CATERPILLAR®

Marine Engines Application and Installation Guide

LEKM7142	Engine Performance & Boat Performance
LEKM7143	Driveline/Mounting/Alignment/Auxiliary
LEKM7144	Control Systems/Instrumentation and Monitoring Systems/Starting Systems/ Start-up Systems/Serviceability
LEKM7145	Cooling Systems
LEKM7146	Ventilation & Exhaust System
LEKM7147	Lubrication Systems & Fuel Systems
LEKM7148	Dredge Engines

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Marine Engines Application and Installation Guide

- Engine Performance
- Boat Performance

LEKM7142-01 (Supersedes LEKM7142) (05-00)

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Engine Performance

Application Guidelines
Rates
Performance Curve Format
Engine Configuration Effects on Ratings
Auxiliary Engine Ratings

Application Guidelines

Knowledge of the engine's operating requirements is essential to establish a proper match of engine rating to boat operating requirements. To help determine the acceptability of a rating for a particular boat's application, the following parameters should be considered:

1. Time at full throttle
2. Annual operating hours
3. Propeller match

Time at Full Throttle

Time at full throttle is the amount of time the engine is operated at rated rpm without load cycling during a normal duty cycle. This is normally specified in terms of percent of total cycle time or in minutes per hour.

Annual Operating Hours

The annual operation hours are based on the accumulated service meter units* during a 12-month period.

Propeller Match

The propeller must be sized to allow the engine to operate slightly above rated rpm under the boat's most severe load conditions: full fuel and water tanks, stores aboard for extended voyaging, and adverse sea conditions.

Ratings

Ratings are statements of the engines' power and speed capability under specified load conditions. The Caterpillar rating system simply matches engines to particular applications. It consists of the following standard ratings.

Continuous A Rating

For heavy-duty service in ocean-going displacement hulls such as freighters, tugboats, bottom-drag trawlers, and deep river towboats when the engine is operated at rated load and speed up to 100% of the time without interruption or load cycling. Expected usage should be from 5000 to 8000 hours per year.

Medium Duty B Rating

For use in midwater and shrimp trawlers, purse seiners, crew and supply boats, ferry boats with trips longer than one hour, and towboats in rivers where locks, sandbars, curves, or traffic dictate frequent slowing and engine load is constant with some cycling. Full power operation to be limited to 80% of operation time. Expected usage should be from 3000 to 5000 hours per year.

Intermittent C Rating

For use in yachts with displacement hulls as well as ferries with trips of less than one hour, fishing boats moving at higher speed out and back (e.g. lobster, crayfish, and tuna), and short trip coastal freighters where engine load and speed are cyclical. Full power operation to be limited to 50% of operation time. Expected usage should be from 2000 to 4000 hours per year.

Patrol Craft D Rating

Continuous power for use in patrol, customs, police, and some fire boats. Full power limited to 16% of operation. Expected usage should be from 1000 to 3000 hours per year.

High Performance E Rating

For use in pleasure craft with planing hulls as well as for pilot, harbor patrol, and harbormaster boats. Full power operation to be limited to 8% of operation time. Expected usage should be from 200 to 1000 hours per year.

Rating Conditions

Ratings are based on SAE J1128/ISO 8665 standard ambient conditions of 100 kPa (29.61 in. of Hg) and 25°C (77°F). Ratings also apply at AS1501, BS5514, DIN6271 and ISO 3046/1 standard conditions of 100 kPa (29.61 in. of Hg), 27°C (81°F) and 60% relative humidity.

*Clock hours are the same as Service Meter Units on all Caterpillar Engines using electric service meters. Some Caterpillar Engines (D399, D398, D379 and earlier engines) used service meters which "counted" engine revolutions. One service meter unit on those engines, corresponds to a clock hour only when the engine is operating at rated speed (rpm). The ratio between clock hours and service meter units is proportional to engine speed.

Power is based on a 35° API [16°C (60°F)] fuel having a LHV of 42,780 kJ/kg (18,390 B/lb) used at 29°C (85°F) with a density of 838.9 g/L (7.001 lb/U.S. gal).

Ratings are gross output ratings: i.e., total output capability of the engine equipped with standard accessories: lube oil, fuel oil and jacket water pumps. Power to drive auxiliaries must be deducted from the gross output to arrive at the net power available for the external (flywheel) load. Typical auxiliaries include cooling fans, air compressors, charging alternators, marine gears, and sea water pumps.

Marine Engine Ratings to DIN Standards

The DIN (Deutsche Industrie Norme) 6270 Standard covers rated output data for internal combustion engines in general applications. When required, DIN 6270 main propulsion ratings can be quoted according to the following stipulations.

Continuous Output A

This is the published Caterpillar "Continuous 'A' Rating" rating in kW units. No additional reference is necessary*.

Output B

Output B is defined as the maximum useful output that the engine can deliver for a definite time limit corresponding to the engine application. The fuel setting is pre-set such that output B cannot be exceeded, so no overload capability need be demonstrated.

On the basis of this definition, we can offer two output B ratings with kW values corresponding to Caterpillar's Medium Duty B Rating or Caterpillar's Intermittent C Rating.

In each case, it is mandatory that reference be made to the applicable rating definitions.

General Comments

DIN 6270 conditions are slightly different from the SAE conditions used in the U.S. We believe that they are virtually equivalent for all practical purposes. No correction to ratings should be made to account for the slightly different reference conditions.

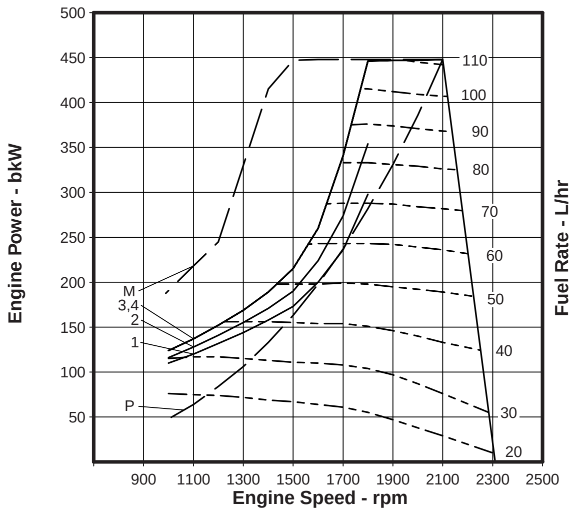
Useful output as described under DIN 6270 is defined as the output available to drive the load after suitable deductions are made for engine driven accessories. This is equivalent to the net rating. Caterpillar ratings indicate gross output. At the kW requirement to drive such accessories as charging alternator and sea water pump are low and well within our rating tolerance, no deductions for main propulsion engine driven accessory loads need to be made.

* A condition in the "Continuous Output A" definition is that the output limiting device must be set to provide a margin of extra capacity. This overload capability can be demonstrated, if required, by increasing the fuel setting from the factory-set continuous output value to the value corresponding to our "B" rating level. With a few exceptions, this increased fuel setting will correspond to an overload capability of approximately 10%. The propeller should be sized for the continuous rating with the appropriate safety margins the Technical Marketing Information File (TMI). The fuel setting must be readjusted to the name-plate value upon completion of the demonstration test.

Performance Curve Format

Caterpillar Performance Curves follow the following format:

Engine Performance - MAR - C Rating	3406 DITA	DM6120-00



ZONE LIMIT DATA

	Speed rpm	Engine Power bkW	Fuel Cons g / kW-hr	Fuel Rate L/hr	Boost Press kPa Gauge	Air Flow cu m/ min	Exh Temp C	Exh Flow cu m/ min
Curve 1	1800	298	203	71.9	934.8	22.9	390	53.6
	1600	200	211	50.5	453.9	15.4	403	36.8
	1400	158	215	40.4	264.2	11.8	404	28.3
	1200	132	215	33.9	179.6	9.4	402	22.3
	1000	110	218	28.6	129.9	7.5	405	17.9
Curve 2	1800	354	200	84.3	122.3	26.5	396	62.3
	1600	224	209	55.7	54.0	16.3	420	40.0
	1400	171	214	43.6	30.5	12.2	422	30.0
	1200	141	215	36.1	20.4	9.6	421	23.4
	1000	116	218	30.2	14.7	7.6	424	18.7
Curve 3	2100	448	208	111.1	201.3	40.9	344	88.3
	1900	447	206	109.7	189.7	35.7	376	82.2
	1700	341	203	82.5	111.0	23.9	427	58.6
	1500	215	210	53.7	47.5	14.8	445	37.5
	1300	169	214	43.0	29.0	11.1	443	28.4
	1100	137	216	35.4	20.0	8.7	444	22.2
Curve 4	2100	448	208	111.1	201.3	40.9	344	88.3
	1900	447	206	109.7	189.7	35.7	376	82.2
	1700	341	203	82.5	111.0	23.9	427	58.6
	1500	215	210	53.7	47.5	14.8	445	37.5
	1300	169	214	43.0	29.0	11.1	443	28.4
	1100	137	216	35.4	20.0	8.7	444	22.2

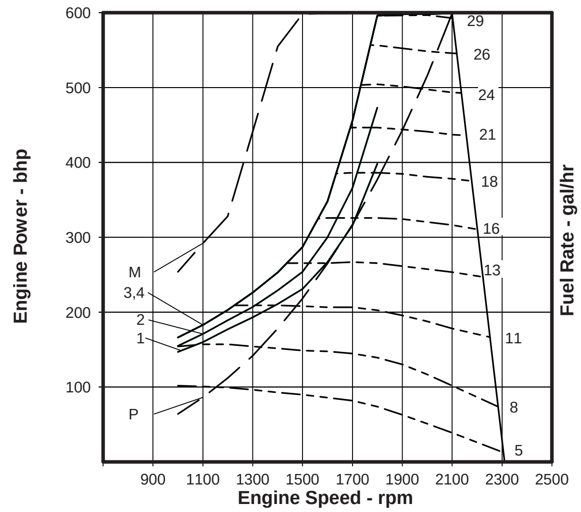
MAXIMUM POWER DATA

Speed rpm	Engine Power bkW	Fuel Cons g / kW-hr	Fuel Rate L/hr	Boost Press kPa Gauge	Air Flow cu m/ min	Exh Temp C	Exh Flow cu m/ min
2100	448	208	111.1	201.3	40.9	344	88.3
1900	448	206	109.8	189.7	35.7	376	82.3
1700	448	203	108.3	182.2	31.8	424	77.6
1500	447	204	108.7	162.7	26.3	513	73.4
1300	330	214	84.3	105.2	18.0	641	56.9
1100	218	231	60.1	55.1	11.4	627	36.8

PROPELLER DEMAND DATA

Speed rpm	Engine Power bkW	Fuel Cons g / kW-hr	Fuel Rate L/hr	Boost Press kPa Gauge	Air Flow cu m/ min	Exh Temp C	Exh Flow cu m/ min
2100	448	208	111.1	201.3	40.9	344	88.3
1900	331	203	80.1	115.7	26.9	370	60.9
1700	237	208	58.7	63.3	18.3	399	43.4
1500	163	215	41.9	29.8	13.0	397	30.5
1300	106	221	27.9	12.0	9.7	332	20.5
1100	64	228	17.5	3.3	7.5	258	13.8

Brake Mean Effective Pressure .... 239 kPa
Heat Rejection to Coolant (total) .... 1746 kW



ZONE LIMIT DATA

	Speed rpm	Engine Power bhp	Fuel Cons lb/ hp-hr	Fuel Rate gal/ hr	Boost Press in. Hg- Gauge	Air Flow cfm	Exh Temp F	Exh Flow cfm
Curve 1	1800	398	.334	19.0	276.7	809	735	1894
	1600	267	.347	13.4	134.4	544	759	1301
	1400	211	.354	10.7	78.2	417	761	1000
	1200	177	.354	9.0	53.2	332	757	788
	1000	147	.359	7.6	38.4	265	763	633
Curve 2	1800	473	.329	22.3	36.2	937	746	2202
	1600	300	.344	14.7	16.0	576	791	1414
	1400	229	.352	11.5	9.0	431	795	1060
	1200	189	.354	9.5	6.0	339	793	827
	1000	155	.359	8.0	4.4	269	799	661
Curve 3	2100	599	.342	29.4	59.6	1446	648	3121
	1900	598	.339	29.0	56.2	1262	708	2905
	1700	456	.334	21.8	32.9	845	804	2071
	1500	287	.346	14.2	14.1	523	838	1325
	1300	226	.352	11.4	8.6	392	835	1004
	1100	183	.355	9.4	5.9	307	836	785
Curve 4	2100	599	.342	29.4	59.6	1446	648	3121
	1900	598	.339	29.0	56.2	1262	708	2905
	1700	456	.334	21.8	32.9	845	804	2071
	1500	287	.346	14.2	14.1	523	838	1325
	1300	226	.352	11.4	8.6	392	835	1004
	1100	183	.355	9.4	5.9	307	836	785

MAXIMUM POWER DATA

Speed rpm	Engine Power bhp	Fuel Cons lb/ hp-hr	Fuel Rate gal/ hr	Boost Press in. Hg- Gauge	Air Flow cfm	Exh Temp F	Exh Flow cfm
2100	599	.342	29.4	59.6	1446	648	3121
1900	599	.339	29.0	56.2	1262	708	2909
1700	599	.334	28.6	53.9	1124	799	2743
1500	598	.336	28.7	48.2	930	966	2594
1300	441	.352	22.3	31.1	636	1208	2011
1100	292	.380	15.9	16.3	403	1181	1301

PROPELLER DEMAND DATA

Speed rpm	Engine Power bhp	Fuel Cons lb/ hp-hr	Fuel Rate gal/ hr	Boost Press in. Hg- Gauge	Air Flow cfm	Exh Temp F	Exh Flow cfm
2100	599	.342	29.4	59.6	1446	648	3121
1900	443	.334	21.2	34.2	951	697	2152
1700	317	.342	15.5	18.7	647	752	1534
1500	218	.354	11.1	8.8	459	748	1078
1300	142	.364	7.4	3.6	343	625	725
1100	86	.375	4.6	1.0	265	486	488

Brake Mean Effective Pressure .... 35 kPa
Heat Rejection to Coolant (total) .... 99293 kW

Features of the Performance Curve:

Vertical Axis [left side] . . . Graduated in units of Power [Brake kW or Brake Horsepower]

Horizontal Axis . . . Graduated in units of Engine Speed [Revolutions per Minute]

Curve P . . . Propeller Demand Curve, describes the power demanded by a fixed pitch propeller used in a displacement hull. Semi-displacement and planing hulls will have higher load demand than shown in the "P" curve. Each semi-displacement and planing hull has different demand, which makes it impossible to show the load demand for each hull. Semi-displacement and planing hulls will need to be sea trialed with fuel measurements taken at different engine speeds to determine actual fuel and load demand.

Curve 1 . . . Continuous Limit Line, describes the upper limit of continuous operation, without interruption or load cycling.

Zone 1-2 . . . Zone 1-2 is located between Curve 1 and Curve 2. It is the zone within which operation is permitted for periods up to 4 hours, followed by a one hour period at combination of power and speed on or under Line 1.

Zone 2-3 . . . Zone 2-3 is located between Curve 2 and Curve 3. It is the zone within which operation is permitted for periods up to 1 hour, followed by a one hour period at combinations of power and speed on or under Line 1.

Zone 3-4 . . . Zone 3-4 is located between Curve 3 and Curve 4. It is the zone within which operation is permitted for periods up to five (5) minutes, followed by a one hour period at combinations of power and speed on or under Line 1.

Curve 4…Maximum Limit Curve, the maximum power available within the rating development limits (cylinder pressure, turbo speed, exhaust temperature).

Curve M…Maximum Power Data, the maximum power capability of the engine without regard to the rating development limits.

Fuel Rate Lines . . . Parallel, slightly curving, dotted lines, with graduations on their right ends, are lines of constant fuel rate. [L/hr or gal/hr]

The most efficient engine rpm to generate any given amount of power will be found directly under the high point of the fuel rate line nearest the required power. This will be most useful in those applications which can vary the engine speed at which power is extracted, such as controllable pitch propellers.

The graphical representation of the engine performance is accompanied by a full set of tabular information. Included is intake manifold pressure, exhaust stack temperature, combustion air flow, exhaust gas flow, fuel rate, engine power and engine speed, and fuel efficiency for all the curves shown.

Each standard rating of the engines will have its performance documented as shown above. There can be a delay of the formal version of the data in the case of new ratings or engine configurations.

Engine Configuration Effects on Ratings

Engine configurations can be altered to allow efficient use of larger amounts of fuel. This is done by increasing the amount of air which can be utilized in an engine. Air flow through an engine is called aspiration. Caterpillar engines have one of the following methods of aspiration:

Naturally Aspirated

In a naturally aspirated engine, the volume of air drawn into each cylinder is moderate, since only atmospheric pressure is forcing air through the cylinder's intake valve. There is no pressurization of the engine's intake manifold by an external device and engine intake manifold pressure is always a partial vacuum.

Turbocharged

Greater amounts of air can be forced into an engine's cylinders by installing a turbocharger. Turbochargers are turbine-like devices which use exhaust energy (which naturally aspirated engines waste) to compress outside air and force it into the intake manifold. The increased amount of air flowing through turbocharged engines does two good things:

1. The greater flow of air cools the valves, piston crowns and cylinder walls, making them better able to resist the firing forces.

2. Fuel can be burned more efficiently, due to the increased amount of air for combustion.

This makes the engine more powerful. Compression does increase the temperature of the intake air, however. It is very useful to remove the heat-of-compression from the intake air, upstream of the combustion chambers. Cooling the air before it enters the combustion chambers makes the air more dense and increases cooling of the combustion chamber components.

Turbocharged/Aftercooled

An air-cooling heat exchanger (aftercooler) is installed between the turbocharger and the combustion chamber on Turbocharged/ Aftercooled engines. The aftercooler cools the incoming air, carrying the heat away with a flow of water. The water can come from two sources. If jacket water (the same water that cools the cylinder head and block) is used in the aftercooler, then the air can only be cooled to approximately 93°C (200°F). Jacket water temperature is thermostatically controlled at approximately 82°C (180°F). Even cooler air can be obtained by cooling the aftercooler with water from a separate circuit, such as sea water or some other circuit, with colder water than the engine jacket water. Lower aftercooler water temperatures permit higher engine ratings because cooler, denser air permits burning more fuel.

Extended Periods of Low Load

Prolonged low load operation should be followed by periodic operation at higher load to consume exhaust deposits. Low load operation is defined as below approximately 20% load. The engine should be operated above 40% load periodically to consume the exhaust deposits. Caterpillar engines can be run well over 24 hours before exhaust slobber becomes significant. The amount of additional time depends upon the engine configuration, water temperature to the aftercooler, inlet air temperature to the engine and type of fuel.

Auxiliary Engine Ratings

Marine engines used for auxiliary power are of the same general configuration as propulsion engines. Their power output is limited by the same design factors. Horsepower ratings are also determined by the type of aspiration, the aftercooling system and by engine application.

Caterpillar prime power ratings are used for marine generator sets when applied as ship-board power and as emergency power at both 60 Hz and 50 Hz. The engine is set at the factory to provide 110% of rated output as required by Marine Classification Societies (MCS).

Normally, other auxiliary power requirements, such as hydraulic pumps, winches, fire and cargo pumps, and compressors, are applied at a rating based on their duty cycle and load factor.

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Boat Performance

Tolerances on Hull, Propeller and Engine

Propeller Sizing

Ducted Propellers (Kort Nozzles)

Hull Types

Rules of Thumb

The performance of the boat is the result of a complex interaction of all three aspects of the installation; the engine, the hull, and the propeller.

Tolerances on Hull, Propeller and Engine

Proper component sizing is very important to the life and performance of the entire propulsion system. There are tolerances in several aspects of the propulsion system. In worst-case conditions, the result can be short life and/or unsatisfactory performance. For example: the effect of these tolerances is shown below in Figure 2.1:

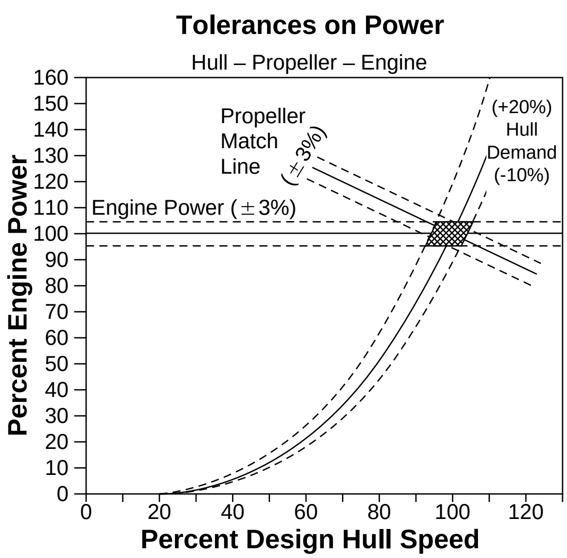


Figure 2.1

The engine power may be expected to vary due to manufacturing tolerance by as much as 3% on either side of its rated or 100% power.

The propeller power absorption may be as much as 5% higher, or lower, than originally expected. This could result from manufacturing tolerance in pitch, surface finish, and blade profile.

The hull resistance may vary as much as 20% from calculated values or previous experience due to inevitable differences in weight and shape.

Propeller Sizing

The propeller is as important as the hull or the engine to the performance of the boat.

The propeller directly influences: top speed, fuel efficiency, and engine life.

General Information

While many operators will choose to operate at reduced throttle settings while cruising, the engine must be able to reach its rated speed (rpm) when the boat is ready for sea; fully loaded with fuel, water, and stores. For the ultimate in engine life and economy, expected engine operating speeds during sea trials should be approximately 1-3% over full load rated engine speed (rpm).

Table of Engine rpm at Sea Trials

Rated Speed (rpm)	Expected Engine Speed During Sea Trials (rpm)
2800	2830-2890
2800	2830-2890
2400	2425-2470
2400	2425-2470
2300	2320-2360
2100	2120-2160
1925	1945-1980
1800	1820-1850
1800	1820-1850

Eliminating Engine Overloading on Overwheeled Vessels

When the engine speed (rpm) measured during the sea trial of a vessel fails to attain the required sea trial speed, the reason generally is one of the following:

Excessive hull fouling — Solvable by cleaning the hull and re-running the sea trial.

Low engine power — Resolved by measuring and recording engine performance parameters such as inlet, exhaust and fuel rate.

Incorrect transmission or propeller — A detailed discussion of the resolution of this condition follows: Simply, this discussion will be restricted to fixed pitch propellers.

Engine fuel setting adjustment — Many vessel operators and shipyards want to increase the engine fuel (rack) setting when their engine does not reach rated speed during sea trials. At first glance, this seems to be the easiest and least costly remedy.

However, in such a situation, this solution is incorrect, even if the engine speed (rpm) does increase to the expected rated rpm. Increasing the fuel (rack) setting will result in reduced engine life, increased wear, or in worst case, early engine failure. The vessel operators engine repair and maintenance costs will likely far exceed the cost of replacing or modifying the existing transmission or propeller.

High idle adjustment — Another often considered alternative is increasing *high idle* engine speed to the specified free running speed. This will not provide the desired results since the fuel stop is already at the maximum fuel position, and an increase in high idle will not result in any appreciable speed change.

Properly sized propeller and/or reduction ratio — The correct, but more costly, remedy is to install a properly matched propeller and/or transmission ratio to allow the engine to operate within it's rating guidelines.

Avoiding driveline component changes — There is another alternative which we will consider in cases where driveline component changes cannot or will not be considered. This method consists of a reduction of both the engine fuel setting and the high idle speed. Of course, the engine power and rated speed are reduced in the process; however, we are taking advantage of the fact that the propellers power demand drops off much faster than the engine power capability when engine and propeller speed is reduced (refer to Figure 2.2).

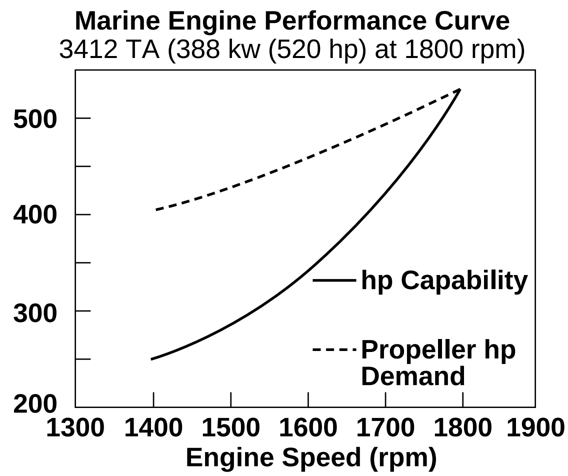


Figure 2.2

The net result is that the engine will perform within its application limits and the engine/propeller match have been optimized. The following formula generally applies for a standard fixed pitch propeller:

$$\frac{\mathbf{hp}_1}{\mathbf{hp}_2} = \left[\frac{\mathbf{N}_1}{\mathbf{N}_2}\right]^3$$

or by rewriting the equation

$$\mathbf{hp}_2 = \mathbf{hp}_1 \times \left[\frac{\mathbf{N}_2}{\mathbf{N}_1}\right]^3$$

Where:

$\mathbf{hp}_1$ = Engine power produced at the full throttle speed recorded during the sea trial. This power level is determined by referring to the appropriate marine engine performance curve corresponding to the original engine rating sold by the dealer and reading the power on the curve at the recorded speed.

$\mathbf{hp}_2$ = Calculated propeller power demand at the new reduced engine speed (rpm) proposed for this application.

$\mathbf{N}_1$ = Engine speed (rpm) observed and recorded during the original sea trial — prior to fuel setting and high idle modifications. (This speed should always be measured with a precision tachometer.)

$\mathbf{N}_2$ = New, reduced engine speed (rpm) which must be determined in order to provide an acceptable engine, transmission, and propeller match.

For example: Consider a 3408B DITA engine, sold at a continuous rating of 365 hp at 1800 rpm; During the sea trial, the maximum attainable engine speed was only 1620 rpm. This engine was operating in an unacceptable overload (or *lug*) condition. The Marine Engine Performance Curve (for a continuous rating of 272 kw (365 hp) at 1800 rpm) indicates that the engine was producing (and the propeller was demanding) 344 hp at the limited speed of 1620 rpm. This power requirement exceeds the approved continuous rating of 330 hp at 1620 rpm. The solution is to further reduce the rpm until the approved engine rating, as shown on the 3408B marine engine rating curve, exceeds the propeller demand.

For this example we will calculate the power required if the rated engine rpm was reduced to 1550.

$$\mathbf{hp}_2 = 334 \times \left[\frac{1550}{1620}\right]^3 = 301 \text{ hp}$$

Reducing the engine speed by 70 rpm has resulted in a decrease in propeller demand of 43 hp. The approved engine continuous rating at 1550 rpm is 314 hp and the propeller demand has been reduced to 301 hp.

At the initial trials, the recorded vessel speed was 10.2 knots for this 21 m long seiner. Resetting the engine from 344 hp @ 1620 rpm to 314 hp @ 1550 rpm would decrease the vessel speed to 9.7 knots, a relatively insignificant difference, especially considering the gain in engine life.

Propeller Pitch Correction

An overpitched propeller must have its pitch reduced to allow the engine to reach rated rpm. The pitch must be reduced by an amount proportional to the engine rpm ratio. The following formula defines this relationship:

$$\mathbf{P}_{\text{required}} = \mathbf{P}_{\text{present}} \times \frac{\textbf{Engine rpm}_{\text{ while over loaded}}}{\textbf{Rated Engine rpm}}$$

Where:

P required = pitch the propeller must have to allow the engine to run at rated rpm

P present = pitch of the propeller which is preventing the engine from reaching its rated rpm

Engine rpm while overloaded = engine rpm under normal working conditions when equipped with the propeller whose pitch is too great

Rated Engine rpm = appropriate engine rpm found on applicable engine specification sheet

Propeller Errors and Propeller Measurement

Fast boats need more precise propellers than slow speed workboats.

Propeller pitch errors which would be insignificant on a 10 knot river towboat, will cost a high speed patrolboat or yacht 2 or 3 knots of its top speed.

Propellers on fast boats must be precisely manufactured if design performance is to be attained and they must remain within nearly new specifications to prevent severe performance deterioration. This is particularly true of propellers' leading and trailing edges. Tiny errors in profile, almost too small to be detected by feel, can constitute sites for initiation of cavitation. In severe cases, this can result in blade failure or loss after as little as 24 hours of high speed running.

Most industry professionals can relate instances where new propellers have been found to be several inches out of the specified pitch. When propellers are repaired or repitched, it is even more difficult to restore the necessary precision for highest performance vessels.

The problem usually is the tooling. Most propeller pitch measurement machines can not resolve or detect the small errors which prevent a boat from attaining first-class performance. All other things being equal, the skill of the propeller finishing machinist will make the difference between barely-adequate and first-class boat performance.

Propeller Measurement Tools

There are several basic types of tools commonly used for propeller pitch measurement:

Swing Arm Type

This machine generally consists of: a stand which supports the propeller in a horizontal position, a vertical column which passes through the center of the propeller's hub, a *swing arm* which rotates around the vertical column, and a vertical measuring rod which can slide in and out on the swing arm.

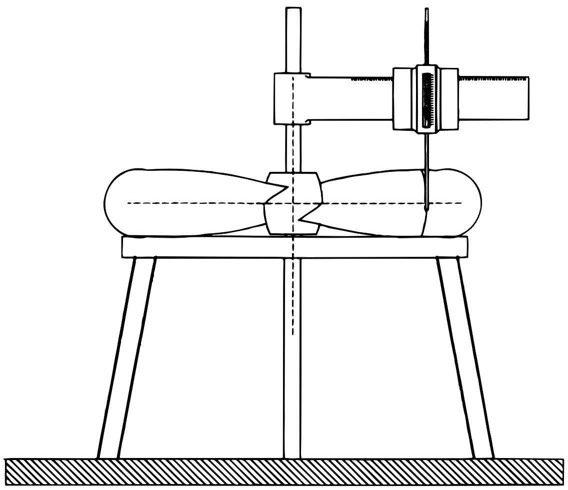


Figure 2.3

This machine reaches down from a horizontally mounted swing arm and "touches" the blade at several radial locations, at some standard increments of angle. The difference in elevation, the radial position, and the angular increment between readings allow pitch to be calculated between any two locations. The accuracy of this device is related to the rigidity of the swing arm and the degree of looseness in the required bearings. The potential accuracy of the propellers measured will be directly proportional to the number of measurements on each blade (places at which it touches each blade). For commercial (workboat) propellers, it is common to examine the blade at six (6) to nine (9) places per blade. On high-performance civilian propellers, it is common to examine each blade at twenty-five (25) to fifty (50) places while military propellers may be examined at several hundred places per blade. The skill of the machinist is applied in smoothing or "fairing" the areas between the measurements.

Pitch Blocks

Pitch blocks are precisely shaped anvils, against which individual propeller blades are hammered to repair or correct their shape.

They can be used to measure propellers by comparing the shape of an unknown propeller to a set of incremental pitch blocks until a match is found.

Angle-Measuring Type

Angle-Measuring Devices relate the angle of a circumferential line on the blade to a horizontal reference plane and calculate the pitch from the angle and the radial position.

Caterpillar markets an angle-measuring pitch measuring tool—Part Number 8T5322.

Ducted Propellers (Kort Nozzles)

The Propeller Duct, sometimes called a Kort nozzle is a ring, wrapped around a generally square-tipped, propeller. The ring has an airfoil-shaped cross section.

The ducted propeller is best used on vessels such as trawlers, tugs, and towboats with towing speeds of 3-10 knots. Ducted propellers should not be used on relatively fast vessels.

To aid in selection, perform the following calculation. If the result is less than 30, the use of the ducted propeller should not be considered as it may result in a net loss of vessel performance.

$$\mathbf{B}_p = (\mathbf{srpm})\,\frac{(\sqrt{\mathbf{shp}})}{(\mathbf{V}_a)^{2.5}}$$

Where:

Bp = Basic Propeller Design Variable
srpm = Propeller Shaft Speed (rpm)
shp = Shaft Horsepower (shp)
Va = Velocity of Advance of the Propeller (knots) generally equals 0.7 to 0.9 times boat speed

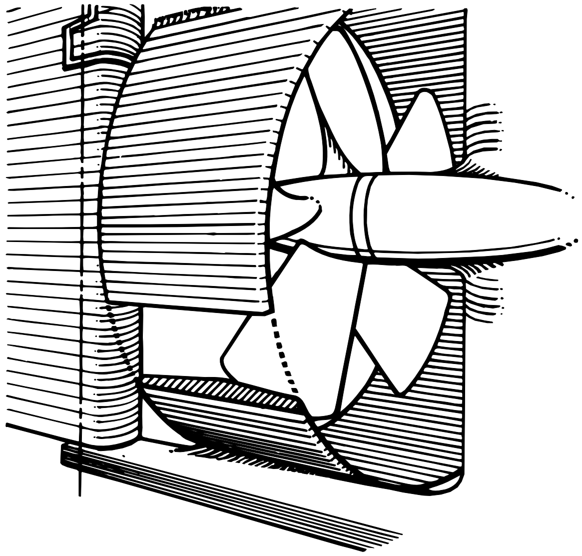


Figure 2.4

The nozzle configuration or profile most often used is a No. 19A nozzle although a No. 37 specifically designed for backing is obtainable. Nozzles are made of mild steel with a stainless steel liner to stand up to erosion. They may be mounted to steel, wood or fiberglass hulls.

A comparison of bollard pull ahead and astern for the open water propeller versus the No. 19A (taken as 100% in ahead) and the No. 37 nozzle follows.

	Ahead	Astern
Nozzle No. 19A	100%	59%
Nozzle No. 37	99%	82%
Open Propeller (B4.70 Type)	69%	55%

These are actual figures for a 1491 kW (2000 hp) installation with 2007 mm (79 inch) diameter propellers. A larger diameter open propeller would show up somewhat better, though not as good as the nozzles.

More specific information on ducted propeller systems generally can be obtained from propeller manufacturers, many of which also manufacture propeller ducts.

Hull Types

All hull types discussed here refer only to the portion of the hull below the waterline. What is above the waterline concerns seaworthiness, seakindliness, stability, comfort, and eye appeal, but has little impact on the propulsion machinery.

There are two basic types of hulls: Displacement Hulls and Planing Hulls. There are also some special types of hulls. These include the Semi-Displacement Hull, Catamaran, Wave-Piercing Catamaran, Hydrofoil, Surface Effects Ship (with both flexible skirts and rigid sidewalls), and the Small-Waterplane-Area-Twin-Hull (SWATH) Ship.

Displacement Hull

A displacement hull can be described in most basic terms as a block, with tapered ends.

Displacement Hull's Most Basic Form

To illustrate the basic shapes this allows, five blocks in what are rearranged to form four simple, but fundamental forms which cover most all displacement hull forms.

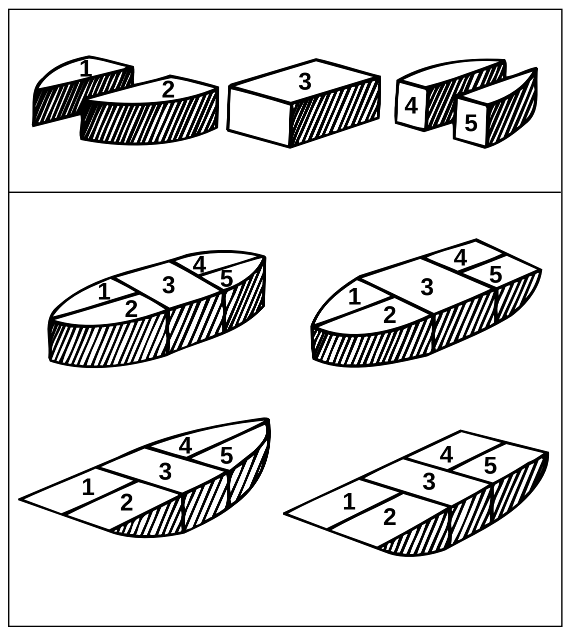


Figure 2.5

Keep in mind that this discussion concerns only the portion of the hull below the waterline and that the blocks represent only the submerged part of the hull.

When any one of the hulls shown above moves through the water, waves are formed. The bow pushes the water aside, forming a bow wave. The momentum imparted to the water carries it beyond the boundaries of the hull, leaving a hollow behind it. The wave surges back, into the hollow. At slow speeds, this causes the return surge to bounce off the hull, starting the familiar diverging pattern of troughs and crests originating with the bow wave.

Relation of Hull Length to Boat Speed

The length of a displacement hull determines its eventual top speed. It is literally possible to measure the length of a displacement hull and calculate its highest practical top speed based on this measurement.

This is due to the relationship of boat speed, boat length and wave length.

Boat Length and Wave Length

Wave length and wave speed are directly proportional: the faster a wave, the longer its length.

Since the movement of the hull causes the bow wave, the faster the hull moves, the faster the speed of the bow wave . . . and the longer its length.

As the boat increases its speed, the length of the bow wave will eventually approach the length of the hull.

The speed at which the length of the bow wave equals the hull length is called the Hull Speed Limit.

Further increases in hull speed, beyond the Hull Speed Limit will cause the stern of the hull to drop into the trough of the bow wave.

This has the following bad effects:

- air can enter the displacement hull's propeller/s (reducing propeller thrust)
- the *belly* of the hull is exposed to the oncoming waves (increasing hull resistance)
- the increased incline of the propeller shaft/s reduces the amount of shaft thrust for forward motion (part of the forward component of propeller thrust is wasted in holding up the stern of the boat).

This greatly increases the hull's resistance-to-further-speed-increase. To go faster, the displacement hull must climb the crest of its own bow wave. For example, the last 10% of a displacement hull's top speed costs 27% of its engine power (and fuel consumption).

Mathematical Representation of Hull Speed Ratio

This relationship can be described mathematically. It is called the Hull Speed Ratio.

$$\text{Hull Speed Ratio (SLR)} = \frac{\text{Boat Speed}}{\sqrt{\text{Hull Length}}}$$

When the bow wave length is equal to the hull length, the speed length ratio formula can be expressed as follows:

$$4.5\sqrt{(\text{Hull Length}_{\text{meters}})} = (\text{Boat Speed}_{\text{km/hr}})$$

or

$$1.34\sqrt{(\text{Hull Length}_{\text{feet}})} = (\text{Boat Speed}_{\text{knots}})$$

Planing Hull

The planing hull skims over the surface of the water with relatively little disturbance of the water. The main resistance to planing hull speed is the skin friction. Hulls of this type are very sensitive to the smoothness of the hull, making good hull maintenance essential for top performance. Planing hulls are very sensitive to boat weight.

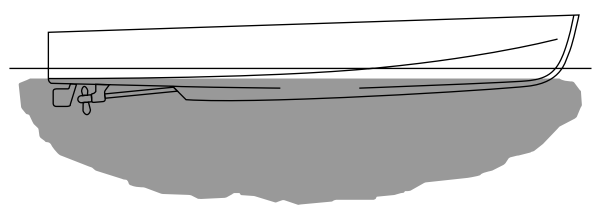


Figure 2.6

Semi-Displacement Hull

The semi-displacement hull looks very much like the planing hull and is easily mistaken for the planing hull. Semi-displacement hulls can be described as having characteristics of both planing and displacement hulls, but are not one or the other.

Displacement hulls have trouble with speed length ratios above 4.5 (1.34) due to their hull shape. The planing hulls have difficulties below speed length ratios of approximately 8.4 (2.5) because of their straight fore-and-aft lines.

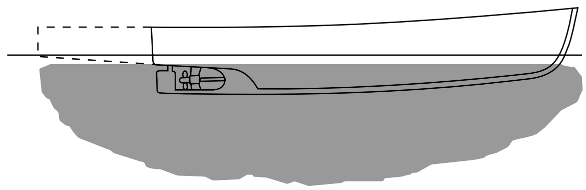


Figure 2.7

Semi-displacement hulls are designed to operate well in this speed range.

Semi-displacement hulls are characterized by the angle of the quarter-beam afterbody buttock line. Visualize a pair of vertical, parallel planes intersecting the hull — midway from the longitudinal center of the hull — to the waterline at the side of the boat. The intersection of the planes — with the bottom of the hull near the stern — form the quarter-beam afterbody buttock line (there are two, one on each side, but they have the same shape). The angle of the quarter-beam buttock line is formed between it and a line parallel to the at-rest waterline, fig. 2.8.

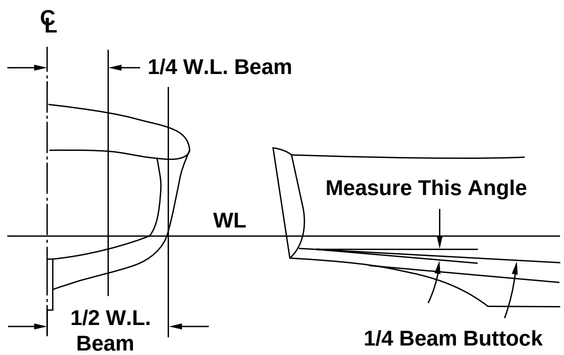


Figure 2.8

If the angle of the quarter-beam buttock line is very small (less than 2 degrees), the hull is capable of planing performance. At an angle of 4 degrees, the limiting speed length ratio will be around 2.0. An angle of 7 degrees will limit the speed to speed length ratios of 1.5, or just above displacement hull speeds. These angles should be measured relative to the hull's waterline at rest.

Rules of Thumb

Power to Reach Hull Speed

A useful rule of thumb for vessels below 100 tons displacement is:

Power to Reach Hull Speed horsepower = 5 x [Displacement long tons]

Fuel Consumption

A useful rule of thumb for basic budgetary purposes is:

Fuel Consumption = 1 Liter per hour per 5 horsepower.

Marine Engines Application and Installation Guide

- Driveline
- Mounting and Alignment
- Auxiliary

LEKM7143-01 (Supersedes LEKM7143) (05-00)

Driveline

Screw Propeller Drivelines
Jet Drive
Torsional Vibration
Driveline Couplings

Drivelines can be subdivided into two groups, depending on how the thrust forces driving the hull are created.

Screw propeller drivelines have a propeller converting engine power to thrust outside the hull. The thrust forces are generated on the propeller and transmitted to the hull through the propeller shafting and marine transmission. *

Jet drives have engine driven water pumps, either within the hull or bolted rigidly to it, which accelerate large flows of water. The thrust of the water leaving the pump propels the hull. The thrust forces are applied to the hull through the pump housing.

Screw Propeller Drivelines

There are several ways screw propeller drivelines connect the engine to the propeller.

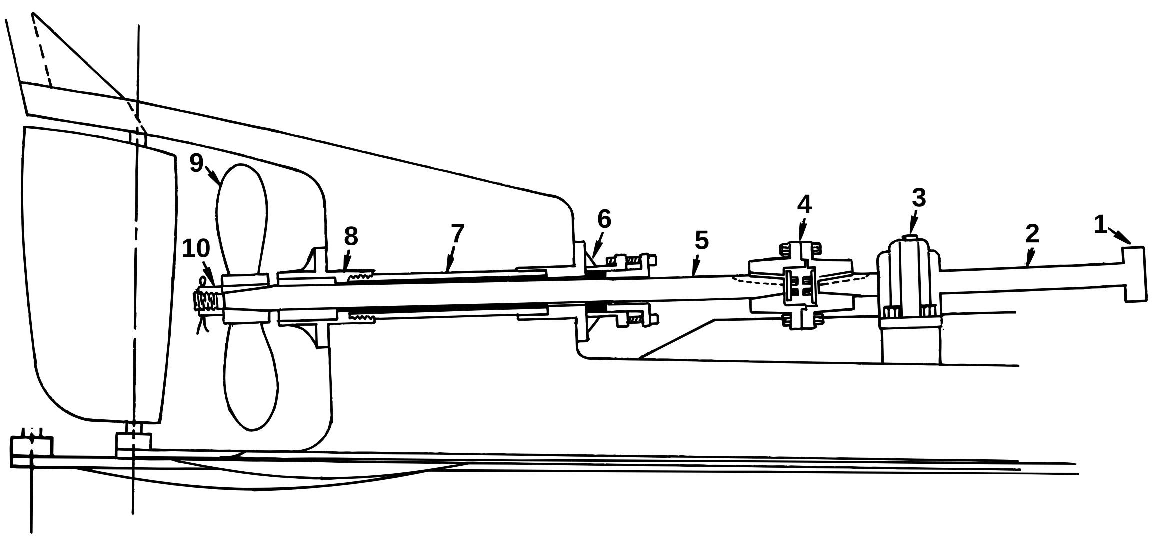
Conventional In-Line Propeller Systems

In conventional in-line propeller systems, the propeller shafting is straight, rigid and transmits the propeller thrust in a direct line from the marine gear output flange to the propeller. The engine(s) is/are located low, near the longitudinal center of the hull, and the marine gear(s) generally accepts the full propeller thrust.

Shaft Diameter and Bearing Spacing Selection

To prevent premature damage to shaft bearings, the shaft bearings should be close enough to prevent shaft whip, but far enough

*Boats with ducted propellers (Kort nozzles) receive a portion of their thrust directly from the hydrodynamic forces on the ducts. Ducted propellers are not common on fast boats due to the high drag of propeller ducts at high boat speeds.



DRIVELINE COMPONENTS – CENTERLINE MOUNTED THROUGH STERN POST

1. **Shaft companion flange**
2. **Intermediate shaft**
3. **Shaft bearing – pillow block, expansion type**
4. **Flange type shaft coupling**
5. **Tail shaft**
6. **Stuffing box – may or may not contain bearing**
7. **Stern tube – one end threaded, the other slip fit**
8. **Stern bearing**
9. **Propeller**
10. **Retaining nut**

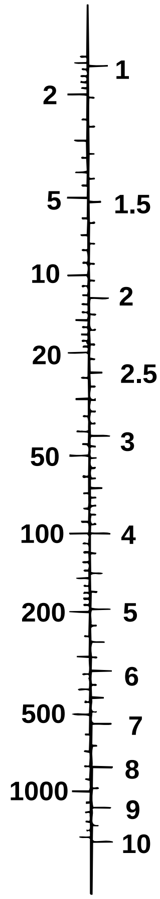
Figure 1.1

apart to permit the shaft to conform to the hull's flexing. For this reason, shafting should be designed for the thrust and torque forces applied. Since the tail shaft is more subject to damage from a propeller's contact with submerged objects, it should be strengthened for this purpose. The following nomograms will serve as a guide for shaft sizes and bearing spacing for commonly used shaft materials consistent with good marine practice.

Figure 1.2 **SHAFT DIAMETER SELECTION NOMOGRAPH**

Horsepower per 100 rpm Propeller Speed

Shaft Diameter (inches)



Tailshafts for heavy duty service should be increased by adding 1% of the propeller diameter to the basic shaft diameter.

Intermediate shafts may be reduced to .95 of the resultant shaft sizes.

Mild steel, read directly.
Tobin bronze, multiply by 1.05.
Monel, multiply by 0.88.

Example 1

Condition
Select bronze propeller shaft for engine with Intermittent rating of 315 hp @ 2000. Reduction ratio, 2:1. Propeller diameter, 32 inches. Light duty operation.

Solution
Intermittent hp = 315

Propeller rpm = $\frac{2000}{2} = 1000$

hp per 100 rpm = $\frac{315}{10.00} = 31.5$

Basic shaft diameter = 2.75 in.

Bronze shaft diameter = 2.75 x 1.05 = 2.88 in.

NOTE: Round off shaft diameter to nearest larger standard size.

Example 2

Condition
Select steel intermediate shaft and bronze tailshaft for heavy duty service. Engine develops 300 continuous hp@ 1800 rpm. Reduction ratio, 4.5:1. Propeller diameter, 54 inches.

Solution
Continuous hp = 300

Propeller rpm = $\frac{1800}{4.5} = 400$

hp per 100 rpm = $\frac{300}{4.00} = 75$

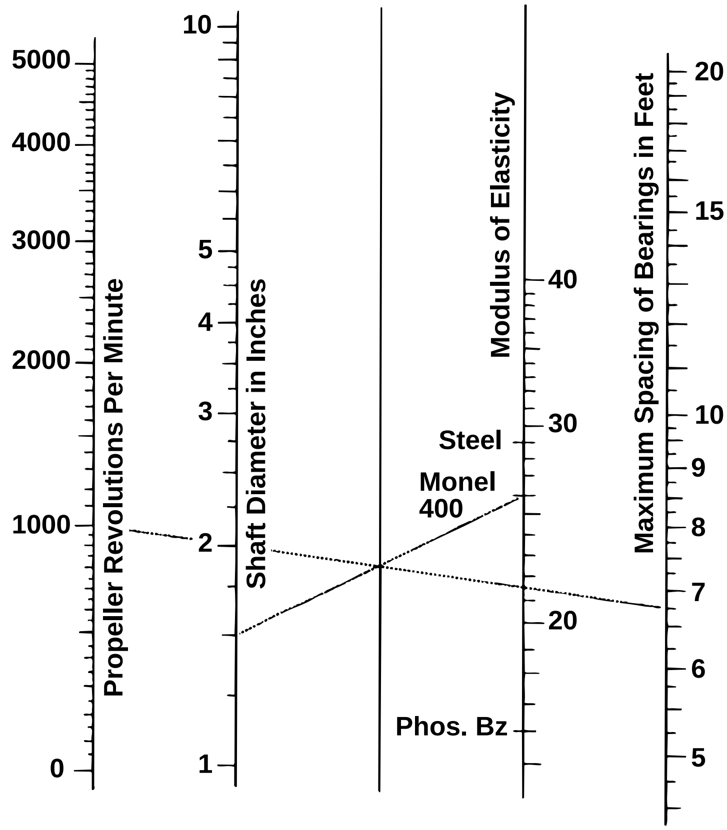
Basic shaft diameter = 3.64 in.

Steel intermediate shaft = 3.64 x 0.95 = 3.46 in.

Bronze tailshaft = (3.64 x 1.05) + (54 x .01)
= 3.82 + 0.54 = 4.36 in.

Figure 1.3

BEARING SPACING



Space bearings — rule a line from shaft in second line scale to modulus in fourth scale (26 for Monel). Then rule a line from point of intersection on center scale to connect propeller rpm on left scale and extend the line to right scale.

"Monel" is a registered trademark of International Nickel Corp.

Chart published courtesy of Paul G. Tomalin

Location of first shaft bearing aft of the marine transmission

The location of the first line shaft bearing from the marine transmission flange is extremely important.

To avoid inducing unwanted forces on the marine transmission thrust bearing, the line shaft bearing should be located at least 12, and preferably 20, or more shaft diameters from the marine transmission output flange. If the bearing must be located closer than 12 diameters, the alignment tolerances must be reduced substantially and the use of a flexible coupling considered.

Vee Drives

In Vee Drives, the propeller shafting is in two sections. The first section of shafting is from the propeller to a *Vee drive unit.*

The Vee drive unit is a bevel-gear box which allows the shafting to change directions. The Vee drive unit accepts full propeller thrust, transmitting the thrust to the hull through its mounting feet.

The second section of shafting turns sharply backward from the Vee drive unit to the engine. The engine is generally mounted as far to the rear of the boat as possible, with its flywheel end facing toward the bow of the boat.

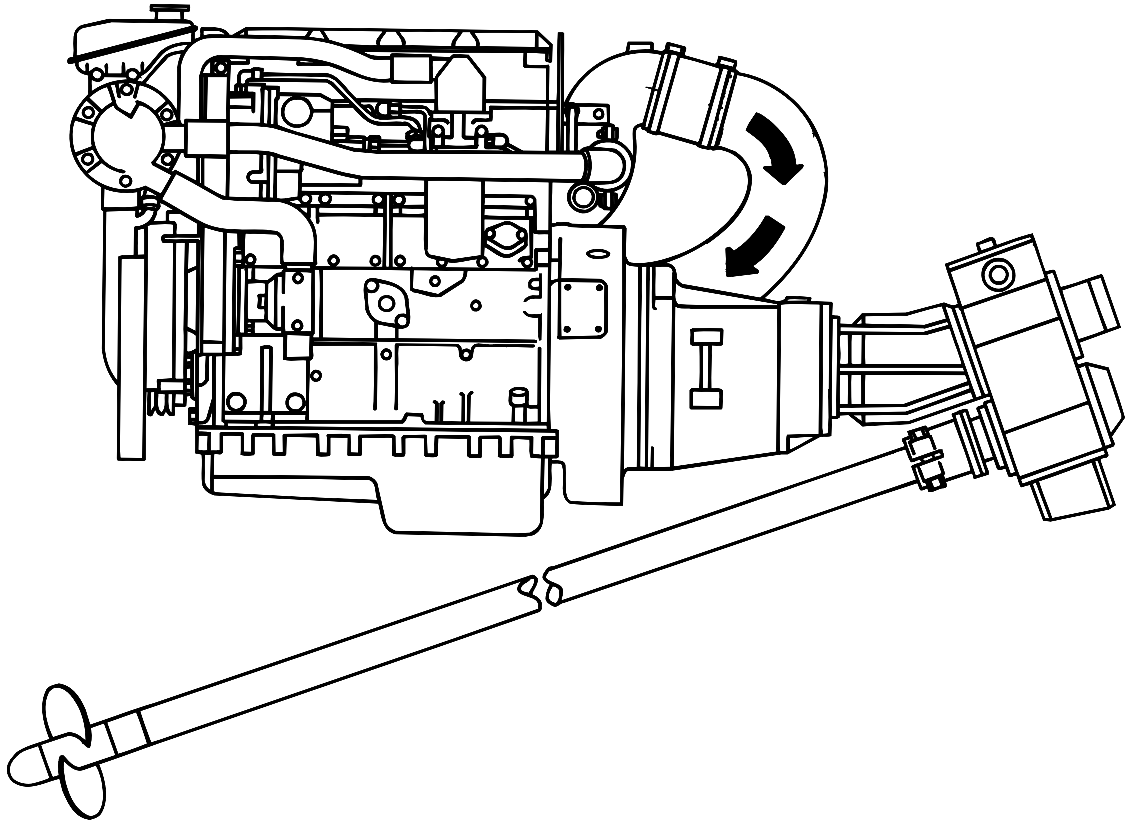


Figure 1.4 Vee Drive

There are a number of advantages of the Vee drive layout:
The engine is located at the extreme rear of the boat, taking minimal usable space within the hull.

Because the shafting between the engine and the Vee drive unit is not loaded with propeller thrust forces, that section of shaft can include universal joints or other soft couplings. The driveline softness allows use of soft engine mounts, resulting in a very quiet installation.

Disadvantage of the Vee drive arrangement:
The engine center of gravity is relatively high. It is further aft than conventional inline drivelines. This reduces stability and adversely affects hull balance.

Z Drive

The Z drive is a drive arrangement where the engine is connected to a right angle gear unit. A vertical drive shaft leads down through the hull to a submerged, second right angle gear unit. The lower right angle gear drives the propeller through a short length of horizontal drive shaft. The engine may face either fore or aft. Transverse engine orientations are not recommended.*

*The rolling of the boat can shorten the life of the crankshaft thrust bearings. When the boat rolls, the crankshaft will slide back and forth, hammering on its thrust bearings. If the engine is running, the motion will be dampened by the engine's oil film. If the engine is not running, the oil film is not present to protect the thrust bearing.

Stern Drives

The stern drive is a drive arrangement wherein the engine is connected to a reverse-reduction unit (to provide the reversing capability) which drives two right angle gear sets (through a double universal joining shaft) and a propeller. The engine flywheel faces aft.

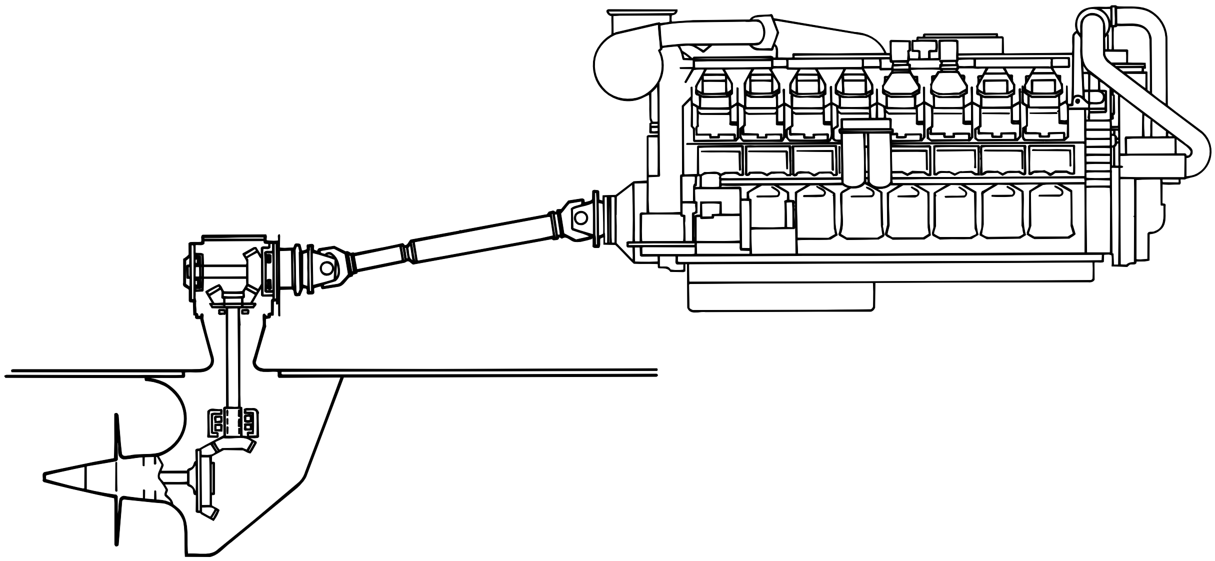


Figure 1.5 Stern Drive

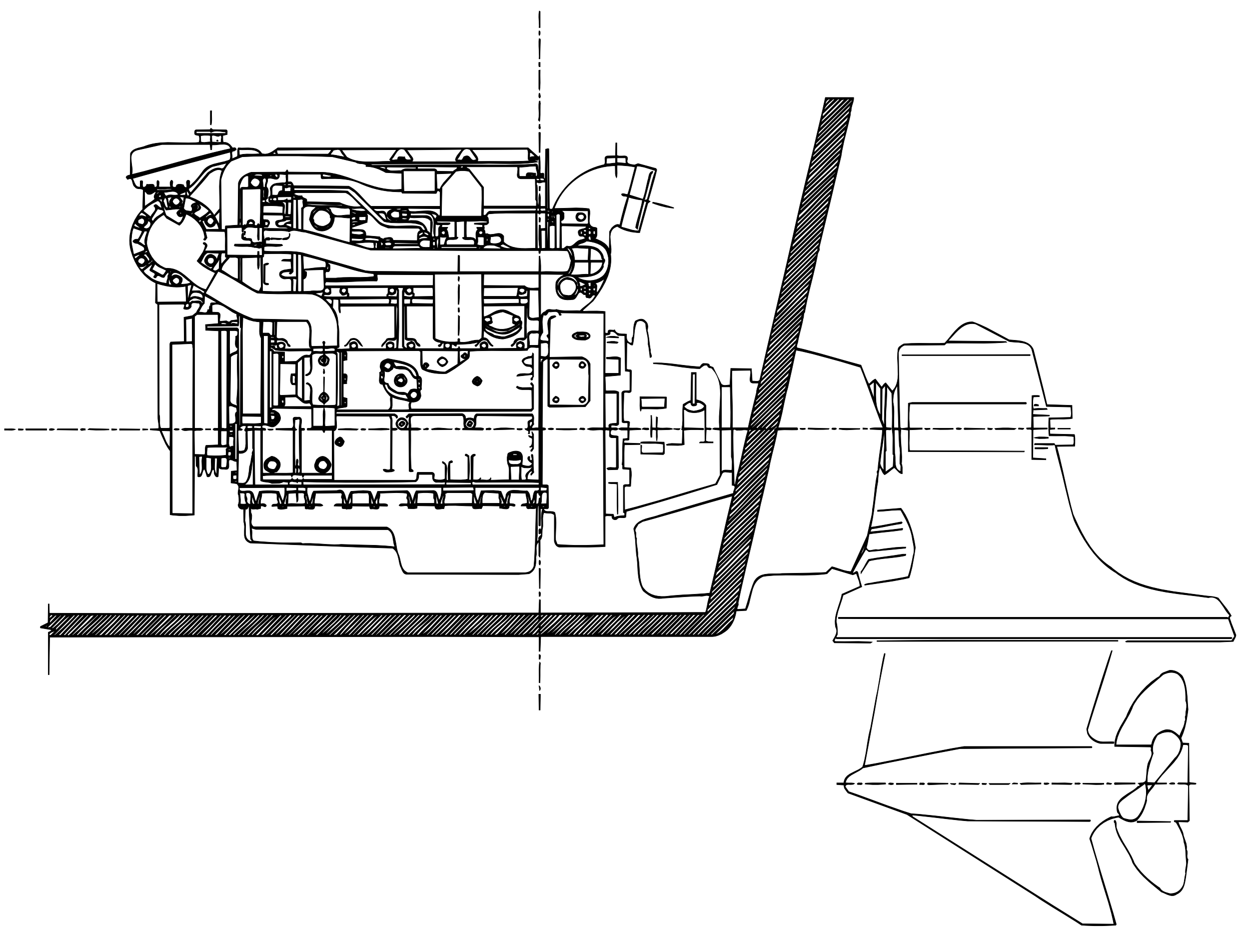


Figure 1.6

Jet Drive

Definition of Jet Drive

The boat is propelled by the acceleration of a flow of water, picked up from the bottom of the hull through an inlet grill and forced out through a nozzle mounted in the stern of the boat. The water flow is powered by an engine driven pump. The pump impeller is driven through a clutched reduction unit or direct from the engine flywheel to a universal joint shaft.

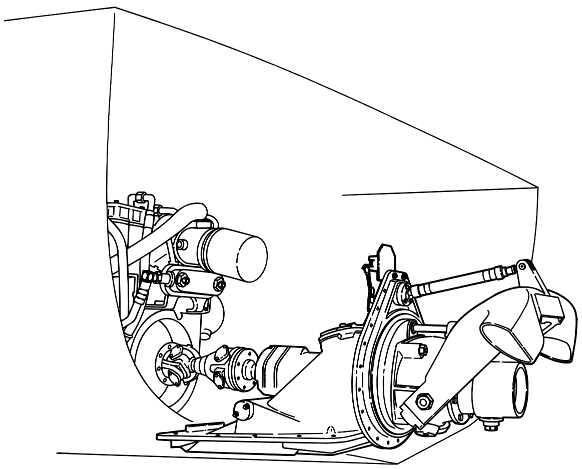


Figure 1.7 Jet Drive

Advantages of Jet Drive

- There is no need for a *reverse* gear. Reverse mode of operation can be accomplished by means of *buckets* or vanes on the discharge nozzle, which redirect the jet's discharge stream forward instead of aft.
- Operation in water too shallow for conventional propeller systems is possible.

Engine load with jet drives normally follow a cubic demand. Jet drive systems are significantly less prone to overload, since jet power demand is not very sensitive to vessel speed. Jet Drive units are less prone to damage from floating debris.

Disadvantages of Jet Drives

Disadvantages of the Jet Drive arrangement include:

- Block loading of the engine if the boat comes off the water ingesting air and loss of load. When the boat comes back in the water, block loading can occur if the engine speed is not matched to the load.
- A tendency to plug the inlet grill with debris. Some jet propulsion units are equipped with built-in cleaning mechanisms.

Torsional Vibration

Definition

Torsional vibration is cyclic irregularity of rotation in a shafting system. It is caused by engine combustion pulses, reciprocating motion and the propeller. As shafts in the system rotate, both the input torque (as each cylinder fires) and the resistance to rotation (caused by the propeller) varies. The torque variation is natural and unavoidable. It is only dangerous when uncontrolled.

Any shaft rotating with a mass attached to each end is capable of torsional vibration if there is any irregularity in the rotation of either mass. Rotation originates with the power stroke of the piston.

The simplified drivetrain below illustrates the relationship of shaft diameter, length and inertia of the natural frequency of the system.

Sources of Torsional Vibration

Many components in the driveline can cause or contribute to torsional vibration:

- Irregularity in the flow of water to a propeller caused by struts, appendages, hull clearance
- Propeller blade-to-blade pitch inaccuracies
- The intermeshing of gear teeth in the marine transmission
- Misaligned flexible couplings
- Universal joints (except for constant velocity types)
- Firing of individual cylinders of the engine
- Auxiliary loads driven from any of the engine's power takeoffs

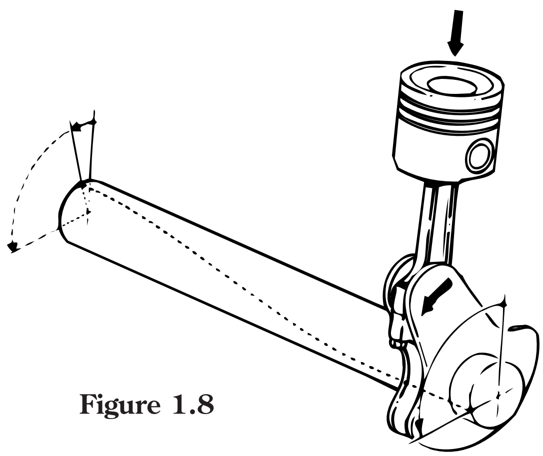


Figure 1.8

Mathematical Torsional Vibration Analysis

To ensure the compatibility of an engine and the drive equipment, a theoretical Torsional Vibration Analysis (TVA) is necessary. Disregarding the torsional compatibility of the engine and driven equipment, can result in extensive and costly damage to components in the drive train or engine failure. The torsional report will show the natural frequencies, the significant resonant speeds, and either the relative amplitudes or a theoretical determination of whether the maximum permissible stress level is exceeded. Also shown are the approximate nodal locations in the shafting system for each significant natural frequency.

Conducted at the design stage of a project, the mathematical torsional analysis may reveal torsional vibration problems which can be avoided by modification of driven equipment, shafts, masses or couplings.

Data Required to Perform Mathematical Torsional Analysis

The following technical data is required to perform a torsional analysis:

1. Operating speed ranges—lowest speed to highest speed, and whether it is variable or constant speed operation.
2. Load curve on some types of installations for application with a load dependent variable stiffness coupling.
3. With driven equipment on both ends of the engine, the horsepower requirement of each set of equipment is required and whether operation at the same time will occur.
4. A general sketch of the complete system showing the relative location of each piece of equipment and type of connection.
5. Identification of all couplings by make and model, along with WR^2* and torsional rigidity.
6. WR^2 or principal dimensions of each rotating mass and location of mass on attached shaft.
7. Torsional rigidity and minimum shaft diameter, or detailed dimensions of all shafting in the driven system, whether separately mounted or installed in a housing.

*WR^2 is a Polar Moment of inertia. It is the way we quantify the tendency of an object to resist changing its rotational speed. A flywheel is an object specifically designed to have a high polar moment of inertia. If its metal were concentrated near its hub, it would have a much lower polar moment of inertia, yet would have the same weight.

8. If a reciprocating compressor is utilized, a harmonic analysis of the compressor torque curve under various load conditions is required. If this is not available, then a torque curve of the compressor under each load condition is required. The WR^2 of the available flywheels for the compressor should be submitted

9. The ratio of the speed reducer or increaser is required. The WR^2 and rigidity that is submitted for a speed reducer or increaser should state whether or not they have been adjusted by the speed ratio squared

10. The WR^2 and number of blades on the propeller.

Availability of Torsion Characteristics of Caterpillar-Furnished Equipment

Upon request, mass elastic systems of items furnished by Caterpillar will be supplied to the customer without charge so that he can calculate the theoretical Torsional Vibration Analysis.

Mass elastic data for Caterpillar Diesel Engines, marine gears and generators is covered in the Technical Information System (TIS) and in Technical Marketing Information (TMI). If desired, Caterpillar will perform torsional vibration analyses. The data required prior to the analysis is described above. There is a nominal charge when this service is provided by Caterpillar.

Timing of Mathematical Torsional Vibration Analysis

The best time to perform a Mathematical Torsional Vibration Analysis is during the design phase of a project; before the driveline components are purchased and while the design can be easily changed if the TVA shows problems are likely to exist.

Responsibility for Torsional Compatibility

Since compatibility of the installation is the system designer's responsibility; it is also his responsibility to obtain the torsional vibration analysis.

Driveline Couplings

There are two types of driveline couplings: Rigid and soft couplings.

Rigid Couplings

Rigid shaft couplings may be classified by the method of attaching the shaft to the coupling.

Society of Automotive Engineers (SAE) Standards—SAE J755

Use of SAE standard shaft ends and couplings is recommended. They represent the highest standards of rigidity and reliability.

Other Rigid Couplings

The couplings using SAE standard shaft ends and hubs are cumbersome to remove and must be machined very carefully, to ensure concentricity and perpendicularity tolerances are met. The following couplings are easier to install and remove. They are also simpler to manufacture.

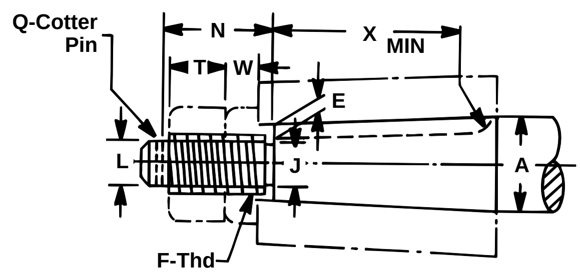


Figure 1.9

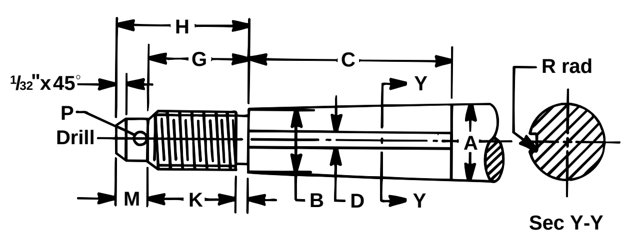


Figure 1.10

Dimensions for Shafts 3/4 to 6 inch in Diameter													
Non Shaft Diameter "A"	Small End Diameter "B"		Taper Length Diameter "C"	Keyway Width Diameter "D"			Keyway Side Diameter "E"			Keyway Fillet Radius "R"	Thread "F"		Taper End to Thread "G" End
	Min	Max		Nom	Min	Max	Nom	Min	Max		Dia	TPI	
.75	.624	.626	2.00	.19	.1865	.1875	.10	.095	.097	.031	.50	13.0	1.063
.88	.726	.728	2.38	.25	.249	.250	.13	.125	.127	.031	.63	11.0	1.250
1.00	.827	.829	3.75	.25	.249	.250	.13	.125	.127	.031	.75	10.0	1.438
1.13	.929	.931	3.13	.25	.249	.250	.13	.125	.127	.031	.75	10.0	1.438
1.25	1.030	1.032	3.50	.31	.3115	.3125	.16	.157	.160	.063	.88	9.0	1.625
1.38	1.132	1.134	3.88	.31	.3115	.3125	.16	.157	.160	.063	1.00	8.0	1.813
1.50	1.233	1.235	4.25	.38	.374	.375	.19	.189	.192	.063	1.13	7.0	2.000
1.75	1.437	1.439	5.00	.44	.4365	.4375	.22	.219	.222	.063	1.25	7.0	2.250
2.00	1.640	1.642	5.75	.50	.499	.500	.25	.251	.254	.063	1.50	6.0	2.625
2.25	1.843	1.845	6.50	.56	.561	.5625	.28	.281	.284	.094	1.75	5.0	3.000
2.50	2.046	2.048	7.25	.63	.6235	.625	.31	.312	.315	.094	1.75	5.0	3.000
2.75	2.257	2.259	7.88	.63	.6235	.625	.31	.313	.316	.094	2.00	4.5	3.500
3.00	2.460	2.462	8.63	.75	.7485	.750	.31	.311	.314	.094	2.33	4.5	3.875
3.25	2.663	2.665	9.38	.75	.7485	.750	.31	.311	.314	.125	2.50	4.0	4.375
3.50	2.866	2.868	10.25	.88	.8735	.875	.31	.310	.313	.125	2.50	4.0	4.375
3.75	3.069	3.071	10.88	.88	.8735	.875	.31	.310	.313	.125	2.75	4.0	4.750
4.00	3.272	3.274	11.63	1.00	.9985	1.000	.31	.309	.312	.125	3.00	4.0	5.125
4.50	3.827	3.829	10.75	1.13	1.123	1.125	.38	.373	.376	.156	3.25	4.0	5.625
5.00	4.249	4.251	12.00	1.25	1.248	1.250	.44	.434	.437	.188	3.75	4.0	6.375
5.50	4.671	4.673	13.25	1.25	1.248	1.250	.44	.435	.438	.188	4.00	4.0	6.750
*6.00	4.791	4.793	14.50	1.38	1.373	1.375	.50	.493	.496	.219	4.25	4.0	7.500
*6.50	5.187	5.189	15.75	1.38	1.373	1.375	.50	.494	.497	.219	4.50	4.0	8.250
*7.00	5.582	5.584	17.00	1.50	1.498	0.500	.56	.555	.558	.250	5.00	4.0	9.000
*7.50	5.978	5.980	18.25	1.50	1.498	1.500	.56	.556	.559	.250	5.25	4.0	9.375
*8.00	6.374	6.376	19.50	1.75	1.748	1.750	.56	.553	.556	.250	5.75	4.0	9.750

*6 in. through 8 in. bore uses 1 in./ft taper. 1/12 in. per taper. Angle with centerline 2° 23 ft 9 in.

Dimensions for Shafts 3/4 to 6 inch in Diameter													
Non Shaft Diameter "A"	Extension Beyond Taper "H"	Undercut		Diameter of Pin End "L"	Length of Pin End "M"	Cotter Pin Hole		Cotter Pin "Q"		Nuts			Keyway Length "X"
		"J"	"K"			"N"	"P" Drill	Nom Dia	Length	Size	Plain Thick "T"	Jamb Thick "W"	
.75	1.313	.391	.125	.375	.250	1.141	.141	.125	.75	.500 - 13	.500	.313	1.500
.88	1.500	.484	.125	.438	.250	1.328	.141	.125	.75	.625 - 11	.625	.375	1.781
1.00	1.750	.594	.125	.500	.313	1.516	.141	.125	1.00	.750 - 10	.750	.438	2.125
1.13	1.750	.594	.125	.500	.313	1.516	.141	.125	1.00	.750 - 10	.750	.438	2.125
1.25	2.719	.125	.625	.375	1.000	.719	.172	.156	1.25	.875 - 9	.875	.500	2.813
1.38	2.250	.813	.125	.750	.875	1.906	.172	.156	1.50	1.000 - 8	1.000	.563	3.188
1.50	2.438	.906	.188	.875	.875	2.094	.172	.156	1.50	1.125 - 7	1.125	.625	3.500
1.75	2.750	1.031	.188	1.000	.500	2.359	.203	.188	1.75	1.250 - 7	1.250	.750	4.219
2.00	3.125	1.250	.188	1.250	.500	2.734	.203	.188	2.00	1.500 - 6	1.500	.875	4.938
2.25	3.500	1.375	.188	1.375	.500	3.141	.266	.250	2.25	1.750 - 5	1.750	1.000	5.625
2.50	3.500	1.438	.188	1.438	.500	3.141	.266	.250	2.25	1.750 - 5	1.750	1.000	6.094
2.75	4.000	1.688	.250	1.688	.500	3.641	.266	.250	2.50	2.000 - 4.5	2.000	1.125	6.656
3.00	4.375	1.938	.250	1.938	.500	4.016	.266	.250	3.00	2.250 - 4.5	2.250	1.250	7.344
3.25	5.125	2.125	.375	2.125	.750	4.578	.375	.375	3.00	2.500 - 4	2.500	1.500	8.500
3.50	5.125	2.125	.375	2.125	.750	4.578	.375	.375	3.00	2.500 - 4	2.500	1.500	9.250
3.75	5.500	2.375	.375	2.375	.750	4.953	.375	.375	3.50	2.750 - 4	2.750	1.625	10.000
4.00	5.875	2.500	.375	2.500	.750	5.328	.375	.375	3.50	3.000 - 4	3.000	1.750	10.500
4.50	6.375	2.750	.375	2.750	.750	—	—	—	—	3.250 - 4	3.250	1.875	9.625
5.00	7.125	3.250	.375	3.250	.750	—	—	—	—	3.750 - 4	3.750	2.125	10.875
5.50	7.750	3.500	.500	3.500	1.000	—	—	—	—	4.000 - 4	4.000	2.250	12.125
*6.00	8.500	3.875	.500	3.875	1.000	—	—	—	—	4.250 - 4	4.250	2.250	13.250
*6.50	9.250	4.375	.500	4.375	1.000	—	—	—	—	4.500 - 4	9.500	2.500	14.375
*7.00	10.000	4.875	.500	4.375	1.000	—	—	—	—	5.000 - 4	5.000	2.750	15.625
*7.50	10.375	5.125	.500	5.125	1.000	—	—	—	—	5.500 - 4	5.500	3.000	16.875
*8.00	10.750	5.375	.500	5.375	1.000	—	—	—	—	5.750 - 4	5.750	3.125	18.125

*6 in. through 8 in. bore uses 1 in./ft taper. 1/12 in. per taper. Angle with centerline 2° 23 ft 9 in.

