CATERPILLAR

DECEMBER 1984

ON-SITE POWER GENERATION

HANDBOOK

CATERPILLAR ENGINE DIVISION

ON SITE POWER GENERATION HANDBOOK

CONTENTS

INTRODUCTION

On-Site power generation using internal combustion engines has gained economic feasibility and attractiveness in recent years largely by virtue of the advances made in the art of recovery and use of rejected heat. Broadly summarizing, the thermal efficiency of these engines, combined with the cost of a suitable fuel has, in the past, imposed limitations upon the competitive capability of on-site power except when reliability or remoteness of location were prime factors. However, when the thermal efficiency of the complete energy system is improved materially, the situation changes — sometimes dramatically.

In the operation of reciprocating, internal combustion engines, approximately 32% of the heat energy in the fuel is converted into work or shaft output power when operating at or near rated load. The remaining energy in the fuel is rejected in the form of heat.

To improve the thermal efficiency of an on-site energy system, part of the rejected heat must be recovered and put to useful purposes. The most convenient source of recoverable heat is the heat rejected to the jacket water or cooling system. Essentially 100% recovery of this normally-wasted heat is feasible. The second source of recoverable heat is the exhaust, approximately 60% of which is economically recoverable. Further heat recovery, while possible, is seldom economically attractive.

How much does heat recovery affect the efficiency of on-site energy systems? The overall or plant efficiency can reach, by utilizing heat recovery, 70% or more as opposed to 25% to 30% without heat recovery. This increased efficiency then enables on-site power to successfully compete with utility power on an economic basis in many instances.

Shaft power may be used to generate electricity, drive a compressor for air conditioning, refrigeration or industrial compressed air, drive a pump, or for any other normal engine load. The recovered heat may be utilized in absorption air conditioning, industrial heating processes, or to heat the building or domestic hot water supply.

Two prime factors affect the economics of any cogeneration application. These are (1) the plant load factor and (2) the relationship of fuel cost to the cost of purchased energy. Considering these factors, the ideal application would be one having a high load factor for electric power usage and a favorable steam or heat utilization factor along with a relatively low fuel cost as compared to purchased electric power. One factor being very favorable can compensate for the other being something less than favorable. A manufacturing plant working three shifts per day would generally represent a very favorable load factor, while a school or office building operating five days a week, eight hours per day, might not present an especially favorable load factor. Each, however, could very well be economically feasible for cogeneration application should the cost of fuel be low compared to the cost of purchased electric power.

The market opportunities for cogeneration application are practically unlimited. Shopping centers, apartment complexes, schools, office buildings, hospitals and industrial facilities all offer excellent potential for cogeneration application. The economic advantage to be realized — the return on invested capital — is the outstanding sales feature of the concept. Tax exempt institutions expecially offer ideal economics as a rule. The need, however, for a reliable source of power becomes of prime importance in many instances.

Hospitals, airports, and many manufacturing concerns must have a reliable source of power. Thus, some of the initial first cost of a cogeneration plant can be "written off" against the cost of this requirement, that is, the cost of standby power when utility power is used.

Acceptance of the cogeneration concept has been substantially accelerated by the performance record of the early or "pioneer" installations, which have demonstrated quite clearly that on-site power, when properly applied can yield very attractive savings over purchased utility power. Other contributing factors to the growth and progress of this movement are (1) the development of compact, dependable heat recovery equipment, (2) greatly improved prime mover governors assuring precise frequency regulation, (3) the wide-spread and increasing demand for year round "climate control", (4) the availability of good quality, attractively priced fuel, and (5) the availability of dependable and efficient prime movers in a wide range of sizes, and (6) improved knowledge of where and how to utilize this sytem.

It shall not be the purpose of this publication to explore all of the design and application possibilities of the cogeneration concept nor to design such a plant. The material and data presented on the pages that follow have been assembled simply to call attention to a number of basic requirements associated with the design of cogeneration systems and to provide, to the extent possible, performance data to assist the designer.

FUELS

Few performance features of a prime mover are as important as fuel economy. Since fuel cost is usually the largest expense item associated with the production of power, an understanding of the chemistry of combustion and a working knowledge of the terms and methods used to define and express performance of internal combustion engines is highly desirable for anyone attempting an appraisal of power costs. This is especially true where the cost of utility electric power must be compared with the cost of power generated by internal combustion engines.

GASEOUS FUELS

Not the least important of the expressions used in connection with gas engine performance are two terms associated with all hydrocarbon fuels and especially with natural gas, namely high heat value (HHV) and low heat value (LHV). An understanding of heat value is an absolute essential when calculating or determining fuel consumption of gas engines.

When natural gas is used as fuel in an internal combustion engine, one of the products of combustion is water. The amount of water formed during combustion varies with different gases, depending upon the kind and mixture of hydrocarbons which constitute the gas. The water so formed obviously is converted into steam before leaving the engine and thus carries with it the quantity of heat which is absorbed in changing water at a given temperature to steam or vapor at the same temperature, called the "latent heat of vaporization," is lost to the engine. Since the exhaust temperature is always above the dew point the engine has no opportunity to convert this heat into work.

The total heat generated by the combustion of a given quantity of gas, usually one cubic foot or one cubic meter, is known as the high heat value (HHV). The low heat value (LHV) of a gas is the high heat value less the heat used to vaporize the water formed by combustion.

Since the amount of heat (per $ft.^3$ or m^3 of gas) lost in vaporizing the water is different for different gases, the only choice the engine manufacturer has if he is to provide reliable fuel consumption data, is to eliminate this variable and use only the low heat value as a basis for published fuel consumption data. Only the low heat value of a fuel can be utilized by the engine to produce work. This explains why all engine manufacturers use the low heat value for gaseous fuels.

A brief examination of the combustion equation using, for example, pure methane (CH_4), the main constituent of natural gas, will illustrate this point further. The equation for combustion of methane is as follows:

$$CH_4 + 2\,O_2 = 2\,H_2O + CO_2$$

To determine the amount of water formed, first determine the molecular weight of each gas as noted here:

$$\begin{array}{c} CH_4 + 2\,O_2 = 2\,H_2O + CO_2 \\ 16 + 64 = 36 + 44 \end{array}$$

The proportions of the mass (weight) of the chemicals used in a chemical reaction are the same as the relationship of their molecular weights. Thus when 16 lb of methane and 64 lb of oxygen are combined during combustion 36 lb of water and 44 lb CO_2 are formed. For each pound of CH_4 burned the amount of water formed is: $36 \div 16 = 2.25$ lb. Likewise 16 kg of methane and 64 kg of oxygen combine to form 36 kg of water and 44 kg of CO_2; 2.25 kg of water per kilogram of CH_4.

To determine the amount of water formed per unit volume of CH_4 burned, divide 2.25 by the ft^3/lb or m^3/kg at standard conditions of temperature and pressure. For methane:

$$1 \text{ lb} = 23.61 \text{ ft}^3$$
$$1 \text{ kg} = 1.474 \text{ m}^3$$

Therefore, $2.25 \div 23.61 = 0.09529$ lb of water is formed per ft^3 of methane burned or $2.25 \div 1.474 = 1.526$ kg of water per m^3 of methane burned. Then the difference between high and low heat value for CH_4 is the heat required to convert the water to vapor at standard conditions (14.696 psi and 60 °F or 101.33 kPa and 15.5 °C). The latent heat of vaporization of water is 1059.9 Btu per pound or 2466.5 kJ/kg. Therefore the difference between HHV and LHV for CH_4 is $0.09529 \bullet 1059.9 = 101$ Btu or $1.576 \bullet 2466.5 = 3763$ kJ. For commercial pipeline gas, the low heat value, if not known, can be calculated with acceptable accuracy by multiplying the heat value by 0.90.

If we are to equate gas engine performance and operating costs with the cost of power from other sources, it follows that we must acquaint ourselves with the methods and practices employed in the industry to determine fuel consumption and cost as related to load. It is customary to express fuel consumption for gas engines in terms of Btu (low heat value) per horsepower hour or kilojoules per kilowatt hour (kJ/kW•h). This is known as specific fuel consumption and is usually published or presented graphically — as illustrated by Figure 1.

Such curves are referred to as "Gas Engine Generator Set Performance Curves." These curves provide important data on the engine at 25, 50, 75 & 100% load.

To determine total fuel consumption per hour for a known load, first determine the specific fuel consumption in energy (Btu or kJ) (LHV) per unit of output (hp/h or kW/h). For example using Figure 1, assume a 375 kW (75%) load. Locating 75% load on the abscissa and following it vertically to the fuel load rate curve we find the two intersecting on a horizontal line corresponding to 4644 ft^3/h or *131.43* m^3/h. This is the fuel rate per hour based on gas having a LHV of F905 Btu/ft^3 or *33.74* kJ/ℓ. This information is listed in 25% load increments under "Performance Data." Brake Specific Fuel Consumption (BSFC) is also listed. A complete study of the information shown by Figure 1 reveals that fuel rate and/or BSFC can be calculated for any incremental load.

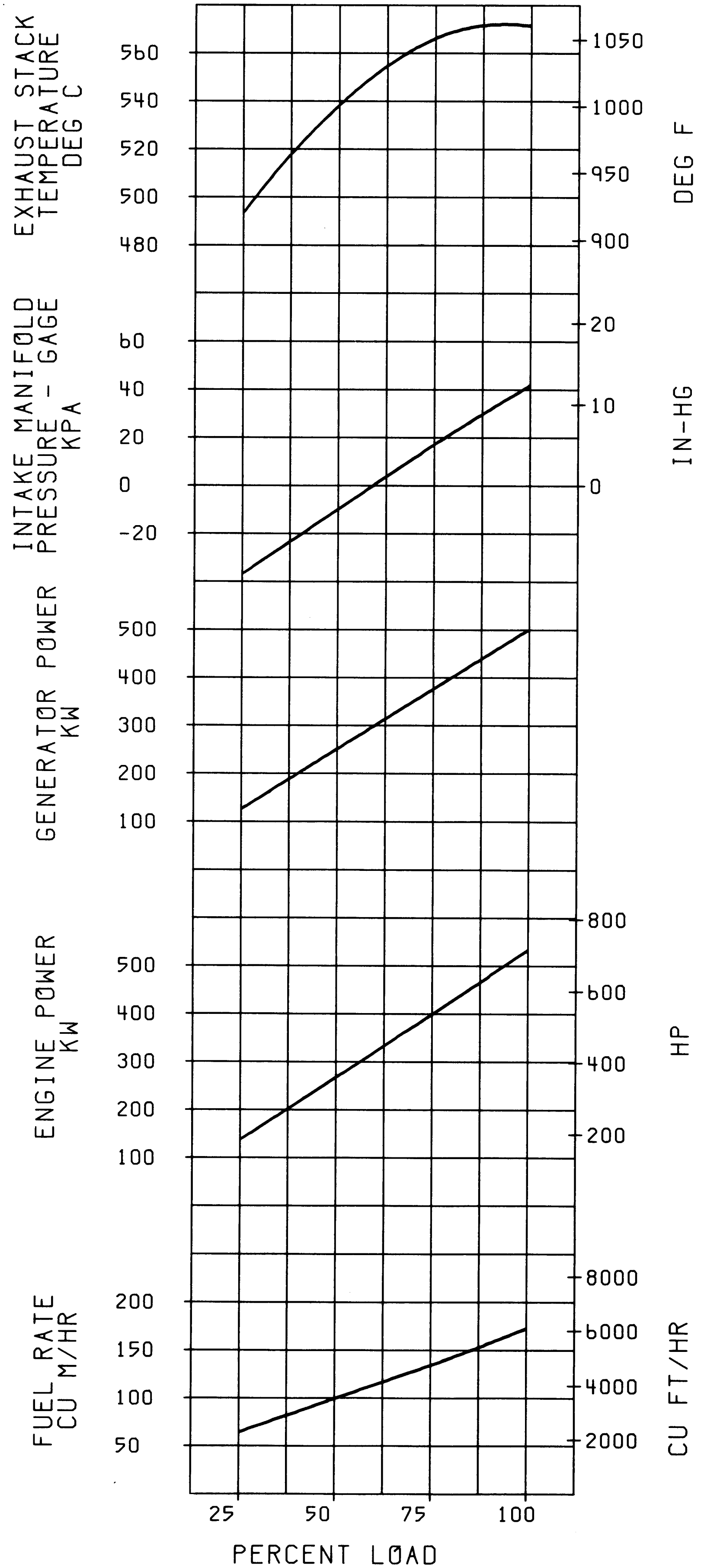
Since gas is sold on the basis of high heat value, it is desirable to convert the Btu or kJ consumption per hour (LHV) to the units of measurement used by the seller. This can be done with the following equation:

$$\text{cu. ft. per.hr.} = \frac{\text{Specific fuel consumption in Btu (LHV)} \times \text{Hp load}}{\text{LHV of fuel to be used.}}$$

$$m^3 = \frac{\text{Specific fuel consumption in kJ (LHV)} \times \text{kW load}}{\text{LHV of fuel to be used.}}$$

PROPANE

Since some gas engine installations will use propane as fuel, comparative power cost studies will often require fuel consumption and cost data for operation on propane. The same fuel consumption curves used for natural gas can be used for propane, however, since propane is sold by the gallon, the fuel consumption must ultimately be expressed in gallons. This may be easily done. It is only necessary to determine the total Btu (LHV) required by the



ENGINE DATA

ENGINEERING MODEL	E162
ASPIRATION	TA
AFTERCOOLER	SCAC-32 (90)
EXHAUST MANIFOLD	WET
COMBUSTION SYSTEM	SI
TURBO MODEL	4MD455-2.6
COMPRESSION RATIO	10.0 TO 1
TYPE OF DUTY	CONTINUOUS
RATED KW	500
RATED HZ	60
RATED RPM	1200
EFFECTIVE SERIAL NUM	73B1056

TOLERANCES

CURVES REPRESENT TYPICAL VALUES OBTAINED UNDER NORMAL OPERATING CONDITIONS. AMBIENT AIR CONDITIONS AND FUEL USED WILL AFFECT THESE VALUES. EACH OF THE VALUES MAY VARY IN ACCORDANCE WITH THE FOLLOWING TOLERANCES

EXHAUST STACK TEMPERATURE	±42 DEG C ±75 DEG F
INTAKE MANIFOLD PRESSURE-GAGE	±1.7 KPA ±0.5 IN-HG
POWER	±3 PERCENT
FUEL RATE	±0.707 MJ/KW-HR ±500 BTU/HP-HR

PERFORMANCE DATA

PERCENT LOAD	ENGINE POWER KW	ENGINE POWER HP	GEN POWER AT 0.8 PF KW
100	532	713	500
75	398	534	375
50	266	357	250
25	136	182	125

PERCENT LOAD	BSFC MJ/KW-HR	BSFC BTU/HP-HR	FUEL RATE CU-M/HR	FUEL RATE CU-FT/HR
100	10.90	7703	171.87	6073
75	11.32	8000	133.54	4719
50	12.52	8848	98.71	3488
25	15.78	11152	63.61	2248

AIR FLOW AT FULL LOAD

INLET 32.8 CU M/MIN 1160 CFM

EXHAUST 93.0 CU M/MIN 3280 CFM

CONDITIONS

PERFORMANCE BASED ON SAE J1686 STANDARD CONDITIONS OF 99.2 KPA (29.38 IN HG) AND 30 DEG C (85 DEG F). PERFORMANCE ALSO APPLIES AT DIN 6270 STANDARD CONDITIONS OF 97.8 KPA (28.97 IN HG) AND 20 DEG C (68 DEG F).

FUEL RATE IS BASED ON GAS HAVING A LHV OF 33.74 KJ/LTR (905 BTU/CU FT)

ENGINE POWER CURVE REPRESENTS THE POWER REQUIRED FOR DRIVING A CATERPILLAR GENERATOR USING AN ENGINE EQUIPPED WITH LUBE OIL AND JACKET WATER PUMPS BUT WITHOUT FAN

THE GENERATOR POWER CURVE REPRESENTS THE ELECTRICAL OUTPUT OF THE GENERATOR

STACK TEMPERATURE, INTAKE MANIFOLD PRESSURE AND FUEL RATE ARE BASED ON ENGINE POWER CURVE

NO ENGINE DERATION IS REQUIRED FOR AMBIENT TEMPERATURES UP TO 52 DEG C (125 DEG F) EXCEPT AS SHOWN ON THE APPLICABLE ALTITUDE DERATING CURVE.

engine per hour for a given load, then divide this figure by the number of Btu (LHV) per gallon of propane (84,190). Thus, for example, a load requiring 4,202,820 Btu/h (LHV) would require:

4,202,820 ÷ 84,190 = 49.92 gal. Propane per hour.

COMPRESSION RATIO (GAS ENGINES)

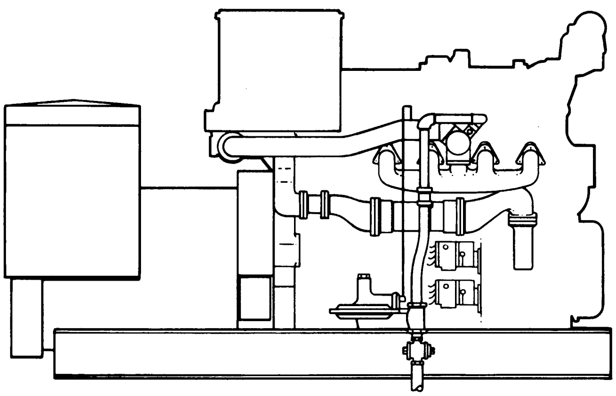


Figure 2

A wide variety of gaseous fuels can be used with Cat gas engines since each model is offered in two different compression ratios. The respective ratios are generally referred to as "high" and "low" and each has its advantages. The variety of range of gaseous fuels which can be used with the high compression ratio units is somewhat limited, however, the specific fuel consumption of these units is very favorable.

While the specific fuel consumption of low compression units is considerably higher, and thus less favorable, they have the advantage of being able to burn a much broader range of fuels.

Naturally aspirated (NA) engines usually require a minimum gas pressure of 2 psi (14 kPa) to the gas regulator, while turbocharged units demand at least 12 psi (83 kPa). These values may be lower only at low altitude installations where response to sudden load changes is not critical.

The addition of another regulator and a change of carburetor may be necessary to accommodate the particular gas.

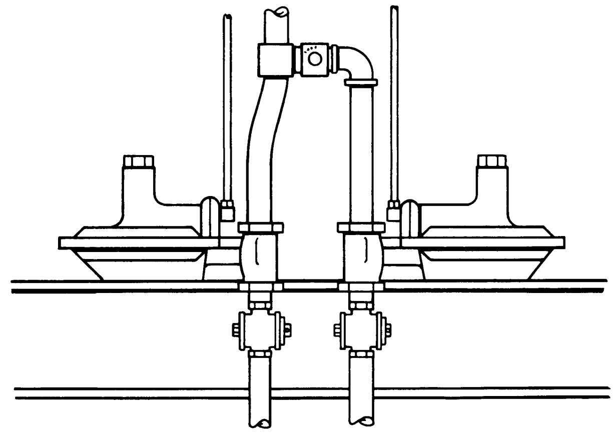


Figure 3

The high compression ratio units are intended primarily for use with fuels that have high anti-knock qualities such as dry or processed natural gas (commercial pipeline gas) and sewage gas which is primarily methane. The low compression ratio units are used for other fuels which are known to be more prone to detonate. Of course, pipeline gas and sewage gas can be used with either high or low compression engines. Fuels which contain more than approximately 5% (by volume) hydrocarbons heavier than propane should be used only in low compression engines in order to avoid serious detonation. Also, since detonation increases with high air-fuel charge temperatures, the aftercooler water temperature for any turbocharged gas engine should be maintained at, or below, the temperature specified by the manufacturer for the particular engine. For example, Caterpillar high compression, turbocharged engines operating on natural gas require aftercooler water 90 °F. (32 °C) or less. For low compression, turbocharged engines, the aftercooler water should be 130 °F. (54 °C) or less. See Table 1 for additional data on fuels.

Gaseous Fuels for Use in Cat Gas Engines:

The following paragraphs describe several fuels along with the suggested types of engine configuration suitable for each.

Dry, Processed Natural Gas — 1000 Btu per Cu. Ft. (HHV) 37.27 MJ/m³

Commercial pipeline natural gas is composed primarily of methane and ethane. Any liquids present are removed prior to pipeline transmission. It can be used in all types of engines. The required aftercooler water temperatures are given in Table 1.

Propane HD5 or Equivalent — 2500 Btu per Cu. Ft. (HHV) 93.18 MJ/m³

Propane of HD5 or equivalent quality can be used in all natural gas engines when necessary adjustments are made; but high compression ratio engines, both naturally aspirated and turbocharged-aftercooled, are limited to non-lug applications. The fuel should not contain more than five percent (by volume) hydrocarbons heavier than propane. The required aftercooler water temperatures are given in Table 1.

Butane — 3200 Btu per Cu. Ft. (HHV) 119.26 MJ/m³

This fuel is recommended only for low compression ratio, naturally aspirated engines. The timing should be retarded from the timing recommended for natural gas, the exact setting being determined by "trial and error" until detonation is eliminated.

Natural Gas with Propane-Air Added — 1000 Btu per Cu. Ft. (HHV) 37.27 MJ/m³

Many gas utilities add a mixture of propane-air to the natural gas during peak demand periods. This "modified" natural gas can be used in *naturally aspirated* engines with either compression ratio without any limitation. If aftercooler water at 130 °F. (54 °C) or less is available, the low compression ratio, turbocharged-aftercooled engine can be used. If the amount of propane-air added does not exceed 35 percent of the mixture volume, the high compression ratio, turbocharged and aftercooled engine can be used with 90 °F. (32 °C) water to the aftercooler. Should the proportion of propane-air mixture exceed 35% of the total volume, the high compression, turbocharged engines would require aftercooler water at a temperature of 70 °F. (21 °C) or less to operate satisfactorily and the timing should be retarded until no audible detonation could be detected.

