OCCASION

This publication has been made available to the public on the occasion of the 50th anniversary of the United Nations Industrial Development Organisation.

DISCLAIMER

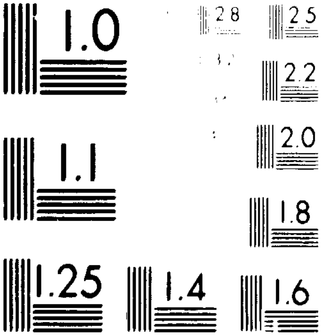
FAIR USE POLICY

CONTACT

Please contact publications@unido.org for further information concerning UNIDO publications.

For more information about UNIDO, please visit us at www.unido.org

UNITED NATIONS INDUSTRIAL DEVELOPMENT ORGANIZATION
Vienna International Centre, P.O. Box 300, 1400 Vienna, Austria
Tel: (+43-1) 26026-0 · www.unido.org · unido@unido.org


1.0
2.8
2.5
2.2
2.0
1.1
1.8
1.25
1.4
1.6

RESTRICTED

12699

DP/ID/SER.B/410
6 July 1983
English

China,

TECHNICAL ASSISTANCE TO INCREASE THE RELIABILITY AND ECONOMY OF LOCOMOTIVE DIESEL ENGINES

SI/CPR/82/801

PEOPLE'S REPUBLIC OF CHINA

Terminal report*

Prepared for the Government of the People's Republic of China
by the United Nations Industrial Development Organization,
acting as executing agency for the United Nations Development Programme

Based on the work of the AVL-Gesellschaft fuer
Verbrennungskraftmaschinen und Messtechnik mbH.,Prof.Dr.hc.Hans List
under UNIDO Subcontract No. 82/66 to AVL

United Nations Industrial Development Organisation
Vienna

* This document has been reproduced without formal editing.

V.83-58024

RESTRICTED

12699

DP/ID/SER.B/410
6 July 1983
English

China. TECHNICAL ASSISTANCE TO INCREASE THE RELIABILITY AND ECONOMY OF LOCOMOTIVE DIESEL ENGINES

SI/CPR/82/801

PEOPLE'S REPUBLIC OF CHINA

Terminal report*

Prepared for the Government of the People's Republic of China
by the *United Nations Industrial Development Organization*,
acting as executing agency for the United Nations Development Programme

Based on the work of the AVL-Gesellschaft fuer Verbrennungskraftmaschinen und Messtechnik mbH.,Prof.Dr.hc.Hans List under UNIDO Subcontract No. 82/66 to AVL

United Nations Industrial Development Organisation
Vienna

---

* This document has been reproduced without formal editing.

V.83-58024

TABLE OF CONTENTS

1. INTRODUCTION

On February 4th and 9th resp., 1983 the UNIDO and AVL signed the contract No. 82/66 for the provision of technical assistance in connection with the improvement of reliability and fuel economy of a locomotive diesel engine manufactured by the

S I F A N G Factory
in the
People's Republic of China

March 30, 1983 the "Interim Report" covering the work of the AVL expert in the project area was mailed to UNIDO.

Based on the drawings received from Sifang and the additional information from the project area AVL's design, calculation, thermodynamic and development department have performed this analysis work.

In this final report all the findings in connection with the aims of the project and the contract are described in detail. The attached drawings show the modifications proposed by AVL to avoid cavitation on cylinder liner and to improve engine fuel economy.

AVL will present the modifications and discuss them with the Chinese delegation during their stay with AVL in the 26th and 27th week of 1983.

2. CAVITATION PROBLEMS ON SIFANG PRECHAMBER ENGINE 12 V 180 ZL

2.1 Thermodynamic recalculation

Part load data at an engine speed of 1500 PRM and part load data corresponding to a propeller characteristic of the locomotive engine were provided by SIFANG for a critical examination. High pressure diagrams measured at rated power with a Farnboro indicator are considered to be incorrect. Consequently, they were not evaluated in detail.

At the rated power of 993 KW at 1500 RPM the measured BSFC of 233 g/KWh is comparable with the specific fuel consumptions of prechamber engines of similar size. At part load, 1500 RPM, BSFC is, on the contrary, considerably higher than compared to the reference engines shown in enclosure 1.

If the engine is operated corresponding to a propeller characteristic BFSC also increases more than usually with decreasing engine speed. Moreover, the smoke level increases from 1.1 Bosch at 1500 RPM to a peak of 3.6 Bosch at 1200 RPM. At this speed the specific fuel consumption is 20 to 24 g/KWh higher than it may usually be expected.

Propeller Characteristic-Data as Measured

RPM	1500	1200	1000
BHP KW	933	508	294
BMEP bar	12.7	8.1	5.6
BSFC g/KWh	233	253	274
Smoke Level (Bosch)	1.1	3.6	0.1

The ample specific air consumption of 8 kg/KWh at 1200 RPM already indicates a sufficient amount of air available in the cylinders for combustion. Accordingly, an excess air ratio of 1.86 (equivalent to an air-fuel ratio of 27:1) based on the amount of air trapped in the cylinder has been calculated.

Such a high excess air ratio should be sufficient for a clean combustion. The excess air ratio calculated at the rated power of 930 KW for comparison amounts to 2.1 due to the lower fuel consumption.

Hence, it may be concluded that both the high BSFC and the high smoke level, are not caused by a lack of combustion air, but are caused by an inefficient combustion which results from prechamber dimensions not suitable for the conditions at part load. Improved prechamber dimensions are proposed in section 2.3

Diagrams of the pressures in the inlet manifold and in the exhaust pipe measured at rated power are typical for a three-pulse system with three cylinders of 240 degr. CA firing distance grouped to a common exhaust pipe. A positive pressure drop between the inlet manifold and the exhaust pipe exists during the entire overlap period of 103 degr. CA. From these low pressure diagrams we draw the conclusion that no disturbances of the scavenging process occur at lower engine speeds.

The overlap is unsymmetrical related to TDC:

Inlet opens	60 degr.CA BTDC
Exhaust closes	43 degr.CA ATDC

AVL prefers a more symmetrical overlap which allows a smaller piston to cylinder head clearance and hence suggests the valve timing:

Exhaust opens	60 degr.CA BBDC	Inlet opens	50 degr.CA BTDC
Exhaust closes	53 degr.CA ATDC	Inlet closes	42 degr.CA ABDC

with exhaust opening and inlet closing of the current valve timing maintained. Although, the exhaust valve is closed later by 10 degr.CA., back flow is not expected from the exhaust into the cylinder. Even if the blow-down pressure pulse of the cylinder subsequent in the firing order would interfere with the scavenging, the amount of air-gas mixture flowing back into the cylinder were to be neglected.

Turbocharger data are not available. Therefore, only a overall efficiency of the turbocharger can be calculated from the power balance of the compressor and of the turbine using the mean pressure and the mean temperature measured before the turbine. At rated power 933 KW an overall efficiency of 56.5 % has been calculated, which corresponds to efficiencies of comparable turbochargers today on the market.

2.2 Comments on the ports of the engine Sifang 180 x 205

Intake ports

- Because of the very low height of the valve guide boss and because of the abrupt change from the straight inlet duct to the guide boss area no efficient guiding of the flow can be achieved in this area; therefore, a flow separation and consequently a loss in flow efficiency is the result.
- Further restrictions arise from the indentation of the ports opposite the nozzle. The branches of the port are not separated enough in the guide boss area, therefore, the flow cannot be guided. AVL designs neutral ports with a decrease in cross sectional area in flow direction in order to guarantee efficient flow conditions and reduce flow separation. For this reason AVL attempts to separate both branches as much as possible.
- The torus radius is too small, thus preventing optimum flow conditions.
- A redesign and development of the port is recommended.

Exhaust ports

- As the two branches already come together just after the valve heads there is a sudden enlargement, whereas in this specific area (of the torus radius of the port) a slight decrease in cross sectional areas in flow direction should prevent flow separation.
- The valve guide boss is very low and protrudes nearly vertical into the port. Therefore, the cross sectional areas are not only too small in the boss area compared to the runner part but probably also have an unfavorable step in the development of areas and of course no optimum flow. Therefore, flow separations have to be encountered in the upper valve lift range in this part of the port. This causes high turbulences almost resulting in a break-down of

the flow in the upper area of the valve guide boss.

- The flange area is far too big in relation to the valve.
- A redesign and development of the port is recommended.

Diagrams of flow parameters are attached as enclosures 2 and 3.

2.3 Fuel injection and combustion aspects of the Sifang IDI-engine

The specific fuel consumption of the Sifang IDI-engine at full load is normal compared with other IDI-engines of the same size, but it increases very steeply at part load. Enclosure 1 shows the specific fuel consumption curves of some large IDI-engines running at constant speed compared with the Sifang engine. A review of the operating data of the Sifang IDI-engine at part load shows no irregularities, indicating a possible improvement of the combustion.
The following modifications should be tested to improve the part load consumption:

- decrease the pre-combustion chamber volume in stages.
- increase the burner hole area in stages.

From the above work, optimum sizes for the combustion chamber and burner holes can be determined.
These modifications will enable the reduction of flow losses between the precombustion chamber and the main combustion chamber by which the specific fuel consumption at part load could be reduced. Further improvements in the fuel consumption may be obtained by optimisation of the injection system components including tunning of the shape of the pintle of the nozzle needle.

All the above suggestions for improving the fuel consumption, will make systematic engine test with a lot of small steps necessary.

2.4 Design Analysis on Cavitation Problems

The modifications, described as follows have been made to avoid cavitation damage on the engine block and the cylinder sleeves, meeting standards of quality, reliability and life time with the 12 V 180 type engine.

2.4.1 ENT 758/Sheet 01 - Installation - cylinder liner

One of the most important parameters for cavitation problems of a cylinder liner is the thickness of the liner and the free length between upper and lower fit of the liner in the motorblock of the engine.
The left hand side of the cross section through the cylinder shows the current production engine with a wall thickness of 9.5 MM, which is about 5.3 % of the bore diameter. Also the distance between upper and lower fit is 310 MM which is about 1.72 times the bore diameter.
Both these dimensions are insufficient due to cavitation. The right hand side of this cross section now shows AVL's proposal to eliminate these problems.
An increase in the wall thickness to 11.5 MM or 6.39 % of bore diameter is a maximum value because of the very tight position of the cylinder head bolts.
According to AVL's experience the wall thickness should be a little bit higher, but in connection with the very short distance between upper and lower fit of the liner (213 MM or 1.18 times the bore diameter) and the maximum peak firing pressure of about 110 bar it should result in a sufficient solution to avoid further cavitation problems.
Because of the increase in the wall thickness, it was necessary to increase also the diamter of the upper and lower fit of the liner.
More detailed information regarding dimensions and tolerances is given in this layout drawing.
In connection with the increased upper fit configuration it was necessary to enlarge the outer diameter of the flange to control the pressure level between flange and motorblock.

