BHP

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EVALUATION OF SEA TRIAL RESULTS

GENERAL ACCEPTANCE CRITERIA FOR SEA TRIALS

THere are usually three basic requirements in a Shipbuilding Contract to be confirmed prior to the delivery of a new ship to her Owners.

(1) Deadweight carrying capacity of the ship.
(2) Fuel Consumption of the Main Engine.
(3) Ships Speed.

Of these three items, only two will be dealt with in this paper, Fuel Consumption of the Main Engine and Ships Speed in relation to the main engine propeller curve developed at the engine trials and its relationship to the propeller design curve and the final development of the power/ships speed curves from sea trials and their evaluation.

OLD ACCEPTED CRITERIA FOR SEA TRIALS/SHIPS

In todays fuel consceious environment, ship owners and operators have now changed their requirements from ships maximum speed to economical speed. In the past a shipbuyer's design criteria would stipulate a ship service speed, D.W.T., proposed dimensions and name the best engine (as their Company saw it) without overdue regard to fuel costs. Fuel costs prior to 1972 world fuel crisis were as low as US$16.00 per ton for heavy fuel oil, whereas now it is no longer surprising to pay US$220.00 per tonne on board.

In the days when high speed was one of the main criteria, with the necessary sea margin built into the propeller design it was not unusual to see cathedral diesel engines sold with a specific fuel consumption of 154 gms/BHP/Hr or above. Imagine if an engine builder now tried to sell a slow speed two stroke cathedral diesel engine with this specific fuel consumption.

TODAYS ENGINES WITH FUEL ECONOMY PRESSURES

The fuel economy pressures placed upon the marine industry have had numerous consequences towards economy, from which two major developments have been produced.

(a) Diesel Engine Builders have developed engines with a markedly better fuel economy.

(b) The Shipowner and operators of ships have reduced ships speed thereby reducing fuel consumption.

Initially with (b) some Shipowners experienced some disastrous side effects with slow speed running such as excessive cylinder wear when full power was reintroduced and turbo chargers overspeeding due to carbon build-up and fires with disastrous results, which had the effect of putting pressure on the engine builders to build more fuel efficient and versatile engines. This reduced speed, and fuel economy running put pressure on the Shipowner/Operator to fully evaluate the utilisation of their ships in their trades and in addition to better plan schedules so as to cut lay-time to a minimum thereby saving fuel.

PRACTICAL EXAMPLE OF ECONOMIC DESIGN

Our Company, by critically evaluating the utilisation factor of our ships in our ore trade and the lay days off the discharge ports coupled together with their increased carrying capacity and our ore requirements arrived at the following design/specification criteria in conjuction with the Shipbuilder.

Bulkcarrier:

D.W.T.	:	140500 tonnes
Service Speed	:	14.25 knots (76.5% of MCR)
Engine Type	:	Sulzer 6RLA90 (20400 BHP metric at 90 RPM)
Engine RPM	:	82.3
B.H.P. required	:	15606 (metric)
Consumption/Day	:	54 tonnes (Based (140 + 3%) grms/BHP/hr)

For comparative the design/specification criteria of a ship designed two years earlier by the same Shipbuilder is listed.

Bulkcarrier

D.W.T.	:	106000 tonnes
Service Speed	:	15.3 knots (76.5% MCR approx)
Engine Type	:	Sulzer 7RND-90M(23450) BHP metric at 122 RPM
Engine RPM	:	111.6
BHP Required	:	17939
Consumption Per Day	:	66 tonnes/day (Based (149+3% grms/BHP/hr)

From the above it can be seen that 34,500 tonnes of extra cargo can be carried at 1.05 knots slower with a fuel saving of 12 tonnes per day. Quite a difference in fuel consumption. This has been achieved by:

    (a) slower ships speed

    (b) improved hull design

    (c) slower propeller speed

    (d) improved engine efficiency

A greater fuel saving could have been achieved had the required ships speed been reduced even further, say to 12 knots, however this further reduction would have put excessive pressure on our scheduling and raw material requirements.

FROM DESIGN CRITERIA TO FINAL SHIP SPECIFICATION AND APPROVED DRAWINGS

During the transition from the buyers Design Criteria to the mutually agreed contract specification between the buyer and the builder a proposed horse power/ship speed curve was developed by the builder. In the case of a bulkcarrier, two curves are developed, one is the ballasted sea trial condition and one is the estimated fully loaded condition. The reason for this is that a bulkcarrier cannot be fully loaded on sea trials, as with a tanker, so an agreed ballasted D.W.T. condition is calculated to give the ship a reasonably level trim and a good propeller immersion factor so as to allow a representative speed curve to be developed during sea trials.

Usually most Shipbuilders have criteria already programmed into their computers, based upon previous trial results, which allows them to produce a relatively close speed curve during Specification discussions. However, as an additional check, model tank tests are conducted at this ballast draft to confirm the horse power required and to allow the Shipbuilder to develop his propeller design to achieve the agreed ships performance.

In the figures attached, the four curves that were obtained from the tank test data are shown for our latest 140,500 D.W.T. bulkcarrier.