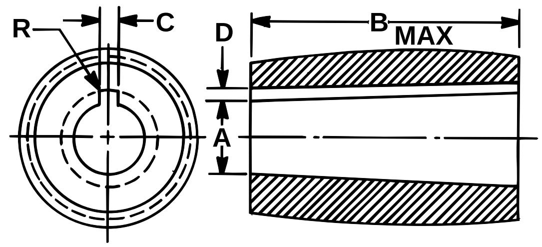
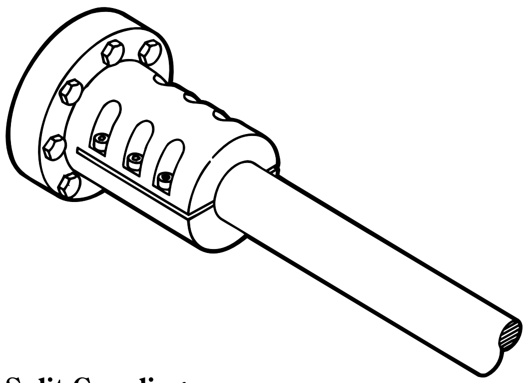
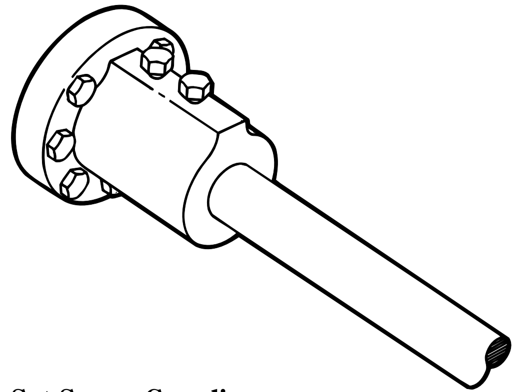


Figure 1.11

Marine Propeller Hub Bore Dimensions									
Nom Bore Diameter	Diameter Small End "A"		Maximum Length "B"	Keyway Width "C"			Keyway Side Depth "D"		
	Min	Max		Nom	Min	Max	Nom	Min	Max
.75	.608	.610	2.250	.188	.1865	.1875	.094	.098	.100
.88	.710	.712	2.625	.250	.249	.250	.125	.129	.131
1.00	.811	.813	3.000	.250	.249	.250	.125	.129	.131
1.13	.913	.915	3.375	.250	.249	.250	.125	.129	.131
1.25	1.015	1.017	3.750	.313	.3115	.3125	.156	.162	.165
1.38	1.116	1.118	4.125	.313	.3115	.3125	.156	.161	.164
1.50	1.218	1.220	4.500	.375	.374	.375	.188	.195	.198
1.75	1.421	1.423	5.250	.438	.4365	.4375	.219	.226	.229
2.00	1.624	1.626	6.000	.500	.499	.500	.250	.259	.262
2.25	1.827	1.829	6.750	.563	.561	.5625	.281	.291	.294
2.50	2.030	2.032	7.500	.625	.6235	.625	.313	.322	.325
2.75	2.233	2.235	8.250	.625	.6235	.625	.313	.322	.325
3.00	2.437	2.439	9.000	.750	.7485	.750	.313	.323	.326
3.25	2.640	2.642	9.750	.750	.7485	.750	.313	.323	.326
3.50	2.843	2.845	10.500	.875	.8735	.875	.313	.324	.327
3.75	3.046	3.048	11.250	.875	.8735	.875	.313	.324	.327
4.00	3.249	3.251	12.000	1.000	.9985	1.000	.313	.326	.329
4.50	3.796	3.798	11.250	1.125	1.123	1.125	.375	.388	.391
5.00	4.218	4.220	12.500	1.250	1.248	1.250	.438	.450	.453
5.50	4.640	4.642	13.750	1.250	1.248	1.250	.438	.450	.453
6.00	4.749	4.751	15.000	1.375	1.373	1.375	.500	.517	.520
6.50	5.145	5.147	16.250	1.375	1.373	1.375	.500	.516	.519
7.00	5.541	5.543	17.500	1.500	1.498	1.500	.563	.579	.582
7.50	5.937	5.939	18.750	1.500	1.498	1.500	.563	.579	.582
8.00	6.332	6.334	20.000	1.750	1.748	1.750	.563	.582	.585



Split Coupling
Figure 1.12



Set Screw Coupling
Figure 1.13

Advantages of the Split Coupling

Split couplings can use shaft ends which require no additional machining after receipt from the shafting supplier. Good quality shafting is generally received with a ground or ground/polished finish. The shaft can generally be removed from a split coupling without use of heat or a press.

The ability to retain the shaft is excellent in a well machined split coupling.

Disadvantages of the Split Coupling

Split couplings use only friction to keep the shaft in the coupling. There is no positive mechanical stop preventing the shaft from pulling out of the coupling.

The inside diameter of the coupling bore should be within 0.025-0.050 mm (0.001-0.002 in.) of the outside diameter of the shaft end to prevent vibration from concentricity error unbalance.

Advantages of the Set Screw Coupling

The set screw coupling allows very easy shaft removal and reinstallation.

Set screw couplings can use shaft ends which require no additional machining after receipt from the shafting supplier; good quality shafting is generally received with a ground/polished finish.

The set screw coupling requires the least work to manufacture. It is the least expensive of all the rigid couplings.

Disadvantages of the Set Screw Coupling

If the fit between the coupling bore and shaft is loose enough to permit convenient installation/removal, it is generally loose enough to allow at least some vibration due to concentricity error.

Set screws in set screw couplings will cause some marring of the propeller shaft.

The heads of the set screws protrude from the outside surface of the set screw coupling. Good safety practice dictates using guards/shields for personnel protection against accidentally contacting the rotating set screw heads.

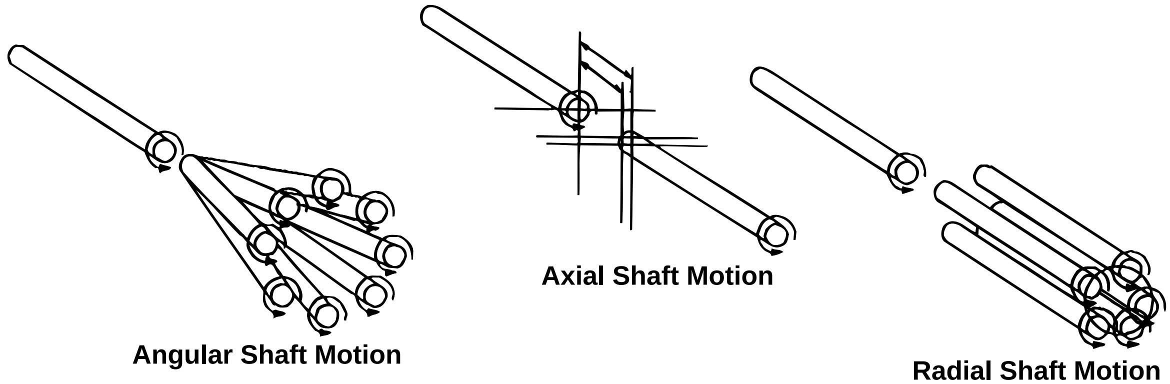


Figure 1.14

Soft Couplings

Soft couplings will accept relative motions between their *driving* and *driven* sides without damage. These relative motions can take the following forms.

Types of Softness

Rotating shafts which will move relative to each other need couplings which permit misalignment without damage. The ability to accept misalignment without damage is often called *softness.*

Radial Softness

Couplings which are radially soft will allow the driving or driven shafts some freedom of motion, so long as their centerlines remain parallel.

Axial Softness

Couplings which are axially soft will allow the driving or driven shafts to vary in their end-to-end spacing. The sliding spline joints in the middle of an automotive universal joint shaft, to allow the effective shaft length to vary are shaft couplings with axial softness.

Angular Softness

Angularly soft couplings allow shaft angle to vary. *Universal Joints* are examples of angularly soft couplings.

Combinations of Softness

Most commercially available shaft couplings are able to accommodate combinations of the above types of relative motion or misalignment. They vary in their tolerance for different types of motion. Good design practice will investigate the potential for the various types of shaft motion/misalignment and confirm the shaft couplings are capable of accommodating the expected shaft motion.

Torsional Vibration Isolation

Torsional vibration is cyclic irregularity of rotation in a shafting system. It is caused by engine combustion pulses, reciprocating motion, and the propeller. As shafts in the system rotate, both the input torque (as each cylinder fires) and the resistance to rotation (caused by irregularities in the velocity of water entering the propeller) vary. The torque variation is natural and unavoidable. It is only dangerous when uncontrolled. Torsionally soft couplings are a way to control torsional vibration.

Protection from Misalignment

Propeller shafts, marine transmissions and engines are mounted to the hull. The hull is not perfectly rigid. Storm waves, temperature changes, propeller thrust, engine torque reaction, vessel loading and other factors result in forces which deform the hull causing misalignment in the shafting. The misalignment is unavoidable. Soft couplings allow the system to accept the misalignment without damage.

Sound Isolation

The driveline of the vessel is a source of noise. One of the methods of reducing shipboard noise is to interrupt the noise path. One path for noise is from the engine down the propeller shaft to the hull via the stuffing box*. A soft coupling in the propeller shaft is an excellent means of introducing resilience into the path between the source of the noise (the engine and transmission) and the receiver of the noise (the ears of the personnel on board).

*There are other equally important noise paths. Another is from the engine to the hull via the exhaust piping. See the Exhaust Section for guidance on use of resilience in supporting exhaust piping.

Another noise path from the engine to the hull is via the engine mounting feet. See the Mounting Section for guidance on use of resilient engine mounts. Cooling lines can transmit engine vibration to the hull where it will be perceived as noise by the crews. Cooling water connections must include flexibility.

CATERPILLAR®

Mounting and Alignment

Marine Gear Installation – Propeller Shaft Droop
Marine Gear Installation – Alignment
Installation/Alignment Instructions
Marine and Engine Gear Mounting

Marine Gear Installation–Propeller Shaft Droop

Introduction

This procedure outlines the first of three basic steps involved in the installation, alignment, and mounting of marine gears. It applies to both free standing gears and gears that are fixed directly to the engine. It has limited application to units which are soft mounted or do not have a shaft support bearing between the gear and stern bushing.

The second and third basic steps concerning:

- Marine gear to propeller shaft alignment.
- Marine gear (and engine) mounting, will be covered in later sections of the application and installation procedures.

Propeller Shaft Droop

Before commencing alignment of the marine gear to the propeller shaft, the shaft *droop,* or *deflection* due to the unsupported shaft and companion flange weight must be compensated for (Figure 2.1). This is an important part of the installation and alignment procedure for marine gears. Otherwise, exceptional loading to the marine gear lower shaft bearings or the first propeller shaft line bearing may result with consequent increased noise or vibration and decreased service life of affected bearings.

Two methods are presented here for eliminating shaft droop as part of the alignment process:

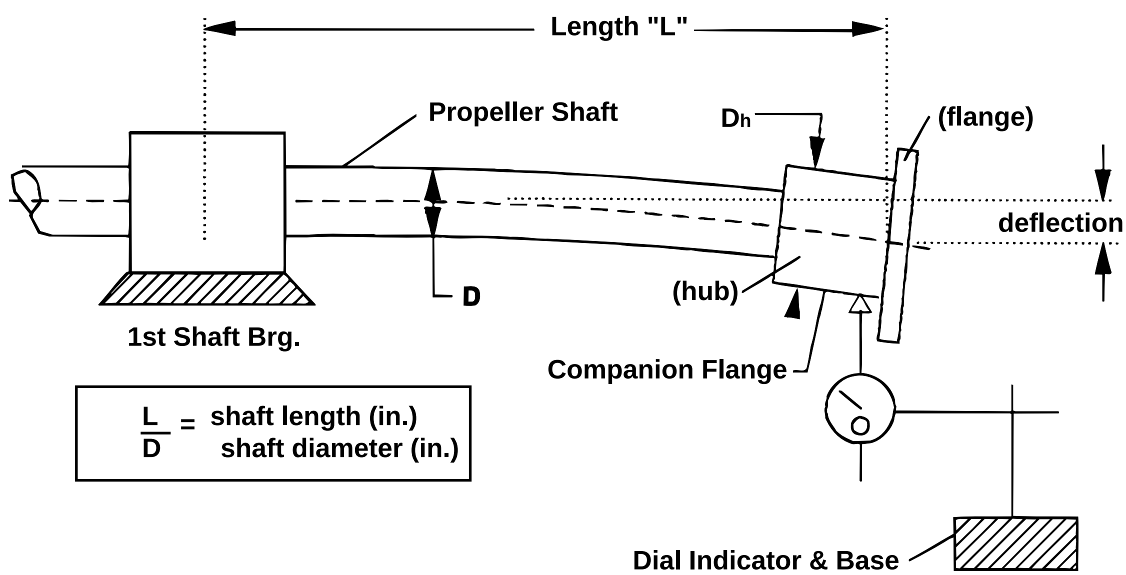
- The *estimated droop* method, whereby droop at the companion flange is estimated from droop tables and directly compensated for.
- The *scaled hoist* method, whereby unsupported shaft weight is directly compensated for.

Both methods give reasonably close approximations of the shafts true center if done correctly.

Note: For both procedures prior connection and alignment of propeller shafting from the first line bearing aft is required. Also, the shaft should be positioned about 13 mm (0.5 in.) aft of its final position when attached to the marine gear.

Estimated Droop Method

This method involves the use of tables which contain calculated deflection or *droop* values for overhung steel shafting with small, medium, or large flanges mounted on the free end (Figures 2.6, 2.7, 2.8; appendix A). Refer to Figure 2.1 for illustration of this



SHAFT DROOP DIAGRAM

Figure 2.1

deflection or droop and the dimensions that apply in using the droop tables.

Three sets of values, as illustrated in Figure 2.1, are used in determining droop (deflection) at the shaft free end (flange end). They are:

1. Flange hub dia. “Dh”;
2. Propeller shaft dia. “D”, and;
3. Shaft overhang length “L”.

Dh/D determines which set of droop tables apply (see Appendix A).

Dh/D = 1.40 to 1.74 use Figure 2.6.
Dh/D = 1.75 to 1.99 use Figure 2.7.
Dh/D = 2.00 to 2.25 use Figure 2.8.

The “D” and “L/D” intersection in the appropriate table determines the estimated droop value.

To obtain shaft droop, first determine the values “L/D”, and “Dh/D”.

Then, go to the proper droop table for the “Dh/D” value and at the intersection of “D” and “L/D” read droop directly in inches.

For example: If Dh = 9.0 in. and D = 6.0 in. then Dh/D = 1.5 and Figure 2.6 should be referred to. If overhung length is 120 in. then L/D = 120/6 = 20.0. The droop is read directly from Figure 2.6 at the intersection of D = 6.0, and L/D = 20 as droop = 0.148 in.

In many cases the actual D and L/D values will fall between those listed in the tables. In those cases droop can be found by interpolation of the data in the tables. An example of this follows:

D = 6.3 in.
L/D = 20.4
Dh/D = 1.5

Since Dh/D is between 1.4 and 1.74 we will use Figure 2.6.

The actual droop is shown as Ya, see Figure 2.2.

Shaft		**Shaft Diameter**	
L/D	**6.0**	**6.3**	**6.5**
20.0	**0.148**	**(a)**	**0.174**
20.4		**Ya**	
21.0	**0.178**	**(b)**	**0.208**
22.0	**0.211**		**0.248**

Figure 2.2

To obtain Ya proceed as follows:

Let D_a = Actual diameter of the shaft
D_1 = Next lower diameter on droop table
L/D_a = Actual length to Diameter ratio
L/D_1 = Next lower L/D ratio on droop table
L/D_2 = Next higher L/D ratio on droop table

Then Y_1 = Droop at L/D_1 and D_1
Y_2 = Droop at L/D_2 and D_2
Y_a = Actual Droop
$Y_a = R\,(b - a) + a$

Where $R = (L/D_a - L/D_1) \,/\, (L/D_2 - L/D1)$
$a = (D_a/D_1)^2 \times Y_1$
$b = (D_a/D_1)^2 \times Y_2$

In this example
$R = (20.4 - 20)/(21 - 20) = 0.4$
$a = (6.3/6.0)^2 \times 0.148 = 0.163$
$b = (6.3/6.0)^2 \times 0.178 = 0.196$
$Ya = 0.4\ (0.196 - 0.163) + 0.176$
$= 0.189$ in.

Scaled Hoist Method

This method involves lifting, with the use of a scale, a weight equal to one half the overhung shaft weight, plus all the companion flange weight with the lifting applied to the companion flange as shown in the illustration (Figure 2.3).

Weights for steel shafts or circular sections can be calculated using the following formula:

WEIGHT (lb) = 0.22 x D^2 x L

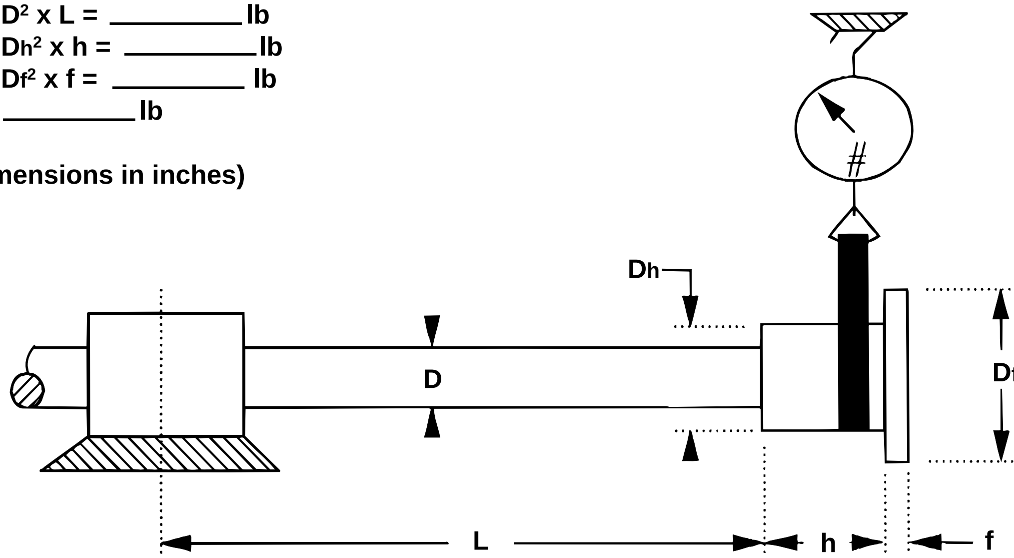
Where:

D = diameter of shaft or circular section in inches
L = length of shaft or circular section in inches

Alternatively, the weights of the shaft and flange sections can be determined by use of Figure 2.4 by simply multiplying the length, in inches, of any cylindrical section by the *lb/in.* value listed for the diameter of that section.

Scale Wt. #(lb) = sum of:
0.11 x D^2 x L = __________ lb
0.22 x D_h^2 x h = __________ lb
0.22 x D_f^2 x f = __________ lb
Wt.# = __________ lb

(all dimensions in inches)



SCALED CORRECTION FOR SHAFT DROOP

Figure 2.3

Weights of Circular Steel Sections Per Inch of Length										
Section Diameter			**2.00**	**2.25**	**2.50**	**2.75**	**3.00**	**3.25**	**3.50**	**3.75**
lb/in. Length			0.88	1.11	1.38	1.66	1.98	2.32	2.70	3.09
4.00	**4.25**	**4.50**	**4.75**	**5.00**	**5.25**	**5.50**	**5.75**	**6.00**	**6.25**	**6.50**
3.52	3.97	4.46	4.96	5.50	6.06	6.66	7.27	7.92	8.59	9.30
6.75	**7.00**	**7.25**	**7.50**	**7.75**	**8.00**	**8.25**	**8.50**	**8.75**	**9.00**	**9.25**
10.0	10.8	11.6	12.4	13.2	14.1	15.0	15.9	16.8	17.8	18.8
9.50	**9.75**	**10.0**	**10.5**	**11.0**	**11.5**	**12.0**	**12.5**	**13.0**	**13.5**	**14.0**
19.9	20.9	22.0	24.3	26.6	29.1	31.7	34.4	37.2	40.1	43.1
14.5	**15.0**	**15.5**	**16.0**	**16.5**	**17.0**	**17.5**	**18.0**	**18.5**	**19.0**	**19.5**
46.3	49.5	52.9	56.3	59.9	63.6	67.4	71.3	75.3	79.4	83.7

Figure 2.4

Since half the shaft weight plus all of the companion flange weight is to be compensated for at the scale, total scale weight can be calculated by the work sheet included in Figure 2.3 (example Figure 2.5).

Example Problem:

(ref. Figures 2.3 & 2.5) If in our illustration Figure 2.3, the following dimensions apply:

Shaft diameter **D = 4.0 in.;** and, shaft length **L = 60.0 in.;**

Companion flange hub diameter **D_h = 6.0 in;** hub length **h = 6.5in.;**

Flange section diameter **D_f = 9.0 in.;** flange thickness **f = 0.75 in.**

Proceed in calculating hoist pull "P" (lb) as follows: Note: In this example both methods of obtaining weights will be shown.

First, overhung shaft weight is calculated as: (per formula-1)

Wt. = 0.22 x $(4.0)^2$ x 60 = 211.2 lb

(per Figure 2.4)

Wt. = 60 in x 3.52 lb/in = 211.2 lb

Half of overhung shaft wt. = **105.6 lb**

Then, companion flange weight, including shaft material inserted into the flange, is calculated:

(per formula-1):

Hub section

wt. = 0.22 x $(6.0)^2$ x 6.5 = 51.5 lb

Flange section

wt. = 0.22 x $(9.0)^2$ x 0.75 = 13.4 lb

(per Figure 2.4)

Hub section

wt. = 6.5 in x 7.92 lb/in = 51.5 lb

Flange section

wt. = 0.75 in x 17.8 lb/in = 13.4 lb

Total companion flange wt. = **64.9 lb**

Finally, the scale reading at the hoist should be the sum of the total companion flange wt. and half the overhung shaft wt.:

"P" = 105.6 + 64.9 = 170.5 lb

Caution: The hoisting mechanism must be set up in such a manner that the direction of pull or lift is straight up, i.e., no force is to be exerted sideways in order to avoid side to side misalignment of the marine gear to the propeller shaft. This can be checked with a plumb line suspended from the overhead connection for the hoisting mechanism.

Also, if the hoist is to be removed and the shaft supported by other means prior to final connection to the marine gear, dial indicators at vertical and side locations should be employed to insure the shaft remains at its proper position.

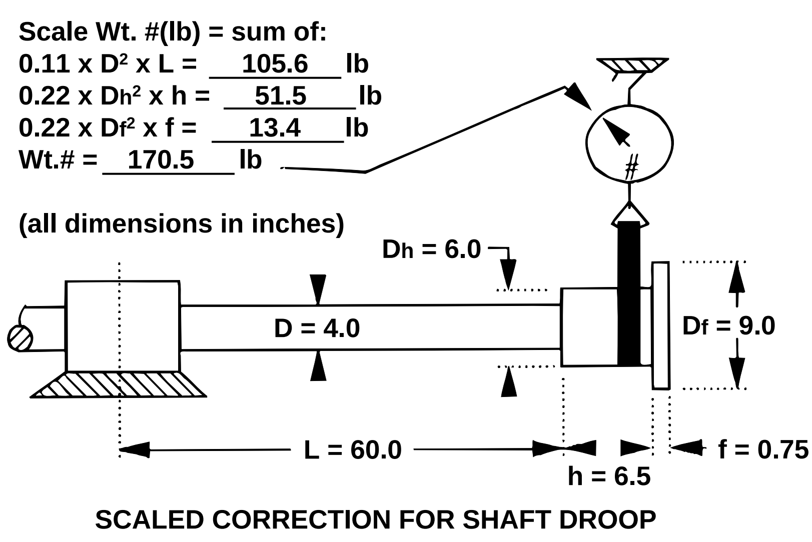


Figure 2.5

Shaft Diameter (inches) for Hub Diameter = 1.40 to 1.74 times Shaft Diameter													
Shaft L/d	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0
8	.001	.001	.001	.002	.002	.003	.004	.004	.005	.006	.007	.008	.009
9	.001	.001	.002	.003	.004	.005	.006	.007	.008	.009	.011	.013	.014
10	.001	.002	.003	.004	.005	.007	.008	.010	.012	.014	.016	.018	.021
11	.002	.003	.004	.006	.007	.009	.012	.014	.017	.019	.023	.026	.029
12	.003	.004	.006	.008	.010	.013	.016	.019	.023	.027	.031	.035	.040
13	.003	.005	.008	.010	.013	.017	.021	.025	.030	.036	.041	.047	.054
14	.004	.007	.010	.014	.018	.022	.028	.033	.040	.047	.054	.062	.071
15	.006	.009	.013	.017	.023	.029	.035	.043	.051	.060	.070	.080	.091
16	.007	.011	.016	.022	.029	.036	.045	.054	.065	.076	.088	.101	.115
17	.009	.014	.020	.028	.036	.046	.056	.068	.081	.095	.110	.127	.144
18	.011	.017	.025	.034	.044	.056	.069	.084	.100	.117	.136	.156	.178
19	.014	.021	.031	.042	.054	.069	.085	.103	.122	.144	.166	.191	.217
20	.016	.026	.037	.050	.066	.083	.103	.124	.148	.174	.201	.231	.263
21	.020	.031	.044	.060	.079	.100	.123	.149	.178	.208	.242	.278	.316
22	.023	.037	.053	.072	.094	.119	.147	.178	.211	.248	.288	.330	.376
23	.028	.043	.062	.085	.111	.140	.173	.210	.250	.293	.340	.390	.444
24	.033	.051	.073	.100	.130	.165	.203	.246	.293	.344	.399	.458	.521
25	.038	.059	.085	.116	.152	.192	.237	.287	.342	.401	.465	.534	.608
26	.044	.069	.099	.135	.176	.223	.275	.333	.396	.465	.539	.619	.704
27	.051	.079	.114	.155	.203	.257	.317	.384	.457	.536	.622	.714	.812
28	.058	.091	.131	.178	.233	.295	.364	.441	.524	.615	.714	.819	.932
29	.067	.104	.150	.204	.266	.337	.416	.503	.599	.703	.815	.936	1.065
30	.076	.118	.170	.232	.303	.383	.473	.573	.681	.800	.927	1.065	1.211

Figure 2.6

Shaft Diameter (inches) for Hub Diameter = 1.75 to 1.99 times Shaft Diameter													
Shaft L/d	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0
8	.001	.001	.002	.003	.004	.005	.006	.007	.008	.009	.011	.013	.014
9	.001	.002	.003	.004	.005	.007	.008	.010	.012	.014	.016	.019	.021
10	.002	.003	.004	.006	.008	.010	.012	.014	.017	.020	.023	.027	.030
11	.003	.004	.006	.008	.010	.013	.016	.020	.024	.028	.032	.037	.042
12	.004	.006	.008	.011	.014	.018	.022	.027	.032	.037	.043	.050	.056
13	.005	.007	.010	.014	.019	.024	.029	.035	.042	.049	.057	.065	.074
14	.006	.009	.014	.018	.024	.030	.038	.045	.054	.064	.074	.085	.096
15	.008	.012	.017	.023	.031	.039	.048	.058	.069	.081	.094	.108	.122
16	.010	.015	.022	.029	.038	.049	.060	.073	.086	.101	.117	.135	.153
17	.012	.019	.027	.036	.047	.060	.074	.090	.107	.125	.145	.167	.190
18	.015	.023	.033	.044	.058	.074	.091	.110	.131	.153	.178	.204	.232
19	.018	.027	.040	.054	.070	.089	.110	.133	.158	.186	.216	.247	.282
20	.021	.033	.048	.065	.084	.107	.132	.160	.190	.223	.259	.297	.338
21	.025	.039	.057	.077	.101	.127	.157	.190	.226	.266	.308	.354	.402
22	.030	.046	.067	.091	.119	.150	.186	.225	.267	.314	.364	.418	.475
23	.035	.054	.078	.107	.139	.176	.218	.264	.314	.368	.427	.490	.558
24	.041	.063	.091	.124	.163	.206	.254	.307	.366	.429	.498	.571	.650
25	.047	.074	.106	.144	.188	.238	.294	.356	.424	.497	.577	.662	.754
26	.054	.085	.122	.166	.217	.275	.339	.411	.489	.573	.665	.763	.869
27	.062	.097	.140	.191	.249	.315	.389	.471	.560	.658	.763	.876	.996
28	.071	.111	.160	.218	.284	.360	.444	.538	.640	.751	.871	1.000	1.137
29	.081	.126	.182	.247	.323	.409	.505	.611	.727	.854	.990	1.136	1.293
30	.091	.143	.206	.280	.366	.463	.572	.692	.823	.966	1.121	1.286	1.464

Figure 2.7

Shaft Diameter (inches) for Hub Diameter = 2.00 to 2.25 times Shaft Diameter													
Shaft L/d	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0
8	.001	.002	.003	.004	.005	.006	.008	.009	.011	.013	.015	.018	.020
9	.002	.003	.004	.006	.007	.009	.011	.014	.016	.019	.022	.026	.029
10	.003	.004	.006	.008	.010	.013	.016	.020	.023	.027	.032	.036	.041
11	.004	.006	.008	.011	.014	.018	.022	.027	.032	.037	.043	.050	.057
12	.005	.007	.011	.014	.019	.024	.030	.036	.043	.050	.058	.067	.076
13	.006	.010	.014	.019	.025	.031	.039	.047	.056	.065	.076	.087	.099
14	.008	.012	.018	.024	.032	.040	.050	.060	.071	.084	.097	.111	.127
15	.010	.016	.022	.031	.040	.051	.062	.076	.090	.106	.122	.141	.160
16	.012	.019	.028	.038	.050	.063	.078	.094	.112	.131	.152	.175	.199
17	.015	.024	.034	.047	.061	.077	.096	.116	.138	.161	.187	.215	.245
18	.019	.029	.042	.057	.074	.094	.116	.141	.167	.196	.228	.261	.297
19	.022	.035	.050	.069	.089	.113	.140	.169	.201	.236	.274	.315	.358
20	.027	.042	.060	.082	.107	.135	.167	.202	.240	.282	.327	.375	.427
21	.032	.049	.071	.097	.126	.160	.197	.239	.284	.334	.387	.444	.505
22	.037	.058	.084	.114	.148	.188	.232	.281	.334	.392	.455	.522	.594
23	.043	.068	.097	.133	.173	.219	.271	.328	.390	.458	.531	.609	.693
24	.050	.079	.113	.154	.201	.254	.314	.380	.452	.531	.616	.707	.804
25	.058	.091	.130	.178	.232	.293	.362	.438	.522	.612	.710	.815	.928
26	.067	.104	.150	.204	.266	.337	.416	.503	.599	.703	.815	.935	1.064
27	.076	.119	.171	.233	.304	.385	.475	.575	.684	.802	.931	1.068	1.215
28	.086	.135	.194	.265	.345	.437	.540	.653	.777	.912	1.058	1.215	1.382
29	.098	.153	.220	.299	.391	.495	.611	.739	.880	1.033	1.198	1.375	1.564
30	.110	.172	.248	.338	.441	.558	.689	.834	.992	1.165	1.351	1.551	1.764

Figure 2.8

Marine Gear Installation

General Information

Alignment of the marine gear to propeller shafting in the vessel warrants close attention. The alignment must be within specified tolerances for satisfactory transmission service life. This discussion outlines the steps in accomplishing such alignment. It applies to both free standing (island mounted) marine gears and those bolted directly to the propulsion engine at the flywheel.

Alignment must be accomplished while the shafting is at, or very near, its true centerline position (ref. discussion in preceding section regarding shaft droop).

After the propeller shaft droop has been compensated for, and the shaft properly supported at the free end as shown in Figure 2.9, the marine gear or gear and engine combination may then be aligned to the propeller shaft.

Alignment Terms and Parameters

The objective of the alignment process outlined herein is good axial alignment of the marine gear to the shafting. Axial alignment is the relationship of the axis of rotation of the members to be coupled, in this case, the propeller shaft and gear output flanges.

There are two basic alignment parameters involved in this process. They are:

1. Parallel or *bore* alignment.
2. Angular or *face* alignment.

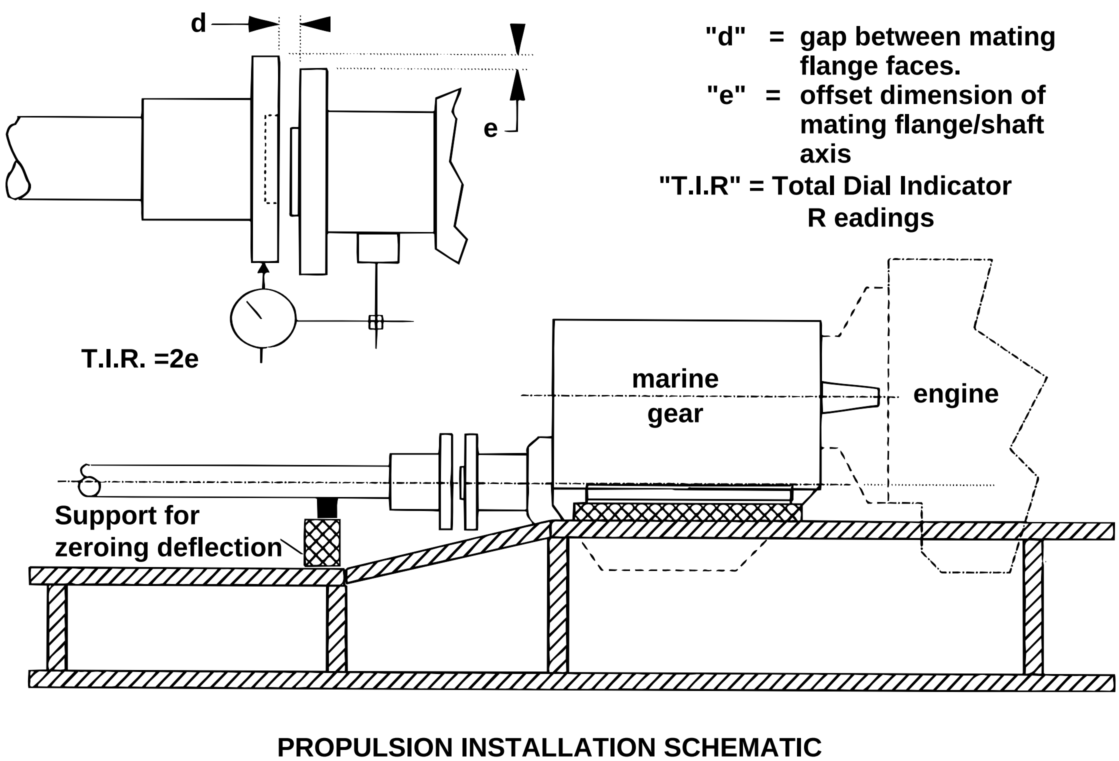


Figure 2.9

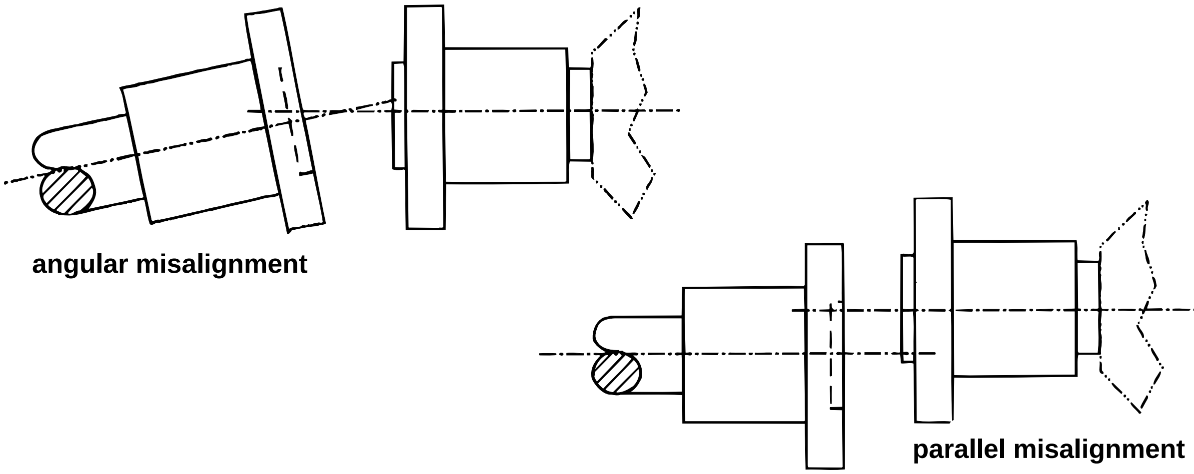


Figure 2.10

Angular misalignment occurs when the centerlines (axis) of the marine gear output shaft and the propeller shaft are not parallel. The limits for parallel and angular misalignment are given in the Alignment Procedure section.

Parallel misalignment occurs when the centerlines, or axis, of the marine gear output shaft and propeller shaft are parallel but not in line.

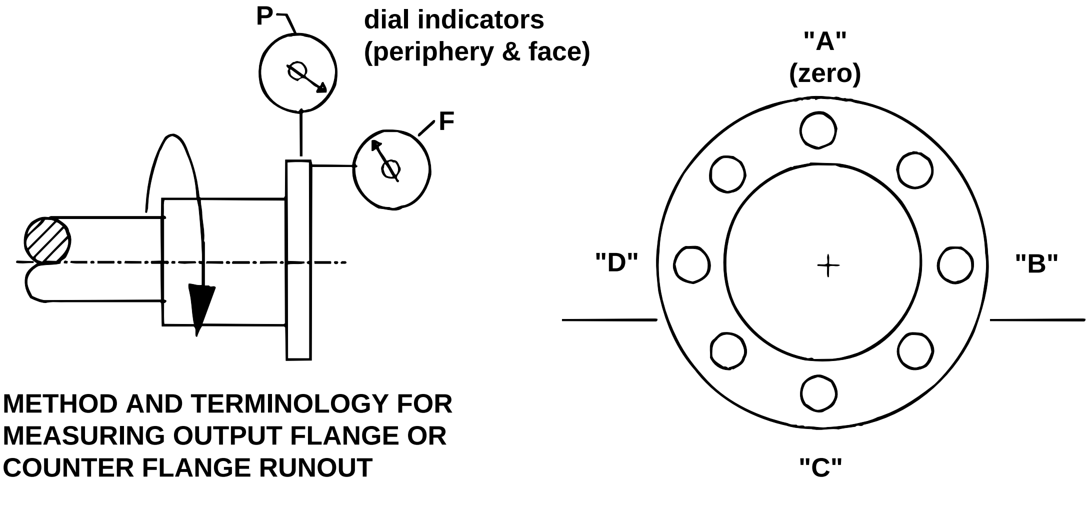
Conditions Required for Alignment

Precision machinery deserves every effort to assure its proper alignment, but it cannot be assumed the machinery bed is a level, stationary, non-deflecting surface. The hull is constantly deflecting, due to daily and seasonal temperature variation, various load and ballast conditions, and sea conditions.

Do not attempt final alignment of propulsion machinery unless the following conditions are met:

- The vessel is in the water.
- All permanent ballast is in place.
- Fuel, Water, and Temporary Ballast Tanks are filled to normal, average, operating levels, generally 1/2 to 3/4 filled. It is not necessary to fill fuel tanks with fuel, since it is only the weight of the fluid in the tanks that is important to load the hull to representative displacement. Water is a suitable alternative to fuel for this purpose.
- All major machinery—weighing over 225 kg (500 lb)—is either installed or simulated by equivalent weights appropriately located.

Note: Where prior experience has shown good results, the alignment process can be substantially completed prior to launch. Make final alignment check immediately prior to sea trials.



METHOD AND TERMINOLOGY FOR MEASURING OUTPUT FLANGE OR COUNTER FLANGE RUNOUT

Figure 2.11

Alignment Procedure

Marine gears and engines with flywheel housing mounted marine gears:

1. With the propeller shaft set on roller block or oiled "V" block sweep the flange face with a dial indicator (Figure 2.11). While watching the indicator, rotate the shaft and note the point of *minimum* indicator reading. Make a reference mark (with paint, center punch, . . .) on the flange rim at the bolt hole nearest the point of *minimum* indicator reading. For all future readings on this flange this will be the "ZERO" reference location "A".

2. With the dial indicator set at the "A" zero reference location, set the dial to zero. Rotate the shaft and measure and record indicator readings at the 3:00, 6:00, and 9:00 positions (facing the flange). These positions correspond to positions "B", "C", and "D" respectively. Continue the rotation to one complete turn, rechecking at the "A", or 12:00 position for zero indicator reading.

3. Using the same zero "A" reference point set the dial indicator to measure and runout of the flange rim (periphery). Repeat the procedure outlined in step 2 to accomplish this.

4. Repeat steps 1 to 3 to measure and record marine gear output flange face and peripheral runout *with the following exception:* Use bolt hole location nearest point of "maximum" indicator reading for zero "A" reference.

Any runout errors must be accounted for in the alignment process. The maximum allowable runout, for most Twin Disc marine gears offered by Caterpillar, is:

Maximum Face Runout = 0.10 mm (0.004 in.)
Maximum Bore Runout = 0.10 mm (0.004 in.)
Runouts shown are total dial indicator readings (T.I.R.).

Note: If face (or bore) runout is excessive on the marine gear flange, some correction may be obtained by removing and reinstalling the flange at a different position (if the type of connection permits).

Compensation may also be obtained by selective match of the mating flanges providing flanges have not already been reamed and matched for fitted bolts connection. For example, if the "A" reference points of both flanges, as determined in steps 1 and 4 are mated then face runout compensation will be realized (Figure 2.12).

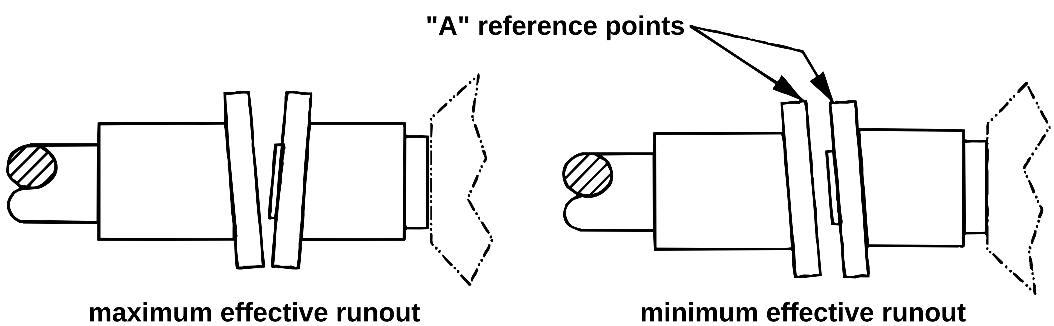
Carrying this example further, if maximum T.I.R. of the shaft flange is +0.008 in. and that of the marine gear flange is –0.005 in. they are individually out of tolerance, however, their effective runout is their sum which is 0.008 + (–0.005) = 0.003 in.. This would be acceptable since it is within the 0.004 in. limit.

5. Position the propeller shaft about 8 mm (0.3 inches) aft of its planned final position.

Note: At this point scantlings or supports for the gear engine mounting should be in place with sufficient gap for poured or metal shims. If not, install these supports now.

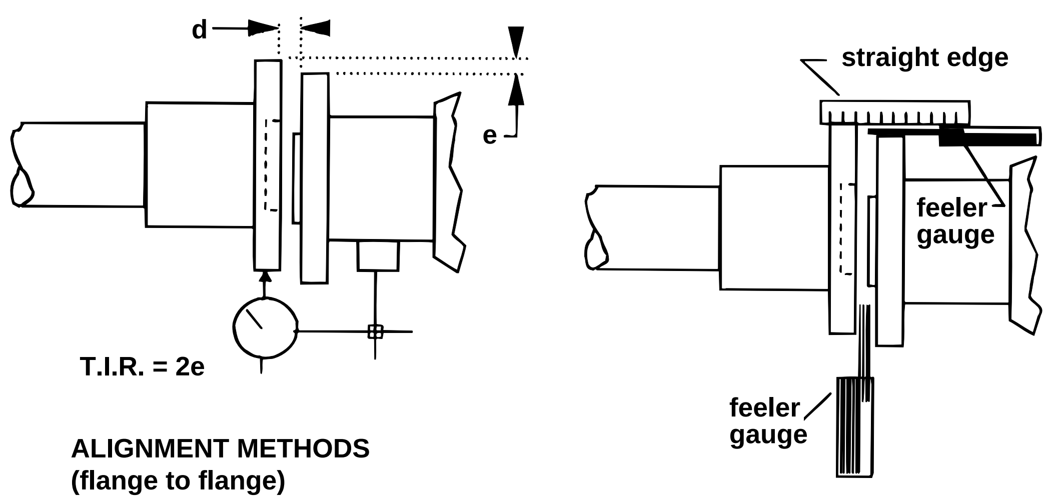
6. Now move the marine gear (or gear engine package) to its approximate final position so that mating flange gap is about 8 mm (0.3 in.), per step 5, without engaging the flange pilot.

7. **Bore alignment:** Take measurements of the diameter gap at four equally spaced points on the flanges' diameters. A straight edge with a feeler gauge, or a dial indicator may be used as shown in Figure 2.13 to accomplish this. If the dial indicator is used, mount it on the gear output flange as shown and sweep the companion flange for T.I.R. Make appropriate position adjustments with wedges or jacking screws to locate the marine gear output flange within 0.127 mm (0.005 in.) of bore alignment (dimension e, Figure 2.13). Maximum T.I.R. would be 0.254 mm (0.010 in.). A slip fit of mating flange pilots on transmissions so equipped—ensures adequate bore alignment.



COMPENSATION FOR MATING FLANGE FACE RUNOUT

Figure 2.12



ALIGNMENT METHODS
(flange to flange)

Figure 2.13

8. When this condition is met, engage the flanges' pilot surfaces by bringing the propeller shaft and companion flange forward to within 4.5 mm (0.180 in.) at all points about the diameter of the flanges.

9. **Angular Alignment:** Position adjustments of the engine and transmission may now be made to align the marine transmission output flange to a proper angular or face alignment position. Using a feeler gauge, or small hole gauge, take measurements of face gap at four equally spaced positions; A, B, C, and D. (Figure 2.13) Then, proceed as follows:

   a. Tabulate the readings of face gap.
   b. Compare diametrically opposite face gap readings (compare readings at A to readings at C; and readings at B to readings at D).
   c. Subtract the smaller of diametrically opposite readings from the larger for both A to C, and B to D.

      Example: if the reading at A is 0.175 in. and the reading at C is 0.165 in., then subtract 0.165 from 0.175 giving a result of 0.010 in.

   d. The resulting differences are proportional to the amount of angular or face misalignment. The face gap difference reading must not exceed 0.005 in. (A to C, or B to D). If the gap difference reading exceeds this value, the engine and marine gear must be moved until the required tolerance is reached.
   e. As the engine is moved for angular alignment, ensure the bore alignment is not altered.

10. Recheck all alignment readings, insert the bolts into the flanges, and prepare the marine gear/engine supports for final securing to the boat's engine foundation.

Except for final alignment checks, after securing the unit to the foundation and prior to sea trial of the vessel, this completes the alignment process.

Installation/Alignment Instructions

Caterpillar Engines and other Free Standing Marine Gears and Vulkan Rato Flexible Couplings

Introduction

The purpose of this instruction sheet is to outline a proven and effective procedure for accurate alignment of Caterpillar Marine Engines to free standing reduction gears, most specifically, where Vulkan Rato flexible couplings are used. The procedure can be adapted to most other coupling types if care is taken to allow relative radial and angular movement between engine and marine gear coupling interface during the alignment process.

Preliminary

The following preliminary steps should already have been done or in place before starting engine to gear alignment:

1. Propeller shafting *in place*. Marine gear aligned to shaft and *secured to foundation* with fitted bolts and thrust stops as required.

2. Flexible coupling inner member mounted to marine gear and outer member mounted to engine flywheel per Figure 2.14.

3. Engine positioned in close proximity (by sight) to the final alignment, but with coupling outer and inner members not touching.

4. Jacking screws are for engine side to side and fore and aft positioning movement in place (provided and installed by shipyard or installer). Jacking screws provided with engine for vertical positioning should be clean and well oiled.

5. Necessary tools and instrumentation at hand, such as: dial indicators with bracketry; yoke for mounting instruments to marine gear hub (ref part numbers 6V2042 & 6V2043); pry bar for moving

crankshaft fore/aft and flywheel up/down; turning tool for rotating engine at flywheel; and miscellaneous hand tools.

Step 1

a. Clean periphery of flywheel and face of coupling/flywheel adapter plate so dial indicators can sweep face and circular runout at flywheel through one complete rotation. Remove any nicks or burrs that interfere with dial indicator readings. A light cleaning with oil may also be beneficial.

b. Mount dial indicators from flywheel housing as shown in Figure 2.14.

c. Rotate engine through one complete revolution and note total dial indicator readings for both face and periphery of flywheel. Record these readings. (Omit this step if coupling has torsional stop.)

d. Remove an engine block side cover and, with the pry bar between the crankshaft web and the main bearing saddle, move the crankshaft fore and aft. Record total indicator movement on data sheet. Leave crankshaft in full forward position.

e. With pry bar, move flywheel up (full travel) and record half of the total movement. This is the flywheel *droop*.

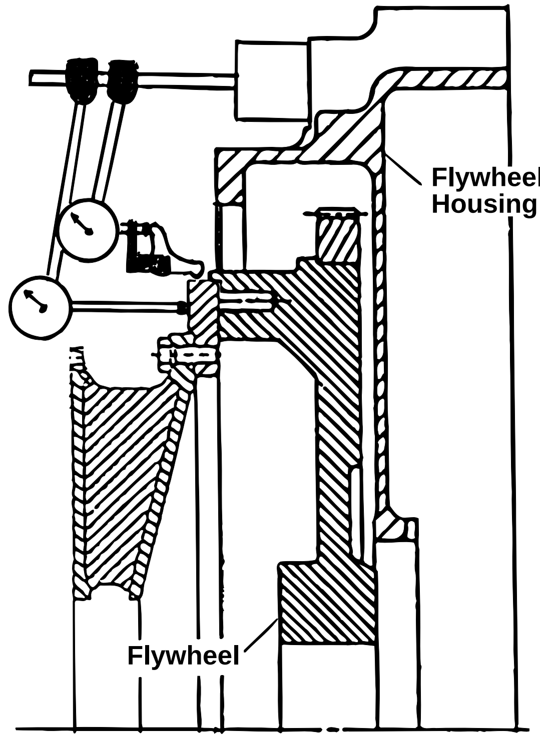


Figure 2.14

f. Determine offset dimension “Y”. This is the amount the marine gear should be set higher than the engine and is equal to flywheel droop, (as found in step “e”), plus the difference in thermal growth of the engine and gear input centers “Xt”.

“Y” = [F. W. droop + Xt]
“Xt” is approximated by the formula:
Xt = [0.009 – (0.0004 x d)] in.
(Where d = gear input to mounting ledge distance in inches.)

Note: For WAF gear models 540 & 560; Xt = + 0.003 in. may be used. For WAF models 860 & 1160; use Xt = – .002 (negative). Xt for WAF models 640, 660, 740, 760, 840, and 1140 is small and can be ignored.

g. Compute “Y” = F.W. droop + Xt = ______ + ______ = ______ Record “Y” on data sheet, see Figure 2.15.

Step 2

a. Remove dial indicators from flywheel housing.

b. With the crankshaft still fully forward, position the engine to the marine gear so that the flexible coupling outer member is aligned with the inner member (by sight) and the gap between the inner and outer members, measured with a feeler gauge, is equal to half the total fore/aft crankshaft movement, measured in step 1d, ±0.002 in. The gap should be measured at the connection points for the bolts marked “X” in Figure 2.16.

Note: Do not install these bolts at this time.

c. Turn the marine gear input shaft and check for freedom of movement. It should turn without much difficulty but may have a slight drag against the coupling outer or engine side member. If it turns with difficulty, or not at all, repeat part “a” of this step.

Step 3

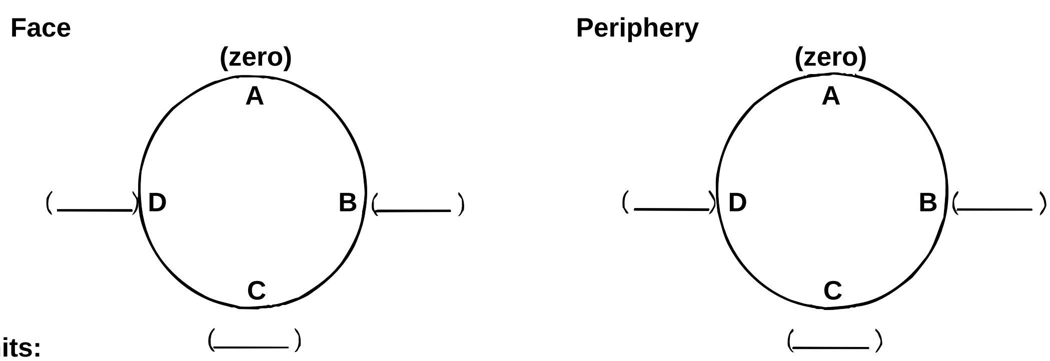
a. Install necessary yoke, brackets, etc., and mount dial indicators as shown in Figure 2.16.

Note: For the procedure outlined here it will be assumed that movement of the indicator tip into the dial case results in a *positive* reading. Also, all dimensions are in inches unless indicated.

Note: If coupling is equipped with torsional stops (come home feature) proceed directly to sub step “e”; If coupling is not equipped with torsional stops (come home feature) proceed as follows:

b. Swing marine gear input shaft through one revolution while making sure no obstructions or protrusions hinder smooth movement of the indicator tips over the surfaces.

c. Set dial indicators at “A” position and set dials to zero readings. Rotate the marine gear input member (or engine and gear together if equipped with torsional stops) and record T.I.R. readings, for both face and periphery, at locations B, C, and D per diagram in Figure 2.15. Recheck for zero at location A.



Limits:
B,C,D = ±0.010 in.
max indicator reading

Max. indicator difference
of B to D is 0.010 in.

Limits:
C = 2Y±0.005 in. (Y = ______ in.)
D + B = C± 0.005 in.
Max. indicator difference
of B to D is 0.010 in.

Figure 2.15

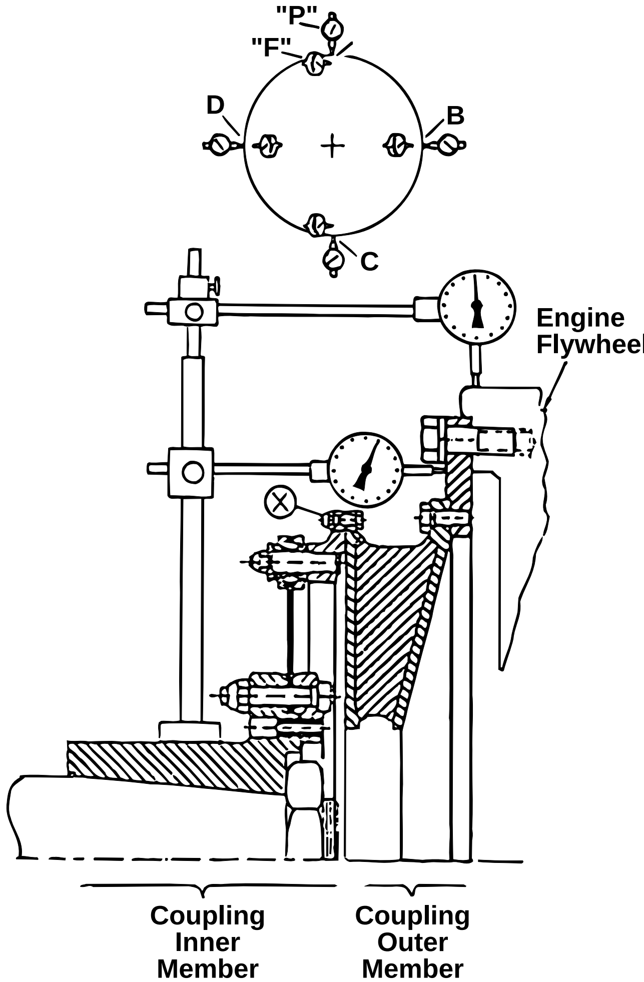


Figure 2.16

d. Using the limits outlined in Figure 2.15, for face and peripheral dial indicator readings, adjust engine position as required to obtain readings within those limits. This will probably require several attempts to achieve the desired readings. *For couplings without torsional stop feature,* refer back to runout readings recorded at step 1c. If those readings are not more than 0.127 mm (0.005 in). face and 0.203 mm (0.008 in.) peripheral proceed directly to step 4, otherwise, continue at “e”.

e. Install several clearance bolts at location “X” so that engine and gear can be rotated together without unduly constraining relative radial movement of the mating coupling members.

f. Repeat sub steps “c” and “d” except rotate engine and gear together by barring or turning the engine flywheel. When readings within the desired range, per Figure 2.15, are obtained remove the clearance bolts at “X” and proceed to step 4.

Step 4

Check axial clearance between coupling members. With crankshaft full forward there should be some clearance between members or, if they are touching, inner member should be rotatable against outer member without excessive drag. (One man should be able to turn gear input shaft by hand or with a .610 mm (2 ft) long bar.) With crankshaft all the way to the rear, coupling members should be touching or gap should not exceed 0.508 mm (0.020 in.). Make sure these conditions are met before proceeding to step 5.

Step 5

a. Pour engine chocks or install solid steel shims and secure engine to foundation by appropriate procedures. *Do Not install fitted foundation bolt at this time.*

Note: Loose metal shims are not recommended for heavy engine mountings.

b. When part “a” is complete (be sure sufficient cure time has been allowed for poured chocks) recheck face and peripheral alignment readings per step 3. If readings have changed, which is often the case, do not re-align if limits outlined in Figure 2.15 have not been exceeded by more than 0.127 mm (0.005 in.) face and 0.254 mm (0.010 in.) periphery.

c. With condition “b” met install fitted bolt at rear engine mount on either left or right side. Install collision chocks if required.

d. Using bore aligning tools (tapered pins) align mating coupling members and install snug fit bolts at locations “X”. Do not hammer bolts in. Press them in or squeeze them in with pliers. They should be installed with the heads to the engine side.

e. Remove instrumentation, reinstall block side cover, etc.

f. This completes the alignment process. If the alignment has been done in dry dock or on a new build in the yard, then alignment should be rechecked when vessel is floated and contains partial stores. This step is critical for vessels with less rigid foundation systems or those more sensitive to float conditions such as flat bottom types.

Record Final Readings

From steps 1 “d” & “e”:

Total flywheel vertical movement . . ___ in.
Total crankshaft movement fore/aft . . ___in.

FACE	PERIPHERY
0.000	0.000
A	A
___D B___	___D B___
C	C

Marine and Engine Gear Mounting

General Information

Preliminary

Proper mounting of the marine gear and propulsion engine in the vessel, once they have been aligned, is critical to maintaining good alignment and consequent smooth, quiet operation and so warrants close attention. This discussion describes the requirements and procedures for mounting marine gears and Caterpillar Engines to the ships foundation and propulsion driveline.

As an engine manufacturer, we can identify the requirements for proper mounting and alignment of the Caterpillar product, however, the responsibility for proper total mounting and alignment always rests with the equipment installer. Primary objectives are:

Mount the *marine gear* so that–

Full propeller thrust can be transmitted to ship structure (except where thrust bearing is separate from marine gear).

Transmitted thrust or other external forces do not adversely affect gear alignment to either the propeller shafting or the engine.

The forces it exerts on its foundation cause no damage.

The *engine* must be mounted so that it is not prestressed.

Movements of the hull cannot reach the engine cylinder block and crankshaft.

Driveline thrust forces are not allowed to reach the crankshaft.

Its natural thermal growth and shrinkage is not restrained.

The forces it exerts on its foundation cause no damage.

Rigid Mounting or *Resilient Mounting* may be used.

Foundations

The marine gear/engine foundation is that portion of the boat's structure which supports the propulsion machinery and holds it in proper relationship to the driveline components. It generally consists of two longitudinal rails–with liberal transverse bracing–which carry the weight, thrust, torque reaction and inertial loads of the gear/engine. It is good design practice to make the foundation members as long as possible. This helps to limit hull deflection by distributing the loads over more of the hull length.

The entire foundation must be strong enough to withstand continued operational forces due to torque, thrust, pitching, rolling, and occasional grounding. Since no structure is absolutely rigid, it is essential that the foundation have greater rigidity than the driveline, so that none of the components of the driveline are stressed beyond their limits when flexing of the hull occurs. Foundation structures may be of metal (steel or aluminum), wood, or fiberglass depending usually on the vessels hull composition. In fiberglass vessel's, foundations will generally be of the *foam filled* type, or wood:

Foam filled foundations require a metal *raft* between the machinery and the fiberglass foundation to distribute machinery loads more evenly.

Wood foundations allow for relatively simple mounting, using lag screws, and does not normally require special load distributing techniques.

Mounting Types

Engine and marine gear mounting generally falls into one of two categories, i.e. *rigid* or *resilient.*

Rigid Mounting. The engine supports and necessary shims are fastened directly to the boat's structure. The shims, used for positioning the engine or gear in proper alignment, are either steel or poured plastic (refer to the section on shims presented later in this document). With no vibration mounts between the engine supports and the boats structure, flexibility must be built into the engine supports to prevent the engine block from becoming stressed by motions of the hull. It is the simplest and least costly way to mount an engine.

Rigid mounted machinery is generally bolted to the engine foundation.

Resilient Mounting of machinery is usually done for isolation of noise and vibration from the ship structures. It is more expensive and requires more attention to detail than rigid mounting. Flexible fittings must be used for all connections (combustion air, coolant, fuel, exhaust gas, controls, etc.) when resilient mounts are used. A section on resilient mounts is presented later in this document.

Mounting Procedures

Note: In the procedures that follow, it is assumed the alignment processes, i.e., marine gear to shafting and engine to marine gear have been accomplished per procedures outlined in previous sections.

Marine Gear Mounting

For rigid mounting of free standing (separate from engine) marine gears proceed as follows:

1. Make a final check of the marine gear support structures for adequate size, strength, shim space, and sufficient clearances for mounting bolts. If the marine gear mounting brackets are of the bolt on type, make sure they are properly secured against further movement relative to the gear case (dowels in place, etc.).

2. If metal shims (chocks) will be used, machine and fit the shims to each side of the marine gear. Refer to the procedure outlined in the appendix for shim fitting and mounting.

   Note: In exception to the caution listed in appendix, it is permissible to use peel type shim packs, or a few loose shims in conjunction with a thick steel chock. In both cases, stainless steel is preferred. This exception is due to the relative lower linear vibration and greater stability of the marine gear as compared to the engine.

3. Taking care not to move or disturb the gear to shaft alignment, drill and ream the holes for the fitted foundation bolts. A minimum of two fitted bolts should be used. They should be installed on each side and directly (not diagonally) opposite each other. For some marine gears it is permissible, but seldom necessary, to use fitted bolts in the remaining bolt hole locations. The thrust capacity of the bolted marine gear foundation results from: (a) the fitted bolts *shear strength* and, (b) the *clamping force* of all the foundation bolts together.

4. Install all fitted and clearance foundation bolts loosely with the threaded ends up. It is best to have sufficient thread showing for installation of double nuts.

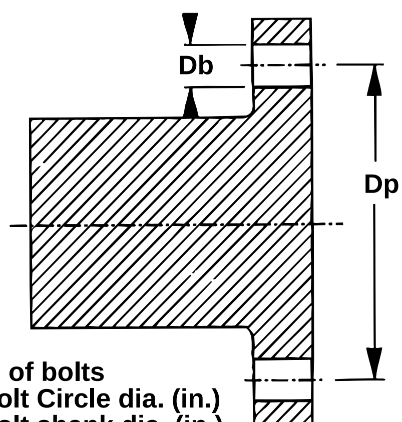
5. If poured *resin* shims are to be used, refer to procedures outlined in the appendix, and to the resin manufacturer's instructions. The basic steps are:
   (a) Prepare for the pour, i.e. dams around pads, spacers around clearance bolts, etc.
   (b) Recheck the gear to shaft alignment and reposition the gear as required to assure proper alignment. (c) Make the pour. Allow the poured shims to cure per manufacturers instructions and tighten the hold down nuts to the torque calculated per appendix, so that the recommended pressure on each chock will not be exceeded. (The pressure applied is the sum of the gear weight and bolt loads.)

6. If metal shims are being used, tighten the hold down bolt nuts per Figure 2.24 in the appendix.

7. Make a final alignment check and, if satisfactory, install the double nuts. Mark the nuts at the thread with a daub of paint for easier periodic visual checks of the bolt connections.

8. Draw the marine gear output flange and propeller companion flange connection tight with the connecting bolts. The connecting bolts are either fitted or clearance type. The fitted bolt connections carry the transmitted torque primarily by bolt shear strength, and by some degree of clamping force. As a rule of thumb, fitted bolts are usually required on propeller flange connections transmitting torque of 0.75 kW/rpm (1 hp/rpm) or greater. Below that torque level bolt load clamping force is often adequate to carry the transmitted torque with a high degree of safety, especially if grade 8 bolts at high tightening torque are used.

   If non fitted bolts are used, grade 8 bolts torqued to the standard *high* torque values should be used since full output torque will be carried by the bolt clamping force. The nominal s*tandard torque* and *high torque* values for 3/8 in. to 1-1/2 in. bolts, along with the resulting bolt load are given in Figure 2.18, (high torque values in bold print). Using these loads and the formula given in Figure 2.17, the torque that can be safely transmitted through the flange connection can be calculated.



N = no. of bolts
Dp = Bolt Circle dia. (in.)
Db = Bolt shank dia. (in.)
To = Torque (hp/rpm)
Bp = Bolt load (lb)

Allowable Transmitted Torque (non fitted bolts):

$$To = \frac{Dp \times N \times Bp}{2{,}800{,}000} \text{ (hp/rpm)}$$

Figure 2.17

Bolt Size	Torque (lb/ft)	Bolt Load "Bp" lb
.375 - 16 & 24	32 **40**	5,200 **8,200**
.738 - 14 & 20	50 **65**	7,100 **11,400**
.500 - 13 & 20	75 **100**	9,200 **15,400**
.563 - 12 & 18	110 **145**	12,000 **20,000**
.625 - 11 & 18	150 **200**	14,800 **25,000**
.750 - 10 & 16	265 **350**	22,000 **37,000**
.875 - 9 & 14	420 **550**	30,000 **49,500**
1.000 - 8 & 14	640 **825**	40,000 **65,000**
1.125 - 7 & 12	800 **1,000**	44,400 **71,000**
1.250 - 7 & 12	1,000 **1,350**	50,000 **87,000**
1.375 - 6 & 12	1,200 **1,700**	55,000 **100,000**
1.500 - 6 & 12	1,500 **2,000**	63,500 **108,000**

Figure 2.18

   If fitted bolts are used, drill and ream the mating bolt holes (if this has not already been done) and install the fitted bolts. Torque the fitted bolts to the standard torque values for the bolt size or to the suppliers' recommended torque if specified.

9. Install thrust or collision blocks if required. Install the blocks on both sides of the marine gear with sufficient clearance for thermal expansion of the gear case. The expansion that needs to be allowed for occurs over the distance from the collision block to the first fitted bolt connection. A clearance of 0.0008 in./in. of that distance should be provided for engine jacket water cooled gears. Provide clearance of 0.0006 in./in. of that distance for sea water or keel cooled gears.

Soft *Resilient* Mounting of free standing marine gears

Soft mounting of free standing marine gears is infrequent and done only in very special cases as a rule. Due to the various cautions and complexities involved in such installations, the procedure will not be addressed here in detail but the following features are common in these installations:

Remote mounted thrust bearing to isolate the marine gear from the thrust force component.

When the above is used, relative motion between the gear and thrust bearing have to be accommodated which, in turn, may require one or more of the following: a cardan shaft (with slip connection); a flexible drive coupling; or a sufficient length of unsupported shaft between the gear and thrust bearing to accommodate the movement.

Since the engine would also be soft mounted in these installations, special provision is required in the engine to marine gear connection to accommodate the relative movement between the engine and gear, which may be substantial.

One advantage to this type of installation is that resilient mounts may be selected specifically for either the engine or marine gear to tune out their particular sound/vibration frequencies. Due to the higher frequencies of the noise or vibration being isolated, the resilient mounts used for the marine gear will most likely be considerably more stiff than those used for the engine. The mounts will normally have stops incorporated in them to limit motion.

Combined Engine/Marine Gear Mounting

Rigid Mounting of pre-assembled marine gear/engine units

These units, already coupled, have either:
(a) continuous rail type supports incorporated on either side (Figure 2.19a), or (b) a three point mounting support system (Figure 2.19b) incorporating a single point front trunnion mount.

When the marine gear and engine are connected at the bell and flywheel housings and mounted on rails, (Figure 2.19a), the package is very strong and can be considered as one unit. Its longitudinal stiffness and length make it very practical for soft mounting if the propeller shaft connection and relative movement are properly accommodated. A fitted bolt, or bolts, is used only at the rear location as indicated in the sketch below.

Further installation procedures for this arrangement are as outlined in the *engine mounting* section which follows this section.

Marine gear and propeller companion flange connections in both of the above arrangements are as covered previously.

If the marine gear and engine are connected but not on common rails, as shown in Figure 2.19b, the marine gear is mounted as previously outlined. The front mount, in this case, is usually a trunnion type. The trunnion

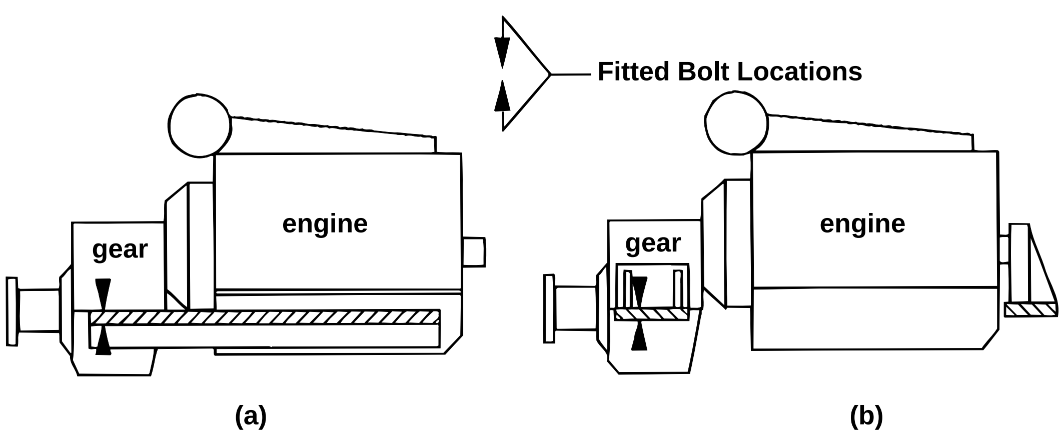


Figure 2.19

mount is attached to both ship rails but has a single swivel connection at the front of the engine with the swivel center on the crankshaft centerline. It often has some degree of fore/aft flexibility as well as a fair amount of rotational freedom. Clearance bolts are normally used for the front mounts. This is especially so if the front mount is rigid (not trunnion). In either case, clearance or flexibility of about 0.0008 in./in. of distance, from the fitted bolt or dowel at the marine gear mount, must be provided for thermal expansion of the engine and gear.

Resilient Mounting of pre-assembled marine gear/engine units

A common method of sound/vibration isolation is to have the gear and engine combined by flywheel housing connection and/or common rails (Figure 2.19) and the whole system then isolation mounted.

Resilient mounts under relatively small, close coupled engine/marine gear packages, such as pictured in Figure 2.19b, are commonly used on planing hull type vessels and performance craft.

However, as the size and power of engines increases, it becomes impractical to use very soft resilient mounts directly under the marine gear. This is because the flexibility of more than 50 mm (2 in.) propeller shafts is not adequate to accept large engine/marine gear motions [6 to 24 mm (1/4 to 1 in.)] without significant likelihood of damage to the stuffing box or the shaft bearings. It is preferable to mount large engines on resilient mounts and mount the marine gear rigidly to the structure of the boat. (This is a rule of thumb. Special shafting arrangements, such as cardan shaft with axial spline, or flexibility mounted stuffing boxes will allow tempering of this guideline.)

When making this type of installation final positioning of the gear and engine is often done by adjustments of the resilient mounts themselves. In any case, follow instructions provided by the mount manufacturer .

In resilient mount installations of gear and engine combined on common rails (Figure 2.19a), the following basic procedures apply:

- The marine gear must be doweled or attached with body fitted bolts to the common rails.
- The engine, *if not directly connected to the marine gear at the flywheel housing,* must be aligned to the marine gear and doweled, or fitted with a body fit bolt, at the left or right rear of the engine. ***Note:*** This step may be done toward the end of the installation process.
- The common rails are then positioned on the resilient mounts to be used and the package is aligned to the propeller shaft companion flange per previously outlined procedures. Appropriate shims or chocks are installed as required.
- Make sure propeller thrust or other externally applied forces are accommodated in the mounting system and limited by stops or other devices. (Refer to later sections in this document regarding resilient mounts and shim types.)

Engine Mounting

Three Point Mounting of Engines

Three point mounting systems are normally associated with combined engine/marine gear units. There are, however, some instances in which separately mounted engines will require the three point system such as in high performance patrol craft where weight reduction and system flexibility are premium factors.

The three point system for the engine normally involves mounts on both sides at the rear of the engine at the flywheel housing plus the trunnion mount at the front. This arrangement is very tolerant of flexing of the ships mounting rails which may be encountered in light high speed craft. Hard mounts are used most often but resilient mounts are not uncommon with this arrangement.

Mounting 3500 Family Engines equipped with Mounting Rails

When mounting engines in a vessel the effects of external and thermally induced stresses to the engine must be considered. This is a most important step in any quality installation.

Ships hulls will flex under the internal stresses of varying displacement and the external stresses of wind, water, and temperature. If the engine is too rigidly mounted to the ship's structure, or if it is restrained from its natural thermal growth, excessive stress may reach the blocks' internal support structure. This could in turn result in distortion of main bearing bores, bore alignment, etc. Severe engine damage or significant reduction of engine life could result.

On the 3500 family of engines, as with other Caterpillar engines, the main structural strength is the cast iron block. The plate steel oil pan which supports the engine is a deep, heavy weldment. Lugs or brackets are welded to the sides of the oil pan for attaching the standard mounting rails to the engine. These rails, when properly mounted to the ship rails, provide the flexibility required to isolate the engine from the hull. The holes in the mounting rails are located so that the rails are allowed to flex, isolating the ship's deflection from the engine.

These rails are also flexible enough to accommodate thermal growth from side to side in most cases but provision for thermal growth front to rear must be allowed for. *Under no circumstances should ground body anchor bolts be used forward of the engine's flywheel housing.*

Note: The steel oil pan to which the mounting rails are attached expands at the rate of about 0.0000063 in./in. for each Fahrenheit degree temperature change (or 0.0000113 mm per Celsius degree).

Example: A 3516 oil pan experiences a rise in temperature from 65°F to 205°F, or 140 F degrees The distance from the fitted bolt at the rear of the rail to the forward most clearance bolt is 80 inches. The expansion, or thermal shift at the forward clearance bolt, is then:

80 x 140 x 0.0000063 = 0.071 in.

The following basic steps may be followed in mounting 3500 family engines with factory supplied mounting rails:

1. Preliminary . . . Marine gear in place, aligned to propeller shaft, and secured to foundation. (Reference gear alignment procedure document and gear mounting procedure in previous section of this document.)

2. Prepare engine bed (foundations) for poured or metal chocks. Foundation pads for metal chocks should be flat, preferably to within plus or minus 0.005 inches flatness tolerance. Foundation pads for poured chocks are less critical for flatness but should be placed for best chock thickness, i.e. 12.7 mm to 44.4 mm (.5 to 1.75 in.)

3. Place the engine on the foundation, supported by the jacking screws in the rails, and at final aligned position by sight. Install side to side and fore to aft positioning screws or devices per Figure 2.20. Align the engine to the gear per instructions in the previous document, *Installation/Alignment Instructions*.

4. *If steel chocks are used, a solid chock is recommended.* Fit and install the metal shims at each of the four rail mounting pads according to the steel shim fitting and installation procedures outlined in the appendix.

   It is not necessary to run the steel shim the full length and width of the machined pads incorporated with the 3500 family engine rails. Smaller steel chocks are easier to machine and fit. The shim's area can be considerably less as shown in the example of Figure 2.21, but, they should encompass the two retaining bolts, which are 152 mm (6 in.) apart, and cover at least 19 of the 101 mm (.75 of the 4 in.) pad width. It is also a good rule of thumb to keep the applied unit pressure on soft steel shims under 34,475 kPa (5000 psi).

5. *If poured chocks are used* refer to the appendix for recommended installation procedures for poured chocks.

Each machined mounting pad on the 3500 family engine rails is about 589 cm^2 (91.4 in^2) in area, minus bolt and jacking screw area, and it is important to utilize all of the pad area when mounting on epoxy resin shims. This is especially so for 3516 engines.

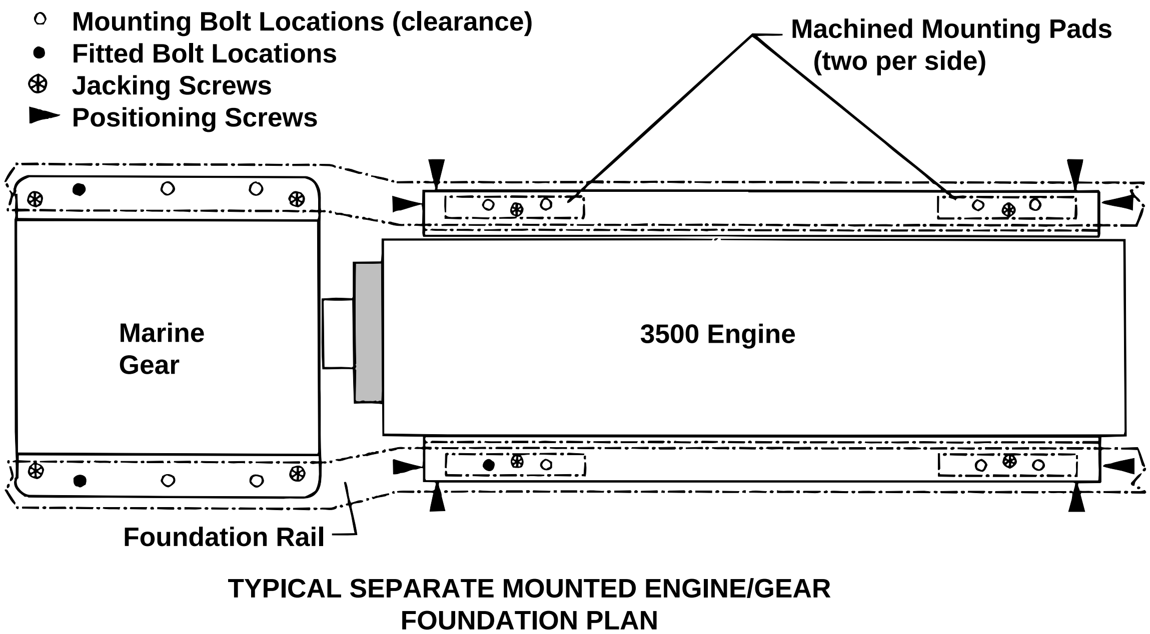


Figure 2.20

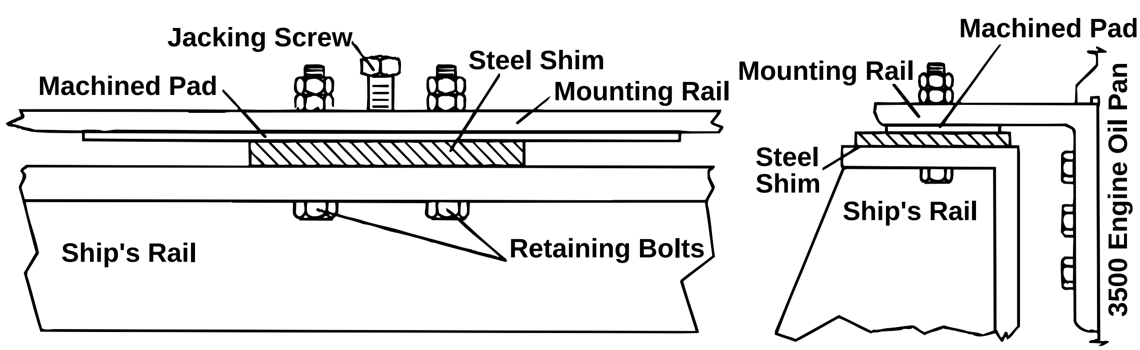
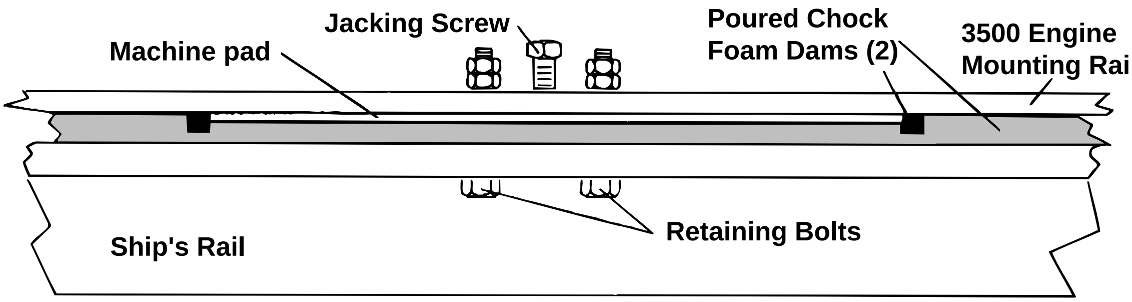


Figure 2.21



EXAMPLE OF CONTINUOUS POUR

Figure 2.22

The shim material can be poured at the mounting pad locations only (an interrupted pour), or it can be used under the full length of the engine rail, *except for immediately fore and aft of the machined mounting pads.*

This is referred to as a *continuous* pour ***Caution:*** (Figure 2.22). In either case, do not pour shim material inboard of the machined pad on the bottom of the mounting rail. Foam rubber strips must be installed on sides and both ends of each pad to provide for expansion. (Figures. 2.22 and 2.23).