The lower fit has also been newly designed and a specification for the 3 O-rings is given on this drawing.
The position of the eight cylinder head bolts has been retained but the size of the thread of the bigger ones has been reduced from M 24 x 2 to M 22 x 2 to get sufficient land between the thread and the water jacket.

The design of the cylinder head gasket has been changed by a little, but with the current thickness of the production engine.
The material of the cylinder head gasket should be from iron with a hardness of about HB 80 - 95 with all over copper coated. The tightening torque for the cylinder head bolts had to be increased to the values shown in the layout, to get a sufficient ratio between the bolt loads and the gas load.
The sealing of the water and oil connections between cylinder head and motorblock has been changed from axial to radial sealing, nevertheless, using of the "soft" cylinder head gasket should also be done with this new design to support the cylinder head and to reduce deformations of the cylinder head bottom surface.

2.4.2 ENT 758/Sheet 02 - Motorblock IDI engine

Because of the modification of the lower fit of the cylinder liner it was necessary to redesign the motorblock.
According to the production equipment of Sifang-Factory the cylinder distance, the position of the camshaft and the axle to the rocker arms and also the total height of the motorblock should not be changed.
Furthermore, it was AVL's endeavour to make as few changes as possible.
Because of the higher position of the lower fit of the cylinder liner it was necessary to redesign the cooling water flow through the motorblock.
The entrance of the cooling water to the motorblock on the vibration damper side has been retained in size and dimension.
In the area between lower fit and the cranking room a water

jacket core is located to distribute the cooling water to each cylinder. The entrance to the water jacket around the cylinder liner is of a tangential type to insure a rotation of the cooling water around the liner.

To support this additonal core some core holes are located, which also allow a careful cleaning from sand.

The machining of the surfaces for the hand hole covers and the supports for oil cooler and -filter are not to be changed in any way.

Also the core inside the V-room has only to be changed in the area of the cylinder head bolts to bring a better connection between these bolts and the wall.
The machining of the camshaft is not influenced by any modification. The bosses for the cylinder head bolts are changed to achieve a better loaded transfer to the bearing wall and to minimize the deformations of the cylinder liner. The seat angle of the cylinder bolt ends has been changed from 180 degr. to 120 degr. for machining purpose.

To relieve the room around the flange of the cylinder liner a drilled hole is provided on the outside of each cylinder. By means of these holes a control of all water and oil connections is possible.

2.4.3 ENT 758/Sheet 03 - Layout piston - IDI

This layout describes a piston modification for the current production engine.
There are two reasons for redesigning the piston:

First, the elimination of the 5th ring groove on the lower end of the piston and second, temperature problems in the first ring groove.
The elimination of the lower oil ring decreases cavitation problems, to assist this measure, the lower length of the piston has been enlarged for 15 MM.
This will not have any restriction for the construction of the power train.
A cast-in ring carrier has been added to control the high temperature in the first ring groove.
All other outside dimensions are equal to the piston of the production engine.

2.4.4 Summary

With the modifications described above it should be possible to avoid further cavitation problems with a minimum of changes on the production engine.

Parts to be changed:

a) Motorblock
Modification with the water jacket core and also some changes in the machining.

b) Cylinder liner
Newly designed part, but it should be possible to machine from the existing casting.

c) Cylinder head gasket ("hard type")
Newly designed parts.

d) Cylinder head gasket ("soft type")
Same part as production engine, but with increased hole diameter for water and oil connections.
Remachining of existing parts is possible.

e) Cylinder head
Remachining of countersunk for oil and water connections.

f) Piston
Has to be replaced by a newly designed one.

g) Intake and exhaust ports
As described in Section 2.2

h) Camshaft
As described in Section 2.1

i) Injection equipment
As described in Section 2.3

3. FUEL REDUCTION BY CONVERSION TO DIRECT INJECTION SYSTEM

3.1. Thermodynamic Calculation

Preliminary cycle parameters have been calculated for a 12 cylinder DI-engine converted from the current prechamber engine. The rated power of the DI-engine is assumed to be the same as with the prechamber engine 933 KW at 1500 RPM.

Additional assumptions are:

- Ambient conditions	1.015 bar	20 degr.C
- Compression ratio	13.75 : 1	
- Maximum cylinder pressure	110 bar	
- Inlet manifold air temperature (after intercooler)	65 degr.C	
- Exhaust and inlet valve lifts remains unchanged but valve timing is as suggested in section 2.1		
- Turbocharger overall efficiency	56 %	
- Compressor efficiency (isentropic)	78 %	
- Frictional losses FMEP	2 bar	

Results of the calculations are listed in enclosure 4. Tota volumentric efficiencies are defined by the amount of the air-mass flow of the engine related to the engine displacement and inlet manifold conditions. Accordingly, trapped volumentric efficiency is the mass of fresh air retained in the cylinder (after closing of the inlet valves) related to displacement and inlet manifold conditions. Excess air ratio of combustions corresponds to the amount of fresh air trapped in the cylinder.

3.2 Fuel Injection and Combustion of the Sifang DI-Engine

3.2.1 Injection System

For the beginning of the tests with the DI-engine AVL proposes the following injection equipment.

3.2.1.1 Injection pump

Two 6-cylinder in-line pumps or, if possible, one 12-cylinder in-line pumpe (i.e. Bosch PE 12 ZW 140...)
With one 12-cylinder pump fewer problems in balancing then with two pumps on one engine would occur. The maximum injection pressure allowable for the pumps should be 800 to 1000 bar. The cam form should have a constant injection velocity during the injection period, if possible. A plunger diameter of 14 MM is recommended. A delivery valve of the constant pressure type with a residual pressure of about 70 bar would be best. If not available, a delivery valve with about 150 MM3 unloading volume in connection with a snubber valve should be used. An injection timing device is not necessary.

3.2.1.2 High pressure pipe

The high pressure pipe should have an inner diameter of 2,5 MM.

3.2.1.3 Nozzle

Injection needle to seat ratio	7 : 4 MM
Nozzle opening pressure	250 bar
Nozzle	8 x 0,31 x 150 deg.

The above fuel injection system specification is only a proposal for the beginning of the optimisation work on the injection test rig and engine.

3.2.2 Combustion

Enclosures 5 show a proposal for the shape of the combustion bowl in the piston of the DI-engine. The average compression ratio should be $\varepsilon \sim 14$. With this ratio, starting the engine is possible at a cooling water temperature of above 0 degr.C. The brim of the combustion bowl is high enough to avoid fuel being injected on the surface of the liner, and no inlet and exhaust valve pockets are needed in the piston.

The right position for the nozzle, the best spray angle and the optimal number of the nozzle holes can only be found by engine test.

The air manifold pressure should be about 2,3 bar (absolute) for a BMEP of 12,7 bar. At this load a firing pressure of about 105 to 110 bar is expected. In order to get a low specific fuel consumption with the DI-engine it is necessary to reduce the performance of the cooling water pumpe. The flow rate should be reduced from 90 M3/h (as currently on the IDI-engine) to 45 - 50 M3/h.
The engine cooling water system pressure should also be reduced to 2 - 2,5 bar above atmospheric. The engine water outlet temperature should be kept to 80 degr. over the whole load and speed range of the engine.

After optimisation of the injection and combustion system, the specific fuel consumption at full load should be between 211 and 218 g/KWh (155 to 160 g/PSh).

3.3. Piston Pin Calculations and Connecting Rod Big End Bearing Analysis

3.3.1. Piston Pin Calculations:

3.3.1.1. Method of calculation:

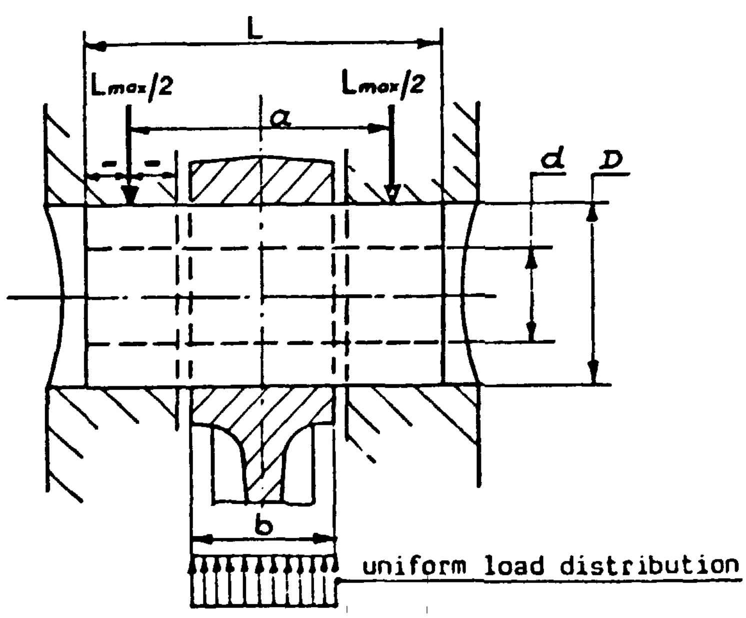
The evaluation of the piston pin assembly is based on calculated longitudinal and oval pin deflections as well as on maximum unit loads in the pin bearings.
The limitation of pin deflections is decisive for the avoidance of cracks in the piston bosses in particular due to an excessive edge loading condition if the longitudinal deflection becomes too high.

The maximum unit loads further must not exceed the load carrying capacity of the respective bearing materials.

3.3.1.1.1. Evaluation of actual pin deflections and unit loads:

The evaluation of the pin deflections is performed according to the calculation method published by M. KUHM in MTZ 25/2 and MTZ 25/6.

This method is based on the loading pattern shown in the sketch below where the load L_{max} is the maximum gas load on the piston.



Longitudinal deflection:

$$f = \gamma \cdot L_{max}\ a^3/48\ EJ$$

where:

$\gamma = 1 - \frac{b}{2a}$ .... a coefficient, which considers the uniform load distribution in the con. rod small end bearing

E .... modulus of elasticity of the pin material

$J = (\pi/64)\ (D^4 - d^4)$ ...... flexural moment of inertia of the pin cross section.

