GE Power & Water
Gas Engines

Waukesha gas engines

Waukesha engines & **Enginator** systems installation Chapters 1-10

form 1091-11
11th edition

Original Instructions (English)
The English version of this manual controls over any error in or conflicting interpretation of any translation.

imagination at work

Waukesha gas engines
Waukesha, Wisconsin 53188
Printed in U.S.A.

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The engine exhaust from this product contains chemicals known to the state of California to cause cancer, birth defects or other reproductive harm.	Certain components in this product and its related accessories contain chemicals known to the state of California to cause cancer, birth defects or other reproductive harm. Wash hands after handling.

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NOTICE

Electrical and electronic equipment can contain harmful substances which can affect the environment and human health.

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You must not dispose any electrical and electronic equipment marked with this symbol (battery-operated electrical appliances, measurement equipment, light-bulbs, etc.) in the domestic waste but dispose of these separately. Always use the waste return and collection systems locally available and contribute to the reuse, recycling and all other forms of use for waste electrical and electronic equipment.

FORM 1091-11

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MOUNTING AND ALIGNMENT OF 275GL SERIES ENGINES

CHAPTER 1

SAFETY

SAFETY INTRODUCTION

The following safety precautions are published for your information. Waukesha does not, by the publication of these precautions, imply or in any way represent that they are the sum of all dangers present near industrial engines or fuel rating test units. If you are installing, operating, or servicing a Waukesha product, it is your responsibility to ensure full compliance with all applicable safety codes and requirements. All requirements of the Federal Occupational Safety and Health Act must be met when Waukesha products are operated in areas that are under the jurisdiction of the United States of America. Waukesha products operated in other countries must be installed, operated and serviced in compliance with any and all applicable safety requirements of that country.

For details on safety rules and regulations in the United States, contact your local office of the Occupational Safety and Health Administration (OSHA).

The words DANGER, WARNING, CAUTION and NOTICE are used throughout this manual to highlight important information. Be certain that the meanings of these alerts are known to all who work on or near the equipment.

Follow the safety information throughout this manual in addition to the safety policies and procedures of your employer.Indicates a procedure, practice or condition that should be followed in order for the engine or component to function in the manner intended.

This safety alert symbol appears with most safety statements. It means attention, become alert, your safety is involved! Please read and abide by the message that follows the safety alert symbol.

DANGER

Indicates a hazardous situation which, if not avoided, *will* result in death or serious injury.

WARNING

Indicates a hazardous situation which, if not avoided, *could* result in death or serious injury.

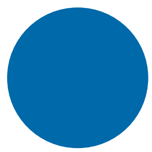
CAUTION

Indicates a hazardous situation which, if not avoided, *could* result in minor or moderate injury.

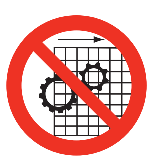
NOTICE

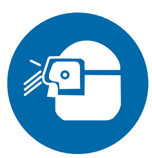
Indicates a situation which can cause damage to the engine, personal property and/or the environment, or cause the equipment to operate improperly.

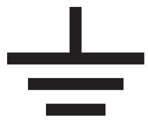
NOTE: *Indicates a procedure, practice or condition that should be followed in order for the engine or component to function in the manner intended.*

Symbol	Description
	A black graphical symbol inside a yellow triangle with a black triangular band defines a safety sign that indicates a hazard.
	A black graphical symbol inside a red circular band with a red diagonal bar defines a safety sign that indicates that an action shall not be taken or shall be stopped.
	A white graphical symbol inside a blue circle defines a safety sign that indicates that an action shall be taken to avoid a hazard.
Warnings	
	Safety Alert Symbol
	Asphyxiation Hazard
	Burn Hazard
	Burn Hazard (Chemical)
	Burn Hazard (Hot Liquid)
	Burn Hazard (Steam)

	Burst/Pressure Hazard
	Crush Hazard (Hand)
	Crush Hazard (Side)
	Crush Hazard (Side Pinned)
	Crush Hazard (Top)
	Electrical Shock Hazard
	Entanglement Hazard
	Explosion Hazard
	Fire Hazard
	Flying Object Hazard

	Hazardous Chemicals
	High-Pressure Hazard
	Impact Hazard
	Pinch-Point Hazard
	Pressure Hazard
	Puncture Hazard
	Sever Hazard
	Sever Hazard (Rotating Blade)
Prohibitions	
	Do not operate with guards removed

	Do not leave tools in the area
	Drugs and Alcohol Prohibited
	Lifting/Transporting only by qualified personnel
	Welding only by qualified personnel
Mandatory Actions	
	Read Manufacturer’s Instructions
	Wear Eye Protection
	Wear Personal Protective Equipment (PPE)
	Wear Protective Gloves

Miscellaneous	
	Emergency Stop
	Grounding Point
PE	Physical Earth
	Use Emergency Stop (E-Stop); Stop Engine

⚠ WARNING

The safety messages that follow have WARNING level hazards.

SAFETY LABELS

All safety labels must be legible to alert personnel of safety hazards. Replace any illegible or missing labels immediately. Safety labels removed during any repair work must be replaced in their original position before the engine is placed back into service.

EQUIPMENT REPAIR AND SERVICE

Always stop the engine before cleaning, servicing or repairing the engine or any driven equipment.

- If possible, lock all controls in the OFF position and remove the key.
- Put a sign on the control panel warning that the engine is being serviced.
- Close all manual control valves.
- Disconnect and lock out all energy sources to the engine, including all fuel, electric, hydraulic and pneumatic connections.
- Disconnect or lock out driven equipment to prevent the possibility of the driven equipment rotating the disabled engine.

Allow the engine to cool to room temperature before cleaning, servicing or repairing the engine. Some engine components and fluids are extremely hot even after the engine has been shut down. Allow sufficient time for all engine components and fluids to cool to room temperature before attempting any service procedure.

Exercise extreme care when moving the engine or its components. Never walk or stand directly under an engine or component while it is suspended. Always consider the weight of the engine or the components involved when selecting hoisting chains and lifting equipment. Be positive about the rated capacity of lifting equipment. Use only properly maintained lifting equipment with a lifting capacity that exceeds the known weight of the object to be lifted.

ACID

Always read and comply with the acid manufacturer's recommendations for proper use and handling of acids.

BATTERIES

Always read and comply with the battery manufacturer's recommendations for procedures concerning proper battery use and maintenance.

Batteries contain sulfuric acid and generate explosive mixtures of hydrogen and oxygen gases. Keep any device that may cause sparks or flames away from the battery to prevent explosion.

Always wear protective glasses or goggles and protective clothing when working with batteries. You must follow the battery manufacturer's instructions on safety, maintenance and installation procedures.

BODY PROTECTION

Always wear OSHA-approved body, sight, hearing and respiratory system protection. Never wear loose clothing, jewelry or long hair around an engine.

CHEMICALS

GENERAL

Always read and comply with the safety labels on all containers. Do not remove or deface the container labels.

CLEANING SOLVENTS

Always read and comply with the solvent manufacturer's recommendations for proper use and handling of solvents. Do not use gasoline, paint thinners or other highly volatile fluids for cleaning.

LIQUID NITROGEN

Always read and comply with the liquid nitrogen manufacturer's recommendations for proper use and handling of liquid nitrogen.

COMPONENTS

HEATED OR FROZEN

Always wear protective equipment when installing or removing heated or frozen components. Some components are heated or cooled to extreme temperatures for proper installation or removal.

INTERFERENCE FIT

Always wear protective equipment when installing or removing components with an interference fit. Installation or removal of interference components may cause flying debris.

COOLING SYSTEM

Always wear protective equipment when venting, flushing or blowing down the cooling system. Operational coolant temperatures can range from 180° – 250°F (82° – 121°C).

Do not service the cooling system while the engine is operating or when the coolant or vapor is hot. Operational coolant temperatures can range from 180° – 250°F (82° – 121°C).

ELECTRICAL

GENERAL

Equipment must be grounded by qualified personnel in accordance with IEC (International Electric Code) and local electrical codes.

Do not install, set up, maintain or operate any electrical components unless you are a technically qualified individual who is familiar with the electrical elements involved.

Disconnect all electrical power supplies before making any connections or servicing any part of the electrical system.

Always label "high voltage" on engine-mounted equipment over 24 volts nominal.

IGNITION

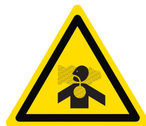
Avoid contact with ignition units and wiring. Ignition system components can store electrical energy, and if contacted, can cause electrical shock.

Properly discharge any electrical component that has the capability to store electrical energy before connecting or servicing that component.

EMERGENCY SHUTDOWN

An Emergency Shutdown must never be used for a normal engine shutdown. Doing so may result in unburned fuel in the exhaust manifold. Failure to comply increases the risk of an exhaust explosion.

EXHAUST



Do not inhale engine exhaust gases. Ensure that exhaust systems are leak-free and that all exhaust gases are properly vented to the outside of the building.

Do not touch or service any heated exhaust components. Allow sufficient time for exhaust components to cool to room temperature before attempting any service procedure.

FIRE PROTECTION

See local and federal fire regulations for guidelines for proper site fire protection.

FUELS

GENERAL

Ensure that there are no leaks in the fuel supply. Engine fuels are highly combustible and can ignite or explode.

GASEOUS

Do not inhale gaseous fuels. Some components of fuel gas are odorless, tasteless and highly toxic.

Shut off the fuel supply if a gaseous engine has been cranked excessively without starting. Crank the engine to purge the cylinders and exhaust system of accumulated unburned fuel. Failure to purge accumulated unburned fuel in the engine and exhaust system can result in an explosion.

LIQUIDS

Use protective equipment when working with liquids and related components. Liquids can be absorbed into the body.

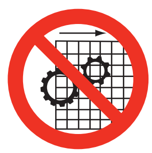
INTOXICANTS AND NARCOTICS

Do not allow anyone under the influence of intoxicants and/or narcotics to work on or around industrial engines. Workers under the influence of intoxicants and/or narcotics are a hazard to both themselves and other employees.

PRESSURIZED FLUIDS / GAS / AIR

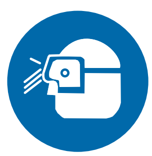
Never use pressurized fluids/gas/air to clean clothing or body parts. Never use body parts to check for leaks or flow rates. Observe all applicable local and federal regulations relating to pressurized fluids/gas/air.

PROTECTIVE GUARDS



Provide guarding to protect persons or structures from rotating or heated parts. It is the responsibility of the engine owner to specify and provide guarding. See OSHA standards on “machine guarding” for details on safety rules and regulations concerning guarding techniques.

SPRINGS



Use appropriate equipment and protective gear when servicing or using products that contain springs. Springs, under tension or compression, can eject if improper equipment or procedures are used.

TOOLS

ELECTRICAL

Do not install, set up, maintain or operate any electrical tools unless you are a technically qualified individual who is familiar with them.

HYDRAULIC

Do not install, set up, maintain or operate any hydraulic tools unless you are a technically qualified individual who is familiar with them. Hydraulic tools use extremely high hydraulic pressure.

Always follow recommended procedures when using hydraulic tensioning devices.

PNEUMATIC

Do not install, set up, maintain or operate any pneumatic tools unless you are a technically qualified individual who is familiar with them. Pneumatic tools use pressurized air.

WEIGHT

Always consider the weight of the item being lifted and use only properly rated lifting equipment and approved lifting methods.

Never walk or stand under an engine or component while it is suspended.

WELDING

Comply with the welder manufacturer's recommendations for procedures concerning proper use of the welder.

CAUTION **The safety message that follows has a CAUTION level hazard.**

Ensure that all tools and other objects are removed from the unit and any driven equipment before restarting the unit.

NOTICE **The safety messages that follow have NOTICE level hazards.**

Ensure that the welder is properly grounded before attempting to weld on or near an engine.

Disconnect the ignition harness and electronically controlled devices before welding with an electric arc welder on or near an engine. Failure to disconnect the harnesses and electronically controlled devices could result in severe engine damage.

CHAPTER 2

PREPARATION FOR MOUNTING

INTRODUCTION

Waukesha engines should be mounted on an inertia block or a concrete pad with spring isolators. These types of mounting are important as they help to isolate the engine and its vibration from the surrounding structure and from other machines. The inertia block or pad provides a level surface on which to mount the engine as well as a high level of isolation, which reduces the noise and vibration level transmitted to surrounding buildings and machines. Isolation is best achieved with well-dimensioned inertia blocks on springs. The concrete upper face shall be painted with hydrocarbon resistant paint to avoid concrete resistance properties alteration and/or finishing coping mortar stratification.

It is strongly recommended that the driven equipment be mounted on a common-skid with the engine (see Figure 2-1). By mounting both units on the same skid, a common plane for the engine and driven equipment is created. The equipment is less likely to lose alignment, because the driven equipment cannot shift relative to the prime mover (engine).

Waukesha strongly recommends the packager analyze skid design to determine that the structural integrity of the skid does not incur harmful natural frequencies for constant speed applications and throughout the speed range for variable speed applications.

To meet these demands, the inertia block or pad (spring isolated) must be of both adequate size and mass to support the engine/driven equipment and to absorb vibration. The engine/driven equipment common skid must rest on a surface of sufficient density to support both the common skid and the equipment mounted on it. The inertia block or mounting pad must have an accurately finished, level mounting surface. To secure the engine/driven equipment to the inertia block or mounting pad, properly sized retaining bolts must be installed in the correct spots to align with the holes in the engine base or common skid.

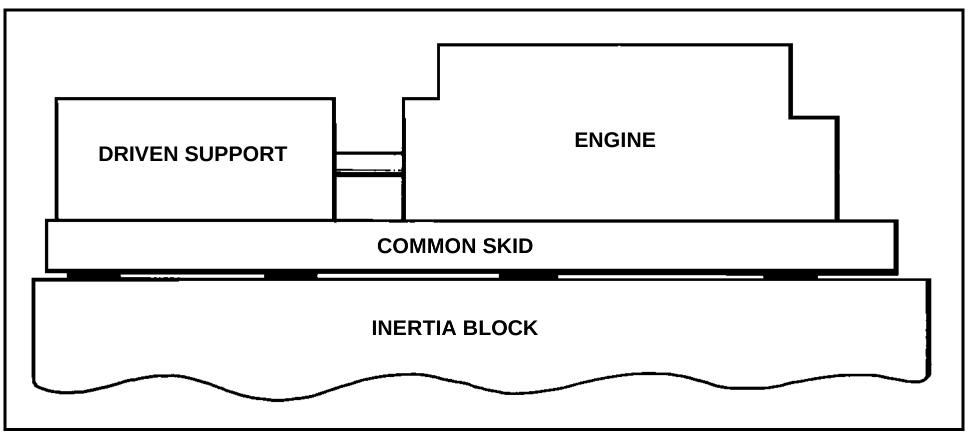


Figure 2-1 Engine and Driven Equipment on Common Skid

SECTION 1 DETERMINING INERTIA BLOCK OR PAD SIZE RECOMMENDED MINIMUM STANDARDS

Width of the inertia block or pad (W)

The inertia block or pad width is to be at least one foot (30.5 cm) wider than the base of the engine or the common skid to be installed.

Length of the inertia block or pad (L)

The inertia block or pad length is to be at least one foot (30.5 cm) longer than the combined length of the base of the engine and driven equipment to be installed.

Height of the inertia block or pad (H)

With the length and width of the inertia block controlled by the package dimensions, the height will be controlled by the desired weight of the block. Waukesha recommends using a foundation specialist to determine what inertia block weight and isolation will be required to minimize vibration transmitted to the surrounding environment. Waukesha provides engine unbalance forces and moments in the *Drive Data* section of the *General Tech Data Binder*. This information, along with the driven machine unbalance information would be required to properly calculate vibration transmission.

In the absence of calculations for the proper inertia block weight, Waukesha recommends the weight of the inertia block equal 1.3 to 1.5 times the weight of all equipment mounted on the inertia block or pad.

This includes accessory equipment and the weight of all liquids (coolant and oil) supported by the inertia block.

Weights of Liquids

Water...8.03 lb/gal (1.00 kg/liter)

Water/Glycol..8.55 lb/gal (1.02 kg/liter)

Lube Oil...7.60 lb/gal (0.91 kg/liter)

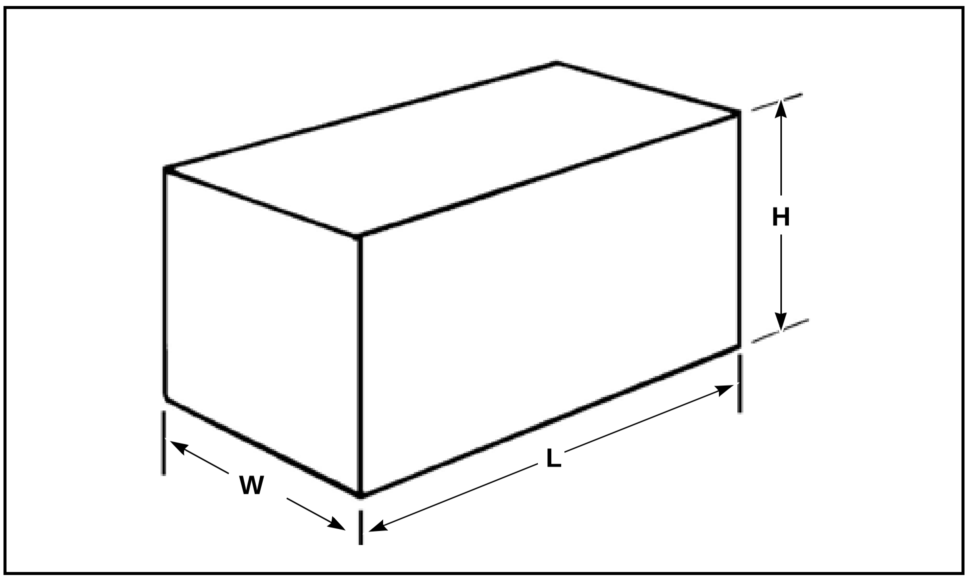


Figure 2-2 Schematic of Inertia Pad

The depth of the inertia block or pad may be found by the following:

U.S. Customary Formula

$$\mathbf{H} = \frac{(1.3 \text{ to } 1.5)M}{(L)(W)135}$$

Metric Equivalent Formula

$$\mathbf{H} = \frac{(1.3 \text{ to } 1.5)M}{(L)(W)\ 2162}$$

Where:

H = Depth of inertia block or pad (width plus one foot) in feet.

1.3 to 1.5 = Constant depending on type of engine. For more vibration prone engines such as diesels and V-8s, use the higher value. This will provide more mass for vibration dampening.

M = Weight of engine in pounds.

L = Length of inertia block or pad (engine or common skid length plus one foot) in feet.

W = Width of inertia block or pad (engine or common skid width plus one foot) in feet.

135 = Density of concrete (lbs. per cubic foot)

H = Depth of inertia block or pad in meters.

1.3 to 1.5 = Constant depending on type of engine. For more vibration prone engines such as diesels and V-8s use the higher value. This will provide more mass for vibration dampening.

M = Weight of engine in kilograms.

L = Length of inertia block or pad (engine or common skid length plus 30 cm) in meters.

W = Width of inertia block or pad (engine or common skid width plus 30 cm) in meters.

2162 = Density of concrete (kilograms per cubic meter).

Example: F3524GSI

M = 15,000 lb

L = 9.3 ft

W = 4.5 ft

$$\mathbf{H} = \frac{(1.4 \times 15,000)}{(9.3)(4.5)(135)}$$

$$\mathbf{H} = \frac{21000}{5650}$$

H = 3.75 ft deep

M = 6808 kg

L = 2.84 m

W = 1.37 m

$$\mathbf{H} = \frac{(1.4 \times 6808)}{(2.84)(1.37)(2162)}$$

$$\mathbf{H} = \frac{9531}{8412}$$

H = 1.13 m deep

The final decision on inertia block or pad size should be made only after calculating the weight of the inertia block or pad plus the weight of the equipment, and comparing this figure to the soil bearing load of the installation site.

SECTION 2 DETERMINING REQUIRED SOIL BEARING LOAD

The next step is to determine if the weight of an inertia block or pad of this size plus the weight of the engine (and driven equipment, if mounted on a common skid) exceeds the safe soil bearing load.

The necessary soil bearing load (S.B.L.) can be determined with the following formula:

$$S.B.L. = \frac{(2.5)(M + F)}{(W)(L)}$$

Where:

2.5 = Safety constant

M = Weight of engine

W = Width of inertia block or pad

L = Length of inertia block or pad

F = Weight of engine (or engine and driven equipment if mounted on a common skid – see Note 1)

The weight of the inertia block or pad (F) must first be determined.

The weight is determined by the following formula:

Weight of inertia block or pad = W x L x H x density of the concrete

Example: F3524GSI

F = 4.5 x 9.3 x 3.75 x 135 lb/ft^3	F=1.37 m x 2.84 m x 1.13 m x 2162kg/mr^3
F = 21187 lb	F = 9505 kg

Now that "F" is known, the required soil bearing load can be determined using the given formula.

$$S.B.L. = \frac{(2.5)(M + F)}{(W)(L)}$$

$$S.B.L. = \frac{(2.5)(15,000 + 21,187)}{4.5 \times 9.3} \qquad S.B.L. = \frac{(2.5)(6808 + 9505)}{1.37\ m\ \times 2.84\ m}$$

$$S.B.L. = \frac{90467.5\text{ lb.}}{41.85\text{ sq. ft.}} \qquad S.B.L. = \frac{(40782.5\text{ kg.})}{(3.89\ m^2)}$$

Required S.B.L.=2161.7lbs/sq.ft. Required S.B.L. = 10483.9 kg/m^2

NOTE: *1. The above example only takes into account the weight and size of the engine. An actual installation would have to include the weight of the engine and the driven equipment, and the weight of a common mounting skid large enough to support both the engine and driven equipment.*

2. An equivalent calculation applies for driven equipment mounted on a separate skid.

Now that the required soil bearing load has been determined, see Table 2-1 to determine if the supporting material at the engine site can support the weight.

If the required soil bearing load exceeds suggested standards, footings may have to be incorporated to give the inertia block or pad a larger support area (see Figure 2-3).

Table 2-1

NATURE OF SUPPORTING MATERIAL	SAFE BEARING CAPACITY	
	(LBS. PER SQUARE FT.)	KG/M^2
Hard rock – Granite, etc.	50,000 – 200,000	240,000 – 980,000
Medium rock – Shale, etc.	20,000 – 30,000	100,000 – 150,000
Hard pan	16,000 – 20,000	80,000 – 100,000
Soft rock	10,000 – 20,000	50,000 – 100,000
Compacted sand & gravel	10,000 – 12,000	50,000 – 60,000
Hard clay	8,000 – 10,000	40,000 – 50,000
Gravel & coarse sand	8,000 – 10,000	40,000 – 50,000
Loose, medium and coarse sand, compacted fine sand	6,000 – 8,000	30,000 – 40,000
Medium clay	4,000 – 8,000	20,000 – 40,000
Loose fine sand	2,000 – 4,000	10,000 – 20,000
Soft clay	2,000	15,000

NOTE: *This table gives approximate values for average conditions. Building code requirements may vary and should be consulted for a particular locality.*

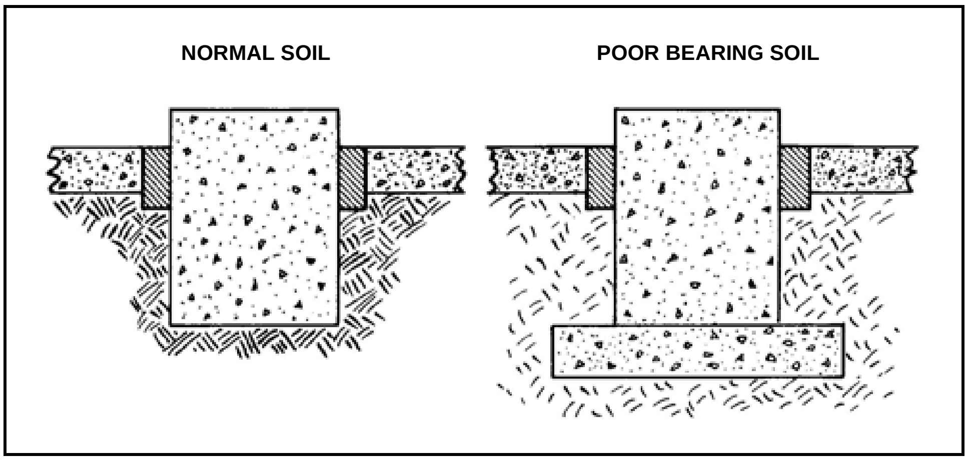


Figure 2-3 Footing for Poor Bearing Soil

SECTION 3

CONCRETE MIXTURE

Use one part cement, two parts sand and three parts aggregrate by volume, with a maximum slump of 4 in. (100 mm) providing a 28-day compressive strength of 3000 psi (211 kg/cm^2).

SECTION 4

INERTIA BLOCK REINFORCEMENT

The concrete reinforcing network should be a 10 in. x 10 in. (254 mm x 254 mm) steel wire fabric or equivalent which is 0.155 in. (3.9 mm) diameter minimum. It should be placed 2 in. (51 mm) from the top and bottom surfaces with each level spaced 6 in. (152 mm) apart.

An alternate method of reinforcing is to place a level of 3/4 in. (19 mm) diameter reinforcing rod, or equivalent, on 6 in. (152 mm) centers in both directions. A level should be placed 2 in. (51 mm) from the top and bottom surfaces. Rod placement should take into consideration interference with inertia block or pad mounting bolts and sleeves.

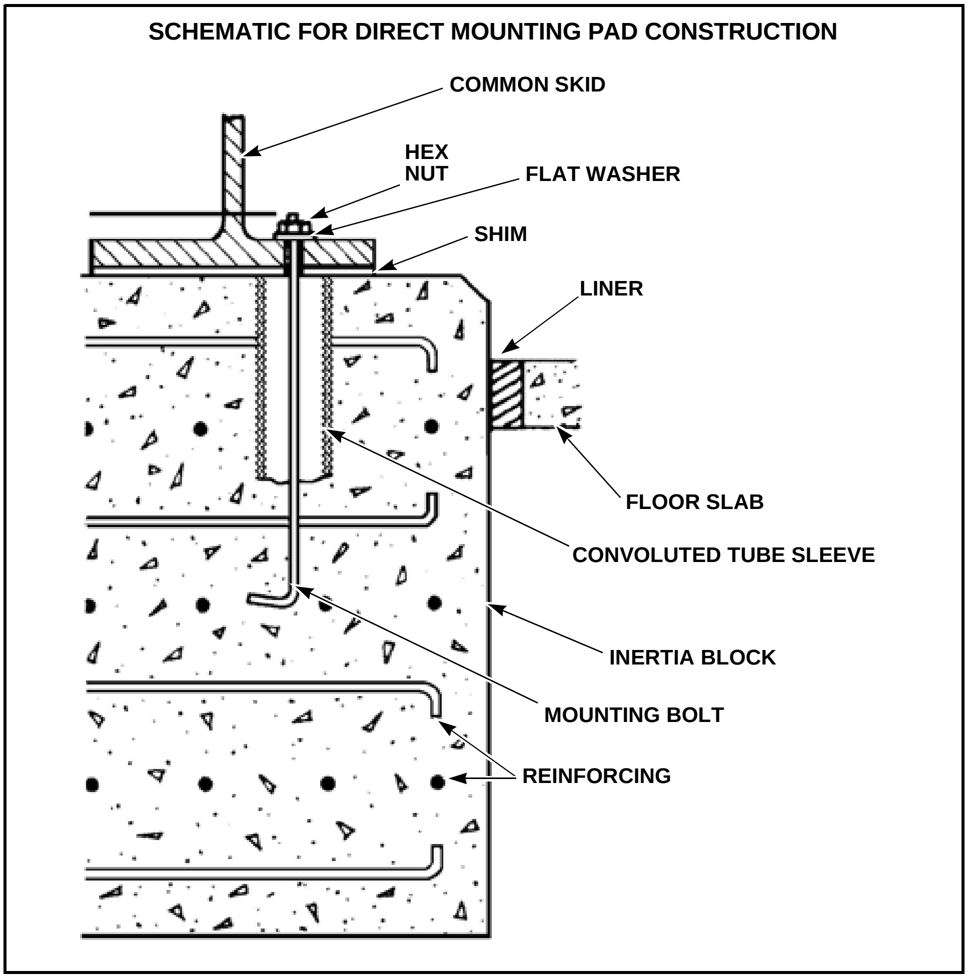


Figure 2-4 Common Skid Mounted Directly to Inertia Block or Pad

SECTION 5

VIBRATION ISOLATION

The inertia block or pad (spring isolated engine) is an important factor in isolating engine vibration from the surrounding structure. Many times however this is not enough. There are several additional techniques that can be used to isolate the vibration.

Isolating Liners

A liner can be fabricated and used to line the pit into which the concrete inertia block is poured (see Figure 2-5). A number of suitable liners are available commercially. Consult the liner manufacturer for specific information. The principle for all liners is the same – line the bottom and sides of the pit, and pour the concrete inertia block inside of the isolator lining. The engine and/or common mounting skid will still vibrate, but the vibration is dampened and largely confined within the liner.

Be sure to construct the liner so that no liquid concrete can flow into gaps between the liner slabs. If concrete seeps between the inertia block and the pit, the vibration absorption value of the liner will be greatly reduced.

Other materials such as sand or gravel may be used as isolating mediums. One foot of well tamped, settled gravel under the inertia block will be satisfactory.

Do not bridge the gap between the inertia block and the surrounding floor with concrete or a similar solid material. If for reasons of neatness or appearance it is necessary to close this gap, use an expansion joint or a similar resilient material.

Isolation of inertia block from the building, convoluted tube sleeve and anchor bolt placement, and a mounting pad area greater than engine base area may be noted in this illustration.

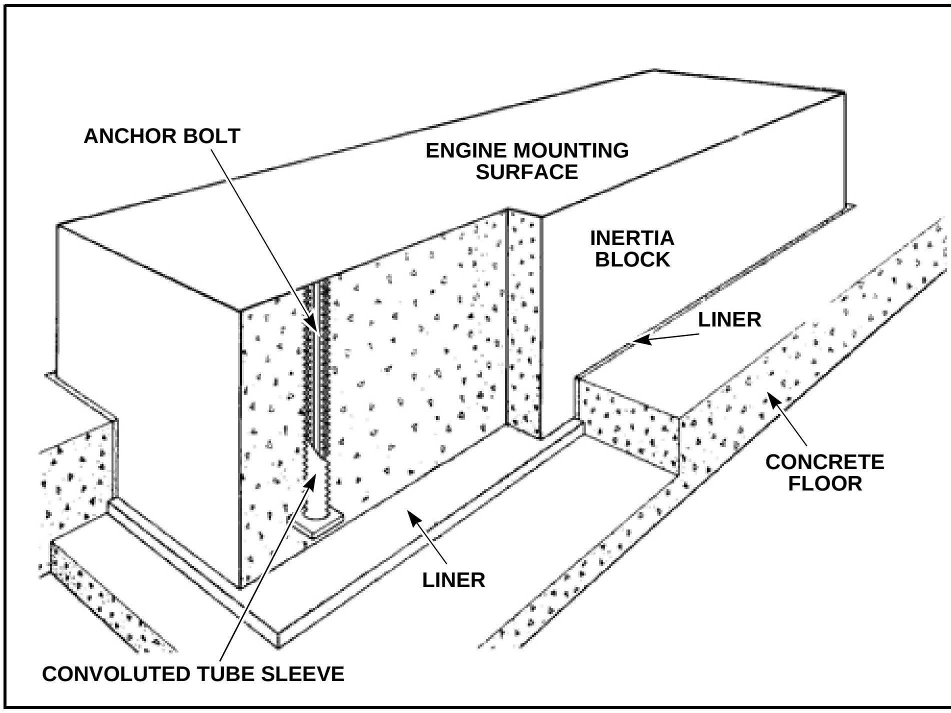


Figure 2-5 Cross Section of Concrete Inertia Block

Spring and Rubber Mounts

Spring and rubber mounts of various sizes and resiliencies are available for installation purposes. These mounts can be positioned between the common skid and the inertia block or pad or between the inertia block and bottom of the pit (see Figure 2-6). As with the isolating liners, we recommend contacting the manufacturer of the mounts for specific instructions.

For units installed in basements or on ground floors (no other floors beneath), neoprene waffle type pads (50% vibration reduction) or the sandwich type pad of rubber and cork (75% vibration reduction) can be used. Where engine-generator sets are to be installed above the ground floor, the more critical type of isolators should be used. For units up to 270 kilowatts (200 kWb), the type of isolator made of rubber bonded to metal can be used and will provide about 90% isolation. Larger units should use spring type vibration isolators that provide about 95% isolation. All percentages are approximate and exact information for your particular application should be discussed with your Waukesha Distributor to be certain that the right type of isolator is selected.

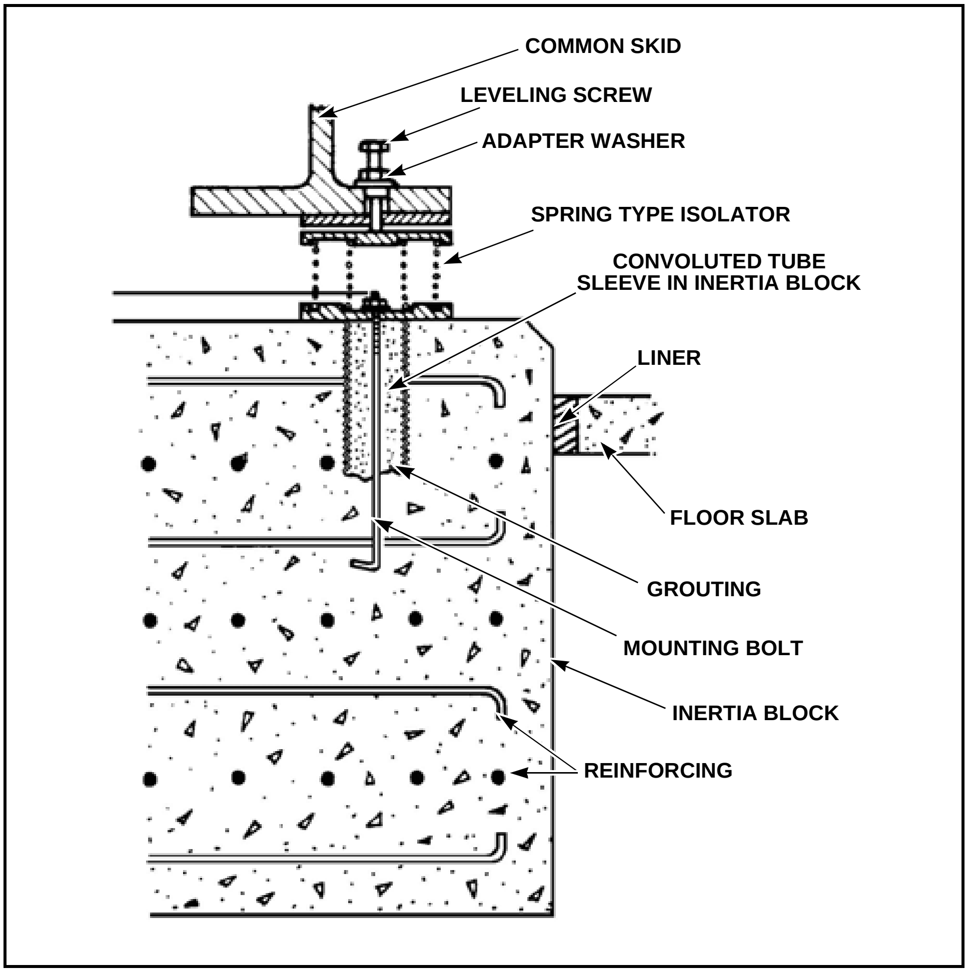


Figure 2-6 Schematic Spring Isolator Mounting Pad Construction

SECTION 6 INERTIA BLOCK BOLT OR PAD MOUNTING BOLT INSTALLATION

The inertia block or pad mounting bolts should be of an SAE grade 5 bolt material. The bolt diameter will be determined by the hole diameter in the engine mounting base or common skid frame. The bolts should be long enough to provide a minimum embedded length of 30 times the bolt diameter, plus 3 – 4 in. (76 – 102 mm) for a hook. (The bolt should have a “J” or “L” shaped hook on the non-threaded end to increase its holding power.) Approximately 7 in. (178 mm) more is needed to protrude above the top surface of the inertia block or pad. This 7 in. (178 mm) will provide the length needed for:

- The grout (if used), 2 in. (51 mm)
- Sole plate (if used), 3/4 in. (19 mm)
- Chock, 1/2 in. (13 mm)

- Shims and engine base, 1-3/4 in. (44.5 mm)
- Washer, nut and small variations in levelness, 7/8 in. (22 mm)

SCHEMATIC FOR PAD VIBRATION ISOLATOR
MOUNTING BASE CONSTRUCTION

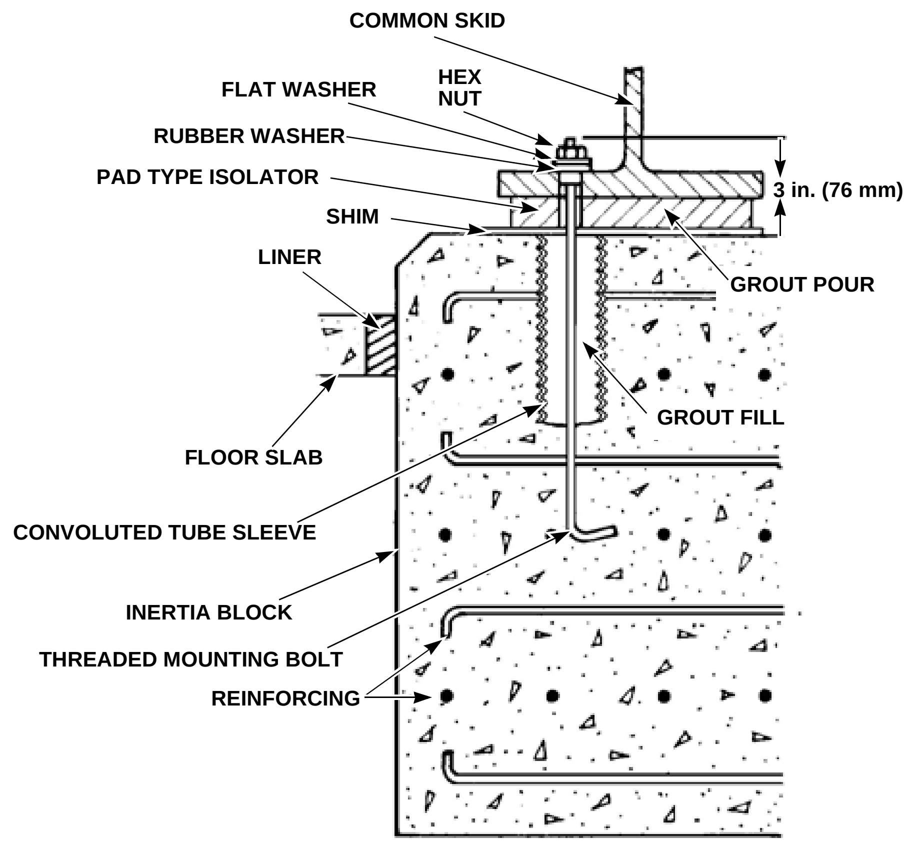


Figure 2-7 Common Skid Mounted on Pad Type Vibration Isolators

For a common skid mounted engine, only 7 in. (140 mm) of bolt need protrude above the inertia block or pad surface (see Figure 2-8).

Bolt placement in the inertia block or pad can be determined by making a template from 1 x 6 in. (25 x 152 mm) boards. Consult a Waukesha installation print for template information. (A certified installation print can be made for your engine if ordered when the engine is ordered.) Suspend the template over the inertia block or pad and hang bolts and sleeves through the template holes (see Figure 2-9). 7 in. (178 mm) of bolt must extend from the top surface of the inertia block or pad.

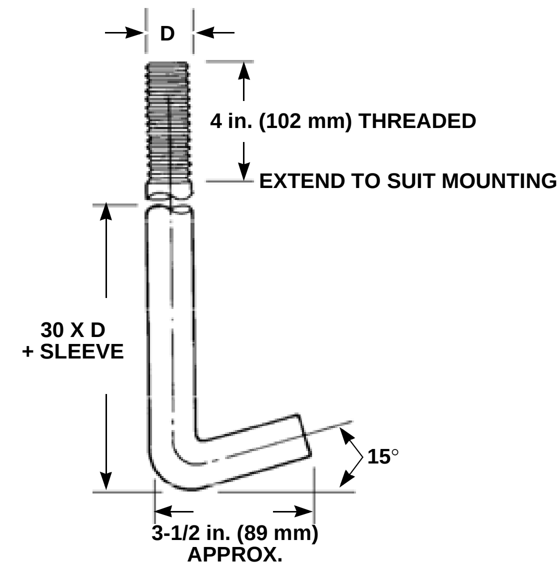


Figure 2-8 Mounting Bolt

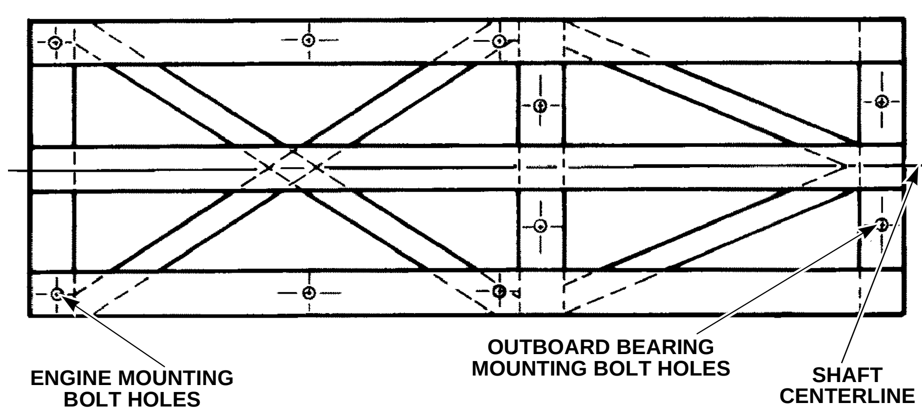


Figure 2-9 Template

A sleeve of convoluted plastic tubing 2 – 3 in. (51 – 76 mm) in diameter, should be placed around the bolts before they are embedded in the concrete (see Figure 2-10). This will allow the bolts to bend and conform to the dimensions of the sole plate (if used) if the template was not exact. The sleeve may be 10 – 12 in. (254 – 305 mm) long. The top end of the sleeve should be slightly above the top level of the inertia block or pad so that the concrete will not spill into the sleeve and interfere with bolt adjustments.

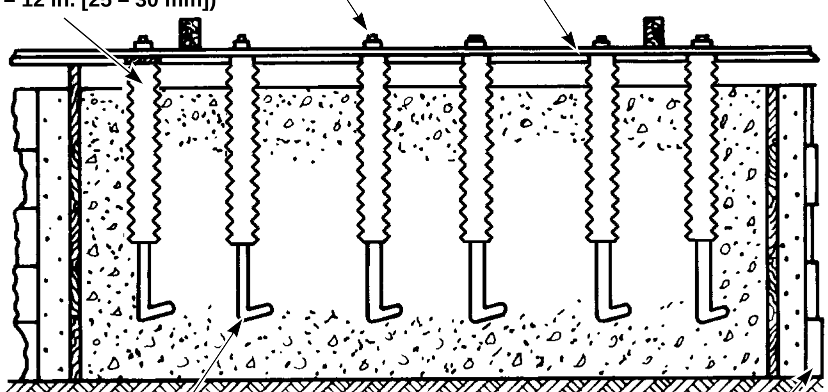


Figure 2-10 Mounting Sleeves Embedded in Concrete

SECTION 7 CURING THE INERTIA BLOCK OR PAD

Once the inertia block or pad is poured, it should be kept moist and protected until fully cured according to the supplier's requirements. A longer curing period may be required in adverse weather.

Inertia blocks or pads poured in the winter must be insulated against the cold or have calcium chloride incorporated into the mix.

Before the concrete curing advances too far, rough up the concrete surface to provide a good bonding surface for the grout (if used).

SECTION 8 SOLE PLATES

Sole plates can be used to mount the engine to the inertia block (see Figure 2-11). The plates distribute the weight of the engine evenly over the top of the inertia block or pad. They also make up for any variations of the concrete from level. When selecting material stock for the sole plates, select cold rolled steel 3/4 – 1 in. (19 – 25 mm) thick, and 4 in. (102 mm) wide minimum. The plates should run the full length of the engine.

If the engine is common skid mounted, it may be less expensive to use several shorter sole plates (if required). The plate should be as wide as the common skid flange. Sole plate lengths are available on Waukesha installation drawings.

The sole plates should be clean and free from rust and scale. Mounting holes in the plates should be drilled and tapped according to the instructions provided. Jack screws are to be used in these holes which keep the sole plates in position while pouring the grout. Before the inertia block or pad is fully cured, the surface should be roughened up to provide for a good bond between the concrete and the grout. Position the sole plate over the inertia block or pad bolts, and level the plates, keeping them a minimum of 2 in. (51 mm) above the inertia block or pad surface. Plates must be level lengthwise, and crosswise, relative to each other. After leveling, tighten the nuts on the inertia block or pad bolts finger-tight. This will help keep the sole plates level while installing the grout.

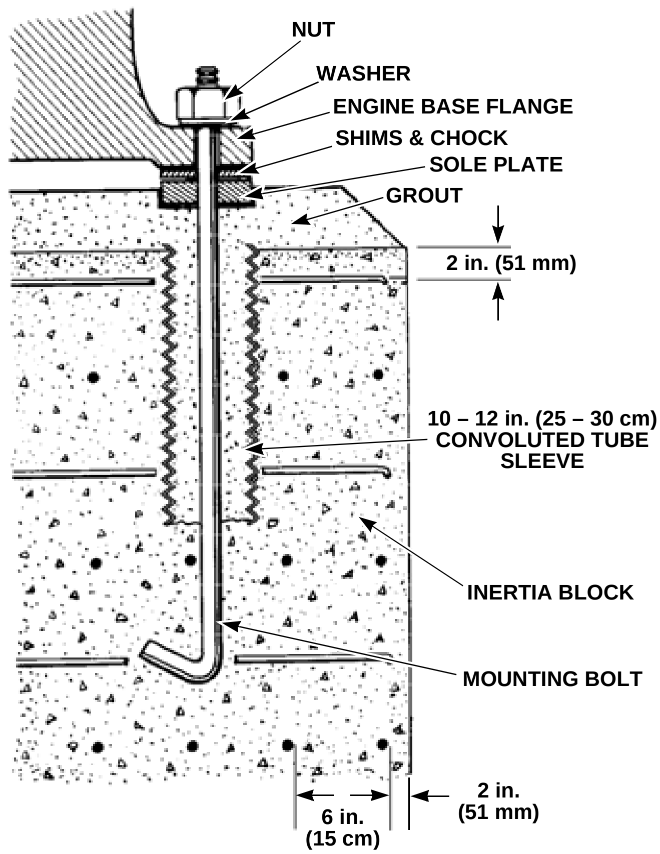


Figure 2-11 Cross Section of Mounting Using Sole Plates

SECTION 9

GROUTING

Grouting can be done only after the installation of the inertia block or pad has fully cured and the sole plates (if used) have been positioned and leveled (see Figure 2-12). On sole plate installations, grouting is important as it anchors the sole plates in place. Since the sole plates support the engine, it is important that the grout be installed properly to hold the plates level.

Engines and common skids can be mounted directly to the grout without the use of sole plates. When this is done, the engine must be mounted and leveled before the grout is poured. Shim and level the engine as described in Chapter 3: Mounting and Alignment. Pour the grout under the engine base or common skid. After all grout has cured, back out the jacking screws and fill with grout.

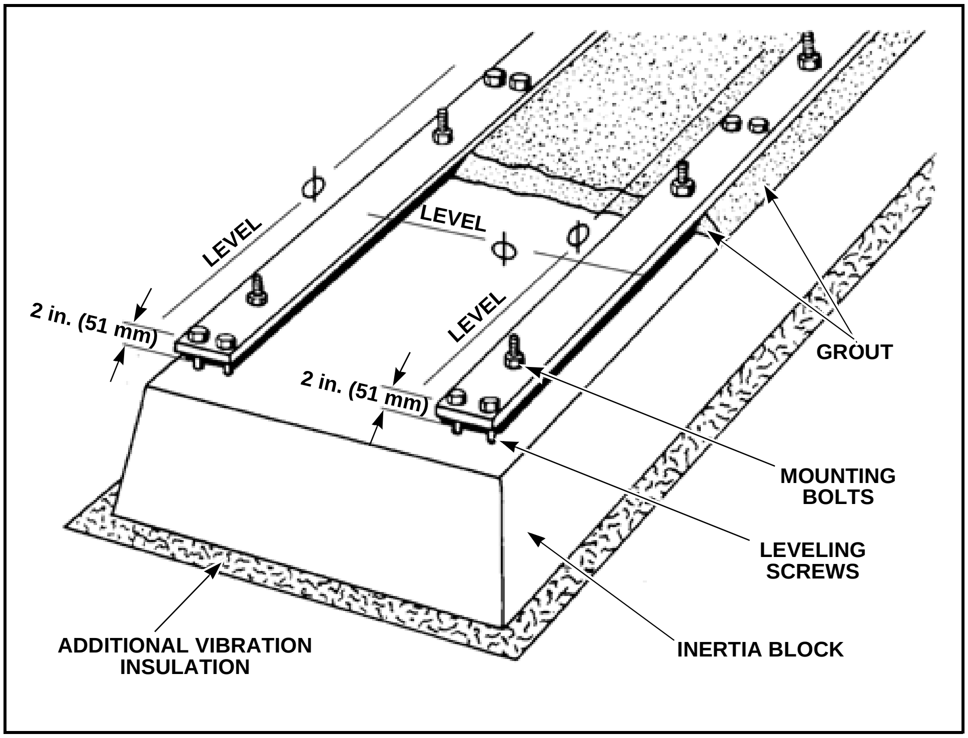


Figure 2-12 Grouting the Inertia Block

SECTION 10

GROUTING PROCEDURE

Make a form around the inertia block or pad. If possible, pour the grout from one point on the inertia block or pad only, and allow the grout to flow under the common skid or engine base rails. This pouring procedure will help lessen the chances of air pockets being trapped between the engine and the inertia block or pad. Air pockets will lessen the contact area between the grouting and the engine base or common skid, reducing support for the engine. Also, a metallic based grout will expand into these spaces and force the engine out of alignment. If the pour point on the engine or common skid is slightly higher than the rest of the inertia block or pad, the grout will flow more easily under the engine or common skid.

The best way to install a concrete, metallic based grout is to form wedge shaped grout pads (see Figure 2-13). These pads should run the length of the engine or common skid. Slope the grout outward in a wedge shape towards the inertia block or pad to provide better support. Sole plates can be embedded in this run of grout, or the engine base can be installed directly on it.

The advantage of this grouting technique is that it will keep grout out from under the engine. The grout will not be able to expand up into the hollow area under the engine base and force the engine out of alignment.

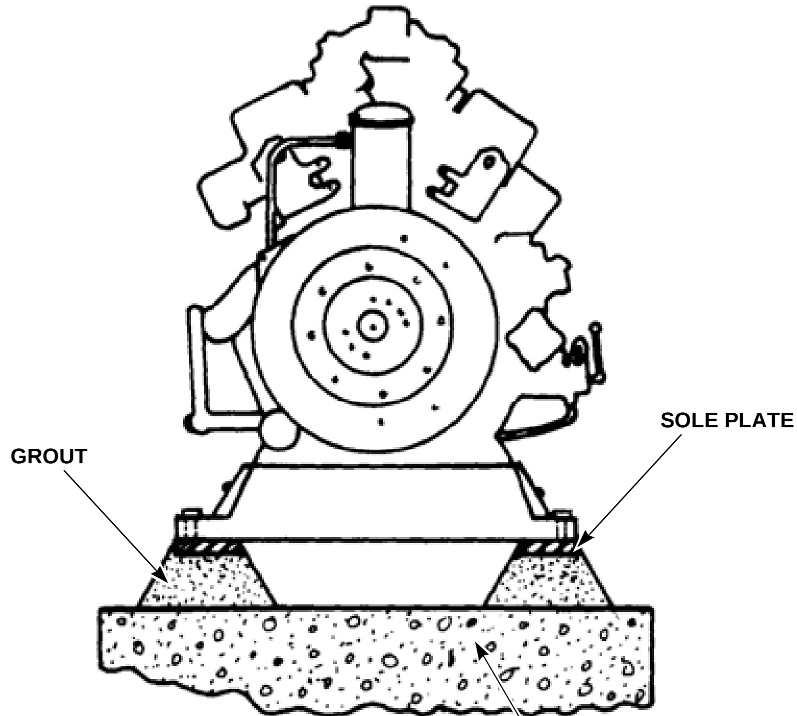


Figure 2-13 Rear View of Mounted Engine

Grouting should be worked into place using rods or chain lengths. Work the material gently to avoid air entrapment.

When using sole plates, pour in enough grout to embed the plates 1/2 in. (13 mm) into the grout. When sole plates are not used, never allow the grout to come up over the engine base or common skid, to allow for future adjustments.

Follow the grout manufacturer's instructions for applying the grout, and recommendations for curing times. Concrete grouts must be sealed after curing. All metallic based grouts should be sealed to prevent rust from destroying the grout.

If the grout is allowed to settle at a slight outward slope, oil and water will be able to run off the inertia block or pad.

After the grout has cured, remove the leveling screws and remove any accumulation from the common skid or engine base. Save enough grout to pour into the inertia block bolt sleeves after the engine has been aligned.

Many epoxy grouts are also available which provide superior performance for these applications.

SECTION 11 ANGULAR OPERATING LIMITS

Generally it is advised that an engine should be placed level which means that the earth's gravity is working perpendicular to the mounting surface of the engine. For applications where this is not possible care has to be taken that the engine, the driven equipment and the periphery are designed to withstand the actual angle of operation. This includes any bending of flex connections and the forces involved. Isolator springs generally are not designed for high angles of operation. A very important item is the engine's lubrication system as the oil level seen by the oil pump pick-up screen will be influenced. Table 2-2 gives the limit values for Waukesha's bare engines. Please contact Waukesha Application Department for any further information on the admissible angle of operation for engines, Enginators and accessories.

Angular operating limits must be maintained to assure a constant supply of oil to the oil pump pickup screen (see Figure 2-14). The oil in the pan will always flow to the lowest possible point. If the engine is not level, the oil will flow to a point where the pickup screen will not be able to pick it up. This means a loss of lubrication at the bearings and other vital engine parts.

See Chapter 6 for specific details on the Lubrication System.

For this reason, maximum permissible angles at which the engine can operate without loss of oil to the oil pickup screen have been established.

Engines can be modified to operate at greater angles by adding deep sump oil pans and special oil pumps. More information on these modifications is available from your Waukesha Distributor.

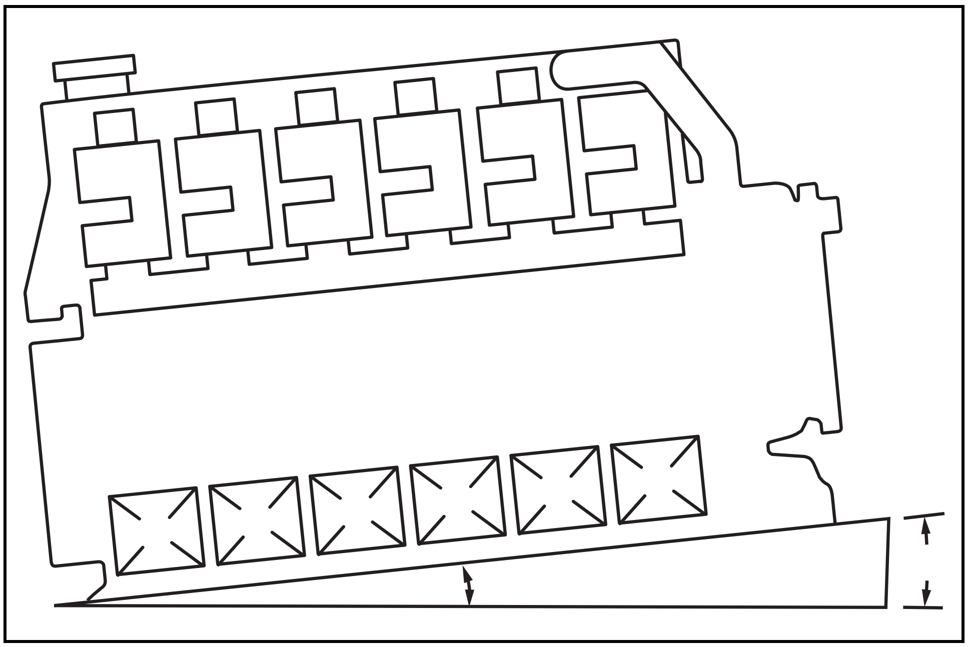


Figure 2-14 Determine Angle of Operation

SECTION 12

DETERMINING THE ANGLE OF ENGINE OPERATION

To determine the angle that an engine is operating at, the length of the engine mounting base and the height of elevation will be needed. The engine mounting base length can be found in an installation drawing available from Waukesha. The height of elevation will have to be measured.

The angle of operation is calculated using the following procedure:

H = height of elevation

L = length of engine mounting base

1. Divide H (height of elevation) by L (length of engine mounting base).

NOTE: *"H" and "L" must always be the same measuring unit (e.g., inches and inches; millimeters and millimeters, etc.) when doing this calculation.*

2. Follow down column A (See Table 2-1) and locate the number calculated in Step 1. If the exact figure is not listed, locate the next highest number.
3. The corresponding figure in column B will be the angle of operation. See the Angular Operation Limits Table (See Table 2-2) to determine the acceptability of the installation.

NOTE: *The above procedure uses the mathematical principles of trigonometry. The figure in column B is the sine of the angle listed in column A.*

Table 2-1 Sines of Angles

A (SINE)	B (ANGLE)
0.0175	1
0.0349	2
0.0523	3
0.0698	4
0.0872	5
0.1045	6
0.1219	7
0.1392	8
0.1564	9
0.1736	10
0.1908	11
0.2079	12
0.2249	13
0.2419	14
0.2588	15
0.2756	16
0.2924	17
0.3090	18
0.3256	19
0.3420	20
0.3584	21
0.3746	22
0.3907	23
0.4067	24
0.4226	25
0.4384	26
0.4540	27
0.4695	28
0.4848	29
0.5000	30

Table 2-2 Angular Operation Limits Table

INDUSTRIAL ENGINES EQUIPPED WITH STANDARD OIL PAN AND OIL PUMP				
MODEL[1]	**FRONT DOWN DEGREES[2]**	**REAR DOWN DEGREES[2]**	**LEFT DOWN[3]**	**RIGHT DOWN[3]**
F11	12	12	12	12
F18, H24	1	1	7	7
L36, P48	1	1	6	6
APG1000/16V150LTD	1	1	6	6
F3521, L5790, L5794, L7042, L7044	2	2	7	7
P9390	1	2	7	7
12V275GL+	7	7	15	15
16V275GL+	7	7	15	15
16V275GL/GL+	5	5	15	15

NOTES 1: *Values apply to all model variations, i.e., G, GSI/D, GL/D, LT/D, unless otherwise noted.*

2: *Tabulated angle operation values are based on unidirectional tilt. For bidirectional tilt or allowable intermittent tilt consult Waukesha's Sales Engineering Department.*

3: *Left and right are as viewed when facing the flywheel.*

NOTE: *Note that operation under an angle will change the load distribution and forces on the mounting. Especially when using spring isolators this has to be accounted for.*

NOTE: *This information was obtained from S3549-J which should be checked for more details and the most recent data.*

SECTION 13 PREPARATION FOR MOUNTING CHECKLIST

Checking the inertia block size and weight

Inertia block size

1. Length ____________________
2. Width ____________________
3. Height ____________________
4. Volume (L x W x H) ____________________
5. Weight (Volume x density of concrete) ____________________

Unit size

6. Length ____________________
7. Width ____________________
8. Weights ____________________
   9. Engine (wet) ____________________
   10. Coupling ____________________

11. Driven equipment ______________________________
12. Other ______________________________
13. Total ______________________________
14. Does inertia block size exceed unit size by at least 1 ft in length and width? ______________________________
15. Is inertia block weight 1.3 to 1.5 times unit weight? ______________

Checking the soil conditions

16. What is the type of soil under the inertia block? ______________
17. What is the soil bearing capacity? ______________________
18. What is the inertia block bearing area? (L x W) ______________
19. What is the total weight of the inertia block plus the unit? __________
20. What is the soil bearing load? (total weight divided by bearing area) _____
21. Is the soil bearing load less than the soil bearing capacity __________
22. If soil bearing load is greater than the soil bearing capacity how is this being resolved? ______________________________

Checking the concrete mix

23. Does the mix have a 28 day compressive strength of at least 3,000 PSI (206 bar)? ______________________________

Checking the inertia block reinforcement

24. What size reinforcement bars or wire mesh is being used? _________
25. How many layers of rebars/mesh are being used? ______________
26. What is the spacing between the surface and the first layer of rebar/mesh? _
27. What is the spacing between each layer of rebar/mesh? __________

Checking the isolation system

28. Has the inertia block been isolated from its surroundings? __________
29. What type of vibration isolation is going to be used:
30. None ______________________________
31. Spring isolators between inertia block and surrounding ________
32. Spring isolators between inertia block and unit base _________
33. Rubber isolators between inertia block and unit base ________
34. Calculated isolation % ______________________
35. Desired isolation% ______________________

Checking the hold down method

36. Quantity of hold down bolts ______________________
37. Grade of hold down bolt______________________
38. What is the diameter of the bolts? ______________________
39. Does the position of hold down bolts match the unit base drawing?________
40. Is enough bolt length exposed to provide adequate thread engagement through grout, sole plate, shims, base, washer, nut, etc? ________________
41. Is enough bolt length, 30 diameters, imbedded to provide adequate strength? ______________________
42. Have provisions, sleeves, been made for location adjustment? ___________

Checking the cure

43. Has the concrete been allowed to cure for at least 7 days? ______________
44. Has the surface been prepared for isolators or grout? _________________

Checking the sole plates

45. Are sole plates going to be used? ______________________
46. Are they wide enough to match the engine foot? ______________________
47. Are they long enough to match the engine foot?______________________
48. Is the material at least 0.75 in. thick? ______________________
49. Are they properly grouted in place?______________________
50. Are they level, flat and their surfaces parallel? ______________________

Checking the grout

51. Is grout going to be used?______________________
52. Type of grout used?______________________
53. Planned grout thickness ______________________
54. Grout manufacturer’s minimum thickness recommendation ______________
55. Grout manufacturer’s maximum thickness recommendation______________
56. Is grout thickness within manufacturer's recommendation? ______________
57. Are grout dams in place?______________________
58. Are manufacturer’s instructions available? ______________________
59. Have manufacturer’s instructions been followed?______________________
60. Have all of the surfaces been grouted, including cross members, if required? ______________________
61. Has care been taken to ensure that no voids are left? _________________
62. Has the grout been allowed to cure?______________________
63. Has all shim stock and blocks been removed? ______________________

Checking the level

64. Will the unit be installed level?______________________

65. At what angle will it operate?____

66. At what angle (°) is the engine installed? ____________________________

CHAPTER 3

ENGINE MOUNTING AND ALIGNMENT

SAFETY

WARNING

Never walk or stand directly under an engine while it is suspended from a hoist.

Always lift engines using their approved lifting eyes. The 6-cylinder engines are equipped with two lifting eyes, one on each end, attached directly to the crankcase.

Twelve- and 16-cylinder engines are equipped with four lifting eyes, one bolted directly to the crankcase on each corner.

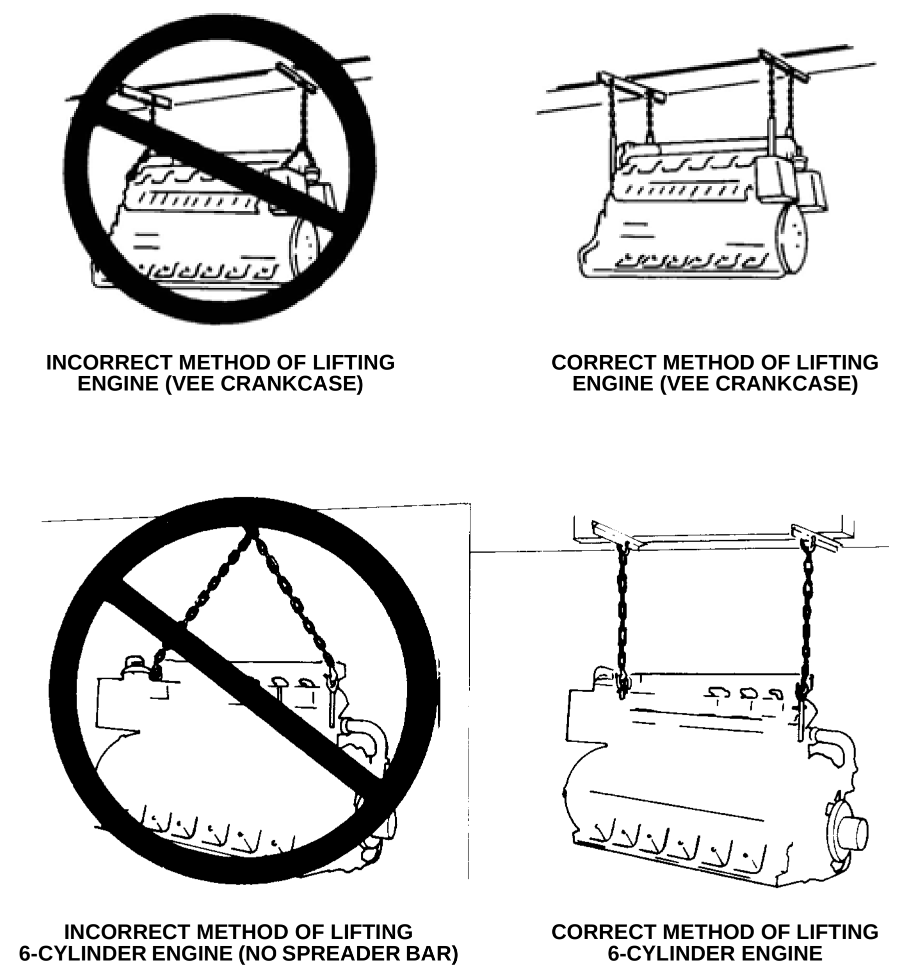


Figure 3-1 Engine Lifting Procedures

INTRODUCTION

Mounting and alignment of engine driven packages is vital to the overall service life and maintenance requirements of the engine, driven equipment, and various package accessories. It is essential that the mounting and alignment procedures are completed accurately. Patience is required during this process because it will require several adjustments to get the mounting and the alignment within specifications. Sufficient time must be provided prior to start-up to ensure that the mounting and alignment procedures are completed accurately.

Prior to beginning the alignment procedure, a visual inspection of the engine mounting system should be conducted. A properly designed inertia block or mounting pad is necessary to ensure adequate support of the equipment.

See Chapter 2 *"Preparation For Mounting".*

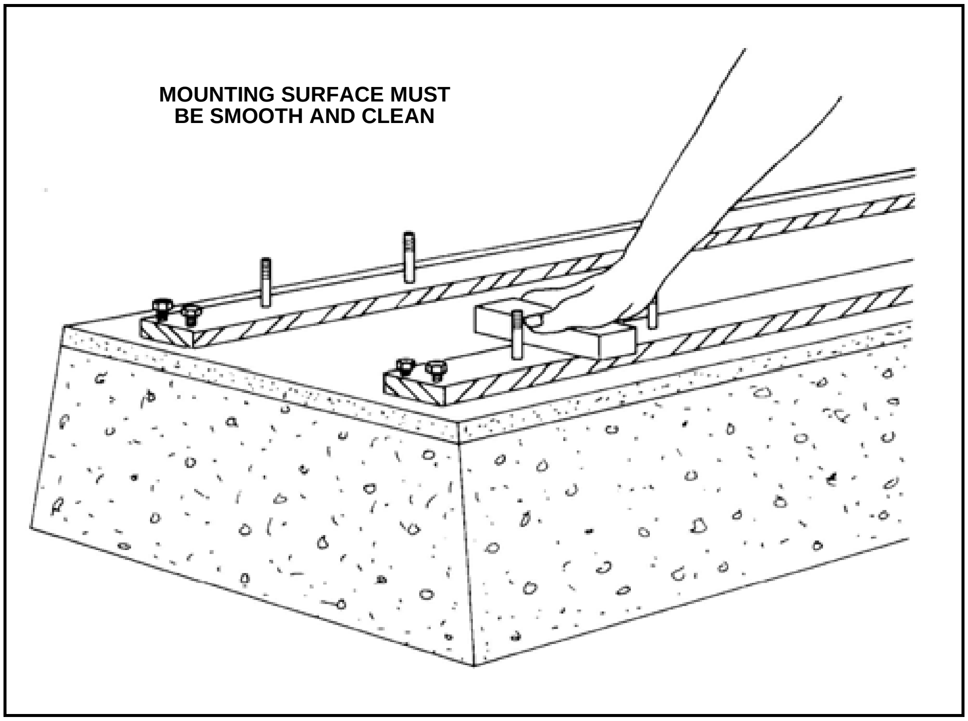
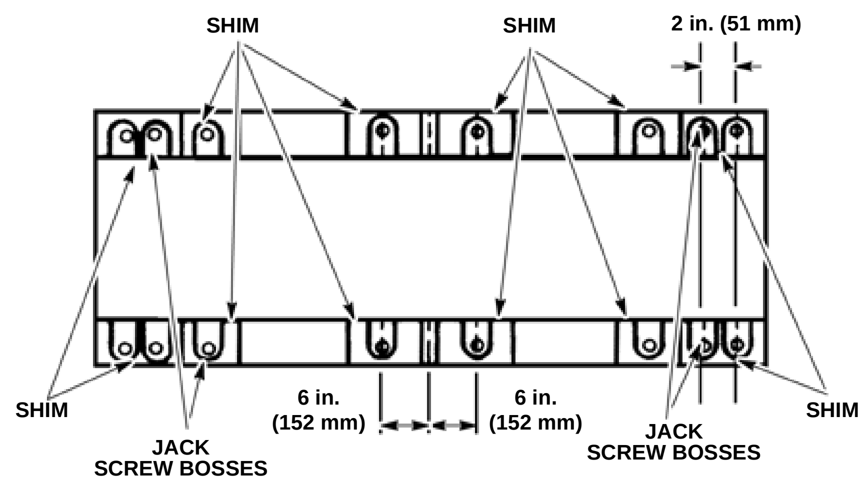


Figure 3-2 Mounting Pad

Engines and driven equipment are brought into alignment with each other through the use of shims used at the driven equipment mountings. Base deflection is adjusted through the use of shims at the engine corner, and center mounting points. The jack screws are used to provide the necessary clearance to install the shims.

Correctly aligning the shafts of two rotating machines will require the proper tools and hardware. A pair of dial indicators and a crankshaft deflection gauge are required for measuring alignment and base deflection. A crankshaft web deflection gauge is a specially designed dial indicator measuring in ten thousandths of an inch 0.0001 in. (0.003 mm), which mounts between the crankshaft webs. A 0 – 1 in. (0 – 25 mm) micrometer should also be available for measuring shim thickness.



NOTE: *Use two separate shim pads under the center mounts.*

Figure 3-3 Engine Shimming Procedure

SECTION 1

VHP STAINLESS STEEL SPACERS AND SHIMS

Shims can be made locally (see Figure 3-4), preferably of stainless steel in a size that adequately covers the engine base mounting pad. They should be sized in thickness so that no more than four of one size are necessary to equal, or surpass, the next larger size.

