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4E

INTERNAL COMBUSTION PISTON ENGINES

by

Charles L. Segaser

ICES

INTEGRATED COMMUNITY ENERGY SYSTEMS

TECHNOLOGY EVALUATIONS

Prepared by:

Oak Ridge National Laboratory
Operated by Union Carbide Corporation
for the U. S. Energy Research and Development Administration

Prepared for:

Argonne National Laboratory
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This is your copy of an Integrated Community Energy Systems Technology Evaluation. This report is one of a series on energy-related components and subsystems which can be used in Community Energy Systems. You will note that the report includes removable yellow-colored Summary Sheets which have been prepared for insertion in a notebook. If you wish to obtain a special Technology Evaluations Summary Notebook please return the enclosed card or write

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9700 S. Cass Ave.
Argonne, Illinois 60439

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INTERNAL COMBUSTION PISTON ENGINES

by

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July 1977

Prepared for

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9700 South Cass Avenue
Argonne, Illinois 60439

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OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830
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The four E's of the cover logo embody the goals of the Community Systems Program of the Energy Research and Development Administration, ERDA, namely:

- to conserve *Energy*;
- to preserve the *Environment*; and
- to achieve *Economy*
- in the design and operation of human settlements (*Ekistics*).

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FOREWORD

The Community Systems Program of the Division of Buildings and Community Systems, Office of Energy Conservation, of the United States Energy Research and Development Administration (ERDA), is concerned with conserving energy and scarce fuels through new methods of satisfying the energy needs of American Communities. These programs are designed to develop innovative ways of combining current, emerging, and advanced technologies into Integrated Community Energy Systems (ICES) that could furnish any, or all, of the energy using services of a community. The key goals of the Community System Program then, are to identify, evaluate, develop, demonstrate, and deploy energy systems and community designs that will optimally meet the needs of various communities.

The overall Community Systems effort is divided into three main areas. They are: (a) Integrated Systems, (b) Community Design, and (c) Commercialization. The *Integrated Systems* work is intended to develop the technology component and subsystem data base, system analysis methodology, and evaluations of various system conceptual designs which will help those interested in applying integrated systems to communities. Also included in this program is an active participation in demonstrations of ICES. The *Community Design* effort is designed to develop concepts, tools, and methodologies that relate urban form and energy utilization. This may then be used to optimize the design and operation of community energy systems. *Commercialization* activities will provide data and develop strategies to accelerate the acceptance and implementation of community energy systems and energy-conserving community designs.

This report, prepared by Oak Ridge National Laboratory, is part of a series of Technology Evaluations of the performance and costs of components and subsystems which may be included in community energy systems and is part of the Integrated Systems effort. The reports are intended to provide sufficient data on current, emerging and advanced technologies so that they may be used by consulting engineers, architect/engineers, planners, developers, and others in the development of conceptual designs for community energy systems. Further, sufficient detail is provided so that calculational models of each component may be devised for use in computer codes for the design of Integrated Systems. Another task of the Technology Evaluation activity is to devise calculational models which will provide part load performance and costs of components suitable for use as subroutines in the computer codes being developed to analyze community energy systems. These will be published as supplements to the main Technology Evaluation reports.

It should be noted that an extensive data base already exists in technology evaluation studies completed by Oak Ridge National Laboratory (ORNL) for the Modular Integrated Utility System (MIUS) Program sponsored by the Department of Housing and Urban Development (HUD). These studies, however, were limited in that they were: (a) designed to characterize mainly off-the-shelf technologies up to 1973, (b) size limited to meet community limitations, (c) not designed to augment the development of computer subroutines, (d) intended for use as general information for city officials and keyed to residential communities, and (e) designed specifically for HUD-MIUS needs. The present documents are founded on the ORNL data base but are more technically oriented and are designed to be upgraded periodically to reflect changes in current, emerging, and advanced technologies. Further, they will address the complete range of component sizes and their application to residential, commercial, light industrial, and institutional communities. The overall intent of these documents, however, is not to be a complete documentation of a given technology but will provide sufficient data for conceptual design application by a technically knowledgeable individual.

Data presentation is essentially in two forms. The main report includes a detailed description of the part load performance, capital, operating and maintenance costs, availability, sizes, environmental effects, material and energy balances, and reliability of each component along with appropriate reference material for further study. Also included are concise data sheets which may be removed for filing in a notebook which will be supplied to interested individuals and organizations. The data sheets are colored and are perforated for ease of removal. Thus, the data sheets can be upgraded periodically while the report itself will be updated much less frequently.