They are:

Fig.1   Propeller Speed/Horse Power Curve

Fig.2   Horse Power/Ships Speed from Tank Test

Fig.3   Estimated Horse Power/Ships Speed Curve Full Loaded
         at 140500 tonnes

Fig.4   M.E. Power – Propeller RPM Curve from Tank Test
         85590 tonnes

At this juncture it is interesting to recall that during our discussions we asked the Builders how they were going to confirm the fully loaded ships speed, horse power requirements, propeller speed and fuel consumption which they guaranteed to achieve. This question brought to light the actual difference from the old school of maximum ships speed/service speed to the current concept of fuel economy/reduced service speed.

The Builders reply was that it would be by the old acceptance method, speed trials carried out at 90% MCR at the ballasted speed trial condition as compared to the tank test developed curve and then these results related to the estimated fully loaded power speed curve. Initially this was not acceptable to us as the fuel economy was now the major factor coupled to the ships reduced speed requirement. We endeavoured to negotiate to carry out confirmation speed trials after the ship was fully loaded on her maiden voyage but this was strongly resisted. It developed that the only way to carry out a fully loaded test would have been to do speed trials with partially ballasted cargo holds which was deemed too dangerous. The final solution was to accept the prescribed method, ballasted speed trial curve transferred to the estimated full loaded condition based on tank test data.

This problem may set minds to work in the future as fuel economy becomes more paramount in the acceptance of a ship and becomes a far more important selling point by the Builders, both engine and ship.

DEVELOPMENT OF CURVES DURING THE BUILDING AND SEA TRIALS OF OUR LATEST 140,000 D.W.T. BULKCARRIER

The first official curves that developed were from the Main Engine shop trials, which were:

Fig.5.   The Main Engine Propeller Law Curve

Fig.6   Main Engine Performance Curve from Shop Trial

Fig.7   Load Indicator - BPME Curve

Fig.8   P Max - Load Indicator Curve

To fully explain the meaning of Fig.5 and Fig.6 please refer to Figs. 9 and 10 which are Sulzer's diagrams with explanations. These curves are applicable to all two stroke diesel engines.

During the Official Shop Trial of the Main Engine the acceptance of the Contractual Specific Fuel Oil Consumption was conducted. The Contract figure was 140 gms/BHP(metric)/hour $\pm$ 3% to be carried out at 90% MCR. The official result was 138.6 gms/BHP/hour corrected to 10,200 KCal/kg of fuel.

It may be interesting to recall at this stage that during internal Company discussions the choice was narrowed to two main engines, Sulzer and Burmeister and Wain. Discussions involving direct participation by Sales Engineers from both engine manufacturers indicated that the methods of calculating specific fuel consumption were inconsistant and not comparable. As a result of these discussions we requested that both engine manufacturers quote the consumption figures based on International Standard Organisation (I.S.O.) parameters, so as a proper comparison could be made. We now note that some engine manufacturers have reverted to the old method of quoting consumption figures in their own parameters which can give fuel consumption advantages i.e. claiming better consumption figures.

The final selection made by the Company was for a Sulzer RLA90 engine. The reason for this selection, in itself, could be the subject of another paper, so the details will not be discussed here.

The next set of curves to be developed were those from the Official Acceptance Sea Trials. These trials were conducted in the inland sea of Japan and the major item was the ships speed trials. These speed trials consisted of two x 1 nautical mile runs in each direction at each of the following horse powers 50% of MCR, 75% of MCR, 90% of MCR and 100% of MCR.

In the design of the propeller the shipyards usually designs about 3 to 6% sea margin into their propellers, in other words they select the 90% MCR/97% R.P.M. as their reference point and they develop their propeller law curve so as to pass through that point and the initial curve is then generated from the tank tests.

With reference to Fig.4 the propeller speeds were determined for the speed runs as follows:

	B.H.P.(Metric)	(Propeller R.P.M.)
50% MCR	10,200	74.5
75% MCR	15,300	85.2
90% MCR	18,360	90.8
100% MCR	20,400	93.8

The contractual guaranteed ship speed was 15.9 knots (-3/10 knots without penalty) ballasted to approximately 86,000 D.W.T. for trials run at normal operating rating (N.O.R.) which was 18,360 B.H.P.)

To obtain the ship speed at these revolutions, two measured mile runs in each direction were carried out using a radio wave measuring device while the designated R.P.M. were maintained.

SEA TRIAL SPEED RESULTS AND EVALUATION

% MCR	B.H.P.	Propeller RPM	Ships Speed
50%	10,328	74.38	13.49 knots
75%	15,327	85.36	15.31 knots
90%	18,464	90.57	16.02 knots
100%	20,750	93.89	16.05 knots

It can be seen from these results of the runs at 90% M.C.R. that the ship obtains her guaranteed speed with some to spare (16.02 knots).

In the case of the 100% MCR run, the maximum revolutions allowable were, for this engine, 99 RPM (110% of revs) or the maximum average horse power 20,400 B.H.P., whichever was attained first.

This run at 100% MCR was carried out to give the maximum ships speed capability, and it is also required that there be a minimum of three different speed runs to draw a representative ships speed curve.
Also conducted during the sea trials was a 2 hour H.F.O. consumption endurance test at 90% M.C.R. This test is basically done to determine a practical Fuel Oil Consumption rate for the ships reference and also to test the H.F.O. heating and purification equipment. The run of H.F.O. showed a Specific Fuel Consumption of 138.08 gms/BHP/Hr corrected to 10,200 Kcal/kg of fuel.

From the speed trials the horse power/ships speed curve, Fig.11 was plotted.