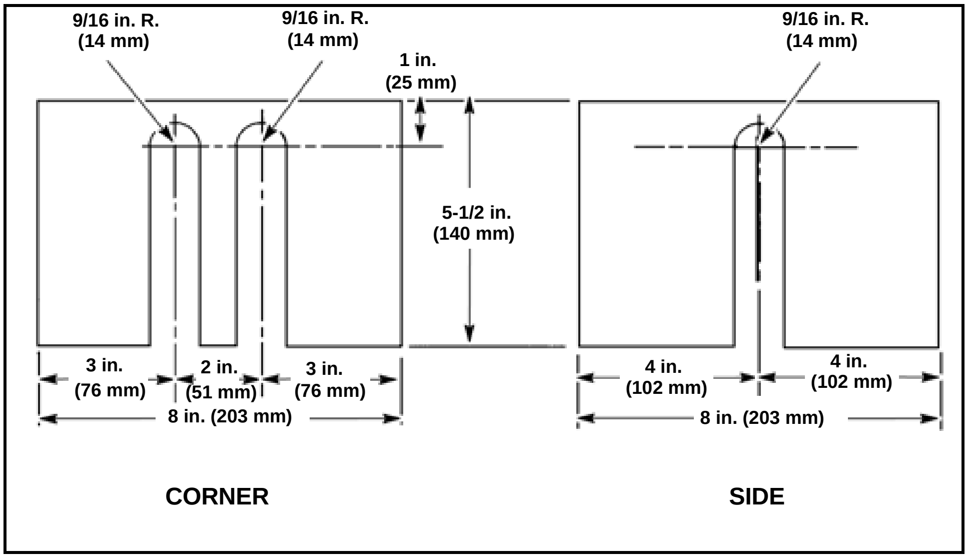


Figure 3-4 Shim Specifications

Stainless steel shims are shipped loose with Waukesha Generator Sets in thicknesses of 0.003 in. (0.076 mm), 0.010 in. (0.254 mm), and 0.025 in. (0.635 mm) for the generator, and 0.005 in. (0.127 mm) for the engine. Assortments of shims and spacers are available as service kits through Waukesha Power Systems under the following numbers:

Table 3-1 Engine Spacers (0.0625 in. [1.587 mm] Thick)

MODEL	QUANTITY USED		
	2 – CENTER POINTS	4 – CORNER POINTS	6 – ALL POINTS
VHP 6 & 12	14 in. (355.6 mm)	17 in. (431.8 mm)	–
VHP 16	–	–	14 in. (355.6 mm)

SERVICE KIT INFORMATION		
SIZE	QUANTITY	KIT NUMBER
14 in. (355.6 mm)	14	940-1
17 in. (431.8 mm)	12	940-2
FULL SET OF ABOVE SPACERS		940-3

Double slot VHP engine shims used on all four corners (second slot for jack bolts). VHP 6-, 12-, and 16-cylinder engines use 12 hold downs (use 4 double slot and 8 single slot). See Table 3-2.

Table 3-2 Engine Shims (Double Slot)

SIZE	QUANTITY	KIT NUMBER
0.002 in. (0.050 mm)	12	940-4
0.005 in. (0.127 mm)	24	940-5
0.010 in. (0.254 mm)	24	940-6
0.030 in. (0.762 mm)	12	940-7
FULL SET OF ABOVE SHIMS		940-8

Single slot VHP engine shims used on the four engine mounting bolts between jack bolts. Total eight bolts. VHP 6, 12, 16. See Table 3-3.

Table 3-3 Engine Shims (Single Slot)

SIZE	QUANTITY	KIT NUMBER
0.002 in. (0.050 mm)	24	940-9
0.005 in. (0.127 mm)	48	940-10
0.010 in. (0.254 mm)	48	940-11
0.030 in. (0.762 mm)	24	940-12
FULL SET OF ABOVE SHIMS		940-13

Table 3-4 Complete Engine Shim/Spacer Kits

MODEL	SHIM/SPACER KIT NUMBER
VHP 6	940-14
VHP 12	940-15
VHP 16	940-16

Each kit in Table 3-4 consists of:

1 SET (6) SPACERS

1 SET (12) 0.002 in. SHIMS (double slot and single slot)

2 SETS (24) 0.005 in. SHIMS (double slot and single slot)

2 SETS (24) 0.010 in. SHIMS (double slot and single slot)

1 SET (12) 0.030 in. SHIMS (double slot and single slot)

Driven Equipment Stainless Steel Shims

Table 3-5 5 x 5 in. (127.0 x 127.0 mm) with Slot for Up to a 1-1/4 in. (31.7 mm) Bolt

SIZE	QUANTITY	KIT NUMBER
0.002 in. (0.050 mm)	100	940-17
0.003 in. (0.076 mm)	100	940-18
0.010 in. (0.254 mm)	75	940-19
0.025 in. (0.635 mm)	50	940-20
0.060 in. (1.52 mm)	40	940-21
0.187 in. (4.74 mm)	25	940-22
FULL SET OF ABOVE		940-23

Table 3-6 4 x 4 in. (101.6 x 101.6 mm) with Slot for Up to a 1 in. (25.4 mm) Bolt

SIZE	QUANTITY	KIT NUMBER
0.002 in. (0.050 mm)	100	940-24
0.003 in. (0.076 mm)	100	940-25
0.010 in. (0.254 mm)	75	940-26
0.025 in. (0.635 mm)	50	940-27
0.060 in. (1.52 mm)	40	940-28
0.187 in. (4.74 mm)	25	940-29
FULL SET OF ABOVE		940-30

NOTE: *An alignment computer is available from Waukesha which will calculate the required adjustments to bring two multibearing machines into alignment. Contact your Waukesha Distributor for further information. This computer is also listed in the Waukesha Tool Catalog, under P/N 494359.*

Adjustable engine shims or chocks are suitable for mounting Waukesha gas engines provided the installer follows the sizing and installation guidelines of the adjustable shim manufacturer. All original engine mounting holes must be used, and the correct size for the size of the engine must be used. It is not acceptable to use a smaller size shim to allow for clearance around the engine mounting pad or original jacking bolt. Adjustable engine shims may not be used on VGF F18 and H24 engines due to the arrangement of the engine mounting foot.

Adjustable engine shims may loosen over time, and engine alignment must be checked periodically to ensure engine is in correct alignment at all times.

SECTION 2

PROCEDURES

Shimming

When shimming to adjust base deflection or alignment specifications, the shim packs should contain no more than four of one size shim. If more than four are required, the next larger thickness shim should be used. On VHP engines, separate shim packs must be used at each mounting bolt and may not always be the same thickness.

Dial Indicator Mounting

On skid mounted packages, tightening, loosening, and jacking of an engine mount during the shimming process will cause deflection of the I-beam flange. Because of this, it is important that the magnetic base or other clamping device for the dial indicator is attached to the web of the I-beam base rather than to the flange (dial indicator kit tool [P/N 494288]).

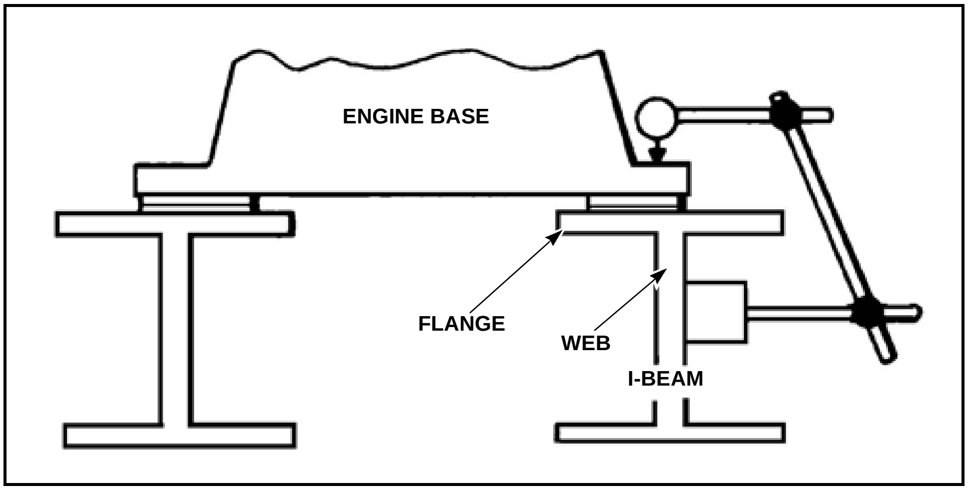


Figure 3-5 Correct Mounting

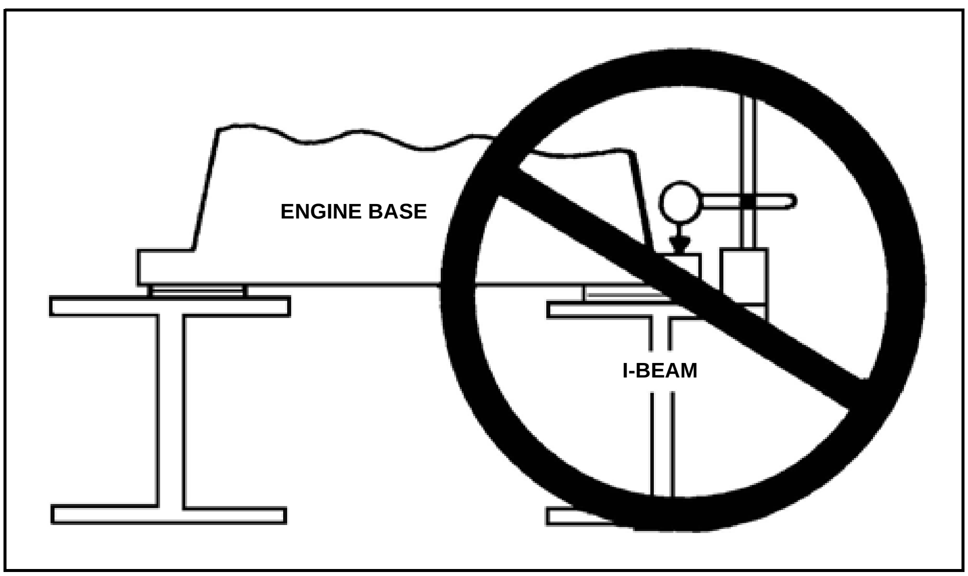


Figure 3-6 Incorrect Mounting

SECTION 3

LEVELING AND BASE DEFLECTION

Solid Mounted Packages

Solid mounted packages can be found in two arrangements:

- Engine and driven equipment are on a common skid which is bolted or grouted directly to an inertia block or support structure.
- Engine and driven equipment are individually bolted or grouted to sole plates on an inertia block.

Leveling – Common Skid-to-Inertia Block

1. Using a glass bubble level, check to see that the inertia block or support structure is even and level at all mounting points. Use spacing plates or shims where necessary.
2. Install the package on the inertia block. Use a glass bubble level to determine if the unit is level front to rear and side to side. Shim as required.
3. When unit is level, use a feeler gauge at each mounting point to determine if any air gaps exist. Shim as required.
4. Add shims under the center mounts of the common skid to eliminate any sag.
5. Tighten the common skid to the inertia block mounting bolts.
6. For grouting, see Chapter 2 *"Preparation For Mounting"*.

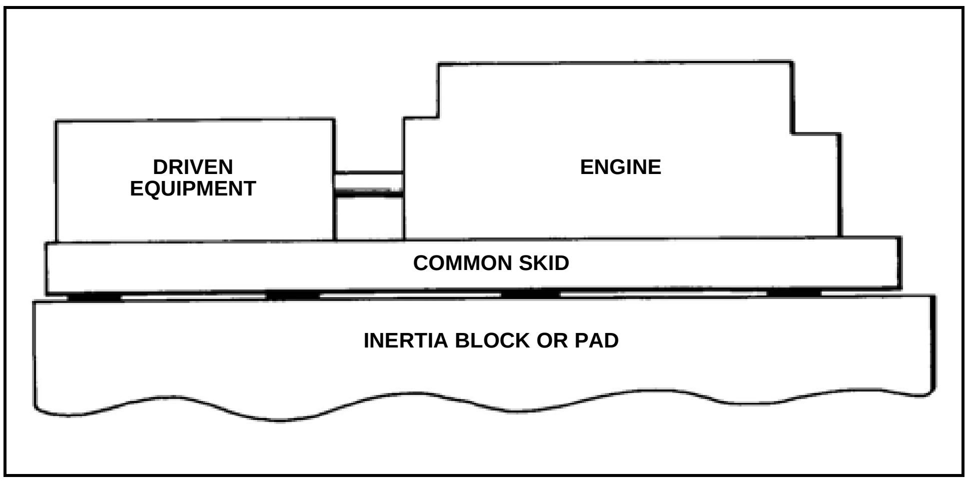


Figure 3-7 Leveling – Common Skid-to-Inertia Block

Leveling – Individual Mounting

Follow common skid procedures for each unit.

Engine Base Deflection

Checking engine base deflection is important to assure that the main bearing bores are in perfect alignment. Misaligned main bearing bores can cause premature failure of bearings and/or bending and breakage of the crankshaft. On solid mounted packages, the “Corner Lift Method” described below is quick and accurate for leveling an engine base and is, therefore, the preferred method. The “Release Method” is described for your information but is not considered as accurate as the “Corner Lift Method” for leveling an engine base on solid mounted packages.

Corner Lift Method

The following procedure provides a simple, quick method for 6 point mounting on solid mounted installations.

1. The engine should be resting on four corner shim packs at least 0.125 in. (3.175 mm) thick. Using the front or rear of the engine as a starting point, tighten the four corner bolts (two each side, on one end). The four bolts at the opposite end should be loosened or removed.

If a single bearing generator is attached, loosen the bolts connecting the generator adapter pilot ring to the flywheel housing.

The center shim packs and mounting bolts must not be used at this point. If they are installed, they should now be removed.

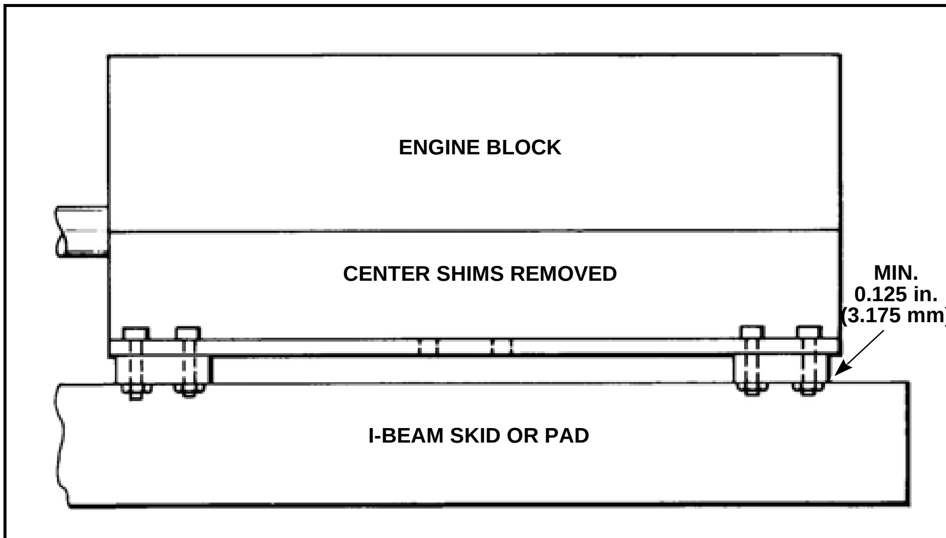


Figure 3-8 Corner Lift Method

2. Set up two dial indicators on the free end as shown below and zero the dials.
3. Using the jack screw, raise the left free corner of the engine until the indicator on the right free corner reads 0.001 in. (0.025 mm). Record the left free corner indicator reading (see Figure 3-9). Lower the left free corner of the engine back onto its shim pack.

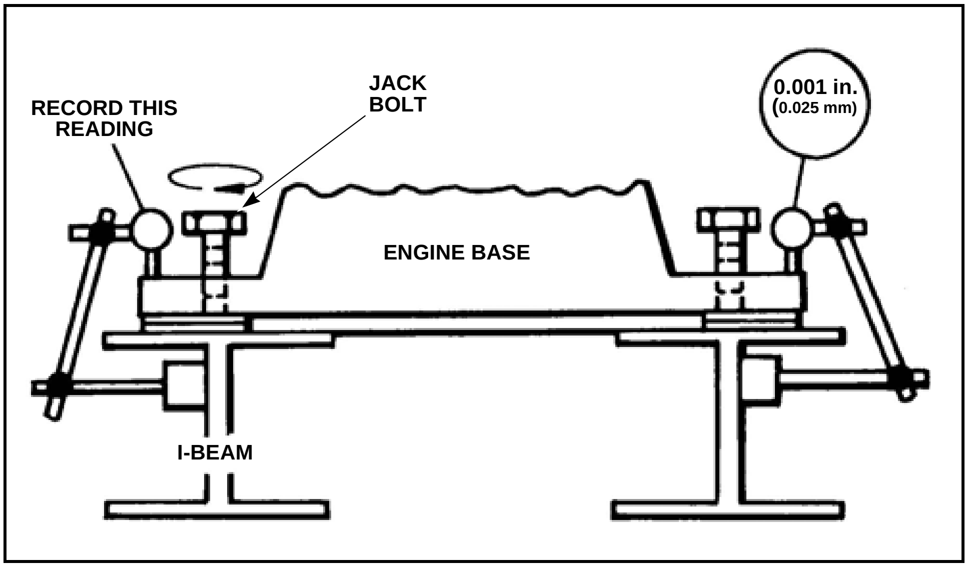


Figure 3-9 Record the Left Free Corner Indicator Reading

4. Raise the right free corner until the left indicator reads 0.001 in. (0.025 mm). Record the right free corner indicator reading (Figure 3-10).

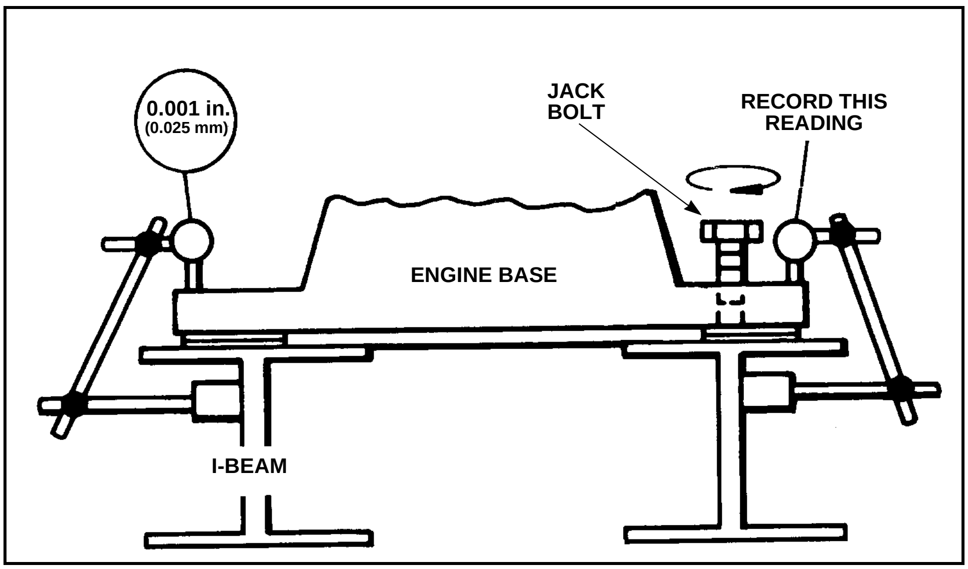


Figure 3-10 Record the Right Free Corner Indicator Reading

5. Calculate the difference between the two recorded corner readings. If the difference is less than 0.010 in. (0.254 mm), the base deflection is satisfactory and the free corners may be bolted down. If the difference is 0.010 in. (0.254 mm) or more, add shims equal to half of this difference under the corner that had the highest reading. Recheck per steps 2 and 3. Readings should now be within 0.010 in. (0.254 mm), and the corners can be bolted down. The four corners are now in the same plane. Checking the opposite end is not necessary.

6. The mounting points in the center of the engine now need to be shimmed. These are the final two points in the six point mounting. These center support points will have some amount of natural crankcase sag. While the engine is supported on the ends, the middle of the case is unsupported, and it may sag (see Figure 3-11). This sag has to be compensated for with the shimming procedure.

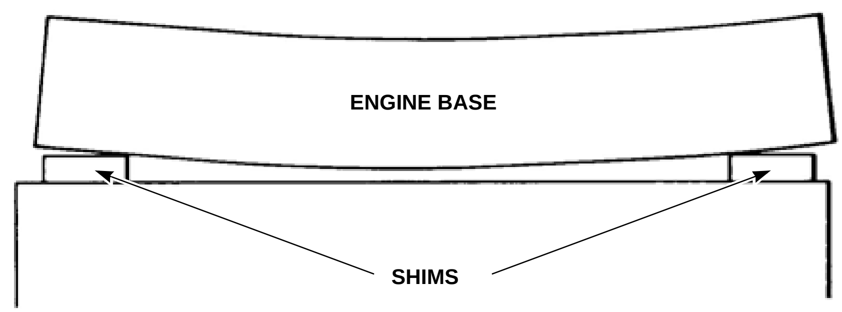


Figure 3-11 Natural Crankcase Sag

- Verify all corner mounts are properly torqued (center bolts removed).
- Set up a dial indicator at the center mount. Zero the dial.
- Add enough shims under the center mounts to fill the air gap. **Be careful not to bump the dial indicator during this procedure.**
- Replace the center bolts and torque the center mounts and then record the dial indicator reading.
- Loosen a front or rear mount and install shims under the center mount as required until the dial indicator reads:

  +0.002 in. (0.051 mm) for a VHP 6-cylinder*
  +0.000 in (0.000 mm) for a VHP 12-cylinder Extender Series*
  +0.004 in. (0.102 mm) for a VHP 12-cylinder Non-Extender Series*
  +0.008 in. (0.203 mm) for a VHP 16-cylinder*
  * **With the center mounts properly torqued.**

NOTE: *VGF* and APG 1000/16V150LTD center mounts should be shimmed to 0.000 in. This does not apply to the 275GL* as the crankcase deflection of these engines should be checked by the web deflection method.*

- If the dial indicator has not been moved or bumped, it should read positive by the amount indicated above, compared to when it was first zeroed. The engine base is now level with all the natural sag removed (see Figure 3-12).

* Trademark of General Electric Company

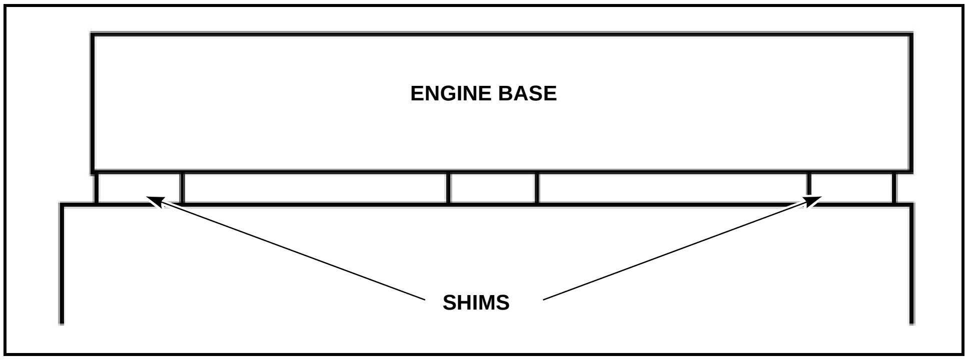


Figure 3-12 Level Engine Base with Natural Sag Removed

Release Method

The release method is used to verify that base deflection is correct by measuring spring up of each mounting point.

1. Starting at any engine mounting point, mount a dial indicator and zero the dial.
2. Loosen the mounting bolts at this point and record the dial reading.
3. Re-torque and verify that the dial indicator returns to zero.
4. Repeat for all mounting points.
5. Compare measurements from all 6 points. The 4 corners should have sprung equally within 0.005 in. (0.127 mm).

NOTE: *Spring-up at the center mounts should be zero because of the shims added to compensate for crankcase sag.*

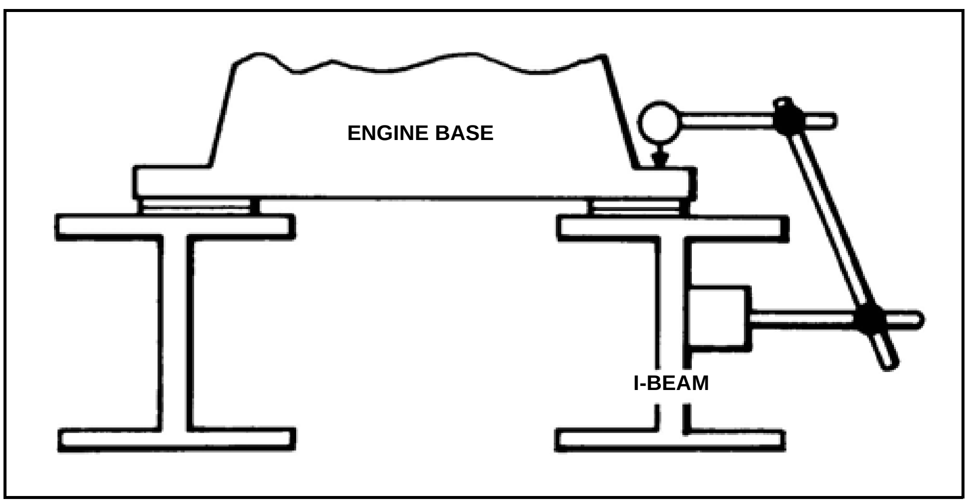


Figure 3-13 Release Method

Crankshaft Web Deflection

This check measures the deflection of a crankshaft during a revolution. It is the most direct method of determining if the shaft is being bent by a deflected crankcase or driven equipment misalignment. Web deflection measurements are required in marine engine applications. This procedure should also be used as a final check for base deflection and alignment especially on packages where the “Corner Lift Method” is too difficult to use.

All current production VHP* crankshafts, 6-, 12-, and 16-cylinder, have center punch marks to indicate the proper web deflection gauge mounting locations. These marks are 5 in. (127.0 mm) from the connecting rod journals and can be added to an unmarked crankshaft by using the counterweight parting line as a reference point. On all fully counterweighted VHP 6- and 12-cylinder crankshafts (12 counterweights), the marks are punched 0.185 in. (4.7 mm) inside the counterweight parting line.

* Trademark of General Electric Company

1. Mount a web deflection gauge tool (P/N 494424 digital or P/N 494292 analog) in the punch marks. Carefully twirl the gauge to make sure it is properly seated. All pistons and connecting rods should be in place during this procedure.

NOTE: *Interference with the connecting rods will not allow measurement during the full 360° shaft rotation.*

2. Position the crankshaft so the deflection gauge hangs freely next to the connecting rod, but as close to the rod as possible. Zero the gauge dial.
3. Slowly rotate the crankshaft until the gauge is in position 2, on the horizontal. Record any positive or negative reading attained.

NOTE: *Always check web deflection by rotating the crankshaft in the direction in which the engine is rotating.*

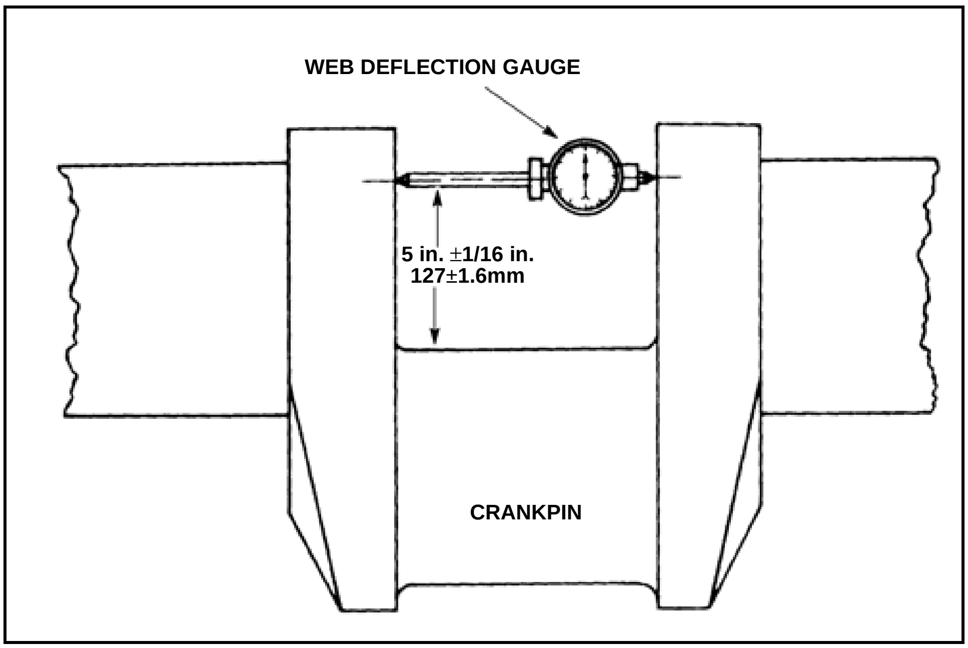


Figure 3-14 Crankshaft Web Deflection

4. Rotate the crankshaft to positions 3 and then 4, recording any readings. Now rotate the shaft further until the gauge is as high as possible, and yet still hangs free, without contacting the connecting rod. Record this reading.
5. Remove the deflection gauge, and repeat this procedure on the other crankshaft webs.
   - A total of 0.001 in. (0.025 mm) deflection, from positive to negative, is allowable on all but the rear crankshaft throw. The rear throw will typically

have 0.0015 in. (0.381 mm) deflection due to the effects of the flywheel weight.

- If deflection of the center throws exceeds 0.001 in. (0.025 mm), this can be corrected by shimming the center mounts. Adding shims will close the crankshaft web at the bottom while removing shims will open the crankshaft web at the bottom.
- High deflection on the rear throws could be caused by drive/driven-shaft misalignment or by an excessively heavy single bearing machine.
- High deflection on the front throws could be caused by overtightened accessory belts.

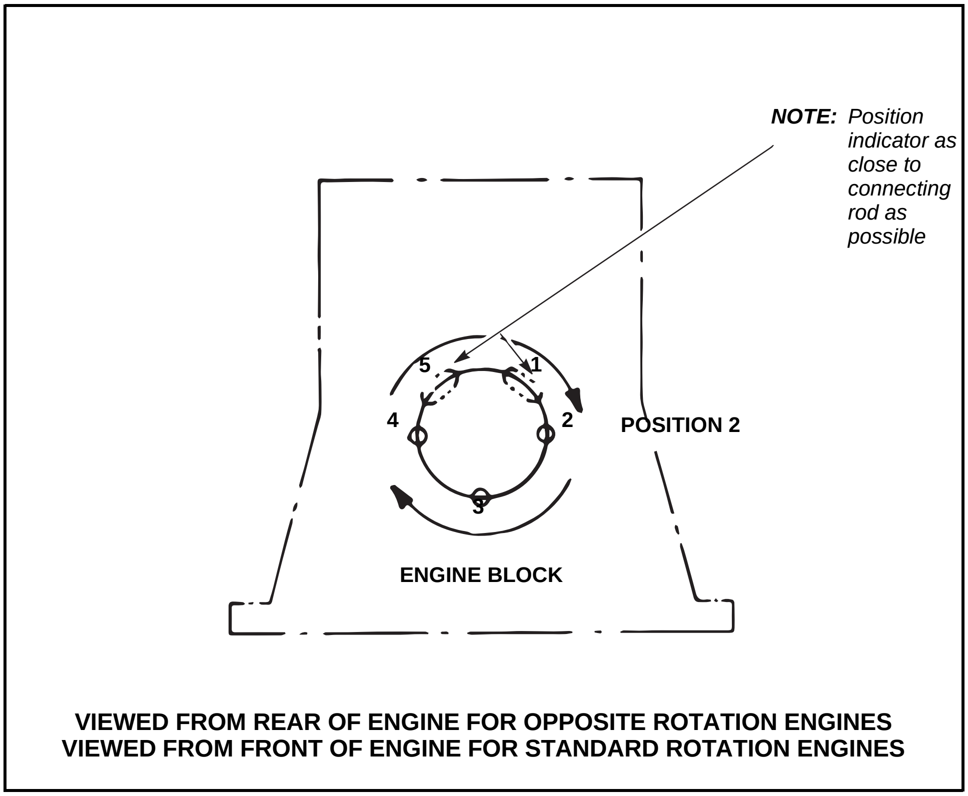


Figure 3-15 Locations for Checking Crankshaft Deflection

Spring Isolated Packages

On spring isolated packages the engine and driven equipment are solidly mounted to a common skid which rests on spring isolators. Beneath the spring isolators is a concrete mounting pad, inertia block, or steel support structure.

Spring isolation is used to isolate the surrounding environment from engine and driven equipment vibration. To do this effectively, the mounting points must be correctly spaced around the center of gravity and the isolators adjusted properly.

Generator sets from Waukesha Power Systems have the isolator mounting holes correctly spaced for uniform support of the package when filled with coolant and lube oil. When supported uniformly, the spring lengths on all the isolators will be equal. The following is a general procedure for adjusting spring type vibration isolators. For more specific instructions, see the spring isolator manufacturer's instructions.

Spring Isolator Installation

1. Check that all points where spring isolators will be fitted are even and level. Build up any low spots using steel chocks until all isolator base plates are within 0.125 in. (3.175 mm) elevation of each other.
2. Install spring isolators and bolt down, if required.
3. Loosen horizontal chocks (snubbers), if used.
4. Place engine/driven equipment package on the isolators. All isolators should have the isolator top plate contacting the isolator base.
5. Turn the adjustment on each isolator down 2 full turns at a time until all isolators have at least 0.125 in. (3.175 mm) between the top plate and the base.

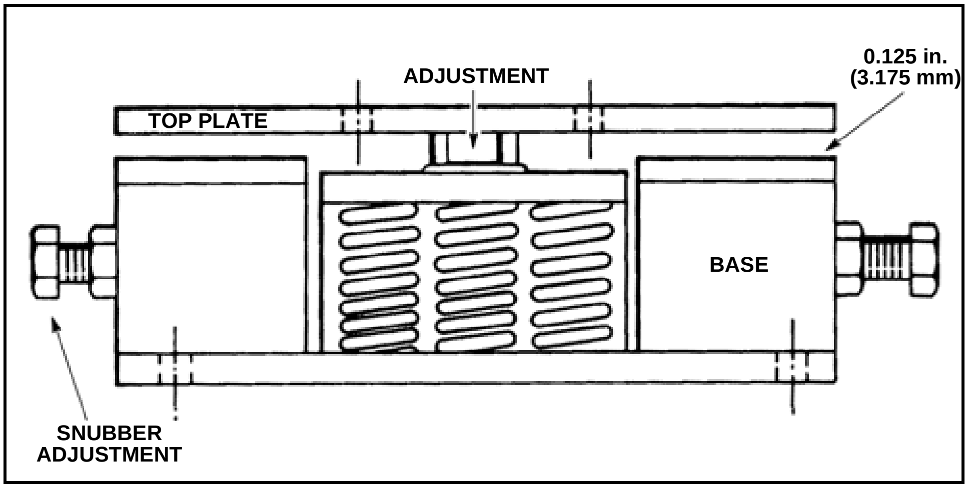


Figure 3-16 Spring Isolator Mount

6. If the package is not level after adjusting the isolators, this will be corrected with further adjustments. To level a unit side-to-side, make equal adjustments to all the isolators on one side. Leveling a unit front to rear, where the isolators are spaced evenly, can be accomplished as follows:
   - Turn the adjustment screw one turn on the pair of isolators next to the high end isolators.
   - Turn the adjustment screw 2 turns on the third pair, 3 turns on the fourth pair, etc. Repeat this as many times as necessary to level the skid.

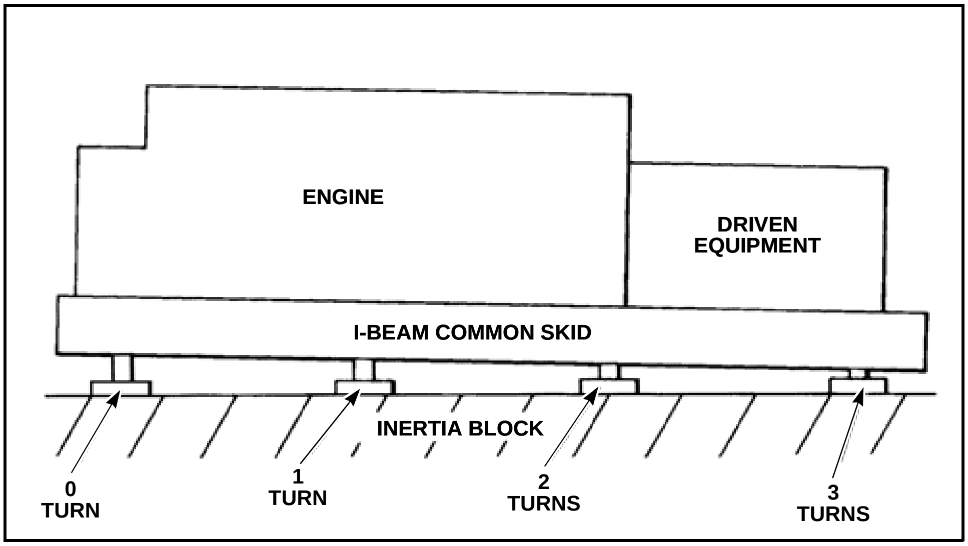


Figure 3-17 Leveling Spring Isolators

7. With the engine running, adjust the horizontal chocks (snubbers), if equipped, for a minimum of horizontal movement (minimal or no gap). Lock the adjustment bolt in place with the locknut.

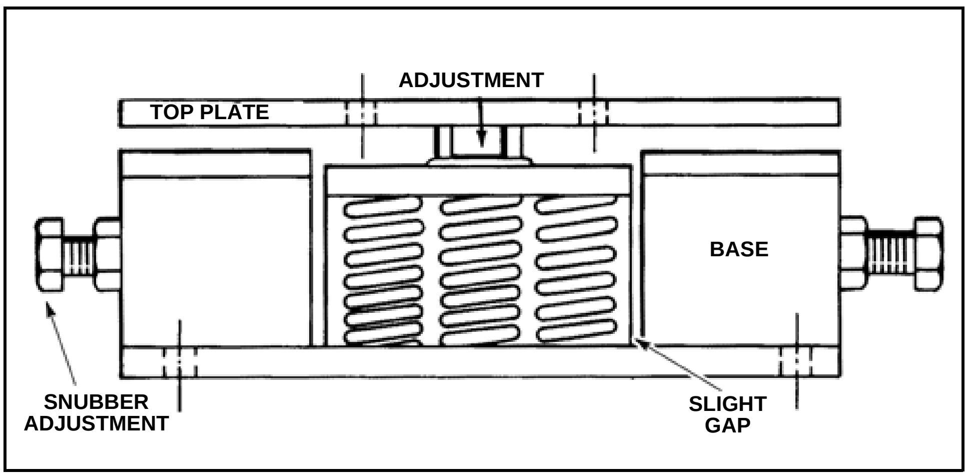


Figure 3-18 Spring Isolator Mount

As stated earlier, when spring isolators are adjusted correctly, the spring lengths on all the isolators will be equal. The formula below calculates what this spring length should be:

$$L_L = F_L - \frac{W}{K \times n}$$

Where:

L_L = Length of springs when engine package is resting on them (inches) - loaded length

F_L = Length of springs while unloaded (inches) - free length

W = Weight of engine package wet (lb)

K = Spring constant of isolators (lb / in.)

n = Number of isolators under package

When one isolator is compressed too far, it can be relieved by adjusting the surrounding isolators down or by adjusting up on the subject isolator. Always maintain a minimum 0.125 in. (3.175 mm) gap between the isolator base and top plate on all isolators.

Engine Base Deflection

Checking engine base deflection is important to assure that the main bearing bores are in perfect alignment. Misaligned main bearing bores can cause premature failure of bearings and/or bending breakage of the crankshaft.

Release Method

This method is used to determine base deflection by loosening each mounting point and measuring spring-up. This procedure may be used when the skid is positioned on the adjusted spring isolators.

1. Remove center shim packs.
2. Starting at any corner, mount a dial indicator and zero the dial.
3. Loosen the mounting bolts at this point and record the dial reading.
4. Re-torque the bolts and verify that the indicator dial returns to zero.
5. Repeat this step at the remaining 3 corners.
6. Compare the measurements from each of the 4 corners and then shim until the corners spring equally within 0.010 in. (0.254 mm).
7. The mounting points in the center of the engine now need to be shimmed. These are the final two points in the six point mounting. These center support points have some amount of natural crankcase sag (see Figure 3-19). While the engine is supported on the ends, the middle of the case is unsupported, and it will sag. This sag will be compensated for in the shimming procedure.

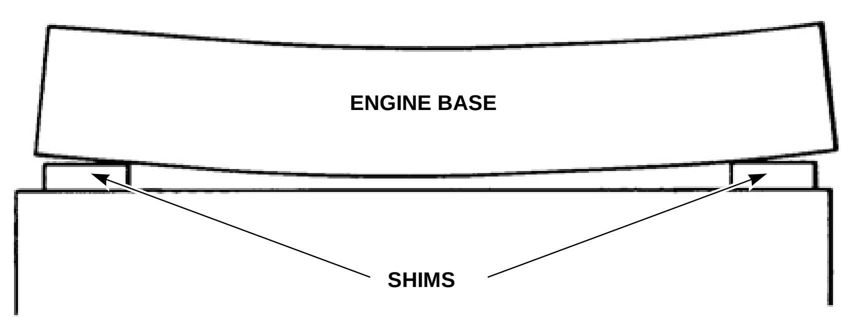


Figure 3-19 Natural Crankcase Sag

- Verify all corner mounts are properly torqued.
- Set up a dial indicator at the center mount. Zero the dial.
- Add enough shims under the center mounts to fill the air gap. **Be careful not to bump the dial indicator during this procedure.**
- Re-torque the center mounts and then read the dial indicator.

- Loosen a front or rear mount and install shims under the center mount as required until the dial indicator reads:

  +0.002 in. (0.051 mm) for a VHP 6-cylinder
  +0.000 in (0.000 mm) for a VHP 12-cylinder Extender Series
  +0.004 in. (0.102 mm) for a VHP 12-cylinder Non-Extender Series
  +0.008 in. (0.203 mm) for a VHP 16-cylinder
  * **With the center mounts properly torqued.**

NOTE: *VGF and APG1000/16V150LTD center mounts should be shimmed to 0.000 in. This does not apply to the 275GL as the crankcase deflection of these engines should be checked by the web deflection method.*

- If the dial indicator has not been moved or bumped, it should read positive by the correct amount from when it was first zeroed. The engine base is now level with all natural sag removed (see Figure 3-20).

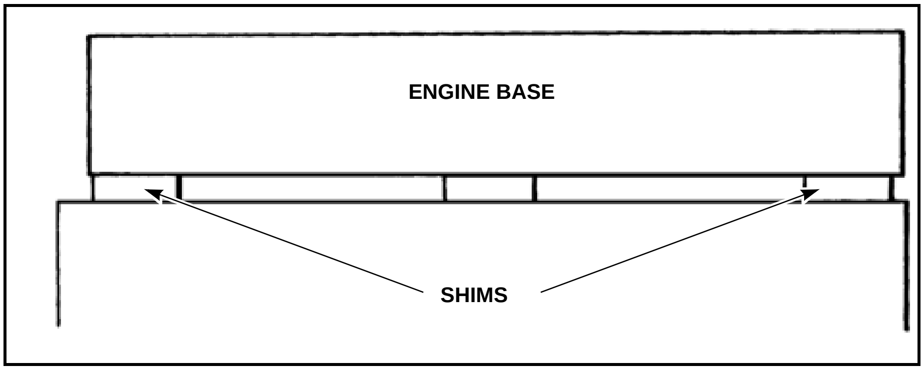


Figure 3-20 Level Engine Base With All Natural Sag Removed

Crankshaft Web Deflection

This check measures the deflection of a crankshaft during one revolution. It is the most direct method of determining if the shaft is being bent by a deflected crankcase or misalignment. Web deflection measurements are required in marine applications. This procedure should be used as a final check for base deflection and alignment on packages where the “Release Method” is too difficult to use.

All current production VHP crankshafts, 6-, 12-, and 16-cylinder have punch marks to indicate proper web deflection gauge mounting locations. These marks are at 5 in. (127.0 mm) from the connecting rod journals and can be added to an unmarked crankshaft by using the counterweight parting lines as a reference point. On all fully counterweighted VHP 6- and 12-cylinder crankshafts (12 counterweights), the marks are punched 0.185 in. (4.69 mm) inside the counterweight parting line.

1. Mount a web deflection gauge tool (P/N 494424 digital or P/N 494292 analog) in the punch marks. Carefully twirl the gauge to make sure it is properly seated. All pistons and connecting rods should be in place during this procedure.

2. Position the crankshaft so the deflection gauge hangs freely next to the connecting rod, but as close to the rod as possible. Zero the gauge dial.

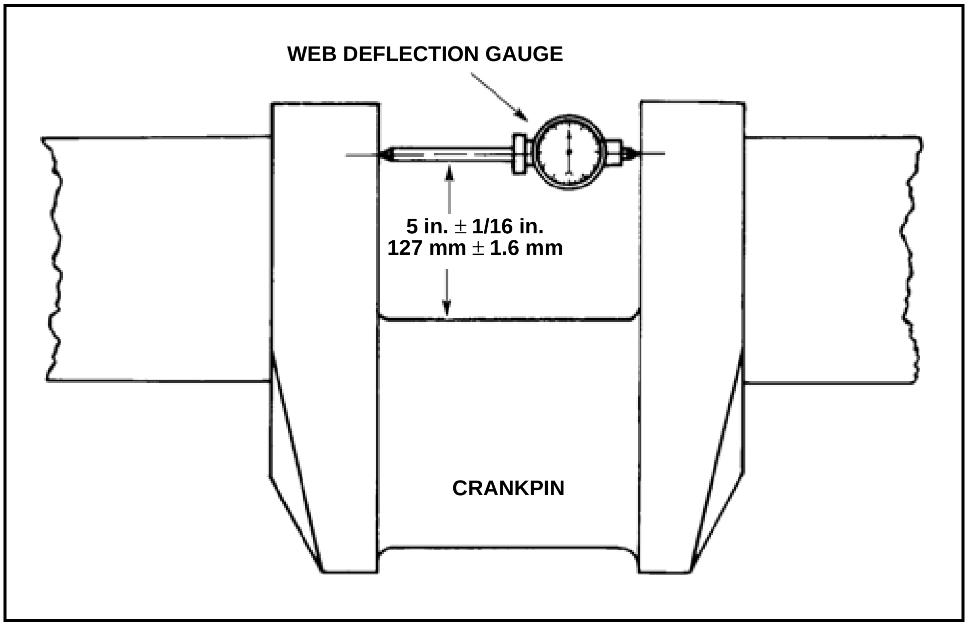


Figure 3-21 Crankshaft Web Deflection

3. Slowly rotate the crankshaft until the gauge is in position 2, on the horizontal. Record any positive or negative reading attained.
4. Rotate the crankshaft to positions 3 and then 4, recording any readings. Now rotate the shaft further until the gauge is as high as possible, and yet still hangs free, without contacting the connecting rod. Record this reading.
5. Remove the deflection gauge, and repeat this procedure on the other crankshaft webs.
   - A total of 0.001 in. (0.025 mm) deflection from positive to negative is allowable on all but the rear crankshaft throw. The rear throw will typically have 0.0015 in. (0.381 mm) due to the affects of flywheel weight.
   - If deflection of the center throws exceeds 0.001 in. (0.025 mm), this can be corrected by shimming the center mounts. Adding shims will close the crankshaft web at the bottom. Removing shims will open the crankshaft web at the bottom.
   - High deflection on the rear throws could be caused by drive/driven shaft misalignment or an excessively heavy single bearing machine.
   - High deflection on the front throws could be caused by overtightened accessory belts.

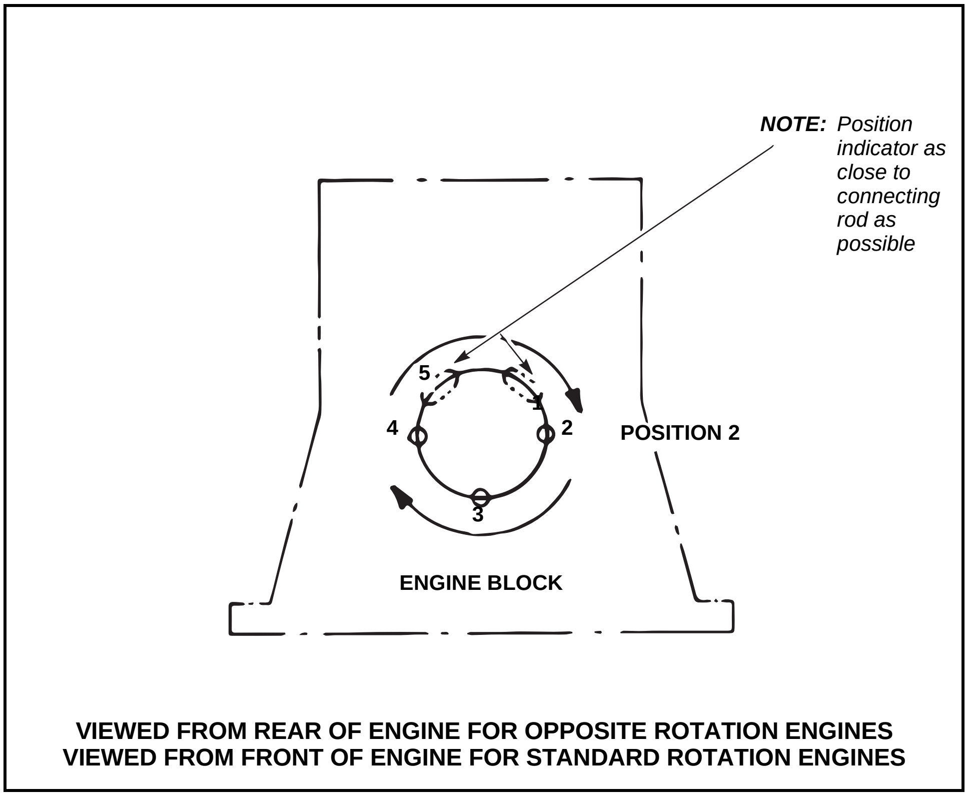


Figure 3-22 Location for Checking Crankshaft Deflection

Driven Equipment Base Deflection

Use the driven equipment manufacturer's procedures and limits if available. Base deflection can also be measured and adjusted using a "Release Method" similar to that described for the engine.

1. Starting at any corner, mount a dial indicator and zero the dial.
2. Loosen the mounting bolts at this point and record the dial reading.
3. Re-torque and verify that the dial indicator returns to zero.
4. Repeat this procedure at the remaining 3 corners.
5. Compare measurements from the 4 corners and shim as required. When all corners spring to within 0.005 in. (0.127 mm) of each other, the procedure is completed.

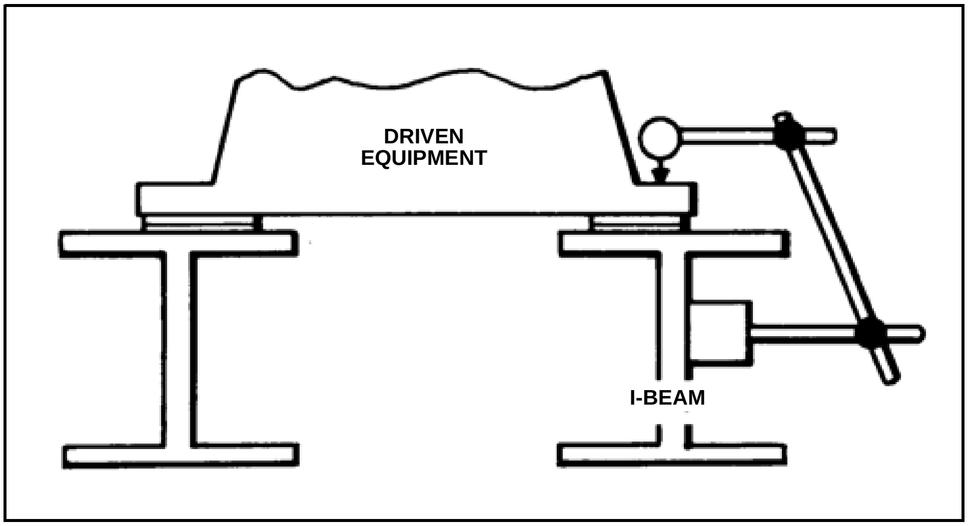


Figure 3-23 Driven Equipment Base Deflection

SECTION 4

ALIGNMENT

Single Bearing Generator and Similar Single Bearing Equipment Alignment

Aligning single bearing equipment involves two steps: first, the driven shaft must be centered in the flywheel pilot and second, the engine crankshaft and driven shaft must form a straight line when viewed both horizontally and vertically.

Centering Pilot (Parallel Alignment)

To measure how well a shaft is centered in the flywheel pilot, a dial indicator must be clamped to the flywheel housing or driven machine body. The dial indicator will then read the total runout of the driven equipment input shaft.

1. Clean the shaft of any dirt, grease, rust or paint. Use emery cloth if necessary to ensure a smooth surface to measure from.
2. Mount a dial indicator to the flywheel housing or generator barrel and take the reading from the shaft. Check for clearance before rotating the shaft.
3. Bar the engine over counterclockwise (facing the flywheel) and take your readings every 90°. A maximum of 0.005 in. (0.127 mm) Total Indicator Runout (TIR) is acceptable.
4. If runout exceeds 0.005 in. (0.127 mm) TIR.
   - Roll the highest point to the top.
   - Loosen the coupling bolts at this point to allow the shaft and coupling to drop in the flywheel counterbore. Once all the bolts are loose, re-torque the bolts.

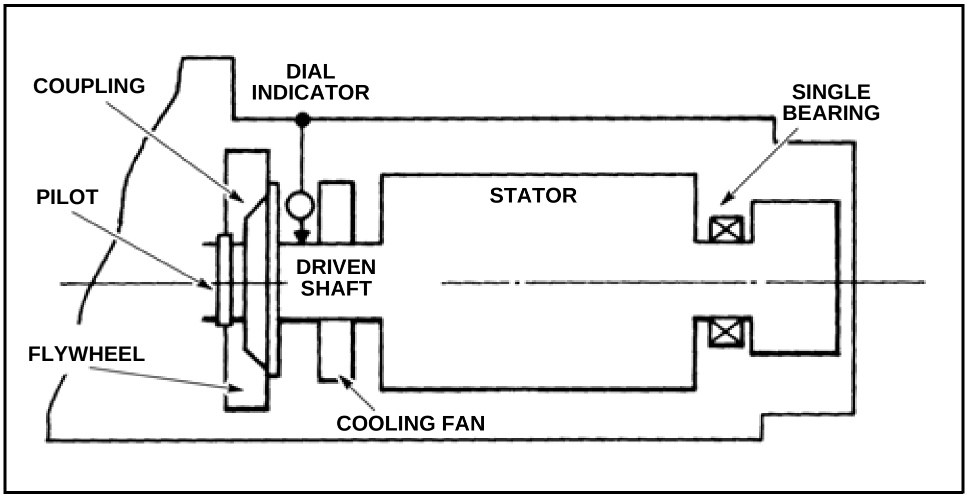


Figure 3-24 Single Bearing Generator

5. Repeat steps 2 and 3, and if TIR is still unacceptable the coupling bolts must be removed and the driven equipment shaft rotated 90° with respect to the engine flywheel. Further adjustments can be made by rotating in additional 90° increments, until the specifications are achieved.

Angular Alignment

To measure angular alignment, a dial indicator is mounted on the shaft of one machine and reads against the shaft face on the other machine. In the case of a single bearing generator, the dial indicator can be clamped to the fan and measures from the flexplate-to-flywheel mounting bolt.

Before taking readings, roll the shaft in reverse rotation 45°, then back 45°, and zero the dial indicator. This sets the axial position of the crankshaft and the driven machine shaft.

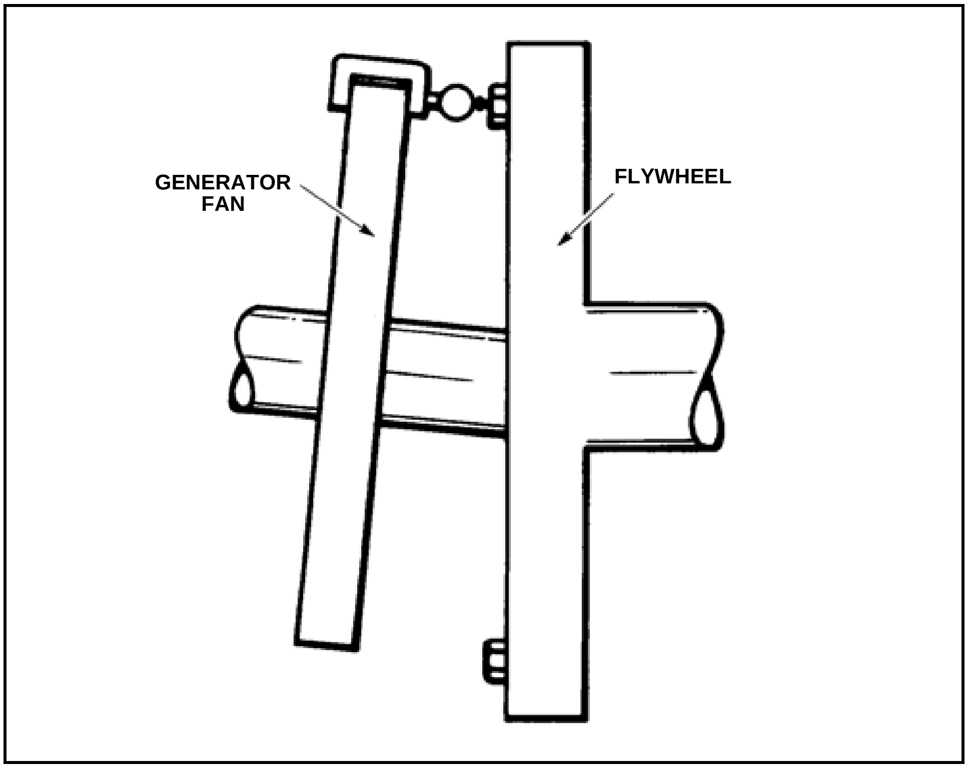


Figure 3-25 Angular Alignment-Single Bearing Generator

To measure the angular alignment, four dial indicator readings are required; one each at the 12, 9, 6, and 3 o'clock positions. Readings at the 12 and 6 o'clock positions determine the vertical alignment and readings in the 3 and 9 o'clock positions determine the horizontal alignment (see Figure 3-26).

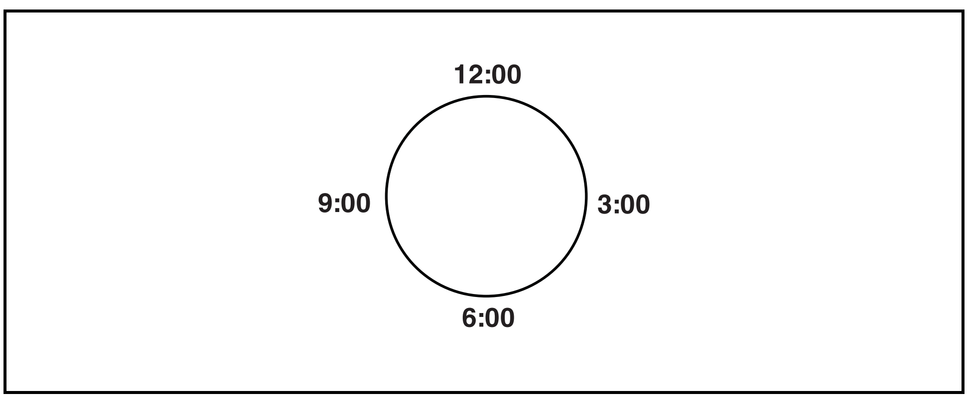


Figure 3-26 Dial Indicator Reading Positions When Measuring Angular Alignment

A total indicator reading (TIR) is the difference between two readings on opposite sides of the shaft. In the example illustrated (see Figure 3-27), the horizontal TIR is (-0.009) and (+0.004) which is a difference of 0.013 in. (0.330 mm) or 13 thousandths of an inch TIR. Vertical TIR is (0) and (+0.005) which is a difference of 0.005 in. (0.127 mm) or 5 thousandths of an inch TIR.

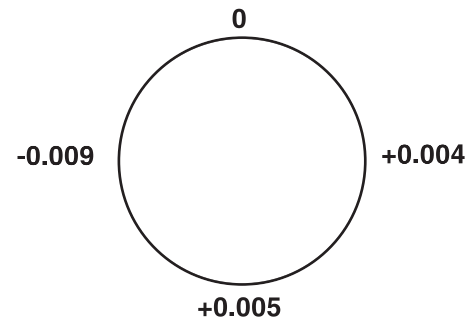


Figure 3-27 Total Indicator Reading (TIR)

The shaft shown (single bearing machine) is angularly misaligned from that of the engine. This could be either vertical or horizontal misalignment. In the case pictured, the distance “S” divided by the distance to the bearing (or rear mount) “L” is equal to 1/2 TIR divided by the radius from the dial indicator to the center of the shaft “R”.

More simply:

$$\frac{S}{L} = \frac{1/2\ (TIR)}{R}$$

Thus, we find that the amount of shimming or horizontal sliding required is:

$$S = L \times (1/2\ TIR)/R$$

This relationship is used with the outboard mount or any inboard mount (closer to the flywheel) as long as the distance to the required mount is used for “L”.

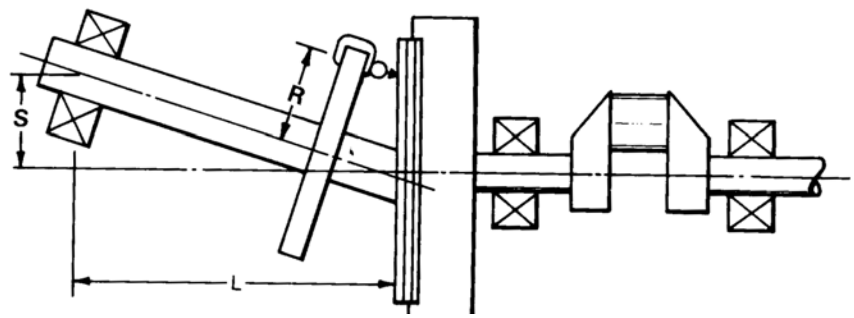


Figure 3-28 Exaggerated Example

Vertical adjustments are made by adding or removing shims from the mounts on each end of the machine. The L.H. and R.H. inboard mounts are adjusted the same, and the L.H. and R.H. outboard mounts are adjusted the same.

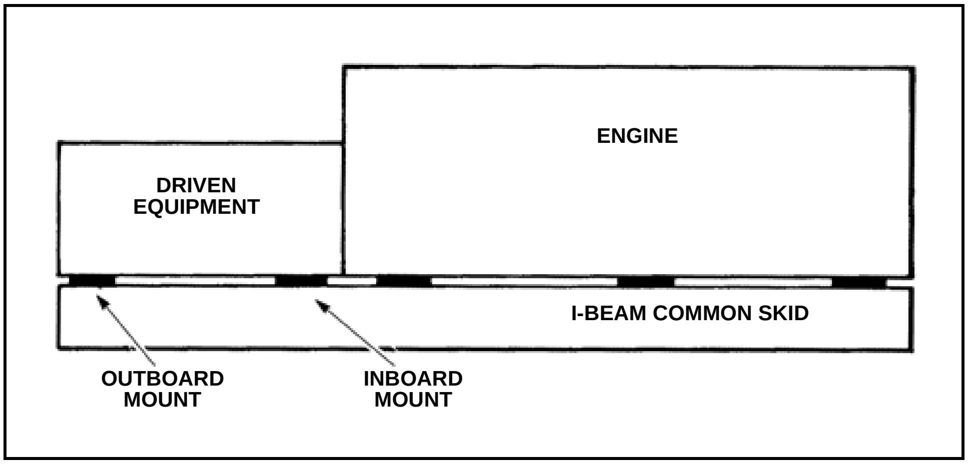


Figure 3-29 Add or Remove Shims from the Mounts on Each End of the Machine to Make Vertical Adjustments

Horizontal adjustment is made by loosening all the mounting bolts and physically forcing the driven equipment to the desired side. This can be done with a jacking screw or a pry bar in the bolt hole. Dial indicators should be set up to monitor how far the machine is moved, or as an alternate method, the shaft can be rotated to the 3 or 9 o'clock position and adjustments made until 1/2 TIR is indicated by the angular dial indicator.

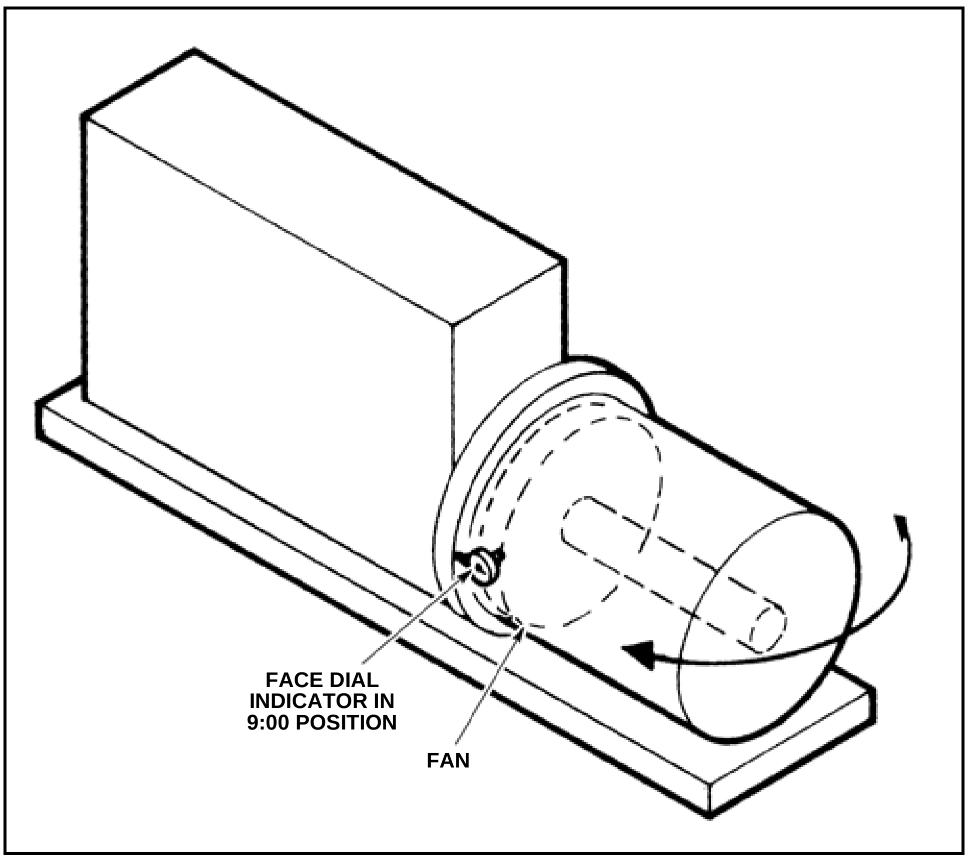


Figure 3-30 Dial Indicator Positioning

Angular alignment is acceptable when the TIR in all directions is less than 0.005 in. (0.127 mm) measured at the flexplate-to-flywheel bolt which is 14 in. (355.6 mm) from the shaft center.

Thermal Growth

Once the drive/driven shaft alignment is acceptable, the vertical thermal growth of the engine and driven machine must be compensated.

Table 3-7 lists the changes in crankshaft height that will occur due to the temperature change from 70° F (21° C) to normal operating temperatures. This is measured from the mounting surface of the base type oil pan on VHP and VGF engines.

Table 3-7 Thermal Growth

ENGINE MODEL	INCREASE IN CRANKSHAFT HEIGHT	
	INCHES	mm
F18/H24	0.012	0.31
L36/P48	0.020	0.51
16V150LTD	0.020	0.51
F2895, F3521, F3524, L5790, L5794, L7042, L7044	0.014	0.36
P9390	0.017	0.43

Thermal growth information for the driven machine should be available from the manufacturer. If not, it can be calculated with the following formula:

$G_m = (T_m - 70) \times h \times E$ for °F or $(T_m - 20) \times h \times E$ for °C

Where:

G_m = amount of growth expected (inches or mm)

T_m = operating temperature of driven machines (°F or °C)

h = height from machine mounting surface to center of shaft (inches or mm)

E = thermal expansion coefficient for material machine is made from:

6.5×10^{-6} (0.0000065) in/in °F or 1.2×10^{-6} mm/mm °C for steel

5.8×10^{-6} (0.0000058) in/in °F or 1.1×10^{-6} mm/mm °C for cast iron

To compensate when there is a growth difference, align the machine with less growth higher than the machine with more growth.

For example, if a generator grows 0.005 in. (0.127 mm) and an engine grows 0.014 in. (0.356 mm), the generator should be shimmed 0.014 in. (0.356 mm) – 0.005 in. (0.127 mm) = 0.009 in. (0.229 mm) higher than the engine. This is done after the machines are initially aligned. The shims go under all mounts of the generator. When checking angular alignment, the vertical TIR will now be off but will fall within the limits once the engine and generator reach operating temperature.

Crankshaft Endplay

After completing the cold alignment, the crankshaft endplay should be checked.

1. Clamp a dial indicator to the flywheel housing and read against the crankshaft or flywheel face.
2. Pry the shaft forward and zero the dial indicator. (It may be necessary to remove an oil pan door and wedge a pry bar between a crankshaft web and main bearing cap to move the shaft forward).
3. Pry the shaft rearward. The shaft should not “bounce” forward and the dial indicator should read within the service manual specifications.

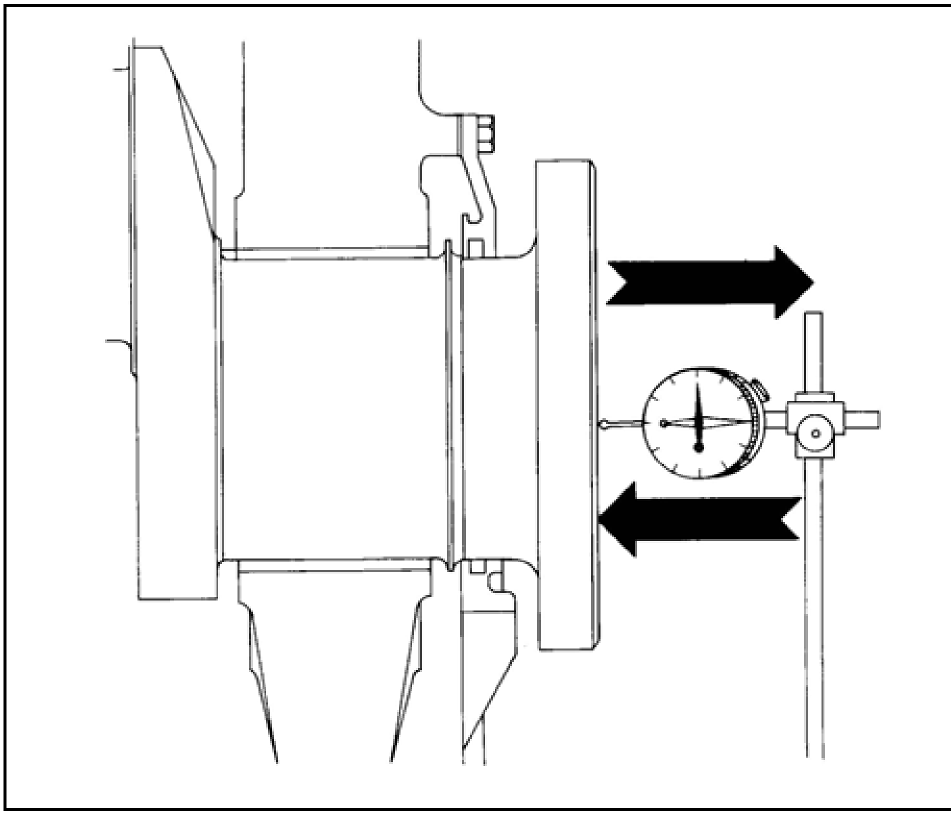


Figure 3-31 Checking Crankshaft Endplay

Air Gap

On single bearing generators, the air gap between the stator and armature and at the exciter should be checked to verify that adequate clearance exists. Correcting the air gap is accomplished by adjusting the position of the inboard feet of the generator. Single bearing induction generators have a very small clearance so it is important that these be checked very carefully.

Some generator fans use setscrews to hold the axial position of the fan. Verify that these setscrews are tight and that the fan hub bolts are properly torqued.

Hot Check

When the alignment, endplay, and air gap are adjusted, the engine and generator set should be run up to operating temperature under load for at least one hour. Then shut down the unit and check alignment, endplay, and air gap. If it is within specifications, then the alignment is complete.

Periodic Inspection

Engine base deflection and alignment must be checked periodically, at least once a year. Installations which are subject to settling of the concrete must be checked monthly initially, to determine if settling is causing any misalignment.