Each document was reviewed by several individuals from industry, research and development, utility, and consulting engineering organizations and the resulting reports will, hopefully, be of use to those individuals involved in community energy systems.

ABSTRACT

Current worldwide production of internal combustion piston engines includes many diversified types of designs and a very broad range of sizes. Engine sizes range from a few horsepower in small mobile units to over 40,000 brake horsepower in large stationary and marine units. The key characteristics of internal combustion piston engines considered appropriate for use as prime movers in Integrated Community Energy Systems (ICES) are evaluated in this report. The categories of engines considered include spark-ignition gas engines, compression-ignition oil (diesel) engines, and dual-fuel engines.

The engines are evaluated with respect to full-load and part-load performance characteristics, reliability, environmental concerns, estimated 1976 cost data, and current and future status of development. The largest internal combustion piston engines manufactured in the United States range up to 13,540 rated brake horsepower. Future development efforts are anticipated to result in a 20 to 25% increase in brake horsepower without increase in or loss of weight, economy, reliability, or life expectancy, predicated on a simple extension of current development trends.

TECHNOLOGY EVALUATION SUMMARY SHEET OF

INTERNAL COMBUSTION PISTON ENGINES

By: Charles L. Segaser, ORNL July, 1977

ICES INTEGRATED COMMUNITY ENERGY SYSTEMS

1. INTRODUCTION

Current worldwide production of internal combustion piston engines includes many diversified types of designs and a broad range of sizes. A total of 119 manufacturers of Diesel, dual-fuel, and spark-ignition gas engines are listed in the 1976 volume of the *Diesel and Gas Turbine Worldwide Catalog*. Engine sizes range from a few horsepower in small mobile units to over 40,000 brake horsepower (Bph) in large stationary and marine units. The largest internal combustion (IC) engines manufactured in the United States range up to 13,540 rated Bhp. Based on a simple extension of current trends, future development efforts are expected to result in a 20% to 25% increase in Bhp without increase in weight or losses in economy, reliability, or life expectancy.

IC piston engines have had wide acceptance as drive units for electrical generators, reciprocating compressors, centrifugal compressors, and various types of pumps. The 1975 Annual Plant Design Report published by *Power* lists 62 internal combustion engine projects utilizing 128 IC engines of various types. Of these engine types, about 70% were full Diesel, 20% were spark-ignition gas, and the remaining 10% were dual-fuel. Fuel availability and fuel cost have apparently greatly influenced the design and selection of plant types.

About 10% of 642 engines characterized in a 1973 annual plant design report by *Power* were used in total energy plants (0 to 1499 Bhp range), approximately another 10% were used for pump and compressor drives (mostly in the 1,500 to 2,499 Bhp range), and the remaining 80% were used for continuous, peaking, and standby power service. The availability of the large engine sizes (5,000 Bhp, and up) makes practical their potential application in integrated community energy systems (ICES) up to 100 MW total capacity.

A generalized empirical equation to correlate part-load performance data of the engines is given by Eq. (DS-1).

$$Y = A + BX + CX^2 + DX^3, \qquad \text{(Eq. DS-1)}$$

where Y represents the value of a particular function, such as brake horsepower, jacket water heat rejection, etc. corresponding to input values of the independent variable X, which can be the percentage of rated load of the engine or other appropriate variable.

2 FUEL CONSUMPTION

Comparative fuel-rate curves for representative large internal combustion engines are shown in Fig. DS-1. Characteristically, the diesel engine has the flattest fuel-rate curve, with performance at partial loads