Oval deflection:

$$\delta = r^3\ L_{max}/12\ EJ_w$$

where:

$r = \frac{D+d}{4}$ .... mean radius of the pin cross section

$J_w = Lw^3/12$ .... flexural moment of inertia of the pin wall

$w = \frac{D-d}{2}$ .... wall thickness.

Unit loads in the pin bearings:

The maximum unit loads in the pin bearings are determined from the formula

$$P_{max} = \frac{L_{max}}{A_p}$$

where:

L_{max} .... maximum gas load on the piston

A_p .... projected bearing area in the piston bosses resp. the con.rod small end.

3.3.1.1.2. Evaluation of permissible deflections:

The recommendations available today for the permissible longitudinal deflection specify a linear dependency from the piston diameter according to the relation

$$f_p = f_o \cdot D_p/100$$

where:

D_p .... piston diameter in mm
f_o .... permissible specific deflection for 100 mm piston dia.

The permissible specific deflection depends on the flexibility of the piston bosses because this flexibility determines the ability of the bosses to follow the pin deflection in order to avoid an excessive edge loading. Stiff piston bosses therefore require smaller deflections than flexible ones.
For pistons of Diesel engines with,in general,stiff bosses,a permissible specific deflection of

$$f_o = 0.022 \text{ mm}$$

is recommended by the German piston manufacturer KARL SCHMIDT GMBH for heavy duty engines.

The permissible oval deflection is expressed by the relation:

$$\delta_p = \delta_o \frac{100 + 0.5\,(D_p - 100)}{100}$$

where:
δ_o = 0.027 mm
is the permissible specific oval deflection for 100 mm piston dia according to recommendations of KARL SCHMIDT GMBH.

3.3.1.2. Calculation results:

The evaluation of the piston pin assembly has been based on a cylinder peak pressure of

$$P_{max} = 125 \text{ bar}$$

which peak pressure is predicted for the direct injection combustion system operating at a BMEP of 12.7 bar at 1500 rpm (1350 PS).

Calculation results for the current piston pin assembly are compiled in enclosure 6 under the title "Current design version". The decisive dimensions of this current design are taken from the following drawings:

Piston and con. rod assembly	SFF9-05-00-000
Piston	SFF9-05-00-001
Con. rod	SFF9-05-01-000

For the reason, that these only available drawings do not specify all required dimensions, some of them had to be measured from the drawings and may be slightly incorrect therefore. Such possible inaccuracies, however, will not considerably affect the calculation results.

For a discussion of calculation results obtained for the current piston pin assembly enclosure 6 also includes the permissible pin deflections as well as reference data for the unit loads in the piston pin bearings.

As per enclosure 6, the current piston pin assembly is inadequate for a 125 bar cylinder peak pressure in particular for the too high longitudinal deflection of 46.8 μm which is 18% above the permissible value. Also the unit load in the small end bearing is on a high level and will be acceptable only for a bearing material with a high load carrying capacity, e.g. steel backed cast copper-lead-tin alloys similar to SAE specification No. 792.

Note: For the currently specified small end bearing material Q Al 9-4 no data for the permissible unit loads are available.

The preferable solution for an effective reduction of the longitudinal deflection would be an increase of the piston pin outer diameter to

$$D_o = 64 - 65 \text{ mm}.$$

It has been ascertained however by con.rod small end stress calculations, that this increased piston pin dia will also require a modified connecting rod forging with an increased outer dia of the small end.

With the current con. rod forging, the maximum piston pin O.D. that can be accommodated is

$$D_o = 62 \text{ mm}.$$

The according reduction of the small end wall thickness by 1 mm already yields small end stresses which are on the permissible limit.

The 62 mm piston pin O.D. alone, however, is not sufficient to reduce the longitudinal deflection to the permissible level and an additional reduction of pin length to 152 mm has been considered therefore to achieve this goal.

As per the calculation results for this combination, which are listed in enclosure 6 under the title "Proposed design version" acceptable pin deflections and unit loads are achieved with this design.

For the small end unit loads this conclusion, however, only applies for the case, that the currently specified bearing material is adequate for the 85.5 N/mm^2 unit load or that another proper material is used.

It is emphasized, however, that this proposed solution is a compromise for retaining the current connecting rod forging and does not include any safety margin for a further uprating of the engine.

3.3.2. Connecting Rod Big End Bearing Analysis:

3.3.2.1. Method of calculation:

For a discussion of crankshaft bearing reliability max. bearing unit loads as well as minimum oilfilm thicknesses are taken into consideration. While max. unit loads must remain below the load rating of the applied bearing alloy, minimum oilfilm thicknesses must exceed a certain level which according to experience is adequate to avoid excessive wear.

The computer program, applied for bearing load and oilfilm thickness computations is based on the calculation methods described below.

3.3.2.1.1. Bearing loads:

For bearing load computations, gas loads and rotating as well as reciprocating inertia forces are taken into account, the latter up to the second order.
For the main bearings, bearing load components are further computed for a statically determinate support of the individual crankthrows and superimposed for the respective main bearing.

Beside the bearing loads, this part of the program also computes the angular position of the load vector relative to the shell as well as relative to the pin, which angles are, among others, required for oilfilm thickness computations.
With the maximum bearing loads obtained from these computations the maximum unit loads become

$$P_{max} = \frac{L_{max}}{W\,D}$$

where:

L_{max} .... max. bearing load
W .... effective bearing width
D .... bearing diameter.

3.3.2.1.2. Oilfilm thicknesses:

The second part of the computer program, applied for oilfilm thickness computations is established on the basis of the Holland-method as published in VDI-Forschungsheft No. 475: "Beitrag zur Erfassung der Schmierverhältnisse in Verbrennungskraftmaschinen" resp. on the publication by A.Eberhard and O.Lang: "Zur Berechnung der Gleitlager im Verbrennungsmotor mittels elelektronischem Digitalrechner", MTZ 22/7.

This calculation method considers the load carrying capacity of the bearing due to the rotational movement and, in case of increasing eccentricity,also due to the squeeze effect.

From the equilibrium condition between these two components, which are expressed by the according Sommerfeld numbers, and the external bearing load, differential equations for the relative eccentricity and the angular position of the minimum oilgap are derived, which are solved by numerical methods.

On account of the fact that this calculation method is based on certain ideal conditions (no misalignment, perfect cylindrical bore and journal, constant oil viscosity etc.) the computations do not yield absolute values for the oilfilm thicknesses but only comparative figures, which have to be discussed on the basis of analogously evaluated values of approved and also failing bearings.

3.3.2.2. Calculation results:

The connecting rod big end bearing analysis has been carried out for the following conditions:

- Speed range:
  800 - 1500 rpm

- Operating conditions:
  Propeller curve corresponding to the rated output of 1350 PS at 1500 rpm. For the evaluation of gas loads at the individual engine speeds, indicator diagrams of a similar D.I. engine have been taken as a basis. The combustion peak pressure of these indicator diagrams is 125 bar at 1500 rpm and decreases gradually with decreasing engine speed to 55 bar at 800 rpm.

- Piston and connecting rod masses:
  For the proposed piston design acc. to drwg. No. ENT 759/Sh.3 a total mass of the complete piston of

$$M_p = 11 \text{ kg}$$

was estimated on the basis of statistical data.

The total connecting rod mass of

$$M_c = 12.838 \text{ kg}$$

was taken from the drawing SFF9-05-00-000.

For the rotating and reciprocating portion of the connecting rod, partial amounts of 64 resp. 36 percent of the total con.rod mass have been estimated from statistics

$$M_{c\ rot} = 8.2 \text{ kg}$$

$$M_{c\ recip} = 4.6 \text{ kg}$$

- Bearing data:

Diameter	125 mm
Min. effective width	51.6 mm
Current clearance range	0.147 - 0.235 mm
Mean clearance used for oilfilm thickness computations	0.191 mm

Above clearance range is calculated from the following specifications, considering the increase of con. rod bore dia due to the pressfit of the bearing shells.

Con. rod bore dia acc. drwg. No. SFF9-05-01-000	$135 + 0.024$
Crankpin dia acc. to operation and maintenance manual	$125 \begin{smallmatrix} -0.014 \\ -0.039 \end{smallmatrix}$
Wall thickness of bearing shells acc. to drwgs. No. SFF9-05-01-005/-006	$5 \begin{smallmatrix} -0.06 \\ -0.08 \end{smallmatrix}$
Pressfit specifications acc. to drwgs. No. SFF9-05-01-005/-006.	

With regard to the bearing clearances obtained from these calculations it has to be noticed, that the current minimum clearance of 0.147 mm which corresponds to 0.001176 x crankpin dia is on an unusually high level. Current con. rod bearing design practice is normally based on a minimum clearance of 0.00075 x crankpin dia, yielding a minimum clearance of only 0.094 mm. Also the current wall thickness tolerance of 0.02 mm is higher than, the commonly applied standard tolerance of 0.015 mm. With these standard design specifications the clearance range could be reduced to

$$C = 0.094 - 0.173 \text{ mm.}$$

To evaluate the effect of this reduced clearance range, oilfilm thickness computations have also been carried out for the according mean clearance of

$$C_m = 0\ 133 \text{ mm.}$$

- Oil viscosity:

The oilfilm thickness computations have been carried out for a dynamic oil viscosity of

$$\eta = 0.012 \text{ PaS} \quad (v \sim 13 \text{ cSt})$$

corresponding to an SAE grade 30 oil at a temperature of 95°C.

The SAE grade 30 oil corresponds to the minimum requirements specified for the lube oil characteristics in the operating and maintenance manual for the current engine.
The assumption of a mean oil temperature in the bearing of 95°C has been based on the maximum lubricating oil temperature in the sump of 86°C which is also specified in the above mentioned manual.

With these basic data, the bearing analysis yields the following results:

3.3.2.2.1. Maximum unit loads, discussion of bearing material:

Computed maximum unit loads are shown in the upper graph of enclosure 7 versus the engine speed.
As per this graph, the highest maximum unit load occuring within the investigated speed range is

$$P_{max} = 39.6 \text{ N/mm}^2$$

at 1500 rpm rated speed.

This maximum unit is on a customary level as compared with similar engines but it is considered too high for the currently specified bearing material, which is a steel backed 20% aluminum-tin-bimetal bearing. (Lining material according to SAE specification No. 783).

