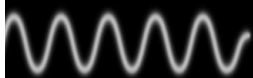




# **Electric Power Application and Installation Guide**

## **Mounting Systems**

LEBX0045



WHERE THE WORLD TURNS FOR POWER



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# Mounting Systems

Proper generator set installation is crucial to ensuring the efficient, long and dependable life of a system, as well as minimizing time spent on maintenance.

Caterpillar Gas Engines are rigid, self-contained structures which will operate and maintain inherent alignment unless subjected to extreme external stresses.

Due to the diversity of installation types, no one mounting system or method is universally acceptable. The engine must be mounted in a manner suited to the specific application, taking into account the characteristics of the engine, the driven loads, and the operating cycle of the machine. One or more of the following results will occur if mounting method is inadequate.

## Foundations

Major functions of a foundation are to:

- Support the total weight of the generator set, which includes accessory equipment and liquids (coolant, oil and fuel)
- Maintain alignment between engine, generator and accessory equipment.
- Isolate generator set vibration from surrounding structures

The equipment foundation is not the responsibility of Caterpillar Inc., nor is the driven equipment attachment to the foundation. The customer or customer's agent familiar with local site conditions and application requirements bears foundation design responsibility. Foundation comments published herein are intended only as general guidelines for consideration. Further engine foundation general guidelines can be found in Engine Data Sheet 30.0, Form LEHQ1172.

## Soil, Gravel, or Rock Mounting

Firm, level soil, gravel, or rock provides satisfactory support for single-bearing generator sets used in stationary or portable service. This support can be used where the weight-bearing capacity of the supporting material exceeds pressure exerted by the equipment package, and where alignment with external machinery is unimportant.

Soil, such as fine clay, loose sand, or sand near the ground water level, is particularly unstable under dynamic loads and requires substantially larger foundations. Information concerning bearing capacity of soils at the site may be available from local sources and must comply with local building codes.

Where support rails or mounting feet have insufficient bearing area, flotation pads can distribute the weight. The underside area and stiffness of the pad must be sufficient to support the equipment.

Seasonal and weather changes adversely affect mounting surfaces. Soil changes considerable while freezing and thawing. To avoid movement from seasonal changes, extend foundations below the frost line.

## Concrete

Several basic foundations are applicable for generator sets. The foundation chosen will depend on factors previously outlined as well as limitations imposed by the specific location and application.

Massive concrete foundations are unnecessary for modern multicylinder medium speed generator sets. Avoid excessively thick, heavy bases to minimize subfloor or soil loading. Bases should be only thick enough to prevent deflection and torque reaction, while retaining sufficient surface area for support. Non-parallel units require no foundation anchoring.

If a concrete foundation is required, minimum design guidelines include:

- Strength must support wet weight of units plus dynamic loads.
- Depth sufficient to attain a minimum weight equal to generator set wet weight (only if large mass, i.e. inertia block, is specified for vibration control).
- Outside dimensions exceed that of the generator set, a minimum of 304.8 mm (12 in.) on all sides.

Before calculating the depth of the concrete foundation, certain issues must be considered:

- When effective vibration isolation equipment is used, floor concrete must only be deep enough to provide structural support of the static load.
- If isolators are not used, dynamic loads transmit to the facility floor and require it to support 125 percent of the generator wet weight.
- If generator sets are paralleled, possible out-of-phase paralleling and resulting torque reactions demand foundations that are able to withstand twice the wet weight of the generator set.

Estimate foundation depth that will accommodate generator set weight using the formula:

$$FD = \frac{W}{D \times B \times L}$$

FD = Foundation depth (m), (ft)

W = Total wet weight of generator set (kg), (lb)

D = Density of concrete (kg/m<sup>3</sup>), (lb/ft<sup>3</sup>)  
(2402.8 kg/m<sup>3</sup>), (150 lb/ft<sup>3</sup>)

L = Foundation length (m), (ft)

Suggested concrete mixture by volume is 1:2:3 of cement, sand, aggregate, with maximum 100 mm (4 in.) slump and 28-day compressive strength of 20 MPa (3000 psi).

## Ground Loading

Initial considerations include generator set weight and material supporting this weight.

The wet weight of the total package must be calculated. This includes accessory equipment and weight of all liquids (coolant, oil, and fuel) supported by the foundation. Engine and attachments dry weights can be found in the price list. Liquid densities are given in Table 1.

Liquid	kg/m <sup>3</sup>	lb/U.S. gal	lb/ft <sup>3</sup>	Specific Gravity
Water, Fresh	994.6	8.3	62.1	1.00
Water, Sea	1018.3	8.5	63.6	1.02
Water/Glycol	1024.4	8.55	64.0	1.03
Diesel Fuel	850.7	7.1	53.1	0.855
Lube Oil	909.7	7.6	56.8	0.916
Kerosene	802.7	6.7	50.1	0.807

Table 1. Densities of liquids [@ 16°C (60°F)].

Material supporting the foundation must carry the total weight (see Table 2) shows bearing load capabilities of common materials.

Material	Safe Bearing Load kPa (psi)
Rock, Hardpan	482.6 (70)
Hard Clay, Gravel and Coarse Sand	386 (56)
Loose Medium Sand and Medium Clay	193 (28)
Loose Fine Sand	96.5 (14)
Soft Clay	0-96.5 (0-14)

Table 2. Bearing load capability.

The area of load-bearing support is adjusted to accommodate surface material. To determine pressure (P) exerted by the generator set, divide total weight (W) by total surface area (A) of the rails, pads, or vibration mounts.

$$P = \frac{W}{A}$$

Where: P = pressure (kPa), (psi)

W = weight (kg), (lb)

A = area (m<sup>2</sup>), (in<sup>2</sup>)

Pressure imposed by the generator set weight must be less than the load-carrying capacity of supporting material.

Where support rails or mounting feet have insufficient bearing area, flotation pads can distribute the weight. The underside area and stiffness of the pad must be sufficient to support the equipment.

With vibration isolation between the baseframe and the floor, if the load is equally distributed over all isolators, the floor loading is:

$$\text{Floor Loading} = \frac{\text{Total Generating Set Weight}}{\text{Pad Area} \times \text{Number of Pads}}$$

Thus, floor loading can be reduced by increasing the number of isolation pads.

If load is not equally distributed, the maximum floor pressure occurs under the pad supporting the greatest proportion of load (assuming all pads are the same size):

$$\text{Max Floor Pressure} = \frac{\text{Load on Heaviest Loaded Pad}}{\text{Pad Area}}$$

Seasonal and weather changes adversely affect mounting surfaces. Soil changes considerable while freezing and thawing. To avoid movement from seasonal change, extend foundations below the frost line.

### Grounding

The generator set must be properly ground before operation startup. NEC recommends maximum 25 Ohm resistance to the ground (reference ANSI/IEEE C37-101-1985 "Guide for Generator Ground Protection").

### Bases

The first design consideration for an engine base is its physical dimensions. The base must provide the proper mounting holes for the engine and all other base-mounted components. The holes must also make allowance for servicing of the engine and other components. They must provide clearance and provisions for proper alignment.

Design the base to maintain the original alignment between engine and driven equipment under all operational and environmental conditions. Misalignment between an engine and driven equipment can cause vibration and

shorten the life of couplings and bearings. Bases designed and fabricated by dealers, or others, must meet design requirements of the Caterpillar supplied base to assure strength and vibration resistance.

The major cause of misalignment is flexing of the base due to lack of torsional rigidity. Other causes are poor installation methods and incorrect alignment procedures.

The base must offer rigidity adequate to oppose the twist due to torque reaction on drives where the driven equipment is mounted on the base assembly, but not bolted directly to the engine flywheel housing (see Figure 1).

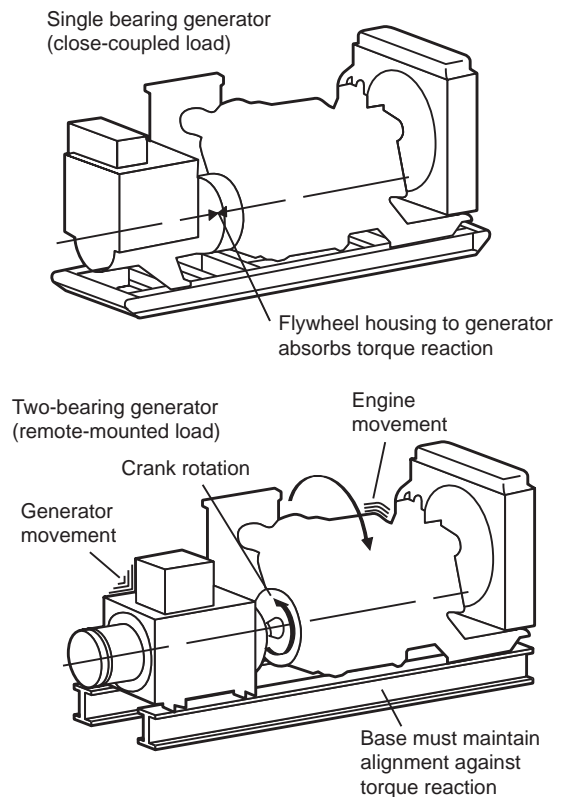


Figure 1. Absorption of torque.

Caterpillar bases are designed to eliminate frequent, periodic realignment of the engine and driven unit. A properly installed Caterpillar base will meet the following criteria:

- Engine torque does not cause excessive misalignment.
- Flexing of the foundation or substructure under the base during operation does not cause bending of the base.

- When the engine and driven unit are mounted on a Caterpillar base, the entire package is able to withstand normal handling during transportation without permanently distorting the base or causing misalignment of the driven unit.
- A Caterpillar base is free of torsional or linear vibrations in the operating speed range of the engine.

## Types of Mounting Systems

The choice of mounting method depends on the configuration and application of the generator set. Various methods are discussed in the following.

### Fabricated Bases

Frequent relocation, initial installation ease, vibration isolation or isolating from flexing mounting surfaces, such as trailers, are major uses for fabricated bases. Do not rigidly connect any base to flexing surfaces.

Bases maintain alignment between engine, generator, and other drive equipment such as radiator fans. Engines with close-coupled single-bearing generators maintain alignment by mounting rails or modest bases. Two-bearing generators, generators driven from either end of the engine, tandem generators, or tandem engines, require substantial boxed bases. Bases must incorporate sufficient strength to:

- Resist outside bending forces imposed on the engine block, couplings, and generator frame during transportation.
- Limit torsional and bending movement caused by torque reactions.
- Prevent resonant vibration in the operating speed range.

Due to thermal expansion, [cast iron  $5.5 \times 10^{-6}$  mm/mm/1.8°C ( $5.5 \times 10^{-6}$  in./in./1.0°F)] engines may lengthen 2.3 mm (0.09 in.) from cold to operating temperature. *This growth must not be restrained.* On single-bearing and most two-bearing generators, no close clearance dowels or ground body bolts are used to limit thermal growth. Single-bearing generators requiring extremely close alignment, use a ground body bolt at the flywheel end on one side of the engine. No other restraint is permitted.

Mounting feet of two-bearing generators can be doweled without harm. Slight expansion within the generator is absorbed in the generator coupling.

### Single of Multiple Bearing

Caterpillar offers different bases for close coupled units (such as single-bearing generators) and for remote mounted units (such as two-bearing generators). The base for a remote mounted two-bearing generator must be sturdy enough to provide support and maintain alignment, meaning that it must be more rigid. The base for a close coupled single-bearing generator can be lighter because the base does not have to withstand torque reaction.

Fastening the driven unit housing to the flywheel housing eliminates the need for the base to absorb the engine torque (see Figure 1). On remote mounted units, the frame of the 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 would be misalignment proportional to the amount of load. This misalignment would not be evident during a static alignment check.

For engines with close-couple units, 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

Caterpillar Engines and single-bearing or close-coupled two-bearing generators with mounting feet can either be mounted on a base or mounted directly on a pair of beams without a base. The mounting feet must be bolted in place. Do not weld the feet to the base or beams. If support beams are used, the beams must be flat and lie in the same plane. Use shims as necessary between feet and base or support beams so all mounting feet are in solid contact at all locations. If the mounting feet are not in equal contact with the base or beams before the anchor bolts are installed, the engine and/or generator can be stressed when the anchor bolts are tightened.

## Mounting Rails

Lugs or brackets have been welded to the sides of the oil pan at the factory and hold the engine to the mounting rails. Shimming is not allowed between the engine oil pan and the mounting rails.

On engines whose mounting rails are extended to mount close-coupled driven units, these rails must not be notched or the cross braces removed to provide clearance for the driven unit. Shims are used as necessary between the mounting feet of the drive unit and the mounting rails to get correct alignment with the engine.

If the driven equipment is close-coupled to the engine, use clearance type bolts at all locations to fasten the mounting rails to the mounting blocks or base. These bolts must have a diameter 1.5 mm (0.06 in.) less than the diameter of the holes in the mounting rails.

If the driven equipment is remote-mounted, clearance type bolts should be used at all mounting locations in the left mounting rail. If a ground body bolt is to be used to control the direction of horizontal thermal growth, install this bolt in the right rail at the end of the rail next to the coupling, or at the rear of the rail if remote mounted equipment is driven from both ends of the engine. Install clearance type bolts at all other locations in the right rail. If a spot weld is to be used to control the direction of thermal growth, spot weld the right rail to the mounting block and/or shims on the side of the rail next to the coupling. Install clearance type bolts at all mounting locations in the right mounting rail. All clearance type bolts must have a diameter 1.5 mm (0.06 in.) less than the diameter of the mounting rails holes.

Each anchor bolt between the mounting rail and the base must be bolted into a mounting block. Distortion of the mounting rails will result if these bolts are fastened from the rails into the base (see Figure 2).

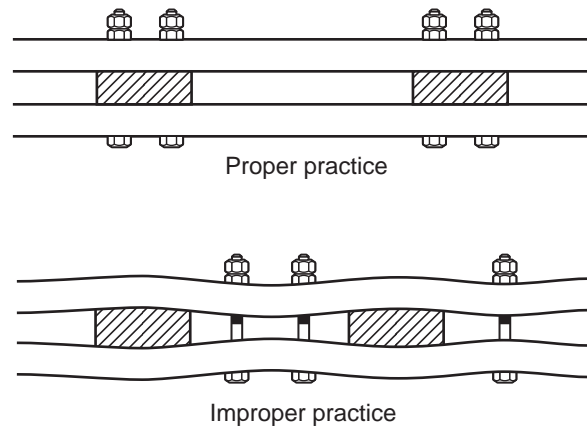


Figure 2. Bolt position.

For engines driving remote-mounted equipment, the mounting rails must be cradled between guide strips which keep the expansion of the rails always parallel to the output shaft centerline. The guide strips are welded to the top of the mounting blocks at the opposite end of the rails from the ground body bolt or spot weld.

Because horizontal thermal growth of the engine and mounting rails will always be away from the ground body bolt or spot weld, never weld stops or chocks against the opposite end of the mounting rails from the ground body bolt or spot weld. If chocks or stops are to be used, there must be a minimum of 0.13 mm (0.005 in.).

## Skid Mounting

Skid mounting is a heavy duty base (see Figure 3). This mounting method is generally most suitable for the semimobile type of power unit or fixed installation which may be subject to the need for occasional relocation. The unit cannot be operated during such movement as the skid base is not supported on a machine subframe.

For these applications the auxiliary fuel tank located below the generator set will have additional wall thickness in order to withstand the handling.

Skid mounting is normally used when the engine drives pumps, blowers, generators, air compressors, or if an outboard bearing is used.

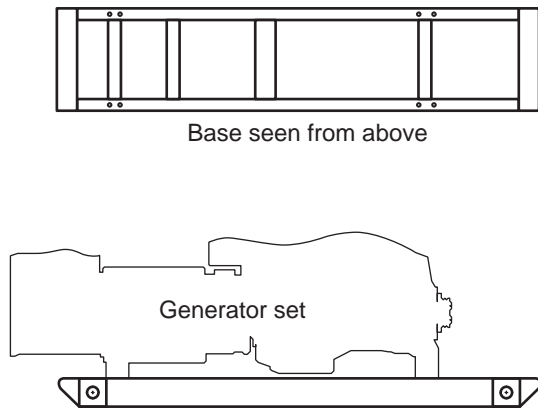


Figure 3. Skid mounting.

### Three/Four Point Mounting

A Caterpillar base for remote mounted equipment is a torsionally rigid structure for mounting the engine and driven unit. The three point suspension design maintains proper relationship and alignment of all equipment by isolating external forces (see Figure 4). The three-point mounting rails can only be used for close-coupled driven units where the rails are extended for the mounting of these units.

The three-point suspension system must be used when there is a possibility the foundation or 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. More than three mounting points can cause base distortion.

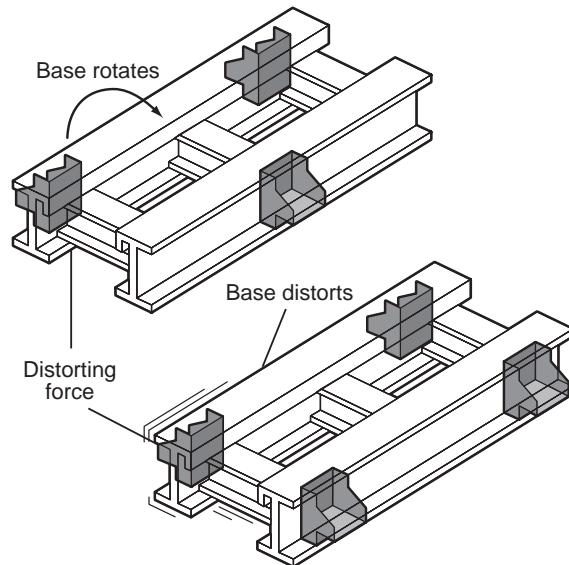


Figure 4. 3-point mounting.

Four-point mounting rails are used if the rails are to be secured to a base or foundation that will not be subjected to external forces that could cause distortion of the mounting rails. These rails can be used for either remote-mounted or close-coupled driven units. For close-coupled driven units, extended mounting rails are available so the driven equipment can be fastened directly to the rails.

Shims only or four drilled and threaded mounting blocks and shims can be used between the engine rails and the base at the four corner location. If mounting blocks are used, these blocks must be welded to the base. The blocks are first fastened to the bottom of the rails at the four corners. Remove bolt clearance with the rails by driving the blocks toward the end of each rail. This will provide clearance for thermal growth at operating temperature. The engine is then put in position on the base and the blocks tack welded to the base. If necessary, remove the engine to complete welding the blocks to the base.

Use shims as necessary to make sure the mounting rails are in solid contact with the mounting blocks or base at all four mounting locations. Distortion of the mounting rails will result when the anchor bolts are tightened, if the mounting rails are not in solid contact.

## Non-Cat Bases

Bases not manufactured by Caterpillar must meet several design criteria. These bases must be rigid enough to limit torsional and bending forces caused by torque reaction and subbase flexing. They must prevent excessive bending forces from passing to the engine block, couplings, and driven unit during shipment. To prevent resonance, they must have a natural frequency out of the operating speed range. They must allow sufficient space for shimming so proper alignment can be accomplished.

## Vibration

All mechanical systems with mass and elasticity are capable of vibration. Engines produce vibration due to combustion forces, torque reactions, structural mass and stiffness combinations, and manufacturing tolerances on rotating components. These forces may create conditions ranging from unwanted noise to high stress levels, and possible ultimate failure of engine or driven components.

The same amplitude and frequency of vibration generated by the engine could result in structural damage if a fixed installation were housed in a building, or close to sensitive instruments or equipment, such as computers.

Other influencing factors are foundation design, soil load characteristics, and other machinery operating in close proximity.

Vibrating stresses can reach destructive levels at engine speeds which cause resonance. Resonance occurs when natural system frequencies coincide with engine excitation frequencies.

Engine vibrations are produced and maintained by regular, periodic driving forces set up by unbalanced moving masses. These are called forced vibrations.

Free vibrations have no driving force. When set in motion such vibrations, is undamped, would continue indefinitely with constant amplitude and natural frequency.

If the frequency of a forced vibration is the same as the natural frequency of free vibrations,

excessive vibration results. This is called resonance and can cause serious problems.

## Mass Elastic System

Engine vibration may be of the following types and causes:

1. Linear vibration — vertical and/or horizontal inertia forces due to lack of balance in reciprocating or rotating machinery.
2. Torque reaction — not a vibratory force, but may excite vibration.
3. Torsional vibration of shafting — occurs in any rotating mass elastic system (two or more masses connected by an elastic shaft) where periodic forces are present. Where these forces recur near the natural frequency of torsional vibration, resonance may develop and cause dangerous stress.
4. Axial vibration of shafting — when torques are applied to a crankshaft, it is alternately shortened and lengthened. This could be troublesome if the natural axial frequency is near a torsional frequency.

Generator sets need no isolation for protection from self-induced vibrations. However, isolation is required if:

1. Engine vibration must be separated from building structures.
2. Vibrations from nearby equipment are transmitted to inoperative generator sets.
3. System is supported on a flexible mounting surface, such as a trailer bed.