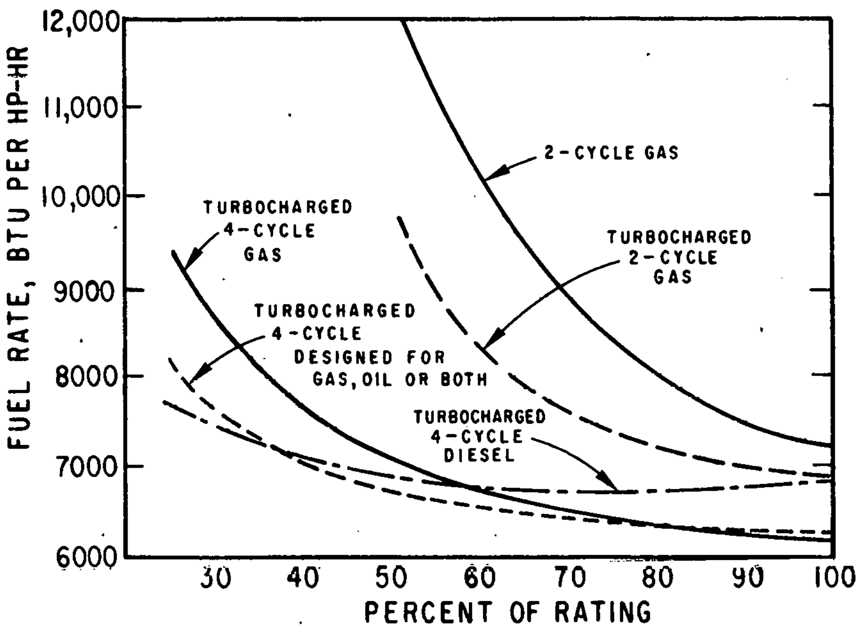

Fig. DS-1 Comparative Fuel-Rate Curves for Representative Internal Combustion Piston Engines

Source: Diesel Engineering Handbook (1976)[7] approaching that at full load. The 2-cycle spark-ignited gas engine has the poorest fuel-rate at full load and the fuel rate increases rapidly at loads below about 60%. In accordance with customary practice, the gas and dual-fuel engine fuel rates are based on the lower heating value of the fuel; whereas, the diesel-fuel rate is based on the higher heating value. Practically all high output IC engines are now turbocharged and intercooled to boost power output and improve efficiency. The performance curves for diesel engines with higher speeds (1200 rpm) are significantly different, with a fuel rate of about 8300 Btu/hr-hp at 80% rating compared to a fuel rate of about 6700 Btu/hr-hp for low speed engines (450 rpm) at 80% rating.

The values of the coefficients to use in the generalized Eq. (DS-1) to model fuel consumption in spark-ignited gas, dual-fuel, and diesel engines are given in Table DS-1.

Table DS-1 Generalized Equation Coefficients - Percent (Y) of Specific Fuel Consumption at Full-Load for Representative Large Gas, Dual-Fuel and Diesel Engines Vs Percent (X) of Rated Load ($25 \leq x \leq 100$.)

Engine Type	Coefficients			
	A	B	C	D
2-cycle gas	506	-10.9	0.098	-2.96×10^{-4}
Turbocharged 2-cycle gas	558	-15.1	0.168	-6.30×10^{-4}
Turbocharged 4-cycle gas	219	-3.74	0.041	-1.54×10^{-4}
Turbocharged 4-cycle gas-diesel	176	-2.51	0.028	-1.09×10^{-4}
Turbocharged 4-cycle diesel	142	-1.61	0.019	-6.94×10^{-5}

3 IC ENGINE HEAT BALANCE

Energy input in the fuel is dissipated as brake horsepower, heat rejected to the jacket cooling water, heat rejected to the lube oil, and heat rejected to the exhaust gas and lost by radiation. A representative part-load heat balance for a four-cycle turbo-supercharged low-speed diesel engine is shown in Fig. DS-2. Data for particular engines can be obtained from engine manufacturers.

