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FUNDAMENTALS OF DIESEL ENGINE

(Questions and Answers)

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Preface

The present reference book is a compilation of questions on diesel engines and answers given to these problems during my term as an instructor of Marine Engineering at the Training Department, SEAFDEC, from 1979 to 1982.

The number of trainees over that period reached almost one hundred both in the joint Fishing and Marine Engineering Course and the Regional courses for Fisheries Extension Officers. A great many engineering questions were dealt with, ranging from relatively easy to sophisticated problems. From among these problems I have selected only those relating to the fundamental principles of the diesel engine and concentrated on the more important aspects so that these may be fully understood. Easier problems whose solution can be found mathematically according to formulae have also been included.

I have not, however, included problems concerning operation, maintenance and trouble-shooting of diesel engine, as these are dealt with in SEAFDEC publications TRB/No.10 and 19.

Although emphasis during practical training was placed on handling of the diesel engine rather than on theory of thermodynamics, combustion etc., I hope that the questions, answers and explanations given here will contribute to a better understanding of the subject, and be thought-provoking as well as conducive to further study of the diesel technology.

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CONTENTS

1. *How is the brake horsepower measured ?*

A dynamometer is used for measuring brake horsepower in the following ways: by absorbing the engine power, by hydraulic brake through the friction of water, by Prony brake (friction between the brake-shoe and the friction wheel), through the friction of air-fan brake and an electric eddy current brake (magnetic lines). In all cases the principle of the dynamometer is the same.

Note 1

The absorbing power of the hydraulic dynamometer is similar to the propeller resistance. The horsepower absorbed by a water brake is a function of the revolution n and the diameter D of the blade:

$$P = C n^3 D^5 \qquad C\text{............constant}$$

The absorbed power is transformed into heat and is carried by a constant flow of pressured water.

Note 2

Theoretically the error of dynamometer comes from the revolution measurement and the weight measurement by spring balance.

2. *How is the horsepower measured by dynamometer ?*

(What is the principle of dynamometer ?)

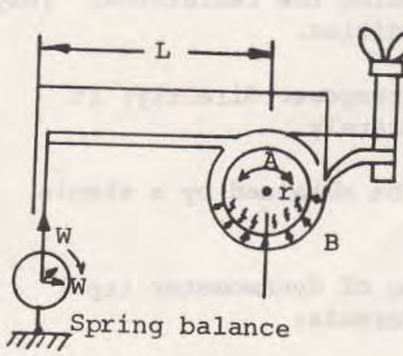


Fig. 1

A ... friction wheel directly connected to engine's driving shaft

N ... the rpm of friction wheel shaft

B ... brake band shoe (friction)

r ... radius of friction wheel

f ... friction forces

W ... spring balance weight (kg)

L ... length of arm lever

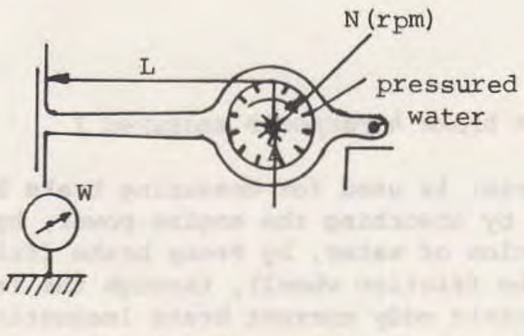


Fig. 2

If by tightening the screw (P) the band is pressed, the surface contact pressure of the brake band is increased; then the total resistance force will be balanced by the moment of the lever (L x W).

This means $L \times W = f \times r$ (moment balanced)

$$f = W.\frac{L}{r}$$

$$BHP = \frac{f \times 2\pi \times r \times N}{75 \times 60} = \frac{W \frac{L}{r} \times 2\pi r \times N}{4500}$$

$$= \frac{W \times L \times 2\pi \times N}{4500} = \frac{W \times N \times L \times 2\pi}{4500}$$

$$= \frac{W \times N}{C} \qquad \frac{L \times 2\pi}{4500} = \frac{1}{C} \qquad C = \text{constant}$$

Note 1 The shaft and brake HP of any kind of internal combustion engine do not indicate the actual power in accordance with the outer resistance, or act without adding the resistance. They only indicate the power at load condition.

Note 2 Dynamometer does not measure the horsepower directly; it measures torque and revolution separately.

The answer to the above question can be obtained by a simple method, as described below.

Measure the weight (kg) and revolution of dynamometer (rpm) simultaneously and calculate by using this formula:

$$BHP = \frac{W^{kg} \times N\,(rpm)}{C} \;.......... \;(1)$$ (Dynamometer at SEAFDEC workshop .. C = 1100)

Note 3 Be careful about the dynamometer's capacity curve. Every dynamometer has a maximum horsepower limit related to its capacity.

Fig. 3 shows the hydraulic dynamometer capacity diagram.

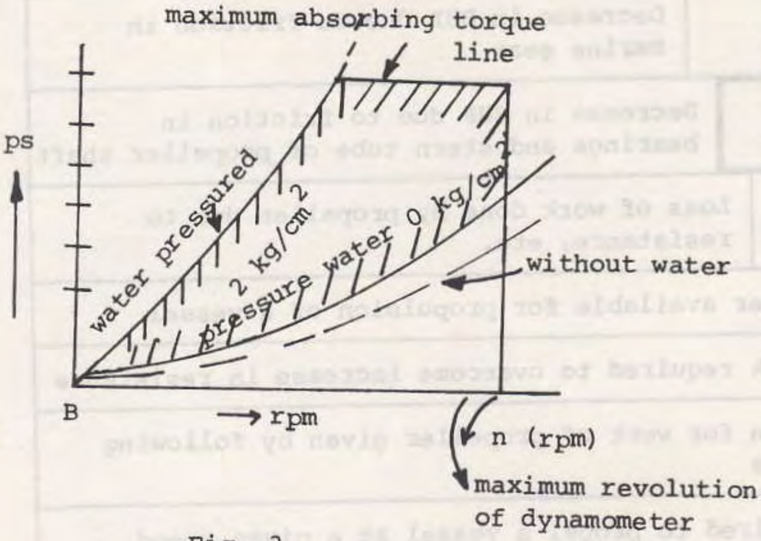


Fig. 3

A dynamometer can measure within the inner zone shown in Fig. 3. The outside of this zone is beyond its capacity.

3. *If the measurements for a diesel engine horsepower indicated by the workshop dynamometer are :*

weight	*20*	*kg*
revolution speed	*550*	*r.p.m.,*

calculate the horsepower.
(The dynamometer constant is 1,100)

$$BHP = \frac{W\,(kg) \times N\,(rpm)}{C} = \frac{20^{kg} \times 550}{1100} = 10 \text{ PS}$$

4. *Tabulate the kinds of horsepower related to the marine engine output and explain each briefly.*

IHP (100%)	Decrease in IHP due to friction in engine and pump
BHP (85% approximately)	Decrease in BHP due to friction in marine gear
SHP (82% approximately)	Decrease in SHP due to friction in bearings and stern tube of propeller shaft
DHP (78% approximately)	Loss of work done by propeller due to resistance, etc.
THP	Power available for propulsion of a vessel
	Work required to overcome increase in resistance
	Gain for work of propeller given by following wake
EHP	Work required to propel a vessel at a given speed
	Remainder to be fed back in the form of following wake

Indicated horsepower (IHP)

The power developed in the engine cylinder is the indicated horsepower. The term is derived from the fact that the cylinder's mean indicated pressure required for computation of this horsepower is measured by an indicator.

Brake horsepower (BHP)

Being the most important factor in engine performance, this power is the actual output available at the end (flywheel) of the crankshaft. The brake horsepower is so called because it was originally measured on a "brake". Today, several types of brakes or dynamometers are used.

Shaft horsepower (SHP)

A part of brake horsepower is expended in overcoming mechanical frictions in the marine gear. The remainder is the shaft horsepower available at the propeller shaft. This power is measured at the output end of marine gear output shaft by using a brake or dynamometer. Generally, SHP is shown in the engine builder's catalogue together with engine speed.

Delivered horsepower (DHP)

The power (shaft horsepower) available from the propeller shaft again encounters friction in the bearing and the stern tube. A decrease in the shaft horsepower due to friction represents the delivered horsepower. The ratio of the delivered horsepower to the shaft horsepower is called "mechanical efficiency" (ηr) or "transmission efficiency" (ηt).

Thrust horsepower (THP)

The power delivered to the propeller shaft to develop "thrust" (propulsive force) with which a vessel moves is the thrust horsepower. The formula for THP is commonly given as

$$THP = \frac{T \times Va}{75}$$

where T = thrust (kilogram)

Va = advance speed of propeller (m/sec)

Effective horsepower (EHP)

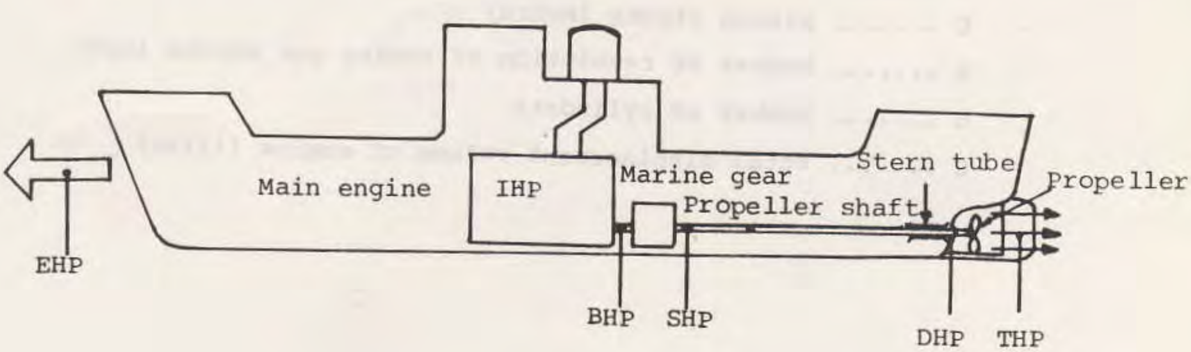
This horsepower is the actual net power required to propel a vessel and may be computed as

$$EHP = \frac{R \times v}{75}$$

where R = total resistance of a vessel (kilogram)

v = velocity of a vessel (m/sec)

Figure below indicates different kinds of horsepower relevant to marine engine.



5. *How is the horsepower of an engine calculated ?*

Explanation

A unit of work does not take into consideration the time required to carry it out. Work is the product of the force and the distance, so whether the work is done in a minute or in a day, the result is the same.

$$\text{Work} = \text{force} \times \text{distance}$$

$$W = F \times S \qquad \text{(unit is kg.m)}$$

However, power is the rate of doing work within a certain time. The formula for power is as follows:

$$\text{Power} = \frac{\text{work}}{\text{time}} = \frac{F \times S}{t} = F \times \nu \qquad \therefore \frac{S}{t} = \nu$$

In the metric system 1 horsepower (PS) = 75 kg.m/sec
One metric PS is equal to 0.9860 HP. Thus one British horsepower (HP) less 1.4% is one metric PS. There are several kinds of engine horsepower (see the preceeding No. 4), such as indicated horsepower (IPS or IHP), brake horsepower (BHP), shaft horsepower (SHP) etc.

Answer: The indicated horsepower of a 4-stroke engine can be calculated in the following way:

$$IPS = \frac{Pmi \times A \times L \times N}{75 \times 60 \times 2} \times Z \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

$$= \frac{Pmi \times A \times 100 \times L \times N \times Z}{75 \times 60 \times 2 \times 100 \times 1000} = \frac{Pmi \times C \times N}{900} \quad \ldots\ldots\ldots (2)$$

Pmi...... Indicated mean effective pressure (kg/cm^2)

A .... Section area of cylinder (cm^2)

L .... Piston stroke (metre)

N .... Number of revolution of engine per minute (rpm)

Z .... Number of cylinders

C .... Total displacement volume of engine (liter)

The indicated horsepower of a 2-stroke engine is calculated as follows:

$$IPS = \frac{Pmi \times A \times L \times N}{75 \times 60} \times Z = \frac{Pmi \times C \times N}{450}$$

In practice the value of Pmi roughly corresponds to Pme brake mean effective pressure considering mechanical loss, also IPS approximates the value of the brake horsepower.

Diesel engine	Pme	5.5 - 6.5 kg/cm^2
Gasoline engine	Pme	6. - 9 kg/cm^2

Note

Formula (2) is very useful for calculating horsepower of a motorcar engine; the total piston displacement or cylinder volume is indicated by e.g. : $1600_{cc(1.6\ \ell)}$, $2000_{cc(2\ \ell)}$, etc.

Example:

The maximum horsepower of a 4-stroke motorcar engine, whose Pme = 9 kg/cm^2, revolution - 5000 r.p.m., cylinder volume - 1600 cc, is :

$$BHP = \frac{9 \times 1.6 \times 5000}{900} = 80\ PS$$

1600 cubic centimetres = 1.6 liters.