Vibration isolators prevent the transmission of possible damaging generator set vibration throughout a building. Noise is also reduced.

When an engine and generator are to be assembled to each other, vibration studies and tests must be completed to assure satisfactory, trouble-free operation on the job site. With factory assembled generator sets, the responsibility is assumed by the manufacturer. In any case, wherever assembly takes place, someone must assure the integrity of the installation from a vibration standpoint.

## Vibration Measurement

Vibration measurements on large engine units should be made using the Caterpillar Vibration Analyzer, Part No. 4C-3030.

If Caterpillar measuring equipment is not available, an equivalent device capable of measuring peak-to-peak displacement at selected frequencies, overall velocity, and overall displacement should be used.

Vibration should be measured at nine points on a two bearing generator set. Comparable points on non-generator driven equipment may be important. These points are illustrated in Figure 5 and are described below.

### Point 1

Horizontal direction at the front of the engine; locate the probe on the side of the block at the crankshaft centerline.

### Point 2

Vertical direction at the front of the engine; locate the probe on the block top deck in the plane of the crankshaft centerline.

### Point 3

Horizontal direction at the rear of the engine; locate the probe on the side of the block at the crankshaft centerline.

### Point 4

Vertical direction at the rear of the engine; locate the probe on the block top deck (or rear housing) in the plane of the crankshaft centerline.

### Point 5

Vertical direction at the generator front bearing; locate the probe on the bearing housing at the shaft centerline.

### Point 6

Horizontal direction at the generator front bearing; locate the probe on the side of the bearing housing at the shaft centerline.

### Point 7

Vertical direction at the generator rear bearing; locate the probe on the bearing housing at the shaft centerline.

### Point 8

Horizontal direction at the generator rear bearing; locate the probe on the side of the bearing housing at the shaft centerline.

### Point 9

Axial direction at the generator rear; locate the probe on the rear right outside edge of the generator structure (not sheet metal) at the shaft centerline.

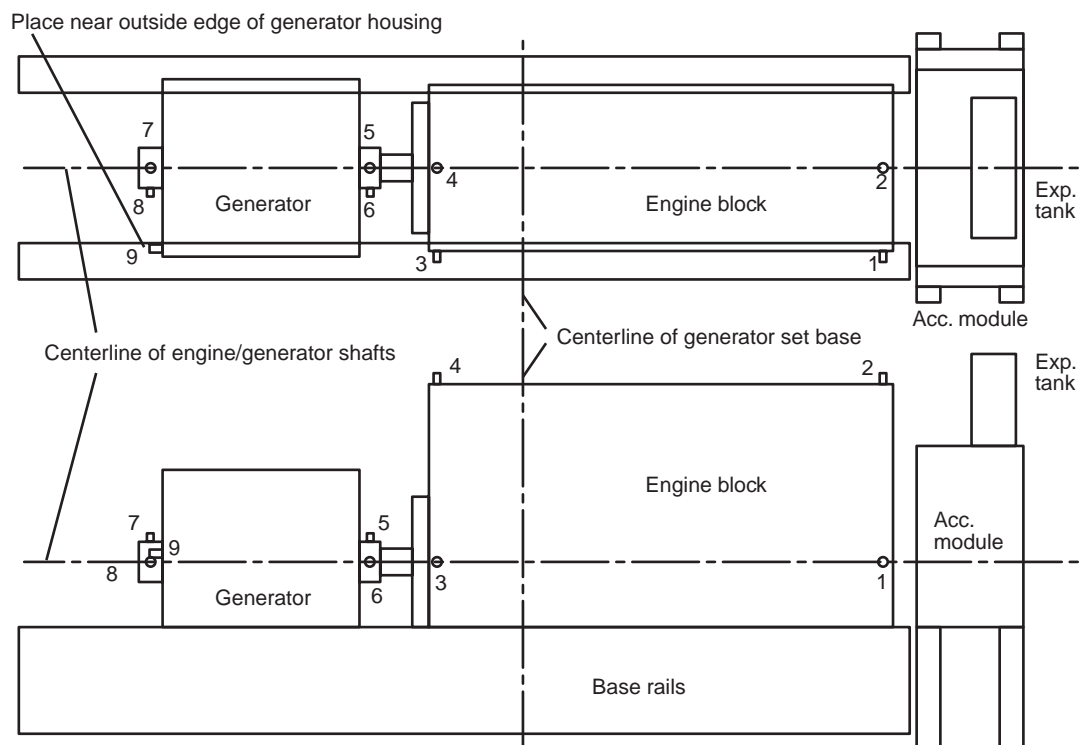


Figure 5. Vibration measuring points.

Vibration measurements must be made at the advertised driven equipment rating (100% load). If additional data is desired, it is recommended that measurements be made at 0% load, 50% load, and 75% load.

Data must be reported in terms of peak-to-peak displacement (mils) at half order frequency, first order frequency, overall velocity level (in/s) and overall displacement (mils) for each of the nine measuring locations. The chart at the end of this section can be used to record and report the measured vibration data.

### Vibration Limits

The vibration levels for any load condition, at any of the nine measuring locations, must not exceed the following guideline limits:

1. Peak-to-peak displacement at half order frequency = 0.13 mm (5 mils)
2. Peak-to-peak displacement at first order frequency = 0.13 mm (5 mils)
3. Overall displacement = 0.22 mm (8.5 mils)
4. Overall velocity = 34.3 mm/s (1.35 in/s)

This is applicable to both Diesel and Gas Engines (reference: EDS 73.1, Linear Vibration).

Consult the manufacturer of the driven equipment for any such vibration limits.

If the measured vibration levels exceed the limits, contact your Caterpillar dealer representative or Caterpillar factory representative for assistance.

Warning: It is not an acceptable practice to lower the package vibration levels when operating at stable conditions by tightening the snubber bolts on the Caterpillar vibration isolators. This practice will only hide vibration problems.

### Linear Vibration

Linear vibration is exhibited by noisy or shaking machines, but its exact nature is difficult to define without instrumentation. Human senses are inadequate to detect relationships between the magnitude of vibration and period of occurrence. A first order ( $1 \times \text{rpm}$ ) vibration of 0.254 mm (0.010 in.) displacement may feel about the same as third order measurement of 0.051 mm (0.002 in.).

Vibration occurs as a mass is deflected and returned along the same place and can be illustrated as a single mass spring system (see Figure 6). With no external force imposed on the system, the weight remains at rest and there is no vibration. But when the weight is moved or displaced and then released, vibration occurs. The weight travels up and down through its original position until frictional forces cause it to rest. When external forces, such as engine combustion, continue to affect the system while it vibrates, it is termed *forced vibration*.

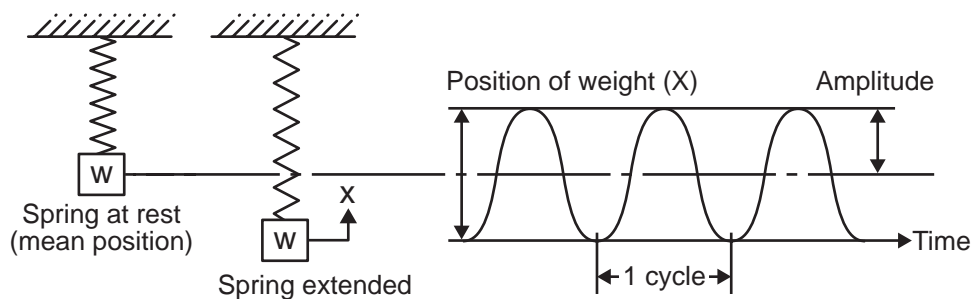


Figure 6. Mass-spring system.

Time required for the weight to complete one movement is called a period (see Figure 7).

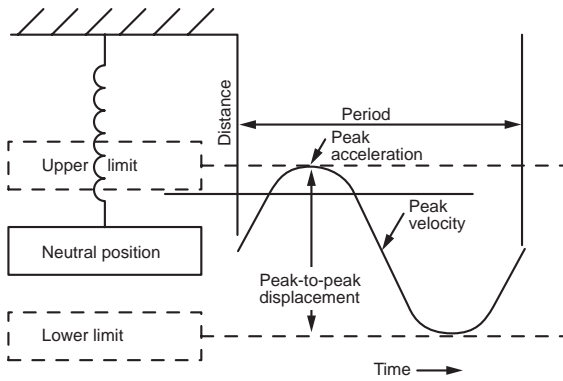


Figure 7. Illustration of a period.

Maximum displacement from the mean position is amplitude; interval in which the motion is repeated is called the cycle.

If the weight needs one second to complete a cycle, the vibration frequency is one cycle per second.

If one minute, hour, day, etc., were required, its frequency would be one cycle per minute, hour, day, etc. A system completing its full motion 20 times in one minute would have a frequency of 20 cycles per minute, or 20 cpm.

Establishing vibration frequency is necessary when analyzing a problem. It allows identification of engine component or condition causing the vibration.

Total distance traveled by the weight, from one peak to the opposite peak, is peak-to-peak displacement. This measurement is usually expressed in mils, one mil equaling one-thousandth of an inch [0.025 mm (0.001 in.)]. It is a guide to vibration severity.

Average and root-mean-square (rms) are used to measure vibration (rms = 0.707 times the peak of vibration.) These terms are referred to in theoretical discussions.

Another method to analyze vibration is measuring mass velocity. Note that the example is not only moving but changing

direction. The speed of the weight is also constantly changing. At its limit, the speed is "0". Its speed or velocity is greatest while passing through the neutral position.

Velocity is extremely important; but because of its changing nature, a single point has been chosen for measurement. This is peak velocity normally expressed in inches per second.

Velocity is a direct measure of vibration and provides best overall indicator of machinery condition. It does not, however, reflect the effect of vibration on brittle material.

Relationship between peak velocity and peak-to-peak displacement is compared by:

$$V \text{ Peak} = 52.3 \times D \times F \times 10^{-6}$$

Where:

V Peak = Vibration velocity in inches per second peak.

D = Peak-to-peak displacement, in mils (1 mil = 0.001 in.).

F = Frequency in cycles-per-minute (cpm).

Acceleration is another characteristic of vibration. It is the rate of velocity change. In the example, note that peak acceleration is at the extreme limit of travel where velocity is "0". As velocity increases, acceleration decreases until it reaches "0" at the neutral point.

Acceleration is dimensioned in units of "g" (peak), where "g" equals the force of gravity ( $980 \times 6650 \text{ mm/s}^2 = 386 \text{ in./s}^2 = 32.3 \text{ ft/s}^2$ ).

Acceleration measurements, or "g's", are used where relatively large forces are encountered. At very high frequencies (60,000 cpm), it is perhaps the best indicator of vibration.

Vibration acceleration is calculable from peak displacement

$$g \text{ Peak} = 1.42 \times D \times F^2 \times 10^{-8}$$

Machinery vibration is complex and consists of many frequencies. Displacement, velocity, and acceleration are all used to diagnose particular problems. Displacement measurements are better indicators of dynamic stresses and are, therefore, commonly used. Note that overall, or total peak-to-peak displacement, described in Figure 8, is approximately the sum of individual vibrations.

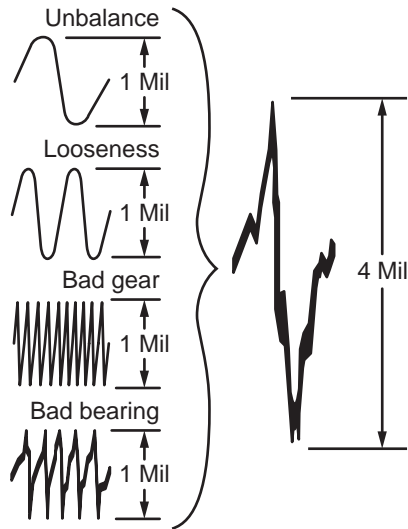


Figure 8. Peak-to-peak displacement.

### Isolation

Generator sets need no isolation for protection from self-induced vibrations. They easily withstand any vibrations which they create.

However, isolation is required if engine vibration must be separated from building structures, or if vibrations from nearby equipment are transmitted to inoperative generator sets. Caterpillar Generator Sets with isolation mounts between the generator set and base already satisfy these requirements. Running units are rarely affected by exterior vibrations. Methods of isolation are the same for external- or self-generated vibrations.

### Isolation Location

Several commercial isolators provide various degrees of isolation. Generally, the lower the natural frequency of the isolator, the greater the deflection (soft) and more effective the isolation. Weight of generator sets can be unequally balanced within the limits of the isolators. However, overloading will eliminate isolator benefits. Isolators are most effective when located under generator mounting and engine front support (see Figure 9). If additional support is desired, place an isolator midway between front and rear mounts and under radiator.

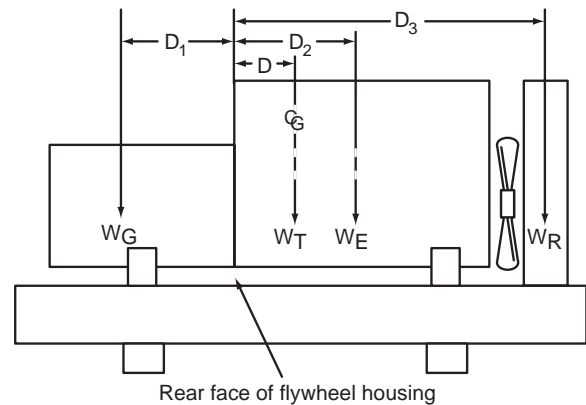


Figure 9. References for distances.

To apply isolators, wet weight and center of gravity of the assembled unit must be established. Assuming engine and generator are assembled to a base, wet weight ( $W_T$ ) and assembled center of gravity can be calculated. A common reference is needed (see Figure 9). 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.

$$W_T (D) = W_E (D_2) - W_G (D_1) + W_R (D_3)$$

$$D = \frac{W_E (D_2) - W_G (D_1) + W_R (D_3)}{W_T}$$

$W_T$  = wet weight

$W_E$  = engine weight

$W_G$  = generator weight

$W_R$  = radiator weight

$D_{1,2,3}$  = distances

If additional equipment is added, the process is repeated to determine a new center of gravity.

Having established center of gravity for the total unit (see Figure 10), loading on each pair of isolators is determined by:

$$S_1 = W_T \frac{B}{C} \quad S_2 = W_T \frac{A}{C}$$

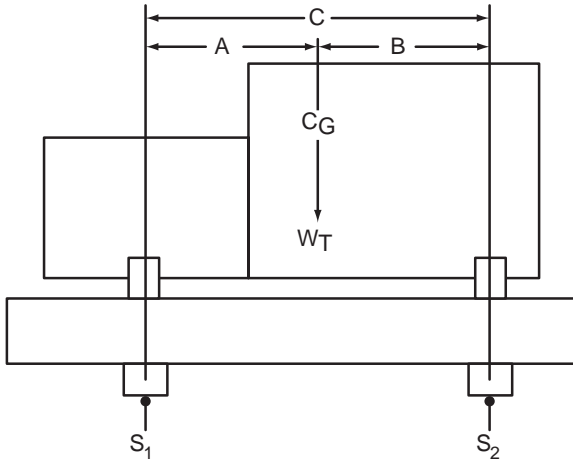


Figure 10. Determination of isolator location.

Isolators are sized to have natural frequencies far removed from engine exciting frequencies. If these frequencies were similar, the entire unit would resonate. The transmissibility chart in Figure 11 depicts this condition. It also shows the significant improvement caused by decreasing the mounting natural frequency to allow a ratio increase above  $\sqrt{2}$ , or 1.414.

### Isolation Methods

Vibration is reduced by commercially available fabricated isolators or bulk isolators. Both techniques utilize static deflection, with increased deflection resulting in greater isolation. Although internal damping of various materials cause performance differences, the vibration chart in Figure 12 describes the general effect deflection has on isolation. By using engine rpm as the nominal vibration frequency, magnitude of compression on isolating materials can be estimated.

The unit can be separated from supporting surfaces by these *soft* commercial devices, i.e., those which deflect under the static weight. Mounting rails or fabricated bases withstand torque reactions without uniform support from the isolators.

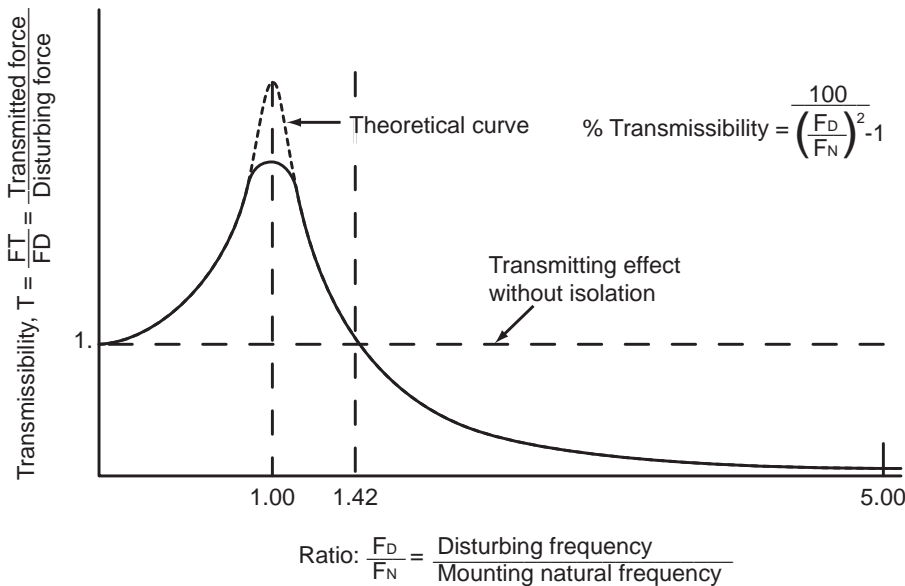


Figure 11. Transmissibility chart.

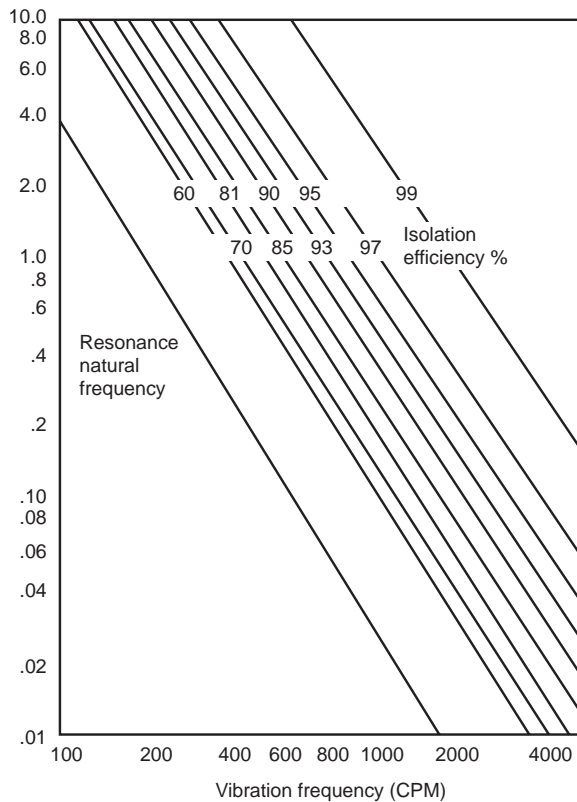


Figure 12. Basic vibration chart.

Piping connected to generator sets requires isolation, particularly when generator sets mount on spring isolators. Fuel and water lines, exhaust pipes, and conduit could otherwise transmit vibrations long distances. Isolator pipe hangers, if used, should have springs to attenuate low frequencies, and rubber or cork to minimize high frequency transmissions. To prevent buildup of resonant pipe vibrations, support long piping runs at unequal distances (see Figure 13).

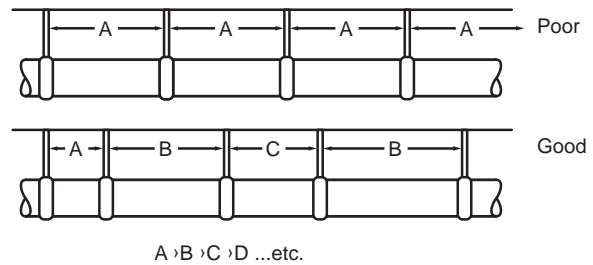


Figure 13. Pipe runs support.

### Rubber

Rubber isolators are adequate for applications where vibration control is not severe. By careful selection, isolation of 90% is possible. They isolate noise created by transmission of vibratory forces. Avoid using rubber isolators with natural frequencies near engine excitation frequencies.

Adding rubber plates beneath spring isolators block high frequency vibrations transmitted through the spring. These vibrations are not harmful but cause annoying noise.

### Spring

The most effective isolators are of steel spring design. They isolate over 96% of all vibrations and noise transmitted from rotating machinery to the foundation or mounting surface. Conversely, isolators can absorb disturbances generated by *adjacent* machinery and prevent damage from being transmitted to idle equipment.

