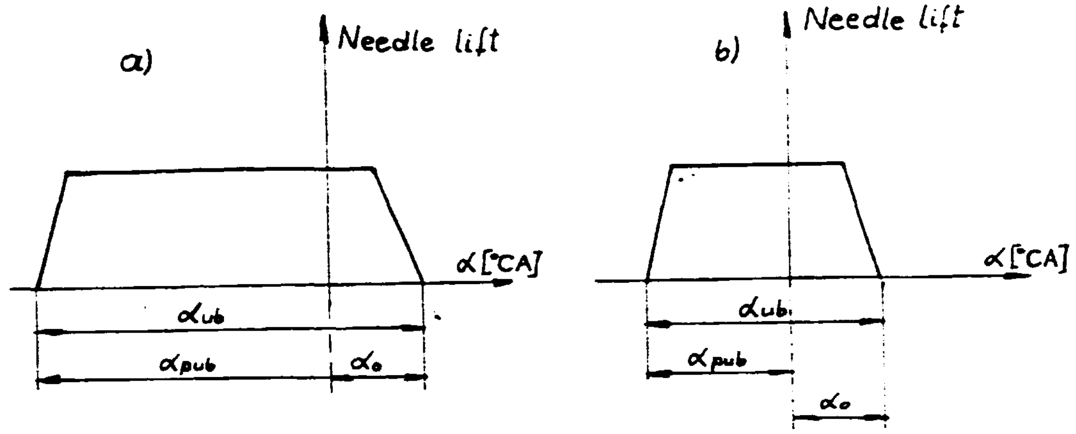


Fig 4 a) conventional injection system
b) HPI system

Let's remember that,about 3 years ago Daimler Benz (DB) - Bosch investigated HPI in cylindrical DB combustion chamber. Injection pressure was increased successively up to 2000 bar, but the concluded that cylindrical DB bowl may not be fit related to HPI.However,we may comment their results as:

"spray penetration was infavourable for the combustion chamber and swirl chosen when pressure drop in nozzle holes increased."Was the Parkers philosophy a wrong one ?

When dealling with HPI still two points have to be considered:

1. When fast injected instanteneons fuel concentrations per unit volume of combustion chamber increase,what may results in droplet coagulation.

2. Very fast plus very fine dispersed fuel requires a lot of energy.Per example: 50 mm^3 fuelling only but despersed uniformly in 2μm required 55 kW.Therefore,CRIDEC system may be a solution related to energy need.Only to mentions that W.Ball (Ricardo) calculated (for conventional system)energy need about 1% of engines power output.

On 24 June 1984 in TH - Karlsruhe the newest results where reported related to HPI (Heinrich and Drescher,TU-München).

The essence of the report was:

- when bad shorter than finer atomized but with the same duration of injection as for conventional system. Increased pressure drop in nozzle holes was used to increase atomization only.

We may anticipate the next step of HPI developement in combustion:

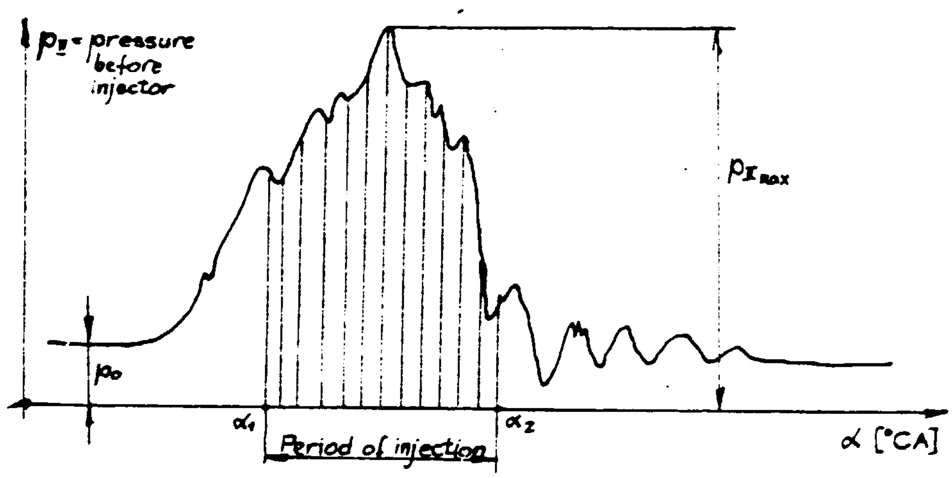
- shorter duration
- finer atomization

but the both matched to the swirl, squish, bowl, outputs and speeds.

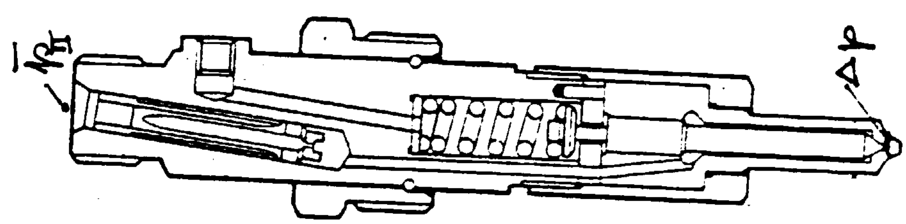
For the purpose of experiments F. and P used one modern (closed design) conventional pump made possible p_{IImax} up to 1300 bar.

Is to be noticed, that HPI does'nt mean a high p_{IImax} only. Morever, the mean effective pressure during effective injection $\bar{p}_{II}$ must satysfy the conditions:

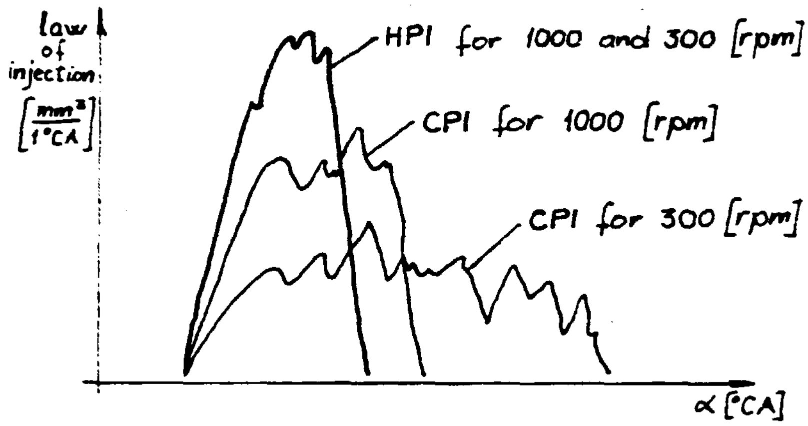
1. $\bar{p}_{II} \geq 0{,}6\ p_{IImax}$



2. $\bar{p}_{II}$ increased, used for Δp increase only



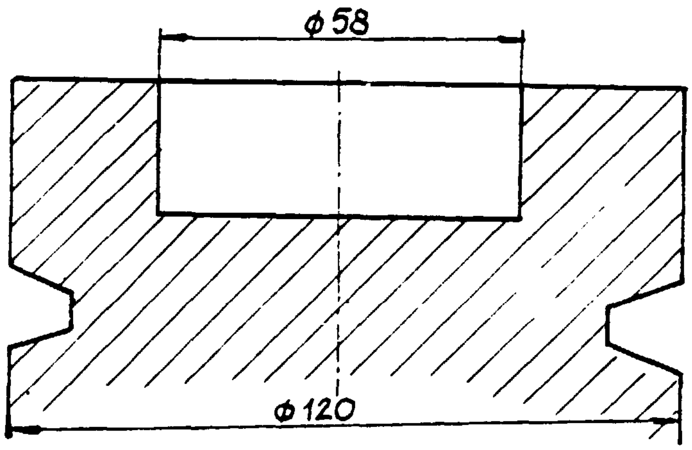
3. $\bar{p}_{II}$ does'nt depend of n /rpm/



The shorter injection period the less important shape of injection law. At very short injection law of injection becomes one pulse only.

For the purpose of experiments Heinrich and Prescher (H&P) used DB single cylinder research NA engine ; V_H=1,8 dm^3 , Ø128x 140 mm, rated speed 2200 rpm, CR=17,4. Related to fuel consumption figure, combustion chambers as well as swirls where separately optimised for CPI-and HPI system. Experiments II in this Report.

For the purpose of analysis of mixture for motion and combustion single cylinder air cooled engine was used ; Ø120x140mm, rated speed 2300 rpm, ε =19,5 ; piston bowl Ø 58 mm.



Injection system used:

<u>CPI</u>

HP pump BOCSH PESIPM Ø10 mm, prelift 2,8 mm

Retraction V_R=const.=50 mm^3

HP tube Ø 2x925 mm

Injector DLLA 150S/86,4 holes, μf=0,27 mm^2

Injector open. pressure 175 bar
Fuelling up to 150 mm^3/cycle was used

HPI

HP pump BOSCH PE6Z..140, Ø 15mm, prelift 1,5 mm
HP tube Ø 1,6x1180 mm
Retraction p_0=const.(const. residual pressure)
Injector DLLA 145 PV, 4 holes
Injector μf=0,11 ÷0,18 mm^2
Injector opening pressure 500 bar

With HPI effective cross-sectional nozzle holes flow area (μf) was varied in order to keep the same period of injection as for CPI. Injector opening pressure was extreme high. However, TAM-Maribor has one export engine with 270^{+8} bar opening pressure.

Here is a new approach in H&P work. With pressure increased we usually increased uf to shorten period of injection, but they kept injection period the same (with reduced μf) as for CPI - - better atomization.

In Fig 5 are $\overline{p}_{II}$ presented for the both CPI and HPI system. It may be seen that:

$$\overline{p}_{II(HPI)} \approx 3 \cdot \overline{p}_{II(CPI)}$$

approaching to 1300 bar at rated power.

