EFFECT OF FLUID AND COMBUSTION MECHANIC MODIFICATIONS ON THE ACTUAL EFFICIENCY OF A MARINE DIESEL ENGINE

By

P. LOSONCI (LOJEK)

Department of Aero- and Thermotechnics, Technical University Budapest

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Fuel economy is a primary interest of diesel operators, especially today, by cutting down the running expenses.

Engine manufacturers are interested in increasing the effective efficiency, of their Diesel engines as they are easier to sell, have a longer life and are less sensitive to breakdown.

Theoretical and experimental examinations have been made to improve the effective efficiency of an existing ship Diesel engine series.

It was assumed that the improvement of one chief component of the fuel injection system, certain fluid mechanical modifications and a novel combustion chamber are likely to increase the effective efficiency of the engine series.

In the design of, and development tests on the high-pressure supercharge unit of the engine series, similar modifications have been introduced as previously for the low-pressure supercharge unit. Component alterations relying on the results of investigations significant improved the effective efficiency of both the low-pressure and the high-pressure supercharge engine. By the way, several regularities have been formulated.

Test results are a contribution to the theory of mixture formation and combustion in Diesel engines.

The investigations and test series were carried out in the LÁNG Engineering Works, whereas the pilot plant tests were sponsored by the Hungarian Shipping Company (MAGYAR HAJÓZÁSI RT). The author feels sincerely indebted to those who promoted the successful completion of the series of experiments with their devoted work and helpful dispositions.

Let us outline the particulars, partial results of the research work and some conclusions drawn from it.

1. The theoretical investigations were expected to yield directly applicable in manufacture

A 736 kW (1000 HP) straight-eight engine series of the engine set has been put at our disposal to carry out the theoretical investigations and experiments.

The chief characteristics of the engines are:

Effective continuous rating	P_e = 736 kW (1000 HP)
Continuous revolution number	n = 400 r.p.m.
Performance at 1 hour overload	P'_e = 810 kW (1100 HP)
Excess load revolution number	n' = 412 r.p.m.
Specific fuel consumption	b_e = 231 g/kW$\times$h; + 5.5% (170 g/HP$\times$h; + 5.5%,
Bore	D = 315 mm
Stroke	S = 450 mm
Total piston displacement	V_t = 281 ls
Compression ratio	ε = 10.8
Break mean effective pressure	p_e = 8.0 bars
Supercharge air pressure	p_s = 0.4 bar
Combustion pressure	p_c = 53 + 2 bars
Compression pressure	p_{cp} = 36 bars
Combustion chamber	undivided disc-shaped with a half-lens-shaped through in the light metal piston crown; direct injection; no regulated air-flow in the cylinder
Injection nozzle type:	Bosch DL T 183 ($6\times 0.4\times 120°$)
Injection pump type:	Bosch PF 1W 190 independent pump for each cylinder
Supercharging system:	pulsating, every two cylinders attached to a common pipe. If mounted on engine, one BBC VTR 320 type supercharger. No air cooling.
Number of valves:	one inlet and one exhaust valve
Degree of valve overlap:	$\Delta\varphi$ = 150 crankshaft degrees
Specific oil consumption:	b_{ek} = 4.08 g/kW$\times$h; + 10%

2. Engine operation observations and conclusions

In the initial of the investigations the engine has been observed operating conditions and its main components uncleaned, after disassembly.

2.1. In general, the components exhibited the following disorders:

— during operation the engine smoked moderately, even under partial loading;
— as many carbon hillocks as the number of bores on the carburettor were deposited on the piston crown;
— on the upper casing of the cylinder-liner an 1 mm thick carbon layer was formed. The carbon deposit filled in the space between the piston crown and the cylinder-liner meant to be occupied by the upper piston ring;
— oily coke and soot was deposited in the suction tube of the cylinder head and in the supercharge air collector.

2.2. These disorders:

— the smoking of the engine, carbon hillocks on the piston crown and the coke deposit on the upper casing of the cylinder-liner motivated investigations into
  — the combustion air ratio,
  — the mixture formation and combustion phases of the engine, and
  — the form of the combustion chamber.
— Contamination of the suction tube and the supercharge air collector with oily soot hinted to flow problems
  — across the inlet valve and the exhaust valve,
  — in the exhaust tube and
  — in the supercharge air collector.