Spring-type linear vibration isolators are available as attachments (not installed) for all generator sets to be used in stationary applications. These isolators permit mounting the generator set on a surface capable of supporting only the static load.

A detail of a spring-type isolator shows the addition of thrust blocks to restrict lateral movement without interfering with the spring function (see Figure 14).

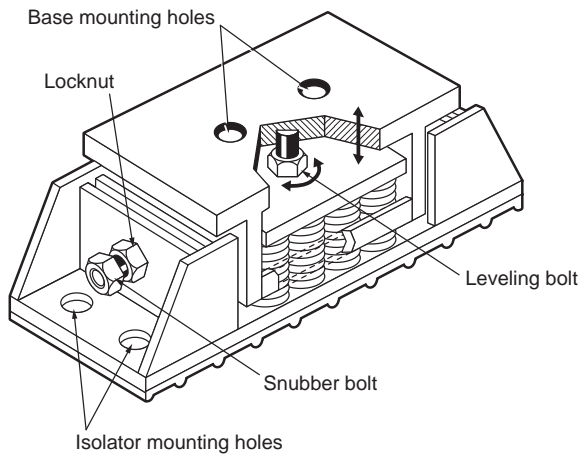


Figure 14. Vibration isolator.

No allowance for torque or vibratory loads is required. As with direct mountings, no anchor bolting is usually required. However, when operating in parallel, vertical restraints are recommended and the isolator firmly fastened to the foundation. Spring isolators are available with snubber for use when engines are side loaded or located on moving surfaces.

### Gravel or Sand

Bulk isolating materials are used between the foundation and supporting surface but are not as foolproof as spring or rubber types.

Isolation of block foundations may be accomplished by 200 to 250 mm (8 to 10 in.) of wet gravel or sand in the bed of the foundation pit (see Figure 15). Sand and gravel can reduce engine vibration one-third to one-half. The isolating value of gravel is somewhat greater than sand. To minimize settling of the foundation, gravel or sand must be thoroughly tamped before pouring the concrete block. Make the foundation pit slightly longer and wider than the foundation block base. A wooden form the size and shape of the foundation is placed on the gravel or sand bed for pouring the concrete. After the form is removed, the isolating material is placed around foundation sides, completely isolating the foundation from surrounding earth.

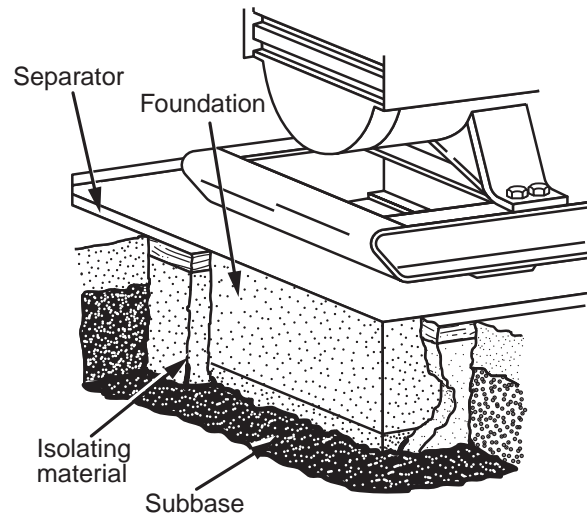


Figure 15. Bulk isolation.

### Other Isolation Methods

Fiberglass, felt, composition, and flat rubber do little to isolate major vibration forces. 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 indiscriminately could force the system into resonance.

Cork is not effective with disturbing frequencies below 1800 cps and, if not kept dry, will rot. It is seldom used with modern generator sets, but is used to separate engine foundations and surrounding floor because of resistance to oil, acid, or temperature changes between  $-18^{\circ}$  and  $93^{\circ}\text{C}$  ( $0^{\circ}$  and  $200^{\circ}\text{F}$ ).

If no isolation is required, the generator set may rest directly on the mounting surface. Factory assembled units are dynamically balanced and theoretically there is no dynamic load. Practically, the surface must support 25% more than the static weight of the unit to withstand torque and vibratory loads. Unless the engine is driving equipment which impose side loads, no anchor bolting is required. This normally applies to all non-parallel generator set mountings. Thin rubber or composition pads minimize the unit's tendency to creep or fret foundation surfaces.

Stationary equipment isolators are used when the package will not be transported, (with isolators attached), or will not experience seismic shock. The isolators contain thrust blocks to control lateral movement.

### External Isolation

Piping connected to generator sets requires isolation, particularly when generator sets are mounted on spring isolators. Fuel and water lines, exhaust pipes, and conduit could otherwise transmit vibrations long distances.

If isolator pipe hangers are used they should have springs to attenuate low frequencies, and rubber or cork to minimize high frequency transmissions.

To prevent buildup of resonant pipe vibrations, long support piping should run at unequal distances (see Figure 13).

### Mobile Equipment

Isolation from a movable platform is desirable to:

1. Reduce vibration.
2. Reduce noise.
3. Prevent torque loading on generator sets caused by platforms or trailer beds.

Mobile equipment isolators have the same features as the stationary equipment isolators. They also have built in restraints capable of withstanding a + 2 G vertical load and + 3 G horizontal load under low cyclic conditions (see Figure 16). They are suitable for use with mobile platforms, but are not generally intended to handle seismic shock.

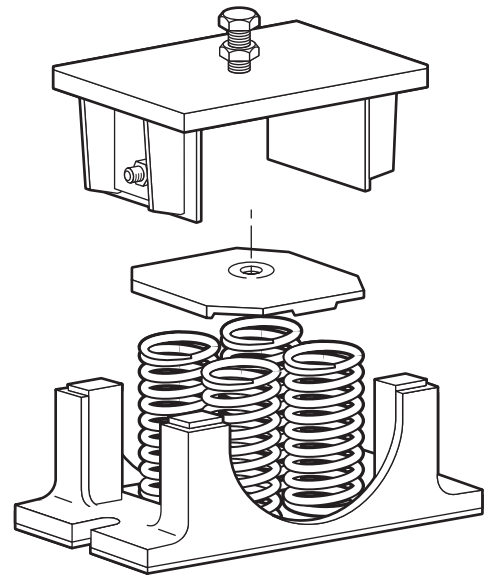


Figure 16. Isolator for mobile equipment.

Vibration carried throughout an enclosure causes early failure of auxiliary equipment. Relays, switches, gauges, and piping are adversely affected.

Noise, while normally only annoying, can attain levels objectionable to owners and operators. If operating near property lines, noise could exceed local ordinances.

Perhaps the most important reason to isolate mobile equipment is to avoid bending of the generator set by movement of the subbase. Unless the platform or trailer bed is extremely rigid, the generator set must not be bolted to it. Deflection of the bed would be transmitted to the engine, causing block bending and possible crankshaft and bearing failures.

Lateral movement of the generator set must be minimized as the trailer is transported. This can be achieved simply by blocking the unit off the isolators during the move. If not practical, snubbers can confine vertical and horizontal movement.

## Trailer Units

Lateral movement of the generator set must be minimized as the trailer is transported. This can be achieved simply by:

- Blocking the unit off the isolators during the move
- Using snubbers to confine vertical and horizontal movement

A spring-type isolator with the addition of thrust blocks will restrict lateral movement without interfering with spring function.

## Seismic Vibration

Seismic shocks are insufficient to harm generator sets resting on the floor. However, isolation devices, particularly spring isolators, amplify small movement generated by earthquakes to levels which would damage equipment. Special isolators incorporating seismic restraining or damping devices are available, but exact requirements must be reviewed by the isolator supplier. Isolators anticipating seismic shock are bolted to the equipment base and the floor. Positive stops are added to limit motion in all directions. Attached piping and auxiliary equipment supports must also tolerate relative movement.

## Torsional Vibration

Torsional vibrations occur as subjects, such as an engine crankshaft, twist and recover. Standard generator set components withstand normal stresses caused by combustion forces and torque reactions. A generator set must prevent the natural frequency of the drive train from approaching the unit's operating speed. Failure of crankshaft, couplings, gears or bearings may result without this attention.

Torsional vibrations originate with the piston power stroke. The simplified drive train in Figure 17 illustrates relationship of shaft diameter, length, and inertia on the natural system frequency.

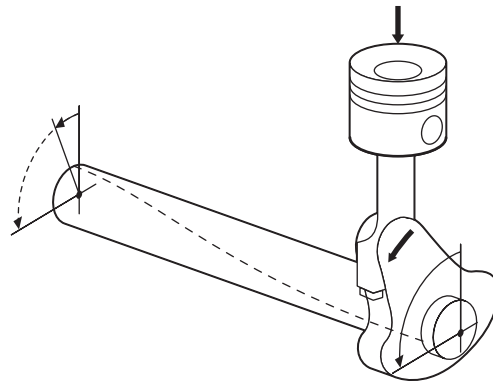


Figure 17. Torsional vibrations.

Even though generator sets, factory packaged on Caterpillar designed bases, avoid critical speeds where resonant conditions occur, all applications, whether packaged by Caterpillar or others, require a torsional vibration analysis. This assures compatibility of the engine and driven equipment. It must be performed by either the customer or by Caterpillar, depending on the customer's preference. Customer performed analyses are subject to Caterpillar review and approval and Caterpillar does not assume responsibility for analysis performed by others without the approval. Without the approval, no warranty can be claimed from Caterpillar. For Caterpillar-performed analyses, a complete set of technical data (see below) must be submitted to Caterpillar before calculations are undertaken. The report will include a mathematical determination of the natural frequency, critical speeds, relative amplitudes of angular displacement, and approximate nodal locations of the complete elastic system (both engine and driven equipment). See the TIR section of the price list for ordering information.

*Note: Consult factory on compound installations. There may be additional charges for analyses of applications where more than one engine drives a single load. A separate torsional analysis is also required for each engine with different driven equipment in multiple engine installations.*

Technical data required:

1. The operating speed range.
2. Load demand curve on generator sets which have a load dependent variable rigidity coupling.
3. General arrangement drawing or sketch of complete system, including data on equipment driven from front of engine.
4. With driven equipment on both ends of the engine, the power demand of each end is required. In addition, simultaneous front and rear power (maximum engine load) is required.
5. Make, model  $WR^2$  (rotational inertia), and torsional rigidity on all couplings used between the engine and driven equipment.
6.  $WR^2$  of each rotating mass. Weight of each reciprocating mass.
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.
8. The ratio of the speed reducer or increaser. The  $WR^2$  and rigidity submitted for a speed reducer or increaser should state whether or not they have been adjusted to engine speed.
9. For reciprocating compressor applications, a harmonic analysis of the compressor torque curve under various load conditions. If not available, a torque curve of the compressor under each load condition through one compressor cycle. The  $WR^2$  of all available flywheels for the compressor.

## Couplings

A coupling must be torsionally compatible with engine and driven load so that torsional vibration amplitudes are kept within acceptable limits. A mathematical study called a torsional vibration analysis should be done on any combination of engine-driveline-load for which successful experience doesn't already exist. A coupling with the wrong torsional stiffness can cause serious damage to engine or driven equipment.

All couplings have certain operating ranges of misalignment, and the manufacturers should be contacted for this information.

Some drives, such as U-joint couplings, have different operating angle limits for different speeds.

As a general rule, the angle should be the same on each end of the shaft (see Figure 18). The yokes must be properly aligned and sliding spline connections should move freely. If there is no angle at all, the bearings will brinell due to lack of movement.

## Cyclic Irregularity

Cyclic irregularity is a nondimensional ratio describing degree of crankshaft twist occurring between two successive firings of cylinders during steady-state operation. Formulas to represent this movement were derived before modern instrumentation allowed measurement. The ratio is expressed as:

$$\text{Cyclic irregularity} = \frac{\text{rpm (maximum)} - \text{rpm (minimum)}}{\text{rpm (average)}}$$

System speed varies with connected rotating mass. Cyclic irregularity differs, therefore, for a basic engine, one driving a generator, or additional equipment.

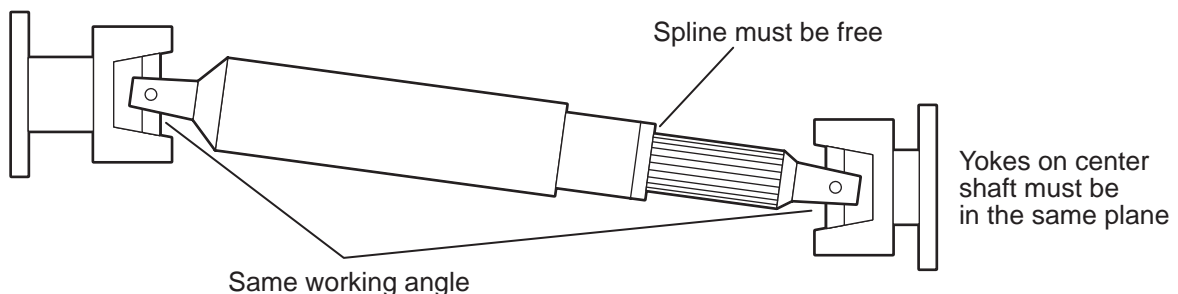


Figure 18. Universal joint shaft drive.

This ratio compares merits of large slow speed engines which were custom made, but has little value applied to modern medium speed engines.

## Out-of-Balance Driven Equipment

The engine itself is designed and built to run very smoothly. Objectionable vibration generally arises from either poor driveline component match to the engine or unbalance of the driven equipment. Reciprocating equipment with large imbalances, for example, can cause premature failure of the mounting structure or undesirable vibration even though the unit is properly mounted and isolated from the engine.

Even though the engine and the driven load are in balance, it is also possible to encounter undesirable and damaging vibration as a result of the driving or connecting equipment having a misalignment or out-of-balance condition. Long shafts, drives, gear assemblies, clutches, or any type of coupling where misalignment, out-of-balance, or mass shifting may occur, are probable sources of vibration.

## Alignment

An unsatisfactory engine mounting nearly always results in alignment problems between the engine and the driven machinery. Assuming that failure of the driven equipment does not occur first, the forces or loads transmitted to the engine in the form of pounding, twisting, flexing, or thrust could result in engine crankshaft and bearing failure. Costly failures of this nature can be avoided if, at the design and installation stage, the importance of proper alignment between the engine and driven load and adequate mounting to maintain alignment is considered.

If this is possible, a suitable flexible coupling must be incorporated into the drive train to compensate for misalignment.

Good alignment practices include proper shimming, correct torque on hold-down bolts, accurate dial indicator usage, allowances for bearing clearances, thermal growth, and accounting for other characteristics of the engine.

## Procedures

Perform final alignment after all major equipment has been installed on the base. Engines should be filled with oil and water and ready to operate. The temperature of the engine and driven equipment must be equalized.

Complete step-by-step alignment procedures are described in Caterpillar Service Special Instructions, Form No. SEHS7073, *Alignment of Two-bearing Generators* or SEHS7654, *Alignment-General Instructions*.

Place driven equipment in its final position as closely as possible without taking indicator readings. There should be a minimum of 0.76 mm (0.031 in.) and a maximum of 1.5 mm (0.060 in.) of shims under each mounting surface of the driven equipment.

Position driven equipment, using the leveling and alignment screws.

For cold alignment, the generator is mounted higher than the engine to compensate for thermal growth, bearing clearances, and flywheel droop.

## 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.

## Runout

Inaccurate flanges cause apparent misalignment and make accurate alignment impossible.

Face runout refers to the distance the hub face is out of perpendicular to the shaft centerline (see Figure 19).

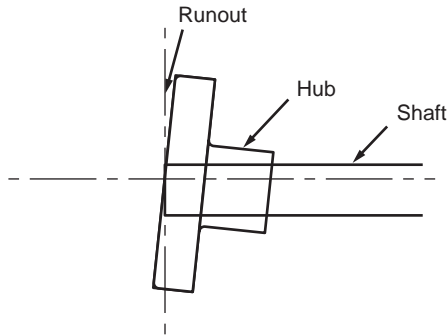


Figure 19. Face runout.

Bore runout refers to the distance the driving bore of a hub is out of parallel with the shaft centerline (see Figure 20).

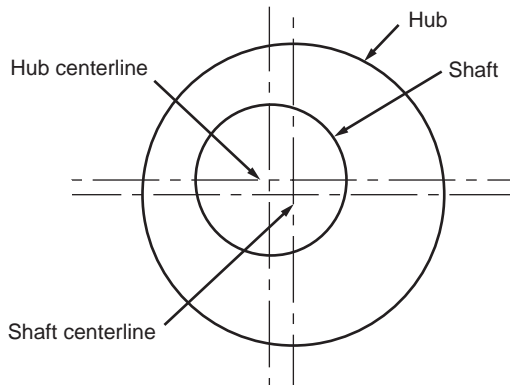


Figure 20. Bore runout.

The face and bore runouts of flywheel, clutch or coupling, driven members, and hubs must be checked when inconsistent alignment results occur. Face or bore errors must be corrected. Bore-to-pilot diameter runout error should not be more than 0.05 mm (0.002 in.) on the flywheel and 0.13 mm (0.005 in.) on adapters bolted to the flywheel. Flange face runout should not be more than 0.05 mm (0.002 in.).

## Parallel/Angular Alignment

Parallel or bore misalignment occurs when centerlines of driven equipment and engine are parallel but not in the same plane (see Figure 21).

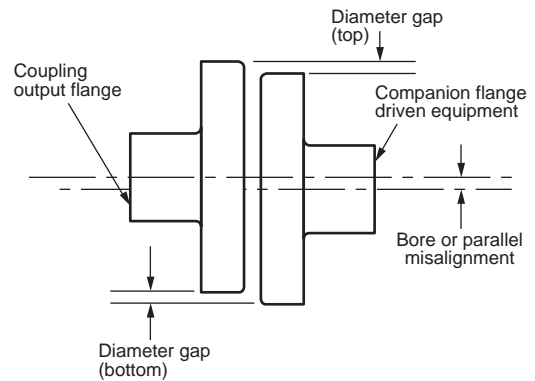


Figure 21. Parallel misalignment.

Angular or face misalignment occurs when centerlines of driven equipment and engines are not parallel (see Figure 22).

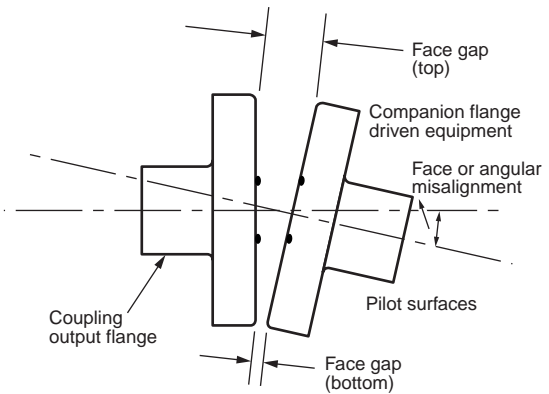


Figure 22. Angular misalignment.

Figure 23 illustrates that misalignment can occur in more than one plane. For this reason, alignment readings must be taken at 90 degree intervals as the units are rotated.

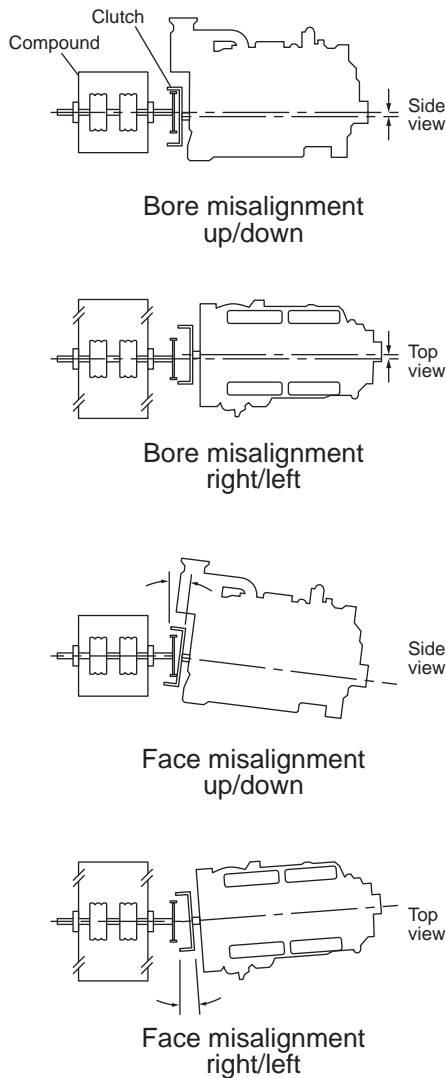


Figure 23. Misalignment.

### Bearing Clearance

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 (see Figure 24).