In addition the Main Engine Performance Curves Fig.12 also the load indicator - BPME curve points were plotted on Fig.7 and on the P Max - Load Indicator Curve Fig.13.

From Figures 6, 12, 7, 13 and 8 (combined) it can be seen how the L.I. (Load Indicator) at the fuel pumps has been reduced due to the propeller design (refer to the dotted lines showing the 76.5% MCR Line for the Maine Engine shop test to the 76.5% MCR Line for the Sea Trial). You will note that at the shop test the L.I. position was 6.3 and at the sea trial it was 6.1.

Today there still appears to be some confusion with some operators regarding the load indicator position in relation to its position at engine trials and sea trials for similar horse power developed by the engine.

Also from the sea trial results the M.E. Power/Propeller R.P.M. curve Fig.14 was plotted.

It can be seen from the M.E. Power - Propeller RPM Curves Fig.14 and 4 and Propeller/Power Curves Figs. 1,2,3,11,15 & 16 that the ship had lost a little on the designed sea margin as the propeller absorbed a little more torque than originally designed (torque richer). Designed RPM at 76.5% MCR was 86.13 RPM. From these graphs it can be seen that at 85.86 RPM the ships speed has increased from the design of 14.25 knots to 14.45 knots.

To gauge the loss of Sea Margin refer to Figures 4 and 14 "Propeller Performance" which shows from the design of the propeller, to the sea trial result, an approximate loss of 9.3% of the designed engine speed margin which represents 0.3% of the propeller speed (loss) however from these two graphs it can be calculated that the sea margin available at the 76.5% of MCR (Sea Trial Results) is approximately 2000 horse power on the constant propeller speed line of 85.86 RPM.

To fully evaluate these results refer to Sulzer's graphs Figs. 9 and 10.

EVALUATION OF GRAPHS OF MAIN ENGINE PERFORMANCE DURING
SPEED TRIALS

With reference to Fig.13, P Max - Load Indicator Curve and relation to the Shop Trial curve Fig.8, it can be seen that the Main Engine Maximum Pressure was 120 kg/cm^2 on Sea Trials where as on Engine Trial was recorded at 112 kg/cm^2. The explanation of this is quite complex as there were a lot of contributing factors.

The RLA90 Sulzer engine P Max is rated at a maximum pressure of 110 kg/cm^2. As can be seen from Fig.8, that the shop trial P Max curve is acceptable, however the curve developed at sea trials is unsatisfactory in that it indicates that the engine went into overload (excessive P max) at 18,000 B.H.P., that is the P Max exceeded 110 kg/cm^2 at that point.

For reference, the new Sulzer RLB90 engine is rated at 120 kg/cm^2 P max and the only difference between the two engines is the engine speed and horse power rating per cylinder which is 102 RPM and 4000 BHP for the RLB as compared to 90 RPM and 3,400 BHP for the RLA. Informatively the RLA90 comes in a 98 RPM version of 3,800 BHP per cylinder.

Referring to Fig.7 Load Indicator - BPME curve, you will note that the curve is constant for Shop Trials and Sea Trials, that is the engine developed the same torque at engine trial and sea trials throughout its whole range of operation. When curves Fig.13 and 7 are viewed together it shows that the cards on sea trials were more "peaky", however, the same power/torque was developed.

The basic reason for this occurrence is that at shop trials the load is constant by means of the Dynamometer, however, at sea trials it is forever varying, from one speed run to the next.

The increased P Max and horse power can be caused by the difference in the hull characteristics on sea trials to the estimated results from the hull used in the tank tests. This can take the form of increased frictional losses of the hull and propeller and is borne out on sea trials by the increased B.H.P. required at the set propeller RPM. This increased friction can be caused by (a) plate rippling, (b) weld reinforcement beads on the hull (c) increased surface roughness of the underwater paint film (d) surface condition of the propeller.

(Design criteria in todays specification should call for controlled hull form roughness of the underwater painting and in pursuit of economy it may be considered necessary that all weld beads and seams under water should be ground off so as to decrease underwater roughness...)

The consequence of (a), (b),(c) and (d) above are as follows:-

As the ships hull roughness increases it causes a drop in ships hull speed, if constant horse power is maintained. This drop in hull speed causes a drop in velocity of the water behind the ship (wake) which effects a change of the water flow into the propeller blade profile and the engine speed, in consequence, is throttled. The hull roughness, in addition, also causes the boundary layer of the water flow to increase in thickness adjacent to the hull which again effects the wake velocity with the same consequence.

Also an increase in the roughness of the propeller blades reduces the propeller efficiency which again throttles the engine speed.

The two conditions as described above have the same effect on the main engine if a constant engine speed is maintained, this is a high torque requirement,which in turn leads to increased BMEP and P Max of the engine.

If torque becomes the constant value then there would be a fall off in revolutions of the propeller, a reduced propeller efficiency and an even bigger drop in the ships speed than would be experienced with constant engine RPM.

With the above explanations as to the reasons for the increase in the torque requirements of the main engine at constant RPM on sea trials we will endeavour to show why the main engine suffered an increase in the P Max as compared to the P Max obtained on the engine trial.