For this bearing material, the maximum unit loading in a heavy duty diesel engine should not exceed a level of

$$P_{max} = 30 - 34 \text{ N/mm}^2$$

For the retention of the current bearing material, an increase of the projected bearing area by at least 16%,preferably however by 25 - 30% would be required therefore.

An according increase of the bearing area which could be accomplished by an increase of the bearing diameter and/or of the bearing width, is however not feasible without major changes to the current engine design.
An increased bearing width would require an increased offset of the cylinders between both banks in order to avoid an unsymmetrical loading of the bearing as well as bending of the connecting rod. It is further questionable, whether an increased width could be accommodated in the current crankshaft design from the point of view of crankshaft strength.

A reduction of the unit load only by an increase of the bearing diameter would require a crankpin diameter of at least 145 mm. This increased pin diameter would further require a new connecting rod forging and probably space problems will arise with an enlarged con. rod design.

It is proposed therefore, to retain the current bearing dimensions and to replace the current bearing material by a bearing material with higher load ratings. The following bearing materials should be satisfactory for the maximum unit load of 39.6 N/mm^2.

Al Sn 6 Cu Ni - Trimetal bearings
lining material similar to SAE spec. No. 770.

Aluminum-Silicon Trimetal bearings
lining material similar to SAE spec. No. 781 or Glacier Code AS 78 (P).

Copper lead trimetal bearings
lining material similar to SAE spec. No. 49.

3.3.2.2.2. Minimum oilfilm thicknesses:

The results of oilfilm thickness computations for the two investigated mean diametral bearing clearances are shown in the lower graph of enclosure 7.
According to AVL's experience with similar engines, a reliable range of minimum oilfilm thicknesses of

$$t_{min} \geq 1.3 \mu m$$

is used for a discussion of these results.

As per enclosure 7, the current bearing clearance of 0.191 mm yields safe minimum oilfilm thicknesses in the lower and medium speeo range. In the upper speed range near the rated speed, however, the minimum oilfilm thicknesses obtained with the current clearance are just below the reliable range and marginal therefore.

It is further shown in this diagram, that a considerable improvement of minimum oilfilm thicknesses within the entire speed range is achieved with the reduced bearing clearance.

The reduced clearance range of

$$C \approx 0.094 - 0.173 \text{ mm}$$

is recommended therefore for an improved bearing reliability in the direct injection version.
These reduced clearances will also contribute to avoid eventual cavitation problems, which probably become more severe in the direct injection version than in the prechamber engine.

3.4 Design analysis on the reduction of fuel consumption

The proposals, described as follows have been worked out in order to decrease fuel consumption rates, increase fuel economy and decrease polution emission rates of the 12 V 180 type engine.

3.4.1 ENT 759/Sheet 01 - Layout - valve pattern and - seats

For the layout of a direct injected cylinder head we want to start with an optimized valve position.
The most important criterias for a layout of the valve position and valve size are:

a) Inner diameter of valve seat (dv)
b) Length of valve seat
c) Angle of valve seat
d) Land between the various valve seat rings
e) Land between the various valve seat rings and the nozzle hole
f) Dimension of nozzle respectively nozzle hole
g) Distance between valve head and cylinder sleeve

ad a)
The inner diameter of the valve seats should be 30 % of the cylinder bore diameter

ad b)
The length of the valve seat exerts an influence on the wear and the durability of the valve train. So a good standard level of engines in this size is a valve seat length of about 3.5MM.

ad c)
For the exhaust valve, a 90 degr. seat angle has been proposed because of the better air flow characteristic of this configuration. For the intake valve, a 120 degr. seat angle has been proposed to insure sufficient durability of the valve train.

ad d)
The land between the valve seat rings is not in a critical high level, because the distance between the two respective ports has to enable a drilled cooling water passage.

ad e)
Also the land between the valve seat rings and the nozzle hole is still on the safe side, because of the installation of the nozzle holder inside of the 4 ports.
In this area there has to remain a sufficient water jacket to guarantee a good cooling water flow through the head.

ad f)
For this size of engines, considering also a growth potential, the installation of a nozzle, similar to a Robert Bosch "T-Type", nozzle should be provided.

ad g)
In AVL's experience the distance should be in a range of 2 MM. A smaller distance would have a bad influence in the port flow efficiency, a bigger one would result in a decrease of the valve seat diameter.

All these criterias mentioned above were taken into consideration for this valve pattern - layout.

3.4.2 ENT 759/Sheet 02 - Layout cylinderhead D.I.-Version I

The main components for this cylinderhead layout are as follows:

- 4 bolt pattern
- valve position acc. to ENT 759/Sheet 01
- Intake ports producing a swirl ratio higher than 1

With the existing cylinder head bolt pattern (4 bolts M16, 4 bolts M22) it is nearly impossible to realize a modern design concept for a direct injected diesel engine.

So, the 4 bolts of a size of M27 x 3 are positioned in a way also to enable an increase of the thickness of the cylinder liner to 12.5 MM, which is sufficient to avoid further cavitation problems.

The valve position has been optimized, taking into consideration the following parameters

a) all aspects, described under ENT 759/Sheet 01
b) configuration of rocker arms in respect to the given push rod position
c) a port design without any restrictions to achieve sufficient flow efficiency and swirl level.

The main design concept, showing a cross flow port design with the intake port on the engines outside has been maintained. The position of the pushrods has been retained in relation to the camshaft and the camfollowers, but has been changed from one engine side to the other. So, the two pushrods are close together (41 MM) without any modification of the camfollower and the camfollower axle.

The cooling water flow through the cylinderhead has been newly designed to satisfy the special cooling conditions of the D.I. design concept. There is only one cooling water connection between the motorblock and the cylinderhead, using the same connection part as for the two pushrod pasages.
The cooling water coming up the motorblock enters into the cylinderhead by a chamber below the exhaust port. From this chamber a drilled hole brings the cooling water to the outer water jacket of the cylinderhead.

The room around the nozzle holder is separated by walls between the ports and the valve guide bosses respectively. The entrance of the cooling water to this inner water jacket will be done by 4 drilled holes between the ports, metering the water quantity by different drilling diameters.

Between the two bosses of the exhaust valve guide there is a passage to lead the cooling water flow above the exhaust port to the cooling water outlet.
This cooling water outlet is of the same size and nearly the same position as used with the current production engine. The oil supply to the cylinderhead will be done by an oil hole equally positioned as in the production engine. Also, the oil return, actuated by the tubes of the pushrod was not changed in any way. To bring the oil down to the push rod holes, a casten groove is provided to avoid oil bags or drilled oil return holes.

A proposal for the typical nozzle holder for a nozzle similar to a "T-type" of Robert Bosch is also shown in this layout.
A more detailed specification for the installation of a nozzle holder sleeve is shown on ENT 759/Sheet 05.
The actuating of the intake and exhaust valves will be done by two rocker arms, pushing down the valve bridge.
For the adjusting of the valve clearance, Sifang standard parts are provided.
The pushrod has only been changed in its length, but still the same end parts are used as with the current production engine.
For a better control of emissions and oil consumption the installation of valve seals is provided.
For inserting of the valve seat rings and the valve guides tolerances according to AVL's experience are proposed.
Also, our propsal for a valve clearance between stem and guide is shown in this layout.

The outside dimensions of the cylinder head, i.e. length and width, have not been changed in relation to the production engine. Only the height of the cylinder head had to be enlarged to 178 MM in order to increase the stiffness of the total design. The weight by itself will increase in relation to the production engine. But a weight of 34 kg for a cylinderhead of 180 MM bore diameter is untypically low and will result in a design with very high deformations.

The location of the cylinderhead on the cylinderliner will be done by the outside of the flange of the liner and the 4 connection tubes for water and oil passages.

3.4.3 <u>ENT 759/Sheet 03 Layout Piston D.I.</u>

Using a piston with the outline contours of the present IDI engine would result in a very poor design for a DI purpose. First of all the very short length will raise problems, but also the land between first ring and top surface of 18 MM is too poor for a sufficient durability of the piston rings. The new design shows a piston with increased length, taking into consideration the lowest possible increase in engine height. The minimum upper height of the piston is influenced by the following parameters:

a) land between 1. ring groove and piston top surface
b) land between the various rings
c) necessity of a piston pin diameter of 62 MM as an absolute minimum size.
   62 MM are the biggest size admissible with the current connecting rod. It is possible to remachine the connecting rod to use the 62 MM piston pin.
   But this pin enables the engine only to run in a range of about 120 to 125 bar of peak firing pressure.
   To give the engine better durability and also a certain growth potential AVL would propose to use a piston pin diameter of 68 MM.
   This increase would reuslt in a modification of the small end of the connecting rod forging.
   More detailed information regarding this problem are given in section 3.3
d) the combustion bowl according to the specification described in enclosure 5.

Also, a cast-in ring carrier is provided to control the high temperatures in the first ring groove.

3.4.4 ENT 759/Sheet 04 Installation Cylinder liner D.I

Because of the newly designed bolt pattern for the cylinder head bolts it was possible to optimize the design of the cylinder liner.
The thickness of the cylinder liner has been increased to 12.5 MM, which is about 7 % of the cylinder bore diameter. This thickness of 12.5 MM in connection with the peak firing pressure of 125 bar and a distance between lower and upper fit of 238 MM (approx. 1.32 times the bore diameter) is a good compromise without any restrictions.
The flange design has also been changed because of the increase in the upper fit diameter. The outside diameter of the flange had to be increased in order to reduce the level of the pressure, caused by the increased bolt loads. The cylinderhead gasket will be of the same type as proposed for the converted IDI engine. But only use of a "hard" type has been proposed.

The location of the cylinderhead will be done by the outside diameter of the flange of the cylinder liner. So, a machining of the deck of the motorblock is not necessary. More detailed information regarding fit dimensions and tolerances also for position and size for the groove of the 0-rings are given in this layout.

A table for bolt loads, tightening torques and ratio between gas load and bolt load is given for 4 bolt and 6 bolt version in this layout.
Some basic information for honing is also given in this layout. The material should be from centrifugal casting with a hardness of about HB 230 - 280.
Some pictures, showing typical micro structures of not heat treated casten cylinder liners are attached to this report as enclosure 8.

3.4.5 ENT 759/Sheet 05 Installation, nozzle holder sleeve

A very critical and very important detail of a direct injected cylinderhead design is the installation of the sleeve for the nozzle holder.
AVL has a very long experience in such kind of installation and this layout is showing a typical design for this size of engines.