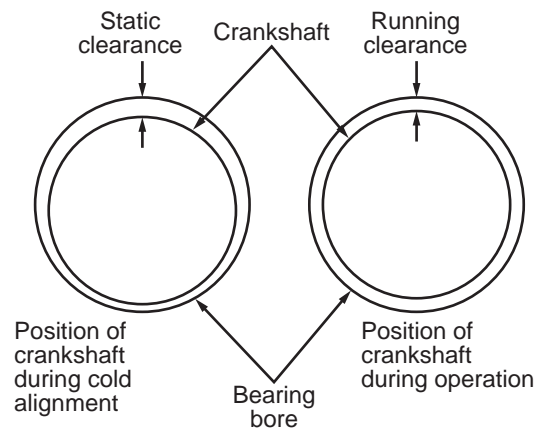


Figure 24. Bearing alignment.

Generally, driven equipment will have ball or roller bearings which do not change their rotational axis between static and running conditions.

### Flywheel Sag

With the engine not running, the flywheel causes the crankshaft to bend. The weight of the coupling increases the bending. This results in the checking surface (pilot bore or outside flywheel diameter) rotating below centerline of the crankshaft bearings. For this reason, Caterpillar recommends alignment checks be performed with the coupling in place (see Figure 25).

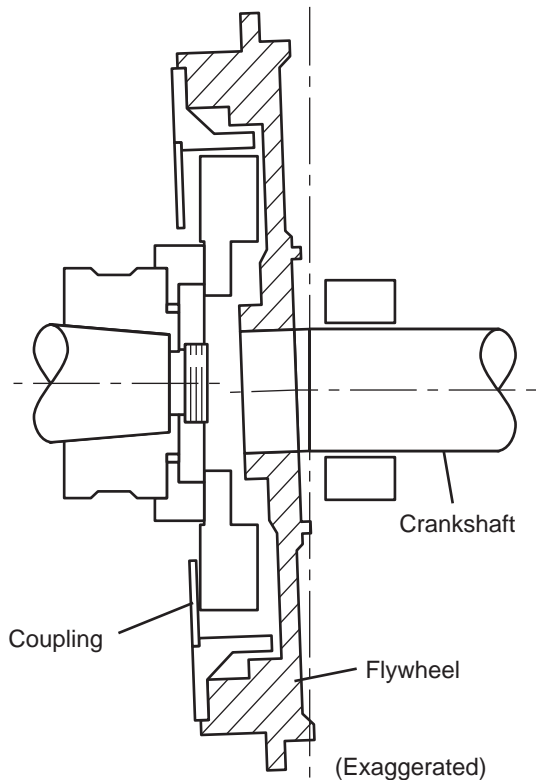


Figure 25. Crankshaft bending.

The values for the maximum crankshaft side loads for the specific engine can be found in TMI.

To calculate the actual load on the shaft, proposed multiple direction drive loads must be resolved into one single direction and load by vector addition.

Side Load:

$$P_{SL} (N) = \frac{kW \times 19099000}{rpm \times pd}$$

$$P_{SL} (lb) = \frac{hp \times 126000}{rpm \times pd}$$

Where:

$P_{SL}$  = force of side load (N), (lb)

pd = pitch diameter of drive sprocket or main belt diameter

D = distance from front or rear side load center to index location (rear face of flywheel housing unless otherwise specified)

*Note: The calculated side load must not exceed the side load limit found in the selected drive type data set for the distance (D) calculated.*

## Thermal Growth

The change in distance between mounting holes due to thermal growth of the engine must be considered when designing the base.

Cast iron has a coefficient of expansion of 0.0000055, and that of steel is 0.0000063. This means that the block of an engine 238.8 cm (94 in.) in length will grow 0.197 cm (0.0776 in.) if its temperature is increased from 10°C (50°F) to 98.8°C (200°F). Using 0.0000063 as the plate steel coefficient of expansion, a steel weldment of 238.8 cm (94 in.) will grow 0.226 cm (0.089 in.) through the same temperature range. The small difference in growth between the block and the lubricating oil pan is compensated for in the design of the engine by making the holes in the flange of the attached component (rails) larger than the attaching bolts.

Due to the growth resulting from thermal expansion, the engine must not be dowel located in more than one location. Failure to do this will cause unnecessary stresses in the engine and base as the engine tries to grow due to temperature changes. It is recommended that a dowel locator be used on one engine mounting rail located at the end with the driven equipment. This will force all thermal growth away from the engine coupling. Clearance between the mounting bolts and the mounting brackets on the base will then allow slip to compensate for thermal growth.

## Shimming

Shims can be used to get correct alignment between the engine and the driven unit (see Figure 26). Depending on the application, shims are required under the engine, under the driven unit, or under both the engine and driven unit.

Shims should be of nonrushing material. The most commonly used shim materials are metal and poured resin shims.

Under no circumstances should lead be used as a shim material. Lead is easily deformed under weight and vibration and has poor support characteristics.

Handle shims carefully. Use only complete width and length shims. Do not use partial shims trimmed to fit. After alignment, each mounting surface must carry its portion of the load.

All mounting points must bear equally on the mounting structure. To determine if shims are required, set the engine on the mounting structure but do not attempt to secure it by bolting it in place. Using a feeler gauge, check all mounting points for clearance between the mounting point and the base. If clearance exists which exceeds 0.127 mm (0.005 in.) compensation must be provided.

If the mounting base is a rigid steel structure, the areas where the engine mounts make contact may be machined to bring them all into a true plane. If this is impractical, shims should be used.

Shim packs under all equipment should be 5 mm (0.2 in.) minimum thickness to permit later corrections requiring the removal of shims, if necessary.

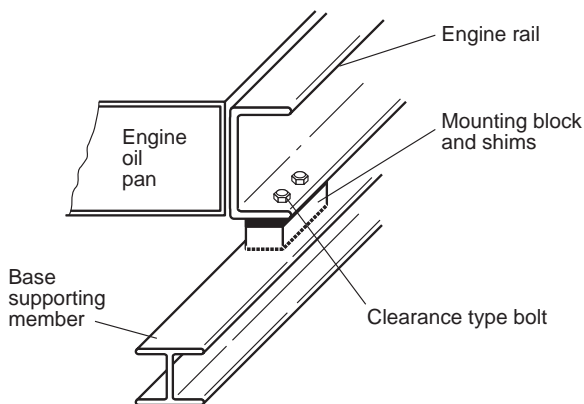


Figure 26. Mounting with shims.

Before the engine and driven equipment can be aligned, each foot must carry its portion of the load. Failure to do this can result not only in misalignment, but also in springing of the substructure causing resonant vibrations, high stress in welds or base metal, and high twisting forces in the engine or generator.

This same requirement for a true plane (flat) mounting is also necessary for most driven equipment. If specific instructions are not provided by the driven equipment manufacturer, the same principles as recommended for the engine can be applied.

**Important:** When metal shims are used between the mounting pads or feet and the base or mounting surface, the mounting surfaces must be flat, free of burrs, and parallel to the bottom surface of the mounting pads or feet.

### Poured Resin Shims

After the engine and the driven equipment have been aligned, poured resin shim material can be used between the mounting pads of the engine mounting rails and the engine bed. When using poured resin shim material, always follow the manufacturer's installation recommendations.

Follow these guidelines:

- Use foam rubber strips of the appropriate thickness to form the dams for pouring the shim material.
- Do not pour shim material inboard of the machined pad on the bottom of the mounting rail.

Shim material can be poured the full length of the mounting rail (a continuous pour) or it can be poured only at the mounting pad locations (an interrupted pour). If an interrupted pour is used, the minimum area of shim material must be 29,000 mm<sup>2</sup> (45 in.<sup>2</sup>) per mounting bolt. If the mounting rails have mounting pads only at the bolt hole locations, foam rubber strips must be installed on both sides of each pad on all pads forward of the flywheel housing to provide for expansion. These expansion strips permit thermal expansion of the mounting rails at operating temperature. On engines whose mounting rails have full length mounting pads, no expansion strips are necessary.

Note: When realigning an existing installation, full length foam rubber strips for a continuous pour can often be difficult because of limited access. It is recommended that a full length continuous pour be used only for new installations where the engine can be raised to install the foam rubber strips.

Do not pour the shim material thicker or thinner than the manufacturer's recommendation. Use steel spacers if the clearance between the bottom of the mounting pad and the top of the engine bed is more than the maximum allowable thickness of the shim material. The spacers must be a minimum of 29,000 m<sup>3</sup> (45 in.<sup>2</sup>) in area per mounting bolt.

Before pouring the shim material, install all anchor bolts finger tight. Put sealing material around the bolt at the bottom of the mounting pad to prevent the shim material from filling the bolt holes in the mounting rails. If shim material is allowed to enter the bolt holes in the mounting rails, thermal expansion of the rails will be prevented.

After the shim material has sufficiently hardened according to the manufacturer's specification, the anchor bolts can be tightened to a torque of 490 N•m (360 lb-ft). Use two nuts on each bolt.

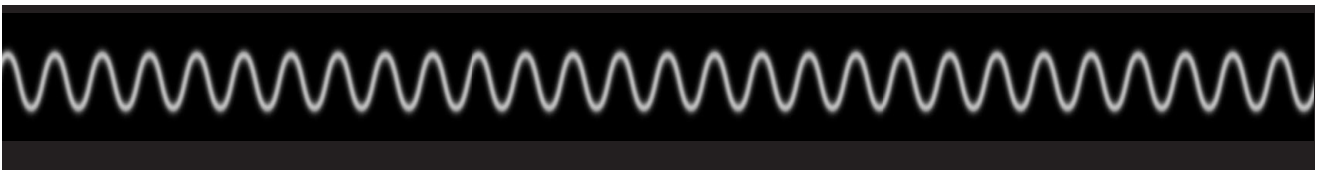


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Materials and specifications are subject to change without notice.

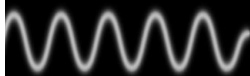




# **Electric Power Application and Installation Guide**

## **Noise**

LEBX0031



WHERE THE WORLD TURNS FOR POWER



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# Definitions

Noise can be defined as all unwanted sounds. Music is sound and can be pleasant to some people and noise to others. Noise and sound are often used to describe the same physical characteristics. Noise is generally random in nature without distinct frequency components. Noise can produce undesirable psychological effects on people and physical damage to the ears. Noise can be annoying to affect verbal communications at work and away. At times, it may impact behavior, including short term and long term hearing loss, muscle tension, respiratory reflexes, stress level, heart function, etc. Recognizing this, many governmental agencies around the world have established regulatory limits for various levels of noise.

The noise from the engine comes mainly from combustion, mechanical, for the exhaust and air intake sources.

Frequency of sound refers to the rapidity or cycles of an oscillation in a unit time. The conventional unit is Hertz (Hz) — one Hz being one cycle per second.

## Sound Waves and Measurement

As sound waves radiate, their strength diminishes (see Table 1). As distance traveled doubles, the wave amplitude is reduced by one-half. This rule applies if the first measuring point is at least two or three times the largest dimension of the noise source, usually about three feet.

Distance	Sound Strength
X	100%
2X	50%
4X	25%

Table 1. Distance vs. wave amplitude.

Sound waves impinging on a microphone produce voltages proportional to sound pressures. The signals measure amplitude or strength, of the *sound pressure* waves. Amplitude and frequency are the only sound properties measurable using ordinary techniques.

## Sound Pressure

The extensive audible range of sound complicates noise ratings. The human ear hears pressure levels that are about 100,000 times stronger than the lowest pressure it is affected by. For this reason, measuring instruments have extraordinary range and are scaled in decibels (dB). The decibel scale is logarithmic, which allows the wide range of sound pressures to be measured in only two- or three-digit numbers.

Sound Pressure Level (SPL) in dB =

$$20 \log_{10} \times \frac{\text{Measured pressure}}{\text{Reference pressure}}$$

The reference pressure is taken as: 20 μPa or 2 x 10<sup>-4</sup> microbars = 0 dB. The relationship between μPa and dB is that when multiplying the sound pressure (μPa) by 10, 20 dB is added to the dB level. Decibel (dB), is the relative measurement of amplitude of sound. Sound is a pressure which makes the membrane in the human ear deflect. The softest pressure the human ear can hear is 20 μPa (1 atmospheric pressure = 1 bar = 100 kPa = 14.5 psi) but the ear can take pressures up to more than 1 million times higher.

## Frequency — Weighting Networks

The ear is more sensitive to high frequencies than low frequencies. To approximate the effect of sound on the average person, measurements are weighted according to frequencies corresponding to the sensitivity of the ear. Loudness can be measured by filtering the microphone signal to reduce the strength of the low frequency signals and give more weight to frequencies in the 5,000-10,000 Hz range. The signal from the measuring microphone is fed to an amplifier, then to an attenuator, which is calibrated in decibels. The signal is then fed to one of four weighting networks, referred to as A, B, C, and D. The response of the network chosen modifies the input signal accordingly.

The most commonly used network is weighting A (A-scale), and it is known as dBA or dB(A).

Figure 1 shows the response characteristics for an “A” filter. The result of adjustments throughout the frequency range is a total decibel rating with a correction for various frequencies to approximate ear’s sensitivity.

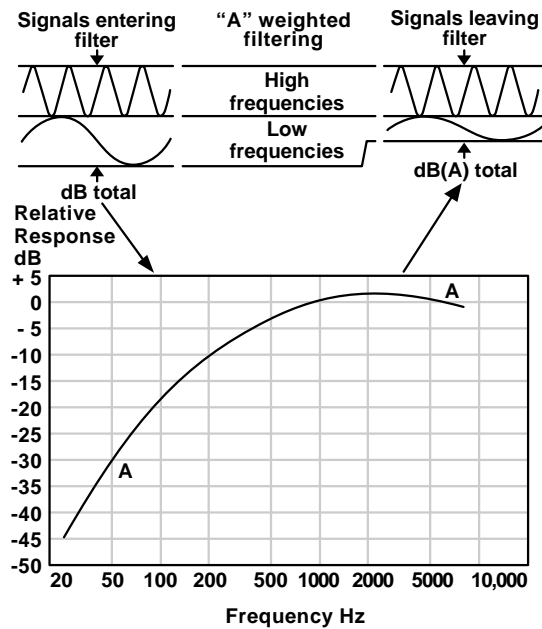


Figure 1. Response characteristics of “A” filter.

### Octave Band Levels

More detail is required of the frequency distribution of a noise than provided by an A-weighted measurement. Measurements are made with filters subdividing sounds over the entire audible range into standardized frequency bands, permitting the pressure levels of only the sound within each subdivision to be measured. Each filter spans an octave; that is, the upper frequency limit is twice the lower limit as shown in Figure 2. Sound levels in each octave are measured in decibels and are referred to as octave band levels.

Band Designation (Center Frequency)	Band Limits
8000 Hz	11300 Hz
4000	5650
2000	2830
1000	1415
500	707
250	353
125	176
63 Hz	88
	44 Hz

Figure 2. Standard octave bands (ANSI Standard S1.11 IEC 225).

### Loudness

The human ear does not use sound pressure decibels to judge loudness. Rating noise loudness is a complex operation because human hearing is also frequency sensitive.

Sounds with frequencies in the 5,000-10,000 Hz range are the easiest to hear; sounds with very low frequencies are the hardest. Hearing loss from exposure to noise is frequency sensitive.

### Direction of Sound

Sound is mostly directional, meaning that the sound tends to move more in one direction than another.

The contour of the sound wave can be complex. By measuring the sound pressure level three dimensionally around the engine, the contour can be determined (see Figure 3).

It is not only the source of the sound which will give the direction, but also any kind of reflective surface in the area of the engine, i.e. floor, walls or ceiling.

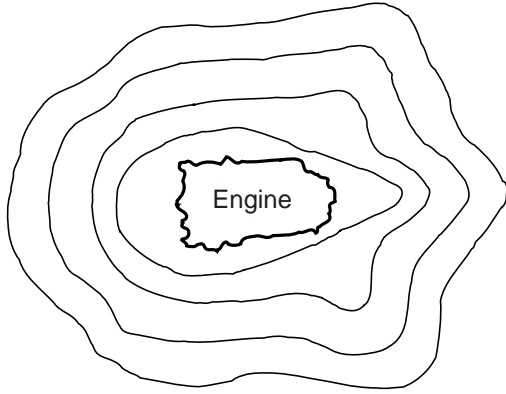


Figure 3. Contour of a sound wave.

### Noise Addition

When standing by an engine, the noise heard from other engines operating in the same area will depend on the spacing of the engines and where the person is in relation to the spacing.

A chart showing the combined effect of up to ten equal sound sources is shown in Figure 4.

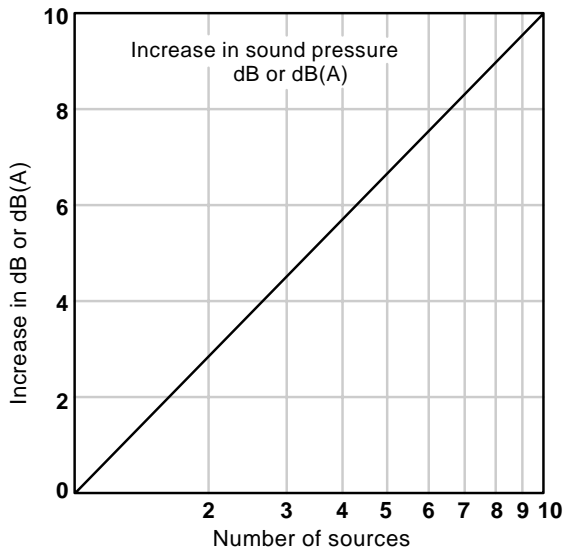


Figure 4. Addition of equal sounds.

Figure 5 shows the versatility of the decibel system. Although calculations are made on the basis of sound power, the system uses measured or calculated sound pressures. Use the difference in the pressure levels of two sounds to find how their combined level exceeds the higher of the two. First adjust the levels for the distances from the source to the spot where the noises are being added. To add a third level, use the same process to combine it with the total of the first two.

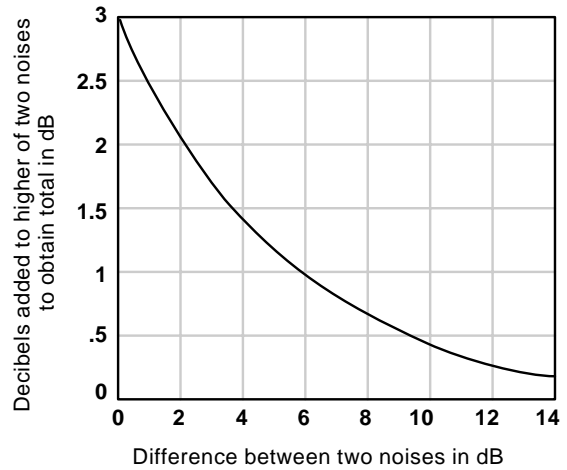


Figure 5. Addition of unequal sounds.

### Sound Level Conversions

Sound level information is presented both in terms of sound power level, SWL, dB(A), and sound pressure level, SPL, dB(A). SWL is the total sound power being radiated from a source and its magnitude is independent of the distance from the source. Relative loudness comparison between engines is simply a comparison of their sound power levels at equivalent operating conditions.

When the sound power level (SWL) is known, the sound pressure level (SPL) at any distance from a point source (such as exhaust noise) can be calculated.

The equation for determining the sound pressure level of exhaust noise without any correction for ambient temperature and pressure, is:

$$\text{Sound Pressure Level, SPL dB(A)} = \text{Sound Power Level, SWL dB(A)} - 10 \times \text{Log}_{10} (C\pi D^2)$$

Where C = 2 For exhaust source adjacent to a flat surface, such as a horizontal exhaust pipe adjacent to a flat roof.

or C = 4 For exhaust source some distance from surrounding surfaces, such as a vertical exhaust stack some distance above roof.

D = Distance from exhaust noise source (m).

$$\text{For } C = 4 = \text{SPL} = \text{SWL} - 20 \text{Log}_{10} D - 10.99$$

SPL measurement requires only a simple sound level meter. However, this being the sum of sound waves arriving from every direction, depends on the acoustic characteristics of the environment and varies with position relative to the noise source. SPL cannot be used to describe the strength of a noise source without specifying relative position and room acoustic properties of the test environment. A disadvantage is that sound pressure level conversion is valid for a point source only. It cannot be used for mechanical noise since the source (overall engine) is quite large.

If the sound pressure level of a point source at some distance is known, the sound pressure level at another distance can be calculated using this formula:

$$\text{SPL}_2 = \text{SPL}_1 - 20 \times \text{Log}_{10} (D_2 \div D_1)$$

Where: SPL<sub>1</sub> = known sound pressure level, dB(A)

SPL<sub>2</sub> = desired sound pressure level, dB(A)

D<sub>1</sub> = known distance, m (ft)

D<sub>2</sub> = desired distance, m (ft)

## Noise Exposure

As mentioned before, exposure to excessive noise causes permanent hearing damage and adversely affects working efficiency and comfort. Recognizing this, the U.S. Government created the Occupational Safety and Health Act (OSHA) which established limits for industrial environments.

When an individual's daily noise exposure, designated D(8), is composed of two or more periods of noise at different levels, the combined effect is calculated by:  $D(8) = (C_1/T_1) + (C_2/T_2) + \dots + (C_n/T_n)$ . Where C<sub>n</sub> is duration of exposure at a specified sound level and T<sub>n</sub> is total time of exposure permitted at a specified sound level (see Table 2). The noise exposure is acceptable when D(8) is equal to or less than 1.