3. Partial examinations and results

3.1. The full air ratio m_t and combustion air ratio m_c of the engine have been investigated for the characteristics

$P_e = 736$ kW,
and $n = 400$ r.p.m.

An earlier measurement of the full air ratio by gas analysis made at the Department of Motor Vehicles, Technical University, Budapest resulted in

$$m_t = 2.4.$$

Full air ratio of the same engine calculated from measured air consumption and specific fuel consumption was:

$$m_t = 2.45.$$

in a rather good agreement with the measurement results of the Department.

Approximate calculations led to the combustion air ratio

$$m_c = 1.75—1.8.$$

These figures improbabilized combustion air insufficiency as reason of engine smoking.

3.2. *Investigation of the elements of the fuel injection system*

The elements of the fuel injection system had been selected on the basis of the recommendations of the Bosch firm. This consideration argued against modifications with these elements. Statements under item 2. pointed, however, to combustion disorders. Combustion being much dependent on injection, it seemed practicable to check the junction of the injection nozzle to the combustion chamber or to the combustion itself.

Accordingly, several different types of nozzles were purchased and tested.

3.3. *Possibilities of simplifying the complicated combustion chamber*

The form of the original combustion chamber in Fig. 1. was caused by the 1 mm annular groove in the upper casing of the cylinder-liner.

The longitudinal dimension of this groove was a function of the compression ratio of the engine unit and of whether the engine had a cast iron or light metal pistons. These factors resulted in four different cylinder-liners, that were not allowed to exchange lest damages resulted.

The elimination of the groove hence of the running-out space of the upper piston ring risked to prevent removal of the piston from the worn cylinder-liner. Nevertheless the elimination of elaborateness seemed practicable.

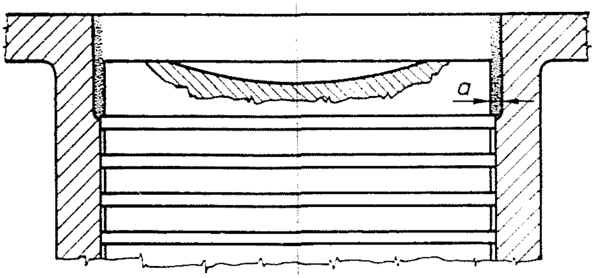

Fig. 1. The original combustion chamber form

3.4. *The combustion chamber form*

The investigation of the combustion chamber forms showed little difference between the effective efficiencies of the naval ship Diesel-engines with disc-shaped combustion chambers (maybe with a lenticular or calotte shaped impression in the piston top ring) and those having Hesselman combustion chambers. Pistons with disc-shaped combustion chambers are easier to manufacture. Marine Diesel-engines with Hesselman combustion chambers exhibit near-optimum specific fuel consumptions in a wider load range, arguing, for the application of the Hesselman combustion chamber.

3.5. *Fluid mechanic investigation of the exhaust valve chamber and the valve*

The oily soot and carbon deposit in the suction tube and the supercharge air collector may have been due to the exhaust valve chamber and the valve itself. The original exhaust valve is shown in Fig. 2. A detailed examination of the cross-section and rate relations seemed necessary. The results are seen in Fig. 3.

The rate changes seen in Fig. 3. imposed to develop a valve with favourable cross-section and rate relations. The flow rate peak of avg. 80 m/sec developing in the smallest cross section has to be eliminated, just as the multiple sudden rate changes inside the valve chamber. Keeping the above mind, a