A detailed specification for the machining of the cylinder head and the sleeve is given in this layout.

The material of MS 72 according DIN 17660 (MS 72 means brass with 71 - 73 % CU and Zn as the rest) should be used.

The assembly procedure for the sleeve into the cylinderhead should be as follows:

a) Clean the cylinderhead and the sleeve carefully with Trichloraethylen
b) Coat the cylinderhead and the sleeve on the marked up areas with "loctite 40"
c) fix the sleeve on the cylinderhead with a holder before punching
d) after punching, remove the protrusion as shown in the layout drawing.

The upper sealing will be provided by an O-ring, which is easy to rechange by removing the M 52 x 2 lock-nut.
This nut should avoid that the sleeve will not fall out when disassembling the nozzle holder after a long period of running the engine.

3.4.6 ENT 739/Sheet 06 Motorblock - DI engine - 4 bolts

The newly designed 4 bolt pattern and, in connection with it, the optimized cylinder liner design results in a new design for a motorblock.

The only restrictions to this concept are the cylinder distance of 250 MM, the position of the camshaft, and the axle for the cam followers and also the total height of the motorblock which should not be changed.
The design concept is very close to the one described under ENT 758/Sheet 02, but with some specific modifications caused by the newly designed cylinder liner.
The total height of the motorblock of 612 MM has been retained but the height of the banks has been changed from 602.55 MM machined surface to 611 MM unmachined surface.

Also the position and the size of the upper and the lower fit of the cylinderliner has been modified. A detailed specification including all tolerances is given in this layout.
The cooling water flow through the motorblock is very similar to the converted IDI motorblock. The entrance of the cooling water to the motorblock on the vibration damper side has been kept in size and position.
In the area between lower fit and the cranking room a water jacket core is located to distribute the cooling water to each cylinder.

The entrance to the water jacket around the cylinder liner is of a tangential type to insure a rotation of the cooling water around the liner.
For the supporting of this additional core some core holes have been made that also allow a careful cleaning from sand.
The machining of the surfaces for the hand hole covers and the supports for the oil cooler and -filter have not been changed in any way. The entrance of the cooling water to the cylinderhead will be done by only one connecting piece, located on the highest position of the water jacket. This is to avoid steam pockets around the cylinder liner.

Also the core inside the V-room has only to be changed to the area of the cylinder head bolts to bring a better connection between these bolts and the wall and for the new water passage to the cylinderhead

The machining of the camshaft is not influenced by any modification. The bosses for the cylinderhead bolts are designed to achieve best load transfer to the bearing walls and to minimize deformations of the cylinderliner. The seat angle of the cylinder head bolts is of a 120 degr. design.

For the water connection and the push rod connection the same pieces have been provided. On this layout detailed information regarding dimensions and tolerances is given also for the connection of the high pressure oil supply. To stiffen up the total design, some wall thicknesses are changed in relation to the current production engine. The specific areas are marked up on this drawing.

3.4.7 ENT 759/Sheet 07 Layout Cylinderhead - 4 bolt Vers.II

The main components for this cylinderhead layout are as follows:

4 bolt pattern according to ENT 759/Sheet 02
Valve positions acc. to ENT 759/Sheet 01
Intake ports producing a swirl ratio higher than 1

The main design concept, showing a cross flow port design with the intake port on the engine out-side has been retained. The position of the pushrods has been retained in relation to the camshaft and the cam followers, but has been changed from one engine side to the other. So, the two pushrods are close together (41 MM) without any modification of the cam follower and the cam follower axle.
The cooling water flow through the cylinderhead has been newly designed in relation to the cylinderhead concept, described under ENT 759/Sheet 02.

The cooling water coming up the motorblock enters the cylinderhead by a chamber below the exhaust port.
From this chamber a drilled hole brings the cooling water to the outer water jacket below the intermediate deck. This

room is separated from the room around the nozzle holder by walls between the valve ports and covered on the top side by the intermediate deck.

The entrance of the cooling water to the inner water jacket will be done by 4 drilled holes between the ports, metering the water quantity by different drilling diameters. From this inner water jacket around the nozzle holder the cooling water comes up and goes out of the cylinderhead through the cooling water outlet.

The cooling water outlet is of the same size and nearly the same position as used with the current production engine.

The oil supply to the cylinderhead will be done by the equally positioned oil hole as in the production engine.

Also the oil return, actuated by the tubes of the push rod has not been changed in any way. To bring the oil down to the push rod holes, a casten groove has been provided in order to avoid oil bags or drilled oil return holes. The installation specification of the sleeve for the nozzle holder is described under ENT 759/Sheet 05.

The vale train, i.e. push rods, rocker arms, rocker arm brakket, valve bridges and valves are the same parts, shown in ENT 759/Sheet 02.

The outside dimensions of the cylinderhead, i.e. length and width, have not been changed in relation to the production engine. Only the height of the cylinderhead had to be enlarged to 178 MM in order to increase the stiffness of the total design. The weight by itself will be increased, also in relation to the design concept described under ENT 759/Sheet 02, but only to a very small extent. This increase is caused by the intermediate deck which by itself will stiffen up the total design. The location of the cylinderhead on the cylinderliner will be done by the outside of the flange of the liner and the 4 connection tubes for water and oil passages.

3.4.8 Discussion of the two cylinderhead concepts according to ENT 759/Sheet 02 and 07

The water jacket core of the cylinderhead acc. to sheet 02 is of a two piece design. When assembling the cores it is necessary to cement the upper and the lower part together after fixing the valve port cores to their supports.
An additional core for the nozzle holder chamber is also necessary. This procedure is a very critical one because of the danger of producing fins when casting the cylinderhead. The water jacket core for the cylinderhead acc. to sheet 07 is of two separate pieces, which have not to be cemented together. Thus, the assembling procedure is very much easier than the one described above because there is no danger of producing fins. Also a larger number of smaller core holes is better for the stiffness of the cylinderhead than only some large holes. The advantage of the one piece water jacket lies in the stiffer design because of the supporting of the 4 walls between the valve guide bosses, but this supporting is not so important.

3.4.9 ENT 759/Sheet 08 Engine cross section and outline contours

This layout drawing has been made to demonstrate that all the modifications, necessary for the conversion of the DI combustion system, do not influence the outline contours of the engine in any area.
Also the increase of the height of the cylinder unit from 870 to 936.5 MM does not influence the outline contours.

In the cross section of the cylinder unit all parts which had to be changed for the conversion are marked up. In the following, a short description of these parts will be done:

1) Motorblock
   modifications caused by the newly designed cylinder liner and the new cylinderhead bolt pattern, described in ENT 759/

Sheet 06, 11.

2) Cylinder liner
new design because of increased liner wall thickness, a new fit design described in ENT 759/Sheet 04.

3) Piston
new design with increased length, ring carrier and combustion bowl for DI purpose acc. to ENT 759/Sheet 03.

4) Cylinderhead
new design for DI combustion system. The respective variations are described in ENT 759/Shet 02, 07, 10.

5) Cylinderhead gasket
new design acc. to ENT 759/Sheet 04

6) Cylinderhead bolts & nuts
new design acc. to ENT 759/Sheet 02, 04, 10, 11.

7) Valves and valve train
the valves and also all valve train parts such as pushrods, rocker arm brackets, rocker arms and valve bridges are shown in ENT 759/Sheet 02.

8) Nozzle holder & bracket
A proposal for a nozzle holder is also given in ENT 759/ Sheet 02.

9) Injection lines
The injection lines have to be of the same length, the inside diameter is specified in 3.2.1.2

10) Cylinderhead cover
new designed cylinder head cover acc. to ENT 759/Sheet 02.

11) Intake manifold

The receiver of the intake manifold can remain in size and position, but the connection to the cylinderhead intake flange has to be adapted to the new port design.

12) Exhaust manifold

The system for the exhaust manifolds is principally the same as used with the production engine, only the flange contours and the cross section have to be adapted to the new exhaust flange position and size.

13) Cooling water pipe

The system for the cooling water pipes has principally been retained; so as the flange contour. There are only some small changes because of the increased top deck height of the engine.

14) Camshaft

The camshaft has to be redesigned because of the exchanging of intake and exhaust cams and the newly defined cam positions and cam profiles.

15) Piston pin

Newly designed piston pin because of the increase peak firing pressure of the DI version, described in ENT 759/ Sheet 03.

16) Connecting rod

remachining of the small end of the connecting rod because of the increased piston pin diameter.

3.4.10 ENT 759/Sheet 09 - Piston grinding shape

This layout is only a proposal for starting the development work. This piston grinding shape is in accordance with a well prooved design but it will be necessary to adapt this grinding shape to the special conditions of the Sifang engine.

In addition to the proposal of a grinding shape, in the following we will give also a preliminary specification for the piston rings.

1st groove:
   plain compression ring, chrome plated, symmetrically spherical form

2nd groove:
   taper faced compression ring

3rd groove:
   napier ring

4th groove:
   coil spring loaded bevelled edged oil control ring
   chrome plated
   surface load 75 N/CM2

Also the piston ring specification may have to be changed during the development work.

3.4.11 ENT 759/Sheet 10 Cylinderhead, 6 bolt pattern Vers.III

The main components for this cylinderhead layout are as follows:

- 6 bolt pattern
- valve positions acc. to ENT 759/Sheet 01
- intake ports producing a swirl ratio higher than 1

The main design, showing a cross flow port design with the intake port on the engine out-side has been retained.
The position of the pushrods has been retained in relation to the camshaft and the cam followers, but has changed from one engine side to the other. So the two pushrods are close together (41 MM) without any modification of the cam follower axle.
The cooling water flow through the cylinderhead is very similar to the one described under ENT 759/Sheet 07.

The cooling water, coming up the motorblock enters the cylinderhead by a chamber below the exhaust port. From this chamber, a drilled hole brings the cooling water to the outer water jacket below the intermediate deck. This room is separated from the room around the nozzle holder by walls between the valve ports and covered on the top side by the intermediate deck.
The entrance of the cooling water to the inner water jacket will be done by 4 drilled holes between the ports, metering the water quantity by different drilling diameters.
From this inner water jacket around the nozzle holder the cooling water comes up and goes out of the cylinderhead around the exhaust port and through the cooling water outlet.
This cooling water outlet is of the same size and nearly the same position as in the current production engine. The oil supply to the cylinderhead will be done by the oil hole equally positioned as in the production engine.
Also the oil return, actuated by the tubes of the pushrod has not been changed in any way. To bring the oil down to the pushrod holes, a casten groove is provided to avoid oil bags or drilled oil return holes.
The installation specification of the sleeve for the nozzle holder is described under ENT 759/Sheet 05.
The valve train, i.e. pushrods, rocker arms, rocker arm brakket, valve bridges and valves are the same parts: shown in ENT 759/Sheet 02.