6. *Calculate the brake horsepower with the following conditions:*

(1) Internal diameter of cylinder .... 145 (mm)

(2) Piston stroke .... 200 (mm)

(3) Engine revolution .... 850 (rpm)

(4) Number of cylinders 5

(5) Indicated mean effective pressure .... 6.5 (kg/cm^2)

(6) Mechanical efficiency .... 80 %

(7) Cycle .... 4-stroke single-acting engine

Displacement volume (V.e) $= \frac{\pi}{4} \times (14.5)^2 \times 20.0 \times 5 \times \frac{1}{1000}$

$= 16{,}504$ liters

from formula (2)

$$BHP = \frac{Pmi \times V^{cc} \times N \times 0.8}{900} = \frac{6.5 \times 16.504 \times 850 \times 0.8}{900}$$

$$= 81.05 \text{ PS}$$

Note   Pmi x mechanical efficiency = Pme = mean effective pressure

$6.5 \times 0.8 = 5.2$ (kg/cm^2) .... Pme

7. *Calculate the horsepower of a 4-stroke diesel engine, if: There are six (6) cylinders, one cylinder's displacement is 1.5 liters, the speed of revolution is 1,000 rpm, the mean effective pressure is 5 kg/cm^2.*

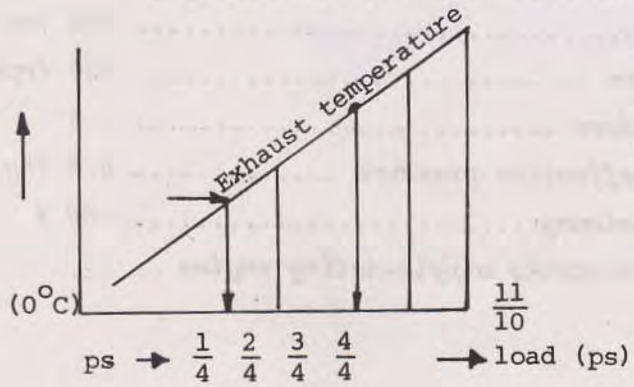
Use formula (2) :

$$BHP = \frac{5 \times 1.5 \times 6 \times 1000}{900} = 50$$

8. *Can the transmitted horsepower be measured when the boat is cruising ?*

Theoretically, the brake horsepower cannot be measured without a dynamometer but in cruising condition, i.e. when there is a load to offer resistance, some methods for measuring BHP do exist.

(1) To estimate the horsepower compare the exhaust temperatures provided in the engine builder's testing data.



If the propeller is a suitable one for the vessel the load condition (horsepower) can be estimated approximately from the exhaust temperature.

(2) By shaft horsepower meter (torque meter)

(Principle) $$T = \frac{\pi d^4 G \theta}{32 \ell} = k \theta$$

θ - measured electrically

T ... torque (kg m)

θ ... torsional angle of shaft

$$BHP = \frac{2\pi T N}{60 \times 75} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

(3) By using an indicator diagram

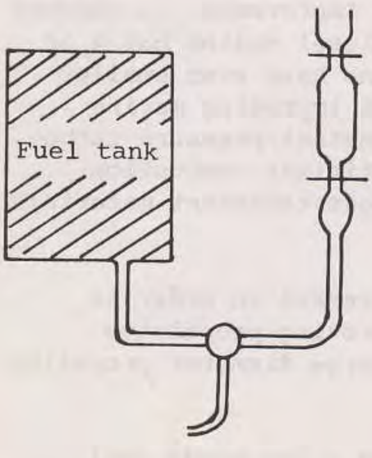
First an indicator diagram must be made. Then calculate the indicated mean pressure and at the same time measure revolution. Then the following formula should be applied:

$$BHP = \frac{Pmi \times C \times N \times \eta}{900}$$

η .... mechanical efficiency obtained experimentally.

9. How is the fuel consumption ratio of an engine measured ?

While measuring horsepower by dynamometer we can measure the fuel consumption ratio (f.c.) by the following method:



F : fuel consumption (liter/hour)

A : volume (cc) between a-b level (correct standard volume; such as 300 cc) SEAFDEC workshop: A = 300 cc

t : time required for consumption of A amount of fuel (sec)

γ : specific gravity of the fuel

P ⊛ : brake horsepower calculated by dynamometer

$$F = \frac{60 \times 60 \times A}{t \times 1000} = \frac{3.6 \times A}{t} \quad (\ell/\text{hour})$$

$$f.c = \frac{F}{P} \times 1000 \text{ (cc)} \times \gamma \qquad \text{(grms/ps. hour)}$$

⊛ See Q No. 2 for method of measuring horsepower.

10. *Calculate the required quantity of fuel for a diesel engine, if:*

Horsepower	*100*	*ps*
fuel consumption ratio	*210*	*g/ps.h.*
operating time	*10*	*hrs.*
specific gravity of fuel	*0.88*	

$$Q = \frac{100\ (PS) \times 210\ (gr/PS.h) \times 10\ (hour)}{0.88\ (S.G) \times 1000^{cc}(liter)}$$

<u>Answer</u> : 238.6 liters.

11. *Which are the methods to lower the consumption/cost of fuel for a marine engine ?*

(1) <u>Improvement in the heat efficiency of engine</u>

Theoretically, according to the laws of thermodynamics, the maximum thermal efficiency is $(\eta) = \frac{T_2 - T_1}{T_1}$

$$\eta = 1 - \frac{1300 - 400}{2400} = 54\% \sim 82\%$$

T_2 ... High heat source

T_1 ... Low heat source

In practice, there is much scope for improvement in thermal efficiency of an engine. A modern diesel engine has η of only 42 percent, other kinds of engine have even smaller values. Some of the possible ways of improving marine diesel engine are: to introduce a constant-pressure turbocharging system which allows more efficient combustion; minimize the loss of heat by using heat-resistant materials such as ceramic, etc.

(2) Improvement in the hull form of the vessel in order to reduce the effective horsepower; improving propulsive efficiency by using low speed with large diameter propeller, etc.

(3) Fuel cost saving by using or blending a low grade fuel oil.

(4) Reduction of boat speed or minimizing the operational cost by economical speed. At present this method is the most effective way to save fuel.

(5) In future it may be possible to reduce the cost by introducing other sources of energy, such as hydrogen fuel.

Note

Daily fuel consumption can be calculated from the formula of fuel coefficient and admiralty constant

$$E = \frac{D^{\frac{2}{3}} \times V^3}{c} \quad \text{............ (1)}$$

$$F = \frac{f.c. \times SHP \times 24}{1000 \times 1000} \text{ (tons)}$$

$$C_A = \frac{D^{\frac{2}{3}} \times V^3}{SHP} \quad \text{............ (2)}$$

where F : fuel quantity per 24 hours (tons)

c : fuel coefficient (constant)

D : displacement tonnage of vessel

V : speed of vessel (knots)

N : engine revolution (rpm)

C_A : admiralty constant

SHP : shaft horsepower

f.c.: fuel consumption ratio grm/ps.h

$$F \propto SHP$$

$$SHP \propto V^3$$

$$SHP \propto N^3$$

$$F \propto N^3$$

How much fuel can be saved by reducing the speed of the vessel ?

If the speed is reduced by 10% from continuous rating, the fuel consumption will be 0.9 x 0.9 x 0.9 = 0.729 or 72.9% of the original consumption. Thus, in theory, the fuel consumption decreases by 27 percent.

However, when the cruising distance is constant this reduced speed will result in an increase in cruising time; consequently, the fuel consumption might become 0.9 x 0.9 = 0.81 → 81 percent of the original amount. Therefore, in practice, the saving rate is about 20 percent.

$$F \propto n^2$$

or

$$F \propto V^2$$

<u>Note 2</u> Economical speed

It is difficult to calculate the economical speed of a fishing vessel but in the case of a cargo vessel the optimum speed <u>Ve</u> can be calculated as follows:

$$\text{Cruising cost/Dw} = Cn = \frac{f + B}{24 \times V \times Dw} \quad \text{....... should be minimized}$$

where
- f : fixed cost per day (interest, depreciation cost, insurance, wages etc.)
- B : cost of fuel oil
- V : speed of boat
- Dw : loading weight
- Cn : cruising cost per unit Dw
- B : kV^3

$$Cn = \frac{f + kV^3}{24\ Dw \times V} = \frac{1}{24\ Dw} \left(\frac{f}{V} + kV^2\right)$$

$$Cn \longrightarrow \text{minimize} \qquad \frac{d(Cn)}{dV} = 0 \qquad Ve = \sqrt[3]{\frac{f}{2R}}$$

At the economical speed Ve, the cost of fuel Be is the lowest.

$$Be = k\,Ve^3 = \frac{f}{2}$$

12. *A fishing boat consumes 12 tons of fuel per day with the cruising speed of 8 knots. If the speed is reduced to 7.5 kt, how many tons of fuel will it consume per day ?*

$Q \propto V^3$ where Q .... quantity of fuel

k .... constant

$12 = k \times (8)^3, \; k = \frac{12}{(8)^3}$ V .... speed of boat

$\therefore \; x = k \times 7.5^3$

$= \frac{12}{(8)^3} \times (7.5)^3 = 9.89$ tons/day

13. *A fishing boat sailing at the speed of 12 knots from the fishing port to a fishing ground consumes 500 liters of fuel. If the same distance is covered at the speed of 10 knots how much fuel can be saved ?*

If the cruising distance is constant

$Q \propto V^2$

Q .... fuel consumed

k .... constant

V .... speed of boat

then $50 = k \times 12^2 \quad k = \frac{50}{12^2}$

$x = \frac{500}{12^2} \times 10^2 = 34.7$ litres

$50 - 34.7 = 15.3$ liters; (30 percent of fuel is economized)

14. *How are diesel engines rated in terms of load ?*

The definition of rated load varies from country to country. In general, there are three forms of rating:

(1) The basic full load.

(2) The overload or intermittent load in a limited period such as 1 hour or 15 minutes.

(3) The continuous rating (lower than (1) : more than 24 hours continuous operating permitted)

Different industrial standards (DIN, SAE, BS, JIS) are compared in the table below.

Comparison of Industrial Standards

	DIN 6270 - 1970	SAE J816b - 1973
Scope	Industrial use Ship propulsion Rail cars, locomotives	4-stroke cycle diesel engines
Output definitions	1. NA: Continuous output with overload margin (code designation: continuous output A)* is the maximum output developed with overload margin for a given application under continuous duty conditions. 2. NB: Output without overload margin (code designation: output B) is the maximum output developed without overload margin for a given length of time in a given application. Adjustments shall be made during the test so that the output B is not exceeded. In cases where no application or load to be driven is specified by the manufacturer, the output B shall be a continuous one-hour output in the course of 6-hour alternate operation.	1. Maximum brake power is the highest power developed at a given speed. 2. Peak brake power is the highest power developed within the engine speed range. 3. Intermittent brake power is the highest power recommended by the manufacturer for satisfactory operation within the manufacturer's specified conditions of load, speed, and duty cycle. 4. Continuous brake power is the power recommended by the manufacturer for satisfactory operation under the manufacturer's specified continuous duty conditions. 5. Rated brake power is the power specified by the manufacturer for a given application at a given (rated) speed.

	DIN 6270 - 1970	SAE J816b - 1973
	3. NU: Marginal output is the maximum output continuously or intermittently developed for a total of one hour in the course of 12-hour operation. Unless otherwise specified, the Nu shall be 110% of the NA. 4. NH: Maximum output is the maximum output developed for 15 minutes without excessive mechanical and thermal strains.	6. Gross power is the power output of a "basic" engine - an engine equipped with the built-in accessories essential to its operation: fuel pump, oil pump, coolant pump, and built-in emission control equipment. A generator or alternator is to be included only if some accessories (such as a fuel pump) are electrically driven. 7. Net power is the power output of a "fully equipped" engine - an engine equipped with all the accessories necessary to perform its intended functions unaided. This includes, but is not restricted to, the basic engine.

	BS649 - 1958 (1960)	JIS F0401
Scope	Reciprocating compression - ignition (diesel) engine, for general purpose	Marine propulsion engines
Output definition	Rated power output 1. British standard rating is the net output (in brake horsepower) continuously developed under the manufacturer's specified standard conditions. 2. Overload rating is the rating of 10% overload of British standard rating an engine is capable of for one hour in the course of 12-hour operation at the same speed (in revolutions per minute) as for British standard rating.	1. Normal output is the output developed to propel a vessel at the specified cruising speed and is economical for both engine efficiency and maintenance. 2. Maximum continuous output is the maximum output developed safely and continuously. The calculation of strength of an engine is based on the maximum continuous output at the stage of design. This output value is used to indicate not only a nominal output but a minute rating output. 3. Overload output is the overload rating an engine is capable of beyond the maximum continuous rating for a short period of time and is 105% to 110% of the maximum continuous rating. 4. Astern output is the output developed to back a vessel astern and is 40% to 60% of the maximum continuous output. (Note) The normal output is usually 80% to 95% of the maximum continuous output.

	DIN 6270 - 1970	SEA J816B - 1973
Standard operating conditions	o Barometric pressure: 736 Torr (mmHg) o Air temperature at inlet: 20°C Humidity: 60% (relative air humidity) o Intake air depression : } depends on individual engine set. o Exhaust back pressure : }	o Barometric pressure: 29.38 inHg (99 kPa) (746.2 mmHg) o Air temperature at inlet: 85°F (29.4°C) o Humidity: 0.38 inHg (1.3 kPa) (50%)

	BS 649 - 1958 (1960)	JISF 0401
Standard operating conditions	o Barometric pressure: 29.5 inHg (749 mmHg) 500 ft at sea level. o Air temperature at inlet: 85°F (29.4°C). o Water vapour pressure: 0.6 inHg (15 mmHg). o Temperature of cooling water at inlet to charge air cooler: 75°F (23.9°C). o Equipped with air silencer. o Equipped with exhaust system recommended by the engine manufacturer.	o Barometric pressure: 760 mmHg o Air temperature at inlet: 20°C o Humidity: 65%

15. *Explain the major advantages and disadvantages of 4-stroke and 2-stroke engines.*

Advantages of 4-stroke engines

(1) Higher thermal efficiency

The 4-stroke engine combustion stroke is a half of the 2-stroke engine at the same revolution speed. Therefore the effects of cooling and scavenging are much better in suction stroke and the quantity of charging air is also increased which results in better combustion and lower fuel comsumption.

Note: Large marine diesel engines such as 10,000 PS and over are 2-stroke engines in order to get better thermal efficiency by lower revolution. In addition, they can use low quality diesel fuel.

(2) The efficiency of supercharging effect is better than for 2-stroke engines and also scavenging action is done perfectly.

(3) The life of cylinder liner is longer.

(4) Higher revolution can be attained than with 2-stroke engine.

(5) The consumption of lubricating oil is much smaller.

Disadvantage of 4-stroke engines

(1) The structure is more complicated because of the necessity for a valve mechanism. It is therefore less compact than the 2-stroke engine.