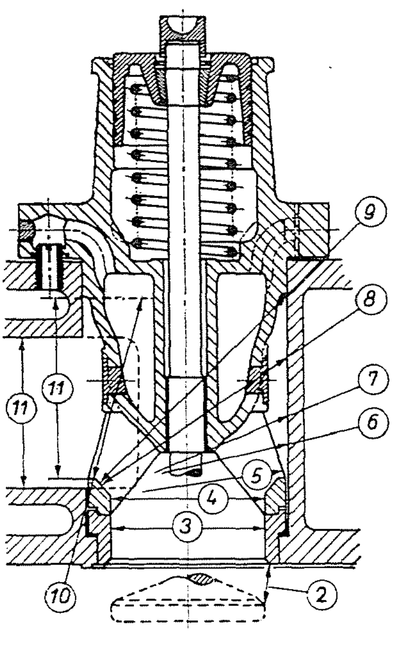

Fig. 2. The original exhaust valve

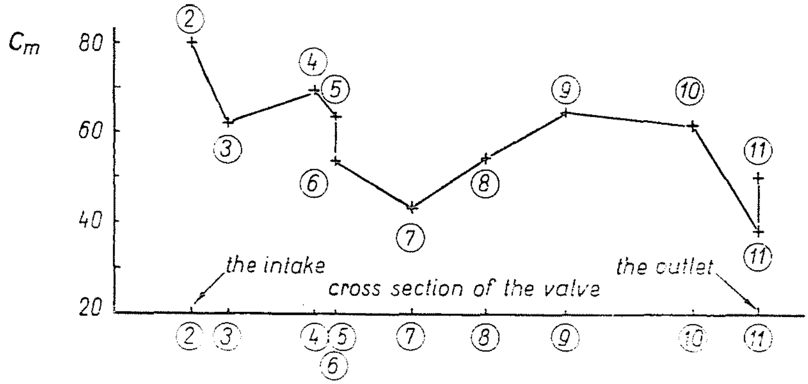

Fig. 3. The average rate relations in the original exhaust valve at a mean piston speed $c_m = 6$ m/sec

unified replaceable suction exhaust valve has been developed, much superior to the original valve from the aspect of fluid mechanics.

3.6. Fluid mechanic examination of the exhaust tube system [1]

— The greatest part of the exhaust valve system was made of rolled steel tube. Some of the shaped pieces were made of grey cast iron, while the others were welded from steel pipe sections. In the latter case six tube sections hence six 15° partial direction changes were applied for a 90° inflection.

— The caulkings and the rims in pipe junctions were not concentrical, so the pipe lengths were printed with 1—3 mm excentricities.

— So-called expansion pieces were inserted into the straight pipe lengths, that provided much useless room for expansion (40—45 mm), with a 25% sudden cross-section change. Besides, the cross-section of the next pipe length in the flow direction had sharp edges.

— Also the pipe junction to the gas trap of the supercharger contained a sudden cross-section change with a more than 60% evasion and a 30% sudden constriction.

— In the exhaust pipes the mean flow rate was 50 m/sec for a mean piston speed $c_m = 6$ m/sec. The former cannot be regarded as an extremely high value. In spite of the fluid mechanic mistakes seeming to be unimportant, yet improvements seemed advisable.

3.7. Fluid mechanic investigation of supercharge air collector and L pipes [1]

Two way examinations of the supercharge air collector imposed themselves, such as the average axial flow rate inside the collector, the air suction being normal to this direction, without baffling the development of the suction pipes.

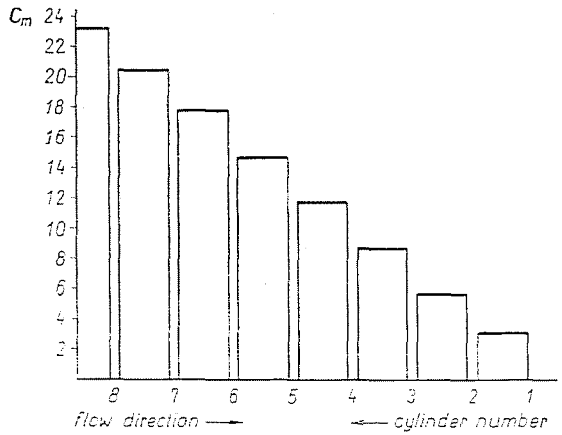

Fig. 4. Average axial rate distribution in the original supercharge air collector. $P_e = 736$ kW, $n = 400$ r.p.m.

The average axial rate distribution in the supercharge air collector is seen in Fig. 4. The operational characteristics of the engine are:

$$P_e = 736 \text{ kW}; n = 400 \text{ r.p.m.}$$

Fluid mechanic characteristics of the suction pipes were inadequate. A suction inlet tube which was an approximately square-based truncated pyramid, was connected through a cylindrical transition piece to the outlet L-pipe. The L-pipe was welded of rolled steel pipe sections, changing the flow direction by 90 degrees.

These investigations justified to reduce the average axial rates in the supercharge air collector, as well as to make tests to develop a simple suction flange and an L-pipe cast of light metal.

4. Constructional modifications relying on test results. Results of tests with the new types of components and evaluation

On the basis of theoretical investigations all the engine parts have been manufactured or purchased that were expected to improve the effective efficiency.