To obtain the ships hull speed readings of the different horsepowers of 50, 75, 90 and 100% of M.C.R. the ship did 2 measured runs in each direction at each set of engine R.P.M., 2 runs against the weather (tide, wind etc.) and 2 runs with the weather. At the end of each run the ship executed a 180 degree turn to do the run in the opposite direction over the same course. Prior to each run over the course, the ship was scheduled to have a good "run up" so as to ensure that hull maximum speed was achieved and that the main engine load had settled down.

At shop trials as well as the engine having a constant load at each setting, the ambient air conditions were relatively constant. During the sea trials the ambient air conditions were always varying, i.e. wind, sea, water temperature, water depth and air temperature, hence the reason for 4 runs to obtain the mean speed. During trial runs, indicator cards were only taken once, that is before one of the speed runs during the time when the engine was settling down to as constant as possible, revolutions and torque loading. Cards were not taken during speed runs so as to ensure no outside unwarranted influences effected the engine. It can now be seen that only 4 indicated horsepower readings (cards) were taken for 16 speed runs or 1 set of cards for each successive 4 speed runs at each designated R.P.M. setting. The resultant P Max sea trial curve therefore is only based on a single fixed reading at a fixed point in time, that is, when the engine was in fact settling down from the turn and coming into the next speed run. At this point the B.H.P. being developed would have been in excess of the actual recorded B.H.P., by the torsion meter for the run and so for these reasons the P Max would also be up. In addition to these factors increasing the P Max, the cards were also taken each time when the ship was running against the weather which again further increased the P Max and B.H.P. This is illustrated for the (100% MCR) 93.8 RPM speed run when the B.H.P. measured by the torque meter was 20,924 BHP, the indicated horse power was measured by cards as 23,249 and at 90% efficiency was calculated at 20,853 B.H.P.

The average horse power developed at the 93.8 R.P.M. speed runs was 20,750 B.H.P. This increased B.H.P., unfavourable sea and weather conditions, insufficient run up to the course all contributed to the P Max of 120 Kg/cm^2 and in addition, the air temperature also had its effect of increasing the P Max. Sea Trial ambient air temperature was $7^{o}C$ and at shop trials was $22^{o}C$ with turbo charger inlet temperature of 17° and 29° respectively. This temperature difference allowed more dense air to be delivered into the engine. It may be advisable to make note at this point that P Max is not directly proportional to the B.M.E.P. or torque. P Max can be varied by injection timing, rate of burning of the fuel, air density etc.

Had the Shipbuilders taken care before each successive speed run and taken the average P Max, I am sure the sea trials P Max Curve Fig.13 would have been far closer to the shop trial P Max Curve Fig.8. This can be born out when you again assess Fig.7 load indicator/BPME curve as stated before.

From the aforesaid and evaluating all the curves developed, the ship, main engine and the propeller achieved the contract criteria.

FUTURE DESIGN FOR INTER-RELATIONSHIP OF MAIN ENGINE AND THE PROPELLER

As stated earlier, under the old criteria "Ships Maximum Speed" propellers were designed so that full horse power could be developed on sea trials so that a maximum value of speed could be obtained.

This can be fully illustrated by a Sulzer statement quote "In the interest of all parties (Owners, Shipyards and engine manufacturers), the propeller should be designed in such a way that the engine can produce full horse power at any time. This is the case when the normal speed (= 100% of revs) is obtained without exceeding the maximum admissable torque and thus the normal brake mean effective pressure (= 100% Pe) of the engine. It is only this way that the engine can develop its guaranteed maximum continuous power (= 100% Pe) during service and the propeller will then meet the engine manufacturers' requirements.

In all such cases, where the described criteria are unknown, we recommend that the propeller be designed in such a way that it absorbs 85-90% of the normal power at normal speed". Unquote.

This particular extract from Sulzers' Technical Bulletin 25.1 in my opinion, adequately displays the Full Engine Speed/Full Torque principle that the Shipbuilder and propeller designers are still adopting. Graphic Illustration of this is shown in Fig.9.

With the ever increasing utilisation of methods of fuel economy, engine builders have now developed many ways to improve economy and one of these now is not to operate at MCR or NOR but to operate the engine further back along the Specific Fuel Consumption/BHP Curve to where the most efficient operation of the engine occurs see Fig.17.

From this curve one could say that the engine has its most efficient fuel consumption at 85% of MCR, however Sulzer in its quest for more economical engines have developed variable injection timing (V.I.T.) and with this facility they are able to develop their M.C.R. - P Max from 85% of BHP to 100% of BHP. See Fig.19. They have now also developed their ERG curve or their Economy Rating Generator Law. See Fig.20.

You will be able to see by this development of 85% MCR and 100% n that the engine designers are gradually moving away from their own initial design criteria to the requirement of the Shipowner /Operator for economy and versatility of the engine. When using the ERG law Sulzer state: Quote: "This relatively elevated revolution version can be interesting for vessels with restricted draughts, which do not allow the use of a very large slow running propeller". Unquote.

In my opinion this statement is somewhat misleading. If one divorces oneself away from the thinking of developing the Maximum Horse Power per cylinder and directs it to the most efficient way of developing the horse power required to produce the torque at the propeller, that is a marrying of ideas between the engine builder and the naval architect to the Shipowner/Operators requirement, we will have achieved today's requirements, efficiency and economy.

In other words, the design programme should take the following form, first the hull dimensions and required hull speed, second the best possible efficient propeller and propeller speed to suit the hull etc., third the engine required tomeet these requirements and today this concept is possible with modern day engines.