(2) The flywheel is bigger than for the 2-stroke engine, because the fluctuations of rotative forces are larger than in the 2-stroke engine. A large flywheel absorbs these fluctuations and makes the operation smooth.

Advantages of 2-stroke engines

(1) It is more powerful than the 4-stroke engine. Theoretically the horsepower should be double compared with a 4-stroke engine because the combustion stroke takes place with each revolution but actually 1.5 - 1.7 times the horsepower can be attained with the same size of engine.

(2) Generally the valve mechanism is unnecessary because the structure is much simpler than for the 4-stroke engine.

(3) The flywheel is smaller because, as combustion strokes take place with every turn of the crankshaft, the fluctuation of rotative forces is smaller.

(4) Reverse rotation is easier than with the 4-stroke engine.

Disadvantages of 2-stroke engines

(1) The mean effective pressure is lower because early opening of exhaust valve is inevitable.

(2) Scavenging function is not sufficiently worked and the loss of scavenging pump reaches 5 - 10 percent of total horsepower.

(3) Combustion is incomplete and the fuel consumption ratio is not as good as for the 4-stroke engine.

16. *Why the horsepower of a 2-stroke engine is not twice that of a 4-stroke engine with the same piston displacement and revolution ?*

Theoretically, a 2-stroke engine horsepower should be double that of a 4-stroke engine horsepower for the same piston displacement and the same revolution.

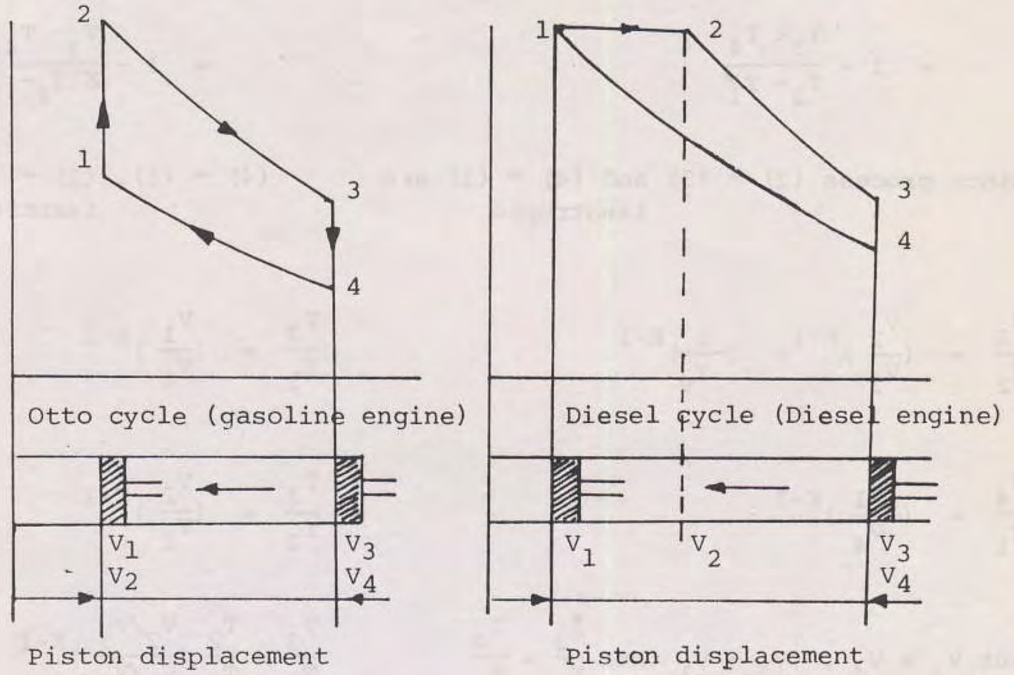
However, experiments show that a 2-stroke engine can attain at the most 1.7 times the horsepower of a 4-stroke engine. One of the reasons for this is that scavenging (removal of burnt gas) does not take place perfectly. Furthermore exhaust blow-down loss is increased which in turn induces a loss of mean effective pressure.

17. *Explain the main differences between the gasoline engine and the diesel engine.*

The main differences can be tabulated as follows:

Items	Gasoline engine	Diesel engine
Theoretical cycle	Otto cycle	Diesel (SABATI) cycle
Ignition equipment	Spark by electricity	Compression ignition (spontaneously)
Compression ratio	6-8-(10)	(13)-15-21
Thermal efficiency	24-28 %	32-34 %
Mean effective pressure	6-9 kg/cm^2	6.5 kg/cm^2 (15 kg supercharged)
Fuel consumption	260-300 g/PS.h	180-220 gr/PS.h
Exhaust temperature	(480 - 540°C) 650 - 760°C	(350 - 400°C) 450 - 540°C
Weight/PS	1.5 - 3 kg/PS	6 kg/PS - 20 kg/PS - (30 kg/PS)
Idling rpm	low	high
Vibration noise	small	large
Torque	small	large
Structure	simple	complex

18. *Calculate the thermal efficiency of Otto cycle and of diesel cycle.*



	Otto cycle	Diesel cycle
1.	Constant volume heat addition (Process (1) - (2))	Constant pressure heat addition
2.	Isentropic expansion (Process (2) - (3))	Isentropic expansion
3.	Constant volume heat rejection (Process (3) - (4))	Constant volume heat rejection
4.	Isentropic compression (Process (4) - (1))	Isentropic compression

$$Q\ in = Q_{1,2} = C_v(T_2 - T_1)$$

$$Q\ out = Q_{3,4} = C_v(T_4 - T_3)$$

$$\eta\ tho = \frac{\overline{W}\ net}{Q\ in} = \frac{C_v(T_2 - T_1) - C_v(T_3 - T_4)}{C_v(T_2 - T_1)}$$

$$= 1 - \frac{T_3 - T_4}{T_2 - T_1}$$

Since process (2) → (3) and (4) → (1) are isentropic

$$\frac{T_3}{T_2} = \left(\frac{V_2}{V_3}\right)^{K-1} = \left(\frac{1}{\gamma_v}\right)^{K-1}$$

$$\frac{T_4}{T_1} = \left(\frac{V_1}{V_4}\right)^{K-1}$$

but $V_3 = V_4$, $V_2 = V_1$ thus $\frac{T_3}{T_2} = \frac{T_4}{T_1}$

$$\boxed{\eta\ thd = 1 - \frac{T_3}{T_2} = 1 - \frac{T_4}{T_1} = 1 - \frac{1}{\gamma_v{}^{K-1}}}$$

where $= \frac{V_4}{V_1}$ = compression ratio

$$Q\ in = Q_{1,2} = C_p(T_2 - T_1)$$

$$Q\ out = Q_{3,4} = C_v(T_4 - T_3)$$

$$\eta\ tho = \frac{C_p(T_2 - T_1) - C_v(T_3 - T_4)}{C_p(T_2 - T_1)}$$

$$= 1 - \frac{T_3 - T_4}{K(T_2 - T_1)}$$

(4) → (1), (2) → (3) are isentropic

$$\frac{T_3}{T_1} = \left(\frac{V_1}{V_4}\right)^{K-1}$$

$$\frac{T_3}{T_2} = \left(\frac{V_2}{V_3}\right)^{K-1}$$

$$\frac{T_3}{T_4} = \frac{T_2}{T_1}\left(\frac{V_2/V_3}{V_1/V_4}\right)^{K-1}$$

$\frac{T_2}{T_1} = \frac{V_2}{V_1}$ (heat addition is constant-pressure)

$$V_3 = V_4$$

$$\boxed{\eta\ thd = 1 - \frac{\gamma_c{}^{K-1}}{\gamma_v{}^{K-1}\ K(\gamma_c - 1)}}$$

where $\gamma_c = \frac{V_2}{V_1}$ is known as the cut-off ratio.

19. *Which has higher thermal efficiency: gasoline engine or diesel engine ?*

Gasoline engine cycle is represented by Otto cycle.

The thermal efficiency of Otto cycle is given as follows:

$$\eta \text{ tto} = 1 - \frac{1}{\varepsilon \text{ k-1}} \quad \ldots\ldots (1)$$

k = polytropic constant

ε = compression ratio

For methods of calculating thermal efficiency of both Otto and diesel cycles refer to Q 18.

Diesel engine cycle is represented by diesel cycle:

$$\eta \text{ diesel} = 1 - \frac{1}{\varepsilon \text{ k-1}} \frac{(\sigma - 1)}{K(\sigma - 1)} \quad \ldots\ldots\ldots (2)$$

σ = cut-off ratio

$\therefore \eta \text{ Otto} > \eta \text{ diesel}$

Efficiency of diesel cycle is always lower than that of Otto cycle having the same compression ratio.

However, engine using diesel cycle usually operates at a higher compression ratio than engine using Otto cycle. A diesel engine has a higher thermal efficy than a gasoline engine according to the following values obtained experimentally.

$\varepsilon = 20$ (diesel cycle)

$\varepsilon = 6$ (Otto cycle)

Thermal efficiency of Otto cycle = 0.511

" " diesel cycle = 0.656

Then, a diesel engine has approximately 28 percent higher thermal efficiency than a gasoline engine.

20. *A diesel engine has the top clearance 8 mm, the engine bore 275 mm and the stroke 420 mm and compression ratio 15. What will be the compression ratio of this engine when the top clearance changes to 10 mm ?*

$$\text{Compression ratio} = \frac{V1 + V2}{V1} = \varepsilon$$

V2 engine's displacement volume

V1 clearance volume

$$\text{Piston area } \frac{\pi}{4} D^2 = \frac{\pi}{4} (275)^2 = 75625$$

$$\text{Compression ratio } 15 = \frac{V1 + 75625 \times 420 \text{ (stroke)}}{V1}$$

$$15\ V1 = V1 + 75625 \times 420$$

$$V1 = \frac{75625 \times 420}{14} = \frac{31762500}{14} = 2268750$$

Top clearance (T.C) 8 mm ⟶ clearance volume 226870

Top clearance (changed 10 mm ⟶ clearance volume X

(T.C. and clearance volume should be proportional)

$$\frac{8}{10} = \frac{226870}{X} \qquad X = \frac{226870 \times 10}{8} = 2835937.5$$

$$\varepsilon \text{ changed} = \frac{2835937.5 + 31762500}{2835937.5} = 12.2$$

$$\underline{\underline{\varepsilon = 12.2}}$$

<u>Note</u>: (1) From this example it is obvious that top clearance has a great effect on the compression ratio.

(2) The tolerance of top clearance for small engines is very small, ± 0.1 mm.

21. *How much air is required to combust 1 kg of diesel fuel ?*

Diesel fuel is a hydrocarbon mixture, such as:

C_nH_{2n} + 2 (gasoline C_7H_{16}) $C_{15}H_{32}$

The chemical reaction equation:

$$C_{15}H_{32} + 23O_2 = CO_2 + 9\ H_2O$$

Molecular weight:

$$(12 \times 15 + 1 \times 32)_{kg} + (23 \times 32)_{kg} = 15(12 + 32) + 16(2 + 16)_k$$

$$(180 + 32) + 736 = 660 + 288$$

Complete combustion of 1 kg of $C_{15}H_{32}$:

$$\frac{736}{212} = 3.47 \text{ kg of oxygen is necessary}$$

100 kg of air contains 23 kg of oxygen

$$736/212 \times \frac{100}{23} = 15.1$$

Answer: 15.1 kg of air is necessary to combust 1 kg of diesel fuel.

Note: The above quantity of air can be expressed in terms of volume ratio:

density of air .... 1.293 kg/m^3

15.1 kg of air volume 15.1/1.293 = 11.83 m^3

1 kg of fuel volume 1/0.85 = 1.176 litres (specific gravity of fuel: 0.85)

volume of air 11.83 m^3 = 11.83 x 10^3 litres = 11,830 litres

then 11,830 ÷ 1.176 = 10,005

Volume of air should be 10,000 times greater than the volume of fuel.

22. *Calculate the caloric power (H_L) of diesel fuel which contains the following components:*

Carbon (C)	*86% (of weight)*
Hydrogen (H)	*12.5% (of weight)*
Sulphur (S)	*1.5% (of weight)*
Oxygen (O)	*1.3% (of weight)*

Total lower caloric power H_L

$$H_L = 8100\ C + 29000\ (H - \frac{O}{8}) + 2500\ S\ (Kcal/kg)$$

<u>Answer</u>:

$$H_L = 8100 \times 0.86 + 29000\ (0.12.5 - \frac{0.013}{8} + 2500 \times 0.015)$$

$$= 10581\ Kcal/kg$$

23. *If 1 kg of diesel fuel is completely combusted in just one hour and its heat converted into work without loss, how much horsepower will be generated ?*

1 PS of 1 hour's work is equal to

75 x 60 x 60 = 270,000 kg.m

If 1 Kcal of heat = 427 kg.m (figure arrived at experimentally)

then 1 PS for 1 hour's work can be converted into Kcal of power:

270,000 kg.m/427 kg.m = 632.3 Kcal/h

Thus, 1 kg of diesel fuel which is 10581 Kcal/kg converted into work during 1 hour means

10581 Kcal - 632.3 Kcal = 16.73 PS

<u>Answer</u>: 16.73 PS

<u>Note</u>: This engine's fuel consumption ratio is:

$$\frac{1000\ g}{16:73} = 59.7\ gr/PS.h$$

The actual minimum fuel consumption ratio now attained is at the most 160 gr/PS.h even with the advanced technology in 1982.

Then $59.7/160 = 37\%$

The other 63 percent of energy is lost.

<u>Note 2</u>: The following table provides data on heat loss (heat balance).