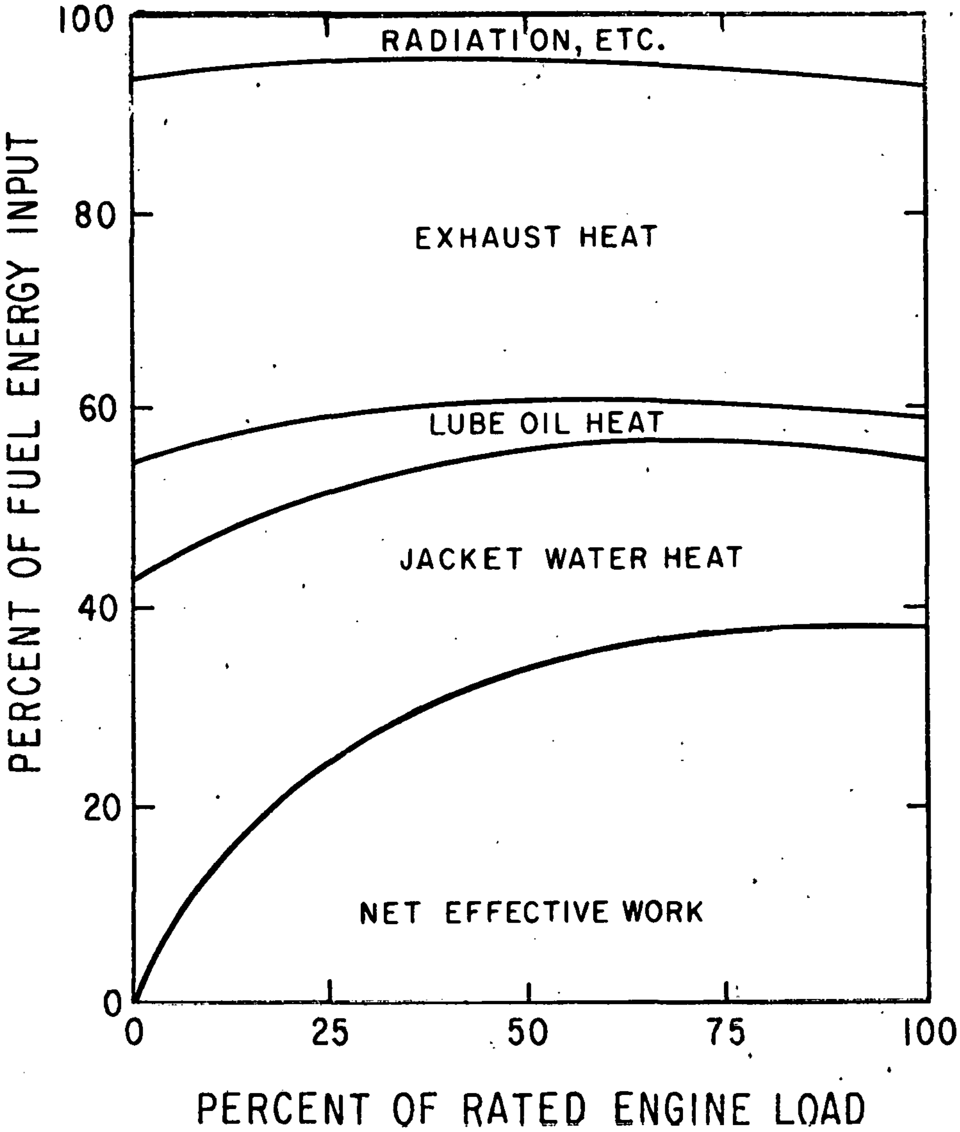

Fig. DS-2 Heat Balance - Turbocharged Low-Speed Compression Ignition Diesel Engine

The ranges of heat energy distribution a representative low-speed diesel engine at full load are given by the heat balance of Table DS-2

Table DS-2 Typical Diesel Engine Heat Balance at Full Rated Load

	General Range (%)	Average (%)
Brake horsepower	33 to 38	36
Exhaust and radiation	37 to 40	38
Jacket water	16 to 20	18
Lube Oil cooler	2 to 6	4
Aftercooler	2 to 6	4

Part load heat balance curves of the spark-ignition gas engine and the dual-fuel engine are similar, but the heat energy distribution varies with the type of engine. The values of the coefficients to use in the generalized empirical Eq. (DS-1) representing the components of the heat balance curves for various engine types as a function of part load are given in Table DS-3. The coefficients listed in Table DS-3 are all for 4-cycle turbo-supercharged engine types.

Table DS-3 Generalized Equation Coefficients — Distribution of Input Fuel Energy (Y, %) Vs Percentage of Rated Load (X) for 4-Cycle Turbo-Supercharged Engines

	Coefficients			
Function	A	B	C	D
A. Coefficients for representative diesel engine heat balance curves				
Brake Thermal Efficiency	0.0	1.449	-1.869×10^{-2}	8.000×10^{-5}
Jacket Water Heat	43.0	-0.886	1.114×10^{-2}	-4.907×10^{-5}
Lube Oil Heat	12.0	-0.194	1.143×10^{-3}	0.0
Exhaust Heat	39.0	-0.159	1.714×10^{-3}	-6.400×10^{-6}
B. Coefficients for representative gas and dual-fuel engines				
Brake Thermal Efficiency	0.0	1.267	-1.633×10^{-2}	6.867×10^{-5}
Jacket Water Heat	41.0	0.613	6.940×10^{-2}	-2.662×10^{-5}
Lube Oil Heat	12.0	0.342	4.489×10^{-3}	-2.026×10^{-5}
Exhaust Heat	32.0	0.232	3.438×10^{-3}	-1.563×10^{-5}