In all our diesel powered ships which we have built in recent years, the propellers have been designed to the Propeller Law applicable to two stroke diesel engine i.e. 90% of B.H.P. and 100% of revs., as shown in Fig.9. As a matter of fact nearly all our diesel ships we have on charter or purchased have been designed to this criteria, which only gives us an approximate 10% horse power and a 2½% RPM sea margin.

Discussions have been held with Shipbuilders with a view to having them increase this sea margin along the following lines. Please refer to Fig.18.
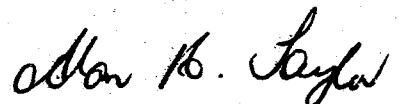
As previously stated when the ship is new we base our operation of the ship at 76.5% M.C.R. see from Fig.4., i.e. 15606 BHP or 85.86 RPM (95.4% n). If we had used the Curve Fig.18, again based on the torque required at 76.5% MCR the propeller speed required would have been 90 RPM (100% n).

If this propeller design curve were used, it would have been possible to achieve Maximum horse power on Sea Trials as the engine manufacturer allows this engine to operate up to 110% of revs., or 99 RPM. At 98 RPM with this propeller the engine would have developed its M.C.R. However if in some instances, the revolutions required, do exceed manufacturers allowances and the 100% M.C.R. cannot be achieved on sea trials, this in itself is of no consequence as the engine achieves its full horse power capabilities at the engine shop trials. By using the 76.5% MCR propeller law curve it does allow us to operate the engine at its designed RPM and at the designed horse power requirement of the propeller. If this curve Fig.18 had been used in the design, it would have given us a better P Max – L.I./horse power curve closer to that curve which was obtained at shop trials (Fig.8), however the best gain is that it would give us a 23.53% sea margin on horse power up to MCR or a 13.5% horse power margin up to N.O.R. at a 90 RPM constant propeller speed. However, if when operated at the required RPM to obtain the desired ships speed it would also give us an 8.5% propeller speed margin. The specific fuel consumption will remain the same as the BHP remains constant at sea trial condition.

In addition to these benefits, in future tonnage with the availability of Variable Injection Time (V.I.T.) on Sulzer Engines and the Area of Maximum P (max) allowable with the B & W Engines, whereby at reduced horse powers the P Max remains constant at the engines 100% MCR value with a reduced B.M.E.P., the fuel savings of 4 to 5 gms per BHP per hour are realistic.

This thinking will undoubtedly give rise to some long and extended arguments between the naval architects and the marine engineers, however if we look at it from the overall approach, the Shipowners/ Operators point of view it means economy of operation and the possibility of longer periods between overhauls by not running the engine into overload. In addition it asks a basic question, did we really need all the horse power that we have been sold in the past.

In closing I hope that the aforesaid ideas, evaluations and experiences will assist in future evaluation of sea trials results and operational efficiencies.

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AHT:CDF:

12.2.82

S.No. 2731 PROP. v POWER CURVE

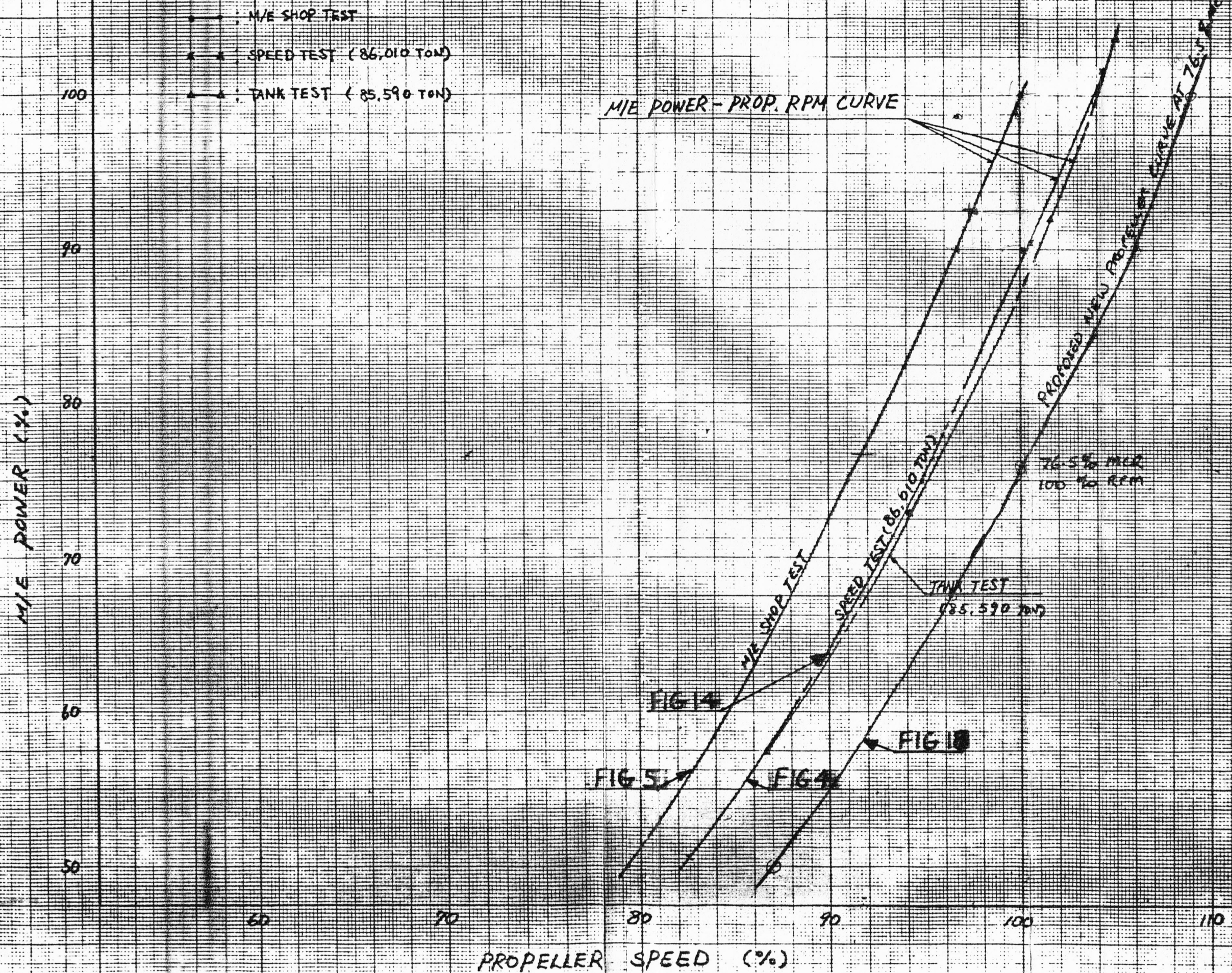FIG 1 — TANK TEST (85,590 TON)
SPEED TRIAL (86,610 TON) — FIG 15