Heat balance and exhaust data (full load):

Item			
Heat balance	Total heat input	100%	Kcal/hr (%)
	Coolant loss	10.4%	
	Air cooler loss	6.1%	
	Lubrication loss	2.7%	
	Friction loss	6.9%	
	Exhaust gas loss	31.4%	
	Others	4.8%	
	Brake horsepower as effective heat output	37.7%	

24. *Calculate the thermal efficiency of a diesel engine which consumes 180 grams of fuel per horsepower per hour (180 gr/PS.h), when the caloric power of the fuel is 10,000 Kcal/kg.*

1. The mechanical equivalent of this caloric power is 10,000 x 0.18 x 427 (kg.m)

2. The work of 1 horsepower in 1 hour is calculated as : 75 x 60 x 60 (kg.m)

therefore, thermal efficiency $= \dfrac{\text{work}}{\text{theoretical work}}$

$$= \frac{75 \times 60 \times 60}{10000 \times 0.18 \times 427} = 0.35 = 35\ \%$$

25. *A diesel engine consumes 18 kg of fuel in an hour. Calculate the following:*

   1) *Horsepower (Nc)*

   2) *Indicated horsepower (Ni)*

   3) *Fuel consumption ratio (bc)*

where, thermal efficiency (ηc) is 31%

mechanical efficiency (ηm) is 82%

caloric power of fuel (H_L) is 10500 Kcal/kg

Answer:

1) $$Nc = \frac{H_L B \eta c}{632.5} = \frac{10500 \times 18 \times 0.31}{632.5} = 92.6 \text{ PS}$$

2) $$Ni = Nc/\eta m = 92.6/0.82 = 112.9 \text{ PS}$$

3) $$bc = \frac{18 \times 10^3}{926} = 194 \text{ gr/PS.h}$$

26. *Compare different types of diesel combustion chamber:*

Direct injection chamber (D.I.)

Pre-combustion chamber (P.C.)

Swirl or Vortex combustion chamber (S. or V.C.)

The main characteristics of different types of diesel combustion chamber are tabulated below.

Item	D.I.	P.C.	S. or V.C.
Structure of chamber	simple	complex	Intermediately complex
Volume of chamber/ total combustion chamber	$\doteqdot 0$	0.15 ~ 0.4	0.5
Cross section area/ piston area	$\doteqdot 0$	0.2 ~ 0.5 %	2 ~ 3.5 %
Compression ratio	13 ~ 16	16 ~ 21	15 ~ 17
Injection pressure of nozzle	170~300kg/cm^2	110~160kg/cm^2	100~140kg/cm^2
Maximum gas pressure of cylinder	100 kg/cm^2	55 kg/cm^2	55 kg/cm^2
Fuel consumption ratio gr/PS.h	165 ~ 200	190 ~ 210	200 ~ 220
Start	easy	little harder	←
Type of injection nozzle	multi-hole	throttle or pintle	←
Atomization	by nozzle	combustion chamber	by nozzle
Loss of heat	small	large	medium
Mean effective pressure	low	high	medium

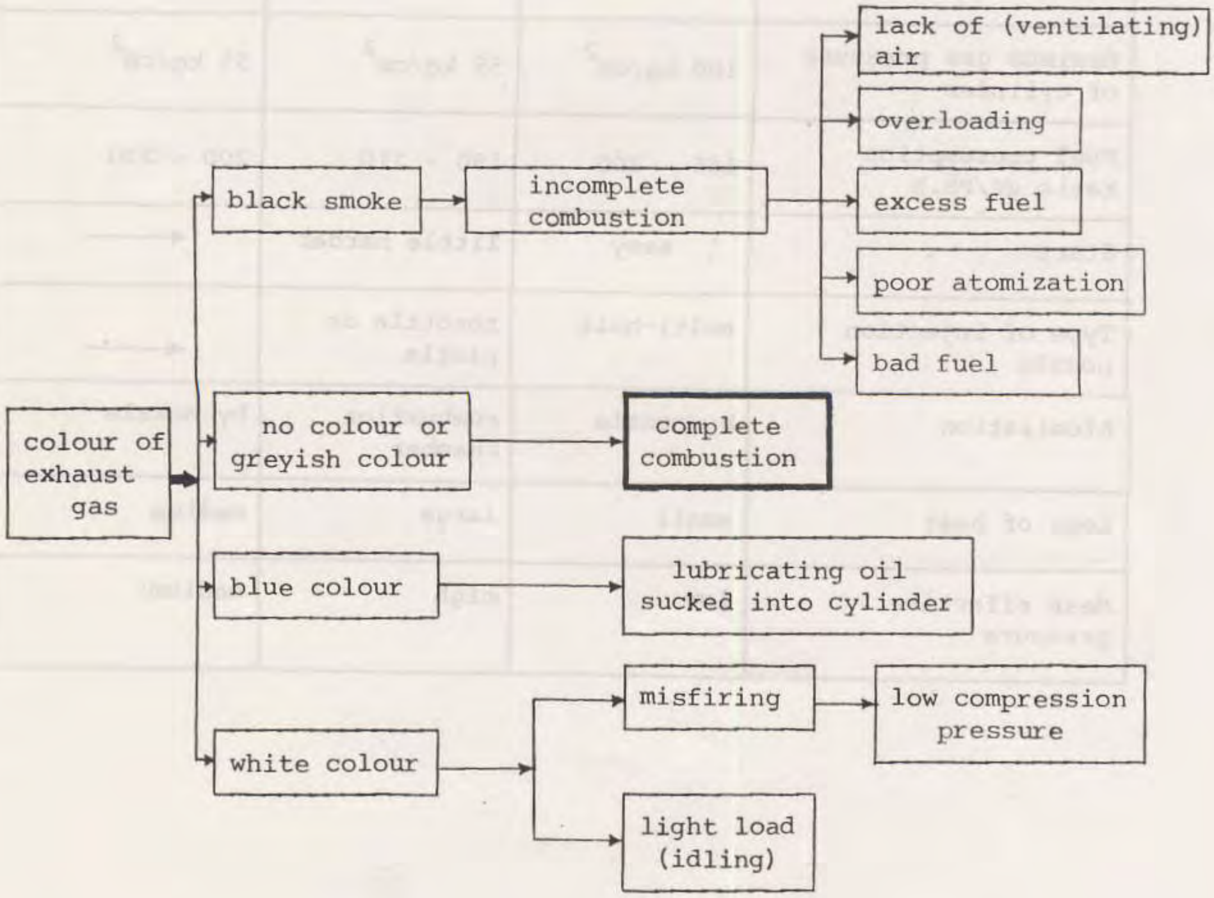
27. *What sort of relationship exists between combustion and the colour of exhaust gas ?*

There are three possible cases:

(1) The black colour of exhaust gas signals overload of engine or maladjusted fuel enjection system of some cylinders which causes load share to be unequally distributed.

(2) Blue smoke mixed with exhaust gas indicates that too much lubricating oil has been sucked into the cylinders and combusted.

(3) White exhaust gas indicates misfiring due to insufficient compression pressure.



28. *What is the cause of abnormal increase in the temperature of exhaust gas ?*

The standard exhaust gas temperature varies for different engines; the operator should check that it corresponds with the level of the lower temperature given in the engine specifications in the instruction manual.

In general, exhaust gas temperature rises for the following reasons.

1. Increasing the load (overload);
2. Delay of injection timing;
3. Injection nozzle dripping (bad atomization);
4. Premature opening of the exhaust valve;
5. Low quality of fuel oil.

29. *What is the relationship between the temperature of exhaust gas and the cylinder pressure in a diesel engine ?*

There is a close connection between the temperature of exhaust gas and the cylinder pressure. The temperature of exhaust gas is one of the most important factors to judge the operational conditions of the engine. When the exhaust gas temperature of all cylinders is higher than the standard this indicates overloading, and if only one cylinder is effected this indicates that there is some trouble with this cylinder itself.

A high exhaust temperature indicates loss of heat which decreases engine efficiency (horsepower) and causes seizure of the valve, cracking the cylinder head and accelerates the wear of the liner and rings.

The relation between the exhaust gas temperature and the cylinder pressure is tabulated below.

Cylinder pressure	Exhaust temp.	Engine condition
low compression pressure	low	clearance of piston liner large; small compression ratio
	normal	clogging of intake passage; shortage of blower pressure
	high	leakage of gas from valves and rings; shortage of intake air
high compression pressure		clearance of piston liner small; compression ratio too high
low combustion pressure	low	injection period short; insufficient fuel quantity
	normal	low cetane number fuel; shortage of air
	high	delay of ignition; malfunction of nozzle; higher back pressure
normal combustion pressure	low	light load
	normal	normal load
	high	overload; higher back pressure
high combustion pressure	low	advanced injection timing
	normal	nozzle opening damaged
	high	duration of injection too long

30. *Why do cracks occur between the bridge of the intake valve and the exhaust valve of the combustion surface of cylinder head ?*

A possible cause of such trouble may be the repeated thermal stress between the inlet valve port (relatively low temperature) and the exhaust valve port (higher temperature) which induces thermal fatigue and cracking of materials.

Prevention method

(1) avoid overload (coverheating) or operating in conditions of incomplete combustion (i.e. when there is black exhaust gas);

(2) avoid overheating of engine;

(3) avoid excessively high temperature of exhaust gas;

(4) clean the water passage of cylinder head and remove the scale;

(5) clean the exhaust passage;

(6) check the cooling water passage and remove the scale or other matter which clogs the passage;

(7) check the water pump performance;

(8) apply proper tightening force to the cylinder head, according to the standard torque recommended by the manufacturer. Tightening of bolts should be done evenly and in proper order.

31. *What is the pumping loss of an engine ?*

The pumping loss is the negative work which is lost by suction and exhaust work. In Fig. 1 the pumping loss is shown by W2 and the effective work is shown by W1.

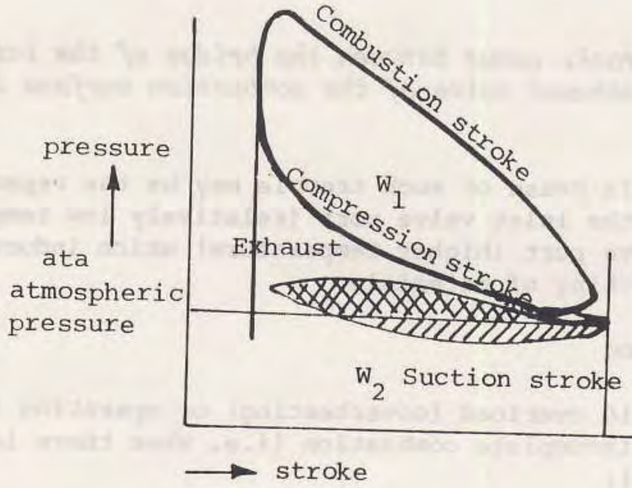


Fig. 1

When the piston goes downward for the suction stroke, the pressure in the cylinder is lower than the atmospheric pressure and some work is required. During the exhaust stroke the pressure in the cylinder is higher than the atmospheric pressure, and some work is required for upward movement of the piston. In both cases this is regarded as negative work (W2) or pumping loss.

Note: The net effective mechanical work is given by W1 - W2.

32. *How does a supercharger increase the power of an engine ?*

An increase in the amount of intake air provides more oxygen for combustion of a larger amount of fuel. Thereby there is an increase in horsepower for the following reasons:

From the formula (2) in Q (5) horsepower (Ps) is indicated

$$Ps \propto Pme \ . N \ . V$$

$$Pme \propto \eta_1 \ \eta_v \ \gamma_2$$

$$\gamma_2 \propto \frac{P}{T}$$

$$Ps \propto N \ V \frac{P}{T} \qquad \ldots\ldots\ldots\ldots\ldots(1)$$

where

Ps ..... engine horsepower

Pme .... effective mean pressure

N ...... engine revolution

V ...... engine total displacement volume

η_v ..... intake efficiency

η_l ..... thermal efficiency

P ...... intake pressure

T ...... temperature of intake air

γ_2 ..... density of air

Formula (1) shows how the engine horsepower is increased by making the intake pressure higher and intake air temperature lower. Intake pressure is increased by supercharger and temperature is decreased by means of air-cooler.

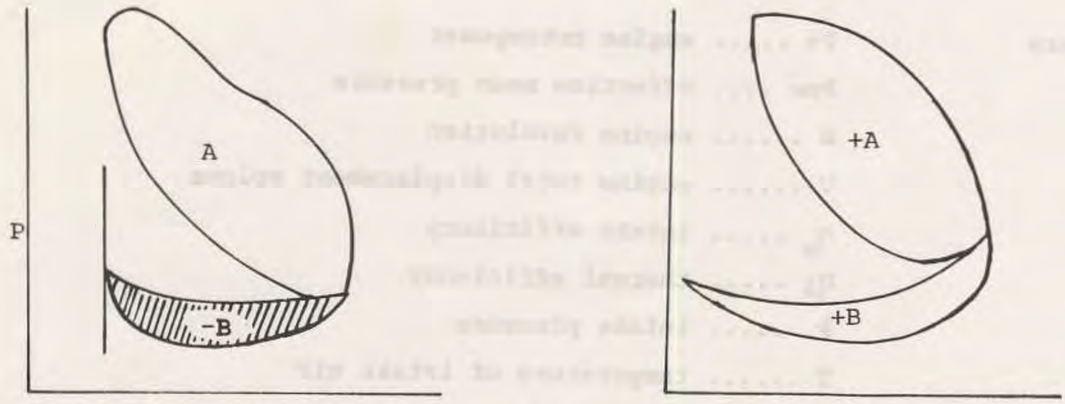
The above is the fundamental principle of supercharging (with air-cooler). Furthermore, the supercharger increases the power in the following ways:

1) Increasing the mean effective pressure;

2) Increasing the volumetric efficiency or charging efficiency by air boosting;

3) Reducing the pumping loss (Refer to Fig. 1). If the intake pressure is higher than atmospheric pressure in suction stroke, the pumping loss does not occur as in a naturally aspirated engine. In a supercharged engine the pumping loss is positive whereas in a naturally aspireted engine it is negative;

4) Maximum combustion pressure is increased.



