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Observations during Operation

1. Symbols and Units

The following designations are used:

Parameter	Symbol	Unit 1	Unit 2
Effective engine power	P_e	bhp	kW
Engine speed	speed	speed	speed
Indicated engine power	P_i	ihp	ikW
Fuel pump index	Index	No.	(mm)
Specific fuel oil consumption	SFOC	g/bhph	g/kWh
Fuel oil lower calorific value	LCV	kcal/kg	kJ/kg
Turbocharger speed	T/C speed	speed	speed
Barometric pressure	p_{baro}	mmHg	mbar
Pressure drop across T/C air filters	Δp_f	mmWC	mbar
Pressure drop across air cooler	Δp_c	mmWC	mbar
Scavenge air pressure	p_{scav}	mmHg	bar ⋆)
Mean indicated pressure	p_i	bar ⋆)	bar ⋆)
Mean effective pressure	p_e	bar ⋆)	bar ⋆)
Compression pressure	p_{comp}	bar ⋆)	bar ⋆)
Maximum combustion pressure	p_{max}	bar ⋆)	bar ⋆)
Exhaust receiver pressure	p_{exhrec}	mmHg	bar ⋆)
Pressure after turbine	p_{atc}	mmWC	mbar
Air temperature before T/C filters	t_{inl}	°C	°C
Air temperature before cooler	t_{bcoo}	°C	°C
Cooling water inlet temp., air cooler	$t_{coolinl}$	°C	°C
Cooling water outlet temp., air cooler	$t_{coolout}$	°C	°C
Scavenge air temperature	t_{scav}	°C	°C
Temperature after exhaust valve	t_{exhv}	°C	°C
Temperature before turbine	t_{btc}	°C	°C
Temperature after turbine	t_{atc}	°C	°C

Conversion factors:

1 bar $= 1.02$ kp/cm^2 $= 0.1$ MPa $= 10^5$ Pa $= 10^5 \frac{N}{m^2}$

1 kg/cm^2 $= 0.9807$ bar

1 kW $= 1.3596$ hp

1 mbar $= 10.2$ mmWC $= 0.75$ mmHg

$\pi = 3.14159$

⋆) ***Note:*** *Pressure stated in **bar** is the measured value, i.e. read from an ordinary pressure gauge. Note: the official designation of **bar** is **ABSOLUTE PRESSURE**.*

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2. Operating Range

2.1 Load Diagram

The specific ranges for continuous operation are given in the 'Load Diagrams':

- For propulsion alone, *Plate 70601*.
- For propulsion and main engine driven generator, *Plate 70602.*

2.2 Definitions

The *load diagram*, in logarithmic scales *(Plates 70601 and/or 70602)* defines the power and speed limits for continuous as well as overload operation of an installed engine having a specified MCR point 'M' according to the ship's specification.

The service points of the installed engine incorporate the engine power required for ship propulsion, *see Plate 70601*, and for main engine driven shaft generator, if installed, *see Plate 70602.*

2.3 Limits for Continuous Operation

The continuous service range is limited by four lines:

Line 3: Represents the maximum speed which can be accepted for continuous operation.
Running at low load above 100% of the nominal speed of the engine is, however, to be avoided for extended periods.

Line 4: Represents the limit at which an ample air supply is available for combustion and gives a limitation on the maximum combination of torque and speed.

Line 5: Represents the maximum mean effective pressure (mep) level, which can be accepted for continuous operation.

Line 7: Represents the maximum power line for continuous operation.

2.4 Limits for Overload Operation

Many parameters influence the performance of the engine. Among these is: overloading. The overload service range is limited as follows:

Line 8: Represents the overload operation limitations.

The area between lines 4, 5, 7 and the heavy dotted line 8 is available as overload for limited periods only (1 hour per 12 hours).

2.5 Recommendations

Continuous operation without limitations is allowed only within the area limited by lines 4, 5, 7 and 3 of the load diagram.

The area between lines 4 and 1 is available for running conditions in shallow water, heavy weather and during acceleration, i.e. for non-steady operation without actual time limitation.

After some time in operation, the ship's hull and propeller will be fouled, resulting in heavier running of the propeller, i.e. loading the engine more. The propeller curve will move to the left from line 6 to line 2 and extra power is required for propulsion. The extent of heavy running of the propeller will indicate the need for cleaning the hull and possibly polishing the propeller.

Note: Point A is a 100% speed and power reference point of the load diagram. Point M is normally equal to point A but may in special cases, for example sometimes when a shaft generator is installed, be placed to the right of point A on line 7.

2.6 Propeller Performance

Experience indicates that ships are – to a greater or lesser degree – sensitive to bad weather (especially with heavy waves, and with head winds and seas), sailing in shallow water with high speeds and during acceleration. It is advisable to notice the power/speed combination in the load diagram and to take precautions when approaching the limiting lines.

3. Performance Observations

Plates 70603 (two pages), 70604

3.1 General

During engine operation, several basic parameters need to be checked and evaluated at regular intervals.

The purpose is to follow alterations in:

- the combustion conditions,
- the general cylinder condition,
- the general engine condition

in order to discover any operational disturbances.

This enables the necessary precautions to be taken at an early stage, to prevent the further development of trouble.

This procedure will ensure optimum mechanical condition of the engine components, and optimum overall plant economy.

3.2 Key Parameters

The key parameters in performance observations are:

- Barometric pressure
- Engine speed
- Ships draught
- Mean indicated pressure
- Compression pressure
- Maximum combustion pressure
- Fuel pump index
- Exhaust gas pressures
- Exhaust gas temperatures
- Scavenge air pressure
- Scavenge air temperature
- Turbocharger speed
- Exhaust gas back pressure in exhaust pipe after turbocharger
- Air temperature before T/C filters
- Δ p air filter (if pressure gauge installed)
- Δ p air cooler
- Air and cooling water temperatures before and after scavenge air cooler.

3.3 Measuring Instruments

The measuring instruments for performance observations comprise:

- thermometers,
- pressure gauges,
- tachometers,
- indicator and planimeter,

It is important to check the measuring instruments for correct functioning.

Regarding check of thermometers and pressure gauges as well as check and functioning of the indicator, see Appendix 1 in this Chapter.

3.4 Intervals between Checks

<u>Constantly:</u>
Temperature and pressure data should be constantly monitored, in order to protect the engine against overheating and failure. In general, automatic alarms and slow-down or shut-down equipment are installed for safety.

Guiding values of permissible deviations from the normal service data are given in *Chapter 701, 'Alarm Limits'*.

<u>Daily:</u> Fill-in the Performance Observation record, *Plate 70603*, except for the values which require the taking of indicator cards.

<u>Every two weeks:</u> Take indicator cards, and fill-in the complete Performance Observation record, *Plate 70603. See also Appendix 1 in this Chapter.*

3.5 Evaluation of Observations

Compare the observations to earlier observations and to the testbed/sea trial results.

From the trends, determine when cleaning, adjustment and overhaul should be carried out.

See Chapter 701, regarding normal service values and alarm limits.

Not all parameters can be evaluated individually.

This is because a change of one parameter can influence another parameter.

For this reason, these parameters must be compared to the influencing parameters to ensure correct evaluations.

A simple method for evaluation of these parameters is presented in the next Section, 'Evaluation of Records',

Evaluation of Records

1. General

Record the performance observations as described in the previous Section 3 'Performance Observations'.

Use the *synopsis diagrams* to obtain the best and most simple method of plotting and evaluating the parameters:

Engine:	Plates 70605, 70606, 70607
Turbocharger:	Plates 70608, 70609
Air cooler:	Plate 70610

Plates 70605, 70606 and 70607 are sufficient to give a general impression of the overall engine condition.

The plates comprise:

Model curve: shows the parameter as a function of the parameter on which it is most dependent (based on the testbed/sea trial results).

Time based deviation curve: shows the deviation between the actual service observations and the model curve, as a function of time. The limits for max. recommended deviation is also shown.

From the deviation curves, it is possible to determine what engine components should be overhauled.

From the slope of the curves, it can be determined approximately when the overhaul should be carried out.

Blank sheets: Blank 'Time based deviation' sheets which can be copied.
Use these sheets for plotting the deviation values for the specific engine.

The following Items describe the evaluation of each parameter in detail.

2. Engine Synopsis

A 6L60MC has been used in these examples.

2.1 Parameters related to the Mean Indicated Pressure (p_i).

Plates 70605 and 70606 (engine synopsis diagrams) show model curves for engine parameters which are dependent upon the mean indicated pressure (p_i)

NB: *Plate 70605* also includes two charts for plotting the draught of the ship, and the average mean indicated pressure as a function of the engine running hours.

For calculation of the mean indicated pressure, *see Appendix 2 in this chapter.*

For engines without indicator drive or MIP-equipment, the estimated mean indicated pressure is read from *Plate 70606 'Average Fuel Pump Index'.*

Mean Draught

The mean draught is depicted here because, for any particular engine speed, it will have an influence on the engine load.

Mean indicated Pressure (p_i)

The average calculated value of the mean indicated pressure is depicted in order that an impression of the engine's load can be obtained.

Load balance: the mean indicated pressure for each cylinder should not deviate more than **0.5 bar** from the average value for all cylinders.

Note: The load balance **must not be** adjusted on the basis of the exhaust gas temperatures after each exhaust valve.

The governor must be steady. Unbalances in the load distribution may cause the governor to be unstable.

It is recommended to apply PMI-system, for easy access to P-V-diagrams (work diagrams), if the indicator drive is not installed.

Engine Speed

The model curve shows the relationship between the engine speed and the average mean indicated pressure (p_i).

The engine speed should be determined by counting the revolutions over a sufficiently long period of time.

Deviations from the model curve show whether the propeller is light or heavy, i.e. whether the torque on the propeller is small or large for a specified speed. If this is compared with the draught (under the same weather conditions), *see remarks in Item 2.1 'Load Diagram'*, then it is possible to judge whether the alterations are owing to:

- changes in the draught,
- or an increase in the propulsion resistance, for instance due to fouling of the hull, shallow water, etc.

Valuable information is hereby obtained for determining a suitable docking schedule.

If the deviation from the model curve is large, (e.g. deviations from shop trial to sea trial), it is recommended to plot the results on the load diagram, *see Item 2.1 'Load Diagram'*, and from that judge the necessity of making alterations on the engine, or to the propeller.

Maximum Combustion Pressure (p_{max})

The model curve shows the relationship between the average p_{max} (corrected to ISO reference ambient conditions) and the average p_i.

NB For correction to reference conditions, *see Appendix 3 in this Chapter.*

Deviations from the model curve are to be compared with deviations in the compression pressure and the fuel pump index *(see further on).*

If an individual p_{max} value deviates more than **3 bar** from the average value, the reason should be found and the fault corrected.

The pressure rise p_{comp}-p_{max} must not exceed the specified limit, i.e. 35 bar.

Fuel Pump Index

The model curve shows the relationship between the average index and the average p_i.

Deviations from the model curve give information on the condition of the fuel injection equipment.

Worn fuel pumps, and leaking suction valves, will show up as an increased fuel pump index in relation to the mean pressure. Note, however, that the fuel pump index is also dependent on:

a) The viscosity of the fuel oil, (i.e. the viscosity at the preheating temperature). Low viscosity will cause larger leakages in the fuel pump, and thereby necessitate higher indexes for injecting the same volume.

b) The calorific value and the specific gravity of the fuel oil. These will determine the energy content per unit volume, and can therefore also influence the index.

c) All parameters that affect the fuel oil consumption (ambient conditions, p_{max}, etc.)

Since there are many parameters that influence the index, and thereby also the p_{max}, it can be necessary to adjust the p_{max} from time to time.

It is recommended to overhaul the fuel pumps when the index has increased by about 10%.

In case the engine is operating with excessively worn fuel pumps, the starting performance of the engine will be seriously affected.

2.2 Parameters related to the Effective Engine Power (P_e)

Plate 70607 shows model curves for engine parameters which are dependent on the effective power (P_e).

Regarding the calculation of effective engine power, *see Appendix 2 in this Chapter.*

For engines without indicator drive, the estimated effective engine power is found by using the fuel pump index and T/C revolutions as parameters, *see Appendix 5 in this Chapter.*

It is recommended to apply PMI-system for easy access to P-V-diagrams (work diagrams) and thereby the effective engine power.

Exhaust Temperature (t_{exhv})

The model curve shows the average exhaust temperatures (after the valves), corrected to reference conditions, and drawn up as a function of the effective engine power (P_e).

NB For correction to ISO reference ambient conditions, *see Appendix 3 in this Chapter.*

Regarding maximum exhaust temperatures, *see also Appendix 3 in this Chapter.*

The exhaust temperature is an important parameter, because the majority of faults in the air supply, combustion and gas systems manifest themselves as increases in the exhaust temperature level.

The most important parameters which influence the exhaust temperature are listed in the table on the next page, together with a method for direct diagnosing, where possible.

Increased Exhaust Temperature Level – Fault Diagnosing:

Possible Causes	Diagnosing
a. Fuel injection equipment: – Leaking or incorrectly working fuel valves (defective spindle and seat) – Worn fuel pumps. If a high wear rate occurs, the cause for this must be found and remedied. **Note:** Inadequate cleaning of the fuel oil can cause defective fuel valves and worn fuel pumps.	As these faults occur in individual cylinders, compare: – fuel pump indexes – Indicator and draw diagrams *See Appendix 2 in this Chapter* Check the fuel valves: – visually – by pressure testing.
b. Cylinder condition: – Blow-by, piston rings *See also Chapter 703, 'Running Difficulties', point 7.* – Leaking exhaust valves *See also Chapter 703, 'Running Difficulties' point 6.*	These faults occur in individual cylinders. – Compare the compression pressures from the indicator and draw diagrams. *See Appendix 2 in this Chapter.* – During engine standstill: Carry out scavenge port inspection. *See Chapter 707, 'Scavenge Port Inspection'.* Check the exhaust valves.
c. Air coolers: – Fouled air side – Fouled water side	Check the cooling capability. *See Section 'Evaluation of Records', Item 'Air Cooler Synopsis' in this Chapter.*
d. Climatic conditions: – Extreme conditions	Check cooling water and engine room temperatures. Correct T_{exhv} to reference conditions. *See Appendix 3, Items 3 and 4 in this Chapter.*
e. Turbocharger: – Fouling of turbine side – Fouling of compressor side	Use the turbocharger synopsis methods for diagnosing. *See Section 'Evaluation of Records, Item 'Turbocharger Synopsis', in this Chapter.*
f. Fuel oil: – Type – Quality	Using heavy fuel oil will normally increase T_{exhv} by approx. 15°C, compared to the use of gas oil. Further increase of T_{exhv} will occur when using fuel oils with particularly poor combustion properties. In this case, a reduction of p_{max} can also occur.

Compression Pressure (p_{comp})

The model curve shows the relationship between the compression pressure p_{comp} (corrected to ISO reference ambient conditions) and the effective engine power P_e.

NB For correction to reference conditions, *see Appendix 3 in this Chapter.*

Deviation from the model curve can be due to:

a) a scavenge air pressure reduction,

b) – mechanical defects in the engine components (blow-by past piston rings, defective exhaust valves, etc. *– see the table on the next page).*

– excessive grinding of valve spindle and bottom piece.

It is therefore expedient and useful to distinguish between 'a' and 'b', and investigate how large a part of a possible compression reduction is due to 'a' or 'b'.

This distinguishing is based on the ratio between absolute compression pressure (p_{comp} + p_{baro}) and absolute scav. pressure (p_{scav} + p_{baro}) which, for a specific engine, is constant over the largest part of the load range (load diagram area).

The ratio is first calculated for the "new" engine, either from the testbed results, or from the model curve.

See the example below regarding:

- Calculating the ratio
- Determining the influence of mechanical defects.

It should be noted that, the measured compression pressure, for the individual cylinders, can deviate from the average, owing to the natural consequence of air/gas vibrations in the receivers. The deviations will, to some degree, be dependent on the load.

However, such deviations will be "typical" for the particular engine, and should not change during the normal operation.

When evaluating service data for individual cylinders, comparison must be made with the original compression pressure of the cylinder concerned, at the corresponding load.

Example:

The following four values can be assumed read from the *model curves:*

The barometric pressure was : 1.00 bar

The scavenge pressure was : 2.25 bar

This gave an absolute scavenge pressure of : 3.25 bar

The average (or individual) compression pressure was : 115 bar

which gave an absolute compression pressure of 115 + 1.00 = 116 bar

$$\frac{p_{comp\ abs}}{p_{scav\ abs}} = \frac{116}{3.25} = 35.7$$

This value is used as follows for evaluating the data read during service.

Service Values

p_{comp} : 101 bar (average or individual)
p_{scav} : 2.0 bar
p_{baro} : 1.02 bar

Calculated on the basis of p_{scav} and p_{baro}, the absolute compression pressure would be expected to be:

$$p_{comp\ abs} = 35.7 \times (2.0 + 1.02) = 107.8 \text{ bar}$$

i.e. p_{comp} = 107.8 – 1.02 = 106.8 bar

The difference between the expected 106.8 bar and the measured 101 bar could be owing to *mechanical defects or grinding of exhaust valve spindle and bottom piece.*

Concerning the pressure rise p_{comp}-p_{max}, *see Item 2.1, 'Maximum Combustion Pressure'.*

Mechanical Defects which can influence the Compression Pressure

Possible cause	Diagnosis / Remedy
a. Piston rings: - Leaking	Diagnosis: *See table `Increased Exhaust Temperature Level - Fault Diagnosis', point b, `Cylinder Condition'.* Remedy: *See Chapter 703, `Running Difficulties', point 7.*
b. Piston crown: - Burnt	Check the piston crown by means of the template. *See Vol. II, Chapter 902.*
c. Cylinder liner: - Worn	Check the liner by means of the measuring tool. *See Vol. II, Chapter 903.*
d. Exhaust valve: - Leaking - The exhaust temperature rises. - A hissing sound can possibly be heard at reduced load. - Timing	Remedy: *See Chapter 703, `Running Difficulties', point 6.* Check: - Cam lead - Hydraulic oil leakages, e.g. misalignment of high pressure pipe between exhaust valve actuator and hydraulic cylinder. - Damper arrangement for exhaust valve closing.
e. Piston rod stuffing box: - Leaking - Air is emitted from the check funnel from the stuffing box.	Small leakages may occur due to erosion of the bronze segments of the stuffing box, but this is normally considered a cosmetic phenomenon. Remedy: *Overhaul the stuffing box, see Vol. II, Chapter 902.*

3. Turbocharger Synopsis

Plates 70608 and 70609
(Turbocharger synopsis diagrams)

NB: Plates 70608 and 70609 should be filled out in a number of copies which corresponds to the number of turbochargers.

Regarding cleaning of the turbochargers, *see Section 'Cleaning of Turbochargers and Air Coolers', further on in this Chapter.*

Scavenge Air Pressure (p_{scav})

The model curve shows the scavenge air pressure (corrected to reference conditions) as a function of the effective engine power (P_e).

See Appendices 2 and 5 regarding the effective engine power.

NB For correction to ISO reference ambient conditions, *see Appendix 3 in this Chapter.*

Deviations in the scavenge air pressure are, like the exhaust temperature, an important parameter for an overall estimation of the engine condition.

A drop in the scavenge air pressure, for a given load, will cause an increase in the thermal loading of the combustion chamber components.

A simple diagnosis, made only from changes in scavenge air pressure, is difficult.

Fouled air filter, air coolers and turbochargers can greatly influence the scavenge air pressure.

Changes in the scavenge air pressure should thus be seen as a "consequential effect" which is closely connected with changes in:

- the air cooler condition.
- the turbocharger condition.
- the cam timing.

Reference is therefore made to the various sections covering these topics.

Turbocharger Speed (T/C speed)

The model curve shows the speed of the turbocharger as a function of the scavenge air pressure (p_{scav}).
Corroded nozzle ring or turbine blades will reduce the turbine speed. The same thing will happen in case of a too large clearance between the turbine blades and the shroud ring (MAN B&W) / cover ring (BBC / ABB).

Deviation from the model curve, in the form of too high speed, can normally be attributed to a fouled air filter, scavenge air cooler, turbine side or compressor side.

A more thorough diagnosing of the turbocharger condition can be made as outlined in the 'turbocharger efficiency' Section below.

Pressure Drop across Turbocharger Air Filter (Δp_f)

The model curve shows the pressure drop across the air filter as a function of the scavenge air pressure (p_{scav}).

Deviations from this curve give direct information about the cleanliness of the air filter.

Like the air cooler, the filter condition is decisive for the scavenge air pressure and exhaust temperature levels.

The filter elements must be cleaned when the pressure drop is 50% higher than the testbed value.

If a manometer is not standard, the cleaning interval is determined by visual inspection.

Turbocharger Efficiency (η T/C)

The model curves show the compressor and turbine efficiencies as a function of the scavenge air pressure (p_{scav}).

In order to determine the condition of the turbocharger, the calculated efficiency values are compared with the model curves, and the deviations plotted.

Calculation of the efficiency is explained in *Appendix 4 to this Chapter.*

As the efficiencies have a great influence on the exhaust temperature, the condition of the turbocharger should be checked if the exhaust temperature tends to increase up to the prescribed limit.

Efficiency reductions can normally be related to "flow deterioration", which can be counteracted by regular cleaning of the turbine side (and possibly compressor side).

4. Air Cooler Synopsis

Plate 70610 (Air cooler synopsis diagrams)

The plate gives model curves for air cooler parameters, which are dependent on the scavenge air pressure (p_{scav}).

Regarding cleaning of air coolers, *see Section 'Cleaning of Turbochargers and Air Coolers', further on in this Chapter.*

<u>Temperature Difference between Air Outlet and Water Inlet</u> ($\Delta t_{(air-water)}$)

The model curve shows the temperature difference between the air outlet and the cooling water inlet, as a function of the scavenge air pressure (p_{scav}).

This difference in temperature is a direct measure of the cooling ability, and as such an important parameter for the thermal load on the engine. The evaluation of this parameter is further discussed in Item 4.1.

<u>Cooling Water Temperature Difference</u> (Δt_{water})

The model curve shows the cooling water temperature increase across the air cooler, as a function of the scavenge air pressure (p_{scav}).

This parameter is evaluated as indicated in Item 4.1.

<u>Pressure Drop across Air Cooler</u> (Δp_{air})

The model curve shows the scavenge air pressure drop across the air cooler, as a function of the scavenge air pressure (p_{scav}).

This parameter is evaluated as indicated in Item 4.1.

4.1 Evaluation

Generally, for the above three parameters, *changes of approx. 50% of the testbed value can be considered as a maximum.* However, the effect of the altered temperatures should be kept under observation in accordance with the remarks under Exhaust Temperature. *(Point 2.2 earlier in this Section).*

In the case of *pressure drop across air cooler,* for purposes of simplification, the mentioned "50% margin" includes deviations caused by alterations of the suction temperature, scavenge air temperature, and efficiency of the turbochargers.

Of the three parameters, *the temperature difference between air outlet and water inlet,* is to be regarded as the most essential one.

Deviations from the model curves, which are expressions of deteriorated cooling capability, can be due to:

a) Fouling of the air side
b) Fouling of the water side

a) Fouling of the air side: manifests itself as an *increased pressure drop across the air side.*

Note however, that the heat transmission can also be influenced by an "oily film" on tubes and fins, and this will only give a minor increase in the pressure drop.

Before cleaning the air side, it is recommended that the U-tube manometer is checked for tightness, and that the cooler is visually inspected for deposits.

Make sure that the drainage system from the water mist catcher functions properly, as a high level of condensed water (condensate) – up to the lower measuring pipe – might greatly influence the Δp measuring. *See also 'Cleaning of Turbochargers and Air Coolers', Item 3, 'Drain System' further on in this Chapter.*

b) Fouling of the water side: Normally involves *a reduction of the cooling water temperature difference,* because the heat transmission (cooling ability) is reduced.

Note however that, if the deposits reduce the cross sectional area of the tubes, so that the water quantity is reduced, the cooling water temperature difference may not be affected, whereby diagnosis is difficult (i.e. lower heat transmission, but also lower flow volume).

Furthermore, a similar situation will arise if such tube deposits are present simultaneously with a fault in the salt water system, (corroded water pump, erroneous operation of valves, etc.). Here again the reduced water quantity will result in the temperature difference remaining approximately unaltered.

In cases where it is suspected that the air cooler water side is obstructed, the resistance across the cooler can be checked by means of a differential pressure gauge.

NB: A mercury manometer pressure gauge should not be used, because of environmental considerations.

Before dismantling the air cooler, for piercing of the tubes, it is recommended that the remaining salt-water system is examined, and the cooling ability of the other heat exchangers checked.

NB: Be careful when piercing, because the pipes are thin-walled.

5. Specific Fuel Oil Consumption

Plate 70611

Calculation of the specific fuel oil consumption (g/kWh, g/bhph) requires that engine power, and the consumed fuel oil amount (kg), are known for a certain period of time.

The method of determining the engine power is illustrated in Appendix 2. For engines without indicator drive, *see Appendix 5 in this Chapter.*

The oil amount is measured as described below.

To achieve a reasonable measuring accuracy, it is recommended to measure over a suitably long period – dependent upon the method employed i.e.:

- If a day tank is used, the time for the consumption of the whole tank contents will be suitable.
- If a flow-meter is used, a minimum of 1 hour is recommended.

The measurements should always be made under calm weather conditions.

Since both of the above-mentioned quantity measurements will be in volume units, it will be necessary to know the oil density, in order to convert to weight units. *The density is to correspond to the temperature at the measuring point* (i.e. in the day tank or flow-meter).

The specific gravity, (and thus density) can be determined by means of a hydrometer immersed in a sample taken at the measuring point, but the density can also be calculated on the basis of bunker specifications.

Normally, in bunker specifications, the specific gravity is indicated at 15°C/60°F.

The actual density (g/cm^3) at the measuring point is determined by using the curve on *Plate 70611,* where the change in density is shown as a function of temperature.

The consumed oil quantity in kg is obtained by multiplying the measured volume (in litres) by the density (in kg/litre).