4 IC ENGINE RELIABILITY

A comprehensive study conducted for the U.S. Army reported an overall availability of about 96% for both internal combustion piston engines and gas turbine prime movers. Of the 4% average downtime observed, about 1% was attributed to forced outages and 3% to scheduled maintenance practices. The mean time observed between failures for piston engines was slightly over 500 hr, and the mean time to repair was about 2.5 hr. Most of the outages were for items such as failures in water hose connections, lube oil, cooling water piping, and ignition systems.

5 COST CONSIDERATIONS

The 1976 uninstalled capital cost of large (>5,000 Bhp) oil-fired diesel engines ranges from an estimated $714,000 for a 5,416 Bhp engine to about $1,526,000 for a 13,540 Bhp engine. These costs include all components normally considered as standard equipment with the engine. They do not include the price of an associated electric generator.

A power function that approximates the cost of large engines (from 5,000 Bhp to 13,540 Bhp) is given by Eq. DS-2.

$$C_D = C_B \times \left(\frac{Q_D}{Q_B} \right)^{0.829} \qquad \text{(Eq. DS-2)}$$

where:

C_D = cost, desired capacity, $

C_B = cost, base or nominal capacity, $714,000

Q_D = desired capacity, Bhp

Q_B = base or nominal capacity, 5,416 Bhp

An ASME 1974 report on diesel and gas engine power costs includes data on performance and production costs of 91 diesel, dual-fuel, and gas engine power plants located within the United States.

A plot of production costs as a function of engine rated Bhp from this report shows widely scattered data points, but there is a definite trend to higher production costs as the size of the engines decreases. Values of coefficients in the generalized empirical Eq. DS-1 to approximate average production costs (variable Y, mills/hp-year) as a function of engine size (variable X, hp) are as follows:

$A = 10.644$ $\qquad$ $C = 6.659 \times 10^{-7}$

$B = -4.031 \times 10^{-3}$ $\qquad$ $D = -3.870 \times 10^{-11}$

A plot of annual engine maintenance costs as a function of engine size also indicates a trend to higher costs as the size of the engines decreases. The following coefficients are appropriate for use in Eq. DS-1 to model engine maintenance cost (variable Y, \$/Bhp-year) as a function of engine size (variable X, Bhp):

$A = 4.963$ $\qquad$ $C = 3.297 \times 10^{-7}$

$B = -1.971 \times 10^{-3}$ $\qquad$ $D = -1.884 \times 10^{-11}$

TECHNOLOGY EVALUATION OF
INTERNAL COMBUSTION PISTON ENGINES

Prepared by Charles L. Segaser, ORNL

Date July, 1977

1 INTRODUCTION

The category of prime movers to which internal combustion (IC) piston engines belong includes compression-ignition oil (diesel) engines, spark-ignition (gas) engines, and dual-fuel engines. The IC piston engines — the most efficient of the currently available prime movers — have both good conversion efficiency for electricity generation (∿30 to 40%) and good overall thermal efficiency (∿80%) when maximum heat recovery from cooling water and exhaust gases is completed. Gas engines are well developed, commercially available items in general use for many applications, including onsite total energy systems. Current production of diesel engines covers many different engine designs and a wide range of sizes and applications. Some users who prefer the dual-fuel engine believe that adequate, advantageously priced gas will be available often enough to defray the additional cost of dual-fuel capability.

The 1975 Annual Plant Design Report[1] for oil and gas engines lists 62 IC engine facilities using 128 engine types. Of these engine types, about 70% were full diesel; 20% were spark-ignition gas; and the remaining 10% were dual fuel. Fuel availability and cost apparently have greatly influenced the selection and design of reciprocating engine plant types.