Naturally aspirated engine

W = A - B

W = A + B

Supercharged engine

Fig. 1

33. Is it easy to modify a naturally aspirated diesel engine into a supercharged engine ?

To modify a naturally aspirated engine into a supercharged engine is not an easy process, and it varies according to engine specifications. The following basic changes are required:

1) Camshaft (cam profiles): to increase the overlapping period of intake and exhaust valve timing;

2) Injection pumps: to increase the fuel quantity;

3) Exhausting manifolds for turbocharging and intake manifolds;

4) Cylinder head including intake valve and exhaust valve, pistons, stronger crankshaft, bearings, and tightening bolts of connecting rod if necessary. (These changes are necessary because of the increased air intake and increased maximum combustion pressure (Pmax))

5) Additional accessories

   1. turbocharger, intercooler, intake manifold, pipe etc.

<u>Note</u>: See Question 32.

34. *What are the advantages and disadvantages of supercharging ?*

Advantages:

1) Increased horsepower by 50-100 percent, compared to the naturally aspirated engine. The additional weight is only 3-5 percent;
2) The engine weight per horsepower is reduced by 30-50 percent;
3) Space is economized;
4) Fuel consumption ratio is decreased by 3-5 percent;
5) Mechanical efficiency is increased by 7-8 percent;
6) Combustion noise is reduced and a simpler silencer can be used.

Disadvantages:

1) Maximum combustion pressure (Pmax) is increased;
2) Starting performance is decreased a little by increasing the compression ratio;
3) Combustion at light load is not so good;
4) Initial cost and maintenance cost are increased;
5) Surging phenomenon might occur (see Question 35).

35. *Explain the valve timing diagram of a supercharged engine. Explain how the timing of a supercharged engine differs from that of a conventional engine.*

(1) The opening time of intake valve is earlier and closing time of exhaust valve is later than for naturally aspirated engine. The period of valve overlap is longer (see figure below).

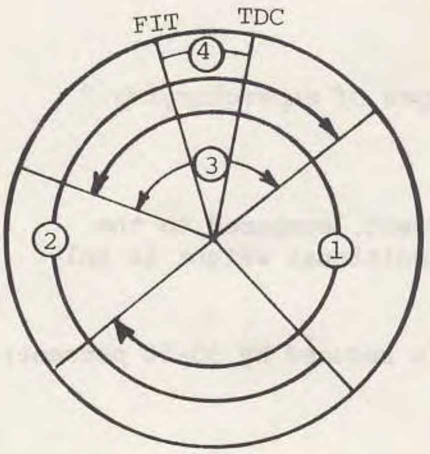


Fig. 1

Timing \ Engine	SC	NA
1 Intake opening angle	285°	230°
2 Exhaust opening angle	300°	230°
3 Valve overlap	130-150°	26°- 45°
4 Fuel injection timing (F.I.T)	18°BTDC	25° BTDC

A large valve overlap improves scavenging and as a result larger amount of combustion gas increases horsepower.

(2) To control the excessive increase in maximum combustion pressure the fuel injection timing (FIT) is later than for the conventional engine. Also, the top clearance is larger than for conventional engines.

36. *One of the limitations of the diesel engine is surging. What is surging and what causes it ?*

In a supercharged engine sometimes the phenomenon of surging (also called blower surging) can be seen. Surging phenomenon is a kind of self-exciting vibration caused by unstable pressure, breakdown of air delivery, accompanying pulsation of air pressure, back blow of air and big noise which occur in operation of blower, or axial flow turbine compressor.

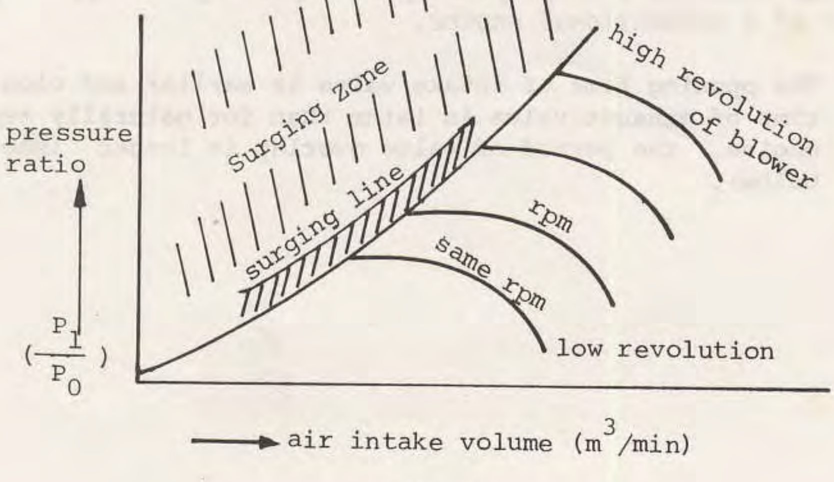


Fig. 1

Causes of surging:

1. Mis-matching of turbocharger and engine;
2. Atmospheric temperature, especially high temperature;
3. Increased resistance due to dirt adhered to blower blades or clogged duct and air-cooler passage.
4. Operation of engine
   (1) Increasing of load, especially torque rich condition.
   (2) Sudden change from heavy load to light load.
5. Malfunctioning of engine
   (1) misfire of a cylinder;
   (2) malfunction of injection timing;
   (3) malfunction of intake valve and exhaust valve sheet fitting (gas leakage);
   (4) malfunction of turbine nozzle;
   (5) others.

Note: Countermeasures in case of surging

1. reduce the revolution of engine,
2. change the diffuser.

Consult the manufacturer if necessary.

The characteristics of the low, medium and high-speed engines are tabulated below.

37. *Compare the characteristics of the low, medium and high-speed diesel engines which have continuous rating 500 horsepower.*

Item	Low speed	Medium speed	High speed	Remark
Revolution (rpm)	< 500	500~900	1000 <	
Piston speed (meter/second)	< 6	6.0~8.00	8 <	
Stroke/bore	1.5~1.7	1.2~1.3	1.1~1.2	
Weight*/horse-power	> 20 (kg/ps)	10~20 (")	< 10 (")	* with clutch
Combustion system	direct injection	direct injection	direct injection or precombustion chamber	
Starting system	compressed air	compressed air	compressed air or electric starting motor	
Lubricating system	forced lub	←	←	
Cooling system	direct sea water	←	direct (sea water) or indirect (fresh water)	
Main metal support	fixed in bed plate	←	hanger bearing type (from cylinder block)	
Connecting rod	horizontal level split	← or angular split	angular split **	** to reduce contact pressure of bearing by larger big end of rod.
Main metal and crank pin bearing	bi-metal (white metal layer)	bi-metal or tri-metal (white)	tri-metal (kelmet)	
Crankshaft	solid forging	←	←	
Crank journal	non-hardening	non-hardening	hardening	
Piston	cast iron	cast iron or cast light alloy	cast light alloy	
Piston pin	fixed	floating type	floating type	

cont'd.

Item	Low speed	Medium speed	High speed	Remark
Intake and exhaust valve	cage system (dismount from head) or valve case box	cylinder head and valve seat: monoblock	←	
Injection nozzle	multi-hole	←	multi-hole or single hole for precombustion chamber	
Lubricating pump	plunger	geared	geared	
Cooling pump	plunger	plunger or centrifugal	centrifugal	
Reverse and reduction	Mietz and Weiss	reduction gear (duplex disc)	←	
Clutch	friction cone	multiple friction plate	←	
Position of flywheel	opposite side of clutch (varies by builder)	clutch side (output side of crank)	←	

38. *What is the general method of correcting the valve timing of an engine under the following conditions :*

   (1) If the valve opens early and closes late ?

   (2) If the valve opens late and closes early ?

   (3) If the valve opens early and closes early ?

   (4) If the valve opens late and closes late ?

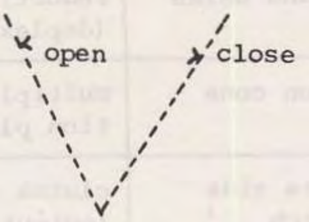
Refer to Figs. 1 - 4 below.

(1) The valve clearance should be adjusted to large by the adjusting screw. Another possible answer is that the length of the push rod should be shortened.

(2) The valve clearance should be adjusted to small by the adjusting screw, or the length of the push rod should be increased.

(3) The crankshaft (pinion) should be rotated (advanced) according to the engine rotating direction against camshaft gear (or camshaft gear should be delayed by one or two teeth). The problem arises from misassembly of the gears of camshaft and crankshaft pinion gear, when the counter marks of gears do not coincide.

(4) The crankshaft pinion should be rotated in reverse direction of engines rotation against camshaft gear.



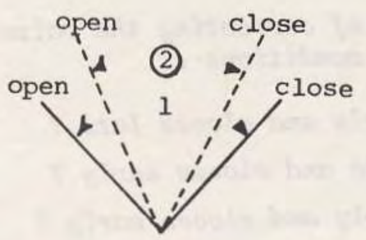


Fig. 1

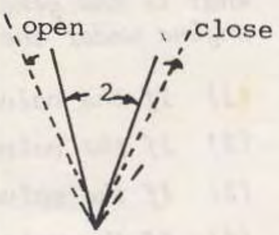


Fig. 2

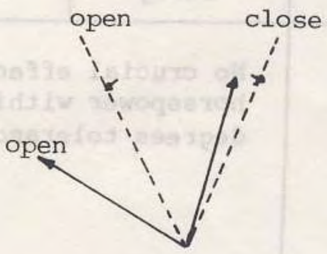


Fig. 3

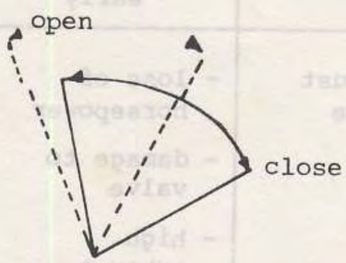


Fig. 4

39. *Describe briefly the effects of valve timing.*
What are the key points to adjust ?

	Open		Close (*)	
	early	late	early	late
Intake valve	No crucial effect on horsepower within 5^{o} degrees tolerance		Reduces the compression pressure	

(*) Key point to adjust : close timing should be accurate for intake valve.

	Open (*)		Close	
	early	late	early	late
Exhaust valve	- loss of horsepower - damage to valve - high exhaust temperature	- suction of new air reduced - loss of horsepower	No crucial effect on horsepower within 5^{o} degrees tolerance	

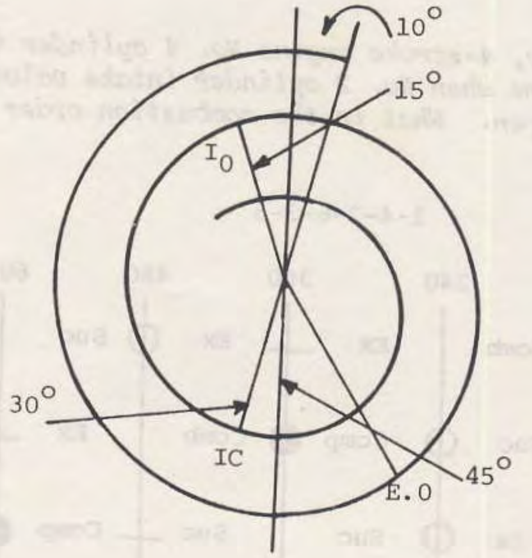
(*) Key point to adjust : open timing should be acourate for exhaust valve.

40. *The table shows a typical valve diagram of a 4-stroke engine.*

Answer the following questions.

Intake valve		*Exhaust valve*		*Valve clearance*
Open	*Close*	*Open*	*Close*	*0.4 mm*
BTDC 15^{o}	*ABDC* 30^{o}	*BBDC* 45^{o}	*ATDC* 10^{o}	

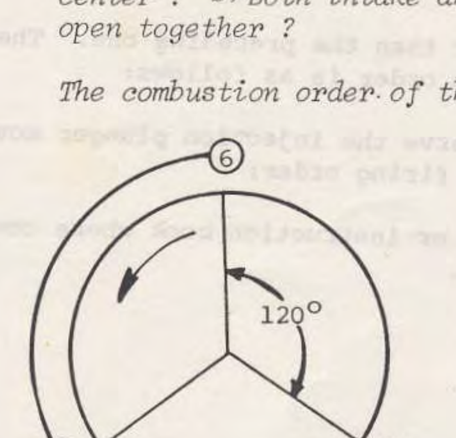
1. *How many degrees is the angle of opening of intake valve ?*
2. *How many degrees is the angle of opening of exhaust valve ?*
3. *How many degrees is the angle of overlap when valves are open simultaneously ?*
4. *How many degrees is the angle when both valves are closed ?*



1. Angle of opening (Intake) $15^{\circ} + 180^{\circ} + 30^{\circ} = 225^{\circ}$
2. Angle of opening (Exhaust) $= 235^{\circ}$
3. Overlap $15^{\circ} + 10^{\circ} = 25^{\circ}$
4. Both valves closed $60^{\circ} + 180^{\circ} + 45^{\circ} = 285^{\circ}$

41. *In a 6-cylinder, 4-stroke engine, we assume that both the intake valve and the exhaust valve of No. 6 cylinder are open at the same time. a) if this engine (crankshaft) turns a further 120°, which cylinder is the nearest to the compression top dead center ? b) Both intake and exhaust valve of which cylinder open together ?*

The combustion order of this engine is 1-4-2-6-3-5.