In order to be able to compare consumption measurements carried out for various types of fuel oil, allowance must be made for the differences in the lower calorific value (LCV) of the fuel concerned.

Normally, on the testbed, gas oil will have been used, having a lower calorific value of approx. 42,707 kJ/kg (corresponding to 10,200 kcal/kg). If no other instructions have been given by the shipowner, it is recommended to convert to this value.

Usually, the lower calorific value of a bunker oil is not specified by the oil companies. However, by means of the graph, *Plate 70611,* the LCV can be determined with sufficient accuracy, on the basis of the sulphur content, and the specific gravity at 15°C.

The corrected consumption can then be determined by multiplying the "measured consumption", by either:

$\frac{\mathbf{LCV_1}}{\mathbf{42,707}}$ (LCV_1 = the specific lower calorific value, in kJ/kg, of the bunker oil concerned)

or

$\frac{\mathbf{LCV_2}}{\mathbf{10,200}}$ (LCV_2 = the specific lower calorific value, in kcal/kg, of the bunker oil concerned)

Example: (6L60MC)

Effective Engine Power, P_e	:	15,600 bhp
Consumption, Co	:	7.125 m^3 over 3 hours
Measuring point temperature	:	119°C
Fuel data	:	Specific gravity: 0.9364 g/cm^3 at 15°C, 3% sulphur

Density at 119°C *(see Plate 70611),*
$\rho119$: 0.9364 – 0.068 = 0.8684 g/cm^3.

<u>Specific consumption:</u>

$$\frac{Co \times \rho119 \times 10^6}{h \times P_e} \quad \text{(g / bhph)}$$

where:

Co = Fuel oil consumption over the period, m^3
$\rho119$ = Corrected gravity, g/cm^3
h = Measuring period, hours
P_e = Brake horse power, bhp

$$\frac{7.125 \times 0.8684 \times 10^6}{3 \times 15,600} = 132.2 \text{ g/bhph}$$

Correction to ISO reference conditions <u>regarding the specific lower calorific value:</u>

LCV_1 = 40,700 kJ/kg, derived from *Plate 70611.*

Consumption corrected for calorific value:

$$\frac{132.2 \times 40,700}{42,707} = 126.0 \text{ g/bhph}$$

or

LCV_2 = 9723 kcal/kg derived from Plate 70611.

Consumption corrected for calorific value:

$$\frac{132.2 \times 9723}{10,200} = 126.0 \text{ g/bhph}$$

Note: The ambient conditions (blower inlet temperature and pressure and scavenge air coolant temperature) will also influence the fuel consumption. Correction for ambient conditions is not considered important when comparing service measurements.

Cleaning of Turbochargers and Air Coolers

1. Turbocharger

1.1 General

We recommend to clean the turbochargers regularly during operation.

This prevents the build-up of heavy deposits on the rotating parts and keeps the turbochargers in the best running condition between manual overhauls.

The intervals between cleaning during operation should be determined from the degree of fouling of the turbocharger in the specific plant.

This is because the tendency to form deposits depends, among other things, on the combustion properties of the actual fuel oil.

Guiding intervals between cleaning are given for each cleaning method in the following items.

Note: If the cleaning is not carried out at regular intervals, the deposits may not be removed uniformly. This will cause the rotor to be unbalanced, and excite vibrations.

IF	THEN
Vibrations occur after cleaning	Clean again.
Vibrations occur after repeated cleaning	*See Chapter 704 'Emergency Running with Cylinders or Turbochargers out of Operation', Item 5 'How to put Turbochargers out of Operation'.* Clean the turbochargers manually at the first opportunity.

Manual overhauls are still necessary to remove deposits which the cleaning during operation does not remove, in particular on the non-rotating parts.

Regarding intervals between the manual overhauls, see the maker's instructions.

1.2 Cleaning the Turbine Side

Dry Cleaning

(Plate 70612)

Intervals between cleaning:
24-50 hours of operation.

The cleaning is effected by injecting a specified volume of crushed nut shells or similar. The "grain size" is to be 1.0 mm in average (max. 1.5 mm).*
Since the cleaning is mechanical, the highest efficiency is obtained at full load, and cleaning should not be carried out below half load.

Carry out the cleaning according to the instruction given on the "instruction plate" located at the turbocharger, *see Plate 70612. See also Vol. II, 'Maintenance', Chapter 910.*

Water Cleaning (Not MET-Turbochargers)

(Plate 90613)

Intervals between cleaning:
Approx. 6 days of operation.

The cleaning is effected by injecting atomised water through the gas inlet, at reduced engine load.

Carry out the cleaning according to the instruction given on the "instruction plate" located at the turbocharger, *see Plate 70613.*

Be aware that water cleaning can cause corrosion on the shroud ring surrounding the T/C turbine blading.

Note that, during normal running, some of the scavenge air is led through a three-way cock, from pipe No. 2 to pipe No. 1, at the turbine outlet drainage hole, whereby this pipe is kept clean.

* In case the turbocharger type is other than NA type, the grain size should follow the manufacturer's recommendation.

1.3 Cleaning the Compressor Side

Guiding intervals between cleaning:
25-75 hours of operation.

Note: Always refer to the maker's special instruction.

The cleaning is effected by injecting water through a special pipe arrangement during running at high load and normal temperatures.

Regarding the cleaning procedure, see the maker's special instructions.

Note: If the deposits are heavy and hard, the compressor **must** be dismantled and cleaned manually.

If the in-service cleaning is carried out when the compressor side is too contaminated, the loosened deposits can be trapped in the narrow passages of the air cooler element.

> This reduces the air cooler effectiveness.

Regarding air cooler cleaning, see Item 2 'Air Cooler Cleaning System', below.

We recommend to wrap a thin foam filter gauze around the turbocharger intake filter, and fasten it by straps.

> This greatly reduces fouling of the compressor side, and even makes in-service cleaning unnecessary.

Replace and discard the filter gauze, when it becomes dirty.

2. Air Cooler Cleaning System

Plate 70614

See Chapter 701, pos. 420 and 421 regarding the basis for intervals between cleaning.

Note: Carry out the cleaning only when the engine is at standstill.

> This is because the water mist catcher is not able to retain the cleaning fluid. Thus there would be a risk of fluid being blown into the cylinders, causing excessive liner wear.

Cleaning of the air side of the scavenge air cooler is effected by injecting a chemical fluid through 'AK' to a spray pipe arrangement fitted to the air chamber above the air cooler element.

The polluted chemical cleaning agent returns from 'AM', through a filter to the chemical cleaning tank.

The procedure is described in the *'Maintenance' instruction book, Chapter 910.*

3. Drain System for Water Mist Catcher

3.1 Condensation of Water from a Humid Atmosphere.

A combination of high air humidity and cold air cooler pipes will cause an amount of condensed water to be separated from the scavenge air in the water mist catcher.

> A typical example is high air temperature and low cooling water temperature.

To give an impression of the amount of condensed water, two examples are shown in *Plate 70713.*

3.2 Drain System

Plate 70614

Condensed water will be drained off from the water mist catcher through the sight glass, the orifice and flange AL to bilge.

The size of the orifice in the drain system is designed to be able to drain off the amount of condensed water under average running conditions.

In case of running under special conditions with high humidity, it can be necessary to open the valves on the discharge line a little.

Close these valves when possible to reduce the loss of scavenge air.

3.3 Checking the Drain System by the Sight Glass

a) A mixed flow of air and water indicates a correctly working system where condensation takes place.

b) A flow of water only, indicates malfunctioning of the system.

   Check the orifice for blocking.

   Check for any restrictions in the discharge pipe from AL.

   Check and overhaul the level alarm.

c) A flow of air is only normal when running under dry ambient conditions

   Note: A sight glass which is completely filled with clean water, and with no air flow, visually looks like an empty air-filled sight glass.

APPENDIX 1

Measuring Instruments

1. Thermometers and Pressure Gauges

The thermometers and pressure gauges fitted on the engine are often duplicated with instruments for remote indication.

Owing to differences in the installation method, type and make of sensing elements, and design of pockets, the two sets of instruments cannot be expected to give exactly the same readings.

Readings taken at shop test and sea trial are to be used as the basis for all evaluations.

Check the thermometers and pressure gauges at intervals against calibrated control apparatus.

Thermometers should be shielded against air currents from the engine-room ventilation.

If the temperature permits, keep thermometer pockets filled with oil to ensure accurate indication.

Keep all U-tube manometers perfectly tight at the joints.

Check the tightness from time to time by using soap-water.

To avoid polluting the environment, do not use mercury instruments.

Check that there is no water accumulation in tube bends.

This would falsify the readings.

If cocks or throttle valves are incorporated in the measuring equipment, check these for free flow, prior to taking readings.

If an instrument suddenly gives values that differ from normal, consider the possibility of a defective instrument.

The easiest method of determining whether an instrument is faulty or not, is to exchange it for another.

2. The Indicator

The indicator is employed for taking indicator diagrams, whereby the combustion chamber pressures can be measured while the engine is running.

2.1 Indicator and Draw Diagrams

The draw diagram is used for measuring the compression pressure and maximum pressure, and for evaluating the ignition characteristics of the fuel oil.

For engines fitted with indicator drive or MIP-equipment:

The indicator diagram (pv diagram: work diagram), illustrates the pressure variations in the engine cylinder as a function of the main piston position. The diagram area can be integrated by means of a planimeter, and the mean indicated pressure calculated.

The power developed in the particular cylinder can then be found by multiplication by the engine speed and the cylinder constant, *see Appendix 2, item 3.*

In order to ensure true indicator/draw diagrams, and correct evaluation of data, the following instructions should be followed in detail.

2.2 Maintenance of the Indicator

Friction in the indicator piston movement, as well as slackness in the stylus (writing) mechanism, will distort both the shape and the area of the diagram.

Test and maintain the indicator in the following way:

Friction and tightness of piston:

Remove the indicator spring.

Dismantle the upper part of the indicator, and remove the piston from the cylinder.

Wipe the piston and cylinder with a clean cloth.

Mount the upper part again.

Note: During mounting, check that the piston sinks slowly down the liner, by its own weight, when the cylinder is held vertically.

Hold the indicator upright.

Pull the piston to the upper position.

Block the bottom of the cylinder with a finger.

Check that the piston fits so tightly that it remains in the upper position.

Push the piston downwards and release.

Check that the piston springs back to the upper position.

Tighten the top screw, which retains the spring, firmly against the ball-head of the spring.

Check that the ball is not loose on the spring (older spring types).

Check that the coils of the spring have not worked loose at the soldered joint in the base.

Stylus (writing) mechanism:

Check that the stylus is sharp.

Check for slackness in the writing mechanism.

Replace any worn parts.

Adjust the stylus so that, with a light writing pressure, a single passage over the paper can just be seen.

To obtain sufficiently distinct work diagrams, trace the diagram two or three times.

Lubricate the mechanism with thin oil.

2.3 Indicator Valve

During the running of the engine, soot and oil will accumulate in the indicator bore.

Clean the bore by opening the indicator valve for a moment.

To protect the valve against burning:

- Open the valve only partially,
- Close the valve after one or two ignitions.

2.4 Fitting the Indicator

Dismantle the upper part.

Give the piston a little cylinder oil.

Check that the various recesses are clean.

Otherwise the parts could be positioned askew, and this would cause the piston to move sluggishly in the cylinder.

Mount the upper part.

Fit the indicator and the cord.

Engage the indicator drive.

Check the cord alignment.

Adjust the length of the indicator cord so that:

- the diagram is traced in the centre of the paper,
- the cord is tight in all positions.

2.5 Taking the Diagrams

For diagram descriptions and nomenclature – *see Plate 70615.*

1. Atmospheric line:

   Keep the indicator valve closed.

   Press the stylus against the paper.

   Release the stylus when the indicator drive has turned the drum one or two times.

2. For engines fitted with indicator drive/ PMI-system.

   Indicator diagram:

   Open the indicator valve.

   Press the stylus against the paper.

   Release the stylus, when the drum has turned two or three times.

   Close the indicator valve.

3. Draw diagram:

   Release the cord from the indicator drive.

   Open the indicator valve.

   Watch the movement of the stylus.

   At the moment it moves upwards, *simultaneously*

   - Press it against the paper.
   - Pull the cord just quickly enough for the stylus to trace the compression and ignition sequence.

   This operation requires some practice to ensure that both compression and maximum pressures are clearly recorded.

Close the indicator valve.

If the indicator quickly becomes very hot, and the piston is black after use, then this means that there is a leakage.

In such a case, exchange the piston and liner.
See also item 2.2 in this Appendix.

4. Check that the diagrams have been correctly taken and are distinct.

   Normal indicator and draw diagrams are shown in the illustration, *Plate 70615.*

Examples of incorrect diagrams and possible causes are shown on *Plate 70617. See also Item 2.6 in this Chapter.*

Regarding pressure evaluation and engine power calculation, see Appendix 2 in this Chapter.

5. Repeat Items 2.3, 2.4 and 2.5 for the remaining cylinders.

Lubricate the piston with a drop of cylinder oil after about six diagrams have been taken.

When diagram taking is finished, unscrew the indicator head.

Clean and lubricate both the cylinder and the piston with cylinder oil.

2.6 Diagram Faults

The most common faults are shown on *Plate 70617, in Figs. 1 to 6.*

Fig. 1 For engines fitted with indicator drive:
Vibrations in the cord, or drive, give a wavy indicator diagram, but a smooth draw diagram.

Fig 2 & 3 For engines fitted with indicator drive:
The drum hits the stop at one of the end points, before the diagram is completed:
The cord is too long or too short.

Fig. 4 The indicator piston works sluggishly in the cylinder, and moves in jerks:
If only the expansion curve is wrong (wavy), the cause may be gas pulsations in the combustion chamber or indicator bore.

Fig. 5 The indicator spring is too weak.
The piston strikes against the top of the indicator cylinder. Change to a more rigid spring.

Fig. 6 The indicator valve leaks:
Gives an untrue atmospheric line.

2.7 Adjustment of Indicator Drive

Plate 70616

The paper drum of the indicator is driven by the indicator drive, which is activated by the indicator cam on the camshaft, in line with the corresponding cylinder.

The indicator drive must be adjusted so that the position of the paper drum at any moment corresponds to the position of the main piston, when taking the diagrams.

This ensures correct indicator diagrams.

Check the adjustment of the individual indicator drives regularly, and after disassembling in the following way:

1. Prepare the indicator valve and indicator for taking diagrams.
   See previous Items 2.3 and 2.4

2. Cut-off fuel injection in one cylinder:
   - Reduce the load to 35-50% of MCR (70-80% of MCR speed).
   - Pull the fuel rack for the cylinder concerned to 'O' index.

   Alternatively, lift the roller guide as described in Vol. II, Procedure 909-5. Start the engine and load to 35-50% of MCR power (70-80% of MCR speed).

3. Trace the compression and expansion lines.
   Follow the procedure in Item 2.5, point 2, 'Indicator Diagram'.
   The compression line is traced when the engine piston moves upwards, and the expansion line is traced when the engine piston moves downwards.

4. Evaluate the diagram:

Do the compression line and the expansion line coincide?	
YES	The indicator drive is correctly adjusted. *See also Plate 70616, Fig. 1.*
NO	The indicator drive is incorrectly adjusted. *See also Plate 70616, Fig. 2.* Adjust the indicator drive. *See Plate 70616, Case A and Case B.*

APPENDIX 2

Indicator Diagram, Pressure Measurements and Engine Power Calculations

Regarding taking the diagrams, see Appendix 1 in this Chapter.

1. Compression Pressure, Maximum Pressure, and Faults

Plate 70618 (See also Plate 70615)

Measure the compression pressure and maximum pressure on the cards.

Use a scale rule which corresponds to the stiffness of the indicator spring used.

Compare the measurement results to the normal values for the actual engine.

Figs. 1-3 show some typical examples of engine maladjustment and faults which can be derived from the indicator and draw diagrams.

Fig. 1

Maximum pressure too low, but compression pressure correct.

Fuel injection delayed, check:

- the fuel pressure at engine (after the filter), *see Chapter 701 'Alarm Limits'.*
- the fuel valves function
- the fuel pump suction valve, puncture valve and shock absorber.
- VIT-index.

If the above are in order, the fuel oil is injected too late in relation to its ignition characteristics.

Note: Exceptionally bad fuels can have very poor ignition qualities.

Increase the fuel pump lead.
See Vol. II, Chapter 909.

Fig. 2

Maximum pressure too high, but compression pressure normal.

Too early injection, check VIT-index.

If this is in order, reduce the fuel pump lead. *See Vol. II, Chapter 909.*

Fig. 3

Compression and maximum pressures both too low. Possible causes:

- piston ring blow-by ★
- leaking exhaust valve ★
- increased combustion space volume (piston crown burnt) ★
- low scavenge air pressure, for instance due to fouling of exhaust and/or air system.
- defective or maladjusted damping arrangement in the exhaust valve ★
- Cooling water inlet and air inlet temperatures deviate from reference ambient conditions.

See also Appendix 3 in this Chapter.

★ *See also section 'Evaluation of Records', Item 2.2 'Compression Pressure', page 706.09.*

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2. Area of Indicator Diagram

(For engines fitted with indicator drive or PMI-system)

Plate 70619

If the planimeter is adjustable, check the setting before use.

For checking, use the reference template, or the area of an accurately drawn rectangle or circle.

Place the planimeter and indicator card on a piece of plane cardboard (not too smooth), as shown in the illustration.
Trace the diagram as described in *Plate 70619.*

Note: Only consider the result satisfactory, when two readings are obtained which do not differ more than `1' on the planimeter vernier scale.

3. Calculation of the Indicated and Effective Engine Power

(For engines without indicator drive or PMI-system, *see Appendix 5 in this Chapter)*

Calculation of the indicated and effective engine power consists of the following steps:

Calculate:

- The mean indicated pressure, p_i
- The mean effective pressure, p_e
- The cylinder constant, k_2
- The indicated engine power, P_i
- The effective engine power, P_e

The mean indicated pressure, p_i

$$p_i = \frac{A}{L \times C_s} \quad \text{(bar)}$$

where:

A (mm²) = area of the indicator diagram, as found by planimetering.

L (mm) = length of the indicator diagram (= atmospheric line).

C_s (mm/bar)= spring constant (= vertical movement of the indicator stylus (mm) for a 1 bar pressure rise in the cylinder).

p_i corresponds to the height of a rectangle with the same area and length as the indicator diagram.

I.e., if p_i was acting on the piston during the complete downwards stroke, the cylinder would produce the same total work as actually produced in one complete revolution.

The mean effective pressure, p_e

$$p_e = p_i - k_1 \text{ (bar)}$$

where

k_1 = the mean friction loss

The mean friction loss has proved to be practically independent of the engine load. By experience, k_1 has been found to be approx. 1 bar.

The cylinder constant, k_2

k_2 is determined by the dimensions of the engine, and the units in which the power is wanted.

For power in kW : $k_2 = 1{,}30900 \times D^2 \times S$

For power in BHP : $k_2 = 1{,}77974 \times D^2 \times S$

where:

D (m) = cylinder diameter
S (m) = piston stroke

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Engine type	For power in kW k_2	For power in BHP k_2
S46MC-C	0.5351	0.7276
L50MC	0.5301	0.7208
S50MC	0.6250	0.8498
S50MC-C	0.6545	0.8899
L60MC	0.9161	1.2455
S60MC	1.0801	1.4685
S60MC-C	1.1310	1.5377
L70MC	1.4547	1.9779
S70MC	1.7151	2.3319
S70MC-C	1.7959	2.4418
L80MC	2.1715	2.9524
S80MC	2.5602	3.4809
S80MC–C	2.6808	3.6449
K80MC-C	1.9268	2.6198
L90MC	3.0918	4.2037
K90MC	2.7037	3.6761
K90MC-C	2.4387	3.3157
S90MC-T	3.3802	4.5958
K98MC-C	3.0172	4.1022
S90MC-T	3.3802	4.5958
K98MC-C	3.0172	4.1022

The indicated engine power, P_i

$P_i = k_2 \times n \times p_i$ (ikW or ihp)

where

n (rpm) = engine speed.

The effective engine power, P_e

$P_e = k_2 \times n \times p_e$ (kW or bhp)

where

n (rpm) = engine speed.

Due to the friction in the thrust bearing, the shaft power is up to 1% less than the effective engine power, depending on speed and load conditions and plant type (FPP/CPP).

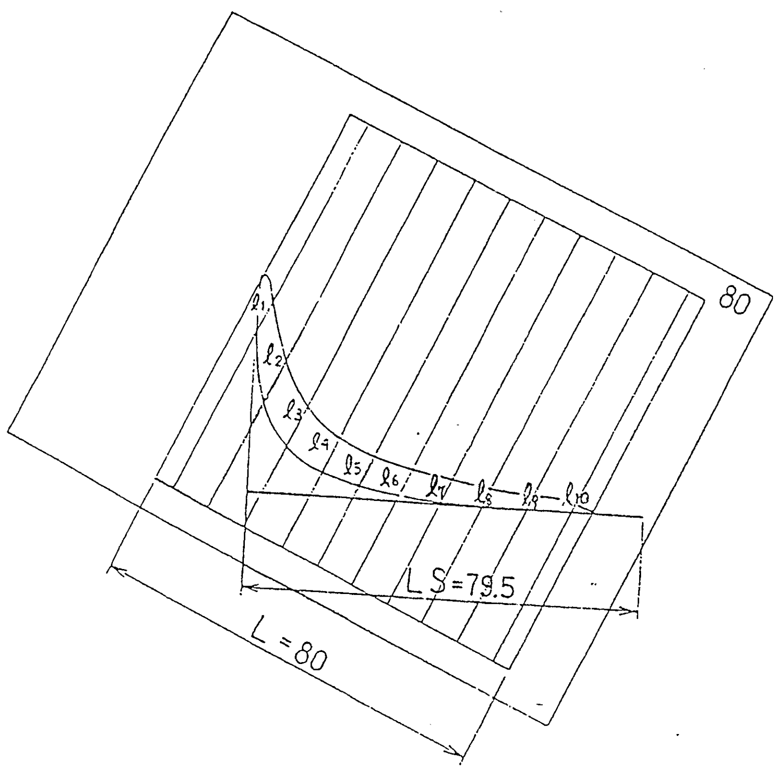
2. Calculation of the engine power (If indicator cam is equipped.)

To obtain mean indicated effective pressure (Pi) from the indicator diagram, the celluloid measuration plates are recommendable.
The celluloid measuration plate whereon stroke (L) nearly equal to the measured stroke (LS) on the indicator diagram should be used.
Put the celluloid measuration plate so that its both end lines coincide with both ends of effective area of the indicator diagram, as shown.
Add the each lengths across the area of indicator diagram from $\ell 1$ to $\ell 10$ with a divider.
The equivalent length (Li) to mean indicated effective pressure can be obtained from following formula.

$$Li = \frac{\ell 1 + \ldots + \ell 10}{10} \times \frac{L}{LS}$$

$\ell 1$ ..... $\ell 10$ = Lengths across the area of indicator diagram
LS = Stroke in the indicator diagram (mm)
L = stroke in the measuration plate (mm)

Convert Li to mean indicated effective pressure (Pi) by the special scale, normally 0.35 mm/kg/cm², in accordance with the spring stiffness.



The indicated horse power in each cylider can be obtained from the following formula.

Indicated Horse Power : IHP = Pi × RPM × C (PS)

In above formula,

$$C = \text{Cylinder constant} = \frac{\text{Piston area } (cm^2) \times \text{piston stroke } (m)}{75 \times 60}$$

Eng. type : Cylinder constant

S60MC: 1.4401	L60MC: 1.22145	S60MC-C: 1.508
S70MC: 2.2868	L70MC: 1.9396	S70MC-C: 2.0183
S80MC: 3.4136	L80MC: 2.8953	S80MC-C: 3.1276

Pi = indicated mean effective pressure (kg/cm^2)

Example,

Pi = 13.36 kg/cm^2 Revolution = 63.0 rpm

Indicated Horse Powe : IHP = 13.36 × 63 × 3.4136 = 2873 (PS/CYL)

The brake house power can be obtained by multiplying the indicated horse power with the mechanical efficiency in "Results of shop trial".

BHP = IHP × η mesh

η mesh : mechanical efficiency (Official shop test result of engine)

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APPENDIX 3

Correction of Performance Parameters

1. General

Some measured performance parameters need to be corrected to ISO ambient conditions to facilitate reliable evaluation.

These parameters are: p_{max}, t_{exhv}, p_{comp} and p_{scav}. *See also 'Performance Observations', page 706.03.*

Making such corrections enables comparison to earlier (corrected) readings or model curves, regardless of deviations of the actual t_{inl} and $t_{coolinl}$ from reference conditions.

I.e. the correction provides the values which would have been measured if t_{inl} and $t_{coolinl}$ had been 25°C.

In extreme cases, the divergencies can be large.

Record the corrected value as described in Section, 'Evaluation of Records' in this Chapter. *See page 706.05.*

Use the following reference conditions:

t_{inl} = Air inlet temperature = 25°C

(The air inlet temperature can vary greatly, depending on the position in which it is measured on the intake filter. Experience has shown that two thermometers situated at ten o'clock and four o'clock positions (i.e. 180° apart) and at the middle of the filter, give a good indication of the average temperature).

$t_{coolinl}$ = Cooling water inlet temp. to air cooler = 25°C.
See also Plate 70610, regarding Δt (t_{scav}-$t_{coolinl}$).

See also Item 1 'Symbols and Limits', earlier in this Chapter.

2. Correction

The correction for deviations of t_{inl} and $t_{coolinl}$ from reference conditions can be carried out in two ways:

By reading

See Plate 70624, which shows how to use Plates 70620-70623 to determine the correction.

By calculation

The corrections can be determined by the general equation:

$$A_{corr} = (t_{meas} - t_{ref}) \times F \times (K + A_{meas})$$

where

A_{corr} = the correction to be applied to the parameter, i.e. to p_{max}, t_{exh}, p_{comp} or p_{scav}.

t_{meas} = measured t_{inl} or $t_{coolinl}$.

t_{ref} = reference t_{inl} or $t_{coolinl}$ (in case of Standard Conditions, 25°C).