Sewage Gas — 600 Btu per Cu. Ft. HHV (Avg.) 22.36 MJ/m³

Sewage or "sludge" gas consists primarily of methane, air, and inert gases. Its HHV is lower than the HHV of natural gas because of the presence of inert gases. Cat engines must be equipped with a digester gas carburetor to utilize low Btu (500-800 Btu/ft³HHV) (18.64 — 29.82 MJ/m³) sewage gas as a fuel. It is necessary to derate the standard high compression ratio naturally aspirated engine by approximately 10% and turbocharged engines by 5% when equipped with the digester gas (DG) carburetors. Standard high compression ratio natural gas engines also require approximately a 5° timing advance from the standard natural gas engine timing.

Natural Gas with Hydrogen Added — 800 Btu per Cu. Ft. Ave. (HHV) 29.82 MJ/m³

"Manufactured gas" is sometimes added to natural gas during peak demand periods resulting in hydrogen being mixed with the natural gas. When the hydrogen content is less than 50 percent by volume, the proper engine configuration can be found from Table 1. If the hydrogen content is more than 50 percent by volume, the fuel is not recommended for use in any type of Caterpillar gas engine.

FIELD GAS

This fuel comes directly from the gas well (unprocessed) and usually contains "heavy ends," butane and heavier, in excess of 5 percent by volume. If any liquids are present, they should be removed in a scrubber. A scrubber will prevent heavy ends from reaching the carburetor in the liquid form; however, it will not remove the vapor from these liquids from the gas stream. This fuel can be burned in low compression ratio engines, naturally aspirated or turbocharged-aftercooled; however, the timing must be retarded sufficiently to eliminate detonation.

STANDBY FUELS FOR NATURAL GAS ENGINES

When the possibility exists that the natural gas fuel supply will be interrupted, provisions can be made to operate the engines on standby propane. Digester gas engines can also be equipped to operate alternately on natural gas when necessary. For this combination, natural gas is supplied to the carburetor at a positive pressure by the regulator. Propane, in the vapor form, must be supplied to the carburetor with a negative gauge pressure. Therefore, a separate regulator and fuel piping system is required for the propane fuel. Propane should always be supplied to the engine in vapor form when the engine is located in a building or structure. For engines operating outdoors, a propane vaporizer can be located at the engine.

Propane fueled engines require that the standard natural gas engine timing be retarded. This operation involves rotating the magneto to adjust the timing. For this reason, the switch-over from natural gas to propane cannot be done automatically. However, when natural gas is used as a standby fuel for sewage gas engines, no change of timing is required although for long periods of operation, some adjustment is usually desirable to insure best fuel economy.

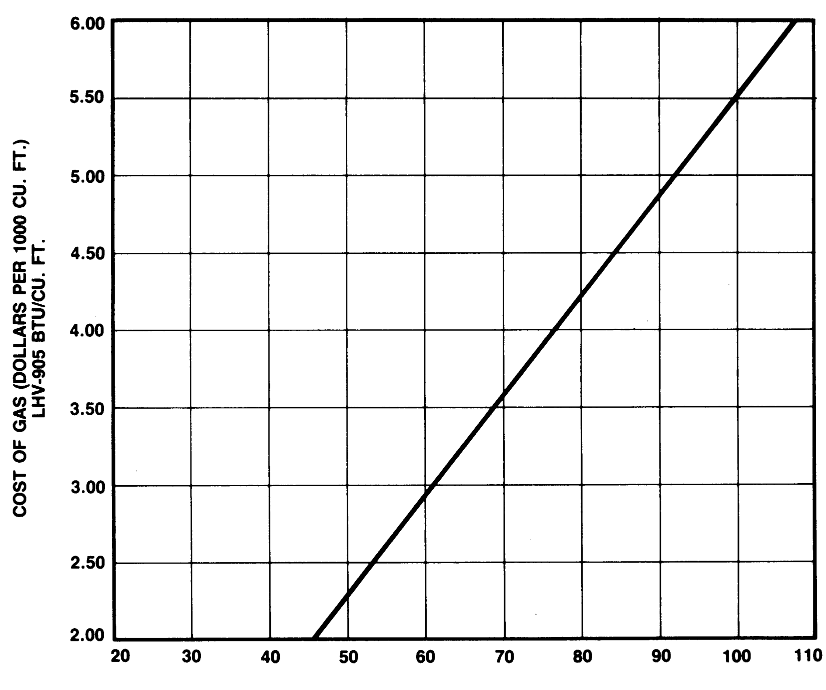


Figure 4 COMPARATIVE COST OF NATURAL GAS & DIESEL FUEL

TABLE 1

SUMMARY — FUELS VS. ENGINE SPECIFICATIONS

	NATURALLY ASPIRATED		TURBOCHARGED-AFTERCOOLED	
Fuel	**High Comp. Ratio**	**Low Comp. Ratio**	**High Comp. Ratio**	**Low Comp. Ratio**
Dry, Processed, Natural Gas . . .	X	X	X (1)	X (2)
Propane	X (3 & 5)	X	X (1, 3 & 5)	X (2 & 3)
Butane		X (3)		
Natural Gas W/Propane-Air .	X (3)	X	X (1, 3 & 4)	X (1 or 2 & 3)
Sewage Gas . . .	X (6)	X (6)	X (1 & 6)	X (2 & 6)
Natural Gas W/Hydrogen, Where:				
H_2 Greater than 50% . . .	Not Recommended			
$H_2 = 50\%$ . . .		X (3)		
$H_2 = 30\%$ . . .	X (3)	X (3)		
$H_2 = 20\%$ . . .	X (3)	X (3)		X (1 & 3)
$H_2 = 10\%$ . . .	X (3)	X (3)	X (1 & 3)	X (1 & 3)
Field Gas		X (3)		X (2 & 3)
Sour Field Gas .	Consult Manufacturer			

Note 1 — Temperature of water to aftercooler not to exceed 90 °F (32 °C).
Note 2 — Temperature of water to aftercooler not to exceed 130 °F (54 °C).
Note 3 — Retarded timing required.
Note 4 — The propane-air added should not exceed 35% of the mixture volume.
Note 5 — For non-lug, standby applications only.
Note 6 — Advance timing. Consult manufacturer for rating.

*Turbocharged engines demand at least 12 PSI gas.

DIESEL FUEL

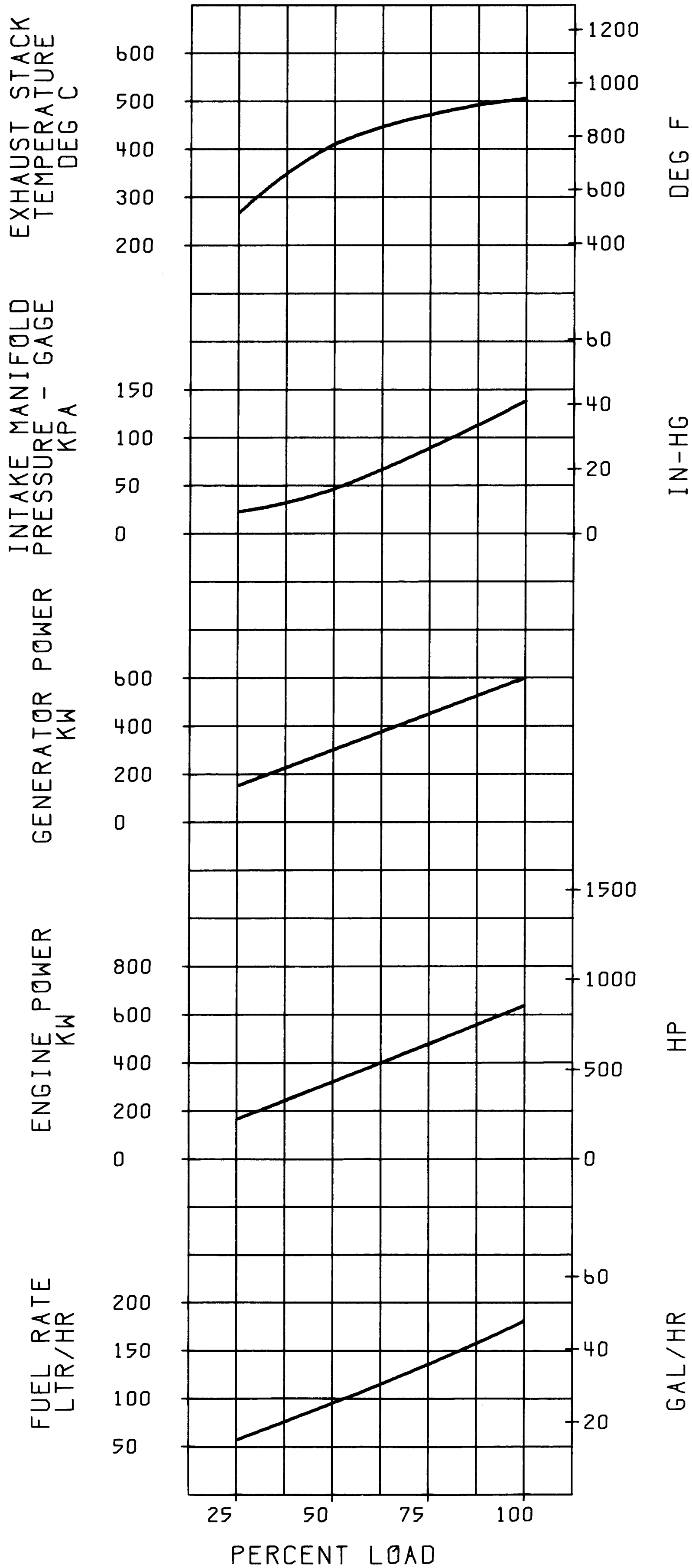
Determining the fuel consumption for diesel engines is a very similar operation to that for gas engines except that it is less complex since all specific fuel consumption data for diesel engines are based on high heat value. Only the high heat value need be considered when diesel fuels are used. All hydrocarbon fuels, whether gaseous, liquid, or solid, or any other fuels which produce water as one of the products of combustion, have their respective high and low heat values. However, in the case of diesel fuel, there is little variation in the amount of water formed per unit of fuel burned for fuels of like gravity, regardless of the source or area of origin. Thus, the difference between high and low heat value is a fairly constant percentage. It has, therefore, become standard practice in the engine industry to publish fuel consumption data for diesel engines based on high heat value, usually expressed in pounds per horsepower-hour or kg/kW•h for diesel generator sets, for a given specific gravity fuel.

Figure 5 illustrates a typical generator set performance curve for a diesel engine.

To use the curves, simply locate the percent load on the abscissa, follow this line vertically till it intersects the fuel curve, then horizontally to the left and read the fuel rate in Gal/h-ℓ/h on the vertical ordinate.

Multiplying this fuel rate by the weight per gallon will provide the total fuel consumption per hour in pounds, of a given gravity fuel, for the respective load. American Petroleum Institute's (A.P.I.) standard for gravity is usually used for petroleum products. To convert pounds to gallons it is only necessary to divide the total pounds used per hour by the number of pounds per gallon corresponding to the given fuel.

The standard unit for merchandising and pricing liquid fuels is the gallon, while the real value of diesel fuel is dependent upon its weight. The "work content" of a gallon of diesel fuel, the Btu content per gallon, is proportional to the weight of the fuel. Heavier and less expensive fuels contain more Btu's for a given volume than lighter fuels. Table 2 lists the density, heat value, specific gravity, and the API gravity of several typical fuels. Instead of stating the fuel weight as a given number of pounds per gallon, at a given temperature, the weight is usually indicated by specific gravity or API gravity. The latter has an arbitrary relationship to the specific gravity. The specific gravity of a fuel is the ratio of its weight to the weight of an equal volume of water at a given temperature. Thus specific gravity increases as the weight per gallon increases. API gravity, however, decreases numerically as the weight increases.



ENGINE DATA

ENGINEERING MODEL	E127
ASPIRATION	TA
AFTERCOOLER	JWAC
EXHAUST MANIFOLD	DRY SHIELDED
COMBUSTION SYSTEM	PC
TURBO MODEL	4HD755
COMPRESSION RATIO	15.1 TO 1
TYPE OF DUTY	PRIME
RATED KW	600
RATED HZ	60
RATED RPM	1200
EFFECTIVE SERIAL NUM	66B3929

TOLERANCES

CURVES REPRESENT TYPICAL VALUES OBTAINED UNDER NORMAL OPERATING CONDITIONS. AMBIENT AIR CONDITIONS AND FUEL USED WILL AFFECT THESE VALUES. EACH OF THE VALUES MAY VARY IN ACCORDANCE WITH THE FOLLOWING TOLERANCES

EXHAUST STACK TEMPERATURE	±42 DEG C ±75 DEG F
INTAKE MANIFOLD PRESSURE-GAGE	±10 KPA ±3 IN HG
POWER	±3 PERCENT
FUEL CONSUMPTION	±6 G/KW-HR ±.010 LB/HP-HR

PERFORMANCE DATA

PERCENT LOAD	ENGINE POWER KW	ENGINE POWER HP	GEN POWER AT 0.8 PF KW
100	636	853	600
75	477	640	450
50	319	428	300
25	164	220	150

PERCENT LOAD	BSFC G/KW-HR	BSFC LB/HP-HR	FUEL RATE LTR/HR	FUEL RATE GAL/HR
100	240	0.395	180.2	47.6
75	240	0.395	135.1	35.7
50	251	0.413	94.5	25.0
25	292	0.480	56.5	14.9

AIR FLOW AT FULL LOAD

INLET 53.5 CU M/MIN 1890 CFM

EXHAUST 139.0 CU M/MIN 4910 CFM

CONDITIONS

PERFORMANCE BASED ON SAE J1349 STANDARD CONDITIONS OF 100 KPA (29.61 IN HG) AND 25 DEG C (77 DEG F). PERFORMANCE ALSO APPLIES AT DIN 6270 STANDARD CONDITIONS OF 97.8 KPA (28.97 IN HG) AND 20 DEG C (68 DEG F).

FUEL CONSUMPTION IS BASED ON FUEL OIL HAVING AN HHV OF 45570 KJ/KG (19590 BTU/LB) AND WEIGHING 848 G / LITRE (7.076 LB / U.S. GAL).

ENGINE POWER CURVE REPRESENTS THE POWER REQUIRED FOR DRIVING A CATERPILLAR GENERATOR USING AN ENGINE EQUIPPED WITH FUEL, LUBE OIL, AND JACKET WATER PUMPS BUT WITHOUT FAN.

THE GENERATOR POWER CURVE REPRESENTS THE ELECTRICAL OUTPUT OF THE GENERATOR

STACK TEMPERATURE, INTAKE MANIFOLD PRESSURE, AND FUEL RATE ARE BASED ON ENGINE POWER CURVE

NO ENGINE DERATION IS REQUIRED FOR AMBIENT TEMPERATURES UP TO 52 DEG C (125 DEG F) EXCEPT AS SHOWN ON THE APPLICABLE ALTITUDE DERATING CURVE NO. TD2018

TABLE 2

PHYSICAL CHARACTERISTICS OF DIESEL FUELS

Weight Fuel		Heat Valve		Sp Gravity	Gravity
LB/Gal	kg/ℓ	BTU/Gal	kJ/ℓ	at 60 °F. 15.5 °C	Deg. API
6.79*	.814	134,700	37,545	.816	42
6.95	.833	137,000	38,186	.835	38
7.29**	.874	141,800	39,524	.876	30
7.48	.896	144,300	40,221	.898	26

* Typical No. 1 Diesel Fuel
** Typical No. 2 Diesel Fuel or Furnace Oil

Note: Caterpillar diesel fuel consumption figures are based on fuel oil having a gross heat value of 19590 Btu per pound and weighing 7.076 pounds (3.2 kg) per U.S. gallon.

Fuel Selection

Engine Requirements

The fuel normally recommended for diesel generator sets is No. 2 furnace oil. When this fuel is also used for heating the building, a common storage tank for both the heating plant and the generator set is practical. In addition to reducing installation costs, this arrangement may reduce fuel costs as a consequence of quantity purchasing and eliminate fuel deterioration concerns.

A Caterpillar Diesel Engine has the capacity to burn a wide variety of fuels. In general, the engine can use the lowest priced distillate fuel which meets the following requirements (fuel condition as delivered to engine fuel filters):

Cetane Number (Precombustion Chamber Engines) . 35 Minimum
Viscosity . . . . 100 SUS at 100 °F Maximum (38 °C)
Pour Point . . . . 10 °F (6 °C) Below Ambient Temperature
Cloud Point . . . . Not Higher Than Ambient Temperature
Sulfur . . . . Adjust Oil Change Period For High Sulfur Fuel
Water and Sediment . . 0.1% Maximum

Some fuel specifications that meet the above requirements:

— ASTM D396 — No. 1 and No. 2 Fuels (Burner Fuels)
— ASTM D975 — No. 1-D and No. 2-D Diesel Fuel Oil
— BS2869 — Class A1, A2, B1, and B2 Engine Fuels
— BS2869 — Class C, C1, and C2 and Class D Burner Fuels
— DIN51601 — Diesel Fuel
— DIN51603 — EL Heating Oil

The following additional information describes certain characteristics and their relation to engine performance:

A. Cetane Number — This index of ignition quality is determined in a special engine test by comparison with fuels used as standards for high and low cetane numbers.

B. Sulfur — Since the advent of high detergent oils, sulfur content has become less critical. A limit of 0.4% maximum is used for Caterpillar Engines, without reducing oil change periods. Oil change periods are reduced with higher sulfur fuel.
C. Gravity — The measurement is an index of the weight of a measured volume of fuel. Lower API ratings indicate heavier fuel which contain more heat value.
D. Viscosity — This factor is a time measure to flow resistance of a fuel. Some low viscosity fuels are lubricants; a viscosity which is too high makes for poor fuel atomization thereby decreasing combustion efficiency.
E. Distillation — This involves the heating of crude to relatively high temperatures. The vapor which results is drawn off at various temperature ranges, producing fuels of different types. The lighter fuel, such as gasoline, comes off first and the heavier fuel last.
F. Flash Point — The lowest temperature at which the fuel will give off sufficient vapor to ignite momentarily when a flame is applied to the vapor.
G. Pour Point — This denotes the lowest temperature at which fuel will flow or pour when chilled.
H. Water and Sediment — The percentage by volume of water and foreign material which may be removed from fuel by centrifuging. No more than a trace should be present.
I. Carbon Residue — Percentage by weight of dry carbon remaining when fuel is ignited and allowed to burn until no liquid remains.
J. Ash — This is percentage by weight of dirt, dust, sand, and other foreign matter remaining after combustion.
K. Corrosion — To determine corrosion, a polished copper strip is immersed in the fuel for three hours at 122 °F (50 °C). Any fuel imparting more than slight discoloration should be rejected.

The customer should order as heavy a fuel as his diesel engine and temperature conditions permit. Fuel costs can represent approximately 80% of total operating costs for an engine. It is good economics to look closely at the largest cost first.

NOTE: Caterpillar Diesel Engine fuel rack settings are based on 35 API (specific gravity) fuel. The use of fuel oil with a higher API (lower specific gravity) number will result in a reduction of power output. When using heavier fuels, a corrected rack setting should be used to ensure against power levels above the engines approved rating. Your Caterpillar Engine Dealer should be contacted to obtain the correct rack setting for fuels which do not comply with the recommendations. Operations above the approved engine horsepower rating level can result in reduced engine life, increased owning and operating costs, and customer dissatisfaction.

Crude Oil Fuels

Crude oil, in some cases, is a practical and economic fuel for diesel engines. Each crude oil must be evaluated individually, and special equipment may be needed to properly condition the fuel. Certain minimum guidelines have been established to determine the suitability of a crude.

API Gravity	45 Maximum
Viscosity at 100 °F (37.78 °C)	100 SUS Maximum
Gasoline and Naptha Fraction	35% Maximum
Kerosene and Distillate Fraction	30% Minimum
Water and Sediment	0.5% Maximum
Cetane Number	35 Minimum

ENGINE MAINTENANCE COSTS

Engine maintenance costs are not as easily computed or estimated as fuel consumption, recoverable heat, or initial cost; they are, however, not entirely elusive. The increasing use of guaranteed maintenance and service contracts has eliminated much of the estimating formerly required in feasibility studies. Maintenance contracts vary from complete maintenance and service, including all parts, supplies, and labor, to contracts that provide only a guaranteed cost for engine rebuild. For this reason, a maintenance cost figure is meaningless unless well defined. Complete maintenance costs are composed of three basic items:

1. The miscellaneous maintenance and service cost, including service manual recommendations plus make-up oil (excluding labor to perform this routine duty).
2. The overhaul maintenance cost, usually expressed in terms of cost per engine operating hour. This item should cover all labor and parts necessary to perform major and minor overhauls at the recommended intervals.
3. The third item is the labor cost necessary to perform the miscellaneous service for Item 1.

Items 1 and 2 will vary considerably with the severity of service the engine must perform. For On-Site Power installations, the conditions under which the engines operate, the quality of fuel, and the routine maintenance the engine receives, are all usually considered to be good. Item 3 will vary with labor costs and the location of the engine plant with respect to the point from which service personnel must be dispatched. Because of the many variables, it is difficult to provide realistic figures that would be useful for all applications. Maintenance costs should be based on past experiences in the area being considered.

HEAT RECOVERY SYSTEMS

Heat may be recovered from engines by a multitude of different systems. The nature of the heat requirements of a facility, however, will usually be the determining factor that makes one system of heat recovery more attractive than another.