ESTIMATED BY TANK TEST RESULT (FULLY LOADED)
ESTIMATED BY SEA TRIAL RESULT (FULLY LOADED)

TANK TEST RESULT (85,590 TON)
SEA TRIAL RESULT (86,610 TON)

FIG 16   FIG 3

FIG 7   FIG 11

M/E BRAKE HORSE POWER (PS)

PROP. SPEED (RPM)

SHIP SPEED (KNOTS)

JIS A3 SIZE 280mm×400mm   ★star

SNo. 2731   PROPELLER PEFORMANCE

Legend:
- M/E SHOP TEST
- SPEED TEST (86,010 TON)
- TANK TEST (85,590 TON)

MIE POWER - PROP. RPM CURVE

PROPOSED NEW PROPELLER CURVE AT 76.5% MCR

76.5% MCR
100 % RPM

M/E SHOP TEST

SPEED TEST (86,010 TON)

TANK TEST
(85,590 TON)

FIG 14

FIG 18

FIG 5   FIG 4

Y-axis: M/E POWER (%) — 50, 60, 70, 80, 90, 100, 110

X-axis: PROPELLER SPEED (%) — 60, 70, 80, 90, 100, 110

SW2731 M/E PERFORMANCE CURVE

SHOP TEST
SPEED TEST
ENDURANCE TEST

MAX. COMPRESSION PRESS. Kg/cm²

MEAN EFFECTIVE PRESSURE Kg/cm² %

(BPME)

100% MCR

ENDURANCE TEST

ENGINE REVOLUTION %

LOAD INDICATOR AT PUMP

FIG 6

FIG 12

FIG 7

FIG 8

FIG 13

RPM-BPME CURVE SHOP TEST

BPME-BPME CURVE SPEED TEST

LOAD INDICATOR-BPME CURVE

JIS A3 SIZE (280mm×400mm)

% Pe (% ENGINE POWER)

110 — Nominal Power — 100

M.C.R.

100 — Nominal Power

90

85

80

(A₁)

(B)

(A)

(C)

Nominal Speed

100% pe

90% pe

85% pe

p

70

60

50

75 80 85 90 95 100 103 105   **% r**

% ENGINE SPEED

A   : Optimum range for continuous operation.
A₁   : Range for engine characteristic on sea trial with fair weather, ship fully laden and clean hull.
B   : Working range for restricted time only (max. 2000 hours).
C   : Upper speed range for sea trial only.
p   : Engine characteristic on shop trial, i.e. approximated propeller curve through the point of M.C.R. Although the engine is capable of working in range B for restricted time, the aim should be to design the propeller in such a way that curve p is not exceeded in continuous service.
t   : Limitation of range B.
M.C.R.: Maximum Continuous Rating, i.e. Nominal Power (100% Pe) at Nominal Engine Speed (100% n)
pe   : Brake Mean Effective Pressure (≙ torque)