The design had to satisfy certain restricting or mutually contradictory requirements.

Among these the most significant ones were the following:
— the new possibility to replace the old component by ones;
— as few parts affected by the constructional modifications as possible, at a minimum cost of manufacture;
— low man-hour and cost demand of experiments;
— maximum improvement in the effective efficiency of the engine.

4.1. *The new nozzle types and measurement results*

Tests were made on light kinds of purchased nozzles with geometrics different from that of the basic nozzle.

The chief geometrical characteristics of these nozzle types are shown in Table 1.

Table 1

Experimental nozzle geometria

No.	Nozzle type	Overall mm²	*Cross-section of bores* percentage	Arrangement of bores
1.	$6\times0.4\times120$	0.756	100	symmetrically on a cone shell
2.	$6\times0.35\times120$	0.576	76	symmetrically on a cone shell
3.	$7\times0.35\times120$	0.6723	89	symmetrically on a cone shell
4.	$8\times0.35\times120$	0.767	101	symmetrically on a cone shell
5.	$8\times0.30\times120$	0.564	75	symmetrically on a cone shell
6.	$9\times0.30\times120$	0.634	84	symmetrically on a cone shell
7.	$10\times0.30\times120$	0.705	92	symmetrically alternating on two cone shells
8.	$12\times0.30\times120$	0.846	112	symmetrically alternating on two cone shells
9.	$12\times0.25\times120$	0.588	78	symmetrically alternating on two cone shells

Nozzle No. 1. had been original part of the basic engines.

The interpretation of the characteristic digits of the nozzles:

— the first digit . . . . the number of bores

— the second digit . . . . the diameter of the bore in mms

— the third digit . . . . its angle (in angle degrees).

Test results for nozzles in Table 1 have been plotted in Fig. 5.

In the nozzle tests

— the initial fuel injection pressure;

— the preliminary injection angle; and

— the injection geometry have been kept constant, but the other characteristics of the injection have naturally changed.

4.2. *Conclusions and suppositions derived from the experimental results:*

— Nozzle brought about the most favourable effective efficiency value. There was neither engine smoking nor deposit of carbon hillocks on the piston crown throughout the range of operation. It can be stated that with medium-speed marine Diesel engines with no orderly air-

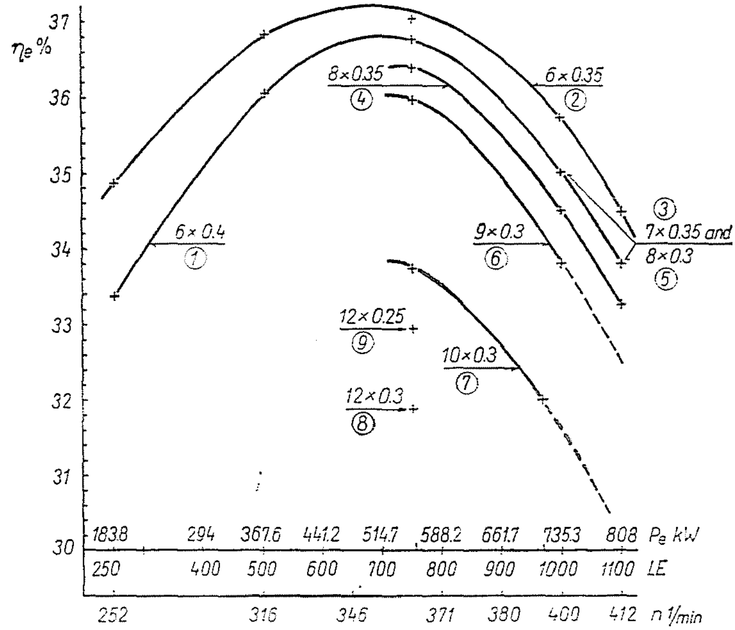

Fig. 5. The change of the effective efficiency of the engine nozzle tests

flow an optimum nozzle ensuring efficient, smokeless combustion can be selected.

— The effective efficiency was not unambiguously affected by varying the cross-section overall of the nozzle.

— Increasing the number of the bores impaired the effective efficiency, first moderately later abruptly.

— Decreasing the bores diameters in nozzle 2 has improved the effective efficiency but where the constriction was concomitant to the increase of bores number, the effective efficiency dropped.