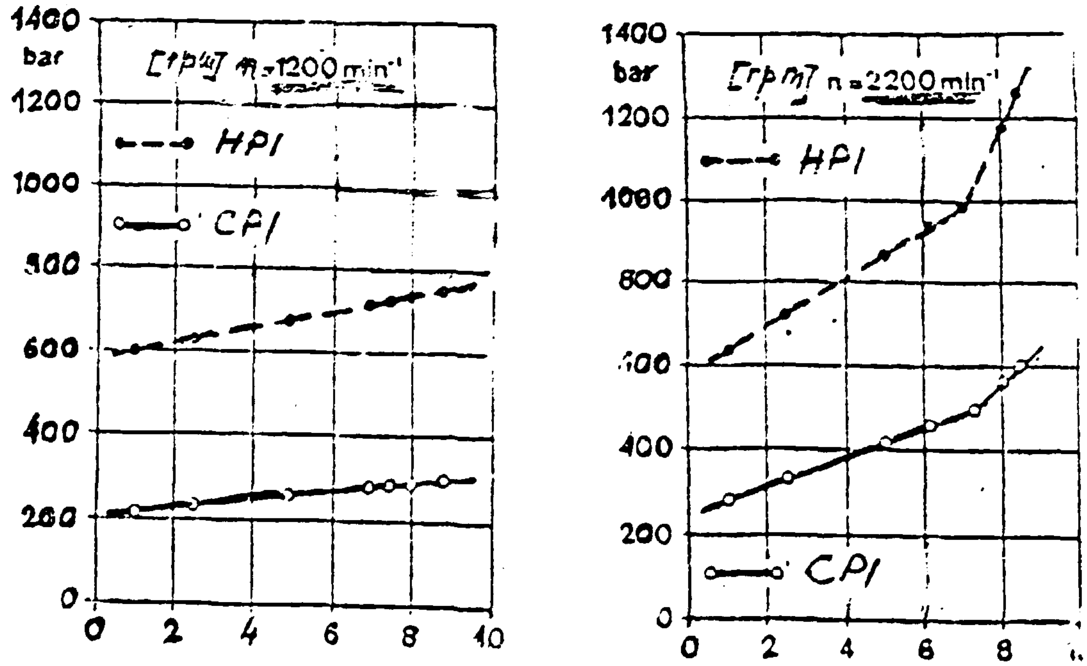


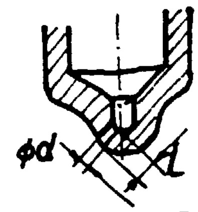
Fig 5

In order to follow spray events in I experiments 3 nozzle holes were pluged and fuelling reduced on $V_{cIexperim.\,spray} = \frac{1}{4} V_{cIexperim.\,engine}$.

It means results obtained may be used qualitatively only.

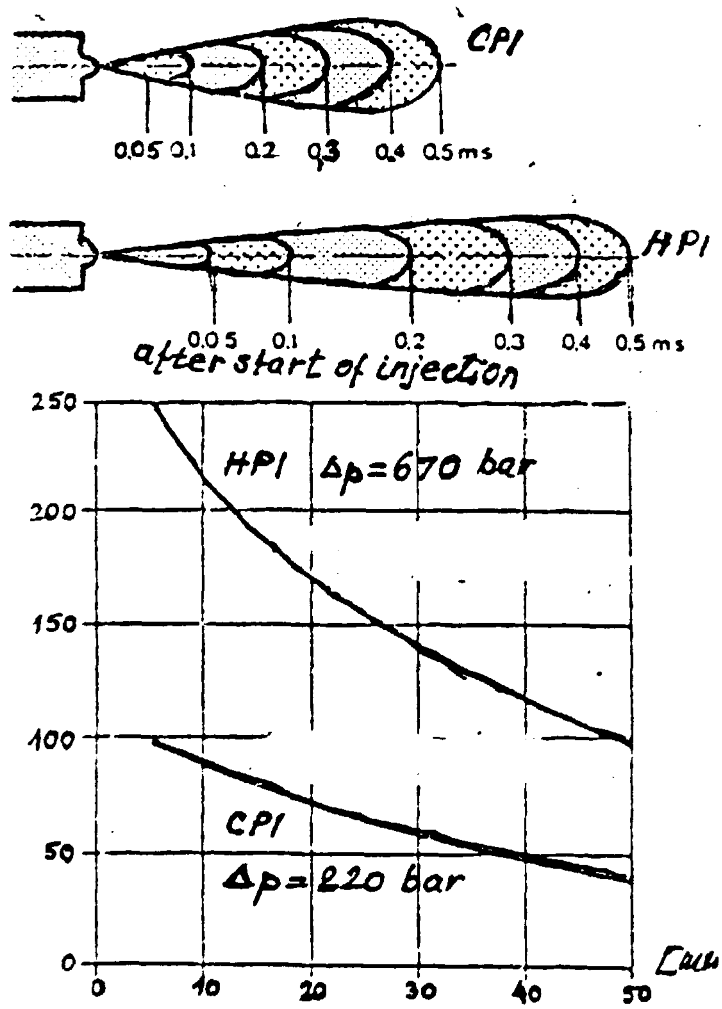
Photograph taken were evaluted for the both systems being operating under conditions:

- same start of injection bTDC=13/°CA/
- same l/d of nozzle hole



- same fuelling at 20 mm³/cycle
- without swirl.

Fig 6 depict the results.

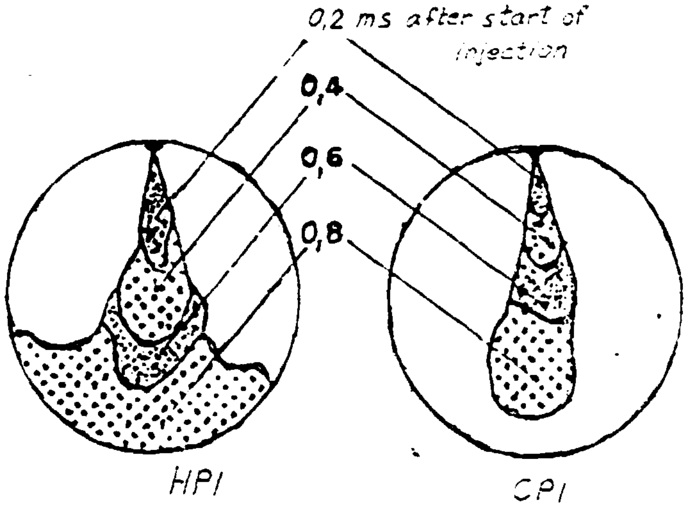


Injection period
10 /°CA/

Speed 1000 /rpm/

Fig 6

With HPI fuel spray got more narrow and it tip velocity was nearly by factor 2 higher than of CPI. HPI formed earlier the fuel vapour phase and in larger quantity, see Fig 7.



fuelling 20 mm^3/cycle
start of injection 15 /°CA/bTDC
injection period 10 /°CA/

Fig 7

With CPI, vapour phase is mostly around spray but HPI vapour phase may be observed into the spray also.

With HPI liquid pfase converts faster into vapour one. Despite vapour penetration in the front of spray, it tip penetrates longer and more early reaches the combustion chamber wall.

With the both CPI and HPI, when swirl of middle intensity combustion started at the same α /°CA/ supposing the same instant of beginning of injection. It means that with the both, CPI and HPI ignition delay was the same, Fig 8.

When HPI, combustion starts abruptly in the larger part of bowl volume and the spray front. When CPI combustion starts around the spray. With HPI after stable start of combustion, vapour phase does'nt develope any more, it means better homogenization and shorter period of combustion, Fig 9. It's in agreement with our investigations. We retarded for 4 ÷ 7 /°CA/ the timing, when HPI. Fig 10 shows the result of H P.

Fuelling 20 mm^3

Start of injection 13° bTDC

Injection period 10° CA

Swirl: middle intensity

liquid phase, vapour, flame

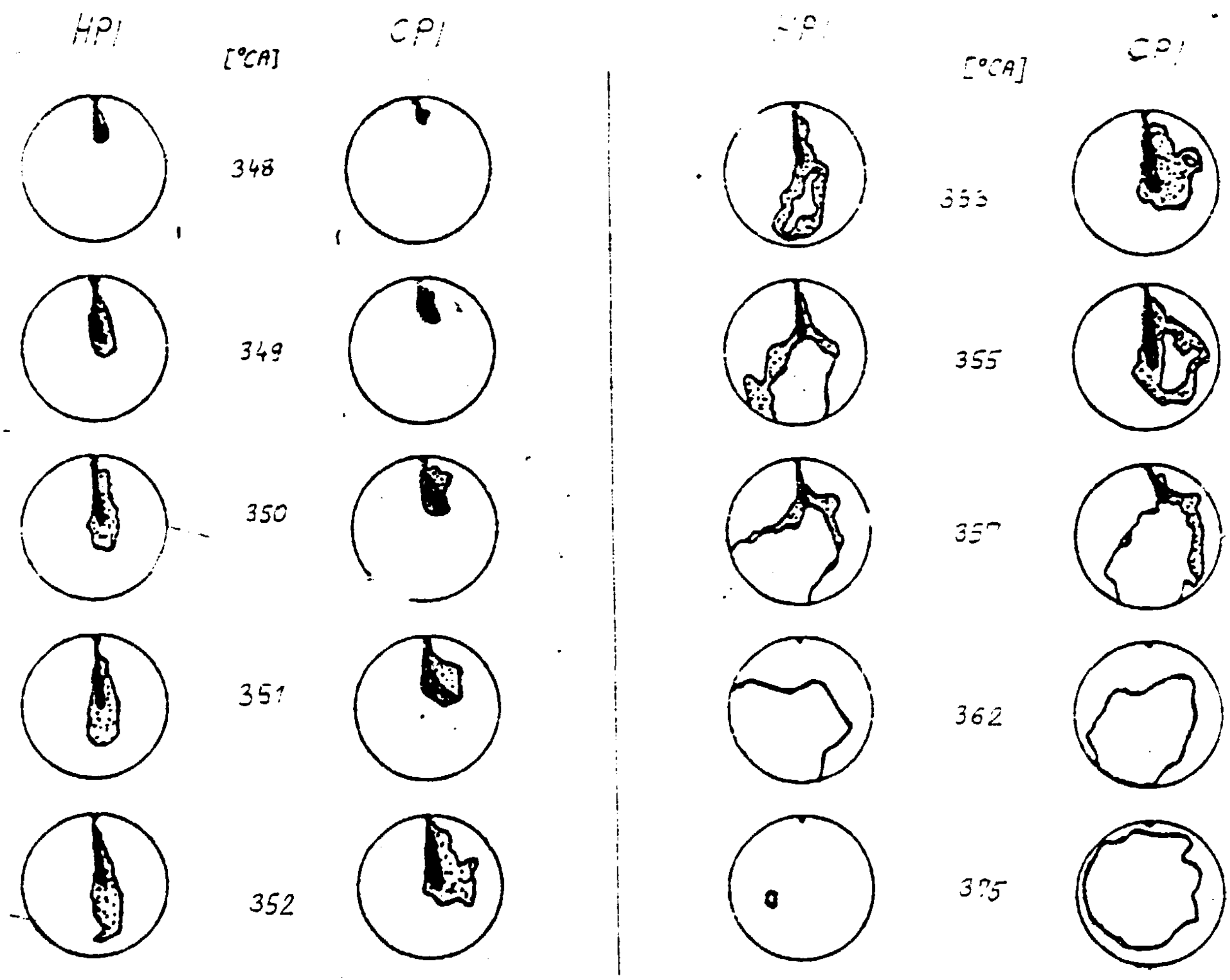


Fig 8 HPI - flame extinction at 375 °CA
CPI - flame extinction at 430 °CA

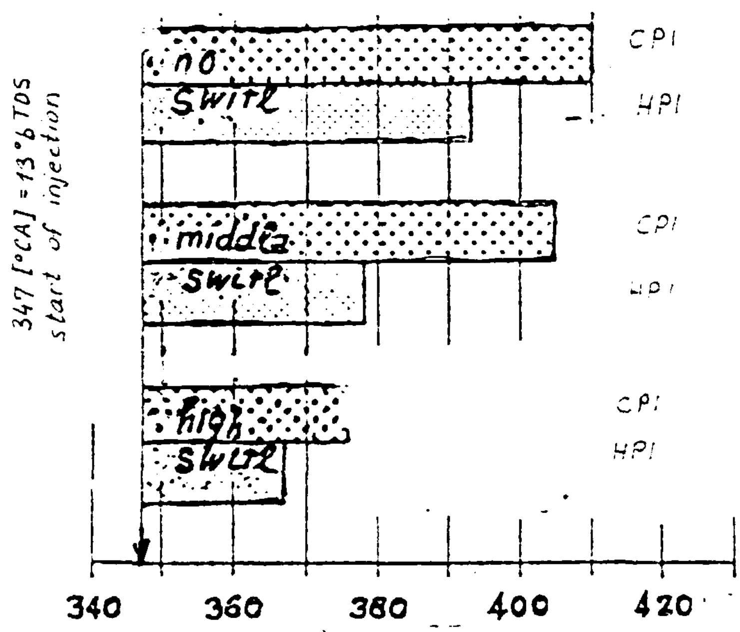


Fig 9 Duration of combustion
Fuelling 20 mm^3
Start of injection 13 °CA bTDC, injection period 10 °CA