FIG 9
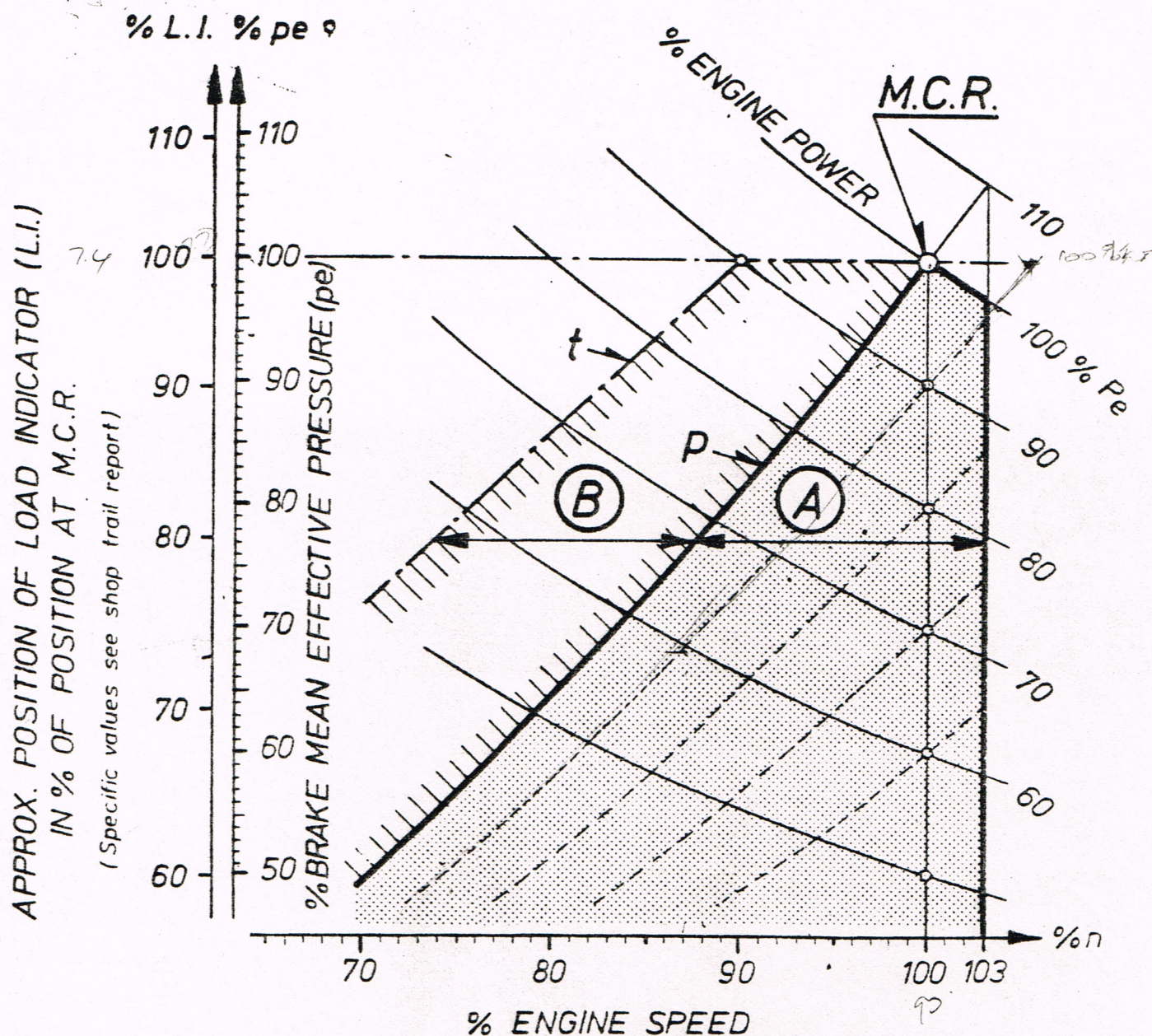
p : Propeller curve through point of M.C.R., i.e. engine
characteristic on shop trial.

t : Limitation of range B.

A : Optimum range for continuous operation.

B : Working range for restricted time only (max. 2000 hours)

M.C.R: Maximum Continuous Rating, i.e. Nominal Power (100%...)
at Nominal Engine Speed (100% rev/min)

FIG 10

Fig. 2 shows the performance curve of a 6RLB90 engine at
its MCR 1 rating, i.e. 2940 kW/cyl. (4000 BHP/cyl.) at 102 rev/
min.

Thanks to the standard VIT-mechanism (variable injection ti-
ming), the specific fuel consumption can be maintained at its
lowest value, i.e. 184 g/kWh (135 g/BHPh) over the most impor-
tant service load range of 80 - 90% (ISO-standard reference conditions, LCV 42 707 kJ/kg (10 200 kcal/kg).