The outside dimensions of the cylinderhead have been changed in relation to the one, described under ENT 759/Sheet 02 and 07. The side surface have been turned for about 6 degr. but have the same width of 248 MM to meet the 250 MM cylinder distance. The position of the exhaust flange has been retained; the position of the intake flange had to be moved for 6 MM to 146 MM.
Also the height of the cylinder head had to be enlarged to 178 MM to increase the stiffness of the total design.
The weight by itself will not be changed in relation to the version described under ENT 759/Sheet 07.

The location of the cylinderhead on the cylinder liner will be done by the outside of the flange of the liner and the 4 connection tubes for water and oil passages.
This cylinderhead design concept with 6 bolt pattern can also be realized with the water jacket core design acc. to ENT 759/Sheet 02 with all its advantages and disadvantages.

3.4.12 Comparison of 4 bolt pattern versus 6 bolt pattern cylinderhead design

a) 4 bolt version

advantages:
- stiffness of the wall in longitudinal direction
- straight side walls
- intake flange 140 MM
- good load transfer from the cylinderhead bolts to the main bearing walls

disadvantages:
- high tightening torque
- tightening of cylinderhead bolts very critically, if not with hydraulic equipment
- connection of a part of the exhaust port with the oil room (near pushrod)
- heating of the 2nd exhaust valve guide boss through the gas flow of the 1st port
- restrictions in the design of the short intake port by the longer one.

b) 6 bolt design

advantages:
- smaller tightening torque
- tightening procedure is not so critical because of the smaller torque and the higher number of bolts
- more flexibility for the design of the exhaust port which result in better flow efficiency
- water jacket around the exhaust port (no connection to oil room)

- no restrictions for the design of the short intake port

disadvantages:
- turned side walls
- intake flange 146 MM (increase of 6 MM)
- load transfer from the 6 bolts to the main bearing wall is more critical also because of the not exactly symmetrical pattern.

c) Discussion

If there is any possibility to tighten the cylinderhead bolts with hydraulic equipment, AVL would propose to use the 4 bolt pattern.
There are two reasons to do this

1) The hydraulic method of bolt tightening by itself
2) The 4 bolt pattern and the better load transfer to the main bearing walls.

But if a bolt tightening from hand has to be realized, we would propose to use the 6 bolt pattern design. The reasons for doing this are:

1) The smaller tightening torque (52.9 MKP versus 95 MKP)
2) The smaller influence of a not correct tightening procedure because of the higher number of bolts.

3.4.13 ENT 759/Sheet 11 Motorblock DI engine 6 bolt pattern

This layout should be understood as a coversheet to the motorblock layout, described under ENT 759/Sheet 06, for the 6 bolt pattern, shown on the cylinderhead acc. to ENT 759/Sheet 10. The design is similar to the one of the 4 bolt pattern; also the cylinder liner design is the same.
The only modification has been made at the 6 bolt pattern and is shown in this layout.
Also the bosses of the M 22 x 2,5 threads and their connection to the mainbearing walls is shown in this layout.

3.4.14 ENT 759/Sheet 12 Increased piston length

As mentioned under ENT 759/Sheet 03 the increase in the piston length has been made for a minimum of increase in the motor-block top deck height. This total length of the piston, and also the ratio of upper to lower length of the piston is only a compromise.
Statistical datas at AVL, given by a number of different manufactors of engines in this size, show a very wide range of practicable designs for pistons.
So, according to AVL's experience a piston design as shown in ENT 759/Sheet 03 is a practicable design, even if it is far from being the average of our statistical datas.

The layout ENT 759/Sheet 12 shows a piston design which is taken into consideration these statistical datas.
This new design will result in a increase of the piston upper length of 26 MM in relation to the one, shown in ENT 759/Sheet 03 and of 42.5 MM in relation to the production engine.

This increased upper length of the piston will however, have some influence on the to deck height of the motorblock and the total height of the engine. Some problems will occur with the cooling water and the push rod connections to the cylinderhead.

The increase in the distance between the upper and the lower fit of the cylinder liner will have a negative influence on the duration against cavitation.

But this negative influence may be compensated by the positive influence of the enlarged total length of the piston.

List of Enclosures

List of attached layouts

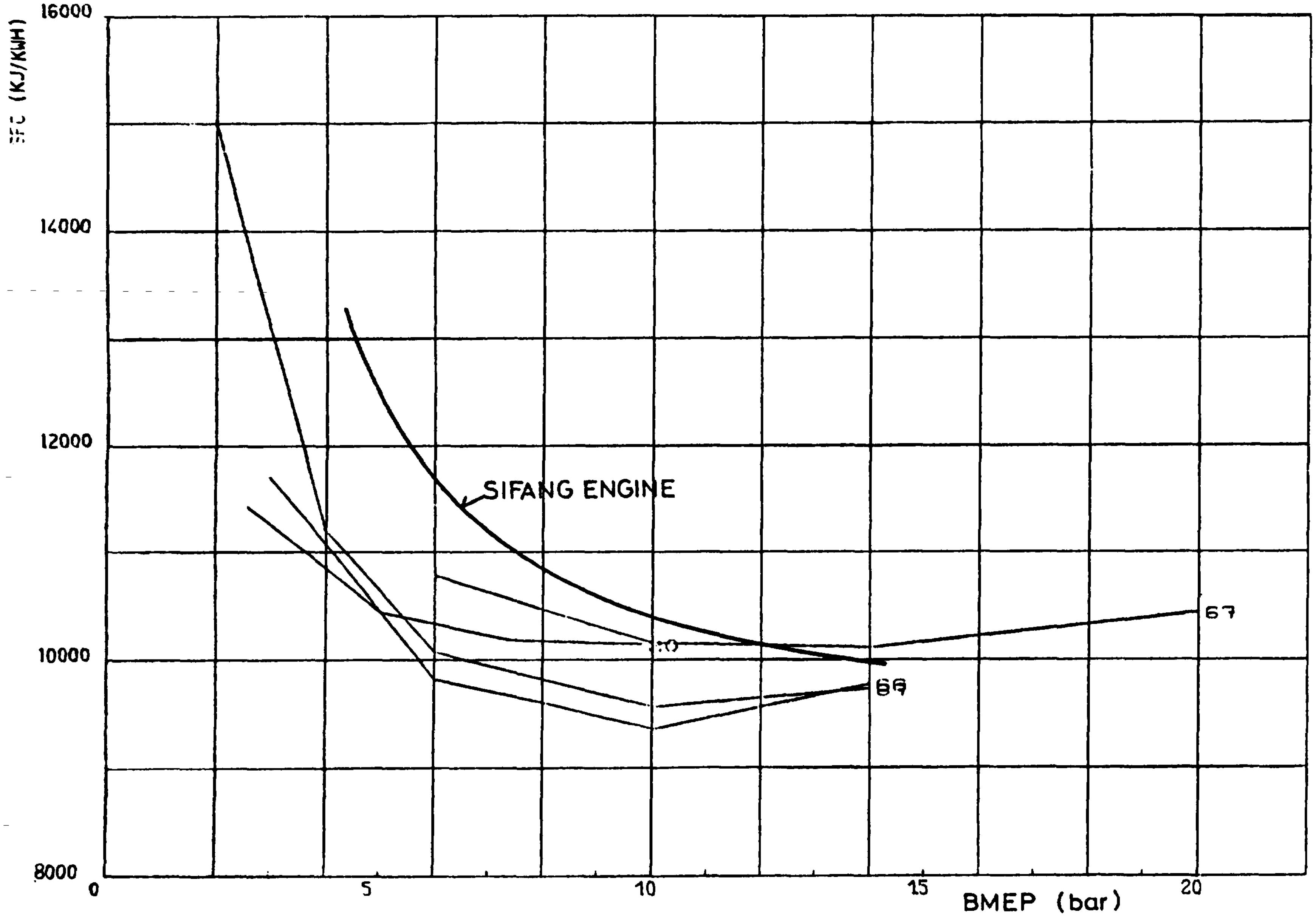
1) IDI engine

ENT 758/Sheet 01	Installation cylinder liner
ENT 758/Sheet 02	Motorblock IDI engine
ENt 758/Sheet 03	Layout piston IDI