F_1, F_2 = constants, *see the table below.*

K = constant, *see the table below.*

A_{meas} = the measured parameter to be corrected, i.e. p_{max}, t_{exh}, p_{comp} or p_{scav}.

See Plates 70620, 70621, 70622 and 70623, which show how to use the formulas.

Parameter to be corrected	F_1: for air inlet temp.	F_2: for cooling water inlet temp.	K
t_{exhv}	-2.466×10^{-3}	-0.59×10^{-3}	273
p_{scav}	$+2.856 \times 10^{-3}$	-2.220×10^{-3}	p_{baro} 1 bar or 750 mm H_g
p_{comp}	$+2.954 \times 10^{-3}$	-1.530×10^{-3}	p_{baro} 1 bar or 750 mm H_g
p_{max}	$+2.198 \times 10^{-3}$	-0.810×10^{-3}	p_{baro} 1 bar or 750 mm H_g

3. Examples of calculations:

See Plate 70624, which states a set of service readings.

1) Correction of t_{exhv} *(Plate 70621).*

Measured:
Exh. temp. after valve = 425°C
Air inlet temp. = 42°C
Cool. w. inlet temp.(air cooler) = 40°C

Correction for air inlet temp.:
$(42-25)\times(-2.466\times10^{-3})\times(273+425) = -29.3°C$

Correction for cooling water inlet temp.:
$(40-25)\times(-0.59\times10^{-3})\times(273+425) = -6.2°C$

Corrected t_{exhv} value = 425–29.3–6.2 = 389.5°C

2) Correction of p_{scav} *(Plate 70623):*

Measured:
Scav. air pressure = 2.0 bar
Air inlet temp. = 42 °C
Cool. w. inlet temp.(air cooler) = 40 °C

Correction for air inlet temp.:
$(42-25)\times(2.856\times10^{-3})\times(1+2.0) = 0.146$ bar

Correction for cooling water inlet temp.:
$(40-25)\times(-2.220\times10^{-3})\times(1+2.0) = -0.10$ bar

Corrected p_{scav} value
= 2.0+0.146–0.10 = 2.046 bar

Alternatively, if p_{scav} is measured in mmH_g:

Scavenge air pressure = 1500 mmH_g

Correction for t_{inl}:

$(42-25)\times(2.856\times10^{-3})\times(750+1500) =$ 109.2 mmH_g

Correction for $t_{coolinl}$:
$(40-25)\times(-2.220\times10^{-3})\times(750+1500) =$ -74.9 mmH_g

Corrected p_{scav} value
= 1500+109.2–74.9 = 1534.3 mmHg

Corrections of p_{comp} *(Plate 70622)* and p_{max} *(Plate 70620)* can be made in a similar manner.

4. Maximum Exhaust Temperature

The engine is designed to allow a limited increase of the thermal loading, i.e. increase of t_{exhv}.

This enables the engine to operate under climatic alterations and under normally deteriorated service condition.

Whether the engine exceeds this built-in safety margin for thermal loading can be evaluated as follows:

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The factors contributing to increased exhaust temperature levels (and thereby thermal loads) and the largest permissible deviation values are:

Factor	Max. temp. increase
• due to fouling of turbocharger (incl. air intake filters), and exhaust uptake, *see also Chapter 701, Item 433A*	+ 30°C
• due to fouling of air coolers	+ 10°C
• due to deteriorated mechanical condition (estimate)	+ 10°C
• due to climatic (ambient) conditions	+ 45°C
• due to operation on heavy fuel, etc.	+ 15°C
Total	110 °C

Regarding increasing exhaust temperatures, *see also – 'Evaluation of Records', point 2.2, page 706.07.*

For new engines it is not unusual to observe a temperature increase of 50–60°C from the shop test to the sea trial.

This is due to the operation on heavy fuel oil and altered climatic conditions.

If the temperature increases further during service:

- Find the cause of the temperature increase.
- Clean, repair or overhaul the components in question at the first opportunity, to improve the engine performance.

Note: The exhaust temperature must not exceed the alarm limit, *see Chapter 701, Item 427.*

To evaluate the exhaust temperature correctly, it is important to distinguish between:

- Exhaust temperature increase due to fouling and mechanical condition, and
- Exhaust temperature increase due to climatic alterations.

The method to distinguish between the factors is shown in the example:

Example:

According to a model curve, the exhaust temperature (approx. 95% engine load) should be 375°C.

The observed exhaust temperature is 425°C.

<u>Correct t_{exhv} according to Plate 70621:</u>

Air inlet temp. (t_{inl}) = **42°C** corresponding to (42–25) = 17°C above the reference value.

Cooling water inlet temp. to the air cooler ($t_{coolinl}$) = **40°C**, corresponding to (40–25) = 15°C above the reference value.

Using the curves, the following temperature corrections are obtained:

Correction due to increased engine room temperature:	-27.0°C
Correction due to increased cooling water inlet temp.	-6.0°C
Total	-33.0°C

<u>Distinguish between the factors:</u>

The total exhaust temperature increase of 425°C–375°C = 50°C, is caused by:

- an increase of 33.0°C on account of climatic alterations,
- an increase of 50°C–33°C = 17°C, due to mechanical conditions and operation on heavy fuel oil.

APPENDIX 4

Turbocharger Efficiency

1. General

To record the turbocharger efficiencies, see 'Evaluation of Records' point 3 'Turbocharger synopsis' earlier in this Chapter.

Plate 70609 shows model curves for compressor and turbine efficiencies, based on the scavenge air pressure.

For general evaluation of the engine performance, it is unnecessary to calculate turbocharger efficiencies.

However, if such calculations are desired, they can be carried out as described below.

2. Calculating the Efficiencies

The total turbocharger efficiency is the product of the compressor, turbine, and mechanical efficiencies.

However, the last one has almost no effect on the efficiency calculations, and is therefore omitted.

When calculating the turbocharger efficiency, it is necessary to distinguish between:

- Plants without turbo compound system (TCS) and exhaust by-pass.
- Plants with TCS and/or exhaust by-pass.

2.1 Plants without TCS and Exhaust By-Pass

Measure the parameters listed in Table 1.

It is essential that, as far as possible, the measurements are taken simultaneously.

Convert all pressures to the same unit.

Use the following conversion factors:

750 mm Hg = 1.000 bar = 0.1 MPa
1 mm H_2O = 0.0001 bar
1 kp/cm^2 = 735 mm Hg = 0.98 bar
1 bar = 0.1 MPa
π = 3.14159

		Unit	Examples of Measurements		
Barometric pressure	p_{baro}	mm Hg or bar	766.5/750	=	1.022 bar
Pressure drop, air filter	Δp_f	mm H_2O or bar	21 × 0.0001	=	0.002 bar
Pressure drop, air cooler	Δp_c	mm H_2O or bar	168 × 0.0001	=	0.017 bar
Temperature before compr.	t_{inl}	°C		=	21°C
Turbocharger speed	n	rpm		=	13350 rpm
Scavenge air pressure	p_{scav}	mm Hg or bar	1900/750	=	2.533 bar *)
Exhaust receiver pressure	p_{exh}	mm Hg or bar	1795/750	=	2.393 bar *)
Pressure after turbine	p_{atc}	mm H_2O or bar	265 × 0.0001	=	0.026 bar *)
Temperature before turbine	t_{btc}	°C		=	400°C

*) "Gauge" Pressure

Table 1: Measurements for calculation of efficiencies

Note that the official designation of bar is "absolute pressure".

Total Efficiency:

The total efficiency η_{tot} is given by the equation

$$\eta_{tot} = 0.9055 \frac{T_1 (R_1^{0.286} - 1)}{T_2 (1 - R_2^{0.265})}$$

	Example of Calculation, η_{tot} *See measurements in Table 1*		
$T_1 = t_{inl} + 273$	$21 + 273$	=	294 °K
$R_1 = \frac{p_{baro} + p_{scav} + \Delta p_c}{p_{baro} - \Delta p_f}$	$\frac{1.022 + 2.533 + 0.017}{1.022 - 0.002}$	=	3.502
$T_2 = t_{btc} + 273$	$400 + 273$	=	673 °K
$R_2 = \frac{p_{baro} + p_{atc}}{p_{baro} + p_{exh}}$	$\frac{1.022 + 0.026}{1.022 + 2.393}$	=	0.307
$(R_1^{0.286} - 1)$ ★		=	0.4311
$(1 - R_2^{0.265})$ ★		=	0.2688
$\eta_{tot} = \frac{0.9055 \times T_1 (R_1^{0.286} - 1)}{T_2 (1 - R_2^{0.265})}$	$\frac{0.9055 \times 294 \times 0.4311}{673 \times 0.2688}$	=	<u>0.634</u>

Compressor efficiency:

The compressor efficiency η_{compr} is given by the equation

$$\eta_{compr} = \frac{3614400 \times T_1 (R_1^{0.286} - 1)}{\mu \times U^2}$$

μ = slip factor, *see Plate 70628*

$U^2 = (\pi \times D \times n)^2$

D = Diameter of compressor wheel, *see Plate 70628*

$U = \pi \times D \times n$ is the peripheral speed of the compressor wheel.

The turbocharger used in this example is an MAN B&W, type NA57/TO7.

From Plate 70628 is taken:
D = 0.656 m
$\mu = 0.77$

	Example of Calculation, η_{compr} *See measurements in Table 1*		
$T_1 = t_{inl} + 273$ °K	$21 + 273$	=	294 °K
$R_1 = \frac{p_{baro} + p_{scav} + \Delta p_c}{p_{baro} - \Delta p_f}$	$\frac{1.022 + 2.533 + 0.017}{1.022 - 0.002}$	=	3.502
$(R_1^{0.286} - 1)$ ★		=	0.4311
$U^2 = (\pi \times D \times n)^2$	$(\pi \times 0.656 \times 13350)^2$	=	757000000
$\eta_{compr} = \frac{3614400 \times T_1 (R_1^{0.286} - 1)}{\mu \times U^2}$	$\frac{3614400 \times 294 \times 0.4311}{0.77 \times 757000000}$	=	<u>0.786</u>

★ *Determine the values of the expressions $(R_1^{0.286} - 1)$ and $(1 - R_2^{0.265})$.*
Use a mathematical calculator or use the curves in Plates 70625 and 70626.

Turbine efficiency:

The turbine efficiency η_{turb} appears from

$\eta_{total} = \eta_{compr} \times \eta_{turb}$

i.e. $\eta_{turb} = \dfrac{\eta_{total}}{\eta_{compr}} = \dfrac{0.634}{0.786} = \underline{\underline{0.807}}$

2.2 Plants with TCS and/or Exhaust By-Pass

The equation $\eta_{tot} = 0.9055 \dfrac{T_1 (R_1^{0.286} - 1)}{T_2 (1 - R_2^{0.265})}$ stated in item 2.1 is based on a situation where the mass flow through the turbine is equal to the mass flow through the compressor plus the fuel oil amount.

If a TCS or an exhaust by-pass is fitted, the mass flow through the turbine is reduced by the mass flow through the TCS or the exhaust by-pass.

The mass flows through the turbine and the TCS or through the turbine and the exhaust by-pass are proportional to the effective areas in the turbines or the orifice in the exhaust by-pass.

Calculate the turbocharger efficiency as described in Item 2.1 'Plants without TCS and exhaust by-pass'.

Then correct the results in accordance with the following:

Total efficiency:

$$\eta_{tot\ TCS/by\text{-}pass} = \eta_{tot} \times \frac{A_{eff} + a_{eff}}{A_{eff}}$$

where

A_{eff} = Effective area in turbocharger turbine
a_{eff} = Effective area in TCS or exhaust by-pass.

See also 'Remarks', below

Turbine Efficiency:

$$\eta_{turb\ TCS/by\text{-}pass} = \eta_{turb} \times \frac{A_{eff} + a_{eff}}{A_{eff}}$$

See also 'Remarks', below

Compressor Efficiency:

η_{compr} is unchanged, as it is not affected by whether the plant operates with TCS/by-pass or not.

Remarks

The relation $\dfrac{A_{eff} + a_{eff}}{A_{eff}}$ can vary from plant to plant, but is most often about 1.07. This value can be used when evaluating the trend of the efficiency in service.

When using a computer program in which the relation $\dfrac{A_{eff} + a_{eff}}{A_{eff}}$ is not introduced, the value for η_{tot} and η_{turb} will have to be multiplied by the above-mentioned factor of about 1.07.

APPENDIX 5

Estimation of the Effective Engine Power without Indicator Diagrams

1. General

The estimation is based on nomograms involving engine parameter measurements taken on testbed.

The nomograms are shown in *Plate 70627.* The following relationships are illustrated:

Chart I – fuel pump index and mean effective pressure.

Chart II – mean effective pressure and effective engine power (BHP), with the engine speed as a parameter.

Chart III – turbocharger speed and effective engine power (BHP), with the scavenge air temperature and ambient pressure as parameters.

A condition for using these charts is that the engine timing and turbocharger matching are unchanged from the testbed.

2. Methods

(See Plate 70627)

2.1 Fuel Pump Index

(an approximate method)

Chart I: draw a horizontal line from the observed fuel pump index to the nomogram curve, and then a vertical line down to the observed engine speed on Chart II. From this intersection a horizontal line is drawn to the effective engine power scale, i.e. 16,400 BHP.

This method should only be used as a quick (rough) estimation, because the fuel oil, as well as the condition of the fuel pump, may have great effect on the index. In particular, worn fuel pumps or suction valves tend to increase the index, and will thus result in a too high power estimation.

2.2 Turbocharger Speed

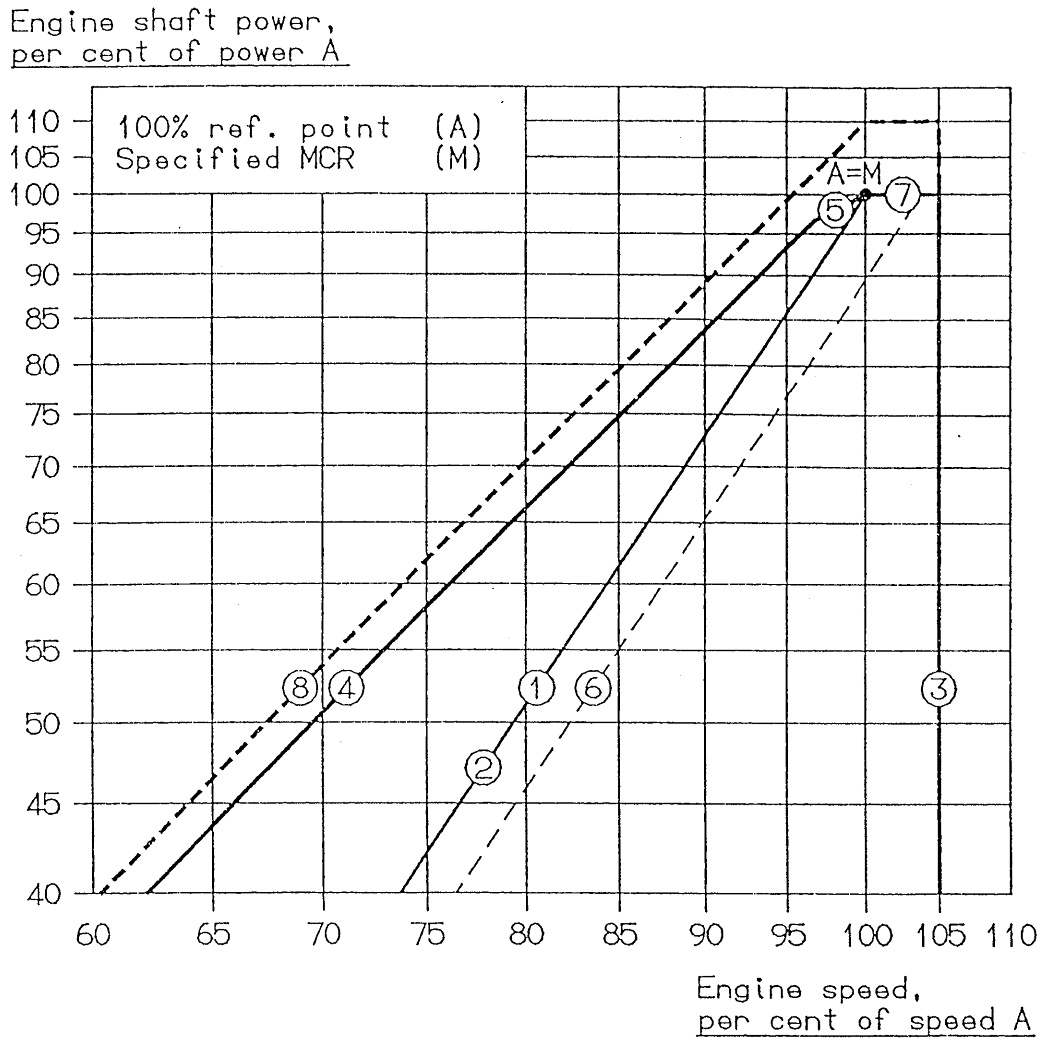
(A more accurate method)

Chart III: draw a horizontal line from the observed t_{scav} value and an inclined line from the observed turbocharger speed.

From the intersection point, draw a vertical line down to the nomogram curve and then a horizontal line to the vertical line from the observed ambient pressure (point x in the ambient pressure scale).

Finally, a line is drawn parallel with the inclined 'ambient pressure correction' lines. The effective engine power can then be read on the scale at the right hand side, i.e. 15,700 BHP.

This method is more reliable, and an accuracy to within ± 3% can be expected. However, the accuracy obtained will depend on the condition of the engine and turbocharger. A fouled or eroded turbocharger will in most cases tend to decrease the turbocharger speed, and thus result in a too low power estimation.
This situation is characterized by increased exhaust gas temperatures and a decreased scavenge air pressure.



Line 1: Propeller curve through point A.

Line 2: Propeller curve - heavy running, recommended limit for fouled hull at calm weather conditions.

Line 3: Speed limit.

Line 4: Torque/speed limit.

Line 5: Mean effective pressure limit.

Line 6: Propeller curve - light running (range: 2.5-5.0%), for clean hull and calm weather conditions.

Line 7: Power limit for continuous running.

Line 8: Overload limit.

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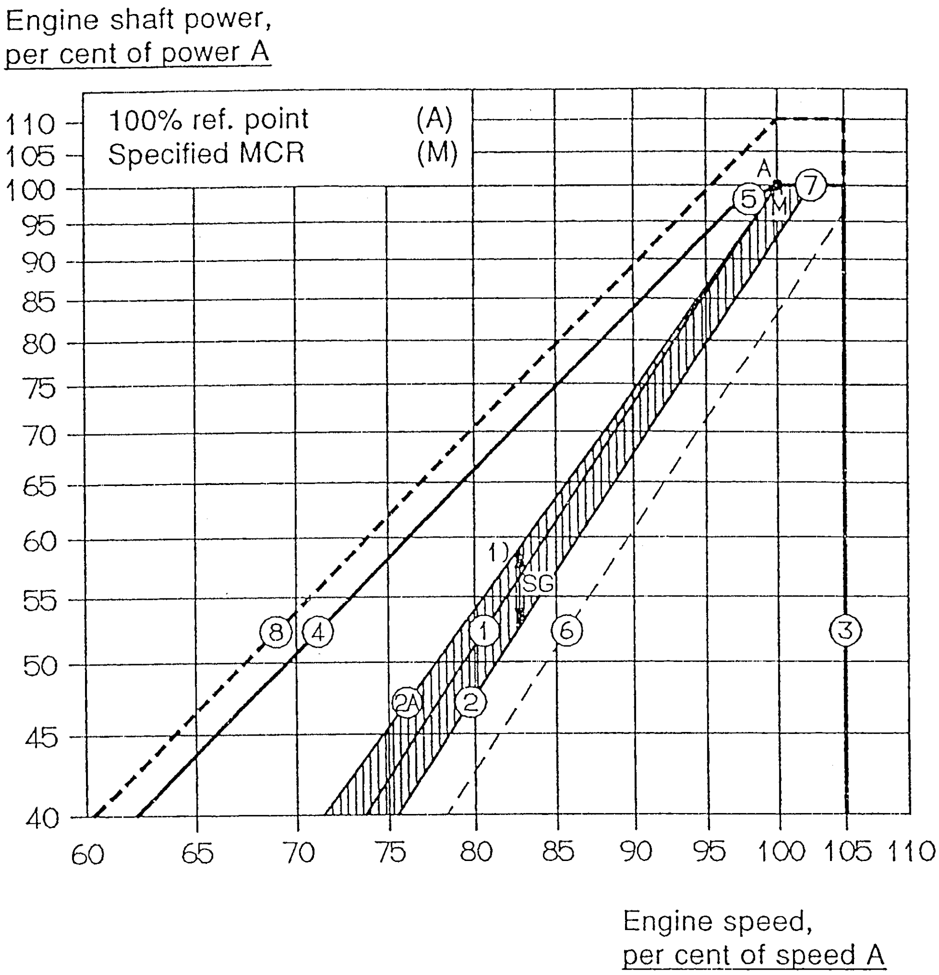
Plate 70602-40 **Load Diagram for Propulsion and Main Engine Driven Generator**



Line 1 : Propeller curve through point A.

Line 2 : Propeller curve for propulsion alone – heavy running, recommended limit for fouled hull at calm weather conditions.

Line 2A : Engine service curve for propulsion (line 2) and shaft generator (SG).

Line 3 : Speed limit.

Line 4 : Torque/speed limit.

Line 5 : Mean effective pressure limit.

Line 6 : Propeller curve for propulsion alone – light running (range: 2.5–5.0%), for clean hull and calm weather conditions.

Line 7 : Power limit for continuous running.

Line 8 : Overload limit.

1) Note : The propeller curve for propulsion alone is found by subtracting the actual shaft generator power (incl. generator efficiency) from the effective engine power at maintained speed.

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Performance Observations

Plate 70603-40
Page 1 (2)

M/V	Engine Type:		Engine Layout:	Checked by:
Yard:	Builder:	Engine	BHP:	
No.:	Built year:	No.:	r/min:	Date:

Turbocharger(s)		Serial No.	
Make:	1		Cylinder constant (HP, bar)
Type:	2		Governor: Type:
Max. RPM:	3		TC specification:
Max. temp. °C:	4		

Lub. oil system: Internal system ☐ External from M. E. system ☐ External from gravity tank ☐

Fuel oil viscosity at °C			Brand	Type
Bunker station		Cylinder oil		
Oil brand	Heat value Kjoule/kg:	Circulating oil		
Density at 15°C:	Sulphur %:	Turbo oil		

Date	Draught fore m	Total running hours	RPM	Wind m/s	Direct
Hour	Draught aft m	Speed setting bar	kW indicated	Wave height m	Direct
Load %	Log Knots	Governor index	kW effective	g/kWh effective	
Barom. millibar	Obs. Knots	Pmaxcontr. pressure bar	BHP	g/BHPh	

P_i bar			P_{max} bar			P_{comp} bar			Fuel pump index		
1	2	3	1	2	3	1	2	3	1	2	3
4	5	6	4	5	6	4	5	6	4	5	6
7	8	9	7	8	9	7	8	9	7	8	9
10	11	12	10	11	12	10	11	12	10	11	12
Average			Average			Average			Average		

P_{max} adjustment index			Exhaust gas temp. °C: Exhaust valve			Turbine Inlet	Turbine Outlet
1	2	3	1	2	3	1	1
4	5	6	4	5	6	2	2
7	8	9	7	8	9	3	3
10	11	12	10	11	12	4	4
Average			Average			Remarks:	

Exh. press. Receiver mmHg	Exh. press. Turbine outlet mmWC	Turbo-charger RPM	Scav. air pressure Δp Filter mmWC	Scav. air pressure Δp Cooler mmWC	Scav. air pressure Receiver	Scav. air temp. °C Inlet blower	Scav. air temp. °C Before cooler	Scav. air temp. °C After cooler	Auxiliary blower
	1	1	1	1	bar	1	1	1	On
	2	2	2	2	mmHg	2	2	2	Off
	3	3	3	3		3	3	3	
					°C				
	4	4	4	4		4	4	4	

Special points: MAN TC p at spiral housing outer dia. mm Hg	Special points: MAN TC/ BBC TC Δp inner and outer dia. mm WC	Cooling water temperature °C: Air cooler Inlet	Air cooler Outlet	Main engine Inlet	Main engine Outlet cylinders			Turbine Outlet
1	1	1	1		1	2	3	1
2	2	2	2	Seaw. temp.	4	5	6	2
3	3	3	3		7	8	9	3
4	4	4	4		10	11	12	4
					Average			

Lubricating oil: Press. bar	Temperature °C	Outlet pistons			Turbochargers: MAN TC inlet/ BBC TC blower end	MAN TC outlet/ BBC TC turbine end	Fuel oil pressure bar
System oil	Inlet engine	1	2	3	1	1	Before filter
Cooling oil	Inlet camshaft	4	5	6	2	2	After filter
Camshaft oil	Outlet camshaft	7	8	9	3	3	Temp. °C Before pumps
Turbine oil	Thrust segments	10	11	12	4	4	
		Average					

Turn

Performance Observations

Power turbine TCS	No.	Make	Type	Serial No.	Remarks:
	1				
	2				
PT specification:					

PTO	No.	Generator make	Type	Volt	Amp.	Effect kW	Crankshaft BHP	DMG		PTO	
	1							RCF		PTI	
	2							RKRV		TCS	

	Exhaust gas temp. °C		Exh. pressure mm Hg	Exh. pressure mm WC	Sealing air pressure bar	TCS RPM	TCS		By-pass		Cooling water temp. °C		Lub. oil pressure bar		Lub. oil temp. °C	
No.	TCS inlet	TCS outlet	TCS inlet	TCS outlet	TCS	(if available)	On	Off	Open	Closed	TCS inlet	TCS outlet	Before filter	After filter	TCS inlet	TCS outlet
1																
2																

Remarks:

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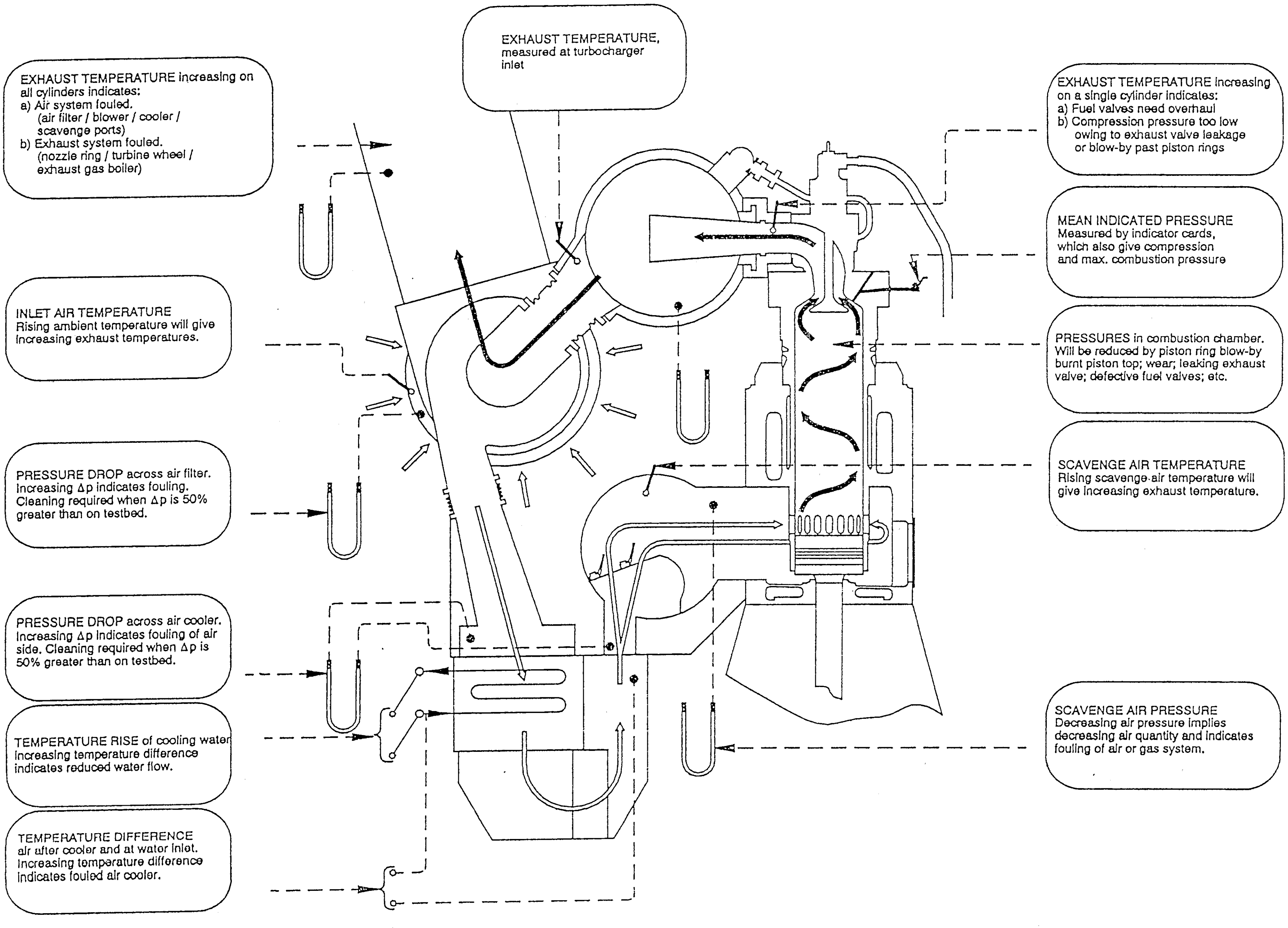
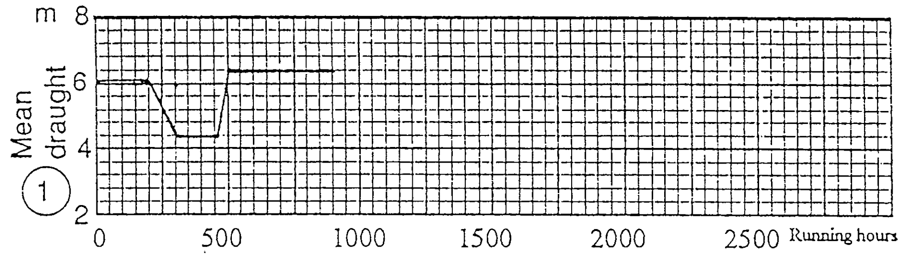


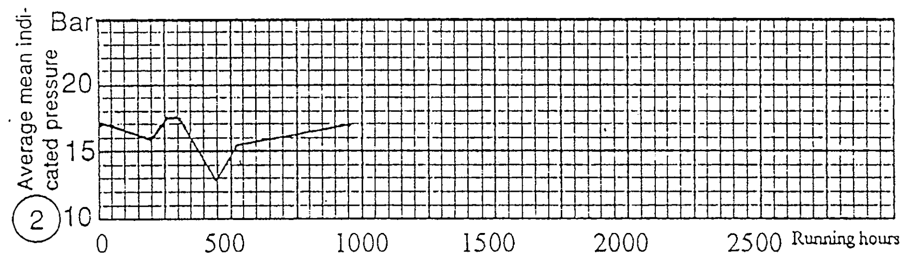
Readings relating to Thermodynamic Conditions — Plate 70604-40

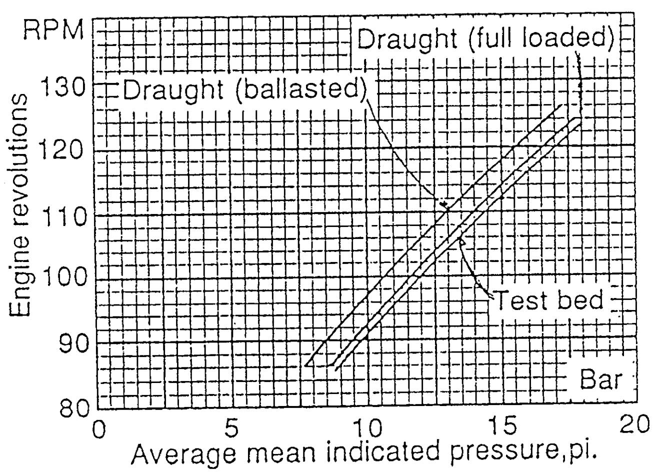
M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:

All the model Curves are based on Test results from Shop-trial or trial trip.

Directions for use:
Service results are plotted faintly in the Model Curve diagrams. The vertical deviations are transferred to the pertaining Time based Deviation-chart (on the right hand side).







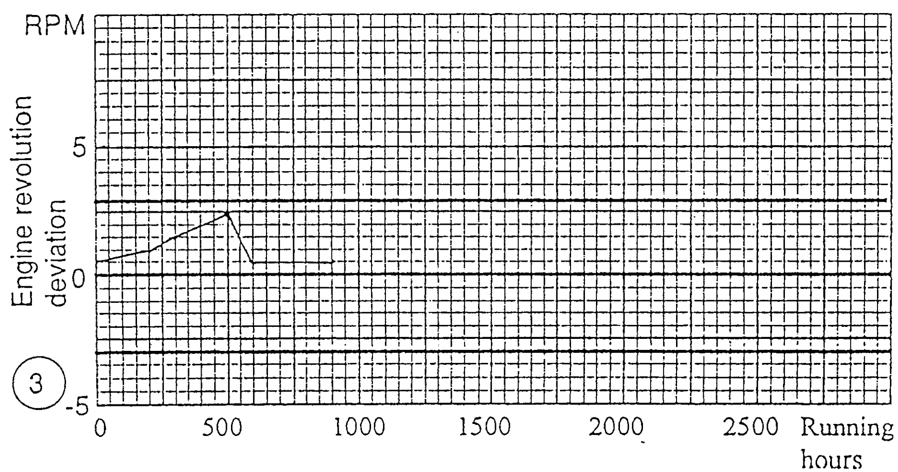
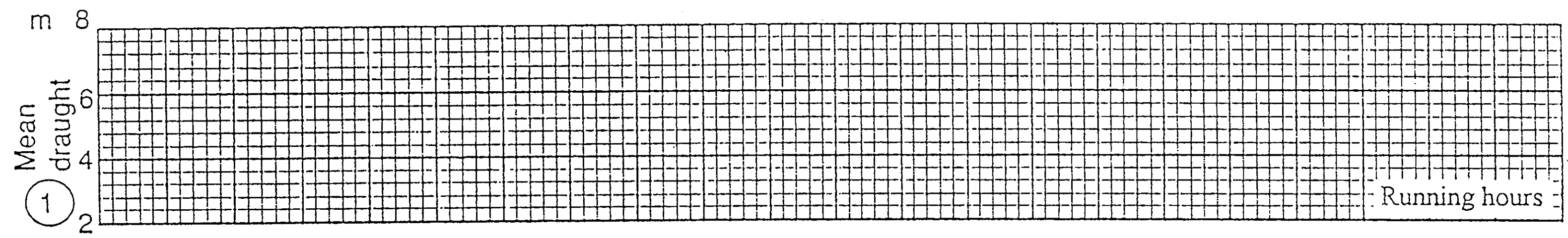
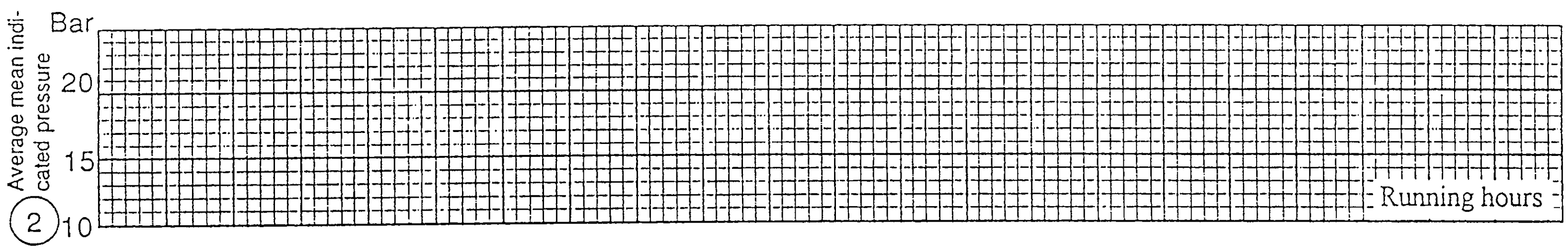


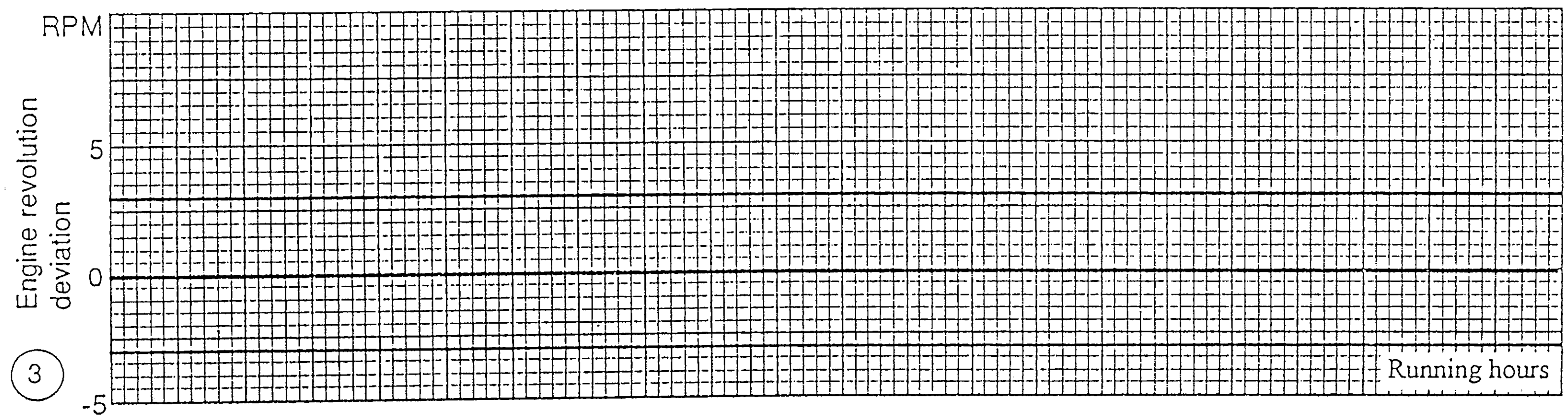
Plate 70605-40 Synopsis Diagrams – for engine

Time based deviation charts for: mean draught and average mean indicated pressure (p_i).
Model curves + time based deviation chart for: r/min as a function of p_i

M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:







Synopsis Diagrams – for engine

Time based deviation charts for: mean draught average mean indicated pressure (p_i) and r/min

Plate 70605-40

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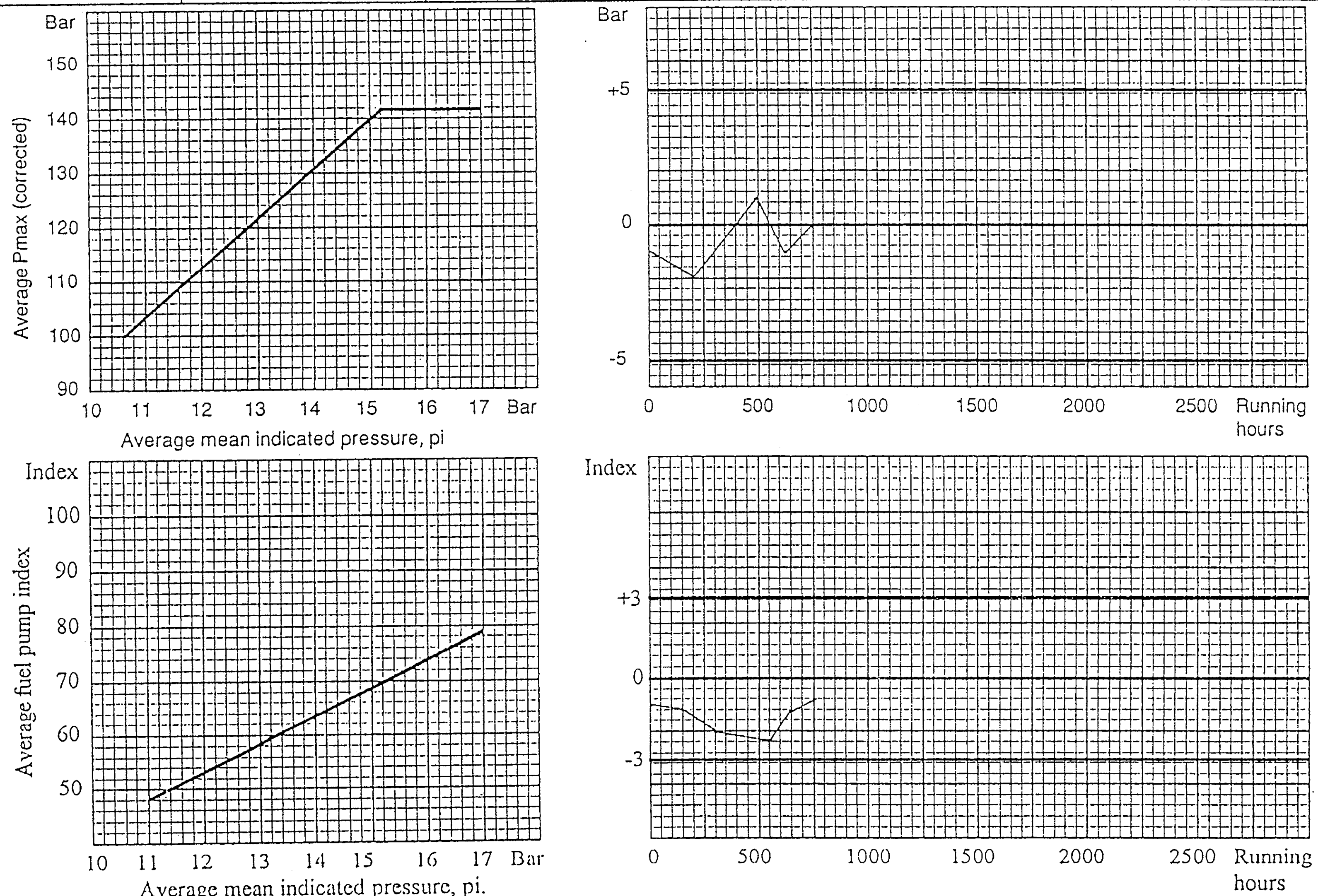
Plate 70606-40

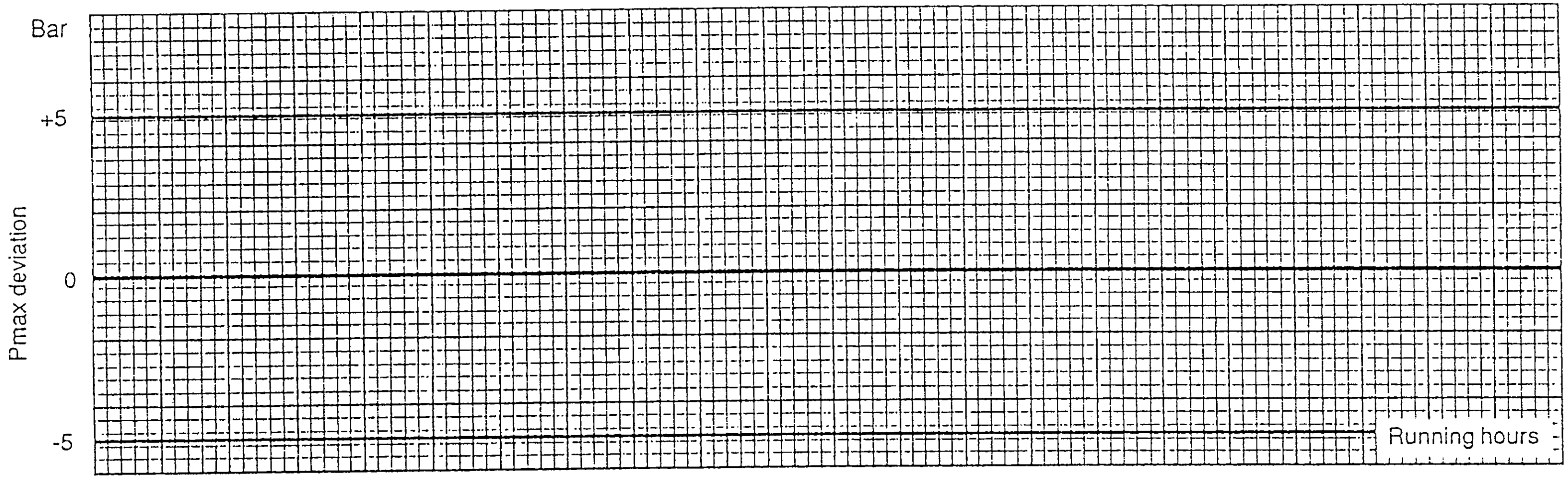
Synopsis Diagrams – for engine

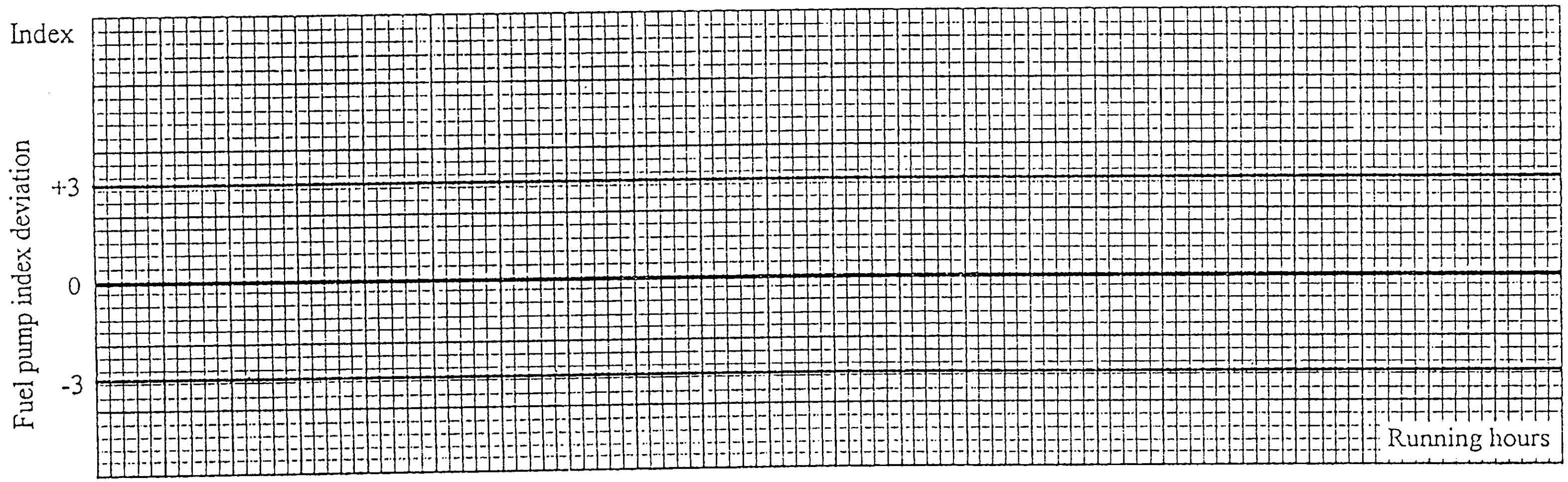
Model curves and time based deviation charts for:
p_{max} and fuel pump index as a function of p_i

M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:



M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:





Synopsis Diagrams – for engine

Time based deviation charts for:
p_{max} and fuel pump index

Plate 70606-40
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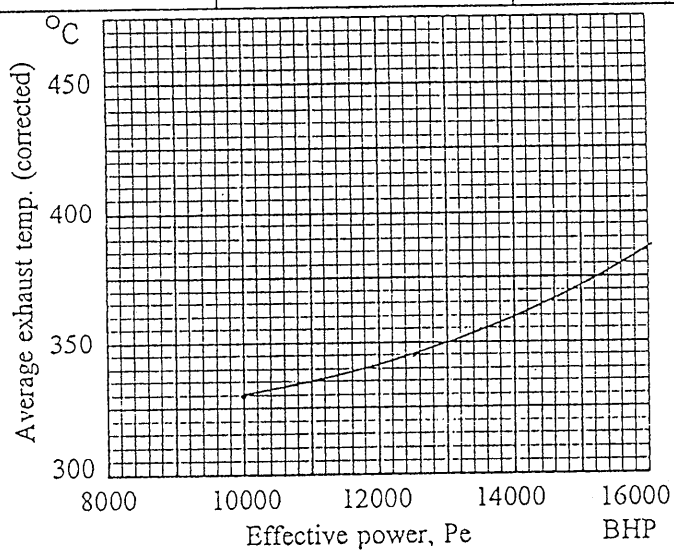
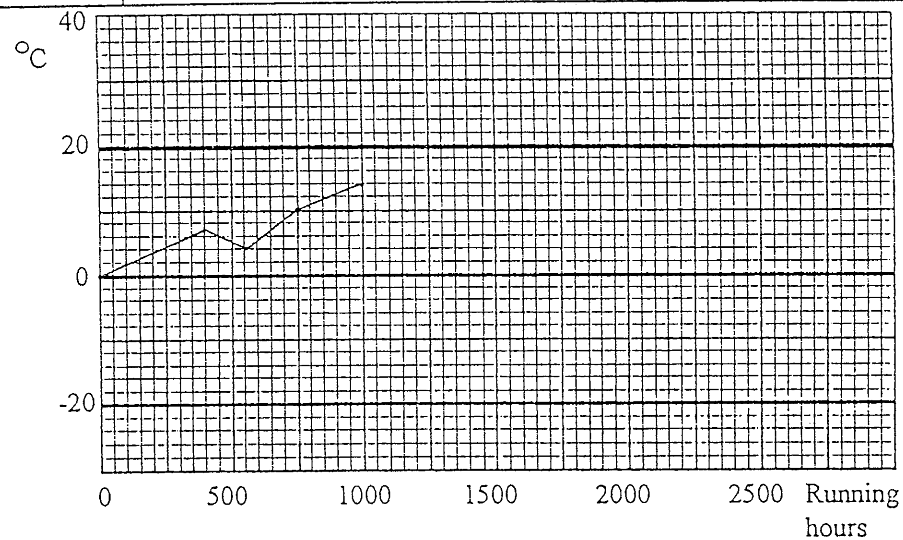
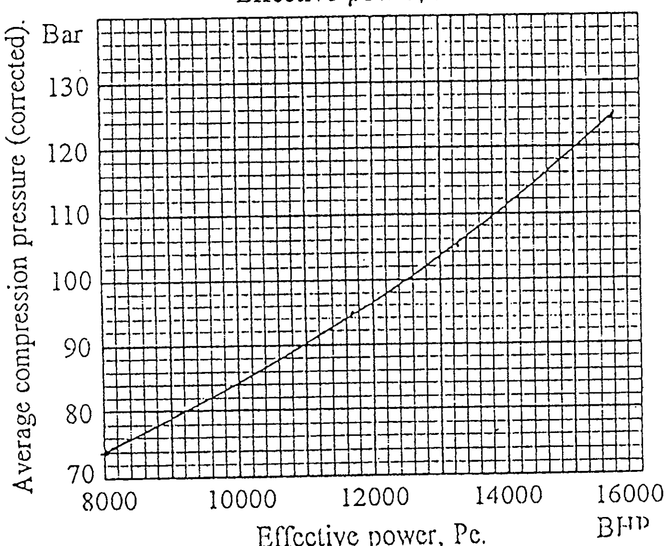
Plate 70607-40 Synopsis Diagrams – for engine