In its simplest form, heat recovery may amount to nothing more than utilization of the heat transferred from an engine radiator to a flow of air. The temperature of such air is usually rather low 100 °F (38 °C) to 150 °F (66 °C). Air so heated can be delivered essentially free from contaminants and thus is quite suitable for pre-heating boiler combustion air, grain and cereal drying, space heating, etc. The cost of such a system is minimal and, under optimum operating conditions, the effect is a relatively efficient system, converting approximately 32% of input fuel energy into work, or power, and 30% into recovered heat energy for an overall efficiency of approximately 62%. This total percentage figure can be further increased by approximately 17% by recovering a portion of the exhaust heat. To do so would add something to the cost except in processes where the exhaust can be used directly.

One of the more popular heat recovery systems in use today for reciprocating engines employs the principle of ebullient cooling for the engine. This system, shown in Figures 10 and 11 has gained its popularity largely because of its simplicity, low operating and maintenance cost, and its capability to produce steam at pressures suitable for conventional absorption air conditioning chillers 12-15 psi (83-103 kPa). This system is obviously more costly and requires more initial capital investment than the simpler systems, however, it also has greater application flexibility.

Because an ebullient cooled system incorporates most of the considerations of a heat recovery system, it will be used as an example:

Primary responsibility of the heat recovery equipment is to cool the engine. Secondary functions are to recover heat and to silence engine exhaust. Individual units for each engine will offer maximum reliability and flexibility. These units include a heat recovery muffler, steam separator, and safety devices necessary to protect the entire cooling system. These units are constructed either as water tube (water inside tube) or fire tube units. The water tube unit generally provides excellent silencing and the highest Btu recovery at the lowest cost, weight and space requirements. The fire tube unit exhibits good life and, because of an integral steam separator, installation piping is simplified.

Either of these units should incorporate the equipment necessary to protect the system. These include:

- Low water level switch
- Minimum water level switch
- High water level switch
- Liquid level make-up valve — float type
- Float and thermostatic trap
- Vacuum breaker
- Pressure control valve
- High steam pressure switch
- Safety steam valve
- Gauge glass
- Pressure gauge

Sizing: When sizing the heat recovery unit to the engine, both water and gas side pressure drops must be considered. On the gas side, pressures should be calculated with the assumption that the unit is operating dry. This will avoid excessive exhaust back pressures.

Between these two extremes in heat recovery system design, lie a host of other systems and combinations of systems. As stated earlier, it is not the purpose of this discussion to explore all of the possible systems nor to design the perfect system, it is rather to call attention to a number of basic requirements associated with the design of such systems and to provide performance data to assist the designer.

The gas turbine offers some latitude in heat recovery system design; however, the most commonly used system likewise generates low pressure steam. Since economics should be the determining factor in practically all on-site power plant design decisions, no simple rule or equation can be given for evaluating heat recovery of turbines versus reciprocating type engines. It can be pointed out, however, that differences in fuel economy and rejected heat are appreciable for the two types of prime movers and thus justify a thorough examination before making a decision on which type of prime mover to use.

Some typical examples of heat recovery systems are:

1. Meat packing plants — hot water for processing
2. Hotels and motels — hot water, heated swimming pools, space heating, domestic hot water, steam absorption air conditioning
3. Schools — hot water and steam, heating and air conditioning
4. Apartments — steam (same as hotels and motels)
5. Brick plants — steam, heat mix, exhaust drying
6. Carpet plant — steam process, exhaust gas drying
7. Mica Reduction — exhaust gas — increases energy of compressed air for reduction process
8. Feed mill — hot water, boiler, feed, water heating, steam
9. Dairy — steam and hot water
10. Heavy manufacturing — steam, air conditioning, space heating and hot water
11. Warehouses — (same as heavy manufacturing)
12. Office buildings — (same as hotels and motels)
13. Ice rinks — hot water, space heating
14. Car wash — hot water, washing, drying
15. Municipal pumping plants — hot water, space heating and water treatment
16. Hospitals — steam, (same as hotels and motels)
17. Sewage plant — hot water, sludge heating
18. Printing plants — steam, (same as hotels and motels), humidity control
19. Ice plants — hot water, cleaning, treating
20. Power generation — steam, binary cycle

BASIC HEAT RECOVERY SYSTEMS

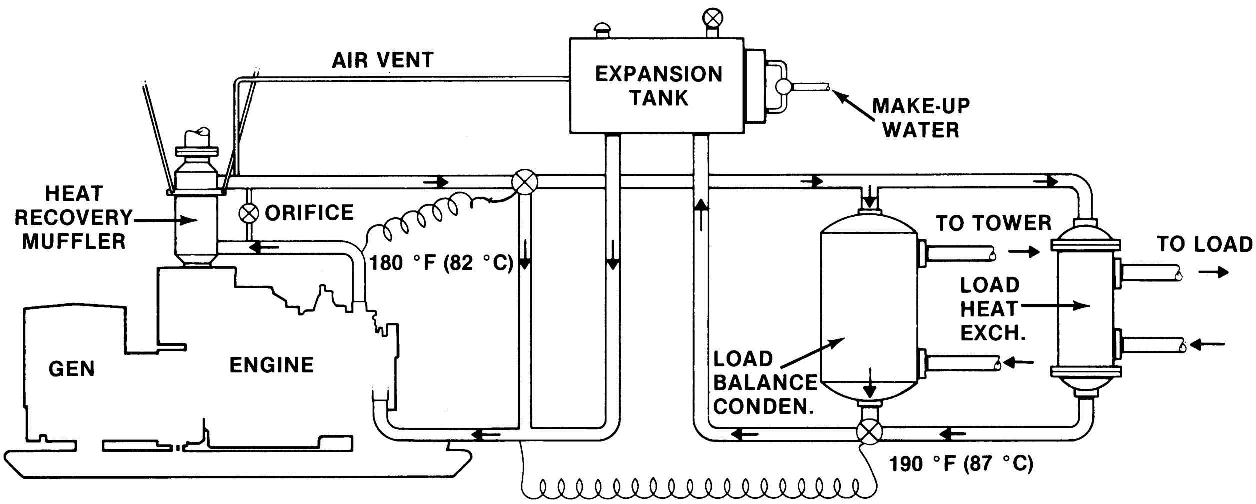
Heat Recovery Systems can be divided into four basic design classifications, as follows:

1. HOT WATER SYSTEM — NORMAL TEMPERATURE*
2. HOT WATER SYSTEM — HIGH TEMPERATURE*
3. HOT WATER AND STEAM SYSTEM WITH FLASH BOILER
4. EBULLIENT SYSTEM

* The temperature classifications used in this discussion evolve from internal combustion engine operating practices and have no relationship to similar classifications used in the building heating industry.

Hot Water System — Normal Temperature

This system utilizes normal jacket water temperature approx. 190-210 °F (88-99 °C) recorded at engine outlet and a shell and tube heat exchanger to transfer rejected engine heat to a secondary circuit — usually water. An exhaust heat boiler may also be included in the system. The primary coolant circuit which serves the engine jacket must be a "closed" system. Figure 6 illustrates a flow diagram for this system.



FLOW DIAGRAM, HOT WATER SYSTEM — NORMAL TEMPERATURE

Figure 6

Critical Design Criteria

a. Provide adequate flow of coolant through engine. If standard engine jacket water pump is used, keep friction and static head on jacket water circuit beyond the engine to a minimum.
b. If a heat recovery muffler is used, controls must be used to assure water flow when engine is operating.
c. Temperature differential between "coolant in" and "coolant out" of engine should not exceed 20 °F (11 °C) and should not be less than 10 °F (6 °C). A 15 °F (8 °C) differential is desirable.
d. Expansion tank in engine coolant circuit must be at highest point in the primary circuit.
e. Heat exchanger location must be at lower level than expansion tank and as near engine as possible.
f. Proper venting to expansion tank of all piping in the primary circuit which might develop either steam or air "pockets."
g. Temperature control of engine coolant to avoid excessively high temperature 210 °F. (99 °C) max.
h. Use only treated water in engine coolant circuit.

Hot Water System — High Temperature

This system utilizes elevated jacket coolant temperatures [220 °F (104 °C) to 250 °F (121 °C) recorded at engine outlet] and functions essentially the same as the normal temperature hot water system except for the pressure required in the circulating systems, especially in the engine coolant circuit. In this system a pressure control must be provided in the engine coolant circuit that will assure a pressure, at all times during operation, of several psi preferably 4 or 5, above the pressure at which steam will form. The source of this pressure may be a static head imposed by an elevated expansion tank or controlled air pressure in the expansion tank. For 250 °F (121 °C) water temperature, this pressure should be approximately 20 psi (138 kPa) at the engine. Also, all water circulating pumps, primary and secondary, must be suitable for use with the elevated temperatures and pressures. Where multiple engines are used with a common circulating pump, a standby pump should be supplied in the interest of reliability. Conventional engine jacket water pumps are not suitable for this service. Figure 7 illustrates a flow diagram for this system.

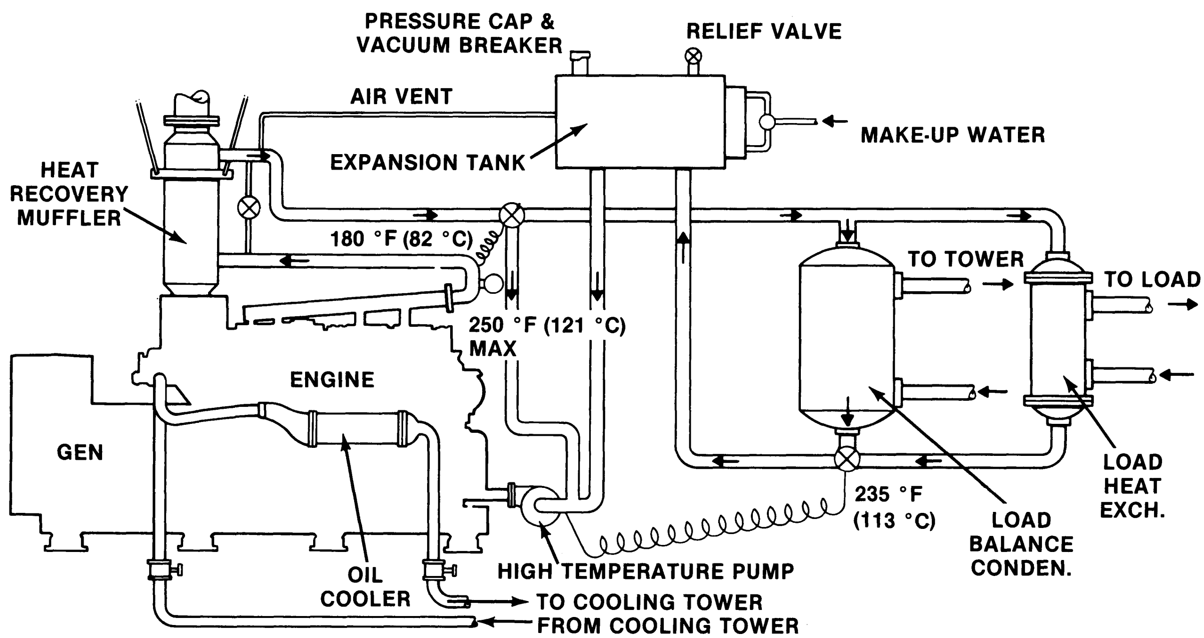


Figure 7 **FLOW DIAGRAM, HOT WATER SYSTEM — HIGH TEMPERATURE**

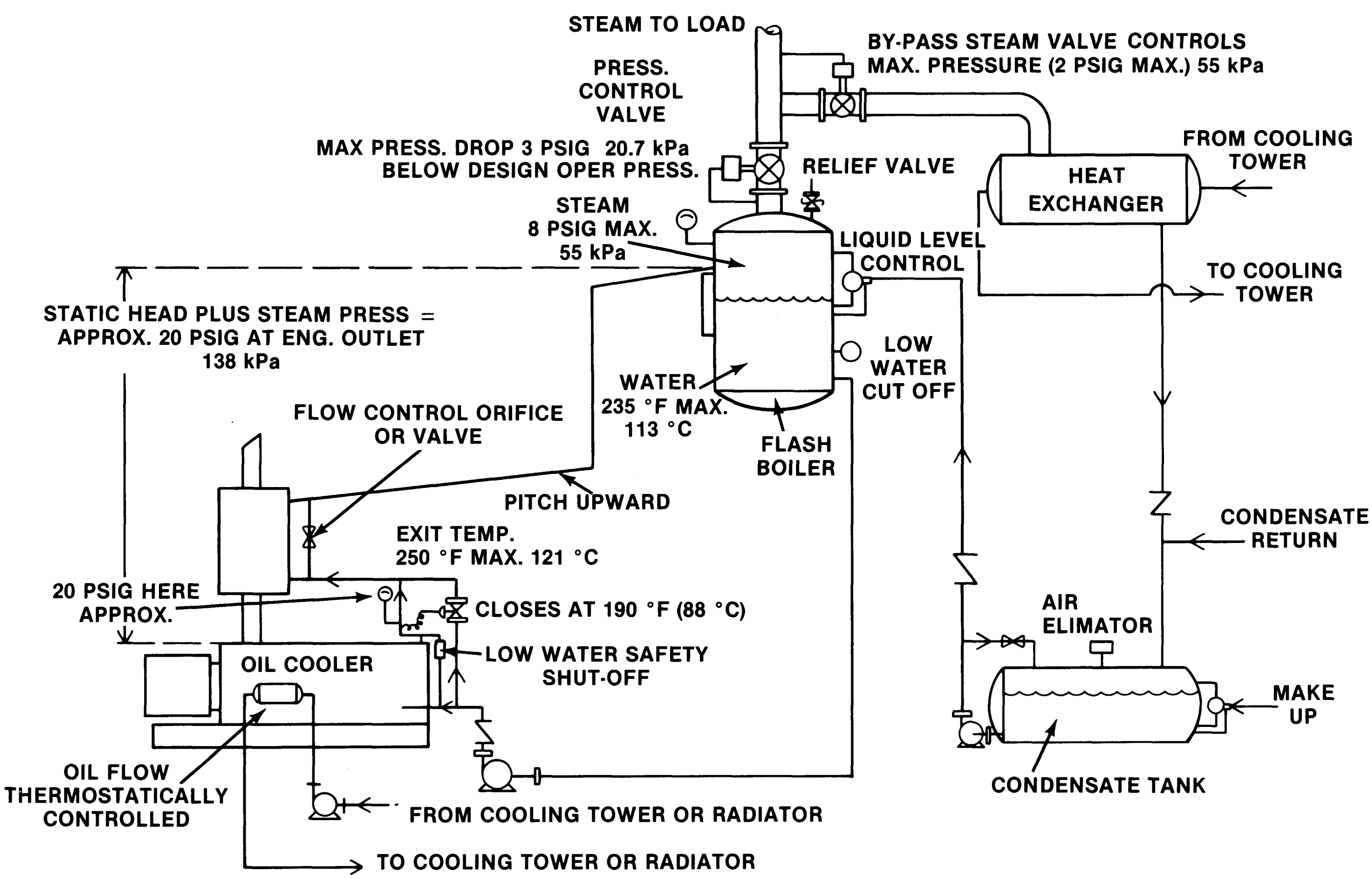
Critical Design Criteria

a. Items B, C, D, E, and F of Normal Temperature Hot Water System.
b. Requires pressure control for engine coolant circuit.
c. Water pump must be suitable for high temperature operation and capable of maintaining adequate flow.
d. Temperature control of engine coolant required to avoid temperatures exceeding 250 °F (121 °C).
e. Engine oil cooler (heat exchanger) and/or aftercooler requires cooling water circuit separate from engine jacket water circuit. Temperature of oil leaving cooler should not exceed 190 °F (88 °C). See Section on installation. Thermostatic control for oil flow is available for some models of Cat engines and should be used to satisfy the above requirements.
f. Use only treated water in engine coolant circuit.
g. Controls must assure flow through heat recovery muffler, if used, when engine is in operation.
h. Remove engine thermostats and eliminate the by-pass circuit.

Hot Water and Steam System With Flash Boiler

This system incorporates many of the features of the high temperature hot water system plus a "flash boiler" for generating low pressure steam. Steam is generated in the flash boiler, and in the piping to the boiler, simply because of the pressure differential which exists by design between the engine outlet and the boiler. A lower pressure obviously prevails in the boiler than at the engine outlet, thus, as the high temperature water from the engine approaches the boiler, the static head is reduced and, as the "heat of the liquid" adjusts to the lower pressure, some heat is released and serves as heat of vaporization to convert part of the water to steam. In the process, the temperature of both the steam, and the remaining water, adjust to the temperature corresponding to the controlled pressure prevailing in the boiler. The steam so formed is delivered to the load while the water, at the reduced temperature returns to the engine to repeat the process.

This type of system is usually designed to operate at steam pressure ranging from 2 to 8 psi (13.8 to 55 kPa). For any predetermined or maximum design engine coolant temperature, the total pressure imposed upon the engine cooling circuit by the combined steam pressure and static head must be adequate to prevent boiling or "flashing" within the engine. When 250 °F (121 °C) water temperature (leaving the engine) is used, the required total pressure at the engine is approximately 20 psi (138 kPa). The additional 5 psi (34 kPa) over the pressure corresponding 250 °F (121 °C) will allow a 3 to 4 psi (20.7 to 27.6 kPa) pressure drop in the flash



FLOW DIAGRAM, HOT WATER and STEAM SYSTEM WITH FLASH BOILER

Figure 8

boiler without danger of "flashing" in the engine. For controlled boiler pressures less than 8 psi (55 kPa), the static head required may be reduced but should always be adequate to prevent "flashing" in the engine when normal pressure fluctuations occur. Figure 8 illustrates a flow diagram for this type of system.

Some flash boiler systems call for a pressure reducing valve or orifice at the inlet to the boiler instead of the static head and controlled steam pressure. While such a system, properly installed and carefully operated, can be made to function, it is not recommended because of the danger of pressure unbalance in the system causing steam to form in the engine jacket water pump and in the engine (because of circulation failure) with disastrous results.

Critical Design Criteria

a. Provide adequate coolant flow through engine. Remove standard engine thermostats and eliminate the bypass circuit.
b. The temperature differential of "coolant in" and "coolant out" of the engine should not exceed 20 °F (– 11 °C) and should not be less than 10 °F (6 °C). A 15° (8 °C) differential is desirable.
c. If a heat recovery muffler is used, controls must be used to assure water flow when the engine is operating.
d. Water pumps must be suitable for high temperature operation.
e. Adequate "make up" water supply and control to accommodate the inherent variations in steam demand peculiar to the particular installation. Make up water should be supplied almost entirely from the condensate return.
f. Use only treated water to supplement condensate for make up water.
g. Flash boiler must be elevated above engine to provide static head. Total pressure at engine outlet must be adequate to prevent "flashing" within engine when pressure variations occur in the boiler.
h. Boiler must be equipped with a pressure control valve located in the steam outlet line to limit pressure drops in the boiler to approximately 3 psi (20 kPa) below design operating pressure.
i. All piping from engine to boiler must be pitched upward.
j. Engine oil cooler (heat exchanger) and/or aftercooler, requires cooling water circuit separate from engine jacket water circuit. Temperature of oil leaving cooler should not exceed 190 °F (88 °C). Thermostatic control for oil flow is available for some models of Cat engines and should be used when available.