Duration of Daily Exposure (hours)	Allowable Level dB(A)
8	90
6	92
4	95
3	97
2	100
1.5	102
1	105
0.5	110
0.25	115

Table 2. Permissible noise exposures.

## TMI Data

TMI contains the specific noise values (SPL) for the specific engine at different ratings. Various definitions are used and most can be found under "HELP" in TMI.

### Free Field

Free field means that it is a 100% open area without any kind of sound reflections or other modifying factors.

## Sound Pressure Level, SPL, — Mechanical or Exhaust

Sound pressure level is presented under two index headings: mechanical or exhaust.

Over one thousand data points per engine are used to prepare this data. There are eight octave bands and one overall reading taken at four engine speeds, four loads, three distances, and four positions around the engine.

### Mechanical

Sound pressure level data is obtained by operating the engine in an open “free” field and recording sound pressure levels at a given distance. The data is recorded with the exhaust sound source isolated.

### Exhaust

Sound pressure level data is recorded with the mechanical sound source isolated.

### Measurements

The instrumentation used are Larsen/Davis and Hewlett-Packard. All measurements are for “without” radiator fan arrangements.

Tolerances for the overall and for the octave band data is shown below:

Overall	Plus or minus 2 dB(A)
60 Hz	Plus or minus 5 dB(A)
125 Hz	Plus or minus 5 dB(A)
250 Hz	Plus or minus 4 dB(A)
500 Hz	Plus or minus 3 dB(A)
1000 Hz	Plus or minus 2 dB(A)
2000 Hz	Plus or minus 2 dB(A)
4000 Hz	Plus or minus 2 dB(A)
8000 Hz	Plus or minus 2 dB(A)

The confidence level of the above data is 99.73%, which means that only 27 out of each 10,000 engines measured of the same configuration as listed could fall outside of the nominal values plus the tolerances shown for the same engine, the repeatability tolerance is  $\pm 1$  dB(A).

## Engine Installations

Engine packages include an engine and some piece of driven equipment, such as a generator or a compressor. Guidelines for installation design are provided, along with

information on using noise data on Caterpillar units from TMI.

Some installations require very little noise abatement (for example, a remote facility far from people). Very sensitive installations, on the other hand, may require extensive noise abatement measures. Because of the variety of noise criteria that may apply to a given site, it is impossible to provide a description of abatement measures meeting all site criteria. It is the responsibility of the facility designer to ensure that the specific criteria of the site are met.

It is strongly advised that a noise control expert be involved in the facility design process from the beginning if the engine unit is to be installed in a building or area that is noise sensitive. Since internal combustion engines produce high noise levels at low frequencies, many traditional noise control approaches are relatively ineffective. Every aspect of facility design must therefore be reviewed with special emphasis on low-frequency attenuation characteristics in order to meet site criteria.

A typical approach to designing an engine installation is as follows:

- *Recognize the special requirements of engine installations.* The first step is to become aware of the special noise characteristics of engine installations. Possible sources, paths, and receivers of large-engine noise are reviewed.
- *Identify site noise criteria.* For example, is the installation in a remote or a populated area? Is it within a building sensitive to noise (for example, a laboratory or a hospital)? What regulations, standards, or restrictions apply to noise? The noise criteria form an essential part of the design goals. Since criteria vary from site to site, this guide cannot identify all the criteria that apply to a particular site. However, some guidelines for site noise criteria are provided.
- *Identify and select appropriate noise abatement measures.* Guidelines for attenuation of noise, both through commercially available equipment and through facility construction, are provided.

# Noise Control

Noise can be either airborne or structure-borne transmitted. Airborne noise is transmitted through air. Structure-borne noise is vibration transmitted through a structure; typically supporting the engine. Noise control methods are different for the two sources. Noise control refers to appropriate technology used for noise attenuation to acceptable levels.

Noise criteria at various frequencies for typical areas are shown in Table 3.

## Airborne Noise Control

Airborne noise control is a straightforward and well-developed area compared with structure-borne noise control. There is abundant information available on sound absorption and transmission properties of common construction materials, and there are accepted and proven procedures for applying that information.

However, it is important to recognize that much of the conventional information and procedures were developed for higher-frequency noise, and thus may not be appropriate for engine units, which produce strong low-frequency acoustic energy. For example, structural and acoustic resonances (conditions of minimum dynamic stiffness) may coincide with pure-tone frequency components of the engine noise, resulting in very efficient transfer of energy. Conventional building acoustics generally is based on statistical descriptions of noise, and therefore does not address resonance effects.

For some installations, airborne noise must be controlled at several receiver points: inside the engine room; in other rooms in the building; and outside the building. The simplest way to reduce airborne noise within a building is through good building layout. Equipment rooms should be situated far from sensitive receiver locations in the building. This takes advantage of the fact that propagating sound energy diminishes with distance from the source. In addition, there are two other methods of controlling airborne noise: with high transmission loss walls and with absorption.

It is helpful to review some terminology before discussing the sound transmission characteristics of walls. The transmission loss (TL) of a partition is a measure of the ratio of energy incident on the wall to that transmitted through the wall, expressed in dB. The less relative sound transmitted through the wall, the higher the TL of the wall. TL is a function of frequency.

The sound transmission class (STC) of a partition is a single-number rating calculated from the partition TL. A reference contour is adjusted against the measured TL data, and the STC rating equals the value of the adjusted contour at 500 Hz. The STC rating does not include information in frequency bands below 125 Hz. This rating is useful for designing walls that provide insulation against the sounds of speech and music; it is inappropriate for industrial machinery with low-frequency energy such as engine units. TL data should be used instead, whenever possible.

Octave Bands in Cycles Per Second	31.5	63	125	250	500	1000	2000	4000	8000
Highly Critical Hospital or Residential Zone	71	63	44	37	35	34	33	33	33
Night, Residential	73	69	52	44	39	38	38	38	38
Day, Residential	76	71	59	50	44	43	43	43	43
Commercial	81	75	65	58	54	50	47	44	43
Industrial-Commercial	81	77	71	64	60	58	56	55	54
Industrial	87	85	81	75	71	70	68	66	66
Ear Damage Risk	112	108	100	95	94	94	94	94	94

Table 3. Noise criteria.

In typical partitions, sounds at higher frequencies are attenuated more than sounds at lower frequencies. The highest transmission loss values are found in cavity wall (two-leaf) constructions, where the two separate wall layers are well isolated. The transmission loss values increase with the masses of the individual leaves, the depth of the airspace, and the characteristics of any sound-absorptive material in the airspace.

It should be noted that noise leaks can severely degrade the performance of a partition. Materials are tested for their transmission loss characteristics in a controlled laboratory setting, with all edges sealed. But in typical construction, sound leaks may occur at the edges of the wall, at openings for pipes or electrical outlets, and across shared ceilings (so-called flanking paths). A wall with a leakage area equal to 0.01% of that of the wall area cannot exceed  $STC = 40$ , no matter how high the  $STC$  of the wall construction.

A partition may include elements with various transmission loss characteristics, for example, windows and doors. The transmission loss of the partition must be calculated taking all elements into consideration.

To estimate the total airborne noise transmission loss of a facility, subtract the noise value for each receiver from the estimated room-average sound pressure level. If there is more than one space, the sum of the individual contributions must not exceed the criterion.

### Mechanical Noise

Many techniques for isolating generator set vibrations are applicable to mechanical noise isolation. Modest noise reductions result from attention to noise sources, i.e., reducing fan speeds, coating casting areas, and ducting air flows. But for attenuation over 10 dB(A), units must be totally isolated. One effective method utilizes concrete blocks filled with sand to house the generator set. In addition, the unit must incorporate vibration isolation techniques. A rough guide comparing various isolation methods is illustrated in Figure 6.

Completely enclosed engines are impractical due to openings required for pipes, ducts, and ventilation. Enclosures with numerous openings rarely attain over 20 dB(A) attenuation.

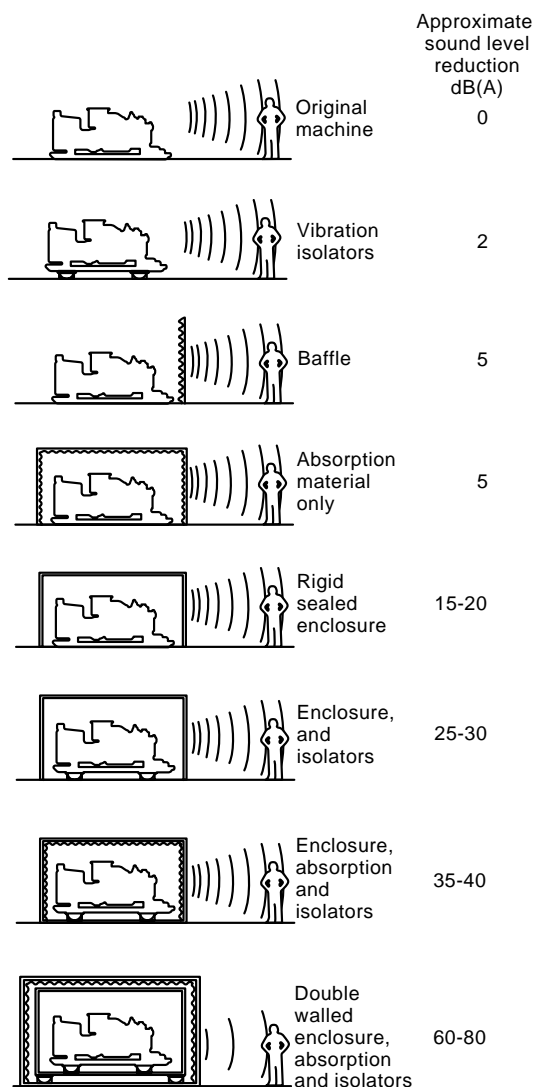


Figure 6. Illustration of isolation methods.

### Intake Noise

Intake noise attenuation is achieved through either air cleaner elements or intake silencers. Noise attenuation due to various air cleaners and silencers can be supplied by the component manufacturer.

## **Exhaust Noise**

Exhaust noise is typically airborne. Exhaust noise attenuation is commonly achieved with a silencer typically capable of reducing exhaust noise 15 dB(A) when measured 3.3 m (10 ft) perpendicular to the exhaust outlet. Locating it near the engine minimizes transmission of sound to the exhaust piping. Since the number of cylinders and engine speeds result in varied exhaust frequencies, specific effects of mufflers must be predicted by the muffler manufacturer.

## **Silencers/Mufflers**

Silencers are used to attenuate airborne noise in piping and duct systems. Their effectiveness generally is frequency sensitive, so it is essential that they be matched to the frequency content of the noise. There are two major categories of silencers; dissipative and reactive. Dissipative silencers use absorptive, fibrous material to dissipate energy as heat. They are effective only for high frequency applications (i.e., 500 to 8000 Hz).

Reactive silencers, on the other hand, use a change in cross-sectional area to reflect noise back to the source. They are typically used for low-frequency applications (such as internal combustion engines), and they may incorporate perforated tubes to increase broadband performance. The effectiveness of a reactive silencer depends on its diameter, volume, and overall design. Multi-chamber silencers provide maximum sound attenuation with some flow restriction. Straight-through silencers offer negligible flow restriction with slightly lower sound attenuation.

Stack silencers are designed to be inserted directly into a stack and withstand a harsh environment. Finally, some manufacturers offer combination heat-recovery silencers for hot gas exhaust.

Most manufacturers offer silencer dynamic insertion loss (DIL) information in octave bands from 63 to 8000 Hz, tested in accordance with ASTM E-477. DIL is the difference in sound level with and without a silencer installed in pipe or duct with air flow. Some manufacturers rate silencers as being “industrial”, “commercial”, or “residential”

grade; in such a case, the DIL of the silencer should still be requested in order to determine the grade of silencer most suitable for the installation.

To determine the DIL required by a particular application, information is required on the actual (unsilenced) and desired noise levels at the emission point. The difference between these values is the silencer DIL. The desired source level is determined from the criteria governing the site.

When used to attenuate exhaust noise, the silencer must be sized to accommodate the specified volume of flow without imposing excessive backpressure. The flow area for a given backpressure can be calculated from the engine exhaust flow (CFM) and the exhaust temperature. The pressure drop will determine the required size of the silencer.

## **Sound Absorption Treatments**

Acoustically absorptive surfaces convert acoustic energy into heat, and are generally described by sound absorption coefficients in octave bands. Absorptive surfaces may be used to reduce the reverberant (reflected) sound field within a room. As mentioned above, reducing the reverberant field within a room can also reduce the noise field outside the room. It should be noted that absorptive materials do not attenuate the direct sound field.

The absorption of a room may be estimated on an octave-band basis from the absorption coefficients and the area of each room surface (ceiling, walls, and floor). Alternatively, the room absorption may be determined through reverberation time measurements. Using this information and the source sound power data, the noise reduction that can be obtained by adding absorption to a room may be determined. Information on the absorption coefficients of a material or element may be obtained from the manufacturer.

A wide variety of commercially available sound absorbing elements are available for almost every application. Ceiling treatments include lay-in tiles or boards (for suspended ceilings), tiles that can be directly affixed to the ceiling surface, and suspended absorbers.

Acoustic wall panels range from “architectural” panels with attractive finishes to perforated metal panels filled with absorbing materials. Concrete blocks with slotted faces and acoustical fill may be used to add sound absorption to normal concrete block wall construction.

Sound absorbing elements are selected on the basis of their sound absorption coefficient in the octave bands of interest. In addition, the elements must survive their environment, be easy to maintain, and offer acceptable flame spread properties.

### **Enclosures and Barriers**

Enclosures and barriers block and reflect direct-radiated sound from a noise source. A barrier provides a “shadow zone” of sound attenuation between the source and the receiver, much as light casts a shadow behind a wall. Full enclosures may be used around the source or around the receiver (e.g., personnel in affected areas). Partial barriers may be used to protect noise sensitive areas, by locating receivers in the shadow zone.

The effectiveness of a barrier in blocking noise transmitted through it is a function of its sound transmission characteristics. Both enclosures and barriers should be lined with absorptive material to be fully effective. In the case of an enclosure without absorption, the reverberant field inside the enclosure can greatly increase the interior sound pressure, so that noise outside the enclosure is also increased. In the case of a barrier without absorption, the noise is simply reflected elsewhere. Transmission loss and absorption are the main selection criteria for barriers and enclosures, and each is a function of frequency.

Opening in enclosures should be acoustically treated, for maximum effectiveness. Also, when using sound barriers it is important to control “flanking path” (sound paths around the barrier).

There are many types of commercially available enclosures and barriers. Complete enclosures for specific types of mechanical equipment are available, some of which include silenced air inlets/exits and a reactive silencer for exhaust noise. Several types of modular panels are available that may include sound absorbing material on one or both sides of the panel. Outdoor barriers, designed to resist wind and seismic forces, are also available to block or reflect noise outdoors.

Along with acoustical performance, practical issues must be considered in using barriers or enclosures. Engine enclosures require ventilation to dissipate the heat that builds up within the enclosure. The enclosure must be accessible for maintenance and inspection, and may require panic latches on doors. Acoustic materials within the enclosure must be fire-resistant.

### **Structure-borne Noise Control**

The purpose of a vibration isolation system (whether simple or compound), or a wave barrier, is to control the transmission of structure-borne noise from the engine unit to the building structure, either directly or through the ground.

Those measures are intended to control noise close to the source, where control measures generally are most effective. However, even with effective isolation mounting of the engine unit it still may be necessary to provide additional structure-borne noise attenuation in the building construction. The simplest way to attenuate structure-borne noise along a path (at least conceptually) is to increase the distance between the source and receiver, since the amplitude of structure-borne noise decreases with increasing distance from the vibration source. The attenuation of noise in concrete-frame buildings has been found to be about 5 dB per floor for frequencies up to 1000 Hz. Attenuation for vibrations traveling along continuous concrete floor slabs typically range from 1.5 to 2 dB/meter. In general, there is less attenuation along horizontal building structures.

Another way to attenuate structure-borne noise is through structural discontinuities. A discontinuity, or impedance mismatch, causes a reflection of energy back toward the source, thereby controlling noise transmission. Such discontinuities are usually filled with a resilient material to prevent debris falling into and “shorting out” the gap. Semirigid fiberglass board is normally used to fill wall gaps, while asphalt-impregnated fiberglass board is normally used between on-grade slabs, foundations, and footings. Many times, large buildings already incorporate expansion joints to allow for thermal expansion and contraction. These may be used to attenuate structure-borne noise by placing the source and receivers on opposite sides of the expansion joint. It is essential that construction elements, pipes, or any other rigid connections do not bridge these discontinuities.

In addition to the source and the path, receiver locations can also be treated to control structure-borne noise in some situations. For example, a “floating floor” construction may be used to isolate the receiver (e.g., a person or some piece of vibration-sensitive equipment) from building vibration.

### **Foundation**

Foundation Design is a very important and often overlooked aspect of large-engine unit facility design. Large-engine units, as noted above, emit relatively strong low frequency energy — structure-borne as well as airborne. If the facility design does not account for both forms of noise, it is likely that site noise criteria will not be met. (Foundation design for installations where noise is not an issue is discussed in the Mounting section.)

Unfortunately, structure-borne transmission and radiation is much more difficult to analyze than airborne noise. Whereas it may

be relatively straightforward to estimate the airborne noise transmission loss of the building structure and various types of noise control systems, and thereby assess the adequacy of a facility design, reliable quantitative estimates of structure-borne noise transmission may be extremely difficult or impossible to obtain with current technology, particularly at low frequencies. Thus, the usual approach for noise-sensitive installations is to over-design for structure-borne noise, to ensure that it is not a problem. This means taking care to control every possible structure-borne noise path. Especially in this area, designers are strongly urged to consult qualified professional noise control engineers for noise-sensitive installations.

Engine units usually are mounted on concrete pad or metal deck foundations, using the spring mounts between the unit base and the foundation. Some of the smaller engine units come with isolators between the engine/generator and base and do not require additional spring mounts for the unit base. Since the unit base provides sufficient stiffness for alignment and relative deflection of the engine and the driven equipment, there is no need to rely on the foundation for additional stiffness. Thus a foundation that is adequate for supporting the static load of the unit will be satisfactory for many installations where noise is not a critical concern.

In installations where noise is a major concern, attention must be directed toward all elements of the isolation system and to the structural paths between the foundation and the rest of the building structure. Adequate isolation often can be achieved with a simple system, but some installations may require a compound isolation system.

# Notes

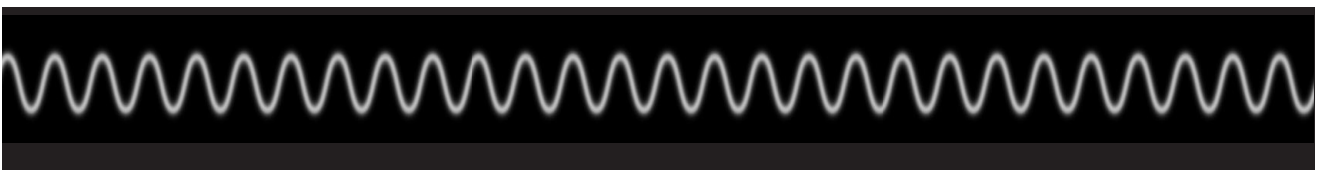


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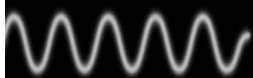




# **Electric Power Application and Installation Guide**

## **Starting System**

LEBX0032-01



WHERE THE WORLD TURNS FOR POWER



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# Starting System

An engine starting system must be able to crank the engine at sufficient speed for fuel combustion to begin normal firing and keep the engine running.

There are three common types of engine starting systems normally used for Caterpillar engines:

- 1) Electric
- 2) Air
- 3) Hydraulic

The choice of systems depends upon availability of the source of energy, availability of space for storage of energy, and ease of recharging the energy banks required for starting the engine.

## Types of Starting Systems

### 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.

Electric and air starters can be used on both diesel and gas engines. Hydraulic starters can only be used with diesel engines.

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.

## Function of Starting System

### Terminology

#### Breakaway Torque

Engine breakaway torque refers to the torque required to begin rotating the crankshaft. The starter must be able to exceed the torque of the engine and auxiliary in order to begin rotating the engine. With an electric starter the voltage and amperage is important, so that the electric starter can reach its full potential which is listed in Table 1.