IC engines have had wide acceptance as drive units for electrical generators, reciprocating compressors, centrifugal compressors, and various types of pumps for space cooling and various industrial or process uses. Figure 1.1 illustrates the energy distribution from an IC engine drive unit that may be coupled to various types of loads as required. Heat recovery units recover waste heat from the jacket cooling water and exhaust gas for use in space and domestic hot water heating and as energy input to absorption chillers for space cooling.

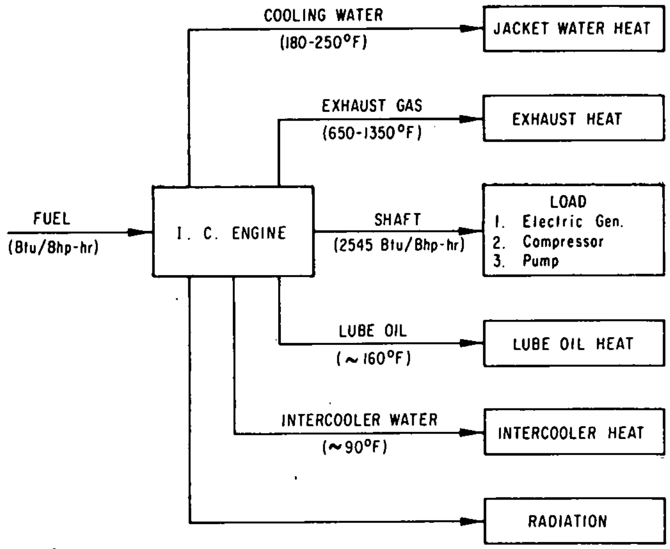

Fig. 1.1 Energy Distribution Diagram for an Internal Combustion Engine

Plant descriptions, capacity data, and engine-generator characteristics of 354 existing oil and gas engine electric power generating projects providing baseload, peaking, and standby services are tabulated in the annual plant design reports[1,2] of *Power* for November, 1975 and November, 1973, respectively. Engine capacity data from these reports are compared for the years 1968, 1973, and 1975 in Fig. 1.2. About 10% of the 642 engines characterized in the 1973 survey were used in total energy plants (0 to 1,499 hp range); another approximately 10% were used for pump and compressor drives (mostly in the 1,500 to 2,499 hp range); and the remaining 80% were used for continuous, peaking, and standby power service. The availability of the large engine sizes (5,000 hp and up) makes practical their potential application in integrated community energy systems (ICES) up to 100-Mw capacity.

Some 119 worldwide manufacturers of oil and gas engines are mentioned in Ref. 3. Table 1.1 lists the basic horsepower ranges of some diesel, dual-fuel, natural gas, and gasoline engines available from manufacturers with headquarters or affiliations in North America.

As shown in Table 1.1, engine speeds can range from about 450 rpm for large engine sizes (∿2000 to 13,540 bhp) up to 1800 rpm and greater for the smaller size units. In the *Advanced Coal-Using Community System (ACUCS)*,