Performance curves of 6RLB90-engine

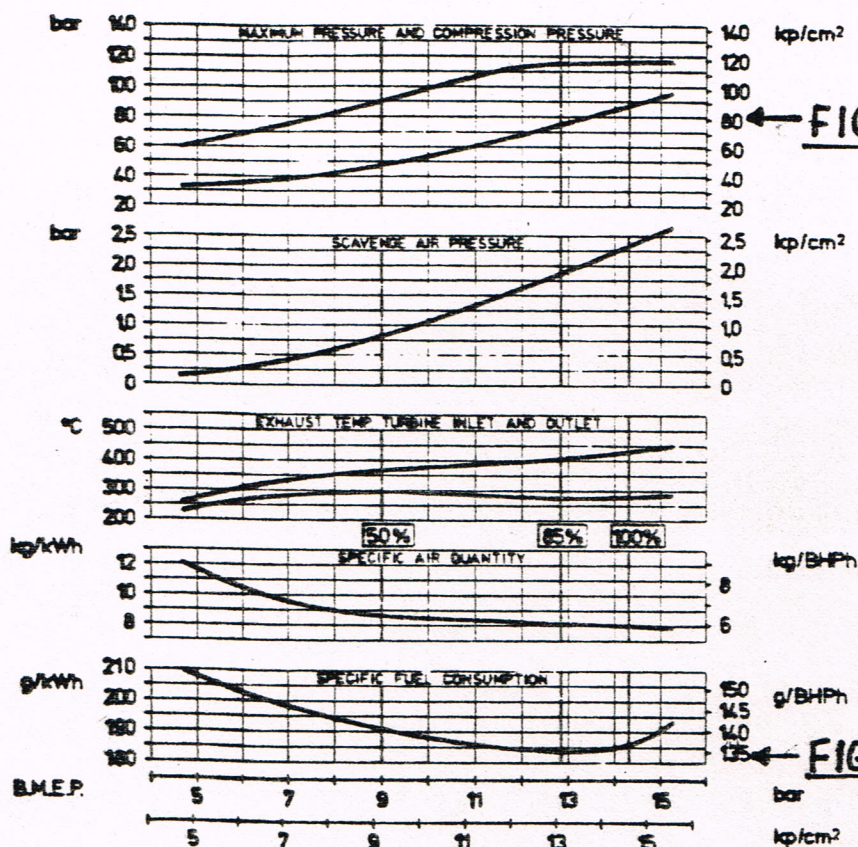### 2940 kW/Cyl.(4000 BHP/Cyl.) at 102 rev/min.
(Propeller law)
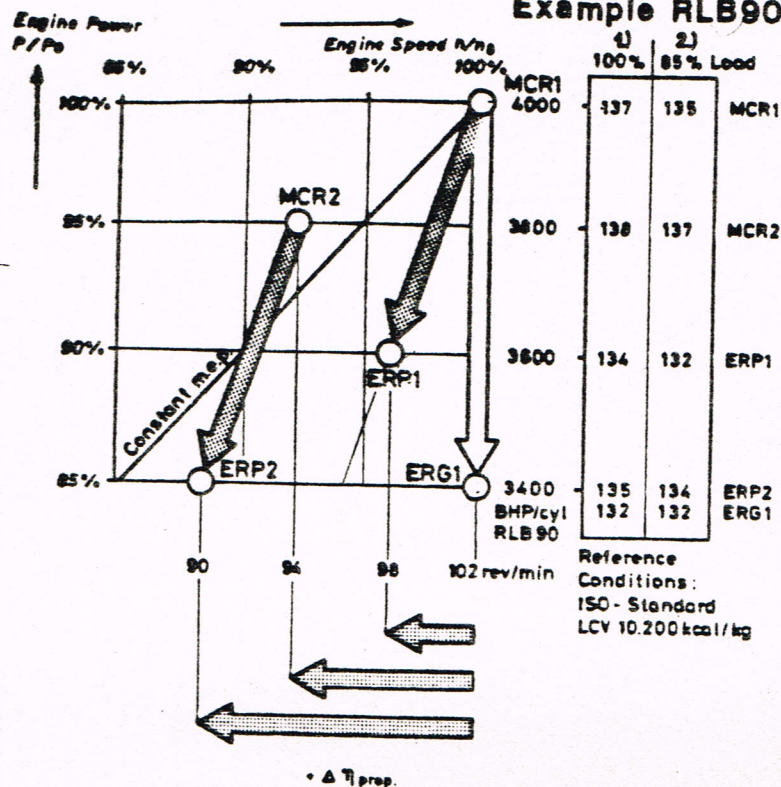

Fig. 2

The lowest possible fuel consumption of RLB90 engines, i.e.

179 g/kWh   (132 g/BHPh)

results from the ERG 1 rating, i.e. 2500 kW/cyl. (3400 BHP/
cyl.) at 102 rev/min. This relatively elevated revolution
version can be interesting for vessels with restricted
draughts, which do not allow the use of a very large, slow
running propeller.

ENGINE LAYOUT   Reduced Specific Fuel Consumption
(BSFC)

Example RLB90

Engine Power P/Po

Engine Speed n/n₀

85%   90%   95%   100%

1.) Specific fuel consumption at 100 % load at the points MCR1, ERP1, ERG1, MCR2 and ERP2.

2.) Specific fuel consumption at reduced load (85 %, propeller resp. generator law) relating to the above mentioned points.

	1) 100%	2) 85% Load	
MCR1 4000	137	135	MCR1
MCR2 3800	138	137	MCR2
ERP1 3600	134	132	ERP1
ERP2 3400	135	134	ERP2
ERG1	132	132	ERG1

BHP/cyl RLB 90

90   94   98   102 rev/min

Reference Conditions: ISO - Standard LCV 10.200 kcal/kg

ΔBSFC - 3.5%

Alternative engine optimizations for 100% maximum combustion pressure at ratings lower than MCR.

Constant mep

+ Δ η prop.

Improved Propulsion Efficiency

Fig. 20

MCR	Maximum Continuous Rating
ERP	Economy Rating Propeller Law
	(P_{ERP}/P_{MCR} = 90%, n_{ERP}/n_{MCR} = 96.5%)
ERG	Economy Rating Generator Law
	(P_{ERG}/P_{MCR} = 85%, n_{ERG}/n_{MCR} = 100%)

The RLB engines are offered each in two MCR and ERP ratings; utilizing a controllable pitch propeller, the ERG1 rating is a further alternative.

Depending on the vessel, the rating with the best cost/benefit ratio with regard to all cost influencing factors, such as capital costs, lowest total fuel consumption of the vessel (- Δ BSFC/+ η prop.), etc. has to be selected.

The propeller has to be layed out according to the Sulzer recommendation, based on the selected nominal rating (MCR or ER).