On the ground of the foregoing it can be established that with the medium-speed direct injection air supercharge marine Diesel engines, without orderly air flow in the combustion chamber, no uniform distribution of the fuel in the combustion chamber during mixture formation has to be strived to.

This conclusion is diametrically opposed to the classical approach to the mixture formation and combustion processes in the combustion chamber of medium-speed marine Diesel engines operating with multi-spray nozzles and having no orderly air flow.

In the tested case, the processes of mixture formation and combustion may be assumed as follows:

At a given engine r.p.m. and loading a multi-spray nozzle produces fuel particles of different geometries, each with a different degree of inflammability and combustibility [2], that still change in the course of combustion, partly with the change of drop size and temperature and of the characteristics of the surrounding steam envelope and partly with the change in the state characteristics of the combustion air + combustion products arounding the fuel droplet, as well as with the appearance of very intensive radiant heat at the moment of ignition.

It seems evident that some among the fuel droplets of different sizes are highly inflammable (optimum particles), others may approximate them, still others may have a poor inflammability.

The above considerations are likely to hold for the combustion properties as well.

The fuel droplets injected into the combustion chamber are unevenly distributed in a conical volume with an apex angle of 19 degrees. There are much less droplets on and near the cone shell than in the core [3].

As there is no orderly air flow in the combustion chamber the injected fuel spray cone is supposed to be little distorted until the ignition and combustion phase starts. This small distortion is caused by the residual, unorderly eddy currents arisen during the suction stroke.

Multy-spray nozzles with as many 12 jets the fuel drops inside the spray cone cannot achieve optimum air for ratio.

Now, let us introduce an arbitrary macro-mixture number, ratio of the compression volume to the total volume of fuel spray cones. Nozzle 2 has a ratio of 6, nozzles 8 and 9 have 3. According to the classical approach to ignition and combustion the most favourable case would be, if the macro-mixture number were 1, that is, the fuel spray filled in the total compression volume. This, however, cannot be realized in practice.

On the ground of the foregoing the processes of ignition and combustion can be commented in the following way:

No doubt, within the spray cone the air ratio is very unfavourable [3]. The air ratio is rather favourable near the cone shell and the base.

The heat absorption and readiness for ignition and combustion of the fuel particles being the most favourable in these zones, the ignition of these particles is expected to start there.

The ignition starts in optimum-fuel particles and this process ends where the combustion process starts.

Diesel engines operating with multi-spray nozzles, where the macro-mixture number is 6, it is in one sixth of the compression volume the fuel that has to be distributed to burn, with a high efficiency.

With no orderly air flow the combustion is likely to proceed as:

From the beginning of the combustion process on the spray cone shell, the radiant heat transfer from the lighting flame to the fuel particles abruptly grows.

Most of the radiant heat is transmitted directly to the still fluid fuel droplets, penetrating their gas envelope. Then total convective and radiant heat absorption may be so high that the fuel particles blast to gas state, instead of evaporating "slowly". As a consequence, the spray expands. Consecutively, the total heat absorption by the new-comer particles grows intensive, with the abrupt rise of temperature and pressure of the working medium. The currents generated by successive "steam explosions" are responsible for that the fuel particles inside the spray cone get to those spots in the combustion chamber, where they can burn smokeless, with a favourable air ratio. This description of the combustion process may be called "a dynamic approach to combustion".

The question arises what is the reason of the important loss of efficiency of 12 jets when the macro-mixture number is about 3.

"Steam blasts" are supposed to take place in this case, too, but their effect on the mixture formation is counteracted by that here the adjacent jets expanding intersect each other, causing a local fuel enrichment, impairing the air ratio, so that fuel burns with intensive soot formation.

Otherwise, this soot formation was clearly observed in 10 and 12 jet nozzle measurements.

4.3 An attempt to bring about orderly air flow

Using the classical approach to mixture formation and combustion as a starting point, orderly air flow was brought about in the engine cylinders with a moderate screening of the suction valve.

The main characteristics of the screening were:

— the intensive spin effect started in the piston position of approximately 50 crankshaft degrees before the lower dead-point of the induction stroke and lasted to the closure of the suction valve;

— the spin direction was opposite in the two cases;

— the circumferential speed of the new air supercharge, calculated with approximations, was 70 m/sec in the vicinity of the cylinder shell envelope.

The experiments failed, namely:

— the engine smoked intensively,

— the effective efficiency dropped by 1.5%, from 37.1% to 35.6%.