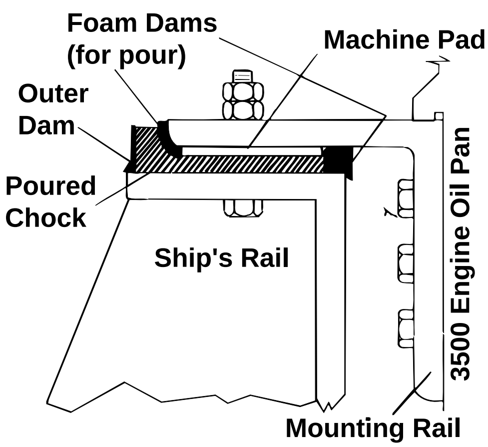


Figure 2.23

Mounting Bolt Torques for use with Poured Shims

After the shim material has sufficiently hardened according to the manufacturer's specification, tighten 22.2, 25.4, and 28.6 mm (7/8, 1 and 1-1/8 in.) mounting bolts to a torque of 490 N•m (360 lb ft). This bolt torque is specified to prevent excessive unit pressure on the poured shims.

Use two nuts on each mounting bolt. Tighten the innermost nut to 490 N•m (360 lb ft.) Use Figure 2.24. in the appendix for torques of the outermost nut.

Resilient mounting of 3500 Family Engines equipped with Mounting Rails

This procedure will be addressed here only in general terms due to the wide variation of resilient mounting systems available. It is very important that the resilient mount manufacturer's directions for installation and operational limits be adhered to. Additional general information regarding soft mounts is included in the appendix.

The following factors are common to most or all resilient engine mount installations, and, should be carefully considered along with the manufacturer's recommendations.

The resilient mounts are to be installed between the mounting pads of the rails and the ship's foundation. (Engine and gear on common rails was covered in an earlier section).

One or both of the rear mounts, nearest the flywheel housing, should be positively located to fix the engine relative to the marine gear. Positive location (doweling) of the engine mounting rail to these mounts will depend on manufacturer's instructions.

Positive stops for excessive vertical or horizontal movement must be provided with or incorporated in the resilient mounts. These are required to limit relative engine motion due to inherent engine motion, misfiring, or vessel pitching and rolling.

Collision blocks or stops will be required, in case of collisions or grounding.

The flywheel mounted flexible coupling must be selected with care. It must routinely tolerate the ranges of side to side and fore/aft motions predicted for the given set of soft mounts.

Similarly, all water, air, exhaust, hydraulic, control, and fuel system connections must be flexible enough to accommodate repeated, day to day movement without failure.

The foregoing applies to all soft mounted engine systems. If the marine gear, which is separately mounted, is also on soft mounts, the prior section on soft mounted gears should also be reviewed for additional factors of concern.

Appendix

Shims, Spacers, Chocks:

General

Shims, spacers, or chocks must be provided to fit the individual locations between the top surface of the supporting members and the bottom surface of the mounting pads of marine engines and gears. Marine Classification Society (MCS) rules often dictate the type of shim to be used. The two most common and effective types will be addressed here. They are: (1) poured epoxy (plastic) shims, and (2) solid steel shims.

Mounting With Poured Shims:
Poured shims, or chocks, have become the shim of choice today in most mounting systems for marine propulsion equipment. The advantages of poured shims over metal are several:

- The time consuming (and costly) process of selective machining and hand fitting of solid steel shims is avoided.
- The surface condition or flatness of mating foundation and support planes is less critical, eliminating further machining operations in many cases.
- The plastic chocking material offers some degree of noise damping between the engine or gear and the foundation member.
- Zero shrinkage of the poured shim material, when mixed and applied properly, allows the most precise retention of position and alignment when the mounting procedure has been properly completed and the retaining bolts secured.

Before installing poured shims, consult the manufacturer’s instructions for using the shim material. Strictly adhere to such critical items as mixing ratios, cure times, temperature effects, allowable thickness, and maximum unit loading. The basic steps in installing poured epoxy shims are as follows:

1. Machine to be mounted is in final aligned position with mating mounting structures in proper place and condition. These mating pads should be of sufficient strength and area, reasonably flat and parallel.
2. Liquid shim materials, damming materials, blowers, grease) releasing agents, tools, etc. at hand per manufacturers recommendation.
3. Drill clearance holes through at all clearance bolt locations.

   Note: It is recommended that holes for the fitted bolts be drilled and reamed after: the chocks have cured; the clearance bolts have been torqued; and alignment rechecked (in case the pour and alignment might have to be repeated).
4. Install the foundation hold-down bolts threaded end up and with nuts hand tight. Sleeve or foam wrap the bolt shanks to provide clearance for thermal expansion of the machinery. An optional method is to grease the bolt shanks with high temperature (non melt) grease, then, remove the bolts and redrill the bolt holes for adequate clearance after the chock has cured.

5. Spray all chock contact surfaces with a suitable releasing agent to prevent adhesion of the chock to those surfaces.

6. Install damming material in preparation for the pour. In this operation, consider the following:

   a. Ship's trim . . . pour from the high end or from the side.
   b. On non pour sides of the chock, dams of non porous foam material is recommended. Use foam tape around edges of mounting pads to provide for pad movement during thermal expansion of the machine or flexing of the foundation members.
   c. Use rigid or semi rigid damming material around the pour area. Width of the pour area generally should not exceed 20 mm (.75 in.), and should provide a riser of at least 12 mm (.5 in.).

7. Mix the resin and hardener for the chocks and make the pour per the resin manufacturers instructions. Be sure to allow sufficient time for the chocking material to cure.

8. After making sure of sufficient clearance around clearance bolt shanks, torque all the clearance bolt nuts to obtain *total* bolt loading on the chocks equal to 2-1/2 times the weight of the mounted machinery but within the unit pressure limits for the individual chocks as set by the resin manufacturer. Unit pressure limits on the poured chock are usually 3447 kPa (500 psi) when maintenance of machinery alignment is required, (as in the case of propulsion engines and gears). To aid in determining the proper torque range the following formulas may be used in calculating bolt loads:

$$\text{Bolt Load (lb)} = \frac{60 \times \text{Bolt Torque (lb ft)}}{\text{Bolt Dia. (in.)}}$$

$$\text{Bolt Load (kg)} = \frac{500 \times \text{Bolt Torque (kg m)}}{\text{Bolt Dia. (mm)}}$$

   Unit load on a given chock is obtained by first summing the weight from the machine plus any machine torque load component plus the total bolt load, all of which is then divided by the chock area.

9. After the clearance bolts have been tightened, recheck for proper alignment or position of the machine. (Keep in mind that some slight shift is normal due to thermal effects on the ship's hull, instrument variations and so on.)

10. If positioning is still satisfactory, install the fitted bolt or bolts.

11. Install the second set of nuts and torque these against the first nuts per Figure 2.24. Be careful that the first, or primary, nut is not torqued to a higher level than that determined in step 8. Mark the nuts with a paint spot to aid in future pass by inspections for loose nuts.

12. Remove damming materials. Clean, dress, or chip away excess chock, where necessary, for appearance and to relieve any unintended restraint of the mounting pad.

Full Torque Values										
Dia (in.)	**.500**	**.563**	**.625**	**.750**	**.875**	**1.000**	**1.125**	**1.250**	**1.375**	**1.500**
Torque (lb ft)	75 ± 10	110 ±15	150 ± 20	265 ± 35	420 ± 60	640 ± 80	800 ± 100	1000 ± 120	1200 ± 150	1500 ± 200

Figure 2.24

Mounting With Steel Shims

Ideally, steel shims, or chocks, are one piece and are made to fit between the top of the ship's rail or machinery foundation and the bottom of the corresponding mounting pad of the engine or marine gear mounting rail or bracket. Mild steel plates are normally used and are surface-machined to specific dimensions at each corner of the plate as determined when the engine or gear has been placed in the final aligned position. The shims can be numbered to avoid confusion during installation. The machining of these shims must provide a uniform fit between the respective machine rails or brackets and the foundation pads.

To fit and install steel chocks the following procedures are suggested:

1. Foundation pads for metal chocks should be flat, preferably to within 0.005 inch flatness tolerance.

2. With the machinery to be mounted in its proper aligned position, measure the vertical gap between the pad and foundation at each corner of the shim area. If both the mounting pad and foundation pad are flat, the four corner gap dimensions will dictate the corner dimensions of the surface machined, finished shim.

3. To facilitate ease of fitting, the chock can be slotted to fit around jacking screws.

4. Check the proper fit of each chock by use of blueing dye, carbon paper, strip gauge (plastigage), or feeler gauges. This is a judgment operation by the fitter/installer. The objective is good even contact, (40% or more), over the major length and width of the mounting faces. Do selective grinding or machining as required to obtain the proper fit.

5. When all of the chocks have been fitted, drill clearance holes for the retaining bolts, making sure of sufficient clearance for thermal growth at all but the fitted bolt locations.

6. Install the bolts from the bottom up and draw the nuts down moderately tight.

7. Recheck alignment of the machine. If the alignment is still satisfactory, torque the nuts on all clearance bolts per torque values as listed in Figure 2.24. Drill and ream for the fitted bolt, or bolts, and install. Torque the nut on the fitted bolt also per Figure 2.24.

8. Install lock nuts on all bolts, torque, and mark with a spot of paint for future visual checks.

Miscellany

Warning Against Lead Shims — Do not use lead metal. Lead is easily deformed under weight and vibration and has poor supporting characteristics.

Warning Against Multiple Piece, Sheet Metal Shims — Using hand cut, sheet metal shims, is discouraged. The edge deformation of the sheet metal shims, caused by the use of hand sheet metal cutters (tin snips) will prevent a stack of such shims from lying flat and will eventually allow an engine, thus shimmed, to drop out of alignment, as its shims relax. Also repeated, small, relative movement between shims, along with inherent engine vibration may cause the shims to *beat out,* especially with 3500 family engines and larger.

Mounting Bolts — There are two types of mounting bolts:

Clearance bolts are nominally 1.5 mm (0.06 in.) diameter smaller than the holes in which they are installed. Clearance bolts are used to insure the engine does not move vertically on its foundation.

Fitted bolts have a tight fit in the holes in which they are installed. Fitted bolts, sometimes called ground body bolts, are used to insure the engine does not move around horizontally on its foundation.

Install both types of mounting bolts with the head down and the threaded end up for ease of routine periodic inspection of the bolted joints.

One fitted bolt must be used at the rear of the engine to maintain position and alignment. **Never use fitted bolts forward of the engines flywheel housing**. Use clearance bolts in all locations forward of the flywheel housing.

Resilient (Soft) Mounts

General

Resilient mounts are used to reduce the transmission of noise and/or vibration to the hull and various compartments of the vessel. This can have considerable effect on crew and passenger comfort, reduced crew fatigue, and consequent increased efficiency. It can even affect, to some extent, increased life of equipment and machinery sensitive to hull borne vibration. However, in installing resilient mounting systems, the following factors must be considered:

Motion Limit Devices

Any resilient mounting system must include some means of limiting the engine motion. Regardless of the type of mount, some means of limiting the overall motion of the engine must be present to prevent breaking the engines cooling and exhaust piping connections during bad weather or after collision/grounding accidents (when the engine might be subject to greater-than-normal inertia forces and motions).

Softness Versus Frequency

All noise/vibration has a frequency.

High frequencies*, such as those produced by turbochargers, gearing, and some hydraulic systems, can be isolated by small amounts of resilience.**

Lower frequencies, such as those produced by the firing of individual cylinders of the engine or driveline unbalance, require much higher amounts of resilience to achieve the best reduction in transmitted engine vibration.

Types of Vibration Mounts

Vibration mounts can be subdivided by the material of the resilient component. Spring mounts which use helical metal springs, rubber mounts which use rubber, either in shear or compression, and Combined Spring and Rubber Mounts-which use both methods of achieving resilience. Some characteristics of these mounts are:

Spring Mounts generally, will achieve the highest amount of resilience. They are also generally more costly. Even with spring mounts, it is good practice to put some rubber in the mounting system. Mounts which use only metal components can still transmit significant vibration/noise in the high frequencies.

Rubber Mounts are excellent at isolating the higher frequencies of engine vibration/noise—often better than the spring type. Rubber mounts are generally cost effective, as well. Rubber mounts require periodic inspection for hardening/cracking of the rubber elements.

Combined Spring and Rubber Type
Vibration mounts combine the best of both types.

Effects of Propeller Thrust on Resilient Mounts

Most inexpensive vibration mounts are designed to accept forces in only one direction, up-and-down. Propeller thrust forces are generally from the side. Make sure the vibration mounts chosen are suitable for the forces to which they will be subjected. Install vibration mounts so the fore-and-aft thrust forces acting on the mount are resisted with the least possible extension of the mount.

Need for Periodic Realignment

Vibration mounts can also settle or *take a set*—which can necessitate occasional realignment of the driveline. Annually check alignment of engines mounted on vibration mounts for misalignment.

* In the audible range of human hearing

** *Amount of Resilience* can be described in terms of the motions of the engine. Engines whose mounting systems permit only small motions—on the order of .5 mm. (0.020 in.) or less can be said to have only a *small amount of resilience.*

Miscellaneous Considerations

Collision Blocks

When marine classification societies or local marine practice requires the use of collision blocks, they should be located with sufficient clearance to allow for thermal growth of the engine. Prefabricate the collision blocks and install them while the engine is at operating temperature with approximately 0.12 mm (0.005 in.) hot clearance. Collision blocks are recommended to resist the shock loads encountered in hard docking collisions and groundings.

Caution: Any electric arc welding of scantlings (ships structural components) or engine support structures for engine or marine gear requires precaution against electrical grounding through engine or marine gear housings. *Under no circumstances is any welding of an engine or marine gear to its foundation* to be done, otherwise, failure within the first few hours of operation is likely to occur.

Crankshaft Deflection Test

General Procedure

To assure the engine block is not unduly stressed during mounting, a crankshaft deflection test is recommended. This test should be performed on 3500 family engines. Marine applications require this test be conducted under hot conditions. To accomplish this test, proceed as follows:

1. Remove an inspection door from the block to expose the center crankshaft throw.

   Rotate the crankshaft in the normal rotation direction. When the cheeks of the center throw just pass the connecting rods, install a Starrett No. 696 distortion dial indicator or similar tool. As a precaution, tie a string to the gauge and secure it outside the engine to facilitate retrieval should the assembly fall into the oil pan.

   Zero the dial indicator's rotating bezel. Properly seat the indicator by rotating it on its own axis until it will hold a zero reading.

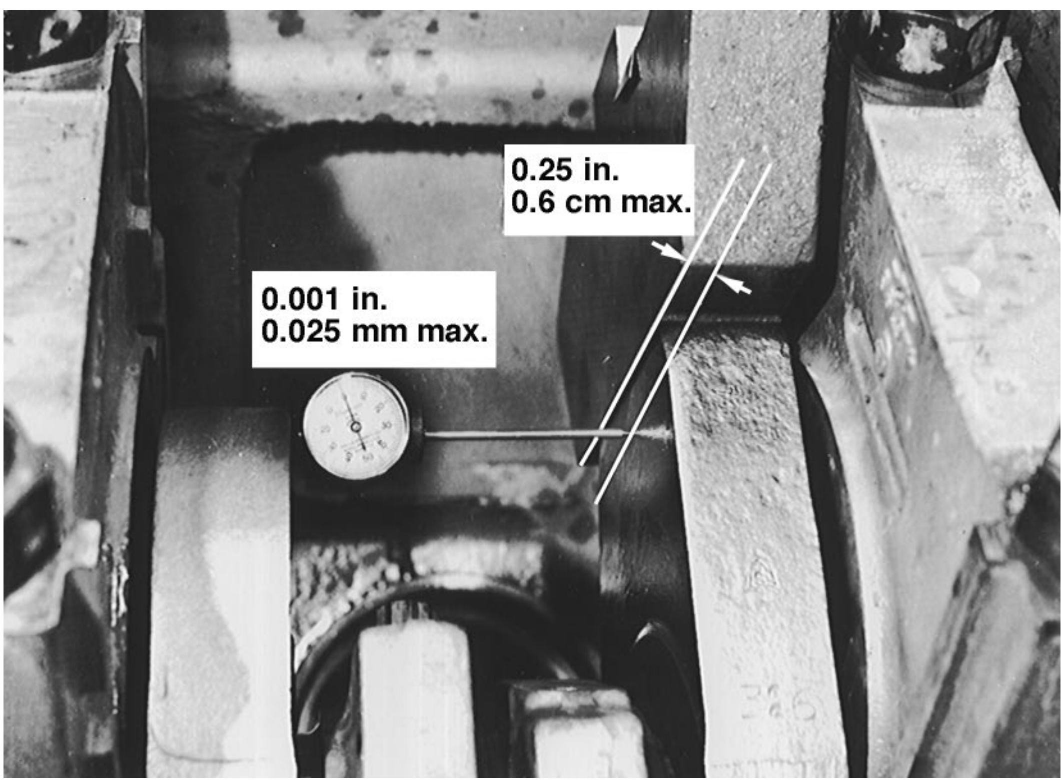


Fig 2.25 CRANKSHAFT DEFLECTION TEST

2. With the indicator still set at zero, rotate the crankshaft in the normal direction until the indicator nearly touches the connecting rods on the other side of the crankshaft. (Do not allow the indicator to touch the connecting rod.)

   The dial indicator reading must not vary more than 0.3 mm (0.001 in.) throughout the approximately 300 degrees of crankshaft rotation.

   Rotate the crankshaft back to its original position in the opposite rotation direction. The indicator must return to its original reading of zero to make a valid test. If not, the indicator shaft points were not properly seated and the test procedure must be repeated.

3. If the gauge reads more than 0.03 mm (0.001 in.), cylinder block distortion has occurred due to improper mounting.

   Loosen the hold-down bolts between the engine rails and mounting blocks. Check carefully for loose shims, improper locations of fitted bolts, interference from clearance bolts, or any other constraints to proper engine block movement.

   Make any needed adjustments and secure the hold-down bolts, making sure alignment of the engine has not been disturbed.

4. Repeat the distortion check procedure. Consult your Caterpillar dealer if the engine block is still bent.

Auxiliary

Bases
Alignment
Mounting Auxiliary Engines

This section is concerned with the mounting and alignment of auxiliary engines.

Auxiliary engines are power packages used to provide onboard power to drive generators, pumps, compressors, winches, etc. The engine driven equipment (load) is either directly mounted to the engine (close coupled) or is remote mounted from the engine and driven through a shaft and coupling. The major application of auxiliary engines is to provide shipboard electrical power. The following discussions, which refer primarily to engines driving generators, also apply to other types of auxiliary power packages.

The Caterpillar diesel auxiliary engine is built as a rigid, self-supporting structure within itself. If the engine is mounted on a foundation which is true (flat) or on a pair of longitudinal beams, the tops of which are in the same plane, the engine will hold its own alignment. If subjected to external forces or restrained from its thermal growth by the mounting, affected tolerances may result in bearing or crankshaft failure.

The power module must maintain the original alignment under all operational and environmental conditions. Misalignment between an engine and driven equipment can cause vibration and shorten the life of coupling and bearings.

The major cause of misalignment is flexing of the mounting structure due to weakness. Other causes are poor installation methods and incorrect alignment procedures.

Bases

Base Design

The most important function of an engine base is rigidity. It must maintain alignment between the auxiliary engine and its driven equipment.

An engine base must:

- Protect the engine block, drive train couplings, and load (generator gear reducer, or pump) from bending forces during shipment.
- Limit torsional and bending moment forces caused by torque reaction and subbase flexing.
- Have a natural frequency such that resonance does not occur during the machinery's normal work.
- Make proper alignment easy. Allow sufficient space for shimming in the alignment process.

Ease of initial installation, vibration isolation, or need of isolating from a flexing mounting surface are major reasons for use of fabricated bases. *No base of any type should be rigidly connected to a flexing surface.*

The type of load will determine the design features required in an engine base:

- If the load is close coupled—such as a single bearing generator, the base is subjected to relatively light twisting loads. Its rigidity need only be moderate.
- But if the load is remote-mounted—such as a two bearing generator, the base is subjected to far greater twisting loads and its rigidity must be very great.

Single Bearing Loads

When single-bearing generators or close coupled loads are used, the base does not have to withstand torque reaction. Bolting the generator housing to the flywheel housing eliminates the need for the base to absorb the driving torque of the engine.

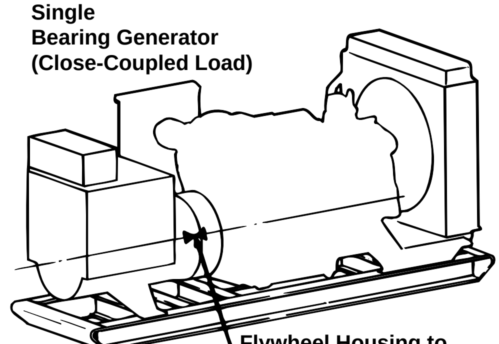


Figure 3.1

Two Bearing Loads

With the load remote-mounted, a more rigid structural base is required. The full load torque between the engine and load has to be absorbed by the base without causing excessive deflection in the coupling.

The stationary frame of the remote-mounted driven equipment tries to rotate in the same direction as the engine crankshaft. If the base were not rigid enough, engine torque would cause the base to flex excessively. The result is misalignment, proportional to the amount of load, which will not show up during a conventional static alignment check.

Severe cases of this problem result in bearing and coupling failures.

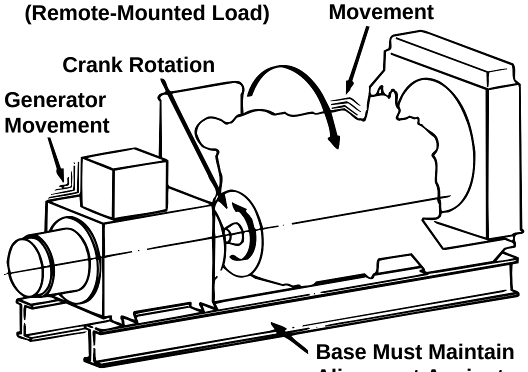


Figure 3.2

Bases for Engines with Close-Coupled Loads

Caterpillar does not recommend a specific section modulus for the longitudinal girders or cross members. Usually "I" beams or channel section steel beams in a ladder-type arrangement are acceptable.

Foot-Mounted Engines

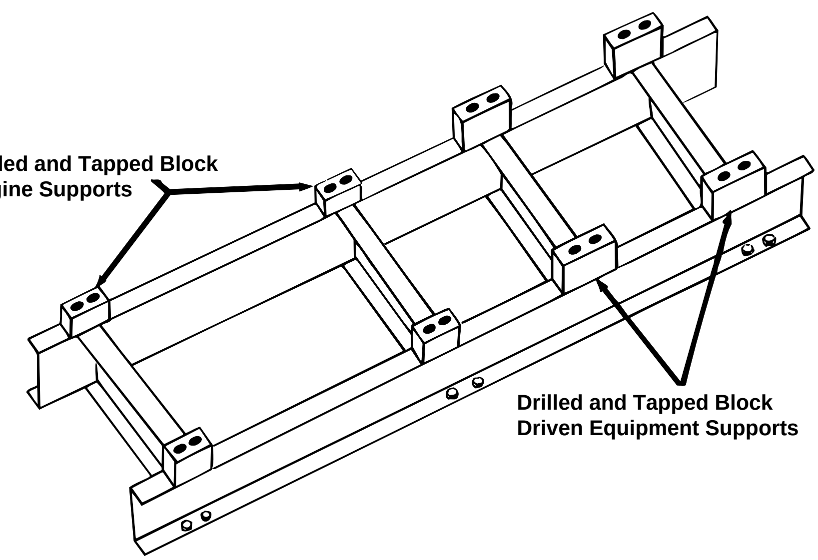
Base cross members must be as substantial as the longitudinal beams.

Place the cross members beneath each engine and generator support location.

Use drilled and threaded steel mounting blocks between the engine/driven equipment and the base. Bolt these blocks to the engine/driven equipment first and then weld to the base providing a flat surface for shimming and mounting. Mounting holes drilled into the structural members of the base are not recommended.

There should be sufficient space for shimming between the mounting blocks and the engine/driven equipment mounting surfaces.

Flexible mounts are not allowed between the mounting blocks and the engine/load mounting foot surfaces.



BASE FOR FOOT-MOUNTED ENGINE WITH CLOSE-COUPLED LOAD

Figure 3.3

Rail-Mounted Engines

In addition to the requirements for foot mounted engines, the following applies for rail-mounted engines:

- The standard engine-mounted support rails (engine length) must be used between the engine and the structural base.
- Locate cross members directly beneath the front and rear engine-to-rail mounting locations.
- Place threaded mounting blocks at the front and rear of the engine-mounted rails with space available for shimming. Bolt these blocks to the engine/driven equipment first and then weld to the base to provide a flat surface for shimming and mounting.
- Do not weld the engine-mounted rail to the structural base.
- Bolt the engine-mounted rails to the threaded mounting blocks through clearance holes to provide for thermal growth.

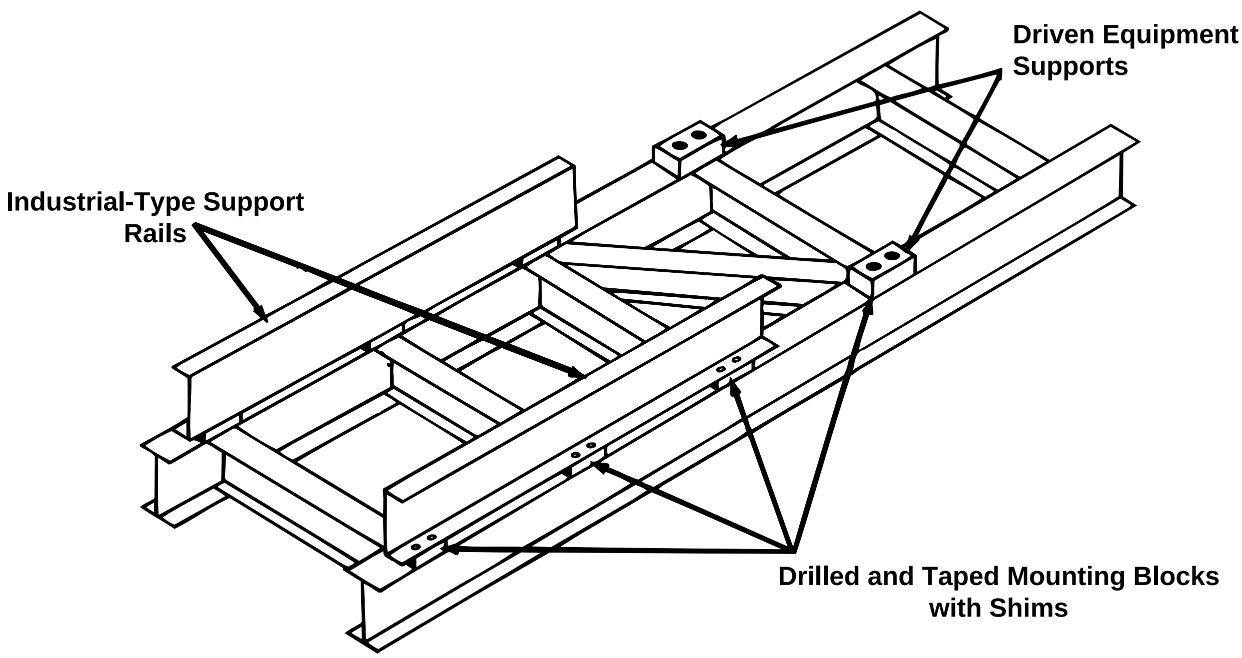
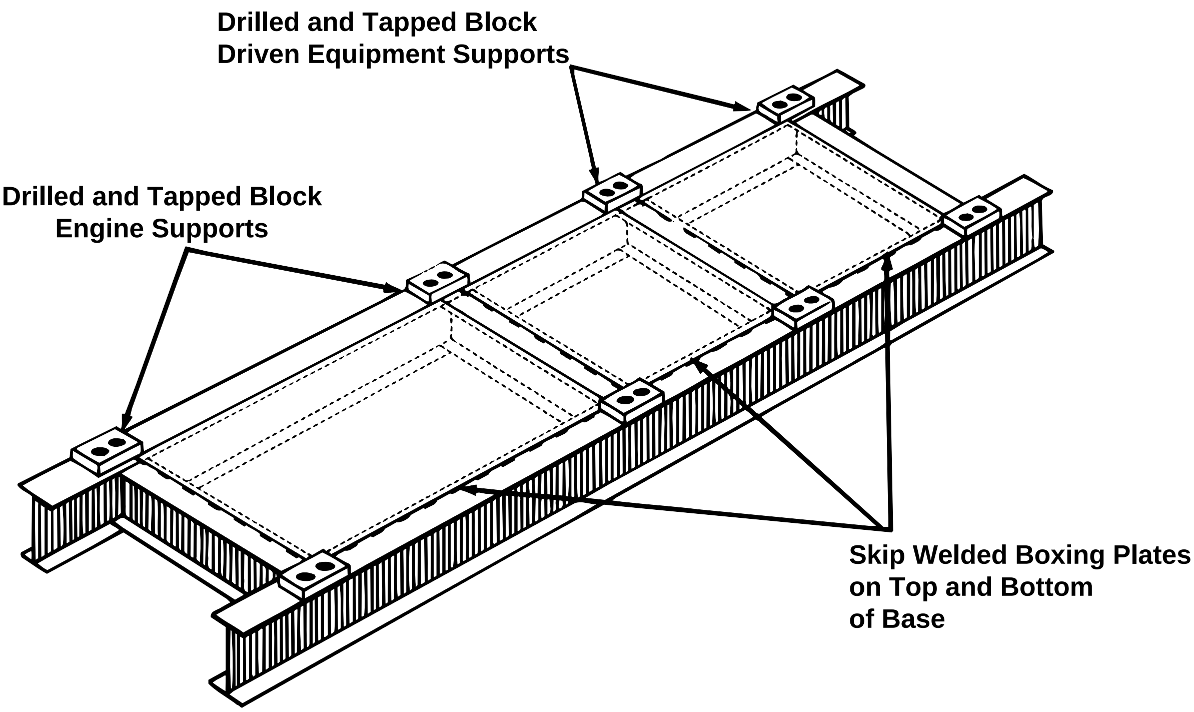


Figure 3.4 BASE FOR RAIL MOUNTED ENGINE WITH CLOSE COUPLED LOAD



BASE FOR ENGINES WITH REMOTE MOUNTED
DRIVEN EQUIPMENT

Figure 3.5

Bases for Engines with Remote-Mounted Loads

The requirements for close-coupled loads also apply to remote-mounted loads. With the load remote-mounted, a more rigid structural base is required. The full load torque between the engine and load has to be absorbed by the base without causing excessive deflection in the coupling.

The base shown above is a boxed beam design which provides a torsionally rigid base.

Boxing consists of welding steel plates on top and bottom surfaces of machinery base girders. The plates should be 5 to 7 mm (3/16 to 1/4 in.) thick. Skip-weld the plates to prevent excessive base distortion during welding. Boxing is done to make the base structure stiffer.

The additional stiffness is necessary to resist torque loads between the engine and remote-mounted driven equipment and to resist possible vibration loads. Vibration-induced base loads are difficult to predict. Experience has shown boxing is effective in preventing base cracking and misalignment.

Recommended Beam Height

The recommended heights of the longitudinal beams for the various engine generator sets are:

Engine Model	Beam Height	
	Metric mm	English in.
3304, 3306	200	8
3406, 3408	260	10
3412	300	12
3508	400	16
3512	450	18
3516	500	20

Alignment

In high speed applications, at normal operating temperatures and load, misalignment between the diesel engine and all mechanically driven equipment must be kept to a minimum. Many crankshaft and bearing failures can be traced to incorrect alignment of the drive systems. Misalignment at operating temperatures and under load will always result in vibration and/or stress loading.

Since there is no accurate and practical method for measuring alignment with the engine running at operating temperature and under load, all Caterpillar alignment procedures must be performed with the engine stopped and the engine and all driven equipment at ambient temperature.

For information on alignment principles and the use of dial indicators, please refer to the Mounting and Alignment section under propulsion engines in this manual.

Refer to the following Caterpillar Special Instruction for more detailed information and specific instructions on mounting and alignment procedures.

Form No.	Title
SEHS7654	Alignment - General Instructions
SEHS7259	Alignment of Single Bearing Generators
SEHS7073	Alignment of Two-Bearing Generators

Alignment of Remote-Mounted Driven Equipment

In order to achieve correct operating alignment, certain factors must be taken into consideration in determining cold alignment specifications.

Factors Affecting Alignment

The input shaft of remote-mounted equipment is always positioned higher than the engine crankshaft. This compensates for vertical thermal growth, flywheel sag, and main bearing oil film lift on crankshaft. These factors cause the relative positions of the crankshaft and load input shaft to shift between static and running conditions.

Bearing Clearances

The generator rotor shaft and engine crankshaft rotate in the center of their respective bearings, so their centerlines should coincide. Alignment is made under static conditions while the crankshaft is in the bottom of its bearings. This is not its position during operation. Firing pressures, centrifugal forces, and engine oil pressure all tend to lift the crankshaft and cause the flywheel to orbit around its true center. Generally, the driven equipment will have ball or roller bearings which do not change their rotational axis between static and running conditions.

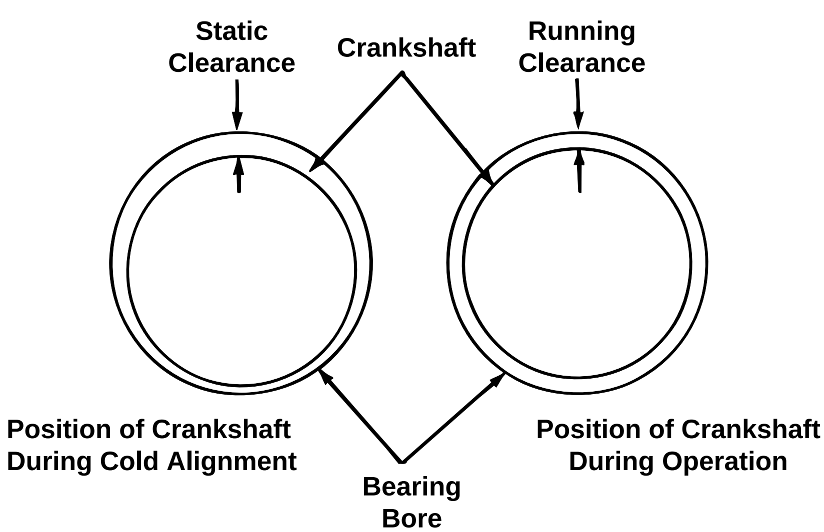


Figure 3.6

Flywheel Sag

With the engine not running, the weight of the overhanging flywheel and coupling causes the crankshaft to bend. This effect must be compensated for during alignment since it results in the pilot bore and outside diameter of the flywheel rotating lower than the true crankshaft bearing centerline during alignment. Caterpillar recommends alignment checks be performed with the coupling in place.

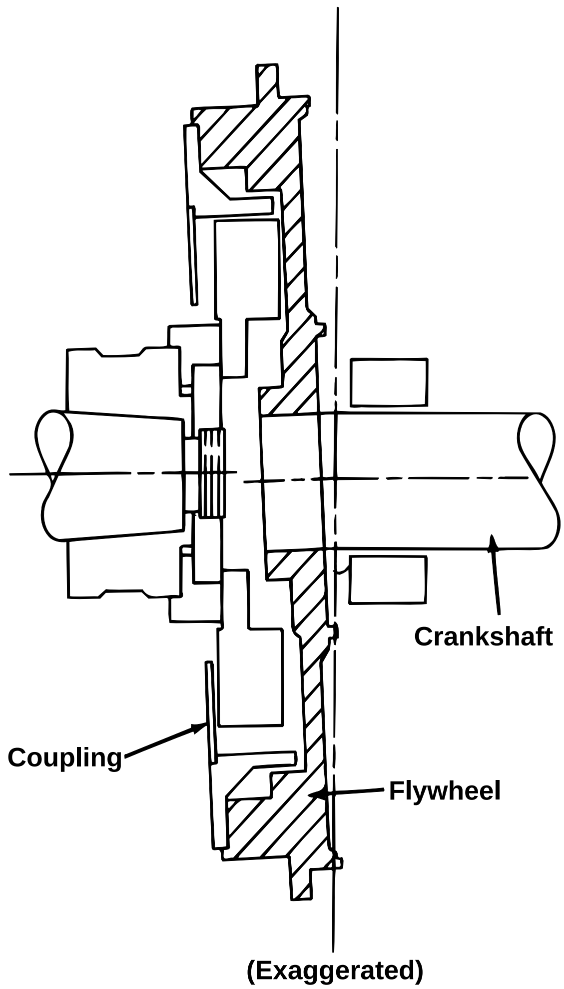


Figure 3.7

Torque Reaction

The tendency of the engine to twist in the opposite direction of shaft rotation and the tendency of the driven machine to turn in the direction of shaft rotation is torque reaction. It naturally increases with load and may cause a torque vibration. This type of vibration will not be noticeable at idle but will be felt with load. This usually is caused by a change in alignment due to insufficient base strength allowing excessive base deflection under torque reaction load. This has the effect of introducing a side to side centerline offset which disappears when the engine is idled (unloaded) or stopped.

Thermal Growth

As the engine and generator reach operating temperatures, expansion or thermal growth will occur. This growth is both vertical and horizontal. The vertical growth increases the vertical elevation between the component mounting feet and the respective centerlines of rotation. This thermal growth depends on the type of metals used, the temperature rise that occurs, and the vertical distance from the center of rotation to the mounting feet.

Crankshaft horizontal growth occurs at the opposite end of the engine from the thrust bearing. The location of thrust bearings on Caterpillar Engines is at the rear of the crankshaft. This growth has to be planned for when driven equipment is connected to the front end of the engine. The growth is slight if the driven equipment is bolted to the engine block, since the block and crankshaft grow at approximately the same rate. An example of this would be a front power takeoff clutch.

End Clearance

Horizontal compensation consists of using a coupling with sufficient end clearance that allows relative movement between the driving and driven members. The equipment must be positioned so the horizontal growth moves into the coupling operating zone, not away from it. Failure to do so will result in excessive crankshaft thrust bearing loading and/or coupling failure. Sufficient clearance has been allowed if it is determined during hot alignment check that the crankshaft still has end clearance.

Cat Viscous Damped Coupling

Caterpillar couplings use an internal gear design with a rubber element between the gears. Silicone grease aids in the dampening characteristics.

The clearances involved in internal gear design allow accurate alignment measurement

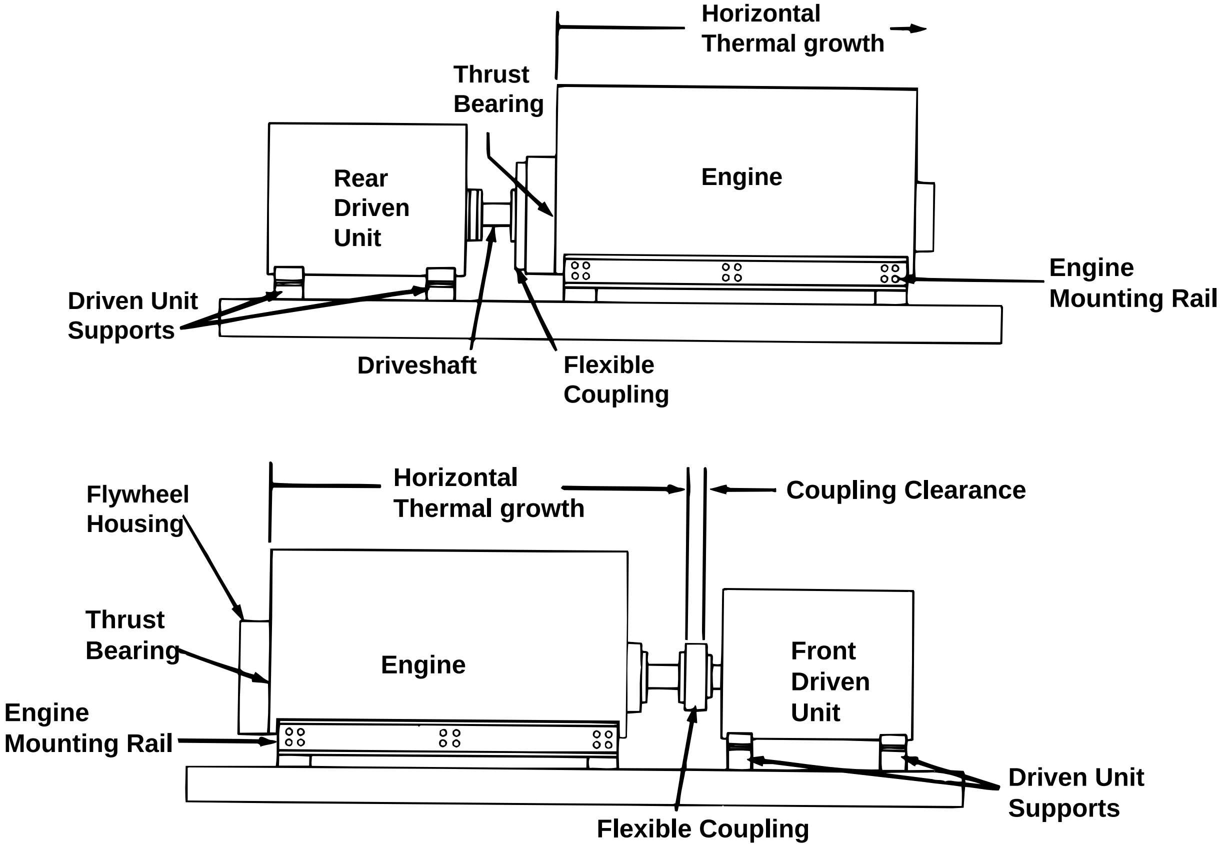


Figure 3.8

to be made without removing the rubber element.

The coupling for front-driven equipment is similar to the rear-drive coupling illustrated below. On front drives, the driven element shown in Figure 3.8 is to be supported on the engine crankshaft as it does not weigh as much as the driving element.

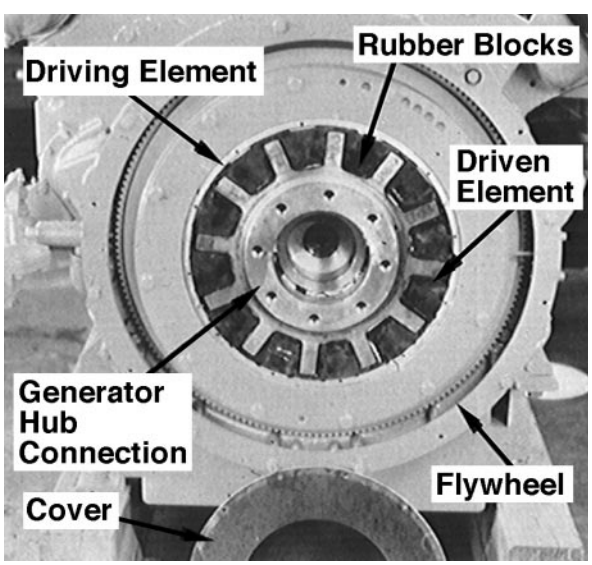


Fig. 3.9

Other Couplings

The flexible element of other couplings must be removed during alignment checks. Element stiffness can prevent accurate alignment readings.

With the coupling element removed, the driving and driven members of the coupling should be rotated together during alignment checks. This prevents face or bore runout of the piece parts from affecting the dial indicator readings. When both members are rotated together, only equipment misalignment will register on the dial indicator readings.

Mounting Auxiliary Engines

The proper engine mounting system will ensure the dependable performance and long life for which the engine was designed and manufactured if all equipment is properly aligned.

The engine should be mounted on a pair of longitudinal beams, the tops of which are in the same plane. If the tops of the beams are not fiat, add sufficient shims between the engine mounting surface and the mounting beams. Bolting the engines to an uneven surface can cause harmful distortions in the engine block, springing of the mounting beams, and high stress in welds or base metal.

If the engine is subjected to external forces, or if restrained from its natural thermal growth, tolerances are greatly affected and could easily result in bearing or crankshaft damage.

Three-Point Mounting

The three-point suspension system should be used when there is a possibility that the substructure supporting the base can deflect due to external forces or settling. Suspending the power unit on three points isolates the unit from deflection of the substructure, thus maintaining proper relationship and alignment of all equipment and preventing distortion of the engine block. More than three mounting points can cause base distortion (Figure 3.10.)

Objectionable vibration can occur if the power module is not mounted on well supported structures or is not anchored securely. In addition to the three-point mounting, vibration isolators may be required to isolate objectionable vibrations.

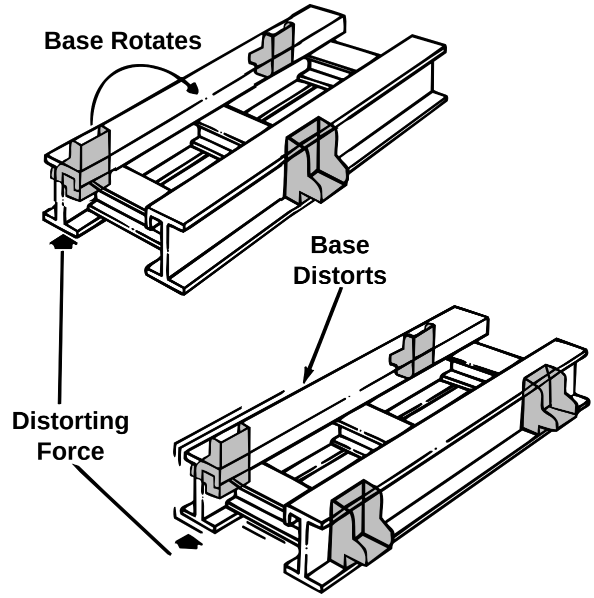


Figure 3.10

Anti-Vibration Mounting

Caterpillar Engines are capable of withstanding all self-induced vibrations and no isolation is required merely to prolong their service life. However, vibrations from surrounding equipment, if severe, can harm a generator set which is inoperative for long periods of time. Bearings and shafts can beat out and ultimately fail if these vibrations are not isolated. A running generator set will rarely be harmed by exterior vibrations. The method of isolating the unit is the same for exterior vibrations as it is for self-induced ones.

Caterpillar recommends the use of flexible mounts on all auxiliary engine installations. Refer to Vibration and Isolation section for information on types of vibration and isolation principles.

Some new generator packages have factory installed vibration isolators. refer to the Price List to determine if they are standard or optional.

Sources of Disturbing Vibrations

Vibrations affecting auxiliary engines may be classified into four groups:

- Propulsion engines.

- Propeller-induced vibration caused by the propeller blades passing the hull, strut, or skeg.

- Twin or multiple screw propulsion installations running out of phase where vibrations will occur at frequencies depending on the differences in engine rpm.

- All first order vibrations caused by other engines and installed pumping equipment.

Vibration Limit (No Load)

The acceptable no load vibration limit for Caterpillar Engines is 0.1 mm (4 mils) peak to peak displacement for the engine only and 0.13 mm (5 mils) for engine and driven equipment.

Protection

In order to protect marine auxiliary engines, flexible spring-type mounts should be installed between the base and the ship's structure. Caterpillar and others can supply flexible mounts. To obtain the correct flexible mount, the supplier must know what protection is required.

Selecting Flexible Mounts

- Contact a suitable supplier and provide him with:

  - Equipment configuration and base drawing.
  - Expected frequency of the forcing vibrations.
  - Weight and center of gravity of the auxiliary unit to be isolated.

- To be effective, static conditions must load isolators close to the center of their optimum deflection range. Therefore, the weight that will rest on each isolator must be known and the isolators properly matched to the load.

- When using resilient materials in addition to spring-type mounts, select the lowest psi loading which gives the highest percentage reduction in transmitted vibration.

- Several types of resilient pads isolate noise but not vibration. Some may even amplify first order vibrations. As a general rule, resilient mounting pads should have at least 6 mm (15/64 in.) static deflection; less than this results in reduced noise, but little or no vibration isolation. Consult the supplier for specific information.

Very Low Frequency Vibration

Vibrations at frequencies of 5 Hz and below are difficult to isolate.

The supplier of the flexible mounts is an excellent source for specific recommendations for very low frequency vibration mounting.

Installation of Flexible Mounts

Flexible mounts must be placed between the structural base of the auxiliary unit and the ship's structure. It is important that the base be of substantial design. When the ship structure is not sufficiently rigid, reinforcing supports should be added. When placing flexible mounts, the directions of the supplier should be followed.

The location of isolation mounts is important. On larger engines requiring three pairs of mounts, install one pair of isolators under the center of gravity and the other two sets equidistant from them at each end of structural base (Figure 3.11).

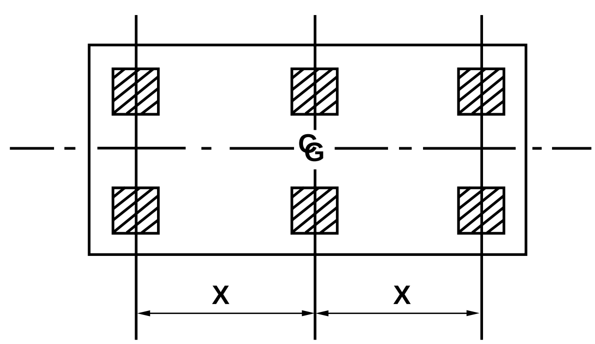


Figure 3.11

On smaller engines requiring only two pairs of mounts, locate one pair under front engine supports and the other pair under load supports.

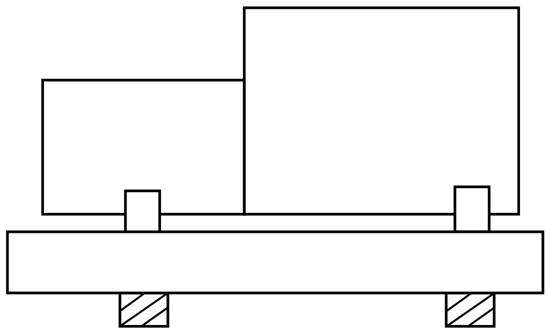


Figure 3.12

For three-point mounting of the engine base, arrange the isolators to obtain three point contact with the load equally distributed.

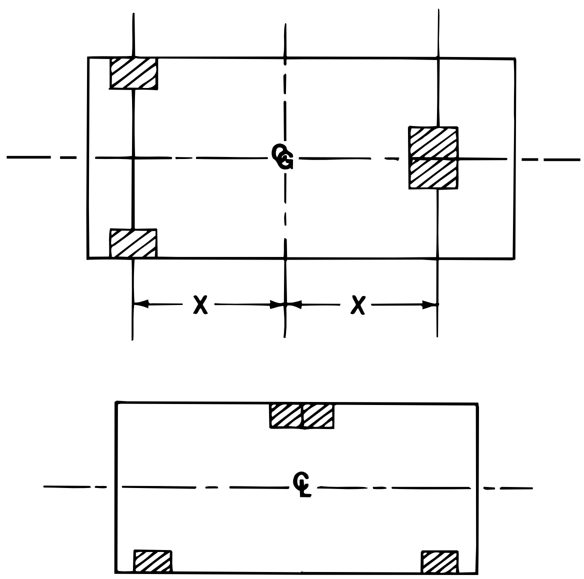


Figure 3.13

Determination of Center of Gravity of Combined Engine and Generator

The location of the center of gravity of the assembled unit can be determined after the total weight of the unit is established.

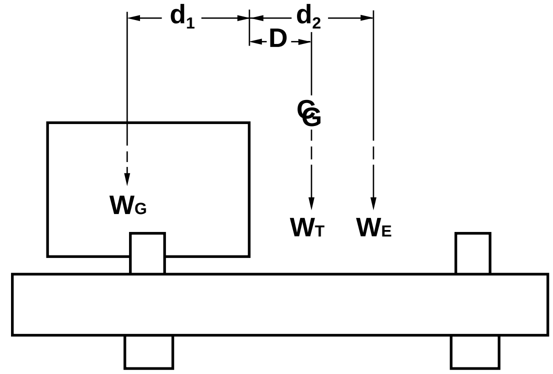


Figure 3.14

Assuming an engine and generator is assembled to a base, the assembled center of gravity (CG) can be calculated. A common reference point is needed. In this case, use the rear face of the flywheel housing. Because measurements are to both sides of the reference, one direction can be considered negative. Therefore:

$$W_T (D) = W_G (-d_1) + W_E (d_2)$$

$$D = \frac{(W_E(d_2) - W_G(d_1))}{W_T}$$

If additional equipment is added, such as front power take-off, the process is repeated to determine a new center of gravity.

Having established the center of gravity for the total unit, the loading on each pair of isolators can be determined (Figure 3.15).

$$S_1 = W_T \left(\frac{b}{c}\right)$$

$$S_2 = W_T \left(\frac{a}{c}\right)$$

The location of the center of gravity of the assembled unit can be determined after the total weight of the unit is established.

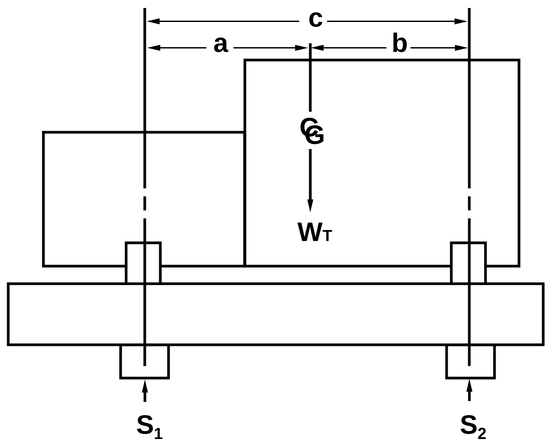


Figure 3.15

Commercial Isolators

Several commercial isolators are available which will provide varying degrees of isolation. Care must be taken to select the best isolator for the application. Generally the lower the natural frequency of the isolator (softer), the greater the deflection and the more effective the isolation. However, the loading limit of the isolator must not be exceeded.

Rubber Isolators

Rubber-type isolators are adequate for applications where vibration control is not severe. By careful selection, isolation of 90% is possible. They will isolate most noise created by transmission of vibratory forces. Care must be exercised to avoid using rubber isolators which have the same natural frequency as the engine-exciting frequencies in both the vertical and horizontal planes.

Fiberglass, Felt, Composition and Flat Rubber (Waffle) Isolators

Fiberglass, felt, composition and flat rubber of a waffle design do little to isolate major vibration forces, but do isolate much of the high frequency noise. The fabric materials tend to compress with age and become ineffective. Because deflection of these types of isolators is small, their natural frequency is relatively high compared to the engines. Attempting to stack these isolators or apply them indiscriminantly could force the total system into resonance. Pad-type isolators are effective for frequencies above 2,000 Hz.

Spring Isolators

The most effective isolators of low frequency vibration are the steel spring type. These can isolate approximately 96% of all vibrations. They also provide overall economy and allow mounting, of all but propulsion machinery, on surfaces that need only support the static weight. No allowance for torque or vibratory loads is required on nonpropulsion machinery. Steel spring-type isolators are effective in the vibration frequency range from 5 to 1,000 Hz.

Marine-type spring isolators should be used for auxiliary engine mounting. This type of isolator is equipped with all directional limit stops designed to restrict excessive movement of the engine and to withstand forces due to roll, pitch and slamming of sea-going vessels.

By the addition of a rubber pad beneath the spring isolator, the high frequency vibrations which are transmitted through the spring are also blocked. These high frequency vibrations are not harmful but result in annoying noise (Figure 3.17).

Follow the installation and adjustment instructions provided by the isolator supplier.

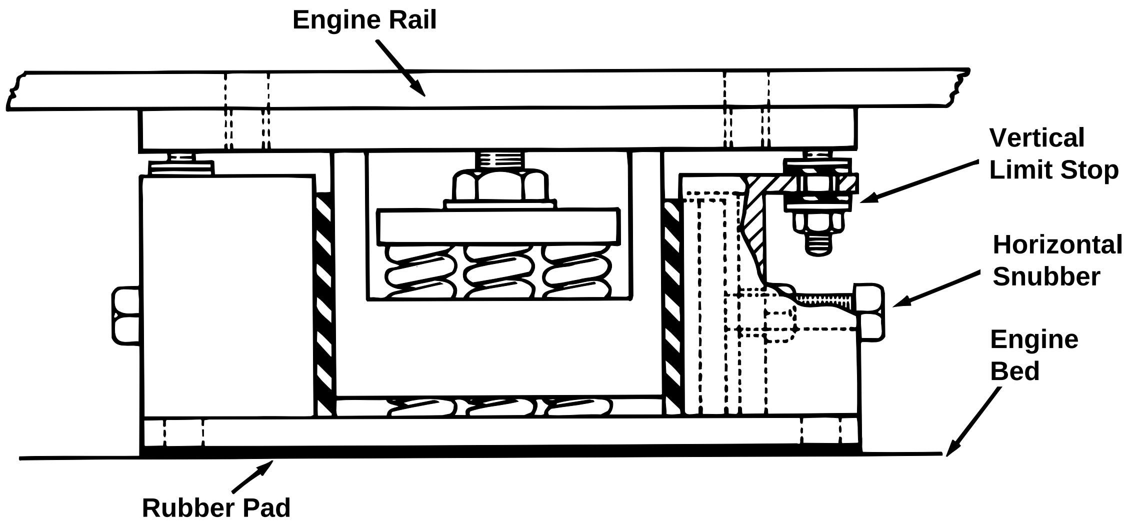
Many spring type isolators are equipped with horizontal limit stops (snubbers) but do not include built in vertical limit stops. If this type of isolator is used, external vertical limit stops should be added between the engine rail, or support, and the ship's engine bed.

Isolator snubbers and limit stops should be adjusted to permit only the amount of motion necessary for isolation purposes.

No matter what type of isolation is used, it should be sized to have its natural frequency as far removed from the exciting frequencies of the engine as possible. If these two frequencies were similar, the entire unit would be in resonance.

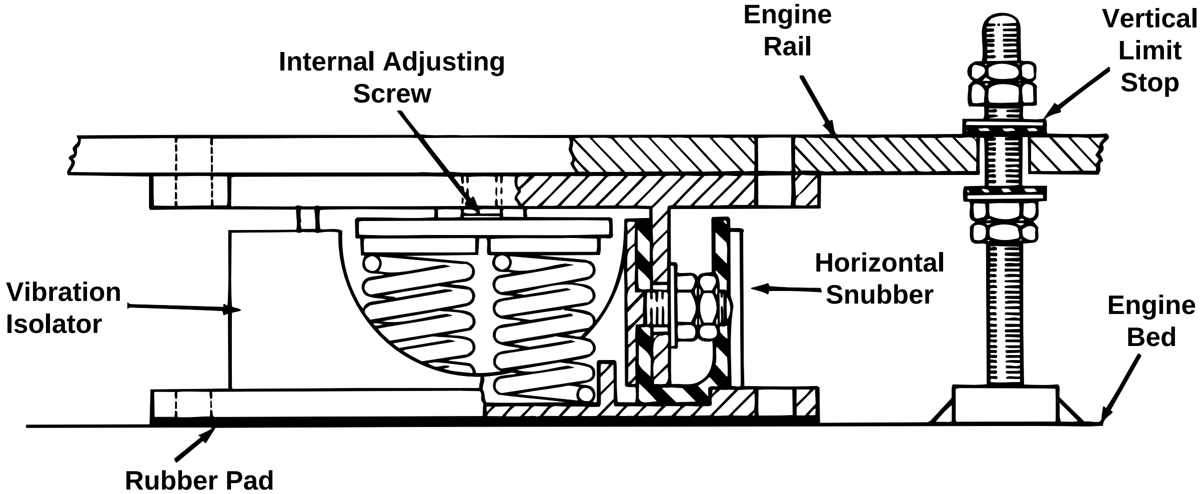
Flexible Connections

When using the marine spring isolator, ensure that each pipe, control system, electrical, and driveline connection is properly designed to allow for maximum engine



MARINE-TYPE ISOLATOR

Figure 3.16



SPRING ISOLATOR WITH EXTERNAL LIMIT STOP

Figure 3.17

motion without overstressing any of the connecting components.

Collision Blocks

All spring-mounted equipment should have stops to restrict vertical and side movement within reasonable limits. Collision blocks may be provided for all auxiliary engine installations if they do not restrict thermal growth.

Shimming

Use shims as necessary between the generator mounting feet and the generator supports to maintain correct vertical alignment with the engine. All generator mounting feet must be in solid contact with the supports before installation of the anchor bolts. If the mounting feet are not in solid contact, distortion of the generator housing can result.

Shim packs under all equipment should be 5 mm (0.2000 in.) minimum thickness to prevent later corrections requiring the removal of shims when there are too few or zero shims remaining.

Shim packs should be of nonrusting material. Handle shims carefully.

Mounting Bolts

The diameter of the clearance-type bolts used to hold the engine rails or feet to the base must be 1.6 mm (0.06 in.) less than the diameter of the holes in the engine rails. This clearance is to allow the engine mounting rails or feet to grow without confinement. Refer to the section on thermal growth.

Mounting Bolt Location

Each engine or generator mounting bolt must bolt through solid material (refer to Figure 3.18).

Procedure for Tightening Equipment Mounting Bolts

1. Torque mounting bolts in sequence shown in Figure 3.19 to 1/2 torque values listed.

2. Install a dial indicator on support bracket as if bore alignment were to be checked. Rotate driving and driven shafts together until dial indicator is at top position.

3. Loosen bolts at mounting surface 1 and retorque bolts at mounting surface 3 to 1/2 torque value as listed in Figure 3.19.

4. If indicator moves 0.05 mm (0.002 in.) or less, retorque bolts at mounting surface 1 and follow steps 6 and 7 below. If indicator moves more than 0.05 mm (0.002 in.), add shims under bolts at mounting surface 1 or 3. Loosen all bolts and repeat steps 1 through 5.

5. Loosen bolts at mounting surface 2 and retorque bolts at mounting surface 4 to 1/2 torque value listed.

6. If indicator moves 0.05 mm (0.002 in.), or less, retorque bolts at mounting surface 2. If indicator moves more than 0.05 mm (0.002 in.), add shims under bolts at mounting surface 2 or 4. Repeat steps 1 thru 6.

7. With indicator and support bracket still at top position, retorque all bolts to full values. Reading should not change more than 0.05 mm (0.002 in.).

Bolt Torque

A bolt is properly torqued when it is stretched a calculated amount. The proper stretch clamps the machinery to the base securely. The clamping force is then maintained during movement caused by vibration(refer to Figure 3.20). A bolt that is undertorqued cannot maintain the clamping force while vibrations are present. It will gradually work loose and allow misalignment to occur.

Bolts of the size used on Caterpillar bases require very high torque values. As an example, a 25.4 mm (1 in.) bolt has a torque of 868±108 N•m (640±80 ft lb).

A torque wrench, extension and torque multiplier are required to obtain this high value. Do not use special bolt lubricants as the effective bolt clamping force can be excessive.

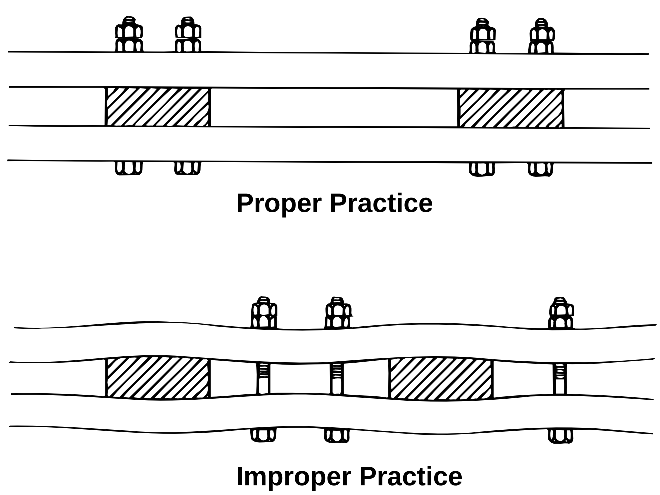
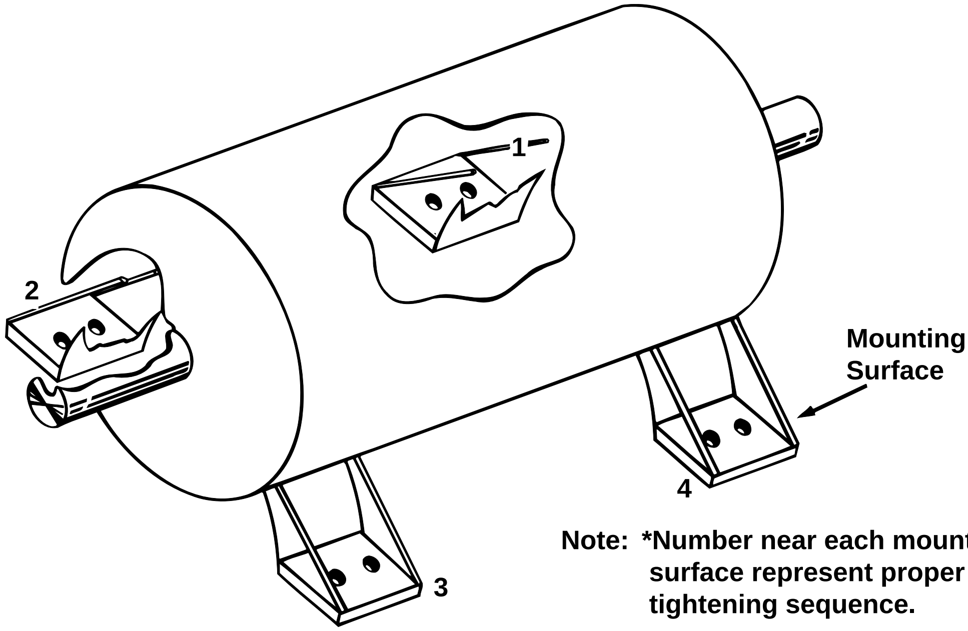


Figure 3.18



Note: *Number near each mounting surface represent proper bolt tightening sequence.

*Procedure described is valid for all independently mounted equipment. i.e., engine, two bearing generator, remote mounted marine transmission, etc.

Full Torque Value

Bolt Diameter		Torque	
	mm	lb-ft	N•m
3/4 Inch	19	265 ± 35	360 ± 50
7/8	22	420 ± 60	570 ± 80
1 Inch	25	640 ± 80	875 ± 100

Figure 3.19

Caterpillar nuts and bolts are made of Grade 8 steel, one of the strongest available. They are identified by six raised or depressed lines on the nut or bolt head. Make sure mounting bolts are not bottomed out in hole, resulting in low effective bolt clamping force.

After completion of the final shimming and bolting operation, recheck the alignment.

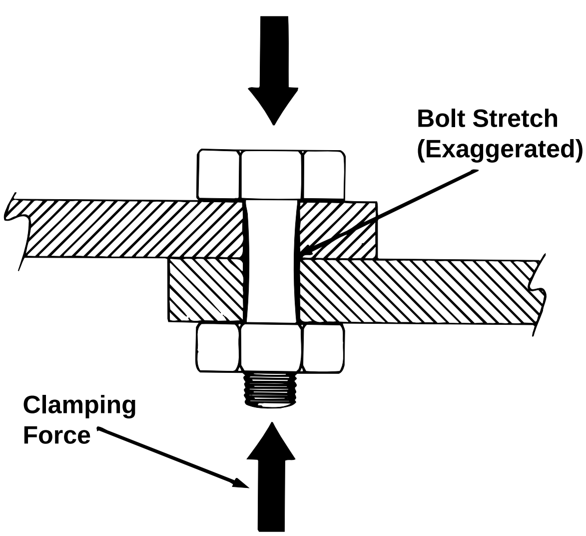


Figure 3.20

Crankshaft Deflection

Crankshaft deflection must be measured both cold and hot on certain specified engines. The allowable deflection must not exceed 0.025 mm (0.001 in.). Deflection must be measured at the center crankshaft throw. See Mounting and Alignment section for details on crankshaft deflection.

CATERPILLAR®

Marine Engines Application and Installation Guide

- Control Systems
- Instrumentation and Monitoring Systems
- Starting Systems
- Start-up Systems
- Serviceability

LEKM7144-01 (Supersedes LEKM7144) (05-00)

CATERPILLAR®

Control Systems

General Information

Engine Stall and Reversal

General Information

The use of a reliable control system is essential. The controls must be precise, dependable, and easy to operate.

The control system, in its most basic form, is the equipment which allows the pilot to adjust the propulsion engine/s throttle (speed) and the marine gear's clutches—from *neutral* to *ahead* or *astern*.

To control throttle setting, a control system must rotate and hold the angular position of the governor control throttle.

To effectively control the marine gear with mechanically actuated hydraulic control valves, a control system must move a short lever on the hydraulic control valve to any of three positions (forward, neutral, reverse) and maintain the selected position without placing undue stress on the linkage or allowing the lever position to creep.

To effectively control the marine gear with solenoid-actuated hydraulic control valves, an electrical signal energizes one of two solenoids to pressurize either the forward or astern clutch. If neither solenoid is energized, the gear remains in neutral and neither clutch is pressurized.

Basic System Features

Two Lever Control

Two lever control systems use two levers for the pilot's control of the engine and the marine gear, hence the name.

The position of one of the levers determines the engine throttle setting.

The position of the other lever determines the transmission direction—neutral, astern, or ahead.

Two lever control systems are most simplified, and most economical, but have the possibility of changing the transmission direction while the engine is at a high throttle setting. Transmission clutch damage is likely if this occurs.

Single Lever Control

Single lever control systems provide automatic sequencing of the control functions, preventing the transmission from changing direction until the throttle lever is moved to the neutral position (refer to Figure 1.1).

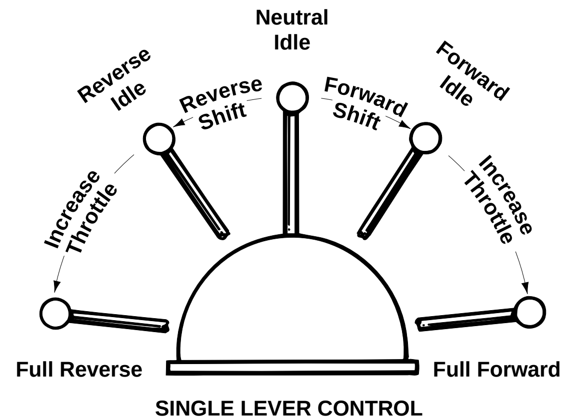


Figure 1.1

Neutral Throttle

Neutral throttle allows independent speed control when the marine gear is in neutral. This feature is useful when controlling the speed of engine-driven accessories such as generators, pumps or winches.

Multiple Control Stations

All vessels require one control station where the pilot controls engine and transmission. It is convenient to have other control stations when specific activities, such as docking and fishing, demand the pilot's close attention.

The simplest and, in most cases, most efficient multiple (dual) station system consists of two lever controls installed in a parallel system. Cables are run from the controls at each station directly to the clutch and throttle levers at the engine, and connected there with the appropriate parallel dual station kits.

A second type of multiple (dual) station system consists of two lever controls in series. Cables are run from the upper control station to the lower control station. A cable attachment kit is required to connect these cables to the lower station controls. Cables are then run from the lower station controls to the clutch and throttle levers at the engine and connected there with the appropriate engine connections kits. Series installations are less precise than parallel systems and should be used only when a parallel installation would be impractical due to long cable runs and excessive or sharp bends in the cable. The system selected is determined by the cable length and total degrees of cable bend required.

Engine/Gear Mounted Bracket Design

Brackets supporting the control systems cables/actuators at the engine/marine gear must be rigid. Good alignment of the cable-ends with the engine's throttle lever and gear's clutch control lever is necessary to avoid binding. Control system manufacturers can provide suitable brackets to the user/installer.

Type

Push-Pull Cable System

Push-pull cable control systems are reliable and economical. The distance between the control head at the pilot station and the engine is limited by friction in the cables.

For best results, keep cable length under 30 ft.

The number and included angle of bends in the control cables add significantly to their internal friction. Avoid all unnecessary bends. Keep all bends in the cables as gradual as possible (minimum 200 mm [8 in.] bend radius).

Stiffness or binding in the operation of the hand lever can usually be traced to:

- Excessive number of bends in cable runs
- Sharp bend in the cables too close to the control head
- Bends smaller than the recommended minimum radius of 200 mm (8 in.)
- Tight or misaligned engine linkage
- Cable compressed too tightly by cable support
- Engine clutch lever hitting its limit stops at forward and/or reverse

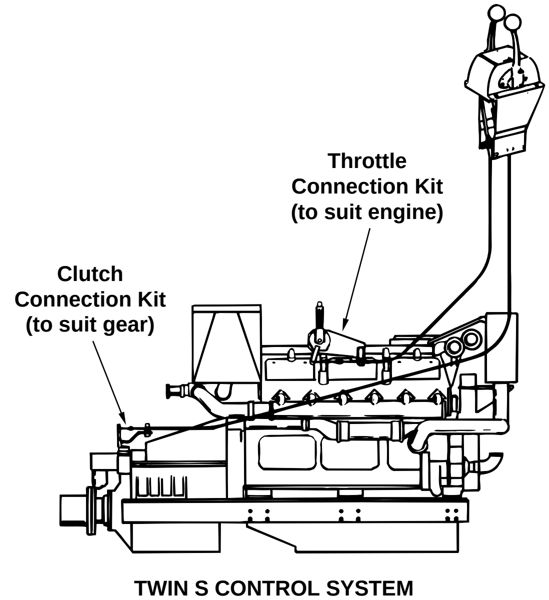


Figure 1.2

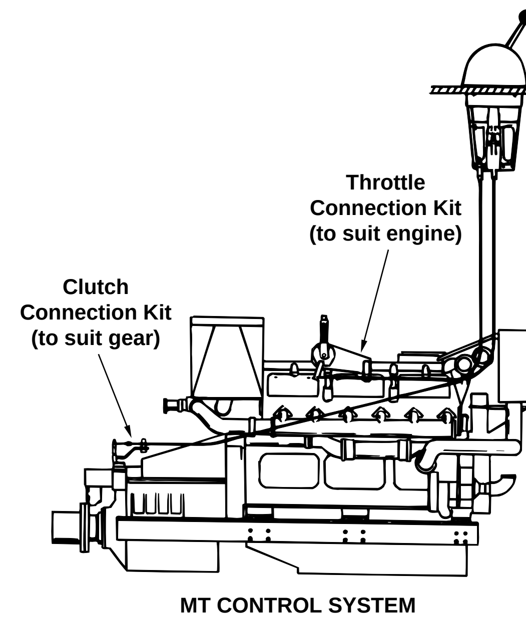
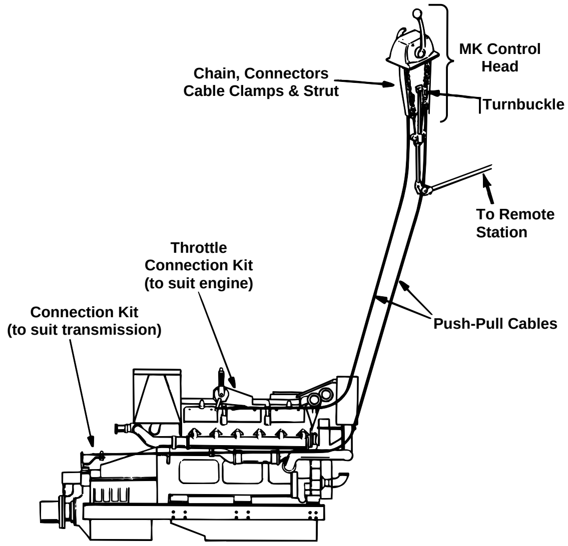
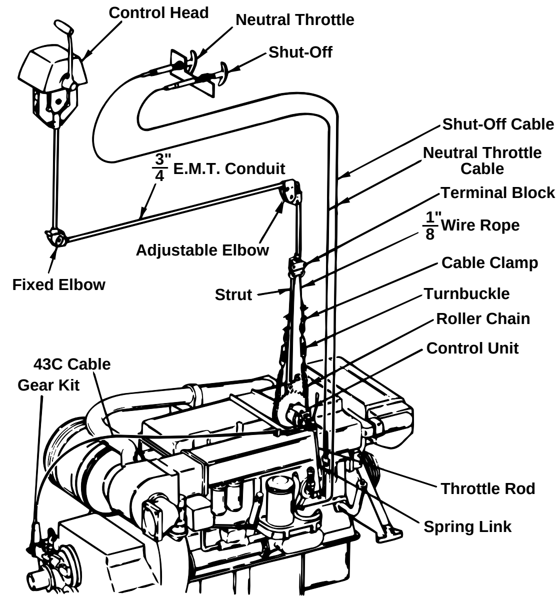


Figure 1.3



MK CONTROL SYSTEM

Figure 1.4



MD 24 CONTROL SYSTEM

Figure 1.5

The installation of push-pull cable control systems is fairly simple.

Manufacturers' installation bulletins for both two-lever (Figure 1.2) and single-lever lever (Figures 1.3 and 1.4) systems illustrate the systems.

Cable-in-Tension System

Where a cable control system is preferred and long runs and numerous bends may be encountered, a system is used utilizing two cables in tension, running over pulleys mounted on antifriction bearings (refer to Figure 1.5). To reduce the number of cables and to maintain precision in the response of the system, a single lever type of control system is used. A control gearbox to the governor and reverse gear is installed on the engine.

Hydraulic Control System

Hydraulic controls offer smooth, precise control of engine/marine gear without significant limitation on number of control stations or distance between control stations and the engines. The cost of hydraulic controls and the number of installation manhours are slightly higher than either of the mechanical cable controls.

Electronic Control System

Electronic control systems should be considered when the following control requirements are encountered.