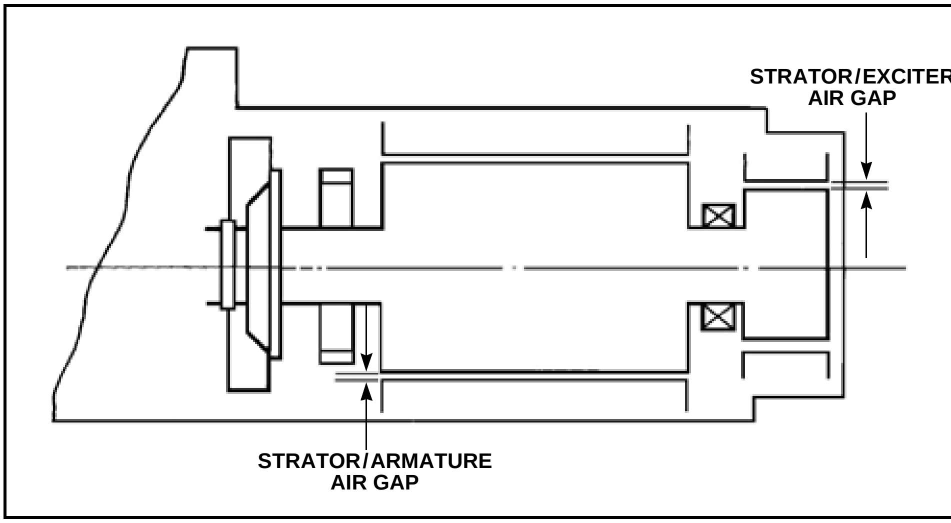


Figure 3-32 Single Bearing Generator

Multi-Bearing Machines

A multi-bearing machine is one which fully supports its own shaft, and does not rely on the engine shaft to support the driven end.

Three areas must be adjusted to accurately align a multi-bearing machine to an engine, which is also a multi-bearing machine. These are: Endplay, Angular Alignment and Parallel Alignment.

When aligning two multi-bearing machines, one machine must be designated as the stationary machine, and one as the movable machine. Deciding which machine will be stationary will depend on size, weight, and connections. All adjustments will be made on the movable machine.

Adjusting angular and parallel alignment on multi-bearing machines requires correcting the angular alignment first and then the parallel. Once alignment is acceptable, the machines must be shimmed to compensate for thermal growth.

The Waukesha alignment computer (P/N 475063 or most current) finds adjustments for angular and parallel alignment as well as thermal growth, after the user inputs the dimension, growth and measuring information. Only one or two adjustments are normally required to place the units within the alignment specifications, when this tool is used.

If the alignment computer is not available, the following procedures will provide an accurate alignment.

Endplay

To adjust endplay:

1. Roughly position the two machines and install the shaft coupling. Adjust the distance between the two machines so that there is no apparent tension or compression on the coupling. Properly space gear type couplings per the coupling manufacturer's specifications.
2. Set up a dial indicator on the machine with the least endplay (normally the engine). Clamp the dial indicator to the engine flywheel housing and read against the flywheel face.
3. Pry the crankshaft fully forward, and zero the dial indicator. (Moving the crankshaft on a VHP engine may require removing an oil pan door and prying between a main bearing cap and crankshaft cheek or web.)
4. Pry the shaft rearward and read the dial indicator. Crankshaft endplay should be within service manual specifications and the shaft should not spring-back when the bar is removed.
5. If there is insufficient endplay or if spring-back occurs, adjust the distance between the machines until it is resolved.

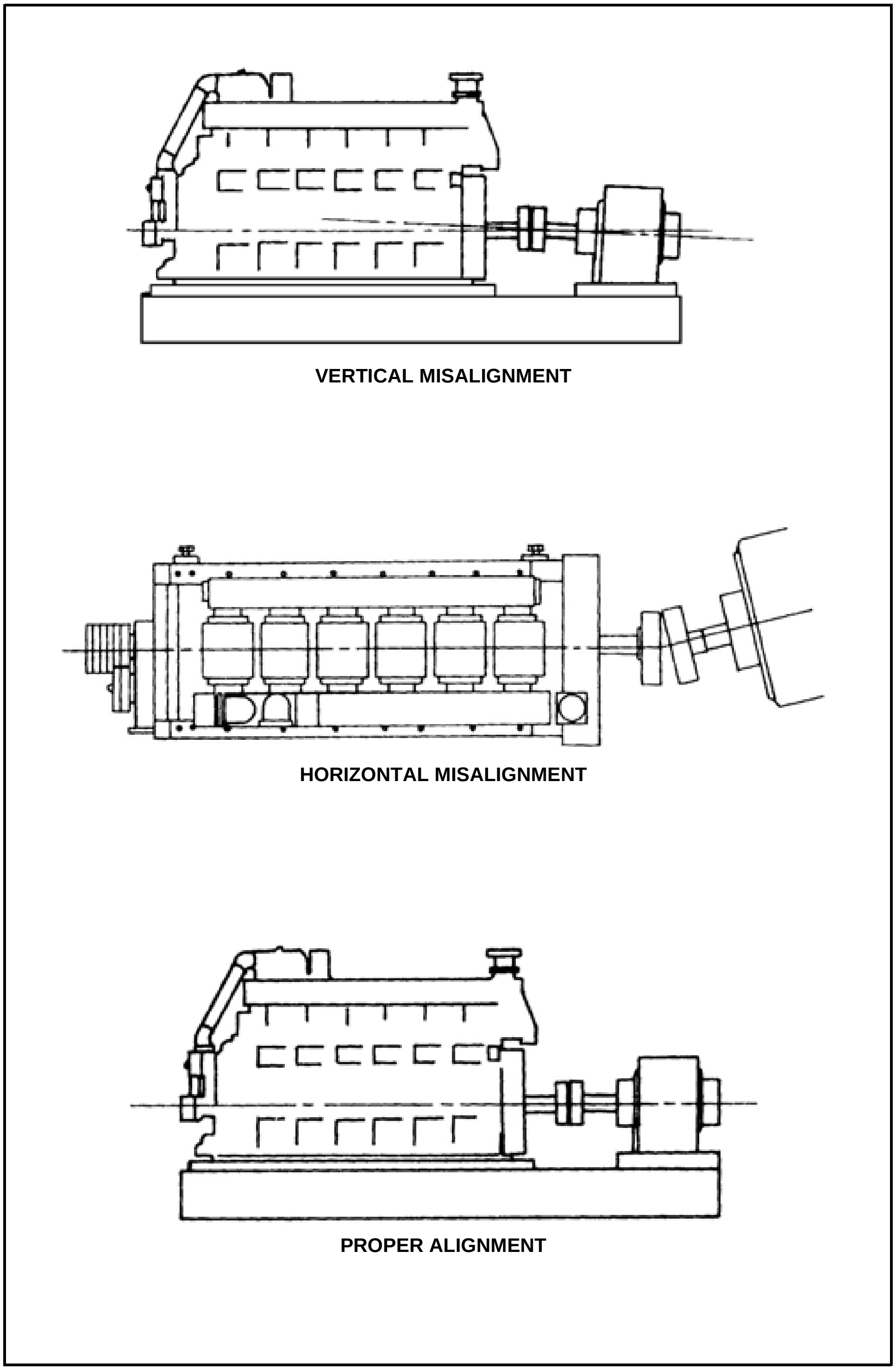


Figure 3-33 Angular Alignment

Angular Alignment

To measure the angular alignment, a dial indicator is mounted to the coupling half of one machine to read against the coupling half face of the other. The coupling should be installed or the shafts bound together so they both turn together while taking the alignment measurements.

The radius “R” from the center of the shaft to the dial indicator should be at least 7 in. (177.8 mm).

Before taking readings, roll the shaft 45° in reverse rotation and then back 45° in standard rotation and zero the dial indicator. This sets the axial position for both the engine and driven machine shafts.

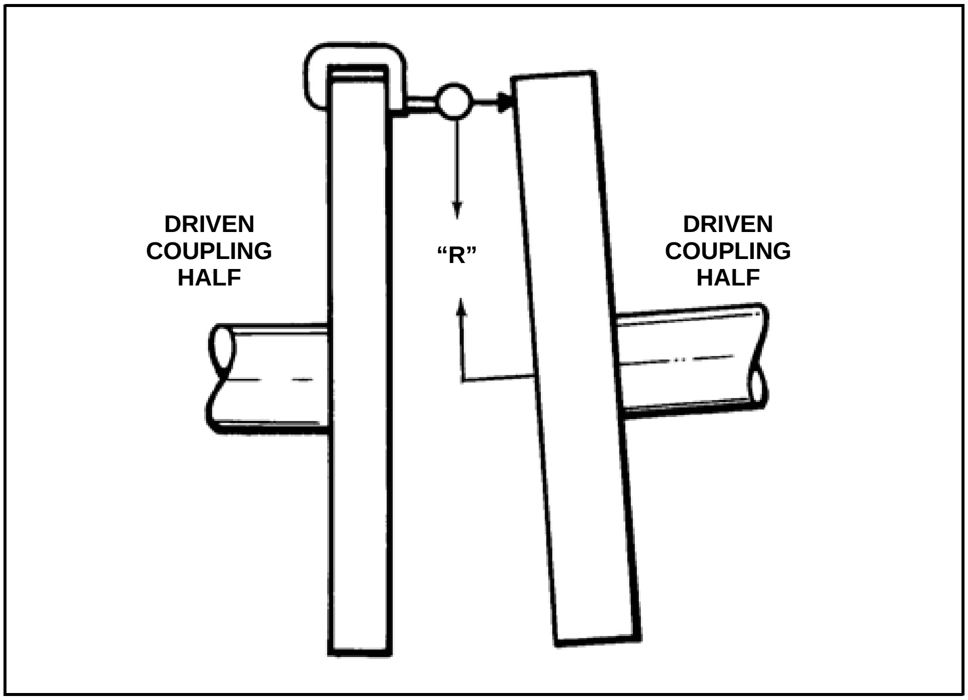


Figure 3-34 Measuring Angular Alignment

To measure angular alignment, four dial indicator readings are required: one each at the 12, 9, 6 and 3 o’clock positions which are taken while turning the engine in the standard direction of rotation.

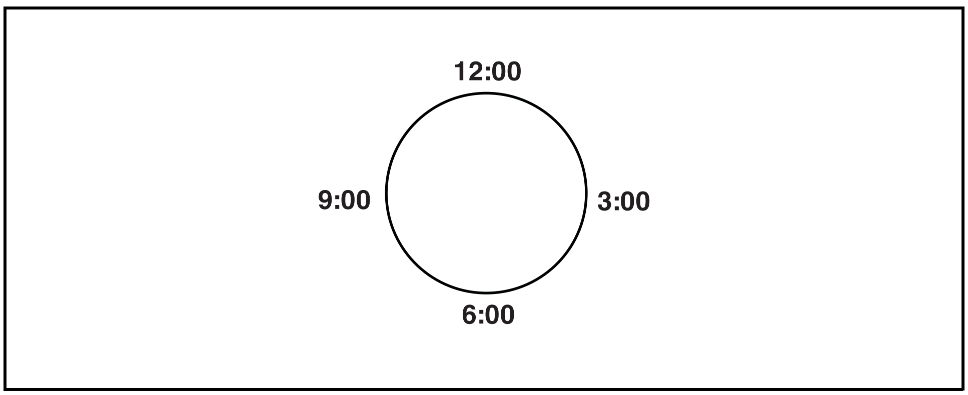


Figure 3-35 Dial Indicator Reading Positions When Measuring Angular Alignment

Readings taken at the 12 and 6 o'clock positions determine vertical angular alignment and readings in the 3 and 9 o'clock positions determine horizontal angular alignment. A total indicator reading (TIR) is the absolute difference between two readings on opposite sides of the shaft. In the illustration, the horizontal TIR is (-0.009) and (+0.004) which is a difference of 0.013. Vertical TIR is (0) and (+0.005) which is a difference of 0.005 in. (0.127 mm).

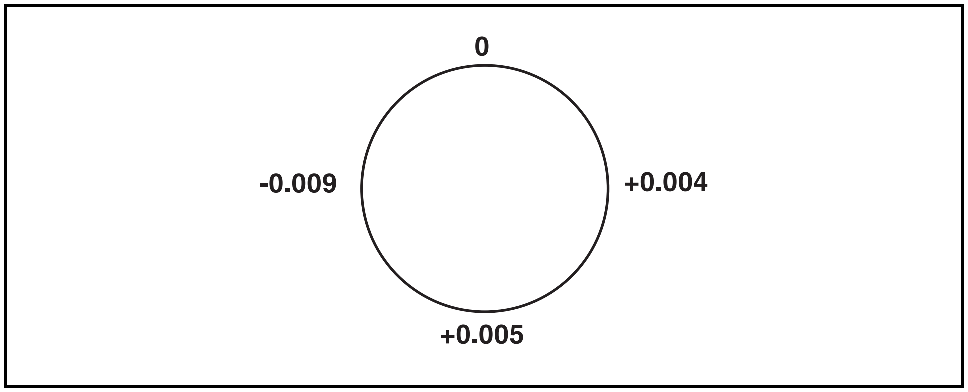


Figure 3-36 Total Indicator Reading (TIR)

The illustration shows the shaft of a multi-bearing machine with both angular and parallel misalignment.

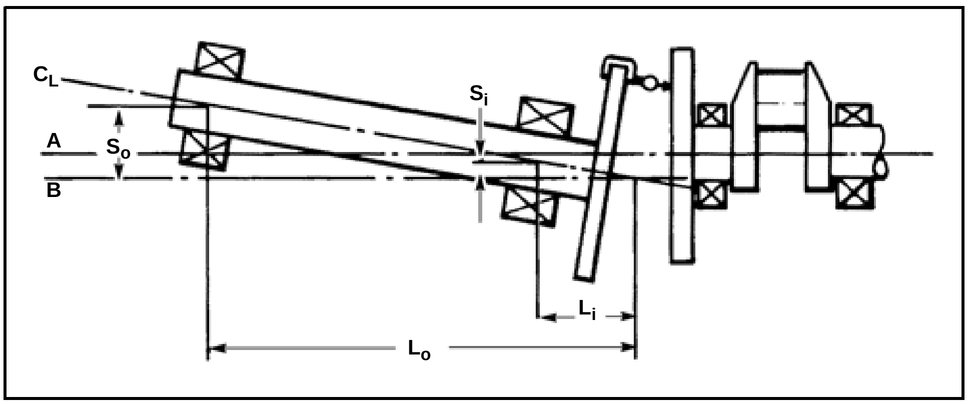


Figure 3-37 Multi-Bearing Driven Equipment

This could represent either vertical or horizontal misalignment since the principles are the same for both.

Correcting this misalignment first involves correcting angular alignment, thus getting the shaft centerline to line up on line B.

The amount of correction required to bring the centerline into alignment with line B, can be determined from the dial indicator TIR, radius to the indicator "R", and distance "L" from the coupling to the mounts.

$$\frac{1/2\,(TIR)}{R} = \underset{\text{OUTBOARD MOUNT}}{\frac{S_o}{L_o}} = \underset{\text{INBOARD MOUNT}}{\frac{S_i}{L_i}}$$

Therefore:

$$S_o = \frac{L_o \times 1/2(TIR)}{R}$$

and

$$S_i = \frac{L_i \times 1/2(TIR)}{R}$$

"S_o" is the amount of adjustment at distance "L_o" which is the distance from the center of the coupling to the center of the outboard mount.

"S_i" is then the adjustment at a mount distance of "L_i" from the coupling.

The adjustment should be made to close the open side of the coupling as pictured.

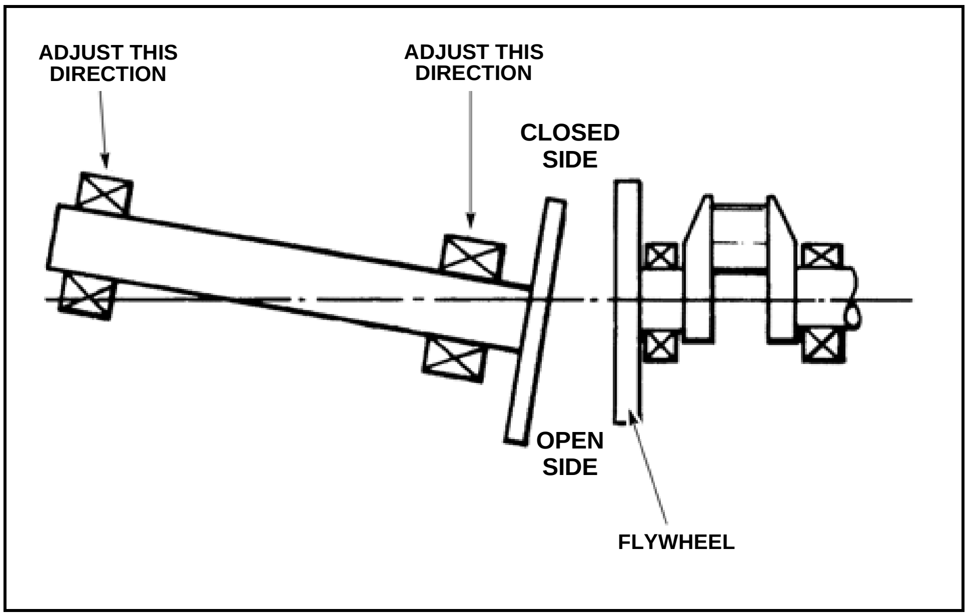


Figure 3-38 Make Adjustment to Close the Open Side of the Coupling

Adjustment for angular alignment should then take place as follows:

1. Set up two dial indicators, one to monitor horizontal movement of the inboard mounts, one to monitor horizontal movement of the outboard mounts. Zero the indicators.

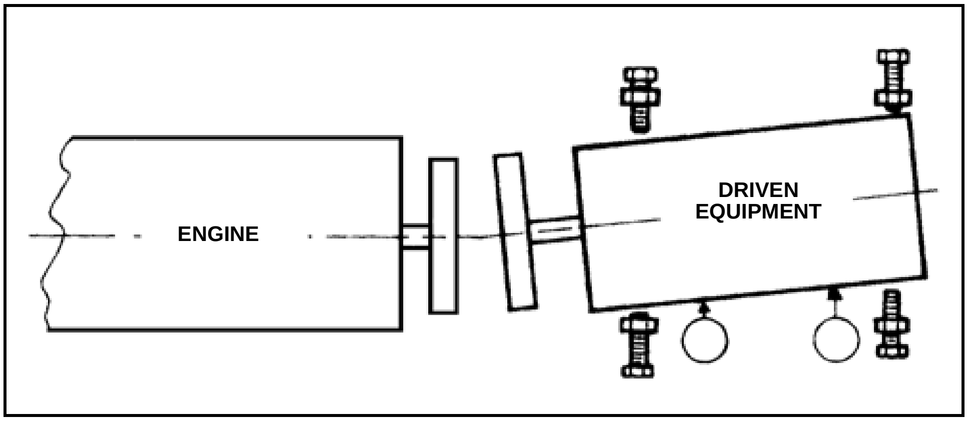


Figure 3-39 Indicator Positioning

2. Going to one corner at a time, loosen the mounting bolt and shim as calculated, then tighten the mounting bolt. Center mounts will have to be shimmed in conjunction with corner mounts. Note any horizontal movement that may occur on the dial indicators.
3. After shimming, loosen both mounts on one end and all center mounts. It may also be necessary to loosen one mount on the fixed end but do not loosen both. Slide the free end the amount calculated then re-torque the bolts.

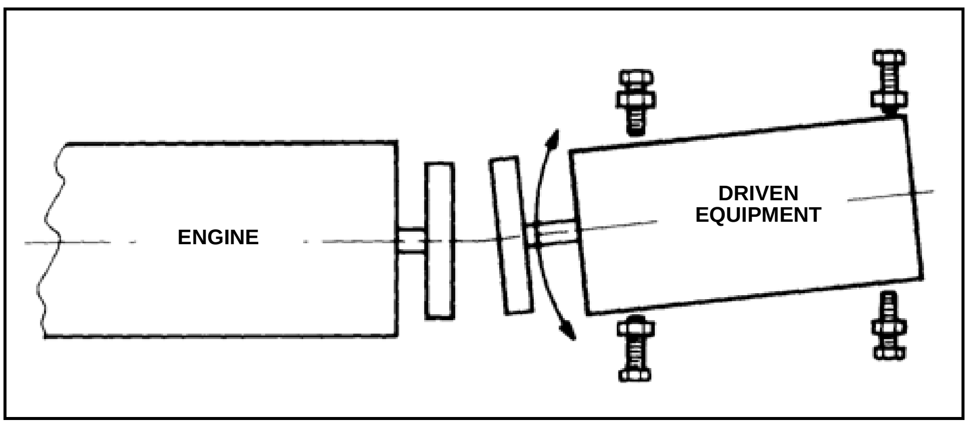


Figure 3-40 Slide Free End

4. Loosen both bolts on the opposite end and move as calculated. Re-torque all mounting bolts.

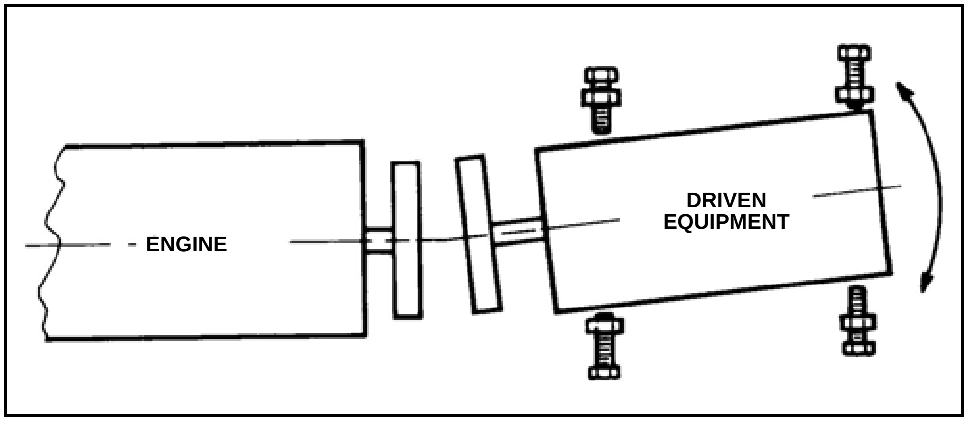


Figure 3-41 Move Opposite End

5. Check angular alignment again using the same procedure as used previously. Angular alignment is correct when total indicator runout is less than 0.005 in. (0.127 mm) per foot of radius from center of shaft to where the dial indicator reads.

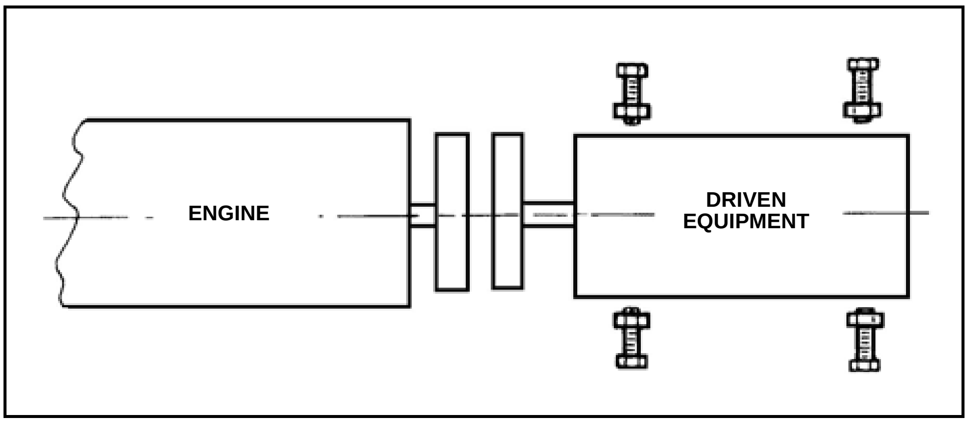


Figure 3-42 Correct Angular Alignment

Parallel Alignment

Parallel alignment can be checked and adjusted after angular alignment has been completed. It will, however, be necessary to re-check angular alignment after each adjustment. The following procedure can be used to measure parallel alignment.

1. Set up a dial indicator to read parallel alignment. If available, set up a second dial indicator to read angular alignment. This will allow you to rotate the shafts only one time to get both readings (see Figure 3-43).
2. Rotate both shafts to the 2 o'clock position (facing the flywheel) then back to the 12 o'clock position. Zero the indicator(s).
3. Rotate the shafts to the 9 o'clock position and record the readings.
4. Rotate the shafts to the 6 and 3 o'clock positions and record the readings.
5. Rotate the shafts back to the 12 o'clock position and verify that the indicators return to zero.

The amount of parallel misalignment is one-half the TIR (total indicator reading) for each direction.

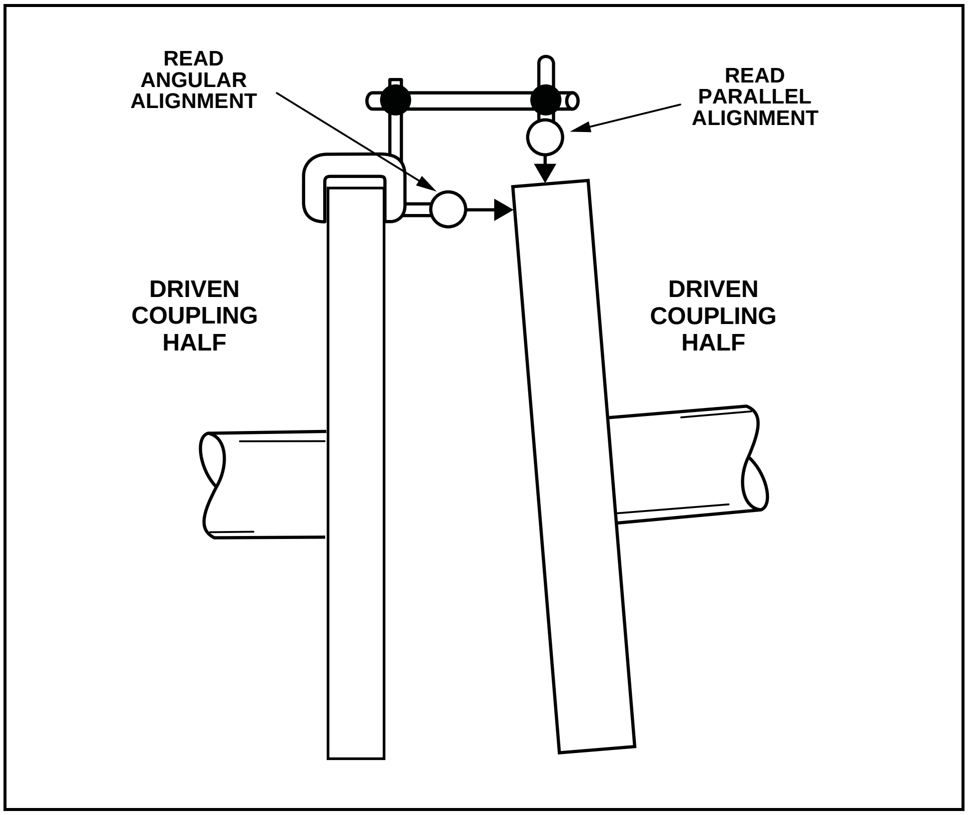


Figure 3-43 Measuring for Parallel Alignment

In this example, the vertical TIR is 0.020 in. (0.508 mm), thus the machines are vertically misaligned by 0.010 in. (0.254 mm). Horizontal TIR is the difference between (+0.015) and (+0.005) which is 0.010 in. (0.254 mm). Horizontal misalignment is half of the TIR which is 0.005 in. (0.127 mm). All mounts should get the same amount of adjustment, 0.005 in. (0.127 mm) in this case, to move the machine without losing angular alignment.

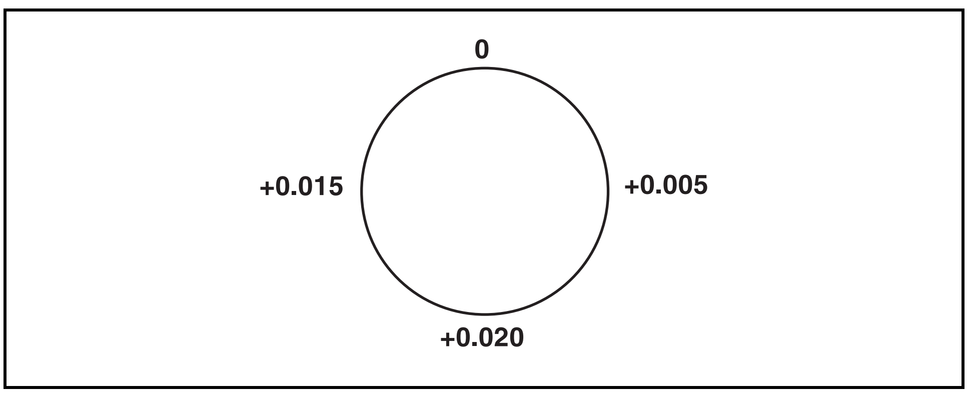


Figure 3-44 Total Indicator Reading (TIR)

Adjustment for parallel alignment is similar to that for angular and should be accomplished as follows:

1. Set up two dial indicators; one to monitor horizontal movement of the inboard mounts, and one to monitor horizontal movement of the outboard mounts. Zero the indicators.
2. Going to one corner at a time, loosen the mounting bolt(s) and shim as calculated, then torque the mounting bolt. Center mounts will have to be shimmed in conjunction with corner mounts.
3. After shimming, loosen both mounts on one end and all center mounts. It may also be necessary to loosen one mount on the fixed end but do not loosen both. Slide the free end the amount calculated then re-torque the bolts.
4. Loosen both mounts on the opposite end and move the same. Retorque all mounting bolts.
5. Check parallel alignment again using the same procedure as used previously. Parallel alignment is correct when total indicator runout is less than 0.005 in. (0.127 mm).

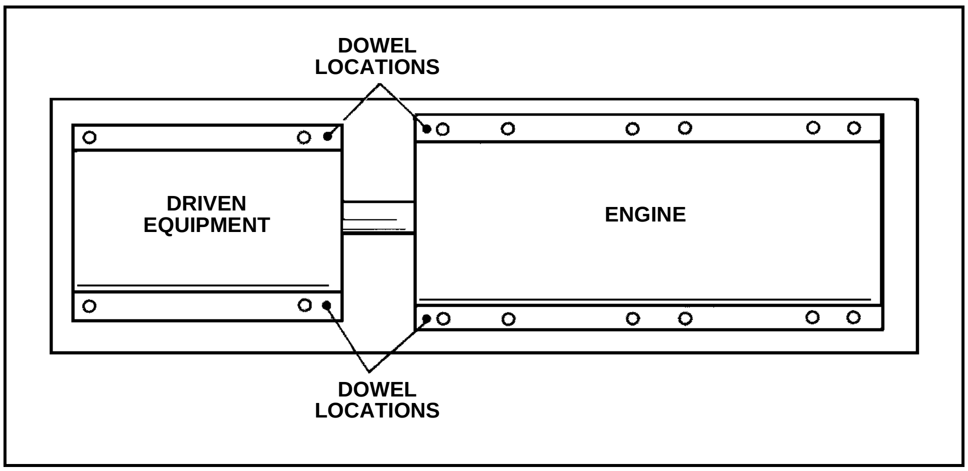


Figure 3-45 Dowel Placement

Thermal Growth

After angular and parallel alignment are satisfactory, it will be necessary to adjust alignment to compensate for thermal growth. This will allow the machines to be in good alignment after they reach operating temperature.

Crankshaft Growth

Table 3-8 lists the changes in crankshaft height that will occur due to the temperature change from 70°F (21°C) to normal operating temperatures (measured from the mounting surface of the base type oil pan).

Table 3-8 Thermal Growth

ENGINE MODEL	INCREASE IN CRANKSHAFT HEIGHT	
	INCHES	mm
F18/H24	0.012	0.31
L36/P48	0.020	0.51
16V150LTD	0.020	0.51
F3521, F3524, L5790, L5794, L7042, L7044	0.014	0.36

See Table 3-9 for the vertical thermal growth in the height of the 275GL crankshaft centerlines from the bottom of the crankcase pan rails (based on temperature change from 21° C (70° F) to normal operation oil temperature).

Table 3-9 Vertical Thermal Growth Crankshaft Centerline

VEES		
12V 275GL+ 16V 275GL+	480 mm (18.898 in.)	0.28 – 0.34 mm (0.011 – 0.013 in.)
		0.31 – 0.39 mm (0.012 – 0.015 in.)

Heat growth information for the driven equipment should be available from the manufacturer. If not, it can be calculated with the following formula:

$G_m = (T_m - 70) \times h \times E$ for °F or $(T_m - 20) \times h \times E$ for °C

Where:

G_m = amount of growth expected (inches or mm)

T_m = operating temperature of driven machines (°F or °C)

h = height from machine mounting surface to center of shaft (inches or mm)

E = thermal expansion coefficient for material machine is made from:

6.5×10^{-6} (0.0000065) in/in °F or 1.4×10^{-6} mm/mm °C for steel

5.8×10^{-6} (0.0000058) in/in °F or 1.1×10^{-6} mm/mm °C for cast iron

To adjust for thermal growth take the difference in machine growths and add that amount in shims under the machine which grows least. In the case of cooling compressors, the compressor gets cold when loaded and shrinks. This will require a further offset to compensate for engine growth and compressor shrinkage. The growth formula still applies for a cold compressor since the growth number will be negative.

To add the shims, loosen one mount at a time and add the shims then re-torque the bolts before moving on to the next mount. This prevents horizontal alignment from changing while adding shims. Parallel dial indicator readings will now indicate the machine which grows least is higher than the machine which grows more but the machines will be aligned when they reach operating temperature.

Check endplay to verify that the alignment procedure did not eliminate end thrust.

Doweling

If doweling of the machines is required, the following information is offered as a guide.

Doweling is a practice often used after aligning two machines to mark their correctly aligned positions. When dowels are placed correctly, they also determine the direction of thermal growth of the machines. Figure 3-45 illustrates where dowels should be placed to cause thermal growth in a direction which will not affect crankshaft endplay and will maintain correct alignment.

Tapered dowels are recommended for this purpose because they have the following advantages over straight dowels;

1. Tapered dowels will not fall through the skid from vibration or a slight gap between the hole and dowel.
2. If alignment changes from shipping of the complete package or settling of its foundation, the machines can be realigned and the tapered holes reamed deeper to fit the dowel in its new position.
3. Tapered dowels are removed easily by driving the pin out the large end.

Dowel holes should be drilled through the mounting foot, shim pack and the skid I-beam flange. No gaps should exist between the engine base and the skid.

Hot Check

Once the machines are aligned and offset for thermal growth, they should be checked when hot.

1. Start the engine and apply load.
2. Allow machines to run for one hour after reaching their operating temperatures.
3. Shut down and immediately check angular and parallel alignment and endplay. Alignment TIR should now be less than 0.005 in. (0.127 mm) for the VHP, both parallel and angular.
4. Adjust alignment and endplay if necessary.

Periodic Inspections

Engine base deflection and alignment must be checked periodically, at least once a year. Installations which are subject to settling of the concrete must be checked often (initially – monthly) to determine if settling is causing misalignment.

SECTION 5

ALIGNMENT CHECKLIST

Single Bearing Machine

NOTE: *Values in the checklist are mentioned for VHP; for others models use the values as mentioned in the preceding sections.*

1. Install and level engine or common base ______________________________
2. Adjust spring isolaters (if used) _____________________________________
3. Adjust base deflection at the four engine corners.

0. in. (mm)	0. in. (mm)	0. in. (mm)
0. in. (mm)	0. in. (mm)	0. in. (mm)

4. Shim center mounts

   +0.002 in. (0.050 mm) for a VHP 6-cylinder

   +0.004 in. (0.102 mm) for a VHP 12-cylinder

   +0.008 in. (0.203 mm) for a VHP 16-cylinder

5. Measure crankshaft web deflection (optional)

   All except rear throw 0.001 in. (0.025 mm) TIR max.

   Rear throw approximately 0.0015 in. (0.038 mm) TIR.

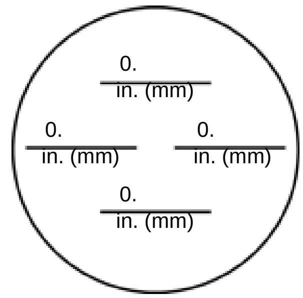
Throw	1	2	3	4
TIR	0. in. (mm)	0. in. (mm)	0. in. (mm)	0. in. (mm)

Throw	5	6	7	8
TIR	0. in. (mm)	0. in. (mm)	0. in. (mm)	0. in. (mm)

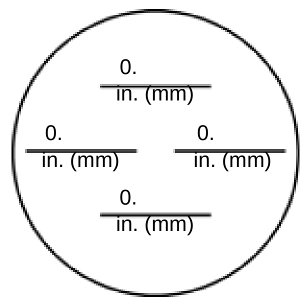
6. Adjust base deflection at four corners of driven machine.

0. in. (mm)	0. in. (mm)
0. in. (mm)	0. in. (mm)

7. Check and adjust shaft pilot centering (parallel alignment). Maximum 0.005 in. (0.127 mm) TIR.



8. Check and adjust angular alignment. Maximum 0.005 in. (0.127 mm) TIR at flywheel bolt.



9. Adjust for vertical growth

   Engine Growth _________ in. (mm) minus

   D. M. Growth _________ in. (mm) =

   Cold Alignment Offset _________ in. (mm)

10. Check crankshaft endplay _________ in. (mm) should be within service manual specifications.

11. Check air gap and fan setscrews (single bearing generator) ____

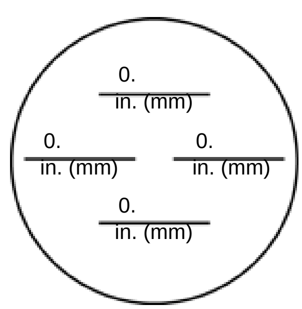
12. Start engine, run loaded, allow to warm up 1 hour minimum ____

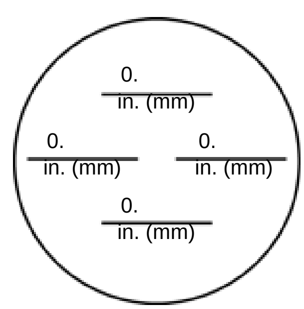
13. Shutdown and check hot angular alignment and endplay

    Endplay (Hot) _________ in. (mm)

    Alignment:

PARALLEL (HOT) **ANGULAR (HOT)**





Multiple Bearing Machine

1. Install and level engine or common skid ______________________________
2. Adjust spring isolaters (if used) ____________________________________
3. Adjust base deflection at the four engine corners.

0. in. (mm)	0. in. (mm)	0. in. (mm)
0. in. (mm)	0. in. (mm)	0. in. (mm)

4. Shim center mounts

   +0.002 in. (0.050 mm) for a VHP 6-cylinder

   +0.004 in. (0.102 mm) for a VHP 12-cylinder

   +0.008 in. (0.203 mm) for a VHP 16-cylinder

5. Measure crankshaft web deflection (optional)

   All except rear throw 0.001 in. (0.025 mm) TIR max.

   Rear throw approximately 0.0015 in. (0.038 mm) TIR.

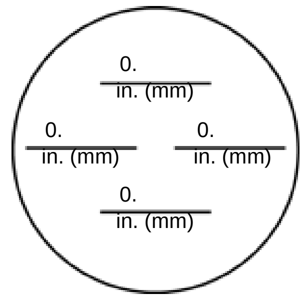
Throw	1	2	3	4
TIR	0. in. (mm)	0. in. (mm)	0. in. (mm)	0. in. (mm)

Throw	5	6	7	8
TIR	0. in. (mm)	0. in. (mm)	0. in. (mm)	0. in. (mm)

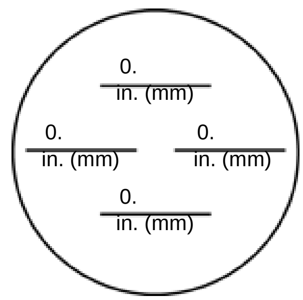
6. Adjust base deflection at four corners of driven machine.

0. in. (mm)	0. in. (mm)
0. in. (mm)	0. in. (mm)

7. Check for crankshaft endplay.
8. Check and adjust angular alignment. Maximum 0.005 in. (0.127 mm) per foot of radius from center of shaft to dial indicator read point.



9. Check and adjust parallel alignment. Maximum 0.005 in. (0.127 mm) TIR.



10. Adjust for thermal growth

    Engine Growth _______ in. (mm) minus

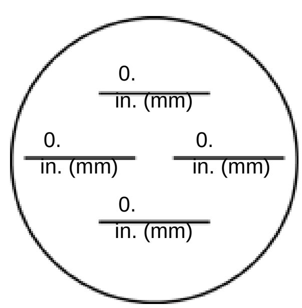
    D. M. Growth _______ in. (mm) =

    Cold Alignment Offset ________ in. (mm)
11. Recheck crankshaft endplay _______ in. (mm)
12. Start engine, run loaded, allow to warm up 1 hour minimum ________
13. Shutdown and check hot angular alignment and endplay

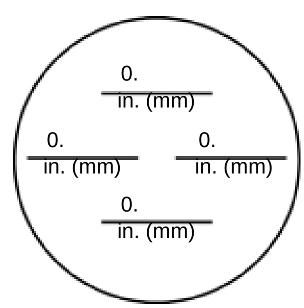
    Endplay (Hot) _________ in. (mm)

    Alignment:

PARALLEL (HOT)

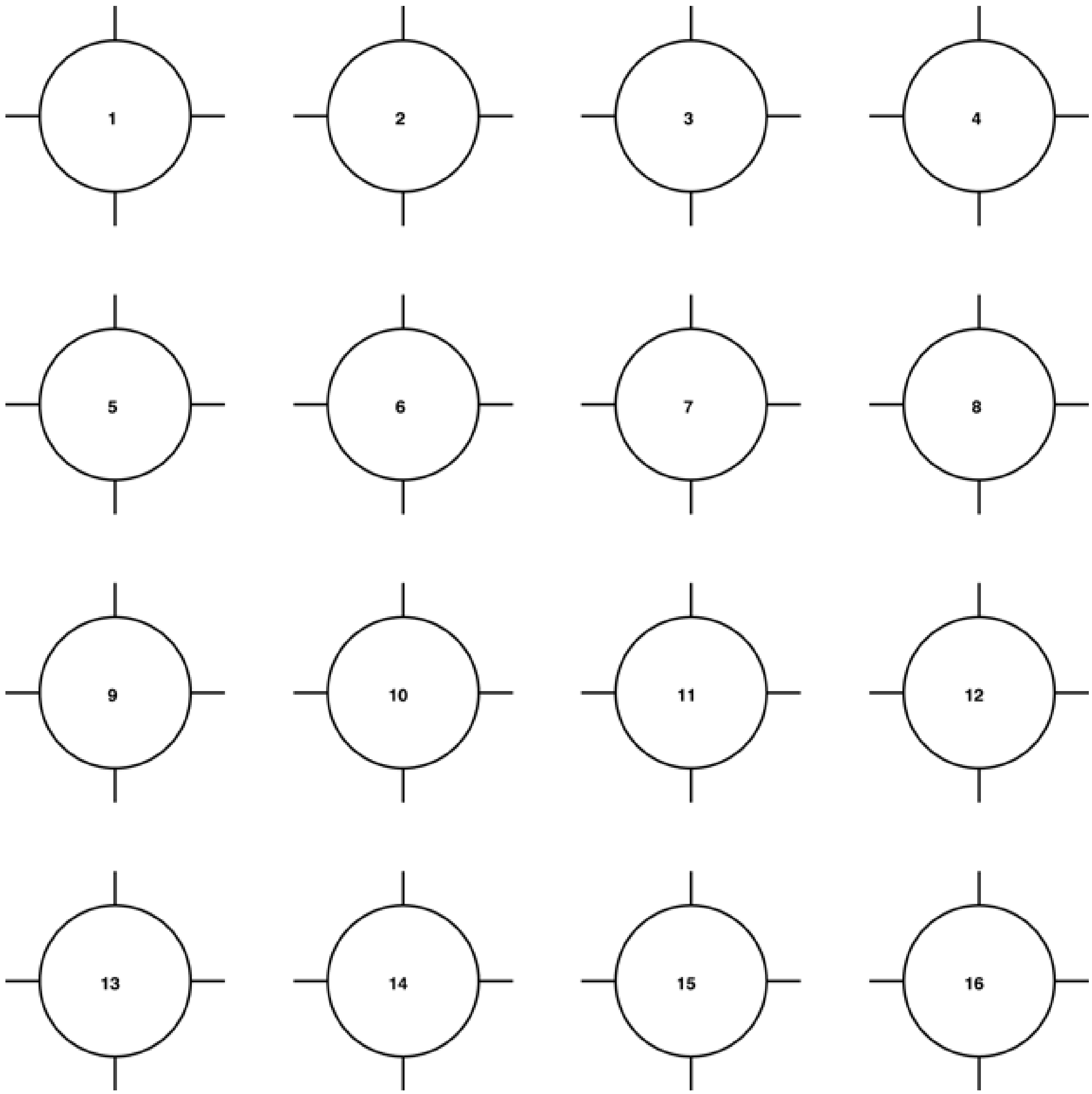


ANGULAR (HOT)



14. Dowel inboard machine mounts (if required).

ALIGNMENT PROGRESSION CHART



ENDPLAY BEFORE ALIGNMENT ____________________ ENDPLAY AFTER ALIGNMENT ____________________

ENGINE MODEL ____________________ SERIAL NO. ____________________

DRIVEN MACHINE MAKE ____________________ MODEL NO. ____________________

CUSTOMER'S OBSERVER ____________________ PERFORMED BY ____________________

DATE ____________________

Figure 3-46 Alignment Progression Chart

CHAPTER 4

VIBRATION, COUPLINGS AND DRIVES

INTRODUCTION

VIBRATION

Vibration is unavoidable in the design and construction of engine installations. However, it is possible to isolate this vibration or sufficiently prevent it from getting into the surrounding structures. This process is known as vibration isolation.

Vibration should be isolated from the surrounding structure for several reasons. Depending on the type of structure, engine vibration can range from being a nuisance to being a safety hazard. For example, hospitals require engine vibration to be isolated from the structure in order to assure patient comfort and proper effectiveness of medical equipment. In some structures, engine vibration can actually cause beams and supports of a building to vibrate harmonically with the engine, causing serious structural damage.

Couplings and Drives

The energy produced by the engine is transmitted to the driven equipment through a system of couplings and drives. These drives must act both as a power transmitter and as an isolation of the engine vibration from the driven equipment. Never install any type of coupling or drive without first consulting the coupling manufacturer. Different companies and the various mechanical designs of their equipment lead to many different lubrication, fluid fill, installation, and cooling recommendations. Improper installation or maintenance of a coupling or drive will not only shorten the life of the components, but will lead to serious engine/driven equipment damage.

SECTION 1

ENGINE BASE DESIGN

When a base is not supplied by Waukesha gas engines, the packager assumes responsibility for the base design. Any package being assembled by a party other than Waukesha gas engines should have a vibration study performed and tests completed for assurance of installation integrity against vibration at the site. Information on engine unbalanced forces and moments can be found in the Waukesha gas engine technical data; vibration limits can be found in the Application and Installation section of Waukesha gas engine Service Bulletins.

When designing bases to be used with Waukesha gas engines, the engine base must be a rigid design to maintain alignment between the engine and the driven equipment. Base flexing due to lack of torsional rigidity is a major cause of misalignment. When designed correctly, the base must offer rigidity adequate to oppose the twist due to torque reaction on drives for driven equipment mounted on the base assembly and not bolted to the engine. The design must prevent any excessive bending forces that could be transmitted to the engine block and any components in the drive train. A modal and torsional analysis must be performed to validate the base design using MESD and maximum unbalanced forces and moments data for the engine. A third-party engineering firm may be required to perform this analysis.

The entire package must be able to withstand normal handling during transportation without permanently distorting the base or causing misalignment of the engine or driven equipment.

The base must limit torsional and bending moment forces caused by torque reaction and flexing of the foundation substructure or vibration isolators under the base.

The base must be free of linear and torsional vibration in the operating load and speed range of the engine, and have a natural frequency such that resonance does not occur during the machinery's normal work. Any significant resonance that might occur must be outside the normal operating speed range of the engine.

The base must maintain engine and driven equipment alignment under all operational and environmental conditions.

Designs that rigidly mount the base to the foundation are preferred over using vibration isolators. The use of isolators causes the base to react all of the transmitted torque and eliminates the “path to ground” for the engine's unbalanced forces.

Special consideration must be taken for bases designed for vibration isolators to ensure the base is designed to limit torsional and bending moment forces and prevent flexing of the base while mounted on vibration isolators. The base must maintain equipment alignment under all conditions.

Vibration isolators between the driven equipment and skid, or engine and skid, are not acceptable for use with Waukesha gas engines.

Skid designs with a step down base feature between the engine and driven are not recommended and as an alternative, it is recommended that the mounting feet of the driven equipment are modified to use a continuous I-beam skid design with one level plane for mounting the engine and driven equipment.

Mounting of any ancillary components by a packager may result in unwanted vibration of those components. Appropriate lifting capability for lifting the complete package must be provided as part of the base design. Engine lifting eyes are not to be used for lifting of a packaged unit.

SECTION 2 VIBRATION: LINEAR VS. TORSIONAL

Torsional

Vibration can be classified as either linear or torsional. One of the largest contributing factors to vibration problems is misalignment. Misalignment accounts for 50 to 75 percent of all machinery vibration problems.

Crankshaft torsional vibration refers to the angular twisting of the crankshaft relative to the center of rotation. Since torsional vibration cannot be seen or felt by hand, it must be measured with special equipment.

The following engine situations may lead to excessive torsional vibration:

- Misalignment
- Bank to bank imbalance
- Uneven firing pressures
- Cylinder misfires
- Uneven ignition timing
- Incompatibility of the engine, couplings, and driven equipment
- Faulty vibration damper

To help limit the possibility of damage to the crankshaft, gear train, or coupling, vibration dampers are mounted on the front of the engine to reduce torsional vibration. Dampers will lose their ability to dampen as they age and therefore must be replaced. However, since damper life cannot easily be determined, it is recommended that they be replaced every five years, 35,000 to 40,000 operating hours or in the event of a crankshaft failure. Inspection and fluid sampling should be done every 15,000 hours.

A torsional analysis must be performed to determine compatibility of the drive line components when the components are used together for the first time. Waukesha can provide a torsional analysis, or the engine mass elastic information, if an outside consultant is used.

Linear

Linear vibration can be described as an oscillating motion or “shaking” about a reference position. When linear vibration is excessive it may be seen or felt by hand. One of the most common causes of linear vibration is misaligned shafts. Other causes of linear vibration are listed below.

- Component imbalance
- Crankcase deflection
- Poor operating conditions

Generally, proper maintenance can help minimize linear vibration. Crankcase deflection and poor operating practices are other common causes of linear vibration because they upset the inherent cylinder to cylinder balance built into the engine.

SECTION 3 FLEXIBLE PIPING CONNECTIONS

Engines and Enginators are generally mounted such that vibrations are isolated from the environment by AVM (Anti Vibration Mounts), isolator springs or the springs carrying the inertia block. This allows movement relative to the surrounding equipment. The engines flexible connections for cooling, fuel, exhaust, intake, etc. have to allow for thermal growth of the engine. Care should be taken that the flexible connections are fit for the intended use. Type and magnitude of movement, stresses, temperature, pressure, transported medium, etc should be considered. Special care has to be taken with spring isolator mounted engines and Enginators as they tend to make larger movements than inertia block mounted engines and Enginators. In all cases the magnitude of the movements depends on the spring rate of the mounting used and the acting forces and moments. Where applicable seismic activity must be considered and seismic snubbers applied.

Flexible connections should be used for the kind of movements they were designed for. Most flexible connections like those available from Waukesha for the APG 1000 are designed for primary application in the lateral mode. The axial mode (compression/expansion) and torsional mode capabilities are minimal and installation for primary operation in this mode must be avoided.

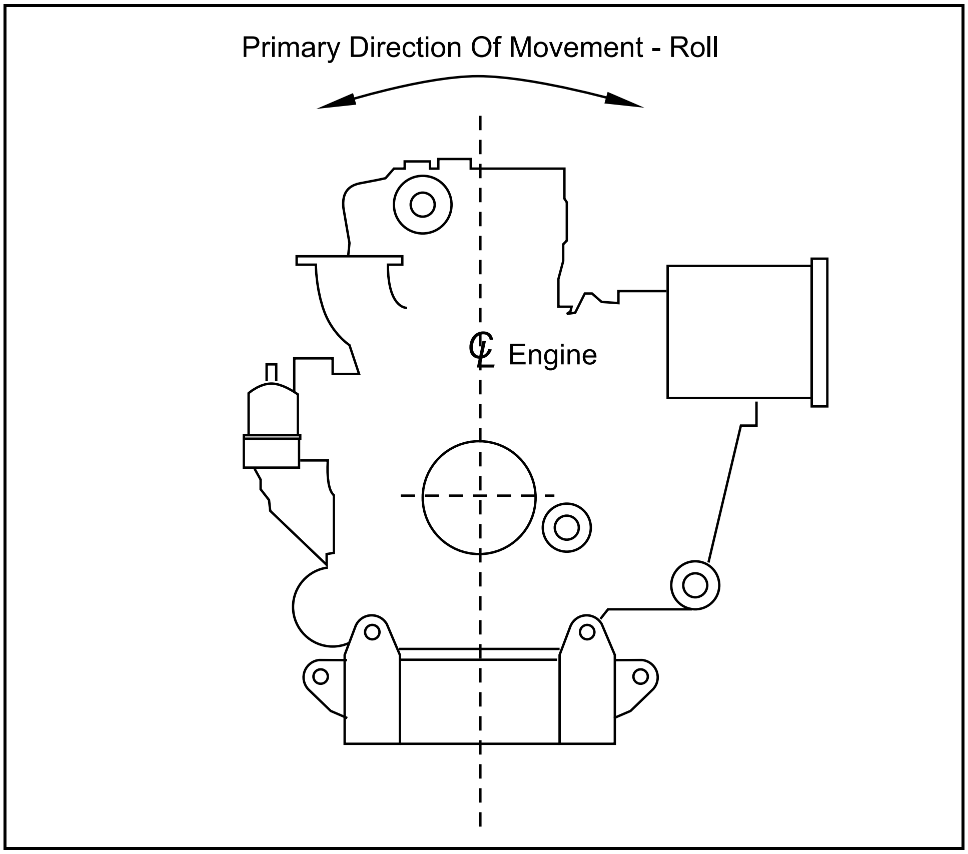


Figure 4-1 Primary Movement of an Engine

The primary direction of movement for an engine or Enginator is side to side (roll). The connections should be installed facing the front or rear of the engine, parallel to the crankshaft, to operate in lateral mode. The exhaust flexible connection may be installed vertically but has to be designed for this application. The exhaust connection optionally supplied by Waukesha for the APG 1000 was designed for this type of application. The material was chosen such that it withstands not only the lateral movements but also the relatively large axial movement caused by the thermal growth of the engine.

See Chapter 8 (Exhaust Systems) for additional information.

Flex connections and especially those meant for isolator mounted engines and Enginator sets should be designed for the displacement at the engine connection point.

For solid mount units the connection point should be lowest as this reduces the magnitude of displacement. Isolator spring mounted units tend to move around their center of gravity which is close to the crankshaft center line. Connections for isolator spring mounted units should therefore be closest to the crankshaft as this is where movements are smallest.

Flex connections delivered by Waukesha included as part of the standard package or ordered optionally were designed to compensate thermal growth of the engine. Additional flex capabilities are required when compensation of thermal expansion and movement of the facility piping. Flex connections delivered by Waukesha will be suitable for solid mount (inertia block) units only unless use for spring isolated units is clearly specified. Flex connections supplied by Waukesha are fit for spring isolated units only when this is specifically stated.

When calculating the restriction of piping it must be considered that flex connections impose higher restrictions than an equal length of pipe. Bends in the flex connection reduce its lifetime and increase the restriction and therefore should be avoided.

A flexible connection generally is a weak link and should be inspected periodically. To allow easy replacement of the flexible connection it is advisable to place them such that isolation for repair and replacement can be easily accomplished.

Do not bend or twist flex connections during installation or transport.

Refer to WPS10/91 for more general information or the individual chapters for more specific information on flex connections for the cooling systems, exhaust, fuel, etc.

SECTION 4 ELECTRICAL CONNECTIONS

Wire harness connectors are not designed to support the weight of the wiring harnesses or conduits. Harnesses, conduits and wire ways should be supported to avoid excessive strain on the connectors. When installed, wires should not be placed under tension by supporting a load or due to engine vibration.

SECTION 5 DRIVE TYPES

Flexible Couplings

Flexible couplings are semi-rigid connectors designed to transmit torque and to absorb torsional vibration being transmitted from the engine to the driven equipment. These flexible couplings will also accommodate small amounts of mechanical misalignment.

NOTE: *Always consult the manufacturer for precise installation specifications.*

When installing flexible couplings, make sure to check the coupling manufacturer's fits and clearances table to allow for sufficient engine crankshaft endplay. The coupling must be able to absorb crankshaft elongation caused by engine heat. In addition, the coupling must also be capable of absorbing the misalignment caused by the heat growth of the engine.

NOTE: *Always consult the engine operators manual or performance specifications for proper crankshaft endplay.*

Flexible couplings will not absorb the effects of significant misalignment. Although flexible couplings can tolerate some degree of misalignment, the amount is so small that the same exacting alignment procedures should always be followed.

Floating Drive Shafts

A floating drive shaft is installed to fill a space between the engine and driven equipment. The shaft is commonly equipped with either universal joints or flexible couplings and can compensate for some degree of misalignment or angularity. When using flexible couplings it is important to use a coupling that is compatible with a floating drive shaft. A splined shaft can also compensate for longitudinal movement. Shaft selection is determined by engine speed, load, torque, distance between engine and driven equipment, degree of angular operation anticipated, and heat expansion of engine components.

NOTE: *Selection of the correct shaft, universal joints or flexible couplings should be done by a supplier fully acquainted with the factors involved.*

Gear Drives

An installation can use gear drives to change the speed or direction output power shaft rotation of driven equipment. Speed increasing or reducing gear drives can be used when the engine speed does not match the required speed of the driven equipment.

One type of application is for a gearing down operation, such as using a high-speed compact standby engine to drive a low-speed pump. By gearing a smaller high-speed engine to run in a more practical speed range, the initial engine cost is reduced and the accessory is run just as effectively as if a slower speed larger engine were used for the same power requirement.

Another practical application for gear drives is using two engines at opposite ends of a common shaft for driven equipment, such as a pump, generator, or similar unit. Unless one engine is of the opposite rotation, a gear drive will be needed to provide the driven unit with the correct rotation. Gear drives can be designed for in-line or side load applications.

NOTE: *Gear applications should be engineered by all parties involved: engine manufacturer, applications engineer, driven unit manufacturer, and gear supplier.*

Hydraulic Drives and Torque Converters

A major advantage of a hydraulic coupling or torque converter is the absence of any solid mechanical connection between the engine and driven equipment. Thus the effects of mechanical shocks, vibrations, and rapid load changes are eliminated or greatly reduced, protecting both the driven equipment and engine. In both hydraulic couplings and torque converters, engine input is absorbed by a turbine pump. The oil or fluid within the pump housing is accelerated as the equipment rotates and the engine power is delivered at the outer edge of the pump in the form of high velocity fluid. The high velocity of the fluid will cause heat and therefore both torque converters and hydraulic couplings may require independent cooling systems. Consult the manufacturer of the coupling or torque converter for cooling requirements.

Hydraulic couplings are used to compensate for shock loads, overloads, and as an aid in no-load starting and load pickup. In a hydraulic coupling, a matching turbine and stator absorb the energy of the moving fluid and deliver it to the driven equipment. The output torque of a hydraulic coupling never exceeds the input torque.

Torque converters are very similar to hydraulic couplings. However, in addition to the turbine, one or more reaction members are positioned in the fluid flow in such a manner as to produce additional torque at the output shaft. Ratios of about 3:1 are most common; however, torque ratios may range from 2:1 to 6:1 or slightly higher. At full engine speed, with an equal output shaft speed, the converter acts as a simple fluid coupling without torque multiplication. The importance of matching the converter to the engine and its intended load and operating range cannot be overemphasized. Below is a typical list of necessary information for proper torque converter selection which is the responsibility of the system designer (Consult the equipment manufacturer).

TORQUE CONVERTER SELECTION INFORMATION

Application ____________________

Type of driven equipment ____________________

Engine model ____________________

Engine BHP @ RPM ____________________

Engine SAE flywheel housing size ____________________

Accessory deductions ____________________

Governed speed set for application ____________________

Stall torque ratio desired ____________________

Other relevant data (minimum stall torque, etc.) ____________________

Clutches

Engines must be started under no-load conditions, or conditions where engine load builds up slowly. Applications with rapid load build up such as piston pumps or air compressors require some method of disconnecting the load from the engine during startup.

Clutches and power take-offs (PTOs) provide a means of disconnecting this load from the engine for starting and warm-up. One commonly used clutch is the friction type, which requires manual or automatic engagement of the clutch and pressure plates. The friction causes the two plates to turn as one, transferring the energy from the engine to the driven equipment. Friction between clutch discs and pressure plates is increased by mechanical pressure during clutch engagement. Acceleration is allowed by the slippage between the discs and clutch plates until friction is sufficient to rotate the driven equipment at engine speed.

Power take-offs are designed for either side load or in-line drive applications. Side load PTOs are made with taper roller bearings which generally cannot be used for in-line applications. PTOs with ball bearings are designed for direct drives, but can accept minor side loads if within the specified limits set by the manufacturer.

Centrifugal clutch assemblies allow for engine acceleration periods before load pick up. As the engine reaches a predetermined speed the clutch will automatically pick up the load or dump load when it falls below this speed.

Clutches are designed for working with side or front load conditions, although the allowable side loads are generally less than straight in-line drives. Clutches are also designed for the type of drive to be used (belt, chain, and sprocket, etc.).

CLUTCH SELECTION

Clutch capacity is based on rated engine torque and the appropriate service factor (see below).

Required Clutch Capacity = Engine Torque x Service Factor

[English] Engine torque (lb-ft.) = HP x 5252/RPM

[Metric] Engine torque (N·m) = kW x 9549/RPM

SAMPLE SERVICE FACTORS

Centrifugal Pump 1.5

Farm Tractor PTO 2.0

Reciprocating Pump.. 3.0

Mud Pump.. 3.0

Rock Crusher 3.0

Reciprocating Compressor.. 4.0

Blower 4.0

NOTE: *Consult clutch manufacturer for specific service factors.*

CLUTCH SELECTION INFORMATION

Type of driven equipment ______________________________

Prime mover (diesel engine, gas engine, propane engine, etc.) ______________

Torque, horsepower, speed at maximum torque, and maximum speed of prime mover ______________________

Torque to be transmitted (and what speed) if other than 100% of prime mover ______________________

Is slip time control required? If yes, amount of time ________________

Maximum expected engagement frequency per hour ________________

Load pulsating or intermittent in nature? ______________________

SECTION 6 SIDE LOAD DRIVES

Side loads derive their power from belts or chain drives running perpendicular to the engine crankshaft. Unless specifically designed for side load applications, the same amount of power cannot be taken off a side load application as can be derived from a straight in-line drive. Side load applications must be carefully designed to avoid excessive wear on the power take-off, clutch, and bearings.

NOTICE **Standard side load applications use belt drives. All side load drives other than belt drives must be analyzed by Waukesha Application Engineering Department for torsional compatibility.**

Many factors have to be considered when designing a side load installation. Engine RPM, sheave design, clutch design, and use of an outboard bearing all affect the success of this type of application. Before using a side load application, it is recommended to contact the Waukesha Application Engineering Department for assistance.

Unsupported pulley overhang (a load carrying pulley mounted far out on the shaft) should be avoided because it will distort the engine flywheel housing and will cause clutch, crankshaft, and/or bearing failure. To avoid such problems, an outboard bearing should be added at the end of the shaft.

Improper pulley selection for the design load is another common source for problems. Too few belts, incorrect belt size or type, or insufficient wrap (contact between belts and pulley) can lead to belt slippage.

Care must also be taken when tightening the belts to prevent slippage. Overtightening of the belts can result in rapid belt wear, engine bearing failure, and poor performance.

SECTION 7 DETERMINATION OF CORRECT PULLEY SIZE

The following formula is used to find an unknown pulley size or operating RPM.

$$\frac{\text{RPM (driven)}}{\text{RPM (driving)}} = \frac{\text{Pulley Diameter (driving)}}{\text{Pulley Diameter (driven)}}$$

NOTE: *When performing calculations, always be certain that either pitch diameters or outside diameters are used for both pulleys. Use the same diameter for each pulley when making this check.*

EXAMPLE:

An engine is governed at 1700 RPM and equipped with an 8 in. (20 cm) driver pulley. If the pump must operate at 3000 RPM, what size pump pulley must be installed?

$$\frac{\text{3000 RPM}}{\text{1700 RPM}} = \frac{\text{8 in. (20 cm)}}{\text{Pulley Diameter (driven)}}$$

1.76 = 8 in. (20 cm) / Pulley Diameter (driven)

Pulley Diameter (driven) = 4.5 in. (11.5 cm)

SECTION 8 BELTS AND PULLEYS TROUBLESHOOTING GUIDE

Table 4-1 Troubleshooting Table

SYMPTOM	PROBABLE CAUSE	REMEDY
1. Belt bottoming in pulley.	a. Belt and/or pulley are worn	a. Replace/Repair.
	b. Belt too narrow for the pulley	b. Use wider belt or correct pulley width.
	c. Pulley split open.	c. Use stronger pulley or lower belt tension.
	d. Pulley groove not deep enough.	d. Increase depth of pulley groove.
2. Belt turn over. Belt turns over in the pulley and runs on its side.	a. Caused by harmonics in the drive.	a. Determine cause of harmonics and try to correct by carefully analyzing installation. Replace belt.
	b. Misalignment	b. Align.
3. Belt flips off pulley.	a. Can have same causes as symptom #2.	a. See symptom #2.
	b. Foreign object forces belt out of pulley groove.	b. Remove object, replace belt.
	c. Belt flips off due to failure of another drive belt.	c. See symptom #2.
4. Belt stretching outside circumference increases abnormally under tension and load. Results in large tension loss or belt running out of adjustment when tensioned.	a. Overstressing the belts.	a. Eliminate tension.
	b. Wrong belt tensile member for particular application.	b. Contact original equipment manufacturer for possible improvements.
	c. Tension is too high.	
5. Clean break of belt.	a. This is an unusual occurrence that is generally due to a foreign object being wedged in a pulley, damage at installation, or as a secondary failure.	a. Replace belt.
6. Belt squeal. Very distinct high pitched noise.	a. Occurs most frequently due to temporary overloading of the belt. This overloading causes the belt to start slipping.	a. Re-tension belt. Replace with new belt if old belt is worn.
7. Excessive whipping of the belt.	a. Most pronounced in long belt spans due to minor torsional and belt dimensional vibrations.	a. Reduce belt span length, dampen torsional or use selected belts. Belt span length can be changed by adding an idler in the long span.
8. Belt vibration, closely related to whipping (7) but produces a distinct beat.	a. Reasons for this condition are unknown, but generally coincide with another vibration source.	a. Eliminate or change the vibration characteristics in the belt or other source. Check pulley condition.

Table 4-1 Troubleshooting Table (Continued)

SYMPTOM	PROBABLE CAUSE	REMEDY
9. Belt squeak (chirping noise).	a. Generally not objectionable. Commonly occurs when starting the engine in the morning, mostly due to humidity. Sometimes pulley surface condition is involved.	a. Contact your distributor. Some belt constructions are more prone to make noise than others. Not much can be done to overcome this situation.
10. Premature wear (accompanied by dark dusty deposits on inside of guards).	a. Misalignment	a. Align.

SECTION 9 V-BELT TIPS

1. Always use new, matched sets of belts in engine installations.
2. When replacing belts, always replace the entire set of belts not just the ones that look worn.
3. To check belt tension, depress the belt with your hand. At proper tension the belts will feel alive and springy. If the belts are too tight they will have no give at all. Loose belts will feel dead.
4. Keep the belts at proper tension. Loose belts will slip, robbing the system of power. Also, the heat produced by the friction of the belt slipping over the sheave will speed up belt decay. Belts that are too tight will wear more rapidly, and also cause the engine shaft bearings to deteriorate.
5. When designing a pulley system, an idler should be incorporated to make belt tension adjustments. The use of idlers is recommended over back idlers to prevent undue stress and wear to the drive belt during operation.
6. To avoid belt damage, always move the idler or pulley when installing belts. Never pry a belt over a pulley.
7. Idler should be positioned on the slack side of the pulley system when designing the system.

SECTION 10 FRONT END DRIVES

On specially engineered applications, power can be taken from the engine front end. The power can be transmitted through a side load or direct drive (torsional) design.

Direct Drive Applications

In a direct drive installation, a stub shaft is mounted to the crankshaft pulley. The power taken off the front end through a direct drive is measured in pound feet (Newton-meters). The torque requirements can be calculated with the following equations.

[English] Torque (lb-ft.) = HP x 5252/RPM

[Metric] Torque (N·m) = 9549 x kW/RPM

Where:

HP (kW) = Power required to drive the accessory

RPM = Engine Speed

The torque value obtained in these calculations is then compared to latest edition of Waukesha Technical Specification S4052-13 to ensure that it meets the established limits for the appropriate engine.

SECTION 11 SIDE LOAD APPLICATIONS

In a side load installation, belts or chain drives are used to transmit power from the crankshaft pulley located on the front of the engine.

Two important factors to consider for side load applications are:

1. The side load bending moment of the drive system
2. The distance from the front face of the crankcase to the mid-point of the drive. The maximum distance for this dimension can be found in latest edition of Waukesha Technical Specification S4052-13. The side load exerts a bending force, or moment on the crankshaft. Waukesha has established a maximum side load bending moment for its line of engines. For these values and sample calculations see latest edition of Waukesha Technical Specification S4052-13.

To bring an unacceptable side load into an acceptable range, either use a larger pulley, decrease the distance from the center of the belt drive group to the front of the crankcase, or decrease the size of the accessory to be driven. For assistance on side load applications, contact the Waukesha Application Engineering Department.

SECTION 12 CHECKLIST

Flexible Couplings

- ❑ Properly secured to driven and driver shafts?
- ❑ Properly aligned?
- ❑ Proper crankshaft endplay allowed for?
- ❑ Coupling cooling system properly installed (if applicable)?
- ❑ Correct coupling selected for application?

Flexible Drive Shafts

- ❑ Is shaft the right length?
- ❑ Properly coupled to driven and driver shafts?
- ❑ Universal joints or flexible couplings properly secured?
- ❑ Universal joints properly lubricated?

Gear Drives

- ❑ Securely mounted to base?
- ❑ Properly coupled and aligned to driven and driver shafts?
- ❑ Oil filled to proper level with the correct oil?
- ❑ Heat exchanger properly installed (if applicable)?

Hydraulic Drives/Torque Convertors

- ❑ Fluid level correct?

- ❑ Correct type of fluid?
- ❑ Cooling system properly installed (if applicable)?
- ❑ Properly lubricated?
- ❑ Properly aligned and secured to driven and driver shafts?

Clutches and PTOs

- ❑ Properly aligned and mounted to flywheel housing?
- ❑ Proper PTO for direct drive or side load application?
- ❑ Was side load limit properly calculated?
- ❑ Outboard bearing properly installed (if applicable)?
- ❑ Clutch/PTO properly sized for application?

V-Belt Drives

- ❑ Was side load limit properly calculated?
- ❑ Belts adjusted to proper tension?
- ❑ Are all belts from a matched set?
- ❑ Proper use of idlers for belt tightening?
- ❑ Is pulley proper size for application?
- ❑ Belts proper width for application?

NOTES

CHAPTER 5

COOLING SYSTEMS

INTRODUCTION

An engine typically converts 30 – 45% of the fuel's energy into mechanical energy. The remaining energy is dissipated as thermal energy (heat). About 30% of this thermal energy is carried away with the exhaust gas and radiation from the engine. The rest of the thermal energy is removed by the engine's cooling systems.

Most industrial engines have up to three heat exchanging systems in the engine for removing thermal energy. These are:

- **Engine Jacket** — A series of cooling passages through the engine which allow coolant to absorb heat from cylinder liners, cylinder heads, and exhaust manifolds.
- **Oil Cooler** — A liquid-to-liquid heat exchanger which cools lubricating oil by transferring heat to the coolant. One function of lubricating oil is to absorb heat from hot internal components, such as pistons.
- **Charge Air Cooler** (turbocharged engines) — A heat exchanger which removes heat from the combustion air charge (or combustion air/fuel charge on draw through) to maintain an acceptable temperature for efficient engine operation.

The heat absorbed from the engine components is contained in the coolant and must be removed. Failure to remove this heat will cause the coolant and engine temperature to increase beyond acceptable levels. The coolant removes heat by an increase in temperature or a change of phase (water to steam). A system which removes heat by increasing the coolant temperature is known as a "Solid Water" system.

Engine heat rejection data for ISO standard conditions and when operating on commercial-quality natural gas is available in Waukesha's Technical Data. Unless the equipment is operating at ISO standard conditions, this standard heat rejection data should not be used for sizing cooling equipment such as radiators or coolers. Site-specific engine heat rejection data is available in the latest version of the EngCalc program and must be used for sizing cooling equipment. For special applications or site conditions, contact Waukesha application engineering for engine heat rejection.