This negative effect argues against promoting mixture formation by means of an orderly air flow in cylinders of medium-speed direct injection supercharge marine Diesel engines, with a macro-mixture number of 6.

4.4. Test results after fluid mechanic modifications of the engine

Fluid mechanic affected

— the suction and exhaust valves,

— the exhaust pipe system and
— the supercharge air collector.

4.4.1. Fluid mechanic improvement of the exhaust valve; testing the unified suction and exhaust valve

Taking test results under 3.5. into account, a new unified suction and exhaust valve chamber with better characteristics has been developed. At the same time the valve has been lowered as much as possible. The new unified suction and exhaust valve and the corresponding average rate relations are seen in Figs 6 and 7.

The fluid mechanic improvement of the valve chamber and the lowering of the valve resulted in a substantial decrease in speed. Besides, the average flow rate got approximately stabilized.

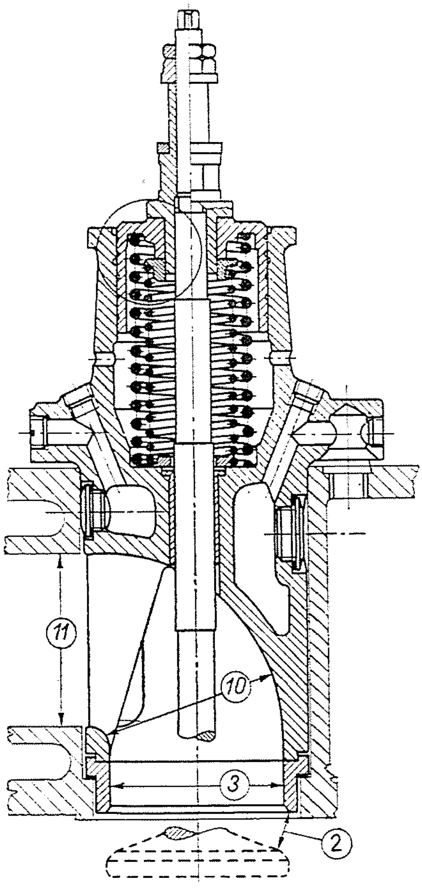

Fig. 6. The new unified suction and exhaust valve

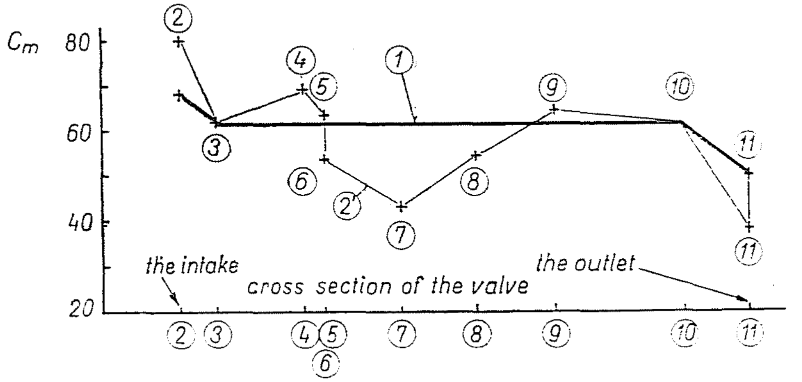

Fig. 7. The average flow rate relations of the new unified suction and exhaust valve at a mean piston speed of $c_m = 6$ m/sec (curve 1). Curve shows the rate relations of the original valve

The application of the unified valves eliminated the penetration of combustion products from the combustion chamber in the direction of the suction valves. The suction valves and the supercharge air collector remained completely clean and dry even in continuous operation.

4.4.2. The modified exhaust pipe system

Fluid mechanical improvement of the original exhaust pipe system eliminated bulging of the caulkings at the flanged pipe connections. The pipe at the flanges in the direction of got slightly evaded to prevent the gas flow from hitting a perpendicular surface in the case of an eventual eccentricity. The extensibility of expansion pieces was set to the necessary minimum. Each pipe nipple of the supercharger gas trap was supplied with a diffuser developed with optimum cone angle, making altogether four. The welded sectional pipes were replaced by bent ones.

Major fluid mechanic modifications of the exhaust pipe system:
— the insertion of seven modified expansion pieces, and
— the insertion of four diffusers with optimum cone angle.