- Electronically controlled engines
- Limiting the engine power during acceleration
- Engine overload protection
- Integration with controllable pitch propeller control systems
- Sharing of load between multiple engines, driving a single load
- Very long distances between the control station and the engine
- Integration with telemetry systems
- Adding additional control stations after vessel completion

Electronic Control System Components Control Station

The control station is generally simpler than a similarly functioned mechanical or hydraulic version. The forces involved in driving rheostats and switches are much less than those to operate push-pull cables or hydraulic cylinders. Electric control stations are very easy to install.

Most electric control systems will install an electric-to-mechanical converter box in the engine room/compartment. The electric-to-mechanical converter box accepts electrical signals from the various control stations and converts them to mechanical forces (generally via push-pull cables), suitable to operate the engine throttle and marine gear shifting valve. On newer mechanically controlled engines, control systems are capable of using electronic engine governors and electric marine gear control valves which eliminate the need for the electric-to-mechanical converter box.

Engine Throttle and Marine Gear Actuator

See electronic installation guide for electronic controlled engine.

3126B	RENR2233	
3176B	SENR6489	
3176C	SENR1187	
3196	SENR1187	
3406E	SENR1187	
3408	SENR6446	(PEEC)
3412	SENR6446	(PEEC)
3412E	SENR5014	

Mather's, ED Electric, Sturdy, Twin Disc and Kobelt are control manufacturers that can provide fully electronic control packages with our electronically controlled engines. They have electronic controls that are programmable for shifting and provide an electronic signal compatible with Caterpillar electronic engines.

Control Logic

Generally, the control logic is contained in the electric-to-mechanical converter box in the engine room/compartment. Larger systems will combine the logic circuitry with a propulsion system monitoring system in a cabinet in the engine room.

System Connectors

Electric control systems are generally interconnected by multiconductor electrical cable. This is much less expensive than mechanical cable or hydraulic tubing.

Pneumatic Control System

Pneumatic controls offer several advantages over other control systems:

- Ability to control engines at long distances. 90 m (300 ft) is a realistic distance to run air lines for pneumatic control. The only real limitation is the speed of response–in the case of very long lines.
- Ability to control from an unlimited number of control stations.
- Ability to add logic to the system, to protect against abuse of the driveline components.
- There are some disadvantages to pneumatic control systems:
- A relatively heavy and expensive compressor with air storage tank is required.
- The tank and lines require regular maintenance (draining of condensation).

Engine Stall and Reversal

When a marine reduction transmission is shifted from forward to reverse or vice versa, sufficient engine torque must be available at idle speed to overcome propeller and driveline inertia, marine transmission inertia, and slip stream torque*.

If sufficient torque is not available or if sufficient engine safeguards are not installed, the engine will stall or reverse itself.

In vessels where rotating masses are moderate to small, clutch modulation and engine torque can control the reversing cycle. Heat buildup caused by the clutch slipping is normally well within the clutch capacity. Heat generated through increased modulation necessary to control large inertia forces can damage clutches. To prevent this buildup of heat, auxiliary devices may be necessary.

* Slip stream torque is the torque generated in a free-wheeling propeller, being turned by the water flowing past the hull. Slip stream torque can be as high as 75% of the engine's rated torque.

Also, under crash reversal conditions, it is conceivable that unless some device is used to counteract the inertia of large masses, the engine could stall or actually be motorized to run in reverse rotation.

Avoiding engine stalling and/or reversal with mechanical controls is difficult. One method is by careful clutch engagement and by allowing the boat to slow down before the shift is made. The adept operator can repeatedly engage and disengage the reversing clutch, until the vessel's speed is checked sufficiently, and then complete the maneuver. Where large, heavy vessels, or those attached to a tow are concerned, this method may cause overheating of the reverse clutch. When this danger exists, other means must be employed.

Engine stalling and reversal problems can be avoided if close attention is paid to the engine and transmission control system.

Pneumatic and electronic controls which provide sequencing and timing of speed and directional signals offer optimum maneuvering as well as protection for the engine and transmission.

When Engine Stall and Reversal Could Be a Problem

The likelihood of this being a problem is significantly increased for vessels equipped with:

- Propulsion engines producing over 500 hp
- Fixed pitch propellers
- Deep ratio reduction gears, usually 4:1 and deeper

What the Operator Can Do

Loss of acceptable engine speed can be prevented by prudent use of the controls by the operator during maneuvering.

Engine Speed Limits During Emergency Maneuvers

It is imperative that engine speed does not drop below 300 rpm for slow speed engines (rated at nominally 1200 rpm) or 400 rpm for high speed engines (rated at nominally 1800 to 2300 rpm) in order to assure adequate lubrication and to prevent the possibility of stalling.

Need for Sequencing Control Systems

Sequencing and timing of the controls when using air control systems is necessary to:

- Reduce vessel maneuvering time
- Prevent excessively low engine speed
- Prevent excessive loading of driveline components
- Reduce the possibility of engine stalling

The possibility of engine speed reduction to the point of stalling due to sudden vessel maneuvering demands will be dependent upon the speed of the vessel when the maneuver is undertaken. During low vessel speed maneuvers, the engine torque capabilities are usually sufficient to respond adequately. However, if a sudden maneuver, such as a crash stop of the vessel, is demanded at full vessel speed, auxiliary driveline devices may be required to prevent stalling and loss of vessel control.

Sequencing Control System Features

To forestall the possibility of engine stall during high speed maneuvers in emergency situations, one or more of the following may be required:

- Raised low idle speed setting
- Throttle boost control
- Shaft brake
- Control system timing

3500 Family Engines equipped with 3161 governors will shut off their fuel if subjected to engine reversal.

Raised Low Idle Fuel System Setting

To increase the engine's low speed torque, the low idle setting may be increased if the vessel's low speed maneuvering is not jeopardized. This will help prevent the engine from stalling or reversing during maneuver. The setting should be accomplished by an authorized Caterpillar Dealer. Excessive shock loading and transmission clutch wear can occur if the engine low idle speed is too high.

Throttle Boost

Throttle boost momentarily raises the idling speed setting of the engine. The engine speed increase comes just before engagement of the marine gear clutch. This momentary speed increase occurs only during maneuvering, not at steady boat speed conditions.

Throttle boost is kept as low as possible because it tends to increase the load on the clutches during maneuvering. The control system should permit adjustment of both the amount and duration of throttle boost. The throttle boost for most Marine Gears should be set no higher than 750 rpm for 1800 rpm engines and 600 rpm for 1200 rpm engines at no load. Sea trials should determine the level of throttle boost necessary to ensure a safe shaft reversal and maintain engine speed above the minimum limits. Consult the marine transmission manufacturer for boosted-shift clutch capability.

Although reversing problems seldom occur with marine transmissions ratios more shallow than those previously mentioned, it is recommended that a throttle boost system be incorporated with more shallow ratioed transmissions as an additional safety feature.

Shaft Brake

In vessel applications where heavy maneuvering is required or if full speed reversals may be encountered, the use of a propeller shaft brake is recommended. A properly controlled shaft brake will stop the rotation of the propeller whenever the transmission clutches are disengaged and the engine is at low idle speed. This action reduces the amount of torque required from the engine in order to complete a shaft directional change. Several advantages are gained with the use of shaft brakes.

1. A propeller shaft brake can safely reduce vessel maneuvering time. A vessel will slow in half the time with a stopped propeller as compared to a windmilling propeller. The propeller slip stream torque, therefore, falls to a level lower than the slow speed torque of the engine in half the time.

2. The propeller shaft brake accepts half the speed reversal loads when maneuvering. The brake brings the propeller to a stop. This load is transmitted directly to the hull. The clutch and propulsion system are only asked to pick up a stopped propeller shaft rather than a windmilling propeller. Because load on the engaging clutch is greatly reduced, clutch life is extended. Transmission gears, engine and other major components of the propulsion system are subject to less shock.

3. The propeller shaft brake will prevent engine stall when attempting crash stops or when high vessel speed shaft reversals are attempted during maneuvers.

A propeller shaft brake should be considered on any marine propulsion system using engines over 500 hp where the reduction ratio is 4:1 or deeper and where high speed maneuvering is a requirement.

Disc brakes and drum-type brakes are available. The brake should be sized to handle at least 75% of the full rated shaft torque and should stop the shaft within three seconds during a crash reversal. Brake size requirements will vary with type of propeller, vessel speed, and vessel application.

Proper control and sequencing of a propeller shaft brake is very important. Overlap can occur if the clutch engages before the brake is released. This would show as an extra load on the engine, slowing it and even stalling the engine. Underlap is releasing the brake well ahead of the clutch making contact. The propeller will quickly begin to windmill in the wrong direction and much of the advantage of the brake is lost.

Event Sequence Timing

Sequencing and timing of engine governor, marine transmission clutch, and shaft brake action are critical and only systems of the following characteristics should be used:

Pilot House Control Movement
Full Ahead to Astern and Full Astern to Full Ahead

Event Sequence

1. Governor to Low Idle
2. Clutch to Neutral
3. Shaft Brake Applied Propeller Shaft Stops
4. Shaft Brake Released

5a. Throttle Boost Applied
5b. Clutch Engaged*
6. Throttle Boost Off, Governor to Full Open

With the above sequencing and timing, the shaft brake will engage any time the pilot house control lever is in the neutral position. Throttle boost will activate each time the pilot house control lever is shifted from neutral to a clutch-engaged position.

A proportional pause-type control system will allow for a variable time between Steps 3 and 4 in the event sequence when the shaft brake is applied. The pause is in proportion to the last-called-for speed. A crash reversal from full speed will leave the brake applied for a longer period than when slow speed maneuvering. This full speed reversal pause in neutral is made just long enough for the vessel speed to slow to a point that the propeller slip stream torque will not stall the engine when the reverse clutch is engaged.

* Timing sequence from brake release to clutch engagement should result in from one quarter to one revolution of the propeller shaft in the wrong direction to ensure there is no overlap between brake release and clutch engagement.

Properly adjusted air controls should provide event sequence time in the area of 5 to 7 seconds for slow speed maneuvering and in the area of 7 to 12 seconds or more for full speed crash reversals. The timing is set as fast as the propulsion system can safely be operated. The timing should be set permanently at the time of sea trials.

Without a propeller shaft brake, a longer pause in neutral in place of Steps 3 and 4 in the event sequence will normally be required to allow reduced vessel speed.

The control system must be carefully maintained. Follow the manufacturer's maintenance recommendations explicitly.

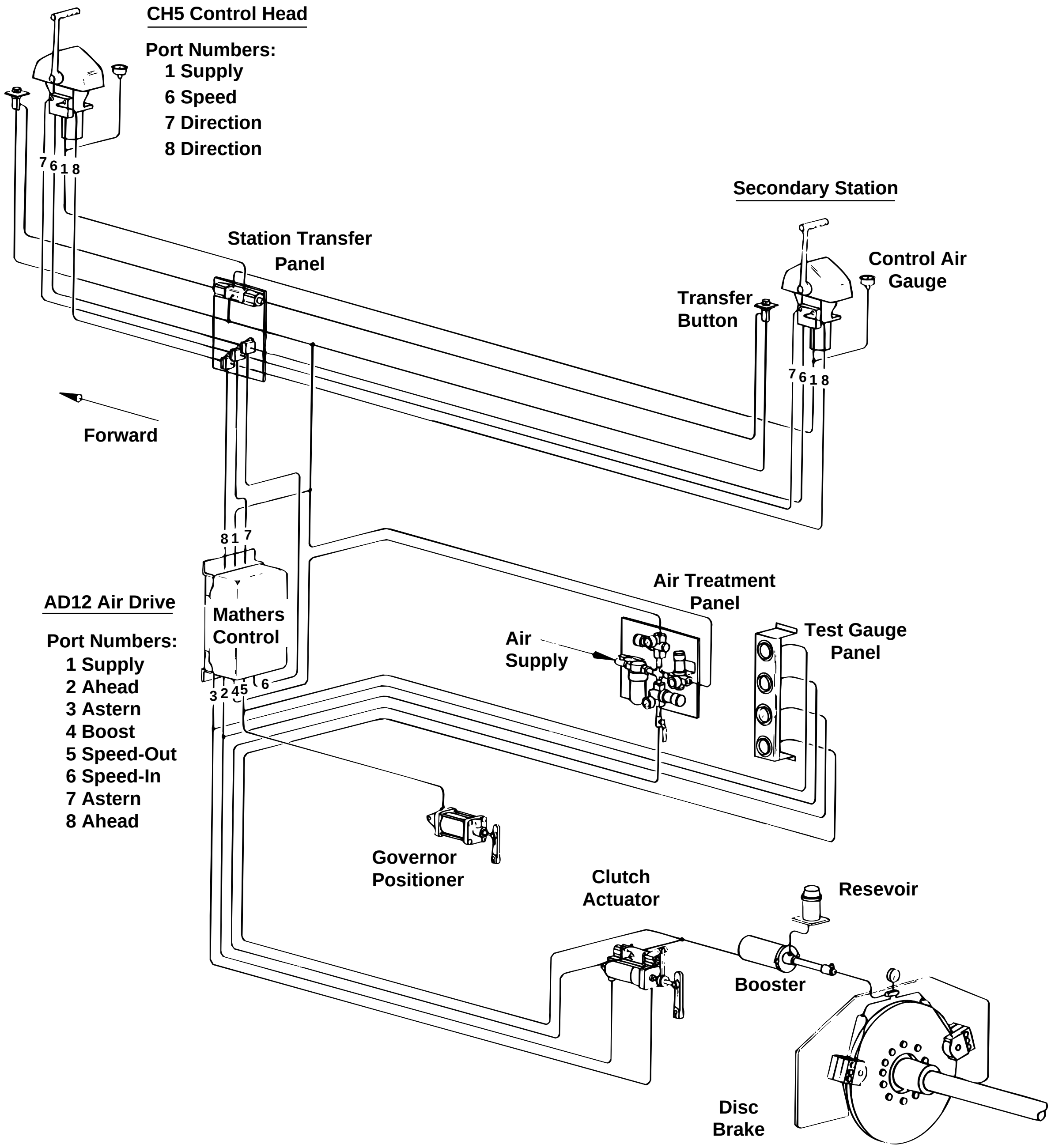
Recommended Systems

Consult with the manufacturers of control systems to determine their availability to provide event sequence systems as described previously.

The shipyard should furnish a low air pressure alarm located at the supply air to the pneumatic control system. The alarm should be audible and visual and should be actuated if the air pressure should fall below a predetermined level–generally 90 psi (620.5 kPa).

Mathers

The Mathers control system offers single lever pneumatic control of speed, clutch, and brake (if required). The system uses fixed orifice timing with option for as many control stations as required. They also offer electronic controls with the same sequence timing adjustments.



Mathers Controls Inc.
AD12 Propulsion Control System
For
Hydraulic Clutches

Figure 1.6

WABCO

WABCO, a division of American Standard, also provide complete sequencing control systems. The following is an example.

LMAC-3C Logicmaster Air Clutch-Control Systems for 3600 Family of Caterpillar Engines

This control system provides interlocked and sequenced operation of proportional timing in ahead and astern clutch engagement and engine speed control to ensure proper operation of the propulsion machinery as the operator manipulates the remotely mounted control lever. The control system incorporates the following interlocks and the optional features.

1. Positive cross engagement interlocks ensure that one clutch is vented below 15 psi before the opposite clutch can be inflated.

2. The clutch engagement system incorporates a three-stage clutch fill. First, an initial quick fill to bring the clutch shoes into contact with the drum. Second, a controlled rate of fill. Third, a full flow *hard fill* inflation at maximum rate up to supply pressure.

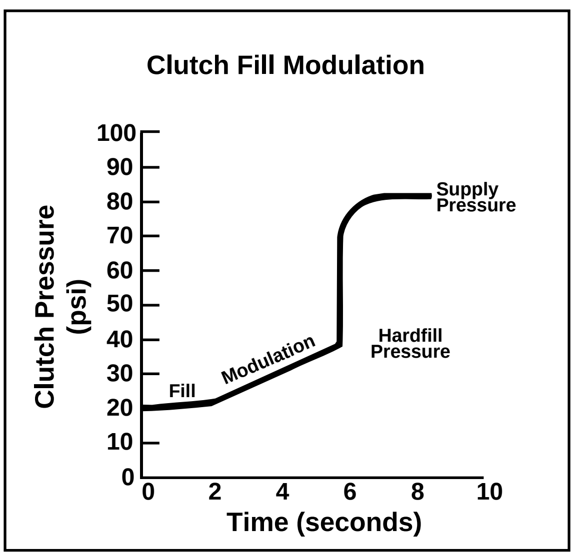


Figure 1.7

3. A governor power boost is applied during initial clutch engagement to prevent engine stalling. This boost is adjustable in magnitude and duration. Adjustment of timer unit controls rate of boost application and regulator controls the magnitude of boost delivered to the governor.

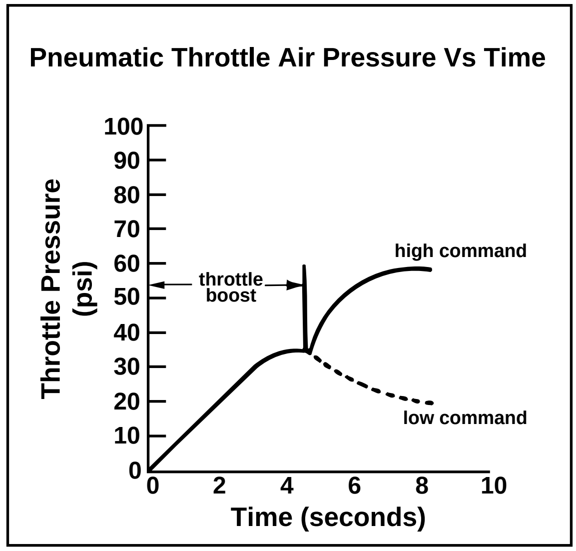


Figure 1.8

4. A clutch pressure-engine speed interlock to ensure that the clutch is inflated to lock-up pressure before engine speed can be increased from the remote operating station.

5. Proportional reversing interlock occurs in both directions. This time is adjustable and provides a neutral hold time proportion to the vessel speed. Normal maneuvers are accomplished with the minimum reversing time increasing in proportion.

6. The ahead clutch hold-in function to shorten the reversing time by holding in the ahead clutch which uses the engine's compression to slow down the propeller during high speed and crash reversals.

7. A shaft brake signal (optional) is provided to actuate a shaft brake in synchronization with the clutch engage/disengage control system. The brake is released when clutch engagement is initiated and is applied when both clutches are fully released. When a brake is used, the "interlocks" provide a neutral hold to permit the brake to be applied and the shaft stopped before reversal is initiated.

Other manufacturers may also be able to provide suitable systems.

Controllable Pitch Propeller to Avoid Engine Stall and Reversal

The controllable pitch propeller allows smooth, well-controlled vessel reversals while the engine rpm and horsepower are kept at optimum levels. This is most desirable on vessels equipped with deep ratio marine transmissions that must be reversed while moving at full vessel speed. To reverse a vessel equipped with a controllable pitch propeller, reduce propeller pitch to the “neutral pitch” position, then increase pitch in the “astern” or reverse direction slowly enough to allow the engine to maintain its full load rpm and horsepower.

Determining Likelihood of Stalling During Sea Trial

Initial sea trials should determine the likelihood of the control system/engine combination stalling during a crash reversal maneuver. Adjustment and timing of air controls can be determined and properly set during sea trials. Suggested procedure is to start with a low forward vessel speed and make crash shifts into reverse at small increments of increased forward vessel speed until it is determined that the system will allow a crash reversal at the most severe condition the vessel will encounter.

CATERPILLAR®

Instrumentation and Monitoring Systems

Instruments

Alarm/Shutdown Contractors

Instrumentation Problems

Instrumentation is a valuable component of a well designed installation.

Instruments

The functions below are listed in their order of desirability for Helmsmans station instrument panel placement.

A = *Must Have* Instrumentation
B = Highly Desirable Instrumentation
C = Useful Instrumentation
D = Questionable, Without Special Requirements

A **Engine Lubrication Oil Pressure**
Loss of lube oil pressure while operating at full power is likely to result in severe engine damage. Quick action by the pilot in reducing stopping the engine can save an engine. To protect the engine, the pilot must be able to see the status of engine oil pressure continuously.

A **Jacket Water Temperature**
Increase of jacket water temperature is almost as serious as loss of lube oil pressure and somewhat more likely. Similar quick action by the pilot can minimize engine damage resulting from a high temperature condition.

A **Engine Speed (rpm)**
Rather than a safety-oriented engine function, engine speed is an operation related measurement. Observing the relationship between engine speed, vessel load, and throttle position will allow the pilot to make informed judgments about engine load and need for maintenance.

B **Transmission Oil Pressure**
Transmission oil pressure measurement shows the pilot when the transmission clutches have engaged and useful information concerning the condition of the pump, filters, or clutches. Excessive pressure can damage components in the hydraulic circuit.

B **Voltmeter**
Voltage of the starter/alternator circuit give the pilot useful information regarding battery condition, alternator condition, state of charge of the batteries, and condition of the battery cables.

C **Transmission Oil Temperature**
Many transmission problems, such as clutch slippage, insufficient clutch pressure, bearing wear, cooler blockage or loss of cooling water flow will be manifested as an increase in transmission oil temperature.

C **Exhaust Stack Temperature**
Changes from *normal* exhaust stack temperatures will give useful information concerning air filter restriction, aftercooler restriction, injector condition, valve problems, and engine load.

D **Individual Cylinder Exhaust Temperature**
While these instruments will give immediate warning of individual injector failure, the inevitable wide tolerance on the standard temperature (±42 C degrees) (75 F degrees) often causes undue operator concern. In general, advantages gained by this instrumentation are overshadowed by high cost (thermocouples need annual replacement) and need for special operator training.

Alarm/Shutdown Contactors

These are preset contactors (switches) that will activate a customer-supplied alarm, light, or engine shutdown solenoid, when certain limits are exceeded.

Alarm switches available from Caterpillar will operate on AC or DC, from 6 volts to 240 volts. These switches are of the singlepole double-throw type.

With the exception of the overspeed function, propulsion engines should not be automatically shutdown. The pilot should be warned of impending failure but should retain the authority to decide whether to shut down the engine or to continue to operate. Overspeed failures will result in loss of engine power. If the engine is equipped with an overspeed protection device, the engine will not be harmed as much as if the overspeed failure had proceeded unimpeded.

A = *Must Have* Instrumentation
B = Highly Desirable Instrumentation
C = Useful Instrumentation
D = Questionable, Without Special Requirements

A **Low Lube Oil Pressure**
There are two conditions that need to be alarmed: low lube oil pressure at low engine load (idle conditions) and low lube oil pressure at high engine speed and/or load. An oil pressure that would be perfectly safe while operating at very low loads and/or speeds would be too low at full load/speed conditions. A suitable system will include two pressure-sensitive contactors and a speed (rpm) switch to decide which pressure switch should have the authority to warn the operator.

A **High Coolant Temperature**
High coolant temperature contactors should be set to actuate within 2.8°C (5°F) of the highest normal temperature of the engine at the point of installation.

A. **Overspeed**
Overspeed faults occur when some part of the engine fails, causing the fuel control mechanism to be locked in a high fuel flow condition. When the engine load goes to a low level, the engine will continue to receive a high fuel flow. Without the load, the engine speed increases to a dangerously high level. Generally, the engine's air supply must be cut off to save the engine.

Overspeed contactors need to be set 12-15% over rated engine speed to avoid nuisance engine shutdowns during sudden reductions in engine load.

B **Water Level Alarm**
Warning of coolant loss could allow the operator to save an engine which would otherwise be lost to overheat failure.

Install level sensors in the highest part of the cooling system–generally in the auxiliary expansion tank. This will give warning of coolant loss at the earliest possible time, before the coolant level has fallen to a dangerous level.

C **Low Sea Water Pump Differential Pressure**
The sea water flow to a heat exchanger cooled engine is very important. It is a good idea to install a differential pressure contactor across the sea water pump to warn of any discontinuity in sea water flow.

C **Intake Manifold Temperature Alarm Switches**
Intake manifold temperature alarm switches are available for use on some engines. High intake manifold temperature will warn of sea water pump failure, sea strainer plugging, or any other condition which reduces or stops aftercooler water flow.

Alarm Panel

Caterpillar recommends the following features in alarm panels:

- Fault light lock-in circuitry keeps the fault light on when intermittent faults occur.
- Lockout of additional alarm lights prevents subsequent alarm lights from going on after the activated engine shutoff stops the engine. This aids in troubleshooting.
- Alarm silence allows the operator to acknowledge the alarm without having to continually listen to the alarm horn. The alarm light is left on.
- If more than one engine is connected to an alarm panel, a fault in a second engine should activate the alarm even though the alarm horn may have been silenced after a fault on another engine.
- Circuit test provides for periodic checking of alarm panel functions.

Instrumentation Problems

Without highly trained personnel and rigorous discipline, too much instrumentation can be detrimental.

The weak link in any instrumentation system is the sensor unit (transducer). Too often, an otherwise fine system is sabotaged because of frequent false alarms. Plan annual replacement of the sensor units unless unusually high quality sensors are used.

High water temperature sensors will not warn of overheat conditions unless their sensing bulbs are submerged in water. High water temperature sensors will not warn of coolant loss.

Electronically controlled engines

The engines have sensors needed to control the engines standard and there are panels available which monitor and display the values these sensors put out. Consult price list for optional pilot house panels for use with these engines.

CATERPILLAR®

Starting Systems

General Information

Starting Aids

Starting Smoke

General Information

Startability of a diesel engine is affected primarily by ambient temperature, engine jacket water temperature, and lubricating oil viscosity. Any parasitic loads (usually associated with the driven equipment) can greatly influence the startability, as well.

The diesel engine relies on heat of compression to ignite fuel. When the engine is cold, longer cranking periods or higher cranking speeds are necessary to develop adequate ignition temperatures. The drag due to the cold lube oil imposes a great load on the cranking motor. Oil type and temperature drastically alter viscosity. SAE 30 oil approaches the consistency of grease below 0°C (32°F).

Starter Types

There are three different types of starting systems normally used for Caterpillar diesel engines. They differ in the method of storing and recharging the energy required for restarting the engine.

Electric Starting Systems

Use chemical energy stored in batteries, automatically recharged by an engine-driven alternator or by an external source.

Air or Pneumatic Starting Systems

Use compressed air in pressure tanks, automatically recharged by an electric motor-driven air compressor.

Hydraulic Starting Systems

Use hydraulic oil stored in steel pressure vessels under high pressure automatically recharged by a small engine-driven hydraulic pump with integral pressure relief valve.

The technology of the systems are well developed. Any of the systems are easily controlled and applied either manually or automatically. Several of the factors which influence the choice of systems have been tabulated below.

Electric

Battery-powered, electric motors utilize low voltage direct current and provide fast, convenient, pushbutton starting with lightweight, compact engine-mounted components. A motor contactor relieves control logic circuits of high cranking currents.

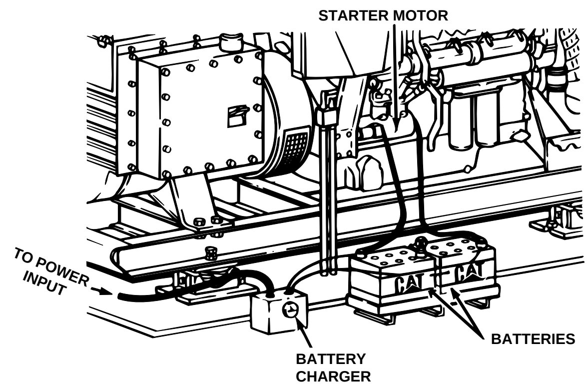


Figure 3.1

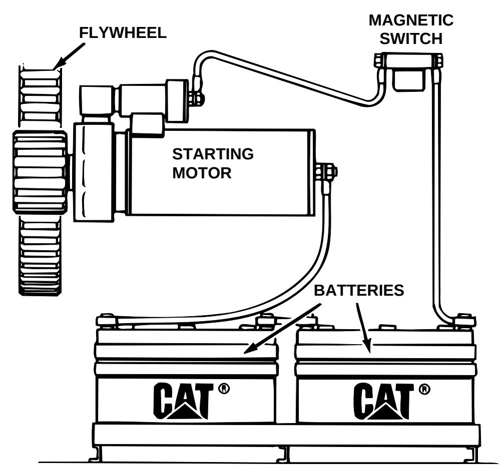


Figure 3.2

Storage Method

Lead/Acid storage batteries.

Relative Cost

Lowest.

Maintenance Requirements

Highest, the batteries require considerable maintenance.

Reliability of Starting

Good.

Special Concerns

Hydrogen gas, released from the batteries during charging, is very explosive, and compartments containing lead acid batteries must be properly vented.

Batteries

Batteries provide power for engine cranking. Lead-acid types are readily available, have high output capabilities, and are relatively inexpensive. Nickel-cadmium batteries are costly, but have long shelf life and require minimum maintenance. Nickel-cadmium types are designed for long life and may incorporate thick plates which decrease high discharge capability. Consult the battery supplier for specific recommendations.

Ambient temperatures drastically affect battery performance and charging efficiencies. When operating in cold climates, the use of battery heaters are recommended. The heaters should be set to maintain battery temperature in the range of 32° to 52°C (90° to 125°F) for maximum effectiveness. The significance of colder battery temperatures is described below.

All battery connections must be kept tight and coated with grease to prevent corrosion.

Temperature vs Battery Output		
°F	°C	Percent of 27°C (80°F) Ampere Hours Output Rating
80	27	100
32	0	65
0	-18	40

Battery Location and Hydrogen Venting

Install batteries only in well ventilated compartments. Visual inspection for terminal corrosion and damage should be easy. Batteries emit hydrogen gas during the recharging cycle. Hydrogen gas is highly explosive and very dangerous in even small concentrations. Hydrogen gas is lighter than air and will escape harmlessly to atmosphere, if not trapped by rising into a chamber from which there is no upward path to atmosphere. Devices which can discharge electrical sparks or cause open flames must not be allowed in the same compartment or in the vent path for the escaping hydrogen gas.

Battery Disconnect Switches (Battery Isolating Devices)

Solid state electrical devices will suffer when installed in vessels whose electrical system includes battery disconnect switches which can interrupt loadbearing circuits. At the instant of a circuit disconnect, transient currents and voltages will often cause failure in any component whose transistors are not otherwise protected.

Use Battery Disconnect Switches (Battery Isolating Devices) which do not cause voltage transients (spikes).

Battery Chargers

Various chargers are available to replenish a battery. Trickle chargers are designed for continuous service on unloaded batteries and automatically shut down to milliampere current when batteries are fully charged. Overcharging shortens battery life and is recognized by excessive water losses.

Conventional lead-acid batteries require less than 59.2 mL (2 oz) of make-up water during 30 hours of operation.

Float-equalize chargers are more expensive than trickle chargers and are used in applications demanding maximum battery life. These chargers include line and load regulation, and current limiting devices which permit continuous loads at rated output.

Chargers must be capable of limiting peak currents during cranking cycles or have a relay to disconnect during cranking cycles. Where engine-driven alternators and battery chargers are both used, the disconnect relay is usually controlled to disconnect the battery charger during engine cranking and running.

Engine-driven generators or alternators can be used but have the disadvantage of charging batteries only while the engine runs. Where generator sets are subject to long idle periods or many short stop-start cycles, insufficient battery capacity could threaten dependability.

Continuous Cranking Time Limit with Electric Starter Motors

An engine should not be cranked continuously for more than 30 seconds to avoid overheating of the starter motors.

Starter Motor Cooling Period Between Cranking Periods

Allow the starter motor to cool for two minutes before resuming cranking.

Battery Cable Sizing (Maximum Allowable Resistance)

The start circuit between battery and starting motor, and control circuit between battery, switch, and motor solenoid must be within maximum resistance limits shown.

Magnetic Switch and Series-Parallel Circuit	Solenoid Switch Circuit	Starting Motor Circuit
12 Volt System .048 Ohm	.0067 Ohm	.0012 Ohm
24 Volt System .10 Ohm	.030 Ohm	.002 Ohm
32 Volt System .124 Ohm	.070 Ohm	.002 Ohm

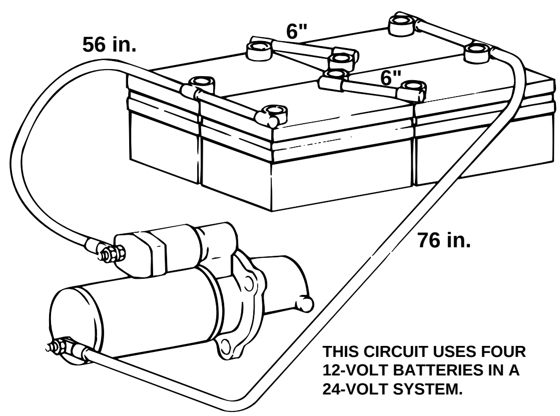
Not all this resistance is allowed for cables. Connections and contactors, except the motor solenoid contactor, are included in the total allowable resistance. Additional fixed resistance allowances are:

Contactors
Relays, Solenoid, Switches
0.0002 Ohm each

Connections
(series connector)
0.00001 Ohm each

The fixed resistance of connections and contactors is determined by the cable routing. Fixed resistance (Rf) subtracted from total resistance (Rt) equals allowable cable resistance (Rc): Rt - Rf = Rc.

Example:



SYSTEM 24-volt
STARTING MOTOR TYPE .... HEAVY DUTY
MAXIMUM ALLOWABLE RESISTANCE .... .00200
MINUS FIXED RESISTANCE—
6 CONNECTIONS @ .00001 .... .00006 OHM
RESISTANCE REMAINING FOR CABLE .... .00194
BATTERY CABLE LENGTH .... 144 in.

Figure 3.3

With cable length and fixed resistance determined, select cable size using the following chart. Only full-stranded copper wire should be used. Arc welding cable is much more flexible and easier to install than full stranded copper wire cable, but welding cable is not so durable and will be damaged from corrosion in a much shorter time.

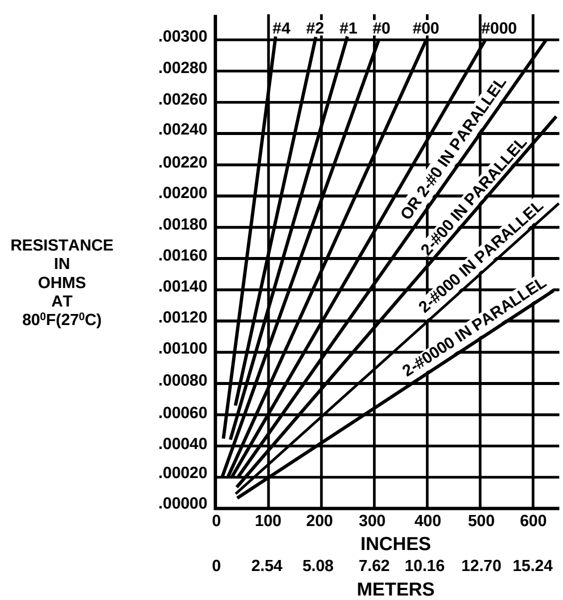


Figure 3.4

To meet cable length and resistance requirements, cable size must be No. 1. To determine fixed resistance in a parallel circuit, only series connections in one leg of the parallel circuit are counted.

Connections/Proper Practices

Electrical connections are often a source of problems for shipboard electrical systems.

Salt air and water are highly corrosive. Electrical connections are almost always made of dissimilar metals. Corrosion is more destructive between dissimilar metals.

The following table lists good practices for marine electrical systems.

When making electrical connections between wires, connect wires mechanically so tugging or pulling can be withstood without any other treatment of the joint. Then, coat the joint and the nearest portions of each wire with electrical solder. Do not expect solder to increase joint strength. The solder is for corrosion protection.

Do not use *crimp-type* connectors for marine service—The plastic sleeve tends to hide the corrosion from view rather than protecting the joint.

Pneumatic

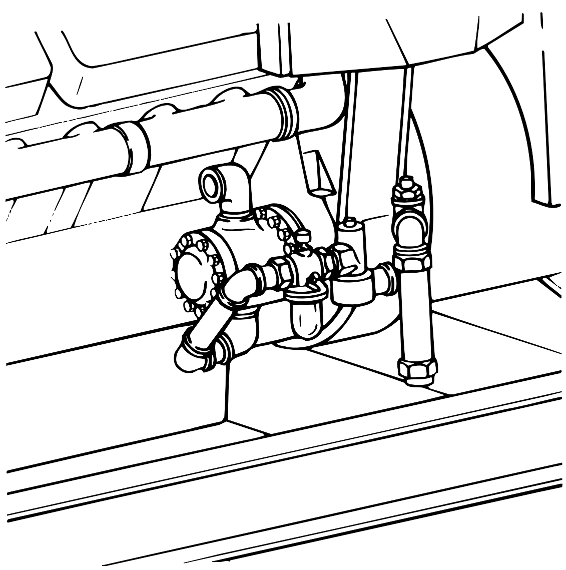


Figure 3.5

Air starting, either manual or automatic, is highly reliable. Torque available from air motors accelerate the engine to twice the cranking speed in about half the time required by electric starters.

Air is usually compressed to 758 to 1723 kPa (110 to 250 psi) and is stored in storage tanks. Stored air is regulated to 759 kPa (110 psi) and piped to the air motor. A check valve between the compressor and the air receiver is good practice, to protect against a failure of plant air which might deplete the air receivers supply. The air compressors are driven by external power sources.

Air starter air supply piping should be short and direct and at least equal in size to the motor intake opening. Black iron pipe is preferred. The piping requires flexible connections at the starter. Deposits of oil and water will accumulate in the air receiver and at low spots in the piping. The accumulation of oil and water must be removed daily to prevent damage to the starting motors. Manual or automatic traps should be installed at the lowest parts of the piping and all piping should slope toward these traps.

Air tanks are required to meet specific characteristics, such as the specifications of the American Society of Mechanical Engineers (ASME). Compressed air storage tanks must be equipped with a maximum pressure valve and a pressure gauge. Check the maximum pressure valve and pressure gauge often to confirm proper operation.

Air Storage Tank Sizing (except 3600 Engines)

Many applications require sizing air storage tanks to provide a specified number of starts without recharging. This is accomplished as follows:

$$V_t = \frac{V_s \times T \times P_a}{P_t - P_{min}}$$

Where:

V_t = Air Storage Tank Capacity (cubic feet or cubic meters)

V_s = Air consumption of the starter motor (m^3/sec or ft^3/sec)–See *Air Starting Requirements Chart*.

T = Total Cranking Time Required (seconds) if six (6) consecutive starts are required, use seven (7) seconds for first start (while engine is cold), and two (2) seconds each for remaining five (5) starts, or a total cranking time of seventeen (17) seconds.

P_a = Atmospheric Pressure (psia or kPaa) normally, atmospheric pressure is 14.7 psia or 101 kPaa.

P_t = Air Storage Tank Pressure (psia or kPaa). This is the storage tank pressure at the start of cranking.

P_{min} = Minimum Air Storage Tank Pressure Required to Sustain Cranking at 100 rpm (psia or kPaa)–See *Air Starting Requirements Chart*.

Cranking Time Required

The cranking time depends on the engine model, engine condition, ambient air temperature, oil viscosity, fuel type, and design cranking speed. Five to seven seconds is typical for an engine at 26.7°C (80°F). Restarting hot engines usually take less than two seconds.

Air Consumption of the Starter Motor

The starter motor air consumption depends on these same variables and also on pressure regulator setting. Normal pressure regulator setting is 690 kPa (100 psi). Higher pressure can be used to improve starting under

Air Starting Requirements				
	Air Consumption of the Air Start Motor Versus m^3/sec (ft^3/sec) of Free Air Air Storage Tank Pressure -Pt-			Minimum Tank Pressure
Engine Model	793 kPaa (115 psia)	965 kPaa (140 psia)	1137 kPaa (165 psia)	-Pmin-
	690 kPag (100 psig)	862 kPag (125 psig)	1034 kPag (150 psig)	kPaa (psia)
3304	.16 (5.8)	.20 (6.8)	.21 (7.7)	345 (50)
3306	.17 (5.9)	.20 (6.8)	.22 (7.8)	352 (51)
3176/96	.17 (6.2)	.21 (7.3)	.23 (8.3)	379 (55)
3406	.17 (6.2)	.21 (7.3)	.23 (8.3)	379 (55)
3408	.18 (6.4)	.21 (7.3)	.24 (8.6)	372 (54)
3412	.25 (9.0)	.29 (10.3)	.33 (11.8)	310 (45)
3508	.26 (9.3)	.30 (10.8)	.36 (12.6)	310 (45)
3512	.28 (9.8)	.32 (11.4)	.38 (13.3)	345 (50)
3516	.30 (10.5)	.34 (12.1)	.40 (14.1)	448 (65)

Note: For engines equipped with pneumatic prelube: add .03 m^3/sec (1 ft^3/sec) air consumption.

adverse conditions up to a maximum of 1034 kPa (150 psi) to the starting motor. The values shown on the air requirements chart assume a bare engine (no parasitic load) at 10°C (50°F).

Operation

The supply of compressed air to the starting motor must be shut off as soon as the engine starts to prevent wasting starting air pressure and prevent damage to starter motor by overspeeding.

Air Storage Tank Sizing (3600 Engines)

The 3600 Air Start Sizing Curve shows required tank volume (for inline or vee engines) versus desired number of starts for different initial tank pressures. The curves for 1600 kPa (230 psi) and less allow for 6% pressure drop between tank and starter. For pressures greater than 1600 kPa (230 psi), the curves assume regulation to 860 kPa (125 psi) pressure at the starter. The pressure regulator should have capacity to flow 280 liters/second (600 scfm) per starter at regulator inlet pressures above 415 kPa (60 psi).

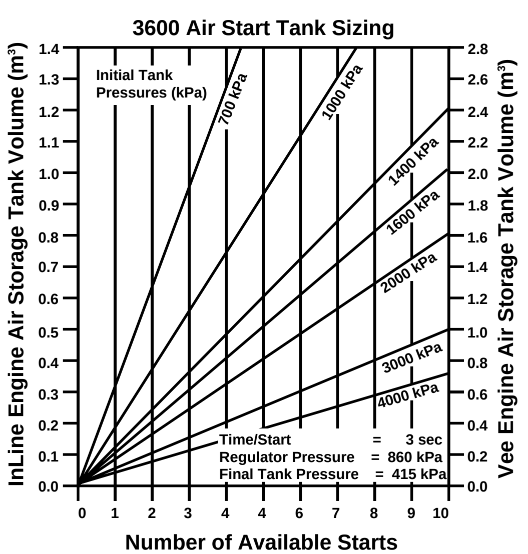


Figure 3.6

Air starters for 3600 Engines are designed to operate at a maximum air pressure of 1550 kPa (225 psi) at the starter. Minimum air pressure to provide adequate cranking speed for engine starting at 25°C (77°F) ambient temperature is 415 kPa (60 psi) at the starter.

The air pressure to the starter must be regulated when tank pressures greater than 1550 kPa (225 psi) are used.

These curves do not include air for a prelube pump.

The Caterpillar air prelube pump consumption rate is 28.2 L/s (60 scfm).

If the engine will be started at ambient temperatures lower than 25°C (77°F), additional storage tank volume will be required.

Prevention of Frost at the Starter Motor Air Discharge

Water vapor in the compressed air supply may freeze as the air is expanded below 32°F (0°C). A dryer at the compressor outlet or a small quantity of alcohol in the starter tank is suggested.

Hydraulic

Hydraulic starting provides high cranking speeds and fast starts. It is relatively compact. Recharging time, using the small engine-driven recharging pump, is fast. The system can be recharged by a hand pump provided for this purpose, although hand recharging is very laborious. The high pressure of the system requires special pipes and fittings and extremely tight connections. Oil lost through leakage can easily be replaced, but because of high pressures in the accumulators (usually 20,700 kPa [3000 psi] when fully charged) recharging the accumulator/s requires special equipment.

Repair to the system usually requires special tools.

Hydraulic starting is most often used where the use of electrical connections could pose a safety hazard.

Hydraulic starting systems are not available from Caterpillar. Contact your local Caterpillar dealer for the nearest available supplier.

The Hydraulic Accumulators, if used, contain large amounts of stored mechanical energy. They must be very carefully protected from perforation or breakage.

Starting Aids

The diesel engine depends on the heat of compression of the air in a cylinder to ignite the fuel. Below some specified temperature, the cranking system will not crank the engine fast enough or long enough to ignite the fuel. One or more commonly used starting aids, such as jacket water heaters and/or ether may be required.

Jacket Water Heaters

Jacket water heaters are electrical heaters which maintain the jacket water at a temperature high enough to allow easy starting of the engine. More heaters of higher ratings may be required in areas of extremely cold temperature.

Ether

Ether is a volatile and highly combustible agent. Small quantities of ether fumes added to the engines intake air during cranking reduce the compression temperature required for engine starting. Caution is required when using ether to prevent spread of fumes to atmosphere. A proper ether system will meter the rate of ether consumption. Not more than 1 cm^3 (0.033 oz) of ether should be released per 100 rated hp for each 10 seconds of cranking. Very low ambient temperatures may require increasing the ether consumption rate. *Under no circumstances should ether be released into an engine while running.*

The 3126 Marine Engine is equipped with an air inlet heater. Under no circumstances should ether be used on this engine.

Manifold Heaters

Heat added to the intake manifold of an engine during cranking will significantly improve startability and reduce any white start-up smoke. Manifold heaters are not available from Caterpillar.

Starting Smoke

High performance engines are prone to have some white start-up smoke. The white smoke is composed of unburned fuel*. Caterpillar Engines have been designed to minimize this problem.

Operators can do several things to improve the situation:

- Use of jacket water heaters to raise the engine water temperature to 32-49°C (90 to 120°F) prior to starting.
- Keep warm-up idle speeds (rpm) low.
- Warm the air to the air cleaners and intake manifold.
- Increase the exhaust pipe restriction-flow (exhaust back pressure) during the warm-up period.

*Diesel engines which are designed to have high output power, yet still be relatively light weight, generally have low compression ratios; i.e., in the range of 12.5 to 16:1. This design factor makes them prone to misfire and run rough until the engine reaches normal operating jacket water temperatures: 80 to 93°C (175 to 200°F).

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Start-up Systems

General Information

Testing/Instruments and Accessories

General Information

Start-up is the *final examination* at the end of the engine installation process.

The start-up is:

- To confirm the engine is operating properly.
- To confirm the boat's systems associated with the engine (exhaust, cooling, ventilation, fuel, starting, etc.) are adequate to allow the engine to operate up to its full potential.
- To determine the causes of performance deficiencies in the engine or boat.

Testing/Instruments and Accessories

The following tools are recommended for measurement of the data for start-up and associated performance testing.

Tachometer

Used to measure the speed (rpm) of the engine.

Caterpillar Part Number—6V3121 Multitach Group, See Special Instruction, SEHS7807

Pressure Measuring Device

Used to measure the pressure of the fluids (air, oil, water, and fuel) in many engine systems. There are both high and low limits on the pressure of all the fluids.

Caterpillar Part Number—6V9450 Pressure Group, six vacuum/pressure gauges permit a check of air cleaner restriction, oil pressure, manifold pressure, and fuel pressure. Provides readings in psi and kPa. Pressure range—100 kPa to 1000 kPa (15 psi to 150 psi), See Special Instruction SEHS8524.

Temperature Measuring Device

Used to measure the temperature of the fluids (air, oil, water, and fuel) in many engine systems.

Caterpillar Part Number–9S9102/8T0470 Thermistor Groups, Temperature Range—30°C to 1370°C (–20°F to 2500°F), See Special Instruction SMHS7140 and SEHS8446.

Fuel Density Measuring Device

Used to measure the relative amount of energy in the fuel. The heavier a quantity of fuel is, the more energy it contains and the more power it will generate in an engine.

Caterpillar Part Number—lP7408 Thermohydrometer and 1P7438 Beaker are used to measure the API gravity and temperature of diesel fuel, so corrected horsepower can be calculated.

Pyrometer

Used to measure the engine's exhaust temperature.

Caterpillar Part Number–lP3060 Pyrometer Group, Temperature Range = 18°C to 870°C (0°F to 1600°F), See Special Instruction SEHS7807.

Pitchometer

Used to measure the propeller's pitch; confirming their accuracy.

Caterpillar Part Number—8T5322 Pitchometer accurately measures propeller blade pitch either on or off the propeller shaft. Pitch can be measured at any shaft angle and at any point along the length of the blade, even under water. See Special Instruction SEHS8643.

Probe Seal Adapters

Used to allow measurement of the various pressures and temperatures of the engine's systems without need to shut down and drain the fluids therein. Works like the valve in a soccer ball. The adapters can be permanently installed and are equipped with a hex head plug to eliminate leakage and debris accumulation.

Caterpillar Part No. 5P2720 1/8 in.-27 NPT Thread
5P2725 1/4 in.-18 NPT Thread

See Special Instruction SMHS7140.

Water Manometer

Measures the exhaust backpressure and intake duct or air cleaner restriction to flow.

Caterpillar Part Number—8T0452 Water Manometer Can be locally made from 1550 mm (5 ft) length of flexible clear plastic 3/8 in I.D. tubing. See Special Instruction SEHS8524.

Fuel Flow Measuring Device

Used in conjunction with the tachometer (engine speed), pressure measuring device (intake manifold pressure), and pyrometer (exhaust temperature) to determine the power demanded of the engine.

Caterpillar Part No. lU5430 Fuel Monitor Arrangement for use with 3100, 3208, 3300, and 3400 Engine Families. Fuel flow range 11.4 L/h to 264 L/h (3 U.S. gal/h to 70 U.S. gal/h). lU5440 Fuel Monitor Arrangement for use with 3500, 3606, and 3608 Engines fuel flow range 151.4 L/h to 3785 L/h (40 U.S. gal/h to 1,000 U.S. gal/h).

Boat Speed Measuring Device

Hand-held radar (like that used by traffic law enforcement officers to check vehicle speed) is effective for this purpose.

CATERPILLAR®

Serviceability

Overhead Clearance for Connecting Rod and Piston Removal

General Information

Well-designed engine compartments will include features which contribute to the serviceability of the machinery. For example, overhead lifting equipment, some engine subassemblies may be heavier than one man can safely lift by hand, particularly in the often close quarters of the machinery space.

Hatches located directly above engines, for simplified removal and reinstallation during overhaul.

Outlets of electricity and compressed air to drive high production mechanic's tools.

Access to those points on the engine which require periodic preventive maintenance such as:

- lube oil filters and drain plug—engine and transmission
- fuel and air filters
- sea water and jacket water pump
- turbochargers
- zinc plugs
- heat exchanger—for core cleaning

Overhead Clearance for Connecting Rods and Piston Removal

The following table gives the height above the crankshaft centerline requirements to allow removal of a connecting rod and piston from the engines. This information is offered to assist designers who wish to provide adequate overhead clearance for piston/connecting rod removal.

In-Line Engines	
Engine	**Height Above Crankshaft Center**
3116, 3126, 3126B	626 mm (24.6 in.)
3304B, 3306B	711 mm (28 in.)
3176, 3196	635 mm (25 in.)
3406B, C&E	786 mm (30.9 in.)
D353	1127 mm (44.37 in.)
3606, 3608	2087 mm (82.16 in.)*

* Distance to remove cylinder liner

Vee Engines	
Engine	**Height Above Crankshaft Center**
3408B&C, 3412C&E	693 mm (27.28 in.)
D346, D348, D349	913 mm (35.94 in.)
D379, D398, D399	1234 mm (48.58 in.)
3508, 3512, 3516	969 mm (38.15 in.)
3612, 3616	1891 mm (74.49 in.)*

* Distance to remove cylinder liner

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Marine Engines Application and Installation Guide

- Cooling Systems

LEKM7145-01 (Supersedes LEKM7145) (05-00)

CATERPILLAR®

Cooling Systems

General Information
Engine Cooling Circuits
System Coolers
Expansion Tanks
Cooling System Protective Devices
Emergency Systems
Central Cooling Systems
System Pressure Drop
Corrosion
Useful Tables to Designers of Cooling Systems

General Information

A properly controlled cooling system is essential to satisfactory engine life and performance. Defective cooling systems and careless maintenance are the direct cause of most engine failures. The factory-supplied cooling system should not be modified since the various circuits of the engine-mounted cooling system have been sized to provide the proper flows to its components. Changes to these circuits can cause flow balance to be disrupted to the point that various engine components may fail.

Need for Cleanliness

All pipe and water passages external to the engine must be cleaned before initial engine operation to ensure there will be flow and that foreign materials will not be lodged in the engine or cooler.

Flexible Connectors

Customer supplied coolant piping **must be attached to the engine with flexible connectors**. The positions of flexible connections and shut-off valves are important considerations. The shut-off valve should be located so that a broken flexible connection can be isolated without having to shut down the whole system.

Engine Cooling Circuits

Caterpillar marine engines generally use one or two cooling water circuits. A closed treated water cooling circuit is always used for cooling the engine jacket. A second circuit is used on turbocharged aftercooled engine arrangements when colder than jacket water aftercooling is required. Cooling of the marine transmission oil is accomplished using either engine jacket water, aftercooler water, or a separate water cooling circuit depending on the model of marine transmission and/or the engine cooling arrangement.

Jacket Water System

Definition

Caterpillar Marine Engines are designed to operate with a jacket water temperature differential of approximately 8°C (15°F) measured across the engine under full load. Minimum jacket water temperatures are controlled by water temperature regulators (thermostats) to provide efficient engine operation. Maximum jacket water temperature limits are controlled by the size of the coolers and flow of coolant. The closed jacket water system consists of engine water jacket (engine block and cylinder heads), the circulating pump, water temperature regulator, oil cooler, engine-mounted expansion tank and heat exchanger.

Water Temperature Regulators

The water temperature regulator (thermostat) and cooler bypass are used to regulate operating temperature. The regulator directs all or part of the water discharged from the engine jacket to the cooler. The remainder is bypassed to the expansion tank on heat exchanger/keel cooled engines or to the water pump inlet on radiator cooled engines where it mixes with cooled water before returning to the engine jacket.

Depending on the engine and configuration, the thermostats may be in a controlled inlet or controlled outlet configuration. The operating temperature of the jacket water will be about the same for either system if the thermostat settings are the same or similar. In either the outlet or inlet control system, thermostat placement and sensing of jacket water temperature (and therefore bypass control) is always at the jacket water outlet.

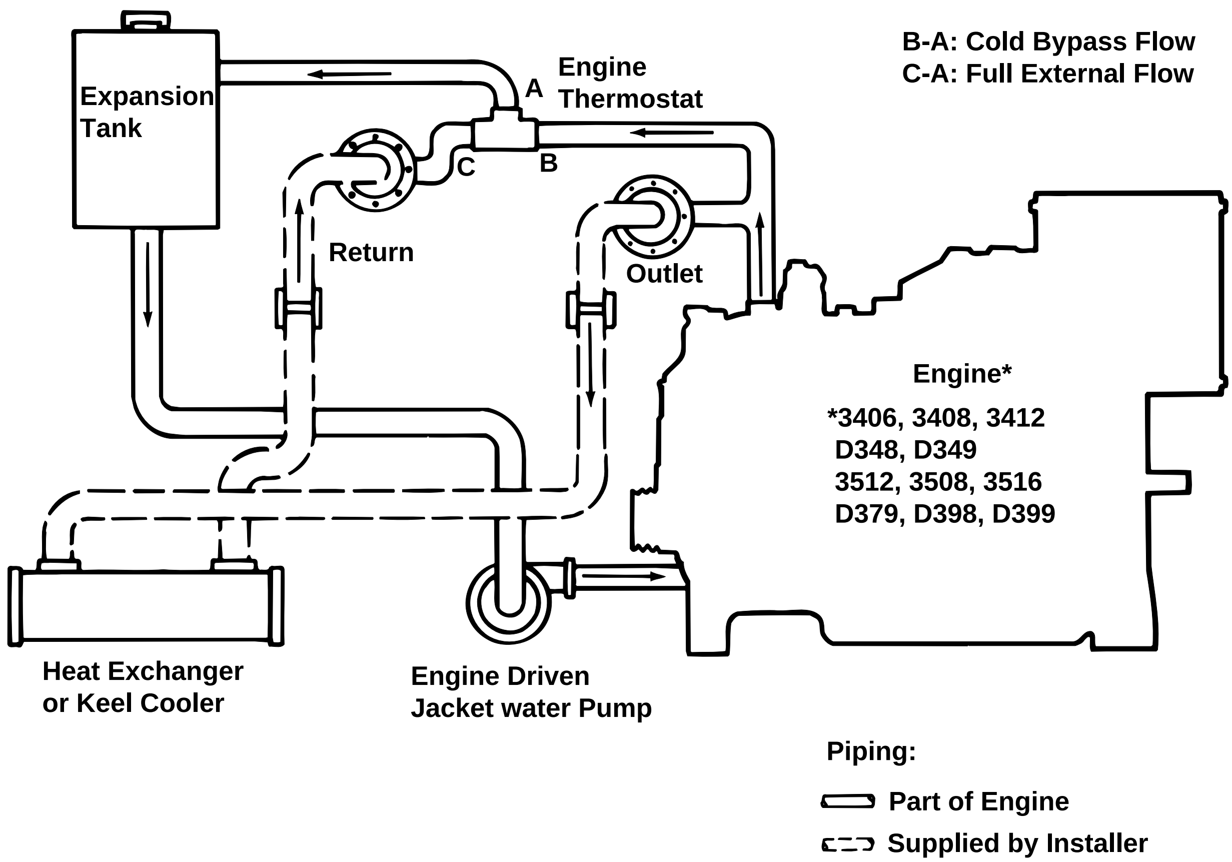


Figure 1.1

— EXPANSION TANK —
CONTROLLED INLET THERMOSTATS

The expansion tank and cooler perform the same function as the radiator. A radiator fan provides air flow through the cooling fins of the radiator to transfer coolant heat to the air. An external water supply is used to accomplish heat transfer when using a heat exchanger or keel cooler.

The inlet temperature controlled system provides less cycle temperature variation because mixing of the by pass jacket water and the cooled water takes place in the expansion tank before passing to the jacket water pump. The volume of water already in the expansion tank dilutes and smooths the temperature change rate.

With the simpler outlet control system, mixing occurs at the water pump inlet and temperature change (or cycles) may be more sudden and drastic. This can pose serious problems where the sea water is very cold and may require some special trimming or modification of the vessels cooling circuit.

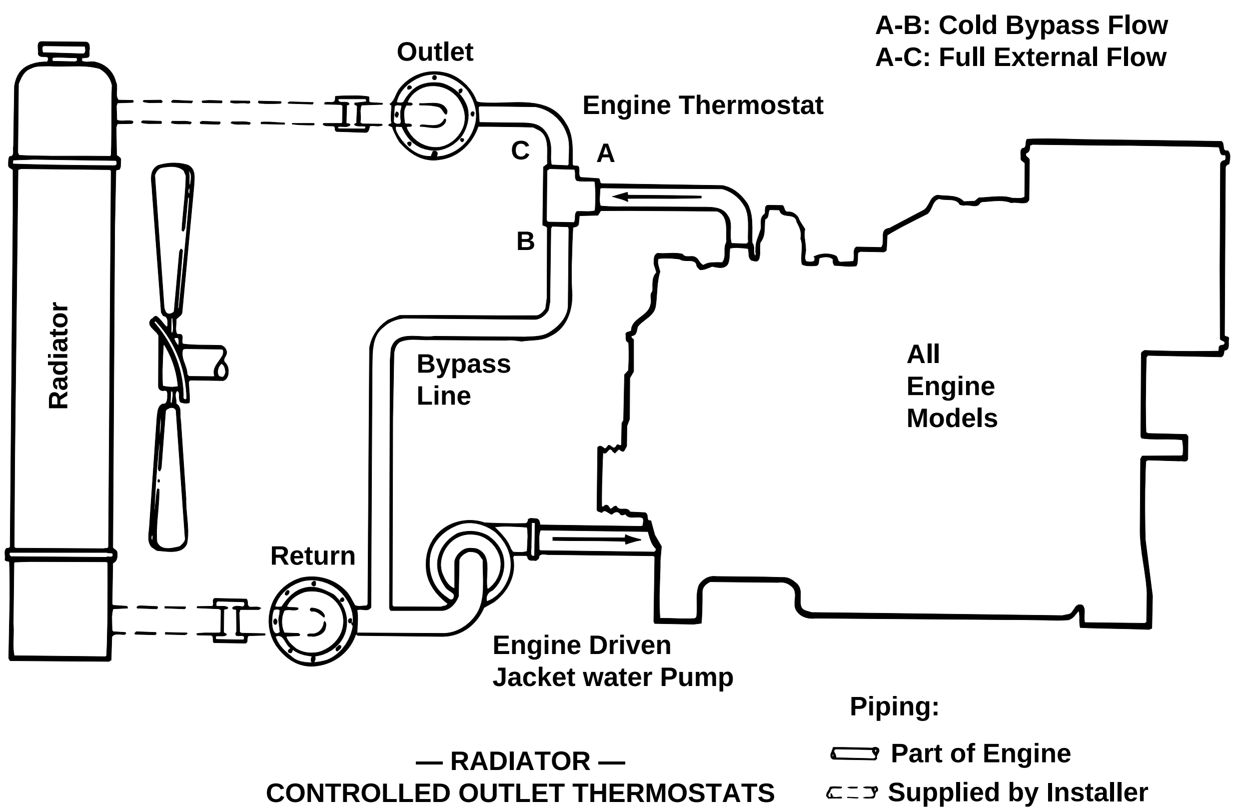
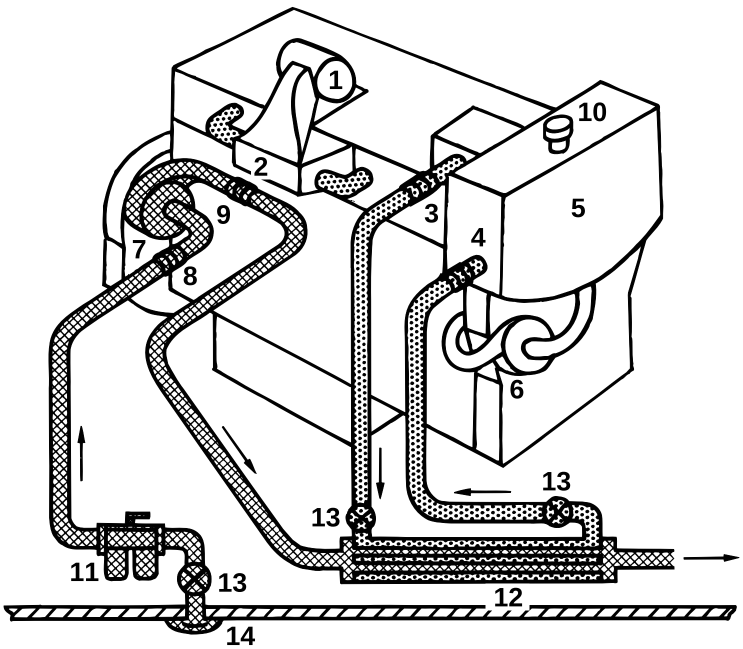


Figure 1.2

Heat Exchanger Cooling for Jacket Water

Heat exchangers can be mounted either on the engine or remote from the engine. Engine-mounted heat exchangers require the least amount of pipe fitting since the jacket water connections to the heat exchanger are provided by the factory.

Remote-mounted heat exchangers require connecting the jacket water inlet and outlet at the engine to the shell side of the exchanger. As shown below, an engine driven seawater pump is used to circulate the cooling water through the tubes of the heat exchanger.



JACKET WATER AFTERCOOLED
Heat Exchanger

1. **Turbocharger**
2. **Aftercooler, jacket water cooled**
3. **Jacket water outlet connection**
4. **Jacket water inlet connection**
5. **Expansion tank**
6. **Jacket water pump**
7. **Auxiliary pump, seawater**
8. **Seawater inlet connection**
9. **Seawater outlet connection**
10. **Pressure cap**
11. **Duplex full-flow strainer**
12. **Heat exchanger**
13. **Shut-off valve**
14. **Seawater intake**

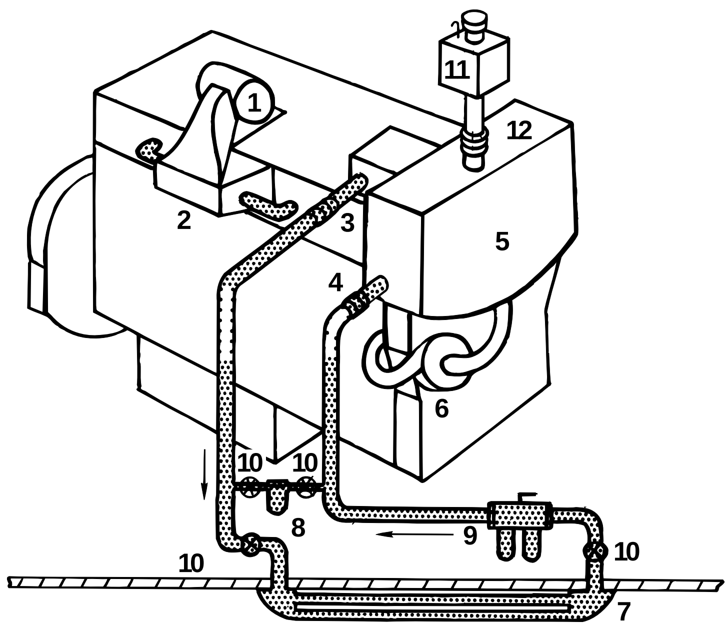
Figure 1.3

Keel Cooling for Jacket Water
A keel or skin cooler is an outboard heat exchanger which is either attached to the submerged part of a vessel's hull or built as a part of it. Jacket water is generally circulated through the cooler by the engine's water pump.

Water Specifications
Caterpillar used two water classifications: fresh water and seawater.

Fresh Water
Fresh water refers to drinkable water. Prior to chemical water treatment for engine corrosion inhibiting, it must be in a pH range of 5.5 to 9.0, containing no more than 40 ppm chlorides. Total dissolved solids must be less than 340 ppm. Total sulfates must be no more than 100 ppm. Total hardness must be less than 170 ppm. This is the cooling water that is used within the engine's jacket water system.

Seawater
Seawater refers to salt water, river water, lake water and all waters that do not meet the fresh-water requirement. Heat exchanger components in contact with this water should be copper-nickel construction, or equivalent, highly corrosion resistant material. This is not the water retained within the engine's jacket water system.



JACKET WATER AFTERCOOLED
Keel Cooler

1. **Turbocharger**
2. **Aftercooler, jacket water cooled**
3. **Jacket water outlet connection**
4. **Jacket water inlet connection**
5. **Expansion tank**
6. **Jacket water pump**
7. **Keel cooler**
8. **Bypass filter**
9. **Duplex full-flow strainer**
10. **Shut-off valve**
11. **Auxiliary expansion tank**
12. **Flexible connection**

Figure 1.4

Chemical Water Treatment for Engine Corrosion Inhibiting

All jacket water must be treated with chemicals for satisfactory engine life. Even potable water is not suitable for use by itself, except for short periods of time (during sea trials or during an emergency). It is good practice to chemically protect or drain engine jacket water before storage or extended transportation of the engine. This is necessary to avoid corrosion and scale from forming in the system. The resulting cooling solution (mixture of proper pH water and corrosion inhibitors) should have a pH in the range of 5.5 to 9.

Water Softener-Treated Water

Water that has been softened (chemically treated to lower the mineral content) by the addition of chlorides cannot be used in the cooling system. Water that is softened by the removal of calcium and magnesium can be used.

Chromate Corrosion Inhibitors

Inhibitors containing chromate compounds should not be used in Caterpillar Engine cooling systems. The concentration of chromate solutions is difficult to control and these solutions are extremely toxic. Before dilution, they can damage human skin. State and local regulations severely limit discharge of chromate solutions into inland and coastal waters.

Soluble Oil

Soluble oil is not recommended for cooling system protection.

Antifreeze

The climate where the engine will be used will determine the need for antifreeze. If antifreeze is needed, Caterpillar dealers can recommend specific types. Corrosion inhibitors are required. Some antifreeze products do not contain corrosion inhibitors. If the antifreeze chosen does not contain corrosion inhibitors, corrosion inhibitors must be added separately. Some antifreeze solutions, because of their higher viscosity (thicker than plain water), will reduce the cooling systems heat transfer capacity. Do not use higher than required concentrations of antifreeze. This practice can cause engine overheating. Antifreeze does not lose its ability to give freeze protection, but the additives in the antifreeze wear out with time and the antifreeze will not give certain other protection to the cooling system. Replace antifreeze periodically or add maintenance quantities of the additives, as directed by the antifreeze manufacturer.

Watermakers, Domestic Water Heaters, Cabin Heaters

Watermakers, domestic water heaters and cabin heaters can put normally-wasted jacket water heat to work. This has the potential for recovery of approximately 15% of the fuel input energy.

Certain aspects of the engine cooling system must be thoroughly understood to avoid misapplication. For example, an engine will only produce significant amounts of waste heat if there is a significant load on the engine. Many engines in marine service are lightly loaded for large parts of their life and are poor choices for installation of watermakers, domestic water heaters, and cabin heaters. When an engine is lightly loaded, almost all of the engine’s jacket water flow goes through a bypass line, from the thermostat housing to the jacket water pump inlet, to maintain a constant high flow through the engine’s cooling passages.

Watermakers, domestic water heaters and cabin heaters can *overcool* an engine.

If the watermaker, domestic water heater, or cabin heater extracts too much heat from the flow of jacket water, the engine’s water temperature sensors/thermostats will sense the engine cooling jacket is operating at a dangerously low temperature. It will attempt to correct the condition by reducing the external flow of cooling water. If there are automatic controls on the watermaker, domestic water heater, or cabin heater, it may shut off having sensed insufficient flow for continued operation. This leads to a troublesome condition of repetitive starting and stopping of the watermaker, domestic water heater, or cabin heater. Automatic control of these devices has proven troublesome and is not recommended. Consider the use of auxiliary jacket water heaters so during periods of light engine load, adequate amounts of heat can be sent to the watermaker, domestic water heater, or cabin heater.

Cooling water piping to and from the watermakers, domestic water heaters, and cabin heaters must not allow entrained air/gases to collect. Trapped air/gases will displace the water required to carry engine heat to the watermakers, domestic water heaters, and cabin heaters and interfere with proper operation. Trapped air/gases can be vented by installing small, approximately 3 mm (0.125 in.), inside diameter-vent lines. The vent lines should carry air/gases from the high points in the domestic water heater and its associated piping to a higher point in the engine jacket water cooling circuit—normally an installer-supplied auxiliary expansion tank.

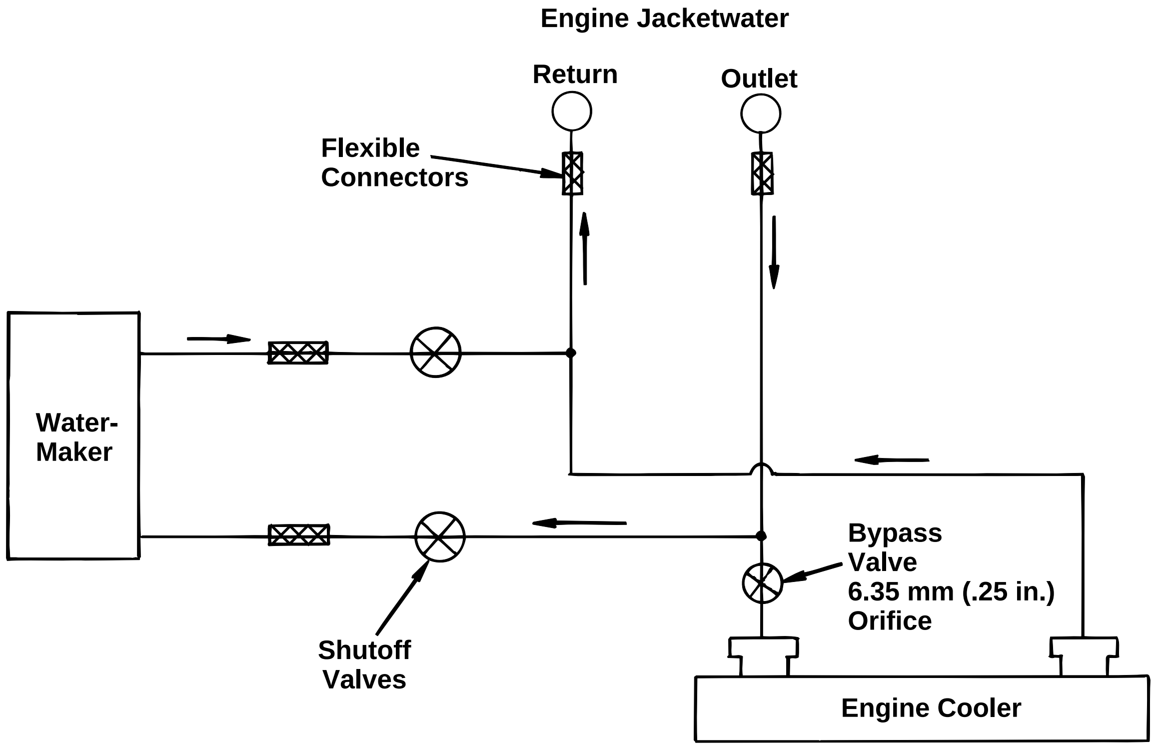
See the engine general dimension drawing for the connection locations of points on the engine where water for this purpose should be extracted and returned.

Watermaker Controls

The watermaker controls may be either manually operated valves or thermostatically controlled valves.

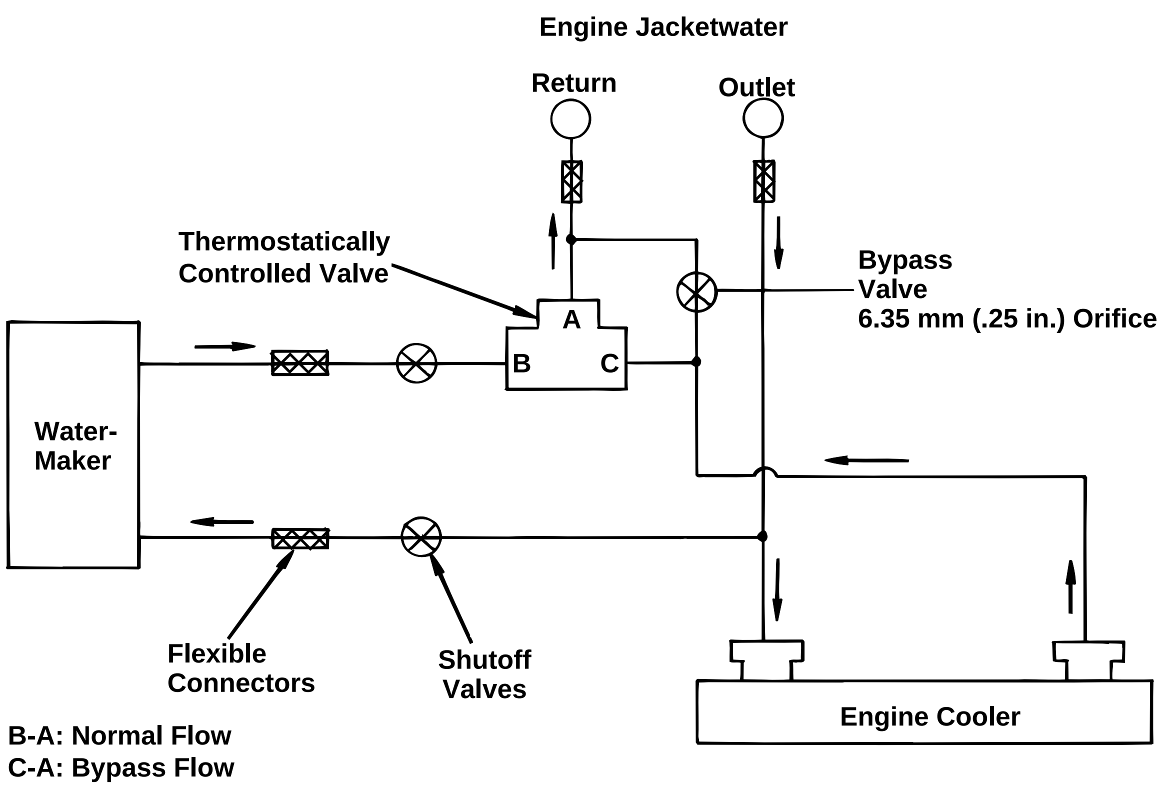
Any failure of the water maker control system (electrical, air, etc.) must shut off jacket water flow to the watermaker and return the flow to the engine heat exchanger.

The thermostat valve, shown in the figure describing automatic control watermaker circuit, would have a temperature setting that will not interfere with the engine thermostats. This valve should begin to divert water flow to the engine heat exchanger at no more than 88°C (190°F) and be fully diverting at 96°C (205°F). For safety, the bypass valve(s) in the engine heat exchanger circuit should contain 6.35 mm (0.25 in.) orifices so there will be a slight water flow in case all valves are inadvertently left closed. This orifice is then required to assure water flow to actuate an alarm system. If the watermaker cannot handle the full heat rejection of the engine and/or cannot handle the full water flow of the engine, the automatic system must be used.



MANUAL CONTROL WATERMAKER CIRCUIT

Figure 1.5



AUTOMATIC CONTROL WATERMAKER CIRCUIT

Figure 1.6

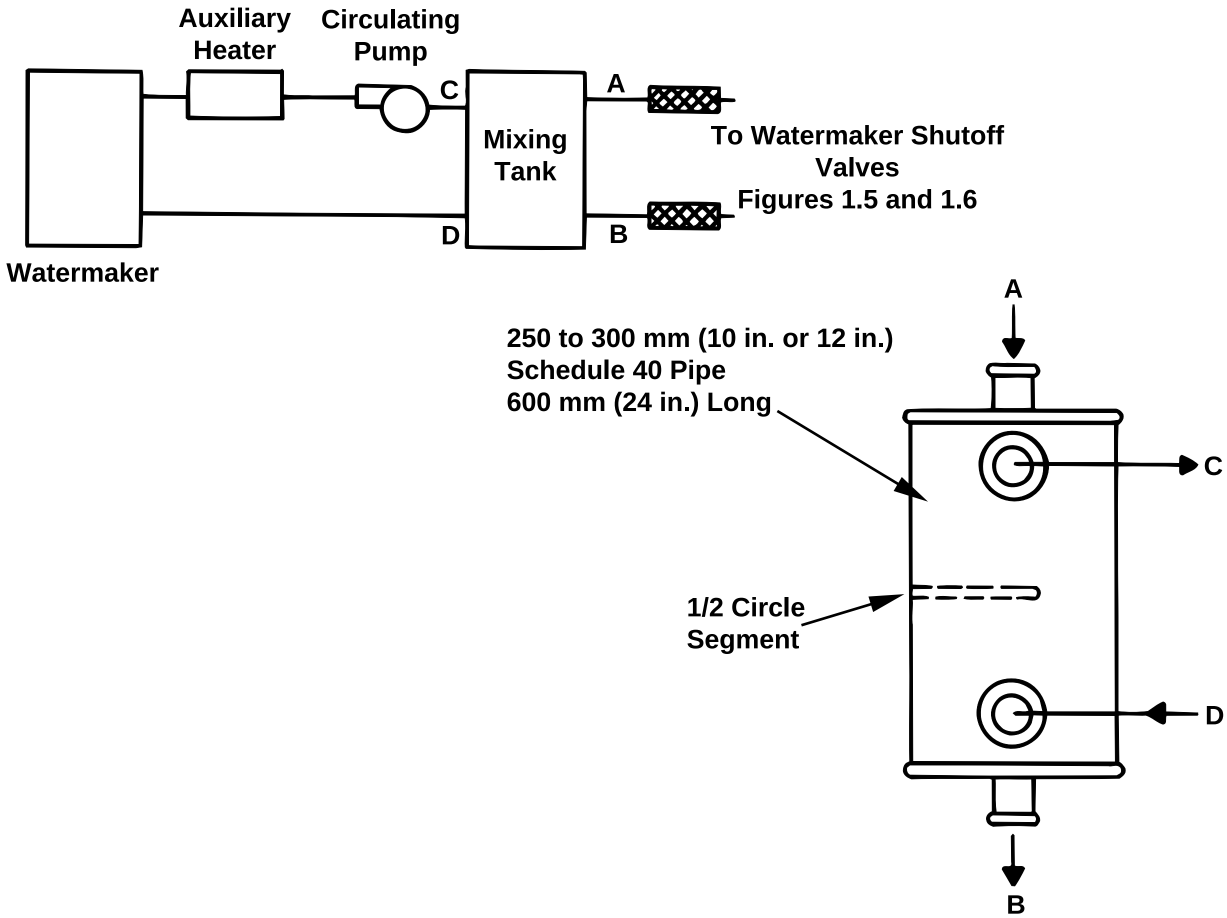


Figure 1.7

Interconnecting Engines

Several problems arise from interconnection of several engines: unequal water flow, one failure shuts down all engines, excessive external head pressures, etc. For these reasons, separate connection of one engine per watermaker is recommended. It is the customer's responsibility to provide a system that is compatible with the engine cooling system in all modes of operation.