	Volts at Starter	Amps to Starter	Speed Starter/ Eng Cranking (rpm)	Potential Torque of Starter N·m (lb·ft)	Potential Torque on Flywheel N·m (lb·ft)
<b>G3304, G3306</b> (156 flywheel teeth)	Starter Part No. — 1W5516 (12 pinion teeth)				
Breakaway*	17	1400	0/0	129 (95)	1674.4 (1235)
<b>G3406</b> (113 flywheel teeth)	Starter Part No. — 2S7227 and 6V0927 (11 pinion teeth)				
Breakaway*	12.5	1400	0/0	135.6 (100)	1392.4 (1027)
<b>G3408, G3412</b> (136 flywheel teeth)	Starter Part No. — 4N1062 (11 pinion teeth)				
Breakaway*	12.5	1400	0/0	135.6 (100)	1675.8 (1236)
<b>G3508, 12, 16</b> (183 flywheel teeth)	Starter Part No. — 7C0527, 6V4246 (11 pinion teeth)				
Breakaway*	13	1400	0/0	124.7 (92)	2074.4 (1530)
	Starter Part No. — 6V0927 (11 pinion teeth)				
	12.5	1400	0/0	135.6/2250.7 (100/1660)	135.6/2250.7 (100/1660)

\*Breakaway torque is theoretically 0 starter and engine rpm. However, the starter amperage limit is approximately 1400 amps. Thus the breakaway torque shown is reduced somewhat from actual to accommodate the amperage limit.

Table 1. Electric starter breakaway performance, gas.

Starter breakaway torque refers to the maximum torque a starter can generate at or near zero rpm. It is important to ensure that the starter breakaway torque is larger than the sum of the engine and auxiliary load breakaway torque. Since breakaway torque can vary widely, it is best to include a breakaway torque margin when handling auxiliary loads. The following equation should be used as a guideline to assure proper breakaway torque margin. The values used for  $T_{eng\ brk}$  and  $T_{aux\ brk}$  should be values representing worst case conditions under which an engine is expected to start.

$$T_{start\ brk} > (1.4)[T_{eng\ brk} + T_{aux\ brk}]$$

where,

$T_{start\ brk}$  = starter breakaway torque output

$T_{eng\ brk}$  = engine breakaway torque req

$T_{aux\ brk}$  = auxiliary breakaway torque req

The relationship of the starter torque to the torque applied to the engine (and auxiliary equipment if direct coupled) is:

$$\begin{aligned} \text{Starter Torque} &= \text{Torque to Engine \& Driven Equipment} \\ &\times \text{Number of Pinion Teeth/Flywheel Teeth} \end{aligned}$$

This formula will work for determining either the breakaway torque needs or the cranking torque needs of the starter.

**Example 1.**

**Starter Sizing for Breakaway Torque**

An air supply source of 620.5 kPa (90 psi) is available to start a 3512 gas engine. The lowest ambient temperature will be 10°C (50°F) and 30 wt oil will be used. The engine will drive a water pump through a clutch so the breakaway torque is negligible. The customer prefers an air starter.

A) What is the breakaway torque required for this application?

B) Does a single vane starter with 620.5 kPa (90 psi) inlet air pressure have sufficient energy to rotate the 3512 engine?

**Solution A:**

Consulting the Engine Breakaway and Cranking Torque Chart (see Table 2). The breakaway torque for the 3512 gas engine is 1247 N•m (920 lb•ft). The auxiliary equipment has negligible breakaway torque. The starter should be able to produce 1.4 times 1247 N•m (920 lb•ft) = 1746 N•m (1288 lb•ft) for torque.

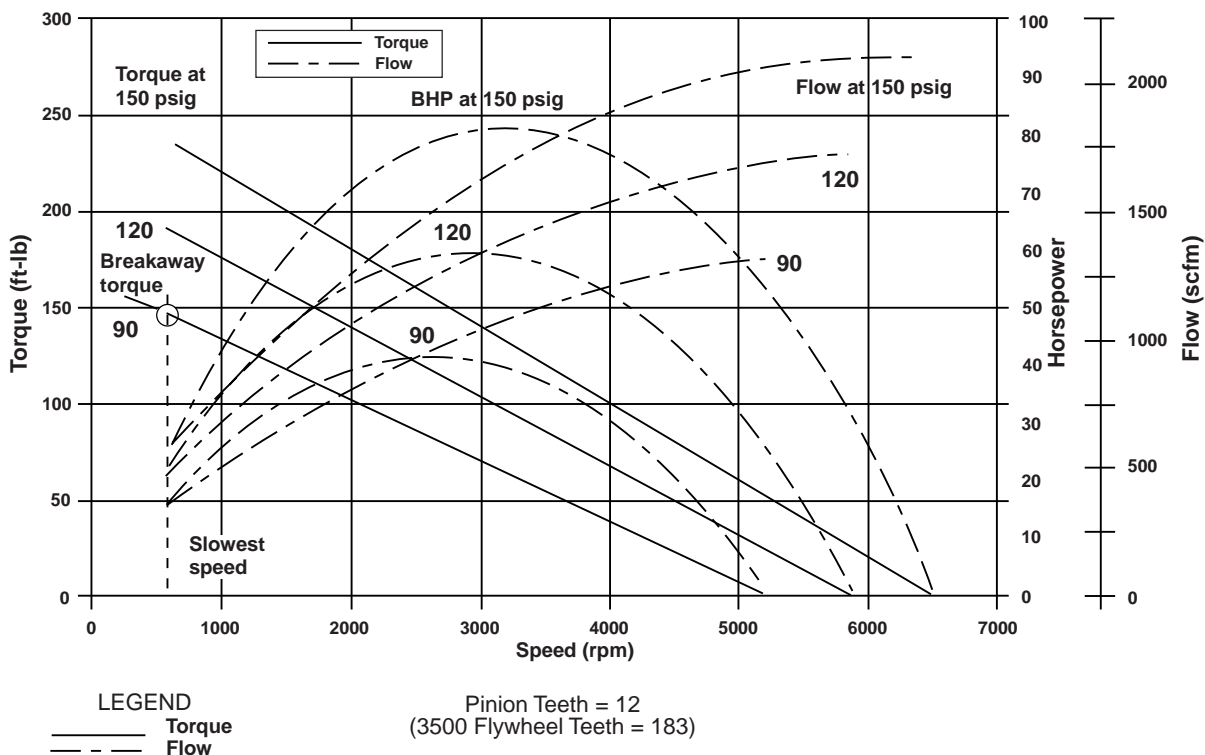


Figure 1. G3500 air starter (vane type) performance curve.

### Solution B:

The starter torque =  $1746 \text{ N}\cdot\text{m}$  ( $1288 \text{ lb}\cdot\text{ft}$ )  $\times$   
 $12$  (pinion teeth)/ $183$   
(number of flywheel teeth)

The starter torque =  $115 \text{ N}\cdot\text{m}$

The starter torque =  $85 \text{ lb}\cdot\text{ft}$

Next consult the Air Starter Torque Chart Figure 1 (can be obtained for the individual air starter from the air starter manufacturer), for the starter's capability at 620.5 kPa (90 psig). Use the torque of the slowest listed speed of the starter as the breakaway torque. The G3500 air starter (vane type) is  $196.6 \text{ N}\cdot\text{m}$  ( $145 \text{ lb}\cdot\text{ft}$ ). Thus one air starter for the 3512 engine will provide sufficient breakaway torque ( $145 \text{ lb}\cdot\text{ft} > 85 \text{ lb}\cdot\text{ft}$ ).

### Cranking Torque

Table 2 shows the torque requirements to crank an unloaded gas engine.

Engine Model	Breakaway Torque		Cranking Torque (@ 100 or 150 rpm)			
			SAE 30 Wt Oil			
			0°C (32°F)		10°C (50°F)	
N·m	(ft·lbs)	N·m	(ft·lbs)	N·m	(ft·lbs)	
G3304	149	110	217	160	149	110
G3306	244	180	278	205	203	150
G3406	366	270	508	375	305	225
G3408	420	310	597	440	352	260
G3412	644	475	813	600	521	385
G3508	1016	750	1085	800	854	630
G3512	1247	920	1573	1160	1037	765
G3516	1545	1140	1898	1400	1288	950

Note: Oil Temperature and Viscosity are major contributors to the actual amount of torque needed to crank the engine at the specific speed. Breakaway torque is independent of oil viscosity or temperature.

Table 2. Engine breakaway and cranking torques, gas.

*Note: The temperature and oil viscosity are the largest factors that affect the cranking torque needed to start an engine. Higher oil temperatures and less viscous oil require less torque to turn the engine.*

Table 3 gives the cranking performance of the electric starters used on the G3500-G3300 engines. Consult these charts to determine starter speed, volts and amps to the starter and the starters torque capability at 100 rpm and 1500 rpm engine rpm (150 rpm should

be used for magneto engines and 100 rpm for Caterpillar Electronic Ignition System equipped engines). The capability of the electric starter can be compared to the total of any particular engine model's and auxiliary equipment's cranking torque needs.

### Crank Terminate

#### Starter Overspeed

Engine overspeed can result from a governor malfunction or fuel rack sticking. With no overspeed protection, a diesel engine can rapidly accelerate to the point of destruction.

### Variables that Affect Length of Engine Start and the Cranking Torque

Size the starting system for an engine to meet the most demanding conditions under which a specific engine is expected to start. To account for the variability, consider the following guidelines:

- 1) **Engine Size:** As the number of cylinders increases, the torque requirements to crank the engine increases. The crank torque curves for each engine must be consulted when sizing a starting system.
- 2) **Engine Application:** Various applications have different auxiliary loads during the start up. The auxiliary load torque curve will affect the time to accelerate the engine to the minimum recommended cranking rpm.
- 3) **Fuel Composition:** Fuel composition varies widely from site to site. Fuels with low energy content will lengthen the cranking time as these fuels burn slower and cylinder firing during start-up is more erratic than when using natural gas.
- 4) **Cold Ambient Conditions:** Cold temperature increases the viscosity of the engine oil. The torque demand of the cranking engine increases as oil temperature decreases. Start length will increase due to cold temperatures. Note: The breakaway torque is not influenced by oil temperature.

### Minimum Requirements for Sizing Support Equipment

When sizing a facility's support equipment for a starting system (i.e. air tank volume, air tank location, pipe diameter, pipe length,

compressor size, battery quantity and capacity), the equipment must be capable of meeting the following minimum requirements for one start attempt:

- 1) The starting system must accelerate the engine and its auxiliary load to the Minimum Cranking RPM in five seconds.
- 2) The starter(s) must be able to sustain the Minimum Cranking RPM for a minimum of 25 seconds.

*Note: Electric starter engagements longer than 30 seconds or repeat 30 second start attempts will overheat starters. Start attempts longer than 30 seconds may indicate engine problems.*

Some applications may require multiple, back-to-back start capability. In this case, multiply the above **one-start minimum requirements** by the total number of required start attempts. Thus, if a site must be capable of three back-to-back start attempts, the support equipment should be capable of three, five second acceleration intervals and three, twenty-five second intervals of sustained engine speed equal to or greater than the Minimum Starting RPM.

### Oil Temperature

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).

### Air Temperature

The ambient temperature is extremely important for the starter system. The colder it is the higher torque is necessary for starting the engine. Take into consideration the temperature when reading a torque graph or table.

### Electric Starting

Electric starting is the most convenient to use. Storage of energy is compact, however, charging the system is slow and difficult in emergencies. Electric starting becomes less

effective as the temperature drops due to loss of battery discharge capacity and an increase in an engine's resistance to cranking under those conditions. It is the least expensive system and is most adaptable to remote control and automation (see Figure 2).

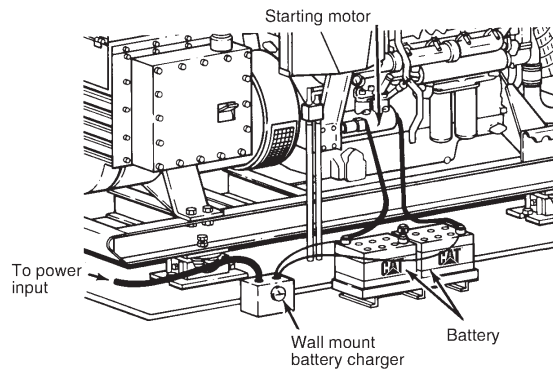


Figure 2. Battery location.

Damage can result if water enters and is retained in the starting motor solenoid. To prevent this, engines stored outside should be provided with a flywheel housing cover. If possible, the starting motor should be mounted with the solenoid in an up position which would provide drainage and prevent water from collecting in the solenoid.

Engines which are subject to heavy driven load during cold start-up should be provided with a heavy-duty starting motor.

### Battery Types

Batteries provide sufficient power to crank engines long and fast enough to start (see Battery recommendations for Diesel, Table 4 and for Gas, Table 5).

Lead-acid types are common, have high output capabilities, and lowest first cost.

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.

	Volts at Starter	Amps to Starter	Potential Torque of Starter N•m (lb•ft)	Potential Torque on Flywheel N•m (lb•ft)
<b>G3304, G3306</b> (156 flywheel teeth)	Starter Part No. — 1W5516 (12 pinion teeth)			
Cranking* @ 100 engine rpm	19	800	70.5 (52)	916.5 (676)
@ 150 engine rpm	22	500	30 (22)	387.8 (286)
<b>G3406</b> (113 flywheel teeth)	Starter Part No. — 2S7227 and 6V0927 (11 pinion teeth)			
Cranking* @ 100 engine rpm	17	900	73 (54)	745.7 (550)
@ 150 engine rpm	19	600	49 (36)	501.7 (370)
<b>G3408, G3412</b> (136 flywheel teeth)	Starter Part No. — 4N1062 (11 pinion teeth)			
Cranking* @ 100 engine rpm	17	860	70.5 (52)	867.7 (640)
@ 150 engine rpm	19	580	40.7 (30)	488 (360)
<b>G3508, 12, 16</b> (183 flywheel teeth)	Starter Part No. — 7C0527, 6V4246 (11 pinion teeth) [Values for 1 starter]			
Cranking* @ 100 engine rpm	19	600	40.7 (30)	678 (500)
	Starter Part No. — 6V0927 (11 pinion teeth) [Values for 1 starter]			
	18.5	600	49 (36)	813.5 (600)

\*Cranking torque is at 100 engine rpm. Cranking torque is very dependent on oil viscosity and temperature. When solving a cranking issue, increasing the temperature of the engine and oil plus lowering the viscosity of the oil will dramatically improve cranking performance.

Table 3. Electric starter cranking performance, gas.

Two considerations in selecting proper battery capacity are:

- The lowest temperature at which the engine might be cranked.
- The parasitic load imposed on the engine. A good rule of thumb is to select a battery package which will provide at least four 30 second cranking periods (total of 2 minutes cranking). An engine should not be cranked continuously for more than 30 seconds or starter motors may overheat.

Ambient temperatures drastically affect battery performance and charging efficiencies. Maintain 32°C (90°F) maximum temperature to assure rated output. The impact of colder temperatures is shown in Table 6.

Minimum Battery Cold Cranking Amperes at -18°C (0°F)				
Model	Voltage	-0.5°C (31°F) and Up*	-18°C (0°F) to -1°C (30°F)**	-31.6°C (-25°F) to -18.3°C (-1°F)
3208	12	1140	1460	1600
	24	570	730	800
3304	12	1140	1500	1740
	24/30/32	570	750	870
3306	12	1140	1500	2000
	24	570	750	1000
	30/32	570	750	870
3406	12	1740	1800	2000
	24	800	870	1000
	30/32	800	870	870
3408/ 3412	24	870	1000	1260
	30/32	870	870	1260
D348	24	870	1000	1260
	30/32	870	870	1260
D349	24/30/32	1260	1260	1260
D353	24	1000	1260	1260
	30/32	1260	1260	1260
D379/ 398	24/30/32	1260	1260	1260
D399	24/30/32	1260	1260	—

\*Below 15.5°C (60°F) use glow plugs if available.

\*\*Below 0°C (32°F) use ether aid for direct injection engines.

Table 4. Caterpillar diesel engine battery recommendations.

Minimum Battery Cold Cranking Amperes				
Model	Voltage	15.5°C (60°F) and Up	-1°C (30°F) to 15.5°C (60°F)	-29°C (-20°F) to -1°C (30°F)
G3304	12 24-32	725	1450 925	1225
G3306	12 24-32	725	1450 925	1225
G3406	24-32		925	1225
G3408	24-32	925	1225	1300
G3412	24-32	925	1225	1300
G3508	24-32	925	1225	1300
G3517*	30-32	725	910	1300
G3516*	30-32	725	910	1300

\*Two motors.

Note: Use aids below -18°C (0°F).

Table 5. Caterpillar gas engine battery recommendations.

## Battery Sizing and Temperature

Low ambient temperatures drastically affect battery performance and charging efficiencies, as well as oil viscosity (see Table 7). High temperatures should also be avoided.

- Maintain 32°C (90°F) maximum temperature to assure rated output.
- High temperatures also decrease battery life.
- Ideally, temperatures surrounding the battery should not exceed 25°C (77°F). Battery life is roughly halved by a 10°C (17°F) rise in temperature; doubled if ambients are reduced 10°C.

°C	°F	27°C (80°F) Ampere Hours Output Rating
27	80	100%
0	32	65%
-18	0	40%

Table 6. Temperature vs. output.

Specific Gravity	% Charge	Voltage per Cell	Freezes	
			°C	°F
1.260	100	2.10	-94	-70
1.230	75	2.07	-56	-39
1.200	50	2.04	-27	-16
1.170	25	2.01	-19	-2
1.110	Discharged	1.95	-8	+17

Table 7. Battery performance.

## 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 (see Table 8).

Magnetic Switch and Series-Parallel Circuit	Solenoid Switch Circuit	Starting Motor Circuit
12 Volt System 0.048 Ohm	0.0067 Ohm	0.0012 Ohm
24 Volt System 0.10 Ohm	0.030 Ohm	0.002 Ohm
32 Volt System 0.124 Ohm	0.070 Ohm	0.002 Ohm

Table 8. Maximum allowable resistance.

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

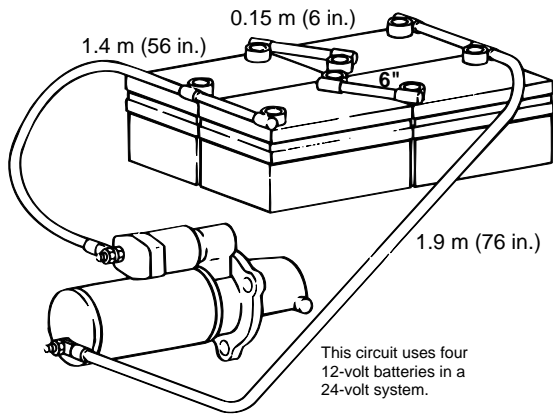
Connections  
(series connector)  
0.00001 Ohm each

The fixed resistance of connections and contactors is determined by the cable routing (see Table 9). Fixed resistance (R<sub>f</sub>) subtracted from total resistance (R<sub>t</sub>) equals allowable cable resistance (R<sub>c</sub>): R<sub>t</sub> - R<sub>f</sub> = R<sub>c</sub>.

Cable Size		12 Volt		24-32 Volt	
AWG	mm <sup>2</sup>	Feet	Meters	Feet	Meters
0	50	4.0	1.22	15.0	4.57
00	70	5.0	1.52	18.0	5.49
000	95	6.0	1.83	21.0	6.40
0000	120	7.5	2.29	27.0	8.24

Table 9. Maximum recommended total battery cable length

**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 .....	3.6 m (144 in.)

Figure 3. Example.

With cable length and fixed resistance determined as shown in Figure 3, the appropriate cable size can be found using Figure 4. 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 as durable and will be damaged from corrosion in a much shorter time.

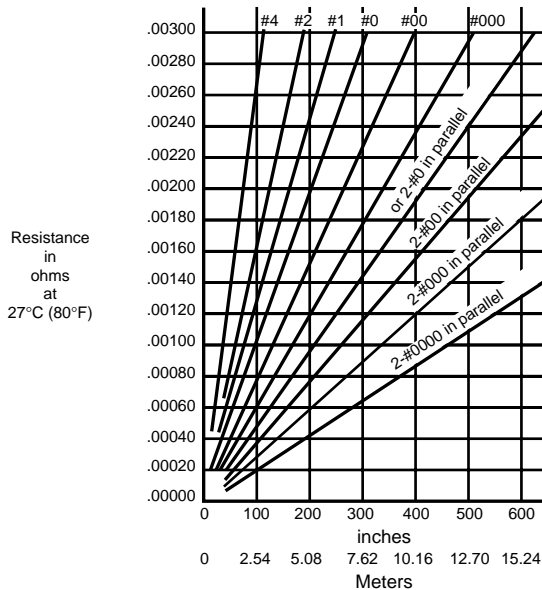


Figure 4. Cable sizing.

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.

**Charging 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 milliamper 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.

Both trickle chargers and float equalize chargers require a source of A/C power while the engine is not running. 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.

**Alternators**

Alternators or engine driven generators can be used as charging systems, but have the disadvantage of charging batteries only while the engine runs. Where generator sets are subject to many starts, insufficient battery capacity could threaten dependability.

**Grounding**

The battery must be ground before running. One battery terminal must be connected to either the engine or the frame work in order to protect against electrical shock.