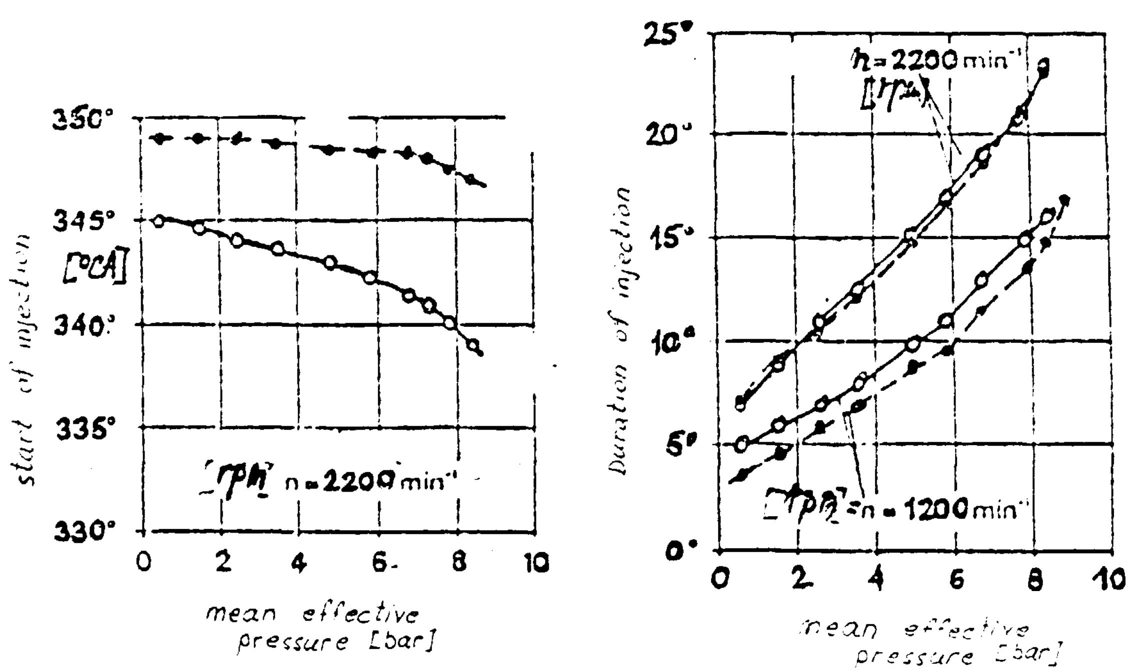


Fig 10

The results of experiments II

The both,HPI and CPI system were tested with 1D single-cylinder engine.However, piston bowl and swirl were matched separately from CPI,but injection period was the same.

As was yet mentioned,the ignition delay was the same the both systems,see Fig 11.

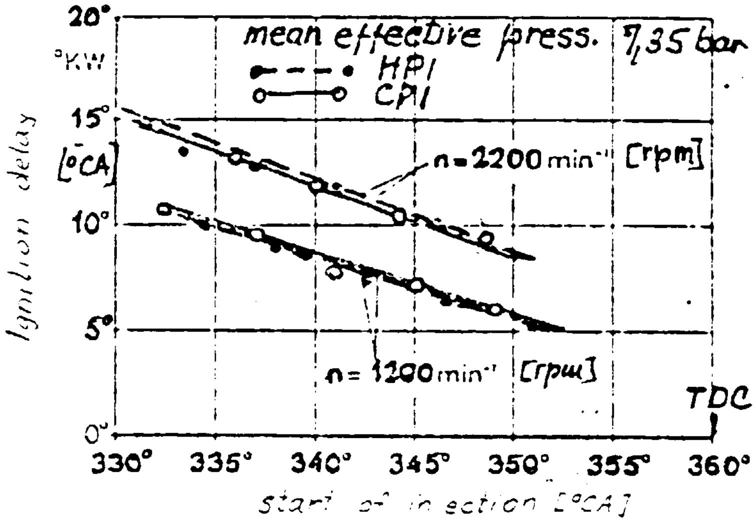


Fig 11 Start of injection was optimized related to fuel consumption in Fig 12

HPI had a high influence on soot emission,see Fig 12.

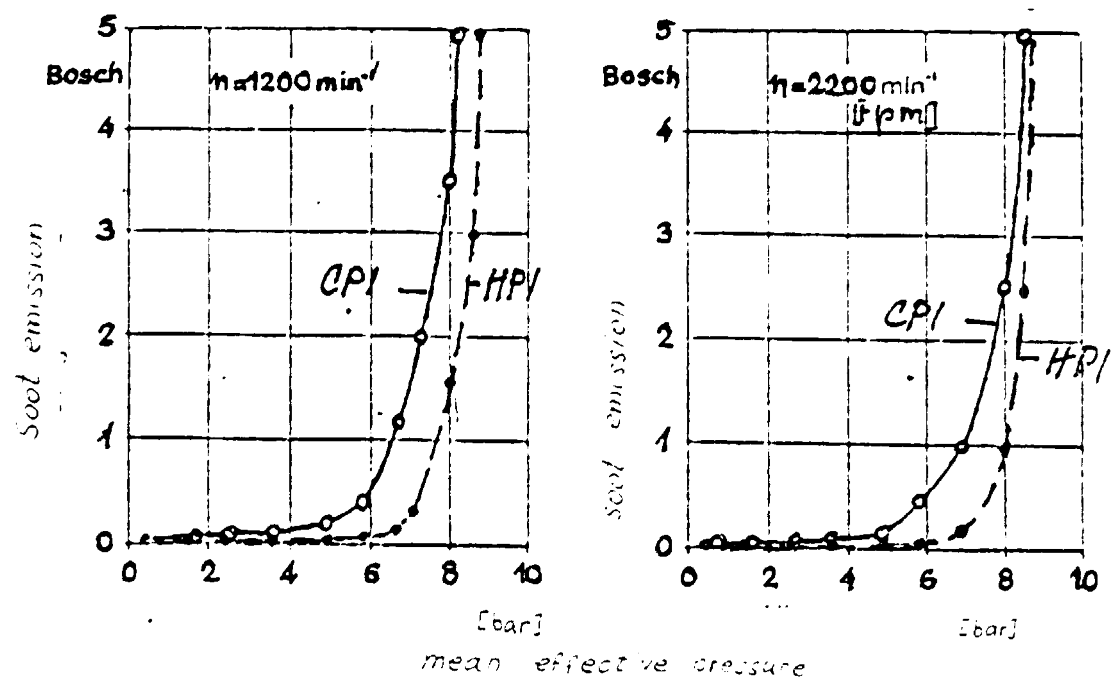


Fig 12

When limited soot emission at 2 BOSCH units, max. effective pressure with HPI was greater for one bar than that with CPI, see Fig 13.

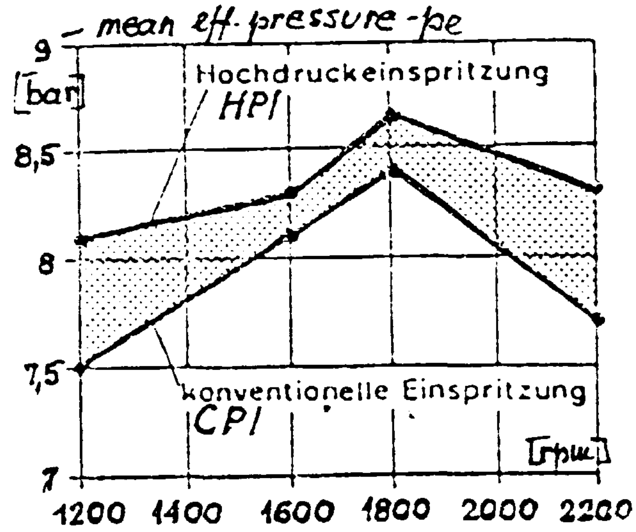


Fig 13

Similary, when soot limited at full load with CPI, application of HPI, for the same power outputs as for CPI, resulted in lower soot emissions for 0,5 ÷ 1,1 Bosch units, see Fig 14.

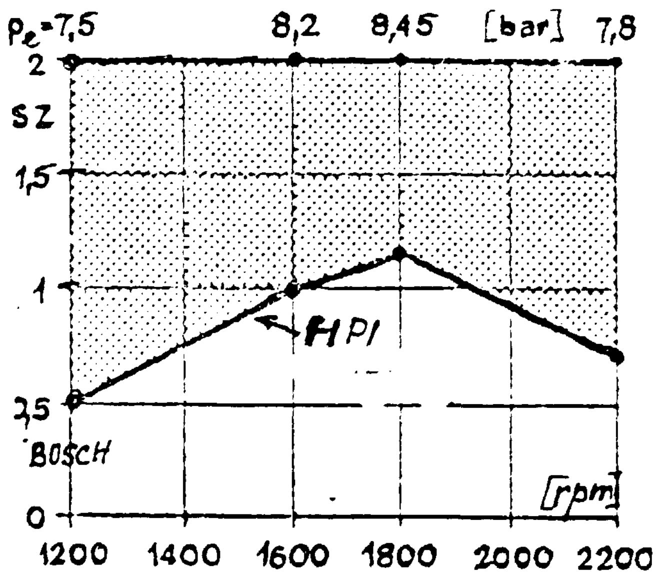


Fig 14

HPI and CPI - the same power outputs

Fig 15 depictes THC emissions for the both injection systems investigated but optimized related to fuel consumption.