When the Watermaker is Far from the Engine

When the watermaker is a long distance from the engine or where the watermaker requires a constant water flow, a mixing tank and circulating pump is required. Do not use a circulating pump by itself, because the circulating pump head pressure will damage the engine thermostats in the event they are closed. Although the mixing tank is not Caterpillar supplied, it can be used with either of the suggested circuits. An auxiliary electrical heater may be installed as shown.

Aftercooler Systems

Caterpillar uses two types of cooling circuits for the aftercooler. One type provides engine jacket water for cooling the air in the aftercooler. The other type provides a separate cooling circuit for the aftercooler. *All aftercooled Caterpillar Engines applied in a marine environment should be equipped with a seawater-type aftercooler core to assure satisfactory core life.*

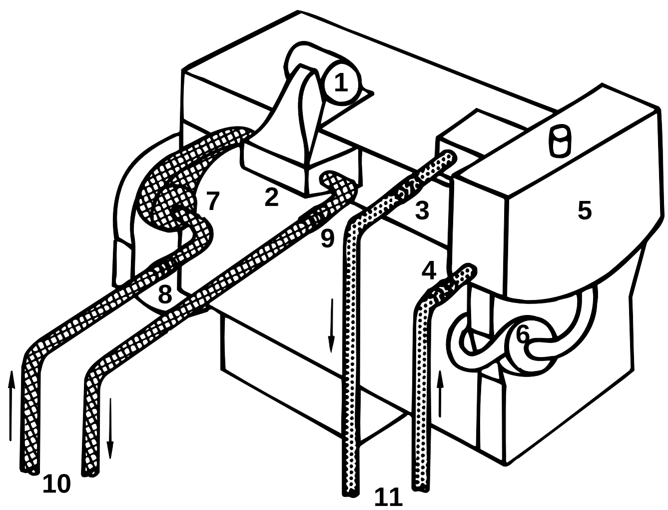
Jacket Water Aftercooling

Jacket water aftercooling uses engine jacket water in the tube side of the aftercooler and results in inlet manifold temperatures lower than those obtained in nonaftercooled turbocharged engines. The lower inlet manifold air temperature allows a jacket water aftercooled engine to achieve a rating higher than either a naturally aspirated or turbocharged-only engine. Jacket water aftercooled circuits are completely installed at the factory.

Separate Circuit Aftercooling

As the name implies, the separate circuit aftercooler circuit, SCAC, provides water to the aftercooler from a source other than engine jacket water. It is used to provide colder water to further reduce inlet manifold air temperatures.

The two arrangements of the separate circuit aftercooled engine configuration provide either an open seawater circuit or a closed fresh-water circuit for the aftercooler water.



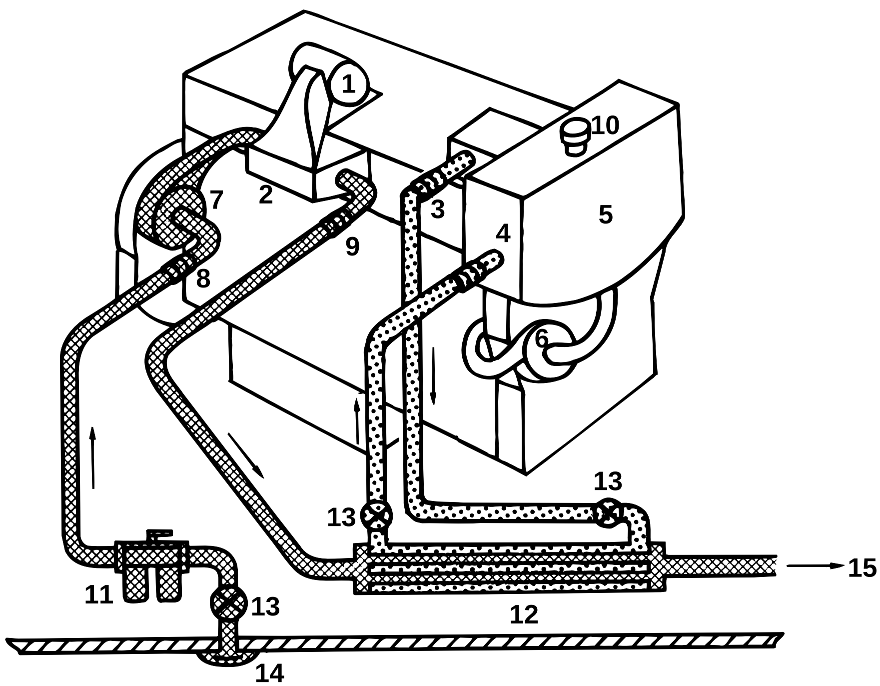
SEPARATE CIRCUIT AFTERCOOLED

1. **Turbocharger**
2. **Aftercooler, auxiliary water cooled**
3. **Jacket water outlet connection**
4. **Jacket water inlet connection**
5. **Expansion tank**
6. **Jacket water pump**
7. **Auxiliary water pump**
8. **Auxiliary water inlet connection**
9. **Auxiliary water outlet connection**
10. **Lines to aftercooler cooler**
11. **Lines to jacket watercooler**

Figure 1.8

Seawater Aftercooling

Engines equipped with seawater aftercoolers use untreated water in the tube side of the aftercooler. Seawater refers not only to salt water but also includes river water, lake water or any source of untreated water. Use of seawater for aftercooling achieves inlet manifold air temperatures lower than those resulting from jacket water or separate circuit fresh water aftercooling. This lower inlet manifold air temperature permits ratings of seawater aftercooled engines to exceed those for jacket water aftercooled engines.



SEPARATE CIRCUIT AFTERCOOLED
Seawater Aftercooled

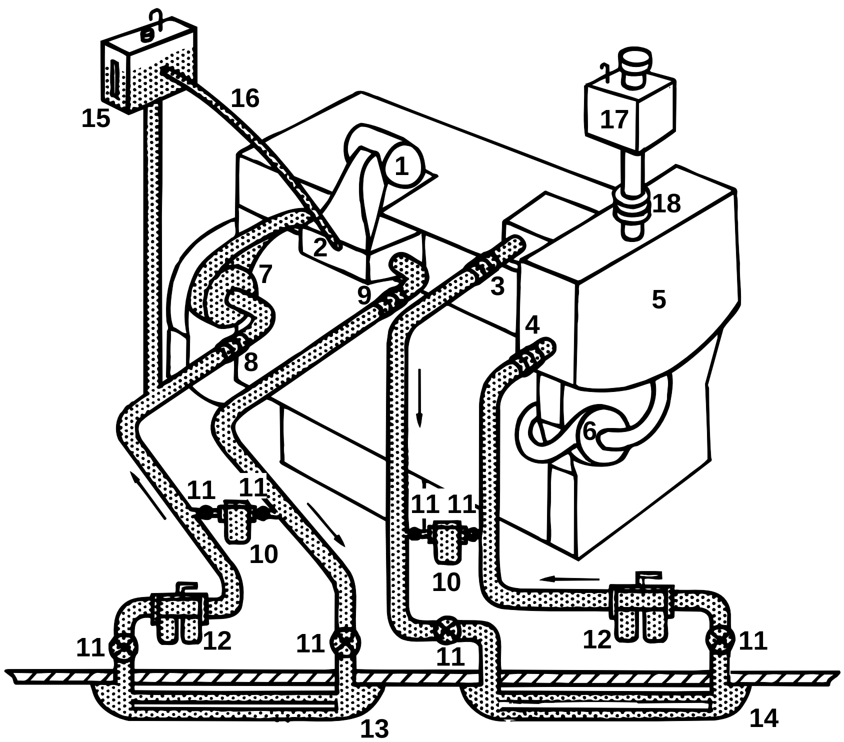
1. Turbocharger
2. Aftercooler, seawater cooled
3. Jacket water outlet connection
4. Jacket water inlet connection
5. Expansion tank
6. Jacket water pump
7. Auxiliary seawater pump
8. Auxiliary seawater inlet connection
9. Aftercooler outlet connection
10. Pressure cap
11. Duplex full-flow strainer
12. Heat Exchanger
13. Shut-off valves
14. Seawater intake
15. Seawater discharge

Figure 1.9

Separate Keel Cooler for Aftercooler

The use of keel or skin coolers in the aftercooler circuit allows a low temperature, fresh water closed circulating system to be used. All closed fresh water aftercooler circuits require the installation of an expansion tank. Refer to the section of auxiliary expansion tanks. The use of an inlet manifold air temperature gauge, or alarm, can provide guidance for required cleaning of the system in order to maintain the desired engine performance. The use of an inlet manifold air temperature sensing device is strongly recommended.

Caution must be used when using the aftercooler keel cooler water circuit to cool an auxiliary piece of equipment (i.e. marine transmission). The auxiliary equipment cooler should be connected to the water circuit after it leaves the engine aftercooler to avoid adding any heat to the water before it enters the aftercooler. The additional resistance of the auxiliary equipment cooling circuit must be held to a minimum to avoid reducing the flow of water to the aftercooler.



SEPARATE CIRCUIT AFTERCOOLED
Keel Coolers

1. Turbocharger
2. Aftercooler, keel cooled
3. Jacket water outlet connection
4. Jacket water inlet connection
5. Expansion tank
6. Jacket water pump
7. Auxiliary fresh water pump
8. Auxiliary fresh water inlet connection
9. Aftercooler outlet connection
10. Bypass filter
11. Shut-off valve
12. Duplex full-flow strainer
13. Keel cooler for aftercooler
14. Keel cooler for jacket water
15. Expansion tank for aftercooler circuit
16. Vent line for aftercooler circuit
17. Auxiliary expansion tank
18. Flexible connection

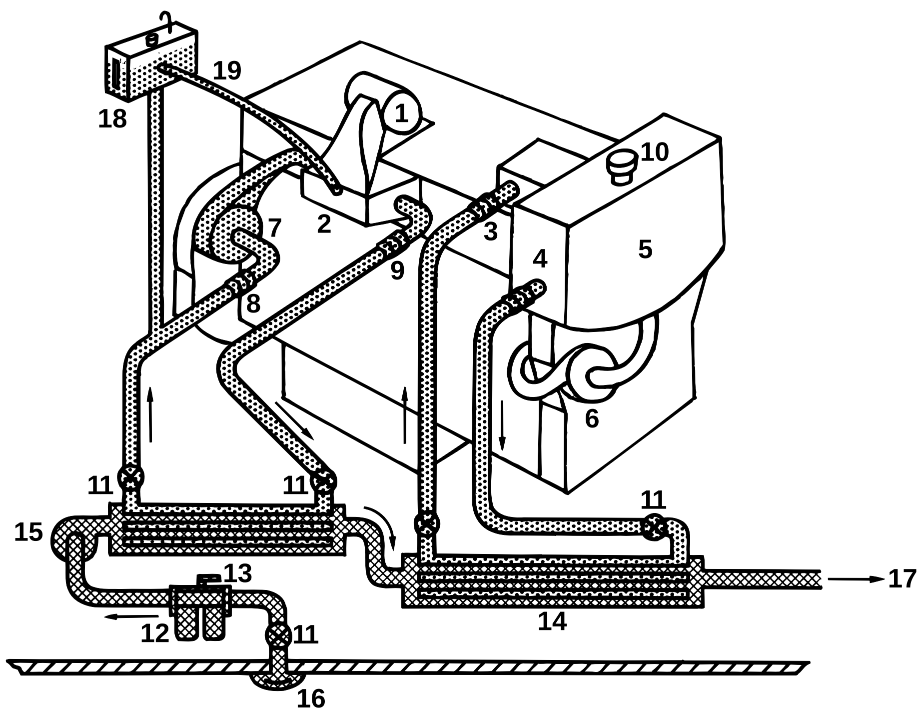
Figure 1.10

Heat Exchanger for Aftercooler

A shell and tube type heat exchanger will also provide cooling for fresh aftercooler water if the seawater temperatures are cold enough to provide adequate cooling. The use of an inboard shell and tube type heat exchanger for the aftercooler circuit requires the use of a seawater pump in addition to the freshwater pump used to circulate water through the aftercooler. An expansion tank is also required for the aftercooler circuit.

Overcooling of Aftercooler Air

The separate circuit aftercooler cooling system must be designed with sufficient capacity for the hottest water and the higher ambient air conditions for operation in climates where both air and seawater temperatures run to extremes. This results in a cooler with excess capacity in cold seawater and warm air conditions. This will result in condensation in the engine's intake system, especially during prolonged light engine load. Extremely cold seawater in the aftercooler can also cause condensation when engine inlet air temperatures are relatively warm with high moisture content. To minimize condensation during light engine load in separate circuit aftercooled systems, it is desirable to maintain the inlet manifold temperature between 38°C and 52°C (100°F and 125°F). This may be achieved by



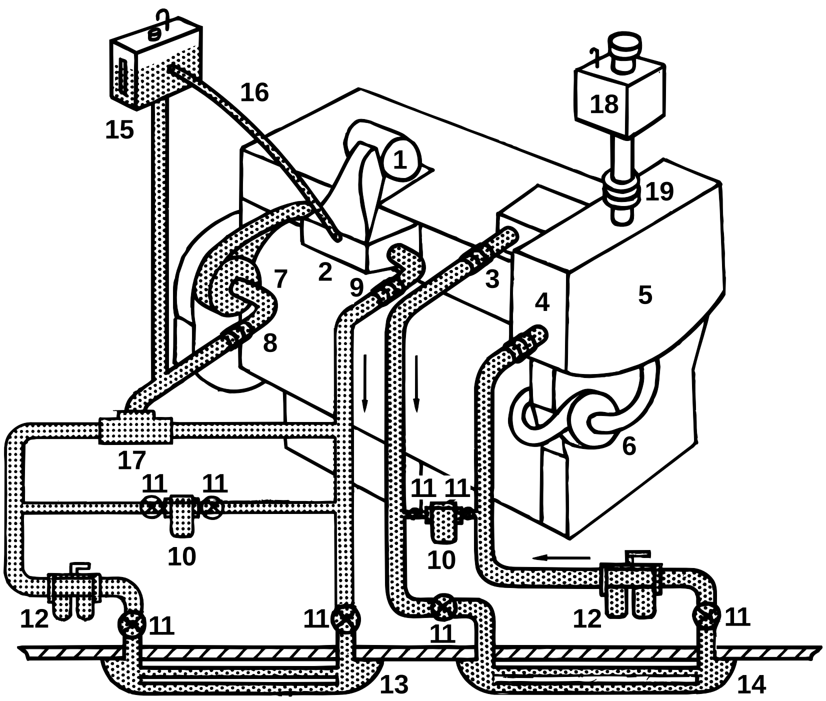
SEPARATE CIRCUIT AFTERCOOLER
Heat Exchangers

1. **Turbocharger**
2. **Aftercooler, heat exchanger cooler**
3. **Jacket water outlet connection**
4. **Jacket water inlet connection**
5. **Expansion tank**
6. **Jacket water pump**
7. **Auxiliary fresh water pump**
8. **Auxiliary fresh water inlet connection**
9. **Aftercooler outlet connection**
10. **Pressure cap**
11. **Shut-off valve**
12. **Duplex full-flow strainer**
13. **Heat exchanger for aftercooler**
14. **Heat exchanger for jacket water**
15. **Customer provided seawater pump**
16. **Seawater intake**
17. **Seawater discharge**
18. **Expansion tank for aftercooler circuit**
19. **Vent line for aftercooler circuit**

Figure 1.11

recirculating the aftercooler cooling water back to the auxiliary water pump inlet until the desired temperature is reached. Cool water should then be mixed with the recirculated water to maintain the temperature. The temperature of the water to the aftercooler can be controlled by using a thermostatically controlled three-way valve.

On closed circuit keel cooled or heat exchanger systems, the aftercooler water is bypassed around the cooler until it reaches the desired aftercooler inlet temperature. On seawater aftercooled engines, the warmed water from the heat exchanger is recirculated to the aftercooler until the desired aftercooler inlet water temperature is obtained. The thermostat valve used must be capable of being used continuously in seawater and be equipped with electrolytically compatible components.



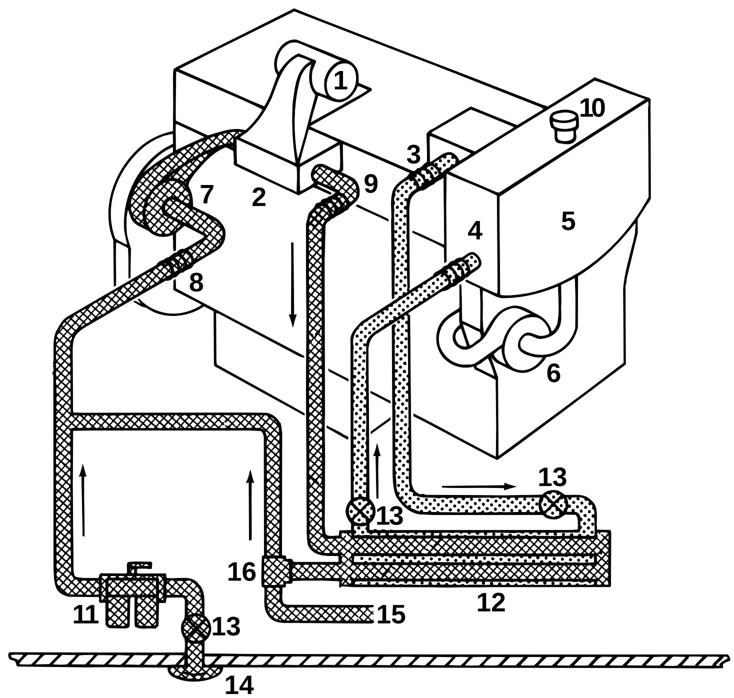
SEPARATE CIRCUIT AFTERCOOLED
Aftercooler Keel Cooler Bypass

1. **Turbocharger**
2. **Aftercooler, keel cooled**
3. **Jacket water outlet connection**
4. **Jacket water inlet connection**
5. **Expansion tank**
6. **Jacket water pump**
7. **Auxiliary fresh water pump**
8. **Auxiliary fresh water inlet connection**
9. **Aftercooler outlet connection**
10. **Bypass filter**
11. **Shut-off valve**
12. **Duplex full-flow strainer**
13. **Keel cooler for aftercooler**
14. **Keel cooler for jacket water**
15. **Expansion tank for aftercooler circuit**
16. **Vent line for aftercooler circuit**
17. **Bypass valve thermostatically controlled**
18. **Auxiliary expansion tank**
19. **Flexible connection**

Figure 1.12

The thermostatic valve used should not allow the temperature of the water to the aftercooler to exceed 30°C (85°F). The keel cooler, heat exchanger and marine transmission oil cooler used must be sized for this maximum temperature. A thermostatically controlled 3-way valve that is equipped with a remote sensor to monitor the inlet manifold air temperature can be used. Adjust the remote sensor to insure that the thermostatic valve does not permit recirculation when the inlet manifold temperature reaches 49°C (120°F).

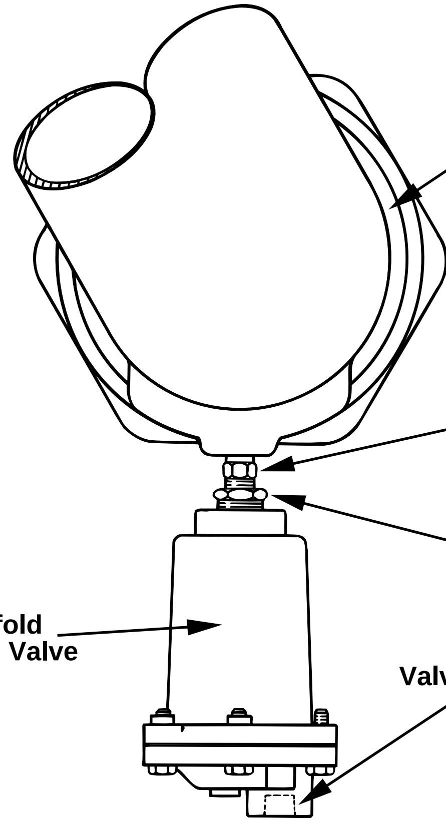
It is important that water be recirculated rather than be throttled to reduce flow. It is essential that unrestricted water flow through the aftercooler be maintained regardless of temperature conditions. Size thermostatic valve plumbing to have internal diameters as large or larger than the inlet connection of the auxiliary pump. Use an air intake manifold temperature alarm set for 52 to 57°C (125 to 135°F) maximum to warn of system malfunction.



SEPARATE CIRCUIT AFTERCOOLED
Aftercooler Seawater Recirculation

1. **Turbocharger**
2. **Aftercooler, seawater cooled**
3. **Jacket water outlet connection**
4. **Jacket water inlet connection**
5. **Expansion tank**
6. **Jacket water pump**
7. **Auxiliary seawater pump**
8. **Auxiliary seawater inlet connection**
9. **Aftercooler outlet connection**
10. **Pressure cap**
11. **Duplex full-flow strainer**
12. **Heat Exchanger**
13. **Shut-off valves**
14. **Seawater intake**
15. **Seawater discharge**
16. **Bypass Valve-themostatically controlled**

Figure 1.13



CONDENSATE VALVE GROUP

Figure 1.14

In situations where condensation can be a problem, a corrosion-resistant water trap can be attached to the intake manifold(s) of the engine.

Seawater System

General

The installation, size and material of the seawater suction lines is extremely important.

Size

Flow restriction in the sea water suction piping will result in abnormally high engine temperatures which can lead to unscheduled shutdowns and, in severe cases, reduced engine life. To minimize flow restriction, pipes and hoses should be at least as large as the sea water pump suction opening.

If the distance to the thru-hull fitting or sea chest is large or if many pipe elbows or bends in the hose are used, the pipe or hose size should be one size larger than the sea water pump opening (suction connection). In no case should the sea water pressure, measured at the sea water pump suction, be less than 24 kPa (3.5 psi) vacuum.

Suction Line Design Considerations

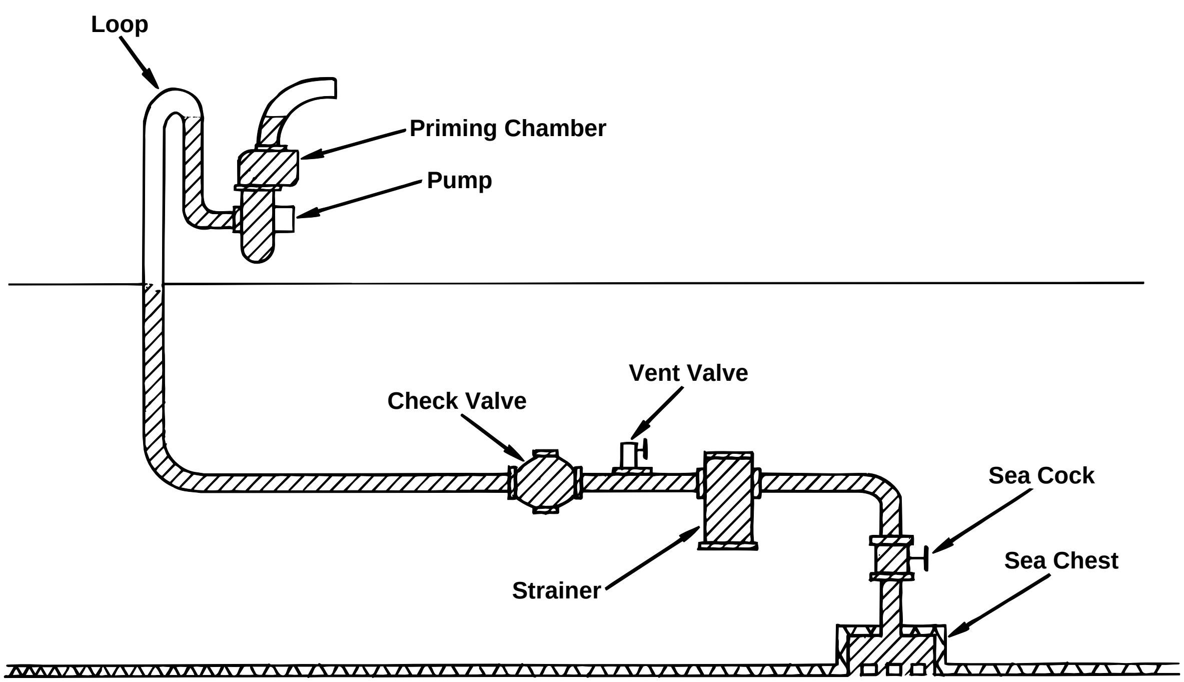
On the inlet side of the pump, as much as possible of the seawater piping, should be below the vessel water line without air traps.

Install a water pressure actuated check valve downstream of the strainer and as close to it as possible. The function of the check valve is to prevent water from draining out of the pump inlet while the pump is not operating and during cleaning of the strainer. Install a vent valve between the strainer and the check valve to allow venting of trapped air after cleaning the strainer and opening the sea cock. If the pump is above the vessel water line, install a piping loop above the pump inlet elbow to trap enough water to keep the pump and priming chamber filled.

Sea Water Inlet Design Considerations

The Sea water inlet serves the following functions:

- Provides a low restriction connection for the seawater inlet plumbing.
- Provides a connection point for the Sea Cock (seawater shutoff valve—installed between the sea water inlet and the seawater inlet plumbing).
- Provides a way to separate air from the sea water required for cooling. Sea chests must have vent connections to allow air, forced under the hull during maneuvering, to be purged *before* it is able to reach the centrifugal seawater pump.



CENTRIFUGAL SEAWATER PUMP INLET PLUMBING

Figure 1.15

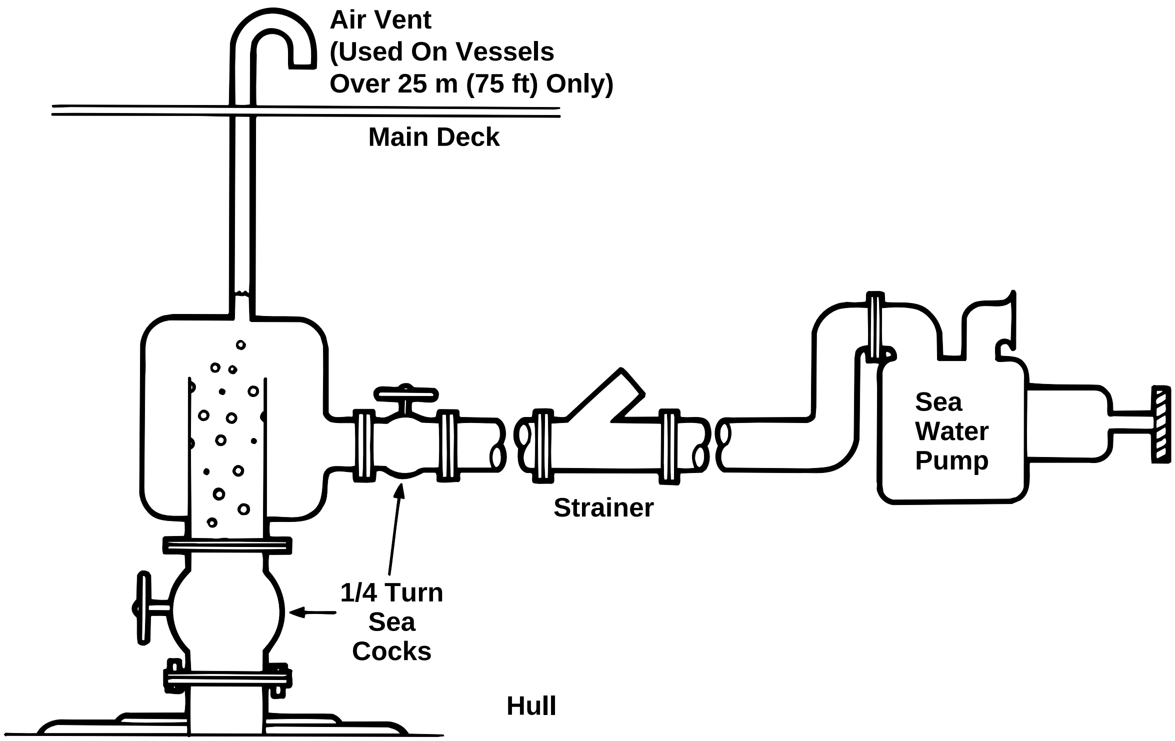


Figure 1.16

Seawater Pumps

Caterpillar offers three types of seawater pumps:

- Rubber Impeller
- Water Ring
- Centrifugal

Rubber Impeller Seawater Pumps

Rubber impeller seawater pumps are characterized by excellent priming characteristics, though they often suffer relatively short life in abrasive waters.

Water Ring Seawater Pumps

Their priming characteristics are less than rubber impellers, but can lift up to 1.5 m (5 ft) **Caution:** A goose neck may be necessary with these pumps to keep water in the pump for priming. They are made entirely of corrosion resistant metals, with no elastomeric components.

Centrifugal Seawater Pumps

Centrifugal Seawater Pumps must be installed with their inlet below the boats *light* waterline. Air allowed to enter centrifugal seawater pumps will likely result in loss of prime and probable engine damage due to loss of cooling. Do not start an engine equipped with a centrifugal pump unless the pump and priming chamber are full of water.

Material

An excellent material for piping carrying seawater is of the copper- nickel alloys. The cost of such piping makes its use unusual for all but the most critical systems.

The material of all the seawater piping should be the same, whenever practical. If parts of the seawater piping, made of different metals, make contact with each other, one of the metals will corrode, sometimes very rapidly.

The materials will corrode according to their position in the electromotive series. See electromotive series chart in section of Useful Tables to Designers of Cooling Systems.

Black iron pipe is often used in seawater service (replacement should be planned every two or three years). If it is necessary to use pipe or other cooling system components of more than one material, avoid letting the dissimilar metals touch, even by mutual contact with an electrically-conductive third material.

Corrosion will be much more severe if a flow of electrons is able to pass freely from one of the metals to the other.

Seawater Strainer

Purpose

Strainers protect the seawater pump, heat exchanger and other cooling system components from foreign material in the seawater. The foreign material can plug or coat heat transfer surfaces, causing overheating of the engine. If abrasive, foreign material will erode pump impellers and soft metal parts, reducing their effectiveness.

Location

Install strainers below the boat's water line and as close to the sea water intake or sea chest as possible (adjacent to the sea cock). The strainer must be installed so it can be easily cleaned, even in the worst weather conditions.

Type

While simplex strainers, which require shutoff of the sea water flow, are adequate to protect the engine, greater safety will result from using duplex strainers, which can be cleaned without interrupting sea water flow or engine power.

Size

Well sized strainers will impose no more than 9 kPa (3 ft of water) restriction to flow at full seawater flow conditions. Suppliers can help in the proper selection of strainer size by providing the flow restriction of each size of strainer at varying water flow conditions.

Mesh Dimensions

Recommended strainer media (screens) should not pass solid objects larger than 1.6 mm (1/16 in.) in diameter.Plate type heat exchangers require a mess less than 3 mm, while the tube type heat exchangers requires a mess less than 5 mm.

Strainer Differential Pressure Sensors

Schools of small fish, floating debris (plastic bags, plant material, etc.) or ice chips can plug a clean strainer in a few seconds. When the differential pressure across the connections of a strainer goes too high, the strainer needs to be cleaned. A differential pressure switch, will provide early warning of strainer plugging and resultant loss of engine cooling. In time, high engine water temperature alarms will also warn of a loss of sea water flow, but the differential pressure sensor will give early warning and the precise location of the problem.

Zinc Plugs

General Information

Zinc plugs are installed in portions of the engine where dissimilar metals must be used in the presence of seawater. Their sacrificial action protects critical cooling system components from corrosion.

Inspection Schedule

Inspect the zinc plugs within 24 hours of filling the piping with sea water. If no significant corrosion is noted, inspect them again after 7 days of sea water submersion. If no significant deterioration is noted, reinspect in 60-90 days. Thereafter, inspect annually and replace if necessary.

Thread Sealant on Zinc Plug Threads

Install zinc plugs with clean threads. Never install zinc plugs using teflon tape or nonelectrically- conductive pipe sealers. The insulating properties of such sealers will stop the protective action of the zinc plugs.

Marine Growth

Marine plants and animals will enter seawater piping and take up residence there. Many forms of sea life are very comfortable within engine cooling system piping and will grow to a size which will threaten adequate flow. The lack of predators, darkness and abundance of suspended food particles combine to create prime growth conditions for sponges, barnacles and like creatures. Strainers are no protection against creatures which are microscopic in size during their infant stages of life. Periodic operation in fresh water will rid boats of salt water life infestation and vice versa. In any case, it will be necessary to remove and clean piping and heat exchanger passages of the corpses. Use of high water temperature alarms, seawater pump pressure switches, and other instrumentation can warn of gradual loss of seawater flow and are recommended. Periodic chemical treatment combats marine growth. Chemical type and concentration must be controlled to prevent deterioration of the seawater cooling system components. Contact a knowledgeable chemical supplier. Continuous low-concentration chemical treatment via either bulk chemical or self- generating electrical processes are offered by various manufacturers.

Seawater Pump Maintenance

Flexible impeller seawater pumps require periodic service. The impellers must be replaced when worn to maintain adequate seawater flow and avoid engine overheating. It is a good idea to put a little soft soap, like that used by mechanics for hand cleaning, on the new impeller just prior to installing it. The soap will lubricate the new impeller long enough for it to fully achieve prime, protecting it from overheating. A spare impeller, for flexible impeller seawater pumps, should be carried on board at all times.

Stern Tube Lubrication/Cooling

It is good practice to divert a small portion of the engine's seawater, before discharging it overboard, to lubricate/cool the stern tube and stuffing box (sometimes called the *packing gland*). The engine's seawater has been strained and the flow of water from the stuffing box end of the stern tube will tend to keep sand or other abrasive material out of the stern tube. Avoid using excessive quantities of the engine's flow of seawater, as this practice tends to increase the seawater system restriction, making the engine more likely to overheat. Generally 4-12 L/min (1-3 gal/min) are adequate.

Potential Problems

Nonreinforced Seawater Pump Suction Hose

The vacuum inside the Seawater Pump Suction Hose can become quite high. If the hose is not internally reinforced, atmospheric pressure will collapse it. That will severely impede the flow of seawater with potentially dangerous results. Use hose which is sufficiently strong to resist collapse due to high suction vacuum.

Internal Hose Deterioration

Some hose will *shred* internally, releasing bits of rubber which can plug cooling passages. It is good practice to use good quality hoses. If users are unsure of their hoses' quality, it is good practice to examine hoses internally at least once during their life. Replace them with good quality hose every three years.

Achieving and Maintaining Seawater Pump Prime

Pump speeds and suction pressures must fall within certain limits for sea water pumps to achieve prime (start pumping water). The priming characteristics of Caterpillar sea water pumps are available from the factory.

Sea Water Discharge through Exhaust System

Wet exhaust systems use sea water, after it has passed through the various heat exchangers and coolers, to cool the hot exhaust gases. After sea water is injected into the hot exhaust gas (generally immediately downstream of the engine's turbocharger), the temperature of the gas is reduced enough to allow use of sections of rubber hose, fiberglass-reinforced plastic pipe or other similar materials to be used as exhaust pipe. **It is critical that nothing interfere with the flow of sea water which cools the exhaust gas. Anytime the engine is operating, the flow of seawater must be present.**

System Coolers

There are two types of heat exchanging systems recommended for use with the Caterpillar Diesel Marine Engines. These involve the use of either inboard mounted heat exchangers or outboard mounted keel coolers.

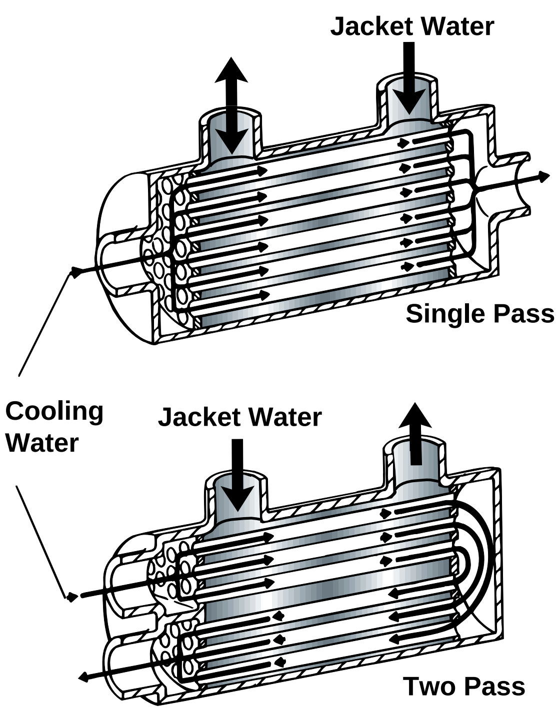
Heat Exchanger Cooling

Caterpillar inboard heat exchangers are shell-and-tube type or plate type. Heat is transferred from the hot, fresh water flowing through the engine to the cold sea water.

Heat exchanger cooled systems require a sea water pump to circulate sea water through the heat exchanger tubes or plates. It is good design practice to "always put the seawater through the tubes". The tubes can be cleaned by pushing a metal rod through them; the shell side requires chemical cleaning which is only available at shore-side facilities.

The fresh water is circulated through the heat exchanger shell, across the tubes, by the engine-driven water pump.

Most shell and tube heat exchangers are of either the single-pass or the two-pass type. This designation refers to the flow in the cold water circuit of the exchanger. In the two-pass type, the cold water flows twice through the compartment where jacket water is circulated; in the single-pass type only once. When using a single-pass exchanger, the cold water should flow through the exchanger in a direction *opposite* to the flow of jacket coolant to provide maximum differential temperature and heat transfer. This results in improved heat exchanger performance. In a two-pass exchanger, cooling will be equally effective using either of the jacket water connection points for the input and the other for return.



HEAT EXCHANGER TYPES

Figure 1.17

Heat exchangers should always be located at a lower level (elevation) than the coolant level in the expansion tank.

Heat Exchanger Sizing

Occasionally, special applications exist which require an inboard heat exchanger size not available as a Caterpillar unit. When these conditions exist, it is necessary to obtain a heat exchanger from a supplier other than Caterpillar. In order to expedite the selection of a nonstandard heat exchanger, a Heat Exchanger Selection Worksheet is included. Heat exchanger suppliers will provide information and aid in selecting the proper size and material for the application.

For a given jacket water flow rate, the performance of a heat exchanger depends on both the cold water flow rate and differential temperature. To reduce tube erosion, the flow velocity of the cold water through the tubes should not exceed 183 cm/s (6 fps).

At the same seater flow rate, the flow resistance and the flow velocity will be greater through a two-pass heat exchanger than through a single-pass heat exchanger. The heat exchanger should be selected to accommodate the cold water temperature and flow rate needed to keep the temperature differential of the jacket water below about 8.3°C (15°F) at maximum engine heat rejection. Thermostats must be retained in the jacket system to assure that the temperature of the jacket water coolant returned to the engine is approximately 79°C (175°F).

Size heat exchangers to accommodate a heat rejection rate approximately 10% greater than the tabulated engine heat rejection. The additional capacity is intended to compensate for possible variations from published or calculated heat rejection rates, overloads or engine malfunctions which might increase the heat rejection rate momentarily. It is not intended to replace all factors which affect heat transfer, such as fouling factor, shell velocity, etc.

Pay particular attention to the shell side pressure drop to ensure that the entire cooling system flow resistance does not exceed the limitations on the engine freshwater pump.

Heat Exchanger Sizing Worksheet

Heat Exchanger Sizing Data

Required by Heat Exchanger Supplier

Engine Jacket Water Circuit:

1. Jacket water heat rejection*	________	kW (Btu/min)
2. Jacket water flow*	________	L/sec (Gpm)
3. Anticipated seawater maximum temperature	________	C° (F°)
4. Seawater flow	________	L/sec (Gpm)
5. Allowable jacket water pressure drop	________	m (ft) water
6. Allowable seawater pressure drop	________	m (ft) water

Drop

7. Auxiliary water source (sea water or fresh water)	☐ ☐	seawater fresh water
8. Heat exchanger material (admiralty or copper-nickel)	☐ ☐	adm. metal cu-ni
9. Shell connection size**	________	
10. Tube side fouling factor***	________	

Aftercooler Water Circuit:

1. Aftercooler circuit water heat rejection*	________	kW (Btu/min)
2. Aftercooler circuit water flow*	________	L/s (Gpm)
3. Anticipated seawater maximum temperature	________	C° (F°)
4. Seawater flow*	________	L/s (Gpm)
5. Allowable Aftercooler Circuit Water Pressure Drop*	________	m (ft) water
6. Allowable seawater pressure drop*	________	m (ft) water
7. Auxiliary water source (sea water or fresh water)*	☐ ☐	seawater fresh water
8. Heat exchanger material (admiralty or copper-nickel)	☐ ☐	adm. metal cu- ni
9. Shell connection size**	________	
10. Tube side fouling factor***	________	

* Refer to TMI (Technical Marketing Information)

** Refer to engine general dimension drawing

*** Fouling Factor, a descriptive quantity often found on heat exchanger specifications, refers to the heat exchangers ability to resist fouling. As defined in Caterpillar literature, fouling factor is the percentage of the heat transfer surface which can be fouled without losing the heat exchanger's ability to dissipate the engine's full heat load.

Maximum Seawater Temperature

Size heat exchangers such that the seawater is not heated above approximately 54°C (130°F). Higher seawater temperatures will result in fouling of the heat transfer surfaces with chalk-like compounds.

Keel Coolers

A keel cooler is an outboard heat exchanger which is either attached to, or built as part of, the submerged part of a ship's hull. The heated water from the engine(s) circuit(s) is circulated through the cooler by the enginedriven water pump(s).

Keel Cooler Types

Fabricated Keel Coolers

Fabricated keel coolers may be made of pipe, tubing, channel, I-beams, angle or other available shapes. The choice of materials used is dependent on the waters in which the vessel will operate. These materials must be compatible with materials used in the vessel's hull in order to prevent galvanic corrosion.

Sizing of Fabricated Keel Coolers

Engine water temperature maximum limits are controlled by size of the keel cooler. Heat transfer rates through any cooler depend mainly on cooling water temperature, cooling water flow and heat transfer surface area. A cooler may have to operate at its maximum capacity at zero hull speed, as in the case of an auxiliary generating set, operating while the vessel is in port. The minimum area calculated includes a fouling factor. Materials used in cooler construction, condition of waters in which the vessel will operate and service life expectancy will influence the size selection of a new cooler.

Keel cooler area recommendations contained in the graphs below *apply only to keel coolers made of structural steel (channel, angle, half pipe, etc.) welded to the ship's shell plating.* These recommendations take into account the thermal resistance to heat transfer of the steel plate, the internal and external water films, and the internal and external surface corrosion factors. The coefficient of heat transfer of the fresh water film flowing inside the cooler is based upon a flow velocity of 0.9 m/sec (3 ft/sec). The coefficient of heat transfer for the raw water film varies with the velocity of water flow past the cooler due to vessel speed. Surface corrosion factors are based on treated fresh water and polluted river water. Miscellaneous factors become so predominant in the resultant heat transfer rate that the type of material used and thickness of metal become minor considerations .

Normal deterioration of the cooler's inner and outer surfaces in the form of rust, scale and pitting progressively reduce a keel cooler's effectiveness over a period of years. Protective coatings and marine growths will also reduce the rate of heat transfer. It can take 4-5 years before deterioration stabilizes in keel coolers. *It must be designed considerably oversize when new.*

Because of the severe deterioration of heat transfer characteristics associated with structural steel coolers, adequate cooler size sometimes becomes impractical. This is particularly true in regions of high seawater temperature (over 30°C [85°F]). *In these regions, the use of "packaged" keel coolers, or box coolers, made of corrosion-resistant materials is suggested.* These coolers can provide more heat exchange surface area in a given volume on, or within the hull, than the coolers made of structural steel.

Keel Cooler Sizing Worksheet

Keel Cooler Sizing Worksheet

Engine Jacket Water Circuit:

1. Jacket water heat rejection* ____________ kW (Btu/min)
2. Jacket water flow* ____________ L/sec (Gpm)
3. Vessel speed classification
   - ☐ 8 knots & above
   - ☐ 3 knots
   - ☐ 1 knot
   - ☐ still water
4. Anticipated seawater maximum temperature ____________ C° (F°)
5. Minimum cooler area required (per unit) ____________ m^2/kW
   ____________ (ft^2/Btu/min)
6. Minimum area required (Line 1 times Line 5) ____________ m^2 (ft^2)

Aftercooler Water Circuit:

1. Aftercooler circuit heat rejection* ____________ kW (Btu/min)
2. Aftercooler circuit water flow* ____________ L/sec (Gpm)
3. Vessel speed classification
   - ☐ 8 knots & above
   - ☐ 3 knots
   - ☐ 1 knot
   - ☐ still water
4. Anticipated seawater maximum temperature ____________ C° (F°)
5. Minimum cooler area required (per unit) ____________ m^2/kW
   ____________ (ft^2/Btu/min)
6. Minimum area required (Line 1 times Line 5) ____________ m^2 (ft^2)

Marine Gear Oil Cooling Circuit:

1. Marine gear heat rejection** ____________ kW (Btu/min)
2. Vessel speed classification
   - ☐ 8 knots & above
   - ☐ 3 knots
   - ☐ 1 knot
   - ☐ still water
3. Anticipated seawater maximum temperature ____________ C° (F°)
4. Minimum cooler area required (per unit) ____________ m^2/kW
   ____________ (ft^2/Btu/min
5. Minimum Area Required (Line 1 times Line 5) ____________ m^2 (ft^2)

* Refer to TMI (Technical Marketing Information)
** See section on Marine Gear Heat Rejection

Marine Gear Heat Rejection

The Twin Disc marine gears offered by Caterpillar are 95% to 97% efficient, depending on the service factor.

Service Factor	Marine Gear Efficiency	Marine Gear Power Loss Factor
I	97%	3%
II	97%	3%
III	96%	4%
IV	95%	5%

Maximum heat rejection to the marine gear cooling system is equal to the transmitted power from the engine multiplied by the power loss factor.

$$H_{\text{(marine gear)}} = P_{\text{(engine)}} \times F_{\text{(power loss)}}$$

Where:

$H_{\text{marine gear}}$ = Heat rejection of the marine gear oil

P_{engine} = Power generated in the engine and transmitted through the marine gear

$F_{\text{power loss}}$ = A factor relating the heat generated in the marine gear oil to the marine gear efficiency

The following conversion factors are tabulated below.

31.63 x kW = Btu/min
42.41 x hp = Btu/min

Aftercooler Circuit Keel Cooler Area Graph

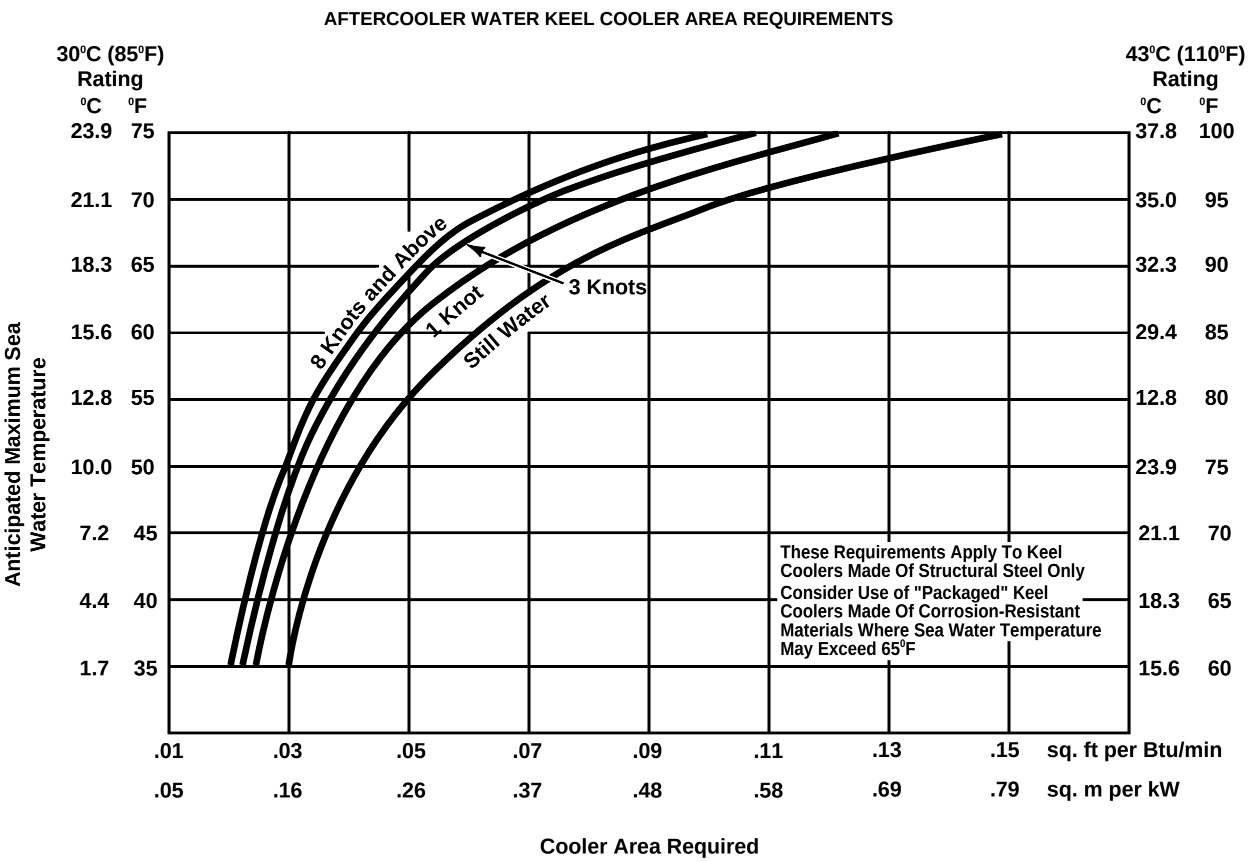


Figure 1.18

Jacket Water Circuit Keel Cooler Area Graph

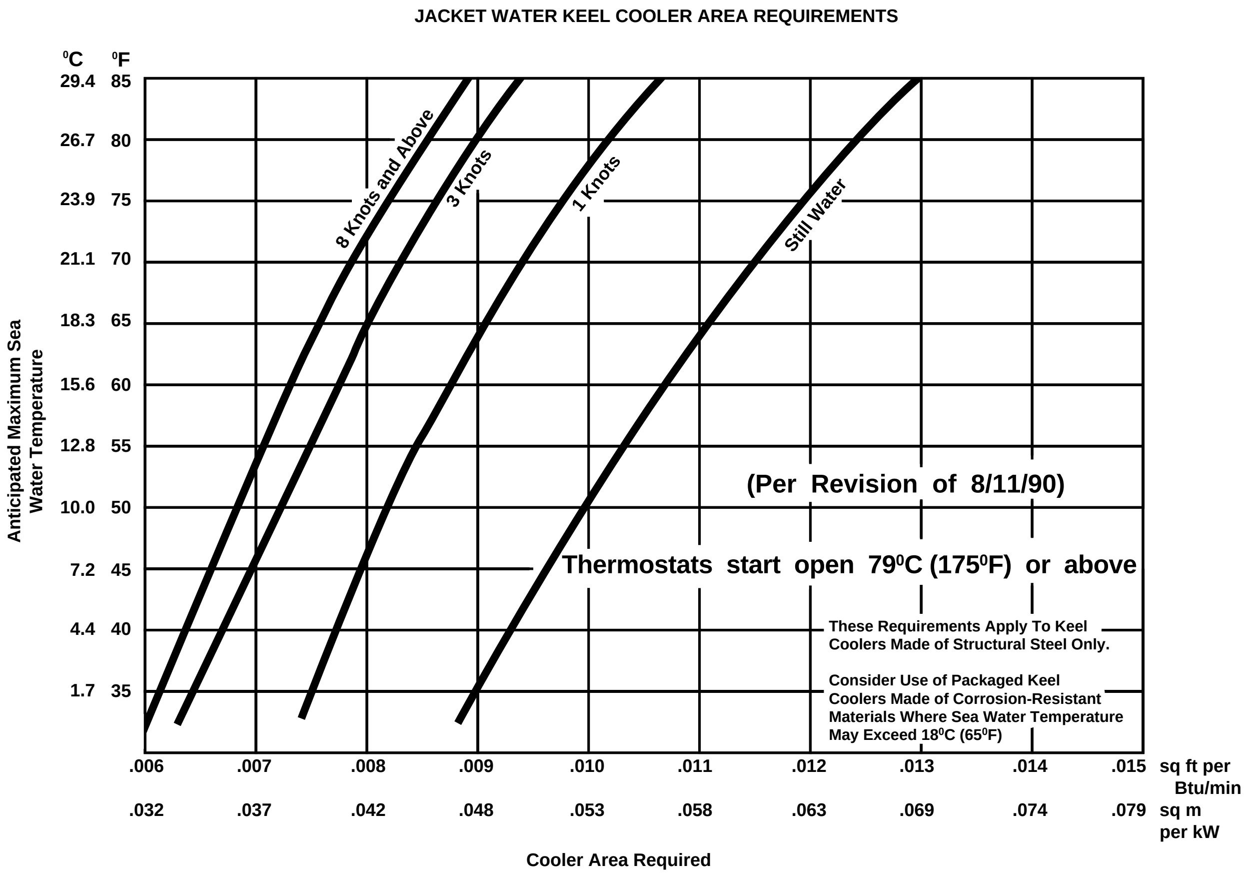


Figure 1.19

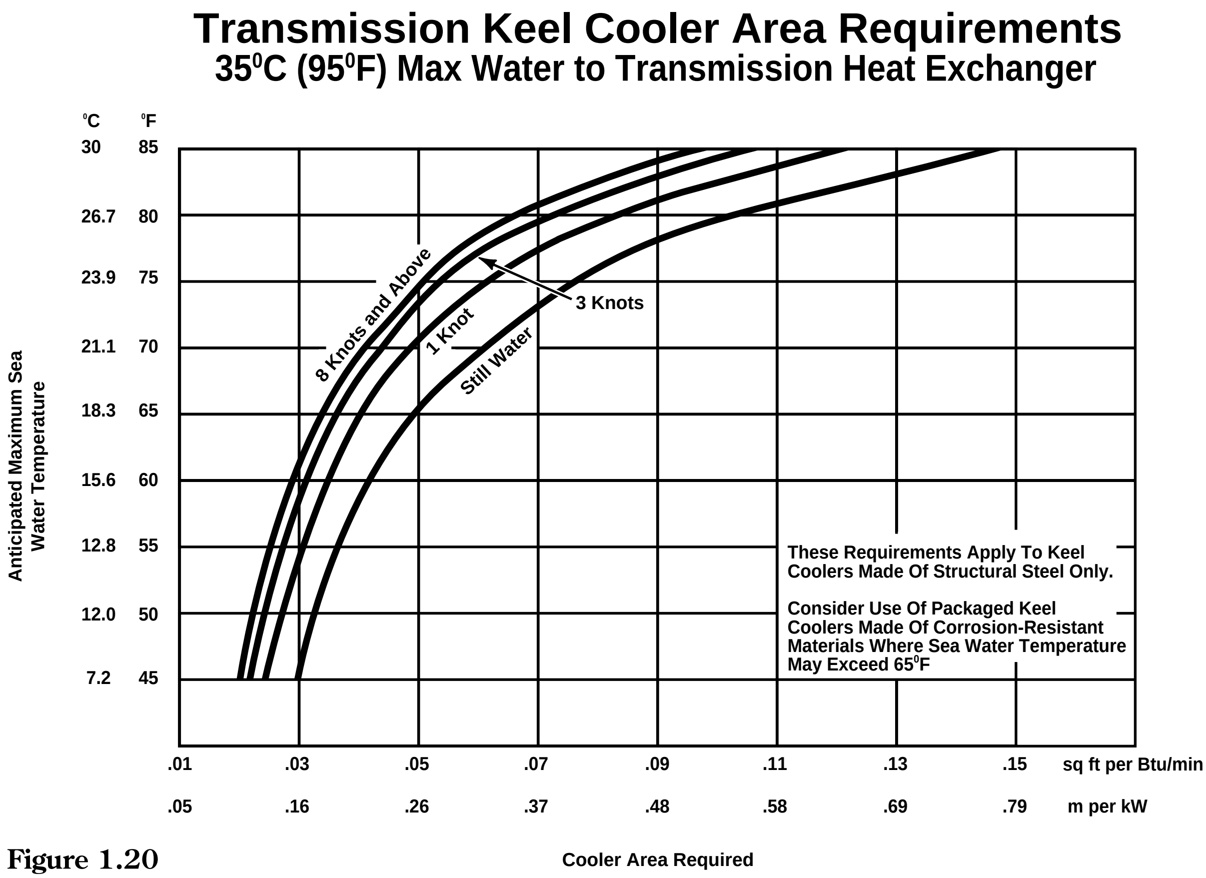


Figure 1.20

Design/Installation Considerations

Water Velocity Inside the Cooler

If the water flows through the keel coolers passages too fast (more then 2.5 m/sec [8 ft/sec]), the internal components will deteriorate (be eroded away), particularly near manifold entrances and exits, elbows and other discontinuities in the water flow. If the water flows through the keel coolers' passages too slowly (less than 0.6 m/sec [2 ft/sec]) rust particles, sand, or other particulate matter in the water will settle out, tend to choke off the flow, and degrade the transfer of heat. Use the following procedure to determine the proper flow pattern through the keel cooler:

- Determine the maximum and minimum expected water flow through the keel cooler. This can be determined from the engines water pump performance data.
- Subtract the minimum expected water flow from the maximum expected water flow.
- Multiply the resultant difference (between the min and max flow) by 2/3. Add 2/3 the resultant difference (from the prior step) to the minimum flow*. This is the *most likely water flow*. Use this figure to determine how to distribute the water flow through the keel cooler passages.
- Determine the cross-sectional area of one keel cooler passage. This is best done by consulting the manufacturer or an engineering reference on shapes of structural channel, pipes, angles, etc.
- Use a good conversion factor table to convert: the *most likely water flow* to units of m^3/min (ft^3/min), and cross-sectional area of one keel cooler passage to units of m^2 (ft^2).

*For design purposes, this is the most likely water flow through the keel cooler. This is dependent on the use of good practice in sizing the connecting piping.

- Divide the most likely water flow by the cross-sectional area of one keel cooler passage.
- The result will be the average velocity through the keel cooler flow passages. If the average velocity through the keel cooler flow passages is greater than 2.5 m/sec (8 ft/sec), arrange the water flow in parallel, so it passes through two or more of the keel cooler passages per pass through the keel cooler. If the average velocity through the keel cooler flow passages is less than 0.6 m/sec (2 ft./sec), use a keel cooler passage with a smaller cross section.

Use of Keel Inserts to Improve Local Flow Velocity

It is economically desirable to use steel channels for keel cooler passages which are so large in the cross-sectional area that water flow is too slow for effective heat transfer. It is useful in this situation to install keel cooler Inserts. Keel cooler Inserts are devices which cause localized high water velocity or turbulence within the keel cooler passage. An effective design for keel cooler inserts is a *ladder-like* device, inserted into the full length of the keel cooler passages.

Using the same metal alloy as the hull and keel cooler*, fabricate a crude ladder of rod** and flat bar***.

The flat bar cross pieces must not restrict flow through the keel cooler flow passages, but simply redirect the flow to avoid laminar flow due to too slow an average velocity.

Insert the ladder into the keel cooler flow passages and weld on the end fittings (inlet and outlet manifolds).

Direction of Flow Through Keel Coolers

Engine coolant should flow through the keel cooler from the rear to the fore end. This is *counter-flow* to the seawater and will significantly increase the effectiveness of the heat transfer. This is rarely practical to implement completely since the flow must be divided through the various flow passages in the keel cooler. If the flow is divided through too many passages, the velocity becomes too slow to maintain turbulent flow conditions.

This will reduce heat transfer. The best compromise is to manifold the coolant in such a way that the flow, in the largest practical number of flow passages, is from rear to the fore end of the vessel.

Bypass Filters

Welded structural steel keel or skin cooler systems require the installation of strainers between the cooler and the pump inlet. Material, such as weld slag and corrosion products, must be removed from the system to prevent wear and plugging of cooling system components. Use a continuous bypass filter to remove smaller particles and sediment. The element size of the continuous bypass filter should be 20 to 50 microns (0.000787 to 0.000197 inches). Do not exceed 19 L/min (5 gal/min) water flow through the bypass and filter.

Strainers

Full-flow strainers are desirable. The strainer screens should be sized no larger than 1.6 mm (.063 in) mesh for use in closed freshwater circuits. The strainer connections should be no smaller than the recommended line size. The use of a differential pressure gauge across the duplex strainers will indicate the pressure drop, and enables the operator to determine when the strainers need servicing.

The pressure drop across a strainer at the maximum water flow must be considered part of the system's external resistance. Suppliers can help in the proper selection of strainers and furnish the values of pressure drop versus flow rate. The strainer should be selected to impose no more than 3 ft (1 m) water restriction to flow under clean strainer conditions.

Packaged Keel Coolers

Packaged keel coolers are purchased and bolted to the outside of a ship's hull.

* For protection against galvanic corrosion.
** Approximately 6 mm (1/4 in) Diameter.
*** Approximately same shape, but 70% of, the cross sectional area of the keel cooler flow passages.

Manufacturers offer keel coolers in many configurations. They are generally made of coppernickel alloys and are initially toxic to marine growth. This is one of their more important advantages. Another important advantage of packaged keel coolers is their compactness and light weight when compared to fabricated keel coolers. It is not uncommon to find packaged keel coolers that are able to cool an engine with less than 20% of the heat transfer surface of an analogous fabricated keel cooler.

Sizing of Packaged Keel Coolers

Manufacturers of packaged keel coolers publish sizing guides which will allow the user to determine the proper size of unit for specific conditions. Caterpillar does not offer guidance outside of manufacturers guidelines.

Packaged Keel Cooler Sizing Worksheet

Collect the information described on the following worksheet. The information thereon is required to accurately size a packaged keel cooler.

Packaged Keel Cooler Sizing Worksheet

Engine Jacket Water Circuit:

1. Jacket water heat rejection* ______________ kW (Btu/min)
2. Jacket water flow* ______________ L/s (Gpm)
3. Vessel speed classification
   - ☐ 8 knots & above
   - ☐ 3 knots
   - ☐ 1 knot
   - ☐ still water
4. Anticipated seawater maximum temperature ______________ °C (°F)

Aftercooler Water Circuit:

1. Aftercooler circuit heat rejection* ______________ kW (Btu/min)
2. Aftercooler circuit water flow* ______________ L/s (Gpm)
3. Vessel speed classification
   - ☐ 8 knots & above
   - ☐ 3 knots
   - ☐ 1 knot
   - ☐ still water
4. Anticipated seawater maximum temperature ______________ °C (°F)

*Refer to TMI (Technical Marketing Information)

Restoration of Toxicity-to-Marine Growth of Copper Nickel Keel Coolers

The toxicity will decline in time. It can be at least partially restored by a thorough cleaning of the cooler and wiping its heat transfer surfaces with a solution of vinegar saturated with salt (sodium chloride).

Paint and Packaged Keel Coolers

Paint will *greatly* reduce the heat transfer capability of packaged keel coolers. ***Never paint them.***

Packaged Keel Coolers and Galvanic Corrosion

Packaged keel coolers are almost never made of the same metal alloy as the rest of the hull*. If the piping is not the same metal alloy as the keel cooler, it is necessary to electronically isolate the packaged keel cooler from the hull metal and the ship's piping.

*There is a manufacturer making packaged keel coolers of aluminum alloy. This will significantly reduce the galvanic corrosion problems associated with such dissimilar metals as aluminum and copper nickel submerged in salt water.

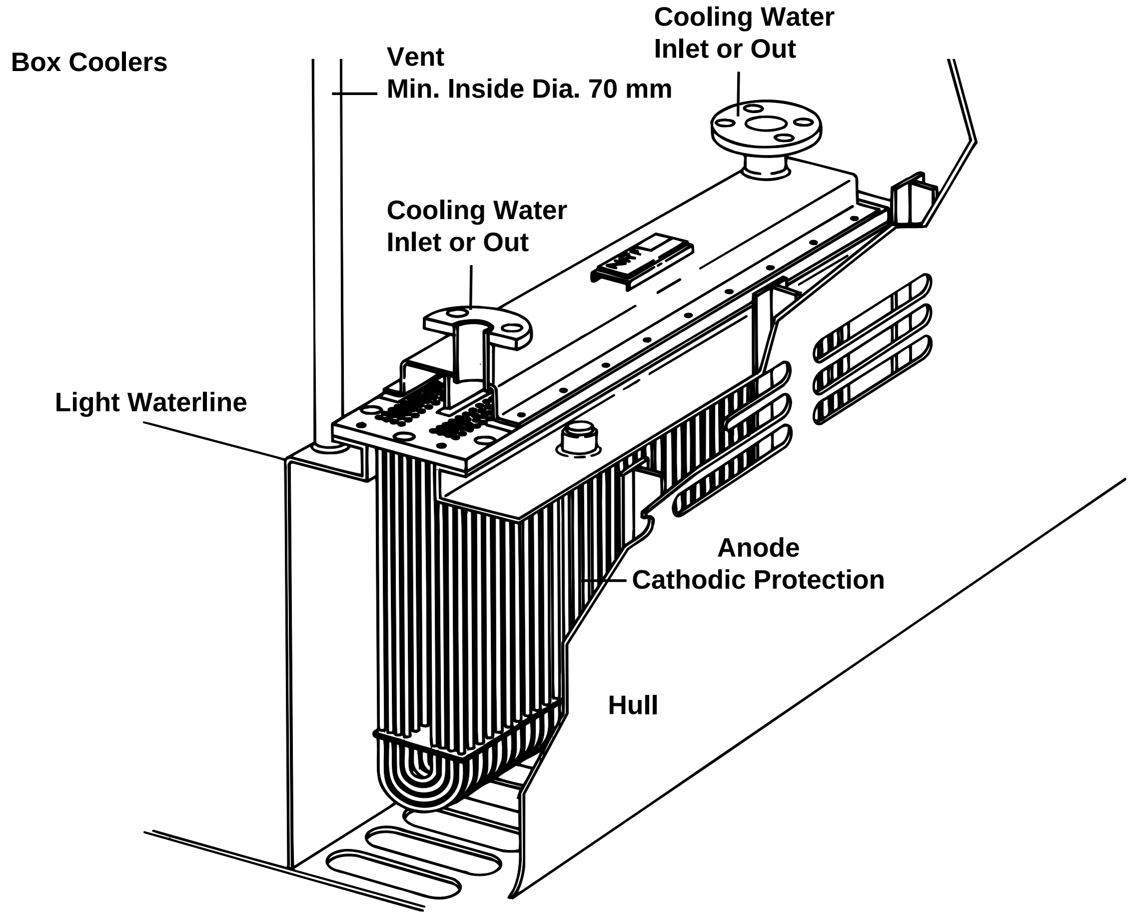


Figure 1.21

See the installation instructions of the packaged keel cooler manufacturer.

Location of Keel Coolers on the Hull

Locate the keel cooler in a well protected area on the hull. This is particularly true of packaged keel coolers which are manufactured of lighter gauge material than fabricated keel coolers.

To achieve the greatest possible heat transfer, locate the separate keel cooler for the aftercooler low on the hull, forward of the keel coolers, for the main and electric set engine jacket water. Heated water from the aftercooler should enter the keel cooler at the rear-most end and be discharged from the cooler for return to the engine at the cooler's forward end. This arrangement assures maximum heat transfer with the vessel either dead in the water or moving ahead.

While the area immediately forward of the propeller(s) is a region of high water velocity and high enough on the hull to be protected from grounding damage, one must consider the effects on the keel cooler from sandblasting (from the propeller(s) during backing maneuvers).

Pumps for Keel Cooler Circuits

Ordinarily, the engine water pump will satisfactorily circulate the engine jacket water through the keel cooler, if the water lines to and from the cooler are relatively short, of adequate size, with minimum bends and if the keel cooler restriction is low. If the total external flow resistance cannot be held within the jacket water pump's capacity, an auxiliary boost pump will be required.

Need for Corrosion Inhibitor

A suitable corrosion inhibitor, carefully maintained, will minimize internal corrosive effects. See the section on cooling system protection.

Venting and Piping of Keel Coolers

Locate the cooler and its through-hull connections so the length of water piping will be kept to a minimum and the cooler will be well vented. Extend water piping downward from the engine to the keel cooler, without high points.

It is very difficult to purge trapped air from the high points of some keel coolers. The air must be bled off during initial fill and whenever the system is completely drained. Vent plugs must be designed into the keel coolers where they rise toward the bow and stern, and any other high points where air may be trapped.

Radiator Cooling

With radiator cooling, the hot water from the engine jacket flows to the radiator core where it is cooled by air being pushed or pulled over the core fins by a fan. The cooled water is then pumped back through the engine; circulation is maintained by a gear- or belt-driven jacket water pump.

When Used

Radiator cooling is used to cool engines that must be located well above the vessel water line or for emergency generator sets that require completely independent support systems.

Radiator Sizing

As with all cooling systems, radiators are usually sized for a heat rejection load a minimum of 10% greater than the maximum full load heat rejection rate of the engine. This allows for overload conditions and system deterioration. This 10% should be added after a careful calculation has been made of the radiator size required to accommodate the maximum heat rejection rate (under normal full load operating conditions) at maximum ambient air temperature.

Keep in mind that radiators lose capacity when operated at altitude or when filled with antifreeze. These conditions should be compensated for and added to the 10% compensation discussed above.

Radiators With Engine-Driven Fans

Some Caterpillar Engines may be ordered with engine-driven fans and close-coupled radiators. They are generally available in two sizes for each engine. The smaller designed for 43°C (110°F) maximum ambient and the larger for 52°C (125°F) maximum ambient temperature.

Caterpillar fan drives are designed to prevent excessive crankshaft loading and to resist vibrations.

Fan Drive Outboard Bearings

Fan drives sometimes require an outboard bearing on the crankshaft pulley. These drives must have a flexible coupling between the pulley and the engine crankshaft. This coupling must not interfere with the longitudinal thermal growth of the crankshaft.

Fan Power Demand

The fan included in Caterpillar radiator systems represents a parasitic load of about 4-8% of the gross power output of the engine.

Radiator System Pressure

Caterpillar radiator cooling systems are designed to work under a pressure of 27.6-48.3 kPa (4-7 psi) to avoid boiling of coolant and allow for best heat transfer.

Remote-Mounted Radiators

On installations where it is desirable to locate the radiator at some distance from the engine—on an upper deck, outdoors or in another room, a remote radiator can be used. Remote-mounted radiator systems require special attention due to the added restriction imposed on the cooling water flow by additional piping. Careful calculations should be made to determine whether a higher output pump is necessary.

Height of Remote Radiators Above Engine

Never remote-mount radiators more than 10 m (33 ft) above the engine. At greater heights, the static head may cause leakage at the engine water pump seals. Consider use of Hotwells in case of need for mounting radiators higher than these guidelines.

Radiators Mounted Below Engines

The radiator top tank loses its air venting capability if it's located below the level of the engine regulator (thermostat) housing.

When a radiator must be mounted lower than the engine, the factory-supplied expansion tank must be used.

Connection Size

Coolant connections must be as large as (or larger than) the applicable engine coolant connections.

Fan Noise

When selecting radiator location, consider fan noise. Noise may be transmitted through the air inlet as well as outlet. As further precaution against noise and vibration, do not rigidly attach ducting to the radiator.

Direction of Prevailing Winds

Also consider the direction of the prevailing winds so the wind does not act against the fan. Another method is to install an air duct outside the wall to direct the air outlet (or inlet). Use a large radius bend and turning vanes to prevent turbulence and air flow restriction.

Hotwell

Hotwell systems are used when static head exceeds 10 m (33 ft) or a boost pump imposes excessive dynamic head.

A mixing tank accommodates total drainback of the remote cooling device and connecting piping. A baffle divides the tank into a hot and cold side but is open sufficiently to assure full engine flow. Baffles are also used where water enters the tank to minimize aeration.

If the Hotwell does not have sufficient volume, the pumps will draw in air during operation. The Hotwell tank must be large enough to accept the full volume of the remote radiator and the interconnecting piping, plus some reasonable amount to prevent air ingestion by the pumps. Generally, 110% of the radiator and piping volume is adequate.

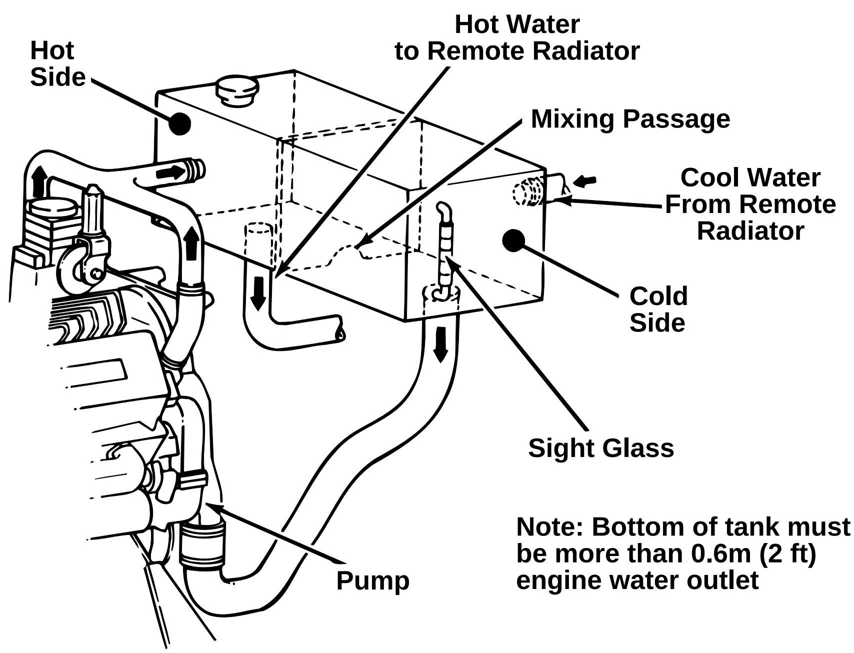


Figure 1.22

Piping Slope for Effective Venting

Piping carrying coolant from the engine to the radiator must have a continual upward slope. This is to allow any gases in the coolant to be separated from the coolant and vented in the radiator top tank.

Recirculation

Care must be taken to ensure engine exhaust gases are not drawing into the radiator. Additionally, the radiators must be arranged so the hot air discharge of one radiator does not recirculate to the inlet of another radiator. Also, for maximum efficiency, the direction of radiator air flow should not be against the direction of strong prevailing winds.

When an engine-mounted radiator is used and the generator set is installed in the center of the room, a blower fan can be used and a duct provided to the outside.

This prevents recirculation and high equipment room temperatures. Some radiator packages have, as standard, a radiator duct flange for ease of installation. The duct is as short and direct as possible; its cross-section area should be as large or larger than the radiator core to minimize backpressure. The anticipated backpressure for a proposed duct design should be less than 12.7 mm (0.5 in) of water.

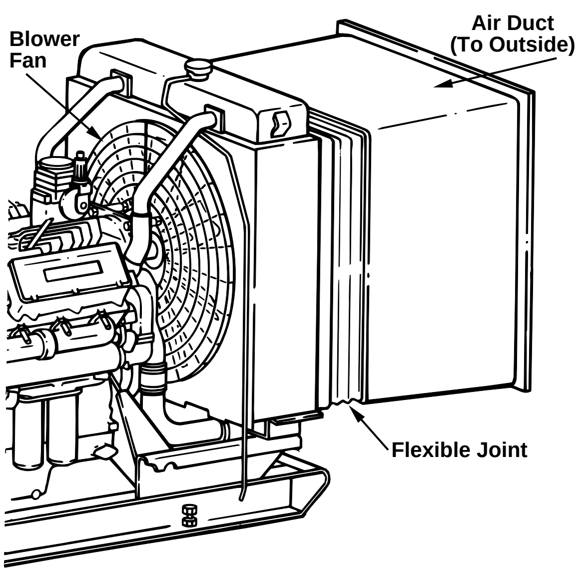


Figure 1.23

Duct Work

Duct work and adjustable shutters can be used to direct some or all of the warmed radiator air for heating purposes. Support duct work independently of the engine or radiator.

Static pressure imposed by the duct work must be determined for each installation. Consult the radiator manufacturer to determine the permissible static pressure.

The fan shroud must be properly positioned for optimum air flow. With a blower fan, two-thirds of the fan width should be outside the shroud. With a suction fan, two-thirds of the fan width should be inside the shroud.

Radiator Air Flow

Backpressure or air flow restriction reduces radiator performance. If radiator air flow is to be ducted, consult the radiator manufacturer regarding the allowable backpressure. An engine installation in an enclosed space requires that the inlet air volume includes the combustion air requirements of the engine unless the air for the engine is ducted directly to the engine from the outside.

Expansion Tanks

Functions

Expansion tanks perform the following functions:

- Vent gases in the coolant
  - —to reduce corrosion.
  - —to prevent loss of coolant due to displacement by gases.
- Provide a positive head on the system pump.
  - —to prevent cavitation.
- Provide expansion volume.
  - —to prevent coolant loss when the coolant expands due to temperature change.
- Provide a place to fill the system, monitor its level, and maintain its corrosion inhibiting chemical additives.

- Provide a place to monitor the system coolant level.
  —an alarm switch located in the expansion tank will give early warning of coolant loss.*

Fill Rate

The Caterpillar engine-mounted cooling circuits are designed to completely vent during the initial fill for fill rates up to 5 gpm (19.0 L/min). Vent lines are located such that the external cooling circuit will also be vented if the customer piping is installed level with, or below, the proper engine connecting points, and if no air traps are designed in the piping.