SECTION 1

SOLID WATER COOLING SYSTEM TYPES

Solid Water Cooling Systems

Three common "Solid Water" methods for removing heat from the coolant are:

– Radiator

– Heat Exchanger

– Cooling Tower

Radiator

A radiator removes heat from the coolant by exchanging it with ambient air which is cooler than the engine coolant.

Radiators are typically used when an application has no use for the engine heat, so it is dumped to the ambient air.

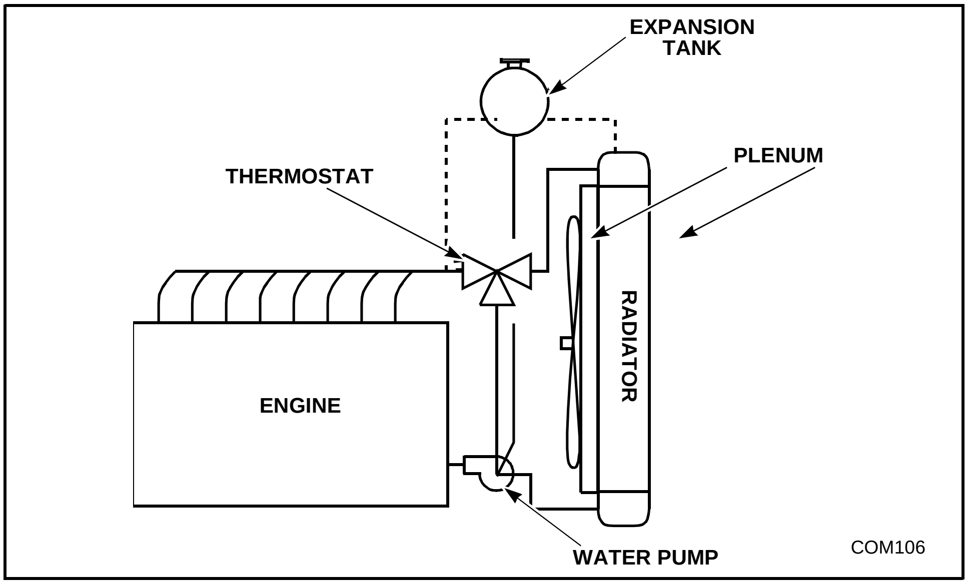


Figure 5-1 Schematic of Engine and Radiator

Cooling fans on a radiator can be engine-driven or powered by electric motors. The drive type is determined by the radiator type and the specific application requirements.

Engine-driven fans are either a pusher or suction type. A pusher fan draws air from the engine side of the radiator core, pushes it into the plenum, through the radiator core, and discharges it to the atmosphere. The temperature of the air entering the radiator is higher than the ambient air temperature due to radiant heat picked up from the engine and driven machine. This increase in air temperature must be considered when sizing the radiator.

Suction fans draw ambient air through the radiator and discharge it on the engine side. This results in an increase in engine room temperature. Engine air intakes, ignition systems, and other electronics must be protected from hot air. Hot air at the engine air intake will decrease available horsepower and promote detonation. Hot air on the ignition and electronic components can cause them to malfunction or break down. Both pusher and suction fans are illustrated in Figure 5-2.

Air flow restriction of an engine room or engine enclosure and louvers must be considered when the radiator fan is used for ventilating the enclosure. The fan supplier will be able to determine the maximum allowable restriction beyond the radiator (typically 0.25 in. [6.35 mm] water column). The power required to drive the fan must be deducted from the available engine power rating.

Electric motor-driven fans are often used when a radiator is located in a position which is difficult to drive from the engine. Remote or horizontal core radiators will use electric motor-driven fans. An electric motor-driven fan can be speed-controlled based upon coolant temperature. Using controls like this may save electric power during light load operation or cold ambient temperature operation. These controls must be adjusted to prevent thermocycling, which may cause thermoshock to the radiator and engine.

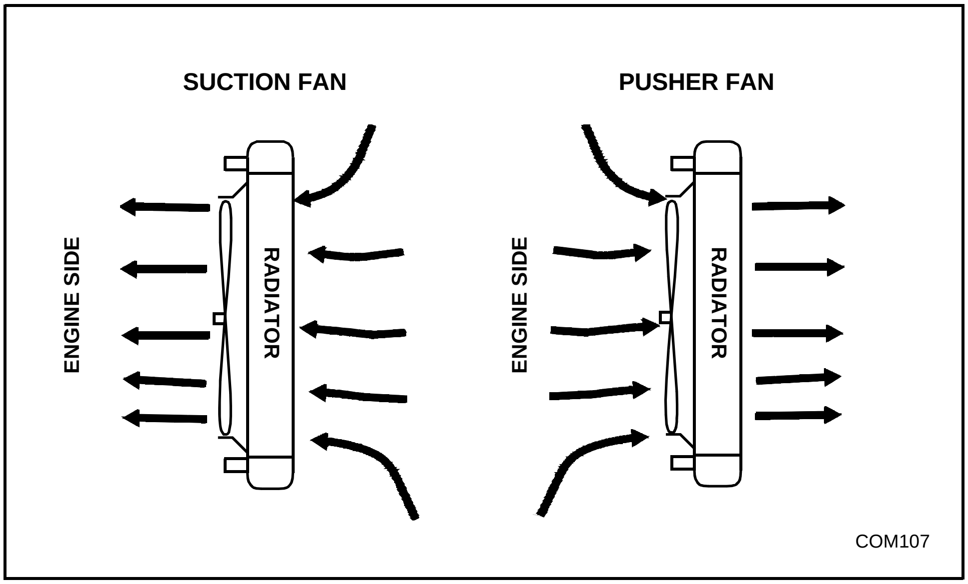


Figure 5-2 Suction and Pusher Fans

A radiator requires a static chamber either as a separate expansion tank or as a static expansion chamber in the radiator top tank. Not all radiator manufacturers provide a static chamber. The packager must confirm that the chamber exists for main and auxiliary circuits, and is properly connected (see “Expansion Tanks and Surge Tanks” on page 5-14).

It is important to consider the effects of the following on the radiator’s heat transfer capacity:

- Wind
- Hot air from other equipment
- Solar heating of surrounding ground
- Hot air recirculation
- Exhaust heat recirculation

See “Installation Concerns” on page 5-23.

Heat Exchanger

A heat exchanger removes heat from the engine coolant and transfers the heat to a secondary fluid. Heat exchangers are often used where installation of a radiator is not practical or where there is a use for the heat.

Common heat exchangers are the shell and tube type and the plate and frame type. A shell and tube heat exchanger consists of a bundle of tubes with coolant flowing through them. The shell is around these tubes and contains the other heat transfer fluid. Heat is transferred through the walls of the tubes. The highest heat transfer rate for a single pass shell and tube heat exchanger is generally achieved with a counterflow design as shown in Figure 5-3.

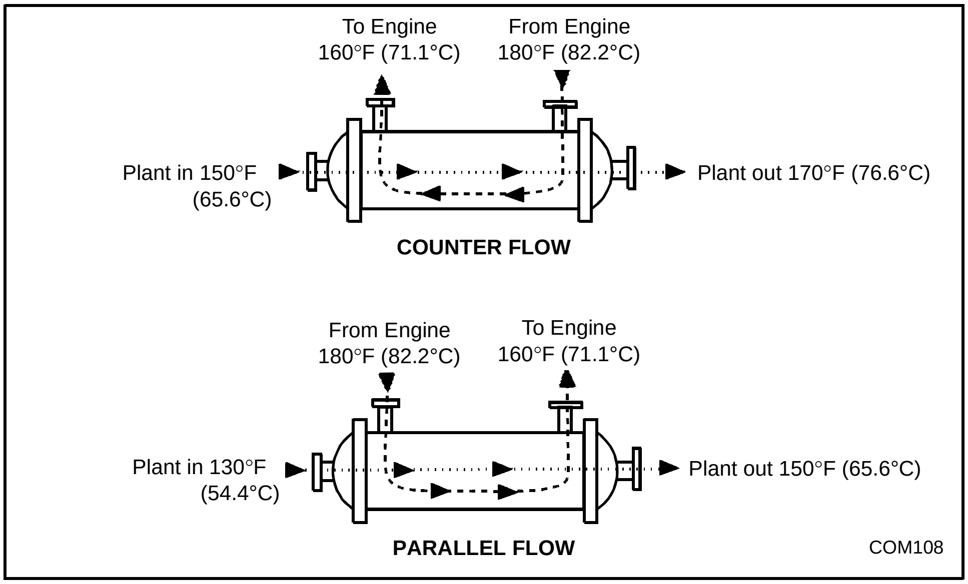


Figure 5-3 Counter and Parallel Flow Heat Exchanger

Counterflow also allows the plant outlet to have a higher temperature than the engine inlet temperature, which provides more useful, higher temperature plant heat.

The counterflow heat exchanger in Figure 5-3 has 180°F (82°C) coolant from the engine, while the plant outlet temperature is 170°F (77°C). This makes an approach temperature of: 180°F minus 170°F equals 10°F (82°C minus 77°C equals 5°C). (The approach temperature is the temperature difference between the coolant from the engine and plant outlet.)

A plate and frame type heat exchanger consists of a series of formed metal plates sandwiched together. Engine coolant is on one side of the plate, while plant coolant is on the other, with counterflow in the plates (see Figure 5-4).

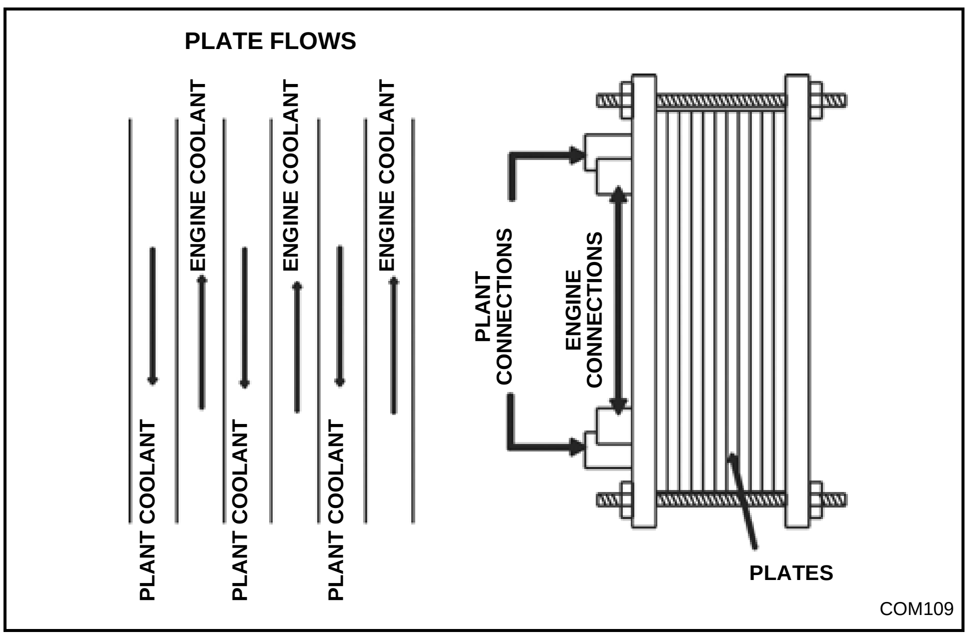


Figure 5-4 Plate-Type Heat Exchanger

A plate and frame type heat exchanger is generally similar in price to a shell and tube type. It is more compact and can have closer approach temperatures. Plate and frame type heat exchangers can have approach temperatures as close as 2°F (1°C).

Cooling Tower

Cooling towers rely on the evaporation of water in a dry climate to remove heat from the engine cooling water. For water to evaporate it must absorb energy which cools the liquid water that is left behind.

Open Cooling Tower: An open cooling tower sprays warm water into the air and collects the remaining cooled water in a pan at the bottom. Because of the energy release caused by evaporation, it is possible to cool the water to a colder temperature than the air it passes through. The evaporated water must be constantly replaced with fresh water resulting in water consumption for the cooling process.

Open cooling towers can often provide water at 85°F (29°C) or less in a sufficiently dry climate. Maintenance of these systems can be very costly. Operating at low temperatures promotes bacterial growth in the cooling system components. The constant introduction of make-up water brings in minerals and contaminants which do not evaporate. These minerals and contaminants can grow to very high concentrations, causing scaling and corrosion in the cooling system. A heat exchanger must be used to isolate a jacket water circuit from a cooling tower circuit to prevent fouling of the engine circuit. A heat exchanger is strongly recommended for the engine auxiliary circuit. An auxiliary circuit which directly uses cooling tower water will require filtering and frequent cleaning and descaling of the cooling system. This cleaning may be required as often as weekly depending on contamination and bacterial growth rate.

Cooling System Component Functions

Water Pumps

Engine water pumps are used to force coolant to flow through the engine, piping, thermostats, and cooling device. These pumps are of the centrifugal type. Flow from centrifugal type pumps is determined by the pump rotating speed and the amount of restriction in the circuit. Increasing the circuit restriction results in a decrease in water pump output. Increasing the water pump speed increases water pump output pressure and flow. The water pump curves in Figure 5-5 illustrate this.

NOTE: *This example uses English units. Calculations using metric units are preformed in the same manner. See Waukesha's Tech Data for data in both English and metric units.*

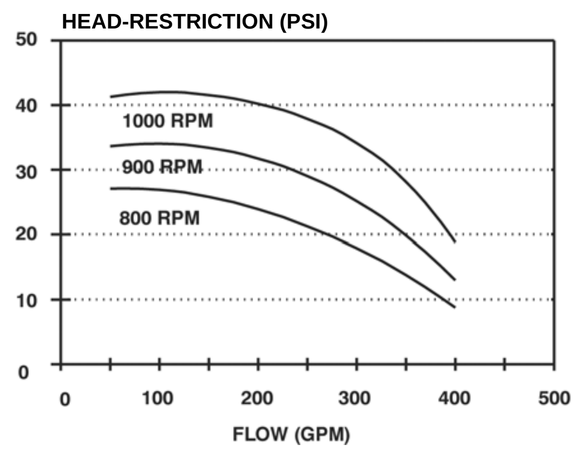


Figure 5-5 Water Pump Output

Components in the cooling circuit will restrict flow of the coolant due to friction. The output of the pump overcomes this friction to allow coolant flow. The engine makes up one portion of the system restriction, while the remaining restriction is from piping and the radiator or heat exchanger. Figure 5-6 shows the engine restriction on the same graph as the water pump output.

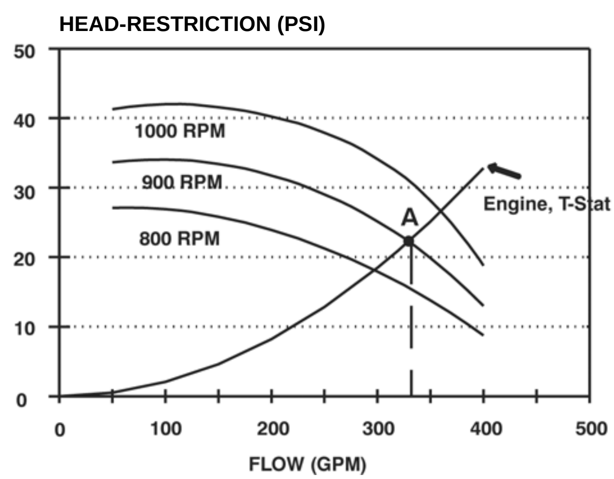


Figure 5-6 Restriction and Water Pump Outlet

Point “A” is where the engine restriction matches the water pump output at 900 RPM. If the engine outlet were connected directly to the engine inlet, the circuit restriction would equal engine restriction. Flow at 900 RPM would be 320 GPM with pump pressure rise and system restriction of 22 psi. The shape of the engine restriction curve is based on turbulent flow. The restriction is proportional to the square of the flow rate. Knowing this, we can calculate restriction for any point off the curve by knowing restriction for a single point on the curve. “Equation 1:” on page 5-7 can be used for any component which has turbulent flow.

Equation 1:

$R_2 = R_1 \times (GPM_2)^2/(GPM_1)^2$

Most cooling systems have additional restriction from the piping and radiator or heat exchanger. The restriction of these components must not reduce cooling circuit flow below the minimum required for the engine. Figure 5-7 illustrates the water pump curves, engine restriction, and now a maximum restriction curve. Total system restriction – including the engine restriction – must not exceed this curve or the coolant flow will be insufficient.

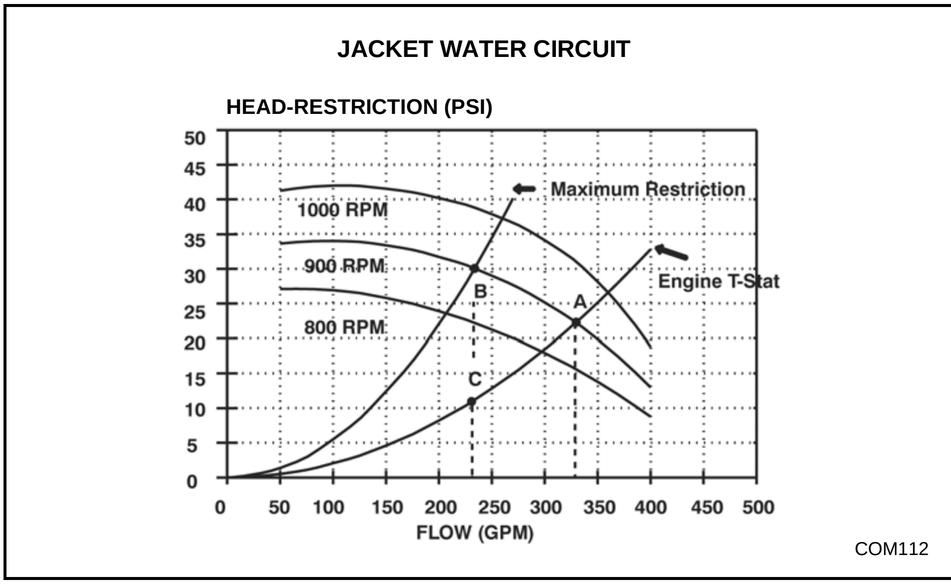


Figure 5-7 Water Pump Curve with Maximum Restriction

Point "B" on the curve represents the maximum restriction for 900 RPM. At this point, the flow rate is 230 GPM and the total restriction is 30 psi. At 230 GPM, the engine restriction is 12 psi, identified at point "C". From this, we can determine the maximum restriction for piping and the radiator or heat exchanger.

Total Restriction - Engine Restriction = External Restriction

30 psi – 12 psi = 18 psi

Radiator and piping restriction (external restriction) at 230 GPM must be 18 psi or less for proper coolant flow.

With some water pumps, the engine restriction is low and can permit coolant flow rates to become excessive. In these situations, a minimum required restriction curve is added. Figure 5-8 illustrates this curve. Point "D" on the curve is the intersection of the minimum required restriction and the 900 RPM pump output curve. This point is at 283 GPM with 27 psi total restriction. The engine restriction at 283 GPM is 17 psi (point "E"). The minimum required external restriction from the radiator/heat exchanger and piping is:

27 psi – 17 psi = 10 psi at 283 GPM

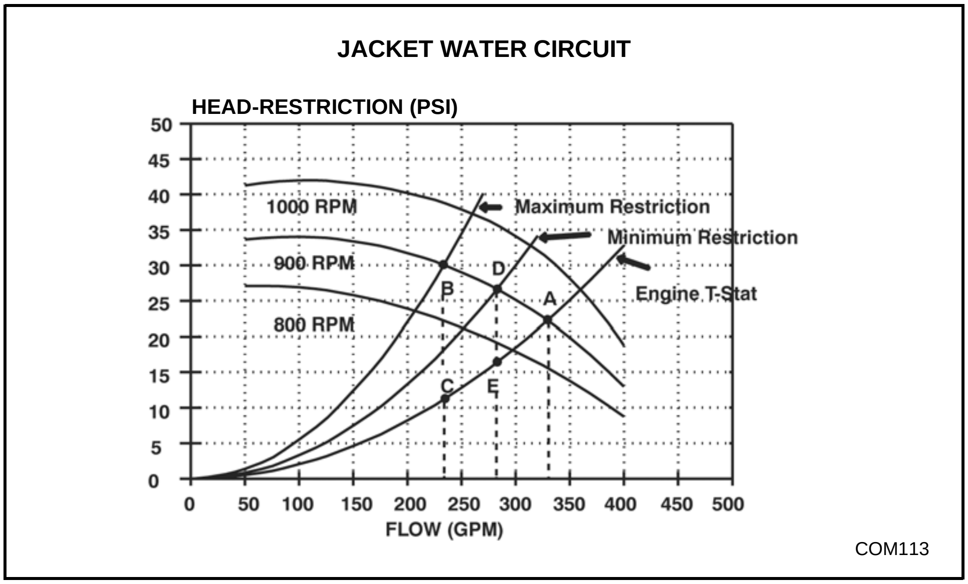


Figure 5-8 Water Pump Curve with Minimum Restriction

For a cooling circuit to be acceptable at 900 RPM for these conditions, the restriction from the external components (radiator/heat exchanger and piping) must be less than 18 psi at 230 GPM and more than 10 psi at 283 GPM.

The cooling system must be designed to accommodate the design flow and external restriction at the engines rated load and speed. If a reduced flow through a radiator is used, a flow bypass pipe will be required across the radiator to make up the remainder of the design flow. It may also be necessary to include an orifice or throttling valve to meet the design flow external restriction limits (see Figure 5-9).

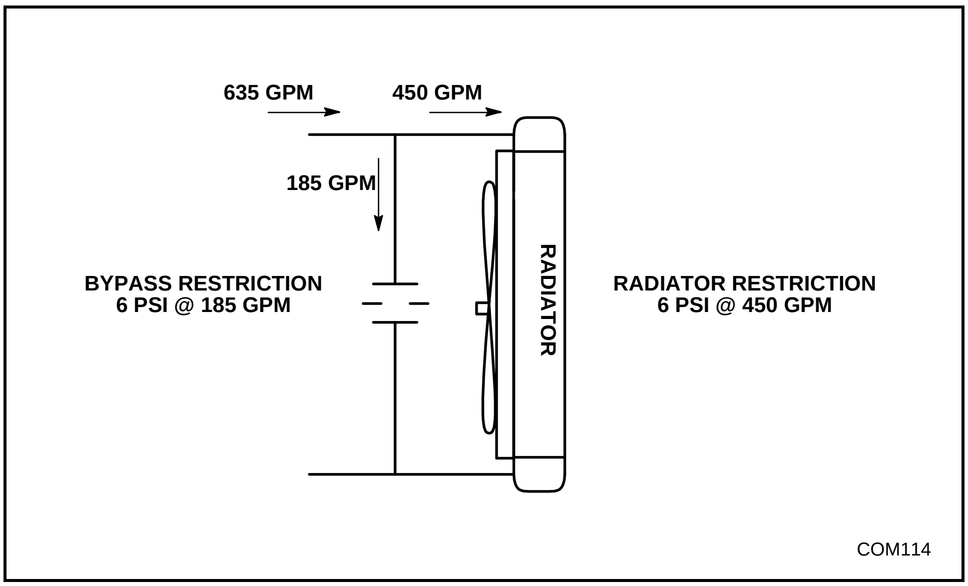


Figure 5-9 Cooling System with Bypass Pipe

Piping

Cooling system piping must be sized to allow the coolant to flow without excessive restriction. The piping material must be suitable for the temperatures and pressures encountered, as well as vibration from the operating engine. Flexible connections (bellows type or rubber hose) are recommended at all connection points to the engine. This will isolate the engine and piping components from high stresses due to vibration. Engines mounted on spring isolators or other soft mounting systems must have cooling system connections with flexibility sufficient to handle the motion normally encountered.

Dresser and Flexmaster couplings have the ability to join pipes which are not closely aligned. These couplings flex to join the pipes. However, they become very stiff when clamped in place. Waukesha does not consider these as flexible couplings for isolating components from vibration.

Piping restriction depends on the pipe diameter, pipe length, number of elbows and transitions, and the piping material used. The following procedure will help determine piping restriction. Use this procedure for a single size of pipe. If more than one pipe size is used, repeat the procedure for each pipe size and include restriction for transitions.

1. Calculate coolant velocity (V) in pipe.

   Equation 2:

$$V(FPM) = \frac{Flow(ft^3/min)}{Pipe\ inside\ area\ (ft^2)} \quad or \quad V(m/sec) = \frac{1000 \times Flow(L/sec)}{Pipe\ inside\ area\ (mm^2)}$$

   Flow(ft^3/min) = Flow (GPM) * 0.1247 (ft^3/gallon) or

   Flow(L/sec) = Flow $\frac{(m^3/hr)}{3.6}$

   Pipe Area(ft^2) = {Diameter (inch)}^{2} * π/4 * 0.00694 ft^2/$inch^2$

   Pipe Area(mm^2) = π/4 * {Diameter (mm)}^{2}

Table 5-1 Pipe Areas for Standard Pipe

PIPE SIZE (in.)	ID (in.)	ID (mm)	AREA ($inch^2$)	AREA (mm^2)	AREA (ft^2)
1.5	1.61	40.894	2.04	1312.77	0.0142
2	2.067	52.502	3.36	2163.80	0.0233
2.5	2.344	59.538	4.32	2782.61	0.030
3	3.068	77.927	7.39	4767.03	0.0513
4	4.026	102.260	12.73	8208.89	0.0884
5	5.047	128.194	20.01	12900.42	0.139
6	6.065	154.051	28.89	18629.39	0.201
8	7.981	202.717	50.03	32259.06	0.347

2. Determine pressure loss (P_L) per 100 ft or meter of pipe for the velocity and pipe size from Figure 5-10.

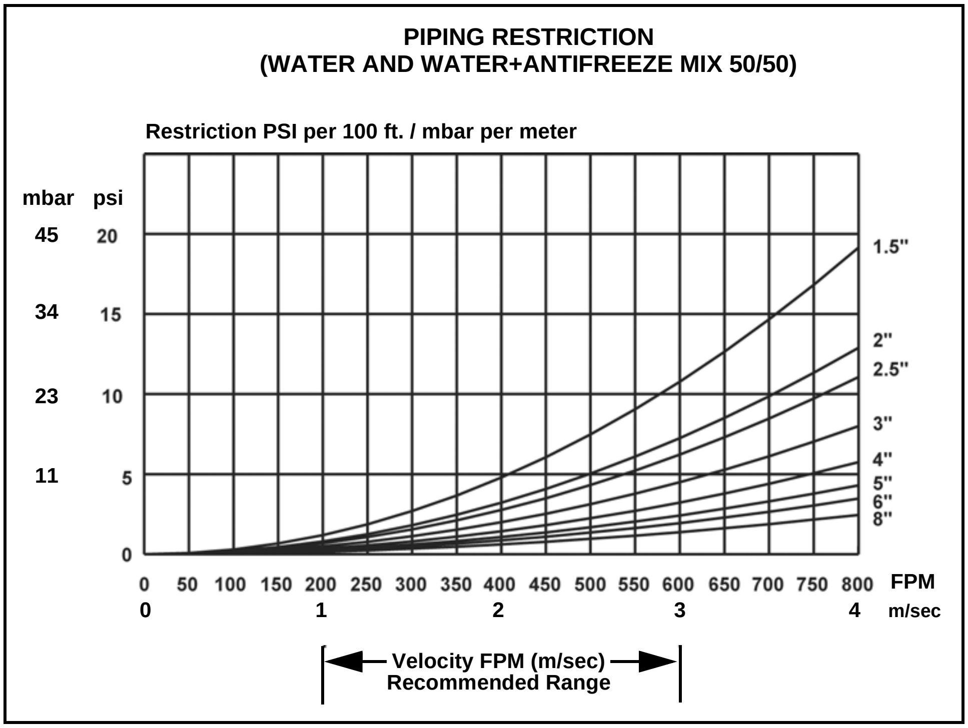


Figure 5-10 Piping Restriction Chart

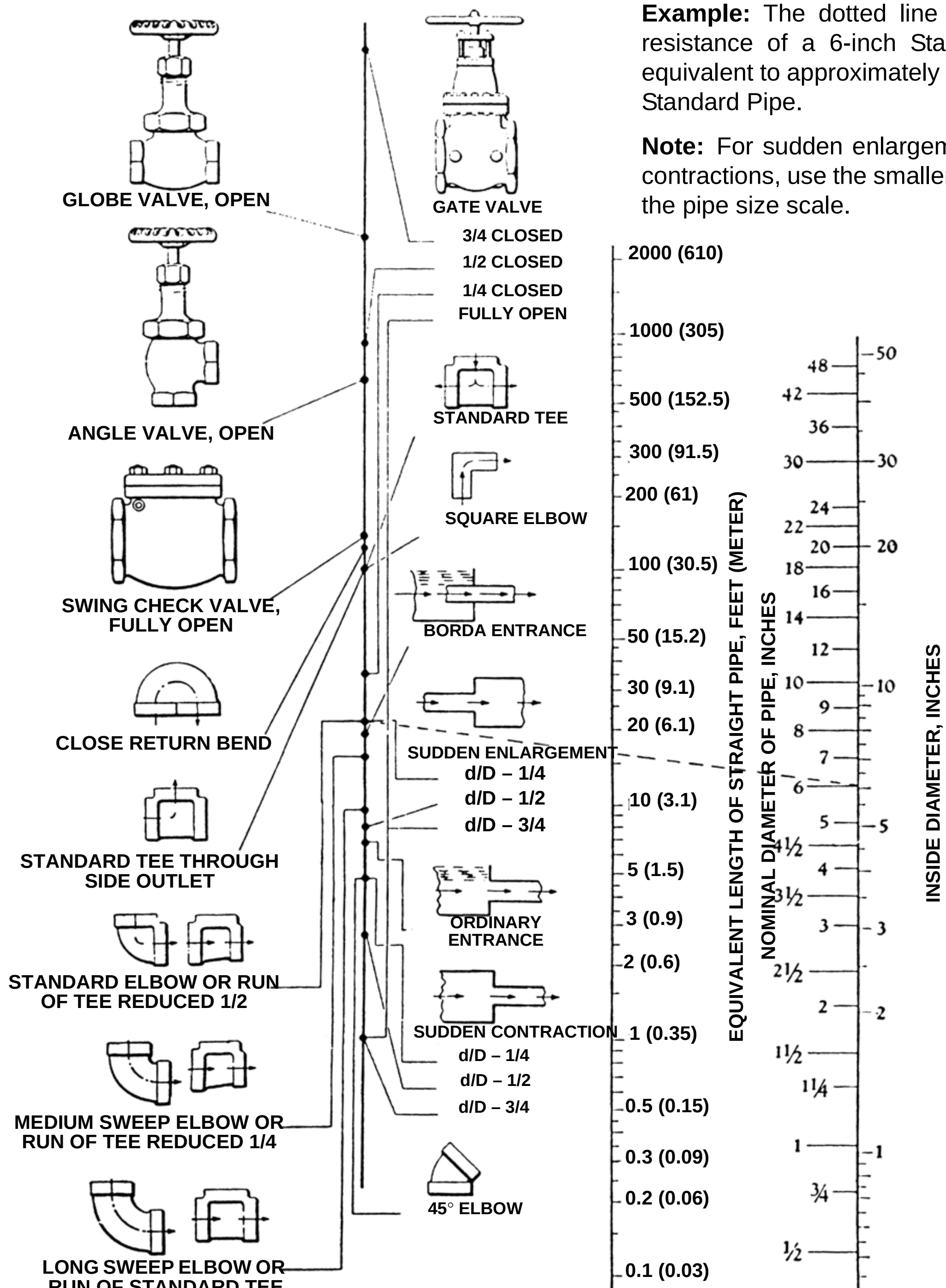
3. Determine the equivalent pipe length (EPL) for all fittings:

   Figure 5-11 gives equivalent pipe length in feet or meters for various pipe fittings. Sum the EPLs and add them to the total length of straight pipe to find the total EPL.

RESISTANCE OF VALVES AND FITTINGS TO FLOW OF FLUIDS IN EQUIVALENT LENGTH OF PIPE

Example: The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16 feet of 6-inch Standard Pipe.

Note: For sudden enlargements or sudden contractions, use the smaller diameter, **d**, on the pipe size scale.



From Crane Co. Technical Paper No. 409. Data based on the above chart are satisfactory for most applications. REPRINTED WITH PERMISSION OF CRANE VALVE GROUP.

Figure 5-11 Equivalent Pipe Length of Fittings in Feet (Meters)

4. Calculate the total piping restriction (R_P):

   Equation 3:

$$R_P = \frac{P_L\ (\text{psi})}{100\ \text{ft}} \times \text{EPL (ft)} \text{ or } \frac{P_L\ (\text{mbar})}{\text{m}} \text{ x EPL (m)}$$

5. Calculate the total cooling circuit restriction:

   Equation 4:

   $R_T = R_P + R_E + R_R$

   Where

   R_T = total restriction (psi)

   R_P = piping restriction (psi)

   R_E = engine restriction (psi)

   R_R = radiator/heat exchanger restriction (psi)

Sample Problem:

An engine which will operate at 900 RPM has pump performance and engine restriction as in Figure 5-8. The radiator chosen has 12 psi restriction at 300 GPM. The contractor intends to use 30 ft of 4 in. flanged steel pipe with 8 regular 90° elbows.

1. Using 230 GPM as a targeted flow rate, determine:
   A. Engine restriction
   B. Piping restriction
   C. Radiator restriction
   D. Total system restriction
2. Find pump pressure rise. Compare to total system restriction R_T. Does R_T exceed the pump pressure rise?
3. Determine the maximum flow rate. Estimate total system restriction at the maximum flow rate. Is flow rate too high?

Solution:

1.
   A. R_E = 11 psi
      (*See Figure 5-8 on page 5-8*.)

   B.
      1) Flow (ft^3/min) = 230 GPM * 0.1247 ft^3/min/GPM = 28.7 ft^3/min
         Pipe Area (ft^2) = 0.0884 ft^2

$$\text{Velocity} = \frac{28.7\ \text{ft}^3/\text{min}}{0.0884\ \text{ft}^2} = 325\ \text{ft/min}$$

(*See “Equation 2:” on page 5-9.*)

2) P_L per 100 ft of pipe = 1 psi
(*See Figure 5-10 on 5-10.*)

3) EPL = 30 ft + (8 elbows * 11.0 ft/EPL = 118.0 ft
(*See Figure 5-11 on page 5-11*.)

4) $R_P = \frac{1 \text{ psi}}{100 \text{ ft}} \times 118.0 \text{ ft} = 1.18 \text{ psi}$

(*See "Equation 3:" on page 5-12.*)

C. $R_R \text{ @ } 230 \text{ GPM } = 12 \text{ psi} \times \frac{(230 \text{ GPM})^2}{(300 \text{ GPM})^2}$

(*See "Equation 1:" on page 5-7.*)

$R_R \text{ @ } 230 \text{ GPM } = 7 \text{ psi}$

D. $R_T = R_P + R_E + R_R = 1.18 + 11 + 7 = 19.2$ psi
(*See "Equation 4:" on page 5-12.*)

2. Pump pressure rise = 30 psi @ 230 GPM, 900 RPM
(*See Figure 5-8 on page 5-8*.)

Pump pressure capacity is greater than the system restriction. Therefore, system restriction is not too high.

3. Maximum flow rate = 280 GPM @ 900 RPM
Pump pressure rise = 27 psi @ 280 GPM

System restriction $R_T = 19.2 \text{ psi} \times \frac{(280 \text{ GPM})^2}{(230 \text{ GPM})^2} = 28.5 \text{ psi}$

(*See "Equation 1:" on page 5-7.*)

The system restriction is greater than the minimum restriction required. Therefore, the components are properly sized.

NOTICE

Use the correct water pump curve for the specific engine model from the "General Technical Data Manual" for water pump performance and restriction information. Do not use the water pump curve in this book. It is an example only and not model specific. Radiator restriction information should be available from the radiator manufacturer.

Expansion Tanks and Surge Tanks

Expansion tanks and surge tanks perform several functions in a cooling circuit. Some of these functions are:

- De-aerate coolant
- Control cooling system pressure
- Allow coolant expansion
- Provide coolant reserve

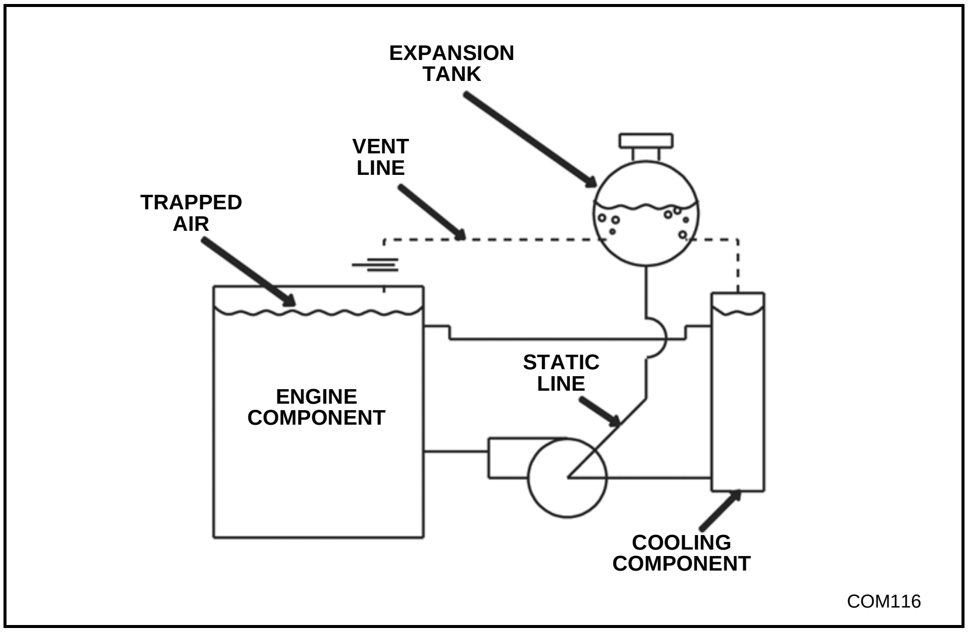


Figure 5-12 Schematic of Cooling Circuit

An expansion tank is a single chamber tank located at the highest point in the cooling system. Vent lines are connected from high points in the cooling system to the expansion tank below the water line. These vent lines allow trapped air to escape to the expansion tank where the air bubbles out of solution, thus de-aerating the coolant (see Figure 5-12).

Vent lines should be 1/4 in. in diameter on systems with vent lines less than 10 feet (3 m) long, or 1/2 in. diameter with a 1/4 in. orifice on systems with vent lines more than 10 ft (3 m) long.

All vent lines must have flex connections, or other provisions, to prevent stress on the lines due to engine vibration. The vent lines must also be properly supported so their weight is not being supported by the flexible connection. Failure to properly relieve these stresses may result in a broken vent line which could cause a glycol fire.

A static line from the bottom of the expansion tank to the water pump inlet controls the pressure there. It is easiest to understand how this pressure is controlled if we ignore the vent lines. Without vent lines, the expansion tank appears as a water tower connected to the water pump (see Figure 5-12).

Pressure at the water pump inlet is equal to the expansion tank pressure, plus the static pressure of the water column from the water pump to the water surface in the expansion tank. If the pressure at the pump inlet were higher, coolant would flow up the static line and raise the coolant level in the expansion tank until pressure equilibrium is reached. If the pump inlet pressure were lower, coolant would flow down the static line until equilibrium is reached. Coolant pressure at points other than the pump inlet depends on pump pressure rise and pressure drop across the various components.

If the static line were incorrectly placed at the water pump outlet, the pressure at the pump outlet would equal the expansion tank pressure plus the static head. The resulting pressure at the pump inlet on a running pump would be very low, possibly even a negative gauge pressure. Operation with low pump inlet pressure causes localized steam bubble formation and collapse, resulting in cavitation erosion of the water pump impeller and a decrease in pump flow. Pump damage and engine overheating will result.

The static line is sized much larger than the vent lines to minimize flow velocity and pressure drop. The static line is typically 1 in. diameter or larger for systems greater than 400 GPM (90m^3/hr), and 3/4 in. diameter or larger for systems less than 400 GPM (90m^3/hr).

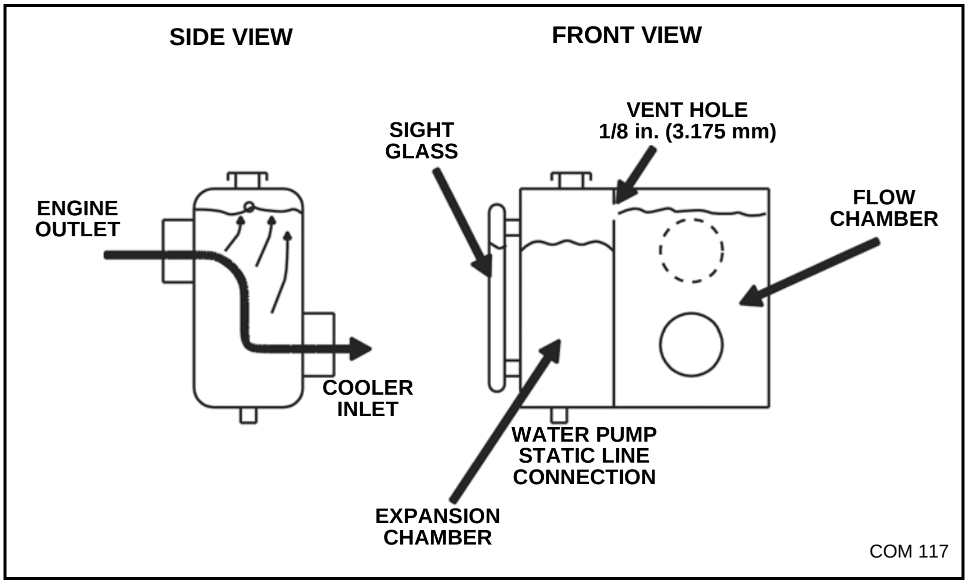


Figure 5-13 Surge Tank

A surge tank performs the same function as an expansion tank, but adds a full-flow chamber. In this chamber, the coolant velocity is reduced, allowing air trapped in the solution to escape to the high point in the chamber. A bleed hole between the full-flow chamber and expansion chamber allows air to escape to the expansion chamber. This bleed hole is small to limit coolant flow down the static line, which connects to the water pump inlet. The full-flow chamber and bleed hole are performing the same function as the vent lines and the 1/4 in. orifice used on the expansion tank system (see Figure 5-13).

A sight glass in an expansion tank or the expansion chamber of a surge tank allows visual monitoring of coolant level. A circuit which is filled to the top of the expansion area will overflow coolant on start-up due to coolant expansion. After that, the coolant level should not overflow. Designing the filler neck to extend into the expansion tank will prevent overfilling the tank. A small vent hole in the filler neck will allow the coolant in the filler neck to drain into the tank after filling (see Figure 5-14).

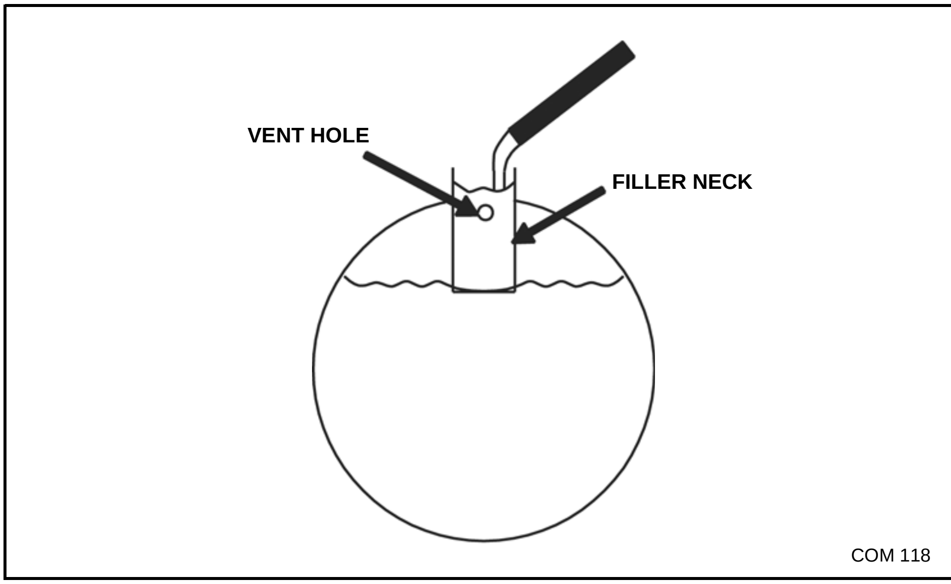


Figure 5-14 Vent Hole in Filler Neck

An expansion tank or expansion chamber of a surge tank should be sized for 6% expansion of the coolant. An additional 5% is recommended for coolant makeup. With these volumes, an expansion tank should be sized to contain 11% of the total cooling system volume.

NOTE: *Separate expansion tanks must be used for separate auxiliary and jacket cooling circuits.*

It is important to provide the correct pressure to the water pump inlet to prevent pump cavitation or over pressurization of the system. Minimum pump inlet pressure requirements are available in the “General Technical Data Manual” on latest editions of instruction sheets S9062 and S9063 for 275GL engines, and S7424-1 for all other Waukesha engines.

Maximum pressure to the water pump is published in the “General Technical Data Manual” Specifications section for the individual engine model. Contact the Application Engineering Department for auxiliary circuit maximum water pump inlet pressure if it is not available in this section.

A pressure cap is required to prevent coolant evaporation losses and to prevent boiling in the system. See latest edition of S6699-7 for pressure cap relief pressures. The pressure cap must have a vacuum relief function to prevent a vacuum from forming in the tank during load reduction or cool-down operation. Only a single pressure cap can be used in a cooling system. The cap must be at the highest point on the expansion tank or expansion chamber of the surge tank.

NOTICE

Do not assume a pressure cap will pressurize the tank to the rating of the cap. Pressure in the tank can range from atmospheric pressure to the pressure cap rating. The exact expansion tank pressure can be affected by many factors, including:

- **ambient temperature**
- **engine load**
- **expansion volume**

It is difficult to predict the exact tank pressure.

Bladder Style Pressurization System With Degassing Tank

As an alternative pressurized expansion tanks can be used for systems that require high pressure levels. These are closed systems that do not allow air entering the cooling system when the engine is cold and the coolant at lowest volume. A pressurized expansion tank has a bladder and uses compressed air or nitrogen to maintain the required pressure. This means that the pressure in the cooling system is not dependent on ambient conditions or engine running conditions but is set by the pressure in the bladder. Air and other gases are removed by a degassing tank with an automatic degasser. A pressurized expansion tank requires a relief valve to prevent excessive pressures in case the system is overfilled with coolant or air is introduced into the system.

Water Filtration

Debris in the coolant can block cooling passages, erode cylinder liner packing ring areas, wear out water pump seals, and cause several other types of damage. This debris may be from fabrication of the engine, cooler or piping. Cleaning the cooling passages during assembly or prior to start up will not remove all of the debris.

Bypass coolant filtration can remove debris from the cooling system on any engine. Bypass coolant filtration sized to remove 15 – 25 micron particles from 2% of the water flow is recommended for Waukesha engines. Figure 5-15 illustrates bypass water filtration systems.

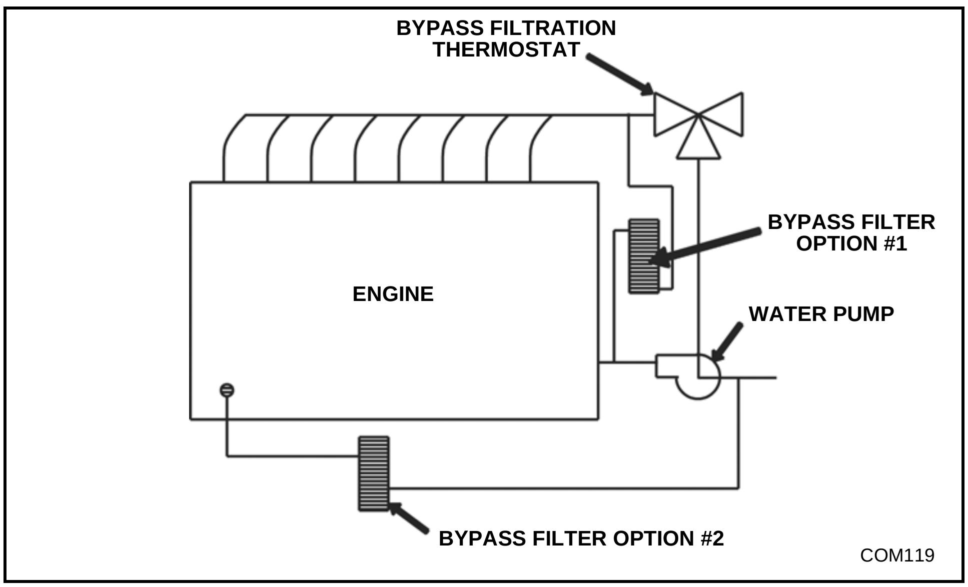


Figure 5-15 Schematic for Bypass Water Filtration Systems

Care must also be taken when welding external cooling system pipes together or when drilling and tapping a hole anywhere in the water system. Ensure that the weld slag and chips are completely cleaned from the cooling system before the engine is operated. A witch-hat filter will accomplish this.

Thermostats

A thermostat controls and maintains temperature by directing coolant in different branches of a cooling circuit. How a thermostat performs and what component temperature it will control depends on its position in a cooling circuit.

A thermostat has 3 ports often labeled “A”, “B”, and “C”, as shown in Figure 5-16. Port “A” receives the full-flow of the circuit and has the temperature sensing element. Port “B” receives all of the flow when coolant is well below the sensing element set point temperature. Port “C” receives the flow when the temperature is well above the sensing element set point. Because of these characteristics, the ports are typically oriented with Port “A” receiving “All” of the flow, Port “B” connected to the “Bypass line” and port “C” connected to the “Cooler”.

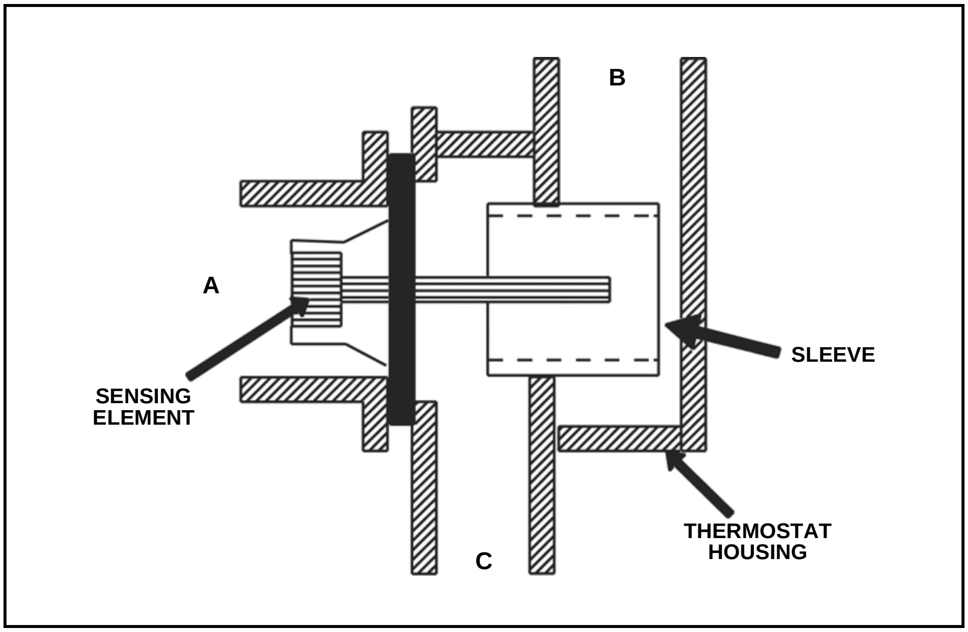


Figure 5-16 Thermostat

Two common thermostat orientations are blending and diverting (see Figure 5-17).

A blending thermostat blends cold coolant from the “C” port with warm coolant from the “B” port to provide the desired temperature at the “A” port. Accurate control of this system depends on proper mixing in the thermostat before the coolant passes over the sensing element. A blending thermostat is commonly used in the intercooler circuit of Waukesha engines to control coolant temperature at the “A” port, which controls intercooler inlet temperature.

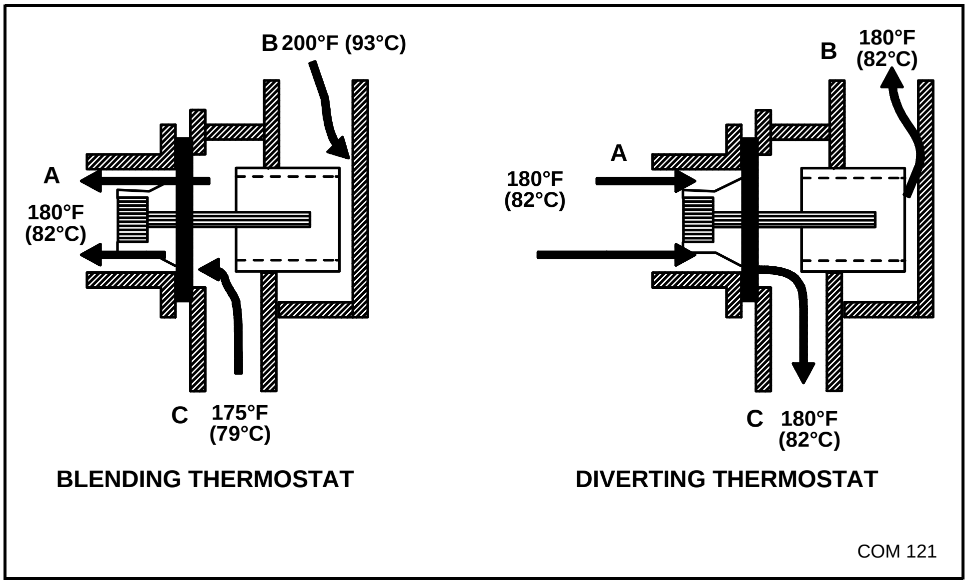


Figure 5-17 Blending and Diverting Thermostats

A diverting thermostat receives the coolant in the "A" port where the temperature is sensed. The valve then positions to send the appropriate amount of coolant to the cooler and through the bypass. With this position, the coolant is the same temperature at all three ports. Therefore, proper mixing is not a concern. Diverting thermostats are commonly used on the jacket water outlet to control engine outlet temperature.

Many thermostats use several elements to meet the flow requirements. Typically, one element has a lower set point than the rest, and acts as a "lead" element. This lead element minimizes thermal cycling, providing steady state temperature control.

Heat Recovery Systems

Heat rejection from an engine can be up to 70% of the total energy input. This energy is lost when an engine is utilized for its crankshaft power only. Heat recovery allows for capture of the majority of this otherwise lost energy and put it to use in place of boilers and other heating devices in a plant.

"Solid Water" heat recovery systems circulate coolant through the engine jacket and the exhaust heat recovery equipment to pick up the heat energy from the engine. Some engines like the APG 1000 use the Jacket Water to cool the (high temperature) first stage of the intercooler or Charge Air Cooler. This increases the amount of heat available at high temperature which is advantageous for CHP and reduces the need for cooling capacity (radiators) at low temperature. This energy is then piped to areas in the plant where it can be used. Several components typically found in a heat recovery system are:

- Engine water jacket
- Engine thermostat
- Engine water pump
- Exhaust heat exchanger (Heat recovery silencer)
- Customer heat load heat exchanger

- Excess heat dump radiator
- Excess heat dump thermostat
- System water pump
- Expansion tank

One Pump System

In the single pump system in Figure 5-18, the engine water pump is used to force coolant through the entire circuit. Engine thermostats and bypass are removed, and a warm-up thermostat and bypass is placed downstream of the exhaust heat exchanger. This assures flow through this device during warm up. (Without flow, the exhaust heat exchanger would be damaged by overheating or thermal shock.) It also allows quicker warm-up of the engine. Once the engine is warm, coolant starts to flow to the customer heat exchanger, where the heat produced in the engine and exhaust will be removed. If more heat is produced than removed by the customer heat exchanger, the coolant temperature will be high and signal the system thermostat (blending) to begin accepting coolant from the heat dump radiator. If the temperature continues to climb, a temperature switch is often used to signal the fan to start.

The warm-up thermostat is used to quickly warm up the engine. The system thermostat limits the circuit temperature. Therefore, the system thermostat is set higher than the warm-up thermostat, but below the maximum engine operating temperature. It is common to use a warm-up thermostat with a nominal opening temperature 20°F (11°C) below the system thermostat. Closer temperatures can cause some heat to be lost unnecessarily to the excess heat dump device. A wider temperature spread may cause overheating damage to the warm-up thermostat.

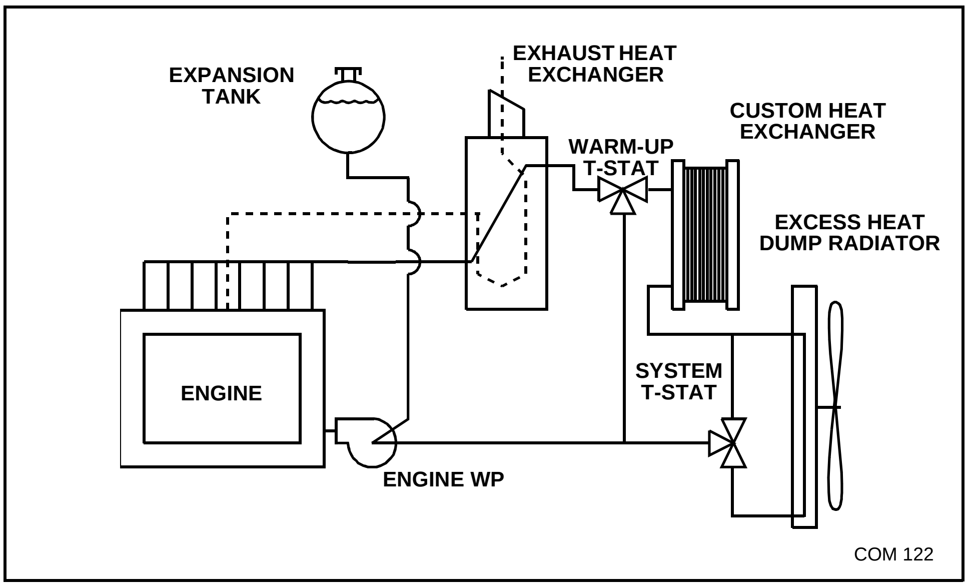


Figure 5-18 Single Pump Heat Recovery System
(vent lines not pictured)

Two Pump System

The two pump system, illustrated in Figure 5-19, is used in circuits where the engine water pump has insufficient capacity to flow coolant through the engine and heat recovery components. The system pump is sized to deliver the required flow through all cooling components other than the engine.

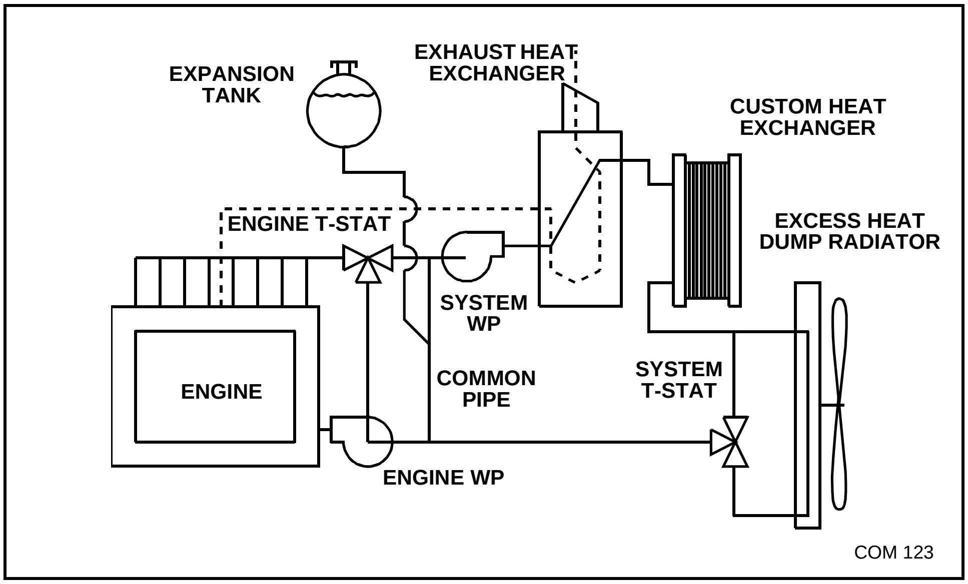


Figure 5-19 Two Pump Heat Recovery System
(vent lines not pictured)

With the two pump system, the engine water pump needs to overcome restriction of the engine, the thermostat, and the common pipe.

During warm up, the engine thermostat is in bypass position, with the engine water pump circulating coolant through the engine and engine thermostat bypass only. The system water pump is also operating, providing coolant to the exhaust heat exchanger. The common pipe is flowing the full system pump output.

NOTE: *The common pipe must be equal in size to other circuit piping.*

When the engine reaches operating temperature, the thermostat begins to open and discharges warm coolant to the system while accepting cold coolant from the system. The common pipe carries system flow, minus the amount of flow discharged from the engine.

The expansion tank static line ties into the common pipe between the two water pumps. If the common pipe is of sufficient diameter and relatively short, the static line will properly control pressure to both water pumps.

The engine thermostat functions as a warm-up thermostat and quickly warms up the engine. Therefore, the system thermostat is set higher than the warm-up thermostat, but below the maximum engine operating temperature. It is common to use a warm-up thermostat with a nominal opening temperature 20°F (11°C) below the system thermostat. Closer temperatures can cause some heat to be lost unnecessarily to the excess heat dump device. A wider temperature spread may cause over heating damage to the warm-up thermostat.

The systems shown here are two common systems used for heat recovery. Other systems are possible. It is important that a system be designed to provide the proper engine temperature and flow rate. Rapid fluctuations in coolant temperature must be avoided to prevent thermal shock. The system must use treated coolant with a closed loop and a properly positioned expansion tank.

Isolated Circuit System

The Isolated Circuit System, illustrated in Figure 5-20, is the preferred method for heat recovery applications. The engine cooling circuit is isolated from the process loop, which helps to minimize the effect to the engine cooling circuit from disruptions which may occur in the heat recovery process loop. This approach allows the engine cooling circuit to remain very similar to standard engine configurations, which can simplify the design and ensure the cooling system parameters (restriction, flows, temperature, etc.) remain within the specified limits. In this type of system, the process loop requires its own circulating pump.

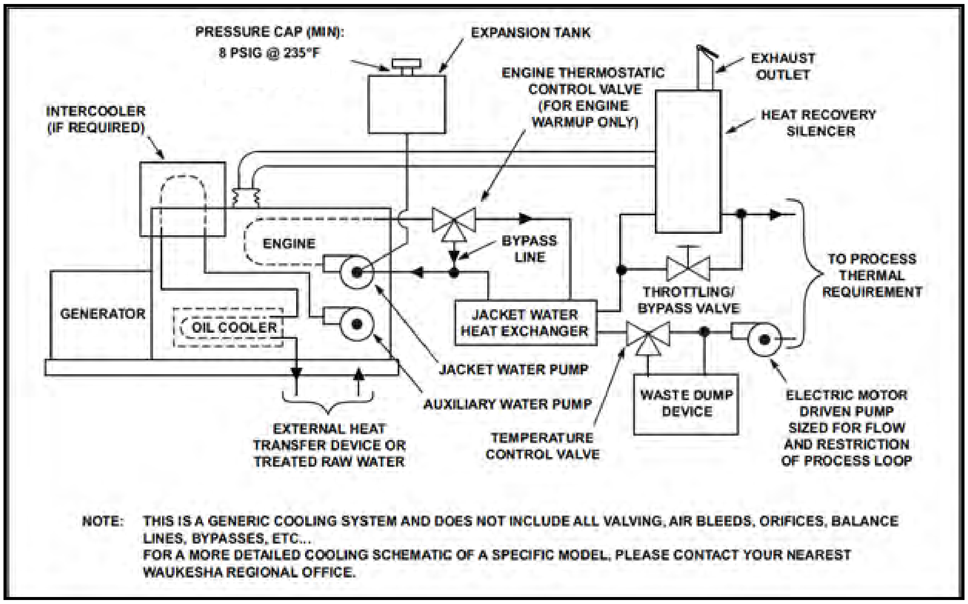


Figure 5-20 Isolated Circuit Heat Recovery System

Coolant Pressure and Treatment

The cooling system is an essential but often overlooked aspect of the engine. The primary purpose of any coolant treatment program is to protect the surfaces of all water passages from corrosion, scaling, or sludge deposits which may impede the heat transfer to or from the coolant. If the system is exposed to low ambient temperatures, antifreeze protection is needed. In addition, cavitation erosion protection is a consideration for engine cooling systems.

Low pressure or poor treatment of the coolant may lead to scaling or fouling of coolant passages, cavitation, overheating, and corrosion which may result in the failure of the engine. See latest edition of S6699-7 for more information regarding coolant pressure and treatment.

Always consult your local environmental legislation concerning coolant and coolant disposal.

SECTION 2

INSTALLATION CONCERNS

Radiator Cooled Units

Performance of a radiator can be affected by many external conditions. When determining radiator location and size the following effects must be considered:

- Wind direction

  A radiator fan pushing against the wind will limit air flow through the radiator and decrease its effective heat rejection capacity.

- Recirculation

  Hot radiator air from the outlet or from radiators of other units located nearby can cause air preheating to the inlet of the radiator, resulting in a decreased effective heat rejection capacity. Engine exhaust recirculation through the radiator must also be considered.

- Reflective Surfaces

  Any surface such as gravel, stones, asphalt, concrete, etc., will reflect/radiate heat resulting in increased temperature into a radiator.

- Enclosures

  When a radiator fan is also used to ventilate an enclosure, restriction beyond the radiator (suction and discharge sides) and its effect on air flow must be considered. Also, preheating of the air from engine and driven machine radiation must be considered for pusher fans.

- Altitude

  Air at high altitudes is less dense than at sea level. This lower density air has less capacity to transfer heat. Therefore, a higher air flow rate and/or larger radiator core may be necessary.

- Coolant

  A radiator sized to cool with water only will have decreased capacity when using water and antifreeze mixes. Coolant mix must be specified when sizing the radiator.

Engine Preheating

In cold weather, lubricating oil viscosity increases considerably. This can make it difficult to crank an engine and cause insufficient flow to lubricated components. Cold weather can also make combustion in the cylinders difficult and prevent starting.

A cold intercooler will allow high mass flow of air, while fuel flow may be normal. This results in a very lean mixture which can prevent starting.

Starting an engine in cold conditions may require preheating of cooling and lubrication circuits. Waukesha requires jacket water and lube oil preheating for starting in temperatures below 50°F (10°C). Heaters should be sized to maintain 70°F (21°C) in these conditions. The intercooler will often be sufficiently warm from the room and conduction from a pre-heated jacket water. Intercooler heating may be necessary if the intercooler temperature falls below 50°F (10°C). Once started, the engine should be allowed to warm up under a light load until water and oil temperatures exceed 100°F (38°C).

Emergency standby engines which are required to start and accept load immediately must be preheated to 100°F (38°C) to 120°F (49°C).

Engine Thermal Shock at Shutdown Due to Thermosiphoning

Thermosiphoning is a process where coolant will circulate in a cooling loop without any assistance from a water pump. As coolant is heated, its density decreases, causing it to rise. As the coolant is cooled, it drops below warmer coolant. These actions create flow in a circuit. A greater difference between engine coolant temperature and radiator coolant temperature will cause a greater flow. A radiator at a higher elevation than the engine will have a greater thermosiphoning flow than one mounted in front of the engine.

When an engine is operating in a cold ambient the thermostat will split the flow be tween the cooler and bypass lines to maintain the proper coolant temperature in the engine (see Figure 5-21).

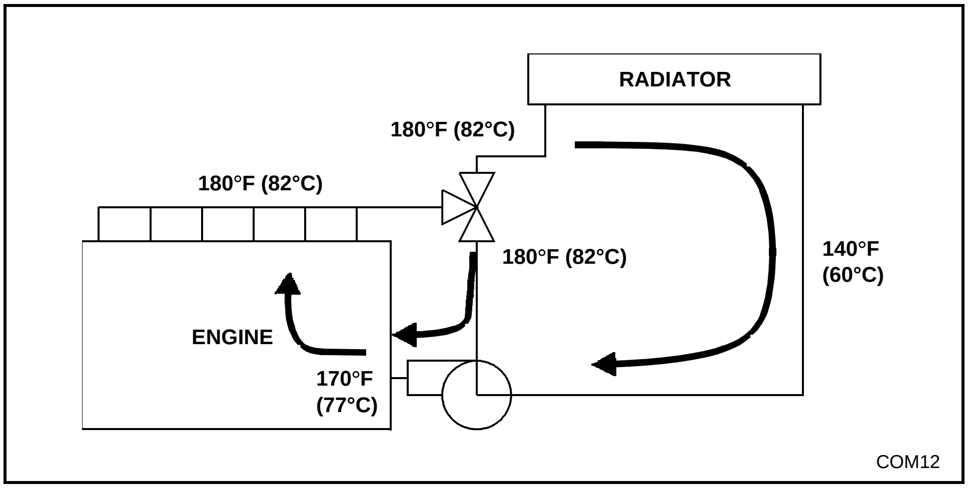


Figure 5-21 Thermostat Partially Open

When the engine is shut down, the coolant circulation stops because the water pump stops circulating. As the coolant in the engine block absorbs heat from the castings, it tends to rise. The thermostat senses warmer coolant and begins to open fully to the radiator and block the bypass. This allows flow to the cooler only. Meanwhile, coolant in the radiator will tend to cool considerably due to the low flow rate. This very cold coolant then slowly circulates back to the engine inlet (see Figure 5-22).

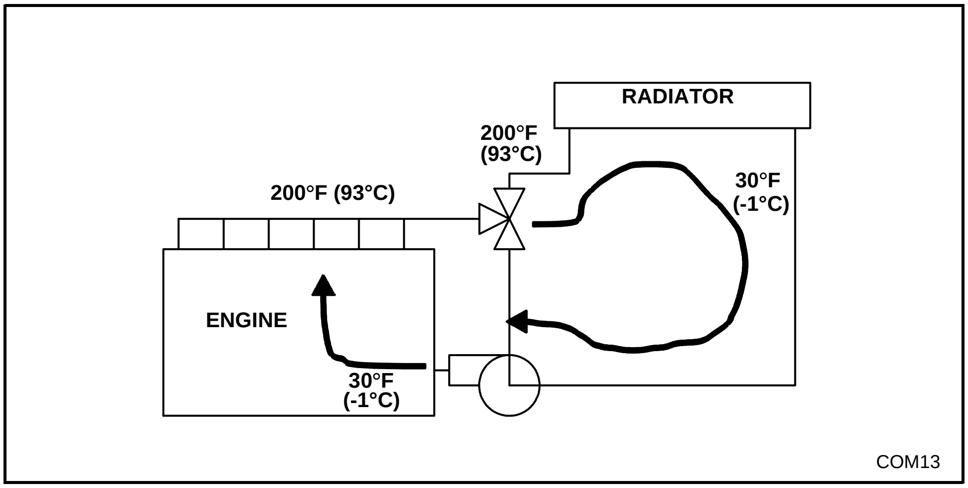


Figure 5-22 Thermostat Fully Open

When this coolant reaches the hot castings, it causes thermal shock. The coolant heats up again as it passes through the engine, but will remain cool enough to cause the thermostat to close its path to the radiator, and open the bypass, briefly stopping the flow (see Figure 5-23). The coolant will again begin to heat up and the process will repeat until an equilibrium is reached.

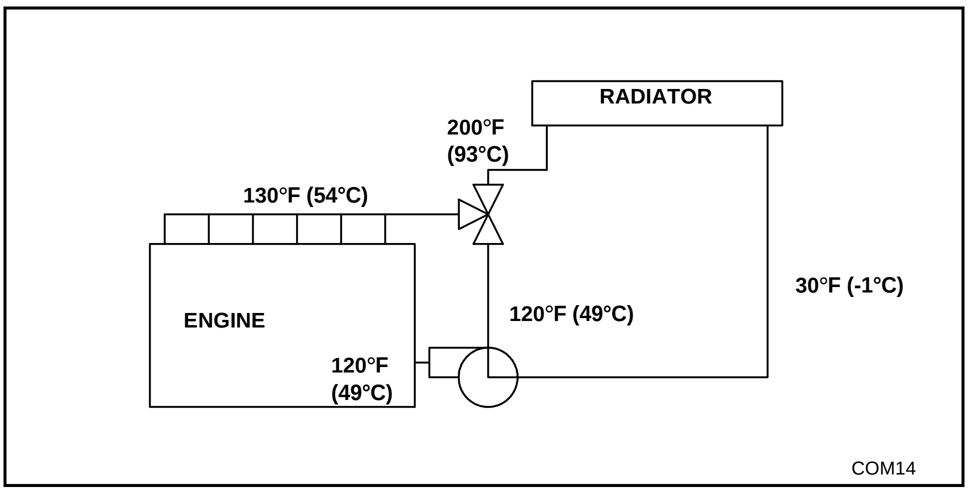


Figure 5-23 Thermostat Fully Closed

An engine subjected to frequent shutdowns (several times a week or daily) in ambients below 40°F (4.4°C) can have numerous thermal shock occurrences. If shocks are severe enough, they may crack cylinder heads and/or other castings.