EBULLIENT SYSTEM

This system utilizes the "heat of vaporization" to remove rejected heat from the engine. Steam, as such, however, is not allowed to collect within the engine but is moved through the water passages, along with the high temperature water by thermal action, to a steam separator located at an elevation somewhat above that of the engine. No jacket water pump is required with this system. While the temperature differential between "water in" and "water out" of the engine in this system is usually quite low 2 °F (1.1 °C) to 3 °F (1.7 °C), flow through the engine is assured by virtue of the change in coolant density as it gains heat from the engine. The higher temperature coolant being lighter creates a pressure differential between the water inlet and water outlet connections to the engine. Almost all of the heat gain in the coolant is added in the form of heat of vaporization. Figure 9 illustrates the basic elements of an ebullient system. Obviously, any number of arrangements are possible. In some instances the exhaust gas boiler, or muffler, and the steam separator are combined into a single "packaged" unit —

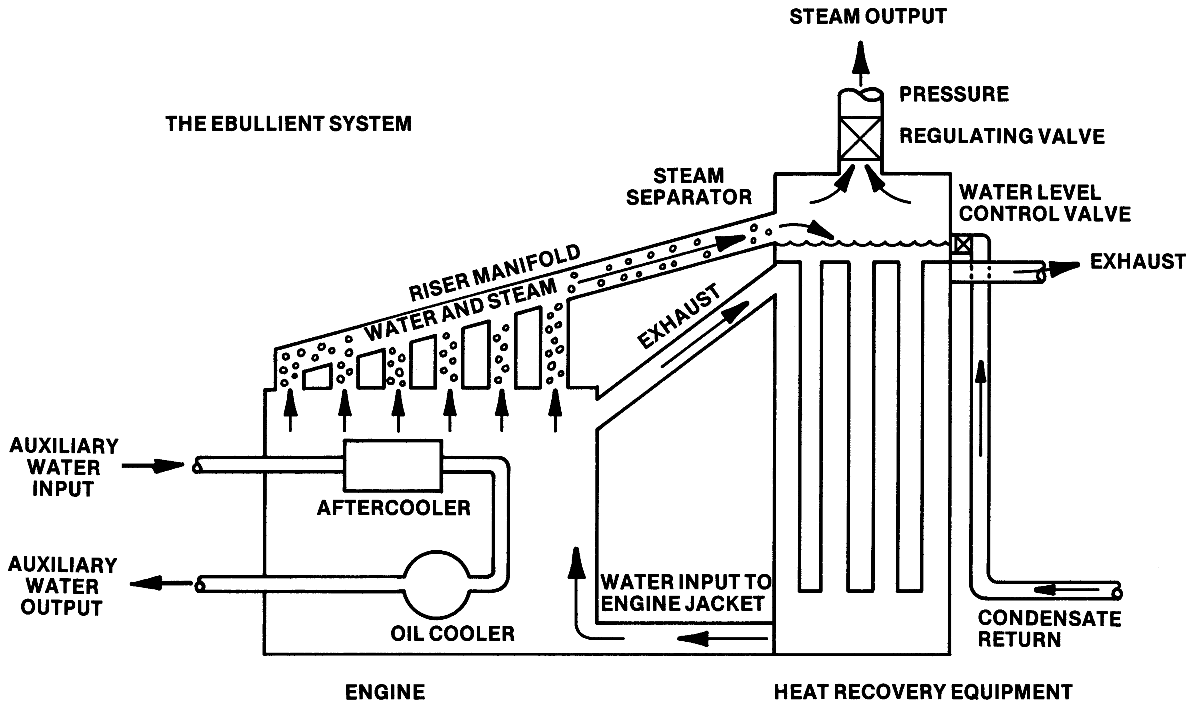
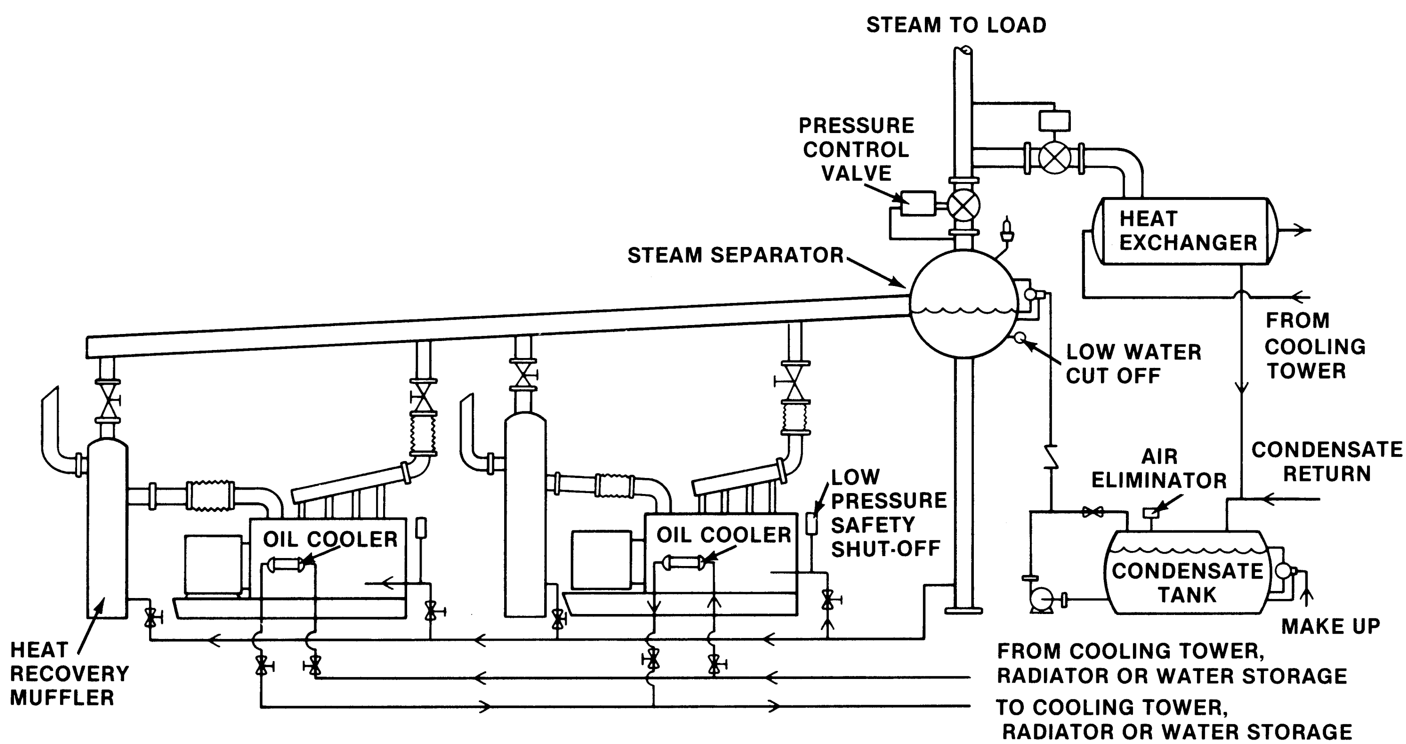


Figure 9



FLOW DIAGRAM, EBULLIENT SYSTEM, MULTIPLE UNITS, SINGLE STEAM SEPARATOR

Figure 10

one packaged unit being used for each engine as illustrated by Figure 11. Other heat recovery equipment available (not illustrated) combines the exhaust gas boiler and the steam separator in a single unit and in addition includes a direct fired section in the exhaust boiler which serves to eliminate the need for an auxiliary boiler. Some units are designed to serve two engines each; however, to do so requires a more complex exhaust piping system. Also, to serve as a steam separator the unit must be located well above the engines.

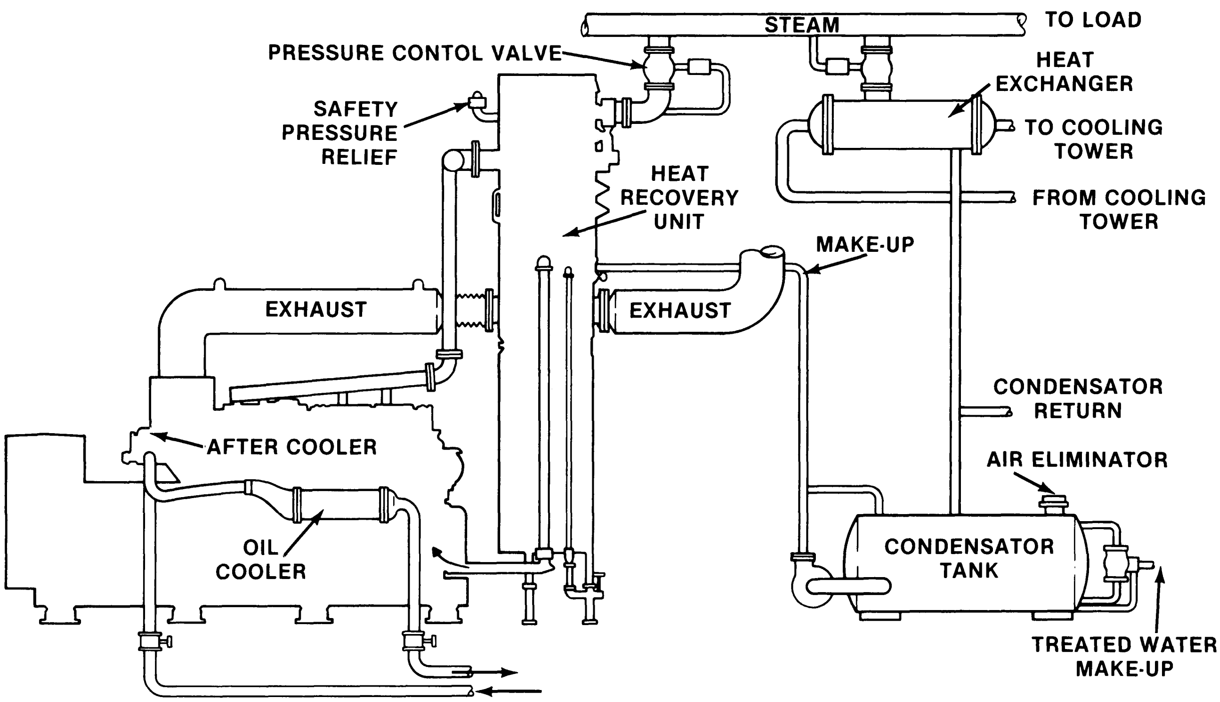


Figure 11

Critical Design Criteria

a. To avoid excessive boiling within the engine and subsequent formation of steam pockets in the water passages, the liquid in the engine must at all times be under a static head of not less than one psi (6.89 kPa) recorded at the coolant outlet of the engine. When measured in feet of water, the static head should be measured from the highest water passage in the engine where heat transfer from the engine or exhaust manifold to the coolant occurs. On some engines this may be the cylinder head while on others it might be the water cooled exhaust manifold.

b. Oil cooler (engine) requires cooling water circuit separate from the engine jacket circuit. Temperature of oil leaving cooler should not exceed 190 °F (88 °C). Thermostatic control of oil flow is provided with Cat ebullient cooled engines.

c. For turbocharged and aftercooled gas engines, the aftercooler requires a cooling water circuit separate from the engine. For high compression ratio (10:1) gas engines the aftercooler water temperature should not exceed 90 °F (32 °C). For low compression ratio (7:1) gas engines the aftercooler water temperature should not exceed 130 °F (54 °C). It is common practice to place the oil cooler and aftercooler in the same cooling water circuit.

d. The engine cooling system must be protected against sudden loss of pressure. Controls must be provided between separator and load which will function to limit the pressure drop to not more than 3 psi (20 kPa). A greater sudden pressure loss will cause "flashing" in the engine which can result in serious damage to the equipment.
e. System pressure should not exceed 15 psi (105 kPa).
f. Pipe size (for coolant) to and from the engine must be in accordance with the engine manufacturer's recommendations, or larger.
g. Coolant piping between engine and steam separator must be so installed that the flow of coolant (a mixture of water and steam) will always be pitched upward — never downward.
h. Make-up water should be provided to compensate for any loss in the system. This water is generally fed into the condensate tank and should always be treated.

PERFORMANCE OF HEAT RECOVERY SYSTEMS

Rejected Heat Available from Engine

The amount of heat rejected by any direct-fired prime mover is directly proportional to the load and inversely proportional to the thermal efficiency. Heat rejection information is available on all Cat engines. Calculation of recoverable heat, however, is not an exact science. While the heat rejected to the jacket coolant can be measured quite accurately and to the exhaust gas not quite so accurately, there are other areas, such as lubricating oil and radiation, where measurements and calculations become more in the category of close approximations.

Heat Rejected to Coolant

For liquid cooled engines, essentially all of the heat rejected to the engine jacket water is recoverable. The quantity of heat available from this source will vary slightly for each engine model. The rate of heat rejection to the jacket water circuit for Caterpillar ebullient cooled engines is approximately 35 Btu/hp•min (36.9 kJ/hp•min) or 48 Btu/kW•min (50.6 kJ/kW•min) for gas engines, 29 Btu/hp•min (30.6 kJ/hp•min) or 41 Btu/kW•min (43.3 kJ/kW•min) for diesel engines.

Heat Rejected to Lubricating Oil

When recovering heat from naturally aspirated or turbocharged-aftercooled engines using any of the high temperature cooling systems, it may also be worthwhile to utilize the heat rejected to the lubricating oil. This is especially so when an auxiliary boiler is used since this heat can be applied to preheat the boiler feed water. It can also be used for domestic hot water or other low temperature requirements. The heat removed by the lubricating oil from engines operating with coolant temperatures above 220 °F (104 °C) is always rejected to a cooling medium other than the jacket water. The rate of heat rejection to the oil cooling circuit for Caterpillar Engines is approximately 5.5 Btu/hp•min (83 kJ/kW•min) for gas engines, 8.5 Btu/hp•min (129 kJ/kW•min) for diesel engines. This figure may be used when calculating the total heat removed by the lubricating oil circuit. See installation section for auxiliary water flow and temperature requirements.

Heat Recoverable from Exhaust

The heat recoverable from engine exhaust gas can be calculated by application of the equation:

English

$Q = C_pM\ (T_1\text{-}T_2)$ Where:

$Q =$ Btu per hour.

$C_p =$ average specific heat of exhaust gas 0.258 Btu/lb. per degree F.

$M =$ The mass or weight of exhaust gas flow in pounds per hour.

$$M = \frac{(CFM \times 60 \times 41.13)}{(Gas\ Temp\ °F + 460°F)}$$

Metric

$Q = C_pM\ (T_1\text{-}T_2)$ Where:

$Q =$ kJ per hour.

$C_p =$ average specific heat of exhaust gas 1.081 kJ/kg per °C

$M =$ The mass or weight of exhaust gas flow in kilograms per hour.

$$M = \frac{(m^3 \times 60 \times 365.5)}{Gas\ Temp\ °C + 273\ °C}$$

T_1 and T_2 represent exhaust gas temperature in and out respectively. Heat recovery mufflers or boilers should be so sized and applied that, when operating at full load, the temperature of the exhaust gas leaving the unit will not be less than 350 °F, ± 25 °F. (177 °C ± 13.9 °C) to eliminate possibility of water vapor in exhaust condensing.

Example: Heat recovery calculation

Given conditions: One G398 turbocharged and aftercooled Electric Set, ebullient cooled, operating at rated prime power output, (500 kW), using 1000 Btu (1055 kJ) Btu (HHV), 905 (955 kJ) (LHV) gas; exhaust gas temperature leaving muffler, 350 °F (177 °C).

Required: Determine amount of recoverable heat per hour from jacket coolant and exhaust gas.
Solution:
Heat rejected to jacket coolant and lubricating oil, Btu/hr 1,440,000
(500 x 48 x 60 = 1,440,000 Btu/kW/hr.) or (kW x kJ x min = kJ/kW/hr.)
Heat rejected to lubricating oil only, Btu/Hr. 237,000
(500 kW x 7.9 x 60 = 237,000 Btu/hr.) or (kW x kJ x min = kJ/kW/hr.)
Heat rejected to jacket coolant only, Btu/hr. . . . . 1,203,000 (1,269,165 kJ)
Heat recoverable from exhaust
$Q = C_pM\ (T_1\text{-}T_2)$
Q = .258 x 5325 x (1060-350) = ... 975,433 Btu/hr
= 1029081 kJ/hr
Total recoverable heat, Btu/hr 2178433
= 2298247 kJ/hr

CONVERSION TO OTHER UNITS

The recoverable heat from prime movers can also be expressed in boiler horsepower, a unit sometimes used in rating the output of small boilers.

Example:
Conditions: G398 TA Industrial Engine operating at 700 hp, ebullient cooled, exhaust temperature leaving muffler 350 °F (177 °C).

Required: Equivalent Boiler HP of recovered heat (as steam)
Solution:
Heat rejected to jacket coolant and lubricating oil, Btu/hr 1,470,000
(700 x 35 x 60 = 1,470,000 Btu/hp/min) or kJ x hp x min
Heat rejected to lubricating oil only, Btu/hr. . . . . 231,000 (243705 kJ/hr)
(700 x 5.5 x 60 = 231,000)
Heat rejected to jacket coolant only, Btu/hr. . . . . 1,239,000 (1307145 kJ/hr)
Heat recoverable from exhaust, Btu/hr . . . . 950317 (1002585 kJ/hr)
[Q = .258 x 5262 x (1050-350) = 950317]
Total Heat Recoverable, (as steam), Btu/hr . . . . 2189317 (2309729 kJ/hr)
Equivalent Boiler hp 69
(2189317 ÷ 33,475 = 69)

It is likewise desirable at times to relate recoverable heat to equivalent tons of air conditioning when using an absorption chiller. Assuming heat input to the chiller per ton of air conditioning to average 18000 Btu/hr. (20045 kJ/hr) and using the recoverable heat at rated load from the previous example:

Air Conditioning = 2189317 ÷ 18000 = 122 tons or 2309729 ÷ 18990 = 122 tons.

Ratio of hp (kW) output to tons of air conditioning from recovered heat = 5.7 hp (4.3 kW) per ton

Since thermal efficiency decreases as the load decreases and more heat is recoverable per hp output, this ratio (hp load per ton) improves as the load is reduced. Thus, since load factors or operating units seldom exceed 75%, it is common practice to use a ratio of one ton per 5.5 hp or per 3.75 kW on the switchboard when estimating the amount of air conditioning available from recovered heat.

STEAM GENERATION

It is often desirable to express heat recovery performance in terms of pounds of steam generated per hour when using either the flash boiler or the ebullient system. The procedure for determining the amount of steam available involves calculating the total heat recoverable from the exhaust gas and the jacket coolant, then dividing this total by the heat required to generate a pound of steam under the prevailing conditions. The heat required per pound of steam may be determined by adding to the heat of vaporization for the prevailing pressure, one Btu for each degree F. difference between the steam temperature and the temperature of the make up water to the flash boiler or steam separator. Table 3 lists the enthalpy, or heat of vaporization, corresponding to the popularly used pressures and temperatures for Total Energy plants.

TABLE 3

ENTHALPY OF STEAM

Psia	kPa	Temp. °F.	Temp. °C.	Heat of Vaporization, Btu/lb.	Heat of Vaporization, kJ/kg
14.696	101	212	100	970.3	2257
15	103	213.03	100.65	969.7	2255
16	110	216.32	102.48	967.6	2250
17	117	219.44	104.22	965.5	2245
18	124	222.41	105.87	963.6	2241
19	131	225.24	107.44	961.9	2237
20	138	227.96	108.95	960.1	2233
21	144.8	230.37	110.29	958.4	2229
22	151.7	233.07	111.79	956.8	2225
23	158.6	235.49	113.14	955.2	2221
24	165.4	237.82	114.43	953.7	2218
25	172.4	240.07	115.69	952.1	2214
26	179.3	242.25	116.90	950.7	2211
27	186.2	244.36	118.07	949.3	2207
28	193	246.41	119.21	947.9	2204
29	200	248.40	120.32	946.5	2201
30	206.9	250.33	121.39	945.3	2198

One G-398 operating at 700 hp and (522 kW) and 12.3 psig (85 kPa) or 27 psia (186 kPa) steam pressure. Make up water temperature at 200 °F (93 °C). Recoverable heat is 2189317 Btu/hr. (2309729 kJ/hr) (see previous example).

Heat of vaporization at 27 psia (186.2 kPa) is 949.3 Btu/lb. (2207 kJ/kg).

Temperature difference, steam and feed water is 244.36-200) = 44.36 °F (118.07-93 = 24.9 °C) (Each degree F difference represents 1 Btu per lb or each degree C difference represents 4.187 J/g.)

Total heat required per lb. of steam is 949.3 + 44.36 = 993.66 Btu or 1001 + 46.8 = 1048 or 2311 kJ/kg

Pounds of steam per hour = 2189317 ÷ 993.66 = 2203 lbs or 2309729 ÷ 2311 = 999.5 kg/hr.

BOILER COST REDUCTION

The ability to recover heat from a prime mover often has a favorable effect on first cost as well as on the operating cost since installation of engines will usually displace some other boiler room equipment.

The total cost of boiler capacity replaced or reduced by the On-Site Power plant should be subtracted from the total cost of the On-Site Power plant when considering rate of return on investment. This is also true for all the other services which are replaced, such as electrical substations, domestic water heating equipment, etc.

SWITCHGEAR AND CONTROLS

The type of switchgear and controls used with On-Site Power generating plants, like heat recovery equipment, varies widely in the degree of sophistication — and in cost. The type of facility and load served by the plant usually dictates the basic design requirements for the switchgear.

When viewed from the standpoint of switchgear and control design, cogeneration plants can be grouped into the following basic categories:

1. MANUALLY CONTROLLED PRIME POWER PLANTS (ATTENDED)
2. TOTALLY AUTOMATIC PRIME POWER PLANTS (UNATTENDED)

MANUALLY CONTROLLED PRIME POWER PLANTS (ATTENDED)

In this type of system, the operator places units on or off the line as the load profile demands. The units are paralleled and the load is divided manually. Generally one unit is designated the master. This unit is set to operate at zero speed droop, with the load limited, and controls the system speed. Its speed control is used to set the system speed. Automatic clock accuracy can be applied if it is desirable to hold clock accuracy. In this type plant, the speed adjustment is usually made manually by comparing a system electric clock to a master clock. Circulating currents are held to a minimum, as in all parallel systems, by cross current compensation but requires manual adjustment occasionally. Load sharing is also controlled by manual adjustment. This type system requires minimum engine safety shutdown controls, however, it should have reverse current trips along with low oil pressure, high oil temperature, and overspeed shutdown devices. Water level and inlet air temperature alarms are also very desirable. See Table 4 for recommended safety devices.

The attractive features of switchgear for this type plant are simplicity, relatively low initial cost, and low maintenance cost.

AUTOMATIC PRIME POWER PLANTS (UNATTENDED)

The totally automatic system has been made practical by the introduction of the electronic governor. By maintaining stable and precise control, these governors allow the units to be automatically paralleled, and to divide load proportionally after being paralleled. The only other device necessary is a signal which will place units on the line or remove them as the load profile demands. Another feature offered by the electronic governor is the ability to allow a large number of engine driven generator sets to be paralleled with reliability and ease.

Since the units in the system are automatic, they can be operated unattended if they have adequate safety and condition readout devices to notify when other than routine servicing is required.

Also, as is true in all power systems, standby capacity must be available in order to provide reliable power. This available standby power should be equivalent to the essential load.