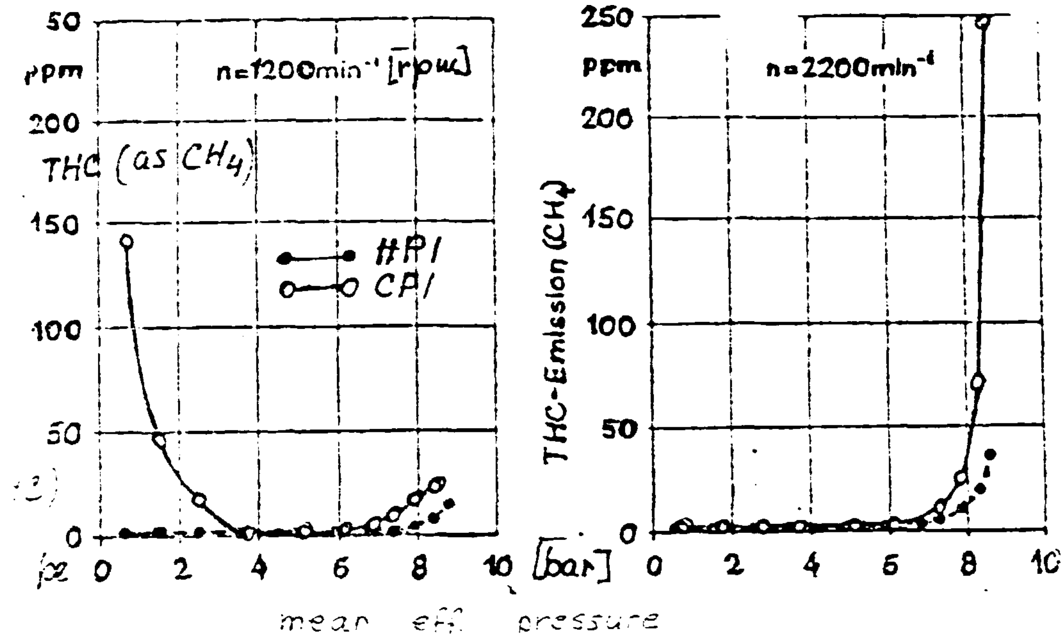


Fig 15

Typical NO_x dependences for diesel engine shows Fig 16.
Namely:

- strong dependence on loading
- weak dependence on speed.

NO_x emission was nearly the some for the both injection system used.

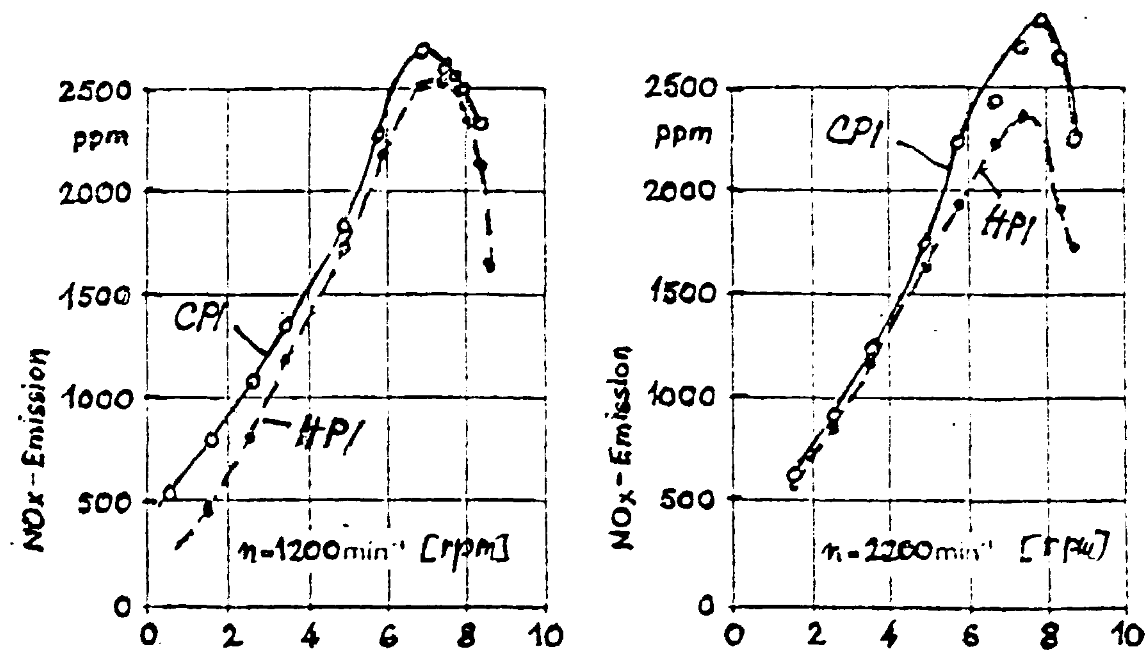


Fig 16 NO_x emission, timing optimized related to fuel consumption

Fig 17 shows that fuel consumption was nearly the same for the both injector systems investigated.

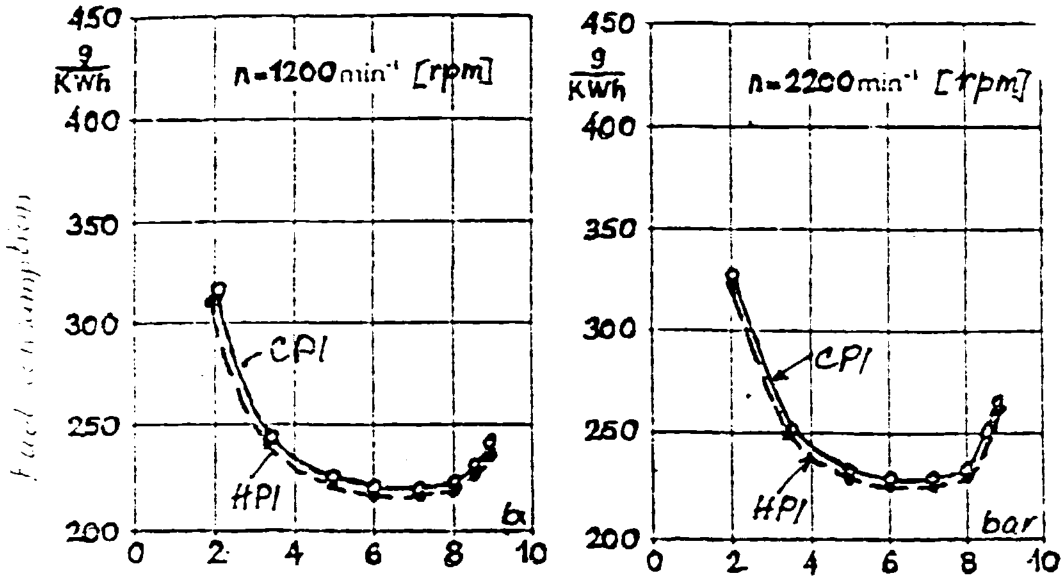


Fig 17

However,Fig's 18 and 19 depicte that pressure rise and heat release rate became higher with HPI,although max in-cylinder pressures were lower.The above said produced higher combustion noise than that with CPI system.Despite of timing retarded (15°bTDC),HPI showed (see Fig 19) a high increase of heat relase at TDC.It may be probably decreased by better matching of injection law.