4. Preventing thermal shock can be accomplished by the method listed below (commercial methods are also available).

NOTICE **Restarting shortly after shutdown should be avoided. Restarting can cause a cold slug of coolant from the radiator to enter the engine because the thermostat may still be fully opened.**

1. Install a post cooling pump as shown in Figure 5-24, and operate it for several minutes after shutdown. This post cooling pump prevents heat soak from fully

opening the thermostat. This allows warm bypass water to blend with the cooler return water which slowly cools the engine and closes the thermostat.

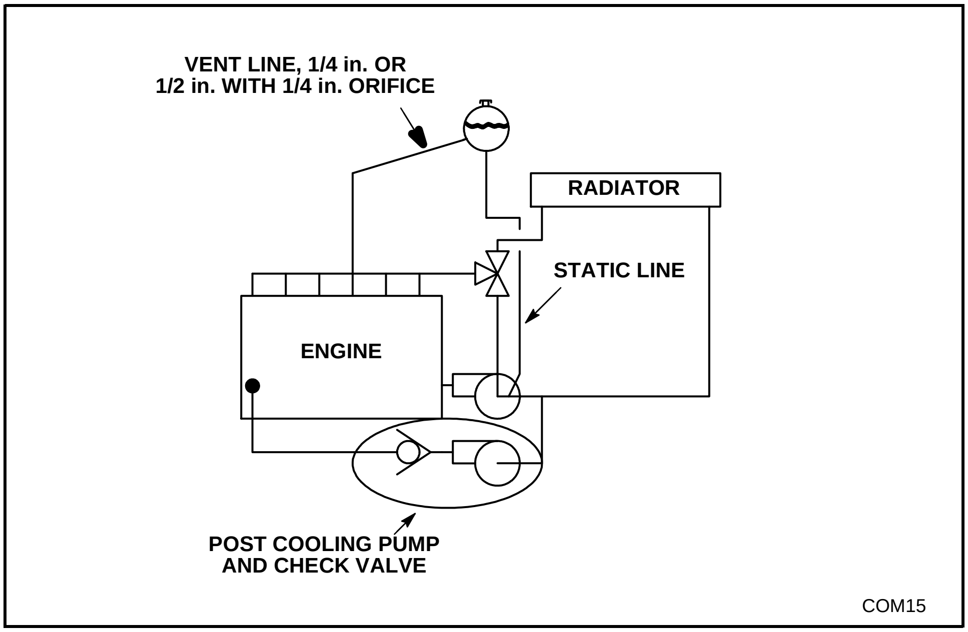


Figure 5-24 Post Cooling Pump

2. For vertical core radiators located above the engine, design the piping with the hot (inlet) radiator connections at the bottom and cold (outlet) connections at the top.
3. Locate the radiator at the same or lower elevation than the engine. This will reduce or prevent thermosiphoning from occurring.
4. Install an automatic or manual valve in the return line from the cooler which is closed between 15 seconds and 1 minute after the engine is shutdown and opened immediately before starting. This allows some heat to be carried out of the engine immediately after shutdown to prevent steam flashing in the higher passages of the engine. All flow is then stopped after the valve closes.

Vent lines from the top water manifolds of an engine to the expansion tank are always recommended for removing trapped air and steam (see Figure 5-24). Trapped steam can be a problem on engines, especially using methods 2, 3, or 4 to prevent thermosiphoning. Failure to bleed off steam will cause a steam pocket to form, forcing the pressure cap to relieve and dump coolant, and drying out the exhaust manifolds and cylinder heads. This may cause them to crack.

Steam can be prevented from forming by allowing the engine to operate unloaded for at least 2 minutes before shutting down. This removes much of the heat which otherwise would soak the coolant after shutdown. A pressure cap rated at a pressure sufficiently higher than the operating temperature's vapor pressure will also help prevent steam from forming.

Jacket Water Return Temperature

Cold cooling water return temperatures at the jacket water circuit inlet may cause engine damage. A sudden change in temperature could cause thermal shock for a hot operating engine or one that has recently been shut down. The suggested minimum return temperature into a hot engine is 150°F (65°C) with a maximum return temperature change of 18°F (10°C) per minute while between the minimum and maximum operating temperatures; temperature oscillations should be kept to a minimum.

The cooling system for an engine without Waukesha-supplied thermostats needs to provide a warm-up function which allows the engine to reach its operating temperature by isolating the engine jacket water from the cooling system components (radiator, heat exchanger, etc.). The system also needs to maintain the designed operating temperature with minimal temperature fluctuations.

Cold water should never be allowed to enter a hot engine; this results in thermal shocking of the engine components. Failure to comply may lead to increased wear or possible engine damage.

Caution for Intercooler Water at 85°F or Below

Consider air dewpoint when rating an engine for 85°F (29°C) intercooler water. Hot, humid air, compressed by the turbocharger, then cooled to 85°F (29°C) in the intercooler will sometimes condense water in the intake manifold. This water will cause unstable engine operation and can wash lube oil off cylinder walls, resulting in liner scoring. Caution must be used when determining an engine rating based on intercooler water temperature. As a guideline, the following pressure vs. dewpoint in formation can be used.

Figure 5-25 is a plot of saturation lines at various intercooler water temperatures for air compressed by the turbocharger. The atmospheric dewpoint where condensation will form can be determined from the chart. The intersection of absolute boost pressure (Uncorrected barometric pressure, plus boost pressure) and intercooler water temperature determines the maximum atmospheric dew point. Condensation will form if the dew point exceeds this temperature.

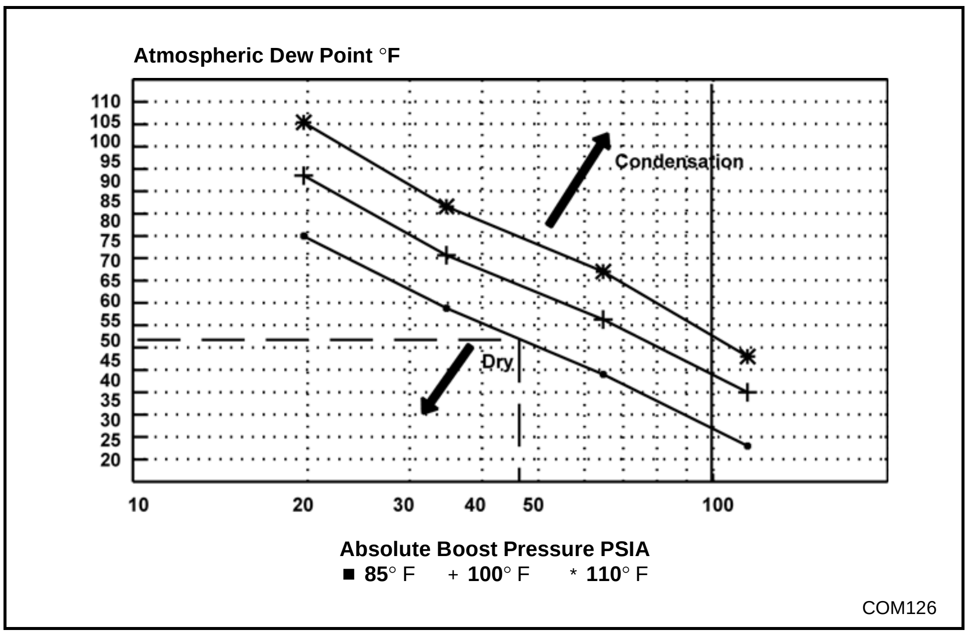


Figure 5-25 Boost Pressure vs. Dew Point to Determine Intake Manifold Condensation

Example:
An engine operates at 2000 ft altitude, which gives an average barometric pressure of 13.7 psia. The boost pressure for the selected engine is 30 psig. Maximum dew point expected during engine operation is 65°F (18°C). Can condensation be expected when 85°F (29°C) intercooler water is supplied?

Answer:
Absolute boost pressure = 13.7 psia + 30 psig ≈ 43.7 psia

From Figure 5-25, at 44 psia with 85°F (29°C) intercooler water, saturation dew point at atmospheric pressure = 52°F (11°C).

Condensation can be expected when operating 85°F (29°C) intercooler water temperature, and dew point exceeds 52°F (11°C).

Piping Installation

Flexible connections at the engine inlets and outlets will reduce stress due to piping alignment and vibration. Flexible connections are recommended for all installations and required for spring isolator mounted units. A Dresser-type coupling can compensate for some misalignment of piping, but once tightened, it is not considered flexible. Braided hose and bellows-type flexible connections are often used for this.

- Hanging Brackets

  All piping runs must be adequately supported or engine and cooler connection points may break. When selecting brackets, remember that the weight of the pipe, as well as the total weight of the coolant within the pipe must be supported.

- Isolation Valves

  In large cooling circuits, it is common to install valves at the inlet, outlet, vent line, and static line connections to the engine. Closing these valves will allow for draining the engine for servicing while retaining coolant in the remainder of the system.

SECTION 3

COOLING SYSTEM CHECKLIST

Engine Model: ___________

_______ BHP (kW) at _______ RPM

_______ °F (°C) Jacket Water Temperature Out of Engine

_______ °F (°C) Auxiliary Water Inlet Temperature

Water Pump Data:

Jacket Water Flow

_____ GPM (m^3/hr) at _____ PSIG (bar) Maximum External Restriction

_____ PSIG (bar) Maximum External Restriction

Auxiliary Water Flow

_____ GPM (m^3/hr) at _____ PSIG (bar) Maximum External Restriction

_____ PSIG (bar) Maximum External Restriction

***Heat Rejection Data:**

Jacket Water __________________________ BTU/hr. (kW)

Lube Oil ______________________________ BTU/hr. (kW)

Intercooler ____________________________ BTU/hr. (kW)

Radiation _____________________________ BTU/hr. (kW)

Combustion Air Flow ___________________ BTU/hr. (kW)

Cooling Media:

Water Source & Chemistry____

Solid Water with inhibitors (List Inhibitors) ________________________________

Glycol Solution ____ %

Other _________ (List) ____

Radiator:

Pressure Cap Rating __________ PSIG (bar)

Design Air Temperature into Radiator__________ °F (°C)

Design Air Flow into Radiator _______ SCFM (m^3/hr) at _______ in. H2O (mbar) External Restriction

Design Flow Rate _________ GPM (m^3/hr) at _________ PSIG (bar) Restriction

Room Design Air Flow ________ SCFM (m^3/hr) at ________ in. H2O (mbar)

Remote Cooling:

Design Air Flow _________ GPM (m^3/hr) at _________ PSIG (mbar) Restriction

System Volume _________ Gal. (m^3)

Expansion Tank Size _________Gal. (m^3)

Static Line Size __________ in. (mm) __________ Aux. Water

Solid Water Systems

1. Has the radiator or heat exchanger been properly sized to reject the engine heat?
2. If a pusher type fan is used with a unit mounted radiator has the temperature increase of the air due to engine radiation been accounted for?
3. If a suction fan is used with a unit mounted radiator has the temperature of the air due to engine radiation been accounted for in the inlet combustion air?
4. Is the engine room designed for adequate air make up?
5. Have the effects of coolant media heat transfer, wind, hot air from other equipment, altitude, enclosures, solar heating from surrounding ground, hot air recirculation and exhaust heat recirculation been accounted for?
6. Does the radiator have a static chamber either as a separate expansion tank or as a static expansion chamber in the top tank?
7. Has the jacket water and auxiliary water cooling circuits been designed to the design flow and external restriction at rated load and speed?
8. Have flexible connections been included to isolate engine and components from high stresses due to vibration?
9. Are there separate expansion tanks for the jacket water and auxiliary circuits?
10. Are there separate static lines to the water pump inlets for each circuit?
11. Are the static lines properly sized for the amount of flow in the water circuits?
12. Is the jacket water pressure according to Waukesha's recommendations?
13. Has a system pressure cap been properly sized for the system?
14. Is the system pressure cap at the highest point in the system?
15. Are vent lines connected from the engine and all high points in the cooling system to the expansion tank below the water line?
16. Is the thermostat control in the proper position relative to its function (mixing or diverting)?
17. Has cooling water quality and water treatment been addressed?
18. Are engine preheaters installed for starting in cold conditions?
19. Has the effects of thermal shock been addressed for engines in cold ambient?

CHAPTER 6

LUBRICATION SYSTEM

INTRODUCTION

The lubrication system could be considered one of the simplest systems on the engine; however, its importance should not be underestimated. This system must be installed correctly to ensure proper engine performance. By circulating properly selected oil throughout the engine, the lubrication system performs three main functions: lubrication, cooling and cleaning.

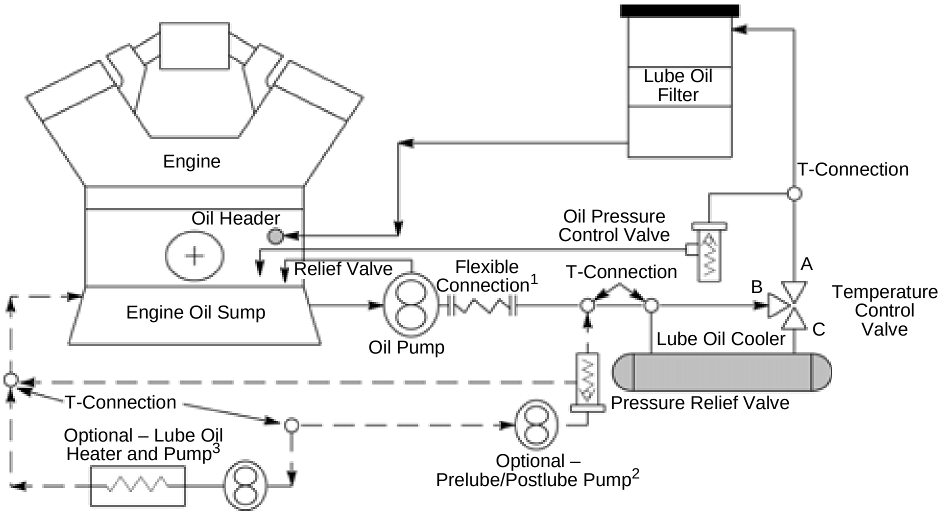
Lubrication systems provide a cushion of oil preventing direct metal to metal contact between engine components. Without a properly functioning lubrication system, moving metal surfaces would come into direct contact with each other. This will create wear and heat, leading to engine failure. If oil does not reach the cylinder sleeves and rings, piston and piston ring scuffing will occur, leading to a loss of ring seal. Excessive blow-by and decreased power would result, ultimately leading to engine seizure/failure.

Oil absorbs heat as it flows through the engine. The combustion chamber is cooled by the jacket water in the cylinder head and around the sleeve and by lube oil on the piston. The heat is then transferred from the lube oil to the auxiliary or jacket water system by the lube oil cooler.

The lube oil contains many additives which enhance specific performance characteristics. Among these additives are dispersants and detergents which suspend dirt and water particles in the oil allowing for removal by the oil filter system. This cleansing action is important for component longevity. (Refer to Waukesha technical data for further details on oil additives.)

While the lubrication system for an industrial engine may or may not come completely assembled from the factory, the installation and maintenance procedures are extremely important. See Figure 6-1 for a general piping schematic for Waukesha Engines.

Today most Waukesha engines are provided with mounted, fully contained lubrication systems. The information given in Sections 2 and 3 is most applicable to engines without mounted lubrication systems.



[1] Standard on all VHP Engines. Customer supplied on all other engines.

[2] Required on all 275GL and VHP engines. Standard equipment on VHP, APG, and VGF GLD/2 engines. Recommended on VGF engines.

[3] Required on 275GL, APG1000/16V150LTD, VGF "Vee", and VHP engines where ambient temperature drops below 50°F (10°C). Must maintain minimum 70°F (21°C) oil temperature, 100°F (38°C) for standby.

Figure 6-1 General Piping Schematic

SECTION 1 LUBRICATION SYSTEM COMPONENTS

Piping

The lubrication system is piped by Waukesha unless the lube oil cooler or filter assembly are shipped "loose." When oil coolers or filters are shipped loose, the packager is responsible for connecting the engine to the cooler and/or filter. If these components are located within eight feet of the engine, the piping size should match the connection size on the engine. If the components are further than eight feet away, the next larger standard pipe size should be used to connect the engine to the cooler and/or filter.

NOTICE **Black iron or steel pipes should be used to carry oil. Never use galvanized or zinc pipes. The sulphur content of the oil will react with the galvanization and zinc metals forming sludge.**

After welding, Waukesha recommends flushing the pipes with muriatic acid (pickle) to remove all welding scale. The pipes then must be rinsed thoroughly to remove the acid and dried before connecting to the engine.

All components and piping must be emptied completely when the oil is drained at a service interval. Used oil trapped by piping that forms a sag below the engine will deteriorate the fresh oil and reduce oil life, leading to an inaccurate oil analysis. All oil should reach the drain by gravity.

Flexible Connections

Flexible connections should always be used between the engine and the off engine mounted cooler and/or filter. Position the connections as close to the engine as possible. Oil temperature and pressure must be considered when selecting flexible connections. Supports should be added under the piping to support the weight of the piping and oil to prevent breakage of the flexible connections.

Dresser couplings, when used in high pressure oil systems, require the piping to be restrained to prevent the coupling from pulling apart.

Fittings

Fittings should consist of a flange and length of pipe. The flange should be welded to the pipe. Care should be taken when using threaded fittings as engine vibration could cause fittings to loosen causing leaks.

Filters and Strainers

Filters and strainers are included in the lubrication system to keep the oil clean. They should be mounted and piped before engine startup to prevent oil slag and piping debris from entering the engine.

Oil filters are designed to catch small particles of dirt and debris and remove them from the oil, preventing premature bearing failure. Many different oil filter designs are available on the market today. The first type of oil filtration, and the most common, is the “full-flow” type. Full-flow means that all the lubricating oil is normally filtered. 275GL, VHP and VGF engines come standard with a “full-flow” type oil filter. In the event the “full-flow” filter element becomes clogged, the housing contains a pressure relief valve which prevents the loss of oil circulation to the engine. Typical pressure relief setting is 28 – 32 psi (193 – 221 kPa). While this lube oil may be unfiltered, the engine is not being starved from lube oil flow. 275GL, VHP, APG, and VGF have pressure ports on the engine-mounted filter housing which can be used to install oil filter pressure differential devices. The 275GL feature ESM pressure transducers to monitor pressure differential and alarm at a predetermined setting. With the engine at operating temperature, the elements should be changed when the pressure difference between the two gauges reaches 12 – 15 psi (83 – 103 kPa) or at every oil change, whichever is first.

Centrifugal type oil filters use the centrifugal force of spinning oil to separate the impurities and force them to compact on the outer walls of the filter shell.

All filters are rated with an efficiency for particle removal. That is, what nominal particle size and larger would be removed during normal operation. Waukesha VHP and 275GL engines have a nominal 15 micron (0.0006 in.), and larger, rating for the filter elements while the VGF engines are rated at a nominal 20 micron (0.0008 in) and larger. The standard sock style filter elements used in the VHP and older VGF (“Vee” only) engines can also be substituted with a cleanable style filter element. The cleanable elements are rated at an absolute 25 micron versus the 15 micron for a standard replaceable sock type filter element.

A bypass filtration system can also be used for further removal of oil contaminating particles and improve the service life of the engine. Bypass filtration is installed so that a small portion of the lube oil flow is continuously bled off either directly before the full-flow engine oil filter or directly after the oil pump. The placement of the bypass filtration depends upon the type used. Waukesha offers two different types – Replaceable cartridge type or the Microspin System.

The Microspin* system, as offered by Waukesha, is an oil pressure driven centrifuge bypass oil filtration system. Engine oil pressure drives the centrifuge, which creates a spinning action from the internal turbine assembly developing a force that exceeds 2000 Gs. This centrifugal force compacts the oil contaminates against the turbine housing. The centrifuge will remove contaminates as small as 0.5 micron. Waukesha offers the Microspin system as a kit for all Waukesha engines.

Oil Cooler

The oil cooler dimensioned for the coolant temperatures as described in the Tech Data is included in the standard scope of supply of most Waukesha engines and Enginators. For CHP applications that have a demand for heat a customer supplied oil cooler with lube oil thermostats is required.

SECTION 2 LUBE OIL FILTER/COOLER INSTALLATION

For engines with remote mounted lube oil filters/coolers, position the filter and cooler as close to the engine as possible. Filter canisters and coolers should be mounted at skid level or lower to prevent excessive oil drain-back into the sump during shutdowns. A drain valve or plug should be placed in this piping circuit to ensure proper oil drainage from the cooler or canister. Keep the filter and cooler in a warm location.

Lube Oil Strainer

The strainer is a final barrier to prevent large particles from entering the engine. The strainer, made of stainless mesh, will only stop large particles and is not to be used in place of the lube oil filter. The strainer is also used during start-up when a paper element is installed to catch welding slag and other fabrication debris. Lube oil strainers are used only on engines with remote mounted oil filters.

NOTICE **Do not put the filter or cooler near the exhaust outlet or other places where the temperature could become excessively warm. Excessive heat will speed oil deterioration. It will also create a fire hazard in the event of an oil spill or line rupture.**

* Trademark of General Electric Company

The lube oil cooler should be mounted horizontally. This prevents air from becoming trapped in the cooler and reduces the amount of oil drain-back into the sump on shutdown. The oil filter should be prefilled with oil prior to engine startup.

SECTION 3

ENGINE PRE/POST LUBRICATION SYSTEM

Engine prelube performs several functions:

- Extends engine life by filling the lube oil cooler and filter prior to the engine starting. This prevents the engine from being starved from the lack of lubricating oil upon immediate startup.
- Purges the lubrication system of air and ensures all moving parts subjected to friction are properly lubricated before the engine is started.

Prelubing is required on 275GL, APG, VGF "Vee", and VHP engine models and is recommended on VGF inline engines. Prelube is required on VGF engines with frequent, more than twice per day, engine start-ups and shutdowns. Waukesha recommends several prelubing methods:

1. Prelube prior to each engine start. This works well on continuous duty applications. See Table 6-1 for prelube time, pressures and flow rates.
2. Continuous Prelube. VHP Series 2 engine models offer continuous prelubing as an option. This method works well on VHP Series 2 engines in standby applications where startup is immediate. Continuous prelube is not available on VGF, VHP Series Four and 275GL engines.
3. Prelube for a set time interval. This method works well for standby applications where engines must start immediately. Prelube is required on VHP 9394 two minutes minimum engines for five minutes every hour. Prelube is recommended for standby VGF engines for 30 seconds every 30 minutes. For 275GL engines, prelube for 15 seconds every hour.

WARNING

There must be NO postlube with any engine emergency shutdown.

Table 6-1 Prelubing/Postlube Specifications

ENGINE MODEL		PRELUBE TIME DURATION	PRESSURE IN HEADER	FLOW RATE
275GL	12V275GL+	90 seconds before starting	5 – 25 psi (34 – 172 kPa)	30 gpm (113 liters/min)
	16V275GL+			
VHP	AC Electric motor driven prelube pump	*Recommended:* 3 minutes before starting -OR- *Required Minimum:* 30 seconds or until pressure is obtained	1 – 4.5 psi* (7 – 31 kPa)	7 gpm* (26 liters/min)
	24 VDC Electric Motor or Air/Gas motor driven prelube pump			
VGF	Inline	30 seconds or until pressure is obtained	5 psi min (34 kPa)	1.75 – 3.5 gpm** (7 – 13 liters/min)
	Vee			3 – 6 gpm** (11 – 23 liters/min)
APG 16V150LTD	Vee	30 seconds or until pressure is obtained	5 psi min (34 kPa)	7 gpm* (26 liters/min)

* *Based on 1200 RPM pump speed.*

** *Based on 1750 RPM pump speed.*

NOTES 1: *If an oil heater is applied, circulate oil to the sump, not the header, on all models except on VHP Series 2.*

2: *When applying a prelube pump to VGF, allow approximately 7 psi loss due to pressure required to open the prelube back flow check valve and for piping losses and 2 – 3 psi on VHP for its check valve and piping losses.*

3: *Pressures may drop in half with hot oil, flow is the determining factor.*

4: *This information was obtained from S7382-56 which should be checked for more details and the most recent data.*

For continuous prelube on VHP engines a check valve (5 psi max) is used to prevent excessive oil flow to the turbocharger and rocker arms. This excessive oil flow to the compressor side of the turbocharger can produce oil leakage in the intake manifold causing oil deposits and fouling. Excessive oil flow to the turbine side of the turbocharger can pose a potential fire hazard due to the extreme heat from the exhaust gases. Also, excessive oiling of the rocker arm assemblies can cause oil to leak past the valve stem seals resulting in the formation of carbon deposits in the combustion chamber. A buildup of oily deposits on the valve stem and guides can lead to stuck valves.

Post Lubrication Requirements for Waukesha Engines

Waukesha recommends post lubrication for 275GL+, VGF and VHP engines. Post lubrication provides cooling to the turbocharger bearings and prevents carbon coking of the oil and extends turbocharger life. Postlube should be performed automatically upon main gas valve closure for 5 minutes after every engine shutdown. For the 275GL+ product line, a maximum post lubrication time of 60 seconds is recommended.

SECTION 4 ENGINE OIL HEATERS

Lube oil heaters and jacket water heaters are required for engines operating at ambient temperatures below 50°F (10°C). Oil must be heated to ensure proper oil flow to ease startability and load application. For engines required to pull load immediately upon startup (standby applications) the oil should be heated to a minimum of 100°F (38°C). For engines that operate continuously other than planned service shutdowns, the oil should be heated to 70° – 100°F (21° – 38°C).

The oil should be heated to ensure proper oil flow, improving the startability of the engine. Cold oil will not flow through the cooler and filter and still provide adequate supply pressure to the engine. Waukesha requires circulating type oil heaters to be used. This prevents the burning or oil coking that can occur with immersion style heaters. The oil circulating pump can also be used for pre/post lubrication.

When piping for engine oil pre/post lubrication and oil heating, see the installation drawing for connection points and sizes. Oil is drawn directly from the engine oil sump drain, and piped to the inlet of the pump/heater. From the heater, the oil flow should be piped back to the engine oil sump. See Figure 6-1 for a general piping schematic.

VGF engines have lubricating oil heating systems available. For 275GL engines, see latest edition of S9064-2 in the Cooling Systems section of the *General Technical Data* book.

SECTION 5 LUBE OIL RECOMMENDATIONS

Lube oil selection is the responsibility of the engine operator and the oil supplier. The refiner is responsible for the performance of the lubricant. Waukesha does not ordinarily recommend lubricants by name or brand. However, our recommendations, based on actual field experience, are listed in latest edition of S1015-30 in the Fuels and Lubrication Systems section of the latest edition of the *General Technical Data* book and also in the Service Bulletin binder – the latest edition of 12-1880.

Lubricating oils are comprised of various additives which enhance various performance characteristics for oil suspension, cleansing, etc. Such additives include the following, but are not limited to:

- Lubricants – creates a slippery “effect”.
- Detergents – aids in cleaning for component longevity.
- Wear Inhibitor – formulated additives to produce ash in the combustion chamber. Small amounts become deposited on the valve face which minimizes valve face and seat wear. See latest edition of S1015-30 in the *General Tech Data* binder or latest edition of Service Bulletin – 12-1880 for the recommended percent (%) ash content in the lube oil for each engine model and application type.
- Corrosion Inhibitor – neutralizes acid content. Ash and zinc are commonly used for neutralizing acid content.
- Viscosity – Maintains oil thickness and thermal stability.
- Dispersants – suspends dirt and wear particles for removal at the oil filter.

- Nitration and Oxidation Inhibitor – reduces sludge and varnish deposits. Nitration can be caused by high amounts of NOx in the exhaust gas, high blow-by with positive pressure in the crankcase, overextending the oil change interval, low operating temperature (minimum 160°F [71°C]). Oxidation can be caused by high operating oil temperatures and also overextending oil change intervals. Zinc is commonly used to reduce nitration and oxidation.

The use of multi-viscosity oils should only be used for engines starting in cold weather applications. Multi viscosity oil may deteriorate in continuous operation permitting the oil to lose viscosity through shearing. In this state the oil may not maintain sufficient oil thickness, thermal stability, and/or pressure. Oil analysis should be used to determine the oil change intervals.

Waukesha recognizes synthetic lubricating oil as being suitable for all Waukesha stoichiometric and lean burn gas engines. When synthetic lubricating oils are being selected, it is suggested that you contact Waukesha for change interval recommendations. Typically, synthetic oil change intervals are 3 – 5 times longer than those of mineral oils. Oil filter change intervals remain at 1000 to 1500 hours of operation. When operating on alternative fuel gas applications, synthetic oils are not recommended without Waukesha's prior approval.

Waukesha recommends regular lube oil analysis be performed to determine lube oil change intervals, to monitor engine wear, and to check for system contamination. Actual engine oil change intervals are determined by engine inspection and oil analysis in conjunction with the condemning limits and recommendations listed in latest edition of S1015-30 of the *General Tech Data* binder or latest edition of Service Bulletin 12-1880. Varnish deposits and sludge conditions in an engine will not be detected by oil analysis and can result in engine damage. Engine inspection is necessary with extended oil change intervals in addition to oil analysis. Special attention should be paid to the Total Base Number (TBN) and wear metals in the analysis. These can be an indication of problems with the engine or auxiliary equipment.

Waukesha strongly recommends changing the lube oil in applications where the engine is shutdown for long periods of time, such as summer cogeneration applications. Used oil within the condemning limits can still harm the engine and lead to failure when left in the engine for long periods of time without running.

SECTION 6 LUBE OIL CAPACITIES

Use the following table (Table 6-2) to determine approximate engine lube oil system specifications. The sump capacities include the standard engine oil sump, oil cooler, oil filter and any lines supplied by Waukesha. Always prefill the lube oil filter and cooler before starting the engine. Prime the oil system on engines equipped with prelube pumps. Engine oil flow curves can be referenced from the *General Tech Data* binder in the *Fuels & Lubrication* section. Engine oil flow requirements are typically dependent upon the speed of the pump, header pressure, oil viscosity, and temperature.

Table 6-2 Lube Oil System Specifications

ENGINE MODEL	LUBE OIL CAPACITY	OIL PRESSURE*
12V275GL+	225 gal (852 L)	60 – 65 psi (414 – 448 kPa)
16V275GL+	275 gal (1041 L)	60 – 65 psi (414 – 448 kPa)
VHP 6-CYLINDER	66 gal. (250 L)	55 ± 5 psi (379 ± 35 kPa)
VHP 12-CYLINDER	90 gal. (719 L)	55 ± 5 psi (379 ± 35 kPa)
VHP 12 EXTENDER	190 gal. (42 L)	65 – 87 psi (450 – 600 kPa)
VHP 16-CYLINDER	155 gal. (587 L)	55 ± 5 psi (379 ± 35 kPa)
VGF 6-CYLINDER (High Capacity)	44 gal. (167 L)	70 ± 8 psi (483 ± 55 kPa)
VGF 8-CYLINDER (High Capacity)	56 gal. (212 L)	70 ± 8 psi (483 ± 55 kPa)
VGF 12-CYLINDER (High Capacity)	86 gal. (326 L)	74 ± 8 psi (379 ± 55 kPa)
VGF 16-CYLINDER (High Capacity)	113 gal. (428 L)	74 ± 8 psi (379 ± 55 kPa)
APG16V150LTD	113 gal. (428 L)	55 ± 8 psi (379 ± 55 kPa)

* *Oil header pressure is checked with the lube oil pressure stabilized at normal operating temperature and the engine fully loaded.*

SECTION 7

ANGULAR OPERATING LIMITS

Angular operating limits must be complied with to assure a constant supply of oil to the oil pump pickup screen. Due to its fluid nature, oil in the sump always flows to the lowest possible point. If the engine is not level, it is possible that the oil pickup screen/tube would not be able to pick up the lubricant (see Figure 6-2).

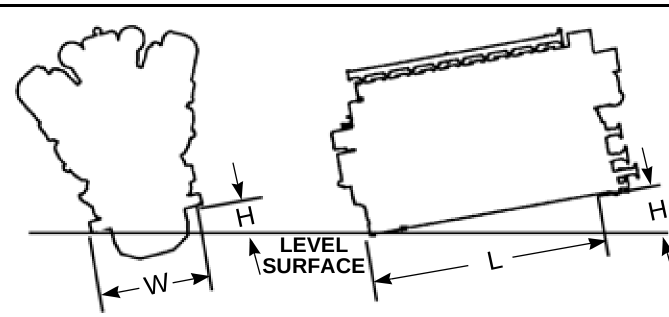


Figure 6-2 Angular Measurement Locale

This would mean a loss of lubrication at the bearings and other vital engine parts.

Waukesha strongly recommends mounting the engine on a level surface. However, Waukesha has established permissible angles at which the engine can operate without loss of oil to the oil pickup screen.

Determining Angle of Operation

H = Height of elevation

L = Length of engine mounting base

W = Width of engine mounting base

To determine the angle of operation, the length (or width) of the engine mounting base and the height of elevation will be needed. The engine mounting base length can be found on the engine outline drawing. The height of elevation will have to be measured.

The angle of operation is calculated using the following procedure:

1. Divide H (height of elevation) by L (length of engine mounting base for front/rear down angle) or W (width of engine mounting base for left/right down angle).
2. In column A, locate the number calculated in Step 1. If the exact figure is not listed, locate the next highest number.
3. The corresponding figure in column B will be the angle of operation. See the Angular Operation Limits Table to determine the acceptability of the installation.

Angle of Operation Table of Angles

The procedure described for calculating the angle of operation uses the mathematical principles of trigonometry (see Table 6-3). The figures in column A are the sines of the angles listed in column B. The sine of the angle of operation is equal to the height of elevation divided by the length of base (H/L) or by the width of base (H/W).

Table 6-3 Angle of Operation

A (sine of b)	**B** (angle, °)
0.0175	1
0.0349	2
0.0523	3
0.0698	4
0.0872	5
0.1045	6
0.1219	7
0.1392	8
0.1564	9
0.1736	10
0.1908	11
0.2079	12
0.2249	13
0.2419	14
0.2588	15
0.2756	16
0.2924	17
0.3090	18
0.3256	19
0.3420	20
0.3584	21
0.3746	22
0.3907	23
0.4067	24
0.4226	25
0.4384	26
0.4540	27
0.4695	28
0.4848	29
0.5000	30

Table 6-4 Allowable Operation Limits for Engines Equipped With Standard Oil Pan and Pump

MODEL[1]	FRONT DOWN DEGREES[2]	REAR DOWN DEGREES[2]	LEFT DOWN[3]	RIGHT DOWN[3]
F18, H24	1	1	7	7
L36, P48	1	1	6	6
APG1000/16V150LTD	1	1	6	6
F3521, F3524, L5790, L5794, L7042, L7044	2	2	7	7
P9390	1	2	7	7
12V275GL+	7	7	15	15
16V275GL+	5	5	15	15

NOTES 1: *Values apply to all model variations, i.e. G, GSI, GL, unless otherwise noted.*

2: *Tabulated angle operation values are based on unidirectional tilt. For bidirectional tilt or allowable intermittent tilt, consult Waukesha's Application Engineering Department.*

3: *Left and right are as viewed when facing the flywheel end of the engine.*

NOTE: *Note that operation under an angle will change the load distribution and forces on the mounting. Especially when using spring isolators this has to be accounted for.*

NOTE: *This information was obtained from S3549-J which should be checked for more details and the most recent data.*

BREATHER SYSTEMS

Industrial engines rely on a crankcase breather system to remove normal engine combustion vapors which accumulate in the engine crankcase. These vapors, if not removed, can cause oil contamination, component damage, crankcase oil leakage, and shortened oil life. Most systems rely on some form of vacuum to evacuate the engine crankcase vapors. The following are the most common types of breather systems available:

- Vacuum behind the air cleaner draws the vapors into the intake, downstream of the air filter element, which are then burned in the combustion process, see Figure 6-3. This type of breather system is typically called a closed breather system because the vapors are directed in the engine.

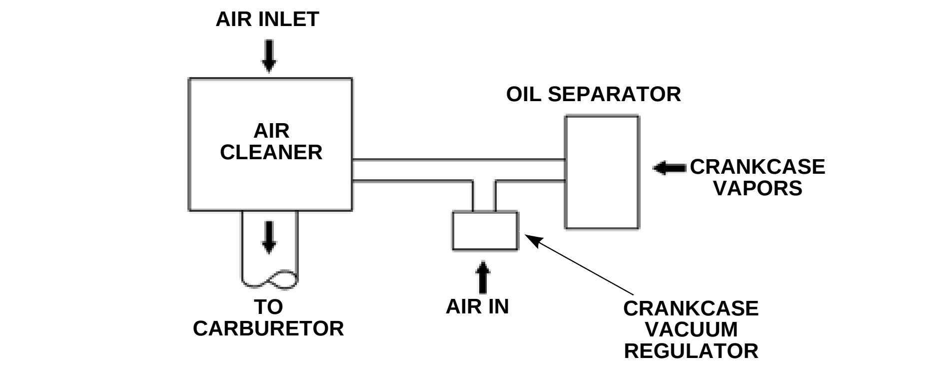


Figure 6-3 Closed Breather System

- F3514GSI, F3524GSI, P9390GSI and P9390GL engines use compressed intake air blown through a venturi to create a vacuum, which draws the vapors out of the crankcase and ejects the vapors into the exhaust system (see Figure 6-4). When a catalyst or heat recovery equipment is used, the vapors are routed downstream of the equipment in use (see Figure 6-5). When piping the crankcase vapors downstream of the high restrictions (i.e., catalytic converter, heat recovery unit, silencer), backpressure limits can be used without BMEP & RPM reductions. *General Tech Data – Intake/Exhaust Systems* should be referenced for exhaust system guidelines (latest edition of S8242) and VHP engine backpressure limitations (latest edition of S7567-3). For a procedure on calculating exhaust system backpressure, see Chapter 8, Exhaust Systems.

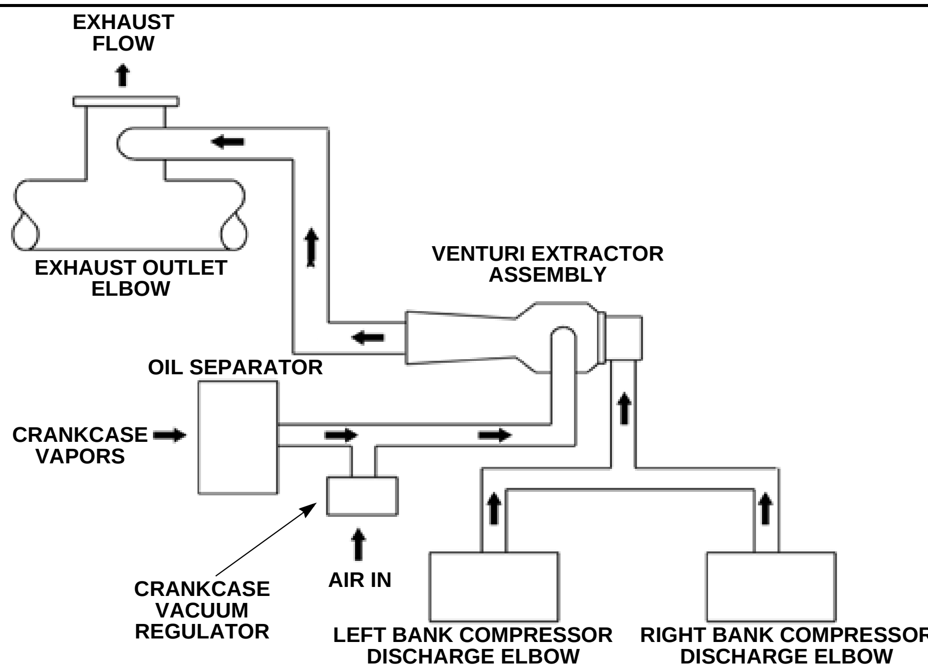


Figure 6-4 Ejector Type System (Standard)

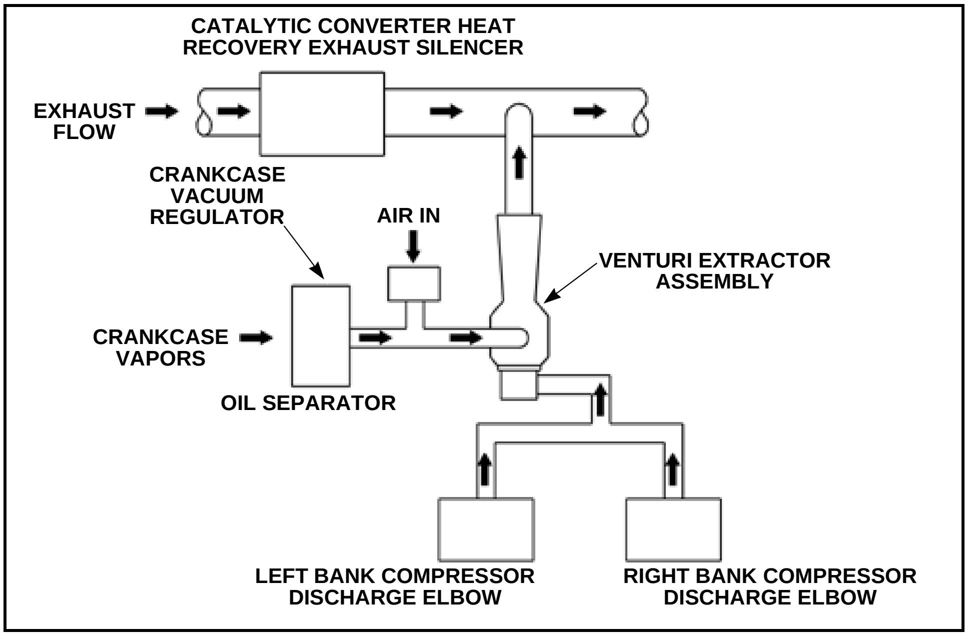


Figure 6-5 Ejector Type Systems (Optional)

- Vapors can be removed via a blower system venting the crankcase vapors to the ambient air. This type of blower system is required on the 275GL. The piping should slope upward from the engine connection to the blower to prevent trapping of oil in the low spots.

Drains should be installed at low spots in the piping, after the blower outlet, to drain condensed vapors. An optional oil separator should be installed after the flexible connection to protect the components from oil fouling. The flexible connection is used to isolate the breather components from engine vibration and to allow for expansion or growth due to heat.

The non-return (check) valve is used to prevent back flow of fresh air into the crankcase, eliminating a source of condensation. The check valve is necessary when there is no oil separator and the breather piping length is less than 10 ft (3 m). An adjustable restriction valve is recommended for adjustment of engine crankcase vacuum.

A “pressure blower” type fan (radial blade wheel) should be used since this type of blower will develop a constant pressure over a given flow range. The specified blow-by rate should fall in the flat portion of the blower flow curve. The blower design should also be spark resistant to prevent a possible fire hazard.

An open breather system is standard on 275GL. The breather system consists of a crankcase ventilation blower with 50 or 60 Hz motor, restrictor valve and check valve.

NOTICE **When this “open” type of breather system is used, the outlet may be considered a second source of emissions for regulator enforcement.**

Table 6-5 Crankcase Breather Specifications

MODEL	MAXIMUM BLOW-BY RATE*	CRANKCASE SUCTION VACUUM
12V275GL+	90 scfm (42.5 liters/sec)	0 – 2 in. H_2O (0 – 5 mbar)
16V275GL+	120 scfm (56.7 liters/sec)	0 – 2 in. H_2O (0 – 5 mbar)

* *Maximum blow-by rates relate to a “used” engine at maximum load. Not to be used as a condemning limit.*

For further information consult Application Engineering.

SECTION 8

LUBRICATION SYSTEM CHECKLIST

Lubrication Systems Components

1. Has black iron or steel pipes been used to carry the oil?
2. Have the pipes been flushed with muriatic acid, rinsed and dried after welding?
3. Have flexible connections been used between the engine and the off engine mounted cooler and/or filter?
4. Was oil temperature and pressure considered when selecting flexible connections?
5. Were supports added under the piping to support the weight of the piping and oil to prevent breakage of the flexible connections?
6. If Dresser couplings were used, are they restrained to prevent the coupling from coming apart?

7. Have the flanges been welded to the pipe?
8. Has a pressure gauge been installed for the purpose of measuring filter housing ΔP?
9. If Microspin centrifuge is used is the drain of the centrifuge at least 12 in. (305 mm) above the oil level?

Lube Oil Filter/Cooler Installation

1. For engines with remote mounted lube oil filters/coolers, have the filter and cooler been positioned as close to the engine as possible?
2. Have the filter canisters and coolers been mounted at skid level or lower to prevent excessive oil drain-back into the sump during shutdowns?
3. Have the filter and cooler been installed in a warm location?
4. Has the oil filter/cooler been installed away from the exhaust outlet or other places where the temperature could become excessively warm?
5. Have fill, make-up, and drain provisions?

Engine Pre/Post Lubrication System

1. Has a prelube system been incorporated into the package?
2. Has an automatic post lube system been incorporated into the package?
3. Has the oil from the prelube pump been piped upstream of the lube oil cooler?

Engine Oil Heaters

1. If the engine will be operating at ambient temperatures below 50°F (10°C), have lube oil heaters and jacket water heaters been incorporated into the package?
2. If engine will be operated in a standby application in which the engine is required to pull load immediately upon start-up, has the oil been heated to a minimum of 100°F (38°C)?
3. If lube oil heater is used, has a circulating type heater been specified?
4. If lube oil heater is required, has the heated oil been repiped to the engine sump?
5. Has the General Tech Data Manual been referenced for heater sizing?

Lube Oil Recommendations

See latest edition of S1015-30 or SB 12-1880.

1. Is the lube oil chosen to run in the engine classified to be run in natural gas engines?

2. Does the oil meet Waukesha lube oil requirements for the particular engine as listed in latest edition of S1015-30 or SB 12-1880?
   (Selected oil brand) ____________________
   (Selected type) ____________________
3. VHP/275GL: Do the filters installed in the engine meet Waukesha's basic filter requirements?
4. If engine is in a cogeneration application, is the engine oil on the Waukesha recommended lube oil list?
5. If synthetic oil is being used has Waukesha been contacted for oil change recommendations?
6. Is a lube oil analysis set-up for the engine?
   List Schedule
   ____

Angular Operating Limits

1. If engine is not to be set level have angular operating limits been complied with to assure constant supply of oil to the oil pick up screen? __________

Breather Systems

1. If the engine is equipped with a catalyst or heat recovery equipment, have the crankcase vapors been routed downstream of the equipment in use or are they ingested by the engine?
2. For customer supplied breather systems:
   List Blower Make
   Model
   Pressure Control Device
   Volume
   Pressure Rating
3. Have drains been installed at low spots in piping after the blower outlet to drain condensed vapors?
4. Has a flexible connection been used to isolate the breather components from engine vibration and to allow for expansion or growth due to heat?
5. If the piping is less than 10 ft (3 m) and no oil separator is used has a check valve been installed?
6. Has an adjustable restriction valve been installed to adjust crankcase vacuum?
7. Has a “pressure blower” type fan (radial blade wheel) been used?
8. Does the specified blow-by rate fall in the flat portion of the blower flow curve?
9. Is the design spark resistant to prevent a possible fire hazard?
10. Have Waukesha’s crankcase breather specifications been met?

NOTES

CHAPTER 7

AIR INDUCTION SYSTEM

INTRODUCTION

The air required for combustion is brought into the engine and cylinders via the air induction system. The temperature of the air entering the air cleaner can vary depending on site conditions. The induction system must be able to provide air within an acceptable temperature range, quality, and quantity to assure proper engine operation. Numerous options are discussed which will provide this type of air supply.

Air filtration principles and filter types are discussed which aid in the selection of the most adequate method for a given application. As part of the filter selection process, site conditions must be considered; such as the types of contaminants expected, desired service intervals, maintenance requirements, and system cost considerations.

The air induction system must be designed not to exceed the maximum permissible inlet air restriction. Exceeding the maximum permissible inlet air restriction will ultimately reduce the available engine power including adversely affecting the efficiency of the filtration system. System design guidelines are presented including a step-by-step procedure and sample problem to aid in calculating this total restriction for a proposed design.

A summary of installation recommendations are summarized in “Installation Summary” on page 7-15, including an Air Induction System Installation Checklist in Appendix A on page 7-A1. It is especially important to take into account all of the pertinent issues outlined in the entire manual during the preliminary design process. An air induction system initially built with the required capability, durability, and serviceability will prevent unnecessary modifications, maintenance, and expensive downtime in the future.

SECTION 1

AIR INDUCTION SYSTEM FACTORS

Air Temperature

The temperature of the engine air intake is usually between -50°F and 100°F (-46°C and 38°C) depending on site conditions. High temperature air is less dense and has fewer molecules per unit volume which reduces engine power output. This results in a horsepower loss of approximately 2% for every 10°F (5.5°C) increase in air temperature. Waukesha rates most engines for a maximum 100°F (38°C) ambient before derates are applied (see the Power Adjustment section in the General Technical Data book for engine specific derate information).

Cold intake air can also adversely affect engine operation, particularly when operating for long periods at light loading. Cold intake air creates a cold combustion chamber which can delay ignition, alter fuel combustion, and result in a loss of power. In cold ambient temperatures, below 50°F (10°C), intake air heating is typically required for effective engine starting. Ducting air from the warm side of the radiator, utilizing radiant engine heat to warm ducted air, or using warm engine room air are common methods of providing warm air in cold climates. Auxiliary water heaters can be used on turbocharged intercooled engines to heat the intake air with the intercooler prior to and during the starting procedure. See the *Starting*

Systems section of the General Technical Data Manual for special 275GL starting and running requirements.

In cold ambient temperatures, below 40°F (4.4°C), intake air heating is required for starting and operation of the 12V275GL+ and 16V275GL+ engines. Ducting air from the warm side of the radiator, using engine jacket water heat to warm ducted air through the use of a packager-supplied heat exchanger, or using warm engine room air are common methods of providing warm air in cold climates. Water heaters for the intercooler circuit are not an effective form of heating the combustion air because they do not heat the air upstream of the turbocharger, which is required to prevent turbo surge.

Intake air temperatures can be kept within a reasonable range if the air induction system is installed properly. Engine rooms must be designed and located to avoid absorbing excessive building heat. The correct air flow in the engine room is extremely important. Care must be taken that the induction air is not heated by the driven equipment such as the generator cooling air. When an engine is equipped with multiple air cleaners and carburetors, all air cleaners must get air at the same temperature. If the air in the immediate vicinity of the engine is not at a low enough temperature (below 100°F [38°C]), some means of ducting cool air directly to the air intake from outside the engine area must be found. If the air temperature to the intake is greater than 100°F (38°C), the heat rejection to the intercooler will increase significantly, resulting in an increase in the radiator or heat exchanger size.

The air intake system should not be installed near hot engine parts or exhaust lines. If installation requirements demand locating air induction lines near hot engine components, insulation must be used to prevent the intake air from being heated.

Air Quantity

Today's engines generally require from 1.1 – 2.6 SCFM (2.5 – 6 Nm^3/kW-hr) of air per HP just for combustion purposes. All installations should be planned with this in mind. An insufficient air supply will reduce power output and may cause structural problems in an engine room.

For enclosed installations, the combustion air supply should be separate from engine room ventilation air. Sometimes however, interior installations must use the air available in the engine room. If so, close attention must be paid to engine room design. In addition to absorbing radiant heat from the engine and supplying combustion air (unless air is ducted into engine), engine room ventilation should maintain a tolerable work environment for the operators. Normally, forced ventilation will be required. The need for ventilation can be reduced by insulating hot surfaces such as exhaust piping. Maintaining proper air flow drawing the ventilator air across the engine toward the exhaust fan, is important, especially when ventilation air is used for combustion. If blowers are used, they should not upset the air flow of the exhaust fans by trapping hot air in corners or isolated areas.

Air Quality

Dirt, if allowed into an engine, can create serious problems by destroying the precision part tolerances that keep an engine running. There can be from 1 – 5 grains (0.065 – 0.32 grams) of dirt in every thousand cubic feet of air. With an average air velocity of 6000 ft/min. (30 m/sec) at an engine air inlet, most of the dirt in the inlet area will be drawn in. An engine drawing 1,000 CFM (1700 m^3/hr) of this air due to leaks in the air induction system and/or leaking filters could end up ingesting enough dirt to destroy an engine in less than 50 hours of operation. It only takes about 1/2 lb (230 g) of dirt, or about one handful, to ruin an engine.

WARNING

Air inlets must be located away from fuel tanks, flammable vapors, tank vents, chemicals, industrial wastes or any other material of explosive nature. An engine backfire could ignite such material causing a dangerous explosion. Also, these volatile fumes could be drawn into the engine.

Air inlets must be located away from concentrations of dirt, dust and mist. If air in the immediate vicinity of the engine is not suitable for engine combustion, fresh, clean air must be brought in through ducting.

Water from rain or mist must be prevented from being drawn into the engine through the intake system. Not all engines are provided with rain guards; it is the customer's responsibility to ensure liquids do not enter the intake system.

All of the air induction system components must be sealed properly to prevent leakage. Once installation is completed, a system pressure test is recommended to locate and seal any air leaks in the system.

When purchasing air cleaner filter/filter elements, they must be capable of removing 99.6% of coarse dust as described in the SAE J726 *Air Cleaner Test Code Standard*.

SECTION 2 AIR FILTRATION PRINCIPLES

If dust enters an engine, it will stick on the oily film covering all internal engine parts. The dirt and dust will ruin bearings, plug lube oil passages and drastically shorten the life of many engine components. Therefore, it is of great importance that no dirt be allowed into the engine via the air intake system. Since no engine operating atmosphere is dirt-free, the only way an engine can be protected is with an effective air cleaner system.

Air cleaners available with Waukesha engines operate on one of two principles: inertial separation or impingement as in Figure 7-1.

Inertial Separation

Inertial or centrifugal separation is based on the law of inertia, stating that bodies set in motion will remain in motion until acted upon by an outside force. Dirt laden air is drawn into the cleaner with a rotational motion induced. The heavier particles of dirt switch direction due to the increase in rotational momentum and travel down into the bottom of the air cleaner box where they are trapped.

Impingement

Impingement filters out dust and dirt particles by having the air "strained" through a filtering medium. This is the operating principle behind a dry or paper type air cleaner. The air is drawn through the cleaner, but the particles of dirt are too big to pass through. Dirt particles get trapped within the filter element. This prevents the dust and dirt from entering the engine.

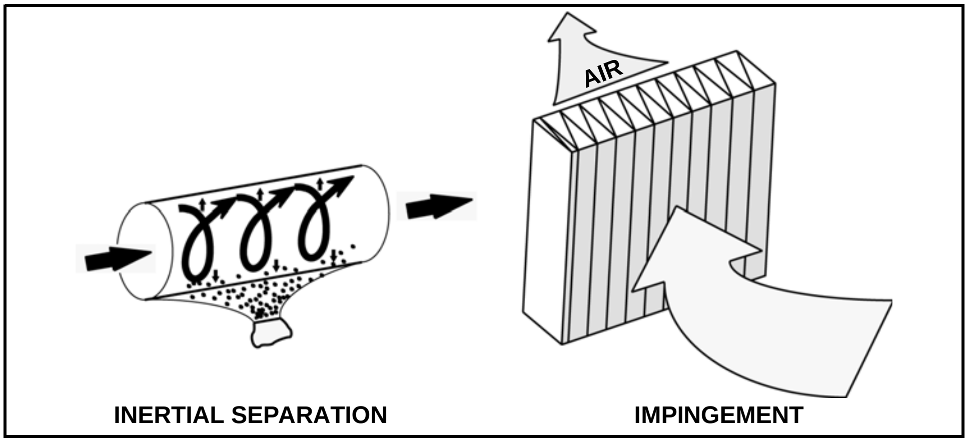


Figure 7-1 Air Filtration by Inertial Separation and Impingement

SECTION 3 AIR CLEANERS

The majority of air cleaning systems used on industrial engines utilize one or both of these inertial separation and impingement filtration principles. Typical air cleaner components are shown in Figure 7-3.

Precleaners

Precleaners remove a portion of the dust in the engines intake air prior to reaching the main filter element which will greatly increase the life of the main filter element. Factory installed precleaners are made of foam rubber or fiber materials and are mounted as shown in Figure 7-3. The precleaner frame assembly is double mounted allowing engine room operators to wash the precleaner without shutting the engine down. The rubber or fiber precleaner can be washed with soap and water.

Another type of precleaner utilizes inertial forces to remove a portion of the dust prior to reaching the main filter element. This inertial type precleaner can be ordered from Waukesha as an integral part of the air cleaner housing. It effectively filters out 70 – 90% of the large dirt particles in the first stage, thereby reducing the dust load passed onto the second stage of the filter. See Figure 7-2 for a description of a typical inertial type precleaner. As shown, the precleaner can be made up of various numbers of cyclone tubes. The air must pass through these tubes. Large dirt particles are spun out of the air as it is drawn through the cyclone tubes and fall into a dust bin located at the bottom of the panel. Inertial precleaners typically employ this type of atmospheric discharge with a dust unloader valve located at the bottom of the dust bin. However, for extremely high dust and/or dirt environments, a scavenge or vacuum type airflow can be used which will further increase the removal of dust particles.

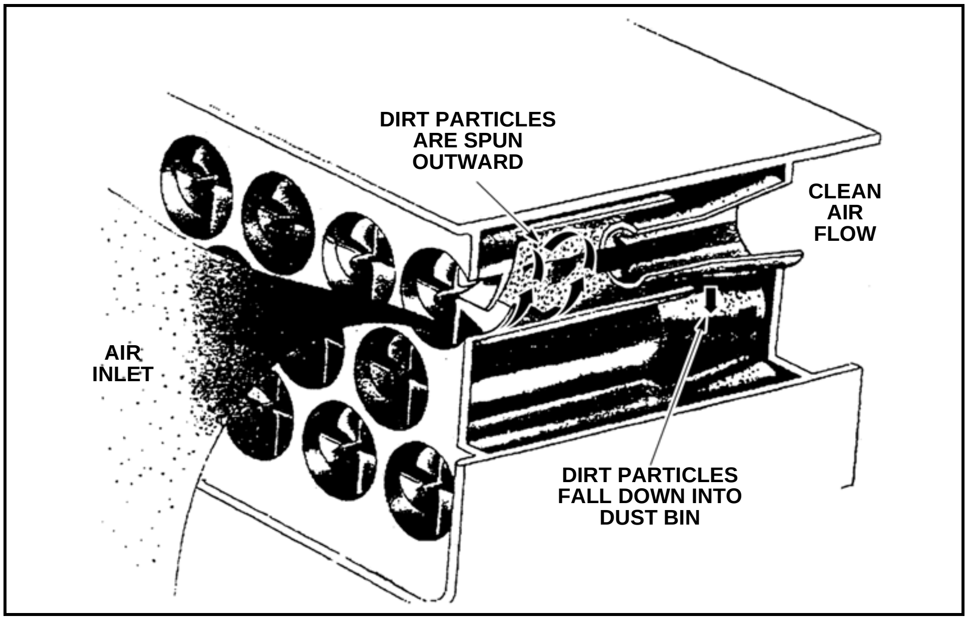


Figure 7-2 Inertial Type Precleaner

Dry Type Air Cleaners

Dry type air cleaners consist of one or more pleated paper elements as in Figure 7-3. Elements are manufactured from fire resistant, waterproof, pleated paper. The total area of the paper is so large that incoming air approaches the paper at a velocity of only 5 – 15 feet per minute (0.025 – 0.076 m/sec). The contaminants in the air are gently deposited on the paper fibers. The particles gradually bridge the openings between the fibers and form a porous filter cake, increasing filter efficiency. Eventually the contaminant buildup reaches a predetermined restriction limit and the element must be replaced. Engines equipped with factory installed air cleaners will have an air restriction indicator. The spring loaded indicator will “pop” red when the differential pressure exceeds its rating. It is recommended customers install an indicator when engines are not factory equipped.

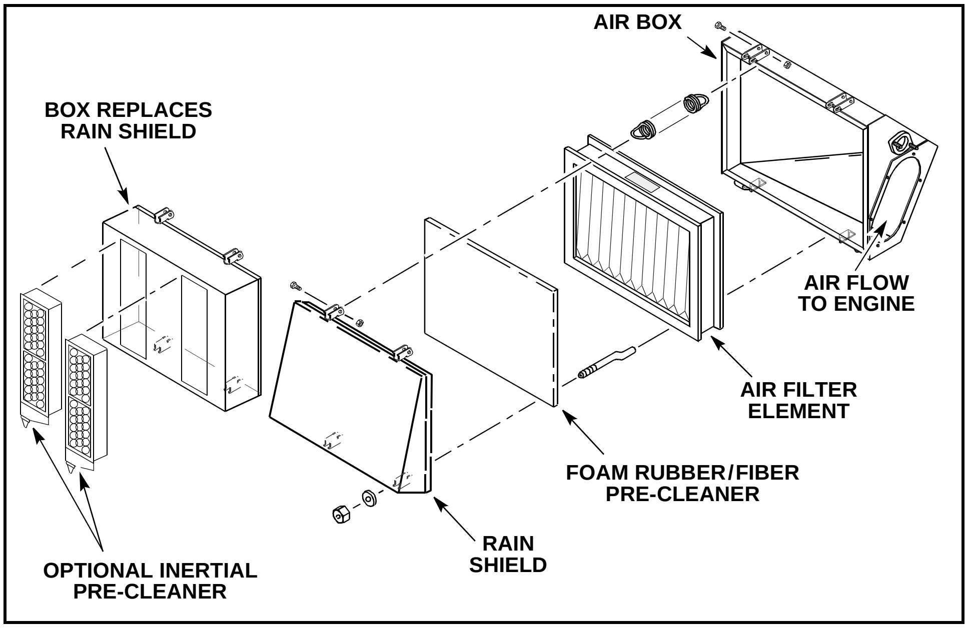


Figure 7-3 Typical Air Cleaner Components

Multi-Stage Air Cleaners

As the name implies, the multistage air cleaner uses two or more stages to clean the air, the first usually being an inertial type precleaner followed by one or more filter elements. This style cleaner is good for use in high dust and/or dirt environments.

Oil Bath Air Cleaners

Waukesha does not recommend using oil bath air cleaners. These air cleaners have an air flow operating range which must be observed to maintain their efficiency. They are also adversely affected by temperature, engine vibration, and tilted mounting positions. Increased maintenance is required including risks of oil injection into the engine intake.

Air Cleaner Effectiveness

Waukesha provides prefilters and dry type air cleaners as standard equipment for all industrial engines except the 275GL series. Due to the size of the 275GL series engines, each package requires a unique air induction system design. As indicated in the chart, the dry type air cleaner can handle all types of contaminants better than the oil bath cleaner, with the exception of soot. The dry type air cleaner has other advantages, which account for its widespread acceptance. These advantages include being:

- Highly versatile. Air cleaner efficiency is not affected by changes in engine air requirements over the load and speed range.
- Highly efficient – Filters out up to 99.9+% contaminants (depending on filter mesh and particle size).
- Unaffected by temperature, vibration or induction system pulsations.
- Unaffected by mounting position or tilt.
- Unaffected by rain, with a properly installed rain shield.
- Can be deliberately oversized for extended service life without affecting efficiency.

Table 7-1 describes the effectiveness of air cleaners exposed to a variety of contaminants.

Table 7-1 Effectiveness Ratings of Air Cleaners

AIR CLEANER TYPE	CONTAMINANTS			
	DUST	OIL VAPOR OR SOOT	ICE SNOW	LINT CHAFF
Oil Bath				
Multistage Dry Type				
Multistage Dry Type With Inertial Precleaner				

EFFECTIVENESS RATING:

= High

= Medium

= Low

SECTION 4 COMBUSTION AIR DUCTING

Air ducts have to be large enough to meet the combustion air requirements of the engine. Air duct restriction should be kept to a minimum. Maximum air inlet restrictions including the air filter (dirty) are listed in the Waukesha Technical Data Manual. Waukesha recommends PVC pipe for air ducting. Seamless or welded steel pipe can be used as an alternative to PVC. Select appropriately sized hose clamps to ensure strong airtight connections.

The pressure in the air ducting will, due to the suction of the engine and the restriction of the air filter and ducting, be lower than ambient. Therefore the ducting must be non-collapsible. The larger the diameter the stronger the ducting has to be.

NOTICE **All pipes and fittings used to bring air into the system must be absolutely free of dirt, scale and slag. Otherwise this material may be drawn into the engine upon startup and will damage engine components.**

Use a flexible connection to mount the air ducting to the engine. This will isolate most engine vibration from the ducting, thereby avoiding stress on the engine air inlets and pipes. Hangers or some other means of independent suspension should be used to install the ducting system. Do not try to support the ducts on the engine, since that would introduce stress to the ducting and the engine. Suspend the air ducting independently, and use vibration absorbing hangers to avoid transmitting engine vibration to the surrounding building.

The best air induction system has ducting as short and straight as possible. Use long radius bends and low restriction fittings only where necessary. The use of pipe fittings should be kept to a minimum. Each fitting added to a ducting system is equivalent to adding a length of pipe. The longer the pipe, the greater the restriction.

SECTION 5 ENGINE REQUIREMENTS

Turbocharger Considerations

Waukesha's turbochargers consist of an exhaust driven radial turbine directly coupled to a radial air compressor. The air compressor compresses large amounts of air in the case of blow-through (high fuel pressure) engines or an air-fuel mixture in the case of draw-through (low fuel pressure) engines. The result is an increase in the volumetric efficiency of the engine. This allows the engine to burn more fuel and develop more horsepower.

As the intake air or air-fuel mixture is compressed by the turbocharger, its temperature increases. This increase in temperature would ultimately reduce engine power output due to the decrease in density. Also, excessively high air or air-fuel mixture temperatures can lead to detonation in the combustion chamber. Waukesha's turbocharged engine installations use intercoolers to lower this temperature. The cooled air or air-fuel mixture exiting the intercoolers flows through the fuel system, intake manifold, and ends up in the combustion chamber at the appropriate air/fuel mixture temperature to ensure efficient combustion with maximum power output.

Turbocharged engines require significantly larger airflow requirements compared to naturally aspirated engines of the same displacement. Waukesha's turbocharged engines typically require a 40 – 130% increase in airflow compared to naturally aspirated engines depending on whether a stoichiometric or lean burn engine is required. Since air consumption for a turbocharged engine will be significantly higher than for a naturally aspirated engine, air induction systems must be designed with the capability of supplying an increased amount of airflow. Determine the required air induction for the appropriate engine model as published for each engine in the Heat Rejection Section (3) found in Volumes 1 and 2 of Waukesha's Technical Data Binders.

When turbocharged engines are operated at elevated ambient temperatures and high altitudes, the turbochargers compress lower density air which results in a lower mass of air available for combustion. Standard engine power ratings are subject to derates at these site conditions. See *Power Adjustments* section in Volume 3 *(General Data*) of Waukesha's Technical Data Binders to determine the available engine power at these elevated ambient temperature and high altitude applications.

NOTICE **Be careful to keep all foreign matter away from the turbocharger compressor. If any hard object struck the blades, metal fragments could fly off of the wheel. The extremely high speeds turbochargers attain could force these metal fragments into the engine resulting in serious engine damage.**

WARNING

Serious personal injury could result from turbocharger failure.

Intake Air Restriction

Intake air ducting restriction should be kept to a minimum. Air induction systems should be designed with at least a 30% pressure loss reserve to minimize required air cleaner change intervals. The maximum permissible intake air restriction for the entire air induction system including the air cleaners are listed in the Waukesha Technical Data Manuals for each engine.

For example, a VGF P48GL engine has a maximum permissible intake air restriction of 15 inch-H_2O (38.1 cm-H_2O) wc (38 mbar). “Clean” prefilters and air cleaners should have no more than approximately 5 inch-H_2O (127 cm-H_2O) wc (13 mbar) of pressure drop. The total pressure drop including all losses from the air induction system inlet to the carburetor inlet (which is typically taken to be the exit of the air cleaner) should not exceed 10 inch-H_2O (254 cm-H_2O) wc (26 mbar).

System Reserve = [1 – (Total System Restriction)/(Maximum Permissible Restriction)] x 100

System Reserve = [1 – (10/15)] x 100 = 33%

Engines equipped with factory installed air cleaning equipment have an air restriction indicator located on the air cleaner box. It is recommended customers install an indicator when engines are not factory equipped.

SECTION 6

DETERMINING AIR INDUCTION SYSTEM RESTRICTION

Once an air induction system design is proposed the total system restriction must be determined. The total system restriction is the total pressure loss the system experiences when the maximum intake air is flowing through the system. This pressure loss results from a number of system characteristics. These can be categorized as follows: pipe wall friction, changes in flow path direction, obstructions in flow path, sudden or gradual changes in cross-section or shape of flow path.

The proposed air induction system must not exceed the “maximum permissible restriction” published for each engine in the Specification Section (1) found in Volumes 1 and 2 of Waukesha’s Technical Data Binders. Exceeding this limit can result in adverse engine performance and increased maintenance requirements. Excessive restriction will have detrimental effects on available engine power, fuel consumption, and air cleaner efficiency.

Before determining the air induction system restriction, the following information is necessary:

- A proposed air induction system design identifying all components, corresponding specifications, and an overall dimensional layout. This includes airflow restriction characteristics of components including louvers, rain guards, pre-cleaners, and air cleaners.
- Engine model and operational settings necessary to select the corresponding engine performance data. Engine performance data can be found either in Waukesha's Technical Data or by contacting Waukesha's Application Engineering Department for special cases.
- Maximum engine airflow requirement. This is found by referencing the above stated engine performance data at the operating point with the highest power and speed. However, selecting an airflow requirement at an engine's maximum power and speed is usually recommended. This assures the air induction system is not undersized if engine airflow requirements increase.

The primary objective is to keep the total air induction restriction, which includes system reserve, below the published maximum permissible restriction. This can be achieved by following the general procedure stated in "Air Induction System Restriction Calculation".

Air Induction System Restriction Calculation

The following procedure implements the EDL (Equivalent Duct Length) Method. This method provides an expedient and typically conservative procedure of estimating overall air induction system pressure losses. The "resistance coefficient K" and "flow coefficient Cv" methods are also commonly used. Also, a combination of these methods can be used depending on the accuracy desired and resources available.

The following steps can be used to obtain an estimate of the maximum system restriction of a proposed air induction system. See also "Sample Problem" on page 7-12

1. Determine the maximum engine inlet airflow at site conditions.

   See Section 4, "Heat Rejection and Operating Data", in Waukesha's Technical Data to find the engine airflow in SCFM (Standard Cubic Feet per Minute or Nm^3/hr) at the stated standard operating temperature and pressure.

   Correct the engine airflow to ACFM (Actual Cubic Feet per Minute or m^3/hr) based on site conditions using the following conversion:

   ACFM = SCFM x (Tsite + 460) / (Tstd + 460) x (Pstd / Psite) or
   m^3/hr = (Nm^3/hr x (Tsite + 273)) / ((Tstd + 273) x (Pstd / Psite))

   Tsite – Highest expected site temperature (°F or °C)
   Tstd – Standard temperature corresponding to SCFM airflow (°F or Nm^3/hr Air Flow °C)
   Psite – Highest expected site pressure (inches mercury or bar)
   Pstd – Standard pressure corresponding to SCFM airflow (inches mercury or Nm^3/hr (bar) Air Flow)

2. Determine the equivalent pressure loss per length of pipe (P_L/L) for each applicable component in the air induction system. Components that are not applicable will be accounted for in Step 4:

   The P_L/L for each component is obtained by referencing the Pressure Loss vs. Airflow graph located in Appendix B. Locate the actual engine airflow in ACFM or m^3/hr from step one on the horizontal axis. Reference the appropriate ANSI Pipe Diameter[1] on the graph corresponding to the system component diameter[2]. The corresponding P_L/L each component can be read on the vertical axis[3]. Tabulate a list of equivalent P_L/Ls for each component.

   [1] ANSI Schedule 40 Pipe Dimensions are listed in Appendix B.

   [2] For system components other than straight ducts the entrance diameter should be used.

   [3] Variations in pressure loss due to differences in component material are considered negligible. This assumption is applicable for materials with a specific surface roughness similar to ANSI schedule 40 commercial steel pipe.

3. Determine the Equivalent Duct Length (EDL) for each applicable component in the air induction system.

   This EDL is the amount of straight pipe that would have the same pressure loss as the component would have when air flows from the inlet to the exit of the component. The EDL for each component is obtained by first locating the component type in Appendix B.