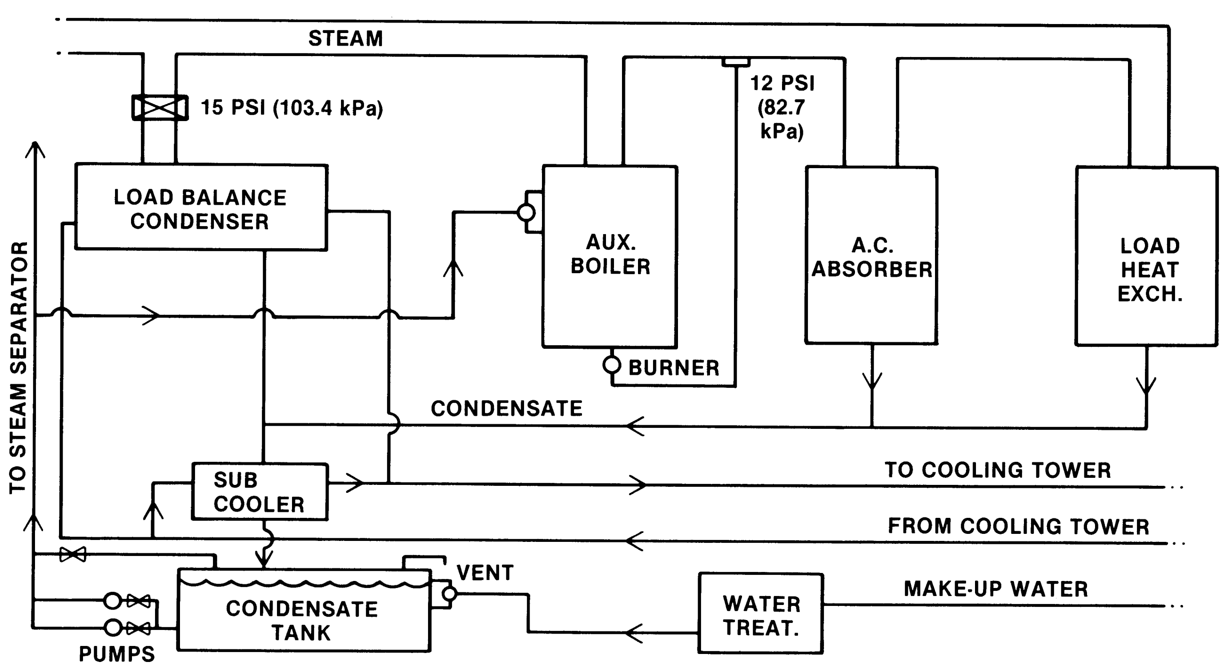
Reliability can be improved by designating a portion of the load, which includes the more critical operations, as "essential" and serving this load with a separate circuit. Then by allowing the remainder, or non-essential load, to be temporarily dropped in the event of an overload condition or unit malfunction, the essential load can be served without interruption. The essential load should not be greater than 90% of the output capability of the unit, or units, assigned to the base load.

If a system is to be totally automatic it must incorporate the capability to drop non-essential load. Dropping non-essential load is not generally a serious disadvantage since this load will normally be picked up in less than 60 seconds and in many cases, the outage will not extend over 10 seconds. The dropping of non-essential load can be avoided, or reduced to a minimum, by arranging for another unit to come on the line at a lower percentage of the load capability of the operating units. This, however, will lower the overall economy of the system.

Fully automatic systems normally require real (kW) *load* sensing to add or delete units running. If the power factor of the system does not vary over ± 3 percent at any given percentage of the total system, then *current* sensing can be used with practically no loss in fuel economy. However, if the power factor variation from the mean is greater than ± 3 percent and maximum economy is desired, then real load sensing should be used.

Clock accuracy can be held to a slow drift of ± 10 seconds about a mean point which provides an accuracy of ± .0005%. The necessity of clock accuracy should be carefully weighed. For instance, if the system is visited daily, slight speed corrections can be made and the system can be held within ± 2 minutes per day or less after initial corrections are made without a clock control circuit. However, if the system is set up for a customary weekly visit by a serviceman, clock accuracy control equipment should be used. Electrically wound mechanical clocks should also be appraised from an economic and performance standpoint.

Automatic systems are generally made automatic simply to provide the capability of operating unattended and it follows that since they are unattended, they should have more safety devices



FLOW DIAGRAM, SUGGESTED AUXILIARY PIPING

Figure 12

than an attended power generating system. The engine should have the customary safety devices such as low oil pressure, high water temperature or high steam pressure, and overspeed. In addition, safety devices should be provided to shut down on high oil temperature, low coolant level, excessive vibration, and high inlet air temperature on turbocharged natural gas engines. It is also wise to program an engine off the line if it is not producing the power that the governor position is calling for.

As these fault conditions vary in their respective seriousness, they can and should be used for immediate shutdown if the fault will cause immediate danger. If, however, the fault is dangerous only when allowed to continue for long intervals, a standby unit can be started and placed on the line before the faulty unit is shut down, thereby avoiding any minor interruption of power. System units should also be equipped with reverse current trips so they cannot be motored.

TABLE 4

CONTROLS AND SAFETY DEVICES EBULLIENT DIESEL ENGINES

Malfunctions:

Unit	**Unattended**	**Attended**
Low oil pressure	I	I
High oil temperature	P	A
Excessive vibration	I	I
Overspeed	I	I
High inlet air temperature	P	A
Overcrank	I	
High water level	A	A
Low water level	I	A
Loss of power	I	A
Reverse power	I	I
Parallel timing	I	
Overcurrent	I	I
High steam pressure	A	A
Sync. relay failure	I	
System		
Low fuel level	A	A
Overload	A	A
Underfrequency	A	
Over/under voltage	A	
Battery charger failure (AC)	A	
Battery failure (DC)	A	
Low condensate level	A	A
Low tower water level	A	A

I — Immediate shutdown — next engine called
P — Programmed shutdown — next engine called (starts before first engine dropped)
A — Alarm only

INSTALLATION PRACTICES

As with all rotating machinery, engines generate noise, radiate heat, develop some vibration, and require periodic maintenance. However, these characteristics prove troublesome only when their existence is ignored — ignored, that is, during the planning and specification stage, for improperly installed machinery has a way of not allowing anyone to ignore it, including the manufacturers and the responsible engineers. On-Site Power installations are simple engine installations coupled to interdependent non-engine components such as steam separators, condensers, heat recovery mufflers, and related piping.

A complete cogeneration system offers few new installation or design problems. The concept does, however, combine installation and design problems of many previously unrelated functions and systems. The engineer who has designed steam power systems is now faced with designing reciprocating engines in a precise balance with the steam system, while the engineer who is familiar with engine drives now finds himself concerned with steam system design and installation situations. A step-by-step approach to the design and installation of the system's components will yield the most direct method of approach to the cogeneration concept.

The On-Site Power installation centers around the prime mover, therefore, the majority of the design and installation criteria is related to the engine, and justly so, for the engine generally accounts for most of the initial cost as well as the largest percentage of the operating costs.

ENGINE COOLING SYSTEM

Cooling is vital to all internal combustion engines, for without proper cooling the engine will be short lived. The heat rejection data reveals that about one third of the input energy is rejected to the engine cooling system. Regardless of how the heat is removed from the engine — by radiator, heat exchanger, or the ebullient system, the primary function is cooling the engine; and heat recovery, while vitally important economically, is of secondary importance operation-wise.

RADIATOR: Selection and Installation

Radiators, when used, should be installed in a manner that will insure a continuous supply of fresh air to the radiator, with particular provisions to prevent recirculation of air (unless recirculation is intentionally used as a means of heat recovery or temperature control, as is sometimes the practice in extremely cold climates). When the radiator is located in an opening in the engine room wall, a "blower" or "pusher" radiator fan is usually used. In any event, consideration should be given to the direction of prevailing winds. If wind direction is changeable, an air duct should be provided outside the wall to direct the air inlet or outlet, as the case may be, in a vertical direction, using a large radius "L" in the duct to avoid air turbulence or restricted flow. Horizontal, remotely mounted radiators using vertical air flow are often used to nullify the effects of changing wind direction. Such radiators are often used in cogeneration plants to dissipate all or part of the rejected engine heat during periods when there is little or no demand for heat by the facility being served.

Radiators should be sized to accommodate the necessary air flow required at the given altitude. At altitudes above sea level, increased air flow in CFM is usually required in order to maintain the equivalent weight of air per unit of time required at sea level. Also, as is the practice for sizing all types of cooling systems, radiators should be sized to accommodate a heat rejection load at least 15 percent greater than the established heat rejection of the engine. The additional 15 percent is intended to compensate for possible variations from published or calculated heat rejection rates, overloads, or engine malfunctions which might increase the heat rejection rate momentarily. It is not intended to replace all factors which affect heat transfer when calculating the heat transfer area required for selecting the proper tube material.

Radiator fan noise should be given adequate consideration when locating the air inlet or outlet. When remote mounted radiators are used, such as roof mounted units, if the static head resulting from the elevated location causes excessive pressure on the engine jacket or jacket water circulating pump 25 psi (172 kPa) or more, heat exchanger (shell and tube type) or hot/cold tank should be used in conjunction with the radiator, thus relieving the static head pressure from the engine. This requires an additional pump for circulating the coolant. In the case of a heat exchanger, an expansion tank will be required. Temperature of the jacket water circuit should always be thermostatically controlled to maintain a minimum inlet temperature of 180 °F (82 °C) with maximum change of temperature across the engine block of 20 °F (11 °C). When remote mounted radiators are used, an alarm system is usually desirable. If the system requires two circuits, (raw water and jacket water) each should be equipped with water level controls or alarms. High temperature (jacket water) safety shut down controls should always be used.

Antifreeze Protection

Installations which expose the engine coolant to subfreezing temperatures must add antifreeze to the water system. Ethylene glycol or Dowtherm 209 are recommended to protect against freezing, and to inhibit corrosion. Borate-nitrite solutions such as Caterpillar inhibitor or NALCO 2000 are compatible only with ethylene glycol, and can be used to replenish the original corrosion inhibitors in the antifreeze.

HEAT EXCHANGER: Selection and Location (Shell and Tube)

As with radiators, heat exchangers should be sized to accommodate a heat rejection rate approximately 15 percent greater than the established engine heat rejection.

The selected heat exchanger should accommodate raw water temperature and flow adequate to cool the engine when operating at maximum anticipated load, with the temperature differential between jacket water in and out of the heat exchanger not exceeding approximately 20 °F (11 °C) and not less than 10 °F (6 °C). Temperature of coolant entering engine should not be below the usually recommended 180 °F (82 °C).

Heat exchangers should always be located at a lower level than the coolant level in the surge tank, preferably several feet lower. The surge or expansion tank must be the highest level in the circuit, and must be located downstream from the heat exchanger. (A heat exchanger system requires a surge tank in the jacket coolant circuit). When the engine is mounted on spring type vibration isolators, it is good practice to install the heat exchanger on the floor near the engine, or at some location free of vibration. This will require flexible fittings in the coolant circuit.

JACKET WATER FLOW

When selecting a heat exchanger, or pump, for engine jacket water, the jacket water flow for a given engine can be calculated by using the following equation, which assumes a 15 °F (8.34 °C) temperature differential between "jacket water in" and "jacket water out":

$$\text{Flow (GPM)} = \frac{\text{Max. Rejected Heat (Btu/min.)} + 15\%}{15\,°\text{F} \times 8.1\ \text{lb./gal.}}$$

$$\text{Flow (L/sec)} = \frac{\text{Max. Rejected Heat (kW} \times 238.85) + 15\%}{8.34\,°\text{C} \times 970\ \text{G/L}}$$

As a safety factor, the equation provides flow to accommodate 15% more heat rejection than would be normally expected under maximum operating conditions. This equation does not apply for ebullient cooling.

AUXILIARY WATER REQUIREMENTS

All cooling systems utilizing jacket water temperatures above 220 °F (104.4 °C) require a cooling water circuit for oil cooler separate from the jacket water circuit. This is illustrated in Figures 7 through 11. Also, all turbocharged and aftercooled gas engines, regardless of cooling system temperature, require a separate cooling circuit for the aftercooler.

AFTERCOOLER AND OIL COOLER (TURBOCHARGED ENGINES)

On turbocharged-aftercooled engine arrangements for high temperature water or ebullient cooling, the aftercooler and oil cooler are connected in series. A good clean source of treated water should be used. The water circuit provided for this service is orificed to provide the recommended water flows, as listed in Table 5. In the event that a pump other than the one normally furnished with the engine is to be used for this circuit, the pressure drop across the aftercooler and oil cooler must be considered when determining the flow resistance or head. The total pressure drop across the aftercooler and oil cooler is as indicated in Table 5.

TABLE 5

AUXILIARY WATER FLOW REQUIREMENTS FOR EBULLIENT COOLED TURBOCHARGED ENGINES

	Water Flow (GPM or L/min) (Aftercooler & Oil Cooler)			Approximate Pressure Drop at Max. Flow	
Engine Model	**Min.**	**Max.**		**psi**	**kPa**
G399	80/302 ℓ/min	130/492 at 90 °/130 °F*	32 °C/54 °C	8.5	58
G398	80/302 ℓ/min	130/492 at 90 °/130 °F*	32 °C/54 °C	8.5	58
G379	80/302 ℓ/min	130/492 at 90 °/130 °F*	32 °C/54 °C	8.5	58
D399	80/302 ℓ/min	130/492 at 130 °F.	54 °C	8.5	58
D398	80/302 ℓ/min	130/492 at 130 °F.	54 °C	8.5	58
D379	80/302 ℓ/min	130/492 at 130 °F.	54 °C	8.5	58

* High Compression gas engines require 90 °F (32 °C) water to the aftercooler. Low compression gas engines require 130 °F (54 °C) water to the aftercooler.

Cat ebullient cooled engines are equipped with thermal bypass oil cooler circuits which maintain the lubricating oil temperature at 180 °F. ± 5 °F. (82 °C ± 2.7 °C) to the engine bearings. This system bypasses oil around the cooler when the engine is cold.

OIL COOLER ONLY
(Naturally Aspirated Gas Engines)

Naturally aspirated gas engines when used with high temperature cooling systems require a separate cooling circuit for the oil cooler; however, since there is no aftercooler in the circuit the maximum cooling water temperature (oil cooler circuit) may be as high as 160 °F (71 °C) as opposed to 90/130 °F (32/54 °C) for turbocharged engines. Flow and temperature data are listed in Table 6.

The minimum water flow is based on providing a sufficient tube velocity in the oil cooler to prevent tube fouling and to provide adequate cooling capacity. The maximum water flow is based on a tube velocity which will normally not cause tube erosion in the oil cooler.

The water flow rates for oil coolers obviously are quite high considering the relatively small amount of heat to be removed, thus the water temperature rise across the oil cooler will be quite low. For example: A G398 using the minimum flow of 65 GPM (246 Lpm) (Table 6) and operating with a 400 hp load would develop a temperature rise of only 4 °F. (2 °C)

$$\Delta T_F = \frac{400 \times 5.5}{65 \times 8.1} = 4.18°F$$

or

$$\Delta T_c = \frac{hp \times 0.0967 \times 238.85}{L/sec \times 970 \ G/L}$$

For applications where the oil cooler is not in the jacket water circuit or in series with an aftercooler on an engine, it is often practical and convenient to use water from a storage tank for this cooling circuit. In such an event, the water is simply pumped from the storage tank, through the oil cooler, and returned to the tank, using another circuit with a much lower flow rate to remove excess heat from the storage tank in the event that the temperature exceeds 160 °F. (71 °C).

TABLE 6

WATER FLOW REQUIRED FOR OIL COOLERS ONLY FOR NATURALLY ASPIRATED GAS ENGINES

Engine Model	Raw Water Flow (GPM) Min.		Max.		Max. Cooling Water Temp.		Approximate Pressure Drop at Max. Flow	
G399	65	245.7 ℓ/min	190	718 ℓ/min	160 °F	71 °C	2 Psig	13.79 kPa
G398	65	245.7 ℓ/min	190	718 ℓ/min	160 °F	71 °C	2 Psig	13.79 kPa
G379	65	245.7 ℓ/min	190	718 ℓ/min	160 °F	71 °C	2 Psig	13.79 kPa

EBULLIENT COOLING

Ebullient cooling entails a few design requirements that are peculiar to this type of cooling. The steam separator, load balancing condenser, and condensate return system become an integrated part of the engine cooling circuit, and the engine depends on these units to remove the heat as it would a radiator or heat exchanger in a conventional cooling system.

The engine should be equipped as an ebullient cooled arrangement with no water pump in the jacket system, proper provision for oil cooling, and proper coolant piping. The steam separator (not a part of the engine arrangement) should include gauges and safety valves to limit pressure and temperature to 15 psi (103 kPa) and 250 °F. (121 °C). The separator should have a high water alarm, a low level alarm, and a low water shutdown along with a pressure regulating valve to maintain minimum steam pressure in the separator when the downstream system pressure drops below a given value. This latter item is of vital importance and is necessary to prevent "flashing" within the engine. The usual practice is to select a valve that closes at a pressure approximately 3 psi (20.7 kPa) below the design operating pressure. See Figures 9 and 10 for valve location. The steam separator should also include modulating liquid level control, water gauge glass, and a vacuum breaker to relieve the vacuum created when the plant is shut down. The low water shut down level should be a minimum of 3 feet (.91 m) above the highest water level in the engine. The height of the steam separator above the engine, however, should be limited to not more than 10 feet (3.0 m) when using 15 psi (103.4 kPa) steam, in order to limit the total pressure on the engine jacket to approximately 20 psi (138 kPa).

COOLANT FLOW ENGINE (Ebullient Cooling)

The flow of coolant through an ebullient cooled engine (with no pump) will normally be approximately 25 lb. (11.34 kg) of coolant per pound of steam produced. All coolant piping to and from the engine should be sized accordingly and long runs between the engine and steam separator should be avoided.

PIPING AND CONNECTIONS

A steel "bellows" type expansion joint, or equal, should be used for the connection between the engine coolant outlet and the piping leading to the steam separator. The expansion joints should be adequate to compensate for dimensional changes in the respective piping systems caused by temperature as well as providing vibration isolation between the engine and all connected systems. Rubber hose should never be used.

The best rule for design and installation is to never use a pipe size smaller than the engine fitting to which it is to be connected. The 3″ (76.2 mm) inlet connection on the G398 is intended to be connected to a 3-inch (76.2 mm) pipe to insure proper operation of the engine. The jacket system pipe sizes provided with Cat ebullient cooled engine arrangements are given in Table 7.

TABLE 7

PIPE SIZES FOR CAT
EBULLIENT COOLED ENGINES

Engine Model	Outlet Connections Std. 150 lb. Flange		Inlet Connections Std. Pipe Thread	
D/G399	Two 4″	101.6 mm	One 3″	76.2 mm
D/G398	Two 4″	101.6 mm	One 3″	76.2 mm
D/G379	Two 4″	101.6 mm	One 3″	76.2 mm

These sizes are provided to limit the water velocity to approximately 5 ft./sec. (1.52 m/sec) or less. Figure 13 can be used to determine the water velocity in standard pipe sizes, or to determine the pipe size required when the flow and velocity are known.

WATER VELOCITY NOMOGRAPH
Figure 13

When sizing pipes for engine cooling systems, the water velocity should be limited to a maximum 5 feet per second (1.52 m/sec). To determine the water velocity, the rate of flow and pipe size is required and any two of the variables will yield the third.

Example 1
Determine the velocity of water flowing through a standard pipe with an internal diameter of 2.067 inches (52.5 mm) [nominal size 2 inches (50.8 mm)] and a rate of 500 lbs. (226.8 kg) per minute. Extend a line from 500 on the flow scale, to 2.067 (52.5 mm) on the internal diameter scale and read 5.70 feet (1.73 m) per second at the intersection with the velocity scale.

Example 2
Find the velocity of water flowing at 400 gpm (1514 ℓ/min) through a standard 6 inch (152.4 mm) internal diameter pipe. Extend a line from 400 on the flow, scale to 6 inches (152.4 mm) on the nominal diameter scale and read 4.4 feet per second (1.34 m/s) at the intersection with the velocity scale.

JACKET WATER TREATMENT

Only soft water treated with a suitable inhibitor should be used in the jacket water system. Make-up water should also be treated. Consult a reputable water treatment specialist in the area for a recommendation. The degree of acidity or alkalinity of the coolant should be maintained at a pH value of 6.5 to 8. A latent heat of vaporization greater than 90 percent of the latent heat of vaporization of water is required. To maintain the proper level of treatment, check the coolant daily, or more often, using methods described in the ASTM Manual on Industrial Water.

LOAD BALANCING HEAT EXCHANGERS

Load balancing condensers are required for heat recovery systems which produce steam and load balancing heat exchangers for systems using high temperature water. In either case the load balancing unit, or units, must be sized to accommodate the maximum heat rejection from the engine, or engines, in order to assure adequate engine cooling during periods when the demand for heat is low or non-existent.