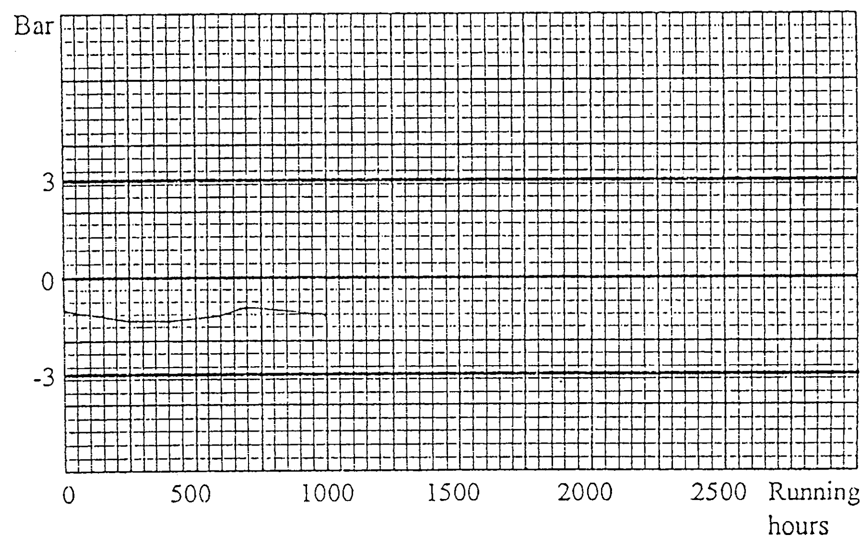
Model curves and time based deviation charts for:
t_{exhv} and p_{comp} as a function of p_e

M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:

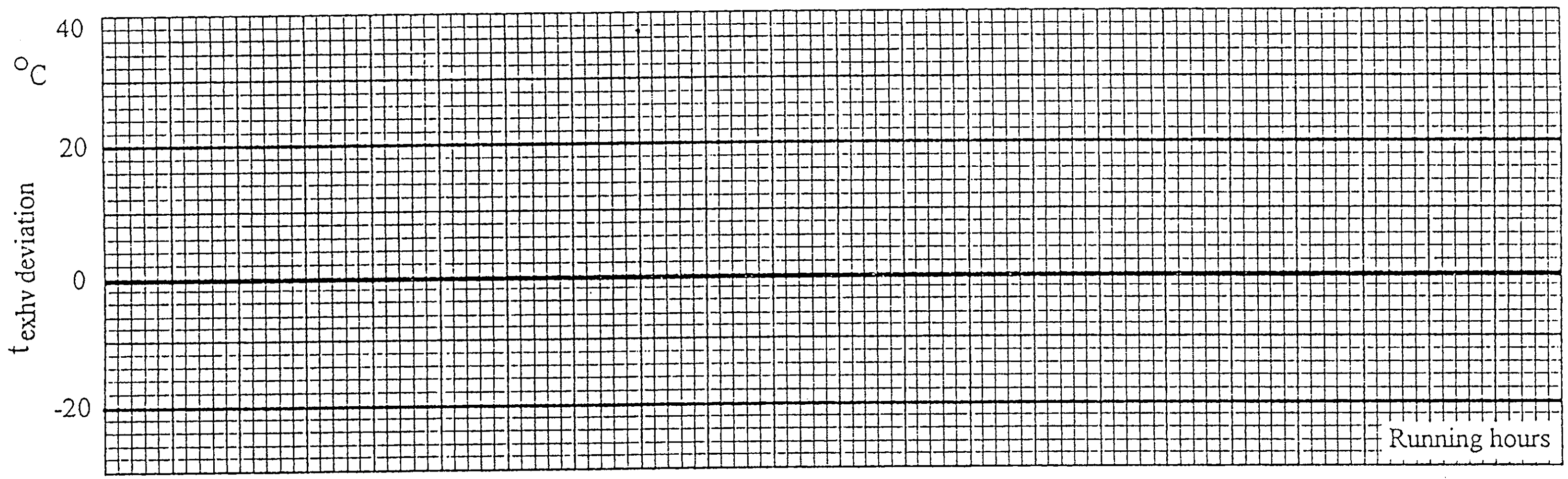


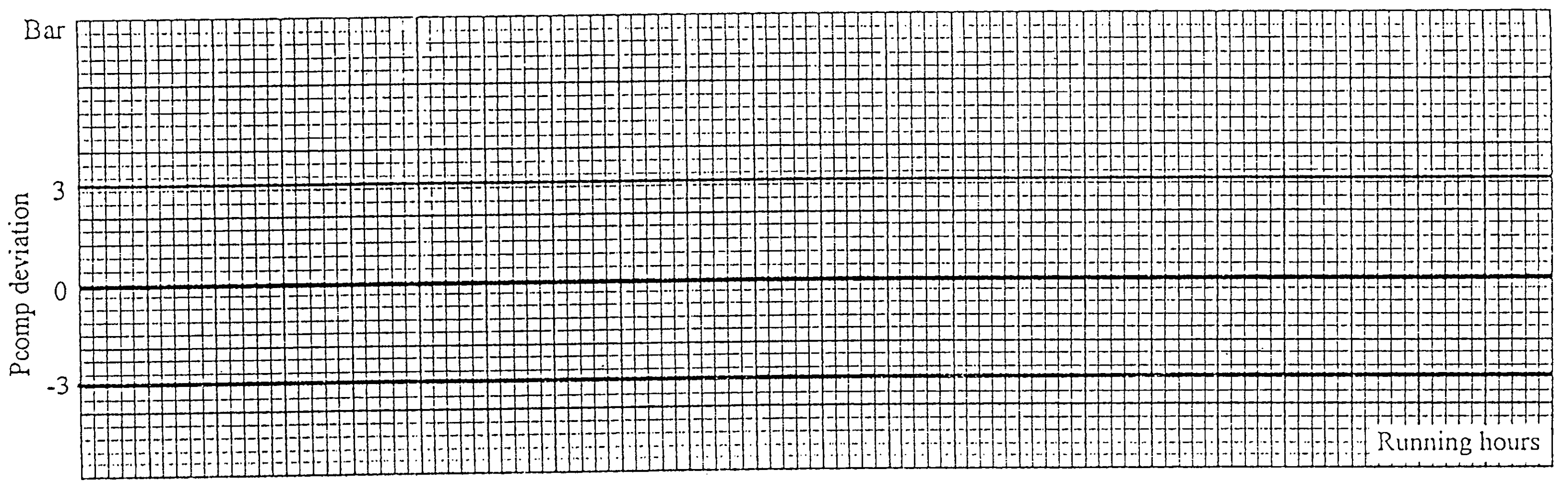






M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:





Synopsis Diagrams – for engine

Time based deviation charts for: t_{exhv} and p_{comp}

Plate 70607-40

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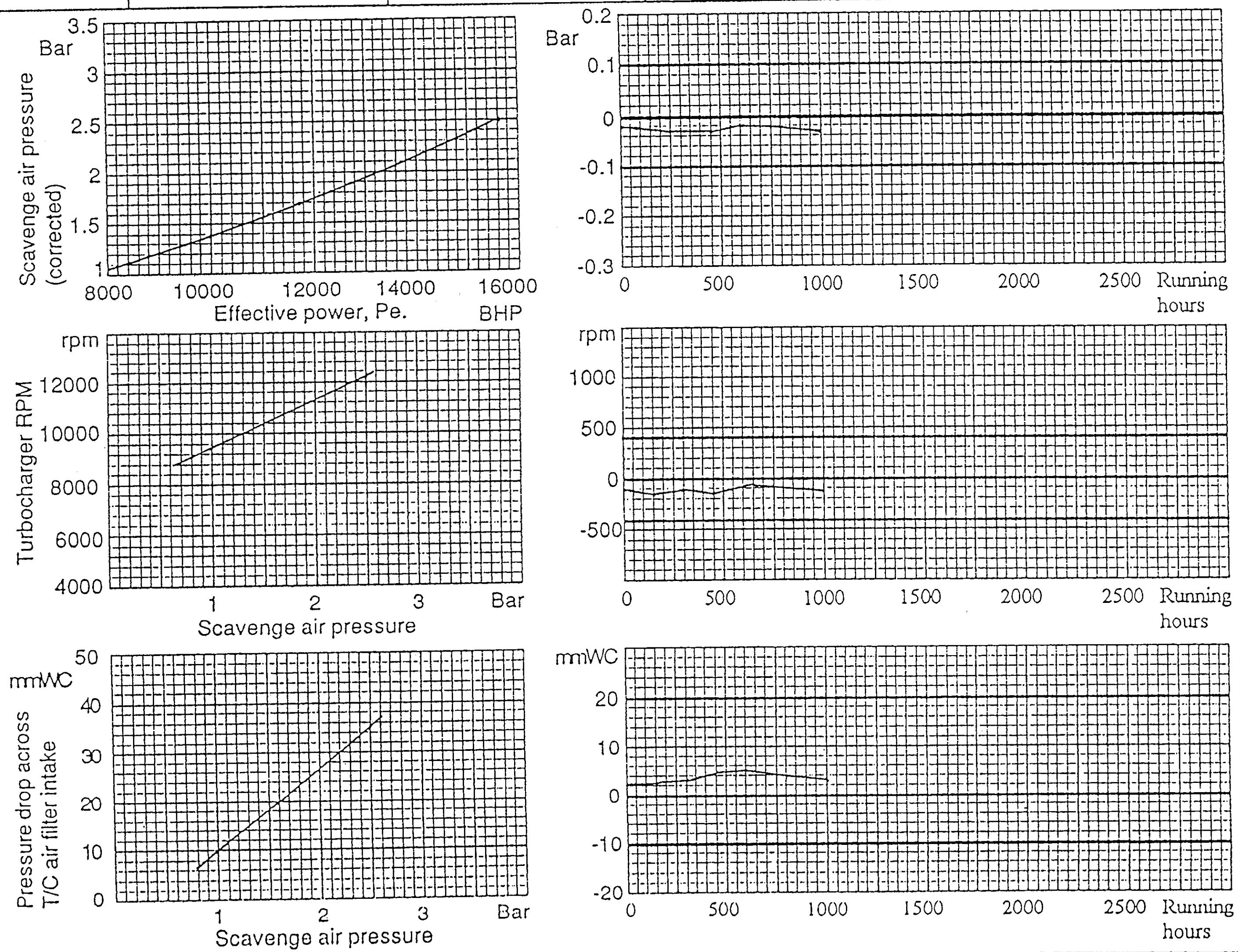
Plate 70608-40

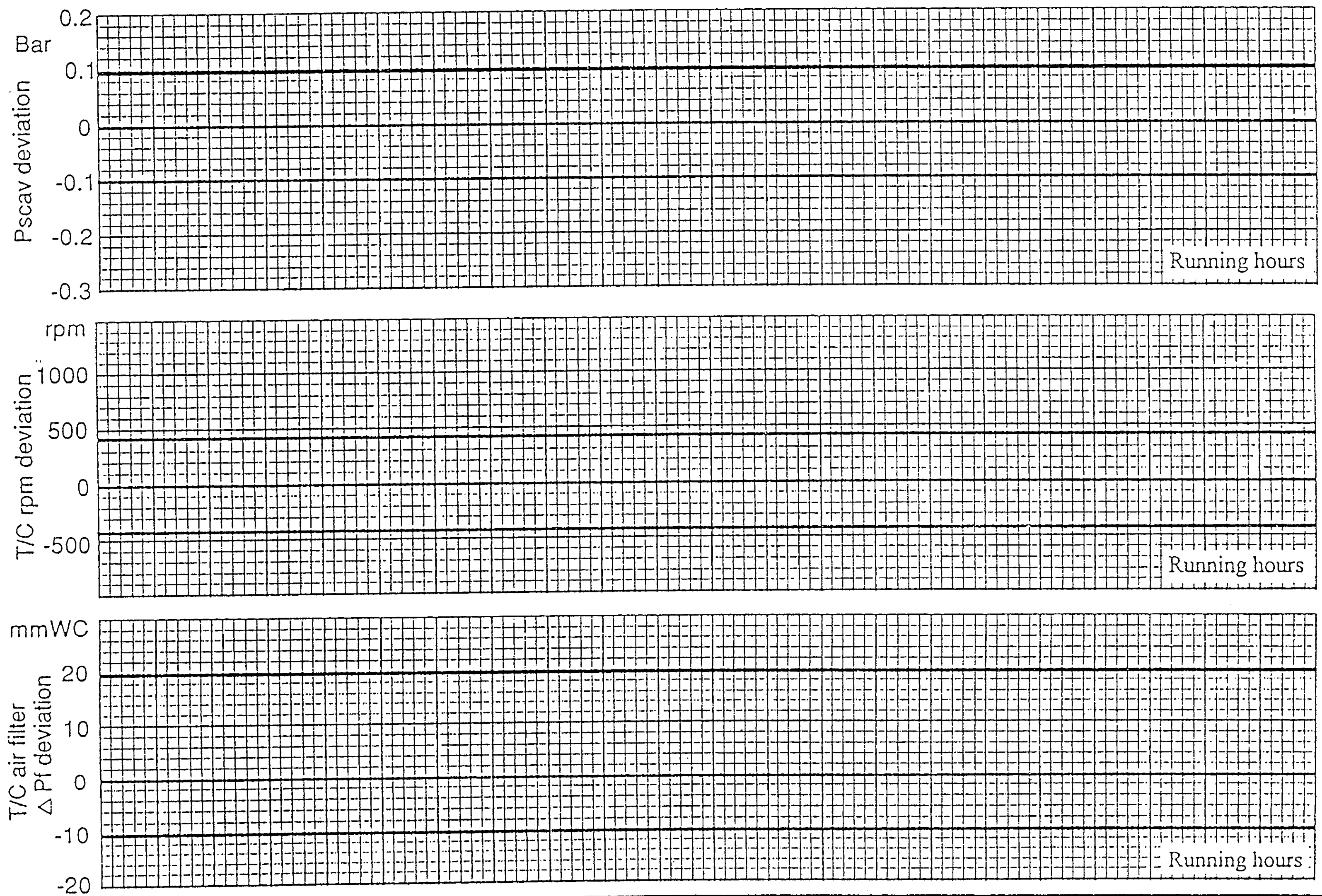
Synopsis Diagrams – for turbocharger

Model curves and time based deviation charts for:
p_{scav} as a function of p_e
T/C r/min and Δp_f as a function of p_{scav}

M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:



M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:



Synopsis Diagrams – for turbocharger

Time based deviation charts for: p_{scav}, T/C r/min and Δp_f

Plate 70608-40
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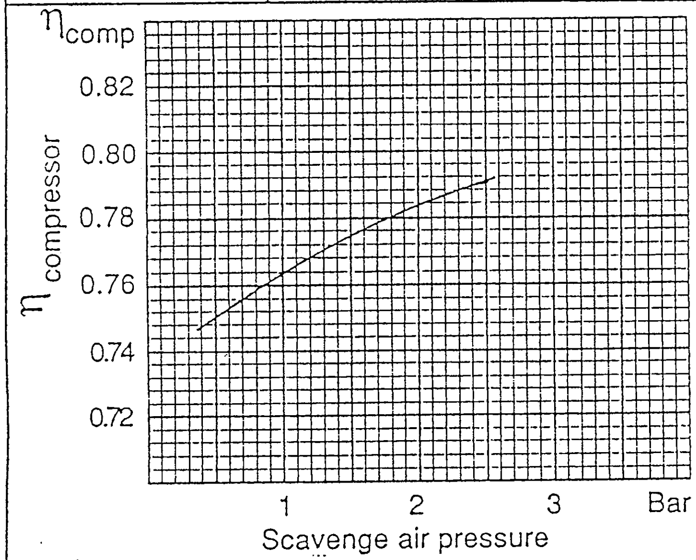
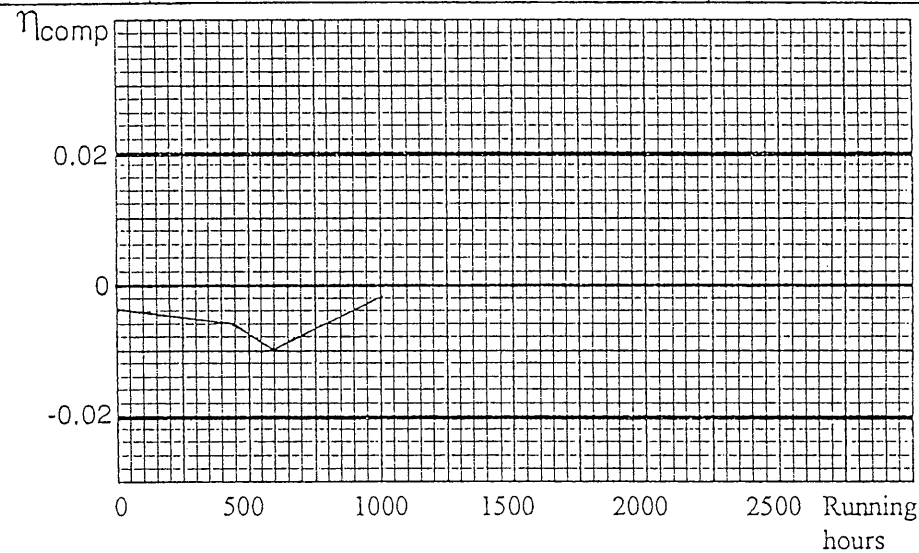
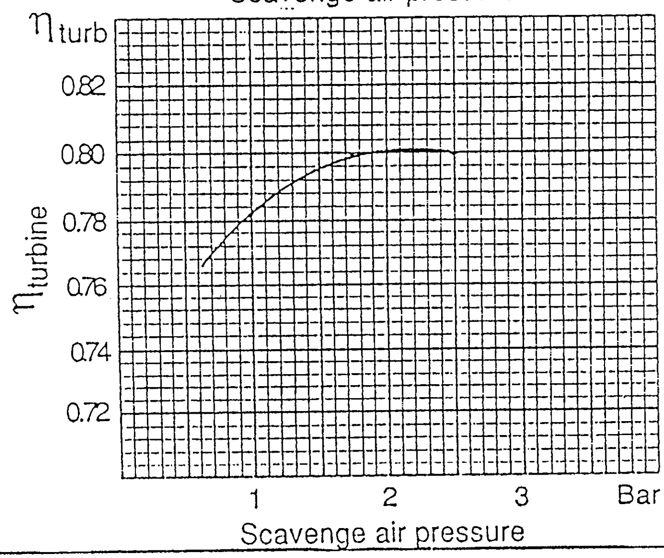
Plate 70609-40 Synopsis Diagrams – for turbocharger

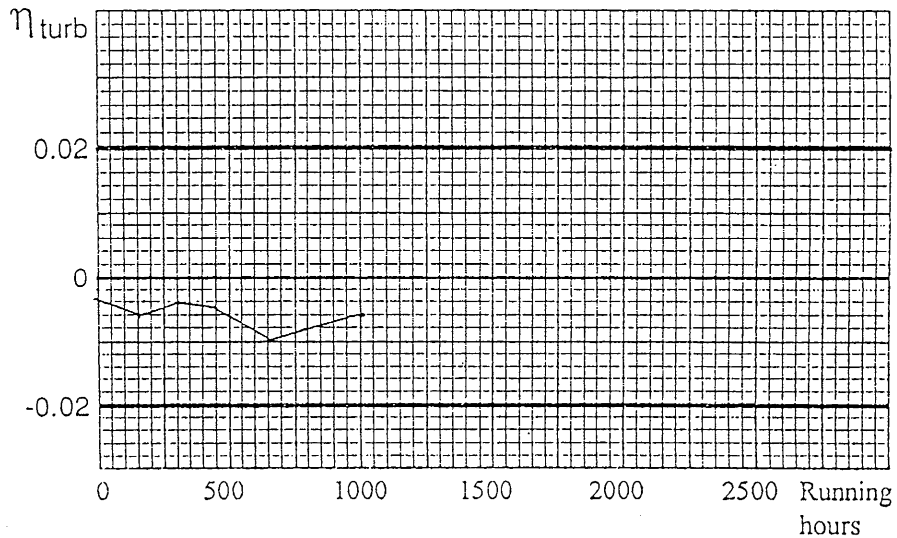
Model curves and time based deviation charts for:
compressor and turbine efficiencies as a function of p_{scav}

M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:

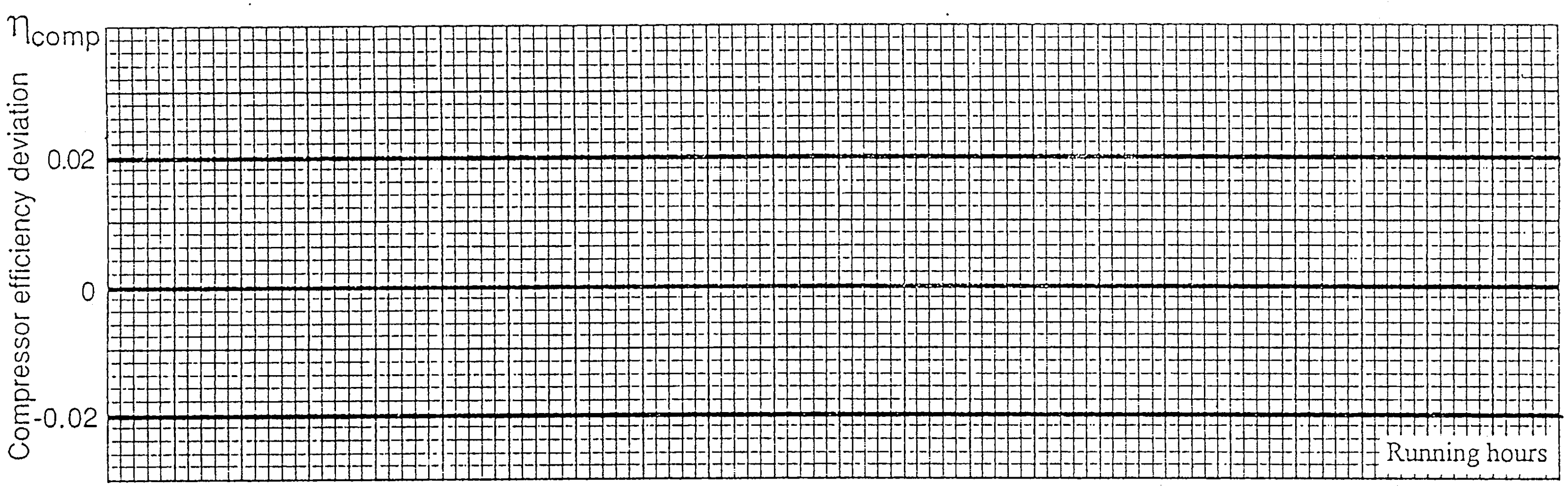


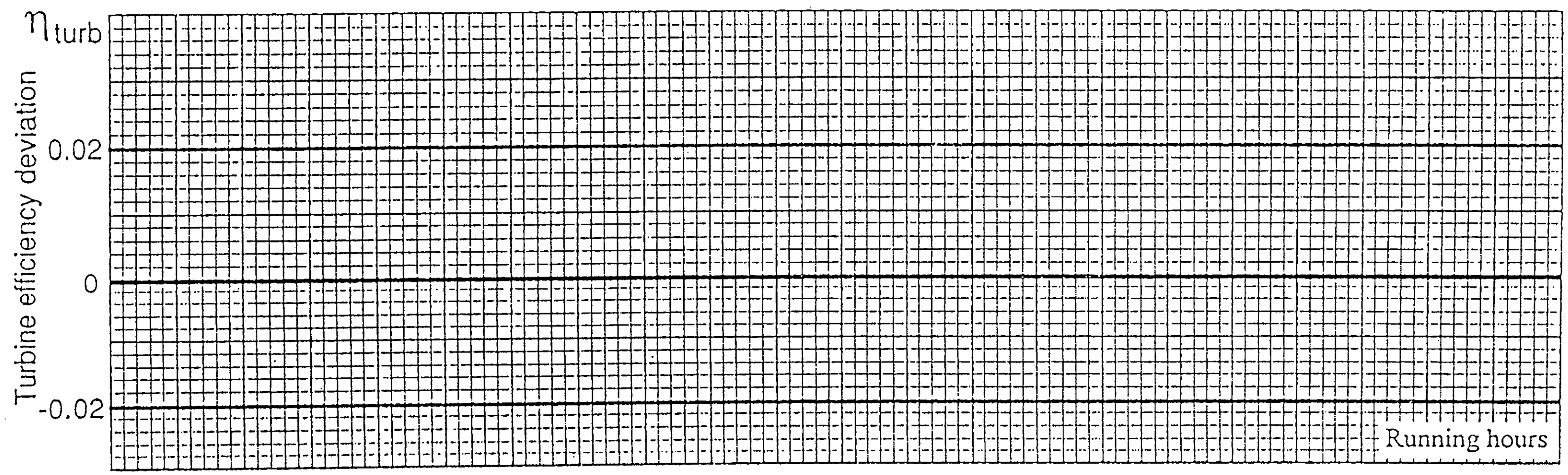






M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:





Synopsis Diagrams – for turbocharger

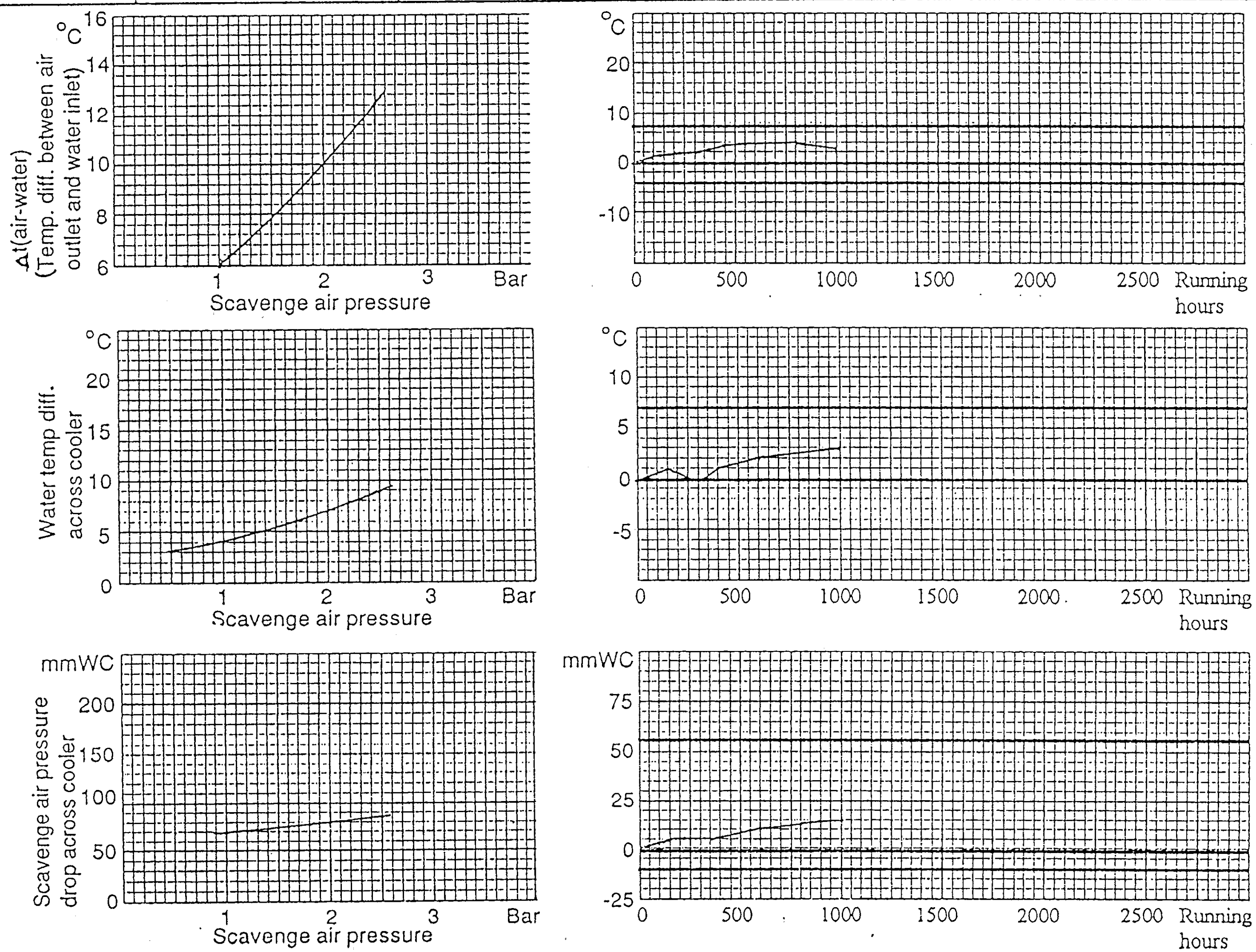
Time based deviation charts for:
compressor and turbine efficiencies

Plate 70609-40
Blank Copy

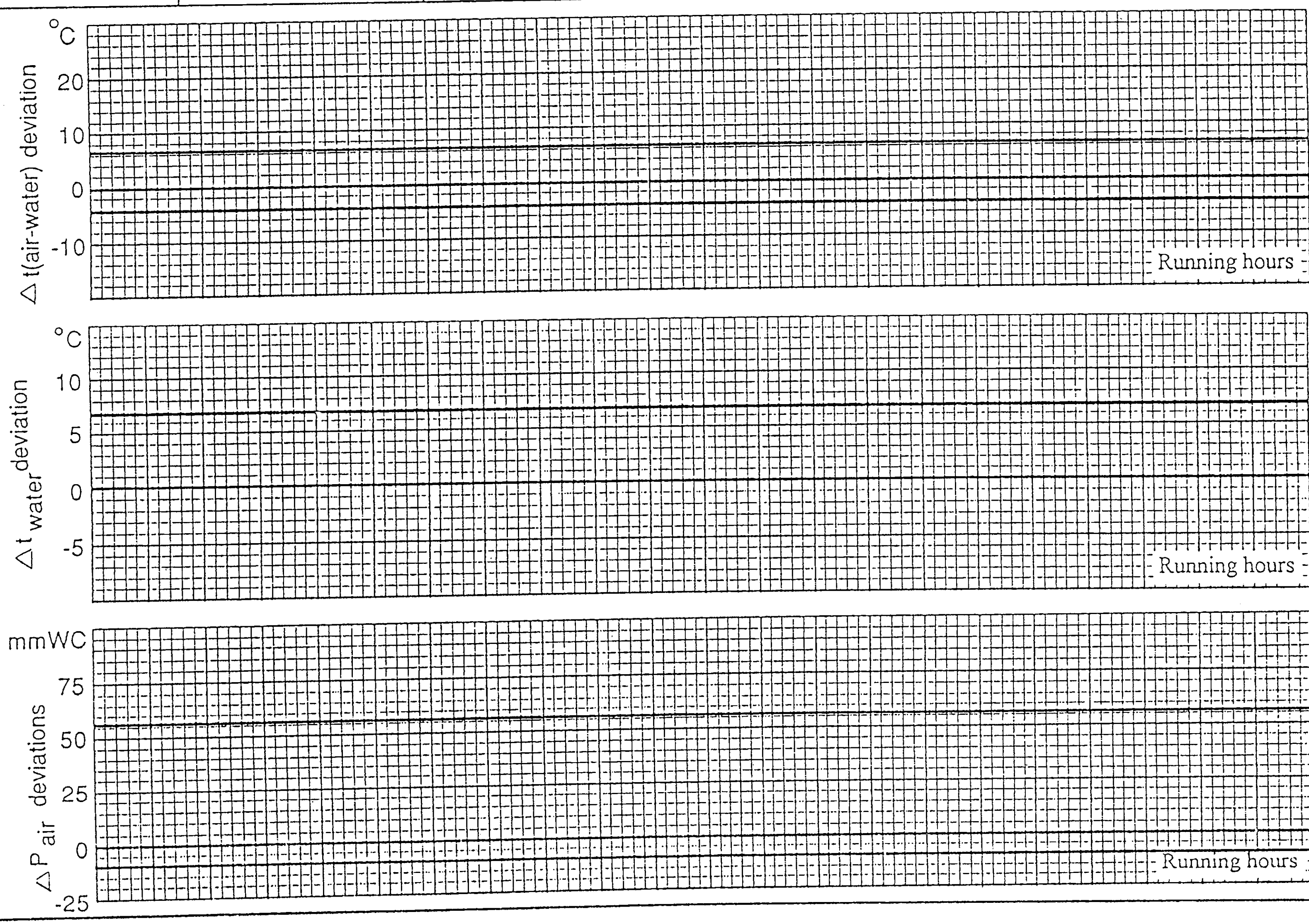
Plate 70610-40 Synopsis Diagrams – for air cooler

Model curves and time based deviation charts for: $\Delta t_{air\text{-}water}$, Δt_{water} and Δp_{air} as functions of p_{scav}

M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:



M / V	Engine Type		Time Based Deviation Charts	Checked by:
	Builder	No.		
Yard	Built Year			Date:



Synopsis Diagrams – for air cooler

Time based deviation charts for:
$\Delta t_{air\text{-}water}$, Δt_{water} and Δp_{air}

Plate 70610-40
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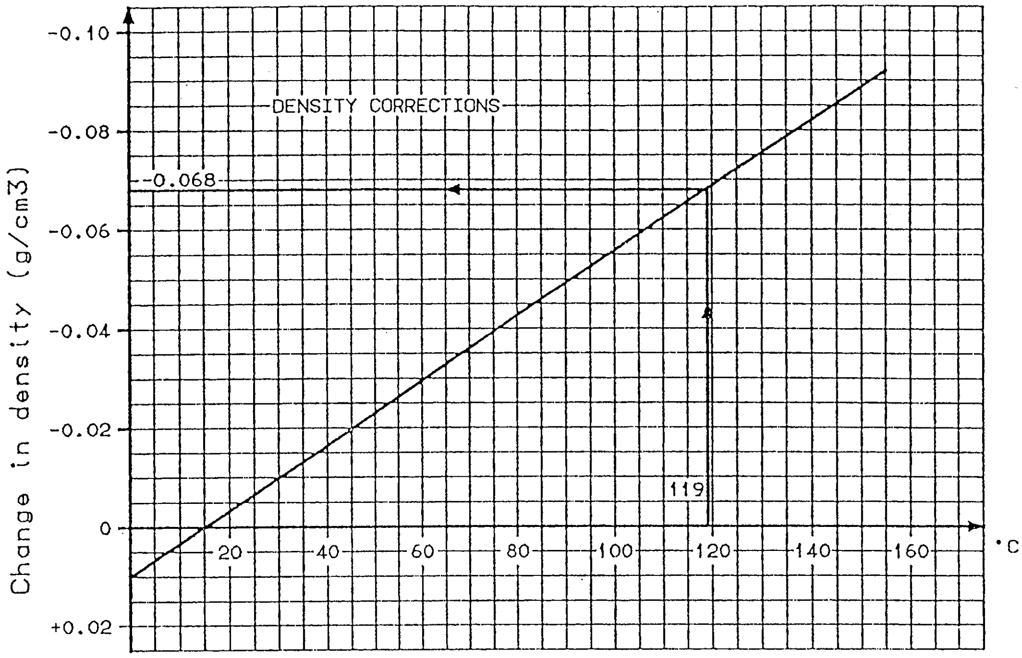
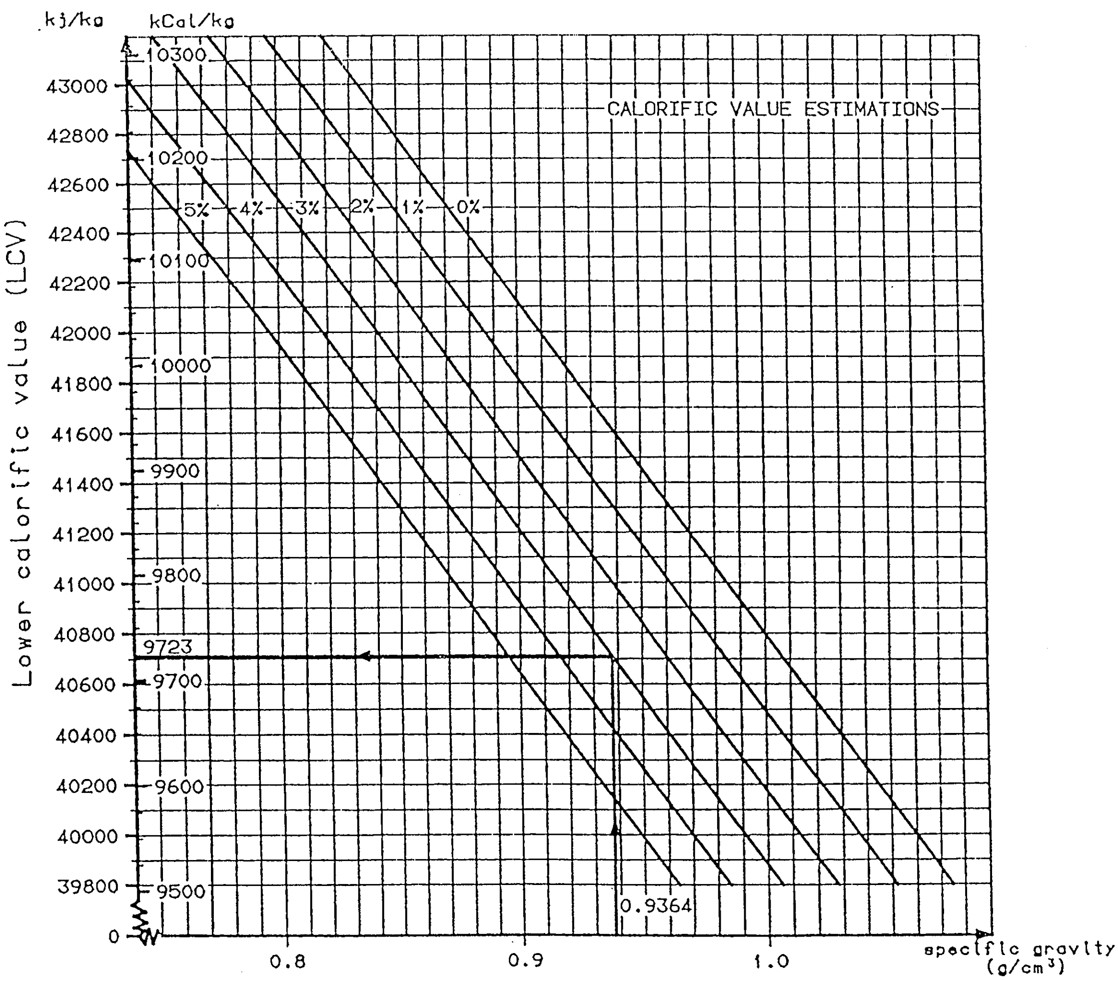
Plate 70611-40B **Specific Fuel Oil Consumption, -**
Correction for Fuel Temperature (Density) and
Sulphur Content (Calorific Value)





12-9

Dry cleaning of turbocharger (turbine side)

1. Carry out cleaning for every 24 to 50 hours of operation, based on observations.

2. Preferably clean the turbocharger at full load. Do not clean below half load.

3. Close plug A.

4. Open valves B and C, to blow out possible deposits and/or condensate in the connecting pipe. After about 2 minutes, close valves B and C.

5. Slowly open plug A to vent the container.

6. Fill the container with the quantity of granules specified in the table.

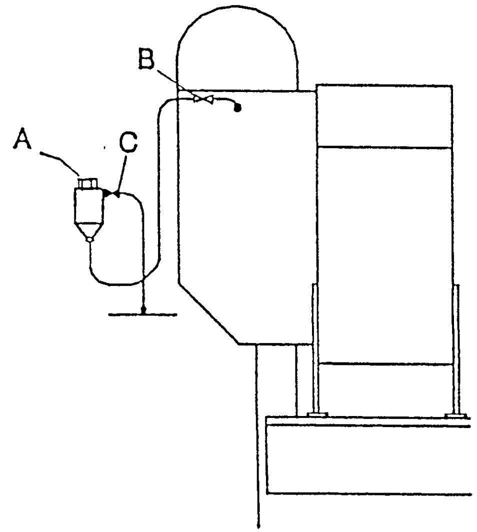
7. Close plug A.

8. Open valves B and C to blow-in the granules. After 1 to 2 minutes, close valves B and C.

9. Slowly open plug A to vent the container.

Note:

The turbocharger manufacturer's recommendation supersedes above description in case of discrepancy.



TC Type	Amount litres
NA34	0.5
NA40	1.0
NA48	1.5
NA57	2.0
NA70	3.0
NA83	3.5
VTR354	1.5
VTR454	2.0
VTR564	2.5
VTR714	3.0
MET 53 SD/E	1.6
MET 66 SD/E	2.6
MET 71 SD/E	2.6
MET 83 SD/E	3.5

Water Cleaning Turbocharger (Turbine side):

ABB TPL (2-stroke) **Plate 70613**

(Interval: approx. every 50 to 500 operating hours)

Instructions

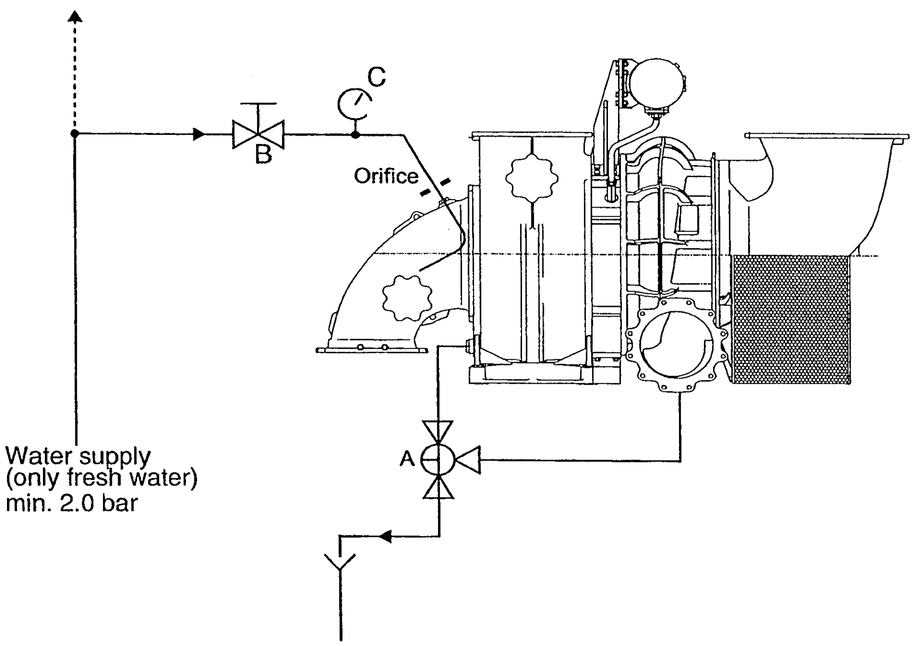
1. Adjust the engine load until the scavenge air pressure is in the range of $\mathbf{p_{scav} = 0.3 - 0.6}$ **bar** (overpressure). The temperature before turbocharger has to be below 430°C and the auxiliary blower should be in operation.
2. Open the drain cock A of the gas outlet casing and check whether exhaust gas emerges
3. Open valve B slowly until the pressure gauge C (water pressure p_{water}) indicates:

$$\mathbf{p_{water} = 1.0\ bar}$$

4. Inject water for 5 minutes while keeping the engine load constant
5. Close valve B
6. Close the drain cock A of the gas outlet casing

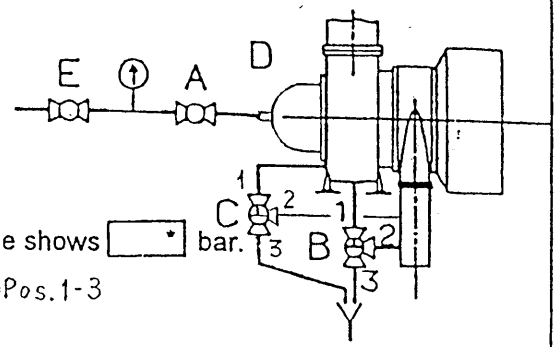
Remarks

- The engine should be run at least for further 10 minutes to prevent corrosion of the internal casing surfaces.
- Depending on the load only little or no water flows out the drain hole A. Water drain is not relevant for the cleaning effect. Water injection can be confirmed with reduced T/C speed during cleaning and lower gas temperature after turbochareger.
- If more than one turbocharger is mounted it is recommend to clean one after the other.



TC-type	TPL65-A	TPL69-A	TPL73-B	TPL77-B	TPL80-B	TPL85-B
Orifice integrated in TC ∅ [mm]	5.7	6.7	7.8	9.3	10.8	13.2

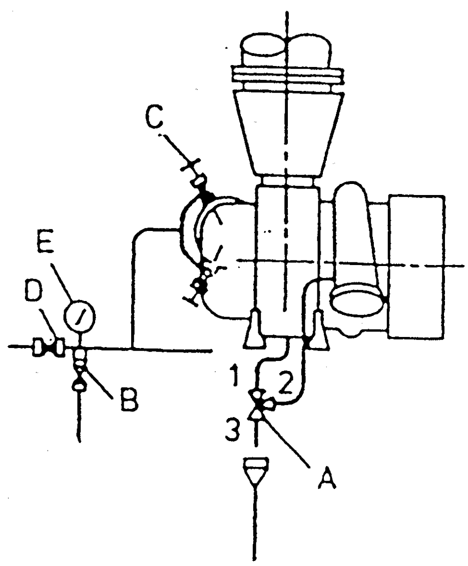
Cleaning of MAN Turbocharger
– Turbine Side –



1. Reduce engine load to T.C.RPM = [*] rpm
2. Wait about 10 minutes.
3. Open cock (E) to the extent that the pressure gauge shows [*] bar.
4. Open drain cock (B) to Pos. 1-3 and drain cock (C)Pos.1-3
5. Open cock (A) to washing.
6. Adjust cock (E) and maintain [*] bar on the pressure gauge.
7. Water wash for 10 minutes.
8. Close cock (E).
9. Close cock (A).
10. Close drain cock (B) (Pos.2-1) and drain cock (C)(Pos.2-1)
11. Run engine at same load for approx. 10 minutes. Then increase load slowly and check for undue vibrations that did not exist before or clean again.
12. Slight noise indicating contact of rotating parts is harmless due to running inlayer of cover ring.
13. Check spray nozzles from time to time.

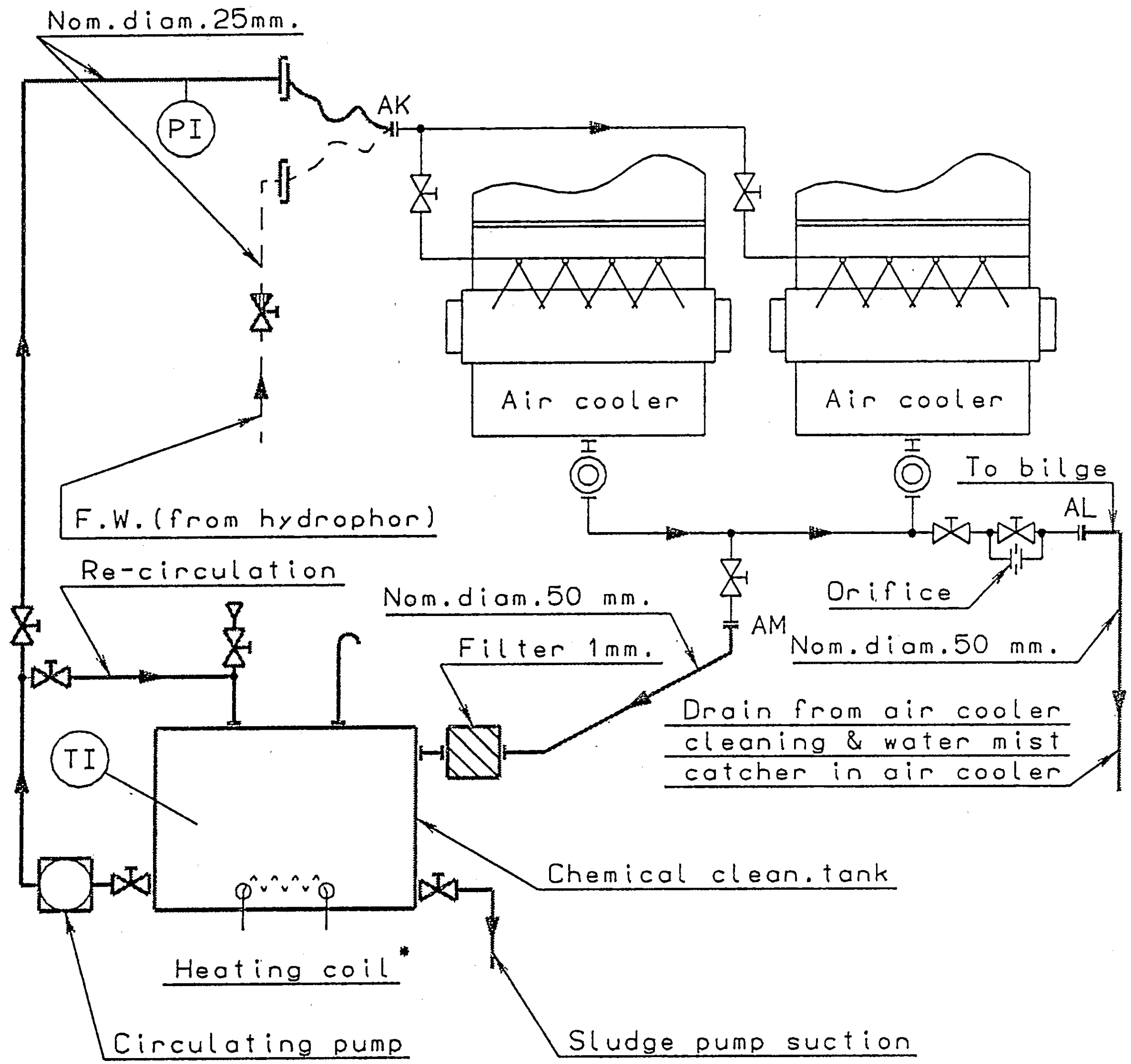
* See the specific data.

Cleaning of BBC Turbocharger
– Turbine Side –



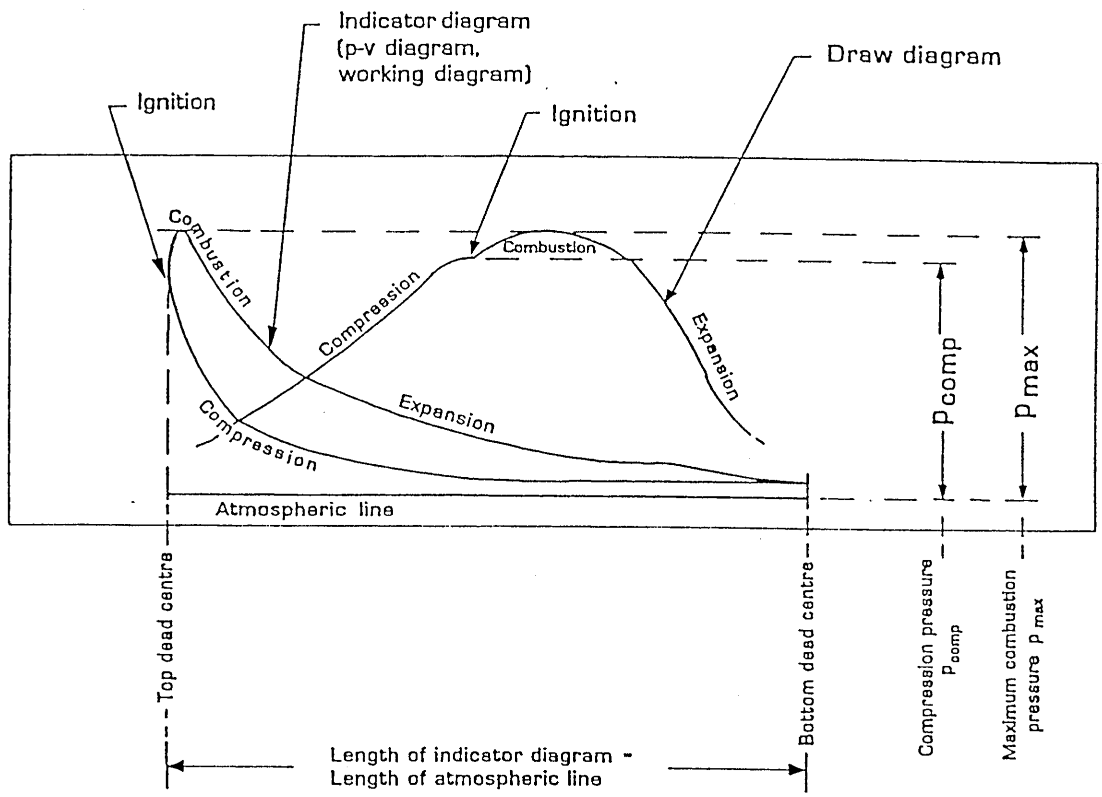
1. Reduce the rpm to 50% load.
2. Open the drain cock (A) (Pos. 1-3).
3. Close the drain cock (B).
4. Open valve (C).
5. Open the valve (D) slowly !
   until the pressure gauge shows 2.5 bar.
6. When the water from the drain pipe 3 appears clean, the cleaning is completed (after about 10 minutes).
7. Close the valve (D).
8. Close the valve (C).
9. Open the drain cock (B).
10. Close the drain cock (A) (Pos. 2-1).
11. After completing the cleaning, operate at the same load for 5 minutes.
12. Listen to the turbocharger, to make sure that it runs smoothly (without vibrations) when increasing the rpm. If not, repeat the cleaning.