   Appendix B lists EDLs for numerous types and sizes of fittings commonly used in air induction systems. Other sources can be referenced to find the EDLs of components not listed.

4. Calculate the pressure loss (P_L) for each applicable component in the air induction system:

   Multiply the tabulated P_L/Ls and EDLs together for to determine the pressure loss (P_L) for each component.

5. Determine the total pressure loss for miscellaneous components, $(dP)_{MISC}$, in the air induction system which could not be converted into EDLs:

   Component losses for items including rain shields, louvers, silencers, precleaners, and air cleaners should be determined based on the available component characteristics at the maximum engine airflow requirement. The total of all of these pressure losses are designated as $(dP)_{MISC.}$

6. Calculate the total air induction system restriction:

   Calculate the Total System Restriction by adding together all the calculated component pressure from Steps 4 and 5:

   Total System Restriction=$[(P_L/L) \times (EDL)]$ component$_1$ + $[(P_L/L) \times (EDL)]$ component$_{2...}$+ $(dP)_{MISC}$.

7. Verify that the Total Restriction does not exceed the Maximum Permissible Restriction including a 30% reserve:

   To minimize precleaner and air cleaner service intervals a pressure loss reserve of approximately 30% should be built into the air induction system. Verify if the proposed system design has this reserve:

   Total System Restriction – Total air induction system restriction as calculated in Step 6.

   Maximum Permissible Restriction – See Waukesha's Specification Section in the Technical Data Gas Volumes for the specific engine.

   System Reserve = [1-(Total System Restriction)/(Maximum Permissible Restriction)] x 100

   The air induction system must be designed properly not to exceed the maximum permissible inlet air restriction published in the *Engine Specification Section* of the Waukesha Technical Data Volumes. Intake duct restriction that exceeds this limit will reduce available engine power, increase fuel consumption, shorten air cleaner service periods and affect the efficiencies of two stage air cleaners. Before designing an air induction system, the following information is necessary:

   - Engine model – Make sure this information is accurate and complete.
   - Maximum engine power and speed expected – Include any intermittent or overload conditions called for in the specifications.
   - Combustion air requirements at maximum power and speed.
   - Engine air inlet pipe size.
   - Air induction system restrictions for louvers, rain guards, precleaners, air cleaners.

   For V-block engines, note if the information is the air inlet size per bank of cylinders or the air inlet size of the air intake header if supplied by Waukesha. If the header is supplied by others, it is recommended that the header diameter be sized so the cross sectional area of the header is equal to or greater than the sum of the areas of the turbocharger inlet connections or the carburetor inlet ducts on naturally aspirated engines. The main objective is keeping the intake restriction below the published maximum permissible include proper system reserve.

SAMPLE PROBLEM

A 12V275GL+ engine is required for continuous duty application. The engine will operate at 3625 BHP @ 1000 rpm. Factory supplied 54°C (130°F) intercooler water is used with a 82°C (180°F) jacket water temperature. The site is at sea level with the highest expected combustion air inlet temperature at 38°C (100°F). The total air induction system restriction must be calculated to determine whether it is acceptable or if it exceeds the maximum permissible restriction. The proposed air induction system is shown in Figure 7-4.

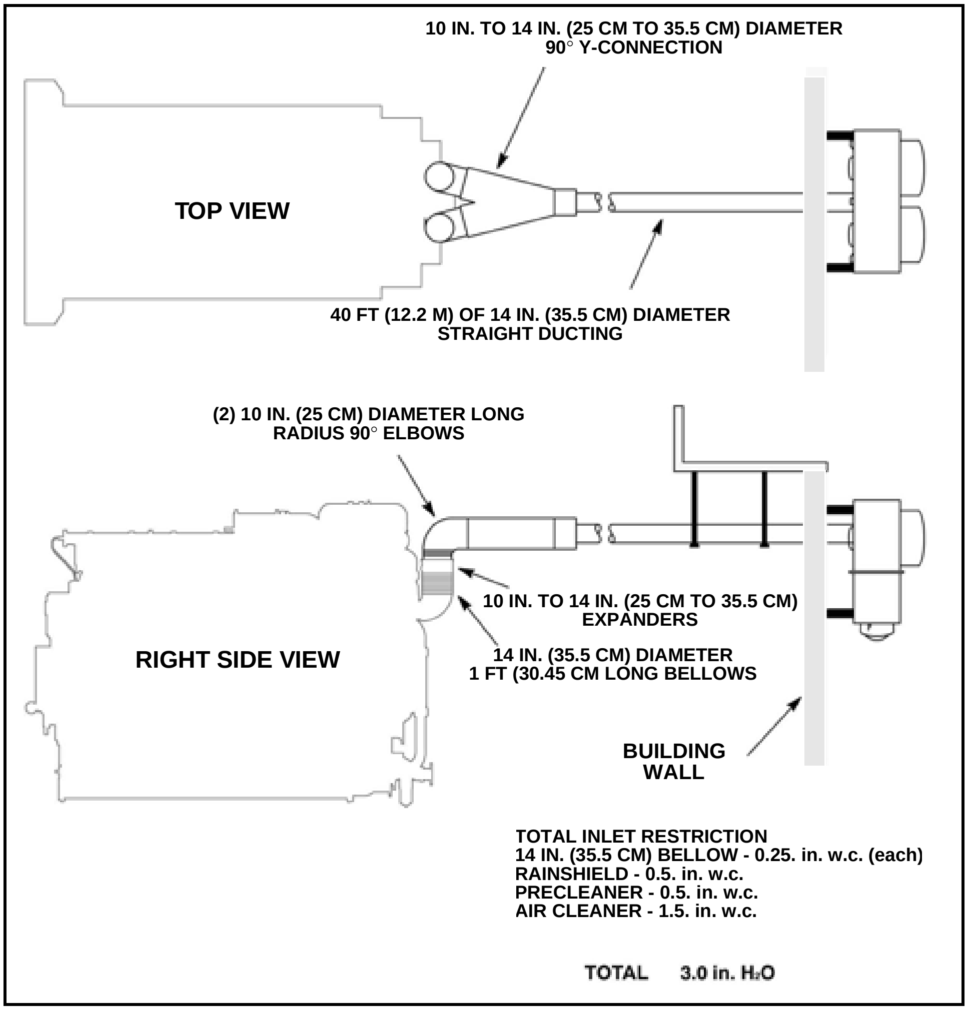


Figure 7-4 Proposed Air Induction System

Table 7-2

<table>
<tr><th>AIR INDUCTION SYSTEM RESTRICTION CALCULATION</th><th>REFERENCE</th></tr>
<tr><td>1. Determine maximum engine inlet airflow at site conditions:
3625 BHP @ 1000 rpm / 130° F Intercooler / 180° F Jacket Water /
Maximum 12V275GL+ engine airflow = 9847 SCFM
Actual Airflow = SCFM x (T_{site} + 460 / Tstd + 460)
= 9847 x (100° F + 460) / (77° F + 460)
= 10268 ACFM (5134 ACFM per turbocharger)</td><td>12V275GL+ Heat Rejection Section of Technical Data S-9062-10 or site-specific data from EngCalc</td></tr>
<tr><td>2. Determine the equivalent pressure loss per length of pipe (P_L/L) for each applicable component in the air induction system. Components that are not applicable will be accounted for in Step 5
<table>
<tr><th>Quantity</th><th>Component</th><th>Airflow (Entrance)</th><th>P_L/L</th></tr>
<tr><td>2</td><td>10" to 14" diameter expansion</td><td>5134 ACFM</td><td>1.0" w.c./10 ft</td></tr>
<tr><td>2</td><td>10" diameter long radius 90° elbows</td><td>5134 ACFM</td><td>1.0" w.c./10 ft</td></tr>
<tr><td>1</td><td>14" to 10" diameter 90° Y connection</td><td>5134 ACFM</td><td>1.0" w.c./10 ft</td></tr>
<tr><td>1</td><td>40 ft. of 14" diameter straight ducting</td><td>10268 ACFM</td><td>0.98" w.c./10 ft</td></tr>
</table></td><td>Appendix B</td></tr>
</table>

AIR INDUCTION SYSTEM RESTRICTION CALCULATION	REFERENCE

3. Determine the Equivalent Duct Length (EDL) for each applicable component in the air induction system:

Component	$EDL_{component}$
10" to 14" diameter expansion	5.4 ft
10" diameter long radius 90° elbows	16.0 ft
14" to 10" diameter 90° Y connection	34.7 ft
40 ft. of 14" diameter straight ducting	40.0 ft

Appendix B

4. Calculate the pressure loss (P_L) for each applicable component in the air induction system:

Component	P_L/L	x	QTY	x	EDL	=	P_L
10" to 14" diameter expansion	1.0" w.c./10 ft	x	2	x	5.4 ft.	=	1.08" w.c.
10" diameter long radius 90° elbows	1.0" w.c./10 ft	x	2	x	16.0 ft.	=	3.2" w.c.
14" to 10" diameter 90° Y connection	1.0" w.c./10 ft	x	1	x	34.7 ft.	=	3.47" w.c.
40 ft. of 14" diameter straight ducting	0.98" w.c./10 ft	x	1	x	40.0 ft.	=	3.92" w.c.
					$(PL)_{TOTAL}$	=	11.67" w.c.

—

5. Determine the total pressure loss for miscellaneous components, $(dP)_{MISC}$, in the air induction system which could not be converted into EDLs:

(2) 14" Bellows	+	Rain Shield	+	Precleaner	+	Air Cleaner	=	$(dP)_{MISC}$
2 x 0.25" w.c.	+	0.5" w.c.	+	0.5" w.c.	+	1.5" w.c.	=	3.0" w.c.

—

6. Calculate the total air induction system restriction:

$(P_L)_{TOTAL}$	+	$(dP)_{MISC}$	=	Total System Restriction
11.67" w.c.	+	3.0" w.c.	=	14.67" w.c.

—

7. Verify that the Total Restriction does not exceed the Maximum Permissible Restriction including a 30% reserve:

Total System Restriction = 14.67" w.c. (From Step 6)
Maximum Permissible Restriction = 15.0" w.c. (From S-9062-2)
System Reserve = [1 - (Total System Restriction) / (Maximum Permissible Restriction)] x 100
= [1 - (14.67" w.c.)/ (15.0, w.c.)] x 100
= 2.2% (Air induction system has too much restriction since minimum reserve is a least 30%)

12V275GL+ Engine Specification Section of Technical Data S-9062-2

NOTE: *The calculation in metric units follows the same guidelines as the sample calculation for English units.*

SECTION 7

275GL

The standard scope of supply of the 275GL, does not include air filtration. Air filters have to be supplied by the customer or ordered optionally at Waukesha. Air inlet piping must be supported such that no weight is carried by the turbocharger inlet adapter. A vacuum tight flexible connection at the turbocharger inlet adapter must be used. A rubber “Hump Hose” specifically designed for internal combustion engines is preferred. This prevents turbocharger damage from piping strain and vibration.

Dry-panel type air filter with rain shields is offered for the 275GL.

See latest edition of S-9200-13 for air filtration and air quality specifications.

SECTION 8

INSTALLATION SUMMARY

Air Supply

Air is ducted to an engine to assure an ample, steady supply of clean, cool air. Ducting in fresh air is always preferred to using engine room ventilation air for combustion. Air ducting pipes should run directly from the engine air intake to a location away from dirt, soot, warm air or exhaust gases. Also, the air inlet and air filter should be easily accessible for servicing and maintenance.

WARNING

Air inlets must be located away from fuel tanks, flammable vapors, tank vents, chemicals, industrial wastes or any other material of explosive nature. An engine backfire could ignite such material causing a dangerous explosion. Also, these volatile fumes could be drawn into the engine.

Cold Weather Operation

In cold ambient temperatures, below 50°F (10°C), intake air heating is typically required to assure proper engine operation. Ducting air from the warm side of the radiator, utilizing radiant engine heat to warm ducted air, or using warm engine room air are common methods of providing warm air in cold climates. Auxiliary water heaters can be used on turbocharged intercooled engines to heat the intake air with the intercooler prior to and during the starting procedure. Once a turbocharged engine is operating, the turbocharger will increase the air temperature as a result of compressing the air. See the *Starting System* section of the General Technical Data Manual for special cold weather 275GL starting and running requirements.

Air Cleaners

Replacement air cleaners must meet Waukesha's requirements of removing 99.6% of coarse dust as described in the SAE J726 Air Cleaner Test Code Standard. Also, refer to the filter manufacturer's recommended final resistance data to determine required change intervals.

System Pressure Loss Reserve

The air induction system should be sized with pressure loss reserve, approximately 30% to minimize required precleaner and air cleaner change intervals. Also, additional system reserve should be considered if the engine power rating may be increased in the future. Engine airflow requirements can increase by over 20% when operating at intermittent power ratings compared to continuous power ratings.

Air Ducting

Flexible connections should be used to mount the air ducting to the engine to isolate engine vibrations from the ducting. Use of vibration absorbing hangers can prevent transmitting engine vibration to the surrounding building. Avoid excessive pipe bends and fittings to minimize the total air induction system restriction. Make certain that all pipes and fittings are free of dirt, scale, and slag before engine startup or damage to engine components could occur. PVC piping and fittings are recommended.

System Pressure Testing

The entire air induction system from the entrance to the intake manifold flange gaskets should be carefully checked for air leaks. Pressure testing is recommended. Any leaks in the system would allow unfiltered air to enter the engine.

CHAPTER 7

APPENDIX A

AIR INDUCTION SYSTEM INSTALLATION CHECKLIST

1. Air intake point:

   ______________ Inside ______________ Manual

   ______________ Outside ______________ Automatic

   ______________ Combination ______________ Splash and rain protected

2. Is air inlet in cleanest possible location? ❑ Yes ❑ No
3. Filter System: ❑ Wet ❑ Dry ❑ Dual
4. Filter(s), Make, Model, Size ______________________________
5. Air line size ______________________________
6. Air line/tube material ______________________________
7. Hose clamps:

   ______________ Make ______________ Style

8. Filter and lines properly supported ______________________________
9. Sharp bends in system eliminated? ❑ Yes ❑ No
10. System free from chafing points? ❑ Yes ❑ No
11. Temperature rise air inlet above filter inlet ______________________________
12. Distance from hot exhaust parts ______________________________
13. Is air filter in hot air stream ______________________________
14. System pressure tight? ❑ Yes ❑ No ❑ Not Checked
15. Vacuum at turbo or manifold inlet:

    ______________ Desired ______________ Observed

16. Vacuum at air cleaner inlet:

    ______________ Desired ______________ Observed

17. On two stage air cleaner, vacuum between 1st and 2nd stage ______________
18. Intake manifold pressure:

    ______________ Desired ______________ Observed

19. Calculated air duct pressure drop (30% Reserve Recommended ______________

NOTES

CHAPTER 7

APPENDIX B

SYSTEM RESTRICTION REFERENCES

Table 7-1 ANSI Schedule 40 Pipe Dimensions

ANSI PIPE DIAMETER (IN.)	ID (IN.)	ID (mm)	AREA (INCH^2)	AREA (mm^2)	AREA (FT^2)
1.5	1.61	40.894	2.04	1312.77	0.0142
2	2.067	52.502	3.36	2163.80	0.0233
2.5	2.344	59.538	4.32	2782.61	0.030
3	3.068	77.927	7.39	4767.03	0.0513
4	4.026	102.260	12.73	8208.89	0.0884
5	5.047	128.194	20.01	12900.42	0.139
6	6.065	154.051	28.89	18629.39	0.201
8	7.981	202.717	50.03	32259.06	0.347
10	10.02	254.508	78.85	50847.84	0.5476
12	12	304.800	113.1	72928.89	0.7854
14	13.25	336.550	137.9	88913.73	0.9575
16	15.25	387.350	182.7	117781.42	1.268
18	16.88	428.752	223.7	144305.20	1.553
20	18.81	477.774	278.0	179190.38	1.931
22	21.00	533.400	346.4	223344.71	2.405
24	22.60	574.040	401.2	258674.71	2.786

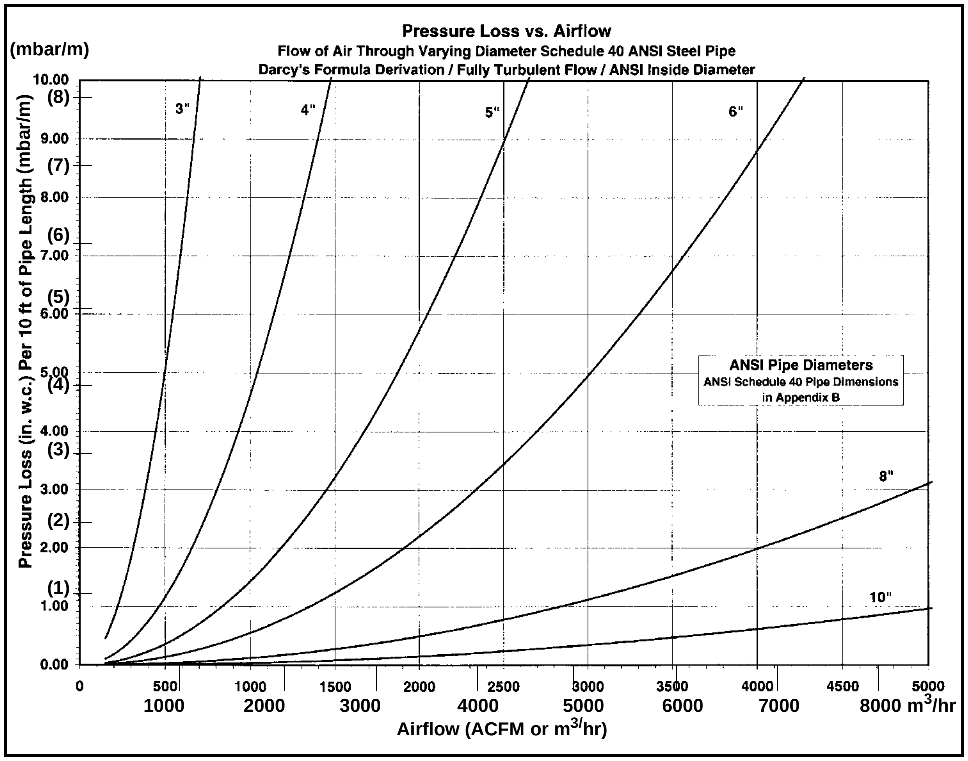


Figure 7-1

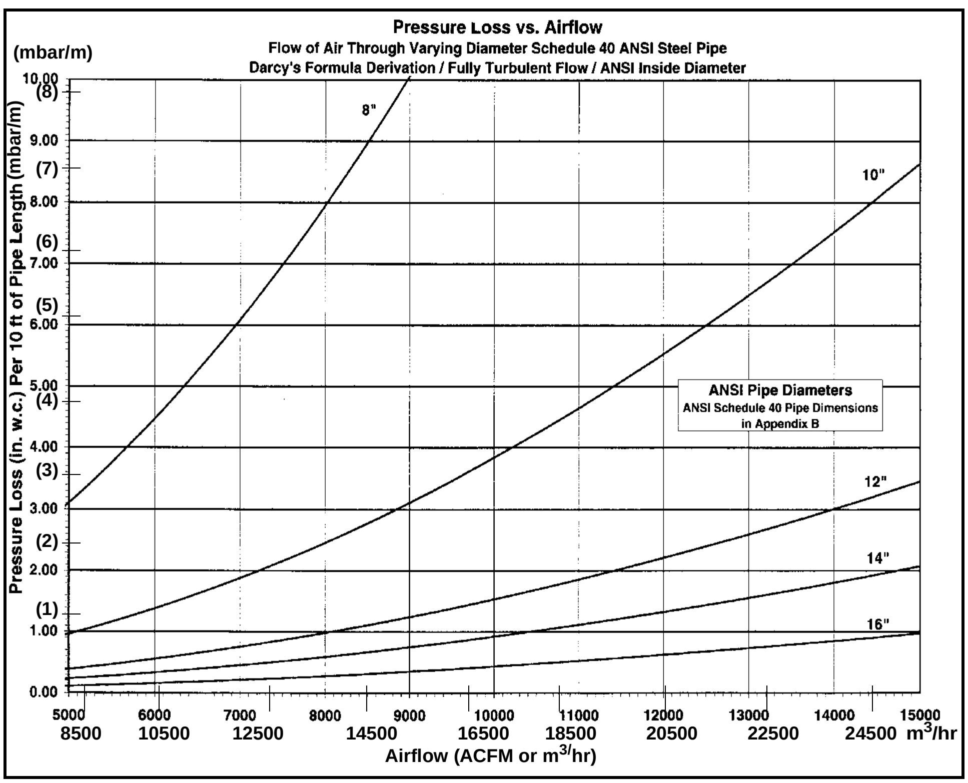


Figure 7-2

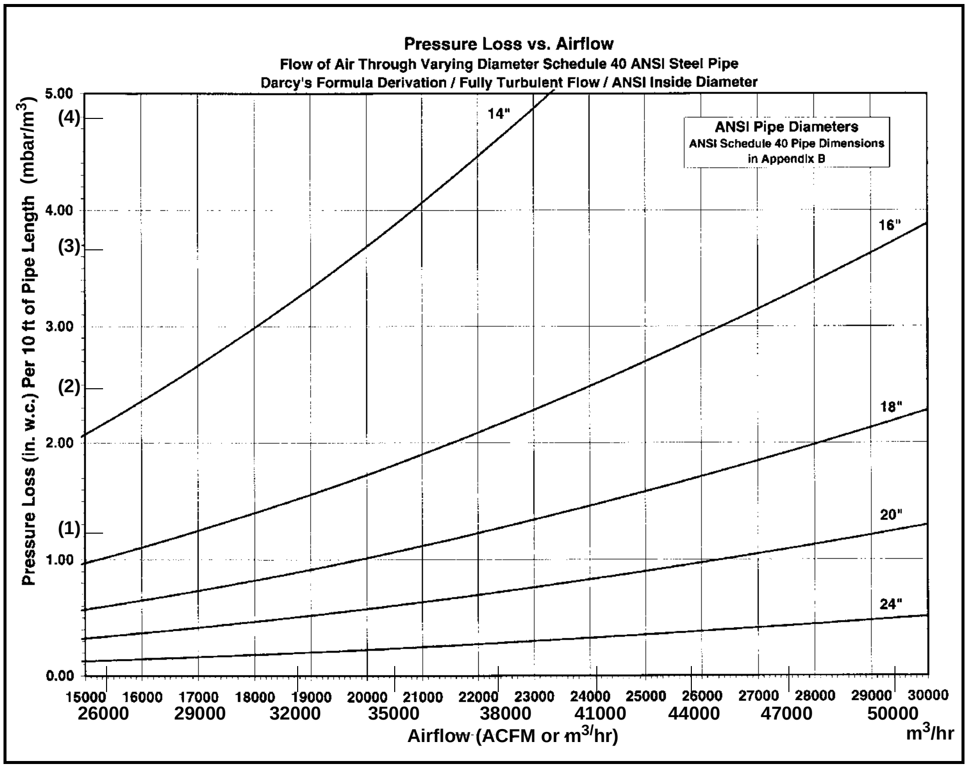
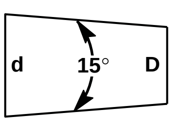
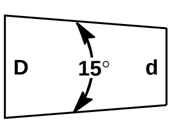
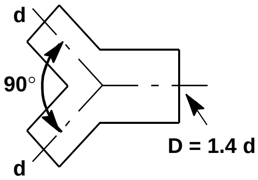
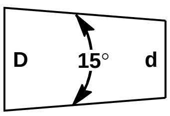


Figure 7-3

Table 7-2 Equivalent Pipe Length of Fittings in Feet (Meter) (Calculated using NTIS Handbook Of Hydraulic Assistance, Form AEC-TR-6630)

FITTINGS		ROUND PIPE DIAMETER											
		3"	4"	5"	6"	8"	10"	12"	14"	16"	18"	20"	24"
 15° DIFFUSER* EPL BASED ON FLOW AT “d”	d/D=1/4 Flanged	3.5 (1)	4.9 (1.5)	6.3 (1.9)	7.9 (2.4)	11.2 (3.4)	14.5 (4.4)	18.3 (5.6)	20.6 (6.3)	24.3 (7.4)	29.7 (9.1)	31.9 (9.7)	39 (11.9)
	d/D=1/2 Flanged	2.4 (0.7)	3.3 (1)	4.3 (1.3)	5.4 (1.6)	7.6 (2.3)	9.9 (3)	12.5 (3.8)	14.0 (4.3)	16.5 (5)	20.3 (6.2)	21.7 (6.6)	27 (8.2)
	d/D=3/4 Flanged	1.1 (0.3)	1.6 (0.5)	2.0 (0.6)	2.5 (0.8)	3.6 (1.1)	4.6 (1.4)	5.8 (1.8)	6.6 (2)	7.8 (2.4)	9.5 (2.9)	10.2 (3.1)	13 (4)
 15° DIFFUSER* EPL BASED ON FLOW AT “D”	d/D=1/4 Flanged	1.3 (0.4)	1.8 (0.5)	2.4 (0.7)	3.1 (0.9)	4.3 (1.3)	5.5 (1.7)	7.0 (2.1)	7.7 (2.3)	8.8 (2.7)	10.7 (3.3)	11.9 (3.6)	14.4 (4.4)
	d/D=1/2 Flanged	1.0 (0.3)	1.4 (0.4)	1.9 (0.6)	2.5 (0.8)	3.5 (1.1)	4.4 (1.3)	5.6 (1.7)	6.2 (1.9)	7.0 (2.1)	8.6 (2.6)	9.6 (2.9)	11.5 (3.5)
	d/D=3/4 Flanged	0.6 (0.2)	0.8 (0.2)	1.1 (0.3)	1.4 (0.4)	2.0 (0.6)	2.5 (0.8)	3.3 (1)	3.6 (1.1)	4.1 (1.2)	5.0 (1.5)	5.6 (1.7)	6.7 (2)

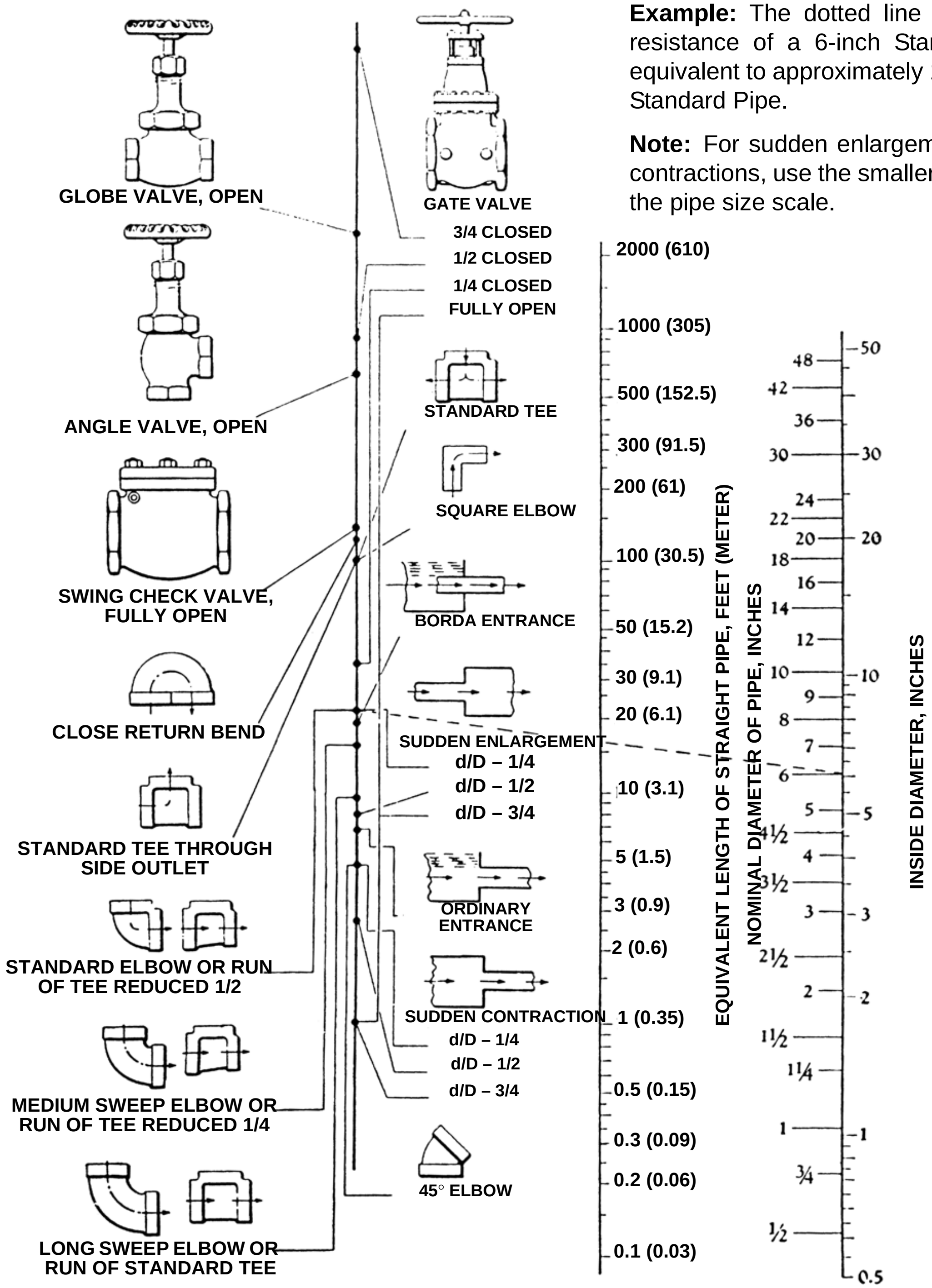
FITTINGS		ROUND PIPE DIAMETER											
		3"	4"	5"	6"	8"	10"	12"	14"	16"	18"	20"	24"
 Y-CONNECTION BASED ON FLOW AT "d"	Flanged	–	–	–	–	–	34.7 (10.6)	43.7 (13.3)	49.1 (15)	58.1 (17.7)	–	–	–
 15° DIFFUSER* EPL BASED ON FLOW AT "D"	Bell Mouth Inlet	0.7 (0.2)	1.0 (0.3)	1.3 (0.4)	1.6 (0.5)	2.3 (0.7)	2.9 (0.9)	3.5 (1.1)	4.0 (1.2)	4.7 (1.4)	5.3 (1.6)	6.1 (1.9)	7.6 (2.3)
	Square Mouth Inlet	6.7 (2)	9.5 (2.9)	13.0 (4)	16.0 (4.9)	23.0 (7)	29.0 (8.8)	35.0 (10.7)	40.0 (12.2)	47.0 (14.3)	53.0 (16.2)	61.0 (18.6)	76.0 (23)

* Minimum restriction is with a 6° diffuser. EPL with a 6° diffuser is approximately half the EPL of a 15° diffuser.

RESISTANCE OF VALVES AND FITTINGS TO FLOW OF FLUIDS IN EQUIVALENT LENGTH OF PIPE

Example: The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16 feet of 6-inch Standard Pipe.

Note: For sudden enlargements or sudden contractions, use the smaller diameter, **d**, on the pipe size scale.



From Crane Co. Technical Paper No. 409. Data based on the above chart are satisfactory for most applications. REPRINTED WITH PERMISSION OF CRANE VALVE GROUP.

Figure 7-4 Equivalent Pipe Length of Fittings in Feet

NOTES

CHAPTER 7

APPENDIX C

PRESSURE CONVERSION TABLE

Table 7-1

PSI	in./H_2O	in./Hg.	mm/H_2O	mm/Hg.	kg/cm^2	bar	mbar	Pa	kPa
0.001	0.0277	0.0020	0.7031	0.0517	0.0001	0.0001	0.0690	6.895	0.0069
0.01	0.2768	0.0204	7.031	0.5171	0.0007	0.0007	0.6895	68.95	0.0690
0.05	1.384	0.1018	35.15	2.586	0.0035	0.0034	3.447	344.7	0.3447
0.10	2.768	0.2036	70.31	5.171	0.0070	0.0069	6.895	689.5	0.6895
0.11	3.045	0.2240	77.34	5.689	0.0077	0.0076	7.584	758.4	0.7584
0.15	4.152	0.3054	105.5	7.757	0.0105	0.0103	10.4	1034	1.034
0.18	4.982	0.3665	126.6	9.309	0.0126	0.0124	12.41	1241	1.241
0.20	5.536	0.4072	140.6	10.34	0.0141	0.0138	13.79	1379	1.379
0.22	6.090	0.4479	154.7	11.14	0.0155	0.0152	15.17	1517	1.517
0.25	6.920	0.5090	175.8	12.93	0.0176	0.0172	17.24	1724	1.724
0.29	8.027	0.5904	203.9	15.00	0.0204	0.0200	19.99	1999	1.999
0.30	8.304	0.6108	210.9	15.51	0.0211	0.0207	21.68	2068	2.068
0.35	9.688	0.7126	246.1	18.10	0.0246	0.0241	24.13	2413	2.413
0.37	10.24	0.7533	260.1	19.13	0.0260	0.0255	25.51	2551	2.551
0.40	11.07	0.8144	281.2	20.69	0.0281	0.0276	27.58	2758	2.758
0.43	11.90	0.8755	302.3	22.24	0.2302	0.0297	29.65	2965	2.965
0.45	12.46	0.9162	316.4	23.27	0.0316	0.0310	31.03	3103	3.103
0.47	13.01	0.9569	330.4	24.31	0.0330	0.0324	32.40	3240	3.240
0.50	13.84	1.018	351.5	25.86	0.0352	0.0345	34.47	3447	3.447
0.54	14.95	1.099	379.7	27.93	0.0380	0.0372	37.23	3723	3.723
0.55	15.22	1.120	386.7	28.44	0.0387	0.0379	37.92	3792	3.792
0.58	16.05	1.181	407.8	29.99	0.0408	0.0400	39.99	3999	3.999
0.60	16.61	1.222	421.8	31.03	0.0422	0.0414	41.37	4137	4.137
0.65	17.89	1.323	457.0	33.61	0.0457	0.0448	44.82	4482	4.482

CONVERSION FACTORS
Note: Conversion factors are rounded.

PSI x 27.68 = in. H_2O PSI x 2.036 = in. Hg	PSI x 703.1 = mm/H_2O PSI x 51.71 = mm/Hg	PSI x 0.0703 = kg/cm^2 PSI x 0.0690 = bar	PSI x 68.95 = mbar PSI x 6895 = Pa	PSI x 6.895 = kPa

Table 7-1

PSI	in./H_2O	in./Hg.	mm/H_2O	mm/Hg.	kg/cm^2	bar	mbar	Pa	kPa
0.69	19.10	1.405	485.1	35.68	00485.	0.0476	47.57	4757	4.757
0.70	19.38	1.425	492.2	36.20	0.0492	0.0483	48.26	4826	4.826
0.75	20.76	1.527	527.3	38.79	0.0527	0.0517	51.71	5171	5.171
0.76	21.04	1.547	534.3	39.30	0.0534	0.0524	52.40	5240	5.240
0.80	22.14	1.629	562.5	41.37	0.0562	0.0552	55.16	5516	5.516
0.85	23.53	1.731	597.6	43.96	0.0598	0.0586	58.60	5860	5.860
0.87	24.08	1.771	611.7	44.99	0.0612	0.0600	59.98	5998	5.998
0.90	24.91	1.832	632.8	46.54	0.0633	0.0620	62.05	6205	6.205
0.94	26.02	1.914	660.9	48.61	0.0661	0.0648	64.81	6481	6.481
0.95	26.30	1.934	667.9	49.13	0.0668	0.0655	65.50	6550	6.550
1.0	27.68	2.036	703.1	51.71	0.0703	0.0690	68.95	6895	6.892
1.5	41.52	3.054	1055	77.57	0.1055	0.1034	103.4	10340	10.34
1.7	47.06	3.461	1195	87.92	0.1195	0.1172	117.2	11720	11.72
2.0	55.36	4.072	1406	103.4	0.1406	0.1379	137.9	13790	13.79
2.2	60.90	4.479	1547	113.8	0.1547	0.1517	151.7	15170	15.17
2.5	69.20	5.090	1758	129.3	0.1758	0.1724	172.4	17240	17.24
3.0	83.04	6.108	2109	155.1	0.2109	0.2068	206.8	20680	20.68
3.2	88.58	6.515	2250	165.5	0.2250	0.2206	220.6	22060	22.06
3.5	96.88	7.126	2461	181.0	0.2461	0.2413	241.3	24130	24.13
4.0	110.7	8.144	2812	206.9	0.2812	0.2758	275.8	27580	27.58
4.3	119.0	8.775	3023	222.4	0.3023	0.2965	296.5	29650	29.65
4.5	124.6	9.162	2164	232.7	0.3164	0.3103	310.3	31030	31.03
4.7	130.1	9.569	3304	243.1	0.3304	0.3240	324.0	32400	32.40
5.0	138.4	10.18	3515	258.6	0.3515	0.3447	344.7	34470	34.47
5.4	149.5	10.99	3797	279.3	0.3797	0.3723	372.3	37230	37.23
5.5	152.2	11.20	3867	284.4	0.3867	0.3792	379.2	37920	37.92
5.8	160.5	11.81	4078	299.9	0.4078	0.3999	399.9	39990	39.99
6.0	166.1	12.22	4218	310.3	0.4218	0.4137	413.7	41370	41.37
6.2	171.6	12.62	4359	320.6	0.4359	0.4275	427.5	42750	42.75
6.5	179.9	13.23	4570	336.1	0.4570	0.4482	448.2	44820	44.82

CONVERSION FACTORS

Note: Conversion factors are rounded.

PSI x 27.68 = in. H_2O PSI x 2.036 = in. Hg	PSI x 703.1 = mm/H_2O PSI x 51.71 = mm/Hg	PSI x 0.0703 = kg/cm^2 PSI x 0.0690 = bar	PSI x 68.95 = mbar PSI x 6895 = Pa	PSI x 6.895 = kPa

Table 7-1

PSI	in./H_2O	in./Hg.	mm/H_2O	mm/Hg.	kg/cm^2	bar	mbar	Pa	kPa
6.9	191.0	14.05	4851	356.8	0.4851	0.4757	475.7	47570	47.57
7.0	193.8	14.25	4922	362.0	0.4921	0.4826	482.6	48260	48.26
7.3	202.1	14.86	5132	377.5	0.5132	0.5033	503.3	50330	50.30
7.5	207.6	15.27	5273	387.9	0.5273	0.5171	517.1	51710	51.71
8.0	221.4	16.29	5625	413.7	0.5625	0.5516	551.6	55160	55.16
8.6	238.0	17.51	6047	444.7	0.6046	0.5929	592.9	59290	59.29
9.0	249.1	18.32	6328	465.4	0.6328	0.6205	620.5	62050	62.05
9.6	265.7	19.54	6750	496.5	0.6749	0.6619	661.9	66190	66.19
10.0	276.8	20.36	7031	517.1	0.7031	0.6895	689.5	68950	68.95
16.0	442.9	32.58	11250	827.4	1.125	1.103	1103	110300	110.3
20.0	553.6	40.72	14060	1034	1.406	1.379	1379	137900	137.9
22.0	609.0	44.79	15470	1138	1.547	1.519	1517	151700	151.7
25.0	692.0	50.90	17580	1293	1.758	1.724	1724	172400	172.4

CONVERSION FACTORS
Note: Conversion factors are rounded.

PSI x 27.68 = in. H_2O PSI x 2.036 = in. Hg	PSI x 703.1 = mm/H_2O PSI x 51.71 = mm/Hg	PSI x 0.0703 = kg/cm^2 PSI x 0.0690 = bar	PSI x 68.95 = mbar PSI x 6895 = Pa	PSI x 6.895 = kPa

NOTES

CHAPTER 8

EXHAUST SYSTEMS

INTRODUCTION

Proper design of an engine exhaust system is important for safe, long lasting service with minimum maintenance. The functions of an engine exhaust system are:

- Safely remove hot exhaust gas and discharge it in a safe area.
- Maintain acceptable noise levels.
- Control exhaust emissions to be within local regulations.

Additionally, exhaust systems are sometimes used to recover exhaust thermal energy in heat exchangers. The exhaust gas may also be recovered for its chemical properties (carbon dioxide) or as an inert gas. Inert exhaust gas is sometimes compressed and injected into an oil well for oil recovery.

The following sections need to be considered when designing and installing the exhaust system.

SECTION 1

EXHAUST PIPING

Selecting the exhaust piping material and strength is very important to get a safe, long lasting exhaust system.

An exhaust explosion is possible if an ignition fails on a natural gas engine. The minimum requirements for the design of the exhaust system should be to contain explosions that could be encountered during the operation of the engine. Waukesha recommends ANSI schedule 10 stainless steel or ANSI schedule 20 carbon steel pipe as a minimum. Stainless steel pipe has greater strength properties at elevated temperatures. Double walled piping and slip joints do not have sufficient strength to contain exhaust explosions and therefore are not allowed for gas engine exhaust systems.

Exhaust Temperature

Exhaust gas temperature from natural gas engines ranges from 600° – 1300°F (316° – 704°C). The exhaust piping is also at this temperature. Insulating the exhaust piping and components is important to avoid fire and personal injury hazards on indoor and some outdoor installations. Insulating the exhaust system also reduces radiant heat in the engine room and allows for more efficient transfer to exhaust heat recovery equipment (when utilized).

Refer to local fire codes for proper insulation material and usage. The safety codes governing installation procedures for exhaust systems are numerous and complex. The designer must consult OSHA for all applications in the United States, and all local and state agencies having jurisdiction in the area in which the installation is located.

Any items which will come in contact with the hot exhaust piping must be suitable for this temperature. Some items to consider are:

- Flexible Connections
- Hanging brackets
- Rollers
- Gasket materials
- Building Walls (where piping passes through)

WARNING

Use high temperature gasket materials and proper room ventilation. Inadequate gaskets can break down allowing poisonous exhaust gas to leak.

Exhaust Velocity

Piping should be sized to keep exhaust velocity less than 12,000 ft/min (60 m/sec). This will keep exhaust restriction and exit noise low.

SECTION 2 THERMAL GROWTH AND EXHAUST FLEXIBLE CONNECTIONS

With exhaust gas temperatures reaching 1300°F, thermal growth can be considerable. The following notes refer to Figure 8-1 on page 8-4.

NOTES:

1. Allow for thermal expansion of the exhaust pipe beyond the engine exhaust flex connection. The Waukesha exhaust flex (when supplied) will accommodate engine thermal expansion but cannot tolerate movement imposed by external thermal growth. Insulated pipes will run hotter and consequently expand more.

   COEFFICIENT OF EXPANSION C_e

   Steel $6.5 \times 10^{-6} \frac{in}{in\ °F}$ $\left(1.17 \times 10^{-5} \frac{mm}{mm°C}\right)$

   Stainless Steel $9.9 \times 10^{-6} \frac{in}{in\ °F}$ $\left(1.7 \times 10^{-5} \frac{mm}{mm°C}\right)$

   Thermal expansion can be calculated with the following formula:

   Equation 1

   $L_e = C_e * L * (T_{exh} - T_{stnd})$

   WHERE:

   Le = Length of pipe expansion (inches or meters)

   Ce = Coefficient of expansion for the material (in/in/ °F or mm/mm/ °C)

   L = Piping length at standard conditions (inches or meters)

   T_{exh} = Exhaust Temperature (°F or °C)

   T_{stnd} = Standard Temperature (°F or °C)

EXAMPLE: How much will 10 ft. of steel pipe expand for an engine with an exhaust temperature of 1060°F based on 60°F ambient.

$C_e = 0.0000065 \frac{in}{in\ {}^\circ F}$

$L = 10\ ft * 12\ in./ft = 120\ in.$

$T_{exh} = 1060^\circ F$

$T_{stnd} = 60^\circ F$

$L_e = 0.0000065\ in./in./{}^\circ F * 120\ in. * (1060^\circ F - 60^\circ F) = 0.78\ in.$

2. An exhaust flexible connection has “spring constants” (lateral, axial, radial, and torsional) that should be considered when engineering the exhaust system. Transmission of forces to the engine exhaust connection must be zero. (Any specific load or bending moment limits shown on an engine installation drawing must not be exceeded.)
3. Design the exhaust system so it will not impose torsional forces on the exhaust flexible connection.
4. The exhaust flexible connection should be designed to allow for flexing caused by engine operation: acceleration, deceleration, starting, and stopping. A Waukesha exhaust flexible connection (when supplied) will accommodate engine vibrations with a solidly mounted unit, but cannot tolerate the additional forces/displacement imposed by mounting on spring isolators. Additional flexible connection capabilities will be required when the unit is mounted on isolators.
5. Consider expected life. Cyclic flexing can lead to premature failure by causing fatigue breakage.
6. Utilize a combination of fixed supports, rollers and flexible connections to provide a well designed exhaust system. See the sketches in Figure 8-1 for additional concepts.
7. Provide water traps/drains to prevent exhaust condensation and/or rain from reaching the engine. This is especially true on long pipe runs. Use rain caps where applicable. Slope piping away from engine.
8. The minimum requirements for the design of the exhaust system should be to contain explosions that could be encountered during the operation of the engine. Waukesha recommends the use of carbon steel schedule 20 pipe as a minimum. Stainless steel schedule 10 pipe is preferred because of its greater strength properties at elevated temperatures. Waukesha does not recommend using double walled piping or slip joints on engine exhausts.
9. Utilize smooth transition to final pipe size when a transition in size is required. Waukesha recommends a diverging angle of 15° for low pressure drop (see sketches for straight and elbowed transitions).
10. Engines with two exhaust outlets combined into one must have a symmetrical design up to and including the point of convergence of the two exhaust streams to produce proper flow and restriction balance. Convergence angle from the center of symmetry must not exceed 45°. The outlet area of the y-connection must be equal to or greater than the sum of the two inlet areas.
11. Size piping and silencer so that exhaust system backpressure, as measured at the engine outlet flange, is less than that indicated in the specifications page in the tech data book.

12. Provide clearance to permit use of a chain hoist for removal of heavy components.

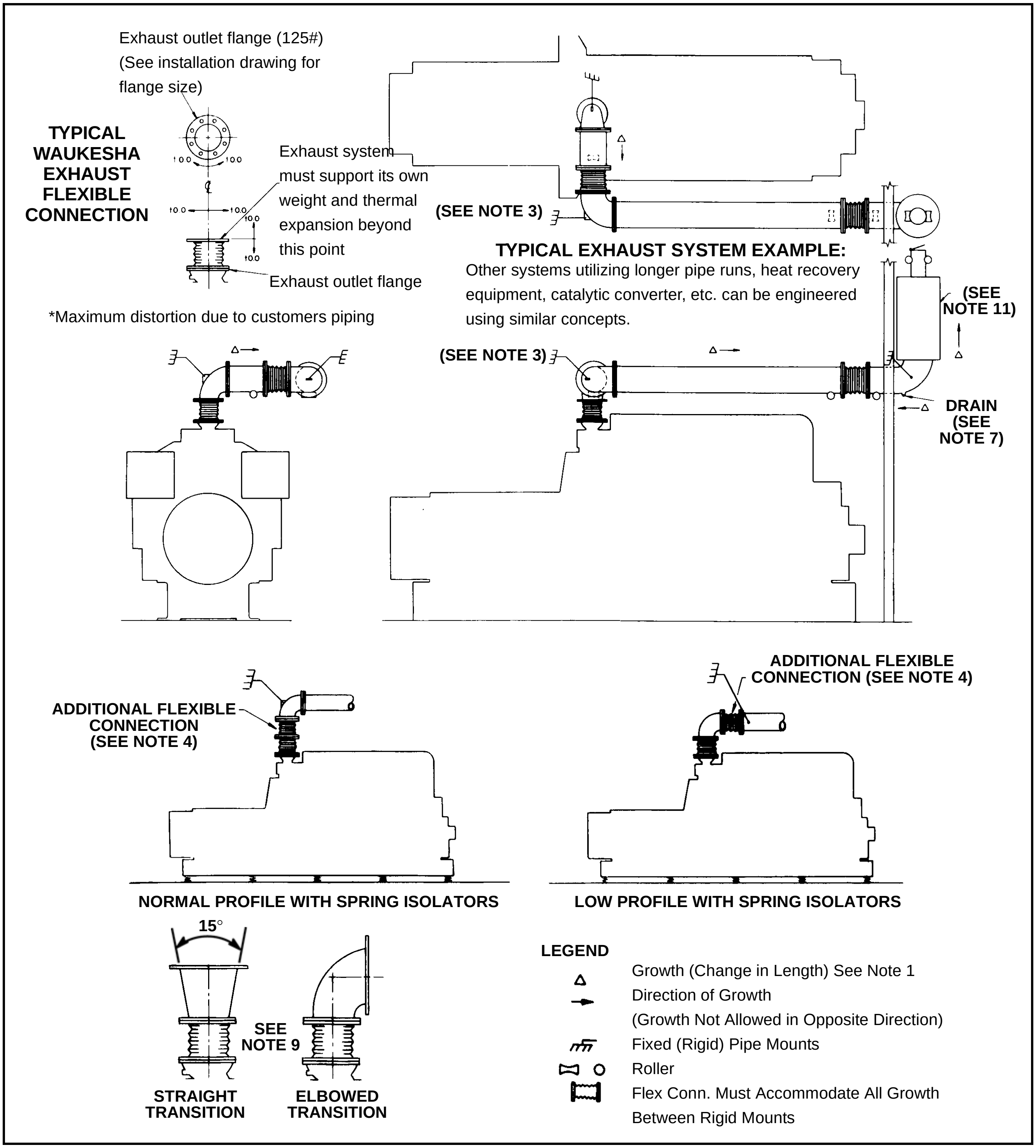


Figure 8-1 Waukesha Installation Schematic

SECTION 3 SOUND ATTENUATION

Engine exhausts are very loud and in most locations require silencing to meet local noise regulations. Numerous silencer designs exist for various levels of sound attenuation and backpressure. Some silencers incorporate exhaust heat recovery. Selecting a silencer for an engine requires knowing local noise level regulations, exhaust flow rate, and allowable engine exhaust backpressure. A silencer supplier can then be contacted to properly specify a silencer.

The exhaust flow rate, temperature and allowable exhaust backpressure information is available for the specific engine model in the Waukesha Technical Data Book. Calculating the entire exhaust system backpressure is addressed later in this chapter.

SECTION 4 EMISSIONS TREATMENT

Emissions regulations for natural gas engines are constantly changing and vary widely among the various air districts. Waukesha offers “Lean Combustion” (GL, GLD, LTD) engines which produce very low exhaust emissions levels. These engines are acceptable in many parts of the United States and the World without further exhaust treatment. Areas where air quality has deteriorated below local air quality standards will often require exhaust emissions treatment.

Emissions treatment for “Stoichiometric” engines (G, GSI, GSID) commonly use a catalytic converter to oxidize or reduce pollutants to harmless, naturally occurring compounds. A precise electronic control of air fuel ratio is necessary to maintain the correct chemical concentrations in the exhaust for optimum oxidation and reduction reactions. Waukesha offers electronic controls for this function.

Emissions treatment of “Lean Combustion” (GL, LT, GLD, LTD) engines normally consists of oxidizing catalysts to oxidize carbon monoxide and hydrocarbons in the exhaust. This application does not require precise electronic control.

Crankcase breathers are often considered a pollution source. Permits for both the exhaust system and breather system can be very time consuming and expensive. The breather emissions can often be piped into the exhaust system downstream of catalysts and heat recovery equipment, or use a closed breather system. The exhaust stack is then considered as the only pollution source.

Emissions requirements must be determined early in a project so permits can be applied for and the correct engine configuration and/or emissions treatment can be selected.

SECTION 5 EXHAUST PURGING

To prevent explosions and personal injury the engine and the exhaust system are purged by cranking the engine for several seconds before the ignition is turned on and the fuel valves are opened. The purge volume of the engine is approximately its displacement for every two revolutions. In case the volume of the exhaust system is such that it will not be purged several times by the cranking of the engine the customer has to use as alternative means to purge the exhaust system.

SECTION 6

EXHAUST BACKPRESSURE

Exhaust system backpressure limits are established for each engine model. Backpressure limits are available in the Waukesha Technical Data Book Engine Specifications section for the particular model of interest. Exceeding these backpressure limits can cause horsepower output to decrease, fuel consumption to increase, and breather backpressure to increase, causing high crankcase pressure. High crankcase pressure will cause numerous oil leaks and may cause operational problems.

Exhaust system backpressure can be calculated using the procedure described below:

Exhaust Restriction Calculation

Sizing exhaust piping and exhaust components (silencers, heat recovery equipment, catalytic converters) will require knowing the exhaust flow rate and temperature at the maximum operating speed and load. The flow rate and temperature must be considered for any intermittent or overload conditions required in the specification.

Piping restriction depends on the pipe diameter, pipe length, number of elbows and transitions, and piping material used. The following procedure will help determine piping restriction.

1. Determine exhaust volume flow rate (ft^3/min or m^3/hr) for the specific engine model from the heat rejection sections in the Technical Data Manual. If exhaust flow is given in terms of mass flow, a conversion is available in the notes section of the heat balance.
2. Calculate exhaust velocity (V) for each pipe size used:

   Equation 2

$$V\,(\text{FPM}) = \frac{\text{Flow}\,(\text{ft}^3/\text{min})}{\text{Pipe inside area}\,(\text{ft}^2)}$$

or

$$V\left(\frac{\text{m}}{\text{sec}}\right) = 277.8 \times \frac{\text{Flow}\,(\text{m}^3/\text{hr})}{\text{Pipe inside area}\,(\text{mm}^2)}$$

$$\text{Pipe Area}\,(\text{ft}^2) = \{\text{Diameter (inch)}\}^2 \times \pi/4 \times 0.00694\ \text{ft}^2/\text{inch}^2$$

Inside diameter and area for common pipe sizes are given in Table 8-1.

Pipe Areas for ANSI Schedule 40 Pipe

Table 8-1 ANSI Schedule 40 Pipe Dimensions

ANSI PIPE DIAMETER (IN.)	ID (in.)	ID (mm)	AREA (inch^2)	AREA (mm^2)	AREA (ft^2)
1.5	1.61	40.894	2.04	1312.77	0.0142
2	2.067	52.502	3.36	2163.80	0.0233
2.5	2.344	59.538	4.32	2782.61	0.030
3	3.068	77.927	7.39	4767.03	0.0513
4	4.026	102.260	12.73	8208.89	0.0884
5	5.047	128.194	20.01	12900.42	0.139
6	6.065	154.051	28.89	18629.39	0.201
8	7.981	202.717	50.03	32259.06	0.347
10	10.02	254.508	78.85	50847.84	0.5476
12	12	304.800	113.1	72928.89	0.7854
14	13.25	336.550	137.9	88913.73	0.9575
16	15.25	387.350	182.7	117781.42	1.268
18	16.88	428.752	223.7	144305.20	1.553
20	18.81	477.774	278.0	179190.38	1.931
22	21.00	533.400	346.4	223344.71	2.405
24	22.60	574.040	401.2	258674.71	2.786

3. Determine pressure loss (P_L) per 10 ft (3 m) of pipe for each velocity and pipe size from Figure 8-3 and Figure 8-4.

4. Determine the equivalent pipe length (EPL) for all fittings of each pipe size:

   Figure 8-5 and Table 8-3 give equivalent pipe length in feet for various pipe fittings. For each pipe size sum the EPLs and add them to the total length of straight pipe of that size to find the total of each pipe size. Exit loss does not need to be considered in these calculations.

Table 8-2 Pipe Wall Thickness

WALL THICKNESS OF SCHEDULE 40 PIPE PER PIPE SIZE		
PIPE ID (IN.)	THICKNESS (IN.)	THICKNESS (mm)
2	0.154	3.9
4	0.237	6.0
6	0.280	7.1
8	0.322	8.2
10	0.365	9.3
12	0.406	10.3
14	0.438	11.1
16	0.500	12.7
20	0.594	15.1
24	0.688	17.5
WALL THICKNESS OF SCHEDULE 20 PIPE PER PIPE SIZE		
8	0.250	6.4
10	0.250	6.4
12	0.250	6.4
14	0.312	7.9
16	0.312	7.9
18	0.312	7.9
20	0.375	9.5
22	0.375	9.5
24	0.375	9.5

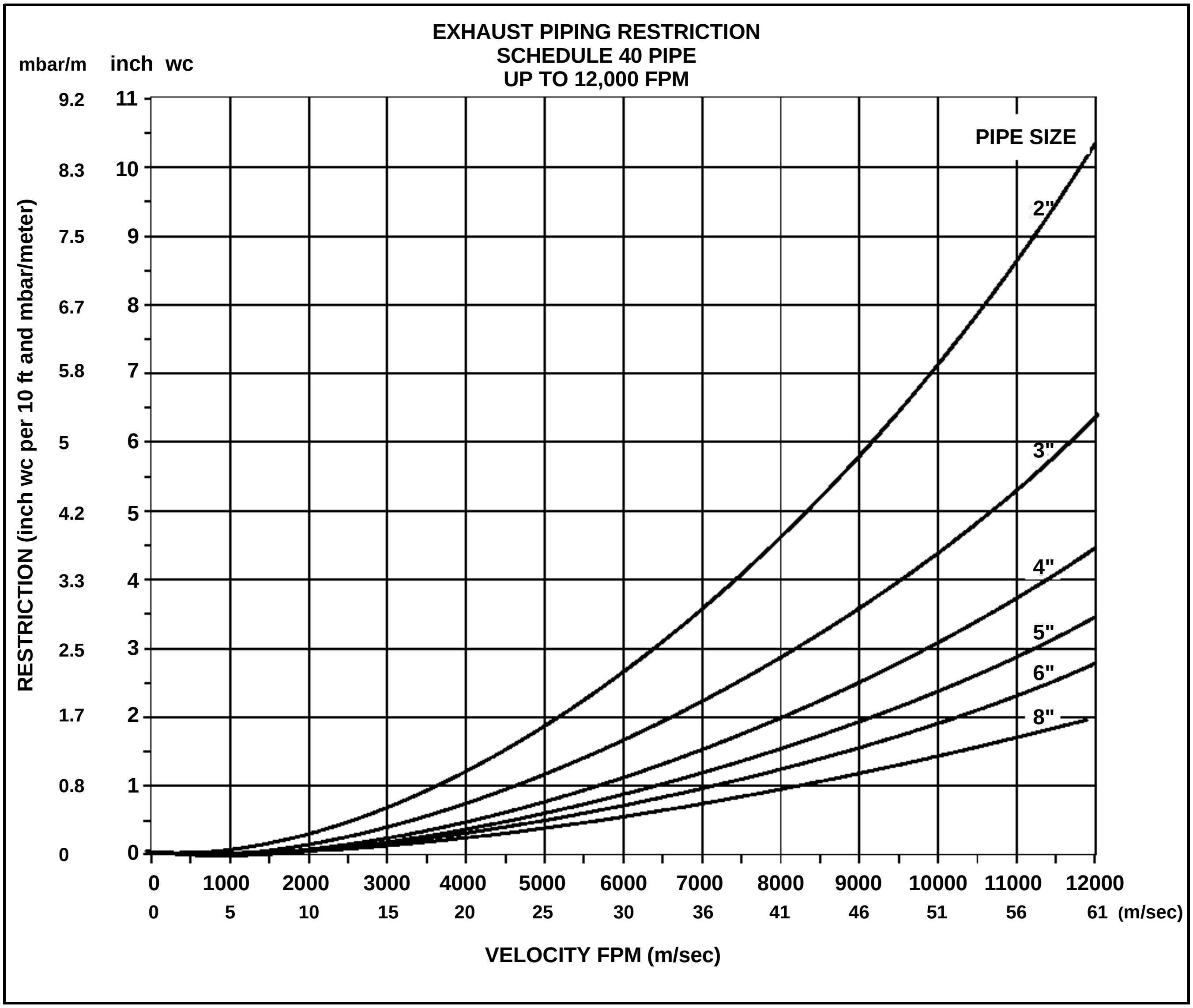


Figure 8-2 Restriction vs. Velocity for Pipe Diameter Up to 8 in.

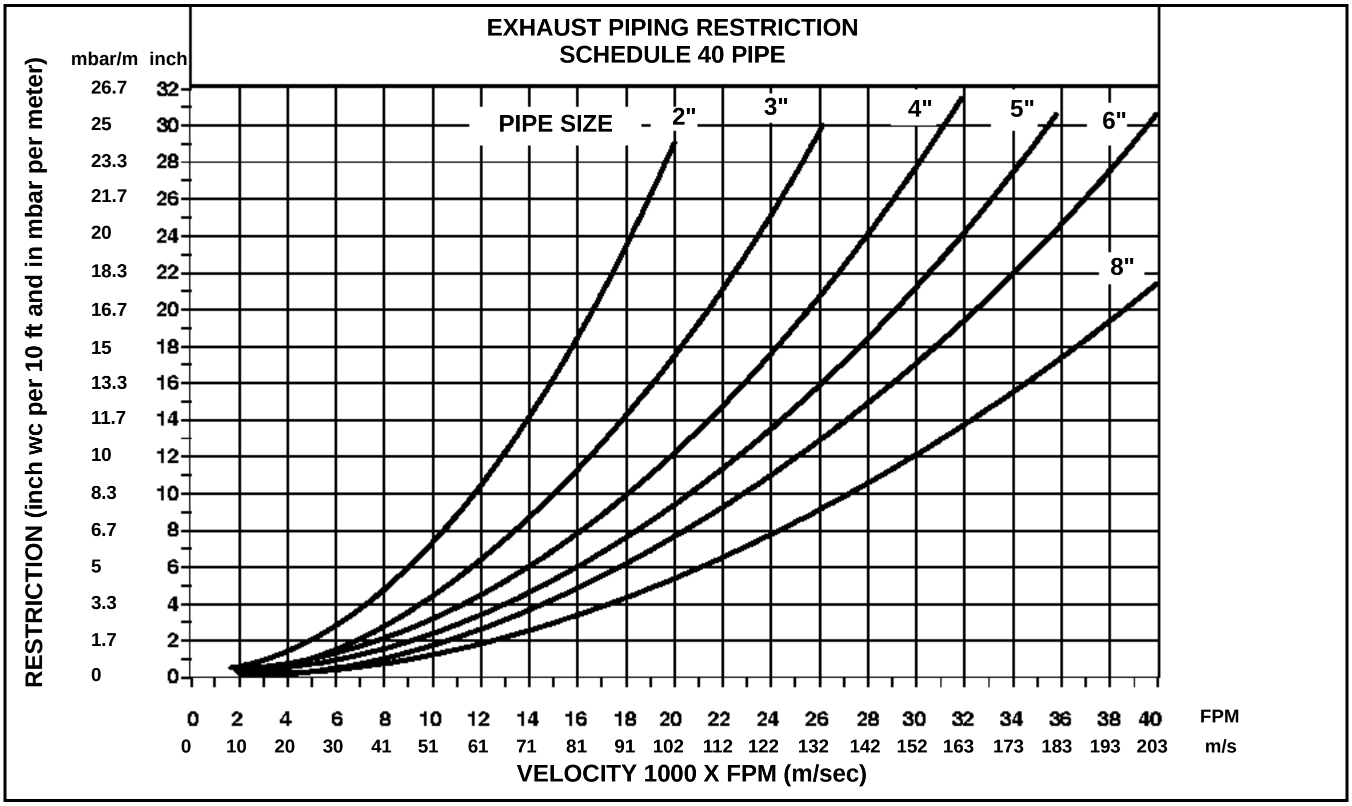


Figure 8-3 Restriction vs. Velocity for Pipe Diameter Up to 8 in. (High Speed)

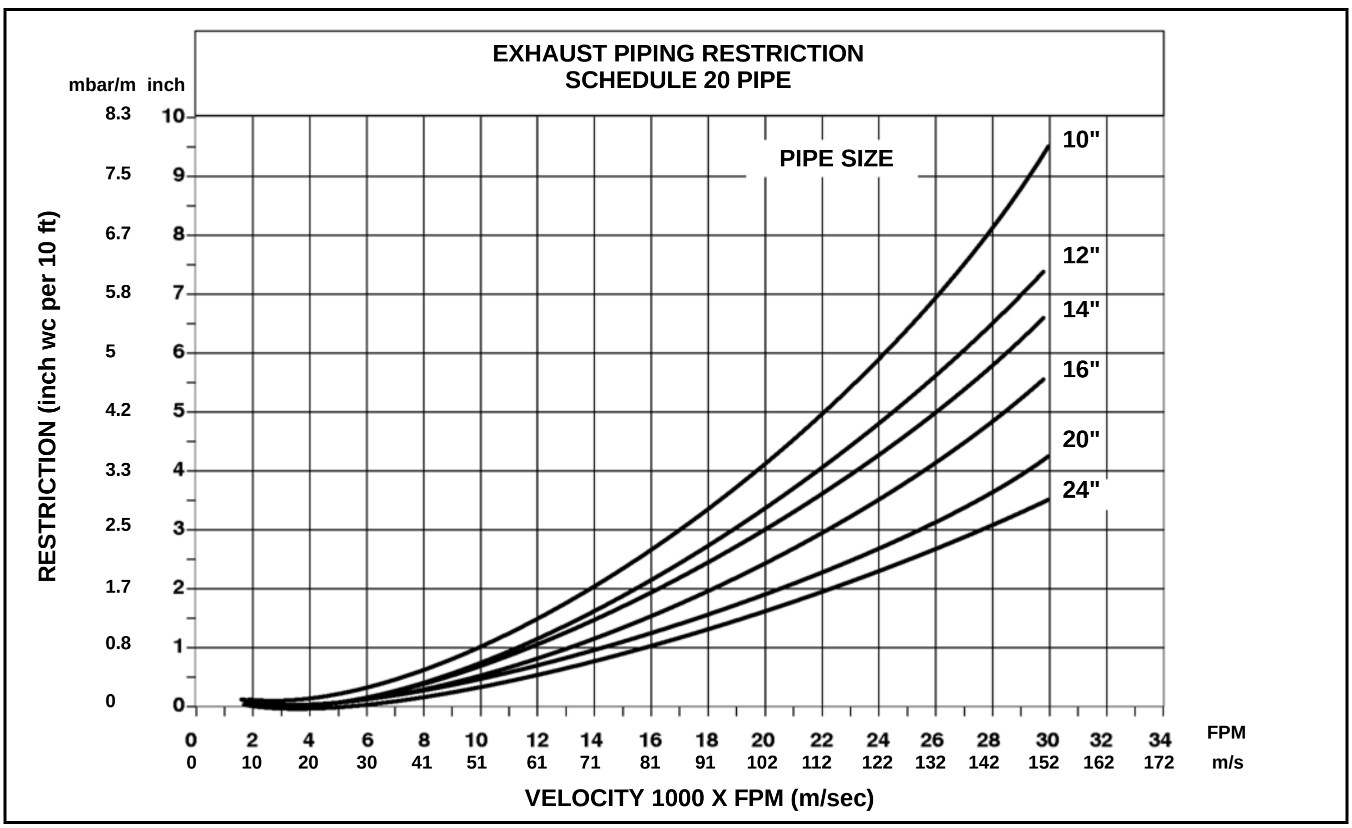
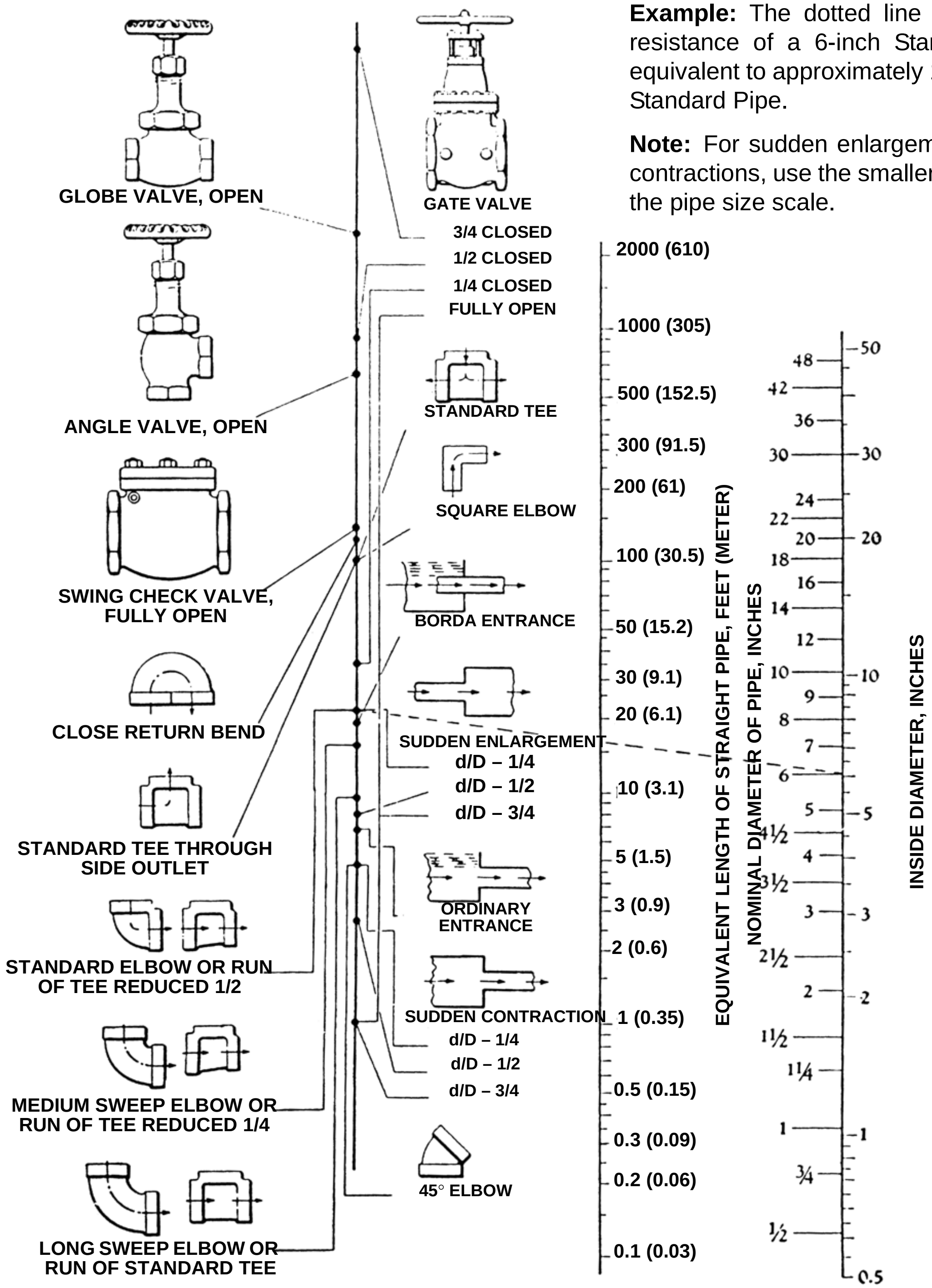


Figure 8-4 Restriction vs. Velocity for Pipe Diameter Up to 24 in.

RESISTANCE OF VALVES AND FITTINGS TO FLOW OF FLUIDS IN EQUIVALENT LENGTH OF PIPE

Example: The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16 feet of 6-inch Standard Pipe.

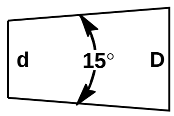
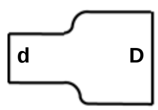
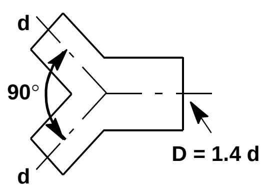
Note: For sudden enlargements or sudden contractions, use the smaller diameter, **d**, on the pipe size scale.



From Crane Co. Technical Paper No. 409. Data based on the above chart are satisfactory for most applications. REPRINTED WITH PERMISSION OF CRANE VALVE GROUP.