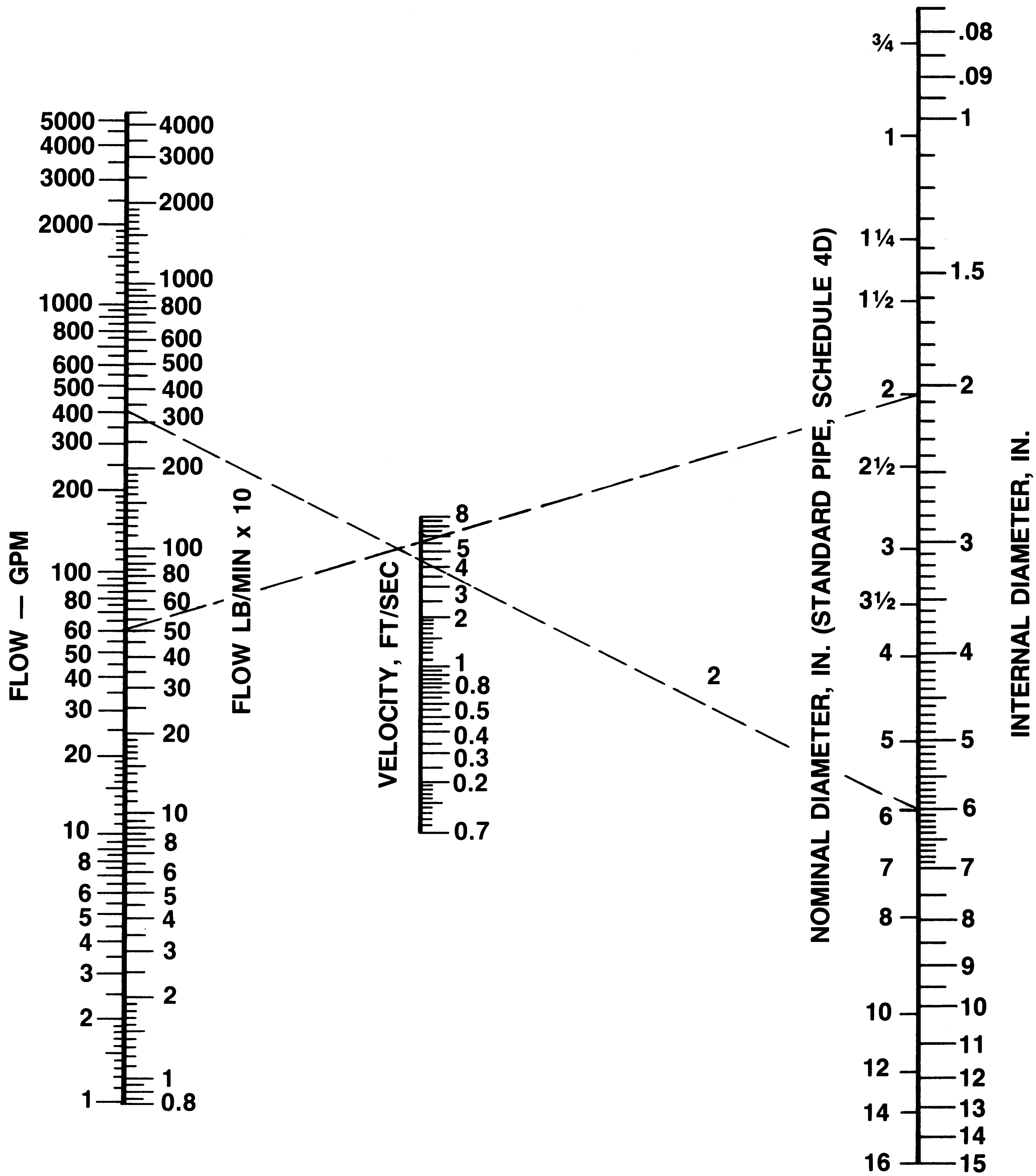

FLOW — GPM
FLOW LB/MIN x 10
VELOCITY, FT/SEC
NOMINAL DIAMETER, IN. (STANDARD PIPE, SCHEDULE 4D)
INTERNAL DIAMETER, IN.
2

Figure 13

CONDENSATE TANKS AND PUMPS

The tank should be sized to provide adequate make-up water when the plant is operating at full capacity. The time required for the steam to pass completely through the system and return must also be taken into account. Any condensate tank make-up water should be treated before being added to the tank.

The condensate pump should be sized about three times the evaporative rate of the heat recovery units. If a boiler is included, an amount 1½ times the boiler evaporative rate should be added to the pump's capacity. Pressure should be calculated with wide open level controller at full flow.

Centrifugal pumps are recommended. They are generally long lasting, jam proof, non-overloading and inexpensive. The pump should run continuously and a standby pump should always be in parallel with the prime condensate pump. In steam systems, a sub cooler may be required to assure that the return to the condensate tank is solid water.

ENGINE EXHAUST SYSTEMS

A good exhaust system should (1) reduce the noise to an acceptable level as determined by the demands of the location; (2) should not impose a backpressure of more than 27 in. H_2O (6.73 kPa) for naturally aspirated and turbocharged units, measured at the engine exhaust manifold outlet or turbocharger outlet; (3) should discharge the exhaust at a point not harmful or annoying to people or industry; and, (4) should dissipate a minimum amount of heat to the engine room.

It is good practice to locate the muffler as close to the engine as practical. A stainless steel "bellows" type flexible connection should be used at the engine exhaust manifold outlet. The weight of the piping system must be supported independently of the engine and should be so installed that the expansion and contraction of the pipe will not impose damaging forces upon either the engine or the muffler (see Figure 14).

If the exhaust system develops excessive back pressure, an induced draft fan should be used at or near the exhaust outlet. This might be necessary where engines are located in the basement of a multi-story building and the environment is such that the exhaust must be discharged at roof level. For such installations, the fan should be sized to reduce the back pressure to acceptable limits.

For naturally aspirated engines, even a small vacuum might be desirable 2 in. H_2O (0.49 kPa); however, for turbocharged engines, a vacuum is not desirable because of possible damage to turbocharger seals, and a small positive pressure should prevail.

Exhaust noises can be reduced somewhat at the point of discharge by cutting the end of the discharge pipe at an angle of approximately 60 degrees to the axis of the pipe.

If possible, a complete exhaust system, engine to atmosphere, should be provided for each engine. When several units discharge exhaust into a common header, exhaust gas will find its way into any non-operating engine; this will usually result in condensation of the water vapor formed by combustion. This deposits water in the piping system and inside the engines not in operation. Although a common exhaust system is not recommended, the damaging effects inherent with such a system can be reduced by employing some rather expensive precautions; (1) install a suction fan at the system outlet to maintain system back pressure within the limits given previously and (2) install automatically operated cut off valves between each engine and the common header. Such valves should close when the engine is not in operation and open when the signal is received to start the engine. Also, to protect the engine in the event of a valve malfunction (failure to open), an engine shut down or interlocking control should be applied for each engine, sensing high back pressure or valve position.

When an exhaust heat recovery muffler or boiler is used, the same requirements for installation apply. In addition to providing support and the necessary flexible fitting, care should be taken that the recovery unit is *never operated dry.* When a steam generating type is used, a modulated make-up water circuit is required to maintain a full water level in the muffler. Failure of the unit is inevitable when the muffler is allowed to operate either dry or partially filled.

Approximately 2.25 lb. (1.02 kg) of water are formed as a product of combustion for every 25 cu. ft. (.703 m^3) of natural gas burned. When gas is burned in an internal combustion engine, the water so formed is discharged with the exhaust gas in the form of steam. Thus, an engine producing 400 hp (298 kW) and using 3500 cu. ft. (99.12 m^3) of fuel (natural gas) per hour will discharge approximately 300 lbs (136 kg) or 36 gal. (136 ℓ) of water per hour. Therefore, any exhaust piping system of such extended length or exposed to cool temperatures to the extent that the exhaust gas temperature may drop below the dew point of water vapor in the exhaust gas, should be equipped with proper traps or drainage features to prevent the condensed water from draining back to the engine or muffler.

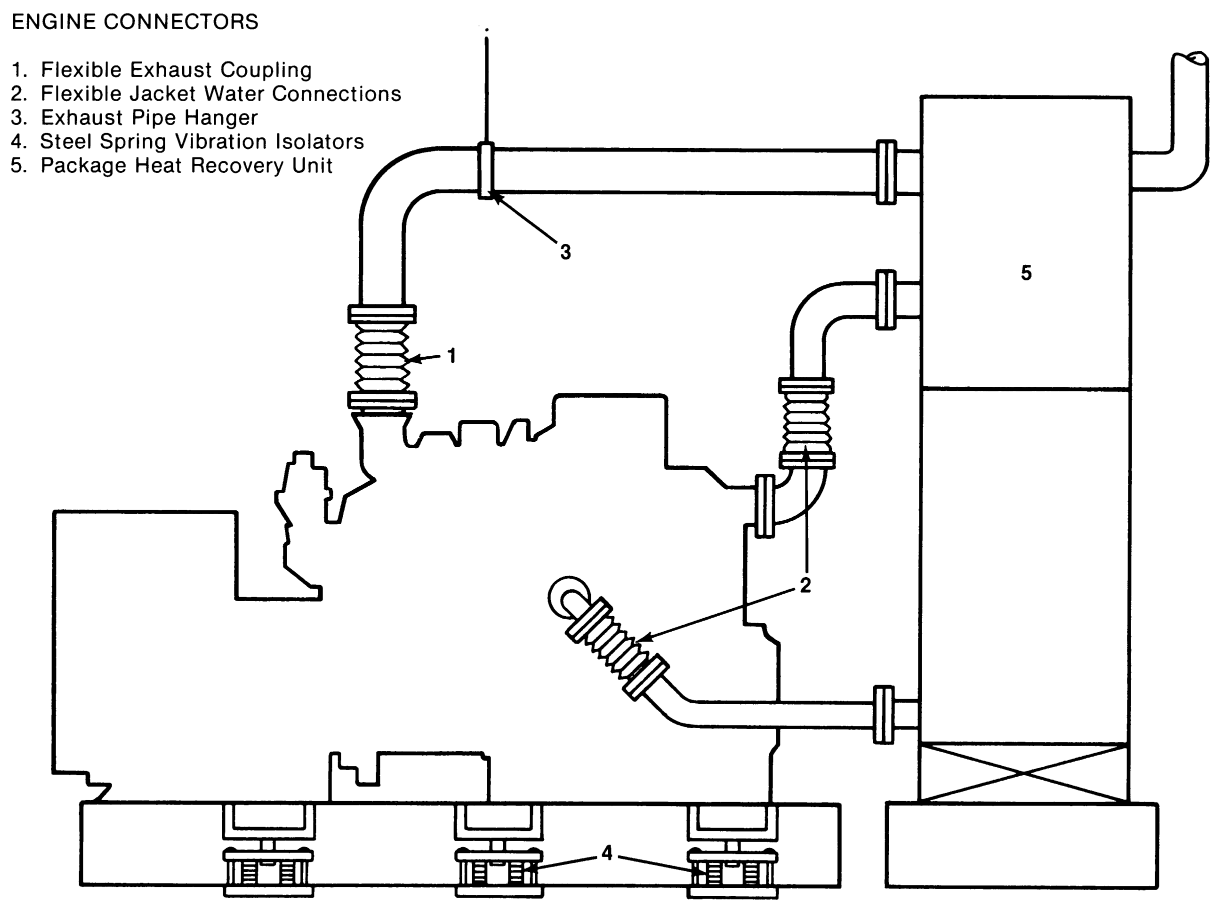


Figure 14 **ENGINE MOUNTING AND FLEXIBLE CONNECTIONS**

FOUNDATIONS AND ENGINE MOUNTING

Modern, multi-cylinder, medium speed engines do not require massive concrete foundations, although concrete often offers advantages in cost and in maintaining alignment for certain types of driven equipment. Fabricated steel bases have proven satisfactory for direct coupled, self-contained units such as generator sets. Steel bases mounted on vibration isolators (steel spring or equal) are completely adequate and need no special foundation other than a floor designed to accommodate the weight (see Figure 14).

Concrete bases are also satisfactory for such units provided such bases are equally well isolated from the supporting floor or sub-floor. Fiber glass blocks have proven quite effective as isolation material for concrete bases. Concrete bases need only be thick enough to prevent deflection. Excessively thick bases increase sub-floor or soil loading. Such bases should be supported by a concrete sub-floor, using some type of acceptable isolation material between the base and sub-floor. An engine base or foundation should never rest directly upon natural rock formations to avoid vibration transmission.

Concrete Base

Several basic foundation designs are applicable for generator sets. The foundation chosen will depend on the factors previously outlined as well as on limitations imposed by the individual location and application. Massive concrete foundations are not necessary for modern multicylinder medium-speed generator sets.

If a concrete foundation is required, some *minimum* design guidelines to consider are:

- The foundation length and width should exceed the length and width of the generator set a minimum of 1 ft (0.305 m).
- The foundation depth should be sufficient to attain a minimum weight equal to the generator set wet weight.

To calculate the necessary foundation depth, use:

$$\text{Foundation Depth (ft)} = \frac{W}{150 \times B \times L}$$

$$\text{(m)} = \frac{W}{2400 \times B \times L}$$

W = Total wet weight of generator set (pounds) — (kg)
150 = Density of concrete (pounds per cubic foot)
2400 = Density of concrete (kilograms per cubic meter)
B = Foundation width (feet) — (meters)
L = Foundation length (feet) — (meters)

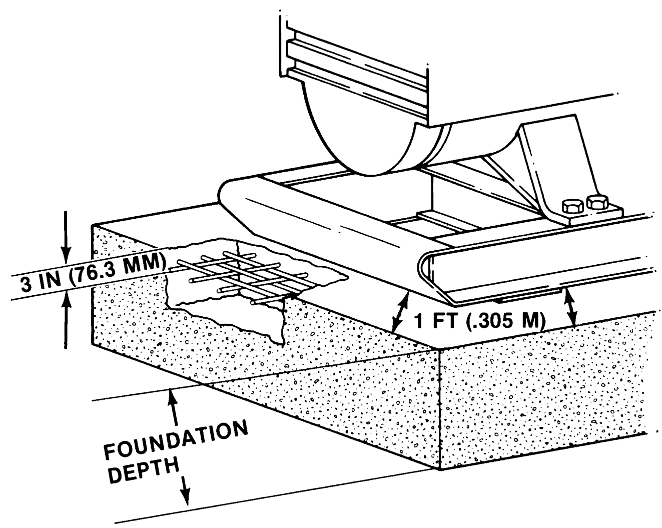


Figure 15

Suggested concrete mixture by volume is 1:2:3 of cement, sand, aggregate with a maximum four-inch — (100 mm) slump with a 28-day compressive strength of 3000 psi (200 MPa).

The foundation should be reinforced with No. 8 gauge steel wire fabric or equivalent, horizontally placed on 6 in (150 mm) centers. An alternate method of reinforcing is to place No. 6 reinforcing bars on 12 in (300 mm) centers horizontally. Bars should clear the foundation surface a minimum of 3 in (75 mm).

When vibration isolation equipment is used, the floor depth required is that needed for structural support of the static load. If isolators are not used, dynamic loads will be transmitted to the facility floor and the floor must be designed to support 125% of the generator set weight.

If the generator set is to operate in parallel with other units, the possibility of out-of-phase paralleling and resultant increased torque reactions demand a stronger foundation. This foundation must be designed to withstand a weight 2 times the wet weight of the generator set.

Excessively thick, heavy bases should be avoided since they increase subfloor or soil loading. They should be thick enough to prevent deflection and torque reaction, while still retaining sufficient surface area to satisfy the supporting material.

FUEL SUPPLY SYSTEMS

The prime function of gaseous fuel supply system is to deliver clean gas to the engine carburetor at the required pressure and in an adequate volume to insure rated engine output, with a minimum of delay in response when the engine is subject to sudden load changes. All piping and accessories must comply with local codes.

An automatic shut-off valve should be placed in the fuel supply line to each engine; it should be located as close to the engine as practical but so located that the line upstream from the cut-off valve will not be exposed to damage in the event of equipment failure. It is also advisable, and required by some codes, to place a manually operated cut-off valve in the gas line upstream from any automatic shut-off valves in the system. A flexible connection should be placed in the fuel line at some point between the cut-off valve nearest the engine and the engine mounted regulator. A cleanable strainer (wire mesh type or equivalent) should be placed in the fuel supply line to each engine, located upstream from the regulator.

When two gaseous fuels of different heat values are to be used alternately through the same carburetor, two sets of regulators and fuel supply lines must be provided, one for each fuel.

GAS PRESSURE

Naturally-aspirated Engines: Cat naturally-aspirated gas engines require gas pressure in the range of 5½ inches (1.37 kPa) of water, or 3½ oz. (99.2 g) gauge, at the carburetor when operating at sea level and at rated output using natural gas with a high heat value of 1000 Btu/cu. ft. (37.28 kg/ℓ). The exact pressure required at the carburetor varies with the heat value of the fuel. Fuels with a high heat value less than 1000 Btu/cu. ft. (37.28 kg/ℓ) will require slightly higher pressure. Precise gas pressure regulation within ± ½ inch (0.12 kPa) of water to the carburetor from no load to full load is essential for best performance and fuel economy. Thus, a pressure regulator should always be used and should be located in close proximity of the carburetor [within 1 to 3 ft. (.3 to .9 m)]. For good engine response to sudden load changes, gas pressure to the regulator should be not less than one psi. Since specifications for the regulator(s) will be dictated by the gas "line" pressure available, the heat value of the fuel, and to some degree, the type of load, either specific recommendations should be obtained from the engine manufacturer or the engine specifications should call for the engine to be equipped with a regulator or regulators suitable for a specified set of prevailing conditions with regard to fuel supply pressure, heat value, type load, etc.

Turbocharged engines: Caterpillar turbocharged natural gas engines are equipped with regulators to accept gas pressures to approximately 25 psi (172 kPa). Turbocharged engines require gas pressure of approximately 10 psi (69 kPa) minimum at sea level for loads not requiring precise engine governing and 12 psi (83 kPa) minimum for electric power generation or service requiring quick engine response. Higher line pressure assures better response. Also, since engine intake manifold pressure must increase with altitude, gas supply pressure must likewise increase with altitude if sea level performance is to be maintained. The minimum gas pressures of 10 (69 kPa) and 12 psi (83 kPa) respectively must, therefore, be increased by an amount equal to the loss in barometric pressure due to altitude.

DIESEL FUEL SYSTEMS

A diesel engine fuel system should include an auxiliary "day tank" in or near the engine room to provide an immediate supply of fuel to the engine driven transfer pump. The auxiliary tank voids the need for lengthy return lines and permits the location of venting, fuel shutoffs, and strainers to be accessible to the operator. The size of the day tank will vary according to the number and size of engines being served; however, the 1981 U.S. National Electrical Code, Section 701-6(b)(2), calls for an on-site fuel supply capable of operating the prime mover at full demand load for at least two hours.

Fuel transfer from the main storage tank to the day tank is usually accomplished by an electrically driven pump, controlled by a float actuated switch at the day tank to maintain the proper fuel level in the day tank.

For Cat Diesels, the day tank should be located above the engine fuel transfer pump to provide gravity flow to the engine.

LUBRICATING OIL SYSTEM

The lubricating oil supply system beyond the engine should provide storage for "make-up" oil in quantities proportionate to the demands of the engines. The supply tank should be sized to accommodate a minimum of 30 days of operation without refilling. If possible, the oil supply tank should be so located that oil will flow by gravity to the engine. It should be conveniently accessible for refilling and, in cold climates, should be in a heated building. An automatic oil level control should be applied to each engine to provide automatic addition of make-up oil as required.

In multiple engine installations the economics quite often justify installing a piping system to facilitate the lubrication oil changes. A line to drain the oil and a second line to provide new oil from a common supply would greatly reduce maintenance time for large installations.

ENGINE ROOM VENTILATION

Approximately 6 to 8 percent of the fuel consumed by the average internal combustion engine is dissipated to the surrounding air by radiation. Removal of this heat is an absolute necessity, and is usually accomplished by the use of induced draft or ventilating fans.

Air should be removed from the immediate vicinity of each engine in a manner that will insure an upward flow of air around each engine. The ventilating system should further provide sufficient air distribution and removal to prevent excessive temperatures in any part of the engine room.

In applications where the engine room noises must be contained within the engine room, the ventilating air should be supplied to, and removed from, the engine room through sound insulated air ducts with properly located inlets and outlets to insure a minimum of noise transmittal. The use of louvered openings in the engine room wall for ventilating air inlets or outlets is, generally speaking, not satisfactory when noise must be contained.

To correctly size the engine room ventilating fan, or fans, the following equation will provide a quick determination of the amount of air required:

$$\text{cfm} = \frac{400 \times \text{hp}}{T}$$

Where: hp = Maximum engine horsepower.
T = Equipment room temperature rise above ambient degrees F.

Or:

$$\text{m}^3/\text{s} = \frac{0.14 \times \text{kW}}{T}$$

Where: kW = Maximum engine kilowatts.
T = Equipment room temperature rise above ambient degrees C.

The air flow should be increased 10% for every 2,500 ft (760 m) above sea level.

It should be recognized that in cold weather the desired temperature rise in the engine room may be as much as 80 °F. (27 °C) instead of the usual 10 °F. or 20 °F. (5.5 to 11 °C) rise when, for instance, the ambient air is at −10 °F (−23 °C). In such a case, theoretically, only 1/8 of the ventilating air is required and, hence, it is good practice to use a number of smaller fans rather than one large unit. This also permits correct ventilation at reduced plant output.

COMBUSTION AIR

A diesel engine requires approximately 2.5 cfm of air per horsepower or 0.09 m³/min per kW produced.

An ample supply of cool, clean air is equally as essential for good engine performance as an adequate supply of fuel. The cooler the air, the higher the potential output; thus, while not absolutely necessary for well ventilated engine rooms, it is always good practice to extend the air intake piping from the engine air cleaners to a suitable outside point, exercising care in locating the air inlet to avoid contaminants such as engine exhaust, process fumes, dust, etc.

As a rule, the air cleaner should remain attached to the engine where it can be serviced with convenience. Restriction in the intake piping to the air cleaner should not exceed three inches of water when the engine is operating at rated output. The air flow required for gas engines can be calculated by multiplying the fuel consumption (in cfm) by ten since the air-fuel ratio remains essentially constant at 9.5:1 to 10:1.

SOUND ISOLATION

Rotating machinery generates noise which usually should be contained within the equipment room. Good practice dictates the provision of a complete enclosure for the engine room.

Eight or ten inch concrete block filled with sand or poured concrete walls and concrete ceiling will reduce the sound pressure level beyond the engine room to acceptable levels for most facilities. Further reduction in sound pressure level within the engine room may be had by insulating the engine room walls and ceiling with a layer of fiber glass or equal, covered with a perforated wall board.