## Design Consideration

The cranking batteries should always be securely mounted where it is easy to visually inspect and do maintenance (see Figure 2). They must be located away from flame or spark sources and isolated from vibration. Batteries should be mount level on nonconducting material and protected from splash and dirt. By positioning batteries near the starting motor, short slack cables can be used and voltage drops minimized. All battery connections must be kept tight and coated with grease to prevent corrosion.

Disconnect the battery charger when removing or connecting battery leads. Solid-state equipment, i.e., electronic governor, speed switches, can be harmed if subjected to charger's full output.

## Air Starting

Air starting, either manual or automatic, is highly reliable. It is generally applied where facilities have existing plant air, or where a combustible gas may be present in the atmosphere.

Air starting usually offers higher cranking speeds than electric starting. This will usually result in faster starts with less cranking time; however, remote controls and automation are more complex. On the other hand, the air system can be quickly recharged. Air storage tanks are prone to condensation problems and must be protected against corrosion and freezing.

The air starting system includes: air starting motor, air storage tank, starting valve, pressure regulator, and oiler (see Figure 5). A starting motor discharge air silencer/vapor arrestor is an optional accessory to the air starting system.

A check valve between plant air and receiver assures that failure of plant air will not deplete the backup supply. The air compressors are driven by gasoline engines and electric motors wired to the emergency power source.

For factory supplied electric motor driven by continuous prelube systems, power to the prelube pump is removed by a signal from the control panel when a start is initiated (see Figure 6).

Venting of the starter system must be inspected before start-up to insure that no backpressure in the vent line builds up. To prevent backpressure, a separate vent line should be provided. Another option would be to install a check valve in the starter vent line. (Check valves are not recommended in applications where freezing of the check valve can occur.)

Tandem or compound engines use two motors and solenoid valves. Valves are equal distance from their respective motors for coordinated motor engagement. When a single solenoid controls air to both motors, piping between valve and each motor must be equal length.

Air motor supply pipes are short, direct, and at least equal in size to the motor intake opening. Black iron pipe is preferable and should be supported to avoid stresses on the compressor. Flexible connections between motor and piping are required.

Deposits of an oil-water mixture accumulating in receiver and piping are removed by traps installed at intervals in the lines. The lines should slope toward these traps.

Air cranking systems may freeze at low ambients. Water vapor in compressed air freezes during expansion in temperatures below 0°C (32°F). A dryer at the compressor outlet or a small quantity of alcohol in the starter air tank prevents freezing. At temperatures below -18°C (0°F) consult the generator set supplier.

Torque available from air motors accelerate the engine to twice the cranking speed in about half the time required by electric starters. Remote controls and automation, however, are more complex. Because the exhaust from the air starter is very loud, an air silencer may be appropriate.

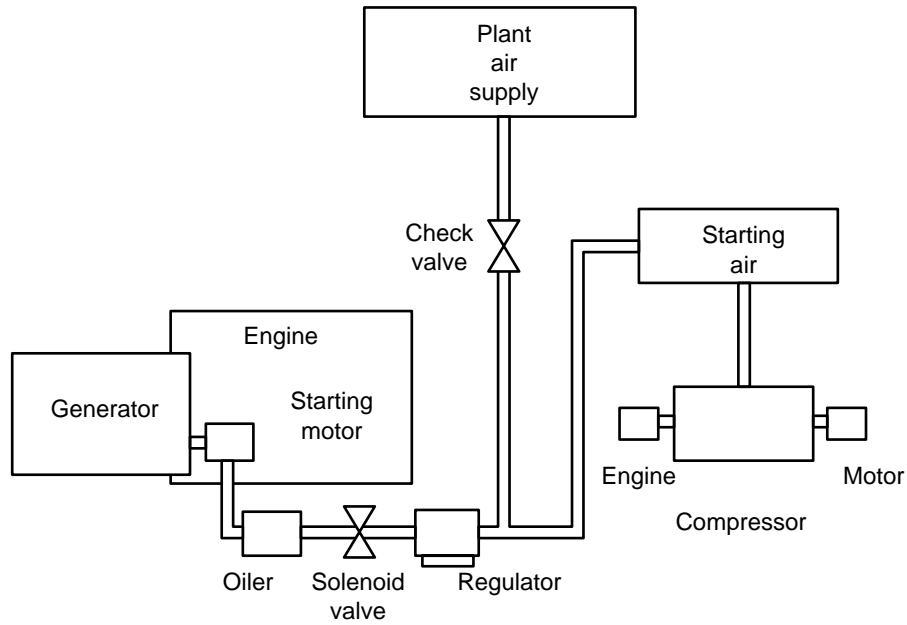


Figure 5. Air starting schematic.

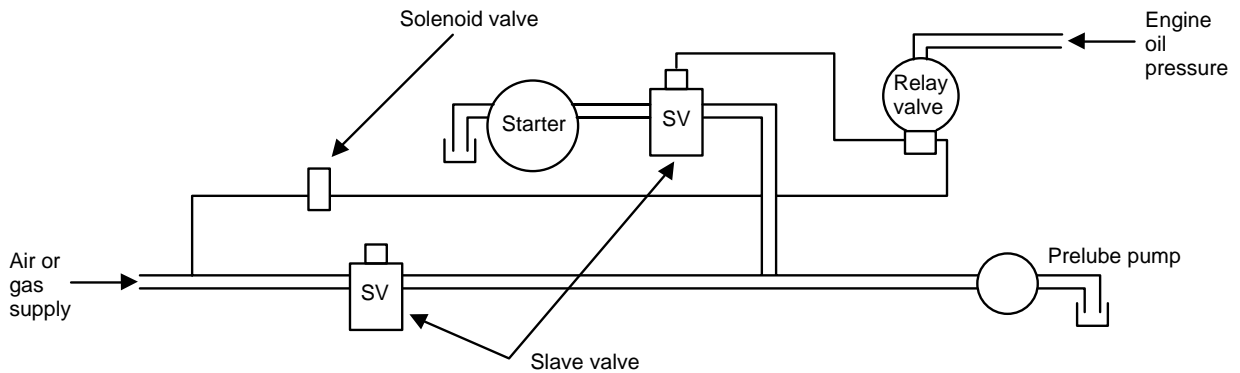


Figure 6. Air start prelude.

### Free Air Consumption

Free air consumption depends to some extent on the same variables as air starting systems. In addition the pressure regulator setting must be taken into consideration.

The correct pressure regulator setting is 620 to 690 kPa (90 to 100 psi), with the higher pressure used to improve starting under adverse conditions. 0.14 to 0.42 m<sup>3</sup> (5 to 15 cu ft) per second is typical for engines from 37 to 1194 kW (50 to 1600 hp).

### System Design Considerations

#### Air Pressure

Air is usually compressed between 758 and 1723 kPa (110 to 250 psi) and is stored in

storage tanks. Stored air is regulated to 758 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.

#### Cranking Time

The cranking time per start depends upon the engine model, engine condition, ambient air temperature, oil viscosity, fuel type, and design cranking speed. Five to seven seconds for the first start is typical for an engine at 26.7°C (80°F). Restart of a hot engine usually takes less than two seconds.

Air Consumption of the Air Start Motor m <sup>3</sup> /sec (ft <sup>3</sup> /sec) -Vs-				Minimum Air Tank Pressure
	Air Pressure to Motor -Pt-			
Engine Model	793 kPaa (115 psia) 690 kPag (100 psig)	965 kPaa (140 psia) 862 kPag (125 psig)	1137 kPaa (165 psia) 1034 kPag (150 psig)	-Pmin- kPaa (psia)
3304B	.16 (5.8)	.20 (6.8)	.21 (7.7)	345 (50)
3306B	.17 (5.9)	.20 (6.8)	.22 (7.8)	352 (51)
3406B	.17 (6.2)	.21 (7.3)	.23 (8.3)	379 (55)
3406B	.18 (6.4)	.21 (7.3)	.24 (8.6)	372 (54)
3412	.25 (9.0)	.29 (10.3)	.33 (11.8)	310 (45)
D379	.26 (9.3)	.30 (10.8)	.36 (12.6)	310 (45)
D398	.28 (9.8)	.32 (11.4)	.38 (13.3)	345 (50)
D399	.30 (10.8)	.34 (12.1)	.40 (14.1)	448 (65)
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 prelude: add 0.03 m<sup>3</sup>/sec (1 ft<sup>3</sup>/sec) air consumption.

Table 10. Diesel engine, air starting requirements at 10°C (50°F).

Air Consumption of the Air Start Motor m <sup>3</sup> /sec (ft <sup>3</sup> /sec) -Vs-				Minimum Air Tank Pressure
	Air Pressure to Motor -Pt-			
Engine Model	793 kPaa (115 psia) 690 kPag (100 psig)	965 kPaa (140 psia) 862 kPag (125 psig)	1137 kPaa (165 psia) 1034 kPag (150 psig)	-Pmin- kPaa (psia)
G3304	.16 (5.8)	.19 (6.8)	.22 (7.7)	242 (35)
G3306	.17 (5.9)	.20 (6.9)	.22 (7.8)	248 (36)
G3406	.18 (6.2)	.21 (7.3)	.23 (8.3)	276 (40)
G3408	.18 (6.4)	.21 (7.5)	.24 (8.6)	269 (39)
G3412	.25 (9.0)	.30 (10.3)	.33 (11.8)	207 (30)
G3508	.26 (9.3)	.31 (10.8)	.36 (12.6)	310 (45)
G3512	.28 (9.8)	.32 (11.4)	.38 (13.3)	344 (50)
G3516	.30 (10.5)	.34 (12.1)	.40 (14.1)	448 (65)
G353	.19 (6.6)	.22 (7.8)	.25 (8.9)	276 (40)
G379	.26 (9.3)	.31 (10.8)	.36 (12.6)	207 (30)
G398	.28 (9.8)	.32 (11.4)	.38 (13.3)	242 (35)
G399	.30 (10.5)	.34 (12.1)	.40 (14.1)	345 (50)

Note 1: Add to the G399 and G3516 0.03 m<sup>3</sup>/sec (1 ft<sup>3</sup>/sec) of air consumption for the air operated oil prelubrication pump. This pump will normally operate 2 to 10 seconds before the engine begins to crank.

Note 2: For start control systems with a purge cycle, include the time of the purge sequence to estimate starting time to obtain total estimated starting time.

\*Minimum air storage tank pressure required to sustain cranking at 100 rpm.

Table 11. Gas engine, air starting requirements at 10°C (50°F).

## Air Tank Sizing

Many applications require the sizing air storage tanks to provide a specified number of starts without recharging. The tanks must meet ASME specifications and be equipped with a safety valve and gauge. The tank size can be calculated from the following:

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

Where:

- $V_t$  = Air Storage Tank Capacity, m<sup>3</sup> (ft<sup>3</sup>) (Figure 7)
- $V_s$  = Air consumption of the starter motor, m<sup>3</sup>/sec (ft<sup>3</sup>/sec) (see Table 10).
- $T$  = Total Cranking time required (seconds) if six consecutive starts are required, use seven (7) seconds for first start (while engine is cold), and two (2) seconds each for the remaining five starts, a total cranking time of seventeen (17) seconds.
- $P_a$  = Atmospheric Pressure (normally atmospheric pressure is 100 kPa (14.5 psia)).
- $P_t$  = Air storage tank pressure kPa (psi). 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.

### Example:

A 3516 Diesel Engine with electric prelude has the following:

- maximum air tank pressure = 1241 kPag (180 psig)
- minimum air to starter pressure = 620.5 kPag (90 psig)
- expected air line pressure drop = 207 kPag (30 psig)
- 6 consecutive starts. First start = 7 sec. the other 5 starts = 2 sec.
- average barometric pressure at this location = 100 kPa (14.5 psi)
- preconditioned engine installation. (cfm × 0.02832 = m<sup>3</sup>/min)

### Solution:

$V_s = 0.40$  m<sup>3</sup>/sec (14.1 ft<sup>3</sup>/sec), found by using Table 10.

$$T = 7 + (5 \times 2) = 17 \text{ sec}$$

$$P_a = 100 \text{ kPa (14.5 psi)}$$

$$P_t = 1241 - 207 = 1034 \text{ kPag}$$

$$(180 - 30 = 150 \text{ psig})$$

$$P_{\min} = 620.5 \text{ kPag (90 psig)}$$

therefore,

$$V_t = \frac{0.4 \times 17 \times 100}{1034 - 620.5} = 1.64 \text{ m}^3$$

$$V_t = \frac{14.1 \times 17 \times 14.5}{150 - 90} = 57.93 \text{ ft}^3$$

### 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) for both gas and diesel. Higher pressure can be used to improve starting under adverse conditions up to a maximum of 1034 kPa (150 psi) to the starting motor (gas and diesel). The values shown on the air starting requirements chart assume a bare engine (no parasitic load) at 10°C (50°F) (see Table 10 and 11).

### Starting Aids

Starting aids are recommended when temperatures fall below certain levels, as shown in the Operation and Maintenance Guides. Glow plugs and/or ether starting aids are sufficient for most conditions, with oil and coolant heating necessary in extremely low ambients (refer to Operations and Maintenance Guides for further data on cold weather procedures).

### Jacket Water Heaters

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

Jacket water heaters are used on both manual and automatic starting systems, but are essential for automatic starting below 21°C (70°F). Provisions should be made to maintain the jacket water temperature at a minimum of 43°C (110°F) for all G3516B applications. Heaters precondition engines for

quick starting and minimize the high wear of rough combustion, by maintaining jacket water temperature during shutdown periods.

Heaters thermostatically control jacket water temperature near 30°C (90°F) to promote fast starts. Higher temperatures accelerate aging of gaskets and rubber material.

### **Flame Start**

In a flame start, glow plugs project into the air inlet manifold and ignite a small amount of diesel fuel during manual starting. The flame is maintained until smooth idling conditions are achieved.

### **Battery Heaters**

Battery heaters are usually recommended in cold ambient temperatures. The heaters should be set to maintain battery temperature in the range of 32 to 52°C (90 to 125°F) for maximum effectiveness.

### **Ether**

Ether facilitates starting because it is a highly volatile fluid which has a low ignition temperature. Many types of ether starting aids are commercially available. The high pressure metallic capsule-type is recommended. When placed in an injection device and pierced, the ether passes into the intake manifold. This has proven to be the best system since few special precautions are required for handling, shipping, or storage.

Caution: When other than fully sealed ether systems are used, ensure adequate ventilation for venting the fumes to the atmosphere to prevent accidental explosion and danger to operating personnel.

Ether must be used only as directed by the manufacturer of the starting aid device. The ether system must be such that a maximum of 3.0 cc of ether will be released, each time the button is pushed. Caterpillar ether systems are designed to release 2.25 cc of ether each time the system is activated. Excessive injection of ether can damage an engine.

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. Only 1 cm<sup>3</sup> (0.033 oz) or less 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.*

Ether starting aids are restricted to manual starting systems and are rarely used for generator sets. Introduced into the intake air, it ignites the mixture at low cylinder temperatures. High pressure capsules are the safest and most dependable injection method. Ether injection should not be used on Cat 3600 Engines.

### **Oil Heaters**

Heating elements in direct contact with noncirculating lubricating oil are usually not recommended due to the danger of oil coking. If specified, heater skin temperatures should not exceed 150°C (300°F) and have maximum heat densities of 0.02 W/mm<sup>2</sup> (13 W/in<sup>2</sup>).

### **Additional Starters**

Hydraulic starting provides the highest cranking speeds and fastest starts. It is relatively compact. Recharging time is fast, and the system can be recharged by a hand pump provided for this purpose. The high pressure of the system requires special pipes and fittings and extremely tight connections. Oil lost through leakage can be replaced, but recharging the pressurized gas, if lost, requires special equipment.

## **Air Impingement System**

The air impingement system is a method of boosting the engine to get it started.

It must be noted that this system is not available for Caterpillar engines. This is due to unacceptable transient response and the fact that the turbo housing must be specially fabricated to accommodate the hose attachment.

Air impingement works by having a compressed air hose which is ported into the turbocharger turbine wheel. Once the solenoid valve is open, a blast of compressed air spins the turbine wheel. This creates more air on the compressor which then feeds more air to the cylinders which results in more combustion.

# Notes

# Notes

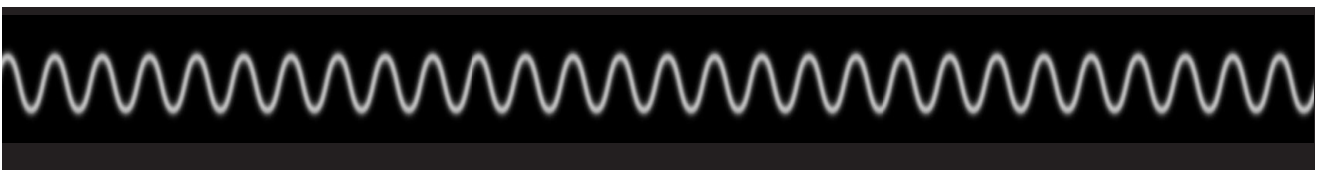


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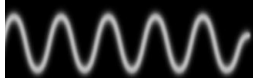




# **Electric Power Application and Installation Guide**

## **Ventilation**

LEBX0033



WHERE THE WORLD TURNS FOR POWER



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# Ventilation

## Ventilation Purpose

Six to ten percent of fuel consumed by an engine is lost as heat radiated to the surrounding air. In addition, heat from generator inefficiencies and exhaust piping can easily equal engine radiated heat. Any resulting elevated temperatures in the engine room may adversely affect maintenance personnel, switchgear, and generator set performance.

## Ventilation Flow

Ideally, clean, cool, dry air circulates around switchgear, flows through the rear of the generator, across the engine and discharges through the radiator (see Figure 1). Cool air should always be available for the engine air cleaner.

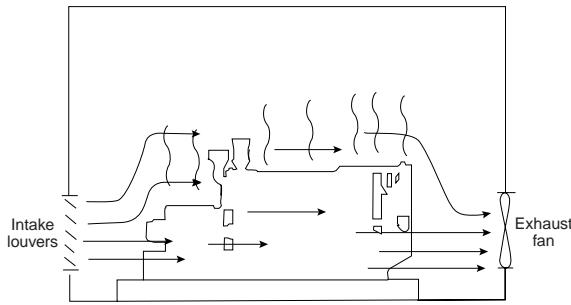


Figure 1. Ideal ventilation.

Locate room air intakes to provide maximum cooling air to the generator set, yet avoid hot, stagnant air in other areas. Multiple generator sets necessitate additional openings and fans.

Units not using radiators require a forced air draft. Openings for intake air should be low, near the rear of the generator set. Outlets should be positioned high on the opposite wall.

Another way of ventilating the engine room is the use of air curtains, although this is not so commonly utilized. The advantage of an air curtain system is that the engine room is exposed to lower air velocity than a system with radiators.

## Ventilation Terms

### 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.

### Crankcase Fumes Disposal:

Elimination of oil laden combustion fumes that collect in the engine crankcase (see "Crankcase Ventilation").

### Radiator Cooling Air:

Air flow passing through the radiator — normally to cool the engine. Certain installations utilize radiator cooling air flow for ventilation purposes.

## Ventilation Considerations

### Routing

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, and leaves areas of the room with no appreciable ventilation.

## Relative Efficiency Routing

The sketches below (see Figure 2) illustrate the relative efficiency of various ventilation routing.  $F_{routing}$  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 left).
- 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 right).

## Horizontal Air Flow

Cool, dry, clean air should enter the engine room as close to the floor as possible using fans/ducts. Allow this air to flow horizontally across the engine room from the entry point

across heat sources such as the engine, exposed exhaust, generator, etc. (see Figure 3).

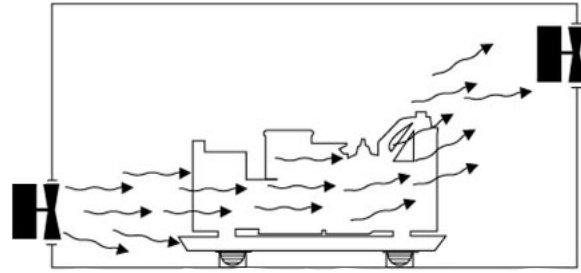


Figure 3. Horizontal air flow.

- For best results, air should flow first across the generator then to both sides of the engine.
- If engine mounted radiators are not used, air discharge fans should be mounted or ducted at the highest point directly over the heat sources.
- Inlet air must circulate between the engines in a multiple engine installation.
- Inlets located at the end of a room with multiple engines will provide adequate ventilation only to the engine nearest the inlet.