Figure 8-5 Equivalent Pipe Length of Fittings in Feet

Table 8-3 Equivalent Pipe Length of Fittings in Feet (Meter) (Calculated using NTIS Handbook Of Hydraulic Assistance, Form AEC-TR-6630)

ADDITIONAL FITTINGS		PIPE SIZE (IN.)											
		3	4	5	6	8	10	12	14	16	18	20	24
 15° DIFFUSER* EPL BASED ON FLOW AT "d"	d/D=1/4 Flanged	3.5 (1)	4.9 (1.5)	6.3 (1.9)	7.9 (2.4)	11.2 (3.4)	14.5 (4.4)	18.3 (5.6)	20.6 (6.3)	24.3 (7.4)	29.7 (9.1)	31.9 (9.7)	39 (11.9)
	d/D=1/2 Flanged	2.4 (0.7)	3.3 (1)	4.3 (1.3)	5.4 (1.6)	7.6 (2.3)	9.9 (3)	12.5 (3.8)	14.0 (4.3)	16.5 (5)	20.3 (6.2)	21.7 (6.6)	27 (8.2)
	d/D=3/4 Flanged	1.1 (0.3)	1.6 (0.5)	2.0 (0.6)	2.5 (0.8)	3.6 (1.1)	4.6 (1.4)	5.8 (1.8)	6.6 (2)	7.8 (2.4)	9.5 (2.9)	10.2 (3.1)	13 (4)
 SUDDEN EXPANSION BASED ON FLOW AT "d"	d/D=1/4 Flanged	13.2 (4)	18.7 (5.7)	24.3 (7.4)	30.1 (9.2)	42.7 (13)	56 (17.1)	70 (21.3)	79 (24)	93 (28)	114 (35)	122 (37)	151 (46)
	d/D=1/2 Flanged	8.5 (2.6)	12.1 (3.7)	15.7 (4.8)	19.5 (5.9)	27.6 (8.5)	35.9 (11)	45.4 (14)	51 (15.5)	60 (18)	74 (23)	79 (24)	97 (30)
	d/D=3/4 Flanged	2.9 (0.9)	4.2 (1.3)	5.4 (1.6)	6.7 (2.0)	9.5 (2.9)	12.3 (3.7)	15.5 (4.7)	17.6 (5.4)	20.8 (6.3)	25.4 (7.7)	27.2 (8.3)	34 (10.4)
 Y-CONNECTION BASED ON FLOW AT "d"	Flanged	—	—	—	—	—	34.7 (10.6)	43.7 (13.3)	49.1 (15)	58.1 (17.7)	—	—	—

NOTE: **Minimum restriction is with a 6° diffuser. EPL with a 6° diffuser is approximately 1/2 the EPL of a 15° diffuser.*

5. Calculate the exhaust gas density correction:

   Equation 3

   $D_C = L_c * 520 / (460 + T_{exh})$ or $D_C = L_c * 273 / (273 + T_{exh}\ °C)$

   WHERE:

   D_c = density correction

   L_c = lambda correction,
   for Lambda = 0.97 to 1.06, L_c = 0.95 (rich burn)
   for Lambda = 1.53 to 2.0, L_c = 0.97 (lean burn)

   T_{exh} = exhaust temperature ° F (° C)

6. Calculate the total piping restriction R_P for each pipe size:

   Equation 4

$$R_{P1} = \frac{P_{L1}(\text{psi})}{10\ \text{ft.}} \times D_C \times \text{EPL (ft)} \quad \text{or} \quad R_{P1} = \frac{P_{L1}(\text{mbar})}{\text{m}} \times D_C \times \text{EPL (m)}$$

7. Calculate the total exhaust system restriction:

Equation 5

$$R_T = R_{P1} + R_{P2} + R_{P3} + R_s + R_A$$

WHERE:

R_T= total restriction (psi or mbar)

$R_{P1,2,3}$ = piping restriction for various pipe sizes (psi or mbar)

R_S = silencer restriction (psi or mbar)

R_A = accessories (catalyst, boiler, etc.) restriction (psi or mbar)

SAMPLE PROBLEM

A 12V 275GL+ engine will operate 3624 BHP @ 1000 RPM with 2.5 in. Water Column silencer restriction. The designer has planned an exhaust system as shown in Figure 8-6.

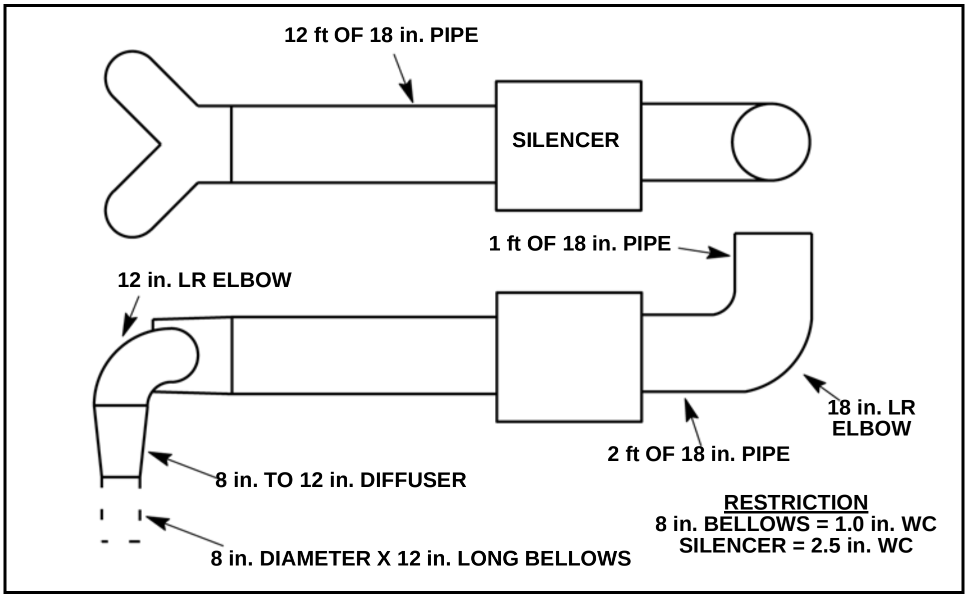


Figure 8-6 Schematic for Sample Problem

Table 8-4

EXHAUST SYSTEM RESTRICTION CALCULATION	REFERENCE
1. Determine exhaust flow: exhaust mass flow = 43,069 Lb/Hr exhaust volume flow: exhaust temperature = 829°F $F_v = \frac{43{,}096 \text{ lb/hr} \times (829°F + 460°)}{2275} = 24{,}418 \text{ CFM}$ per bank *Fv = 24,418/2 = 12,209 CFM*	S-9062-10 Heat Rejection Section of Technical Data or site-specific data from EngCalc
2. Calculate the exhaust velocity for each pipe size.	Equation 2
Velocity in 8" pipe (per bank) $V_8 = \frac{12{,}209 \text{ ft}^3\text{/min}}{0.347 \text{ ft}^2} = 35{,}184 \text{ ft/min}$ Velocity in 12" pipe (per bank) $V_{12} = \frac{12{,}209 \text{ ft}^3\text{/min}}{0.7854 \text{ ft}^2} = 15{,}545 \text{ ft/min}$ Velocity in 18" pipe (whole engine) $V_{18} = \frac{24{,}418 \text{ ft}^3\text{/min}}{1.553 \text{ ft}^2} = 15{,}723 \text{ ft/min}$	Table 8-1
3. Determine pressure loss per 10 feet of pipe: Pressure loss in 8" pipe @ 35,184 ft/min $P_{L8} = 17'' \text{ wc/10 ft}$	Figure 8-3
Pressure loss in 12" pipe @ 15,545 ft/min $P_{L12} = 1.8'' \text{ wc/10 ft}$	Figure 8-4
Pressure loss in 18" pipe @ 15,723 ft/min $P_{L18} = 1.5'' \text{ wc/10 ft}$	Figure 8-4
4. Determine the equivalent pipe length: EPL_8 2(8" to 12" diffuser) 2(5.04ft)= 10.1 ft	Table 8-3
$EPL_{12} = 2(12'' \text{ LR elbow}) + (\text{Y-Connection})$ $= 2(20 \text{ ft}) + 43.7 \text{ ft} = 83.7 \text{ ft}$	Table 8-3 Figure 8-5
$EPL_{18} = (12 \text{ ft} + 2 \text{ ft} + 1 \text{ ft}) + (18'' \text{ LR elbow})$ $= 15 \text{ ft} + 31 \text{ ft} = 46 \text{ ft}$	Figure 8-5
5. Calculate the exhaust gas density correction: $D_2 = 0.97 \times 520/(460 + 829) = 0.391$	Equation 3
6. Calculate the total piping restriction R_p for each pipe size: $R_{P1} = 17\text{" wc/10 ft} \times 0.391 \times 10.1 \text{ ft} = 6.71\text{" wc}$ $R_{P2} = 1.8\text{" wc/10 ft} \times 0.391 \times 83.7 \text{ ft} = 5.89\text{" wc}$ $R_{P3} = 1.5\text{" wc/10 ft} \times 0.391 \times 46 \text{ ft} = 2.70\text{" wc}$	Equation 4
7. Calculate the total exhaust system restriction (R_S = 2.5" wc and R_B = 1.0" wc): $R_T = 6.71\text{" wc} + 5.89\text{" wc} + 2.70\text{" wc} + 2.5\text{" wc} + 2\,(1.0\text{" wc}) = 19.8\text{" wc}$	Equation 5

NOTE: *The calculations in metric units follows the same guidelines as the sample calculation in English units.*

SECTION 7

INSTALLATION CONCERNS

Exhaust Discharge

The end of the exhaust line must be designed to keep out rain water, dust and dirt. Vertical discharge outlets should have a rain cap to prevent moisture, dust and dirt from entering while an engine is shutdown. Exhaust discharge and flow path must be away from radiators and engine air intakes. Miter cut exhausts will reduce exit noise.

Exhaust Gas

Exhaust gas is poisonous and must be discharged to a harmless location. Do not discharge gases near the engine air intake system, ventilation ducts, windows or buildings. The discharge location must have sufficient natural ventilation to carry away the exhaust gas preventing an unhealthy concentration. Observe safety and other applicable codes.

The discharge must also be at a safe distance from flammable waste dump sites and fuel storage areas. Observe local fire codes and other applicable codes.

Moisture Traps and Drains

Exhaust gas contains 10 – 20% water in the form of steam. During startup of a cold system large amounts of water will condense and collect in low spots of the exhaust piping which can result in corrosion. Moisture traps with drains in these low spots will provide a convenient way for removing this water. Many silencer and heat recovery equipment manufacturers provide traps and drains in their equipment.

Common Exhaust Systems

Using a single exhaust system fed by several engines is discouraged by Waukesha. An engine which is shutdown becomes a path for exhaust gas to leak. This exhaust gas will condense water in the shutdown engine causing hydraulic lock, valve sticking, ring sticking, rust, and other engine damage. The exhaust leak path could cause poisonous exhaust gas to collect in an engine room.

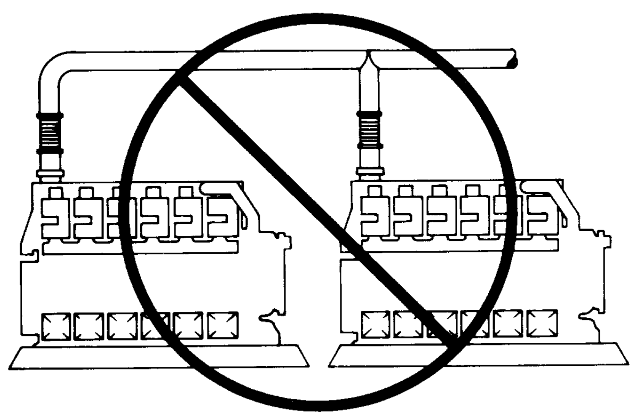


Figure 8-7 Common Exhaust System

Ducting

Never run hot exhaust pipes near flammable materials or fuel supply lines.

Explosion Relief Valves

Waukesha gas engines recommends the use of an explosion relief valve such as a rupture disk to prevent possible damage caused by exhaust explosions. Valve design must provide a leak-free seal. This is especially important when a catalyst is being used as any leaking could reduce the effectiveness of the NOx reduction in the catalyst. Consult with the catalyst supplier for recommendations for your specific application.

NOTICE **Never discharge engine exhaust into a brick, tile, or cement block chimney, or a similar structure. Exhaust pulsations could cause severe structural damage.**

SECTION 8

EXHAUST SYSTEM CHECKLIST

1. Engine exhaust flow______ CFM (m^3/hr) @ ____ °F (°C) @_____ in H_2O (mbar) max restriction.
2. Exhaust system length ______ ft (m)
3. Quantity of elbows in exhaust system ______
4. Exhaust system equivalent pipe length ______ ft (m)
5. Diameter of exhaust pipe ______
6. Silencer restriction ______ in H_2O (mbar)
7. Catalyst restriction ______ in H_2O (mbar)
8. Exhaust system restriction ______ in H_2O (mbar)
9. Exhaust gas velocity ______ FPM (m/sec) (should be < 12,500 FPM (60 m/sec)
10. Exhaust system thermal growth ______ in. (m)

Checklist for Exhaust Systems

1. Size piping for pressure drop / restriction ______________________________
2. Size exhaust silencer for noise and flow requirements.___________________
3. Does the piping require insulating?__________________________________
4. Does the piping meet latest edition of Waukesha spec S-8242? __________
5. Is a pressure relief valve required? _________________________________
6. Has the relief valve been piped to a safe location?_____________________
7. Are flexes installed in the proper locations? ___________________________
8. Has the piping been supported properly – independent of engine? ________
9. Is the piping growth away from the engine? ___________________________
10. Is a catalyst required? ____
11. Has the catalyst been sized correctly? ________________________________
12. Has the exhaust outlet been located to minimize noise? ________________
13. Are all transitions smooth? ____
14. Has the exhaust outlet been designed to keep out rain, dust and dirt?______

15. Has the exhaust been piped to a safe location? ______________________
16. Has the exhaust system been designed to withstand an exhaust explosion? ____
17. Are all elbows long radius? ______________________________________

NOTES

CHAPTER 9

FUEL SYSTEMS

INTRODUCTION

Proper design and installation of the fuel system is required to meet the expected performance and operation of the engine. The information in this chapter will help with understanding the function of the fuel system and properly using it.

SECTION 1

EXPLOSION SAFETY

Preventing Gas Leakage

Air leaking into gaseous fuels can create explosive mixtures. This can cause explosions resulting in material damage, severe personal injury and death. To prevent this the following guidelines should be observed.

Engine specific fuel gas supply should automatically be closed in a shut-down mode with a fail safe shut-off valve located outside the engine room (before gas pressure control equipment). By isolating fuel gas outside, the gas quantity in the engine room is limited and the likelihood of getting dangerous concentration of engine fuel inside the engine room is very low. Also, fuel gas piping should be welded with limited amount of flange joints to minimize the leaking possibility in the fuel gas piping.

To minimize the risk of explosions, gas sensors for the fuel and hazardous gas are to be installed in the engine room. The engine is to be stopped and the fuel train is to be depressurized when gas is detected.

Forced ventilation must be used to minimize the risk of creating a hazardous environment. If a ventilation failure occurs, the engine specific fuel supply is to be closed and the fuel gas piping to the engine is to be depressurized.

An interlock in the control system must secure that the engine cannot be started without sufficient ventilation.

Ventilation can be off in a stand-by mode. (This means for instance that there is no need for heating ventilation air in cold ambient temperatures, which would contribute to operational costs of the power plant.)

SECTION 2

COMBUSTION TYPES

There are two common types of combustion in gaseous fueled engines, each to accomplish different goals. These combustion types are:

Stoichiometric Combustion (Rich Burn)

Air and fuel are mixed at a ratio that during combustion completely consumes all the fuel and all the oxygen. This ratio produces the highest amount of power for the air consumed resulting in fast load response. It is often referred to as “rich burn” because it operates at a richer air/fuel ratio than a “lean combustion” fuel system.

Lean Combustion

Lean combustion is when excess air (50% or more than required) is delivered to the combustion chamber. The oxygen in this extra air does not burn but merely passes through as an inert. The inert oxygen and nitrogen absorbs heat of combustion resulting in lower temperatures in the combustion chamber and exhaust parts, as well as lower fuel consumption.

SECTION 3 ENGINE FUEL SYSTEMS

Mechanical Systems

Waukesha uses a variety of fuel systems on the engines offered to best meet the requirements of a project. These fuel systems are:

– Naturally Aspirated

– Turbocharged Blow-Thru

– Turbocharged Draw-Thru

– Lean Combustion Prechamber

A brief description of each system follows.

Naturally Aspirated

Air is drawn through an air cleaner and enters the mixing valve assembly (carburetor). Fuel pressure to the carburetor is regulated at approximately 5" wc (inches water column) above intake air pressure (atmospheric pressure less the air filter restriction). This fuel mixes with air in the carburetor, then travels to the intake manifold and combustion chambers (see Figure 9-1).

Naturally aspirated engines typically have stoichiometric (rich burn) combustion.

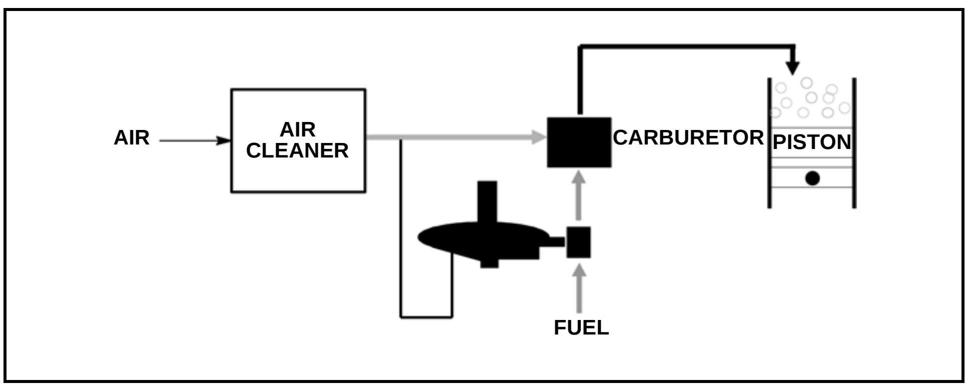


Figure 9-1 Naturally Aspirated Engine

Turbocharged Blow-Thru

Air enters the turbocharger at atmospheric pressure and is boosted to a higher pressure to allow higher horsepower from the engine. Fuel is mixed with this pressurized air downstream of the intercooler. Fuel pressure to the carburetor must be approximately 5" wc higher than the air pressure in the carburetor to get proper mixing. The regulator senses air pressure to control at 5" wc gas/air. The fuel supply to the regulator must be higher than the boosted air pressure in order to supply 5" wc gas/air (see Figure 9-2).

Turbocharged blow-thru engines are available in stoichiometric (rich burn) and lean combustion.

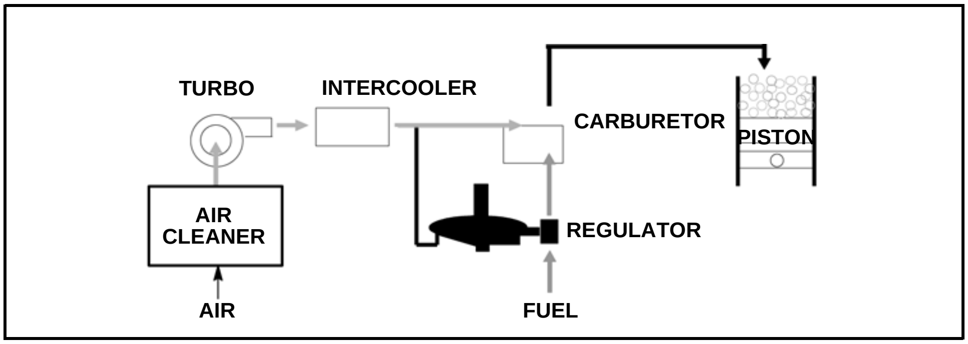


Figure 9-2 Turbocharged Engine

Turbocharged Draw-Thru

The draw-thru system mixes air and fuel before the charge gets pressurized in the turbocharger. With this system a much lower fuel pressure can be supplied to the regulator (see Figure 9-3).

Turbocharged draw-thru engines are available in stoichiometric (rich burn), and lean combustion (open chamber).

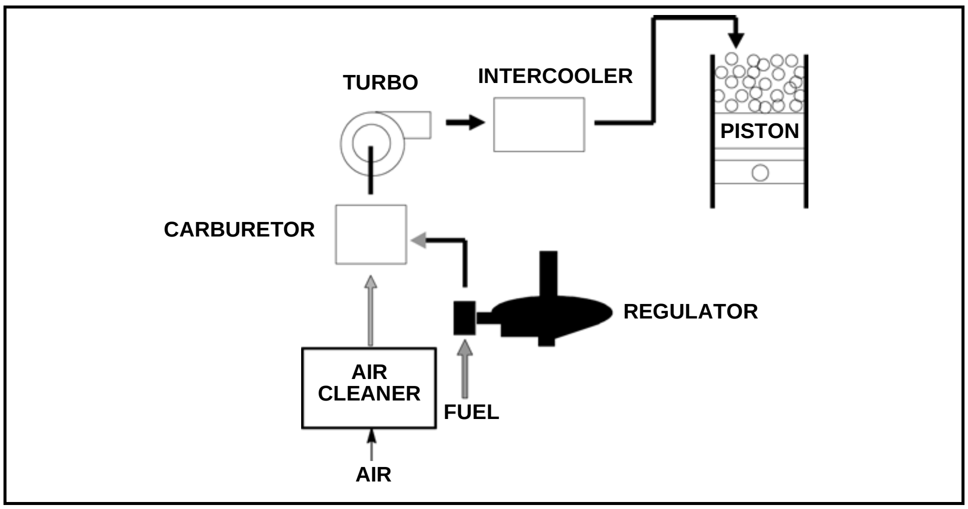


Figure 9-3 GSID Rich Burn Turbocharged Draw-Thru Fuel System

Lean Combustion Prechamber

A prechamber fuel system is used when the combustion charge contains more than 60% excess air and generally with bore sizes above 152 mm. The prechamber regulator maintains gas pressure over intake manifold pressure to obtain a stoichiometric mixture in the prechamber. The stoichiometric mixture ignites easily and delivers a strong torch into the main chamber to ignite the very lean mixture (see Figure 9-4).

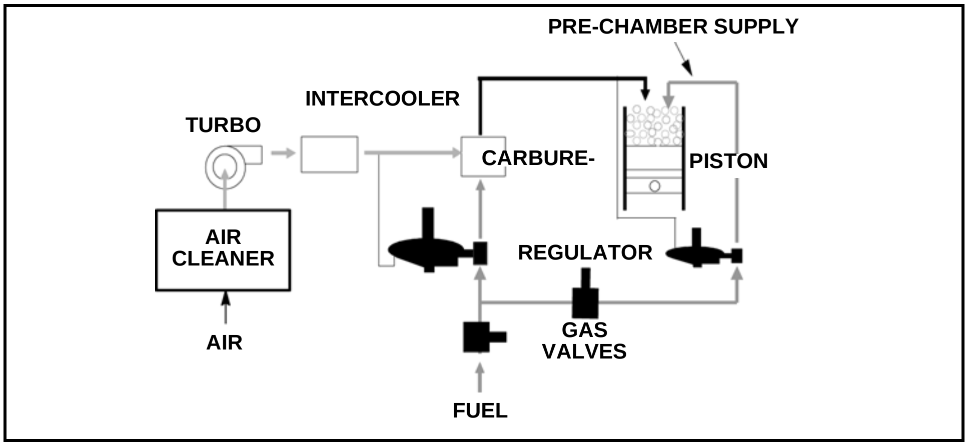


Figure 9-4 Pre-Chamber Lean Combustion Engine (Blow-Thru)

Electronic Air/Fuel Ratio Controls

In addition to the basic mechanical fuel systems, there are electronic air/fuel ratio controls which can be applied to rich burn and lean burn combustion engines. The electronic air/fuel ratio controls measure oxygen content in the exhaust gas and adjust the gas regulator pressure setting. The change in regulator pressure setting causes a change in the air/fuel ratio (see Figure 9-5).

Electronic air/fuel ratio controls compensate for large changes in fuel temperature, ambient air temperature, pressure and fuel heating value, which could affect engine emissions, fuel consumption, and power. On stoichiometric engines this control will maintain a very precise air/fuel ratio which is required for proper chemical reactions in a three-way catalytic converter.

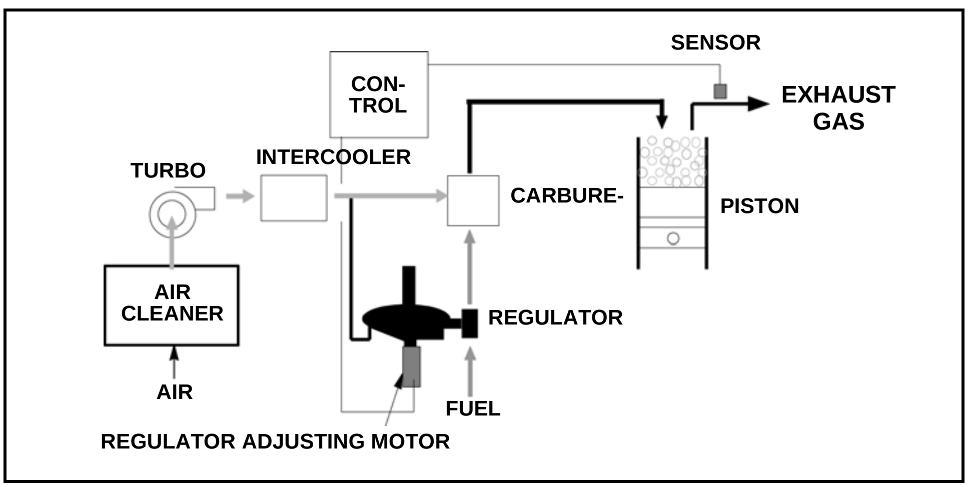


Figure 9-5 Electronic Air/Fuel Ratio Control

SECTION 4 BASIC CARBURETOR TYPES

Currently Waukesha uses two basic types of carburetors. One is manufactured by IMPCO. The other is a venturi type carburetor. The venturi type has no moving parts and works strictly on the venturi principle (see Figure 9-6).

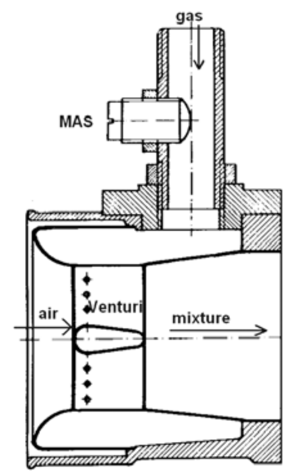


Figure 9-6 Venturi Type Carburetor

WARNING

Never operate the engine without a positive fuel shutoff valve installed in the gas train. The valve is used to stop fuel from leaking in the engine and exhaust system when the engine is not running.

Air/fuel ratio is determined by the dimensions of the venturi and the area of the gas holes in the venturi wall. Fine tuning is controlled by the MAS (Main Adjusting Screw). The venturi type carburetor has low restriction and low service requirements. It requires the gas pressure to be close to the air pressure. This carburetor has no fuel valve and therefore always requires a positive fuel shut off valve in the gas train to stop the fuel from leaking in the engine and exhaust system when the engine is not running.

The IMPCO carburetor uses the venturi effect and a spring to position a fuel valve (see Figure 9-7).

The spring and the valve characteristics can be changed. This type of carburetor requires approximately 5" wc gas/air.

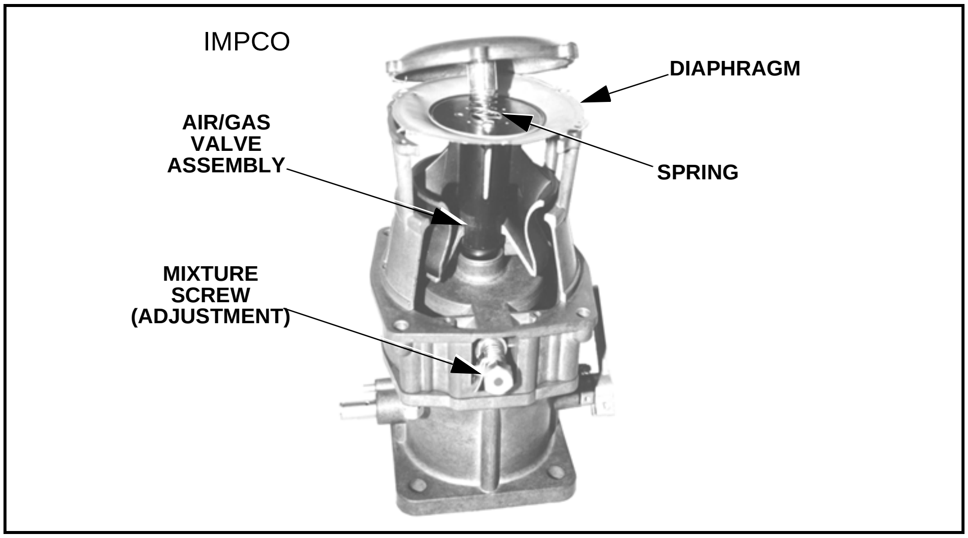


Figure 9-7 IMPCO Carburetor

SECTION 5

MULTIPLE FUEL SOURCES

Many installations require a secondary fuel source to assure the engine can continue operating when the primary fuel source is temporarily lost or fuel production drops off. These fuels may be any combination of low Btu gas (digester gas, landfill gas), natural gas, and/or HD-5 Propane. Waukesha uses several different methods for providing a back-up fuel source dependent on which type of fuel system is used, and requirements of the project. In the following sections the 5" wc gas/air specification only applies to the IMPCO carburetor. The venturi type carburetor requires the gas/air pressure to be closer to equal.

Dual Fuel – 2 Pipe System

Single Carburetor

The Lower Btu gas is regulated at approximately 5" wc gas/air with full load air/fuel ratio trimming handled with the carburetor load adjust valve. High Btu gas is regulated at approximately -1" wc gas/air with final air/fuel ratio trimming handled with the line load adjust valve (see Figure 9-8).

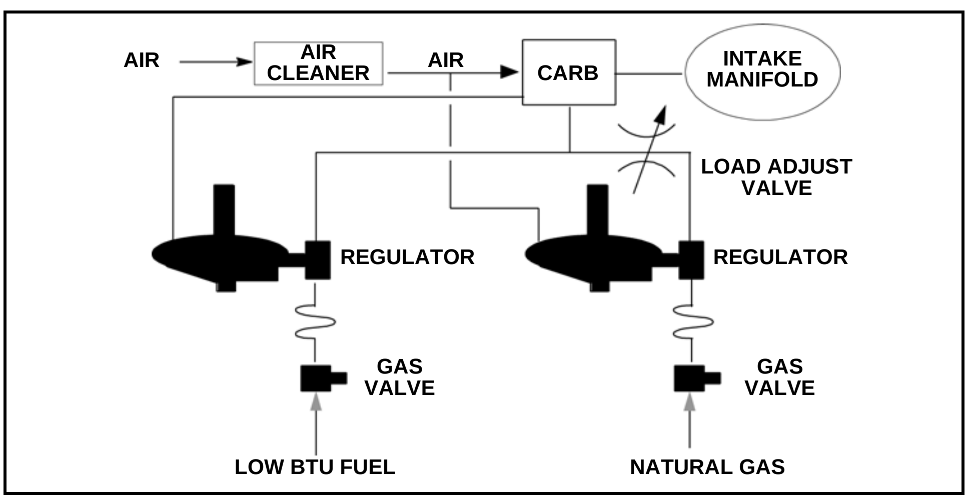


Figure 9-8 Single Carburetor

Switching between fuels involves simultaneously opening the oncoming fuel valve, closing the outgoing fuel valve, and changing the ignition timing. Change-over requires load reduction and/or special sequencing controls for switching fuel sources and ignition timing.

SECTION 6 FUEL SYSTEM COMPONENTS

The fuel train of an engine consists of numerous components each with important functions (see Figure 9-9). These components are discussed here.

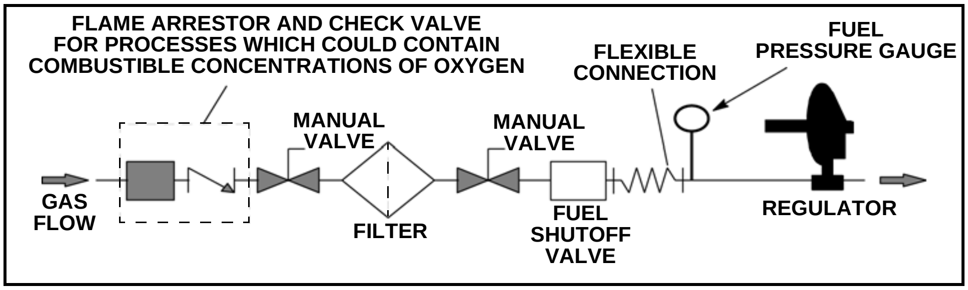


Figure 9-9 Typical Fuel Train

Fuel Valves

Fuel shutoff valves are required on gaseous fueled engines to stop fuel flow when the engine is shut down. Failure to close a valve can result in fuel bleeding through the engine into the exhaust stack which wastes the fuel and becomes a fire and explosion hazard. There are several types of fuel shutoff valves discussed here:

Solenoid Valves

Electric solenoid valves use electric power to either hold the valve open (energize to open) or hold it closed (energize to close). For safety reasons an energize to open solenoid valve is recommended.

Pneumatic Valves

Pneumatic valves use air or gas pressure to hold the valve open. Pneumatic valves are common in CSA Group D, Class 1, Division 2, hazardous locations instead of electric solenoid valves. Electric components present an arcing hazard which could start a fire.

Mechanical Valves

A mechanical valve is a type which is manually reset to the open position immediately before starting. The valve closes automatically when ignition system energy is momentarily channeled away from firing spark plugs, to energize a trip coil on the valve. The valve trips closed and circuit opens which again directs energy to fire the spark plugs. The intent of firing the spark plugs after the valve closes is to burn residual fuel during the engine's coast down. This type of valve is common where there is no on-site source of electric or pneumatic power.

Regulators

Gas regulators are used to provide a consistent pressure to the engine carburetor regardless of engine operating load and fuel flow. The regulator typically provides gas at 5" wc pressure above the air pressure entering the carburetor (see Figure 9-10). Air pressure to the carburetor varies considerably depending on air cleaner restriction, and turbocharger boost pressure at varying speeds and loads.

To account for these pressure variances, the regulator vent port is connected near the air inlet side of the carburetor.

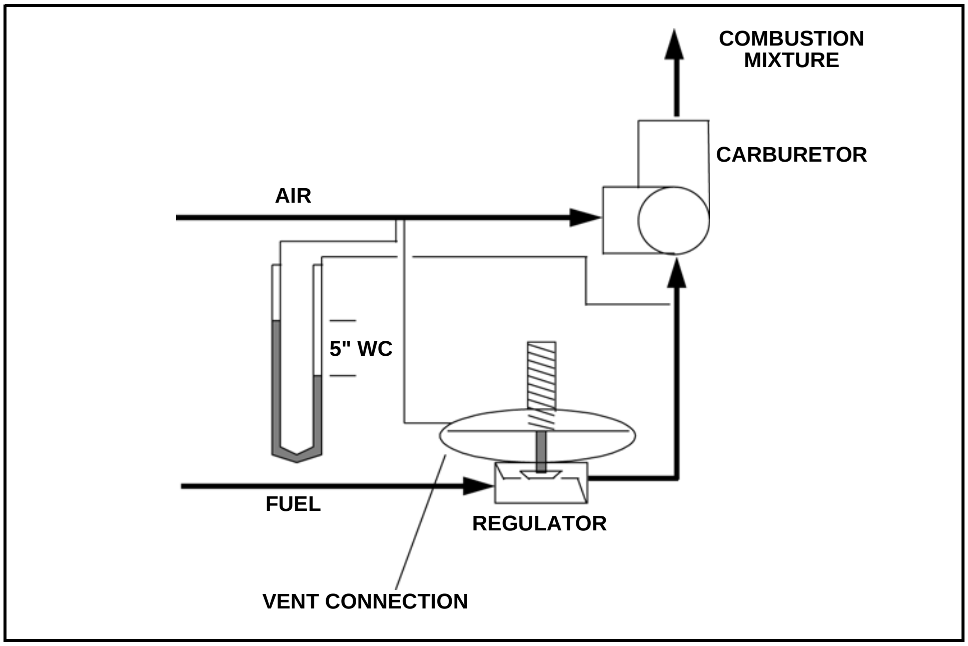


Figure 9-10 Engine Fuel Regulator

Waukesha provides gas regulators for many of its engines; however, in some cases, it is necessary to obtain a regulator from another source. To properly select a regulator you need to consider the outlet pressure, inlet pressure, fuel flow requirements, fuel specific gravity, and droop (proportional band).

Outlet Pressure

The regulator outlet/carburetor inlet pressure depends on air pressure to the carburetor and the required gas over air pressure for proper function of the carburetor. For naturally aspirated and draw through fuel systems, the air pressure is near atmospheric pressure. Gas over air requirements and air pressure to the carburetor on other fuel systems are available in the Tech Data Manual or by contacting Waukesha Application Engineering.

Inlet Pressure

Minimum inlet pressure to the regulator must be sufficiently higher than the outlet pressure to flow enough fuel. Maximum inlet pressure is determined by the regulator limitations. Gas regulators must be selected for specific inlet pressures.

WARNING

Exceeding the maximum inlet pressure rating of the regulator may cause the regulator housing to burst.

Fuel Flow

The fuel flow rate depends on the engine fuel consumption (Btu/hour or kW) and the saturated lower heating value (SLHV) of the fuel (Btu/standard cubic foot SCF). Fuel flow rate is then determined as given below:

Fuel flow rate (SCF/hr) = Fuel cons. (Btu/hr) / SLHV (Btu/SCF)

or

$$\text{Fuel flow rate} \quad (\text{nm}^3/\text{hr}) = \frac{\text{KJ/hr}}{\text{SLHV}(\text{KJ/nm}^3)}$$

This consumption must then be divided by the number of regulators. There are two regulators required on VHP 12- and 16-cylinder engines and on some VGF 12- and 16-cylinder engines.

Example: An engine with 2 gas regulators consumes 5,000,000 Btu/hr of a 600Btu/SCF SLHV gas. The fuel flow rate per regulator is then:

$$\text{F (SCF/hr)} = \frac{5{,}000{,}000 \text{ Btu/hr}}{600 \text{ Btu/SCF} \times 2 \text{ Regulators}} = 4167 \text{ SCF/hr}$$

Fuel Specific Gravity

Most published regulator performance is based on 0.6 specific gravity commercial quality natural gas. Field gases and low Btu gases tend to have higher specific gravities which decreases regulator capacity. A correction factor to be applied in these cases is given below:

CF = (NG spec.gr.)$^{0.5}$ / (site fuel spec.gr.)$^{0.5}$

For a 0.9 specific gravity fuel the correction factor would be:
CF = $(0.6)^{0.5}$ / $(0.9)^{0.5}$ = 0.7746 /0.9487 = 0.8165

Regulator capacity (site gas) = Reg. Cap. (NG) $\times$ CF

Droop

For an engine with a nominal gas over air setting of 5" wc, the regulator droop should be 1" wc or less. Droop at higher gas over air settings can be proportionately higher. For instance, an engine with a nominal gas over air setting of 10" wc could have up to 2" wc droop.

Filters

Gas filters help protect regulators, carburetors, and combustion chamber components by catching piping debris, grit, etc. Liquid hydrocarbons and water tend to wash lube oil off of cylinder walls and cause heating value and knock index swings. Coalescing filters remove water and liquid hydrocarbons from the fuel stream.

Manual Valves

Having 1/4 turn ball valves near the engine allows technicians and operators to cut gas flow manually in an emergency. It also allows them to positively turn off gas flow when servicing the engine. Pneumatic and electric solenoid valves connected to automatic or remote controls cannot be trusted for keeping gas flow off during servicing.

Fuel Piping

Use schedule 40 black iron or stainless steel pipe as a minimum. Schedule 40 piping can withstand high pressure spikes caused by engine backfires. Do not use galvanized piping or fittings. Galvanized coatings tend to flake off when exposed to corrosives sometimes found in gaseous fuels. All piping joints and connections must be gas tight and meet local regulations.

Flexible Connection

Flexible connections at the engine fuel inlet will reduce stress due to piping alignment and vibration. Flexible connections are recommended for all installations and are required for spring isolator mounted units. A Dresser-type coupling can compensate for some misalignment of piping, but once tightened, it is not considered flexible. Braided hose and bellows-type flexible connections are often used for this. Do not use flexible connections as elbows or to compensate for severe piping misalignment. The stress placed on these connections when severely deformed will compromise their service life.

Fuel Piping Check Valves and Flame Arrestors

For digester gas and some other processes it is possible to have air enter the process or storage sphere. With sufficient quantities of air, a combustible mixture is possible which becomes a huge explosion hazard. An engine backfire in such an instance may cause a flame to reach the process or storage sphere and ignite the mixture. To prevent this, the piping should include a check valve to prevent flow from reversing and a flame arrestor in case the flame gets past the check valve.

Volume Tanks

Volume tanks located at the engine can provide a temporary fuel supply for absorbing sudden gas demands caused by rapid loading of an engine. As a general rule these tanks can be sized using the following formula:

$$\text{VT (gallons)} = \frac{125 \times \text{Power [BHP]}}{\text{SLHV [Btu/ft}^3\text{]}} \quad \text{or} \quad \text{VT(m}^3\text{)} = \frac{1000 \text{ x Power(kW)}}{\text{SLHV(KJ/m}^3\text{)}}$$

Where

VT = Volume Tank

Power = maximum operating horsepower

SLHV = Saturated Lower Heating Value of the fuel

The tank size can be further adjusted considering the fuel source pressure as follows:

$$\text{VTc} = \text{VT} \times \frac{(\text{MFP} + 14.7)}{(\text{SFP} + 14.7)}$$

Where

VTc = Volume tank size corrected for site fuel pressure.

MFP = Minimum allowed Fuel Pressure for the engine.

SFP = Site fuel pressure to the engine regulator(s).

SECTION 7

FUEL TREATMENT

Fuel quality is important for stable, reliable, long lasting service from a gas engine. There are many contaminants which can affect the performance of an engine. This section discusses these contaminants and existing technologies for removing them. Pipeline quality natural gas generally does not contain many contaminants other than particles. Most other gaseous fuels, particularly landfill and digester gas, contain contaminants that are harmful to your engine.

Liquid Removal

If saturated gas is allowed to condense, this moisture can lead to corrosion and liquid-loading problems within the fuel train.

Removal of liquids

- Water
- Compressor oil carryover
- Liquid hydrocarbons
- Glycols

There are four basic methods for removing this moisture from the gas:

– Droplet Interception
– Cooling/Condensing
– Absorption
– Adsorption

A brief description of each method follows.

Droplet Interception

To remove liquid droplets (also referred to as “aerosols”), both a scrubber vessel and coalescing filter are often used. A scrubber generally consists of a large cylindrical vessel containing numerous serpentine baffles. As the gas stream enters this vessel, the gas velocity is reduced, causing the heavy water droplets to fall to the vessel bottom due to gravity. The gas stream then enters the baffled section, where gas velocity increases. Inertia will cause the remaining droplets to be thrown against the baffle walls with vigorous force, thus “scrubbing” the droplets from the gas stream. These droplets eventually trickle down the baffle wall and drop to the vessel bottom. A drain is then used to remove the collected liquids.

Droplet interception is effective for removing all liquids (while in the liquid state) including water, compressor oil carryover, and liquid hydrocarbons.

A coalescing filter consists of a tube shaped cartridge of randomly oriented glass fibers. As the gas stream flows through this media, the liquid droplets will impinge and adhere to these fibers. These droplets then travel along the fibers to a point where several fibers intersect. Here, the droplets unite or “coalesce” to form larger droplets, as shown in Figure 9-11. This process continues until the droplets have enough mass to trickle down the length of the cartridge. At this point, the droplets fall into a low turbulence area at the bottom of the coalescer housing, where a drain is used to remove the collected liquid.

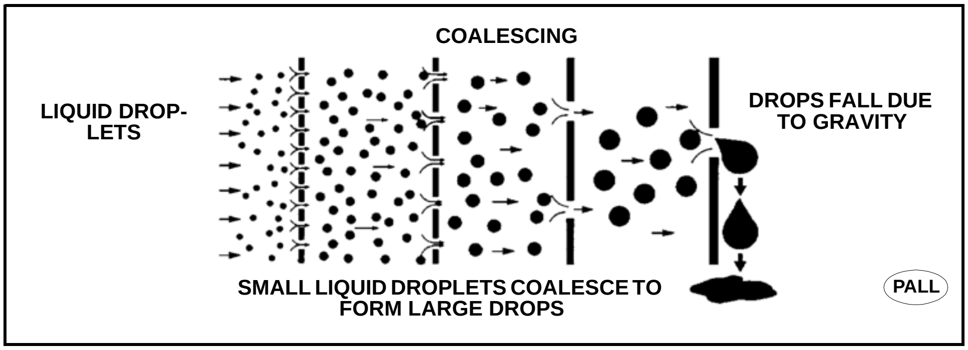


Figure 9-11 Coalescing Filter

A coalescing filter has, in effect, an infinite capacity for liquid droplets, since these droplets can be coalesced and drained from the filter housing as quickly as they enter. Due to its construction, however, a coalescing filter will also trap solid particles in its media, which will eventually create a large enough pressure drop to necessitate change-out of the cartridge.

Cooling/Condensing

Another method of drying the gas stream involves the use of a heat exchanger. Various devices and thermodynamic processes can be used for this purpose, with gas-to-air and gas-to-refrigerant being the most common. The function of this device is simply to reduce the temperature of the site gas. As the gas temperature is reduced, the gas loses its capacity to hold moisture. This moisture then condenses out, and is separated from the gas stream through the use of a droplet interception device (i.e., dropout tank, scrubber, coalescer, etc.)

The effectiveness of this process depends on condensing temperatures of the liquids which must be removed. It is generally very effective for removing water has been proven very valuable and is required for landfill gas treatment.

Absorption

The absorption (or “deliquescent”) type of dryer utilizes a vessel containing salt beads (or “desiccants”) which have a high affinity for moisture. As water vapor comes in contact with this desiccant, a reaction takes place, causing the water vapor to turn liquid and the desiccant to dissolve. The resulting liquid solution then flows to the bottom of the vessel, where it is drained away. Because they are consumed as they absorb moisture, the desiccant beads must be replenished on a regular bases.

For this type of device, the gas stream must be filtered prior to reaching the desiccant to prevent compressor oil carry-over from fouling the desiccant. Post filtration is also required to prevent any desiccant dust from traveling downstream where it could cause abrasive damage to internal engine parts.

Another form of absorbent drying involves the use of a liquid agent to attract moisture. This liquid (usually glycol based) is brought into intimate contact with the gas stream, where the water vapor is drawn out of the gas stream due to its attraction to the glycol. This glycol/water solution is then separated from the now dehydrated gas stream and is fed to a regeneration system where the water and other contaminants are removed (see Figure 9-12). The regenerated glycol is then used again for further absorption.

Absorption works well for water removal but is not suitable for removal of liquid hydrocarbons or compressor oil carryover.

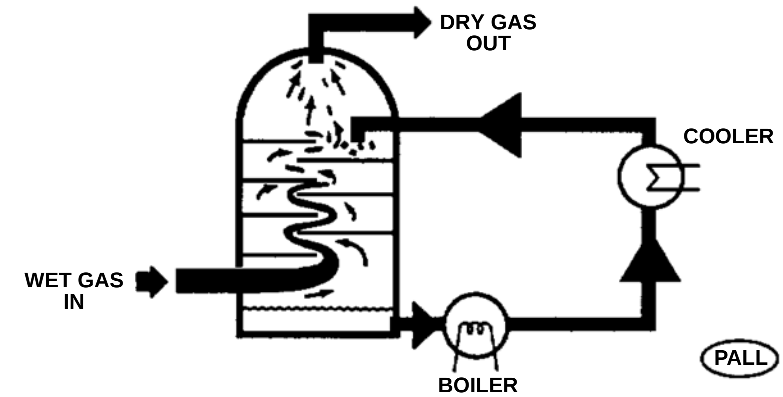
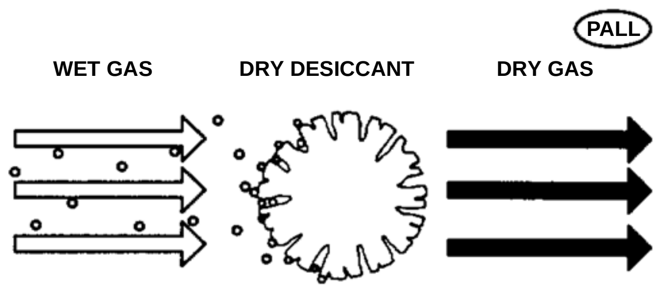


Figure 9-12 Absorption Deliquescent Dryer (Liquid)

Adsorption

Unlike deliquescent desiccants (which dissolve when moisture is absorbed), an adsorbent desiccant remains solid. With this type of desiccant, the water vapor is molecularly attracted to the surface layer of the beads; there are no chemical interactions (see Figure 9-13). Activated alumina and molecular sieve catalysts are commonly used as adsorbing desiccant materials.



Adsorption Occurs Because:

1. Moisture in air is greater than moisture on desiccant.
2. Molecular attraction (Van Der Waals forces.)

Figure 9-13 Adsorption Drying

Once these desiccant beads are fully saturated, they are then dried or "regenerated" so that they may be used again. The method of regeneration is usually of the heated "thermal-swing" variety. Simply stated, thermal-swing dryers pass dry, heated gas through the desiccant bed to evaporate and remove water from the desiccant beads. Once dry, the beads are then able to be reused for further water adsorption.

To provide uninterrupted dehydration capabilities, an adsorbing dryer must utilize two desiccant vessels: one vessel dehydrates the saturated gas stream while the other vessel is regenerating, then vice-versa.

Like absorption, adsorption is not suitable for removal of liquid hydrocarbons or compressor oil carryover.

In addition to liquids, a properly designed filtration system must also remove solid particulates from the gas stream. These particulates usually consist of silicons or iron oxides of sub-micronic size, which are highly abrasive within the engine. To remove these solids, a particulate or coalescing filter is relied upon.

Halogenated Hydrocarbon Removal

Halogenated hydrocarbons are compounds containing halogen atoms such as chlorine and fluorine. Halogenated hydrocarbons react in the combustion chamber with oxygen and water to form acids such as hydrochloric and hydrofloric acid. This acid causes corrosion of bearings and bear metal parts causing failure.

Presently, there are two processes used to remove halogenated hydrocarbons from landfill gas. Both methods are based on processes previously described in this section.

The first method utilizes the adsorption process: desiccant beads of activated carbon are used to molecularly attract and adsorb halogenated hydrocarbons from the gas stream. Thermal-swing regeneration is then used to strip the halogens from the desiccant. The halogen-laden purge gas is then sent downstream to a cold-water heat exchanger/condenser, where the halogens are removed in liquid form.

The second method of halogen removal utilizes the glycol absorption technique: A glycol solvent is brought into contact with the landfill gas stream, where both the water and halogenated hydrocarbons are absorbed into the glycol solvent. The contaminated glycol is then sent to a regeneration vessel, where the solution is flashed to a low pressure and high temperature. This boils off the water and halogens, which are then sent to an incinerator for destruction.

Further information regarding halogenated hydrocarbons can be found in the Waukesha Tech Data Book (see latest edition of S7884-7).

H_2S Removal

Hydrogen sulfide and other sulfur bearing compounds react in the combustion chamber with oxygen and water to form sulfuric acid. This acid causes corrosion of bearings and bare metal parts causing failure of these.

There are a variety of commercial systems available for removal of H_2S from fuel. The most common of these is referred to as an iron sponge.

In the iron sponge process, the gas containing H_2S flows across a bed of hydrated iron oxide where a reaction takes place forming iron sulfide and water. The sulfur compounds are thus removed from the fuel gas.

The iron sponge material can be regenerated a limited number of times by flowing air across the bed which converts iron sulfides back to iron oxide. The iron media eventually permanently coats with elemental sulfur, insulating the active iron oxide from the gas and increasing pressure drop.

Other commercial systems are available from various vendors. The process to use depends on the amount of gas to be processed, contamination level, and availability of required chemicals and/or media.

Siloxanes Removal

Siloxanes are man-made compounds which are commonly used in down-the-drain products such as shampoos, deodorants, cosmetics, and medicine. Because of their uses and disposal, siloxanes often migrate into landfill gas from these products. Siloxanes can also be found in digester gas, especially when an industry in the sewage circuit manufactures products using siloxanes.

Siloxanes in the combustion chamber burn and form silicon dioxide (SiO_2), also known as “silica” or “sand.” If the siloxane concentration in the fuel is great enough, the silicon dioxide can form significant deposits on cylinder heads, exhaust valves, spark plugs, and gas admission valves. These deposits generally take on a fine, white, powdery appearance, but have also been seen as either a glaze-like coating or a ceramic-like coating.

Experience has shown that lean burn engines tend to have more difficulties with deposit formation than stoichiometric (“rich burn”) engines.

LANDFILL GAS FILTRATION SYSTEM RECOMMENDATIONS

At a minimum, the fuel treatment system should be similar to that of Figure 9-14. This type of system addresses the removal of liquids and solids, not halogenated hydrocarbons (for systems with halogenated hydrocarbon removal equipment, contact the equipment manufacturer for specific fuel system recommendations).

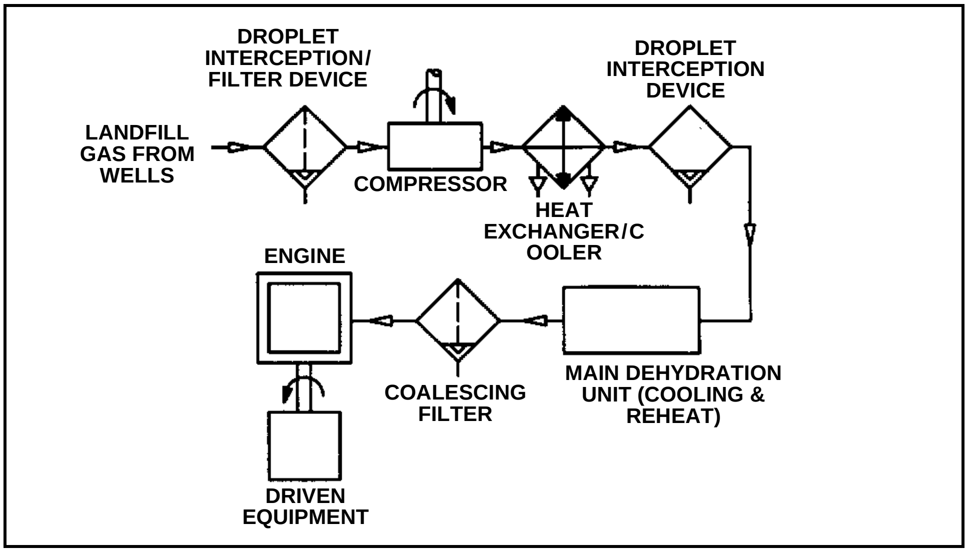


Figure 9-14 Typical Landfill Gas Fuel Treatment System

(Actual fuel composition may dictate more, fewer, or different components than illustrated in Figure 9-14.)

The system inlet should utilize a droplet interception device, i.e., a scrubber or coalescer. A scrubber is the most commonly used device, as it is simple and virtually maintenance free. A coalescer, on the other hand, is more effective than a scrubber, but will require periodic filter changeouts.

Immediately downstream of the compressor, a heat exchanger/cooler (gas to air) is recommended to remove the heat absorbed by the landfill gas during the compression process. This reduction in temperature will also cause liquids to condense out of the gas. These liquids can then be removed from the gas stream through the use of a droplet interception device.

At this point, a dehydration unit is placed in the system to further remove water vapor from the gas stream. It is difficult to recommend the selection of the dehydration unit due to the vast differences in operation, maintenance, and expense. Table 9-1 offers some guidelines to the advantages and disadvantages of each type.

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Table 9-1 Dehydration Unit

UNIT	ADVANTAGES	DISADVANTAGES
Absorbent: Deliquescent Desiccant	• Low initial cost • Simplicity-no moving parts. • No energy input required.	• Offers limited moisture removal (20°F reduction of inlet dewpoint). • Requires pre- and post-filtration. Recurring cost of desiccant replacement.
Absorbent: Glycol	• Removes water and some halogenated hydrocarbons (40°F reduction of inlet dewpoint).	• Requires regeneration process to restore glycol quality.
Adsorbent Desiccant	• Offers greatest moisture removal ability (-40°F absolute pressure dewpoint). • No water to drain.	• High initial cost • Requires pre- and post-filtration. • Requires periodic change out of the desiccant. • Requires regeneration process to restore desiccant quality.
Cooling/Condensing (Gas to Refrigerant)	• Offers consistent absolute pressure dewpoint of 35°F. • No post-filtering required.	• Requires energy input for refrigeration circuit.

A high-quality coalescing filter is highly recommended as the final piece of equipment in the fuel treatment system. The coalescer should be placed upstream of the engine-mounted regulator (as close as practical) to eliminate any liquids which have condensed out of the gas stream. The following guidelines are recommended for the selection of the coalescing filter:

1. The coalescer must be specifically designed to remove liquids and solids from a gaseous stream.
2. It must utilize an inside-to-outside flow path through the coalescing media.
3. It must have a 1 micron absolute particulate rating, or a 1 micron Beta ratio of no less than 10,000. (Beta ratio is determined by a test, where a known number of particles of a given size are placed upstream of a filter and the resulting number of these particles which pass through the filter are counted. The Beta ratio is calculated by dividing the number of particles sent into the filter by the number of particles which passed through it.)
4. The entire coalescer assembly (including the housing and drain) must be compatible with any liquids it may come in contact with.

To further ensure that no liquid water condenses out within the fuel train components, the fuel treatment system must reduce the moisture content of the fuel such that the pressure dewpoint is at least 20°F (-6°C) below the measured temperature of the fuel at the engine-mounted regulator inlet.

NOTE: *Pressure dewpoint is defined as the temperature at which water vapor will begin to condense out of a gas at its given pressure; i.e., a low pressure dewpoint value indicates a relatively dry gas.*

For solid particulate filtration of the landfill gas, Waukesha recommends a 1 micron absolute rated fuel filtration system. As previously stated, a quality coalescing filter will meet this requirement.

In addition to fuel filtration, a Waukesha 1 micron bypass lube oil filter is mandatory for landfill applications, so as to remove any solid particles which have entered the crankcase via blow-by gases.

SECTION 8 PIPING RESTRICTION CALCULATION

Fuel Piping Restriction Calculation

Sizing fuel piping and components (valves, filters, regulators) will require knowing the fuel flow rate, density, pressure and temperature at the maximum operating speed and load. The flow rate must be considered for any intermittent or overload conditions required in the specification.

Piping restriction depends on the pipe diameter, pipe length, number of elbows and transitions, and piping material used. The following procedure will help determine piping restriction.

1. Determine fuel consumption (Btu/hr or kW) for the highest speed and load condition expected. This information is available in the Ratings and Standards section or Heat Rejection section of the Waukesha Tech Data Manual. If working with the Brake Specific Fuel Consumption “BSFC” (Btu/hp-hr) then multiply this figure by the maximum horsepower to get fuel consumption in Btu/hr.

   Equation 1

   Fuel Consumption (Btu/hr) = BSFC (Btu/hp-hr) x BHP

   or

   $$\text{Fuel Consumption (kW)} = \frac{\text{KJ}}{\text{kW/hr}} \times \frac{\text{kW}}{3600}$$

2. Determine the flow volume at standard temperature (60°F) and pressure (29.92 inch-Hg) with the following formula:

   Equation 2

   $$\text{Standard Flow (ft}^3\text{/min)} = \frac{\text{Fuel consumption (Btu/hr)}}{\text{Fuel SLHV (Btu/SCF SLHV)}} \div 60$$

   or

   $$\text{Standard Flow (nm}^3\text{/hr)} = \frac{\text{Fuel consumption (kW)}}{\text{Fuel SLHV (KJ/nm}^3\text{)}}$$

3. Determine the flow volume at the site supply temperature and pressure:

   Equation 3

   $$F_{sup} = F_{std} \times \frac{14.7\ [\text{psia}] \times (460^\circ\ [\text{R}] + T_{sup})}{(14.7\ [\text{psia}] + P_{sup}) \times 520^\circ [\text{R}]}$$

   or

   $$\text{ACTUAL FLOW (m}^3\text{/s)} = \frac{101.3\ \text{kPa} \times (273 + T_{sup})}{(101.3\ \text{kPa} + P_{sup}) \times 273}$$

Where

F_{std} = Flow at standard conditions (ft^3/min or Nm^3/sec)

F_{sup} = Flow at supply conditions (ft^3/min or Nm^3/sec)

Tsup = supply temperature (°F or °C)

P_{sup} = supply pressure (psig or kPa)

4. Calculate fuel velocity (V) based on the supply flow F_{sup} for each pipe size used:

Equation 4

$$V\ (FPM) = \frac{F_{sup}(ft^3/min)}{\text{Pipe inside area }(ft^2)} \text{ or}$$

$$V\ (m/sec) = \frac{F_{sup}(m^3/sec) \times 1{,}000{,}000}{\text{Pipe inside area }(mm^2)}$$

$$\text{Pipe Area }(ft^2) = [\text{Diameter (inch)}]^2 \times \pi/4 \times 0.00694\ ft^2/inch^2$$

Inside diameter and area for common pipe sizes are given in Table 9-2.

As a general rule, gas velocities over 12,000 FPM (60 m/s) are unacceptable because of the high resulting restriction.

Table 9-2 Pipe Areas for Schedule 40 Pipe

PIPE SIZE (in.)	ID (in.)	ID (mm)	AREA (inch^2)	AREA (mm^2)	AREA (ft^2)
1	1.049	26.644	0.864	557.42	0.00600
1.25	1.380	35.53	1.496	965.16	0.01039
1.5	1.610	40.894	2.04	1312.77	0.0142
2	2.067	52.502	3.36	2163.80	0.0233
2.5	2.344	59.538	4.32	2782.61	0.030
3	3.068	77.927	7.39	4767.03	0.0513
4	4.026	102.260	12.73	8208.89	0.0884
5	5.047	128.194	20.01	12900.42	0.139
6	6.065	154.051	28.89	18629.39	0.201

5. Determine pressure loss P_L per 10 ft or 1 meter of pipe for each velocity and pipe size from Figure 9-16.

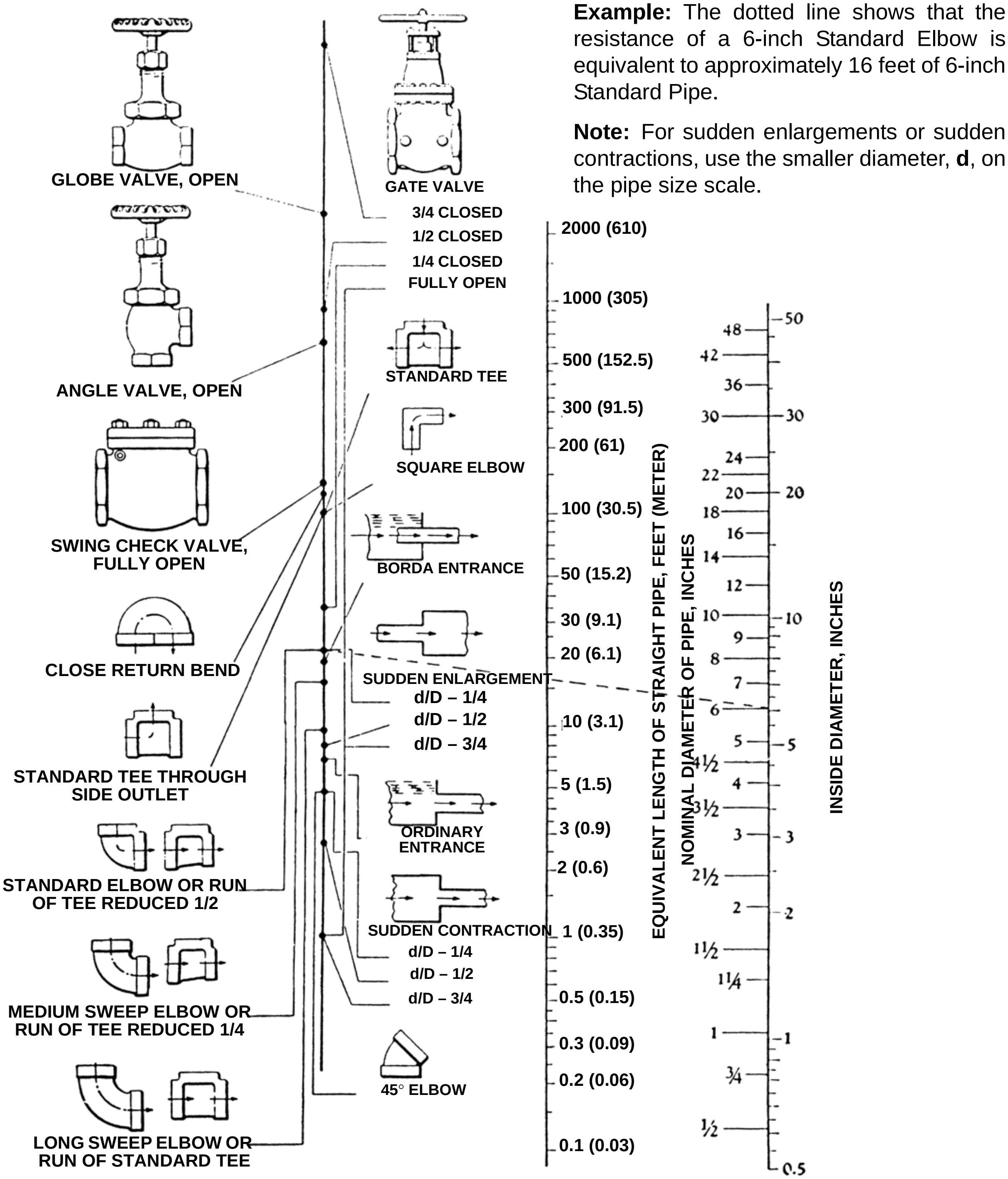
6. Determine the equivalent pipe length (EPL) for all fittings of each pipe size.

   Figure 9-15 gives equivalent pipe length in feet for various pipe fittings. For each pipe size, sum the EPLs and add them to the total length of straight pipe of that size to find the total of each pipe size.

RESISTANCE OF VALVES AND FITTINGS TO FLOW OF FLUIDS IN EQUIVALENT LENGTH OF PIPE

Example: The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16 feet of 6-inch Standard Pipe.

Note: For sudden enlargements or sudden contractions, use the smaller diameter, **d**, on the pipe size scale.



From Crane Co. Technical Paper No. 409. Data based on the above chart are satisfactory for most applications. REPRINTED WITH PERMISSION OF CRANE VALVE GROUP.

Figure 9-15 Equivalent Pipe Length of Fittings in Feet (Meters)

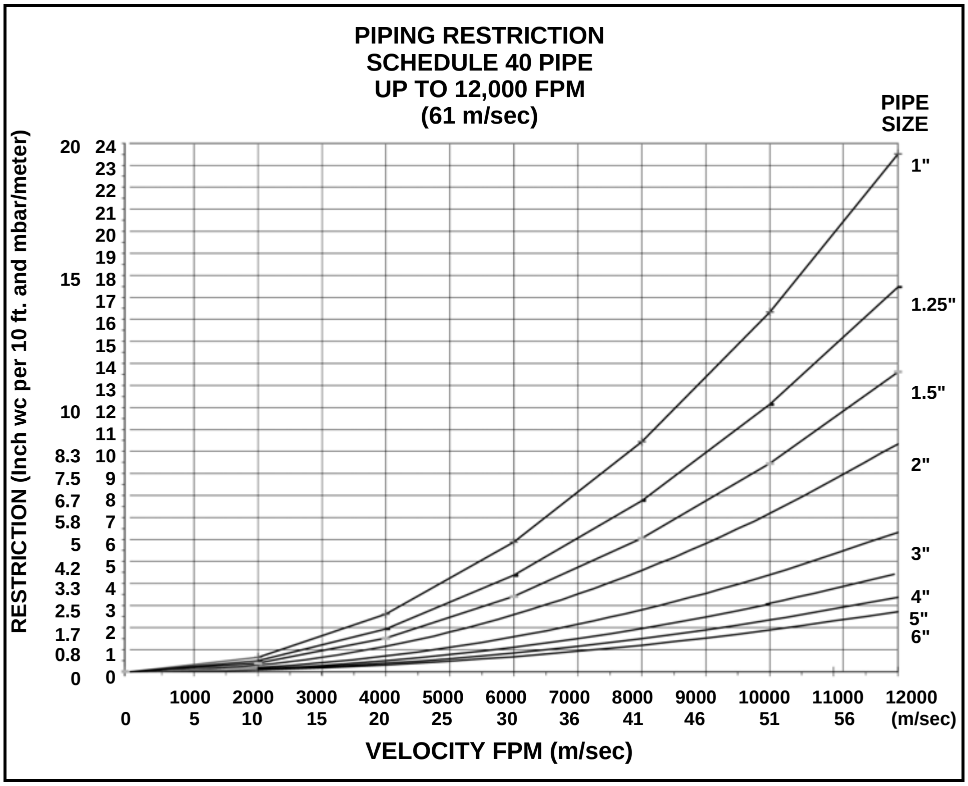


Figure 9-16 Restriction vs. Velocity for Pipe Diameters Up to 6 in

7. Calculate the fuel gas corrected specific gravity:

Equation 5

$$SG_C = SG \times \frac{(14.7\ [psia] + P_{sup}) \times 520°[R]}{14.7\ [psia] \times (460°[R] + T_{sup})} \quad \text{or}$$

$$SG_C = SG \times \frac{(101.3\ [kPa] + P_{sup}) \times (273)}{(101.3 \times (273 + T_{sup}))}$$

Where

SG_c = specific gravity corrected for pressure and temperature

SG = fuel specific gravity

Natural Gas	SG ≈ 0.6
600 Btu Digester Gas	SG ≈ 0.9
400 Btu Landfill Gas	SG ≈ 1.1
Field Gas	SG ≈ 0.6 to 1.0
HD-5 Propane	SG ≈ 1.5

T_{sup} = supply temperature (°F or °C)

P_{sup} = supply pressure (psig or kPa)

8. Calculate the total piping restriction R_P for each pipe size:

Equation 6

$R_{P1} = P_{L1}$ (" wc /10 ft) $\times$ $SG_c \times$ EPL(ft) or P_{L1} (mbar/m) x SG_C x EPL (m)

9. Calculate the total fuel piping restriction:

   Equation 7

   $R_T = R_{P1} + R_{P2} + R_{P3} + R_A$

   Where

   R_T = total restriction (" wc or mbar)

   $R_{P1,2,3}$ = piping restriction for various pipe sizes (" wc or mbar)

   R_A = accessories (filters, solenoid valves, etc.) restriction (" wc or mbar)

Sample Problem:

A 12V275GL+ engine will operate 3624 bhp @ 1000 rpm with natural gas (900 Btu/SCF LHV). The designer plans to use 50 ft of 2 in. pipe with 7 short radius 90° elbows and a globe valve at the engine for servicing. The following data applies to this fuel system:

Fuel

Type .... Natural Gas

LHV .... 900 Btu/SCF

Specific Gravity 0.6

Pressure .... 50 psig

Temperature .... 90° F

Piping

Length .... 50 ft

Short radius elbows 7

Valves .... 1 globe

Table 9-3 Fuel System Restriction Sample Problem

CALCULATE THE FUEL SYSTEM RESTRICTION	REFERENCE
1. Determine fuel consumption 24,074,000 Btu/hr	S-9062-10 Heat Rejection Section of Technical Data or site-specific data from EngCalc
2. Determine flow volume. $F_{Std}(ft^3/min) = \frac{24{,}074{,}000 \text{ Btu/hr} \times 1 \text{ hr/60 min}}{900 \text{ Btu/SCF}} = 446 \text{ SCFM}$	Equation 2
3. Determine the flow volume at the site supply temperature and pressure: $F_{sup} = 446 \text{ SCFM} \times \frac{14.7 \text{ [psia]} \times (460°\text{[R]} + 90°\text{F})}{(14.7 \text{ [psia]} + 50 \text{ [psig]}) \times 520°\text{[R]}} = 107.18 \text{ ACFM}$	Equation 3
4. Calculate fuel velocity (V) based on the supply flow F_{sup} for each pipe size used: $V \text{ (FPM)} = \frac{107.18 \text{ ft}^3/\text{min}}{0.0233 \ (\text{ft}^2)} = 4{,}600 \text{ FPM}$	Equation 4 Table 9-2
5. Determine pressure loss per 10 feet of pipe: Pressure loss in 2" pipe @ 4,600 ft/min $P_{L2} = 1.5\text{" wc/10 ft}$	Figure 9-16
6. Determine the equivalent pipe length: $EPL_6 = 50 \text{ ft. pipe} + (7 \text{ elbows} \times 5 \text{ ft./elbow}) + 50 \text{ ft./globe valve} = 135 \text{ ft.}$	Figure 9-15
7. Calculate the fuel gas corrected specific gravity: $SG_C = 0.6 \times \frac{(14.7 \text{ [psia]} + 50) \times 520°\text{[R]}}{14.7 \text{ [psia]} \times (460°\text{[R]} + 90)} = 2.50$	Equation 5
8. Calculate the total piping restriction R_P for each pipe size: $R_{P1} = \frac{1.5\text{" wc}}{10 \text{ ft.}} \times 2.5 \times 135 \text{ ft.} = 50.6\text{" wc}$	Equation 6
9. Calculate the total fuel system piping restriction $R_T = R_{P1} = 50.6\text{" wc}$	Equation 7

SECTION 9

INSTALLATION CONCERNS

Fuel Piping Cleanliness

Fuel piping often contains welding slag, mill scale, rust, and other debris from fabrication and the environment. The piping must be cleaned to remove these contaminants before final assembly on site. Additionally the piping should be momentarily blown out initially to remove any debris which may have entered during assembly.