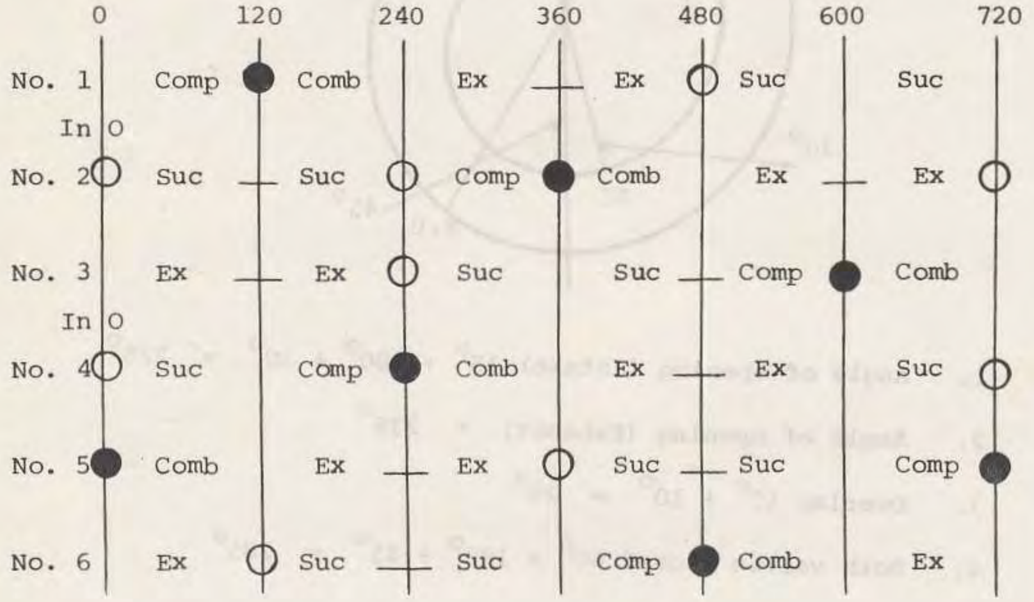
Combustion order 1-4-2-6-3-5

a) No. 4 cylinder compression TDC.

b) No. 3 cylinder suction TDC.

42. *In a 6-cylinder, 4-stroke engine No. 4 cylinder intake valve is open at the time when No. 2 cylinder intake valve is just beginning to open. What is the combustion order of this engine ?*

1-4-2-6-3-5



NG 1-3-5-6-2-4
" 1-3-2-6-4-5
" 1-5-2-6-4-3

$\boxed{\text{1-4-2-6-3-5}}$

NG 1-5-3-6-2-4

Note: This problem is a little harder than the preceding one. The best way to find the combustion order is as follows:

(1) Turn the flywheel and observe the injection plunger movement order which indicates the firing order;

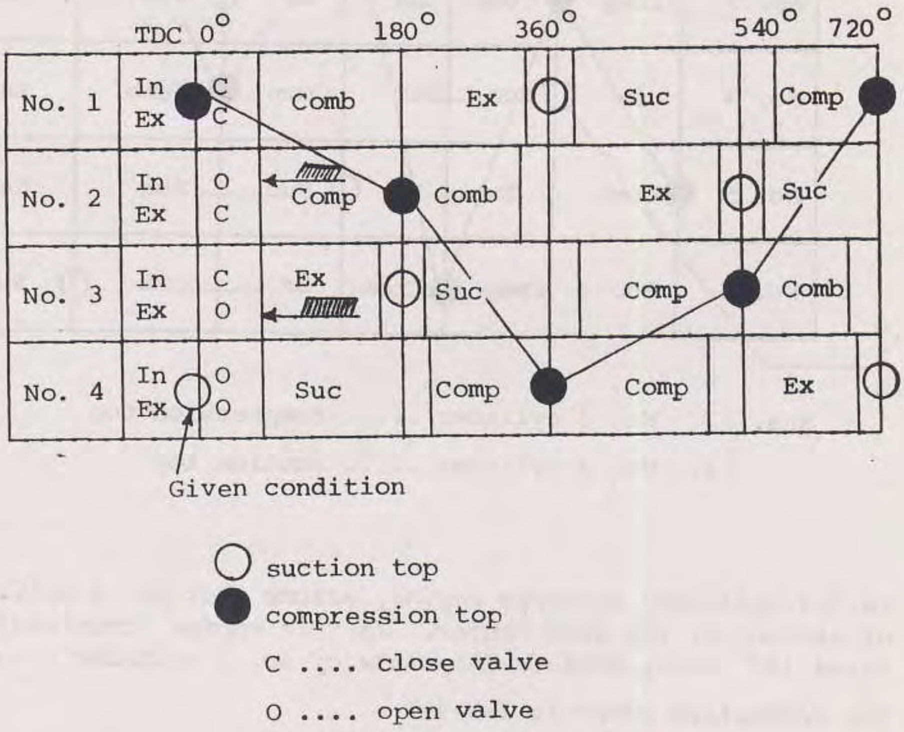
(2) See the engine name plate or instruction book where combustion order should be indicated.

43. *In a 4-cylinder, 4-stroke engine, both the intake valve and the exhaust valve open at No. 4 cylinder: which cylinders' valves are open ?*

The combustion order is 1-2-4-3.

Answer:

$720^\circ \div 4 = 180^\circ$ interval of combustion

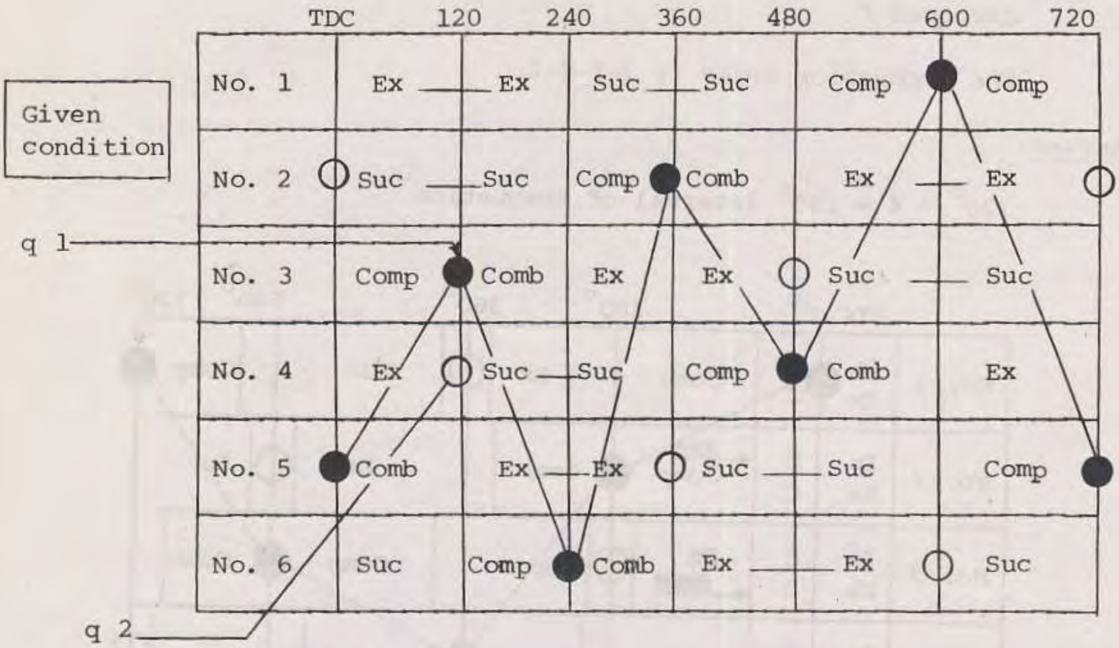


Ans. No. 2 Intake valve (Open) ← shows

No. 3 Exhaust valve (Open) ← shows

44. *In a 6-cylinder, 4-stroke engine, assume that No. 2 cylinder is at suction of top dead center. If this engine (crankshaft) turns a further 120°, which cylinder is the nearest to the compression top dead center ? Both inlet valves and exhaust valves of which cylinders (0) open together ?*

The combustion order is 1-5-3-6-2-4.

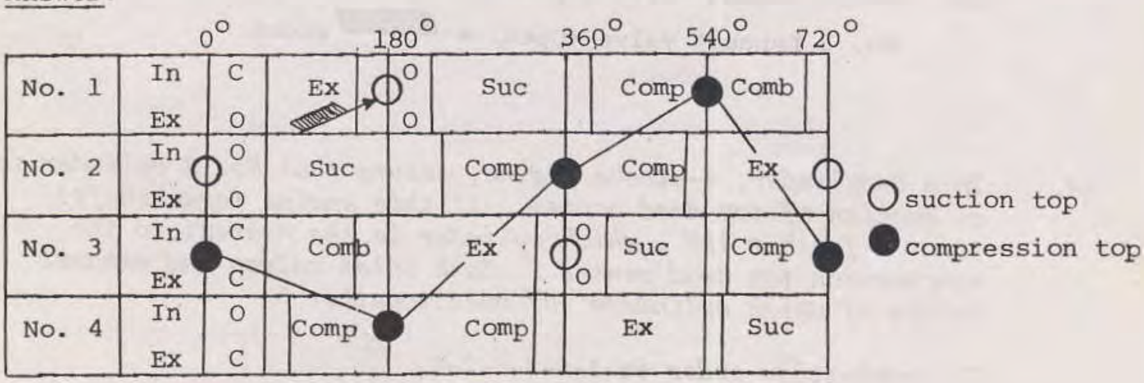


Ans. { 1. No. 3 cylinder ..... compression top
2. No. 4 cylinder ..... suction top

45. *In a 4-cylinder, 4-stroke engine, assume that No. 2 cylinder is at suction of top dead center. If this engine (crankshaft) turns 180° more, what is the state of No. 1 cylinder's valves ?*

The combustion order is 1-3-4-2.

<u>Answer</u>:



Ans. { Intake valve .... open
Suction valve ...... open

46. *The cam gear of a 4-stroke engine has 36 teeth, the exhaust valve closes 20^o early and opens 20^o early; how can the timing be corrected ?*

One revolution of cam gear (36^o) corresponds to 36 teeth (cogs)

$$1 \text{ tooth (cog)} = \frac{360}{36} = 10^o$$

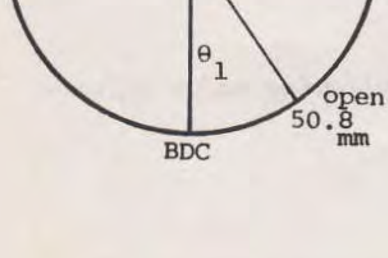
Therefore, 20^o corresponds to 2 teeth of the cam gear.

In order to correct a 20^o mis-timing, the cam gear should be delayed by 2 teeth (or the crankshaft should be turned in reverse direction by 2 teeth of its gear).

Note:

This is a concrete example of Q. 35.

47. *The exhaust valve of a 4-stroke engine with a 32 cm flywheel, opens 50.8 mm as measured on the flywheel before the bottom dead center, and closes 12.7 mm after the top dead center. How many degrees is the angle of opening ?*



The angle of $\theta 1$ (Open exhaust valve before BDC)

$$\frac{\theta 1}{360} = \frac{5.08}{32 \times \pi} \quad \theta_1 = \frac{5.08 \times 360}{32 \times 3.14} = 18.20$$

$$\theta_2 = \frac{12.7 \times 360}{32 \times 3.14} = 4.55$$

θ_2 (close exhaust valve after TDC)

Total angle of exhaust opening

$$18.20 + 180 + 4.55 = 202.8^o$$

48. *a) How is injection timing checked ?*

b) What are the effects of incorrect timing ?

a) The methods of checking and adjusting will differ with the engine type. Consult the instruction manual accordingly.

A general method for Bosh type engine is as follows:

(1) The control rack or operation lever should be positioned to nearly maximum power and the screw holder should be disassembled;

(2) Take off the delivery valve and its spring;

(3) Reassemble the screw holder;

(4) Open the fuel cock;

(5) Turn the flywheel until the fuel just stops flowing (this position is the nominal fuel injection position;

(6) Check that this position coincides with the mark of fuel injection on the flywheel exactly.

b) Effect of inadequate injection timing

(1) If the fuel injection is too early the engine loses power and sometimes induces diesel knocking;

(2) If the fuel injection is delayed the engine also loses horsepower causing incomplete combustion and the appearance of black smoke.

Note: Generally, the tolerance of fuel injection timing is $\pm$ 1 degree

49. *How much tolerance is permissible for the following major adjustments in an engine:*

1. *Injection timing,*
2. *Valve timing,*
3. *Difference of exhaust temperature between cylinders at rated horsepower,*
4. *Clearance of crankpin and its bearing,*
5. *Deflection of crankshaft.*

The values differ somewhat depending on the make of the engine but in general they are as follows:

1. Fuel injection timing tolerance is $\pm 1^{\circ}$

2. Valve timing $\pm 1^{\circ} \sim 5^{\circ}$
   and valve clearance $\pm$ 0.1 mm

3. Differences of exhaust temperatures between cylinders with engine type (the instruction manual should be consulted, but generally $\pm$ 15 - 20° C will be permissible)

4. Clearance of crankpin and its bearing (in general):

   d = standard diameter of pin

   The material of bearing ...... white metal

   $$\left(\frac{3}{10000} \sim \frac{10}{10000} \right) d$$

   $$\left(\frac{10}{10000} \sim \frac{14}{10000} \right)^{d} \quad \text{kelmet metal}$$

   repairing limit approximately 0.13 + 0.001d

5. Deflection of crankshaft

   $$\frac{1}{10000} \times s \qquad s \text{ ..... stroke of engine (mm)}$$

   Repairing limit is $\frac{2}{10000} \times s$

50. *What is the important point in converting a road vehicle diesel engine into a marine engine ?*

Generally speaking the engine which is used for road vehicles should have its maximum horsepower reduced by at least 20 percent by restraining the maximum injection quantity of injection pump, otherwise engine troubles might occur because of heavy load. Alternatively, a lighter propeller should be used for the same reason.

And clutch capacity should be carefully examined. See also Question 51 on clutch capacity.

51. *What is the clutch capacity and how is it calculated ?*

Note:

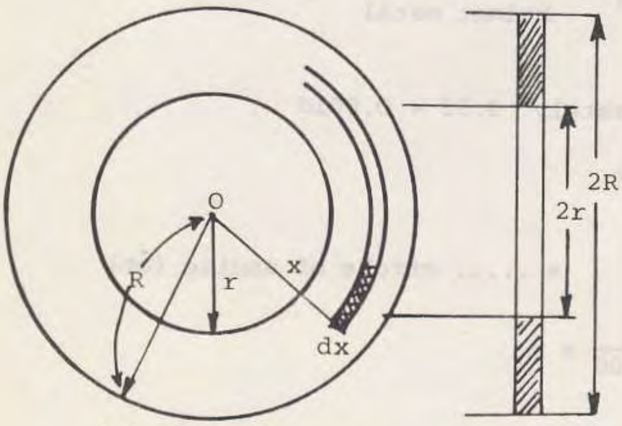
See also Q 50 on how to convert a road vehicle diesel engine into a marine engine.