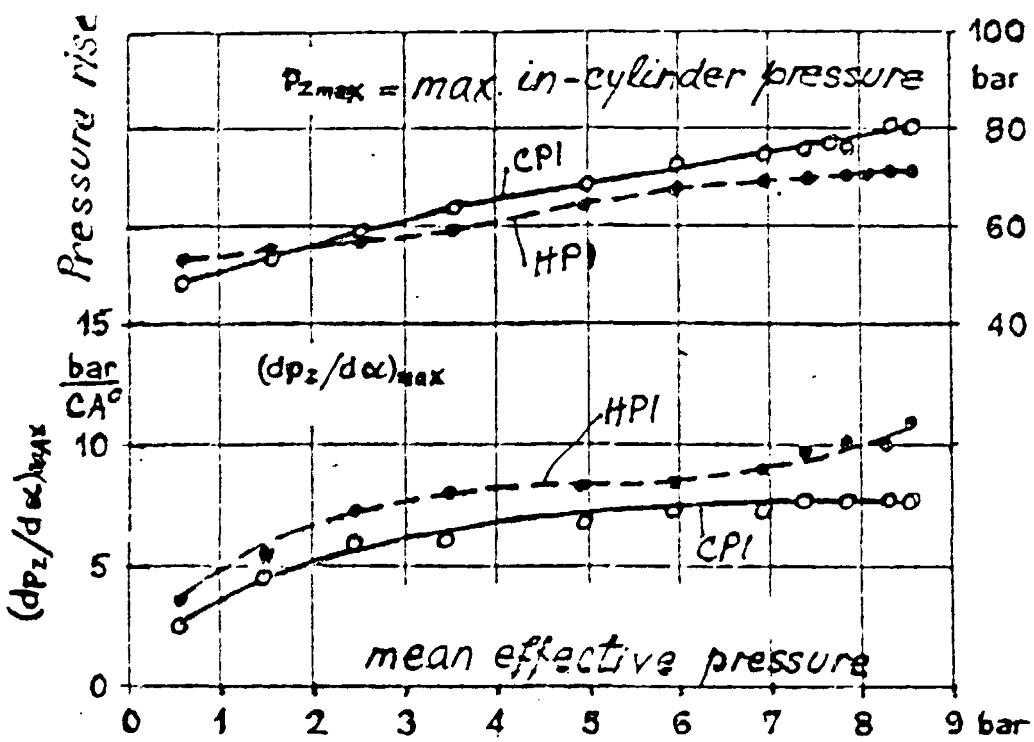


Fig 18

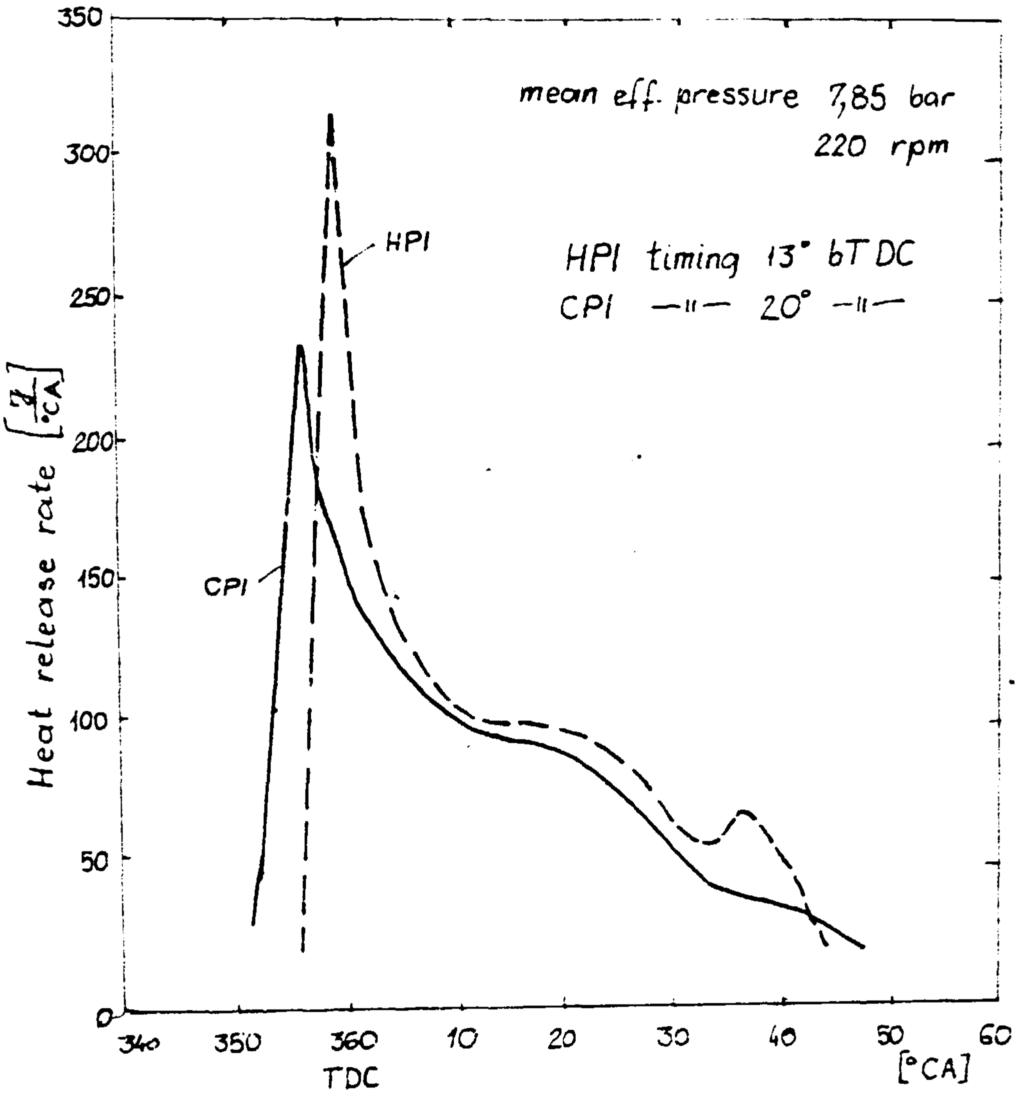


Fig 19

Heinrich – Prescher's conclusions:

- when Δ p=670 bar instead of 220 bar at nozzle holes, the velocity of fuel spray tip increases by factor 2 , so does fuel vapour also
- at const. timing, combustion delay does'nt depend on pressure drop in nozzle holes (in the range investigated)
- when Δ p higher, timing may be retarded and with this, ignition delay may be shorter
- with HPI the first apperance of the flame is around the flame front. Besides that, high temperature radiation drop faster and the combustion period becomes shorter.
- soot emission decreases with Δp increased.

Matsouka, Kamimoto and Kobayashi, as well as Kahn earlier, showed the "flame figure" when fuel spray penetrated to the wall of piston bowl, Fig 20.

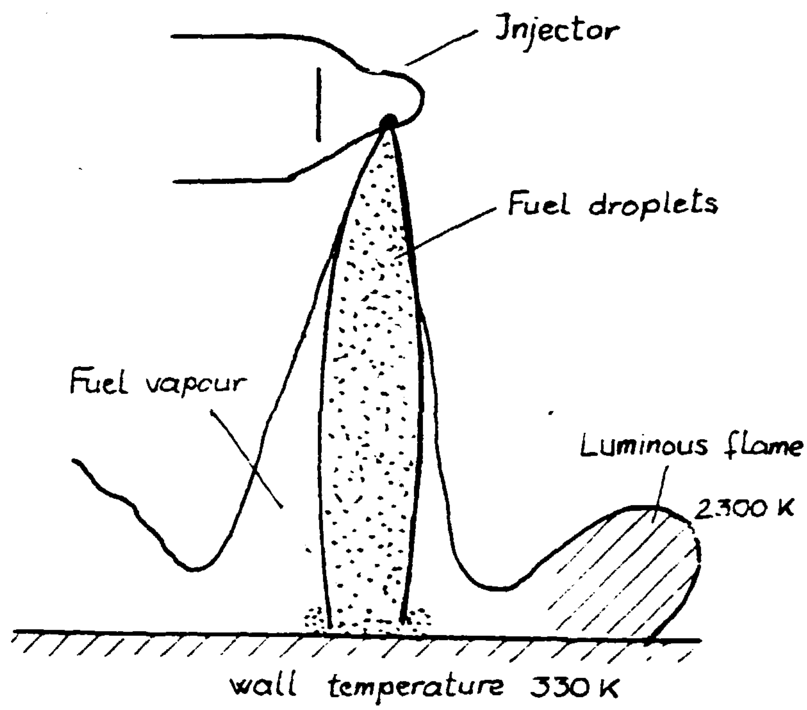


Fig 20

It's obviously clear that, having a larger nozzle hole dia., swirl intensity must be higher because of larger deposition of the fuel. It means to desperse more in air , smaller nozzle hole has to be applied or for less intense swirl, fuel must be finer dispersed in air.

Unavoidable shortage of CPI is a long spray distance at low and high speed operations. However, CDI (MAN) system claims to solve this problem, as was yet explained.

To disperse finer asks for higher pressure drop at nozzle holes. Here is to be mention.

1. $\bar{p}_{II}/p_{IImax}$ satisfied must be obtained at a high p_{IImax}.

2. Increasing p_{IImax}, resistances and elastic deformation of HP tube increase also. Thus, HPI may be applied when the corresponding stiffness of the whole injection system exists.

3. a high $\bar{p}_{IImax}$ we need for high Δp at nozzle holes only.

The expression for a part of fuel injected ΔV in a time Δt may be expressed as follows:

$$\Delta V = \mu f \sqrt{\frac{2}{\rho}} \sqrt{\Delta p} \cdot \Delta t$$

$$\Delta V = \mu f \sqrt{\frac{2}{\rho}} \sqrt{\Delta p} \cdot \frac{\Delta \varphi}{6n}$$

$$\Delta V = const \cdot \underline{\mu f \cdot \Delta p^{1/2} \cdot \Delta \varphi} \qquad for\ n = const\ [rpm]$$

where : μf– flow cross-sectional area of nozzle holes

Δp– pressure drop at nozzle hole

$\Delta \varphi$– duration in $^{\circ}$CA for the injection of volume ΔV

Thus:

a) $\Delta \varphi$ may be shorten with increased - shorter period of injection.

b) Still shorter period of injection may be obtained combining a higher Δp with μf increased - a larger hole dia.

c) When decreased μf and increased Δp; $\Delta \varphi$ may be kept unchanged--finer dispersion.

d) Every time is possible to combine μf and Δp for optimum $\Delta \varphi$

In order to increase Δp the below cited may be used:

- increase of plunger dia.
- decrease of μf
- increase of plunger pre-lift
- new shape of the cam in HP pump

but:

- after injection
- too high wear of cam-follower
- leakage at connection retraction valve cap-pump body
- cavitation wear

have to be avoided.

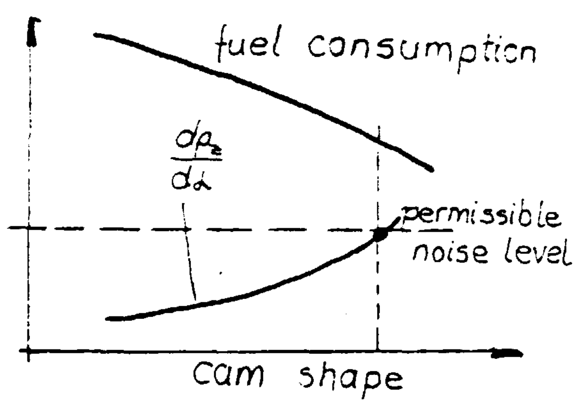


Fig 21

Fig 21 explains cam shape matching $dp_z/d\alpha$=in-cylinder pressure rise.

In DI-NA diesel engine may be of benefit to accelerate the combustion within 10 ÷ 15 °CA aTDC, let say for about 2 ÷ 4 kcal/°CAm^3.It may be done by:

- reduction of nozzle hole dia.'s
- application of nozzle design acc. to Fig 22.

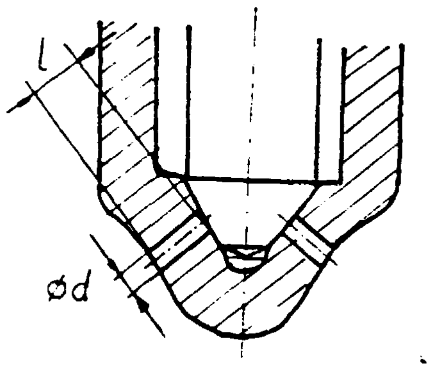


Fig 22 Sackless nozzle

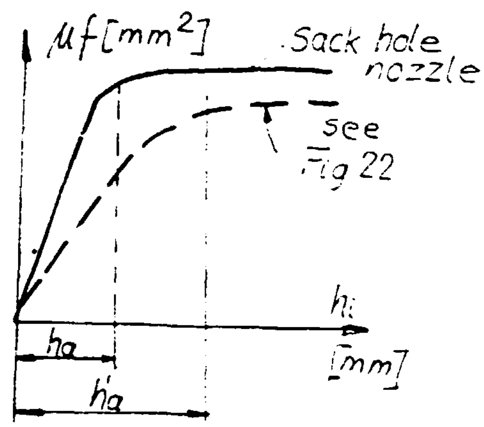


Fig 23

Sackless nozzle offers some advantages:

-less fuel injected in the period of ignition because of retarded increase of effective flow area of nozzle (μf) in dependence of needle lift (h_i),see Fig 23.(of interest for CH_3OH also).

- THC emission may be drasticaly reduced.

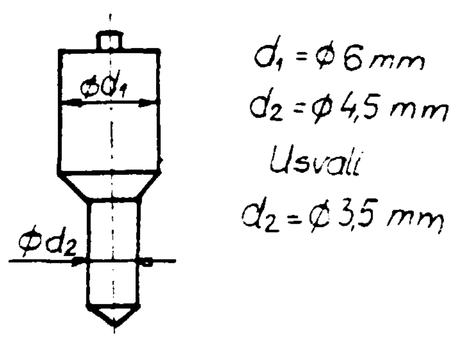


Fig 24

Potential drawbacks:

-reduced service life of nozzle

-nonuniform fuel distribution of nozzle holes with multi-hole injector

-l/d more difficulty to match.

With HPI injector opening pressure has to be increased. Doing that injector closing pressure increases also.Sometimes reduced hydraulic ratio of needle helps to obtain higher opening pressure.Doing that closing pressure decreases but it may be not of importante when injected faster (see Fig 24). One practical example of toward HPI application shows Fig 25 as the influence on engine characteristics.As for engine:ø120 x140 mm,6 cyl. in-line,water cooled,combustion chamber of ω type (see Fig 3 b,page in this report),production "Steyr".