Type of Expansion Tank

Engine-Mounted Expansion Tank (Manufactured by Caterpillar)

The engine-mounted expansion tank provides all of the above functions for the engine's jacket water circuit. Caterpillar does not provide expansion tanks for the engines auxiliary water circuit (the aftercooler circuit). It can provide adequate expansion volume for only a modest amount of jacket water. Table 1.1 describes the allowable external volume using only the engine-mounted expansion tank. Consult TMI for engine coolant capacity.

*In case of a system leak, the water in the auxiliary expansion tank must be completely drained before the engine is in danger from coolant loss. Therefore, a water level switch and sight glass will give early warning of coolant loss and significantly protect an engine from this problem.

Table of Cooling System Volumetric Data

Cooling System Volumetric Data		
Engine Model	Allowable External Volume With Engine Mounted Tank	
	Liters	U.S. Gal
3116	0.0	0.0
3126	0.0	0.0
3208NA	7.5	2.0
3208T&TA	7.5	2.0
3304B	7.5	2.0
3306B	7.5	2.0
3176/96	0.0	0.0
3406C	38.0	10.0
3406E	0.0	0.0
3408C	53.0	14.0
3412C	53.0	14.0
3412E	0.0	0.0
3508	243.0	64.0
3512	182.0	48.0
3516	122.0	32.0
3606	365.0	95.0
3608	210.0	55.0
3612	550.0	145.0
3616	260.0	65.0
3606	4710.0	1245.0
3608	4550.0	1200.0
3612	4890.0	1290.0
3616	4600.0	1210.0

Table 1.1

Deaerator

All the functions of engine mounted expansion tanks may be fulfilled by a simple volume chamber, a few feet higher than, and connected to the jacket water pump, by a continuously upward-sloping standpipe—all *except* for the need to continuously vent combustion gas bubbles.

A deaerator is a device to separate gas bubbles from engine coolant in the absence of a factory-designed, engine-mounted expansion tank.

It is mounted, in series, in the main flow of jacket water between the engine and its heat exchanger.

A simple volume chamber is still required to handle coolant expansion and for venting the separated bubbles to atmosphere.

Engines installed with a simple, volume tank and a deaerator can avoid the use of the engine-mounted expansion tank.

Jacket Water Circuit Auxiliary Expansion Tank (Fabricated by the Engine Installer)

An auxiliary expansion tank is needed when additional expansion volume is required in the cooling system. This generally occurs when keel coolers are used and may occur when remote-mounted heat exchangers are used.

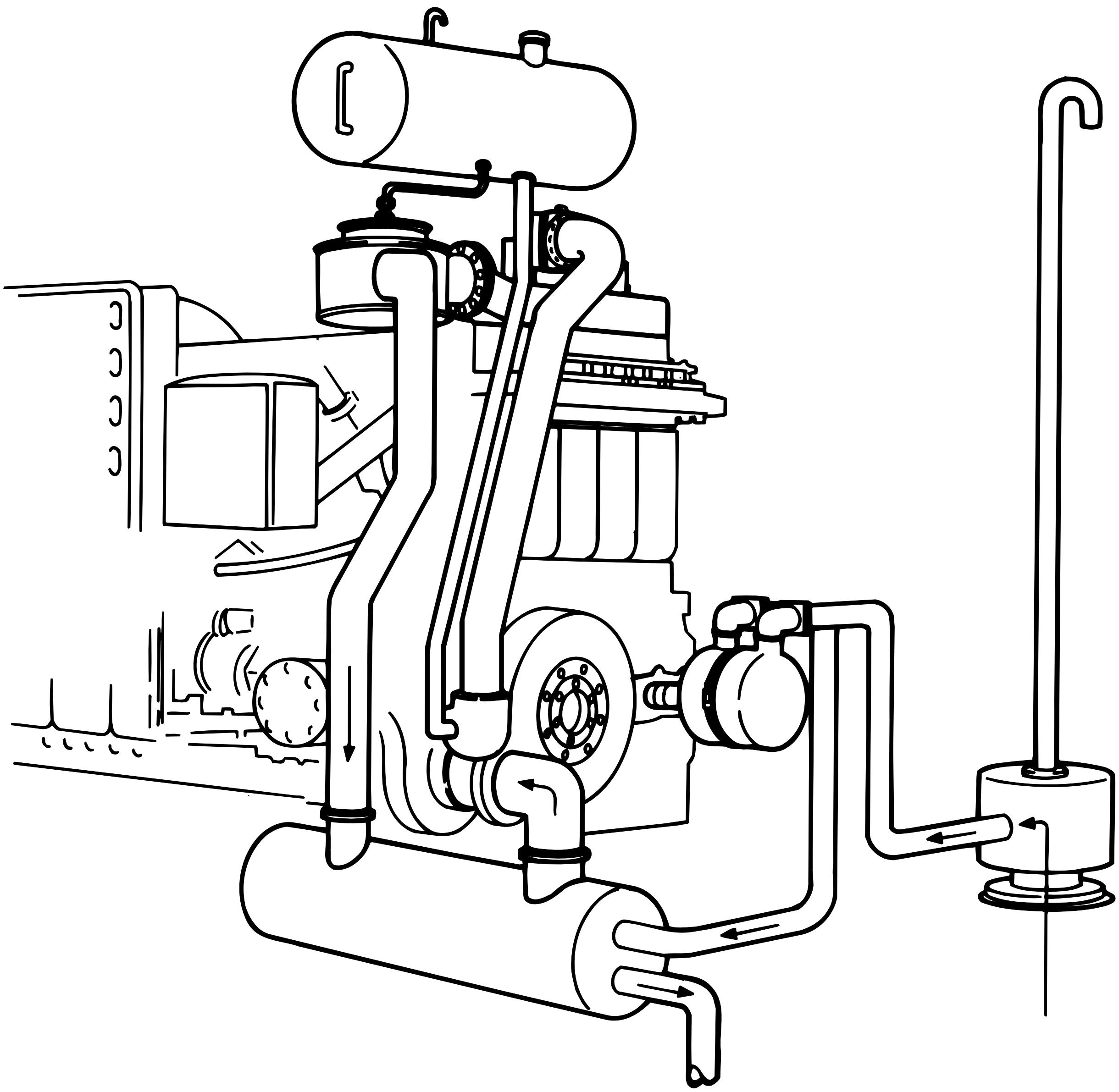
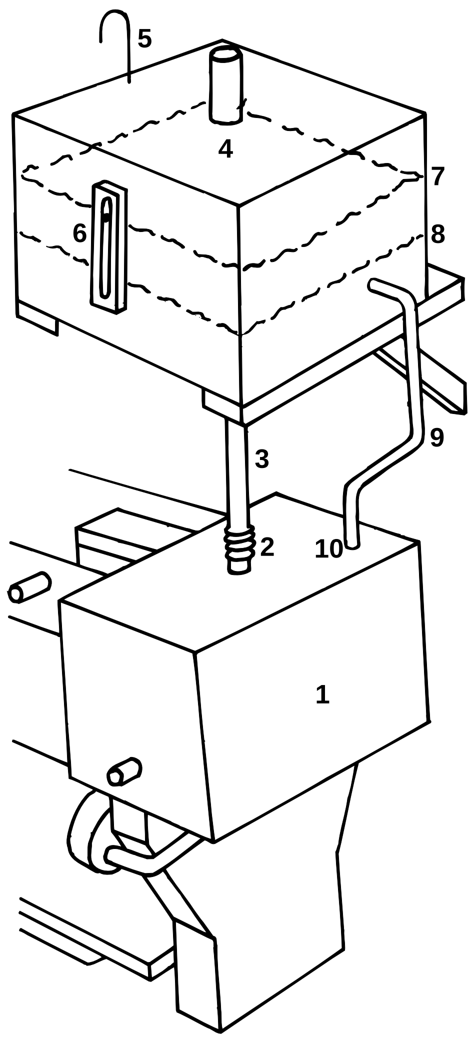


Figure 1.24

The auxiliary tank can consist of a simple tank. Internal baffles are not required.

The engine-mounted components of the cooling system will adequately separate gases from the coolant. However, the gases, once separated, must by allowed to rise by a continuous upward sloped standpipe to the auxiliary expansion tank. Additional air vent piping may be required if the auxiliary expansion tank is not located directly above the engine-mounted expansion tank.



AUXILIARY EXPANSION TANK
Engine Jacket Water

1. Engine mounted expansion tank
2. Flexible connection
3. Connecting pipe
4. Auxiliary expansion tank
5. Tank vent
6. Level gauge
7. Engine mounted expansion tank
8. Flexible connection
9. Connecting pipe
10. Connect lower end of fill vent to vent piping entering rear side of engine mounted expansion tank

Note: Do not drill engine mounted expansion tank

Figure 1.25

Aftercooler Circuit Auxiliary Expansion Tank (Fabricated by the Engine Installer)

All closed fresh water aftercooler circuits require an expansion tank. The tank provides coolant expansion volume, allows system venting and provides a positive pressure on the inlet side of the circulating pump. The expansion tank must be the highest point in the aftercooler water circuit.

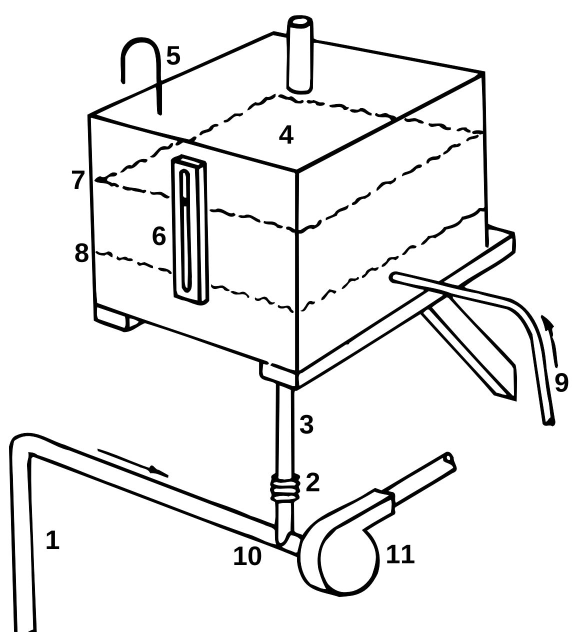
This tank is a simple reservoir with the connecting pipe placed as close to the pump inlet as possible. See Auxiliary Expansion Tank Sizing Worksheet to determine the minimum volume required.

Separately support and isolate the auxiliary tank from the pump inlet and cooler by flexible connections. Install a vent line from the high point on the engine aftercooler circuit to a point in the tank below the tank's low water level. This line must be sloped upwards from the engine to the tank.

All closed separate circuit aftercooler circuits require installation of a vent line. A tapped hole is provided at the high point in the engine-mounted aftercooler circuit. Install a vent line from that point to the aftercooler circuit expansion tank. Vent line size of 6.3 mm (0.25 in.) is adequate. The vent line should enter the tank below the low water level. If possible, water lines connecting to the aftercooler circuit should be level with or below the connecting points on the engine. If the water lines must run above the connection points on the engine, it will be necessary to vent the high points in the external system. Air traps in the external system piping should be avoided.

Sizing the Volume of Auxiliary Expansion Tanks

The minimum volume of the auxiliary tank should include the total jacket water system expansion volume required, plus the volume for the water to the low water level in the tank. The worksheet on page 38, *Auxiliary Expansion Tank Sizing*, can be used to determine the minimum volume required.



1. Return line from cooler
2. Flexible connection
3. Connecting pipe
4. Auxiliary expansion tank
5. Tank vent
6. Level gauge
7. Operating level
8. Cold fill level
9. Vent line from aftercooler
10. Connecting line to auxiliary pump inlet
11. Auxiliary fresh water pump

AUXILIARY EXPANSION TANK
Separate Circuit Aftercooler—Fresh Water

Figure 1.26

Auxiliary Expansion Tank Sizing

Engine Model ______________________ Rating _________ hp at _____________ rpm

For Engine Jacket Water, Figure 1.25:

Auxiliary jacket water expansion tanks are not always required.

1. Allowable external volume _________ L/gal, with engine mounted tank. (This value shown in Table 1.1, Column A, on page 37.)
2. Total volume of jacket water contained in external cooling circuit (not furnished as part of engine) _________ L/gal. See Table 1.2, page 47, for volume per length of standard iron pipe.
3. Line 2 minus Line 1 _________ L/gal.
   If this value is zero or less, additional tank is not required.
   If this value is greater than zero, an auxiliary tank is required.
4. If required, the *minimum* volume of the auxiliary expansion tank can be determined by:

   a. Engine volume, Table 1.1, Column B ______________________________
   b. External volume Line 2 ______________________________
   c. Total volume—
      sum of line a and line b ______________________________
   d. Multiply line a by 0.06 ______________________________
   e. Multiply line b by 0.04 ______________________________
   f. Multiply line c by 0.01 ______________________________
   g. Total of lines d, e and f ______________________________

   (This is the minimum volume of the jacket water auxiliary expansion tank.)

For Separate Circuit Aftercooler, Figure 1.26:

1. Total volume of aftercooler external water _________ L/gal.
2. Multiply Line 1 by 0.02 _________ L/gal.
3. Add the cold fill volume desired in auxiliary expansion tank to Line 2.
   Total of Line 2 and cold fill volume _________ L/gal.
   (This is the minimum volume of the aftercooler circuit auxiliary expansion tank.)

Mounting of Auxiliary Expansion Tank

Separately support and isolate the auxiliary tank against vibration from the engine-mounted tank with a flexible connector.

Pressurization of Systems Containing Auxiliary Expansion Tanks—Afterboil

Generally, pressure caps are not required or desirable on auxiliary expansion tanks. This is to allow free venting and refilling, when required.

An exception exists in the situation of high performance craft, such as fast ferrys, yachts and patrol craft: vessels of this type are prone to have their engines stopped immediately after periods of hard use. In this circumstance, a phenomenon known as *afterboil* can occur.

Afterboil is the boiling (change of liquid to vapor) of the coolant, caused by hot engine components which have lost coolant flow and pressure when the engine is hastily shut off. This can result in sudden loss of coolant out the vents and fill openings of the expansion tank. This can be dangerous to personnel in the area if they are not expecting it.

Afterboil Hazard—How to Avoid It

System pressurization with pressure caps on the auxiliary expansion tanks will minimize afterboil but cannot completely avoid it. It is strongly recommended that auxiliary expansion tank vents and fill openings be arranged so any hot coolant being discharged during afterboil will not present danger to personnel. Vents should carry the vented hot water directly into the bilge.

Use of the *Burp Bottle*

This is just like the overflow bottle system found on most modern automobiles and for the same reasons.

After each occurrence of afterboil, the system will need to be refilled. This can be avoided by using a *burp bottle*. The burp bottle is a reservoir for temporary storage of the discharged coolant. The jacket water circuit auxiliary expansion tank vent leads to the bottom of the burp bottle. As soon as the steam bubbles condense within the engine, the displaced coolant will be drawn back into the system by the resultant vacuum. Use of the burp bottle requires the jacket water circuit auxiliary expansion tank be fitted with a double-acting pressure cap*.

Filling of Auxiliary Expansion Tanks

Auxiliary expansion tanks in vessels operated so their engines are not subjected to afterboil should have permanently installed provisions to add water from the ship's portable water supply plumbing. It should be possible for the operator to add water to the system by opening and shutting a valve. This is to minimize danger to the operator when adding system water during severe sea conditions.

Care should be taken to ensure that the coolant full level in the tank is above all piping in order to fill the system.

If it is necessary to design a cooling system that will not purge itself of air when being filled, provide vent lines from high points to the expansion tank. These vents should enter the tank below the normal low water level to prevent aeration of the water which will circulate through these lines when the engine is running. Slope the vent line upwards with no air traps. The vent line should be 6.3 mm (0.25 in.) tubing. Use of a smaller size will clog and may not provide adequate venting ability. Too large a vent tube may introduce a circuit that could contribute either to subcooling or overheating, depending on the location.

If at all possible, avoid external piping designs that require additional vent lines.

Cooling System Protective Devices

A most common problem associated with properly installed cooling systems is loss of coolant, generally due to breaking a water hose and overheating, which can have many causes. As with many engine safety devices, the decision to automatically shut down the engine without warning, or to continue operation risking total engine destruction, is for the careful consideration of the owner. In conditions where the entire boat and the lives of those on board are at stake, it may be appropriate to use a safety system which does not have automatic shutdown capability. The boat's pilot has the option to continue operation of a distressed engine to provide a few more minutes of engine power to escape a more present danger.

* A double-acting pressure cap is one which will hold a certain pressure or vacuum.

Coolant Level Switches

Coolant level switches are devices which can give early warning of coolant loss. They generally consist of a sealed single pole-double throw switch, actuated by a float which rides on the surface of the coolant in the expansion tank. It is good design practice to locate the coolant level switch in the highest part of the cooling system—to give earliest warning of a drop in coolant level. High water temperature switches will not give warning of coolant loss; their temperature sensing portion works best when surrounded by liquid water rather than steam.

High Water Temperature Switches

High water temperature switches are devices which continuously monitor the temperature of some fluid, generally coolant, and actuate switch contacts when the fluid temperature goes above some preset limit. In the case of jacket water coolant, the set point is usually between 96° and 102°C (205° and 215°F), depending on the engine, cooling system type, and whether *alarm of impending problems* or *actuation of engine shutdown systems* is desired. Switches can be set for either condition.

Emergency Systems

The worldwide marine classification societies require in certain applications that, for unrestricted seagoing service, engines be equipped with a separate emergency supply of cooling water flow. The requirement applies to both the engine jacket water and auxiliary (sea or fresh) water systems. The purpose of the emergency systems is to ensure cooling if either the jacket water or auxiliary (sea or fresh) water pump should fail. The customer-supplied emergency pumps should provide flow equal to the failed pump to permit operation at full, continuous power with the emergency systems. For pump flow requirements of engine-mounted pumps, refer to Technical Marketing Information (TMI) or consult the Caterpillar Dealer. If reduced power operation is acceptable, reduced flows can be utilized. Use flexible connectors at the engine to protect the piping and engine.

Jacket Water Pump Connections

The optional Caterpillar emergency jacket water connections (available for the large Vee engines) meet the requirements of the engine and the marine classification societies. Use of these connections permits the emergency system to utilize the normal jacket water as the coolant and to bypass the engine-mounted jacket water pump. The system includes a blanking plate or valve to direct jacket water to the emergency system and flanged connection points on the engine for the emergency system piping. Figure 1.27 is a schematic diagram of the system properly connected. The customer-supplied emergency water pump should provide flow equal to the failed pump.

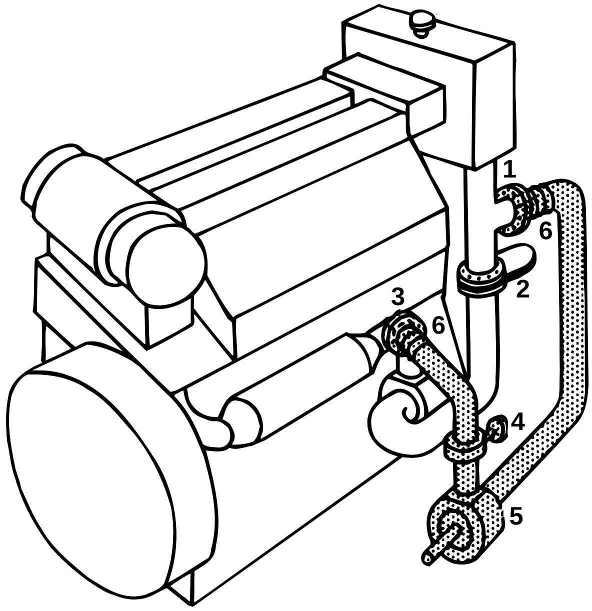
The use of seawater in the engine jacket water system is not recommended. If seawater must be used in the jacket water system to ensure the safety of the ship in an emergency situation, use the lowest engine power level commensurate with the sea state. On reaching port, the jacket water system must be thoroughly flushed and cleaned.

Auxiliary Seawater Pump Connections

All emergency seawater cooling connections are to be provided by the installer and connected as indicated in the figure illustrating emergency auxiliary pump connections. The emergency seawater pump should provide flow equal to the failed pump.

Auxiliary Freshwater Pump Connections

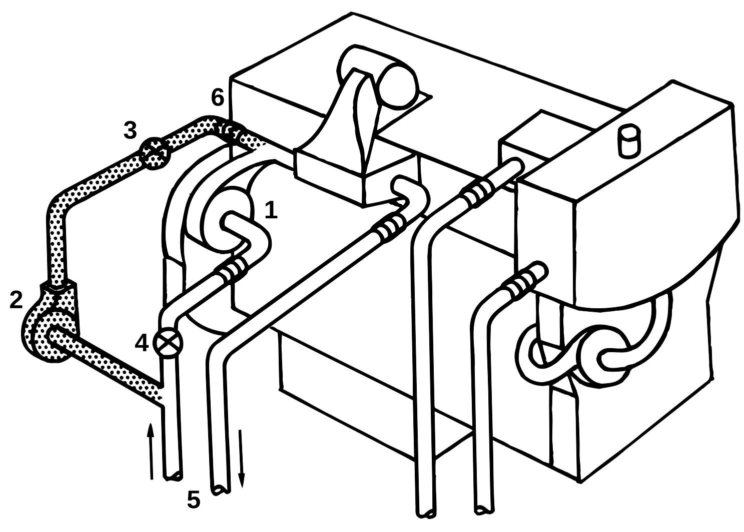
All emergency connections for separate keel cooled aftercooler circuits are to be provided by the installer, and connected as indicated in the figure illustrating emergency auxiliary pump connections. The flow required for the emergency separate keel cooled aftercooler pump should equal the failed pump. The use of seawater in the separate keel cooled aftercooler circuit is not recommended. The engine- mounted pump and lines are of ferrous material and have low corrosion resistance in seawater. If seawater must be used in an emergency to ensure the safety of the ship, thoroughly flush the system as



EMERGENCY JACKET WATER PUMP CONNECTIONS
(Location of Jacket Water Pump May Vary)

1. **Flanged tee connection — to emergency pump**
2. **Blanking plate — closed for emergency pump operation**
3. **Flanged tee connection — from emergency pump**
4. **Valve — customer supplied (open for emergency pump operation)**
5. **Emergency pump — customer supplied**
6. **Flexible connector — customer supplied**

Figure 1.27



EMERGENCY AUXILIARY PUMP CONNECTIONS

1. **Engine mounted auxiliary pump**
2. **Customer provided emergency auxiliary pump**
3. **Customer provided valve (normally closed)**
4. **Customer provided valve (normally open)**
5. **Auxiliary cooling circuit**
6. **Flexible connection**

Figure 1.28

mentioned in the jacket water section, and inspect the parts for corrosion damage and deposits.

Central Cooling Systems

A central cooling system is defined as one which cools multiple engines and which combines many individual system components (heat exchangers and pumps) into large *central* ones. There are economic advantages to such systems.

Advantages of a Central Cooling System

There are fewer lines to install, significantly reducing the amount of shipyard labor required to install such a system.

The smaller number of components cost less to procure, inventory, and support with repair units.

Larger components are generally more robust and can be expected to last longer.

Disadvantages of a Central Cooling System

It is very difficult to diagnose problems in such a system because there are so many modes of operation possible:

For example: with a system containing three engines, one heat exchanger, and two pumps, there will be 162 possible combinations or modes of operation.

Following is a list of common errors in designing such systems:

Flow Control

There are upper and lower limits to the allowable flow through an engine. The system must be able to throttle the flow through each engine independently.

Temperature Control

The heat exchanger must be capable of delivering the proper amount of cooling, proportional to engine load.

Load Control

The amount of *external* water flow through a Caterpillar Engine is directly proportional to the engine's load. The greater the load, the greater the amount of cooling required and the more water the engine's internal cooling circuitry will discharge for cooling. At light loads, the engine's temperature controls will bypass the external portion of the engine cooling system, recirculating virtually all of the coolant. If the water pressure presented to an engine by a central cooling system is too high, the proper operation of the engine's temperature controls may be overridden, and the engine will suffer over or under cooling problems. It is very difficult to adequately balance and control the flow through several engines, all of which might be operating at widely varying loads.

The water pressure on an engine jacket water inlet cannot be allowed to exceed 172 kPa (25 psi). Economic factors encourage many designers to use higher pressures. Do not use higher pressures. Higher pressures will significantly reduce water pump seal life.

<table>
<tr><th>Running</th><th>Engine Load</th><th>Maintenance Condition</th><th>Redundancy Required for Reliability</th></tr>
<tr><td>Yes</td><td>High</td><td rowspan="2">Operational shutdown for maintenance but still connected to the system</td><td>In the heat exchanger</td></tr>
<tr><td>No</td><td>Intermediate</td><td>In the interconnecting plumbing*</td></tr>
<tr><td></td><td>Low</td><td>Overhaul in process, disconnected from the system</td><td>In the pump(s) and their controls/switchgear</td></tr>
</table>

* In areas of severe marine growth problems, it is a good idea to have two parallel sets of plumbing so that one set can be in process of being cleaned at any given time.

Suggestions for Design of a Successful Central Cooling System

Keep each engine's jacket water system independent of all others. The load control problems are not economically solvable.

Use a separate heat exchanger at each engine for cooling of the engine jacket water.

Provide a *ring main* of freshwater, circulated by at least two, parallel water pumps. A third water pump should be kept in reserve to maintain operation when either of the other pumps require maintenance. Each pump should be identical for ease of parts inventory and maintenance. The ring main is the water source for each engine's independent cooling system. The temperature and pressure of the water in the ring main do not need precise control. Each engine should have an engine-driven, auxiliary (not jacket water) water pump. This pump will draw water from the ring main and return it back to the ring main, downstream.

System Pressure Drop

The total external system resistance to flow must be limited in order to ensure adequate flow. The resistance to flow is determined by the size and quantity of pipe, fittings and other components in the portion of the cooling system which is external to the engine. As the resistance (pressure drop) increases, the engine-driven water pump flow decreases.

The external resistance imposed on the pump (also called external head) includes both the resistance ahead of the pump inlet and the resistance downstream of the engine. The resistance to flow in the external circuit of a closed circulating system consists only of the frictional pressure drop. The resistance to flow in an external open cooling circuit consists of not only the frictional pressure drop but also the height of suction lift on the pump inlet and the heights of the lift on the engine outlet.

Curves showing water flow versus total external system head for engine-driven pumps are available. The value for the maximum external resistance must not be exceeded in the cooling circuit added by the customer in order to maintain minimum water flow. Flows lower than the minimums will certainly shorten the life of the engine.

When designing the engine cooling systems, pressure drop (resistance) in the external cooling system can be calculated by totaling the pressure drop in each of the system's components. The section of useful tables to designers of cooling systems can be used to determine the pressure drop through pipe, fittings and valves. Suppliers of other components, such as strainers and sea cocks, can provide the data required for their product.

It is always necessary to evaluate the design and installation of the cooling circuits by testing the operation and effectiveness of the completed system to ensure proper performance and life.

Corrosion

Galvanic Corrosion in Seawater

When two dissimilar metals are electrically connected and both submerged in saltwater, they form a battery and an electrochemical reaction takes place. In this process, one metal is eaten away. The rate of deterioration is proportional to a number of factors:

- The differential potential between the two metals on the electrochemical series (see Useful Tables to Designers of Cooling Systems).
- The relative areas of the two metals: If there is a small area of the more noble metal relative to the less noble metal, the deterioration will be slow and relatively minor. If there is a large area of the more noble metal such as copper sheathing on a wooden hull, and a much smaller area of the less noble metal, such as iron nails holding the copper sheathing to the wood, the wasting away of the iron nails will be violent and rapid.

Dissimilar Metal Combinations to Avoid

- Bronze Propeller on Steel Shaft
- Mill Scale on Hull Plate (Internal or External)
- Aluminum Fairwaters Fastened to a Steel Hull
- Steel Bolts in Bronze Plates
- Bronze Unions and Elbows Used With Galvanized Pipe
- Bronze Sea Cocks on Iron Drain Pipes
- Brass Bilge Pumps on Boats With Steel Frames
- Brass, Bronze, or Copper Fasteners in Steel Frames
- Stainless Steel Pennants on Steel Mooring Chains
- Bronze or Brass Rudder Posts With Steel Rudders
- Bronze Rudders With Steel Stopper-Chains
- Steel Skegs (Rudder Shoes) Fastened With Bronze or Brass Leg Screws
- Steel and Brass Parts in the Same Pump

Rule of Thumb:

Do not put iron or steel close to or connected with alloys of copper under salt water.

The Protective Role of Zinc

If alloys of copper (bronze, brass), iron (steel), and zinc are all connected together and submerged in salt water, the zinc will be eaten away, protecting the iron (steel). It is necessary to have a metallic electrical connection to the metals to be protected. This is usually easy to accomplish on a steel hull. It is more difficult on a fiberglass hull, since special electrical connection may be required unless the zincs are connected directly to one of the metals, preferably the copper alloy.

The Zinc Must Never Be Painted! When electrical contact is made through the fastening studs, it's desirable to put galvanized or brass bushings in the holes in the zincs so that contact will be maintained as the zincs corrode.

Zincs should be periodically inspected. As they work, a white, crust-like deposit of zinc oxides and salts form on the surface. This is normal. If it does not form and the zincs remain clean and like new, they are not protecting the structure.

Useful Tables to Designers of Cooling Systems

Pipe Dimensions — Standard Iron Pipe

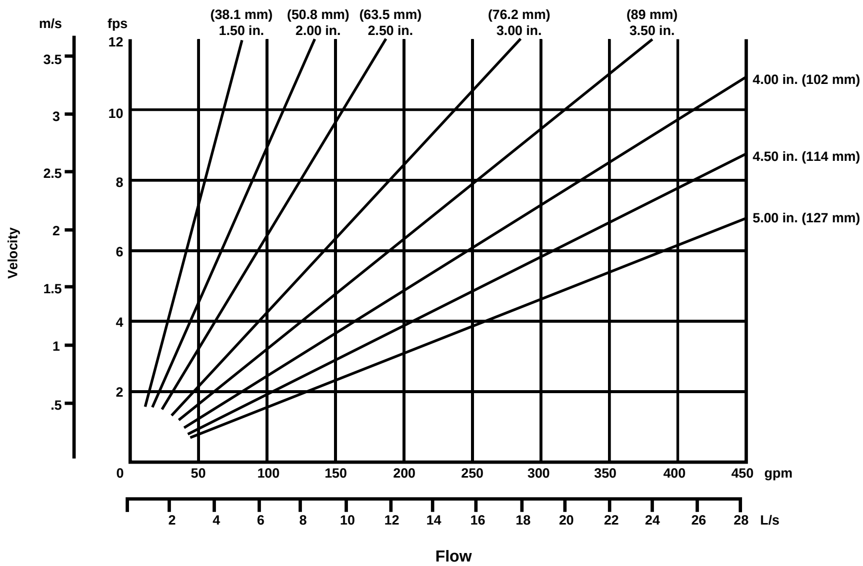
Nominal Size		Actual I.D.		Actual O.D.					
in.	mm	in.	mm	in.	mm	ft/gal	m/L	ft/cu ft	m/cu m
.125	3.18	.270	6.86	.405	10.29	336.000	27.000	2513.000	27.049
.250	6.35	.364	9.25	.540	13.72	185.000	16.100	1383.000	14.886
.375	9.53	.494	12.55	.675	17.15	100.400	8.300	751.000	8.083
.500	12.70	.623	15.82	.840	21.34	63.100	5.000	472.000	5.080
.750	19.05	.824	20.93	1.050	26.68	36.100	2.900	271.000	2.917
1.000	25.40	1.048	26.62	1.315	33.40	22.300	1.900	166.800	1.795
1.250	31.75	1.380	35.05	1.660	42.16	12.850	1.030	96.100	1.034
1.500	38.10	1.610	40.89	1.900	48.26	9.440	.760	70.600	760.000
2.000	50.80	2.067	52.25	2.375	60.33	5.730	.460	42.900	462.000
2.500	63.50	2.468	62.69	2.875	73.02	4.020	.320	30.100	324.000
3.000	76.20	3.067	77.90	3.500	88.90	2.600	.210	19.500	210.000
3.500	88.90	3.548	90.12	4.000	101.60	1.940	.160	14.510	156.000
4.000	101.60	4.026	102.26	4.500	114.30	1.510	.120	11.300	122.000
4.500	114.30	4.508	114.50	5.000	127.00	1.205	.097	9.010	97.000
5.000	127.00	5.045	128.14	5.563	141.30	.961	.077	7.190	77.000
6.000	152.40	6.065	154.00	6.625	168.28	.666	.054	4.980	54.000
7.000	177.80	7.023	178.38	7.625	193.66	.496	.040	3.710	40.000
8.000	203.20	7.982	202.74	8.625	219.08	.384	.031	2.870	31.000
9.000	228.60	8.937	227.00	9.625	244.48	.307	.025	2.300	25.000
10.000	254.00	10.019	254.50	10.750	273.05	.244	.020	1.825	19.600
12.000	304.80	12.000	304.80	12.750	323.85	.204	.160	1.526	16.400

Table 1.2

If the zincs are not working, look for the following conditions:

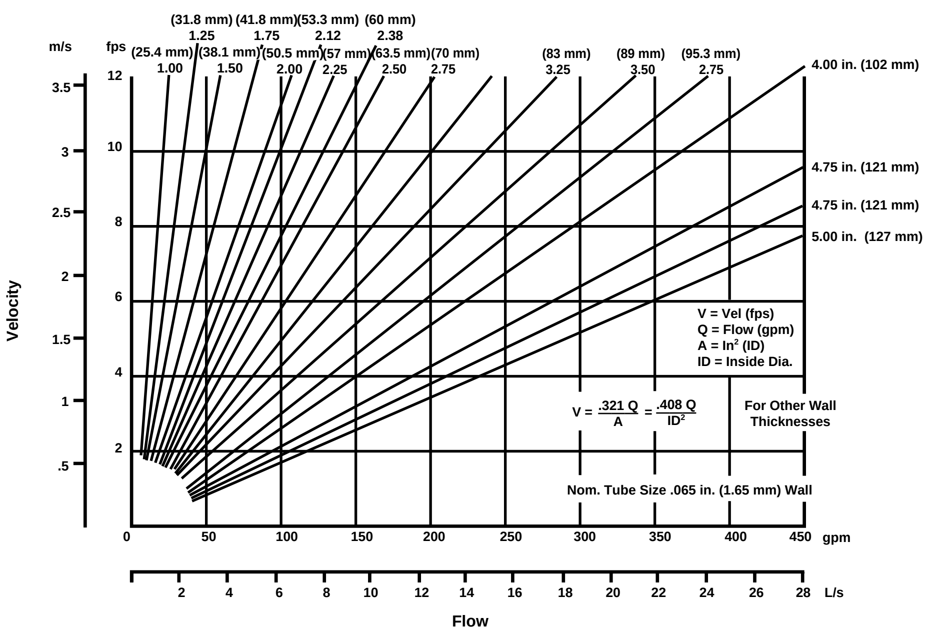
- The anode is not electrically bonded to the structure.
- The paint on the structure is still in near perfect condition.

Velocity Versus Flow (Standard Pipe Sizes)



VELOCITY vs FLOW
Standard Pipe Sizes 1.5 to 5 in.
(38.1 to 127mm)

Figure 1.29



VELOCITY vs FLOW
Tube Sizes From 1 in. to 5 in. O.D. (25.4 mm to 127 mm)
(Common Usage Wall Thickness)

Figure 1.30

Typical Friction Losses of Water in Pipe — (Old Pipe)

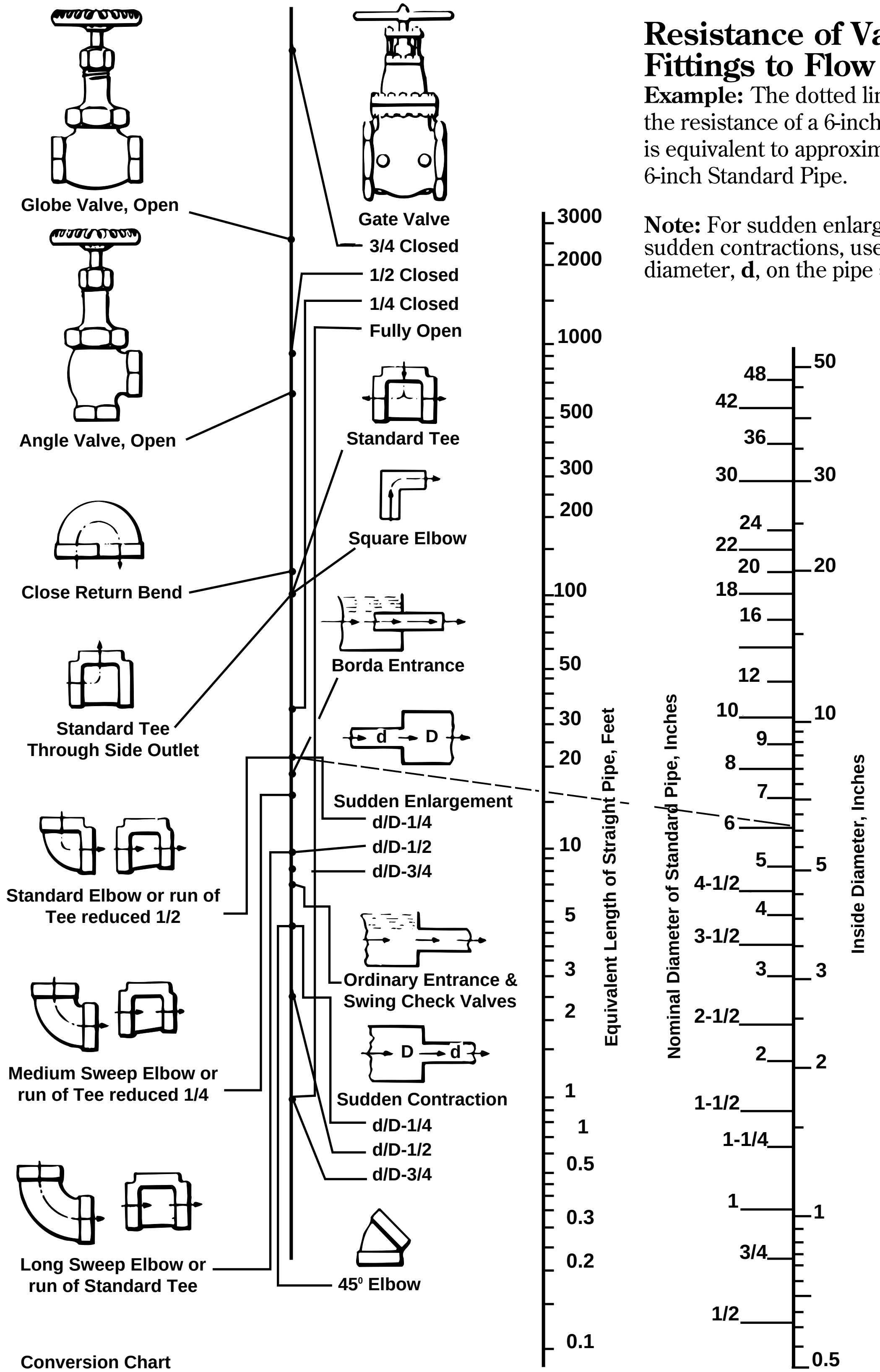
Flow		Head Loss in ft/100ft (m/100m)							Flow	
gpm	(L/s)	.75 in. (19.05 mm)	1 in. (25.4 mm)	1.25 in. (31.75 mm)	1.5 in. (38.1 mm)	2 in. (50.8 mm)	2.5 in. (63.5 mm)		gpm	(L/s)
5	.34	10.50	3.25	.84	.40	.16	.05	3 in. (76.2 mm)	5	.34
10	.63	38.00	11.70	3.05	1.43	.50	.17	.07	10	.63
15	.95	80.00	25.00	6.50	3.05	1.07	.37	.15	15	.95
20	1.26	136.00	42.00	11.10	5.20	1.82	.61	.25	20	1.26
25	1.58	4 in. (101.6 mm)	64.00	16.60	7.85	2.73	.92	.38	25	1.58
30	1.9	.13	89.00	23.00	11.00	3.84	1.29	.54	30	1.90
35	2.21	.17	119.00	31.20	14.70	5.10	1.72	.71	35	2.21
40	2.52	.22	152.00	40.00	18.80	6.60	2.20	.91	40	2.52
45	2.84	.28	5 in. (127 mm)	50.00	23.20	8.20	2.76	1.16	45	2.84
50	3.15	.34	.11	60.00	28.40	9.90	3.32	1.38	50	3.15
60	3.79	.47	.16	85.00	39.60	13.90	4.65	1.92	60	3.79
70	4.42	.63	.21	113.00	53.00	18.40	6.20	2.57	70	4.42
75	4.73	.72	.24	129.00	60.00	20.90	7.05	2.93	75	4.73
80	5.05	.81	.27	145.00	68.00	23.70	7.90	3.28	80	5.05
90	5.68	1.00	.34	6 in. (152.4 mm)	84.00	29.40	9.80	4.08	90	5.68
100	6.31	1.22	.41	.17	102.00	35.80	12.00	4.96	100	6.31
125	7.89	1.85	.63	.26	7 in. (177.8 mm)	54.00	17.60	7.55	125	7.89
150	9.46	2.60	.87	.36	.17	76.00	25.70	10.50	150	9.46
175	11.05	3.44	1.16	.48	.22	8 in. (203.2 mm)	34.00	14.10	175	11.05
200	12.62	4.40	1.48	.61	.28	.15	43.10	17.80	200	12.62
225	14.20	5.45	1.85	.77	.35	.19	54.30	22.30	225	14.20
250	15.77	6.70	2.25	.94	.43	.24	65.50	27.10	250	15.77
275	17.35	7.95	2.70	1.10	.51	.27	9 in. (228.6 mm)	32.30	275	17.35
300	18.93	9.30	3.14	1.30	.60	.32	.18	38.00	300	18.93
325	20.50	10.80	3.65	1.51	.68	.37	.21	44.10	325	20.50
350	22.08	12.40	4.19	1.70	.77	.43	.24	50.50	350	22.08
375	23.66	14.20	4.80	1.95	.89	.48	.28	10 in. (254 mm)	375	23.66
400	25.24	16.00	5.40	2.20	1.01	.55	.31	.19	400	25.24
425	26.81	17.90	6.10	2.47	1.14	.61	.35	.21	425	26.81
450	28.39	19.80	6.70	2.74	1.26	.68	.38	.23	450	28.39
475	29.97		7.40	2.82	1.46	.75	.42	.26	475	29.97
500	31.55		8.10	2.90	1.54	.82	.46	.28	500	31.55
750	47.32			7.09	3.23	1.76	.98	.59	750	47.32
1000	63.09			12.00	5.59	2.97	1.67	1.23	1000	63.09
1250	78.86				8.39	4.48	2.55	1.51	1250	78.86
1500	94.64				11.70	6.24	3.52	2.13	1500	94.64
1750	110.41					7.45	4.70	2.80	1750	110.41
2000	126.18					10.71	6.02	3.59	2000	126.18

Table 1.3

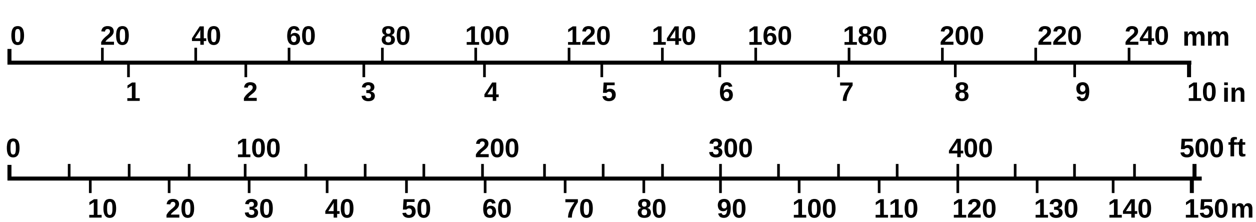
Resistance of Valves and Fittings to Flow of Fluids

Example: The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16 feet of 6-inch Standard Pipe.

Note: For sudden enlargements or sudden contractions, use the smaller diameter, **d**, on the pipe size scale.



Conversion Chart



Electrochemical Series

Corroded End — Least Noble

Magnesium
Magnesium Alloys
Zinc
Beryllim
Aluminum Alloys
Cadmium
Mild Steel or Iron
Cast Iron
Low Alloy Steel
Austanitic Cast Iron
Aluminum Bronze
Naval Brass
Yellow Brass
Red Brass
18-8 Stainless Steel (Active)
18-8-3 Stainless Steel (Active)
Lead-Tin Solders
Lead
70-30 Copper Nickel
Tin
Brasses
Copper
Bronzes
Copper-Nickel Alloys
Monel
Admiralty Brass, Aluminum Brass
Manganese Bronze
Silicon Bronze
Tin Bronze
Silver Solder
Nickel (Passive)
Chromium-Iron (Passive)
18-8 Stainless Steel (Passive)
18-8-3 Stainless Steel (Passive)
Silver
Ni-Cr-Mo Alloy 8
Titanium
Ni-Cr-Mo Alloy C
Gold
Platinum
Graphite

Protected End — Most Noble

Corrosion Rates of Various Metals in Seawater

Representative Corrosion Rates in Seawater

Metal	Corrosion Rate in Quiet Seawater* mm/yr
Aluminum	.02 to 1.20
Zinc	.02 to .25
Lead	> .02 to .38
Iron (Steel)	.10 to .25
Silicon Iron	0.00 to .07
Stainless Steel**	0.00 to .12
Copper Alloys	.01 to .38
Nickel Alloys	0.00 to .02
Titanium	Nil
Silver	Nil
Platinum	Nil

* Rates are ranges for general loss in seawater at ambient temperatures and velocities no greater than 1 m (3 ft) per second. Pitting penetration is not considered.

** Many stainless steels exhibit high rates of pitting in stagnant seawater.

Marine Engines Application and Installation Guide

- Ventilation
- Exhaust System

LEKM7146-01 (Supersedes LEKM7146) (05-00)

Ventilation

General Information

There are three aspects to ventilation:

Ventilation Air

The flow of air required to carry away the radiated heat of the engine(s) and other engine room machinery.

Combustion Air

The flow of air required to burn the fuel in the engine (propulsion and auxiliaries).

Crankcase Fumes Disposal

The crankcase fumes of the engine must be either ingested by the engine or piped out of the engine room.

Ventilation Air

Engine room ventilation has two basic purposes:

- To provide an environment which permits the machinery and equipment to function dependably.
- To provide a comfortable environment for personnel.

Radiated heat from the engines and other machinery in the engine room is absorbed by engine room surfaces. Some of the heat is transferred to atmosphere or the sea through the hull. The remaining radiated heat must be carried away by the ventilating system.

A system for *discharging* ventilation air from the engine room must be included in the construction of the vessel. Do not expect the engine(s) to carry all the heated ventilation air from the engine room by way of the exhaust piping.

Routing

Correct Ventilation Air Routing is Vital

Comfortable air temperatures in the engine room are impossible without proper routing of the ventilation air.

Fresh air should enter the engine room as far from the sources of heat as practical and *as low as possible.* Since heat causes air to rise, it should be discharged from the highest point in the engine room, preferably directly over the engine. Avoid incoming ventilation air ducts which blow cool air toward hot engine components. This mixes the hottest air in the engine room with incoming cool air, raising the temperature of all the air in the engine room.

Relative Efficiency of Various Routing of Ventilation Air

The sketches on page 2 illustrate the relative efficiency of various ventilation routing:

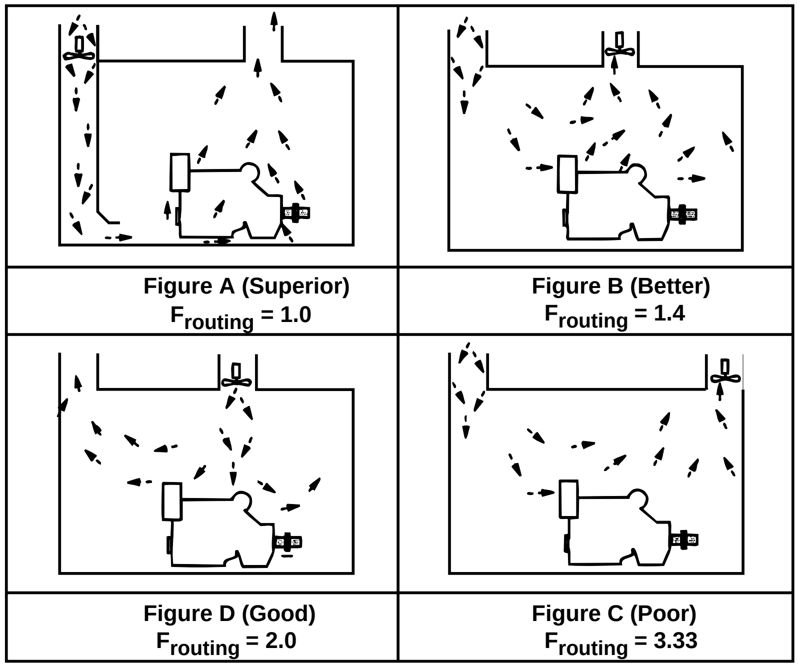


Figure 1.1

Where: $F_{routing}$ is a factor which relates the relative efficiency of various ventilation air routing.

Example:
If the routing in Figure A (upper left) is used as a base to which the others are compared:

- 1.4 times more air is required (duct cross-sectional area and fan capacity) to adequately ventilate the machinery space illustrated in Figure B (upper right).
- It takes twice as much air (duct cross-sectional area and fan capacity) to adequately ventilate the machinery space illustrated in Figure C (lower right).
- 3.3 times more air is required (duct cross-sectional area and fan capacity) to adequately ventilate the machinery space illustrated in Figure D (lower left).

Engine Room Temperature

A properly designed engine room ventilation system will maintain engine room air temperatures within 9°C (15°F) above the ambient air temperature (ambient air temperature refers to the air temperature surrounding the vessel). Maximum engine room temperature should not exceed 49°C (120°F).

Quantity Required

In general, changing the air in the engine room every one or two minutes will be adequate, if flow routing is proper.

Provisions should be made by the installer to provide incoming ventilation air of 0.1 – 0.2 m^3/min (4-8 cfm) per installed horsepower (both propulsion and auxiliary engines). This does not include combustion air for the engines. (See following remarks on engine combustion air, page 7.)

Engine exhaust ventilation air should be 110 to 120% of the incoming ventilation air. The excess exhaust ventilation air accomplishes two things:

- It compensates for the thermal expansion of incoming air.
- It creates an in draft to confine heat and odor to the engine room.

Operation in extreme cold weather may require reducing ventilation air flow to avoid uncomfortably cold working conditions in the engine room. This is easily done by providing ventilation fans with two speed motors (100% and 50 or 67% speeds).

Through-Hull Opening Design

There must be openings for air to enter the engine room and openings for air to leave the engine room.

There should be an *inlet* for cool air to enter, and a *discharge* for hot air to leave, on each side of the hull. If it is impractical to have two separate openings per side, then avoid having hot discharged air mix with cool air entering the engine room.

Design Features

Size

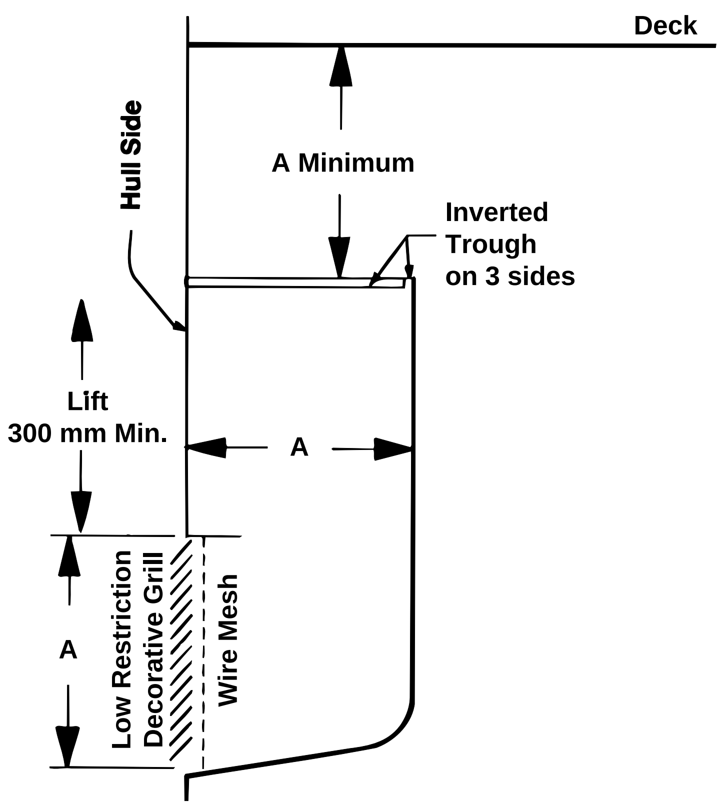
Size the openings (A) to keep the air velocity (in the openings) below 610 m/min (2,000 ft/min).

Air Entering the Engine Room

The engine room must have openings for air to enter. The air may also enter from the accommodation spaces (staterooms, galley, salon, companionways, pilot house, etc.)* or directly through the hull or deck. Engine room air inlets through accommodation spaces can be troublesome.

If air is to enter the engine room from the accommodation spaces (staterooms, galley, salon, companionways, pilot house, etc.), good design practice will include sound deadening treatments for the opening(s) which conduct air from the accommodation spaces to the engine room.

*Heating and/or air conditioning of accommodation spacecs will be made much more complicated if the engines must rely on that heated/cooled air for combustion. Engine room air inlets through accommodation spaces simplify the task of ensuring the engine room inlet air is kept clean and free from rain or spray.



Features of Through-Hull Ventilation Openings

Figure 1.2

Air Leaving the Engine Room

The through-hull or through-deck openings for discharge of heated ventilation air should be located aft of and higher than all intake openings to minimize recirculation.

- The intake air opening should be located forward of — and, if convenient — at a lower elevation, than . . .
- The ventilation air opening, discharging heated ventilation air, which should be located aft of — and at a higher elevation than the intake air opening — to minimize recirculation. Cross- and following-winds make total elimination of ventilation air recirculation impossible.

Fans

In modern installations, natural draft ventilation is too bulky for practical consideration. Adequate quantities of fresh air are best supplied by powered (fan-assisted) systems.

Location

Fans are most effective when they withdraw ventilation air from the engine room (suction fans) and discharge the hot air to the atmosphere.

Type

Ventilating air fans may be of the axial flow type (propeller fans) or the centrifugal type (squirrel cage blowers). When mounting fans in ventilating air discharge ducts (most effective location), the fan motors should be outside the direct flow of hot ventilating air for longest motor life. The design of centrifugal fans (squirrel cage blowers) is ideal.

Sizing

The *name plate* ratings of fans do not necessarily reflect their *as-installed* conditions. Just because a fan's name plate says it will move 1000 cfm of air does not mean it will move 1000 cfm through an engine room which has severely restricted inlet and/or outlet openings. Fans are often rated under conditions which do not reflect *as-installed* flow restrictions. In general, the *as-installed* conditions will be more severe than the fans name plate rating conditions.

Combustion Air

Quantity Required

A diesel engine requires approximately 0.1 m^3 of air/min/brake kW (2.5 ft^3 of air/min/bhp) produced.

Combustion Air Ducts

Design combustion air ducts to have a minimum flow restriction.

Very large amounts of air flow through the combustion air ducts.

Air Cleaners

Engines must be protected from ingesting foreign material. The engine-mounted air filter elements must never be remote-mounted, without factory approval.

If large amounts of sea spray, dust, or insects are expected, external, remote-mounted, precleaners may be installed at the inlet to a duct system to extend the life of the engine-mounted filter elements.

Air Cleaner Service Indicators

Air cleaner service indicators signal the need to change air filter elements when a restriction of 7.47 kPa (30 in.) of water — measured while the engine is producing full rated-power — develops. This allows an acceptable operating period before air filter service or replacement is required.

Duct Restriction

Total duct air flow restriction, including air cleaners, should not exceed 2.49 kPa (10 in.) of water measured while the engine is producing full rated power. It is good design practice to design combustion air ducts to give the lowest practical restriction to air flow, since this will result in longer times between filter element service or replacement.

Velocity of Air in Combustion Air Ducts

Combustion air duct velocity should not exceed 610 m/min (2,000 ft/min). Higher velocities will cause unacceptable noise levels and excessive flow restriction.

Water Traps

Traps should be included to eliminate any rain or spray from the combustion air. Rain and spray can cause very rapid plugging of the paper air filter elements on some Caterpillar Engines. This will reduce the flow of air through the engine, raising the exhaust temperature with potentially damaging effects.

Temperature

A well designed engine room ventilation system will provide engines with air whose temperature is not higher than 8.5°C (15°F) above the ambient temperature though derating of Caterpillar Marine engines is not required so long as combustion air temperatures remain below 49°C (120°F).

Rain and Spray

The combustion air should be free of liquid water, though water vapor — humidity — is acceptable.

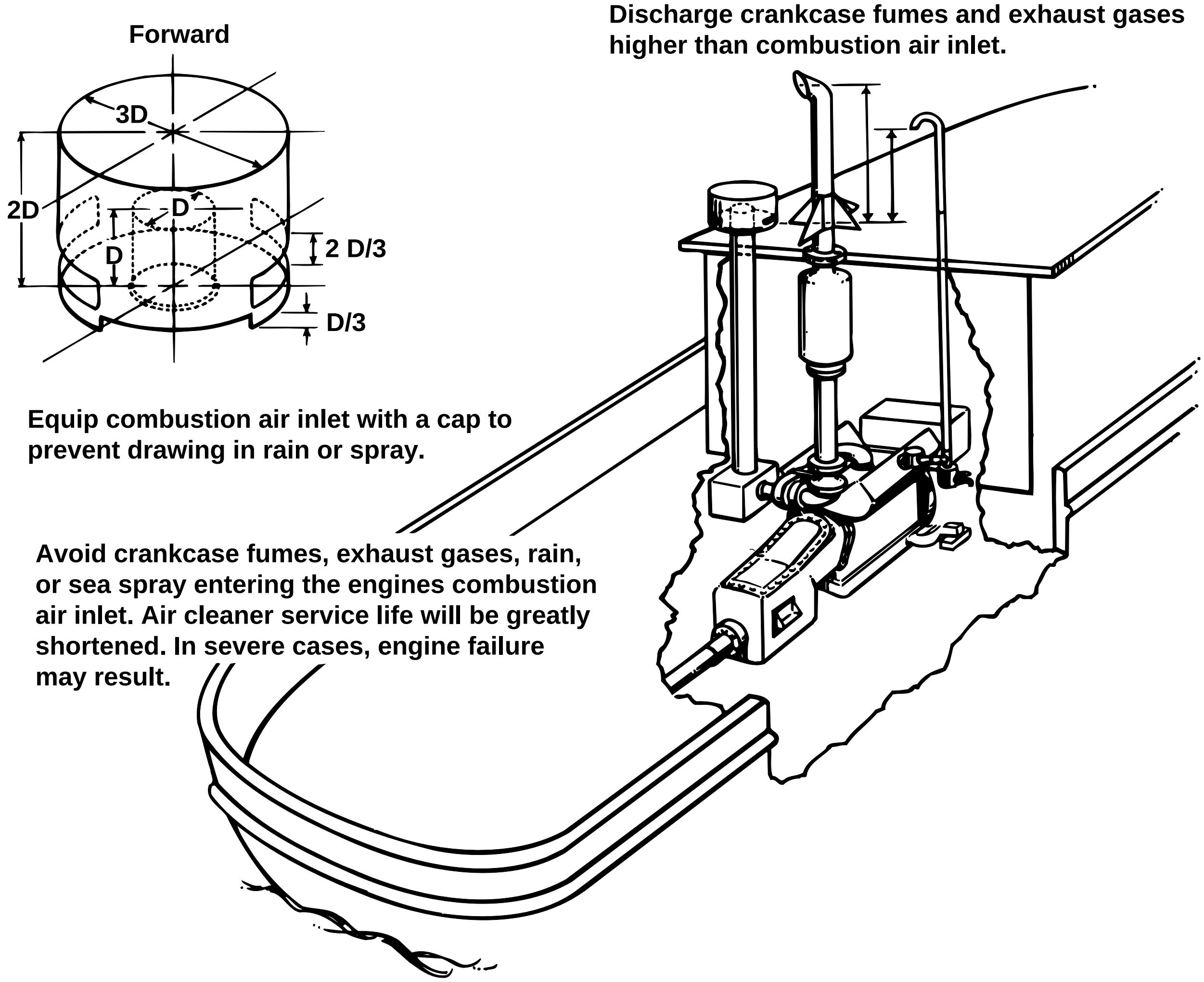


Figure 1.3

Sizing of Combined Combustion and Ventilation Air Ducts — Rule of Thumb

Air Must Be Allowed to Enter the Engine Room Freely.

A useful rule of thumb is:

- Use 4-6 sq cm of duct cross-section area per engine kW and no more than three (3) right angle bends. A larger area allows more air flow into the engine room.
- Use 0.5-0.75 sq in. of duct cross-section area per engine horsepower and no more than three (3) right angle bends. A larger area allows more air flow into the engine room.

If more right angle bends are required, increase the pipe diameter by one pipe size.

Crankcase Fumes Disposal

Normal combustion pressures of an internal combustion engine cause a certain amount of blowby past the piston rings into the crankcase. To prevent pressure buildup within the crankcase, vent tubes are provided to allow the gas to escape. 3100 and 3208 and high performance 3176, 3406, 3408, and 3412 marine engines consume their crankcase fumes, by drawing the fumes into the engine's air intake system. Larger Caterpillar marine engines must have their crankcase fumes piped away from the engine to prevent the fumes from plugging their high-efficiency paper air filter elements.

Pipe Sizing

Generally use pipe of the same size as the crankcase fumes vent on the engine. If the pipe run is longer than approximately 3 m (10 ft) or if there are more than three (3) 90° elbows, increase the pipe inside diameter by one pipe size.

Common Crankcase Vent Piping Systems

A separate vent line for each engine is required. Do not combine the piping for multiple engines.

Location of Crankcase Vent Termination

Crankcase fumes must not be discharged into air ventilating ducts or exhaust pipes. They will become coated with oily deposits.

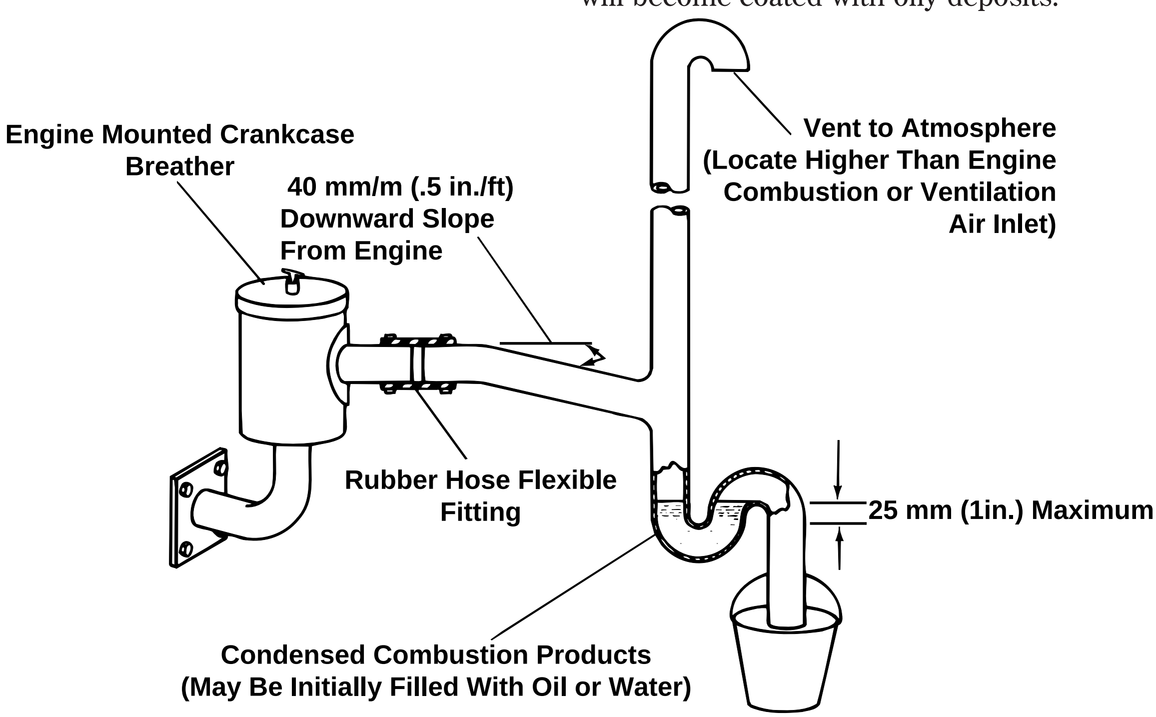


Figure 1.4

The crankcase vent pipe may be directed into the exhaust gas flow at the termination of the exhaust pipe.

Preferably, the crankcase vent pipe will vent directly to the atmosphere. The vent pipe termination should be directed to prevent rain/spray entering the engine.

Condensation/Rainwater in Crankcase Fumes Piping

Loops or low spots in a crankcase vent pipe can collect rainwater and/or condensed combustion products. These liquids may be trapped in a drip collector and drained to minimize the amount of oily discharge through the vent pipe and prevent restriction of normal discharge of fumes.

Required Slope of Crankcase Fumes Disposal Piping

Avoid horizontal runs in crankcase vent piping. Install the vent pipe with a minimum slope of 40 mm/m (.5 in./ft).

Maximum Pressure in the Engine Oil Sump

Under no circumstances should crankcase pressure vary more than 25.4 mm (1 in.) of water from ambient barometric pressure when the engine is new*. Higher crankcase pressures will cause oil leaks. A powered crankcase fumes disposal system should create no more than 25.4 mm (1 in.) of water vacuum in the crankcase.

Crankcase Volumes

The volume of an engine's crankcase is required for the sizing of crankcase pressure relief valves. See the following table.

Crankcase Volumes	
Model	**Crankcase Volume**
3116	.045 m^3 (1.59 ft^3)
3126	.045 m^3 (1.59 ft^3)
3176	.14 m^3 (4.9 ft^3)
3304B	.05 m^3 (1.76 ft^3)
3306B	.063 m^3 (2.22 ft^3)
3208	.0011 m^3 (.039 ft^3)
3406B	.25 m^3 (8.8 ft^3)
3408B	.3 m^3 (10.5 ft^3)
3412	.4 m^3 (14.1 ft^3)
D399	1.28 m^3 (45.2 ft^3)
D398	.96 m^3 (33.9 ft^3)
D379	.64 m^3 (22.6 ft^3)
3508	.65 m^3 (22.9 ft^3)
3512	.98 m^3 (34.6 ft^3)
3516	1.41 m^3 (49.8 ft^3)
3606	2.3 m^3 (81.2 ft^3)
3608	3.06 m^3 (108 ft^3)
3612	3.12 m^3 (110.2 ft^3)
3616	4.08 m^3 (144 ft^3)

Many marine classification societies (MCS) expect crankcase pressure relief valves to be installed on engines with crankcase volumes more than 0.61 m^3 (21.5 ft^3) or cylinder bores over 200 mm (7.89 in.) in diameter. Caterpillar offers crankcase pressure relief valves on engines larger than the 3408.

* As the engine approaches its overhaul interval, blowby (one of the causes of crankcase pressure) will tend to increase. Careful monitoring of crankcase pressure will provide valuable guidance on the condition of an engine's valve guides and piston rings.

Special Ventilation Considerations

Refrigeration Equipment

Prevent refrigerant leakage into the engine's air intake system. Freon or ammonia will cause severe engine damage if drawn into the engine's combustion chambers. The chemicals in refrigerants become highly corrosive acids in the engine's combustion chambers.

If refrigeration equipment is installed within the same compartment as a diesel engine, the diesel engine must take its combustion air from a shipyard-supplied ductwork system which carries air to the engine from an area free of refrigerant fumes.

Exhaust Pipe Insulation Recommended

Long runs of hot, uninsulated exhaust piping will dissipate more heat into the engine room than all the machinery surfaces combined. Completely insulate all exhaust piping within the engine room area. All hot surfaces within the engine room should be insulated if high air temperatures are to be avoided.

Test With Doors and Hatches Closed

Ventilating systems must be designed to provide safe working temperatures and adequate air flow when hatches and doors are secured for bad weather conditions. Test the ventilation system with the vessel fully secured for bad weather. This condition will reflect the most severe test of the ventilation system.

Air Velocity for Personnel Comfort

Maintain air velocity of at least 1.5 m/s (5 ft/s) in working areas adjacent to sources of heat, or where air temperatures exceed 100°F (35°C). This does *not* mean that all the air in the engine room should be agitated so violently. High air velocity around engines and other heat sources is not good ventilation practice. High velocity air aimed at engines will hasten transfer of heat to the air, raising average engine room air temperature.

CATERPILLAR®

Exhaust System

General Information
Wet Exhaust System
Dry Exhaust System
Exhaust Backpressure Limits
Measuring Backpressure
Warning Against “Common” Exhaust Systems
Slobber (extended periods of insufficient load)

General Information

The exhaust system carries the engine's exhaust gases out of the engine room, through piping, to the atmosphere.

A good exhaust system will have minimum back pressure.

Exhaust back pressure is generally detrimental as it tends to reduce the air flow through the engine. Indirectly, exhaust back pressure tends to raise exhaust temperature which will reduce exhaust valve and turbocharger life. There are two general types of exhaust systems seen on boats, wet exhaust systems and dry exhaust systems.

Wet Exhaust System

Wet exhaust systems are characterized by the following:

- Generally the exhaust gases are mixed with the sea water which is discharged from the sea water side of the engine's jacket water heat exchanger.
- Particulate and condensable/soluble gaseous emissions from the exhaust system are effectively scrubbed from the exhaust gases, reducing the possibility of atmospheric pollution. Exhaust piping which is cool enough to be made of uninsulated, fiberglass reinforced plastic (FRP) or rubber.
- The moisture of exhaust gases and sea water is discharged from the boat at or slightly below the vessels waterline.
- With the relatively small elevation difference between the engine's exhaust discharge elbow and the vessels waterline, it is difficult to design a system which will always prevent water from entering the engine through the exhaust system. While a number of proprietary exhaust components are available to help avoid this problem, the most common generic methods are exhaust risers and water lift mufflers.

Exhaust Risers

One way to minimize the possibility of water entering the engine from backflow in the wet exhaust system is to have a steep downward slope on the exhaust piping, downstream of the engine.

Exhaust risers are pipes which elevate the exhaust gases, allowing a steeper slope in the downstream piping.

The risers must be insulated or water-jacketed to protect persons in the engine compartment from the high temperatures of the exhaust gas in the riser. The sea water is not injected into the exhaust gases until downstream of the top of the riser, so the upward-sloping portion of the riser is dangerously hot if not insulated or water-jacketed.

The weight of locally fabricated risers (not provided by Caterpillar) must be supported from the engine and marine transmission. Do not attempt to carry the weight of the risers from the boat's overhead or deck structure. The risers will vibrate with the engine-transmission. The risers must be supported independently from the hull to avoid transmitting those vibrations into the boat's structure and passenger compartments.

Exhaust risers (for other than the smallest Caterpillar Engines) are not available from Caterpillar; the diversity of the various boat builders engine compartments prevents designing a riser with wide usability. See fabricators of custom exhaust components for exhaust risers.

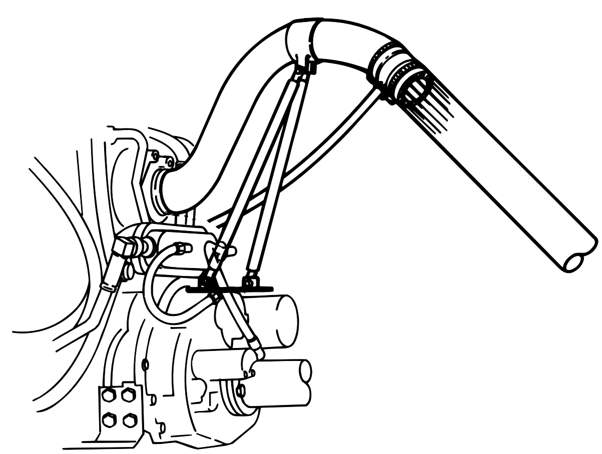
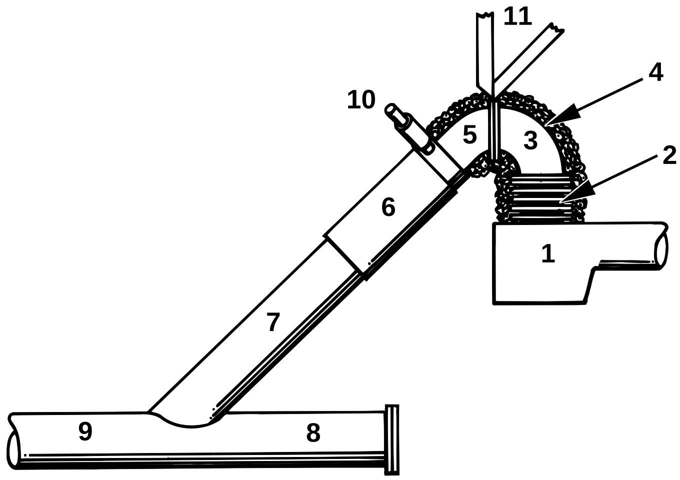


Figure 2.1



**WET EXHAUST SYSTEM USING DRY EXHAUST ELBOWS
AT ENGINE EXHAUST DISCHARGE**

1. **turbocharger heat shield**
2. **flexible pipe connection**
3. **elbow—℄ bend radius ≥ diameter of pipe**
4. **insulation, must not restrict flexibility of 2**
5. **elbow (min 15⁰) with water discharge ring**
6. **exhaust hose**
7. **connecting exhaust pipe**
8. **surge pipe**
9. **discharge pipe**
10. **raw water discharge connection**
11. **support from overhead structure**

Figure 2.2

Water Lift Mufflers

Another way to minimize the possibility of water entering the engine from backflow in the wet exhaust system is by using a water lift muffler.

Water lift mufflers are small, sealed tanks, mounted to the deck in the engine compartment. The tanks have two (2) connections, an inlet connection and an outlet connection. An additional small drain connection in the bottom is often provided. The inlet enters the tank through the top or side.

The tubing of the inlet connection does not extend past the tank walls. The tubing of the outlet connection enters the tank walls, through the top, and extends to the bottom of the tank, where it terminates on an angle.

As the mixture of seawater and exhaust gas enters the tank from the inlet connection, the water level rises in the tank. As the water level rises, the water surface gradually reduces the gas flow area entering the discharge pipe. The reduced area for gas flow causes a great increase in gas velocity. The high speed of the gases, entering the outlet pipe, finely divides the water. The finely divided water is transported to the highest elevation of the exhaust piping as a mist of water droplets.

If good design practice is not followed, the engines exhaust back pressure limit is easily exceeded. The vertical (upward sloping) portion of piping immediately downstream of a water lift muffler must be designed as a pneumatic conveyer, using high exhaust gas velocities to lift finely divided droplets of the sea water to a point from which the gas/water mixture can be safely allowed to drain to the thru-hull fitting.

The designer should size the diameter of the upward sloping portion of the exhaust piping—between the water lift muffler and the highest system elevation—such that the velocity of the exhaust-gas-and-water droplet mixture is not below 25.4 m/sec (5000 ft/min), with the engine running at rated load and speed.

If this velocity is not maintained, the water droplets will not remain in suspension. The water will be forced out of the reservoir of the water lift muffler as a solid *slug* of water. This will cause the exhaust back pressure to be the same as a column of water the height of the upward sloping muffler discharge piping.

If the velocity in the upward sloping muffler discharge piping is kept above 25.4 m/sec (5000 ft/min), then the exhaust back pressure will be much lower.

Hose vs Rigid Exhaust Pipe

The weight and heat of the water and exhaust gases can cause nonrigid exhaust piping to sag or deform, leaving low spots between pipe supports.

If the slope of the piping is too shallow, water will collect in the low spots and choke off the flow of exhaust gas. This will lead to excessive exhaust back pressure, smoke, high exhaust temperatures and, in severe cases, premature engine failures.

Hose and other nonrigid piping must be evenly supported over its entire length.

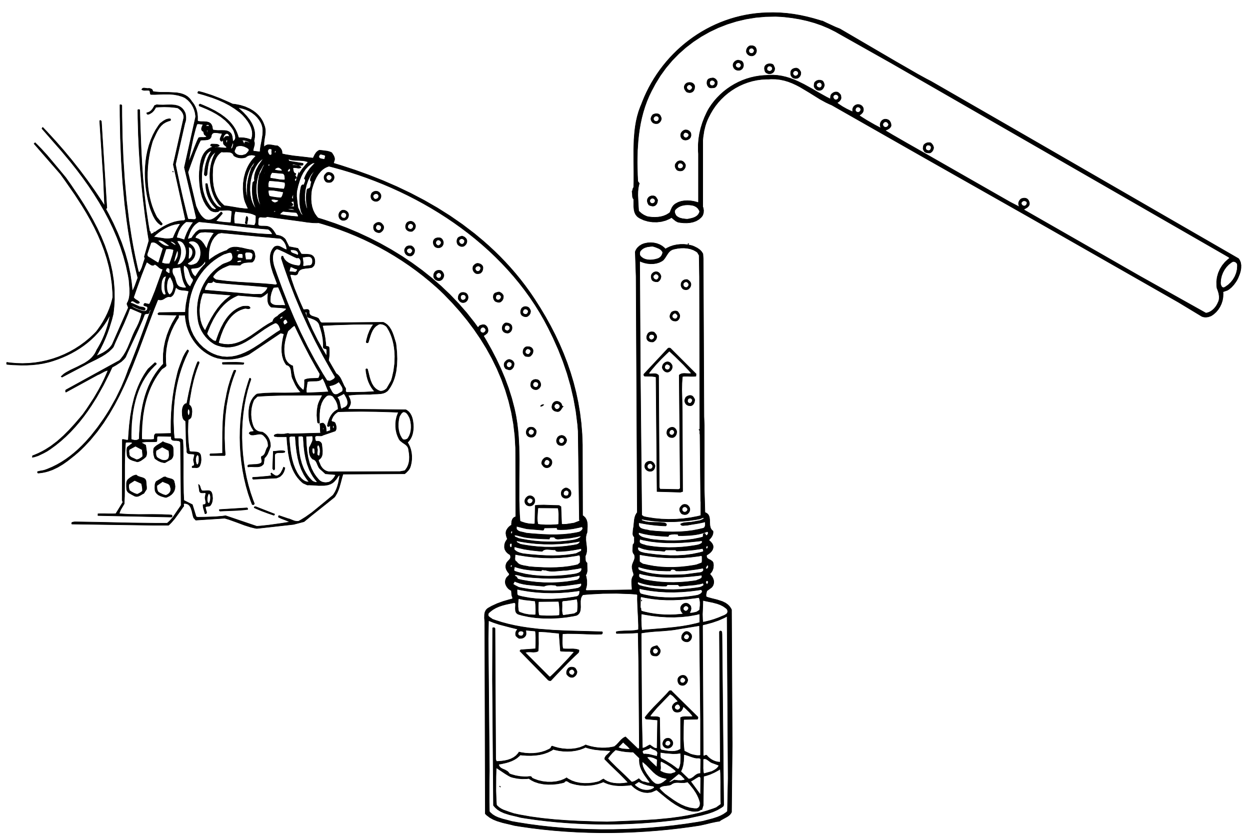


Figure 2.3

Preventing Wave Action From Forcing Water Into Wet Exhaust Systems

Waves, striking the hull's exhaust opening, can force water up into the exhaust system. If the waves are severe, or if the exhaust system design allows, the water can reach the engine. Early turbocharger failure or piston seizure may result.