2) DI engine

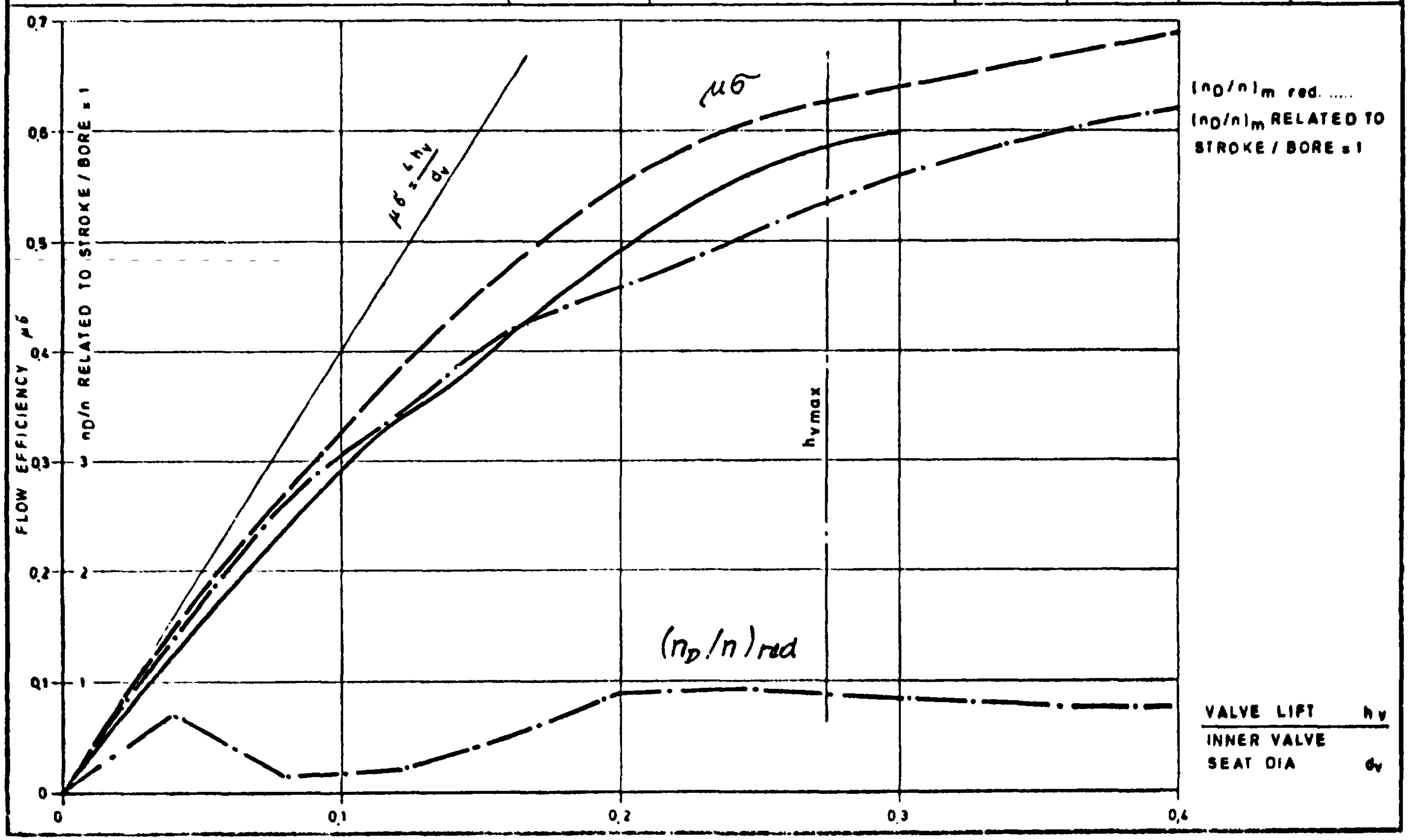
ENT 759/Sheet 01	Layout valve pattern & seats
ENt 759/Sheet 02	Layout cylinderhead DI Vers.I
ENT 759/Sheet 03	Layout piston - DI
ENT 759/Sheet 04	Installation cylinder liner - DI
ENT 759/Sheet 05	Installation nozzle holder sleeve
ENT 759/Sheet 06	Motorblock - DI engine - 4 bolt pattern
ENT 759/Sheet 07	Layout cylinderhead - DI - Vers.II
ENT 759/Sheet 08	Engine cross section and outline contours
ENT 759/Sheet 09	Proposal, piston grinding shape
ENT 759/Sheet 10	Layout cylinderhead 6 bolt pattern,Vers.III
ENT 759/Sheet 11	Motorblock DI engine 6 bolt pattern
ENT 759/Sheet 12	Increased piston length

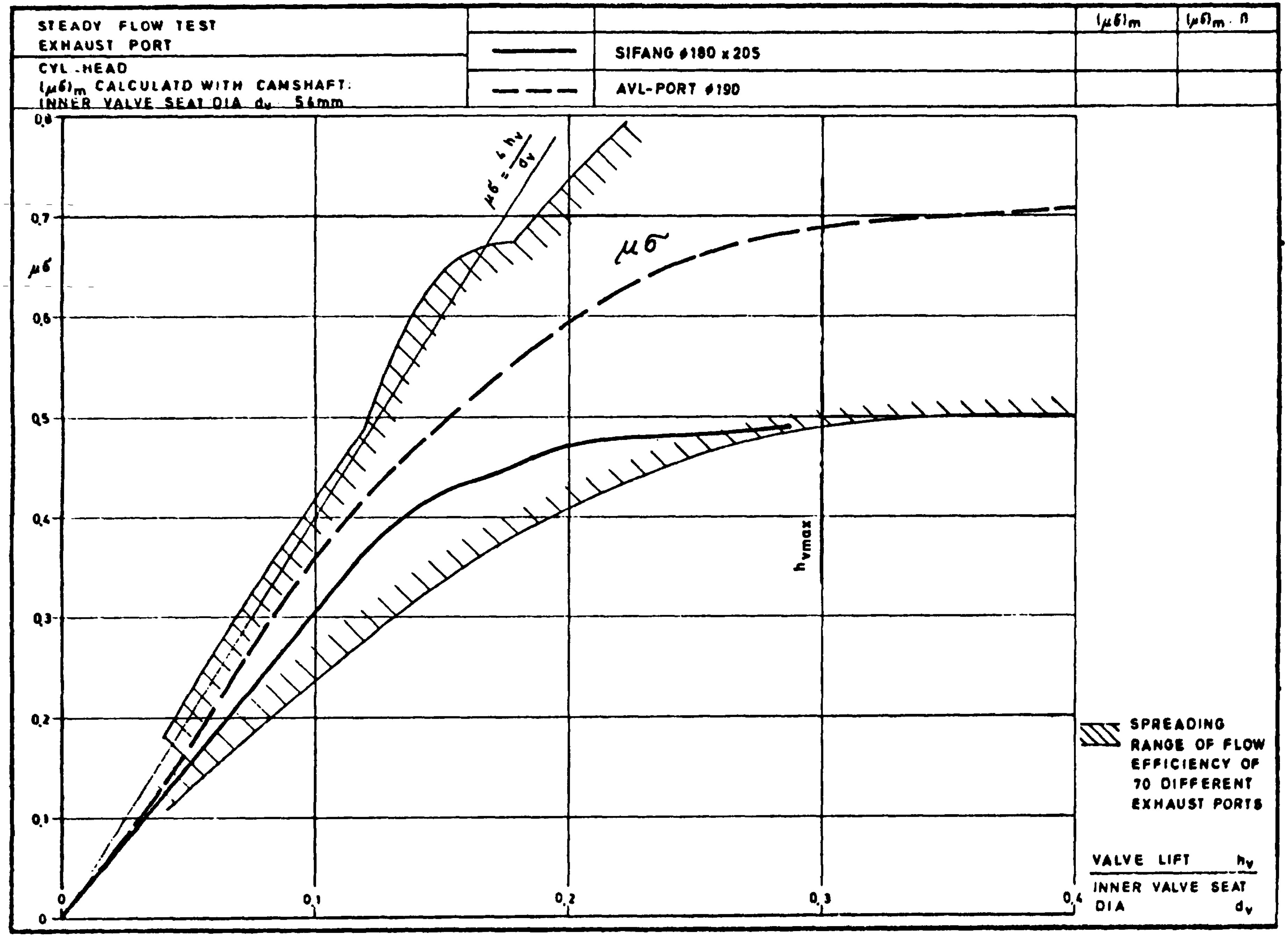
SPECIFIC FUEL CONSUMPTION ON GENERATOR APPLICATION



4-cycle IDI engines, turbocharged/aftercooled D >170 MM

STEADY FLOW TEST INTAKE PORT			$(n_D/n)_m$	$(n_D/n)_m$ red	$(\mu\sigma)_m$	$(\mu\sigma)_m \cdot n$
	————	SIFANG ⌀180x205				
CYL.-HEAD: $(n_D/n)_m$, $(\mu\sigma)_m$ CALCULATED WITH CAMSHAFT:	— — —	NEUTRAL AVL-PORT ⌀190	0	0	0,411	0,0739
INNER VALVE SEAT DIA d_v: 55mm	—·—	SWIRL PRODUCING AVL-PORT ⌀190	2,162	2,162	0,323	0,0627





STEADY FLOW TEST
EXHAUST PORT
CYL. HEAD
(μσ)m CALCULATD WITH CAMSHAFT:
INNER VALVE SEAT DIA dv 56mm
(μσ)m
(μσ)m · n
SIFANG ø180 x 205
AVL-PORT ø190
μσ = 4 hv / dv
μσ
hvmax
SPREADING RANGE OF FLOW EFFICIENCY OF 70 DIFFERENT EXHAUST PORTS
VALVE LIFT hv
INNER VALVE SEAT DIA dv
0,8
0,7
0,6
0,5
0,4
0,3
0,2
0,1
0
0,1
0,2
0,3
0,4

DI-Engine 933 KW/155 RPM

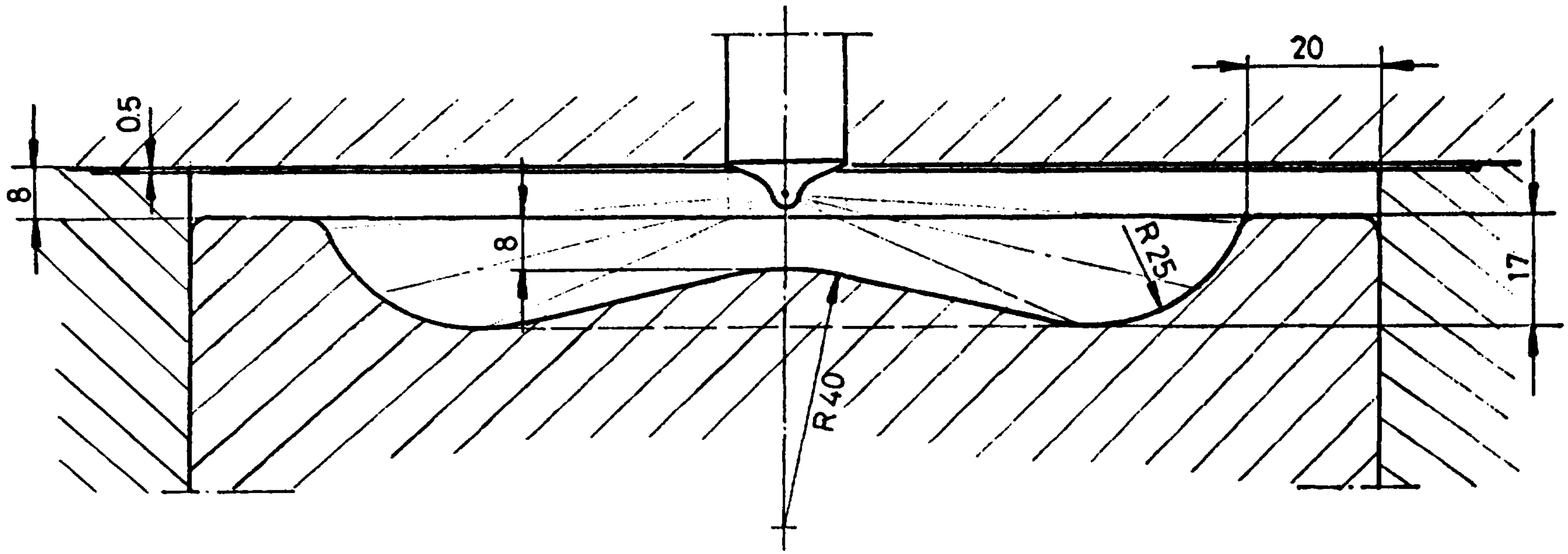
BMEP bar	12.7
FMEP	2.0
IMEP	14.7

Inlet manifold-pressure bar	2.3
-temperature degr.C	65
Volumetric efficiency-total	1.15
-traped	0.98
Injection fuel mg/cycle/cyl.	397.8
Excess air ratio of combustion	2.07
Indicated thermal efficiency %	45.2
ISFC g/KWh	186.5
BSFC	216
Compressor air flow kg/s	2.13
Compressor pressure ratio	2.35
Temperature after compression degr.C	124
Temperature before turbine	520
Pressure before turbine bar	2.0