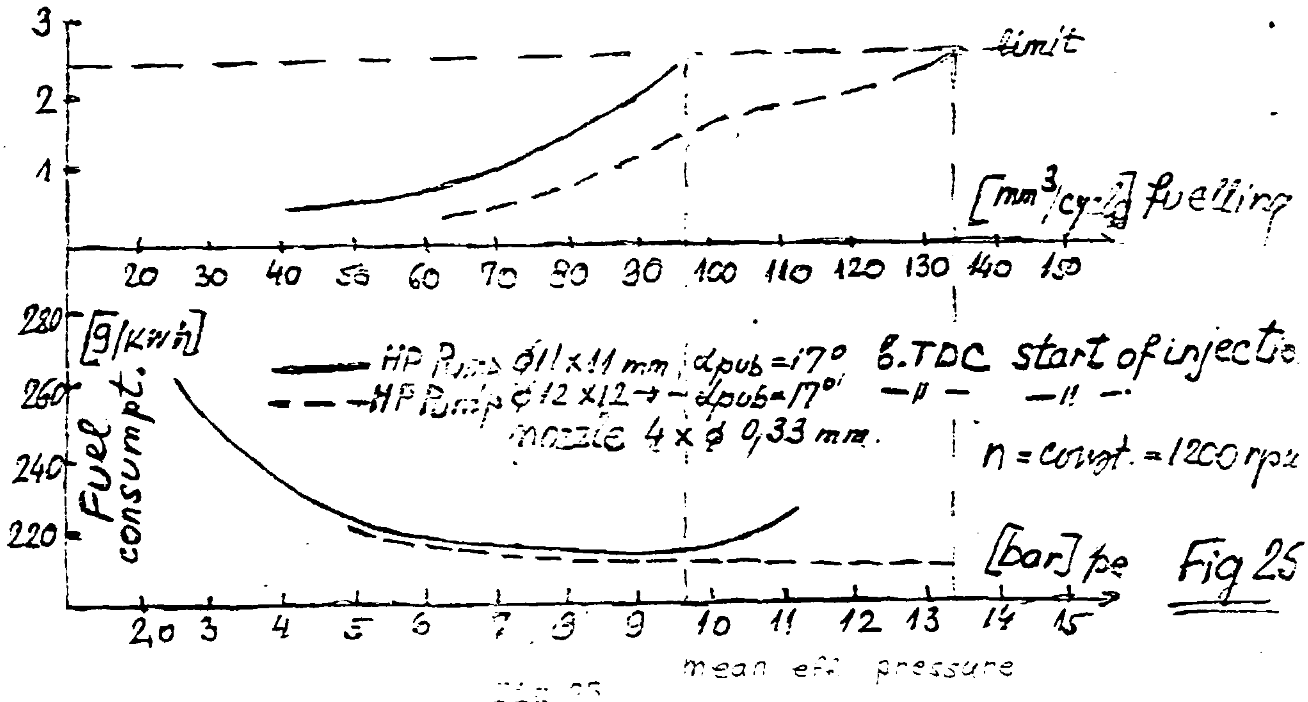


Fig 25

Fig 25

It may be seen that:
- increasing plunger dia.
- increasing plunger lift (higher pre-lift possible)

injection velocity increased and it had positive effect upon engine outputs.

However, after injection may be the problem, to avoid, snubber valve may be applied (see the 1[st] Report). To avoid after injection without snubber valve the next expression must be used:

$$A_k v_k \leq 5 \cdot a \cdot (\mu f) \cdot A_c^{-1}$$

where: A_k - cross-sectional area of plunger, v_k - plunger velocity at the end of delivery, a - velocity of sound (1320 m/s), μf - nozzle hole flow area, A_c - cross-sectional flow area of HP tube. Tendency to after injection increases when (see the above expression):

- plunger dia. increases ($\pi d_k^2/4=A_k$)
- n /rpm/ increases
- pre-lift increases
- more steeper cam shape

} V_k increases

- μf /mm^2/ decreases
- A_c /mm^2/ increases (or dead volumes of HP system)

Appendix II 3 related to subject 5

During discussion related to subject 5 more informations were desired but concering deeper cylindrical piston bowl as well as fuel arrangement.

Fig 1 shows hyperboloid combustion chamber (HPA) with three possibilities related to single hole directions of nozzle.

In-cylinder pressure diagrams shown on the right in Fig 1 correspond to spray arrangement 1,2 and 3.Spray marked with number 3 offers more multi-fuel tendency.

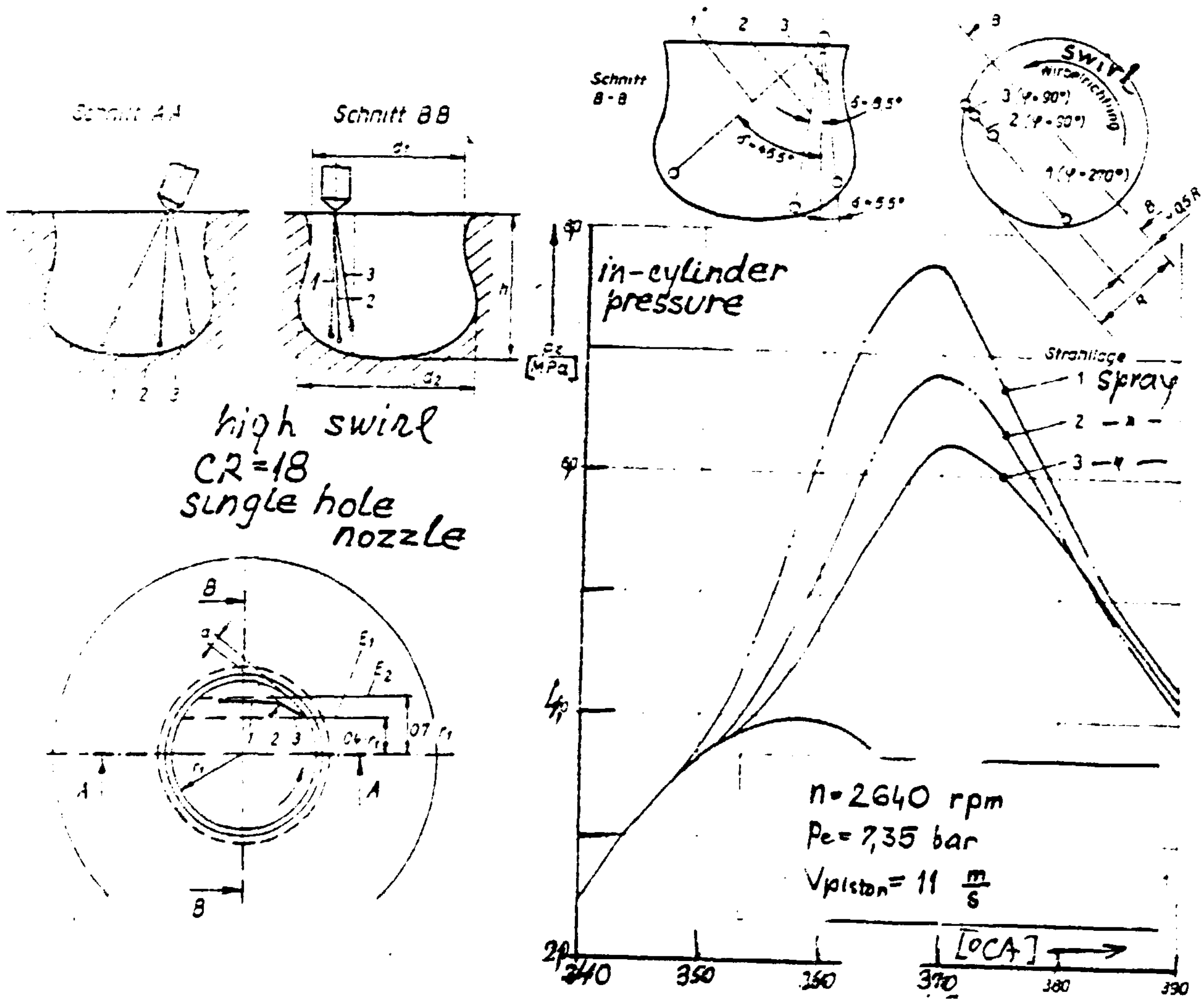


Fig 1

However,in-cylinder pressure diagram when gasoline,depicts characteristics figure of late combustion,see Fig 2,on the right. Is to be mentioned that missfiring was not experienced but late combustion produced a bad fuel consumption figure.Thus this combustion chamber with nozzle arrangement 3 (Fig 1) also asks for spark plug to control combustion.Otherwise,hyperboloid combusion chamber as well as Elsbett are suitable for kerosine combustion also, without any modifications.