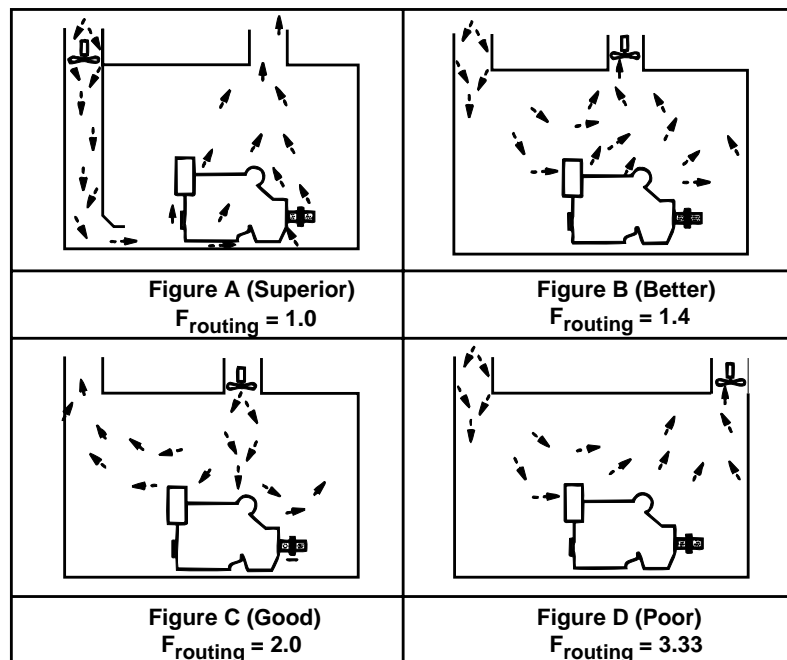


Figure 2. Efficiency of various routings.

Providing horizontal air flow will dissipate engine heat but a certain amount of heat will still radiate and heat the engine room.

### Air Curtains

Air curtains, totally enveloping the generator set, provide ventilation without exposing the equipment room to high air velocities (see Figure 4). Radiated heat is removed with approximately half the air flow of a horizontal flow system.

It is important to stretch the air curtain inlet the full length of generator set.

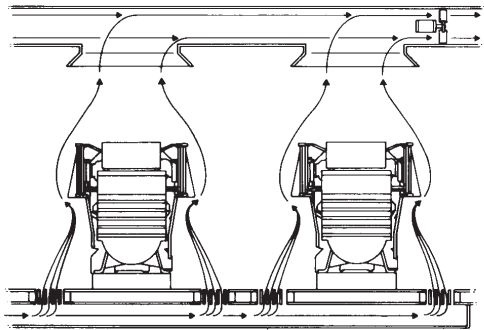


Figure 4. Air curtains.

Air curtains present ducting challenges when local fan radiators are used.

### Vertical Air Flow

The least desirable ventilation system discharges outside air directly down on the engines with inlet fans (see Figure 5). Exhaust fans should be mounted in the corners of the room.

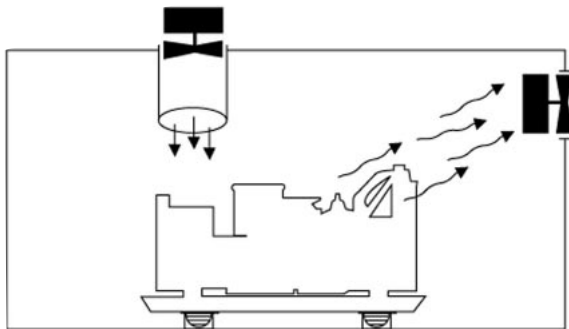


Figure 5. Vertical air flow.

Because this system interferes with the natural rising of hot air, ducting should be used to prevent air from taking the shortest path out of the engine room and bypassing the engine.

### Radiator Air

Installations utilizing local free standing or engine-mounted radiators may provide sufficient air flow for cooling. Total ventilation air flow requirements must be compared to radiator fan capabilities.

Intake and exhaust ventilators may have movable or fixed louvers for weather protection. If movable louvers are used, they should be actuated by pneumatic, electric, or hydraulic motors. Do not depend on air pressure developed by the radiator fan to open the vanes. In cold climates, movable louvers can be arranged to provide circulation inside the room until jacket water temperatures reach 88°C (190°F) (see Figure 6).

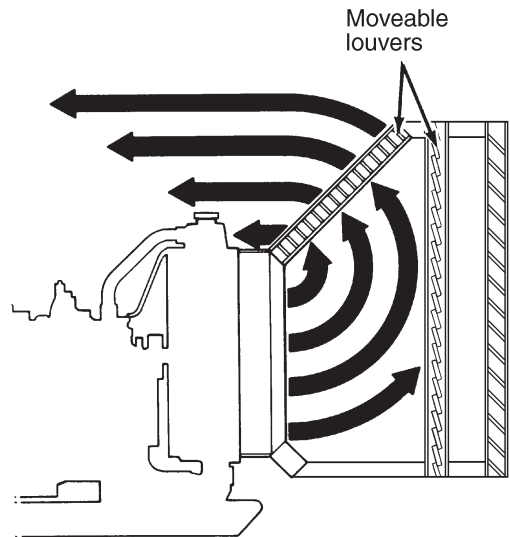


Figure 6. Moveable louvers for recirculation.

Once jacket water temperatures reach 88°C (190°F), the radiator must be furnished with sufficient cooling air. Use a number of small ventilating fans rather than a single large unit. Selective fan operation compensates for varying ambient temperatures while maintaining engine room temperatures.

Increase air flow 10 percent for every 763 m (2500 ft) above sea level to maintain original cooling capability. Final ventilation calculations must use precise heat radiation of selected engine, generator, and power output.

### **Engine Driven Fans**

For most local radiator generator sets, a blower fan (air discharges from engine through radiator) is recommended. The blower fan will move the incoming ventilation air from the generator, past the engine block and manifolds and use that air to cool the radiator.

If a suction fan is used, special considerations must be considered to avoid recirculation of the generator cooling air. It is also much more efficient to use the engine driven blower fan as a room evacuation device than to try to force heated air from a room by pressurizing the room with preheated air from the radiator.

### **Air Quantity Required**

In general, changing the air in the engine room every one or two minutes will be adequate, if flow routing is proper.

Combustion air would be required solely for burning the fuel in the combustion process. Hence the total air intake into the engine room would be the sum of both the combustion air and the cooling air. The quantity of air required for combustion is explained below.

Provisions should be made by the installer to provide incoming ventilation air of 0.1 – 0.2 m<sup>3</sup>/min (4-8 cfm) per installed horsepower. This does not include combustion air for the engines.

Engine room 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 reduced 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).

### **Ventilating 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.

#### **Fan Location**

Fans are most effective when they withdraw ventilation air from the engine room and discharge the hot air to the atmosphere.

#### **Fan 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.

#### **Fan 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

### Required Air Flow

Engine room ventilation can be estimated by the following formula, assuming 38°C (100°F) ambient air temperature:

$$V = \frac{H}{D \times C_p \times \Delta T} + \text{engine combustion air}$$

Where:

V = Ventilating Air (m<sup>3</sup>/min), (cfm)

H = Heat Radiation i.e. engine, generator, aux (kW), (Btu/min)

D = Density of Air at 38°C (100°F)  
(1.099 kg/m<sup>3</sup>), (0.071 lb/ft<sup>3</sup>)

C<sub>p</sub> = Specific Heat of Air  
(0.017 kW × min/kg × K), (0.24 Btu/°F)

ΔT = permissible temperature rise in engine room (°C), (°K)

Note: If duct work is used to bring in air for the engine's combustion air, the last term in the equation can be dropped.

### Example:

A 3412 DITA genset has the following data:

Heat rejection: 659 kW (37 478 Btu/min)

Temperature rise: 11°C (20°F)

### Solution:

The estimated engine room ventilation required for this arrangement:

$$V = \frac{659}{1.099 \times 0.017 \times 11} = 3206.61 \text{ m}^3/\text{min}$$

$$V = \frac{37478}{0.071 \times 0.24 \times 20} = 109970.7 \text{ cfm}$$

### Combustion Air Ducts

Design combustion air ducts to have a minimum flow restriction. Note that 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.

### Location

The combustion air ducts should be located close to the engine. Usually, flexible connections are used to reduce noise from the ducting system. In addition, all duct work must be supported on the engine to avoid unnecessary loading on the turbochargers.

### Duct Restriction

Total duct air flow restriction, including air cleaners, should not exceed 2.49 kPa (10 in. H<sub>2</sub>O) 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.

### Air Velocity in Combustion Air Ducts

Combustion air duct velocity should not exceed 610 m/min (2000 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 engines. This will reduce the flow of air through the engine, raising the exhaust temperature with potentially damaging effects.

### Crankcase Ventilation

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. Each engine's fumes disposal should have separate discharge pipes.

Crankcase fumes must not be discharged into air ventilating ducts or exhaust pipes. They will become coated with oil deposits. Crankcase fumes must be either ingested by the engine or piped out of the engine room.

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.

## Design Considerations

### Heat Rejection

#### Engine Radiant

The heat input into the engine is the sum of the work output and the heat generated. Besides the work output, heat is rejected to the atmosphere, into the oil cooler, aftercooler, through the exhaust stack and also through the jacket water. Hence, while designing the ventilation system in a room, the engine generated heat should be taken into consideration. This information can be found in TMI.

#### Generator Radiant

The heat radiated by the generator can be calculated by the following formulas:

$$\text{HRG (kW)} = P \times \left[ \frac{1}{\text{EFF}} - 1 \right]$$

$$\text{HRG (Btu/min)} = P \times \left[ \frac{1}{\text{EFF}} - 1 \right] \times 56.9$$

Where:

HRG = Heat Radiated by the Generator  
(kW), (Btu/min)

P = Generator Output at Maximum Engine  
Rating (ekW)

Eff = Generator Efficiency %/100%  
(Example: Eff = 94%/100% = 0.94)

#### Example:

A 3512B, 975 ekW standby generator set has a generator efficiency of 92%. What is the generator radiant heat for this genset?

#### Solution:

$$P = 975 \text{ ekW}$$

$$\text{Eff} = \frac{92\%}{100\%} = 0.92$$

$$\text{HRG} = 975 \times \left( \frac{1}{0.92} - 1 \right)$$

$$\text{HRG} = 84.78 \text{ kW}$$

$$\text{HRG} = 975 \times \left( \frac{1}{0.92} - 1 \right) \times 56.9$$

$$\text{HRG} = 4824 \text{ Btu/min}$$

#### Engine Room Auxiliaries

The consultant should take all auxiliary equipment besides the generator set into consideration while designing the ventilation system. Auxiliary equipment such as switchgear, compressors, pumps, lighting, piping etc. are sources of heat. Thus the design should cover all main and auxiliary equipment so that the room temperature is maintained.

#### Engine Room Temperature

A 8.5°C (15°F) temperature rise above the ambient temperature is a reasonable target for engine rooms. (Ambient air temperature refers to the air temperature surrounding the engine room.) However, in cold climates this may cause discomfort from the flow of cold air. Restrict flow only if engine combustion air is available and engine jacket water is adequately cooled. In general, engine room temperature should not exceed 49°C (120°F).

#### 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.

Table 1 lists typical air motions:

Air Velocity		Conditions
m/min	(fpm)	
15.2	50	Offices, seated worker
30.5	100	Factory, standing worker
45.7	150	Capture velocity, light dust
61	200	Maximum continuous worker exposure
396	1300	Capture velocity, rain
306 – 610	1000-2000	Maximum intermittent exposure

Table 1. Air velocity.

## Radiators

Installations with engine-mounted radiators using engine room air for cooling (Figure 7), generally provide more air flow than is needed for adequate ventilation. The high air flow combined with low ambient temperatures, below 21°C (70°F), may cause water to condense inside exposed engine components, like valve covers. This can result in oil and maintenance problems. Therefore, special installation considerations must be made in cold climates.

There are two methods that can be used to overcome this problem.

- Remote mounted and specially ducted engine-mounted radiators do not require engine room air for cooling (see Figure 8). One advantage of such a system is that

the air used to cool the radiator is not pre-heated by the engine, thus increasing the ambient capability (or reducing the size) of the unit. The disadvantage is that motor-driven fans must be installed to provide ventilation for the engine, generator and other equipment which increases the overall cost of the system. This system is suitable for continuous duty applications.

- Thermostatically controlled louvers can be installed to recirculate some of the radiator exhaust in order to maintain a warm air flow across the engine (see Figure 9). This also maintains a comfortable working environment for maintenance personnel. Caution must be exercised so that the recirculated air is reintroduced upstream of the engine and is well mixed by the time it reaches the radiator.

For any arrangement where a radiator fan is used to ventilate an engine room, the vacuum created in the engine room must not exceed 0.12 kPa (0.5 in. H<sub>2</sub>O). Any restriction above this limit could reduce air flow through the radiator and overheat the engine.

## Radiator Sizing

Radiator core frontal area should be as large as possible to minimize restriction to airflow. Low radiator core restriction usually results in being able to provide a larger slower turning fan.

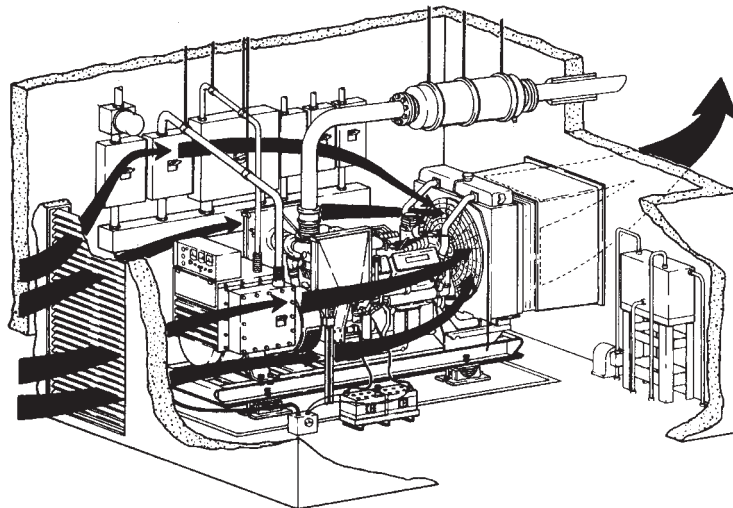


Figure 7. Engine driven fan arrangement.

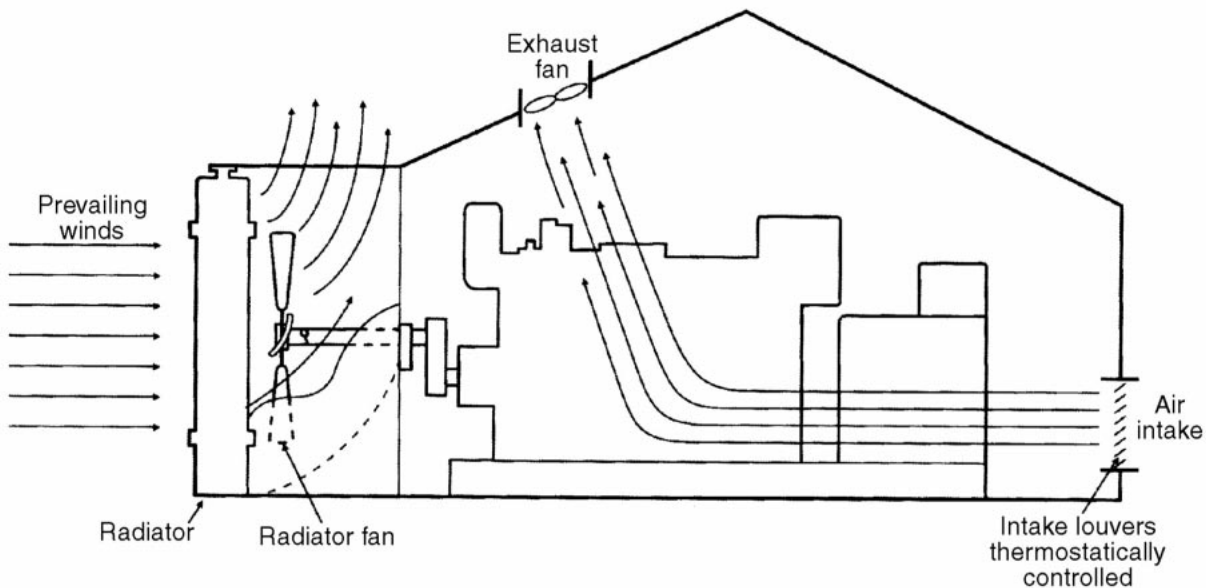


Figure 8. Engine driven fan arrangement.

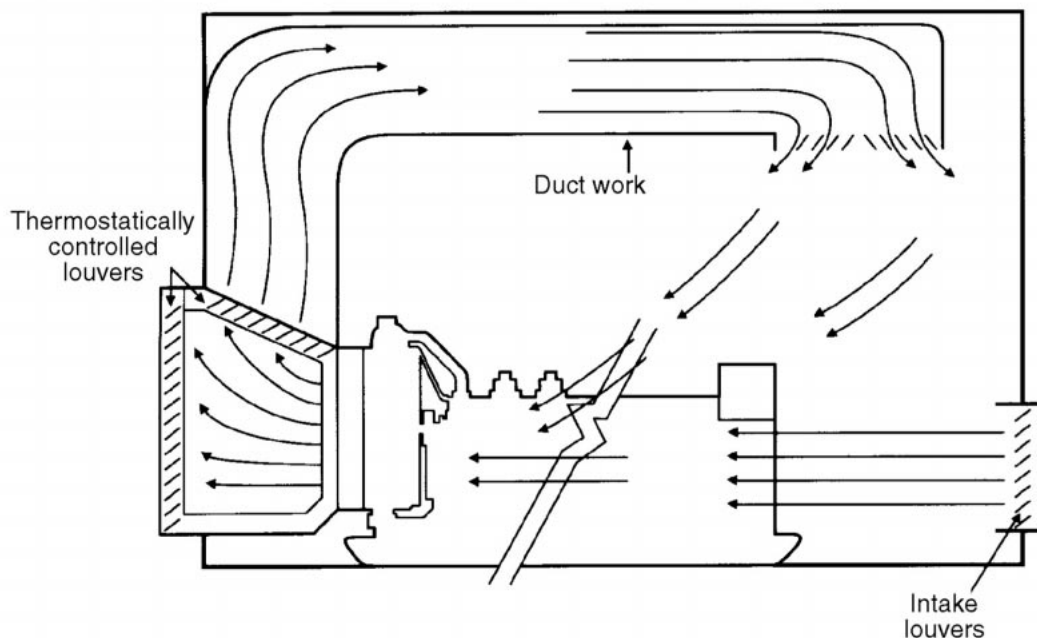


Figure 9. Radiator with thermostatically controlled louvers.

Radiators which are nearly square can provide the most effective fan performance. They can be installed with a minimum of unswept core area. As a general rule, keep core thickness to a minimum with a maximum of 11 fins per 2.54 cm (1.0 in.). Increasing the number of fins per cm (in.) does increase the radiator heat rejection for a given air velocity through the core but also increases the resistance to air flow.

While the most economical initial cost will be maximum core thickness and fins per cm (in.),

this involves higher fan horsepower with consequent operating cost and noise penalties throughout the life of the installation. In addition, a radiator with more fins per cm (in.) is much more susceptible to plugging from insects and debris.

#### Fan Sizing

As a general rule, the most desirable fan is one having the largest diameter and turning at the lowest speed to deliver the required airflow. This also results in lower fan noise and lowest fan horsepower draw from the engine.

Blade tip speed, while being only one of the elements of cooling fan design, is an item easily changed by choosing an appropriate fan drive ratio. An optimum fan tip velocity of 6096 cm/s (12,000 fpm) is a good compromise for meeting noise legislation requirements and cooling system performance requirements. Maximum acceptable tip speed is 7620 cm/s (15,000 fpm) for Caterpillar fans.

### Moveable Louvers

If moveable louvers are used, specify those which open in a positive manner. Pneumatic and electric-actuated louvers are satisfactory (see Figure 10).



Figure 10. Moveable louvers.

### Louver Operation

- Louvers which open from the discharge pressure of the radiator fan are discouraged. Rain, ice, and snow can render them inoperative within a short time and result in engine overheating and shutdown.
- Do not wait to activate the louvers until the engine warms up. In an emergency, the engine will be loaded immediately and require full air flow. Open the louvers as soon as the engine starts and install them to open fully in case of an emergency.
- Heat sensors needlessly complicate the system and their malfunction can reduce air flow to the engine which can cause shutdown.

### Enclosures

Enclosures trap radiated heat and direct it through the radiator decreasing cooling capabilities 8° to 10°C (14° to 18°F). Even with doors open, radiators can derate 5° to 7°C (9° to 13°F) when enclosed.

### Special 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 specially 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. Do not insulate engine turbochargers.

#### Test With Doors and Windows Closed

Ventilating systems must be designed to provide safe working temperatures and adequate air flow when windows, doors, and other normally closed ports are secured for bad weather conditions. Test the ventilation system fully secured for bad weather. This condition will reflect the most severe test of the ventilation system. Remember that a small room suction can exert a large pressure on an entrance door or window.

#### Ducting Considerations

Design all ducting to withstand extremes of vacuum or pressure and still maintain tight joints.

Provide inspection ports (or areas that are easily disassembled) to allow removal of foreign objects — especially for standby applications.

# Notes

# Notes



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