In general, the clutch capacity is indicated by the maximum torque (kg.m).

If the clutch capacity is smaller than the maximum torque, slipping will occur and the clutch will not be able to transmit the horsepower of the engine.

Therefore the capacity of clutch should be 25-35 percent larger than the maximum torque of the engine.

The principle of clutch



friction plate

The figure shows a friction plate whose diameter is 2R with the hole diameter 2r

small area .... A at x (radius $= 2\pi x \cdot dx$

T_A ..... Torque of centre 0 around A

$T_A = A \times X$

Total torque of friction plate A x R

$$A \times R = \int_r^R 2\pi r \cdot dx \cdot x = 2\pi\left[\frac{R^3}{3} - \frac{r^2}{3}\right]$$

slip torque ..... T_c (kg.cm) $= AR \times \mu \times q \times z$

where μ = friction coefficient 0.1 ~ 0.3 varies according to material

q = contact pressure kg/cm^2 2 ~ 3.5 kg/cm^2

z = number of friction surfaces

$$T_c = AR \times \mu \times q \times z \quad \ldots\ldots\ldots (1)$$

$$E_t \ldots\ldots \text{Engine maximum torque (kg. m)} = \frac{BHP}{N \text{ rpm}} \times 716.2 \ldots\ldots (2)$$

where N .... engine (rpm)

if $E_t \leqq T_c$ there is no slipping

$$\frac{T_c}{E_t} = \geqq 1.4 = e \quad \ldots\ldots\ldots \text{safety factor}$$

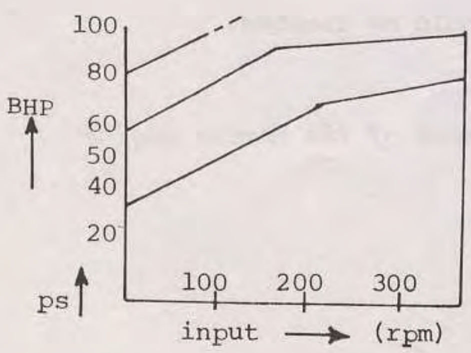
From formulas (1) and (2) $AR \times z \times 0.1 \times 3.5 \geqq \frac{BHP}{N} \times 716.2$

find AR by empirical method by trial and error.

assuming z = 12, 8, 6, (number of friction plate x z)

Notes:

(1) Request the clutch manufacturer to provide a catalog with the clutch capacity diagram (Fig. 1).



Choose the most suitable type according to the diagram.

BHP .... engine output horsepower

Clutch capacity diagram

(2) To select a clutch it is not sufficient to examine only torque. The following items should be examined.

1. Torque and total connecting amount of work.
2. Initial period (in seconds) before full operation is attained.
3. Disposal of generated friction heat.
4. The life of friction plate.

52. What is the cause of excessive engine vibration ?

Excessive vibration may be caused by any of the following **factors:**

(1) Engine bed not hard enough or tighting bolts are damaged;

(2) Operation at critical speed which comes from torsional vibration (manifested especially as gear noise);

(3) Overloading;

(4) Engine knocking;

(5) Variation of maximum combustion pressure between cylinders;

(6) Increased clearance of bearings, or damaged bolts;

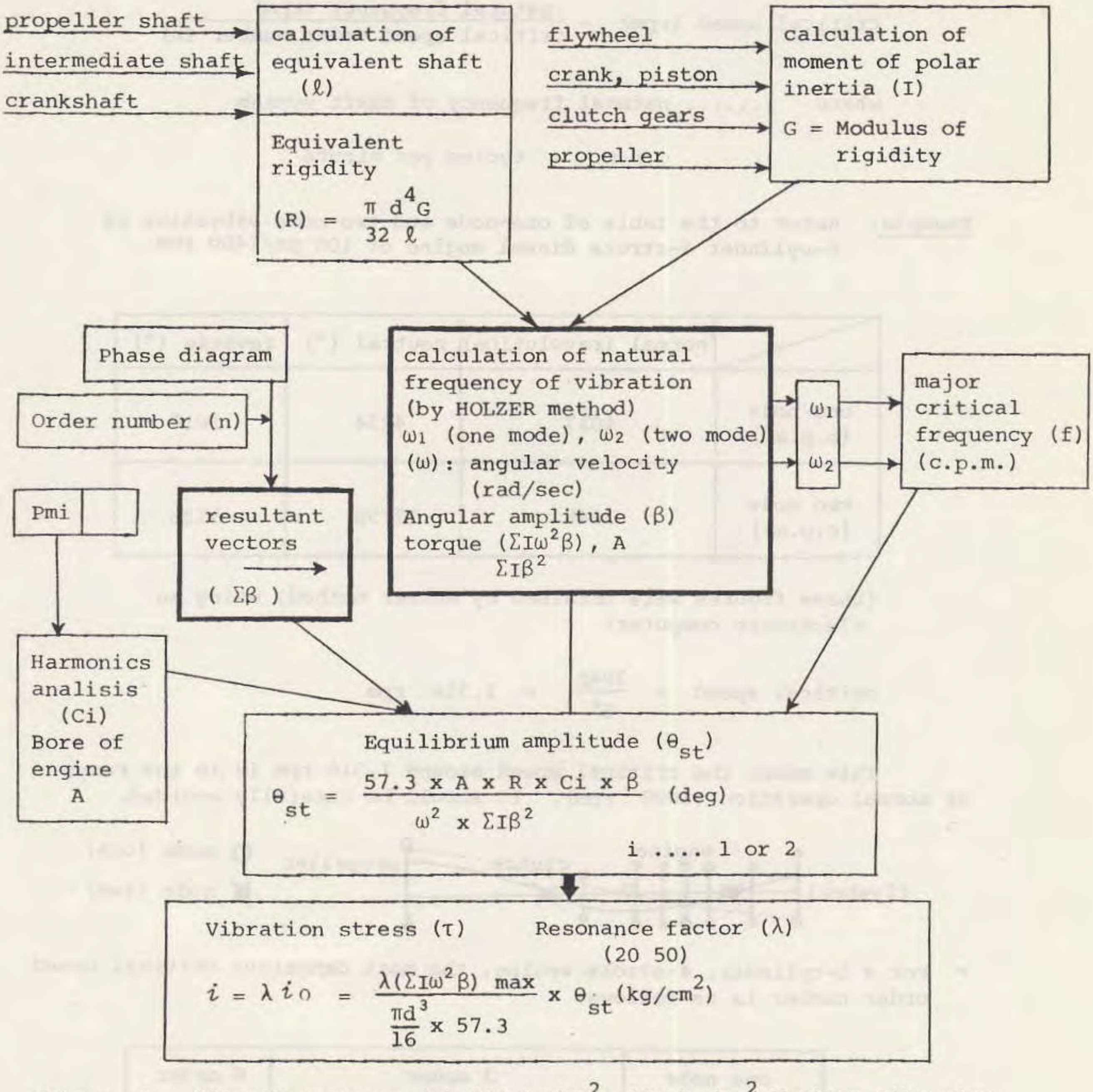
(7) Cylinders worn out;

(8) Incomplete combustion in cylinders.

Note: The amount of crank deflection should be checked.

53. How is the torsional vibration stress of the marine engine calculated ?

The calculation is shown by the flow chart. Stress is indicated by (τ)



Notes:
1. Vibration stress (τ) 250 kg/cm^2 ~ 300 kg/cm^2. Maximum vibration stress occurs near the node.
2. Twisting maximum angle should not exceed ± 0.3°.
3. ☐ usually calculated by a programed electronic computer.

54. *How is the critical speed of the main engine calculated and observed in practice ?*

$$\text{critical speed (rpm)} = \frac{\text{natural frequency (cpm)}}{\text{critical speed order number (n)}}$$

where .... natural frequency of shaft system

c.p.m. = cycles per minute

Example: Refer to the table of one-node and two-node vibration of 6-cylinder 4-stroke diesel engine of 100 ps/1400 rpm.

	normal (revolution)	neutral (")	reverse (")
one node (c.p.m.)	1017	4234	1019
two node (c.p.m.)	3948	12750	5536

(these figures were obtained by Holzer method, using an electronic computer)

$$\text{critical speed} = \frac{3948}{3^*} = 1{,}316 \text{ rpm}$$

This means the critical speed around 1,316 rpm is in the range of normal operation (1400 rpm). It should be carefully avoided.

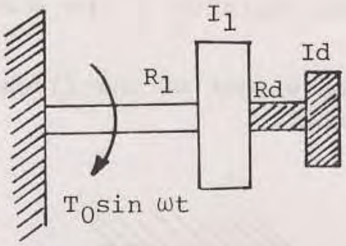
* For a 6-cylinder, 4-stroke engine, the most dangerous critical speed order number is as follows:

one node	3 order		6 order
two node	3* order	7.5 order	9 order

Generally speaking the critical speed is indicated on the tachometer by a red line, and if the engine revolution reaches this red line abnormal noise of gear may occur because of resonance. (Revolution should be quickly increased or decreased, below or above the red line on the tachometer).

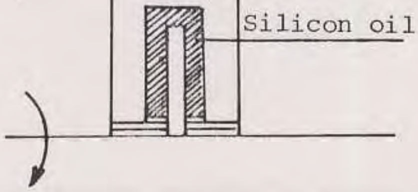
55. *Are there any measures to avoid torsional vibration ?*

(1) By dynamic damper



By adding a new vibration system Rd, Id against the initial node location (R_1, I_1). If we choose $_d = \sqrt{\frac{Rd}{Id}}$ force of $T_0 \sin \omega_t$ to coincide with ω the amplitude of R_1, I becomes zero and Rd, Id becomes equal force but of reverse direction. Such addition of R_d, I_d is called dynamic damper.

(2) By friction damper



Resonance energy is absorbed by the cohesive resistance of silicon oil.

(3) By moving the critical speeds farther away from the engine speed for ordinary use, by changing the dimension of its system estimated frequency by original formula (a)

$f \propto G \frac{d^2}{r} \sqrt{\frac{1}{WL}}$ .... (a)

see Fig. 1 below

G .... modulas of elasticity of shaft

W .... weight of flywheel

d .... diameter of shaft

L .... length of shaft

r .... radius of weight

To increase critical revolution:

(1) Increasing the diameter of the shaft

(2) Decreasing the length of the shaft

(3) Increasing the rigidity of the shaft (G) by changing material

(4) Decreasing the weight of the flywheel and counterweight of the crankshaft.

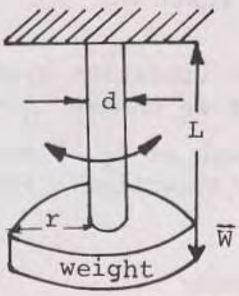


Fig. 1

56. *What is the cause of piston expansion trouble ?*

The cause of trouble may be as follows:

1. Deformation of cylinder or piston when cylinder liner is mounted/overhauled or replaced. The difference of inside diameter should be within a certain limit such as 0.02 mm at the top, middle and bottom position, measured by cylinder gauge.

2. Overloading for a long period or abnormal combustion such as high combustion pressure or high temperature.

3. A wrong combination of piston and liner, causing the clearance of liner and cylinder to become smaller than recommended by the manufacturer; this occurs especially when the liner or piston are replaced.

4. Overheating or lack of cooling water, especially when slowing down suddenly after a long period of overloading.

5. Too much carbon deposit from low quality fuel or lubricating oil.

<u>Note:</u> In general, the clearance of piston and liner (D) should be approximately:

$$D \doteqdot 0.03^{mm} + D(\beta_p \Delta t_p - \beta_c \Delta_{tc})$$

where D .... standard bore (mm)

β_p .... expansion coefficient of piston material

β_c .... expansion coefficient of liner material

tp ...... temperature of piston at maximum load

tc ...... temperature of liner at maximum load

57. *a) What are the external signs that a cylinder liner is worn out ?*

b) When must a cylinder liner be replaced ?

a) 1. The compression pressure and horsepower decrease and there is an increase in leakage of combustion gas from the piston ring.

2. Lubricating oil deteriorates faster and its consumption is increased.

3. Combustion of fuel becomes bad and consumption increases.

4. Turning of flywheel by a turning bar becomes easier because of lack of compression pressure. For under 250 mm bore of engine, the cylinder liners must be replaced when approximately 1/100 of diameter is worn out.

b) For a larger diameter bore the limit is when wearing off reaches 8/1000 of the original bore. If the liner is chrome-plated the limit is when this plated surface is worn out or partially stripped off.

58. *How much of gap is necessary for the piston ring with 300 mm bore if the ring temperature is supposed to rise by $120^{o}C$ and the expansion coefficient of the ring is 11×10^{-6}?*

The circumference of the ring's outer surface is

$$\pi \times D = 300 \times 3.1416$$

difference of temperature : 120^{o}.

The total amount of expansion of ring

$$\frac{300 \times 3.1416 \times 11 \times 120}{1{,}000{,}000} = 1.244 \text{ (mm)}$$

Then, the minimum ring gap is 1.244 mm.

59. a) *What is the standard side clearance between the piston ring and piston grooves ?*

b) *What is the standard piston ring gap ?*

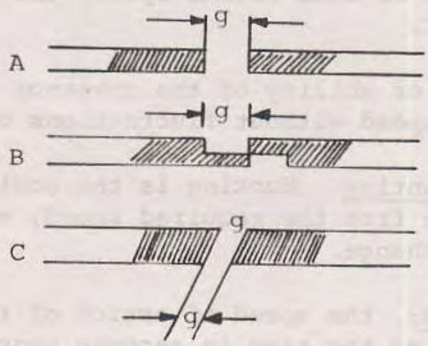
The recommended side clearance can be found in the engine builder's repairing manual. In general it should be as follows:

(a) Side clearance for small engines (bore size under 350 mm) is approximately 0.05 mm.