WARNING

Use extreme caution when venting gas in this way to avoid open flames or other ignition sources. Observe local fire and safety codes.

A 15 micron strainer or filter located at the engine can collect any additional solid contaminants in the piping which were not collected earlier.

Fuel Temperature Considerations

Fuel temperature is important for the successful operation of the engine.

Minimum Temperature

Low fuel temperatures may cause condensation of water or heavy hydrocarbons in the fuel resulting in fuel system control problems, lubrication problems in the cylinder, corrosion, and detonation problems. The minimum fuel temperature should be less than 20°F (11°C) above the dewpoint of these liquids to prevent condensation. Extremely low temperatures of -20°F (-29°C) or less will cause hardening of the elastomeric components in the regulators and carburetors resulting in stiff operation and possible cracking of diaphragms. Therefore, the minimum allowed temperature is -20°F (-29°C).

Maximum Temperature

Maximum fuel temperature must not exceed 140°F (60°C). Higher temperatures affect regulator capacity and cause deterioration of elastomeric components in the regulators and carburetors, as well as affect the knock resistance limit of an engine.

Temperature Fluctuations

Fluctuations in fuel temperature cause a change in the density of the fuel. Since a carburetor is a volume mixing device, temperature changes will change the operating mass air fuel ratio. A 50°F (28°C) increase in fuel temperature causes an initial air/fuel ratio of 28:1 to move to 30.7:1 which may result in lost power and unstable operation. A 50°F (28°C) decrease in fuel temperature will richen the engine causing detonation and/or higher NO_x emissions. To avoid these problems fuel temperature fluctuations should be less than 10°F (5°C). An electronic air/fuel ratio controller can compensate for higher fuel temperature swings (50°F or 28°C).

Gas Meter Installation

Positive displacement gas meters can be sensitive to sudden surges of gas pressure caused by instantaneous opening of the engine gas shutoff valve. The meters often jam up preventing fuel flow and damaging the meter. To prevent the pressure surges Waukesha recommends installing a bypass line around the meter with valves located as shown in Figure 9-17. Before opening the fuel shutoff valve on the engine, valve 1 is opened and valves 2 and 3 are closed. Once the engine is started, valves 2 and 3 are opened, then valve 1 is slowly closed (3 seconds). This strategy should minimize the pressure shock at the gas meter.

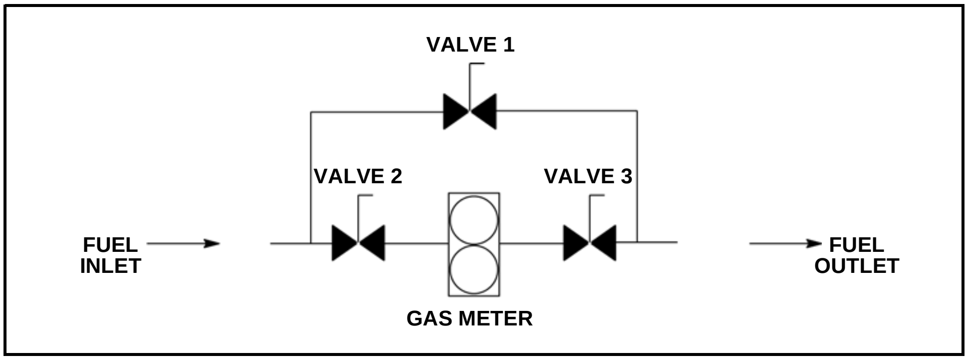


Figure 9-17 Gas Meter Bypass Piping

SECTION 10

FUEL SYSTEM CHECKLIST

1. Coalescing filter – make and model
2. Fuel filter – make and model
3. Fuel meter – make and model
4. Solenoid valves – make and model
5. Gas regulators – make and model
6. Flame arrestor – make and model
7. Refrigeration – make and model

Checklist for Fuel Systems

8. Does fuel meet latest edition of Waukesha spec S-7884-7? ______________
9. Has the piping been correctly sized? ______________
10. Is the piping clean? ______________
11. Is fuel filtration required? ______________
12. Does it require refrigeration? ______________
13. Is there a coalescing filter? ______________
14. Is there an adequate fuel supply and pressure? ______________
15. What is the WKI and timing setting? ______________
16. Is a fuel meter required? ______________
17. Are there solenoid valves installed? ______________
18. Is there any timing requirement for dual fuel? ______________

19. Is the engine specification for the correct fuel? ______________________
20. Have the gas regulators been correctly sized? ______________________
21. Is there a flexible connection at the engine? ______________________
22. Is a volume tank required?______________________
23. Is an AFM required? ______________________
24. Has the AFM been designed into the package? ______________________
25. Is a flame arrester required? ______________________
26. Are there any siloxanes, H_2S or chlorine concerns? ______________________
27. Is the fuel gas within Waukesha temperature spec?

CHAPTER 10

MOUNTING AND ALIGNMENT OF 275GL SERIES ENGINES

SECTION 1 MOUNTING SURFACE

This section discusses mounting surface requirements for Waukesha 275GL engines.

Waukesha 275GL engines require a very smooth and level mounting surface. This is to prevent distortion of the main bearing bores in the crankcase and prevent movement from vibration and thermal growth. Using shims to correct a rough distorted surface does not provide adequate support under the engine. Figure 10-1 illustrates a surface leveled by machining then shimmed and a surface leveled by shims alone.

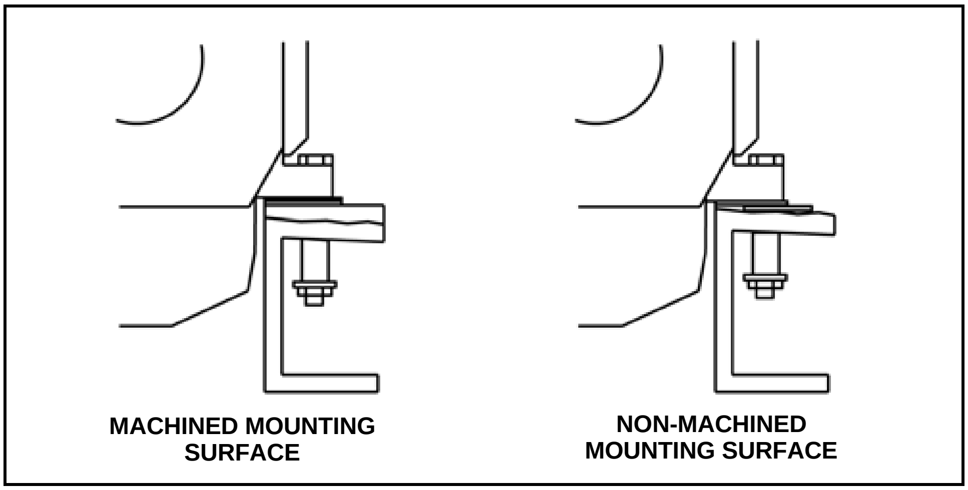


Figure 10-1 Machined Surface Mounting

The machined surface provides a much better support.

A level mounting surface can be provided by attaching 175 x 175 x 65 mm (7 x 7 x 2.5 in.) chocks to the skid by welding or grouting. The engine mounting surface of the chocks must be flat, smooth, and their planes parallel within 0.08 mm (0.003 in.) with a surface finish of 500 RMS.

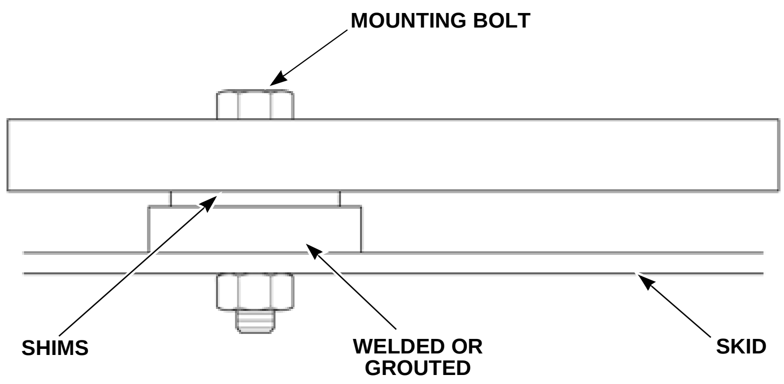


Figure 10-2 Shimming an Engine

Shims of 127 x 127 mm (5 x 5 in.) are then used at each mounting bolt to correct base deflection and alignment. Appendix A *"Shimming Information"* describes proper shimming procedures and lists shims available from Waukesha.

All of the mounting bolt positions are required to properly secure the engine. The jacking bolts are used to raise the engine to shim for final crankshaft web deflection and alignment. An anti seizing dry lubricant must be applied to the jacking bolts before adjusting to prevent the threads from locking. The jacking bolts can be removed and mounting bolts installed once the engine is aligned to provide additional clamping force. If the jacking bolts are to remain in place, they must be backed off to allow proper forging of the mounting bolts.

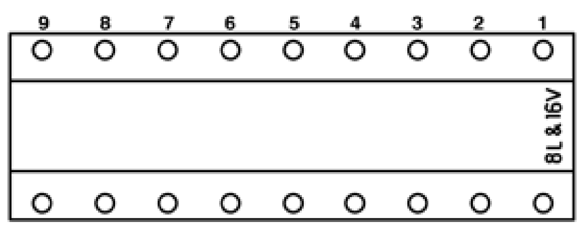


Figure 10-3 Bolt Locations for 16-Cylinder Models

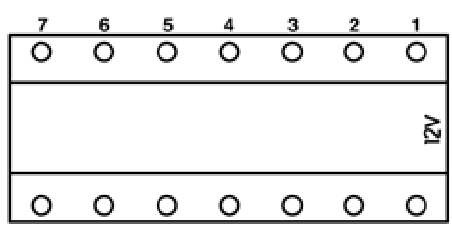


Figure 10-4 Bolt Locations for 12-Cylinder Model

Table 10-1 Bolt Location

ENGINE	NO. OF MOUNTING BOLTS	POSITIONS	NO. OF JACKING BOLTS	POSITIONS
12-Cylinder	8	1, 3, 5, 7	6	2, 4, 6
16-Cylinder	10	1, 3, 5, 7, 9	8	2, 4, 6, 8

SECTION 2 FLYWHEEL MOUNTING & CRANKSHAFT WEB DEFLECTION

Flywheel Mounting

WARNING

Always consider the weight of the item being lifted and use only properly rated lifting equipment and approved lifting methods.

The following procedure must be used for mounting the flywheel to the crankshaft:

1. The mating surfaces of the crankshaft flange and flywheel must be carefully cleaned prior to assembling. They must be dry and free from oil, grease, chips and other impurities. Special attention must be given to this point as power transmission is effected through friction contact.
2. Insert the flywheel bolts from the engine side or push them in with a light hammer blow.
3. Prior to final insertion of the coupling bolts, the nut bearing surfaces as well as the bolt and female threads must be smeared with grease.
4. Using a wrench, tighten the nuts in a diagonal pattern until metal to metal contact is established between all the parts being fastened.
5. Using a torque wrench, tighten the nuts in a diagonal pattern to 400 Nm (290 ft-lb).

   Next tighten the nuts in the same pattern to 800 Nm (580 ft-lb).

   Finally tighten the nuts to 1180 Nm (870 ft-lb).
6. After having torqued all the nuts to the final value, check whether they can be secured with split pins.
7. If necessary, tighten down further (do not slacken back) until a slot in the castle nut is in alignment with a hole in the bolt so that the split pin can be easily pushed through. The further tightening down of the nuts should also be done in a diagonal pattern as far as possible.
8. Secure castle nuts with split pins.

Crankshaft Web Deflection

WARNING

Turn off the air/gas supply to the starters and bleed off air/gas pressure to prevent accidental rotation of the crankshaft before entering the engine crankcase.

Engine crankshaft distortion caused by mounting is determined by measuring deflection of crankshaft webs. This procedure measures the deflection of a crankshaft during one revolution. It is the most direct method of determining if the shaft is being bent by a deflected crankcase or misalignment. Web deflection measurements are required for 275GL engines.

Waukesha 275GL engines have center point marks located on the counterweights to indicate proper web deflection gauge mounting locations (see Figure 10-5).

1. Mount a web deflection gauge, P/N A292683, into the center point marks. Carefully twirl the gauge to make sure it is properly seated.
2. Position the crankshaft so the deflection gauge hangs freely next to the connecting rod, as close to the rod as possible, but not touching. Zero the gauge dial.
3. Slowly rotate the crankshaft until the gauge is in position 2, on the horizontal. Record any positive or negative reading attained.
4. Rotate the crankshaft to positions 3 and then 4, recording any readings. Now rotate the shaft further until the gauge is as high as possible to position 5 without contacting the connecting rod. Record this reading.
5. Remove the deflection gauge and repeat this procedure on the other crankshaft webs. Acceptable deflection figures are given in Table 10-2.

Table 10-2 Deflection of Web

NEXT TO FLYWHEEL	ALL OTHER WEBS
0/-0.105 mm (+0/-0.0040 in.)	+0.035 mm/-0.035 mm (+0.0015 in./-0.0015 in.)

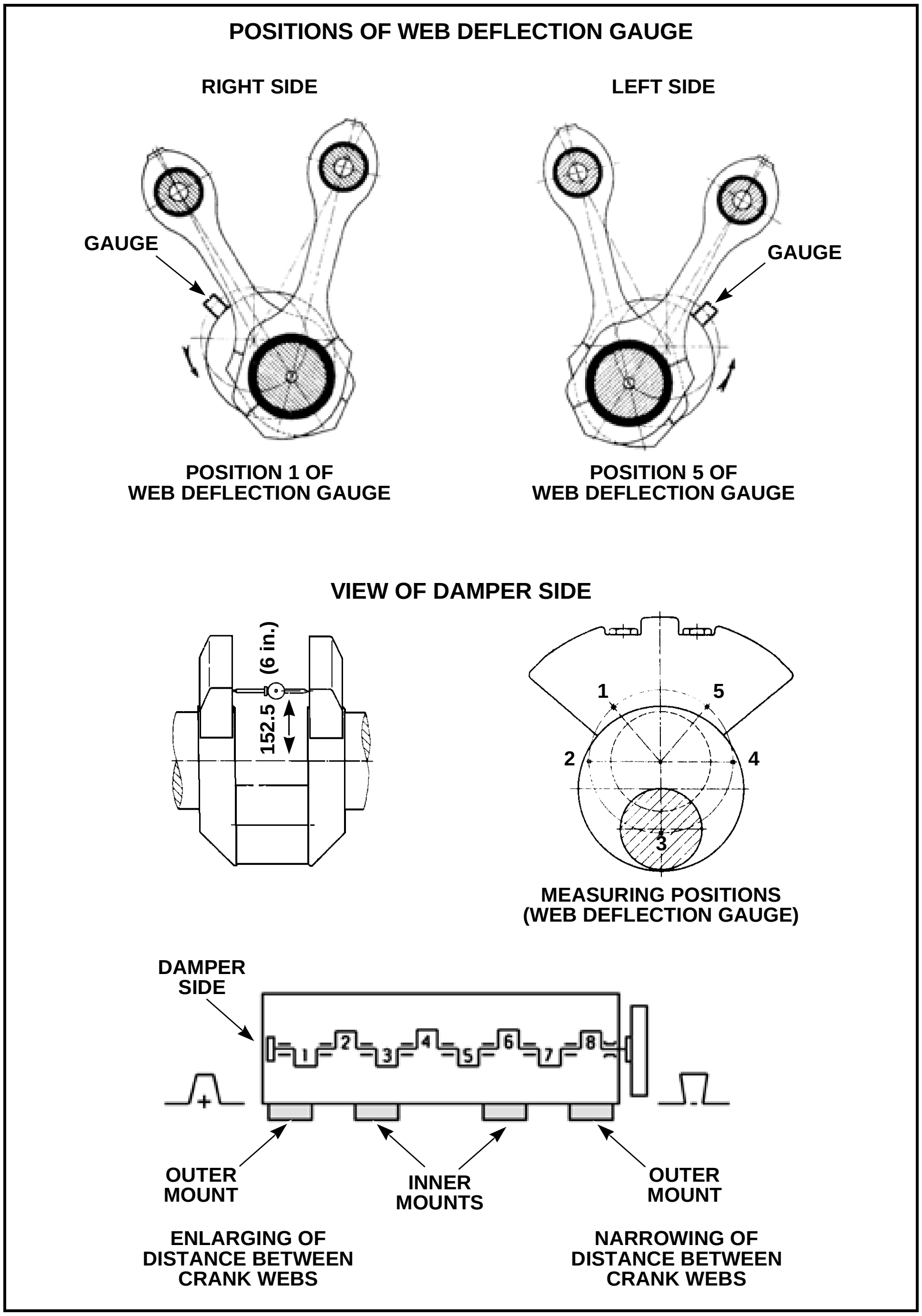


Figure 10-5 Checking the Crankshaft Deflection

Crankshaft web deflection is corrected by adjusting the mounting foot height adjacent to the deflected web. A web which is closed at the bottom (6 o'clock) position is corrected by lowering an inner mount or raising an outer mount, if it is adjacent to an outer mount. Conversely, a web which is open at the bottom is corrected by raising an inner mount or lowering an outer mount. After an adjustment the deflection at the other crankshaft webs must again be measured.

SECTION 3

ALIGNMENT

Multi-Bearing Machines

A multi-bearing machine is one which fully supports its own shaft and does not rely on the engine shaft to support the driven end.

Three areas must be adjusted to accurately align a multi-bearing machine to an engine, which is also a multi-bearing machine. These are endplay, angular alignment and parallel alignment.

When aligning two multi-bearing machines, one machine must be designated as the stationary machine and one as the movable machine. Deciding which machine will be stationary will depend on size, weight, and connections. All adjustments will be made on the movable machine.

Adjusting angular and parallel alignment on multi-bearing machines requires correcting the angular alignment first and then parallel. Once alignment is acceptable, the machines must be shimmed to compensate for thermal growth (see Figure 10-6).

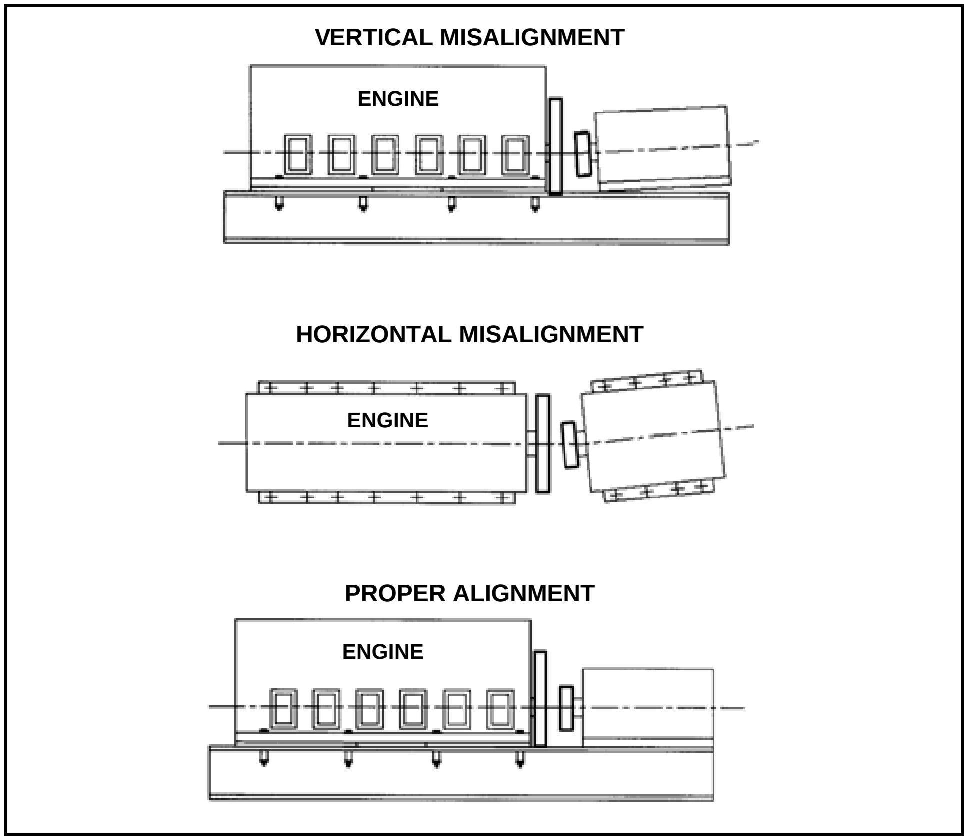


Figure 10-6 Angular Alignment

Waukesha offers a low cost computer program (P/N 475063) which finds adjustments for angular and parallel alignment as well as thermal growth. The program user inputs dimensional, growth, and measuring information. Using this program, only one or two adjustments are normally required to place the units within the alignment specifications. If the alignment program is not available, the following procedure will provide an accurate alignment.

Endplay

⚠ WARNING

Turn off the air/gas supply to the starters and bleed off air/gas pressure to prevent accidental rotation of the crankshaft before entering the engine crankcase.

To measure endplay:

1. Roughly position the two machines and install the shaft coupling. Adjust the distance between the two machines so that there is no apparent tension or compression of the coupling. Properly space gear type couplings per the coupling manufacturer's specifications.
2. Set up a dial indicator on the machine with the least endplay (normally the engine). Clamp the dial indicator to the engine crankcase and read against the flywheel face.
3. Prelube engine until gauge reads a pressure, if oil is available.
4. Pry the crankshaft fully forward, and zero the dial indicator. (Moving the crankshaft will require removing an oil pan door and prying between a main bearing cap and crankshaft cheek or web.)
5. Pry the shaft rearward and read the dial indicator. Crankshaft endplay should be within 0.15 – 0.45 mm (0.006 – 0.018 in.). The shaft must not spring back when the pry bar is removed.
6. If there is insufficient endplay or if spring-back occurs, adjust the distance between the machines until it is resolved.

Angular Alignment

⚠ WARNING

Turn off the air/gas supply to the starters and bleed off air/gas pressure to prevent accidental rotation of the crankshaft before entering the engine crankcase.

To measure the angular alignment, a dial indicator is mounted to the coupling half of one machine to read against the coupling half face of the other. The coupling should be installed or the shafts bound together so they both turn together while taking the alignment measurements.

The radius "R" from the center of the shaft to the dial indicator should be at least 180 mm (7 in.) (see Figure 10-7).

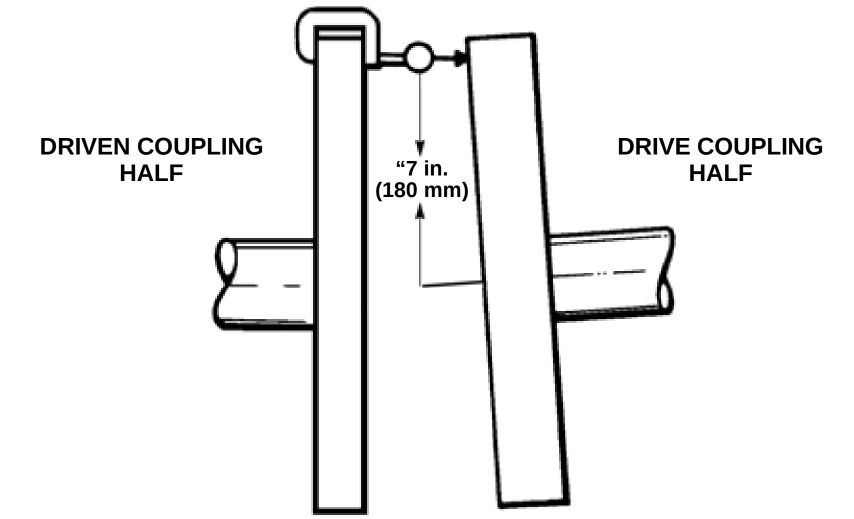


Figure 10-7 Distance from Center of Shaft to Dial Indicator

Before taking readings, roll the shaft 45° in reverse rotation and then back 45° in standard rotation and zero the dial indicator. This sets the axial position for both the engine and driven machine shafts.

To measure angular alignment, four dial indicator readings are required: one each at the 12, 9, 6 and 3 o’clock positions, which are taken while turning the engine in the standard direction of rotation (see Figure 10-8).

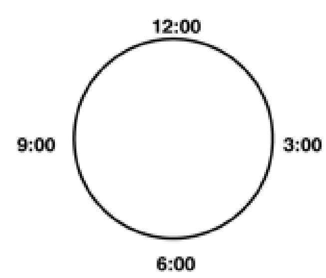


Figure 10-8 Dial Indicator Reading Positions When Measuring Angular Alignment

Readings taken at the 12 and 6 o’clock positions determine vertical angular alignment and readings in the 3 and 9 o’clock positions determine horizontal angular alignment. A total indicator reading (TIR) is the absolute difference between two readings on opposite sides of the shaft (see Figure 10-9). The horizontal TIR is (-0.009 in.) and (+0.004 in.) which is a difference of (0.013 in.). Vertical TIR is (0) and (+0.005 in.) which is a difference of 0.127 mm (0.005 in.).

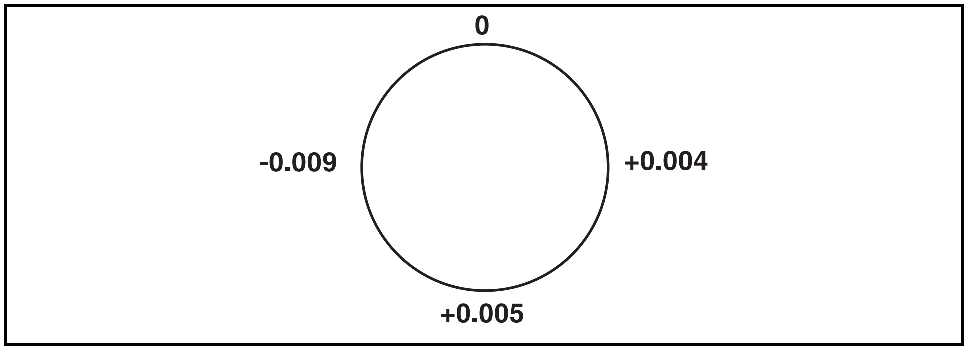


Figure 10-9 Total Indicator Reading (TIR)

Figure 10-10 shows the shaft of a multi-bearing machine with both angular and parallel misalignment.

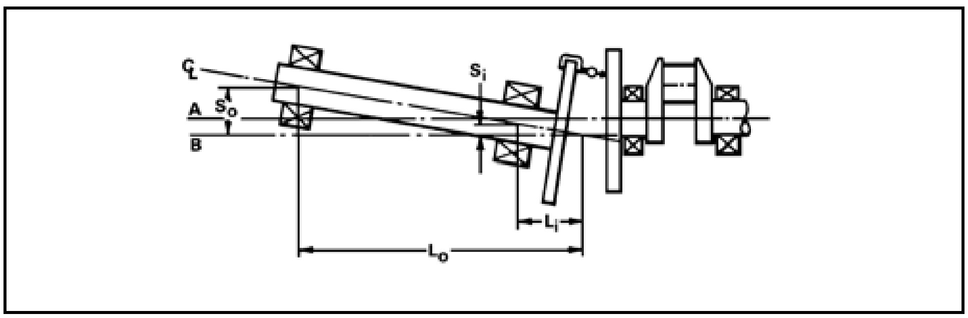


Figure 10-10 Multi-bearing Driven Equipment

This could represent either vertical or horizontal misalignment since the principles are the same for both.

Correcting this misalignment first involves correcting angular alignment, thus getting the shaft centerline to line up on line B.

The amount of correction required to bring the centerline into alignment with line B, can be determined from the dial indicator TIR, radius to the indicator "R", and distance "L" from the coupling to the mounts.

$$\frac{1/2\,(TIR)}{R} = \underset{\text{OUTBOARD MOUNT}}{\frac{S_o}{L_o}} = \underset{\text{INBOARD MOUNT}}{\frac{S_i}{L_i}}$$

Therefore:

$$S_o = \frac{L_o \times 1/2(TIR)}{R}$$

and

$$S_i = \frac{L_i \times 1/2(TIR)}{R}$$

"S_o" is the amount of adjustment at distance "L_o" which is the distance from the center of the coupling to the center of the outboard mount.

"S_i" is then the adjustment at a mount distance of "L_i" from the coupling.

The adjustment should be made to close the open side of the coupling (see Figure 10-11).

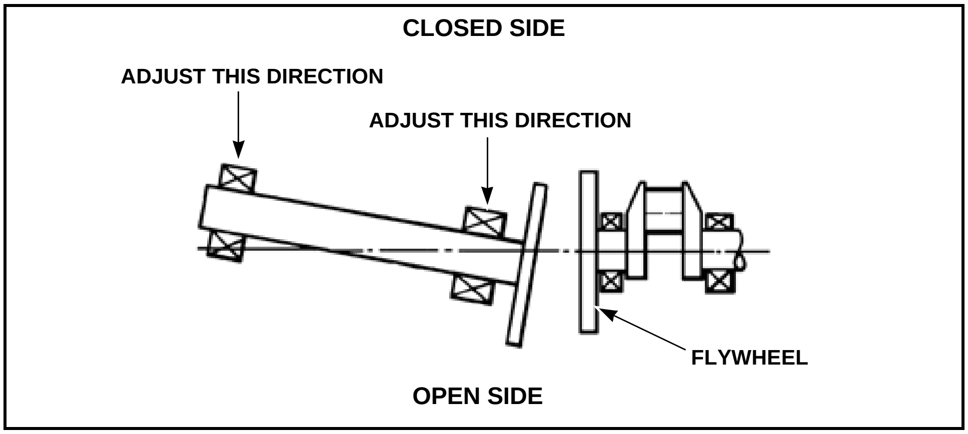


Figure 10-11 Adjusting Coupling

Adjustment for angular alignment should then take place as follows:

1. Set up two dial indicators: one to monitor horizontal movement of the inboard mounts, one to monitor horizontal movement of the outboard mounts. Zero the indicators (see Figure 10-12).

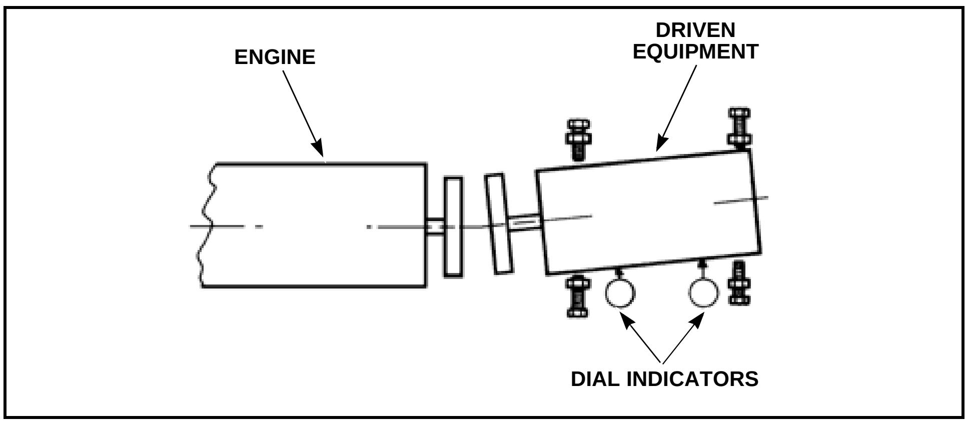


Figure 10-12 Adjusting for Angular Alignment

2. Going to one corner at a time, loosen the mounting bolt and shim as calculated, then tighten the mounting bolt. Center mounts will have to be shimmed in conjunction with corner mounts. Note any horizontal movement that may occur on the dial indicators.
3. After shimming, loosen both mounts on one end and all center mounts. It may also be necessary to loosen one mount on the fixed end but do not loosen both. Slide the free end the amount calculated, then re-torque the bolts (see Figure 10-13).

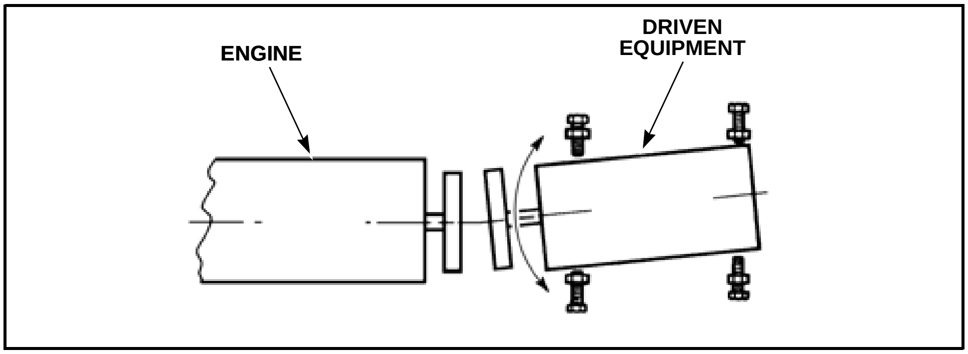


Figure 10-13 Slide Free End

4. Loosen both bolts on the opposite end and move as calculated. Re-torque all mounting bolts (see Figure 10-14).

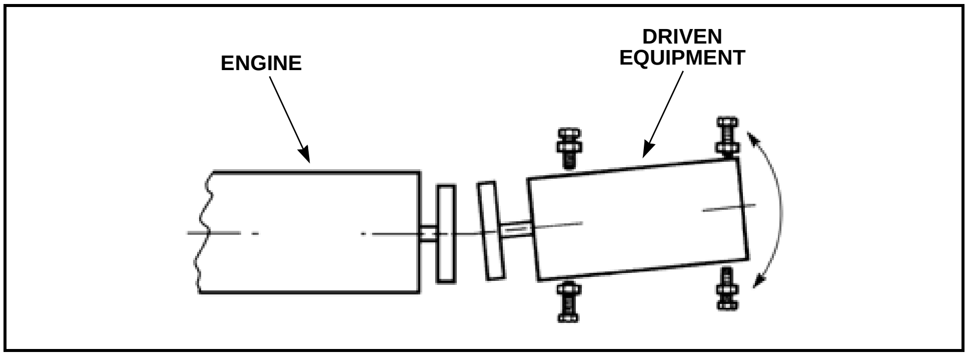


Figure 10-14 Move Opposite End

5. Check angular alignment again using the same procedure as used previously. Angular alignment is correct when total indicator runout is less than 0.127 mm (0.005 in.) per foot of radius from center of shaft to where the dial indicator reads (see Figure 10-15).

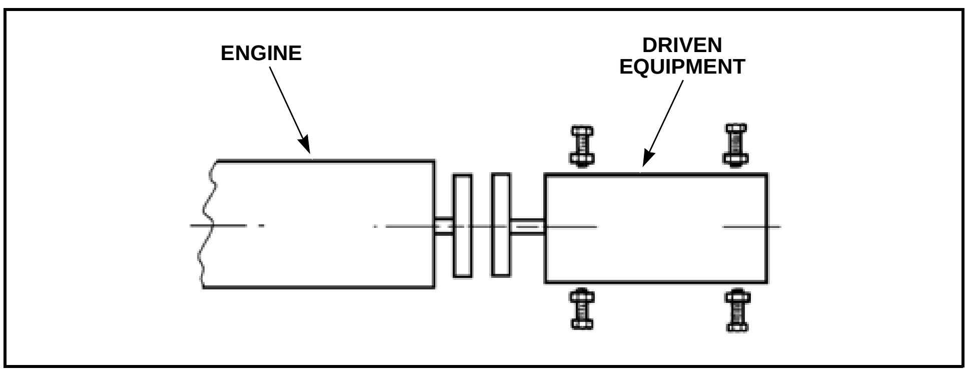


Figure 10-15 Correct Angular Alignment

Parallel Alignment

WARNING

Turn off the air/gas supply to the starters and bleed off air/gas pressure to prevent accidental rotation of the crankshaft before entering the engine crankcase.

Parallel alignment can be checked and adjusted after angular alignment has been completed. It will, however, be necessary to re-check angular alignment after each adjustment (see Figure 10-16).

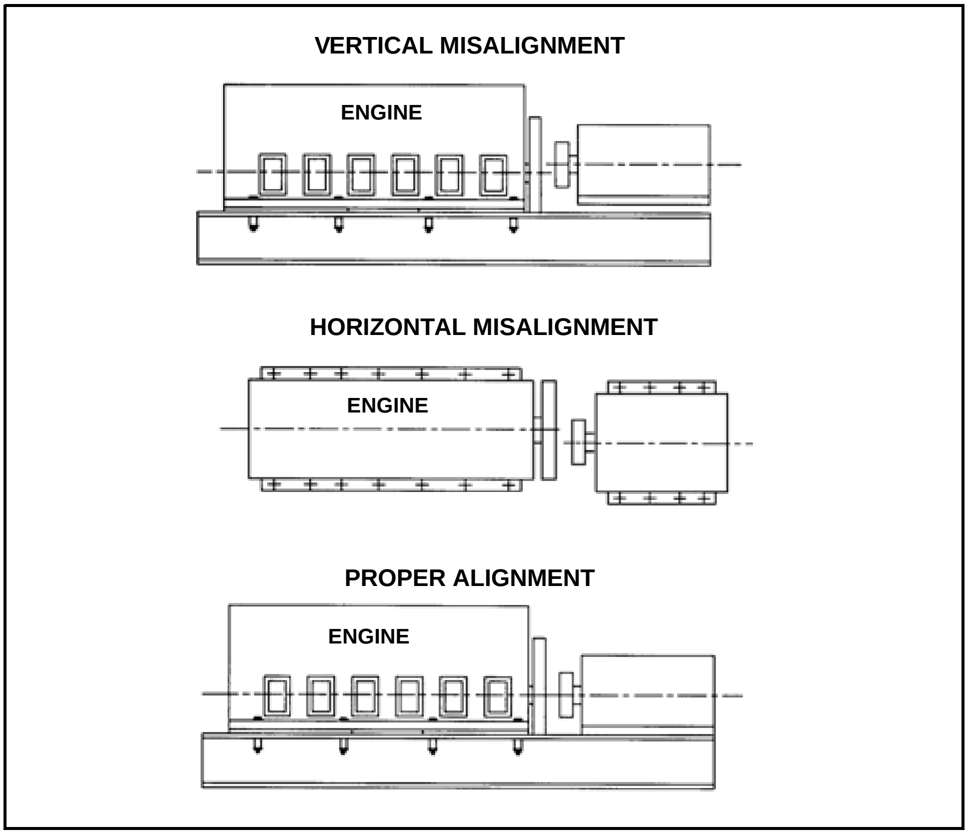


Figure 10-16 Parallel Alignment

The following procedure can be used to measure parallel alignment.

1. Set up a dial indicator to read parallel alignment. If available, setup a second dial indicator to read angular alignment. This will allow you to rotate the shafts only one time to get both readings (see Figure 10-17).

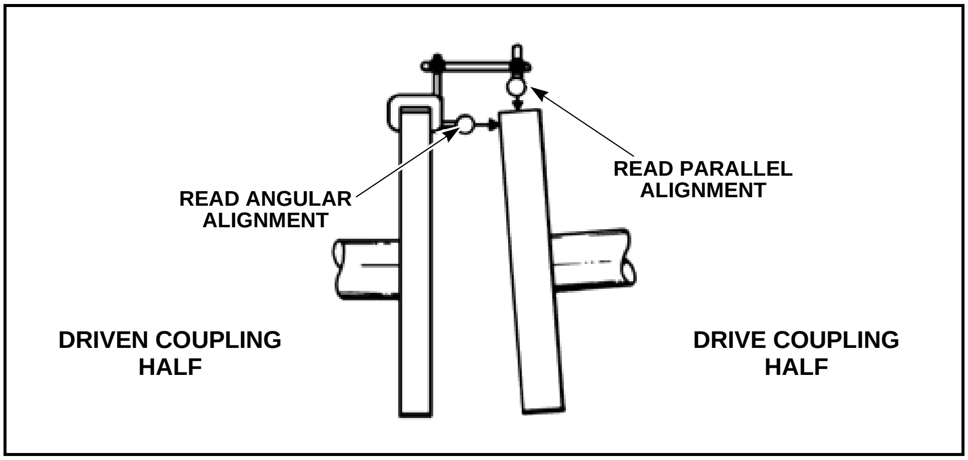


Figure 10-17 Position of Dial Indicator

2. Rotate both shafts to the 2 o'clock position (facing the flywheel) then back to the 12 o'clock position. Zero the indicator(s).
3. Rotate the shafts to the 9 o'clock position and record the readings.
4. Rotate the shafts to the 6 o'clock and 3 o'clock positions and record the readings.
5. Rotate the shafts back to the 12 o'clock position and verify that the indicators return to zero.

The amount of parallel misalignment is one-half the TIR (total indicator reading) for each direction.

See Figure 10-18. In this example, the vertical TIR is 0.508 mm (0.020 in.), thus the machines are vertically misaligned by 0.254 mm (0.010 in.). Horizontal TIR is the difference between (+0.015 in.) and (+0.005 in.) which is (0.010 in.). Horizontal misalignment is 1/2 of the TIR which is (0.005 in.). All mounts should get the same amount of adjustment, 0.127 mm (0.005 in.) in this case, to move the machine without losing angular alignment.

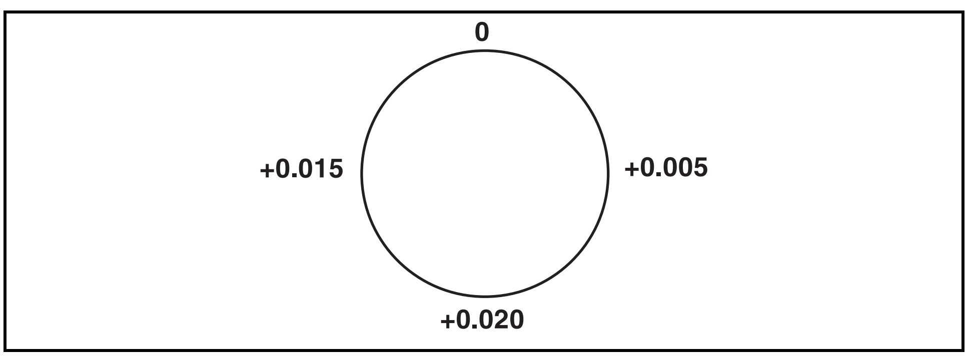


Figure 10-18 Total Indicator Reading (TIR)

Adjustment for parallel alignment is similar to that for angular and should be accomplished as follows:

1. Set up two dial indicators; one to monitor horizontal movement of the inboard mounts, and one to monitor horizontal movement of the outboard mounts. Zero the indicators.
2. Going to one corner at a time, loosen the mounting bolt(s) and shim as calculated, then torque the mounting bolt. Center mounts will have to be shimmed in conjunction with corner mounts.
3. After shimming, loosen both mounts on one end and all center mounts. It may also be necessary to loosen one mount on the fixed end but do not loosen both. Slide the free end the amount calculated then re-torque the bolts.
4. Loosen both mounts on the opposite end and move the same. Re-torque all mounting bolts.
5. Check parallel alignment again using the same procedure as used previously. Parallel alignment is correct when total indicator runout is less than 0.127 mm (0.005 in.).

Thermal Growth

After angular and parallel alignment are satisfactory, it will be necessary to adjust alignment to compensate for engine block thermal growth which affects the height of the crankshaft centerline after the engine is hot. This will allow the machines to be in good alignment after they reach operating temperature.

Table 10-3 lists the changes in crankshaft height that will occur due to the temperature change from 21°C (70°F) to normal operating temperatures (measured from the mounting rail of the crankcase).

The vertical thermal growth in the height of the 275GL crankshaft centerline from the bottom of the crankcase pan rails are listed here in Table 10-3 (based on temperature change from 21°C (70°F) to normal operation oil temperature).

Table 10-3 Vertical Thermal Growth Crankshaft Centerline

ENGINE MODEL	HEIGHT OF CRANKSHAFT CENTERLINE TO BOTTOM OF PAN RAIL	VERTICAL GROWTH IN CRANKSHAFT HEIGHT (Over Range of Normal to High Lube Oil Temperatures)
12V275GL+ 16V275GL+	480 mm (18.898 in.)	0.31 – 0.39 mm (0.012 – 0.015 in.)

Heat growth information for the driven equipment should be available from the manufacturer. If not, it can be calculated with the following formula:

$G_m = (T_m - 70) \times h \times E$ for °F or $(T_m - 20) \times h \times E$ for °C

Where:

G_m = amount of growth expected (inches or mm)

T_m = operating temperature of driven machines (°F or °C)

h = height from machine mounting surface to center of shaft (inches or mm)

E = thermal expansion coefficient for material machine is made from:

6.5×10^{-6} (0.0000065) in/in °F or 1.2×10^{-6} mm/mm °C for steel

5.8×10^{-6} (0.0000058) in/in °F or 1.1 x 10-6 mm/mm °C for cast iron

To adjust for thermal growth take the difference in machine growths and add that amount in shims under the machine which grows least. In the case of cooling compressors, the compressor gets cold when loaded and shrinks. This will require a further offset to compensate for engine growth and compressor shrinkage. The growth formula still applies for a cold compressor since the growth number will be negative.

To add the shims, loosen one side at a time and add the shims then re-torque the bolts before moving on to the next mount. This prevents horizontal alignment from changing while adding shims. Parallel dial indicator readings will now indicate the machine which grows least is higher than the machine which grows more, but the machines will be aligned when they reach operating temperature.

Check endplay to verify that the alignment procedure did not eliminate end thrust.

Hot Check

WARNING

Engine components and fluids are extremely hot after the engine has been shut down. Contact with hot components or fluids can cause severe personal injury or death. Wear protective clothing and eye protection during the hot check of crankshaft deflection.

Once the machines are aligned and offset for thermal growth, they should be checked when hot.

1. Remove alignment measurement instruments.
2. Install safety guards over flywheel and coupling.
3. Start the engine and apply load.
4. Allow machines to run for one hour after reaching their operating temperatures.

WARNING

Ensure that all tools and other objects are removed from the unit and any driven equipment before starting the unit. Running equipment can eject objects at great force.

5. Shut down and immediately check angular and parallel alignment and endplay. Alignment TIR should now be less than 0.203 mm (0.008 in.) both parallel and angular.
6. Adjust alignment and endplay if necessary.

Periodic Inspections

Engine base deflection and alignment must be checked periodically, at least once a year. Installations which are subject to settling of the concrete must be checked often (initially – monthly) to determine if settling is causing misalignment.

Alignment Worksheet

1. Install and level engine or common skid.
2. Measure crankshaft web deflection.

Table 10-4 Web Deflection

DEFLECTION OF WEB	
NEXT TO FLYWHEEL	**ALL OTHER WEBS**
0/-0.105 mm (+0/-0.004 in.)	0.035 mm/-0.035 mm (+0.0015,/-0.0015 in.)

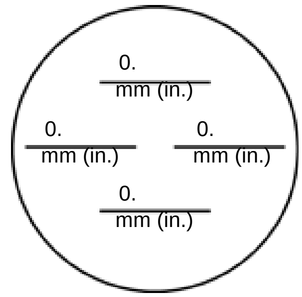
Throw	**1**	**2**	**3**	**4**
TIR	0.____ mm (in.)	0.____ mm (in.)	0.____ mm (in.)	0.____ mm (in.)

Throw	**5**	**6**	**7**	**8**
TIR	0.____ mm (in.)	0.____ mm (in.)	0.____ mm (in.)	0.____ mm (in.)

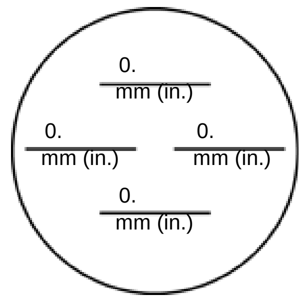
3. Adjust base deflection at four corners of driven machine.

0. mm (in.)	0. mm (in.)
0. mm (in.)	0. mm (in.)

4. Check for crankshaft endplay 0.________ mm (in.)
5. Check and adjust angular alignment. Maximum 0.127 mm (0.005 in.) per foot of radius from center of shaft to dial indicator read point.



6. Check and adjust parallel alignment. Maximum TIR 0.127 mm (0.005 in.).



7. Adjust for thermal growth.

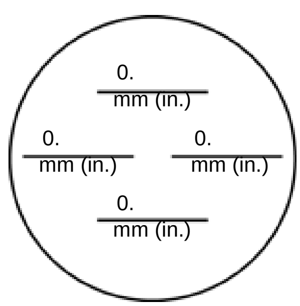
   Engine Growth 0._______mm (in.) minus

   D.M. Growth 0._______mm (in.) = Cold Alignment Offset 0._______mm (in.)
8. Recheck crankshaft endplay 0._______mm (in.)
9. Start engine, run loaded, allow to warm up 1 hour minimum 0._______
10. Shutdown and check hot angular alignment and endplay.

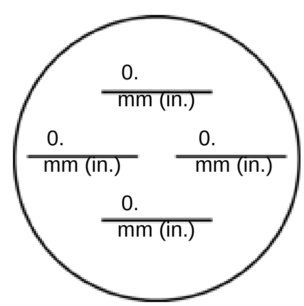
    Endplay (Hot) 0._______mm (in.)

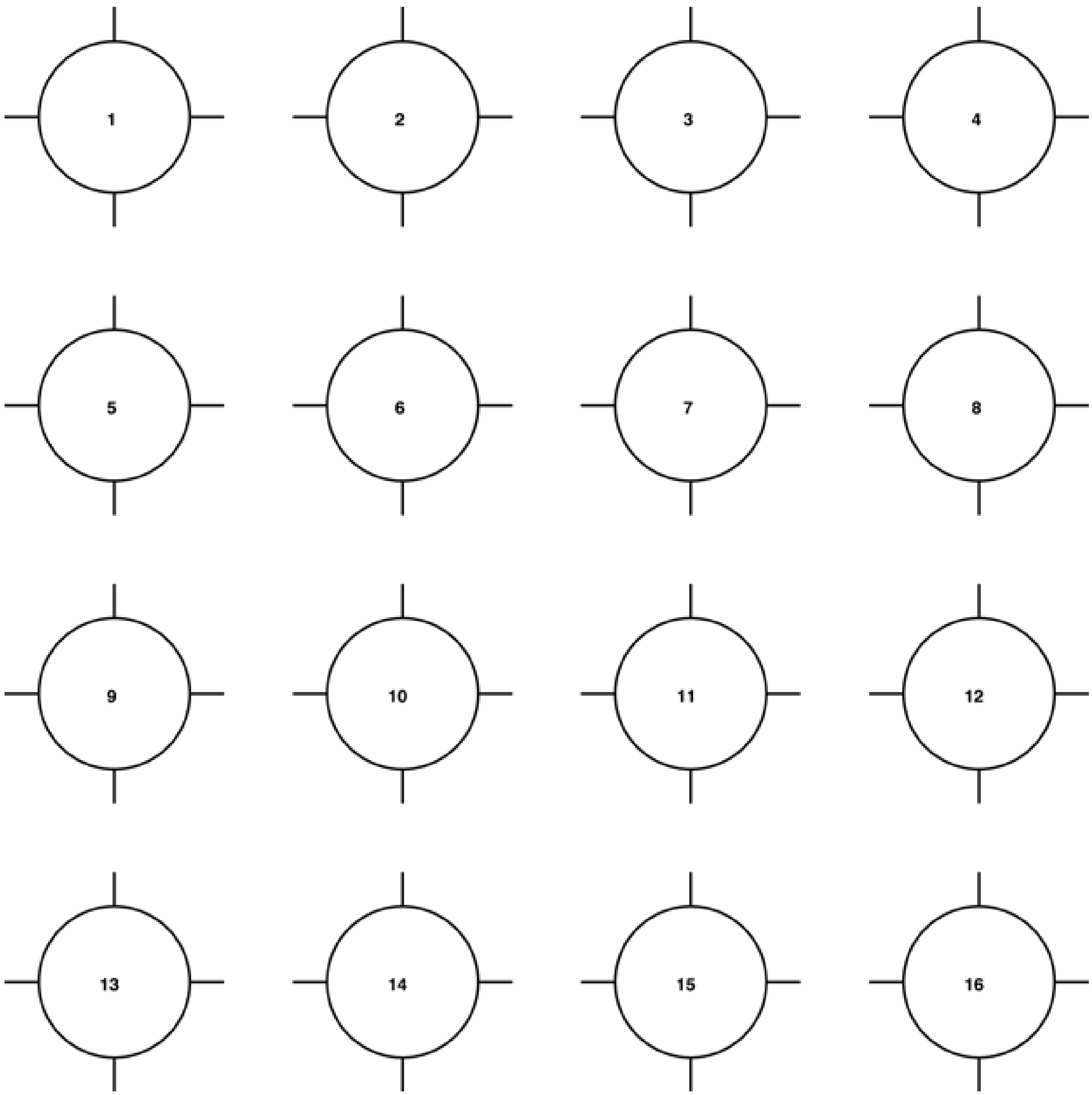
    Alignment:

PARALLEL (HOT)



ANGULAR (HOT)





ENDPLAY BEFORE ALIGNMENT ____________________ **ENDPLAY AFTER ALIGNMENT** ____________________

ENGINE MODEL ________________________________ **SERIAL NO.** ________________________________

DRIVEN MACHINE MAKE __________________________ **MODEL NO.** ________________________________

CUSTOMER'S OBSERVER __________________________ **PERFORMED BY** _____________________________

DATE ____

Figure 10-19 Alignment Progression Chart

SECTION 4

SUMMARY OF MOUNTING AND ALIGNMENT

The intent of this chapter is to establish a step by step procedure to properly install the 275GL engines both in the fabrication shop and at the final operating site.

Fabrication Shop

1. Level the prefabricated skid in the fabrication shop.
2. Prepare the mounting surfaces.
3. Install the engine on the skid. Level the engine using jacking screws (Section 1). Use crankshaft deflection measurements for engine leveling.
4. With the engine in its leveled position, install shims.
5. Remove or loosen the jacking bolts and fasten the engine down with the mounting bolts. Recheck crankshaft deflection and make corrections if necessary.
6. Align the driven machine.

On Site

1. Level the unit skid using crankshaft web deflection and alignment to verify the skid is properly leveled (see Figure 10-20) (see “Mounting Surface” on page 10-1).

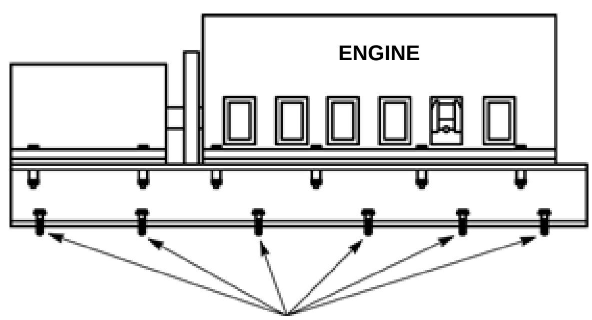


Figure 10-20 Adjust for Minimum Crankshaft Web Deflection per Section

2. Shim or grout under the skid to hold its level position.
3. Fasten the skid with anchor bolts or welding, whichever is applicable.
4. Recheck alignment and crankshaft deflection to verify fastening did not cause distortion. Correct it if necessary.

WARNING

Ensure that all tools and other objects are removed from the unit and any driven equipment before starting the unit. Running equipment can eject objects at great force.

5. Hot Check – Operate the engine for 1 hour minimum after reaching operating temperature. Shut down the engine and recheck crankshaft deflection and alignment.

CHAPTER 10

APPENDIX A

SHIMMING INFORMATION

Waukesha recommends using die cut stainless steel shims for final adjustment of base deflection and alignment when a “Machined skid” or “Sole plates leveled in grout” mounting surface is used. Die cut stainless steel shims provide and accurate mounting and good corrosion resistance. Shims should be available in thicknesses of approximately:

0.05 mm (0.002 in.)

0.125 mm (0.005 in.)

0.250 mm (0.010 in.)

0.750 mm (0.030 in.)

Die cut shims available from Waukesha are listed in Table 10-1.

Table 10-1 Die Cut Shims Available from Waukesha

PART NUMBER	LENGTH	WIDTH	THICKNESS	BOLT SLOT WIDTH
P310062	127 mm (5 in.)	127 mm (5 in.)	4.76 mm (0.1875 in.)	41 mm (1.625 in.)
P310063	127 mm (5 in.)	127 mm (5 in.)	1.52 mm (0.060 in.)	41 mm (1.625 in.)
P310064	127 mm (5 in.)	127 mm (5 in.)	0.635 mm (0.025 in.)	41 mm (1.625 in.)
P310065	127 mm (5 in.)	127 mm (5 in.)	0.254 mm (0.010 in.)	41 mm (1.625 in.)
P310066	127 mm (5 in.)	127 mm (5 in.)	0.076 mm (0.003 in.)	41 mm (1.625 in.)

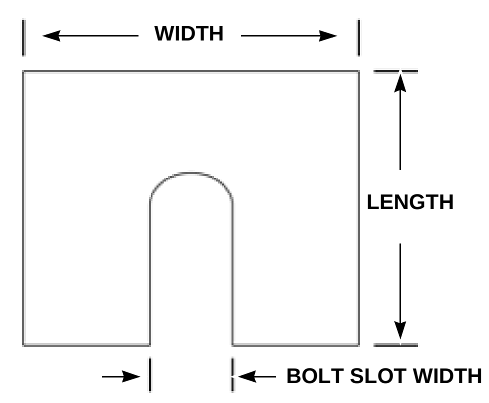


Figure 10-1 Shim Schematic

Shim packs should use thick shims on the outside sandwiching the thin shims on the inside. Adding or removing shims is accomplished by removing the complete shim pack and adding or removing shims as required. (See Figure 10-2.)

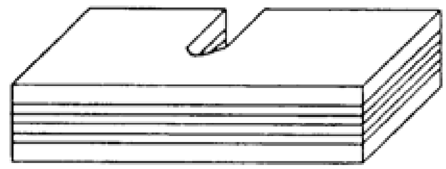


Figure 10-2 Stacking Shims

GROUT INFORMATION

An oil resistant and heat resistant pourable epoxy grout is required for grouting. Grouting must be sized and applied per the grouting manufacturers recommendations. The engine cannot be secured with the mounting bolts until the grout has had ample time to cure.

MOUNTING BOLT AND TORQUING INFORMATION

Grade 5 or 8 mounting bolts 32 mm (1-1/4 in.) should be utilized for engine mounting. The minimum bolt length allowed is 215 mm (8.25 in.). Spacers must be used if necessary to meet the bolt stretch length requirement. Hardened washers are required under the nut (see Figure 10-3). Torque should be based on grout strength and bolt torque limits. Waukesha recommends a minimum of 62.2 kgf·m (450 ft-lb).

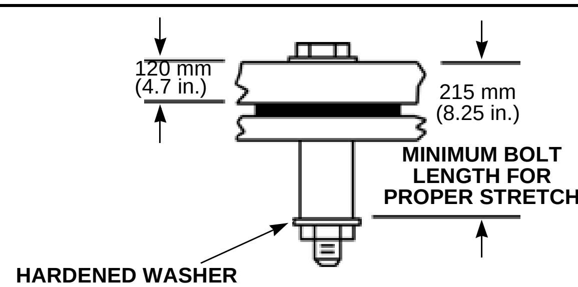


Figure 10-3 Hardened Washer

After the engine has acceptable crankshaft web deflection and alignment and the grout has cured or shims have been installed, the jacking bolts can be replaced with mounting bolts. Alignment and web deflection must be measured again after the mounting bolts are properly torqued to verify the measurements remain within acceptable limits.

Mounting bolts must be torqued in three increments (1/3 torque, 2/3 torque, and full torque) in the pattern indicated in Figure 10-4.

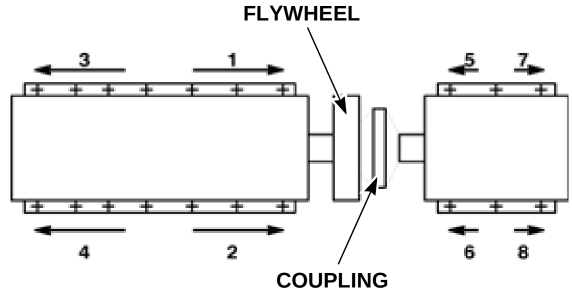


Figure 10-4 Belt Tightening Pattern

When jacking bolt positions will be used for mounting bolts a special provision must be made to block the mounting hole during leveling. This can be accomplished by threading the mounting hole in the skid and inserting a blocking bolt with a locking nut. (See Figure 10-5.)

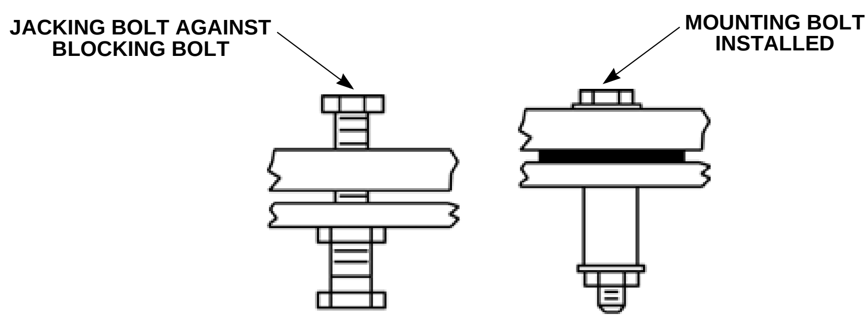


Figure 10-5 Jacking Using Mounting Bolt

SIDE JACKING BOLT INFORMATION

NOTICE **Side jacking bolts must not push between end mounting bolts or crankcase. Rail damage will result.**

Horizontal jacking bolts should be installed for positioning the engine horizontally and axially. The jacking bolts must push against the area indicated in Figure 10-6. Pushing on a different area of the mounting rail will result in damage to this hollow rail.

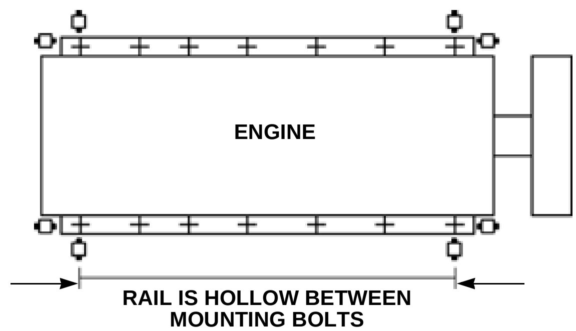


Figure 10-6 Horizontal Jacking

Side jacking devices should be of sufficient size and strength to slide the engine where necessary. Jacking devices designed as shown in Figure 10-7 are suitable for moving the engine.

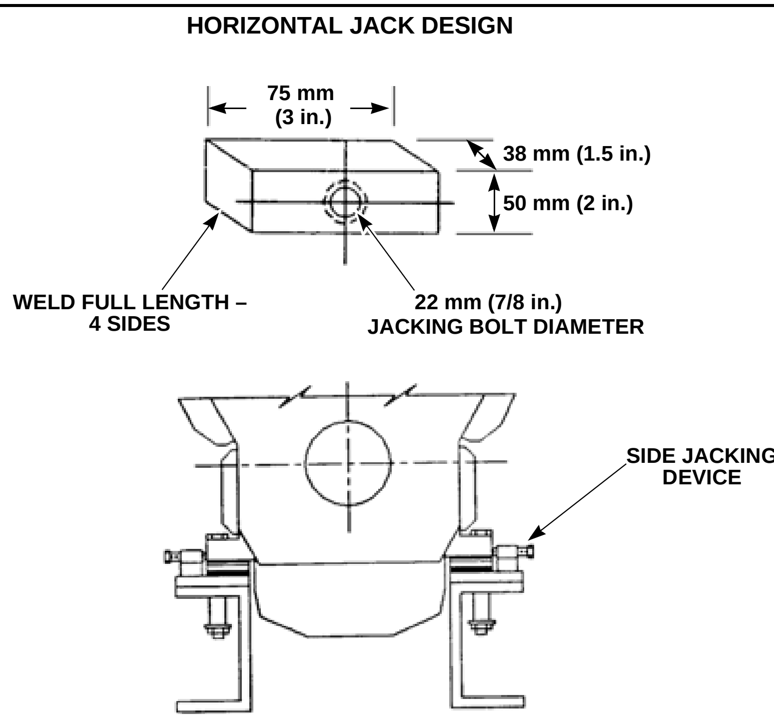


Figure 10-7 Jacking Devices

SKID DESIGN INFORMATION

Proper skid design varies greatly with the type of driven machine, location, skid mounting requirements etc. For this reason, Waukesha cannot make specific skid design recommendations. However, the following general guidelines are provided to assist packagers in designing a skid to fulfill their needs.

The Waukesha 275GL engines have oil sumps which are underslung below the mounting surface of the engine. A flush mounting surface is not possible therefore, and the skid must be designed to accommodate this. To get good support under the engine, the mounting rail should be positioned directly over the web of the skid beam. A C-channel beam or I-beam can be used for the beam. For wide flanged I-beams, it will be necessary to cut off the inner flange at the top of the beam to allow the engine to mount directly above the web. (See Figure 10-8.)

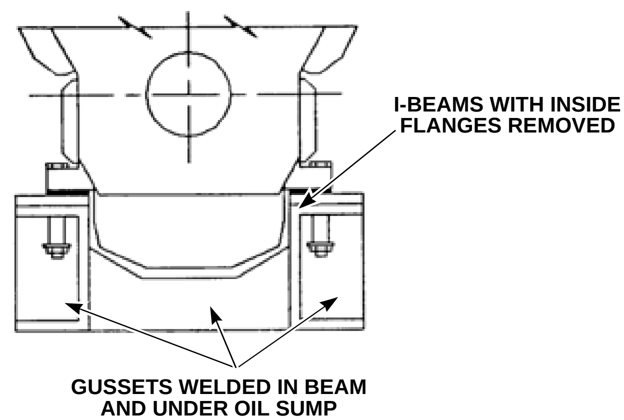


Figure 10-8 Engine Mounted Directly Above Web

Waukesha strongly recommends the packager analyze skid design to determine that the structural integrity of the skid does not incur harmful natural frequencies for constant speed applications and throughout the speed range for variable speed applications.

For more information on skid design and preparation for mounting reference Chapter 2.

Side to side support in the skid must be provided by gussets between the main beams and in the channel or by gussets welded to a wider outside beam. There should be gussets near each mounting bolt position, but sufficient space must be provided for tools to reach the mounting bolts. (See Figure 10-9.)

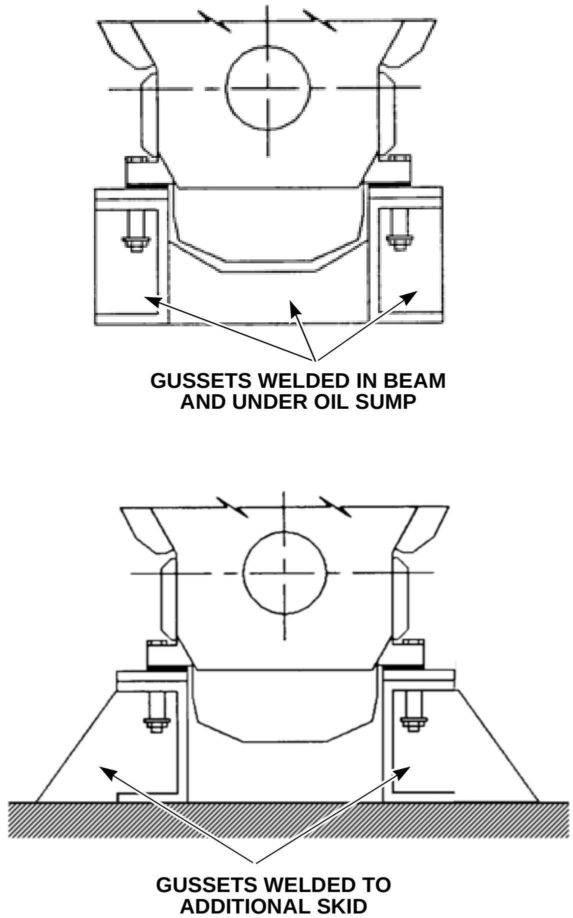


Figure 10-9 Additional Gussets Added

Torque reactions between the engine and driven machine are transferred by the skid. The skid design must have sufficient torsional rigidity to transmit the torque reactions. (See Figure 10-10.)

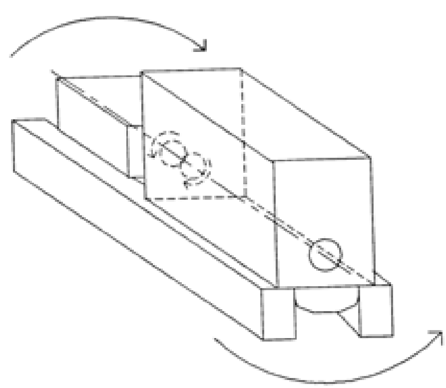


Figure 10-10 Torsional Rigidity in Skid

MAXIMUM UNBALANCED FORCES AND MOMENTS FOR 275GL ENGINES

The tabulated data are based on the possibility that engine components (pistons, rods, crankshaft, etc.) may be machined to maximum and/or minimum permissible weight limits, and that assembly may occur in such a way that maximum unbalanced forces or couples may be produced. Note that these are maximum theoretical values and that standard assembly procedures will rarely produce such extreme conditions. Normally expected values would not exceed 50% of the calculated values shown. Note also that an assembly that produces a maximum unbalanced force will produce a minimum unbalanced couple and vice versa.

Table 10-2 Maximum Unbalanced Inertia Force – Lb (N)

ENGINE	900 RPM			1000 RPM		
	HORIZONTAL	VERTICAL	AXIAL	HORIZONTAL	VERTICAL	AXIAL
12V275GL+	917 (4079)	1550 (6894)	0	1132 (5035)	1913 (8509)	0
16V275GL+	1277 (5680)	2027 (9016)	0	1577 (7015)	2503 (11133)	0

Table 10-3 Maximum Unbalanced Inertia Force – Lb-Ft (N·m)

ENGINE	900 RPM			1000 RPM		
	HORIZONTAL	VERTICAL	AXIAL	HORIZONTAL	VERTICAL	AXIAL
12V275GL+	2620 (3552)	4154 (5632)	0	3235 (4386)	5128 (6952)	0
16V275GL+	4684 (6350)	7128 (9664)	0	5783 (7840)	8800 (11931)	0

Unbalanced forces or moments at any other speed, N, may be calculated as:

$Force_N$ $(Moment_N)$ = $Force_{1000}$ $(Moment_{1000})$ x $N/1000)^2$

NOTES

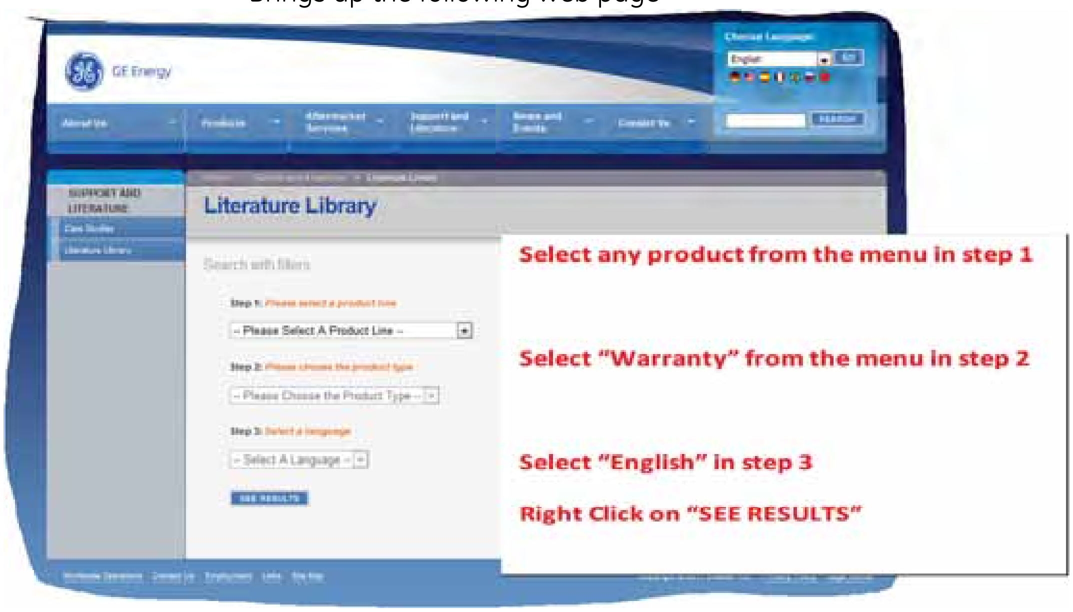
The Waukesha Express Limited Warranty is available by:

Requesting a copy from:

GE Energy
1101 West Saint Paul Avenue
Waukesha, WI 53188-4999
P: 262.547.3311
F: 262.549.2759

OR

Enter the following into your browser:
http://www.dresserwaukesha.com/index.cfm/go/literature-search/
Brings up the following web page



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