SPACE FOR ENGINE MAINTENANCE AND SERVICE

Floor space between an engine and a parallel wall or between engines should not be less than the width of the engine. Overhead space, i.e., space between the top of the engine and the nearest obstruction or ceiling, should be adequate to permit convenient removal of cylinder heads, manifolds, exhaust piping, etc., for service. For larger engines, provisions should be made to permit the use of a chain hoist or overhead crane to remove the heavier components.

Space between either end of the engine and the nearest wall or other obstruction should be adequate to permit removal of certain components or parts such as camshafts, which may have to be "pulled" from one end of the engine.

When installing package type heat recovery units, ample space should be provided between the engine and recovery unit to allow maintenance on the engine or the recovery unit.

CONTROL AND SAFETY DEVICES

See Table 4 and section on switchgear.

LOCATION AND PROTECTION OF SWITCHGEAR

It is always desirable to provide a separate enclosure or room for all switchgear and control panels located remotely from the engines and other equipment. Such a room usually consists of an enclosure within the equipment or engine room, isolated and insulated against equipment room noise and affording a window for visual observation of the equipment. The room should be well ventilated or air conditioned to protect the switchgear from engine room heat, miscellaneous vapors, etc.

LOAD ANALYSIS

A load analysis for a plant or facility to be complete must encompass a study of the energy requirements for each of the three major services: air conditioning; heating; and electric power and lighting. Complete treatment of all three areas is beyond the scope of this publication; however, much of the basic data employed when making an analysis of the electric load can be assembled and presented in a limited space. The material offered on the pages that follow is an attempt to provide some of the more useful of such data for those engaged in the preparation of load analyses and economic studies for on-site power applications.

THE ELECTRIC LOAD

The first step in making an analysis of the electric load is to develop and plot a family of load profile charts. A minimum would consist of one chart for a representative 24-hour period and one for a representative 12-month period. The 24-hour chart should reflect the average kilowatt load for each hour, or in some instances each half hour, while the 12-month chart should represent the average kW load for each month. In some instances an additional set of charts should be made to reflect anticipated growth.

The development of load profile charts is equally essential and useful whether they apply to existing loads or to proposed loads for facilities not yet constructed — only the method of development differs. Such charts provide a basis for the selection of power generating equipment as well as data for feasibility and economic studies.

For existing loads which are being served by an electric utility company or a similar source of power, either power bills or power consumption records for a representative 12-month period will provide most of the data needed for a 12-month average load profile. The average kW load for each month can be determined by dividing the total kilowatt hours used during the month, by the total number of hours of operation during that month.

$$\text{Avg. kWh Load} = \text{total kWh used} \div \text{Total hour of operation}$$

The average kilowatt load will always be lower than the maximum kilowatt demand of the plant. Therefore, the average load as determined above cannot be used without reservation to establish the engine and generator requirements, unless the load is known to be steady, such as a single air conditioning unit or pump to be operated by the generator. However, when the average monthly kilowatt load is determined for any month, it can be plotted on a bar graph similar to Figure 16. This, when completed for a year, can illustrate seasonal variation in load and help in the selection of a proper size and number of electric sets.

Most power bills show a demand charge. However, unless the bill clearly points out how this figure is determined, it is not advisable to use it. Some bills will give the maximum kW demand. This figure is usually the highest 15- or 30-minute average demand during the month, and does not show momentary peak kW demands which may be caused by starting large motors and certain other equipment. *The average demand is the normal load on the electric set.* Because the generator has a momentary overload capacity, it is capable of absorbing the peak demand provided it does not exceed its momentary overload capacity. However, the generator must be sufficiently large to continuously supply the power necessary for the average demand.

Acquiring load data from existing plants is a relatively simple operation. Developing similar data, however, for a plant or facility not yet constructed is quite another operation. Most load analyses associated with on-site power generation will be in the latter category. The data available from existing facilities provides a most important source of information for use when estimating loads and power consumption for new facilities. A limited amount of these data are given in Table 8. While the figures given are admittedly average and subject to some variation in different geographic locations because of climate and operating conditions, they are quite reliable and are particularly useful for making preliminary or exploratory feasibility and economic studies of proposed installations.

TABLE 8

LOAD AND POWER CONSUMPTION ESTIMATING DATA

	Watts/Square Foot	**Power Usage $kWh \bullet /ft^2 \bullet yr$**
Schools	4-5 average (43.06-53.82 W/m^2) (Total)	11 to 17 (1.02 to 1.58)
Class Rooms	5-6 (53.87-64.58)	
Locker Rooms, Auditoriums	2-3 (21.53-32.29)	
Halls and Corridors	20 Watts per running foot (65.62 W/m)	
Shopping Centers	5 average (Total)	28 to 34 (2.6-3.2)
Stores, Large Department and Specialty Stores	5-6 (53.87-64.58)	
Show Windows	500 Watts per running foot (1640.5 W/m)	
Office Buildings	5-6 average	28 to 34 (2.6-3.2)
Private and General Offices	4 (43.06)	
Professional Offices	6-7 (64.58-75.35)	
Dentist, Drafting Rooms, etc.	7 (75.35)	
Hotels and Motels	32.29-43.06 average (Total)	12 to 17 (1.1 to 1.58)
Lounge	2 (21.53)	
Rooms	3 (32.29)	
Dining Rooms	4 (43.06)	
Exhibition Halls, Shops, Lobby, Kitchen	3 (32.29)	
Hospitals	1.5 to kW per bed average	8500 to
Lobby, Wards, Cafeterias	3 Watts/Sq. Ft. (32.29 W/m^2)	11400 kWH
Private Rooms, Operating Rooms	5 Watts/Sq. Ft. (53.82 W/m^2)	per bed
Operating Tables Major surgeries	3000 Watts each	per year
Minor surgeries	1500 Watts each	
Apartment Houses	2-3 kW per unit (Total)	11 to 17 (1.02 to 1.58)
Lobby	2 Watts/Sq. Ft. (21.53 W/m^2)	
Apartments	3 Watts/Sq. Ft. (32.29 W/m^2)	
Small Appliances	1.5 kW/unit	

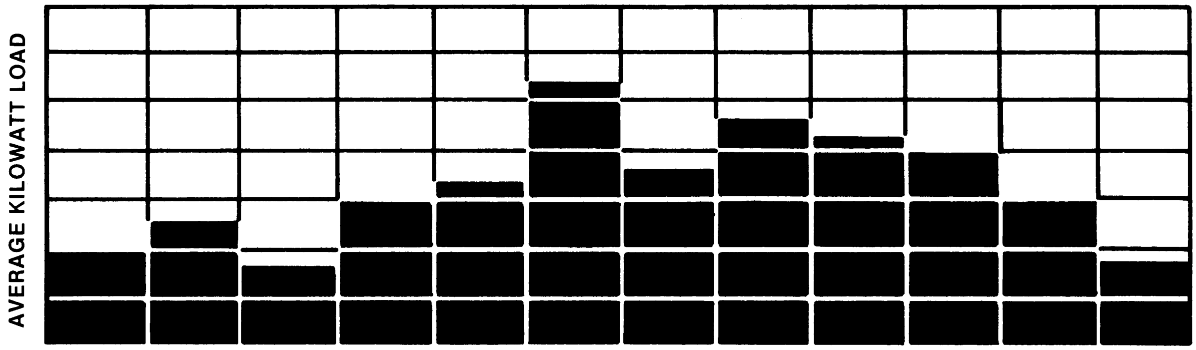


Figure 16 **Average monthly kW load curve showing seasonal variation**

ENGLISH TO METRIC CONVERSION FACTORS

SYMBOL	WHEN YOU KNOW	MULTIPLY BY	TO FIND	SYMBOL
Btu	BRITISH THERMAL UNIT	1055.06	JOULE	J
Btu/hp•h	BRITISH THERMAL UNIT/ HORSEPOWER-HOUR	0.001 42	MEGAJOULES/KILOWATT-HOUR	MJ/kW•h
Btu/h	BRITISH THERMAL UNIT/ HOUR	1055.06	JOULES/HOUR	J/h
Btu/min	BRITISH THERMAL UNIT/ MINUTE	0.017 58	KILOWATT	kW
Btu/ft³	BRITISH THERMAL UNIT/ CUBIC FOOT	8.8906434	KILOCALORIES/CUBIC METER	Kcal/m³
°C	CELSIUS (DEGREES)	[(1.8 C) + 32]	FAHRENHEIT (DEGREES)	°F
cu ft	CUBIC FEET	0.028 32	CUBIC METER	m³
cu ft/h	CUBIC FEET/HOUR	0.028 32	CUBIC METER/HOUR	m³/h
cfm	CUBIC FEET/MINUTE	0.028 32	CUBIC METER/MINUTE	m³/min
cu in	CUBIC INCH	0.016 39	LITER	L
cu in	CUBIC INCH	0.000 02	CUBIC METER	m³
°F	FAHRENHEIT (DEGREES)	[0.5555 (F-32)]	CELSIUS (DEGREES)	°C
ft/min	FEET/MINUTE	0.3048	METER/MINUTE	m/min
ft	FEET	0.3048	METER	m
ft H_2O	FEET OF WATER	2.988 98	KILOPASCAL	kPa
gph	GALLON/HOUR	3.785 41	LITER/HOUR	L/h
gpm	GALLON/MINUTE	3.785 41	LITER/MINUTE	L/min
hp	HORSEPOWER	0.7457	KILOWATT	kW
in Hg	INCH OF MERCURY	3.376 38	KILOPASCAL	kPa
in	INCH	25.4	MILLIMETER	mm
in H_2O	INCH OF WATER	0.249 08	KILOPASCAL	kPa
kW	KILOWATT	56.869 03	BRITISH THERMAL UNIT/MINUTE	Btu/min
L	LITER	61.0237	CUBIC INCH	cu in
μ	MICRON	1.0	MICROMETER	μm
lb	POUND	0.453 59	KILOGRAM (MASS)	kg
lb	POUND	4.448 22	NEWTON (FORCE)	N
lb ft (ft-lb)	POUND FOOT	1.355 82	NEWTON METER	N•M
lb in (in-lb)	POUND INCH	0.112 99	NEWTON METER	N•M
lb/in	POUNDS/INCH	0.175 13	NEWTON/MILLIMETER	N/mm
lb/in	POUNDS/INCH	175.127	NEWTON/METER	N/m
lb/HP-h	POUND/HORSEPOWER-HOUR	608.277	GRAM/KILOWATT HOUR	g/kW•h
lb/h	POUND/HOUR	0.453 59	KILOGRAM/HOUR	kg/h
m³	CUBIC METER	61 023.7	CUBIC INCH	cu in
psi	POUNDS/SQUARE INCH	6.894 76	KILOPASCAL	kPa
US qt	US QUART	0.946 35	LITER	L
ft²	SQUARE FEET	0.0929	SQUARE METER	m²
in²	SQUARE INCH	6.4516	SQUARE CENTIMETER	cm²
US gal	US GALLON	3.785 41	LITER	L

AREA EQUIVALENTS

UNIT	SQ. CM.	SQ. IN.	SQ. M.	SQ. FT.
1 Sq. Cm.	1	0.155		
1 Sq. In.	6.4516	1	.00064516	.006944
1 Sq. M.	10,000	1550	1	10.764
1 Sq. Ft.	929	144	0.0929	1

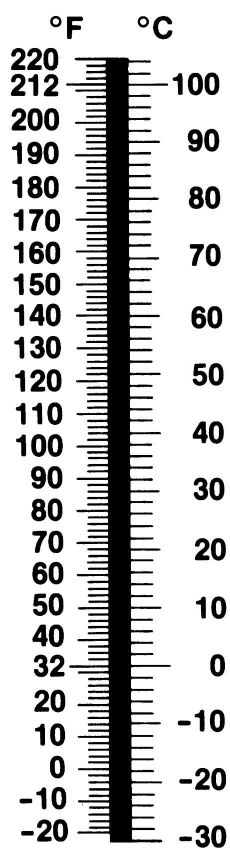
VOLUME AND CAPACITY EQUIVALENTS

UNIT	in^3	ft^3	yd^3	cm^3	m^3	US gal	Imp gal	liter
in^3	1	0.000 58	0.000 02	16.3871	0.000 02	0.004 32	0.003 61	0.016 39
ft^3	1728	1	0.037 04	28 316.8	0.028 32	7.480 52	6.228 83	28.3169
yd^3	46 656	27	1	764 554	0.764 55	201.974	168.178	764.555
cm^3	0.061 02	0.000 04	—	1	—	0.000 26	0.000 22	0.001
m^3	61 023.7	35.3147	1.30795	1 000 000	1	264.172	219.969	1000
US gal	231	0.133 68	0.004 95	3785.41	0.003 78	1	0.832 67	3.785 41
Imp gal	277.419	0.160 54	0.005 95	4546.09	0.004 55	1.200 95	1	4.546 09
liter	61.0237	0.03531	0.001 31	1000	0.001	0.264 17	0.219 97	1
acre — ft	—	43 560	1613.33	—	1233.48	325 851	271 335	—

LENGTH EQUIVALENTS

UNIT	cm	in	ft	yd	m	km	mile
cm	1	0.3937	0.032 81	0.010 94	0.01	0.000 01	—
in	2.54	1	0.083 33	0.027 78	0.0254	0.000 03	—
ft	30.48	12	1	0.333 33	0.3048	0.000 30	—
yd	91.44	36	3	1	0.9144	0.000 91	—
m	100	39.3701	3.280 84	1.093 61	1	0.001	0.000 62
km	100 000	39 370.1	3280.84	1093.61	1000	1	0.621 37
mile	160 934	63 360	5280	1760	1609.34	1.609 34	1

TEMPERATURE CONVERSION



TYPICAL FRICTION LOSSES OF WATER IN PIPE
(OLD PIPE)

FLOW		HEAD LOSS IN FEET PER 100 FT. (m per 100 m)							FLOW	
gpm	**(l/s)**	**3/4" (19.05 mm)**	**1" (25.4 mm)**	**1-1/4" (31.75 mm)**	**1-1/2" (38.1 mm)**	**2" (50.8 mm)**	**2-1/2" (63.5 mm)**		**gpm**	**(l/s)**
5	.34	10.5	3.25	0.84	0.40	0.16	0.05	**3" (76.2 mm)**	5	.34
10	.63	38.0	11.7	3.05	1.43	0.50	0.17	0.07	10	.63
15	.95	80.0	25.0	6.50	3.05	1.07	0.37	0.15	15	.95
20	1.26	136.0	42.0	11.1	5.20	1.82	0.61	0.25	20	1.26
25	1.58	**4" (101.6 mm)**	64.0	16.6	7.85	2.73	0.92	0.38	25	1.58
30	1.9	0.13	89.0	23.0	11.0	3.84	1.29	0.54	30	1.9
35	2.21	0.17	119.0	31.2	14.7	5.10	1.72	0.71	35	2.21
40	2.52	0.22	152.0	40.0	18.8	6.60	2.20	0.91	40	2.52
45	2.84	0.28	**5" (127 mm)**	50.0	23.2	8.20	2.76	1.16	45	2.84
50	3.15	0.34	0.11	60.0	28.4	9.90	3.32	1.38	50	3.15
60	3.79	0.47	0.16	85.0	39.6	13.9	4.65	1.92	60	3.79
70	4.42	0.63	0.21	113.0	53.0	18.4	6.20	2.57	70	4.42
75	4.73	0.72	0.24	129.0	60.0	20.9	7.05	2.93	75	4.73
80	5.05	0.81	0.27	145.0	68.0	23.7	7.90	3.28	80	5.05
90	5.68	1.00	0.34	**6" (152.4 mm)**	84.0	29.4	9.80	4.08	90	5.68
100	6.31	1.22	0.41	0.17	102.0	35.8	12.0	4.96	100	6.31
125	7.89	1.85	0.63	0.26	**7" (177.8 mm)**	54.0	17.6	7.55	125	7.89
150	9.46	2.60	0.87	0.36	0.17	76.0	25.7	10.5	150	9.46
175	11.05	3.44	1.16	0.48	0.22	**8" (203.2 mm)**	34.0	14.1	175	11.05
200	12.62	4.40	1.48	0.61	0.28	0.15	43.1	17.8	200	12.62
225	14.20	5.45	1.85	0.77	0.35	0.19	54.3	22.3	225	14.20
250	15.77	6.70	2.25	0.94	0.43	0.24	65.5	27.1	250	15.77
275	17.35	7.95	2.70	1.10	0.51	0.27	**9" (228.6 mm)**	32.3	275	17.35
300	18.93	9.30	3.14	1.30	0.60	0.32	0.18	38.0	300	18.93
325	20.5	10.8	3.65	1.51	0.68	0.37	0.21	44.1	325	20.5
350	22.08	12.4	4.19	1.70	0.77	0.43	0.24	50.5	350	22.08
375	23.66	14.2	4.80	1.95	0.89	0.48	0.28	**10" (254 mm)**	375	23.66
400	25.24	16.0	5.40	2.20	1.01	0.55	0.31	0.19	400	25.24
425	26.81	17.9	6.10	2.47	1.14	0.61	0.35	0.21	425	26.81
450	28.39	19.8	6.70	2.74	1.26	0.68	0.38	0.23	450	28.39
475	29.97		7.40	2.82	1.46	0.75	0.42	0.26	475	29.97
500	31.55		8.10	2.90	1.54	0.82	0.46	0.28	500	31.55
750	47.32			7.09	3.23	1.76	0.98	0.59	750	47.32
1000	63.09			12.0	5.59	2.97	1.67	1.23	1000	63.09
1250	78.86				8.39	4.48	2.55	1.51	1250	78.86
1500	94.64				11.7	6.24	3.52	2.13	1500	94.64
1750	110.41					7.45	4.70	2.80	1750	110.41
2000	126.18					10.71	6.02	3.59	2000	126.18

	Flow Restriction of Fittings Expressed as Equivalent Feet of Straight Pipe										
Size of Fitting	**2"**	**2-1/2"**	**3"**	**4"**	**5"**	**6"**	**8"**	**10"**	**12"**	**14"**	**16"**
90 Ell	5.5	6.5	8	11	14	16	21	26	32	37	42
46 Ell	2.5	3	3.8	5	6.3	7.5	10	13	15	17	19
Long Sweep Ell	3.5	4.2	5.2	7	9	11	14	17	20	24	27
Close Return Bend	13	15	18	24	31	37	51	61	74	85	100
Tee — Straight Run	3.5	4.2	5.2	7	9	11	14	17	20	24	27
Tee — Side Inlet or Outlet	12	14	17	22	27	33	43	53	68	78	88
Globe Valve Open	55	67	82	110	140						
Angle Valve Open	27	33	41	53	70						
Gate Valve Fully Open	1.2	1.4	1.7	2.3	2.9	3.5	4.5	5.8	6.8	8	9
Gate Valve Half Open	27	33	41	53	70	100	130	160	200	230	260
Check Valve	19	23	32	43	53						

UNITS OF PRESSURE AND HEAD

UNIT	mm Hg (0° C)	in Hg (0° C)	in H_2O (39° F)	ft H_2O (39° F)
mm Hg	1	0.039 37	0.535 25	0.0446
in Hg	25.4	1	13.5954	1.132 96
in H_2O	1.868 27	0.073 55	1	0.083 33
ft H_2O	22.4193	0.882 65	12	1
psi	51.7151	2.036 03	27.6807	2.306 73
kg/cm^2	735.561	28.9591	393.712	32.8094
bar	750.064	29.5301	401.474	33.4562
atm	760	29.9213	406.794	33.8995
kPa	7.500 64	0.295 30	4.014 74	0.334 56

UNIT	psi	kg/cm^2	bar	atmospheres	kPa
mm Hg	0.019 34	0.001 36	0.001 33	0.001 32	0.133 32
in Hg	0.491 15	0.034 53	0.033 86	0.033 42	3.386 38
in H_2O	0.036 13	0.002 54	0.002 49	0.002 46	0.249 08
ft H_2O	0.433 51	0.030 48	0.029 89	0.029 50	2.988 98
psi	1	0.070 31	0.068 95	0.068 05	6.894 76
kg/cm^2	14.2233	1	0.980 67	0.967 84	98.0665
bar	14.5037	1.019 72	1	0.986 92	100
atm	14.6959	1.033 23	1.013 25	1	101.325
kPa	0.145 04	0.010 09	0.010 00	0.009 87	1

UNITS OF FLOW

Cubic foot per second, also written second-foot, is the unit of flow in the English system used to express rate of flow in large pumps, ditches, and canals. Flow in pipe lines, from pumps and wells is commonly measured in gallons per minute.

Rates of water consumption and measurement of municipal water supply are ordinarily made in million gallons per day. The Miner's Inch is still used in some localities for irrigation and hydraulic mining, but is not suitable for general use.