Heat rejected to cooling water:

Intercooler KJ/KWh	485
Engine (oil cooler included)	1760
Total	2245

COMBUSTION CHAMBER



COMPRESSION VOLUME 408 cm^3

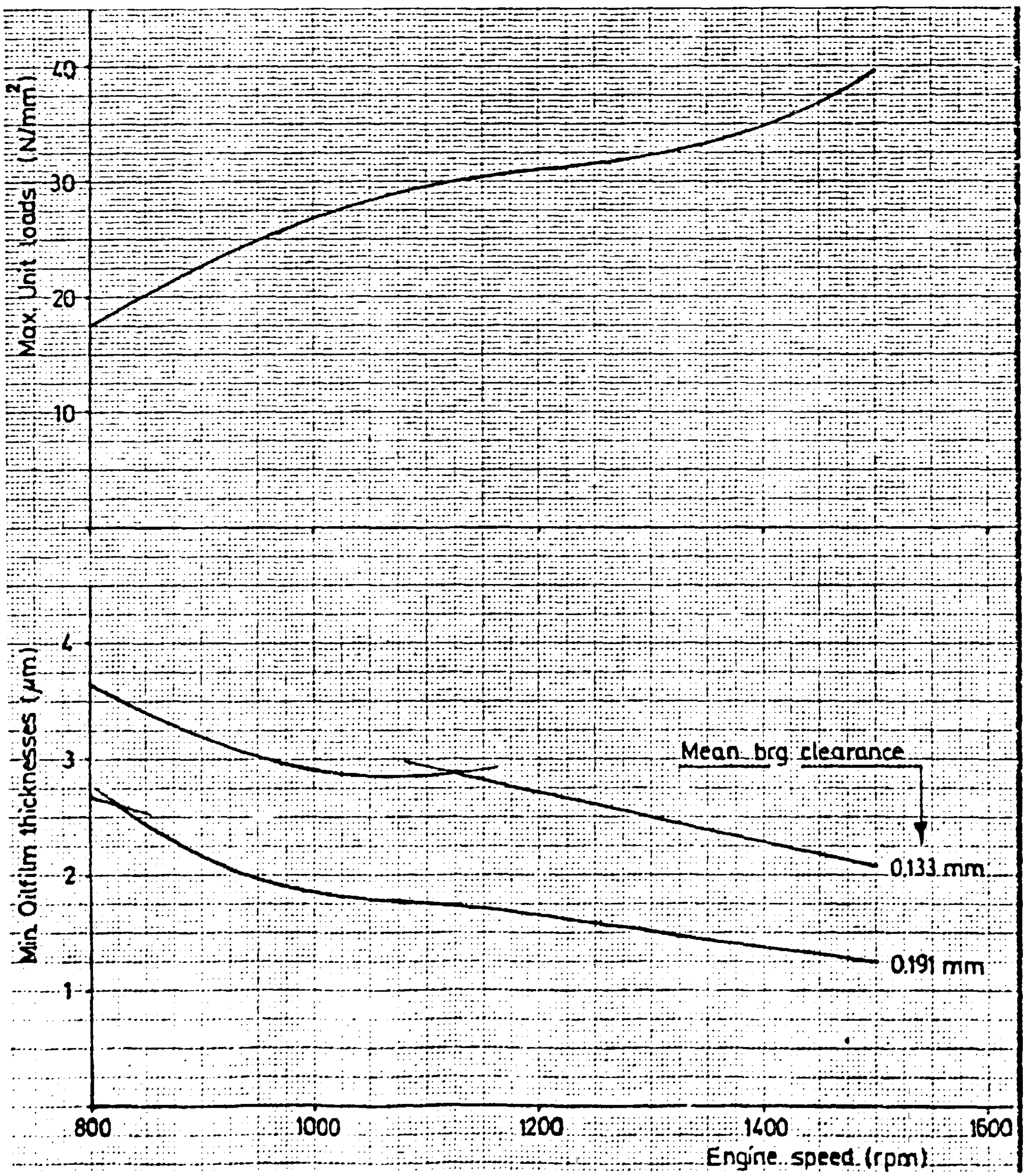
COMPRESSION RATIO 13.8

Piston Pin
Design and Performance Data

Design version		-	Current	Proposed
Cylinder peak pressure		bar	125	
Pin diemnsions	Outer dia	mm	60	62
	Inner dia	mm	24	30
	Overall length	mm	158	152
	Effective length	mm	156	150
Small end brg. width		mm	71	71
Width between inner edges of piston bosses		mm	64	64
Pin deflections	Longitudinal deflection	μm	46.8	38.4
	Permissible long. deflection	μm	39.6	
	Oval deflection	μm	15.5	30.2
	Permissible oval deflection	μm	37.8	
Unit loads	Unit load in piston	$\frac{N}{mm^2}$	57.6	59.65
	Reference values for unit load in piston	$\frac{N}{mm^2}$	60 - 65	
	Unit load small end	$\frac{N}{mm^2}$	88.4	85.5
	Reference values for unit load S.E.	$\frac{N}{mm^2}$	85 - 90	

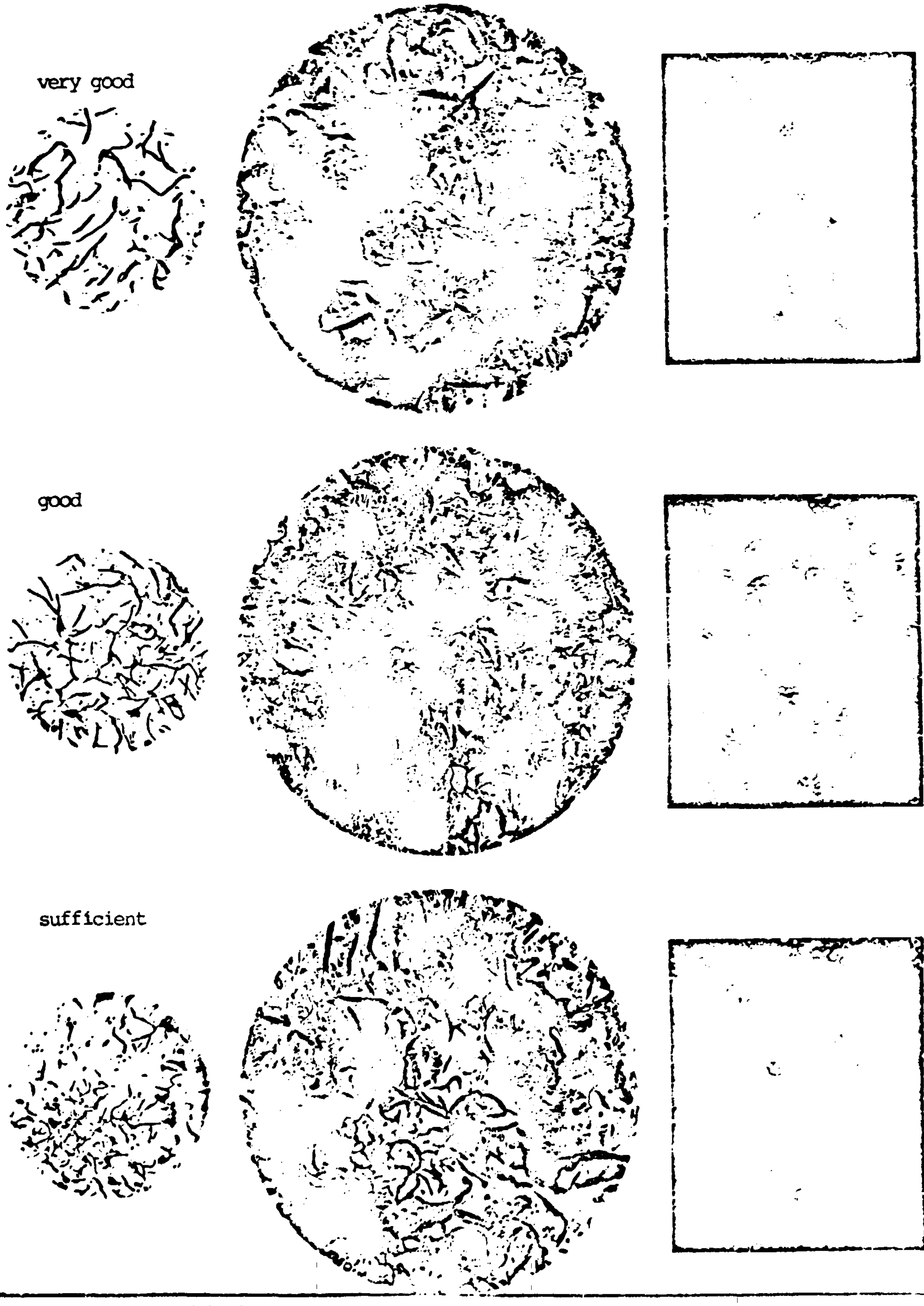
$1 \mu m = 1/1000$ mm

Maximum Unit Loads and Minimum Oilfilm Thicknesses
in the Connecting Rod Big End Bearing



GENERAL INSTRUCTIONS FOR STRUCTURE OF NOT HEAT TREATED CASTEN CYLINDER LINER

GRAPHITE 100:1 BASIC STRUCTURE 200:1 PHOSPHITE MATRIX 20:1



SOME FIGURES OF THIS DOCUMENT ARE TOO LARGE FOR MICROFICHING AND WILL NOT BE PHOTOGRAPHED.