There are a number of ways the kinetic energy of waves entering the engine's exhaust system can be harmlessly dissipated.

The traditional method of preventing water from entering an idle engine is to locate the engine far enough above the water line that breaking waves do not reach the height of the exhaust elbow. While the relative elevation of the engine to the water line is fixed and unchangeable, it is possible to design an exhaust system which protects the engine from ingesting water.

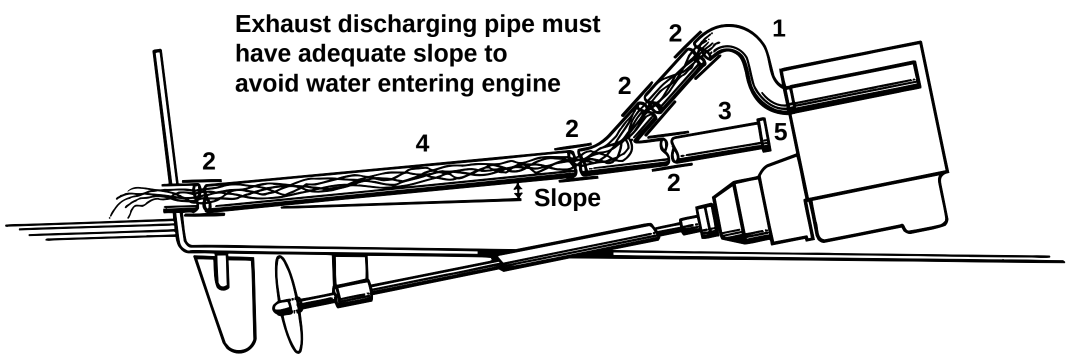
Features of such an exhaust system will include the following:

- Sufficient elevation difference between the water line and the highest point in the exhaust piping to prevent even small amounts of water from reaching the engine.
- Some method of dissipating the kinetic energy of the waves as they enter the exhaust piping. The more effective the method of wave energy dissipation, the lower the elevation difference required.

In no case should the elevation difference between the water line and the highest point in the exhaust piping be less than 560 mm (22 in.).

Surge Chamber

A surge chamber is a branch of the exhaust piping, near the engine, which has one end closed off. When a wave of water enters the exhaust pipe and moves toward the engine, the air trapped in front of the wave will be compressed into the surge chamber. The cushion of compressed air in the surge chamber will force almost all waves back out.



WET EXHAUST SYSTEM
(Engine Mounted Above Water Line)

1. **water cooled exhaust elbow on engine - sea water cools elbow, then discharges through peripheral slot at discharge end of elbow into exhaust pipe**
2. **rubber exhaust hose flexible connection - must be oil and heat resistant**
3. **backwater surge chamber - prevents sea water surging into engine exhaust when vessel at rest with stern exposed to oncoming waves**
4. **exhaust pipe - should have slight downward gradient toward discharge end**
5. **end cover plate - removable for inspection and cleanout purposes**

Figure 2.4

Valve in Exhaust Discharge

A valve located where the exhaust piping penetrates the hull can keep waves from entering the exhaust piping when the engine is not running. The valve mechanism should not include any components which rely on sliding contact to maintain flexibility. This type of action has proven troublesome in an atmosphere of salt water and exhaust gas. A flexible strip of one of the chemically inert plastics can provide hinge action.

Valves in Exhaust Water Cooling Lines

The cooling water which is injected into the exhaust gas stream must not be interrupted, for any reason, while the engine is running. Without a dependable supply of cooling water, the high temperature of the exhaust gases will cause severe and rapid deterioration of plastic or rubber exhaust pipe, with potentially disastrous consequences.

Therefore, to protect against inadvertent loss of exhaust system cooling, **shut-offs or valves of any kind must never be used in the lines supplying cooling water to water cooled exhaust fittings.**

Location of Exhaust Discharge Opening

All diesel engines will eventually discharge some smoke through their exhaust systems if not when they are new, then certainly near the end of their useful time before overhaul. Locating exhaust discharge openings as far aft as possible will minimize the hull and deck area exposed to the eventual discoloration.

Dry Exhaust System

Dry Exhaust System Warnings

Insulation

It is the responsibility of the engine installer to protect combustible parts of the boat and provide personnel protection from the heat of dry exhaust systems piping. **Exposed parts of dry exhaust piping can exceed 650°C (1200°F).**

Rain/Spray

It is the responsibility of the engine installer to provide appropriate drain connections, rain caps or other means to protect the engine from rainwater or sea spray entering the engine through the dry exhaust piping. Long runs of exhaust piping require traps to drain moisture. Traps installed at the lowest point of the line near the exhaust outlet prevent rain water from reaching the engine.

Slope exhaust lines from engine and silencer to the trap so condensation will drain.

Traps may be built by inserting a vertical pipe, with a drain petcock, down from a tee section in the line.

Slope the last few feet of the exhaust pipe discharge to prevent rain water or spray from entering the pipe. Alternatively, fit some form of rain cap to a vertical exhaust pipe section.

Saw cuts in the exhaust pipe to allow rain/spray to drain harmlessly. Deform the edges of all slots. Use a punch on engine side slot edges. Bend inward. Cut through no more than 60° of the pipe circumference.

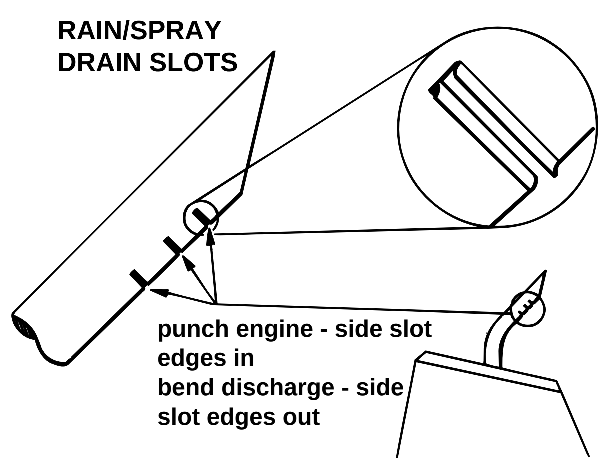


Figure 2.5

Exhaust Gas Recirculation

Exhaust stacks must be designed so engine exhaust is discharged high enough, and in a direction to keep it clear of the air turbulence created by wind swirling around the vessel's superstructure. Engine air cleaners, turbochargers and aftercoolers clogged with exhaust products will cause engine failures.

Ventilation

Mufflers and other large dry exhaust system components would be best mounted outside the engine compartment.

This is suggested to minimize the additional and unnecessary load on the machinery compartment's ventilation system.

Flexible Connections

The exhaust pipe must be isolated from the engine with flexible connections. They should be installed as close to the engine's exhaust outlet as possible. A flexible exhaust connection has three primary functions:

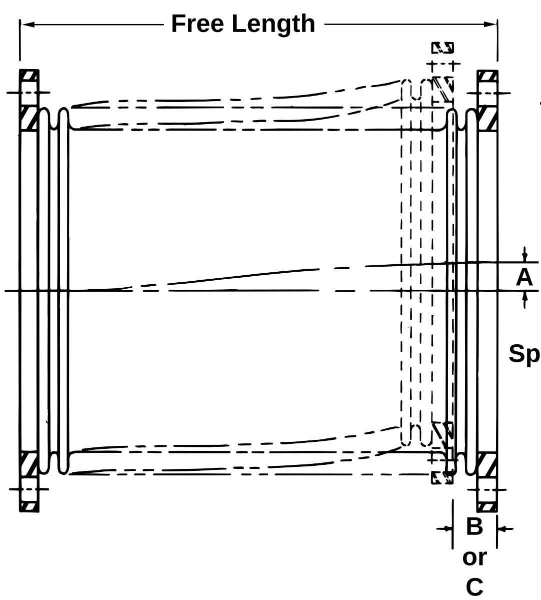
- To isolate the weight of the exhaust piping from the engine. No more than 28 kg (60 lb) of exhaust piping weight should be supported by the engine.
- To relieve exhaust components of excessive vibrational fatigue stresses.
- To allow for relative shifting between reference points on engine exhaust components. This shifting has numerous causes. It may result from expansion and contraction of components due to temperature changes, or by slow but continual creep processes that take place throughout the life of any structure.

Softness or flexibility is very important to prevent excessive vibratory stresses. The flexible connector must have high fatigue life to enable it to survive for indefinite periods. Softness prevents transmission of vibration beyond the connection. Resistance to fatigue keeps it from breaking under vibratory or recycling stresses.

To prevent the exhaust coupling from flexing during exhaust system construction, it is recommended that straps be tack welded between the two flanges to make the coupling rigid. Remove these straps before starting the engines.

The growth and shrinkage of the exhaust pipe must be planned or it will create excessive loads on exhaust piping and supporting structure. Long runs of dry exhaust pipes can be subjected to very severe stresses from expansion and contraction. From its cold state, a steel exhaust pipe will expand about 0.11 mm/m for each 100°C rise of exhaust temperature (0.0076 in./ft of pipe for each 100°F). This amounts to about 52 mm

INSTALLATION LIMITATIONS
OF BELLOWS - TYPE FLEXIBLE EXHAUST FITTING



A. max. offset between flanges mm	A. in.	B. max. compression from free length mm	B. in.	C. max. extension from free length mm	C. in.
0	0	38.1	1.50	25.4	1.00
3.05	.12	22.86	.90	12.7	.50
6.35	.25	15.24	.60	7.11	.28
9.65	.38	10.16	.40	5.08	.20
12.7	.50	5.84	.23	.00	.00
19.05	.75	0.0	.00	.00	.00

Spring Rate of Bellows = 19.21 N/m (170 lb/in.) Approx.

If bellows-type exhaust fittings are distorted beyond limits shown in table while engine is operating at full throttle, service life will be greatly reduced.

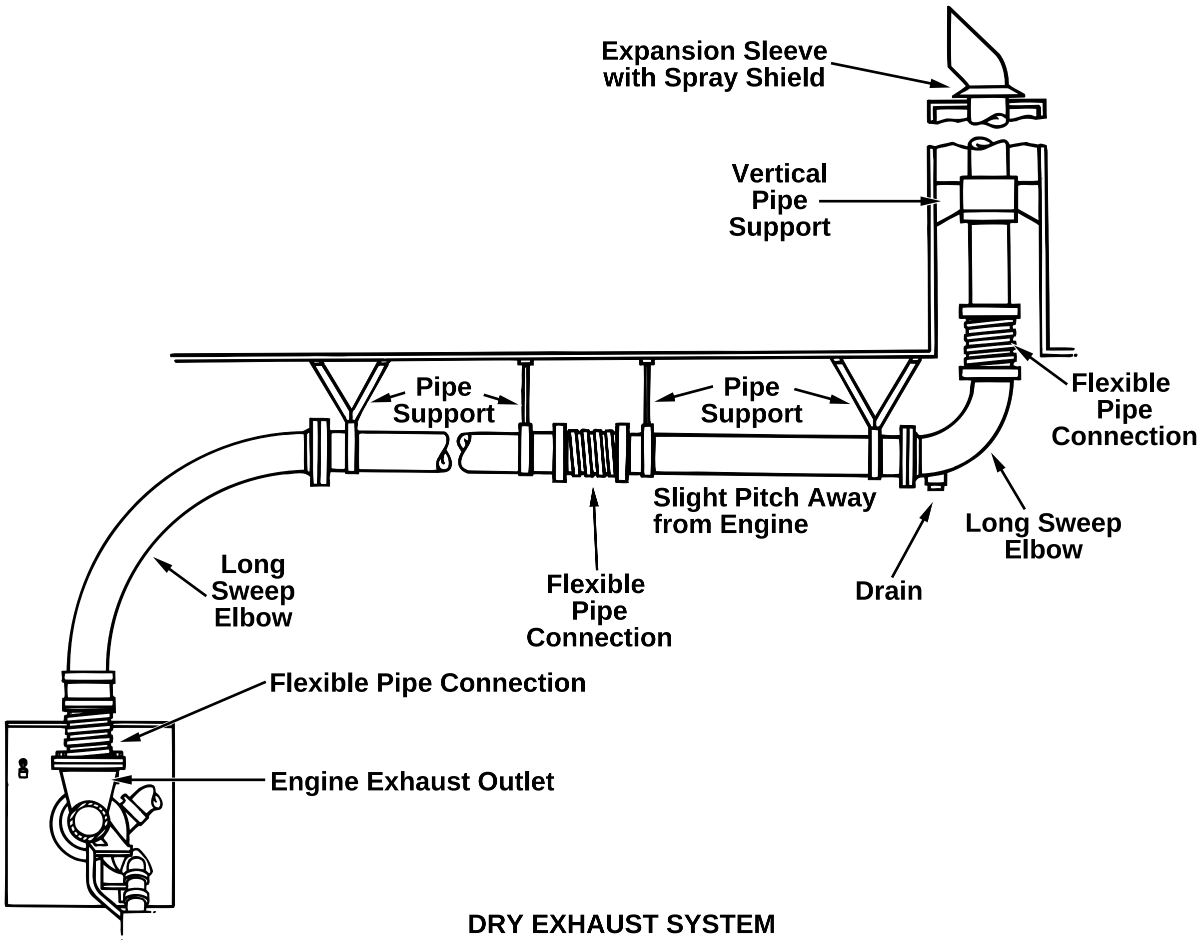
"Lagging" or insulation must not restrain flexibility of bellows.

Figure 2.6

expansion per m from 35°C to 510°C (0.65 in. expansion for each 10 ft of pipe from 100°F to 950°F).

Divide long runs of exhaust pipe into sections having expansion joints between sections. Each section should be fixed at one end and be allowed to expand at the other.

It is of utmost importance that the flexible pipe connection, when insulated, be insulated in such a way that the flexible pipe connection can expand and contract freely within the insulation. This generally requires either a soft material or an insulated sleeve to encase the flexible pipe connection.



DRY EXHAUST SYSTEM

Figure 2.7

Dimension Chart							
Flange Type	P/N	Nominal ID	Flange OD	Bolt Circle	# of Bolts	Hole Diameter	Overall Length
		mm (in.)	mm (in.)	mm (in.)		mm (in.)	mm (in.)
Rect.	3N3015	127 (5)	163* (6.42)	127** (5)	4	16.7 (.657)	457 (18)
Rect.	3N3017	152 (6)	203*** (8.00)	152**** (6)	4	16.7 (.657)	609 (24)
Circ.	5L6297	203 (8)	274 (11)	250 (9.875)	8	15.875 (.625)	304 (12)
Circ.	5N9505	304 (12)	400 (15.75)	375 (14.75)	12	13.8 (.500)	304 (12)

* This dimension is the length of each side of a square flange.
** This dimension is from bolt center to bolt center along a side of the rectangular flange.
*** This dimension is the length of each side of a square flange.
**** This dimension is from bolt center to bolt center along a side of the rectangular flange.

Dry Exhaust System Pipe Supports

The exhaust piping supports/hangers are very important. If the piping is supported with some flexibility between the piping and the structure of the boat, the boat will be much quieter and more comfortable for the occupants.

Exhaust Ejector-Automatic Ventilation

A relatively simple system utilizing an engine's exhaust for ventilating an engine room can be utilized with most dry exhaust systems.

Utilizing the normally wasted kinetic energy of discharging exhaust gases, this system may draw out a quantity of ventilating air approximately equal to the flow of exhaust gas.

Air must be allowed to enter the engine room freely.

A useful rule of thumb is:

- Use 10 cm^2 of duct cross section area per engine kilowatt and not more than three (3) right angle bends.
- Use 1.25 $in.^2$ of duct cross section area per engine horsepower and no more than three (3) right angle bends.

If more right angle bends are required, increase the pipe diameter by one pipe size.

For best results, the intake air openings should discharge cool air into the engine room near the floor level. After the intake air has been heated by contact with hot surfaces in the engine room, draw the ventilating air out from a point directly over the engines, near the engine room overhead.

Place the ejector in the exhaust system just prior to the exhaust's discharge to atmosphere to avoid back pressure on the mixture of exhaust gas and hot air through any length of stack. Any bends in the exhaust stack following the mixture can seriously affect the system's performance.

Furthermore, the exhaust stack will remain cooler and cleaner if the engine exhaust is contained within the exhaust piping throughout its run through the stack. The discharged ventilation air will tend to cool the exhaust stack upstream of the point where it is mixed with the exhaust gases.

Exhaust ejectors are most effective on vessels with only one propulsion engine. On multiple engine installations, if one engine is operated at reduced load, the ejector air flow for the engine with reduced load may reverse, pulling exhaust gas from the more heavily loaded engine into the engine room. The following diagrams illustrate methods of laying out the system:

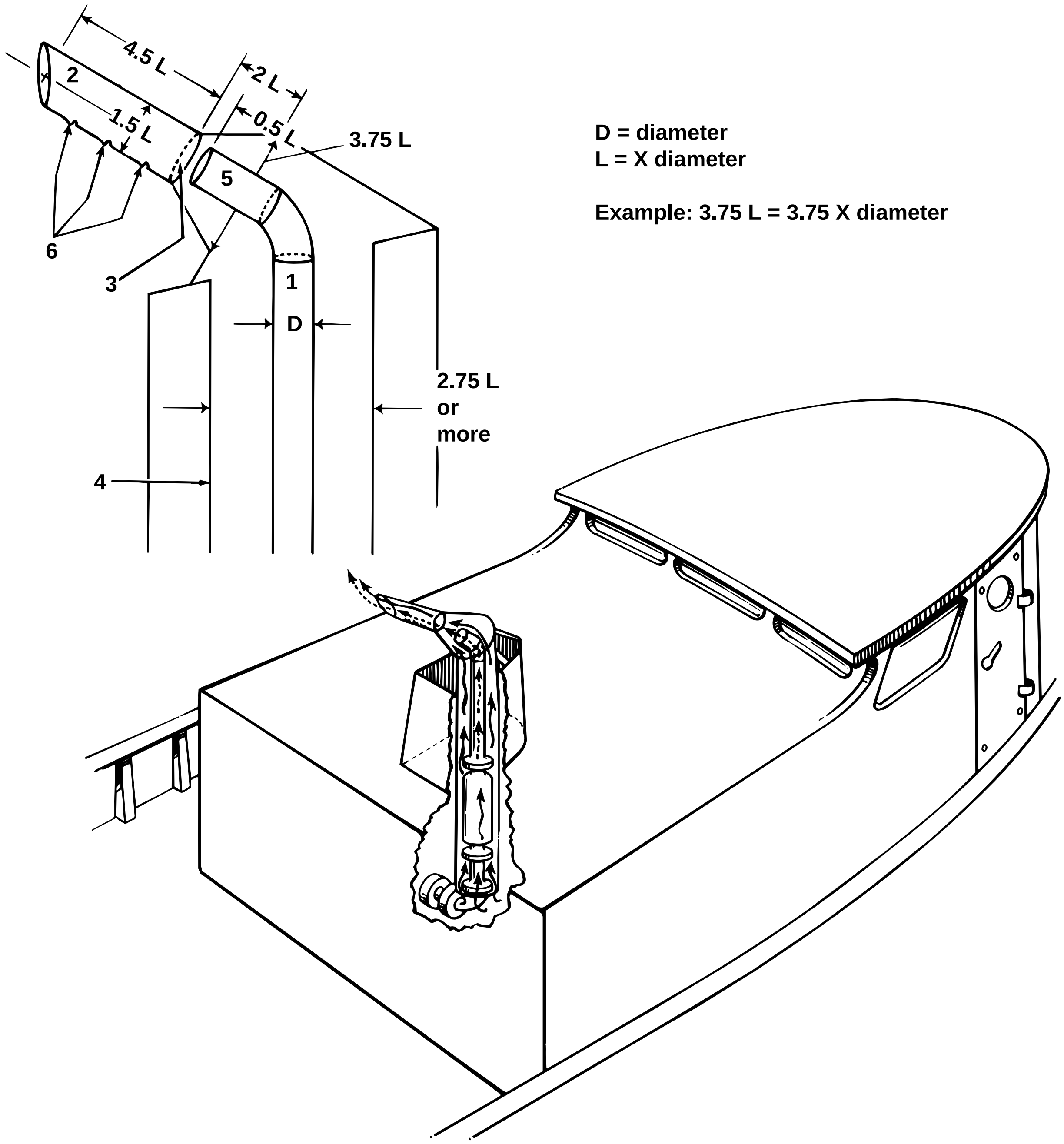

4.5 L
2
1.5 L
2 L
0.5 L
3.75 L
5
6
3
1
D
2.75 L
or
more
4
D = diameter
L = X diameter
Example: 3.75 L = 3.75 X diameter

Figure 2.8

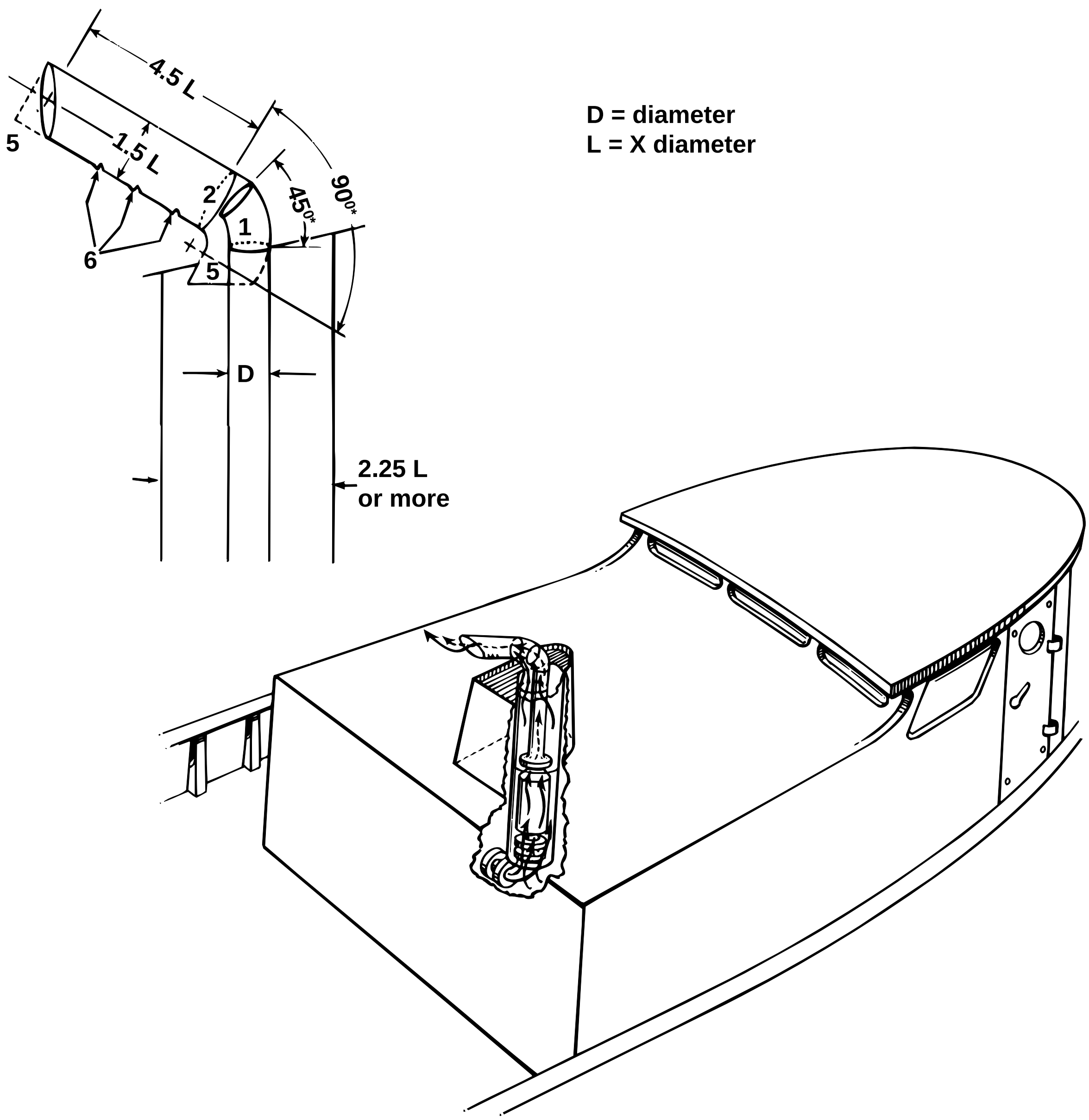


Figure 2.9

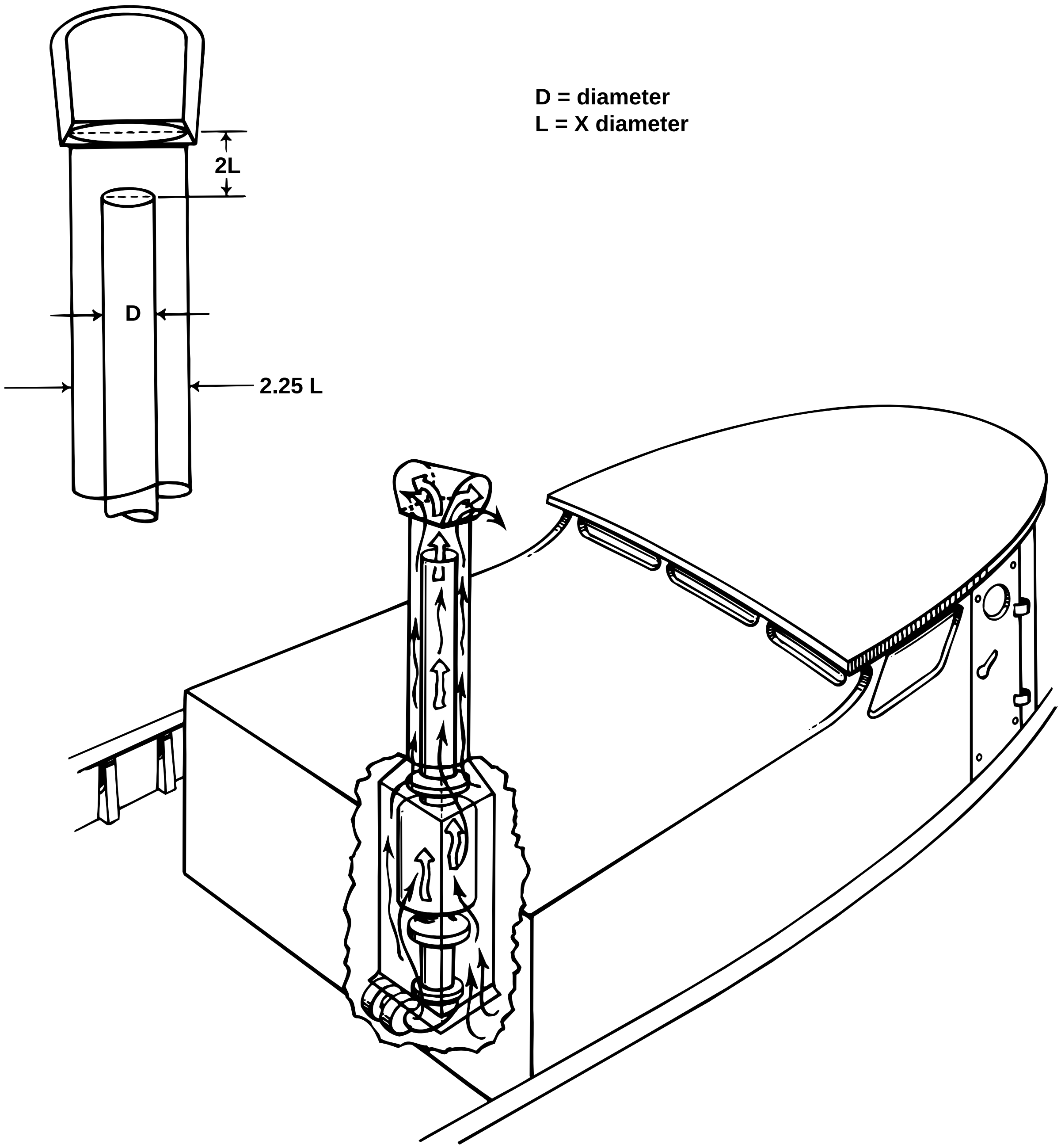


Figure 2.10

Formulae for System Diameter to Back Pressure Limits.

These formulae allow the exhaust system designer to calculate a pipe diameter which, when fabricated into an exhaust system, will give exhaust back pressure less than the appropriate limit.

Calculate the pipe diameter according to the formula, then choose the next larger commercially available pipe size.

Exhaust Pipe Diameter to Meet Back Pressure Limits

(Metric Units System)

P = Back pressure limit (kPa) See section on exhaust back pressure limits for specific engine.

$$D = \sqrt[5]{3600000\,\frac{LSQ^2}{P}}$$

D = Inside diameter of pipe (mm)

Q = Exhaust gas flow (m^3/min). See engine performance curve.

L = Length of pipe (m). Includes all of the straight pipe and the straight pipe equivalents of all elbows.

$$\mathbf{S}\ (kg/m^3) = \frac{352}{\text{Exhaust Temperature} + 273°F}$$

S = Specific weight of gas (kg/m^3)

Exhaust Pipe Diameter to Meet Back Pressure Limits

(English Units System)

P = Back pressure limit (inches of water). See section on exhaust back pressure limits for specific engine.

$$D = \sqrt[5]{\frac{LSQ^2}{187P}}$$

D = Inside diameter of pipe (inches).

Q = Exhaust Gas Flow (ft^3/min). See engine performance curve.

L = Length of pipe (feet). Includes all of the straight pipe and the straight pipe equivalents of all elbows.

S = Specific weight of gas (lb/ft^3).

$$S\ (lb/ft^3) = \frac{39.6}{\text{Exhaust Temperature} + 460°F}$$

Formulae for Straight Pipe Equivalent Length of Various Elbows

To obtain straight pipe equivalent length of elbows:

English Units	Metric Units
Standard Elbow (radius of elbow equals the pipe diameter)	
$L = 33\,\frac{D}{12}$	$L = 33\,\frac{D}{1000}$
Long Radius Elbow radius greater than 1.5 pipe diameters	
$L = 20\,\frac{D}{12}$	$L = 20\,\frac{D}{1000}$
45° Elbow	
$L = 15\,\frac{D}{12}$	$L = 15\,\frac{D}{1000}$

Where:

L = Straight Pipe Equivalent Length of Elbows

D = Pipe Diameter

Useful Facts for Exhaust System Designers

Conversion Factors

psi = 0.0361 x in. of water column

psi = 0.00142 x mm of water column

psi = 0.491 x in. of mercury column

kPa = 6.3246 x mm of water column

kPa = 4.0 in. of water column

kPa = 0.30 x in. of mercury column

kPa = 0.145 psi

Mufflers

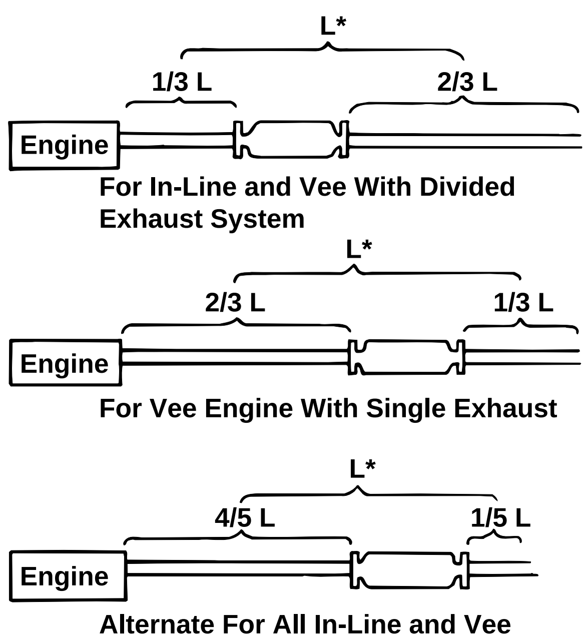
Exhaust noise attenuation is best performed with a quality muffler. However, the attenuation characteristics of a muffler are not the same for all frequencies. The effect of a given muffler could be quite different if the engine runs at two different speeds. The manufacturer must be contacted for any specific muffling characteristics.

The location of the muffler within the exhaust piping, whether close to the engine or nearer the exhaust outlet, and the number of engine cylinders is important. It will affect the efficiency of the muffler. The sketch below offers suggestions for the most efficient muffler location.

Exhaust Back Pressure Limits

As the exhaust gas moves through the exhaust system, it experiences frictional resistance—causing back pressure on the engine's turbocharger discharge. Exhaust system back pressure has a number of bad effects on the engine.

Excessive back pressure will shorten exhaust valve and turbocharger life due to increased exhaust temperatures. Excessive exhaust back pressure wastes fuel as well.



***L Is Total Length of Exhaust Piping Excluding The Muffler**

MUFFLER LOCATION

Figure 2.11

Maximum Exhaust Back Pressure Limits		
	Metric Units	**English Units**
Naturally Aspirated Engines	8.47 kPa	34 in. H_20
Turbocharged Engines	6.72 kPa	27 in. H_20
Except for those specifically listed below:		
435 hp 3208, 3116, 3126 Propulsion Engines	9.96 kPa	40 in. H_20
3600 Family Engines	The limits on 3600 are competitive with other medium speed diesels. Higher pressure will affect performance and fuel consumption.	

To ensure the above limits are not exceeded during operation, it is recommended the design limit be not more than one-half the specified back pressure limits.

Measuring Back Pressure

Measure exhaust back pressure by a water manometer at the fitting provided in the engines exhaust discharge location. Use a system similar to that shown below.

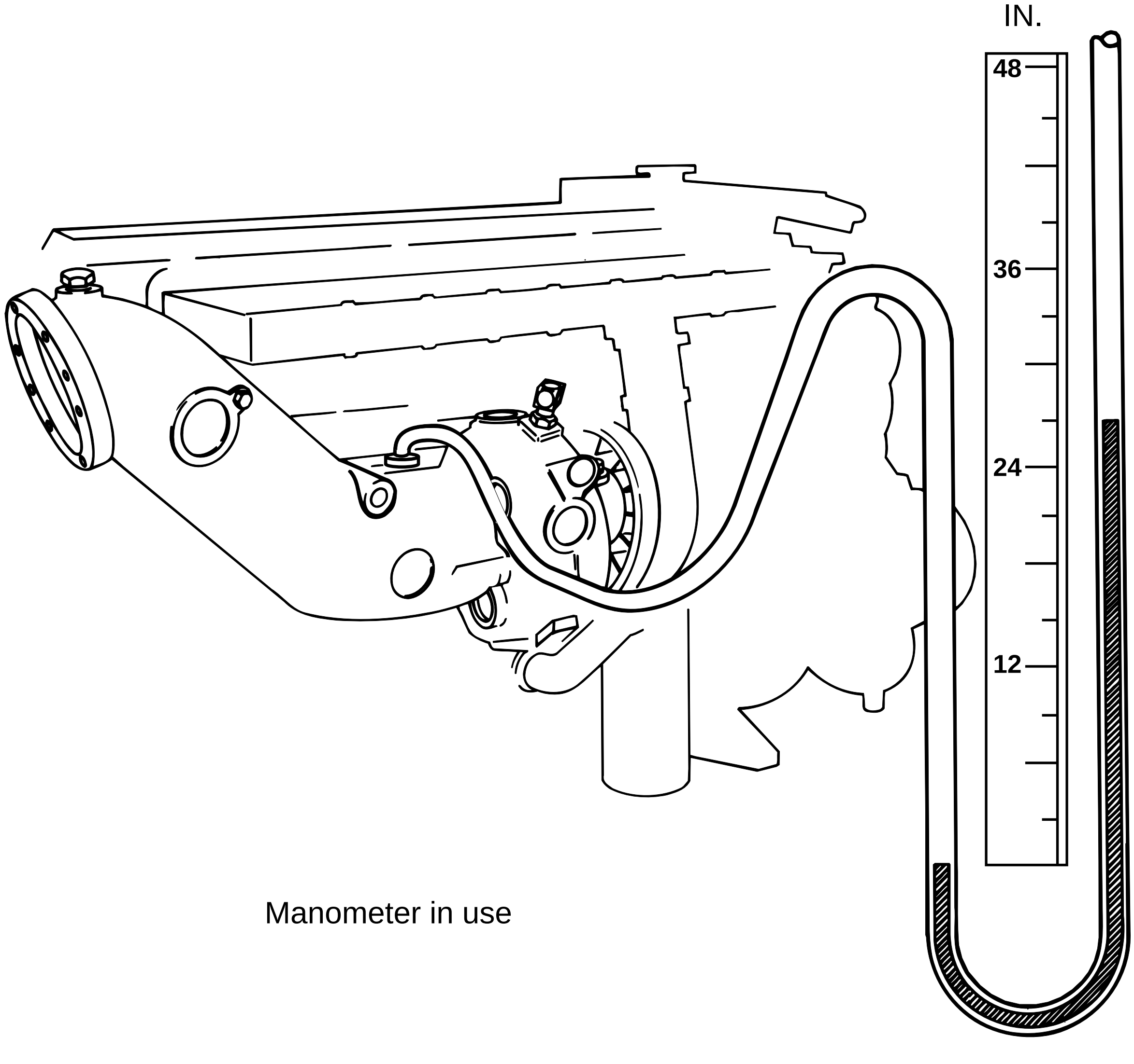


Figure 2.12

Warning Against *Common* Exhaust Systems

Although economically tempting, a common exhaust system for multiple engine installations is rarely acceptable. Combined exhaust systems allow operating engines to force exhaust gases into engines not operating. Every gallon of fuel burned provides about one gallon of water in the exhaust. This water vapor condenses in cold engines and quickly causes engine damage. Soot clogs turbochargers, aftercoolers or air cleaner elements. Duct valves separating engine exhausts is also discouraged. High temperature warp valve seats or soot deposits cause leakage.

Slobber (Extended Periods of Insufficient Load)

Extended engine operation at no load or lightly loaded conditions (less than 15% load) may result in exhaust manifold slobber. Exhaust manifold slobber is the black oily fluid that can leak from exhaust system joints. The presence of exhaust manifold slobber does not necessarily indicate an engine problem. Engines are designed to operate at loaded conditions.

At no load or lightly loaded conditions, the sealing capability function of some integral engine components may be adversely affected. Exhaust manifold slobber is not usually harmful to the engine; the results can be unsightly and objectionable in some cases.

Exhaust manifold slobber consists of fuel and/or oil mixed with soot from the inside of the exhaust manifold. Common sources of oil slobber are worn valve guides, worn piston rings and worn turbocharger seals. Fuel slobber usually occurs with combustion problems.

A normally operating engine should be expected to run for at least one hour at light loads without significant slobber. Some engines may run for as long as three, four or more hours before slobbering. However all engines will eventually slobber if run at light loads. External signs of slobber will be evident unless the exhaust system is completely sealed.

If extended idle or slightly loaded periods of engine operation are mandatory, the objectionable effects of the engine slobber can be avoided by loading the engine to at least 30% load for approximately ten minutes every four hours. This will remove any fluids that have accumulated in the exhaust manifold. To minimize exhaust manifold slobber, it is important that the engine is correctly sized for each application.

CATERPILLAR®

Marine Engines Application and Installation Guide

- Lubrication Systems
- Fuel Systems

LEKM7147-01 (Supersedes LEKM7147) (05-00)

CATERPILLAR®

Lubrication Systems

General Information

Bearing failure, piston ring sticking and excessive oil consumption are classic symptoms of oil related engine failure. There are numerous ways to avoid them. Three of the most important are Scheduled Oil Sampling (S•O•S), regular maintenance of the lubrication system, and the use of correct lubricants. Taking these measures can mean the difference between experiencing repeated oil related engine failure and benefiting from a productive and satisfactory engine life. The following information will acquaint the reader with oil; what it is composed of and what its functions are, how to identify its contamination and degradation, typical consequences, and some preventive measures to help you protect your engine against the devastating effects of oil related engine failure.

Function

Engine oil performs several basic functions:

It cleans the engine by carrying dirt and wear particles until the filters can extract and store them.

It cools the engine by carrying heat away from the pistons, cylinder walls, valves, and cylinder heads to be dissipated in the engine oil cooler.

It cushions the engine's bearings from the shocks of cylinder firing.

It lubricates the wear surfaces, reducing friction.

It neutralizes the corrosive combustion products.

It seals the engine's metal surfaces from rust.

Additives

Lubricating oil consists of a mixture of base oil fortified with certain additives. Depending on the type of base, paraffinic, asphaltic, naphthenic or intermediate (which has some of the properties of the former), different additive chemistries are used.

Additive Types

The most common additives are: detergents, oxidation inhibitors, dispersants, alkalinity agents, anti-wear agents, pour-point dispersants and viscosity improvers.

Detergents help keep the engine clean by chemically reacting with oxidation products to stop the formation and deposit of insoluble compounds.

Oxidation inhibitors help prevent increases in viscosity, the development of organic acids and the formation of carbonaceous matter.

Dispersants help prevent sludge formation by dispersing contaminants and keeping them in suspension.

Alkalinity agents help neutralize acids.

Anti-wear agents reduce friction by forming a film on metal surfaces.

Pour-point dispersants keep the oil fluid at low temperatures by preventing the growth and agglomeration of wax crystals.

Viscosity improvers help prevent the oil from becoming too thin at high temperatures.

Total Base Number (TBN)

Understanding TBN requires some knowledge of fuel sulfur content. Most diesel fuel contains some degree of sulfur. One of lubricating oils functions is to neutralize sulfur by-products, retarding corrosive damage to the engine. Additives in the oil contain alkaline compounds which are formulated to neutralize these acids. The measure of this reserve alkalinity in an oil is known as its TBN. Generally, the higher the TBN value, the more reserve alkalinity or acid-neutralizing capacity the oil contains.

Viscosity

Viscosity is the property of oil which defines its thickness or resistance to flow. Viscosity is directly related to how well an oil will lubricate and protect surfaces that contact one another. Oil must provide adequate supply to all moving parts, regardless of the temperature. The more viscous (thicker) an oil is, the stronger the oil film it will provide. The thicker the oil film, the more resistant it will be to being wiped or rubbed from lubricated surfaces. Conversely, oil that is too thick will have excessive resistance to flow at low temperatures and so may not flow quickly enough to those parts requiring lubrication. It is therefore vital that the oil has the correct viscosity at both the highest and the lowest temperatures at which the engine is expected to operate. Oil thins out as temperature increases. The measurement of the rate at which it thins out is called the oil's viscosity *index* (or VI). New refining techniques and the development of special additives which improve the oil's viscosity index help retard the thinning process.

The Society of Automotive Engineers (SAE) standard oil classification system categorizes oils according to their quality.

Cleanliness

Normal engine operation generates a variety of contamination—ranging from microscopic metal particles to corrosive chemicals. If the engine oil is not kept clean through filtration, this contamination would be carried through the engine via the oil.

Oil filters are designed to remove these harmful debris particles from the lubrication system. Use of a filter beyond its intended life can result in a plugged filter.

A plugged filter will cause the bypass valve to open releasing unfiltered oil. Any debris particles in the oil will flow directly to the engine. When a bypass valve remains open, the particles that were previously trapped by the filter may also be flushed from it and then through the open bypass valve. Filter plugging can also cause distortion of the element. This happens when there is an increase in the pressure difference between the outside and inside of the filter element. Distortion can progress to cracks or tears in the paper. This again allows debris to flow into the engine where it can damage components.

Recommended Oils for Various Caterpillar Products

Refer to Operation and Maintenance Manual for lubricaiton specifications.

Contamination

Contamination refers to the presence of unwanted material or contaminants in the oil. There are seven contaminants commonly found in contaminated oil.

1. Wear Elements
   Wear elements are regarded as those elements whose presence indicates a part or component which is wearing. Wear elements include: copper, iron, chromium, aluminum, lead-tin, molybdenum, silicon, nickel, and magnesium.

2. Dirt and Soot
   Dirt can get into the oil via air blowing down past the rings and by sticking to the oil film and being scraped down from cylinder walls. Soot is unburned fuel. Black smoke and a dirty air filter indicate its presence. It causes oil to turn black.

3. Fuel

4. Water
   Water is a by-product of combustion and usually exits through the exhaust stack. It can condense in the crankcase if the engine operating temperature is insufficient.

5. Ethylene Glycol/Antifreeze

6. Sulfur Products/Acids

7. Oxidation Products
   Oxidation Products Oxidation products cause the oil to thicken; oxidation rate is accelerated by high temperature of the inlet air.

Diagnostic Tests

Caterpillar's Scheduled Oil Sampling (S•O•S) program is a series of diagnostic tests designed to identify and measure contamination and degradation in a sample of oil. S•O•S is composed of three basic tests:

1. Wear Analysis

2. Chemical & Physical Tests

3. Oil Condition Analysis

A brief explanation of what each of these tests involves is in order.

Wear Analysis

Wear analysis is performed with an atomic absorption spectrophotometer. Essentially, the test monitors a given component's wear rate by identifying and measuring concentrations of wear elements in oil. Based on known normal concentration data, maximum limits of wear elements are established. After three oil samples are taken, trend lines for the various wear elements can be established for the particular engine. Impending failures can be identified when trend lines deviate from the established norm.

Wear analysis is limited to detecting component wear and gradual dirt contamination. Failures due to component fatigue, sudden loss of lubrication or sudden ingestion of dirt occur too rapidly to be predicted by this type of test.

Chemical & Physical Tests

Chemical and physical tests detect water, fuel and antifreeze in the oil and determine whether or not their concentrations exceed established limits.

The presence and approximate amount of water is detected by a *sputter test*. A drop of oil is placed on a hot plate controlled at 110°C (230°F). The appearance of bubbles is a positive indication (0.1% to 0.5% is the acceptable range).

The presence of fuel is determined with a Setaflash Tester. The tester is calibrated to quantify the percentage of fuel dilution.

The presence of antifreeze can also be determined by a chemical test. (Any indication that is positive is unacceptable.)

Oil Condition Analysis

Oil condition analysis is performed via infrared analysis. This test determines and measures the amount of contaminants such as soot and sulfur, oxidation and nitration products. Although it can also detect water and antifreeze in oil, infrared analysis should always be accompanied by wear analysis and chemical and physical tests to assure accurate diagnosis. Infrared analysis can also be used to customize (reduce, maintain, or extend) oil change intervals for particular conditions and applications.

Recognizing the Causes & Effects of Contamination

S•O•S identifies and measures various contaminants in the oil which cause engine failure. For example, a high concentration of copper indicates thrust washer or bushing wear. A high concentration of chromium indicates piston ring damage (with the exception of plasma coated rings). S•O•S gives you an opportunity to inspect the condition of these parts and, if necessary, take action to prevent further damage. Here are some examples of typical contaminants and what effect they have on the condition of your engine.

Silicon

Above normal readings of silicon can indicate a major problem. Oil loaded with silicon becomes, in effect, a grinding compound which can remove metal from any number of parts during operation.

Sodium

A sudden increase in sodium readings indicates inhibitor leaking from the cooling system. Inhibitor may indicate antifreeze in the system which can cause oil to thicken and become like sludge, leading to piston ring sticking and filter plugging.

Silicon, Chromium, Iron

A combination such as this signals dirt entry through the induction system, possibly causing ring and liner wear.

Silicon, Iron, Lead, Aluminum

This combination indicates dirt in the lower portion of the engine, possibly leading to crankshaft and bearing wear.

Aluminum

This can be critical. Concentrations of aluminum suggest bearing wear. Relatively small increases in the levels of this element should receive immediate attention because once rapid wear begins, the crankshaft may produce large metal particles which will become trapped in the oil filters.

Iron
Iron can come from any number of sources. It can also appear as rust, after engine storage. Frequently, when accompanied by a loss of oil control, increases in iron contamination indicate severe liner wear.

Soot
A high soot content is not usually the direct cause of failure but as solid particles which will not dissolve in the oil, it can plug oil filters and deplete dispersant additives. Soot indicates a dirty air cleaner, engine lug, excessive fuel delivery, or repeated acceleration in the improperly set rack limiter (smoke limiter). It can also indicate a poor quality fuel.

Water
Water combined with oil will create an emulsion which will plug the filter. Water and oil can also form a dangerous metal corroding acid. Most instances of water contamination are the result of condensation within the crankcase. More serious contamination occurs when a leak in the cooling system allows water to enter from outside the engine oil system.

Fuel
Fuel contamination decreases the oil's lubricating properties. The oil no longer has the necessary film strength to prevent metal-to-metal contact. This can lead to bearing failure and piston seizure.

Sulfur
The presence of sulfur signals danger to all engine parts. The type of corrosive wear attributed to high sulfur content can also cause accelerated oil consumption. Also, the more fuel consumed during an oil change interval, the more sulfur oxides are available to form acids. Therefore, an engine working under heavy loads should be checked more often. Also, its TBN should be checked more frequently. Fuel sulfur damage can cause piston ring sticking, and corrosive wear of the metal surfaces of valve guides, piston rings and liners.

Engine operating conditions can also play a major role in the type and degree of oil contamination. A dry environment will for instance affect silicon readings. Another example is engines which stand idle for long periods at a time. The liners in such engines will rust at an unusually rapid rate; oil samples will reveal high iron readings.

Changing Lubrication Oil

Changing lubrication oil can be simplified by using a system as described below: Install a machine thread-to-pipe-thread adapter* in the oil pan drain.

Connect a length of flexible, oil and temperature resistant hose to the adapter. Engine vibration working against a *rigid* pipe can cause drain boss failure in the oil pan in short time.

Connect the other end of the hose to the inlet of a small, electric motor-driven pump. Control the pump motor with a *key-operated switch*. Use a key-operated switch to prevent unauthorized operation of the pump.

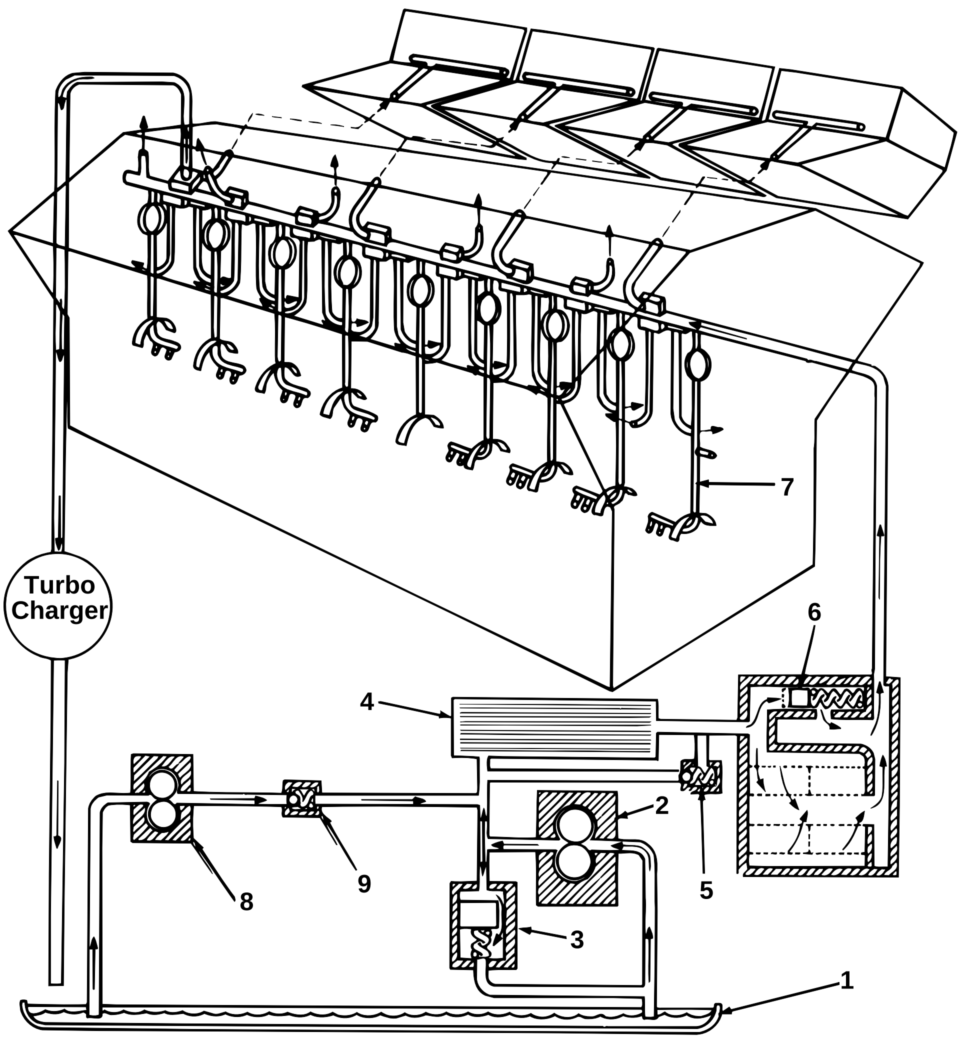
Connect the discharge of the motor-driven pump to a dirty oil tank for storage of the used lube oil until proper disposal is practical.

Keep the Key on the Captain's Key Ring.

Check oil level prior to every engine start.

* Adapter 6L6155 is available to convert the oil pan machine threads to 1/2-14 NPTF female threads for remote oil draining of the 3304B, 3306B, 3406B, and 3408B Marine Engines. Seal 8M4432 and washer 1S3889 are required to prevent leakage between the pan drain boss and the adapter.

Adapter 3N9442 and Gasket 3B1925 are available to convert from machine threads to 1/2-14 NPTF female threads on 3208 Marine Engines.



Engine Lube Oil Flow Schematic

1. **Sump – lube oil is drawn from the sump through a strainer into the inlet of the lube oil pump.**
2. **Lube Oil Pump – the quantity of lube oil delivered by the lube oil pump exceeds the engine's needs when the engine is new. As the engine clearances increase through normal wear, the flow required to properly lubricate the engine will remain adequate.**
3. **Oil Pressure Regulating Valve – this valve regulates oil pressure in the engine and routes excess oil back to the sump.**
4. **Lube Oil Cooler – the oil to the engine is cooled by jacket water in the engine oil cooler .**
5. **Oil Cooler Bypass Valve – when the viscosity of the oil causes a substantial pressure drop in the oil cooler, the bypass valve will open, causing the oil to bypass the cooler until the oil is warm enough to require full oil flow through the cooler.**
6. **Lube Oil Filter – Caterpillar lube oil filters are the full-flow type with a bypass valve to provide adequate lubrication should the filter become plugged. The filter system may have the replaceable element type or the spin-on type. The oil filter bypass valve is a protection against lube oil starvation if the oil filter clogs.**
7. **Engine Oil Passages – the main oil flow is distributed through passages to internal engine components. The oil flow carries away heat and wear particles and returns to the sump by gravity.**
8. **Prelubrication Pump – used only during starting cycle on largest engines.**
9. **Check Valve.**

Figure 1.1

Filter Change Technique

Spin-on oil filters are conveniently changed by gripping the loosened used oil filter with a plastic garbage bag. As the used oil filter is then removed, the oil which might have soiled the engine compartment can be caught by the garbage bag.

Lubricating Oil Heaters

Caterpillar does not recommend the use of immersion-type lubrication oil heaters due to their tendency to overheat the oil in contact with the heating element. This overheating causes deterioration and sludging of the lubricating oil and may lead to premature engine failure.

Emergency Systems

Some marine applications require the capability to connect an emergency lubricating oil pump into the engine's lube system.

This is a specific requirement of some marine classification societies for seagoing single propulsion engine applications. The purpose is to ensure lube oil pressure and circulation if the engine lube pump fails.

Requirements for emergency lube system operation:

1. Keep pressure drops to a minimum by using short, low restriction lines.
2. Use a line size at least as large as the engine connection point.
3. Install a low restriction strainer in front of the emergency oil pump.
4. Install a low restriction check valve between the emergency pump discharge and the engine inlet connection.
5. Use a pressure limiting valve in the emergency system set at 8.8 kg/cm^2 (125 psi).

Transmissions

Some marine classification societies require emergency lube oil pumps for marine transmissions to meet unrestricted service classification.

Duplex Filters

The optional Caterpillar Duplex Oil Filter System meets the requirements of the standard filter system plus an auxiliary filter system with the necessary valves and piping. The system provides the means for changing either the main or auxiliary filter elements with the engine running at any load to speed. A filter change indicator is included to tell when to change the main filter elements. A vent valve allows purging of air trapped in either the main or auxiliary system when installing new elements. *Air must be purged from the changed section to eliminate possible turbocharger and bearing damage.* **The auxiliary system is capable of providing adequate oil filtration for at least 100 hours under full load and speed operation.**

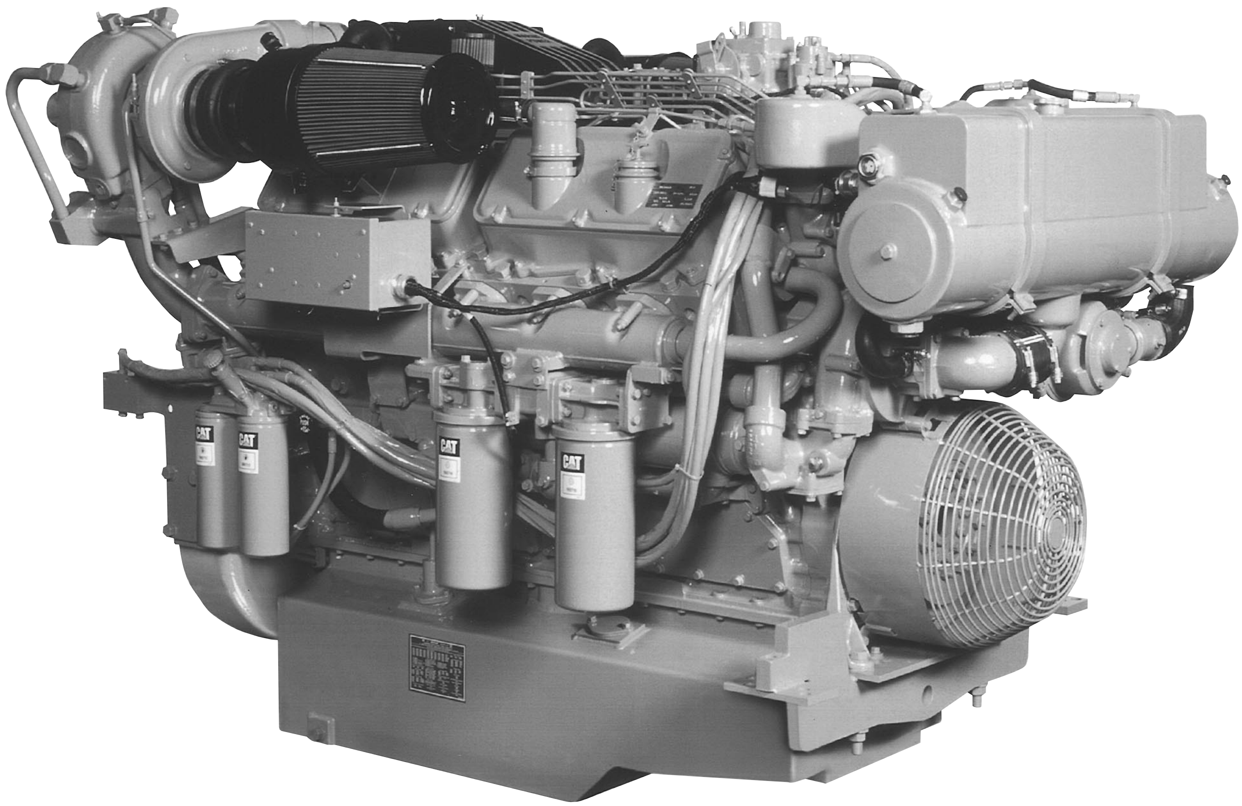


Figure 1.2

Auxiliary Oil Sump

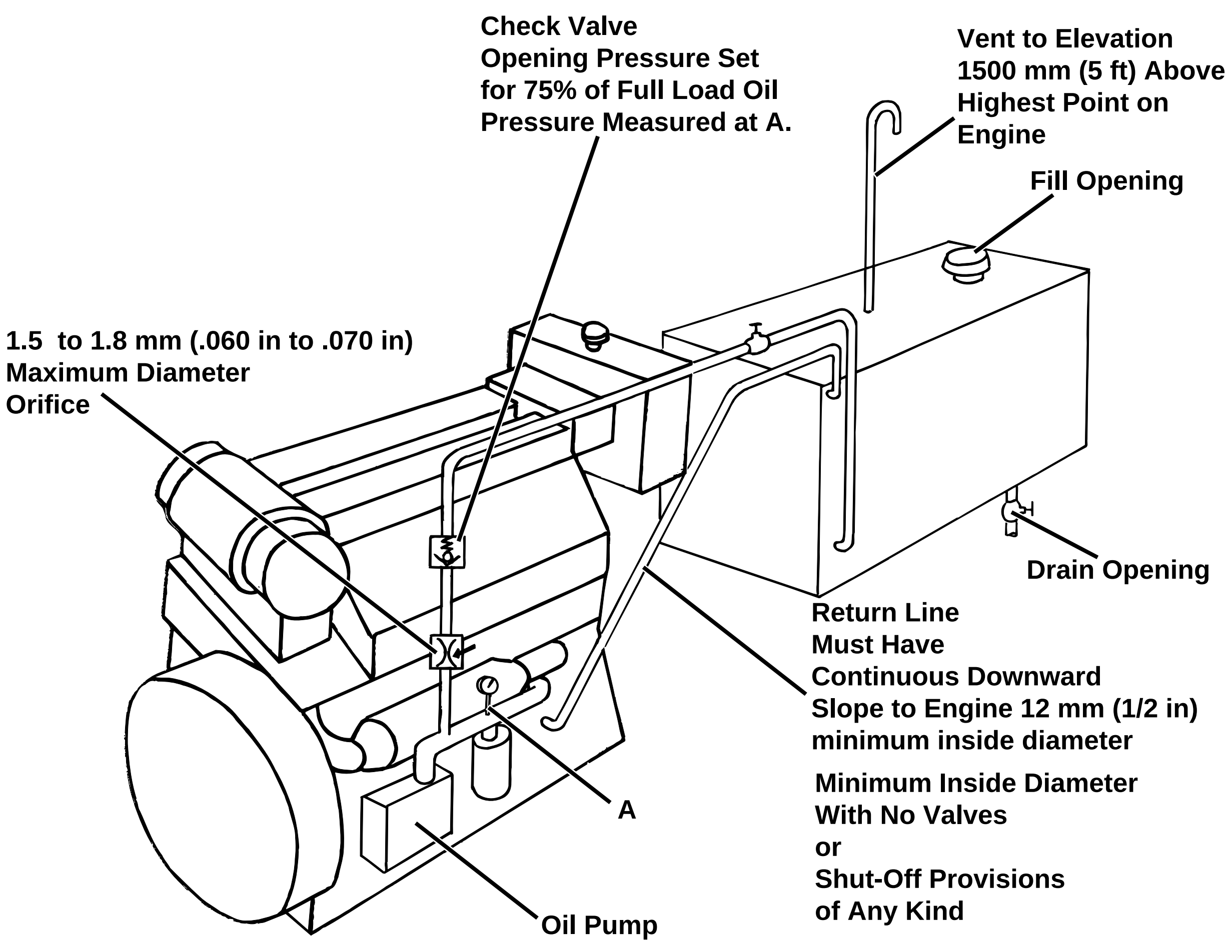
If longer oil change periods are desired, consider the auxiliary oil sump. Engine oil change period is directly proportional to total oil quantity, all other factors remaining equal. This is: to double oil change period, add an auxiliary oil sump with a capacity equal to the engine mounted oil sump. This will double the amount of oil available to be contaminated/diluted/neutralized and allow proportionately longer periods between oil changes. Previously mentioned considerations regarding fuel sulfur, oil TBN and oil analysis should be followed. After a basic change period has been confirmed, then an auxiliary oil sump may be used to increase the basic period-based on fuel quality, oil TBN and oil analysis.

Auxiliary Oil Sump System Considerations

1. Connect the oil source line to the auxiliary tank as close to the engine oil pump as possible. **The auxiliary oil sump tank must be full prior to starting the engine.** The auxiliary tank must remain full of oil at all times. As soon as the engine starts, the auxiliary oil sump will overflow, returning the oil to the engine; exactly compensating for the oil removed through the oil source line to the auxiliary tank.

2. Use a 1.5 to 1.8 mm (0.060 to 0.070 in.) orifice in this line to flow approximately 3.8 L/m (1 Gpm).

3. Put a check valve in the oil pump discharge line, set to open at 75% of the measured pressure at the line connection point, when the engine is up to temperature and at maximum operating speed.



AUXILIARY OIL SUMP CONNECTION SCHEMATIC

Figure 1.3

Prelubrication

A prelube system provides the capability to prelubricate all critical bearing journals before energizing the starting motors.

The *automatic* system utilizes a small pump which fills the engine oil galleries from the engine oil sump until the presence of oil is sensed at the upper portion of the lubrication system. The starter motors are automatically energized only after the engine has been prelubricated.

The *manual* system uses the engine's manually operated sump pump and allows the engine operator to fill all engine oil passages after oil changes, filter changes, periods of idleness, and before activating the starter motors.

Either prelube system will allow the engine operator to fill all engine oil passages after oil changes, filter changes, and before activating the starter motors. Either system will allow the engine user to minimize the sometimes severe engine wear associated with starting an engine after periods of idleness.

Special Marking of Engine Crankcase Dipstick

Sometimes marine engines are installed and operated in a tilted position. If the tilt angle is significant (5° or more) the amount of oil needed to fill the engine crankcase to the full mark on the dipstick (usually marked for level operation) may be more or less than the correct amount the oil pan was intended to accommodate without uncovering the suction bell or flooding the crankshaft seal.

The maximum safe tilt angle is dependent upon the design of the oil sump as well as the dipstick location—both of which are not necessarily uniform for all engine models. Therefore, where a tilted engine installation is encountered it is wise to check and, if necessary, remark the standard dipstick in order to make certain that the high and low marks will really reflect the proper amount of oil for safe engine operation. Oil pressure may be lost due to an uncovered suction bell, a flooded crankshaft seal may leak excessively, and engine vibration can be caused by crankshaft counterweights dipping into the oil. These are all problems that can be caused by an improper amount of oil in the sump.

Procedure

1. Drain engine crankcase and remove oil filter elements.

2. Install new oil filter elements.

3. Fill crankcase with a given volume of oil

(V_f) which can be determined as follows.

$V_f = V_r - V_m$

Where:

V_r = Volume of oil required to refill to *Full* mark with filter change.

V_m = Volume of oil between *add* mark and *Full* mark for level operation.

Note: Both Vr and Vm values for a specific engine model are published in the current Technical Marketing Information (TMI) on *Marine Engine Systems and Performance Specification Data* microfiche under subject heading *Pan-Oil Capacity*.

Add to this any additional oil volume required for special filters, oil lines, or coolers which are additions to the standard engine or unique to the installation.

4. Insert the dipstick to make certain that the oil shows on the dipstick. Be certain that the correct dipstick is used and that it does not hit the bottom of the sump or is otherwise improperly installed.

5. Start the engine and operate it at one half rated rpm until the oil has reached normal operating temperature. Reduce engine speed to low idle and mark the level indicated on the dipstick. This is the *add oil* or low mark.

6. Add additional oil equivalent to Vm as shown in the Technical Marketing Information (TMI) and let the engine operate at least another five minutes in order to bring all the oil up to temperature. Mark the new oil level on the dipstick. This is the full oil mark. Refer to Operation and Maintenance manuals for dipstick marking base on intalled tilt angles.

Synthetic Lubricants and Special Oil Formulations

Some producers of synthetic lubricants imply their products have properties which allow extended oil life.

Caterpillar Inc. neither endorses nor recommends a brand or type of *extended oil drain interval* crankcase oil for its engines.Caterpillar recommends (SOS) Schedules Oil Sampling to determine if extended oil drain periods can be achieved with synthetics. Caterpillar offers both petroleum and synthetic oils which are formulated for maximum wear conditions and long life because the additive package is on the high side of the tolerance range of CG4/CF4 specification.

Crankcase oil is changed because it becomes contaminated with soot (unburned carbon), wear products, partially burned fuel, acids, dirt, and products of combustion. The additive components included in the oil become depleted as they perform their intended functions of dispersing soot, preventing oxidation, wear, foaming, etc. Caterpillar requires petroleum and synthetic engine crankcase lubricants to meet Engine Service Designation CG4/CF4.

Types of Synthetic Oil

Two widely used synthetic oil types use base stocks made of synthetic hydrocarbons or di-basic acid testers. Both types of synthetic base oils have high viscosity indexes which make them advantageous in cold weather operations. Their use in any other application should be treated with caution.

The cost of these synthetic base oils range from three to four times the price of petroleum based lubricants, and makes the economics of their general use questionable.

Another type oil is called a partial synthetic engine oil. This is a petroleum base oil with some synthetic base oil which is blended for good cold weather performance.

Special Oil Formulations

Caterpillar does not recommend the use of additives to extend oil change periods. Oil additives such as graphite, teflon, molybdenum disulfide, etc., which have been properly blended into an oil that meets API CG4/CF4 specification can be used in Caterpillar Diesel Engines. These additives are not necessary to achieve normal life and performance of the engine.

Normal engine life and performance can be achieved by properly applying the engine, by servicing at recommended oil change period, by selecting the correct oil viscosity, by using a API CG4/CF4 oil and performing and maintenance as outlined in the engine operation and maintenance guide.

Caterpillar does not recommend the use of molybdenum dithiophosphate friction modifier additive in the engine oil. This additive causes rapid corrosion of bronze components in Caterpillar Diesel Engines.

Oil Publications Available From Caterpillar

The following publications are available through your local Caterpillar Dealer. Some of the publications may have a nominal charge. Some may be revised or discontinued in the future. These publications should be ordered directly from your dealer. Your dealer can also assist you in answering questions concerning available oils in your operating area.

All Engine Data Sheets are included in the Caterpillar Engine Technical Manual, Volume I, Form No. LEKQ2030.

Synthetic Lubricants and Special Oil Formulations LEKQ2051 (Engine Data Sheet 90.3)

Special Dipstick Marking of Engine Crankcase Dipstick LEKM3272 (Engine Data Sheet 94.0)

Oil Consumption Data LEKQ4028 (Engine Data Sheet 96.2)

Oil and Your Engine SEBD0640

Full Synthetic Diesel Engine Oil PEHP7062

Introduction of New Full Synthetic CAT Diesel Engine Oil SAE 5W-40 PELE0580

CATERPILLAR®

Fuel Systems

General Information

Caterpillar Engines have three different types of fuel systems.

The earliest has the high pressure pumps (for all the cylinders) in a single housing. The input shaft for that housing is driven by the engine's gear train. The high pressure fuel pump housing provides the high pressure fuel to the fuel valves at each cylinder, at the proper time, and in precisely metered amounts. The fuel valves at each cylinder are simple and easily replaced.

A later design combines each cylinder's high pressure pump and the fuel valve in a single unit; therefore, the term *unit injector*. The power to generate the high pressures for injection is taken from the engine's camshafts by way of pushrods and rocker arms.

Electronic unit injectors use engine camshaft and push rods to generate injection press but use electronics to time the fuel delivery and the amount.

Caterpillar Diesel Engine fuel delivery systems are designed to deliver more fuel to the engine than is required for combustion. The excess is returned to the fuel tanks.

Cleanliness

Clean fuel meeting Caterpillar's fuel recommendations provides outstanding engine service life and performance; anything less is a compromise and the risk is the user's responsibility. Dirty fuel and fuels not meeting Caterpillar's minimum specifications will adversely affect:

- The perceived performance of the combustion system and fuel filters.
- The service life of the fuel injection system, valves, pistons, rings, liners and bearings.

Heat in Fuel

Excess fuel (returned to the fuel tanks) picks up engine heat and can raise the temperature of the fuel in the tanks.

To avoid decreased injector life, fuel temperature to the engine must not exceed 66°C (150°F). Heat will also increase the specific volume of the fuel, resulting in a power loss of 1% for each 6°C (10°F) above 29°C (85°F) .

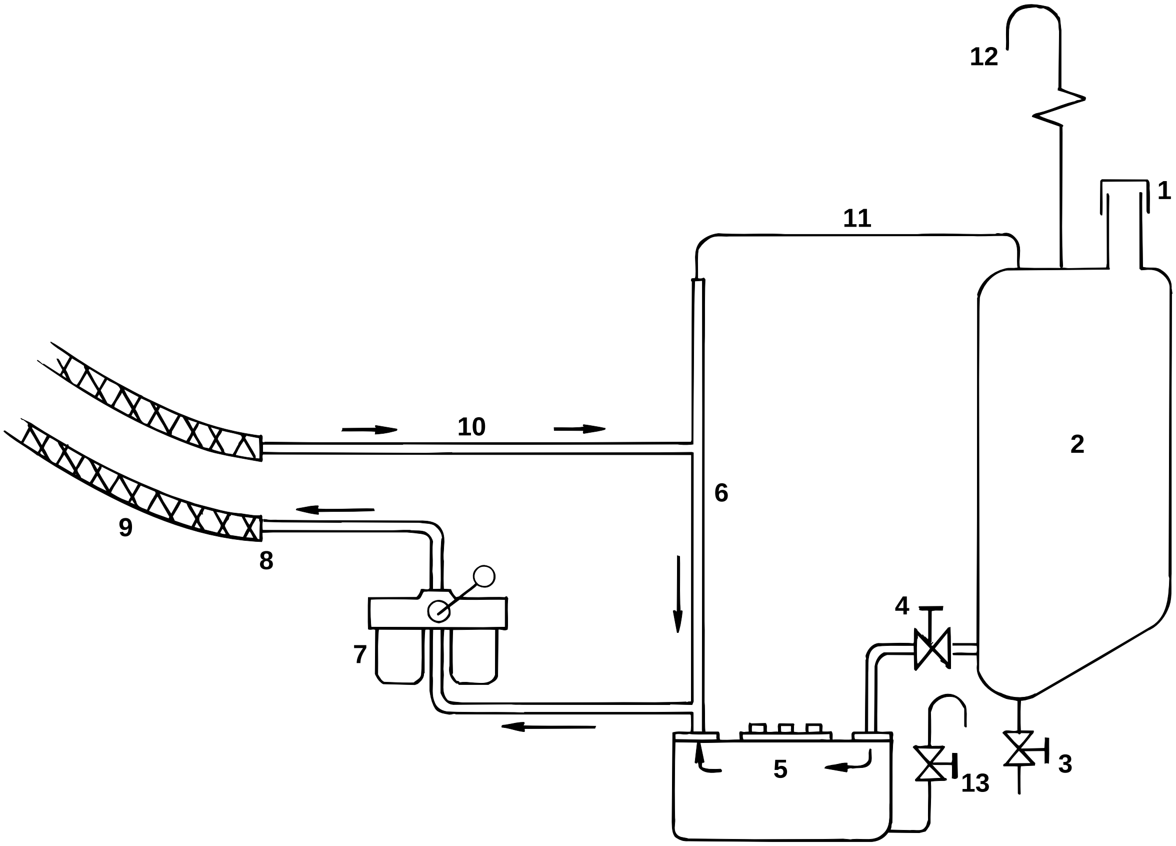
If the tank is so located and is of such size that the accumulated heat will not be objectionable when temperature stabilizes, then nothing more needs to be done. If the stabilized fuel tank temperature is high, the returning fuel should be cooled. See section on Fuel Coolers.

Air in Fuel

Gases entrained in the supplied fuel are discharged from the engine in the returned fuel. The gases (generally, air introduced through leaks in the fuel suction plumbing) must be vented to prevent engine power loss.

Standpipe Systems

The simplest method for eliminating the air problems is to install a standpipe between the fuel tank and the engine. Fuel will flow from the tank to the bottom of the standpipe by gravity. This is the point where the engine picks up fuel. The fuel return line must enter the standpipe at a point a few inches above the higher of either the supply or delivery point. The top of the standpipe can be vented into the top of a fuel tank or to atmosphere. This system works satisfactorily with any number of fuel tanks. There must be no upward loops in piping between the fuel tank and the standpipe, as entrapped air may block fuel flow.



FUEL SUPPLY SYSTEM-SINGLE TANK OR DAY TANK

1. Fuel filler
2. Fuel tank or day tank
3. Drain valve – install at lowest part of tank to enable draining of all water and sediment. Outlet of valve should be plugged when not in use to prevent fuel dripping.
4. Fuel discharge valve
5. Water and sediment trap – must be lowest point of system.
6. Fuel return standpipe
7. Primary fuel filter – to be cleanable without shutting down engine
8. Fuel supply line to engine
9. Flexible fuel lines connecting to basic fuel delivery system
10. Return from engine to standpipe
11. Vent from top of standpipe to top of fuel tank
12. Vent from top of fuel tank atmosphere-must be high enough above deck to prevent water washing over deck from entering pipe
13. Cleanout drain for water & sediment-valved and kept below fuel discharge from tank, to allow flushing water & sediment tank by gravity feed from supply

Figure 2.1

Fuel Coolers

The excess fuel returned from engines equipped with unit injectors (1.7 liter, 3500 and 3600 Family Engines) can absorb considerable heat from the injectors and the surrounding jacket water. Fuel coolers may be necessary for proper engine performance. The following factors affect the need for fuel cooling equipment:

- **Length of periods of continuous operation—**If the operating periods are short, the amount of heat returned to the fuel tanks will be relatively small. Fuel coolers are not generally required for engines used in high performance applications.

- **Length of time between periods of operation—**If the time between periods of operation is long, the heat will have an opportunity to dissipate.
- **Volume of the fuel tank—**If the volume of the fuel tank is large (larger than 11 000 L [3,000 gal]), it will accept a great deal of heat before the temperature of the fuel leaving the tank increases significantly.
- **Ability of the fuel tanks to dissipate the heat of stored fuel—**If the fuel in the tank is in contact with shell plating*, the fuel heat will be easily dissipated and stored fuel temperature will remain within a few degrees of the ambient water temperature.

Day Tanks (Auxiliary Fuel Tanks)

Auxiliary or day tanks are required if the main fuel tanks are located:

- More than 15.25 m (50 ft) from the engine;
- Are located above the engine;
- Or are more than 3.65 m (12 ft) below the engine.

Auxiliary or day tanks also provide a settling reservoir so air, water, and sediment can separate from the fuel.

The auxiliary or day tank should be located so that the level of the fuel is no higher than the fuel injection valves on the engine. If the fuel level is higher, the static pressure may allow fuel to leak into the combustion chambers when the engine is not running. *The presence of liquid fuel in the combustion chamber at the instant of engine starting is very likely to cause engine failure.* The tank should be close enough to the engine so the total suction lift is less than the 3.65 m (12 ft). The smaller this figure, the easier the engine will start.