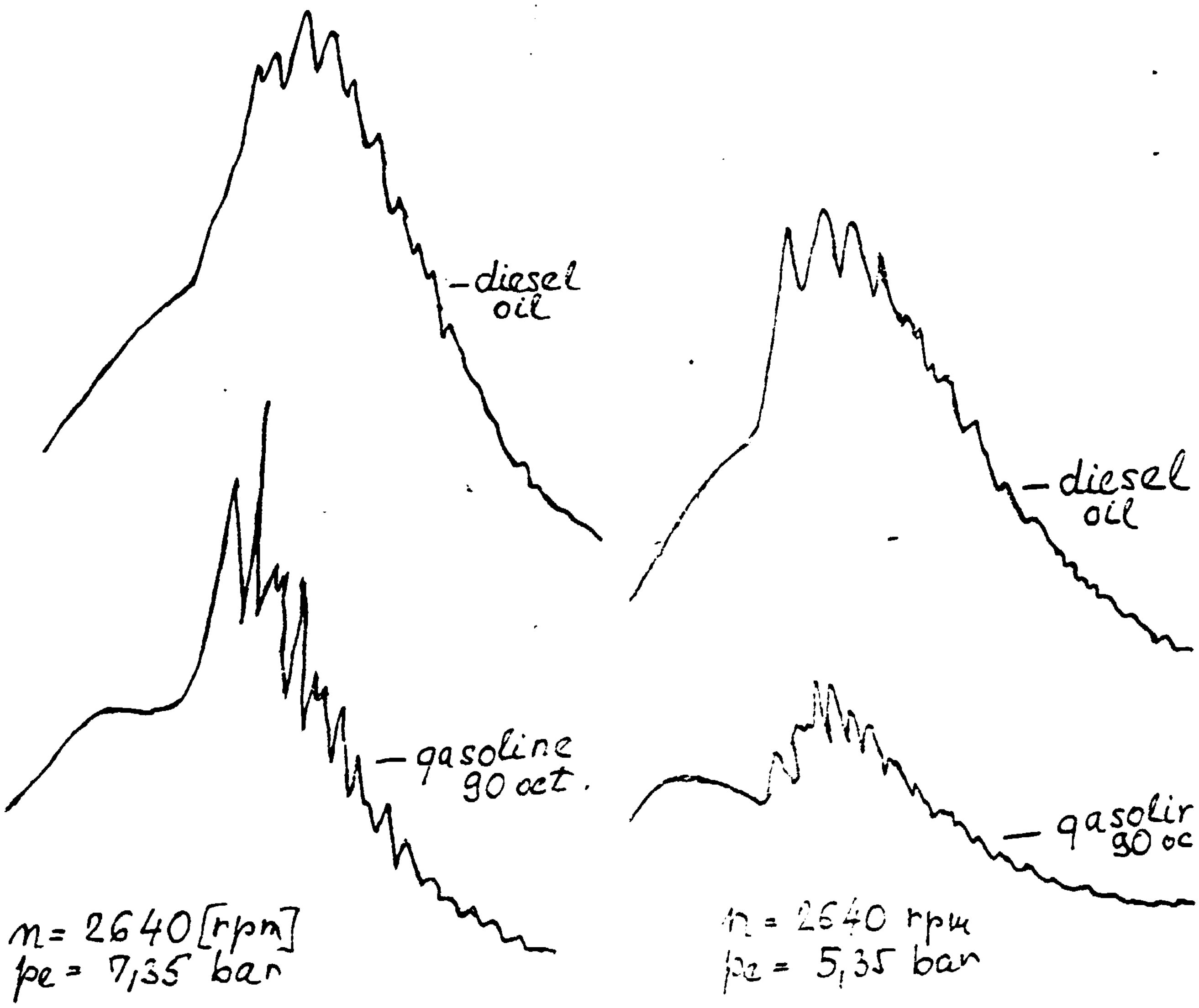


Fig 2 In-cylinder pressure diagrams

As was yet said, spark plug is anavoidable not only because of late combustion in low load operation with gasoline or for better starting. Morever, spark plug must control combustion when fuel with low cetane number. It's known that heat release rate in amount of 120 kJ/kg°CA results in knocking. Using gasoline 95 and hyperboloid combustion chamber (arrangement 3, Fig 1) without spark plug, max. heat release rate approaches 270 kJ/kg°CA what is by factor 3,5 higher than in the case of diesel oil combustion. Fig 3 shows peak in-cylinder pressure, pressure rise $dp_z/d\alpha$ and fuel consumption v.s. mean effective pressure at rate speed. The both, gasoline and diesel oil were tested in hyperboloid combustion chamber (arrangement 3, Fig 1) without spark plug. It may be seen that,with gasoline peak in-cylinder pressure increases at higher loads beacuse of knocking.At low load operation,fuel consumption increase because of late combustion,when gasoline used as fuel.Knocking tendency may be seen from $dp_z/d\alpha$ traces.

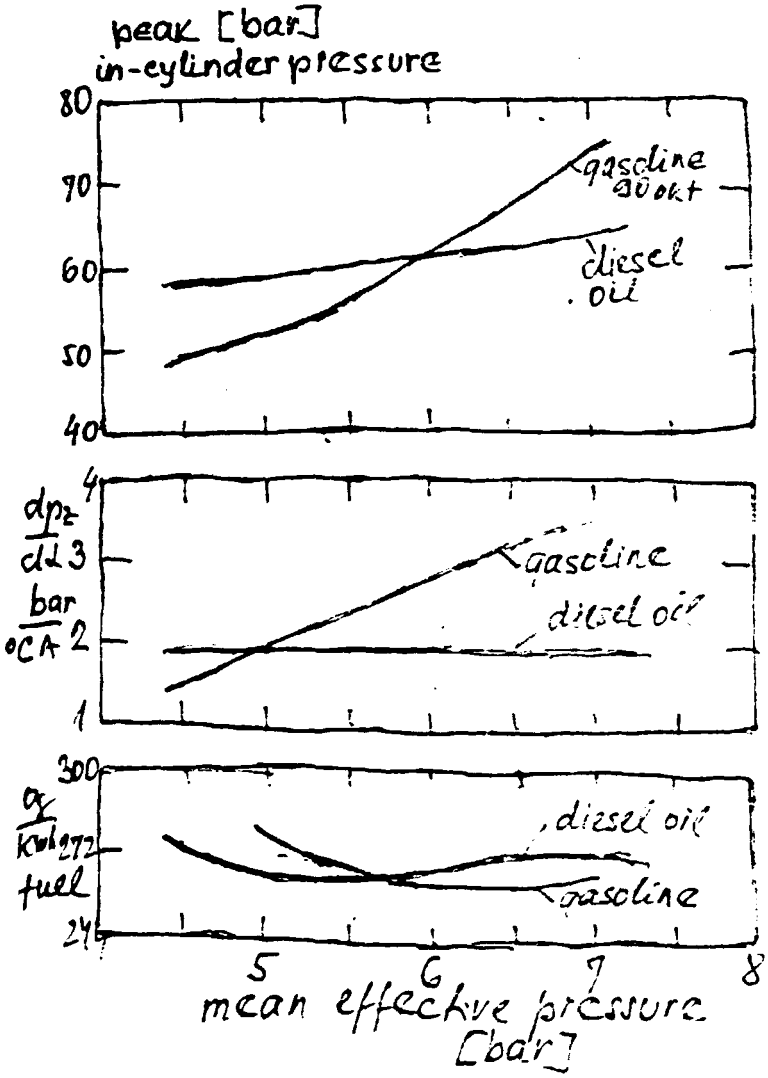
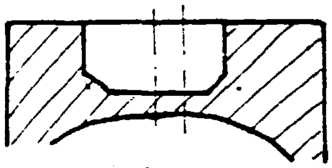
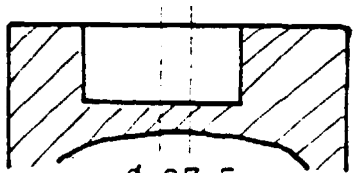


Fig 3

Still one evidence that without spark plug,fuels with low cetane number may not be used in diesel engine operation.

Appendix X 4 Miscellaneous subjects

1. Mixed diesel oil-CH_3OH was used in TATA engine (see Topic 2). However, data collected showed that fuel consumption demonstrated a high rate with diesel fuel also. It was the reason to compare existed TATA engine with one of new DB engines series OM 366.

	TATA engine	DB,OM 366
engine type	6 cyl., 4 stroke in line, water cooled, vehicular application, NA	the same
Type of comb. chamber (nearly the same)		
Piston dia. mm	Ø 92	Ø 97,5
Stroke mm	120	133
CR	17	17
Rated power /kW/	82	100
Rated speed /rpm/	2800	2800
Swept volume /cm^3/	4786	5958
Specific rated power output /$\frac{kW}{dm^3}$/	17,13	16,78
Mean piston velocity at rated power/m/s/	11,2	12,4
Max. torque /Nm/	?	402
Speed at max. torque /rpm/	1800	1400
Engine weight /kg/	?	445
Specific weight /kg/kW/	?	4,5
Number of piston rings	5	3
Inlet duct	?	Resonant matched
Injector number of nozzle holes	4	4

Injection pump (BOSCH)	A		A
Min. fuel consumption rate /g/kWh/	265	$\xleftrightarrow{+25\%}$	212
Fuel consumption rate at rated power/g/kWh/	~ 294	$\xleftrightarrow{+25\%}$	235
Comparison between MA-TATA and OM366LA version at rated power /g/kWh/	294	$\xleftrightarrow{+33,6\%}$	220
at $M_{max.}$ position	265	$\xleftrightarrow{+33,8\%}$	198

OM366LA is TC-IC version of OM366 NA version.

As was yet suggested in the 1st Report, before CH_3OH application fuel consumption must be reduced with diesel fuel at first.Here is a many reasons to do that.Per example capacity limit of injection pump,etc.

Without to change to much fuel consumption figure may be better and at the same time engine less expensive.

It's known that about 75% of neat friction loss belongs to piston/piston rings - cylinder sliding.Simple calculation may show,that replacing existed piston ring set with only 3 piston rings (1st keystone, 2nd tapered , 3nd spring supported elastic oil ring) neat friction loss may be decreased for obout 12% at rated power and so does fuel consumption rate also.Wear and oil consumption not to mention.

Thus with very small change:

- inlet duct matched
- cylindrical combustion chamber
- injector and pump better matched
- 3 piston rings and shorter piston skirt

fuel consumption in diesel operation may be reduced for about 20%.

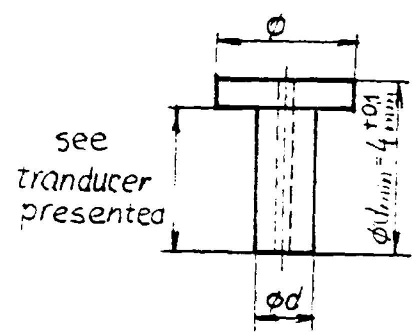
Reached a good fuel consumption figure with diesel fuel at first it's reason to apply methanol as replacing fuel,but with spark plug ignition control.

(For DB-OM366 engine see ATZ 4(April)1984.,Arthur Mischke and Dietrich Koppenhöfer.May 1984 we tested OM364 (4 cylinder) and found data given were correct).

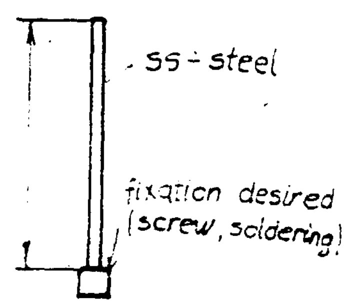
2. Proforma invoice was desired for our inductive tranducers and bridge-needle lift measurements.

To do that need:

- official letter
- dimensions of tranducer body



- dimensions of tranducer needle



3. For CH_3OH application as well as for ... system testings - F&M P7 pump may be used. When asked, we may contact F&M and purchased for JJP P7 pump gratis for the time of six months.

Offical letter and all data of existing Bosch pump are needed. When Bosch original type indication anyother date are unecessary.

4. As was explained during commons experiments, IAIA engine has not timer at all (when correct informed). Reaching 2800 rpm any diesel engine must have time advancer for the better fuel consumption figure, better torque back-up and lower mech. loading.

We apply timing advancer when

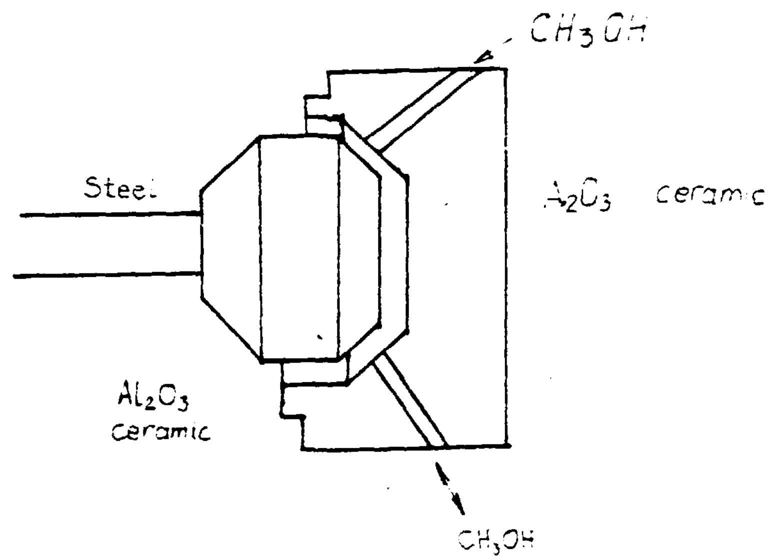
$$n_{rated} \geqslant 2400 \text{ rpm}$$

5. In some experiments with methanol we decreased the flow of cooling water and improved low-load operation.

6. As was yet mentioned calculation tehnique is to be recommended for matchings. Many programmes are today on desposal. For the optimisation matchings, calculation may not be avoided. Thus, we use for matching the calculation at first accompanied

by experiments after. If any need some of the programs may be sent to IIP as well as expert for practical application.