UNITS	U.S. GALLONS PER MINUTE	MILLION U.S. GAL. PER DAY	CUBIC FEET PER SECOND	CUBIC METERS PER HOUR	LITER PER SECOND
1 U.S. Gallon per Minute (U.S. G.P.M.)	1	.001440	.00223	.2270	.0631
1 Million U.S. Gal. per Day (M.G.D.)	694.5	1	1.547	157.73	43.8
1 Cubic Foot per Second	448.8	.646	1	101.9	28.32
1 Cubic Meter per Hour	4.403	.00634	.00981	1	.2778
1 Liter per Second	15.85	.0228	.0353	3.60	1

GENERATOR RATING

3 PHASE AMPERES — 80% POWER FACTOR

kV•A	kW	208V	220V	240V	380V	400V	440V	450V	480V	600V	2400V	3300V	4160V
6.3	5	17.5	16.5	15.2	9.6	9.1	8.3	8.1	7.6	6.1			
9.4	7.5	26.1	24.7	22.6	14.3	13.6	12.3	12	11.3	9.1			
12.5	10	34.7	33	30.1	19.2	18.2	16.6	16.2	15.1	12			
18.7	15	52	49.5	45	28.8	27.3	24.9	24.4	22.5	18			
25	20	69.5	66	60.2	38.4	36.4	33.2	32.4	30.1	24	6	4.4	3.5
31.3	25	87	82.5	75.5	48	45.5	41.5	40.5	37.8	30	7.5	5.5	4.4
37.5	30	104	99	90.3	57.6	54.6	49.8	48.7	45.2	36	9.1	6.6	5.2
50	40	139	132	120	77	73	66.5	65	60	48	12.1	8.8	7
62.5	50	173	165	152	96	91	83	81	76	61	15.1	10.9	8.7
75	60	208	198	181	115	109	99.6	97.5	91	72	18.1	13.1	10.5
93.8	75	261	247	226	143	136	123	120	113	90	22.6	16.4	13
100	80	278	264	240	154	146	133	130	120	96	24.1	17.6	13.9
125	100	347	330	301	192	182	166	162	150	120	30	21.8	17.5
156	125	433	413	375	240	228	208	204	188	150	38	27.3	22
187	150	520	495	450	288	273	249	244	225	180	45	33	26
219	175	608	577	527	335	318	289	283	264	211	53	38	31
250	200	694	660	601	384	364	332	324	301	241	60	44	35
312	250	866	825	751	480	455	415	405	376	300	75	55	43
375	300	1040	990	903	576	546	498	487	451	361	90	66	52
438	350	1220	1155	1053	672	637	581	568	527	422	105	77	61
500	400	1390	1320	1203	770	730	665	650	602	481	120	88	69
625	500	1735	1650	1504	960	910	830	810	752	602	150	109	87
750	600	2080	1980	1803	1150	1090	996	975	902	721	180	131	104
875	700	2430	2310	2104	1344	1274	1162	1136	1052	842	210	153	121
1000	800	2780	2640	2405	1540	1460	1330	1300	1203	962	241	176	139
1125	900	3120	2970	2709	1730	1640	1495	1460	1354	1082	271	197	156
1250	1000	3470	3300	3009	1920	1820	1660	1620	1504	1202	301	218	174
1563	1250	4350	4130	3765	2400	2280	2080	2040	1885	1503	376	273	218
1875	1500	5205	4950	4520	2880	2730	2490	2440	2260	1805	452	327	261
2188	1750			5280	3350	3180	2890	2830	2640	2106	528	380	304
2500	2000			6020	3840	3640	3320	3240	3015	2405	602	436	348
2812	2250			6780	4320	4095	3735	3645	3400	2710	678	491	392
3125	2500			7520	4800	4560	4160	4080	3765	3005	752	546	435
3750	3000			9040	5760	5460	4980	4880	4525	3610	904	654	522
4375	3500			10550	6700	6360	5780	5660	5285	4220	1055	760	610
5000	4000			12040	7680	7280	6640	6480	6035	4810	1204	872	695

POWER FACTOR OF TYPICAL AC LOADS

UNITY (OR NEAR UNITY) POWER FACTOR

Load	Approximate Power Factor
Incandescent Lamps (Power factor of lamp circuits operating off step-down transformers will be somewhat below unity.)	1.0
Fluorescent Lamps (With built-in capacitor)	0.95 to 0.97
Resistor Heating Apparatus	1.0
Synchronous Motors (Operate at leading power factor at part loads; also built for leading power factor operation.)	1.0
Rotary Converters	1.0

LAGGING POWER FACTOR

Load	Approximate Power Factor
Induction Motors (Rated load and speed.)	
Split Phase Below 1 hp	0.55 to 0.75
Split Phase, 1 hp to 10 hp	0.75 to 0.85
Polyphase, Squirrel Cage	
High Speed, 1 hp to 10 hp	0.75 to 0.90
High Speed, 10 hp and Larger	0.85 to 0.92
Low Speed	0.70 to 0.85
Wound Rotor	0.80 to 0.90
Groups of Induction Motors	0.50 to 0.90
Welders	
Motor Generator-Type	0.50 to 0.60
Transformer-Type	0.50 to 0.70
Arc Furnaces	0.80 to 0.90
Induction Furnaces	0.60 to 0.70

LEADING POWER FACTOR

Load

Synchronous Motors
(Are designed in standard ratings at unity, 0.9 and 0.8 leading power factor.)

Synchronous Condensers
(Nearly zero leading power factor. Output practically all leading reactive kV•A.)

Capacitors
(Zero leading power factor. Output practically all leading reactive kV•A.)

THREE-PHASE AC MOTORS — 80% POWER FACTOR
FULL-LOAD CURRENT IN AMPERES
INDUCTION-TYPE SQUIRREL CAGE AND WOUND ROTOR

Horsepower	110 V	208 V	220 V	440 V	550 V
1/2	4	2.1	2	1	.8
3/4	5.6	3	2.8	1.4	1.1
1	7	3.7	3.5	1.8	1.4
1-1/2	10	5.3	5	2.5	2
2	13	6.9	6.5	3.3	2.6
3		9.5	9	4.5	4
5		16	15	7.5	6
7-1/2		23	22	11	9
10		29	27	14	11
15		43	40	20	16
20		55	52	26	21
25		68	64	32	26
30		83	78	39	31
40		110	104	52	41
50		133	125	63	50
60		159	150	75	60
75		198	185	93	74
100		262	246	123	98
125		330	310	155	124
150		380	360	180	144
200		510	480	240	192
250		697	657	328	262
300		837	790	394.5	315
350		976	922	461	368
400		1114	1051	526	421
450		1254	1192	592	473
500		1393	1317	657	526
600		1672	1578	789	632
700		1950	1842	921	737
800		2220	2103	1051	842
900		2504	2365	1194	947
1000		2789	2639	1316	1050

SINGLE-PHASE AC MOTORS
FULL-LOAD CURRENT IN AMPERES

Horsepower	kW	100 V	200 V	208 V	230 V	380 V	400 V	415 V
1/2	0.45	4	2.2	2.1	1.9	1.1	1.1	1.1
3/4	0.68	5.6	3.1	3	2.7	1.6	1.6	1.5
1	0.9	7	3.8	3.7	3.3	2	1.9	1.9
1-1/2	1.5	10	5.5	5.3	4.8	2.9	2.8	2.7
2	1.8	13	7.2	6.9	6.2	3.8	3.6	3.5
3	2.6		9.9	9.5	8.6	5.2	5	4.8
5	4.4		16.6	16	14.5	8.8	8.3	8
7-1/2	6.6		24	23	20.8	12.6	12	11.5
10	8.8		30	29	26	16	15	14.5

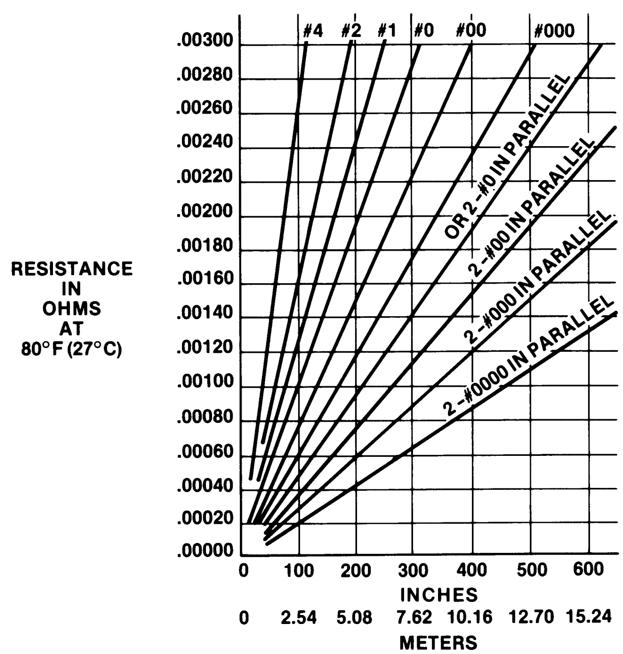
DIRECT CURRENT MOTORS
FULL-LOAD CURRENT IN AMPERES

Horsepower	115 V	230 V	550 V
1/4	3	1.5	
1/3	3.8	1.9	
1/2	5.4	2.7	
3/4	7.4	3.7	1.6
1	9.6	4.8	2
1 1/2	13.2	6.6	2.7
2	17	8.5	3.6
3	25	12.5	5.2
5	40	20	8.3
7-1/2	58	29	12
10	76	38	16
15	112	56	23
20	148	74	31
25	184	92	38
30	220	110	46
40	292	146	61
50	360	180	75
60	430	215	90
75	536	268	111
100		355	148
125		443	184
150		534	220
200		712	295

APPROXIMATE EFFICIENCIES — SQUIRREL CAGE INDUCTION MOTORS

Horsepower	Full-Load kW Required	Full-Load Efficiency
1/2	0.6	68%
3/4	0.8	71
1	1	75
1-1/2	1.5	78
2	1.9	80
3	2.7	82
5	4.5	83
7-1/2	6.7	83
10	8.8	85
15	13	86
20	16.8	89
25	21	89
30	24.9	90
40	33.2	90
50	41.5	90
60	49.2	91
75	61.5	91
100	81.2	92
125	101.5	92
150	122	92
200	162.5	92
250	203	92
300	243	92
350	281	93
400	321	93
450	362	93
500	401	93
600	482	93

Cable Length and Resistance



CONVERSION TABLES

UNITS OF POWER

UNIT	hp	ft lb/min	W	kW	metric hp	Btu/min
hp	1	33 000	745.70	0.745 70	1.014	42.456
ft lb/min	—	1	0.0226	—	—	0.001 28
W	0.001 34	44.25	1	0.001	0.001 36	0.056 87
kW	1.341 02	44 250	1000	1	1.359 62	56.8690
metric hp	0.986 32	32 550	735.498	0.735 49	1	41.8271
Btu/min	0.023 58	778.2	17.5843	0.017 58	0.023 91	1

MISCELLANEOUS EQUIVALENTS

1 Btu = Heat required to raise 1 lb water 1° F = 778 ft lb = 0.000 293 kW-h = 0.252 kg-cal = 0.0039 hp-h

1 hp = 746 watts = 33 000 ft lb/min = 550 ft lb/sec = 42.45 Btu/min = 1.014 metric hp

1 kW = 1000 watts = 1.341 hp = 3412 Btu/h

1 hp-h = 2544 Btu

BRAKE MEAN EFFECTIVE PRESSURE:

$$\text{BMEP psi (4-cycle)} = \frac{792{,}000 \times \text{hp}}{\text{RPM} \times \text{Displacement}}$$

$$\text{BMEP psi (2-cycle)} = \frac{396{,}000 \times \text{hp}}{\text{RPM} \times \text{Displacement}}$$

$$\text{BMEP psi} = \frac{150.8 \times \text{Torque}}{\text{Displacement}}$$

TORQUE:

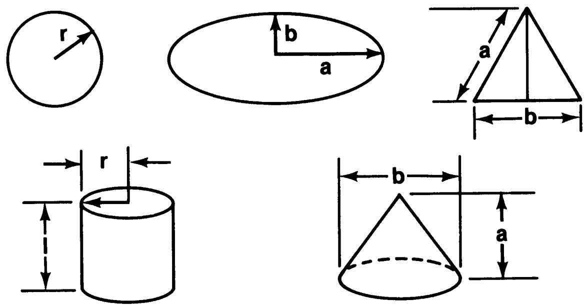
$$\text{T (lb ft)} = \frac{\text{Displacement} \times \text{BMEP}}{150.8}$$

$$\text{T (lb ft)} = \frac{33000 \times \text{hp}}{2\pi \times \text{RPM}} = \frac{5252 \times \text{hp}}{\text{RPM}}$$

BAROMETRIC PRESSURES AND BOILING POINTS OF WATER AT VARIOUS ALTITUDES

	BAROMETRIC PRESSURE			
ALTITUDE	INCHES MERCURY	LB. PER SQUARE INCH	FEET WATER	POINT WATER BOILING
Sea Level	29.92 In.	14.69 P.S.I.	33.95 Ft.	212° F
1000 Ft.	28.86 In.	14.16 P.S.I.	32.60 Ft.	210.1° F
2000 Ft.	27.82 In.	13.66 P.S.I.	31.42 Ft.	208.3° F
3000 Ft.	26.81 In.	13.16 P.S.I.	30.28 Ft.	206.5° F
4000 Ft.	25.84 In.	12.68 P.S.I.	29.20 Ft.	204.6° F
5000 Ft.	24.89 In.	12.22 P.S.I.	28.10 Ft.	202.8° F
6000 Ft.	23.98 In.	11.77 P.S.I.	27.08 Ft.	201.0° F
7000 Ft.	23.09 In.	11.33 P.S.I.	26.08 Ft.	199.3° F
8000 Ft.	22.22 In.	10.91 P.S.I.	25.10 Ft.	197.4° F
9000 Ft.	21.38 In.	10.50 P.S.I.	24.15 Ft.	195.7° F
10000 Ft.	20.58 In.	10.10 P.S.I.	23.25 Ft.	194.0° F
11000 Ft.	19.75 In.	9.71 P.S.I.	22.30 Ft.	192.0° F
12000 Ft.	19.03 In.	9.34 P.S.I.	21.48 Ft.	190.5° F
13000 Ft.	18.29 In.	8.97 P.S.I.	20.65 Ft.	188.8° F
14000 Ft.	17.57 In.	8.62 P.S.I.	19.84 Ft.	187.1° F
15000 Ft.	16.88 In.	8.28 P.S.I.	18.07 Ft.	185.4° F

GEOMETRIC FORMULAS

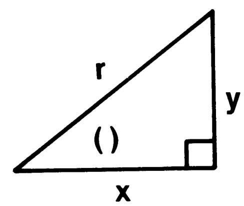


Circumference: Circle $2\pi r$

Area:	Circle	πr^2
	Ellipse	πab
	Shpere	$4\pi r^2$
	Cylinder	$2\pi r\,(r + l)$
	Triangle	$\frac{1}{2}ab$
Volume:	Ellipsoid of revolution	$4/3\pi b^2 a$
	Sphere	$4/3\pi r^3$
	Cylinder	$\pi r^2 l$
	Cone	$\frac{\pi b^2 a}{12}$
Analytical:	Circle	$\frac{x^2}{r^2} + \frac{y^2}{r^2} = 1$
	Ellipse	$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1$
	Hyperbola	$\frac{x^2}{a^2} - \frac{y^2}{b^2} = 1$
	Parabola	$y^2 = \pm 2px$
	Line	$y = mx + b$

MATHEMATICAL EXPRESSIONS

Trigonmetric Relations



$\sin O = \frac{y}{r}$

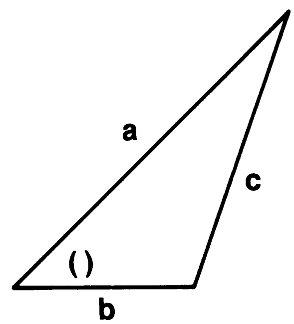
$\cos O = \frac{x}{r}$

$\tan O = \frac{y}{x}$

$\sin^2 O + \cos^2 O = 1$

$e^{in} = \cos O + i \sin O \quad i = \sqrt{-4}$

Law of Cosines



$a^2 + b^2 - 2ab \cos O = c^2$

RULES OF THUMB

Laws of Exponents

$a^x \times a^y = a^{x-y} \qquad \frac{1}{a^x} = a^{-x}$

$(ab)^x = a^x \times b^x \qquad \frac{a^x}{a^y} = a^{x-y}$

$(a^x)^y = a^{xy} \qquad a^o = 1$

Laws of Logarithms

$\ln (y^x) = x \ln y$

$\ln (ab) = \ln a + \ln b$

$\ln \left(\frac{a}{b}\right) = \ln a - \ln b$

RULES OF THUMB

HEAT REJECTION:

% of Fuel Energy Consumed

BHP	30%
Jacket Water	30%
Exhaust	30%
Radiation	10%

Jacket Water

Turbocharged Engines

Btu/min = 42 × BHP

Naturally-Aspirated, Roots Blown and Spark-Ignited Engines

Btu/min = 45 × BHP

Oil Cooler	Btu/min = 5 × BHP
Watercooled Manifold	Btu/min = 7 × BHP
Torque Converter	Btu/min = 42.4 × BHP input × $\frac{(100 - \text{conv. eff.})}{100}$

FUEL CONSUMPTION — BHP:

BHP = GPH fuel × 15	Diesel	1/15 gal. per BHP-h
BHP = GPH fuel × 9.5	Gasoline	1/10 gal. per BHP-h
BHP = cu ft/h fuel × 1/8	Natural Gas*	7 to 8 cu ft/BHP-h
kW = GPH fuel × 10	Diesel	1/10 gal/kW-h

*100 Btu gas.

GAS COMPRESSOR:

BHP = 22 RcVS

Where: Rc = Stage Compression Ratio

V = Million cu ft/day

S = Number of Stages

COOLING:

Heat Exchanger Flow Rate
Raw water to jacket water 1:1 to 2:1
Submerged Pipe Cooling
½ sq. ft. surface area per HP
With 85° F flowing water

ELECTRICITY:

Generator Capacity Required
Motors:
1 kW per nameplate hp (motor running cool or warm to touch)
1¼ kW per nameplate hp (motor running hot to touch)
Horsepower Requirements

1½ BHP per kW of load or $\dfrac{\text{kW}}{0.746 \times \text{Gen. Eff.}}$

ELECTRIC SETS:

Motor Starting Requirements
Inrush kV•A (Code F motor) = 5.5 × BHP
Inrush Current (Code F motor) = 6.2 × Full load rated current
1 kV•A per HP at full load
Generator full load rated current capacity

Voltage	Rated Current
120	6.01 × kW
208	3.47 × kW
240	3.01 × kW
480	1.50 × kW
2400	0.30 × kW
4160	0.17 × kW

Generator Cooling Requirements
Air Flow = 20 CFM per kW
Circuit Breaker Trip Selection
1.15 to 1.25 × full load generator amp rating
Single Phase Rating of 3-Phase Generator
60% of 3-phase rating
Generator Temperature Rise
Increase 1° C for each 330 feet above 3300 feet

ON SITE POWER REQUIREMENTS:

Based on 100,000 sq. ft. of office bldg., etc., and 40° N. latitudes
Electric Requirements:
600 kW continuous load
(Air conditioning is absorption)
Use three - 300 kW units
(2 prime and 1 standby)
Air Conditioning Compressor:
400 tons prime load
Use two - 200 hp engines
(No standby)

REFRIGERATION:

One ton refrigeration = 200 Btu/min = 12,000 Btu/h
One boiler HP = 33,475 Btu/h
One ton compressor rating = One Engine hp
Auxiliary air conditioning equipment requires
¼ hp per ton of compressor rating

Ice Plant:
Complete power requires 4-5 hp per daily ton capacity

AIR COMPRESSORS:

hp = ¼ × cu ft per minute at 100 psi
Increase BHP 10% for 125 psi
Decrease BHP 10% for 80 psi

CONVEYORS: 15 to 20° Incline.

$$\text{BHP} = \frac{\text{Vertical lift in feet} \times \text{tons per hour}}{500}$$

PUMPS:

$$\text{Deep Well BHP} = \frac{\text{Feet of lift per 1000 GPM}}{3}$$

Pipe Line BHP = Barrels per hour × psi × 0.00053

$$\text{Any Liquid BHP} = \frac{\text{GPM} \times \text{lb/gal (Liquid)} \times \text{feet of head}}{33{,}000 \times \text{pump efficiency*}}$$

*Efficiency: Centrigugal

Single impeller, double suction	65-80%
Single impeller, side suction	55-75%
Deep well turbine	65-80%
Reciprocating	75%

OILFIELD DRILLING:

Hoisting

$$\text{BHP} = \frac{\text{Weight} \times \text{FPM (assume 100 is unknown)}}{33{,}000 \times 0.85 \text{ (eff.)}}$$

Mud Pumps

$$\text{BHP} = \frac{\text{GPM} \times \text{lb/gal} \times \text{(feet of head)}}{33{,}000 \times \text{pump efficiency (see pumps)}}$$

Dry Table

Depth in feet	BHP Required
0 - 4000	75
4000 - 8000	100
8000 - 12000	150
12000 - 16000	200

SAWMILL:

1½ BHP per inch of saw diameter at 500 RPM
Increase or decrease in proportion to RPM
Swing Cut-Off Saw
24-inch 3 BHP
36-inch 7½ BHP
42-inch 10 BHP
Table Trimmer 7½ to 10 BHP
Blower Fan, 12-foot sawdust 3 to 5 BHP
Planer Mill 2 to 4 BHP per 100 board feet per hour
24 to 30-inch planers 15 to 25 BHP
Edgers
2 saws 12 to 15 BHP
3 saws 15 to 25 BHP
Slab Saw 10 BHP
Jack Ladder 10 BHP
Approximate fuel consumption
Softwood 1 gal. per 1000 board feet
Hardwood 1 gal. per 750 board feet

TORQUE CONVERTERS:

Peak output shaft horsepower:
Normally 80% of input horsepower for either single or three-stage converter.

Output shaft speed at peak output horsepower:
Single-stage — 0.7 to 0.85 engine full load speed
Three-stage — 0.5 to 0.6 engine full load speed

Torque multiplication at or near stall:
Single-stage — 2.2 to 3.4 times engine torque
Three-stage — 3.6 to 5.4 times engine torque

CATERPILLAR

LEBX4457
Supersedes LEBX9212

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