4.4.3. Simplified supercharge air collector of increased cross-section

— Availing of the constructional dimensions of the engine, the cross-section of the original supercharge air collector could be increased by more than 50%.
— The intricate welded, tapered suction pipes got replaced by simple suction rims with burnished surfaces, with 20 mm rounding-off radius.
— The speed of the air arriving from the delivery pipe of the supercharger compressor was reduced to the degree required by the new air collector by a diffuser with an optimum cone angle.

— The suction L-pipes welded of sections were replaced by light-metal L-pipes cast smooth sides for conveying the air into the suction channels of the cylinder heads.

The increasing of the cross-section of the supercharge air collector decreased the average axial flow speed of the air in the pipe. This new speed distribution is seen in Fig. 8.

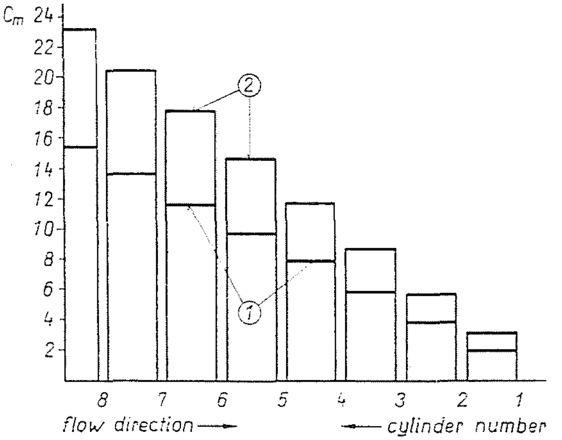

Fig. 8. Average axial speed distribution in the supercharge air collector with an increased cross-section, curve, confronted with the original speed relations, curve 2. Load: $P_e = 736$ kW; $n = 400$ r.p.m.

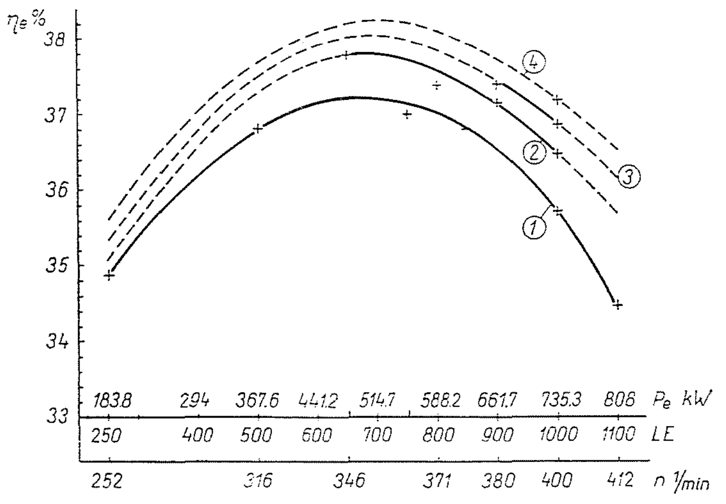

Fig. 9. Effect of different fluid mechanical modifications on the effective efficiency of the engine

4.5. The effect of fluid mechanic improvements on the effective efficiency of the engine

The component modifications under 4.4.1., 4.4.2 and 4.4.3. have improved the effective efficiency of the engine (Fig. 9).

— Test results of nozzle 2 have been plotted in curve 1;
— Joint test results of nozzle 2 and the unified suction and exhaust valve chamber have been plotted in curve 2;
— Joint test results of nozzle 2, the unified suction and exhaust valve chamber and the new exhaust pipe system have been plotted in curve 3;
— Joint test results of nozzle 2, the unified suction and exhaust valve chamber, the new exhaust pipe system and the new supercharge air collector have been plotted in curve 4.

Plots in curves 1, 2, 3 and 4 for a power of 736 kW have been averaged from 6 measurements each.

Marked plots in curves 2, 3 and 4 have been averaged from two measurements each.

Curve limbs in dark line are based in estimates.