7. Fuel distributor suggested in Fig 6, page 44 as well as it connection in Appendix X 2, Fig/page 118 may be made from two different materials. Namely, where methanol ceramic may be used.



Is to be noticed that leakage of methanol in the pump may produces:

- a high wear at pump bearing
- a high wear at cam and follower
- very fast deterioration of lubricant when integrated oil circut

Service life of plunger, barrel, retraction valve, nozzle body and needle cold'nt be a long one when neat methanol used. Therefore in any way, some lubricant to methanol must be added. Is to be mentione, the higher pressure in the pump the better lube property of the liquid. Sulphorous components exhibit a good lube property when stable mixed without separation.

APPENDIX A-84

PREPARED FOR IIP

"Estimation of Wiebe's Heat-Release Parameter"

(Bulaty - Glanzmann, MTZ 86 (1984) 7/8)

Elaborated by:

Černej Anton
Filipović Ivan
Gebert Krešimir

Maribor, september 1984.

Based on kinetic, Wiebe law of combustion enables, when proper matched, the accomodation of law of heat release to the various conditions in engine operation. Thus, Woschni showed the re-calculation of Wiebe parameters ($m, \Delta\varphi_v$) from one to another conditions.

$$\frac{dQ_B}{d\varphi} = \frac{Q_B}{\Delta\varphi_v} a\,(m+1)\,Y^m e^{-aY^{m+1}} \qquad \dots (1)$$

$$Y = \frac{\varphi - \varphi_{vA}}{\Delta\varphi_v}$$

where:

Q_B - integrated input heat
$\Delta\varphi_v$ - period of combustion
a - const (for 99,9 % of combustion a=6,905)
m - shape parameter
Y - relative instanteneous period of combustion
φ_{vA} - beginning of combustion

However, problem appears to define Wiebe-equivalent law of combustion when usual test bench results on desposal without in-cylinder pressure diagram. We used "trial and error" method as time consuming and unaccurate one. Here is one basic method described to estimate Wiebe's heat-release parameters by means of systematic cycle calculations (Thomas Bulaty and Walter Glanzmann).

Basic methode

Experimental data:
- b_e - fuel consumption
- $p_{z,max}$ - peak in-cylinder pressure
- p_{me} - mean eff. pressure

have to be reproduced by systematic cycle calculation as accurate as possible.

In some examples by cycle calculations low pressure events (LW - gas exchange process) remain unchanged. Thus, high pressure part of working cycle (HD) may be independently calculated supposing the connection points of the both parts (HD and LW) known (Fig 1). The connection points may be approximately defined as shown in Fig 1, point ac and ex. The end points of HD part of cycle in BTC (UTP) are connected by line of isentropic change of state with ac and ex.

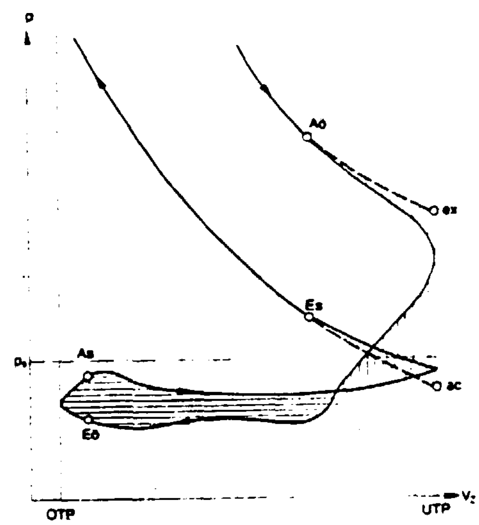


Fig 1 Connection of gas exchange process to high pressure part of working cycle of four stroke engine

p_s - scavenging pressure
AÖ - exhaust opens
As - exhaust closes
EÖ - intake opens
Es - intake closes
ex - end of expansion
ac - start of compression
Solid line: real pressure curve
broken line: isentropic change of state

State in point ac depends on:
- state in scavenging receiver (p_s, T_s)
- typ of engine
- valve timing

When no LW calculation performed p_{ac} and T_{ac} may be estimated approximately as follows:

$$\left.\begin{aligned} p_{ac} &= (0{,}96 \div 0{,}985)\ p_s \\ T_{ac} &= 0{,}833 \cdot T_s + 86 \end{aligned}\right\} \qquad \ldots \quad (2)$$

Thus, for calculation of HD part of working cycle we have on desposal:
- starting point or ac introduced by fuel
- upper limit of calculation given by peak in-cylinder pressure.

The area of in-cylinder pressure diagram of the working part of cycle(HD, Fig 1) without gas exchange process (p_{iOLW}) may be estimated by means of:

- mean eff. pressure (measured) p_{me}
- supposed friction of engine p_{mr}
- gas exchange δ_{piLW} acc. to Eq 3.

$$p_{mi} = p_{iOLW} + \delta_{piLW} = p_{me} + p_{mr} \qquad \ldots \quad (3)$$

Now, with one reasonable value of start of combustion chosen (φ_{vA}) (per example φ_{vA}= 714 ^{o}CA with midlle speed four stroke engine)systematic calculations may be performed, acc. to Eq 1. For three periods of combustion chosen ($\Delta\varphi_{v\,A,B,C}$) at φ_{vA}=const three shape parameters ($m_{A,B,C}$) are supposed. Thus, we have nine combinations on desposal and for every case $m, \Delta\varphi_v$ at φ_{vA}=const. HD diagram has to be calculated. The selection of shape parameter is explained in Fig 2a. Cross-sections A,B,C of each peak in-cylinder pressure curve and line $p_{z,max}$ soll given, correspond to the shape parameters $m_{A,B,C}$.

At the same time the mean pressures p_{iOLW} vs. shape parameters are presented in Fig 2b. Again, the points A,B,C are defined by means of shape parameters in Fig 2a. The line A-B-C in Fig 2b satisfies the condition. $p_{z,max}$ soll but only the point R satisfies the second condition also p_{iOLW}soll (given). Point R defines the shape parameter m_R demanded. Using Fig 2c the second Wiebe parameter $\Delta\varphi_{vR}$ may be defined.

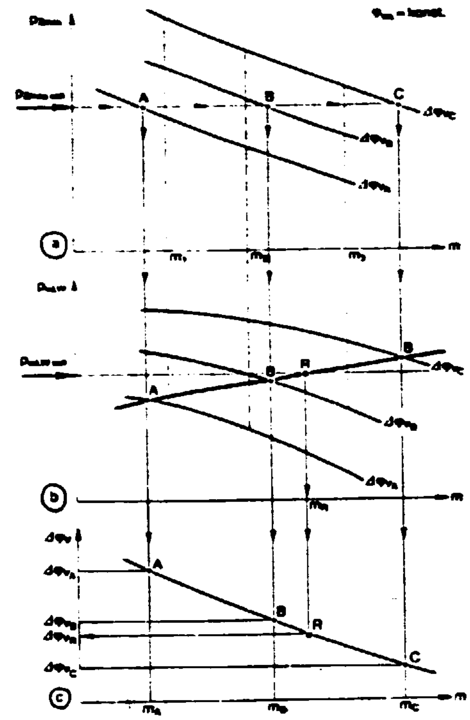


Fig 2 Graphical representation of basic method

Thus we have:

$$\varphi_{VA},\ m_R \text{ and } \Delta\varphi_{VR}$$

under condition given:

- fuel consumption
- peak in-cylinder pressure ($p_{z,max}$)
- mean pressure (p_{iOLW} soll)
- start, point <u>ac</u> (Fig 1).

The corresponding combinations of m, $\Delta\varphi_V$ (see Fig 3 and Table 1) is possible to define for almost every start of combustion and typ of engine given.

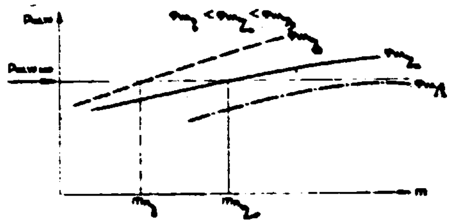


Fig 3 Dependence of m.e.p. on shape parameter at different timings of heat release

Table 1 Results of estimation of the heat-release parameters for different heat-release timing and parameter conversion along propeller operation line

P_e	n	p	T	φ	$\Delta\varphi$	m	p	p_{Zmax}	p	T	Q	λ
kW	min⁻¹	bar	K	°KW	°KW	–	bar	bar	bar	K	kJ/AS	–
727.6 100 %	430	3.385	340.3	710.95	82.11	1.108	24.612	126.927	11.557	1122.3	372.4	2.049
				714.34	80.60	0.901	24.628	126.880	11.557	1122.3	372.4	2.049
				717.02	80.11	0.738	24.630	126.846	11.559	1122.6	372.4	2.049
582.1 80 %	399.2	2.753	338.5	710.99	80.58	0.890	20.990	115.091	9.402	1115.6	314.2	1.987
				714.34	79.18	0.726	20.977	113.886	9.421	1117.8	314.2	1.987
				716.99	78.61	0.594	20.944	112.762	9.438	1119.9	314.2	1.987
438.6 60 %	362.7	2.102	337.1	711.19	79.87	0.660	17.168	99.313	7.321	112.99	257.2	1.863
				714.47	78.45	0.541	17.142	97.184	7.348	1134.0	257.2	1.863
				717.08	77.89	0.444	17.114	95.355	7.378	1138.7	257.2	1.863
291 40 %	316.8	1500	335.7	711.70	77.93	0.449	13.048	79.910	5.315	1141.4	198.0	1.733
				714.84	76.56	0.370	13.066	77.592	5.351	1149.2	198.0	1.733
				717.38	76.03	0.305	13.072	75.569	5.391	1157.7	198.0	1.733
145.5 20 %	251.5	1.144	335.1	711.50	62.41	0.356	8.358	62.719	3.424	969.0	126.4	2.07
				714.56	61.33	0.294	8.383	60.486	3.447	975.4	126.4	2.07
				717.07	60.89	0.243	8.431	58.728	3.482	985.5	126.4	2.07

Concerning that gravity centre of combustion of every combination remains nearly unaffected, the shape parameter m increases with earlier start of combustion, see Fig.4

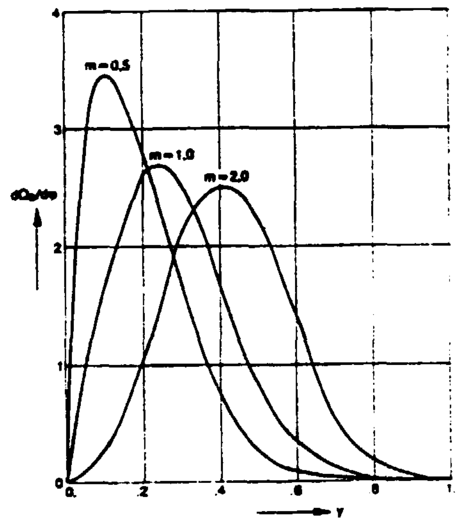


Fig 4 Different heat-release patterns depending on shape parameter m (see Eq 1)

When start of combustion to early (per example φ_{VA1} - Fig 3) desired mean eff. pressure can not be reached although extreme high value of m may be applied.