* The shell plating should be approximately 10% of the inside surface area of the fuel tank.

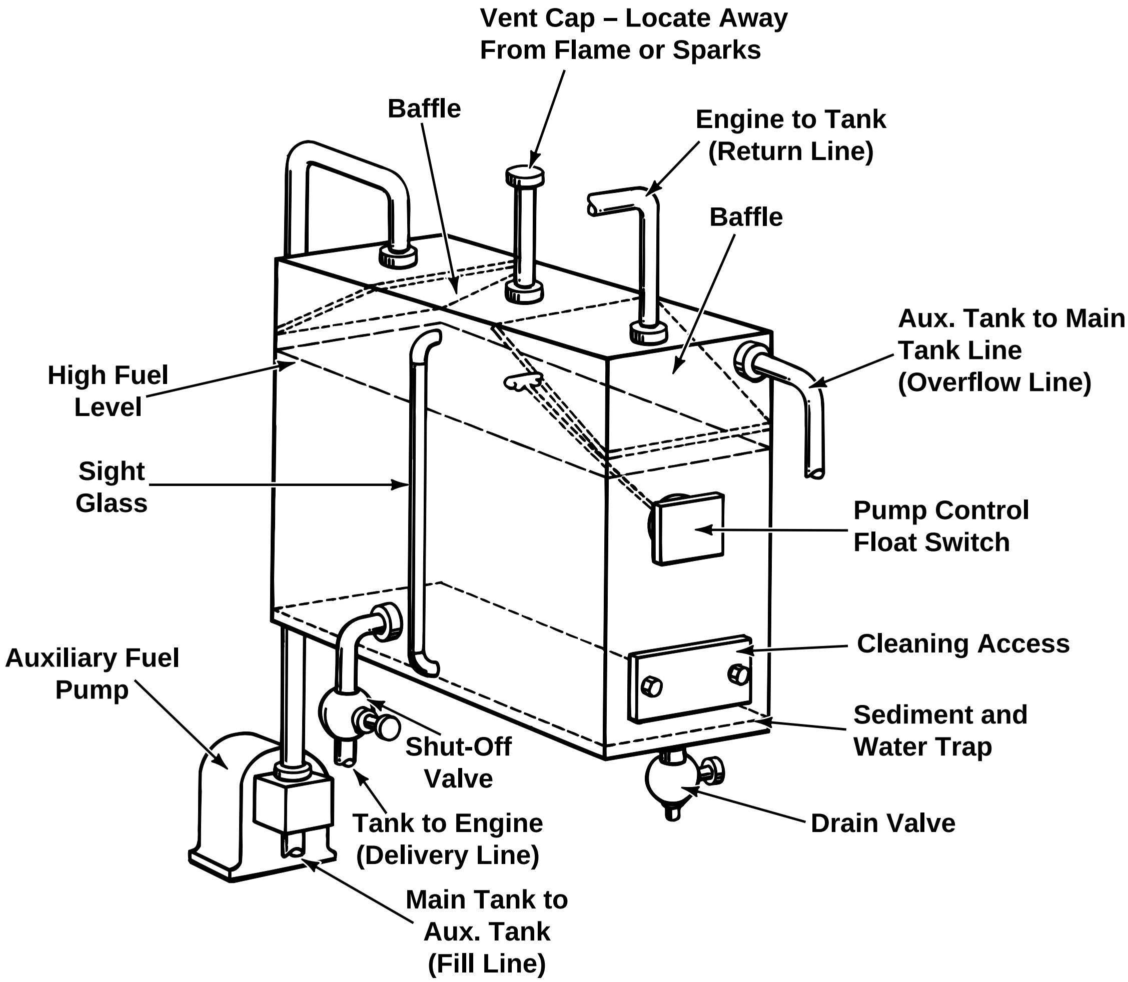


Figure 2.2

Fuel Return Line Pressure Limits

Engine fuel pressure measured in the fuel return line should be kept below 27 kPa (4 psi), except for the 3300 Engine Family, which is 20 kPa (3 psi) and the 3600 Engine Family, which is 350 kPa (51 psi). The fuel return line must be at least the same size as the supply line. A shutoff valve is not recommended.

Cleanliness

All connecting lines, valves, and tanks should be thoroughly cleaned before making final connections to the engine. The entire fuel supply system should be flushed prior to engine start-up.

Tank Design

Material

Fuel tanks are best made from low carbon rolled steel. Zinc, either in the form of plating or as a major alloying component, should not be used with diesel fuels. Zinc is unstable in the presence of sulfur, particularly if moisture is present in the fuel. The sludge formed by chemical action is extremely harmful to the engine's internal components. Zinc should be avoided where continuous contact with diesel fuel is involved.

Sizing

The capacity of a fuel tank or tank system can be estimated by multiplying the average horsepower demand by the hours of operation between refuelings, and divide the result by 16 for U.S. gallons and by 4 for liters.

This calculation does *not* allow for any reserve capacity which should be added to this basic requirement.

Grounding/Bonding (Electrical Connections)

The filler attachment and tank should be connected with a *ground cable* if they are not already connected electrically. The tanks should also be connected to the vessel's bonding system.

This is necessary to reduce the fire hazard of sparks discharged from static electricity buildup during refueling operations.

Drains

All fuel tanks should have easily accessible drain connections. Water and sediment which collects in the bottom of the tank must be eliminated regularly.

Provide clean-out openings for periodical removal of sediment and trash which settles out of fuel tanks.

Well-designed tanks have large enough clean-out openings so the lowest part of the fuel tank can be accessed with cleaning equipment.

Fuel Lines

Material

Black iron pipe is best suited for diesel fuel lines. Copper pipe or tubing may be substituted in sizes of 13.0 mm (0.5 in.) nominal pipe size or less. Valves and fittings may be cast iron or bronze (not brass). Zinc, either in the form of plating or as a major alloying component, should not be used with diesel fuels. Zinc is unstable in the presence of sulfur, particularly if moisture is present in the fuel. The sludge formed by chemical action is extremely harmful to the engine's internal components.

Routing

Whenever possible, route fuel lines under any machinery, so any leakage will be confined to the bilges. Leaks from overhead fuel system components may fall onto hot machinery, increasing the likelihood of fire danger.

Sizing

Determine the fuel line sizing by the supply and return line restriction. The maximum allowable restriction is published in the Engine Performance books (green books). Supply and return lines should be no smaller than the fittings on the engine.

Fuel Specifications

Caterpillar Diesel Engines have the capacity to burn a wide variety of fuels. See your Caterpillar Dealer for current information on fuel recommendations. In general, the engine can use the lowest priced distillate fuel which meets the following requirements:

Properties

Cetane Number or Index

The cetane index is a measure of the ignition quality of fuel which affects engine starting and acceleration.

The fuel supplier should know the cetane number or index of each fuel shipment.

Precombustion chamber fuel systems require a minimum cetane number of 35.

Direct injection engines require a minimum cetane number of 40 for good starting characteristics.

Engine Effects

Fuel with a low cetane number usually causes an ignition delay in the engine. This delay causes starting difficulties and engine knock. Ignition delay also causes poor fuel economy, a loss of power and sometimes engine damage. A low cetane number fuel can also cause white smoke and odor at start-up on colder days. Engines running on fuels with low cetane numbers may need to be started and stopped using a good distillate fuel.

Blended fuels or additives can change the cetane number. The cetane number is difficult and expensive to establish for blended fuels due to the complexity of the required test.

White exhaust smoke is made up of fuel vapors and aldehydes created by incomplete engine combustion. Ignition delay during cold weather is often the cause. There is not enough heat in the combustion chamber to ignite the fuel. Therefore, the fuel does not burn completely.

Using a cetane improver additive can often reduce white smoke during engine start-up in cold weather. It increases the cetane number of diesel fuel which improves ignition quality and makes it easier for fuel to ignite and burn. Contact your local fuel supplier for information on where to obtain cetane improvers.

The cetane number sensitivity can also be reduced in an engine by raising the inlet air temperature, if practical.

Cetane number is usually calculated or approximated using a *cetane index* due to the cost of more accurate testing. Be cautious when obtaining cetane numbers from fuel suppliers.

Flash Point

The flash point is the temperature at which fuel vapors can be ignited when exposed to a flame. It is determined by the type of fuel and the air-fuel ratio. It is important for safety reasons, not for engine operating characteristics.

The minimum flash point for most diesel fuels is about 38°C (100°F).

WARNING! For safety, maintain storage, settling and service fuel tanks at least 10°C (18°F) below the flash point of the fuel. Know the flash point of the fuel for safe storage and handling, especially if you are working with heavy fuels that need heating to a higher temperature to flow readily.

Cloud Point

The cloud point of a fuel is that temperature at which a cloud or haze appears in the fuel. This appearance is caused by the temperature falling below the melting point of waxes or paraffins that occur naturally in petroleum products.

Engine Effects

The cloud point of the fuel must be at least 6°C (10°F) below the lowest outside (ambient) temperature to prevent filters from plugging.

The fuel’s cloud and pour points are determined by the refiner. Generally, the cloud point is most important to you since it is at this temperature that fuel filter plugging begins to occur and stops fuel flow to the engine.

Steps to Overcome a High Cloud Point Temperature

Three steps can be taken to cope with high cloud point fuels.

1. Use a fuel heater when the outside temperature is below the cloud point of the fuel. Since the cloud point is also the wax melting point, when your fuel temperature is maintained above the cloud point, the wax will remain melted in the fuel. The heater should warm the fuel before it flows through the filter(s). Fuel heaters often use the engine coolant to heat the fuel and prevent wax particles from forming. Make sure the heater is capable of handling the maximum fuel flow of the engine. When the ambient temperature is low enough to require the use of a fuel heater, start and run the engine at low idle until the fuel temperature is high enough to prevent wax formation in the engine fuel filter circuit. Otherwise, high fuel rates with cold fuel will increase the risk of plugging.

Note: Do not allow the fuel to get too warm because fuel above 29°C (85°F) will affect the power output of the engine. Never exceed 66°C (150°F) with straight distillate fuel. The high fuel temperatures also affect the fuel viscosity. When the fuel viscosity falls below 1.4 cSt, pump damage may occur.

2. You can also dilute high cloud point fuels with a low cloud point fuel like kerosene.

3. The fuel manufacturer can also add flow improvers (wax crystal modifiers) to the fuel. These do not change the cloud point of the fuel, but they do keep the wax crystals small enough to pass through the fuel filter.

Caterpillar does not recommend the use of aftermarket fuel flow improvers because of occasional compatibility problems.

Pour Point
The pour point of a fuel is that temperature which is 3°C (5°F) above the temperature at which the fuel just fails to flow or turns solid. Usually the pour point is also determined by the wax or paraffin content of the fuel.

Steps to Overcome a High Pour Point Temperature
The pour point can be improved with flow improvers or the addition of kerosene. Fuel heaters cannot normally solve problems related to a high pour point temperature.

Viscosity
Viscosity is a measure of a liquid's resistance to flow. High viscosity means the fuel is thick and does not flow as easily. Fuel with the wrong viscosity (either too high or too low) can cause engine damage.

When comparing viscosity measurements, be sure they are taken at the same fuel temperature. Caterpillar recommends a viscosity between 1.4 cSt and 20 cSt delivered to the fuel injection pump. Engines with unit injectors can expect a 20°C (68°F) temperature rise between the transfer pump and the injector.

Engine Effects
High viscosity fuel will increase gear train, cam and follower wear on the fuel pump assembly because of the higher injection pressure. Fuel atomizes less efficiently and the engine will be more difficult to start.

Low viscosity fuel may not provide adequate lubrication to plungers, barrels, and injectors; its use should be evaluated carefully.

Steps to Correct Viscosity Problems:
The viscosity of fuel will vary with the fuel temperature.

Heating or cooling can be used to adjust viscosity somewhat.

Blending fuels is another way to adjust viscosity.

Viscosity and Heavy Fuel
The Caterpillar 3500 and 3600 Families of Engines can run on a blend of heavy and distillate fuels. Viscosity is a key factor. Heavy fuel must be diluted or heated until it reaches a viscosity of 20 cSt or less before it reaches the fuel system. Unless the engine has extremely low rpm, there is little economic benefit to trying to treat fuel with a higher viscosity than 380 cSt.

Steps to Correct Viscosity Problems
To handle high viscosity fuel, some additional installation requirements may be needed, depending on the exact viscosity. The installation may require:

- Fuel tank and fuel line heating.
- Centrifuging and back flush filtering.
- Externally driven fuel transfer pumps.
- Additional fuel filtering.
- Washing of the turbocharger exhaust turbine. (3600 Family Engines)

Specific Gravity
The specific gravity of diesel fuel is the weight of a fixed volume of fuel compared to the weight of the same volume of water (at the same temperature). The higher the specific gravity, the heavier the fuel. Heavier fuels have more energy or power (per volume) for the engine to use.

Effects on Engine

Light Fuels

When comparing fuel consumption or engine performance, always know the temperature of the fuel measurement for correct gravity and density.

- Lighter fuels like kerosene will not produce rated power.
- Do not adjust engine fuel settings to compensate for a power loss with lighter fuels (with a density number higher than 35 API). There is a likelihood of inaccuracy in the compensation process (if not done by authorized personnel) and the service life of a compensated engine might be seriously reduced if occasionally subjected to denser fuel.
- Fuel system component life can be decreased with very light fuels because lubrication will be less effective (due to low viscosity). Lighter fuels may also be a blend of ethanol or methanol with diesel fuel. Blending of alcohol (ethanol or methanol) or gasoline into a diesel fuel will create an explosive atmosphere in the fuel tank. In addition, water condensation in the tank can cause the alcohol to separate and stratify in the tank. Caterpillar recommends against such blends.

Heavy Fuels

A heavy fuel tends to create more combustion chamber deposit formations which can cause abnormal cylinder liner and ring wear.

Correct Specific Gravity

- Blending is the only way to correct fuel density problems.

Contaminants

Sulfur

Sulfur, in diesel fuel, is converted to sulfur trioxide during combustion. Sulfur trioxide will exhaust from the engine (without causing serious problems for the engine), as long as it does not come in contact with liquid water. If the sulfur trioxide gas does contact liquid water, the result is H_2SO_4 or sulfuric acid; a highly corrosive compound which will cause severe engine damage.

Engines should maintain jacket water temperatures above 74°C (165°F) at all times to minimize internal condensation of water vapor (from combustion).

Fuels containing higher sulfur levels can be utilized in Caterpillar Marine Engines. This does require proper lubrication oil selection. Consult the appropriate lubrication and maintenance manual, published by the Caterpillar Service Department, for specific recommendations.

Maintain the crankcase breather system to prevent condensation in the crankcase oil which will cause rapid TBN depletion.

Maintain a regular Scheduled Oil Sampling (S•O•S) oil analysis program. Infrared (JR) analysis is valuable as well.

Follow standard oil change intervals unless S•O•S or known sulfur content indicates differently.

Caterpillar Fuel Sulfur Analyzer

The Caterpillar 8T0910 Fuel Sulfur Analyzer will allow the vessel operator to immediately analyze fuel containing up to 1.5% sulfur. Caterpillar recommends checking each bulk fuel delivery, especially if fuel quality is questionable.

Vanadium

Vanadium is a metal present in some heavy fuels. It is impractical to remove or reduce this element at the refinery.

Vanadium compounds accelerate deposit formation.

Vanadium is not present in distillate fuels.

Engine Effects

Vanadium in the fuel quickly corrodes hot components. It will often first appear in the form of molten slag on exhaust valve seats.

Vanadium forms highly corrosive compounds during combustion. These compounds attach to hot metal surfaces, like exhaust valve faces, injector tips and turbocharger blades. Vanadium compounds melt and remove the oxide coating. When component temperatures rise, vanadium corrodes even faster. For example, exhaust valves can wear out in a few hundred hours when vanadium content in a fuel is high.

Steps to Help Prevent Vanadium Corrosion Damage

Vanadium compounds must reach their melting point to become active. The best corrosion control is to limit exhaust system component temperatures by controlling the temperature of the exhaust gas. Cooler exhaust gas temperatures can allow an engine to tolerate more vanadium in the fuel.

Some of the measures utilized to deal with high vanadium fuels include:

- Using special heat resistance materials.
- Rotating exhaust valves (standard on Caterpillar Engines).
- Engine derating to lower exhaust temperatures.
- Special cooling of high temperature parts.
- Blending the fuel with low vanadium fuel will reduce effects.

Water

Water can be introduced into the fuel during shipment or as a result of condensation during storage.

Engine Effects

Water (both fresh and salt) can cause:

- Excessive separator sludge after the fuel has been centrifuged.
- Piston ring groove deposits.
- Wear in fuel system plunger and barrel assemblies.
- Power loss from fuel starvation; the water causes fuel filter media to swell, cutting off the engine's fuel supply.

Steps to Overcome Effects of Water

- The effects of water in fuel can be minimized by draining water from the fuel tank daily.
- Obtaining fuel from reliable sources.
- Removal of salt water may require centrifuges.

Water Separators

There are two types of water separators.

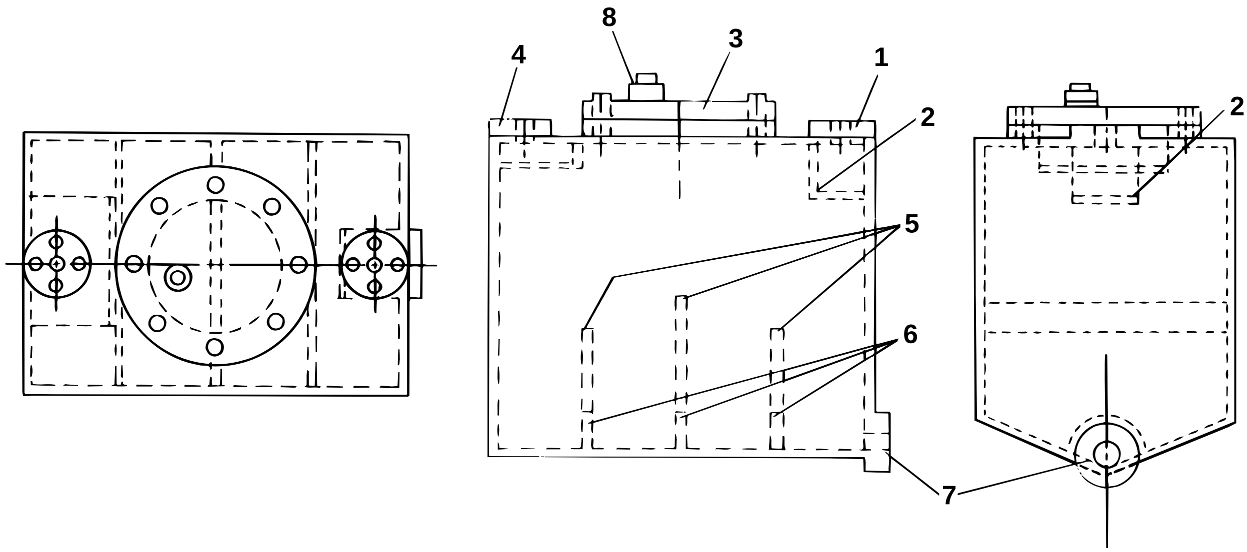
Sediment-Type Water Separator

The sediment type is installed ahead of the engine's fuel transfer pump. For water and sediment to separate properly, the sediment-type water separator should not be subject to violent motion.

A sediment water separator does not have a filtering media in the element. It does not normally need scheduled element replacement.

The water and sediment trap should be large enough to reduce the fuel flow rate to a velocity less than 0.61 m/s (2 ft/s). The larger particles of sediment and water will settle out at this flow rate.

Locate the water and sediment trap as close to the fuel tank as possible. This is to minimize the length of ship's fuel lines which are subject to water and sediment contamination. It will minimize any problem with water freezing in fuel lines.



SUGGESTED ARRANGEMENT FOR WATER AND SEDIMENT TRAP

1. **Fuel inlet**
2. **Inlet baffle**
3. **Hand hole and cover**
4. **Fuel Outlet**
5. **Baffles**
6. **Openings at bottom of baffles for drainage**
7. **Drain Opening**
8. **Air bleed plug**

$$\text{Sediment Trap Capacity} = \frac{\text{Fuel Cons.}}{25} + \frac{\text{Tank Cap.}}{5000}$$

Note: Suggested minimum size of trap

Figure 2.3

Coalescing Water Separator

The coalescing type of separator must be used if the water in the fuel is broken into such small particles that they make the fuel cloudy.

A coalescing-type separator will separate all water from fuel. It can be put anywhere in the fuel line, such as next to the components that need the most protection from water. The elements are composed of two-stage paper media that are replaceable. You can tell the element is plugged when there is a lack of fuel pressure.

Catalytic *Fines*

Catalytic *Fines* are small, hard particles which originate at the refinery. They are usually composed of aluminum and silicon particles and can cause very rapid abrasive wear.

Engine Effects

Catalytic *Fines* will severely damage injection pumps, injectors, piston rings and cylinder liners.

Proper fuel treatment methods (centrifuging and filtration) will remove these particles.

Volatile Fuel Components

Certain liquids are present in fuels in a vapor or gaseous state. This can cause an interruption of fuel supply to the fuel pump.

Lighter fuels and crude oils will have greater tendency to this problem than heavier fuels.

An air eliminator, or vapor trap, can be used to minimize the effect of interrupted fuel supply due to vapor lock.

The vapors and gases, thus separated are combustible and must be disposed of according to safe venting practice.

Simple venting to atmosphere is not adequate, since some of the vapors and gases may be heavier than air and tend to collect or *pool* in low spots, forming a severe safety hazard.

Asphaltenes

Asphaltenes are components of asphalt that are insoluble in petroleum naphtha and hot heptanes but are soluble in carbon disulphide and benzene. They are hard and brittle and are made up of long molecules with high molecular weight. In high concentrations, asphaltenes can cause filter plugging. They often contain heavy metals such as nickel, iron and vanadium. Asphaltenes are not present in distillate fuels.

Microorganisms in Fuel

All water and fuel offer a medium for microorganism growth. These simple life forms live in the water and feed on fuel.

Engine Effects

Microorganisms in fuel cause corrosion and filter plugging. They may be any color but are usually black, green or brown. They grow in long strings and have a slimy appearance. A biocide added to the fuel will kill the microorganisms but will not remove the remains of their bodies. Extensive filtering of the fuel after using the biocide is required to eliminate engine-mounted filter plugging.

Steps to minimize problems with bacterial growth:

- Avoid long fuel storage periods.
- Drain water from fuel tanks daily.
- Purchase fuel from reliable sources.
- Dose all fuel inventory with biocide at the first sign of microorganism contamination.

Air

Air can be dissolved in fuel, and it can also be pulled into the fuel lines by a leak on the suction side of the fuel transfer pump.

Engine Effects

Air in the fuel will cause starting problems, *missing,* low power and smoke problems. Air can also cause excessive white smoke in some engines.

Reducing the effects of air in the fuel system.

Remove air by bleeding the fuel system. Check for dissolved air in fuel with a 2P8278 Fuel Flow Tube. Correct suction piping leaks.

Filters

Primary Fuel Filter Element Specification

The primary fuel filters elements should have the following properties:

Mesh Size—	32 x 28 strands per cm (70 x 80 strands per in.)
Element—	Monel wire cloth material or equivalent
Element Area—	645 cm^2 (100 $in.^2$) or greater
Opening Size—	0.1778 mm x 0.2235 mm (0.007 in. x 0.0088 in.)

Duplex Fuel Filters

Many Caterpillar Engines can be equipped with duplex fuel filters. These filters may be serviced (change elements), without shutting off the engine. There are two types: the *symmetrical* type—which has two identical filter sets and the *main-auxiliary* type—which has a *main* filter set and a smaller capacity *auxiliary* filter set. A special valve connects the two sets of filters in each type. The valve routes the fuel to be filtered through either or both sets of filters.

Both filter sets can be used simultaneously to extend running time in an emergency.

Filter Micron Ratings

Caterpillar does not specify filter or filter paper by micron rating.

Caterpillar specifies actual filter capability, rupture strength, the capacity for holding dirt, flow resistance, filter area, etc.

Micron ratings are easily confused for the following reasons:

- The test for micron ratings is not repeatable at different labs. One manufacturer may give a rating of 10 microns (0.00039 in.), another at 2 microns (0.000079 in.), and a third may rate a particular filter media (paper) at 15 microns (0.00059 in.).

- There is no consistent relationship between micron rating and actual filtration efficiency. The entire filter needs to be tested, not just the media (paper).

- The micron rating does not show what happens to a filter over time. The test provides no information about how a filter will stand up under continual use.

Micron ratings are overemphasized; a 10 micron filter will not always stop a 10 micron particle. Many reputable filter manufacturing firms are drifting away from micron ratings to more conclusive tests.

Smaller micron ratings are not necessarily better.

If all other factors (area) were equal, a smaller micron number media (paper) has a severe drawback: less capacity before plugging, needs to be replaced more often. The size of the pores in the paper needs to be balanced against the costs of the filter replacement.

Common questions are:

- What is the maximum particle size which can pass through Caterpillar filters?
- What is the difference between nominal size and absolute size filters?

For example: A nominal 10 micron filter media (paper) will pass some particles up to about 50 microns in size. Theoretically, an absolute rating of 10 microns will stop all particles larger than 10 microns. In fact, filters with absolute micron ratings of 10 will pass some particles larger than 10 microns due to the irregularity of the paper weave. New filters may pass larger particles than they will after only a few hours of use.

As a general rule, Caterpillar fuel filter media (paper) is about 3 microns nominal, 20 microns absolute. Oil filter media (paper) is about 10 microns nominal, 50 microns absolute. These are approximate values only.

Filters are not effectively compared on the basis of micron rating alone. Evaluate filters on the basis of their ability to collect foreign material as a whole.

Fuel System — Miscellaneous

Disposal of Used Lube Oil

It is necessary to collect, store, and dispose of used crankcase oil from engines correctly. It is not acceptable to dump used crankcase oil into the oceans, rivers and harbors from vessels or offshore drilling and production platform installations. It may be necessary for engine operators to consider burning crankcase oil in their Cat Engines. This can be done, providing the precautions below are carefully followed:

- Only diesel engine crankcase oils can be mixed with the diesel engine fuel supply. The ratio of used oil to fuel must not exceed 5%. Premature filter plugging will occur at higher ratios. *Under no circumstances* should gasoline engine crankcase oil, transmission oils, hydraulic oils, grease, cleaning solvents, etc., be mixed with the diesel fuel. Also, do not use crankcase oils containing water or antifreeze.

- Adequate mixing is essential. Lube oil and fuel oil, once mixed, will combine and not separate. Mix used filtered crankcase oil with an equal amount of fuel, then add the 50-50 blend to the supply tank before new fuel is added (maintaining the 5% used oil to fuel ratio). This procedure should normally provide sufficient mixing. Failure to achieve adequate mixing will result in premature filter plugging by *slugs* of undiluted oil.

- Filter or centrifuge used oil before putting it in the fuel tanks to prevent premature fuel filter plugging, accelerated wear or plugging of fuel system parts. Soot, dirt, metal and residue particles larger than 5 microns (0.000197 in.) should be removed by this process.

  If filtering or centrifuging is not used before adding the oil to the fuel, primary filters with 5 microns (0.000197 in.) capability must be located between the fuel supply and engine. These will require frequent servicing.

- Clean handling techniques of the used crankcase oils are essential to prevent introducing contaminants from outside sources into the diesel fuel supply. Care must be taken in collecting, storing and transporting the used crankcase oil to the diesel fuel tanks.

  Diesel fuel day tank sight glasses may become blackened in time due to the carbon content in the crankcase oil. Ash content of the lube oil added to the fuel may also cause accumulation of turbocharger and valve deposits more rapidly than normal.

Corrosion

Copper Strip Corrosion

Corrosion is commonly tested by examining the discoloration formed on a polished copper strip when immersed in fuel for three hours at 100°C (212°F). Any fuel showing more than slight discoloration should be rejected.

Many types of engine parts are of copper or copper alloys. It is essential that any fuel in contact with these parts be noncorrosive to copper. There are certain sulfur derivatives in the fuel that are likely sources of corrosion.

Sodium or Sodium Chloride (Salt)

Sodium is an alkaline, metallic element. It is very active chemically. Sodium's most common form is table salt.

Sodium is frequently introduced during storage or because of incorrect handling procedures. Sodium can come directly from sea water or salt air condensation in fuel tanks. It can also be present in crude oil in its natural state.

Engine Effects

Sodium acts as a catalyst for vanadium corrosion. When sodium and vanadium combine, they react to form compounds which melt within normal engine operating temperatures.

The sodium/vanadium combination causes high temperature corrosion of exhaust valves. It can also cause turbocharger turbine and nozzle deposits.

Steps to Reduce the Effects of Sodium

Fuel can be blended to reduce the concentration of sodium.

Fuel contaminated with sodium can be *washed* by blending fresh water with the contaminated fuel in one centrifuge and separating the two (with the sodium now dissolved in the added fresh water) in a second centrifuge.

Handle and store fuel in a manner which minimizes the exposure to salt water and salt water laden air.

Crude Oils

Description

Crude oil is used to describe unrefined oils/fuels. Crude oil is basically the same as it was when pumped from the ground. Certain types of crude oils can be burned in Caterpillar Engines. See the Crude Oil Chart (Limits of Acceptability for Use in Caterpillar Engines) in the Fuel Section Appendix.

Heavy/Blended/Residual Fuels

Description

Heavy/Blended/Residual Fuel is composed of the remaining elements from crude oil after the oil has been refined into diesel fuel, gasoline or lubricating oils, etc. After the more desirable products have been refined, the remaining elements (which resemble tar and contain abrasive and corrosive substances) can be combined or diluted with a lighter fuel *(cutter* stock) so they can flow. These are called blended, heavy or residual fuels.

Caterpillar 3500 and 3600 Family Engines can be modified to run on fuels which meet the specifications in the Heavy/Blended/ Residual Fuel Chart in the Fuel Section Appendix.

Caterpillar 3500 and 3600 Diesel Engines can atomize many heavy fuels because of the unit injector fuel system. This system does not have high pressure fuel lines and can withstand higher injection pressures. 3500 Family Engines are capable of operating on blended fuels up to 180 cSt at 50°C at 1800 rpm and lower, usually without changing engine timing. However, engine aerating may be required to keep the exhaust temperature below maximum limits. 3600 Family Engines are capable of operating on blended fuels up to 380 cSt at 50°C.

There are many other considerations to keep in mind when making the decision to switch to heavy fuel. Because heavy fuel is the heavy residue which is left over from the refining process, it has concentrated contaminants. In the best situation, using heavy fuel will increase the workload of the operating personnel. In the worst situation, heavy fuel could cause extremely short engine and component life. For your engine to operate successfully on heavy fuels, you must have a thorough maintenance program and high quality fuel treatment equipment.

It is recommended that you always consult with your local Caterpillar Dealer when considering fuel changes.

The Economics of Using Heavy Fuel

Lower fuel costs make heavy fuel appear to be more economical. Blended fuels can lower costs for some customers, but there are often significant tradeoffs. Fuel price must be compared to fuel contaminants, effects of reduced engine component life, higher maintenance and personnel costs. Conduct a thorough analysis of all the costs involved before you decide to use heavy fuel.

Caterpillar or your Caterpillar Dealer will aid you in this evaluation.

Also, investigate other fuel-saving methods. The following is a list of some fuel-saving alternatives:

- More modern, fuel-efficient engines.
- Lower speed (Engines can operate at 1200 rpm instead of 1800 rpm; 1000 rpm, instead of 1500 rpm; etc.).
- More efficient propeller (larger diameter with reduced pitch) or more efficient generator or other driven unit.
- Waste heat recovery.
- Lighter blends.
- Crude oil instead of diesel fuel.

Installation Costs Associated With Using Heavy Fuel

Installation costs for an engine using heavy fuel may range from 25-85% more than an engine using No. 2 diesel fuel or marine diesel fuel. Other costs result from the need for fuel treatment equipment.

Downtime is also typically increased. Operators must spend more time taking care of engine and fuel handling equipment. They must understand the system and have training on the engine as well as on the actual fuel preparation equipment.

How Your Caterpillar Warranty Applies to Using Heavy Fuel

When you decide to use heavy fuel, you are making an economic tradeoff. Though your fuel costs may be 5-40% lower when using blended fuels, this savings does not come free. Because of contaminants, fuel injector, valve and piston ring life could be significantly shorter. These worn components may have to be replaced during the warranty period, but they are not covered by Caterpillar.

Caterpillar does not offer a warranty on replacement of parts which have a shortened service life because of the use of heavy fuel. The Caterpillar warranty which applies to your engine is available from your dealer.

Fuel Blending

Many fuel characteristics can be tailored by blending different fuels. A blended fuel can help improve engine starting and warm-up, reduce deposits and wear, improve emissions and sometimes have an effect on power and economy.

In general, lighter fuels are cleaner and help engine starting. Heavier fuels have higher heating values (per volume), better cetane quality, etc.

The 3600 and 3500 Family Engines can use blended fuel economically as long as the fuel treatment facilities are adequate, and there are trained personnel to run this equipment.

Blended Fuel Should Be Analyzed

Chemical labs can evaluate fuel properties. Some oil companies and regulating agencies also provide fuel analysis services.

Fuel System Maintenance

Filter Maintenance

First clean around the filter housing, then unscrew or otherwise remove the old filter(s) without introducing dirt into the housing.

Lubricate and clean the new filter gasket with *clean* diesel fuel.

Install the new filters *dry*.

Prime the fuel system.

Never pour fuel into the new filter element before you install it. Contaminated fuel will cause fuel system damage.

Always bleed the fuel system to remove air bubbles after changing the fuel filters and before starting the engine.

Check the fuel pressure differential which can indicate a restricted or plugged fuel filter.

Inspect all new filters (especially check the threads on spin-on filters) for debris or metal filings. Any filings already in the filter will go directly to the fuel pumps and injectors.

Use genuine Caterpillar fuel filters to ensure quality, consistency and cleanliness. There are great differences in fuel filters. Even if the filter fits your engine, it might not be the correct filter. There are a lot of important differences between Caterpillar filters and nongenuine filters. For more information on fuel filter differences and considerations, see your Caterpillar Dealer.

Properly store new filters to prevent dust from direct entry into the filter before use.

Cut apart used filters after every filter change. A way to thoroughly inspect filters is to use the 6V7905 Filter Cutting Tool to cut them apart after they have been used (every filter change period). This will allow you to inspect internal filter components, see contaminants, and to also compare brands of filters for quality and filtering effectiveness.

Storage Tank Maintenance

Fill the fuel tank after each day of operation to minimize condensation of water. A full fuel tank helps prevent condensation by driving out moisture laden air. However, don't fill the tank too full; if the temperature increases, the fuel will expand and may overflow.

Drain water and sediment from the fuel tank at the start of every shift or after the tank has been filled and allowed to stand for 5-10 minutes. Be sure to drain a cupful at the start of every shift for inspection. Drain storage tanks every week.

Install and maintain a water separator before the primary fuel filter.

As Needed Periodic Activities

Test fuel as it is delivered. Identify contaminant levels immediately and notify appropriate operations personnel.

Before storage, test for compatibility between fuel in the tanks and the fuel being purchased. Keep the fuel in separate tanks, if possible.

Use regular S•O•S oil analysis to determine if there are wear particles in the oil and maintain the proper Total Base Number (TBN) level.

Request infrared analysis on used oil to determine the effects of burning heavy fuel on the crankcase oil.

Daily Activities

Maintain and monitor fuel treatment equipment.

Record engine temperatures to assure adequate jacket water temperature, aftercooler temperature and air intake temperature.

Check exhaust thermocouples and record exhaust temperatures. Be alert for worn exhaust valves.

Note: Measure valve stem projection when new; use a stationary point such as the valve cover gasket surface for a reference point. Record the measurements for each valve for later follow-up measurements. If valve *stem* projection moves more than 1.25 mm (0.050 in.), consider disassembly to find the reason. Another way to observe valve face wear is to measure and record changes on valve lash over a period of time.

Fuel Publications Available From Caterpillar

The following publications are available through your local Caterpillar Dealer. Some of the publications may have a nominal charge. Some may be revised or discontinued in the future. These publications should be ordered directly from your dealer. Your dealer can also assist you in answering questions concerning available fuels in your operating area.

All Engine Data Sheets are included in the Caterpillar *Engine Technical Manual*, Volume I, Form No. LEKQ2030.

Mixing Used Crankcase Oil With Diesel Fuel LEKQ6070 (Engine Data Sheet 62.0)

Fuel Recommendations for Caterpillar Diesel Engines LEKQ4219 (Engine Data Sheet 60.1)

Alcohol Fuels for Caterpillar Diesel Engines LEKQ0287 (Engine Data Sheet 61.2)

Fuel Heaters for Cold Weather Operation LEKQ4065 (Engine Data Sheet 64.5 for No. 1 and No. 2 Diesel Fuel Only)

Installation of 8N9754 Fuel Heater Group SEHS7653-02 (Special Instruction)

Fight Fuel Sulfur, Your Diesel's Silent Enemy SEBD0598

Analyzing Fuel Nozzle and Fuel Line Failures SEBD0639

Using Diesel Fuel Thermo-Hydrometers GMGO0977 (Special Instruction)

Using 2P8278 Fuel Flow Tube to Check for Entrained Air in Diesel Fuel GMGO0825 (Special Instruction)

Heavy Fuel Contaminant Levels for 3500 and 3600 Engines LEKQ2314 (Engine Data Sheet 61.1)

Sizing Fuel System Components for Heavy Fuels LEKQ9173 (Engine Data Sheet 61.3)

Heavy Fuel Operating Procedures for 3500 and 3600 Engines LEKQ1177 (Engine Data Sheet 61.4)

Fuel Water Separator for Use With 3208 and 3300 Engines Equipped With Sleeve Metering Fuel System LEKQ3383 (Engine Data Sheet 64.1)

Fuel Conservation Practices LEKQ3106 (Engine Data Sheet 60.2)

Other Publications

ABS Notes on Heavy Fuel Oil (1984)
American Bureau of Shipping
45 Eisenhower Drive Paramus, NJ 07652
U.S.A. Telephone: (201) 368-9100
Attention: Book Order Department

Appendix

Table of Specific Gravity Versus Density			
Gravity		**Density**	
Degrees API @ 15°C (60°F)	**Specific Gravity @ 15°C (60°F)**	**kg/L**	**lb/gal**
25	.9042	.902	7.592
26	.8984	.897	7.481
27	.8927	.891	7.434
28	.8871	.886	7.387
29	.8816	.880	7.341
30	.8762	.874	7.296
31	.8708	.869	7.251
32	.8654	.864	7.206
33	.8602	.858	7.163
34	.8550	.853	7.119
35	.8498	.848	7.076
36	.8448	.843	7.034
37	.8398	.838	6.993
38	.8348	.833	6.951
39	.8299	.828	6.910
40	.8251	.823	6.870
41	.8203	.819	6.830
42	.8155	.814	6.790
43	.8109	.809	6.752
44	.8063	.804	6.713
45	.8017	.800	6.675
46	.7972	.795	6.637
47	.7927	.791	6.600
48	.7883	.787	6.563
49	.7839	.782	6.526

Crude Oil Chart		
Fuel Properties and Characteristics	**Permissible Fuels as Delivered to the Fuel System**	
Cetane number or cetane index (ASTM D613 or calculated index) (PC Engines)	Minimum	35
(DI Engines)	Minimum	40
Water and sediment % volume (ASTM D1796)	Maximum	.5%
Pour Point (ASTM D97)	Minimum	6°C (10°F) below ambient temperature
Cloud point (ASTM D97)		Not higher than ambient temperature
Sulfur (ASTM D2788 or D3605 or D1552)	Maximum	.5% — See page 15 to adjust oil TBN for higher sulfur content
Viscosity at 38°C (100°F)	Minimum	1.4 cSt
(ASTM D445)	Maximum	20 cSt
API gravity (ASTM D287)	Maximum Minimum	45 30
Specific gravity (ASTM D287)	Minimum Maximum	.8017 .875
Gasoline and naphtha fraction (fractions boiled off below 200°C)	Maximum	35%
Kerosene and distillate fraction (fractions boiled off between 200°C and cracking point)	Minimum	30%
Carbon residue (ramsbottom) (ASTM D524)	Maximum	3.5%
Distillation — 10%	Maximum	282°C (540°F)
— 90%	Maximum	380°C (716°F)
— cracking %	Minimum	60%
— residue (ASTM D86, D158 or D285)	Maximum	10%
Reid vapor pressure (ASTM D323)	Maximum	20 psi (kPa)
Salt (ASTM D3230)	Maximum	100 lb/1,000 barrels
Gums and Resins (ASTM D381)	Maximum	10 mg/100 mL
Copper strip corrosion 3 hrs @ 100°C (ASTM D130)	Maximum	No. 3
Flashpoint °C°F (ASTM D93)	Maximum	Must be legal limit
Ash % wieght (ASTM D482)	Maximum	.1%
Aromatics % (ASTM D1319)	Maximum	35%
Vanadium PPM (ASTM D2788 or D3605)	Maximum	4 PPM
Sodium PPM(ASTM D2788 or D3605)	Maximum	10 PPM
Nickel PPM (ASTM D2788 or D3605)	Maximum	1 PPM
Aluminum PPM (ASTM D2788 or D3605)	Maximum	1 PPM
Silicon (ASTM D2788 or D3605)	Maximum	1 PPM

PPM = parts per million

Heavy/Blended/Residual Fuel Chart			
Fuel Properties and Characteristics	**Permissible Fuels as Delivered to the Fuel System**		
Water and sediment percent volume (ASTM D1796)	Maximum	3500	3600
		.5	.5
Sulfur (ASTM D2788 or D3605 or D1552)	Maximum	4%	5%
Viscosity	Minimum	1.4 cSt	1.4 cSt
(To the Unit Injector)(ASTM D445)	Maximum	180 cSt @ 50°C	380 cSt @ 50°C
Carbon Residue (Conradson Carbon Residue) (ASTM D189)	Maximum	15	18
Vanadium	Maximum (PPM)	250	300
Aluminum (ASTM D2788 or D3605)	Maximum (PPM)	1	2
Silicon (ASTM D2788 or D3605)	Maximum (PPM)	1	2

PPM = parts per million

Heavy/Blended/Residual Fuel Viscosity Chart	
Viscosity (cSt @ 50°C)	**Viscosity (Redwood Seconds @ 100°F)**
30	200
40	278
60	439
80	610
100	780
120	950
150	1250
180	1500
240	2400
280	2500

CATERPILLAR®

Marine Engines Application and Installation Guide

- Dredge Engines

LEKM7148-01 (Supersedes LEKM7148) (05-00)

CATERPILLAR®

Dredge Engines

Definitions

Pump Engine Considerations

Engine Installation

Basically, dredging is the removal of material from under water, and its disposal elsewhere. It includes two distinct operations: first, excavating the material, and second, transporting it to a disposal area. There are two ways of doing this— mechanically and hydraulically.

Definitions

Mechanical Dredges

Mechanical dredges were the first to be developed. Today three basic types are used:

1. Grapple Dredge
2. Dipper Dredge
3. Bucket Dredge

The Grapple Dredge

The grapple dredge is essentially a derrick mounted on a barge and equipped with a clamshell bucket for dredging. It is most suitable for excavating soft and cohesive materials.

This type of dredge does not give the best results in very soft deposits where the material is likely to be washed out of the bucket or in very hard materials where the penetration is not sufficient to fill the bucket. Grapple dredges have the advantage of being able to work in confined areas near docks and breakwaters.

The Dipper Dredge

The dipper dredge is essentially a barge-mounted power shovel. Its main advantage is in the strong crowding action of the bucket as the dipper stick forces it into the material to be moved. Its best use today is for excavating hard compact materials, rock and other solid formations after blasting. For its size, a dipper dredge can handle larger pieces, thus reducing the amount of blasting. For most other work it has been replaced by more efficient, faster working hydraulic dredges.

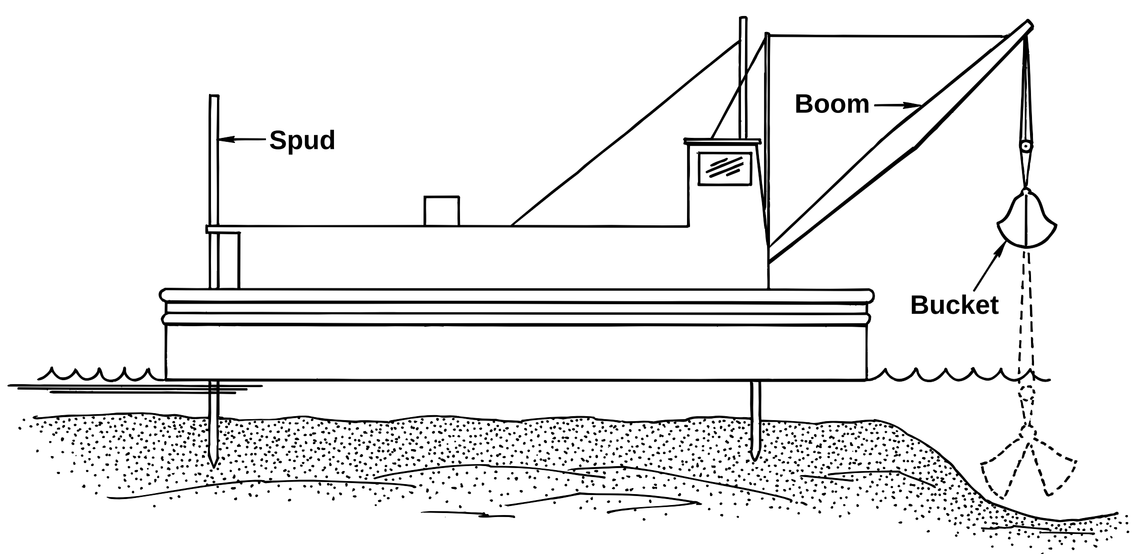


Figure 1.1

The Bucket Dredge

The bucket dredge consists of an endless chain of buckets moving from the work face to a point above the surface of the water. Each bucket digs its own load, carries it to the surface and, as it rotates over the top tumbler, dumps its load and goes back for another. Bucket dredges are more efficient than dipper or grapple dredges because the work cycle is continuous. Dredges of this type have found wide use in commercial production of sand and gravel and in the the recovery of various ores and precious metals such as tin and gold.

All three types of mechanical dredges have their advantages; however, each fulfills only one part of the two-phase dredging operation of excavation and disposal. Mechanical dredges remove material, but to dispose requires a fleet of barges and tugs to move the material to its disposal point. Hydraulic dredges handle both phases of the dredging process.

Hydraulic Dredges

Unlike the mechanical dredges, hydraulic dredges use the water on which they float to make dredging more efficient. A hydraulic dredge mixes the material to be removed with water and pumps it as a fluid. Hydraulic dredges are usually more versatile, efficient and economical to operate than mechanical dredges because the digging and disposing operation is performed by one self- contained unit.

The Plain Suction Dredge

The plain suction dredge consists of a dredge pump which draws in a mixture of water and excavated material through the suction pipe lowered to the working face of the deposit. The mixture is discharged through a pipeline to the spoil area or into barges or hoppers. The use of units of this type is limited to digging soft and free-flowing materials, such as clay, sand, silt, or gravel.

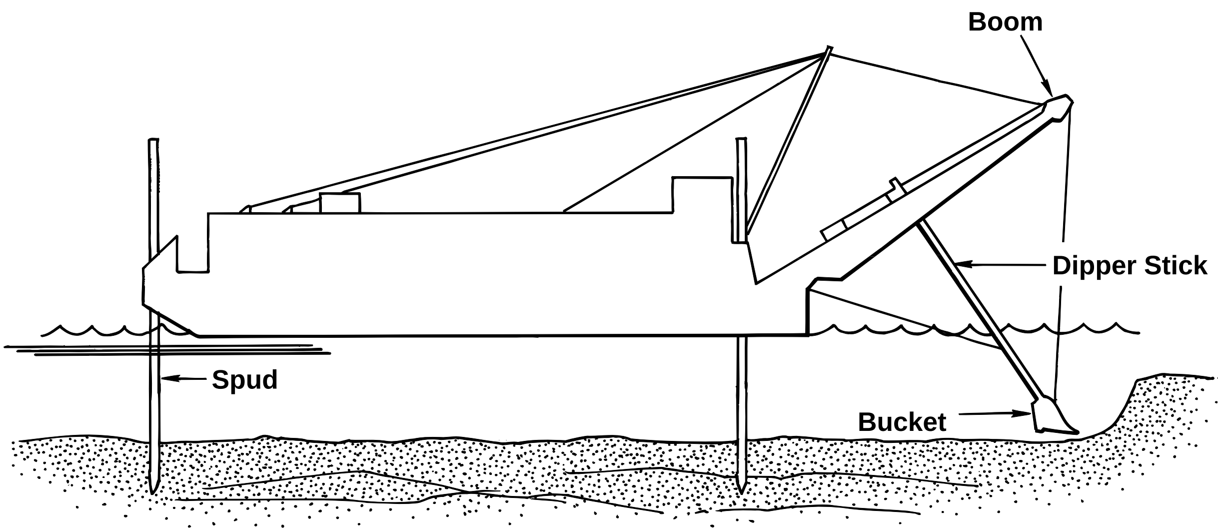


Figure 1.2

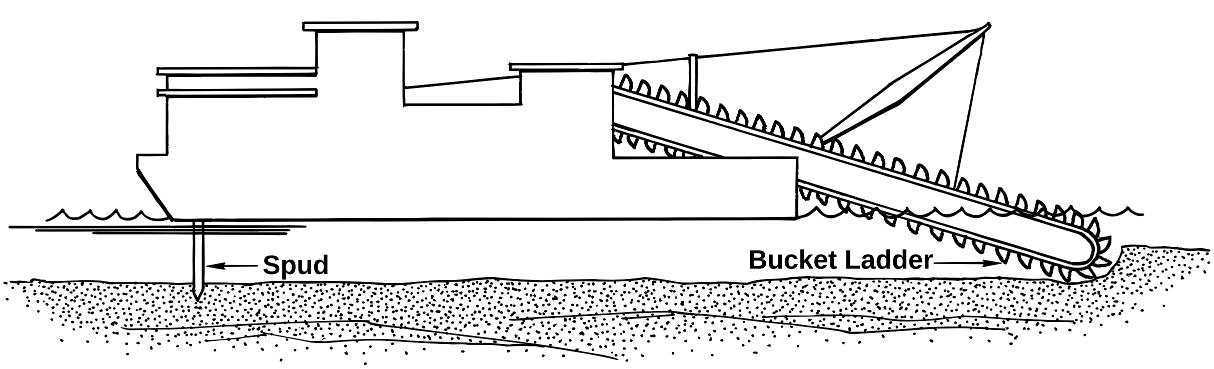


Figure 1.3

As a further development, dredges of this type are sometimes equipped with a special suction head, using water jets or other devices to agitate the material. One particular adaptation of this principle is the Dustpan dredge, so named because of the shape of the suction head. Units of this type are used extensively on large rivers where accumulated materials must be rapidly removed from the navigation channel.

The Self-Propelled Hopper Dredge

Resembling an oceangoing ship, the self-propelled hopper dredge functions in a similar manner to the plain suction dredge. In operation, as the suction pipe or pipes are dragged along the bottom while the dredge is moving ahead at a slow speed, a mixture of water material is picked up and conveyed to the pump or pumps installed on the dredge. The discharge pipes are connected to the dredge pump or pumps to carry the materials to the hoppers which are built into the hull. When the hoppers are filled, the dredge proceeds at full speed to the dumping grounds in deep water. Here, the hopper doors built in the bottom of the hull are opened and the material dumped. The dredge then returns to the site of work and repeats the cycle.

Dredges of this type are necessary for maintenance work and improvement in exposed harbor entrances where traffic and operating conditions will not permit use of stationary dredges. These dredges have been built with hopper capacities ranging up to 6116 m^3 (8,000 $yd.^3$).

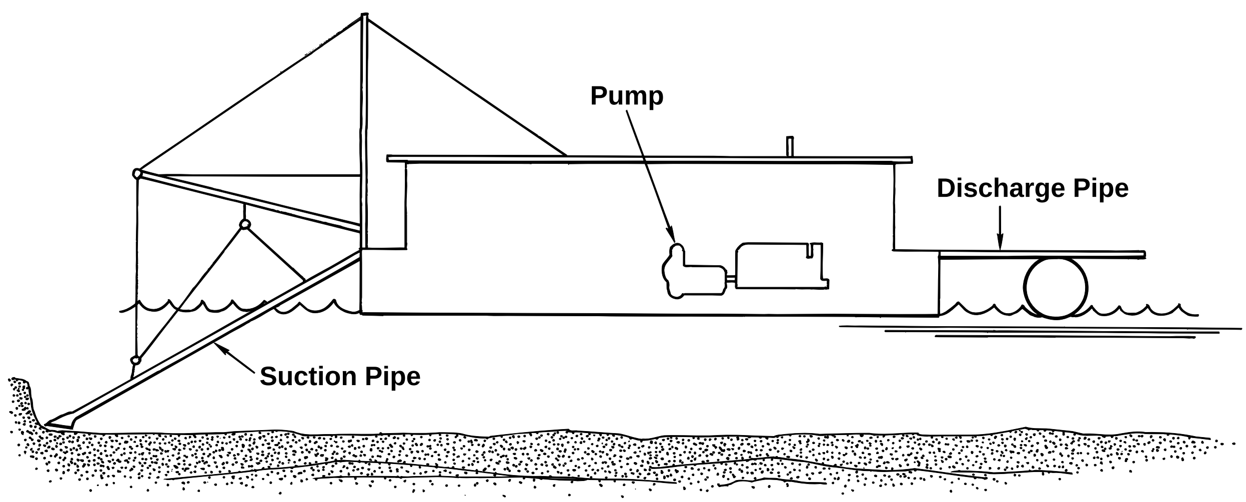


Figure 1.4

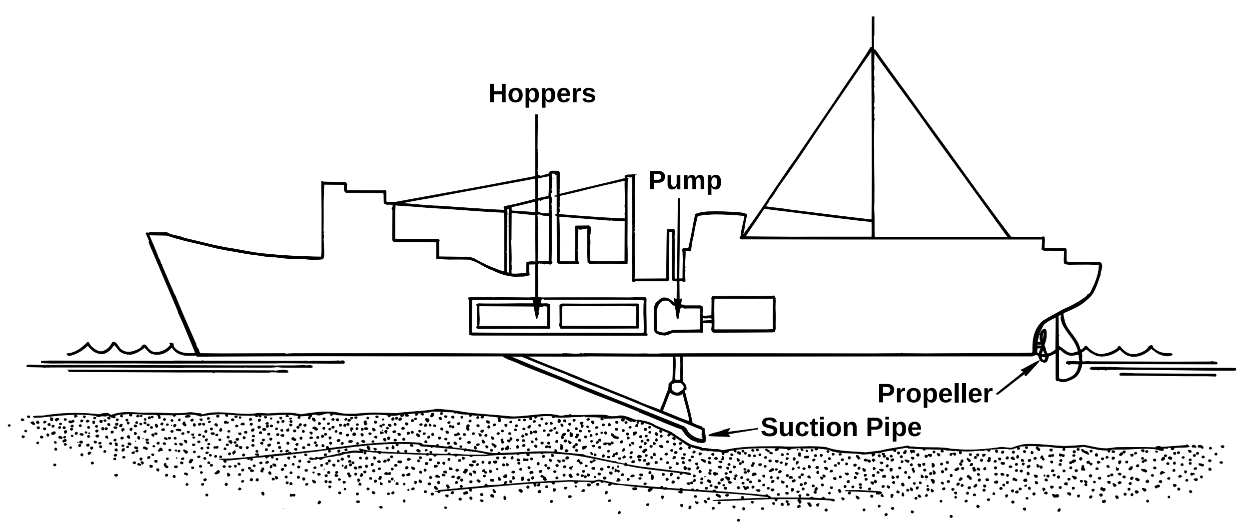


Figure 1.5

The Cutterhead Pipeline Dredge

The cutterhead pipeline dredge is the most versatile and widely used marine excavating unit. It is similar to the plain suction dredge, but is equipped with a rotating cutter surrounding the intake end of the suction pipe. This cutter loosens the material which is then sucked in through the dredging pump, delivered to the stern of the dredge and conveyed to the disposal area by means of a pipeline. Hydraulic pipeline dredges can efficiently dig and pump loose materials as well as compacted deposits such as clay and hard pan. The larger and more powerful machines are used to dredge rocklike formations, such as coral and softer types of basalt and limestone, without blasting.

The cutterhead pipeline dredge, like several other types, is held in working position by spuds and advances by walking itself on these spuds. The advantages of the cutterhead pipeline dredge are its versatility and nearly continuous operating cycle, resulting in maximum economy and efficiency.

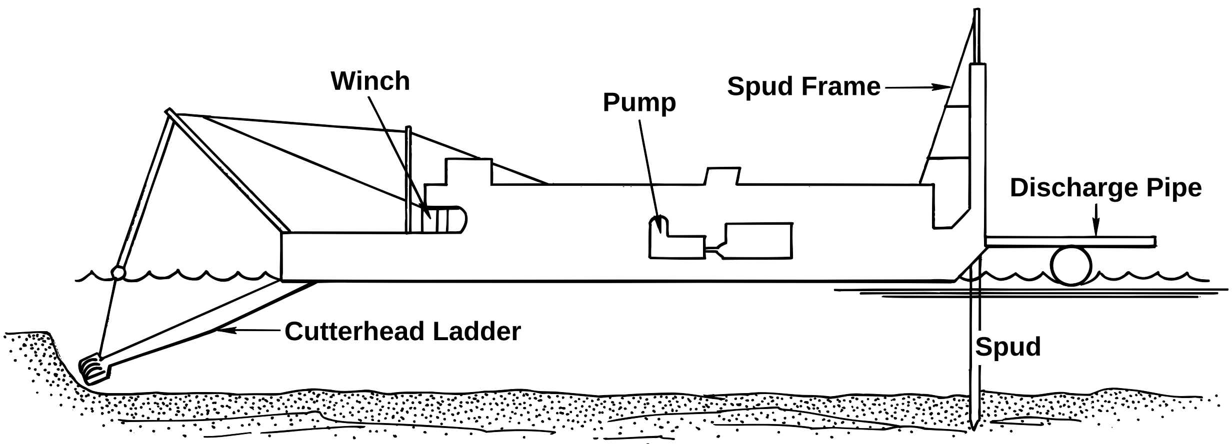


Figure 1.6

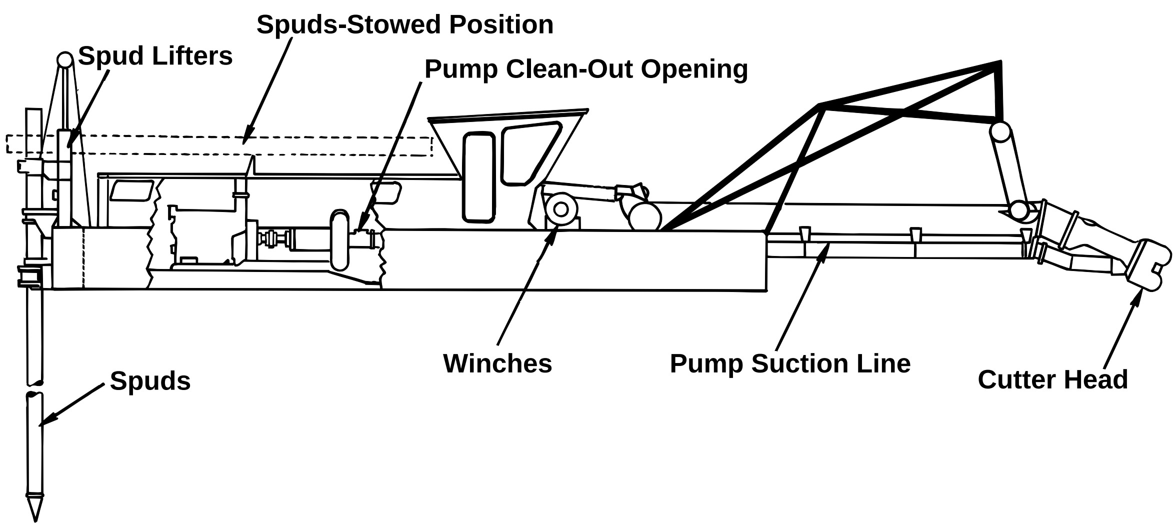


Figure 1.7

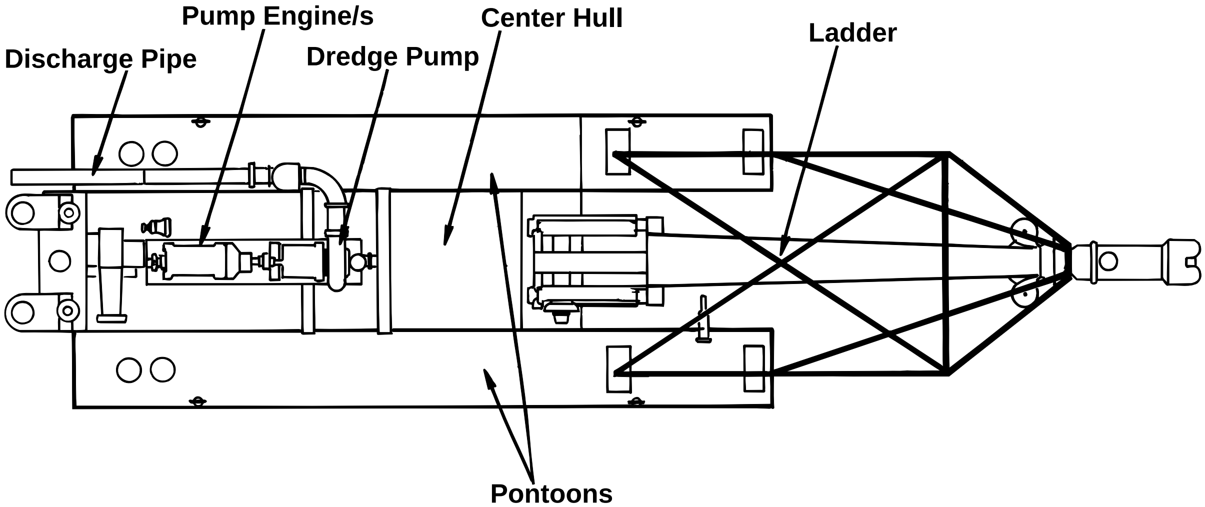


Figure 1.8

Dredge Nomenclature

Cutterhead—A rotating toothed auger for dislodging material. The cutterhead contains the pump suction inlet. The cutterhead is usually driven by a separate diesel engine through hydraulic motors, electric motors, or a shafting drive line. Winches and spud lifters may also be driven by this engine.

Ladder—A horizontally hinged boom, rigidly constructed, which provides structural support for the cutterhead, the cutterhead driving mechanism, and the dredge pump suction line. It is hinged to the front of the dredge and may be lifted or lowered to control digging depth.

Main or Center Hull—A rectangular-shaped hull which contains the dredge pump and its associated reduction gearing, clutches, engine(s), and controls. Mounted on the center hull is the control station, swing and ladder winches, spuds, hydraulic drive engine, and the dredge's service power generation machinery.

Spuds—The dredge is equipped with two, long, tubular, sharp pointed poles, vertically mounted on the rear of the dredge. These spuds are raised and alternately lowered into the bottom material. The spuds provide pivot points around which the dredge may swing as the cutterhead advances into the bottom materials.

Pontoons—The pontoons are longer than the center hull and are mounted on either side. They provide flotation stability and fuel and water storage capacity.

Pump Engine Considerations

Horsepower (Engine Load) Versus Discharge Line Length

A common misunderstanding is that more horsepower is required to pump against a long distance line than a short one. The horsepower requirement of a pump is proportional to the gallons per minute (gpm) being pumped. The longer the discharge line, the greater the resistance to flow, therefore, the fewer gpm and a lower horsepower requirement results. Conversely, as the line length is reduced, resistance to flow is reduced, more gpm are being moved and a higher horsepower required.

If a dredge must be operated with a discharge line shorter than its design length, the engine rpm must be reduced. This reduces the pump rpm, causing a decrease of the horsepower requirement. Throttling back will relieve the engine from an overload situation and may even result in an increase in dredge production.

Horsepower Versus Specific Gravity (Percent Solids)

The heavier the material being pumped, the greater the horsepower requirement. It takes less horsepower to pump pure water than it does to move a mixture of solid material and pure water. The pump horsepower requirement is directly proportional to the specific gravity of the pumped fluid.

Example: If a pump engine is called on to produce 100 hp when pumping clear water, the same pump engine must be capable of developing 150 hp while pumping the same flow rate (gpm) of a slurry (water and solid mix) whose specific gravity is 1.5.

Horsepower Versus Pump Speed

The load on the dredge pump engine is proportional to the cube of the pump speed. This means a small increase in pump rpm will result in a much greater horsepower demand on the engine. For example: To double the speed of the pump impeller would require eight times more horsepower. Stated another way, a pump impeller which demands 100 hp to turn at 150 rpm will require 800 hp at 300 rpm.

Engine Operation to Avoid Overload

If dredging conditions are such that the pump engine(s) are not able to reach rated rpm while at full throttle, then throttle position must be reduced to avoid engine overload. Reduce throttle position from full throttle—while digging—until engine speed drops approximately 50 rpm. This will result in approximately the same horse power output delivered to the pump, but will allow the engine to deliver that horsepower safely, without overfueling.

Engine Installation

Many marine engine installation practices apply equally to dredges. When this is the case, the reader will be referred to the appropriate marine engine section. Only those practices and recommendations that are unique to the dredge application will be discussed in this section.

Mounting and Alignment Mounting Rails

All large bore Vee-type engines should be mounted with angle section, ledge-type marine mounting rails. Engines can be successfully installed using industrial channel section mounting rails, but mounting flexibility is sacrificed. See Marine Mounting Recommendations section for further details on shimming and bolt fit.

Tandem Engine Thermal Growth Considerations

The thermal expansion of engines must not be restrained. The flywheel end of the engine mounting rails should be fixed by a ground body, fitted bolt on either or both sides of the engine. The diameter of the mounting bolts—fixing the engine's rails to the dredge structure—forward of the flywheel must be 1.6 mm (0.06 in.) less than the diameter of the holes in the mounting rails. This clearance will1 allow the engine and mounting rails to grow without confinement.

When installed properly, there is sufficient axial clearance within the Caterpillar viscous damped engine-to-engine coupling to allow the engine nearest the load to grow without restraint. The axial clearance dimension can be checked on a new installation by measuring from the outer face of the grease retaining plate (of the Caterpillar viscous damped coupling) to the nearest surface of the coupling inner member. This dimension should be 8.6 ± 0.76 mm (0.34 ± 0.03 in.).

Tandem Engine Timing Considerations

Timing Recommendations are contained on Tandem Engine Coupling Arrangement drawings. These directions must be followed to avoid possible torsional vibration problems.

Tandem Engine Governor Settings (Low Idle rpm)

Some dredge pump drive applications require a special engine low idle setting to

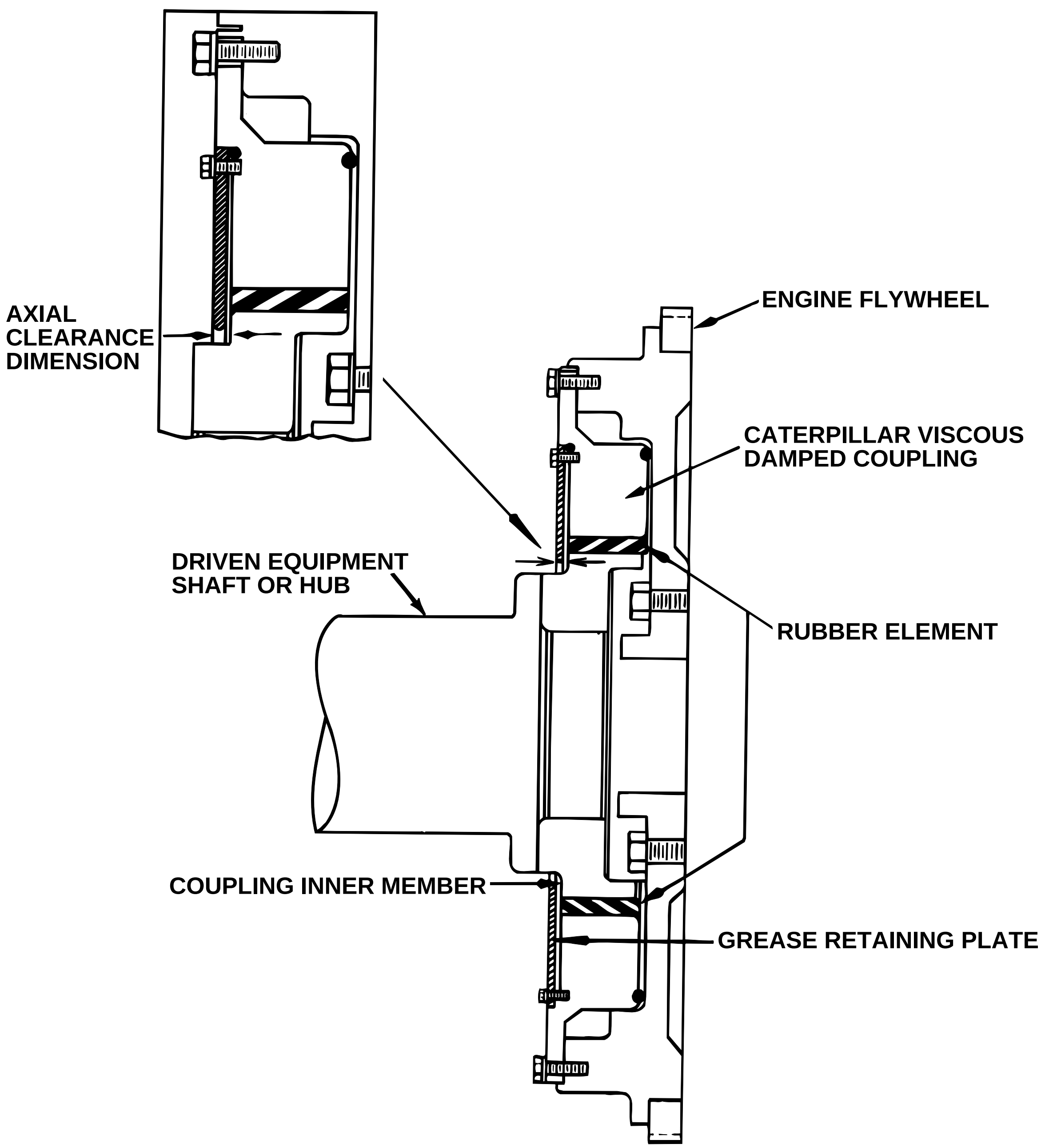


Figure 1.9

avoid torsional resonance. Many Caterpillar engines used in tandem service must have a low idle setting of not less than 600-650 rpm. Always check the engine data plate to determine proper governor settings. Many dredge engine settings are special and not listed in standard Caterpillar Service literature.

Fuel Treatment and Plumbing

Since dredges are normally equipped with very large fuel tanks, condensation, fungus/ bacteria growth, and contamination of the fuel may be troublesome. Fuel system maintenance is especially important in dredge applications.

The lowest point within the dredge's fuel tanks should be drained or pumped daily to eliminate condensed moisture and sediment.

Water and sediment traps should be used in fuel supply lines.

Terminate engine fuel supply plumbing at least 300 mm (12 in.) above the lowest point in fuel tanks.

See fuel section for information on detection and prevention of fungus/bacteria growth in fuel tanks.

Exhaust, Ventilation, and Crankcase Vent Systems

All diesel engines require large quantities of clean, cool air for long trouble free life.

Combustion Air

Equip dredge engines with combustion air inlet ducts, located and routed to prevent recirculation of exhaust gases and crankcase fumes. Locate combustion air inlets so they do not ingest heated engine room ventilation air rising through removable roof caps. Exhaust gases must be discharged to atmosphere high enough above the combustion air inlet openings to prevent rebreathing of exhaust gases. Equip exhaust stacks with joints which allow addition of extra sections of exhaust pipe if exhaust recirculation proves to be a problem on operating location. An unrestricted elbow-type exhaust discharge fitting is preferred over the counterbalanced flapper valve because there will be less chance for downward deflection of exhaust gases.

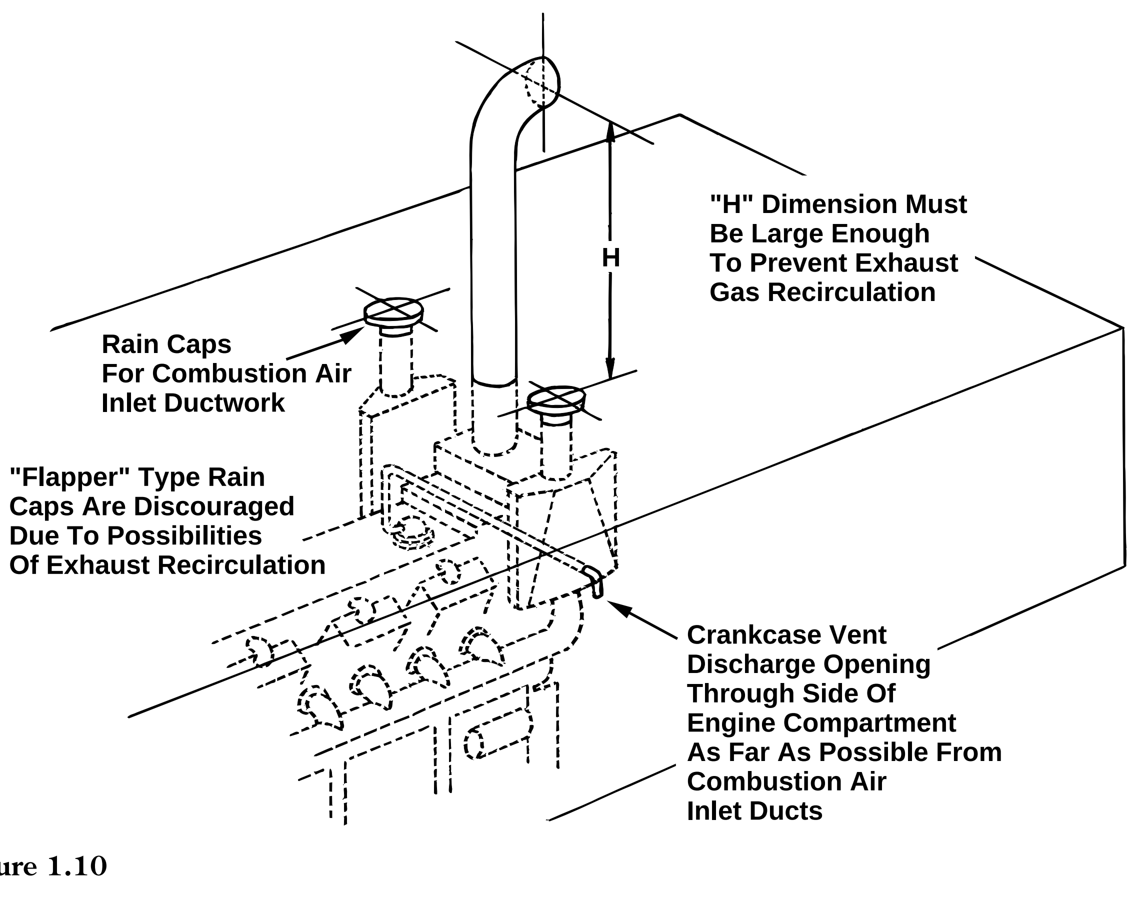


Figure 1.10

See *Ventilation* and *Exhaust* sections for flow, pressure, and temperature information.

Ratings

Engines in dredge pump drive service should be applied at the continuous ratings.

Engines driving electric or hydraulic cutterheads and/or winches may carry the intermittent or light duty commercial rating since neither cutterheads nor winches are continuous loads.

Engines driving generators which supply lighting and service pumping power should be rated for prime power due to the continuous nature of lighting and pumping loads.

For additional explanation of Caterpillar engine rating philosophy, see *Ratings* section.

Cooling

The dredging application may place severe demands on the engine cooling system.

Keel Cooling Considerations

The dredge is normally stationary on its digging location with little or no water flowing past the hull. The efficiency of keel coolers is greatly reduced under these conditions. Keel coolers operating in *dead water* conditions require more than twice the surface area that would be required if the cooling surface had a five knot water velocity over the outside of the cooler. See Keel Cooler Area Requirement curves in the Marine Cooling System section for added information.

Heat Exchanger Cooling

Using inboard (shell and tube-type) heat exchangers may be troublesome due to the highly abrasive particles suspended in the water common to dredging operations. Cooling water suction line strainers are a necessity to minimize damage to pumps and heat exchangers tubes. Use Caterpillar engine-mounted sea water pumps (particularly rubber impeller pumps) with the knowledge that their service life, when pumping water containing abrasive particles, will be significantly shortened.

Hydraulic Dredges will have, as one of their normal components, a "service water" pump. This pump is usually designed for abrasive water service. The service water pump provides clean sea water, at higher pressure, for lubrication and flushing of the dredge pump and cutterhead lineshaft bearings. Use of excess flow from the service water pump is a superior alternative to Caterpillar engine mounted sea water pump.

Box Coolers

Box-type coolers offer many of the advantages of keel cooling and shell and tube coolers, particularly in dredging applications.

Aftercooler Cores

Although many dredges operate in freshwater ponds, lakes, and rivers, experience has proven that engine aftercooler cores suitable for sea or salt water are a necessity. The moisture-laden air surrounding any floating equipment will corrode the fins on the air side of nonmarine aftercooler cores severely limiting the heat transfer and possibly even restricting the combustion flow.

Controls

Single Engine Drive

Dredge engine controls are the same as conventional controls for engines in other pumping or electric power generation applications, with the exception of tandem or compound engines driving a single load.

Tandem/Compound Engine Drive

When multiple engines are tandemed (nose-to-nose configuration) or compounded side-by-side configuration with flywheel outputs (combined in gearing or with chains), the capability to share load equally at full load becomes important.

Load Share

Engines must share load equally so one engine, the one taking most of the load, will not wear out prematurely or fail. The precision of the sharing of the load is only important at or near the engines full power capability (large fractions of the engine rating).

Hydra-Mechanical Governors

A way to ensure load share at full load is to adjust the air actuators on Caterpillar standard hydra-mechanical governors so both engines reach the same high idle speed (rpm) with the same air actuator pressure. The adjustment is normally done at the factory when pairs of engines are specified to be used in tandem or compound.

Isochronous Governors

Load sharing is more easily attained if governors capable of isochronous operation are avoided or adjusted to operate in a droop, or non-isochronous mode. Generally 5-10% droop is satisfactory.

Safety System Considerations on Tandem/Compound Engines

Prelubrication System

Wire the oil pressure sensors included with prelubrication systems in series to prevent either tandem/compound engine's cranking motor from engaging before both engines are prelubed.

Shut-down Devices

Sensors connected to automatic shut-down devices must be interconnected on tandem/compound engines to insure both engines shut-down in the event of a malfunction in either engine.