Plate 70614-40D Air Cooler Cleaning System (Option)



* Capacity for heating coils according to requirement from supplyer of the chemical.

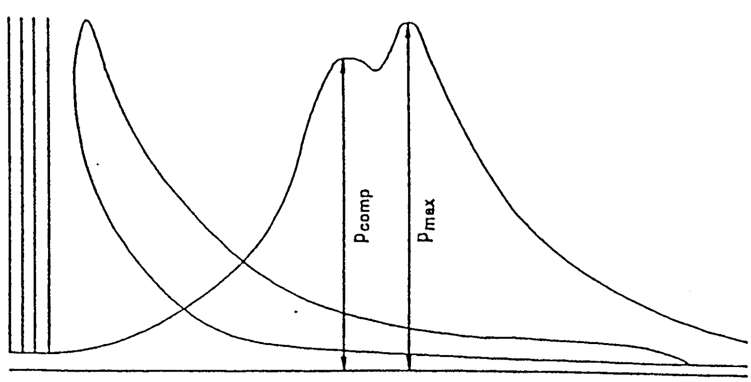
K/L-MC Engines:



S-MC Engines:

For this type of engine it has been necessary to delay the point of ignition to 2-3° after TDC, in order to keep the pressure rise, $p_{comp} - p_{max}$, within the specified 35 bar, while still maintaining optimum combustion and thereby low SFOC.

Due to this delay in ignition, the draw diagram will often show two pressure peaks, as shown in the figure below.



Note:
P-V diagram is measured in case the engine is furnished with the optional indicator cam.

Plate 70616-40A Adjustment of Indicator Drive

Correctly adjusted indicator drive/cam

The compression and expansion lines coincide, no area visible between the curves. (Fig. 1)

See note.

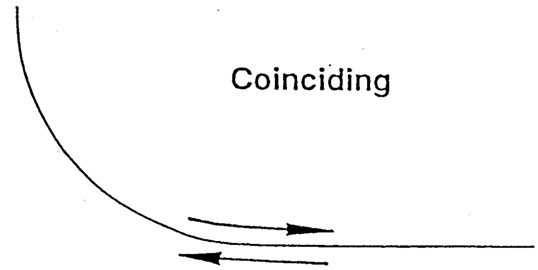


Fig. 1

Incorrectly adjusted indicator drive/cam

The compression and expansion lines do not coincide - an area is visible between the curves. (Fig. 2)

NB: The compression line is normally thicker than the expansion line.

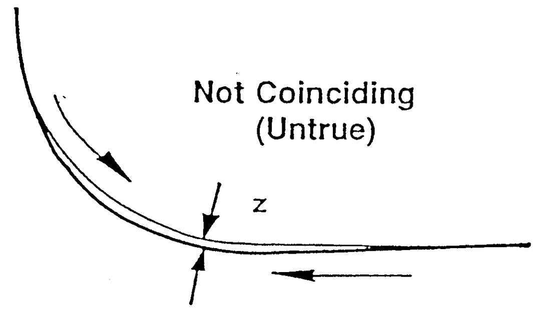


Fig. 2

Case A
The expansion line above the compression line, as shown.
The discrepancy 'Z' is positive: Displace the indicator drive or turn the cam "back", Fig. 3.

Case B
The expansion line below the compression line. 'Z' is negative: Displace the indicator drive or turn the cam "forward", Fig. 3.

(For z = 1 mm, turn the cam 2 mm or displace the drive approx. 2.5 mm, when using an indicator spring of 0.5 mm or 0.6 mm per bar).

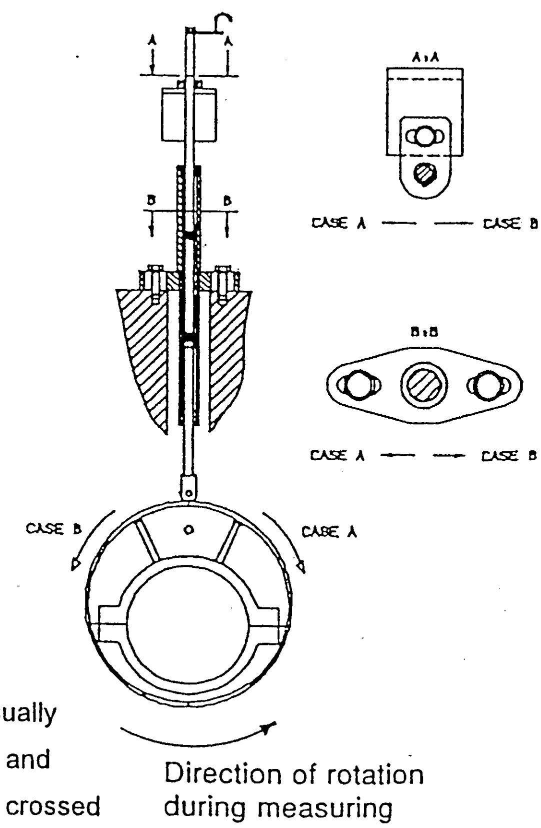


Fig. 3

Note:

(1) The adjustment of the indicator cam is relevant only when the engine is furnished with the optional indicator cam.

(2) Above Fig-1 shows an ideal condition of adjustment, and usually the lines do not coincide. It is allowed to have a small area, and the adjustment is to be made in such a way that the area is crossed in the middle, and the area on both sides to be equal.

- - - - - - - correct diagram
———— measured diagram

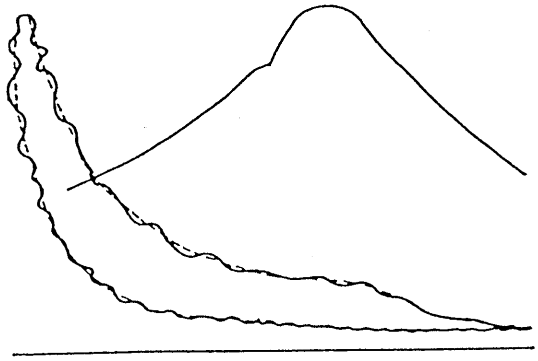


Fig. 1. Vibrations in drive.
Draw-diagram not affected.

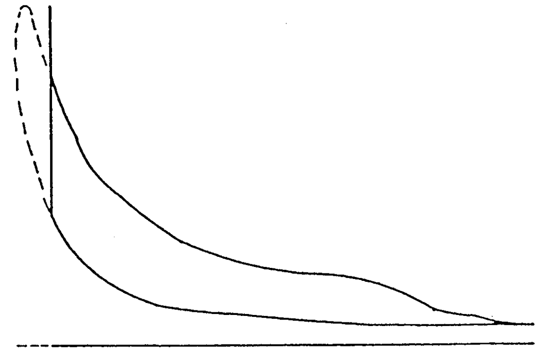


Fig. 2. Length of cord too long.
T.D.C.-part missing.

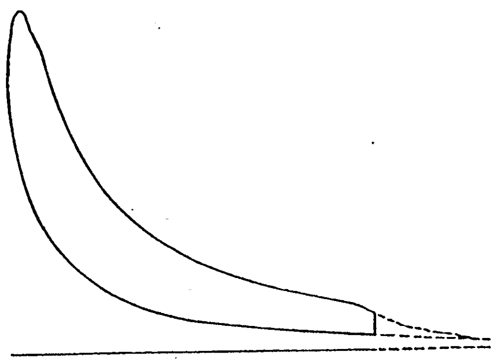


Fig. 3. Length of cord too short.
B.D.C.-part missing.

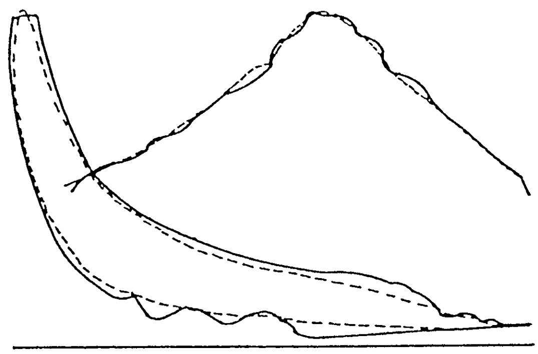


Fig. 4. Friction in indicator piston.
Draw-diagram also affected. This fault gives a too large working diagram area.

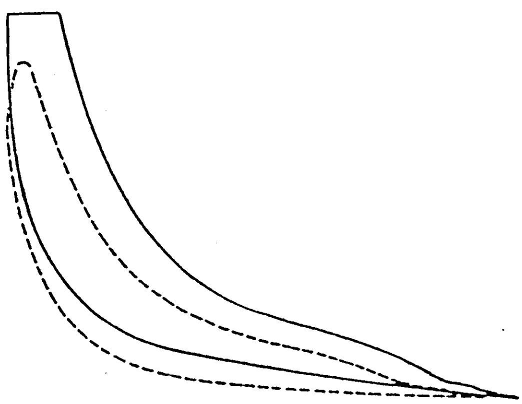


Fig. 5. Spring too weak. Indicator piston strikes top end of cylinder.

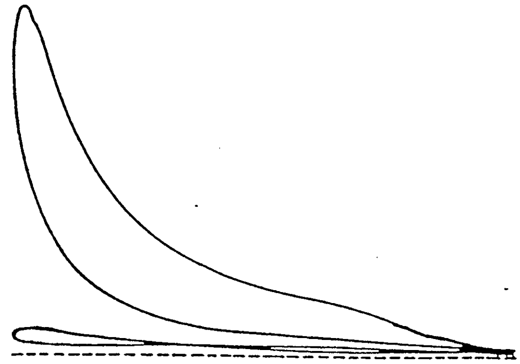


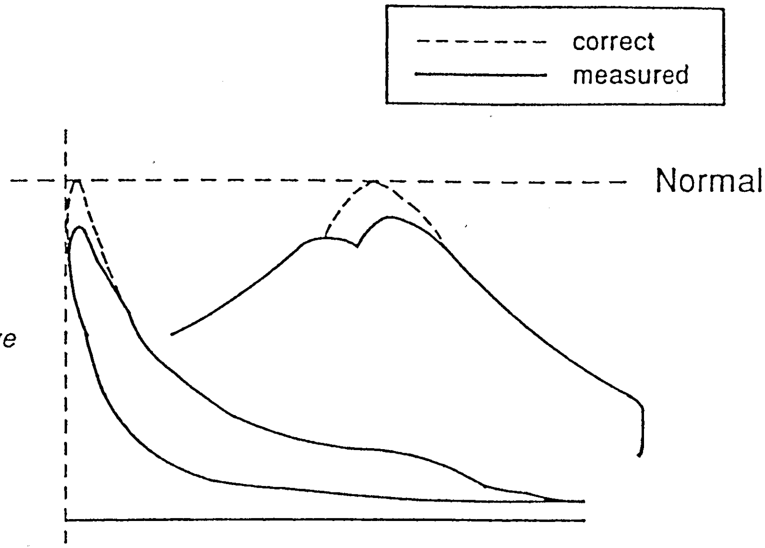
Fig. 6. Indicator cock leaking.
Atmospheric line untrue.

Note:
P-V diagram is measured in case the engine is furnished with the optional indicator cam.

Fig. 1:
Fuel injection too late.

- *Fuel pressure too low.*
- *Defective fuel valve(s).*
- *Defective fuel pump suction valve or shock absorber.*
- *Exceptionally poor fuel (bad ignition properties)*
- *Fuel pump lead too little.*

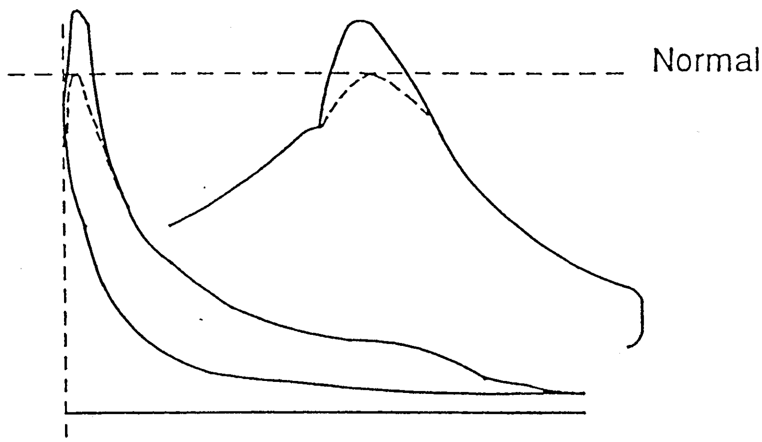
(see also the text)



p_{max} low, but p_{comp} normal

Fig. 2
Fuel injection too early.

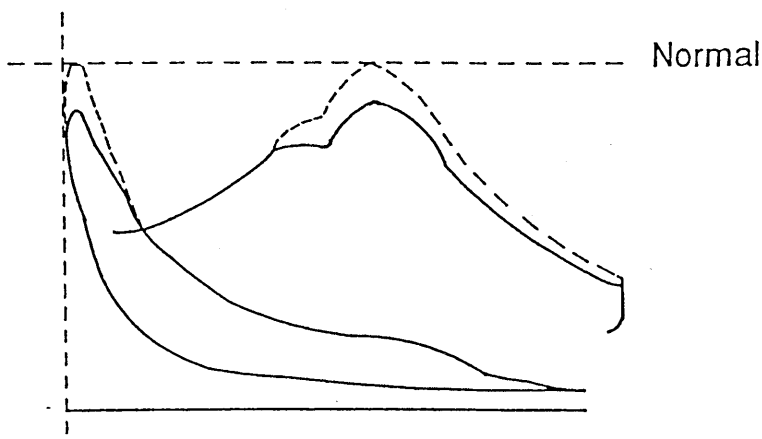
- *VIT index wrong.*
- *Fuel pump lead too large.*



p_{max} high, but p_{comp} normal

Fig. 3:
Leakages, increased cyl. volume, or fouling.

- *Piston ring blow-by.*
- *Exhaust valve seat leakage.*
- *Piston crown burnt.*
- *Low scavenge pressure, fouling of exhaust and/or air system.*



both p_{comp} and p_{max} are low.

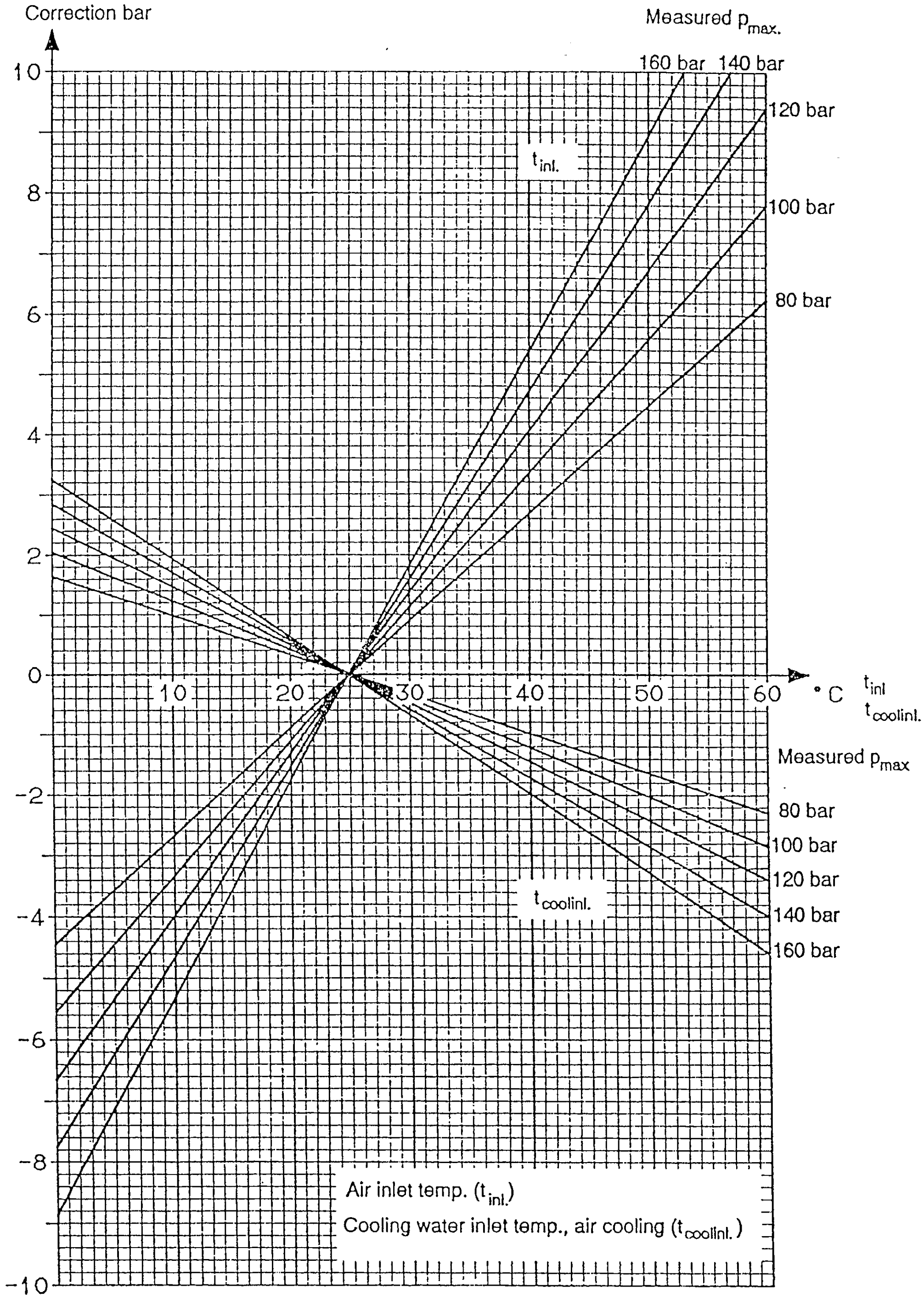
Note:
P-V diagram is measured in case the engine is furnished with the optional indicator cam.

Correction to ISO Reference Ambient Conditions

Plate 70620-40B

Maximum Combustion Pressure

Correction of measured p_{max}
because of deviations between t_{inl} / $t_{coolinl}$ and standard conditions.



Calculating the corrections:

t_{inl} : $A_{corr} = (t_{meas} - 25) \times 2.198 \times 10^{-3} \times (1 + A_{meas})$ Bar

$t_{coolinl}$: $A_{corr} = (t_{meas} - 25) \times -0.810 \times 10^{-3} \times (1 + A_{meas})$ Bar

See also Plate 70624.

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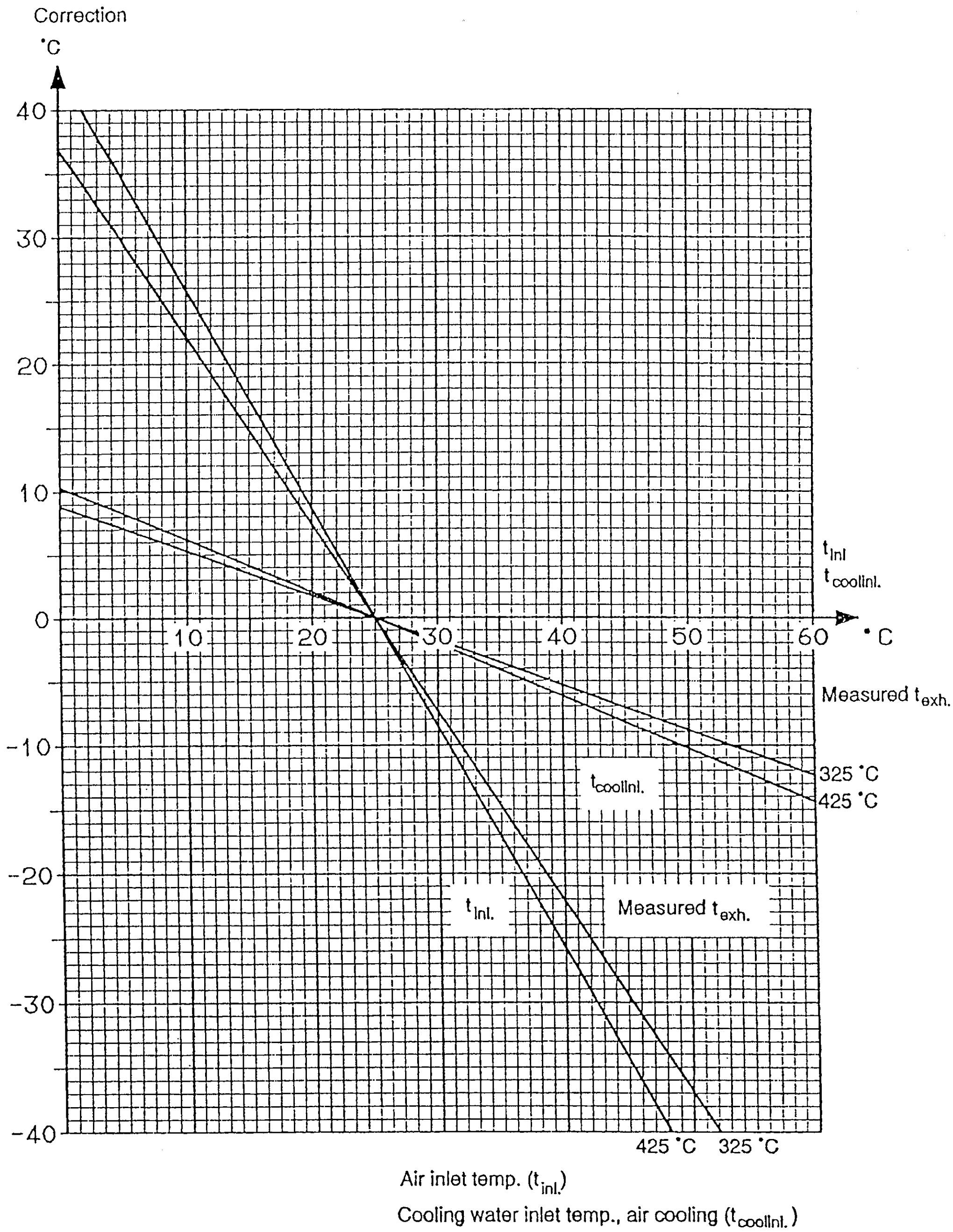
Plate 70621-40B Correction to ISO Reference Ambient Conditions

Exhaust Temperature (after exhaust valves)

Correction of measured exhaust temperature (t_{exhv}) because of deviations between t_{inl} / $t_{coolinl}$ and standard conditions.



Calculating the corrections:

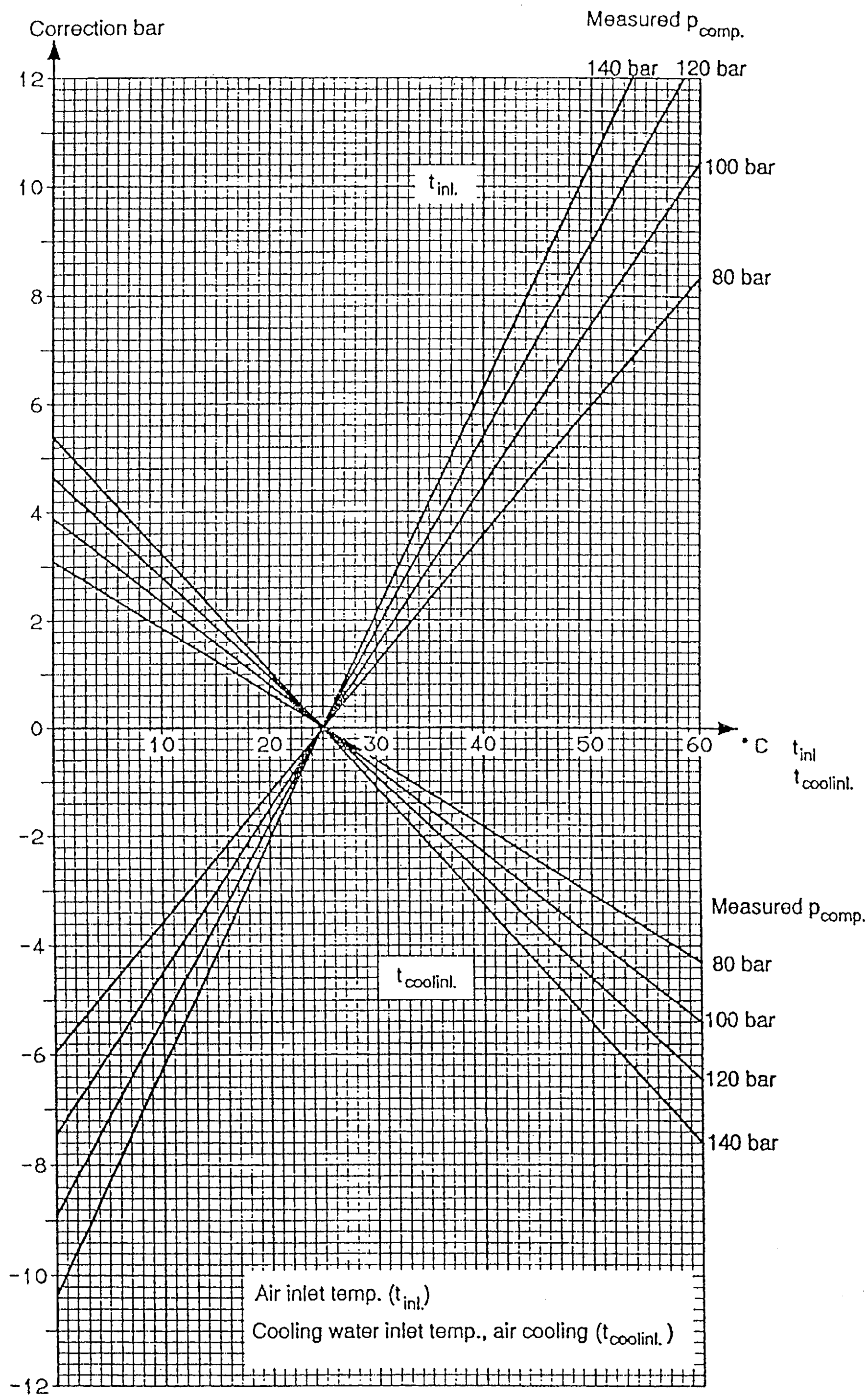
t_{inl} : $A_{corr} = (t_{meas} - 25) \times -2.466 \times 10^{-3} \times (273 + A_{meas})$ °C

$t_{coolinl}$: $A_{corr} = (t_{meas} - 25) \times -0.590 \times 10^{-3} \times (273 + A_{meas})$ °C

See also Plate 70624.