For engines whose bore size is over 350 mm it is approximately as follows:

1st and 2nd ring	0.08 - 0.10 mm
3rd and over	0.05 - 0.06 mm.

(b) The ring gap should be decided according to differences of temperature between ring and liner. Sectional view of common types of ring gaps are illustrated in A, B and C:



$g = (0.0025 - 0.0035) \times$ Bore (mm) (straight joint)
(step ")
$g = (0.035 - 0.0045) \times$ Bore (mm) (Angle ")

60. Enumerate the main characteristics of a governor that determine the degree of control of the engine.

Diesel-engine governor must have certain characteristics to fit the type of load of a particular engine.

(1) Speed droop: a decrease in the speed of engine from no load to full load, usually expressed as a percentage of normal speed or sometimes of average speed. If the speed of droop is designated by P, N_0 is no-load speed N_1 is full load or rated speed.

$$P = \frac{N_0 - N_1}{N_1} \times 100$$

(2) Isochronous governing: the speed of engine is truly constant at all loads, i.e. there is a perfect speed regulation with zero speed droop.

(3) Sensibility: the change in speed necessary before the governor will make a corrective movement in the fuel supply. It is generally expressed as a percentage of the normal or mean engine speed; the smaller the value the better.

(4) Stability or ability of the governor to maintain the required speed without fluctuations or "hunting".

(5) Minimal Hunting. Hunting is the continual fluctuation of the engine from the required speed, even when the load does not change.

(6) Promptness: the speed of action of the governor, usually expressed as the time in seconds required for the governor to move the fuel control mechanism.

(7) Power of the governor: the force with which the governor can overcome the resistance in the fuel control mechanism.

Note: There is a kind of speed droop which is indicated by the instantaneous speed change (δ)

$$\delta = \frac{n_i - n_r}{n_r} \times 100$$

where N_i .... maximum speed at the moment of no-load speed from rated speed

N_r .... rated speed (loaded)

this δ should be within 20% for the main marine engine and after setting of no-load speed, the percentage of increased speed should be within 10% (Japanese government safety rules). For auxiliary engine the values are 10% and 5% respectively.

61. *What is the easy way to determine the quality of diesel fuel oil ?*

(1) The analysis of diesel fuel oil is not easy to carry out; instead, we should compare the component tables provided by oil companies.

(2) Specific gravity is an essential factor in judging the quality of fuel oil - it should be as small as possible (0.88 - 0.94 at 150°C).

(3) The presence of acids or caustic substances can easily be detected with litmus paper. An indication of acid means that the fuel contains sulphur which accelerates the wear of cylinder liners. If a caustic indication is obtained, it signals ash content in the fuel.

(4) The water content can be checked by precipitation in a glass test tube.

The table below shows the limiting requirements for diesel fuel oils according to ASTM (American Society for Testing Materials).

Limiting Requirements for Diesel Fuel Oils

Grade of diesel fuel oil	Flash point, deg F (deg C)	Cloud point, deg F (deg C)	Water and sediment, volume percent	Carbon residue on. 10 percent residuum, percent	Ash, weight percent	Distillation temperatures. deg F (deg C) 90 percent point		Viscosity at 100°F (37.8°C) Kinematic, cSt (or SUS)		Sulfur, weight percent	Copper strip corrosion	Cetane number
	Min	Max	Max	Max	Max	Max	Min	Max	Min	Max	Max	Min
No. 1-D A volatile distillate fuel oil for engines in service requiring frequent speed and load changes.	100 or legal (37.8)		0.05	0.15	0.01	-	550 (287.8)	1.4	2.5 (34.4)	0.50 or legal	No. 3	40
No. 2-D A distillate fuel oil of lower volatility for engines in industrial and heavy mobile service.	125 or legal (51.7)		0.05	0.35	0.01	540 (282.2)	640 (338)	2.0 (32.6)	4.3 (40.1)	0.50 or legal	No. 3	40
No. 4-D A fuel oil for low and medium speed engines.	130 or legal (54.4)		0.50		0.10	-	-	5.8 (45)	26.4 (125)	2.0	-	30

62. *How do we choose the appropriate lubricating oil ?*

The engine builder's instruction manual usually gives the API service classification and SAE viscosity number. Refer to the tables below for service classification number and SAE viscosity number. Note that SAE Nos. 30 and 40 are recommended for tropical zones.

Service classification

Letter designation	API engine service description
CA	Light duty diesel engine service Service of diesel engines operated in mild to moderate duty with high quality fuels.
CB	Moderate duty diesel engine service Service of diesel engines operated in mild to moderate duty, but with lower quality fuels which necessitate more protection from wear and deposits.
CC	Moderate duty diesel engine service Service of lightly supercharged diesel engines operated in moderate to severe duty.
CD	Heavy duty diesel engine service Service of supercharged diesel engines in high speed, high output duty requiring highly effective control of wear and deposits.

SAE Viscosity Numbers for Crankcase Lubricating Oils

SAE viscosity number	Viscosity rating (Centistokes)			
	At $0^{o}F$ ($-18^{o}C$)		At $210^{o}F$ ($99^{o}C$)	
	Min	Max	Min	Max
5W	-	1.3	-	-
10W	1.3	2.6	-	-
20W	2.6	10.5	-	-
20	-	-	5.7	9.6
30	-	-	9.6	12.9
40	-	-	12.9	16.8
50	-	-	16.8	22.7

63. *The pressure gauge shows that the air pressure in the tank is $14 k/cm^2$ at standard atmosphere. What will the gauge pressure be if the volume of the tank is doubled (i.e. if the tank is connected to another tank of the same capacity) ?*

Answer:

$$\frac{PV}{T} = \frac{P'V'}{T'}$$

$$T = T'$$

$$PV = P'V'$$

$$(14 + 1.033) \times V = P'x \times 2V$$

$$P'x = \tfrac{1}{2}(14. + 1.033) = 7.517$$

$$\text{gauge pressure} = 7.517 - 1.033 = 6.5\ (kg/cm^2)$$

64. *Calculate the temperature (^{o}C) if the pressure is increased from normal pressure, at temperature $0^{o}C$ to $35\ kg/cm^2$. (The compression ratio is 1/13).*

Answer:

$$P = 1.033$$
$$V = 13$$
$$T = 273^{o}K$$
$$P = 35 + 1.033$$
$$\frac{PV}{T} = \frac{P'V'}{T'}$$

$$\frac{1.03 \times 13k}{273} = \frac{(35 + 1.033) \times 1k}{Tx}$$

$$T'x = 732.5^{o}\ K$$
$$T'x = 732.5 - 273$$
$$= 459.5^{o}C$$

65. *A drum of diesel fuel whose inner volume (capacity) is 200 liters contains 180 liters of fuel at $0^{o}C$ and normal atmospheric pressure.*

(a) Calculate the pressure inside the drum at $30^{o}C$.

(b) Calculate the pressure if expansion coefficient of fuel is $0.00075/^{o}C$.

Apply the Boyle's and Charles's laws:

(a)

$$\frac{PV}{T} = \frac{P'V'}{T'}$$

$$\frac{1.033 \times 20\ L}{273} = \frac{P' \times 20\ L}{273 + 30}$$

$$P' = \frac{273 + 30}{273} = 1.109$$

$$\text{Gauge pressure} = \text{absolute } P - 1 \text{ atm} = 1.109 - 1.033$$
$$= 0.076 \doteqdot 01\ kg/cm^{2}$$

(b)

	$0^{\circ}C$ Standard condition	at $30^{\circ}C$
Volume	200 L - 180 L	20 L - x L
Pressure	P = 1 atm	P'x
Absolute temp	$273 + 0^{\circ}$ ($^{\circ}$K)	273 + 30 ($^{\circ}$K)

x L ...... expanded volume

$$x^{L} = 180^{1} \times 0.00075/^{\circ}C \times 30^{\circ}C = 4.05^{1}$$

$$\frac{PV}{T} = \frac{P'V'}{T'}$$

$$\frac{1 \times 20}{273} = \frac{(20 - 4.05)\ P'x}{273 + 30}$$

$$P'x = 1.39\ kg/cm^{2}$$

gauge pressure = absolute P - 1.033 kg/cm^2 = <u>0.35 kg/cm^2</u>.

66. *At $29^{\circ}C$ the pressure in a tank is indicated as 26 kg/cm^2. If the temperature is reduced to $18^{\circ}C$, what will be the pressure in the tank?*

<u>Answer</u>:
$$\frac{27.033 \times V}{273 + 29} = \frac{P' \times V'}{273 + 18} \qquad V = V'$$

$$P = \underline{27.033 \times 291} = 26.05\ kg/cm^{2}\ abs$$

gauge pressure = 26.05 - 1.03

= <u>25 (kg/cm^2)</u>

67. *It takes 40 minutes to fully charge the air tank whose volume is 10 m^3 at 30 kg/cm^2 pressure, by operating compressor driven by a diesel engine. Calculate the capacity of this compressor (m^3/h) under the following conditions:*

(i) *Normal atmospheric pressure (1.033 kg/cm^2);*

(ii) *Efficiency of compressor is 70%;*

(iii) *Temperature of air sucked in by compressor is 15^oC;*

(iv) *Temperature of inlet port of air tank is 21^oC.*

Answer:

P1 = 1.033

T1 = 15 + 273 = 288^oK

V1 = air volume (accumulated in 40 minutes) m^3

P2 = tank pressure = 30 + 1.033 = 31033 kg/cm^2

T2 = temperature of air tank = 21 + 273 = 294^oK

V2 = volume of air tank = $10m^3$

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$V_1 = \frac{P_2 \times V_2T_2}{P_1T_2} = \frac{31.033 \times 10 \times 288}{1.033 \times 294} = 294.29/m^3 \text{ (40 min)}$$

$$294.29 \times \frac{60}{40} = 441.43 \ (m^3/h)$$

$$\text{capacity of air compressor } \frac{441.43}{0.7} = \underline{\underline{630.6 \ m^3/h}}$$

68. *Calculate the pressure and temperature under the conditions (a), (b) and (c) given below, if the initial conditions are as follows:*

(i) *Initial volume of air (V_1) .... 30 liters*

(ii) *Initial temperature (T_1) .... 10^oC*

(iii) Compression ratio (ε) 20

(a) Isothermal change

(b) Adiabatic change

(c) Polytropic change

(a) from $\varepsilon = \frac{V_1}{V_2}$ $\quad V_1 = 30\ell = 0.03\ m^3$

$$V_2 = \frac{V_1}{V_2} = \frac{\frac{30\ell}{1000}}{20} = 0.0015\ m^3$$

Isothermal change $P_1V_1 = P_2V_2$

Pressure: $P_2 = \frac{1.0332 \times 0.03}{0.0015} = 20.66\ (kg/cm^2)$

Temperature: There is no temperature change because of isothermal change

$$T_2 = T = 10^{\circ}C$$

(b) Adiabatic change

$P_2 = P_1(\frac{V_1}{V_2})^k$ $\quad$ where k of air is 1.4

$k = \frac{Cp}{Cv}$

$= 1.0332 \times 20^{1.4^{*}} = 1.0332 \times 66.28908$

(specific heat at constant pressure)

Pressure: $P_2 = 68.48\ (kg/cm^2)$

* Use an fx - model scientific calculator.

Temperature after adiabatic compression .... T_2

$$T_2 = T_1(\frac{V_1}{V_2})^{k-1} = (10 + 273) \times 20^{1.4-1} = 938.0^{\circ}K$$

$$t_2 = T_2 - 273 = 665^{\circ}C$$

(c) Polytropic change

Pressure after compression .... P_2

$$P_2 = P_1(\frac{V_1}{V_2})^n \qquad n = 1.3$$

$$P_2 = 1.0332 \times 20^{1.3} = 50.76\ (kg/cm^2)$$ (n = value of the exponent of polytropic change)

Temperature after compression T_2

$$T_2 = T_1(\frac{V_1}{V_2})^{n-1} = (10 + 273) \times 20^{1.3-1} = 695.2$$

$$T_2 = 695.2 - 273 = 422.2^{\circ}C$$

The summary of the above result can be tabulated as follows:

	Isothermal change	Adiabatic change	Polytropic change
Compressed pressure	20.66 kg/cm^2	68.48 kg/cm^2	50.76 kg/cm^2
Temperature after compression	$10^{\circ}C$	$665^{\circ}C$	$422.2^{\circ}C$

Polytropic change is similar to the actual cycle condition of an ordinary diesel engine.

69. *How does a change in the diameter and the pitch of a propeller affect the engine revolution ?*

$$DHP \;\alpha\; n^3 (D + P)^{5*} \qquad \dots\dots\dots \; (1)$$

where DHP .... delivered horsepower

n .... propeller revolution (rpm)

D .... diameter of propeller

P .... Pitch of propeller

D', P' .... changed diameter and pitch

* Note $\quad$ thrust $\alpha \frac{1}{2} PV^2 S \rightarrow (nD)^2 \times D^2 \rightarrow n^2 D^4$

$DHP = \text{thrust} \times \Upsilon = n^2 D^4 \times nD = n^3 D^5$ approximately, if engine horsepower is the same

from (1) $\quad n^3 (D + P)^5 = n^{3'} (D' + P')^5$

$$n^{3'} = n^3 \left(\frac{D + P}{D' + P'} \right)^5$$

$$n' = n \times \sqrt[3]{\left(\frac{D + P}{D' + P'} \right)^5}$$

Example:

A small FRP boat is equipped with a propeller whose diameter is 660 mm and pitch is 670, and an engine with the maximum revolution of 2200 rpm. The propeller is damaged and has to be replaced by a new one whose diameter is 640 mm and the pitch 570. What is the expected revolution of the engine with the new propeller if the horsepower remains the same ?

Answer: $\quad n' = 2220 \times \sqrt[3]{\left(\frac{660 - 670}{640 - 570} \right)^5} = 2200 \times \left(\frac{1330}{1210} \right)^{\frac{5}{3}} = 2200 \times 1170$

$= 2575$ rpm

(the actual observed rpm was 2550 rpm).