4.6. The advantageous effects of applying a Hesselman combustion chamber

The simplification of the complicated combustion chamber form in Fig. 1 could be achieved by eliminating the 1 mm annular groove slit on the upper part of the original cylinder-liner barrel. For experimental purposes a Hessel-

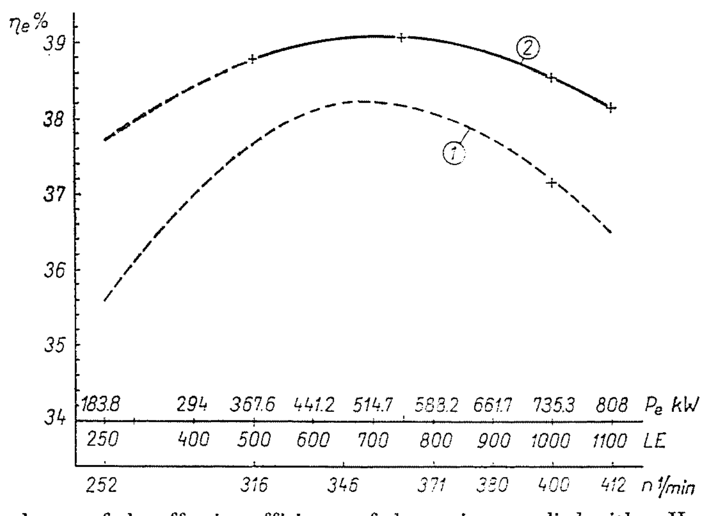

Fig. 10. The change of the effective efficiency of the engine supplied with a Hesselman combustion chamber (curve 2) points to the significant difference in character between curves 2 and 1. Namely in curve 2, load variations either direction from, the optimum zone impair effective efficiencies substantially less, than in the case of curve 1. The differences between the two curves are rather marked

man combustion chamber has been developed. Its application permitted to accommodate some 45% of the compression volume in the piston crown. This solution has resulted in substantially fewer fuel particles getting into the "cold" cylinder-liner wall, which is likely to improve the effective efficiency.

The measurements taken on an engine with a Hesselman combustion chamber have led to results beyond all expectations.

The change of the effective efficiency of the engine with a Hesselman combustion chamber is seen in Fig. 10. points to the significant difference in character between curves 2 and 1. Namely in curve 2, load variations either direction from, the optimum zone impair effective efficiencies substantially less, than in the case of curve 1. The differences between the two curves are rather marked.

The effective efficiency values corresponding to permanent operation load (100%), have been averaged from six measurements each. The other plots are averages of two measurements each. The clash line is based on estimates.

4.7. Some further advantageous effects of the fluid and combustion mechanic improvements of the engine

— Comparison of the weak spring indicator diagrams recorded unambiguously pointed to the improvement of the gas exchange performance of the engine with fluid mechanic modifications. The purity of the fresh supercharge air portion has increased, the initial pressure and initial temperature of the compression have decreased, and so did the average pressure and the average temperature of the cycle.

Measurements of the total volume of air passing through the engine showed the scavenging air volume to grow accompanied by about 4% temperature decrease of exhaust gases.

— Combustional modifications practically eliminated soot formation. Soot level recorder gave zero. The formation of carbon deposit has been minimized. The exhaust temperature of the engine has fallen by at least 6%.

— The combined modification effects increased the combustion air ratio of the engine from1.8 to 2.2 according to approximative calculations. This growth, as well as the decrease in the average pressure and average temperature of the cycle reduced the heat load on the engine hot parts.

— The elimination of the complexity of the combustion chamber and the application of the new piston type have decreased the specific oil consumption of the engine from 5 g/kW$\times$h to 1.4 g/kW$\times$h.

The elimination of soot formation and the minimization of the carbon deposit permitted to substantially increase the oil change intervals.

— The improvement of the effective efficiency of the engine involved the expansion of the travelling range of the ship, permitting, at the same time, refuelling in domestic, rather than in foreign, ports.

Summary

Fluid and combustion mechanic alterations on the 8LD 315 RF type ship master engine — except the implementation of orderly air flow — have resulted in an evident and properly documentable improvement in the effective efficiency of the engine. The tests suggest that modifications made in order to improve the effective efficiency up-to-date engines seldom produce spectacular results such as, those of e.g. the Hesselman combustion chamber. The majority of the modifications are likely to entail modest if not negative results, such as the realization of orderly air flow in the engine cylinder.

Since the measurement range covered most part of the load range, the results may lead to some specific conclusions, which would take the form of expectations, or trends if approached theoretically.

The endeavours to improve the effective efficiency cannot do without wide-range theoretical investigations, which will be facilitated by the measurement results presented.

These measurement results can be made good use of with engines under design or already operating.

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Pál LOSONCI (Lojek) H-1521 Budapest