Compression Pressure

Correction of measured compression pressure
because of deviations between t_{inl} / $t_{coolinl}$ and standard conditions.



Calculating the corrections:

t_{inl} : $A_{corr} = (t_{meas} - 25) \times 2.954 \times 10^{-3} \times (1 + A_{meas})$ Bar

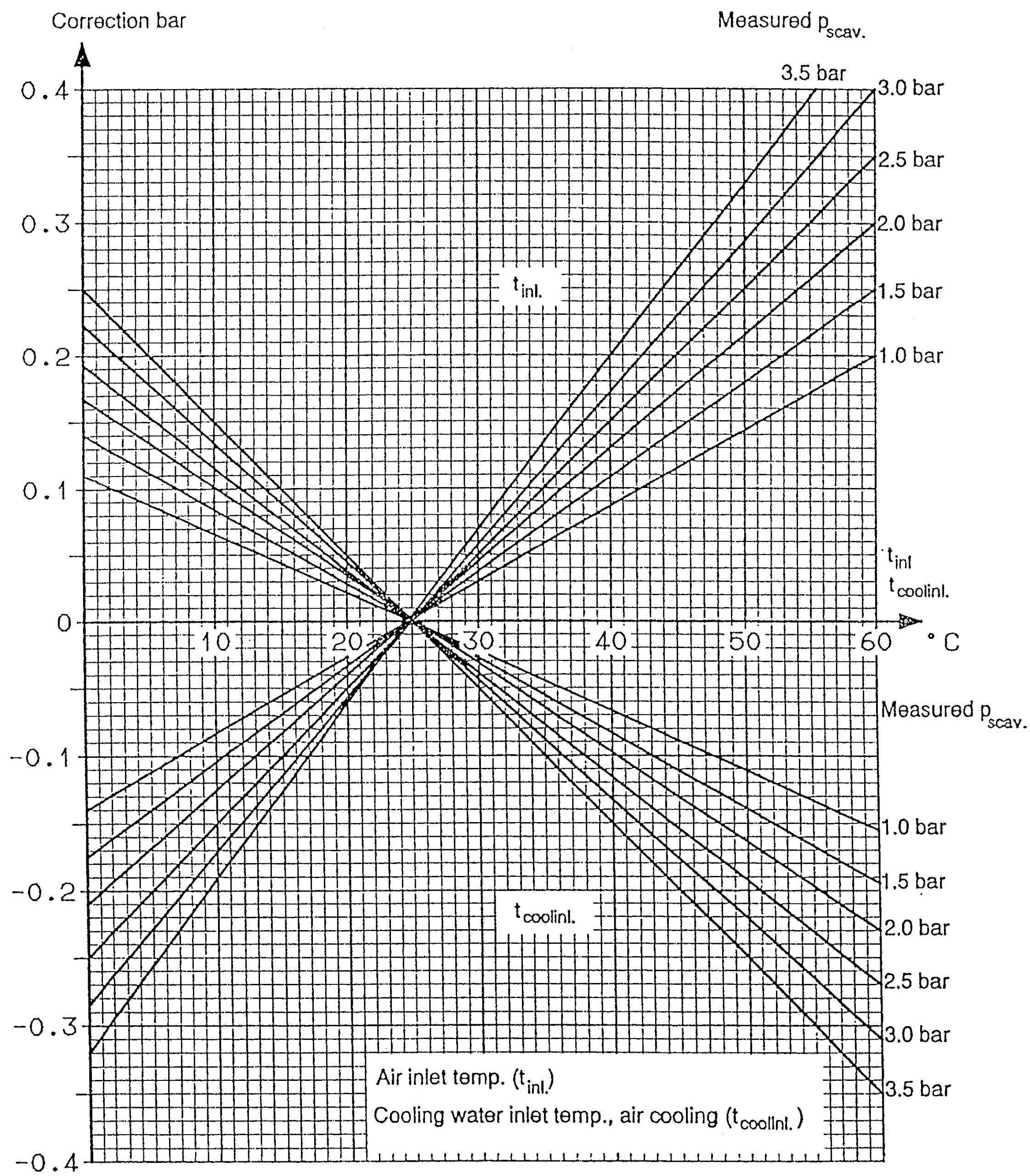
$t_{coolinl}$: $A_{corr} = (t_{meas} - 25) \times -1.530 \times 10^{-3} \times (1 + A_{meas})$ Bar

See Plate 70624.

Plate 70623-40B Correction to ISO Reference Ambient Conditions

Scavenge Pressure

Correction of measured scavenge pressure
because of deviations between t_{inl} / $t_{coolinl}$ and standard conditions.



Calculating the corrections:

t_{inl} : $A_{corr} = (t_{meas} - 25) \times 2.856 \times 10^{-3} \times (1 + A_{meas})$ Bar

$t_{coolinl}$: $A_{corr} = (t_{meas} - 25) \times -2,220 \times 10^{-3} \times (1 + A_{meas})$ Bar

See Plate 70624.

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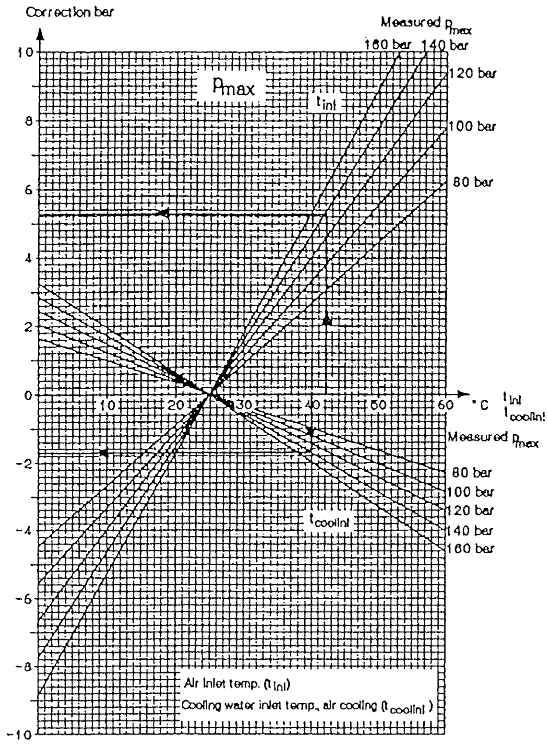
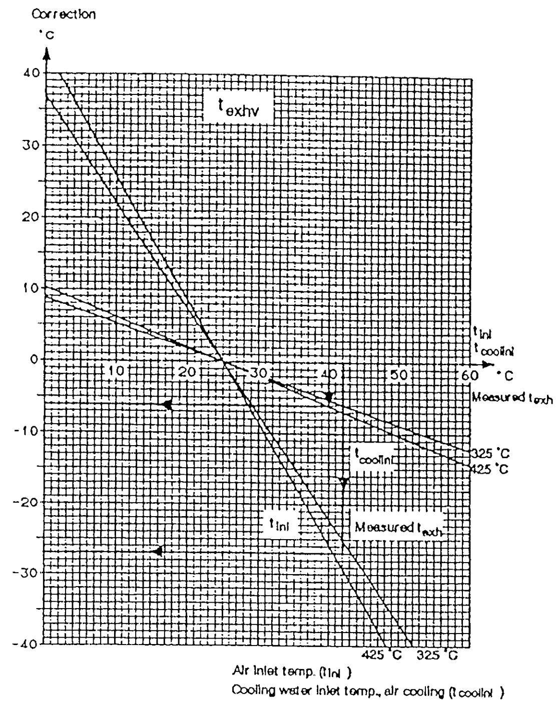
Correction to ISO Reference Ambient Conditions

Example of readings:

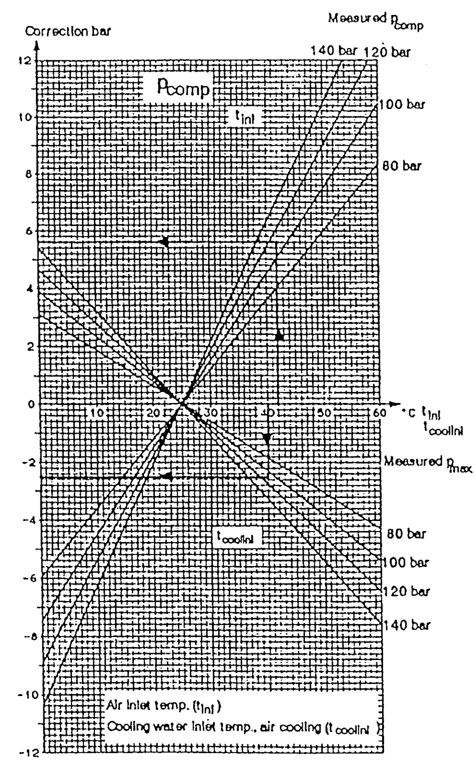
p_{max}	: 140 bar	p_{scav}	: 2.0 bar
t_{exhv}	: 425 °C	t_{inl}	: 42 °C
p_{comp}	: 110 bar	$t_{coolinl}$	: 40 °C



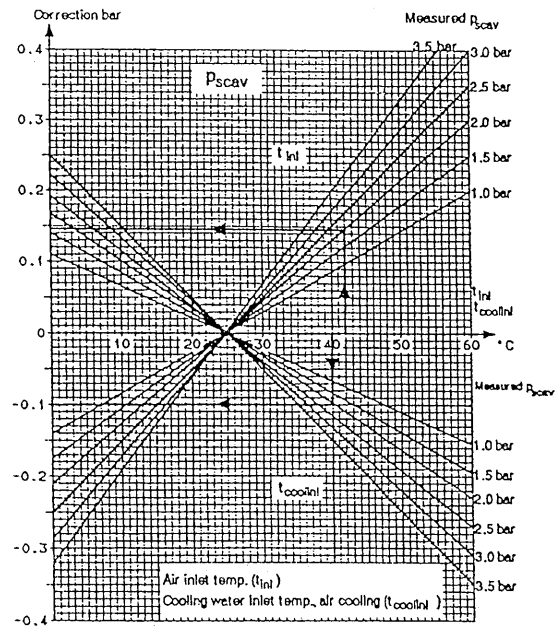
Correction for t_{inl} : +5.3 bar
Correction for $t_{coolinl}$: −1.7 bar
Correction 5.3−1.7 : +3.6 bar



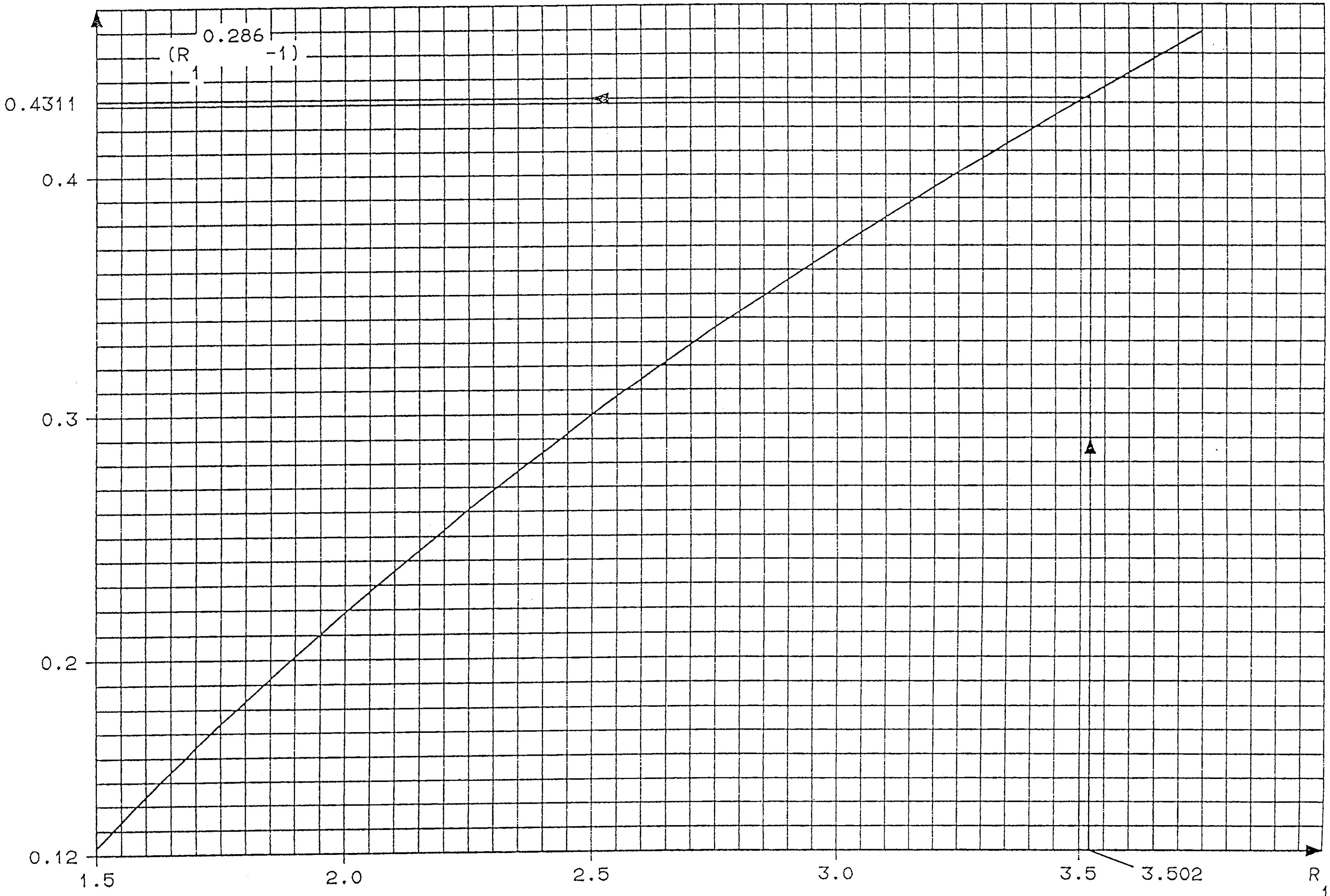
Correction for t_{inl} : −27 °C
Correction for $t_{coolinl}$: −6 °C
Correction −29−6 : −33 °C



Correction for t_{inl} : +5.6 bar
Correction for $t_{coolinl}$: −2.5 bar
Correction 5.6−2.5 : +3.1 bar



Correction for t_{inl} : +0.145 bar
Correction for $t_{coolinl}$: −0.1 bar
Correction 0.145−0.1 : +0.045 bar



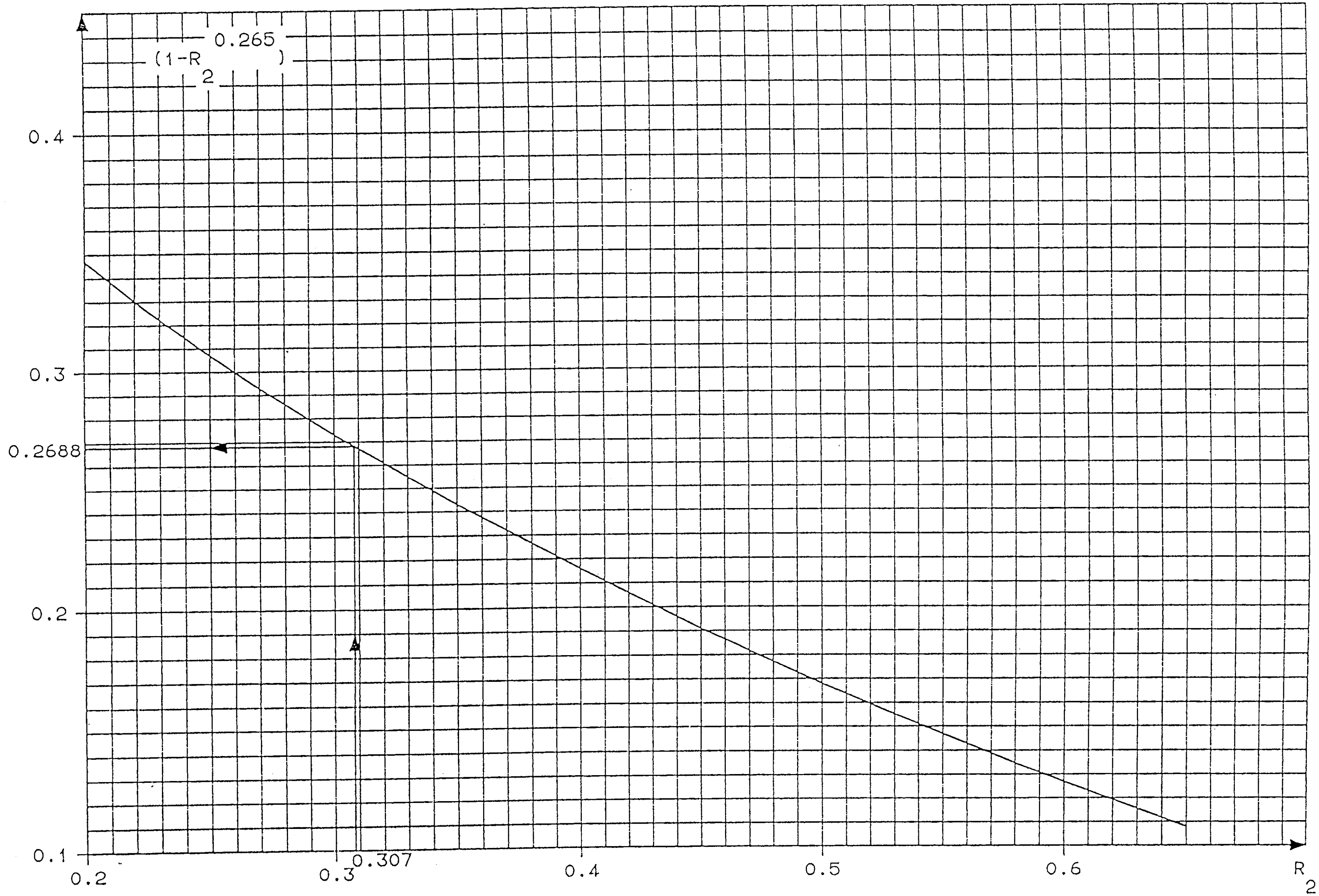
Calculation of Compressor Efficiency

Curve for the factor $(R_1^{0.286}-1)$

Plate 70625-40B

Plate 70626-40B Calculation of total Turbocharger Efficiency

Curve for the factor $(1-R_2^{0.265})$



The Charts and values, in the example below, refer to a particular 7L60MC engine. For a specific plant, see the actual Testbed Report concerned.

ESTIMATION OF EFFECTIVE ENGINE POWER FOR 7L60MC

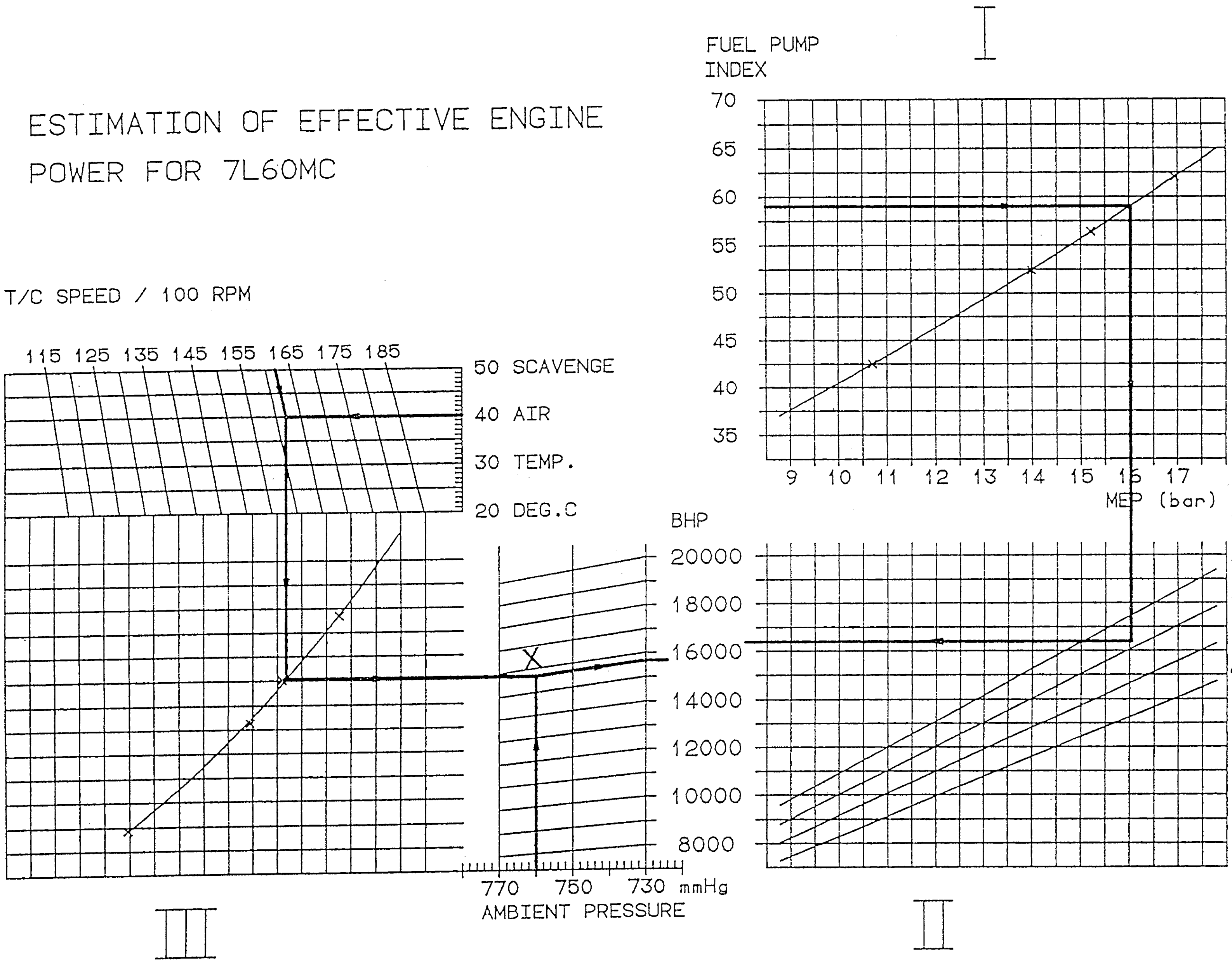


Plate 70628-40F Turbocharger Compressor Wheel Diameter and Slip Factor

Turbocharger Make: MAN B&W

Type Designation	Diameter, D (m)	No. of Blades	Slip Factor, μ
NR 24/R	0.276	–	0.76
NR 26/R	0.324	–	0.75
NA 34/S	0.408	–	0.70
NA 40/S	0.480	20	0.70
NA 48/S	0.576	20	0.70
NA 57/T9	0.684 0.684	20 18	0.70 0.74
NA 70/T9	0.840 0.840	22 18	0.76 0.74

Turbocharger Make: BBC/ABB

Type Designation	Diameter, D (m)	Slip Factor μ	Type Designation	Diameter, D(m)	Slip Factor μ
VTR254	0.29420	0.790	TPL65-A10	0.33900	0.715
VTR304	0.34970		TPL69-A10	0.39994	
VTR354	0.41570		TPL73-B11	0.48793	
VTR454D-VA12	0.52330		TPL73-B12	0.50652	
VTR454D-VA13	0.57560		TPL77-B11	0.57991	
VTR564D-VA12	0.65880		TPL77-B12	0.60200	
VTR564D-VA13	0.72460		TPL80-B11	0.67290	
VTR714D-VA12	0.82940		TPL80-B12	0.69854	
VTR714D-VA13	0.91230		TPL85-B11	0.82388	
			TPL85-B12	0.85527	

Turbocharger Make: Mitsubishi H.I. (MET)

Type Designation	Diameter, D (m)		Slip Factor μ			
Impeller Profile	V, S or R		V		S or R	
Impeller Size	2	3	2	3	2	3
MET33SD,SE	0.352	0.373	0.72		0.69	
MET42SD,SE	0.436	0.462				
MET53SD,SE	0.553	0.586				
MET66SD,SE	0.689	0.730				
MET83SD,SE	0.873	0.924				