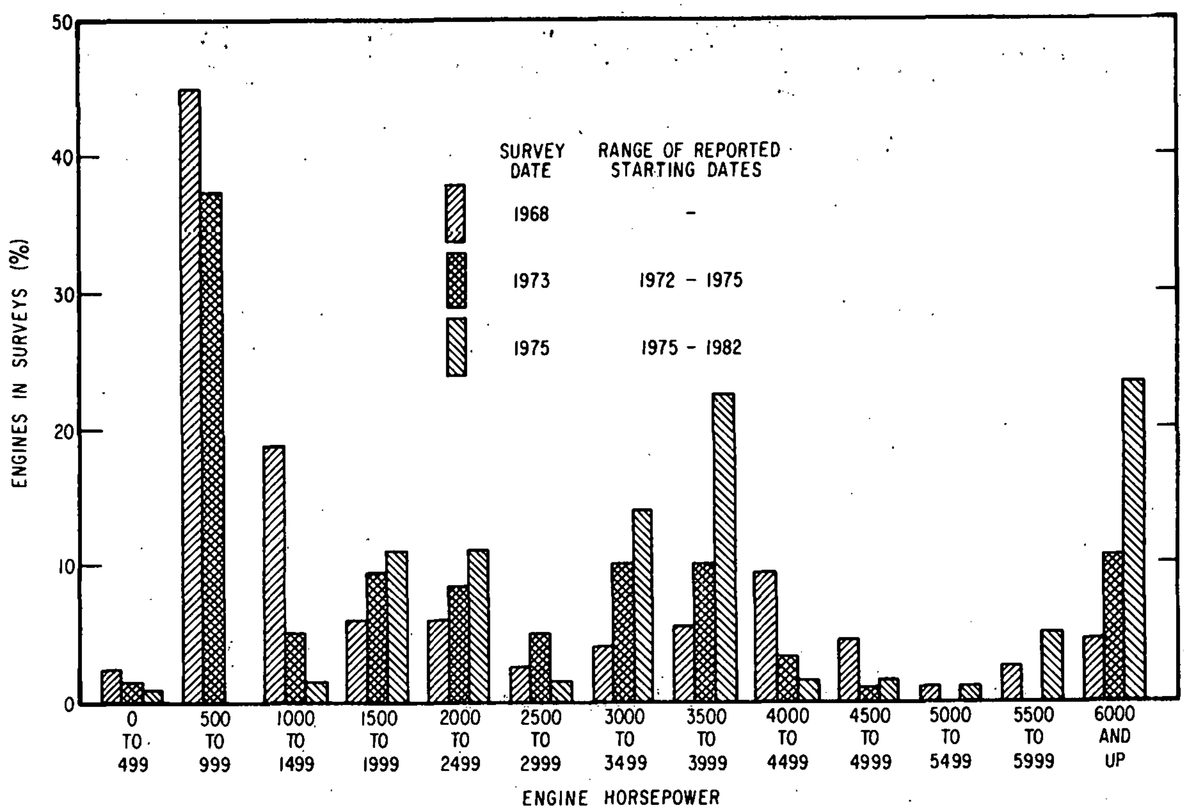

Fig. 1.2 Size Distribution of Oil and Gas Engine Installations Reported in 1968, 1973, and 1975 Surveys

Preliminary Draft Final Report - Task 1A,[14] it was pointed out that internal combustion engines can be characterized by their operating rpm value, and that performance characteristics are significantly different for each rpm range. Values of 1200 rpm for high-speed engines, 900 rpm for medium-speed engines, and 450 rpm for slow-speed engines were given as typical speed values. In general, it is shown in Ref. 14 that electrical efficiency (for conventional engine generator sets), total efficiency, output capacity, and per-unit-output investment costs increase as the rpm's decrease, and maintenance costs and per unit output fuel consumption decrease.

Table 1.1 North America-Based Manufacturers Offering Diesel Dual-Fuel, Natural Gas, and Gasoline Engines

Manufacturer	Diesel (hp)	Gas (Natural Gas and/or LPG or Gasoline)	Dual Fuel	Speed Range (rpm)
Alco Engines Division, White Industrial Power, Inc.	1,000- 4,500	-	-	900-1,200
Allis-Chalmers Corp., Engine Division	49- 900	-	-	1,200-2,600
Avco Lycoming Industrial Products Operations	4- 40	-	-	1,800-3,000
Case, J. I. Co., Component Sales	30- 240	-	-	1,500-2,200
Caterpillar Tractor Co., Industrial Division	35, 1,550	125- 930 hp	-	1,200-2,200
Colt Industries, Fairbanks Morse Engine Division	640-11,700	770- 2,114 kW	450- 6,445 kW	514- 900
Cummings Engine Co., Inc.	149- 1,200	-	-	1,800-3,300
De Laval Turbine Inc., Engine & Compressor Division, (Enterprise Engines)	1,667-13,540	2,000- 6,850 hp	1,667-13,540 hp	450- 630
Detroit Diesel Allison Division, General Motors Corp.	148- 1,600	-	-	1,500-2,800
Dresser Clark Division, Dresser Industries, Inc.	-	2,400-12,000 hp	-	300- 360
Electro-Motive Division, General Motors Corp.	800- 3,950	-	-	720- 900
Ford Motor Co., Industrial Engine Operations	33- 123	38- 180 hp	-	2,200-2,800
GEC Diesels Limited	10- 4,950	46- 550 hp	-	428-2,500
Ingersoll-Rand Co.	-	1,080- 3,500 hp	-	330- 550
ONAN Division, Onan Corp.	7.2-27.5	12.9-43.5 hp	-	1,800-3,600
Perkins Engines	25- 165	65- 80 hp	-	1,800-3,000
Stirling Engine Co., Inc.	20- 150	-	-	1,200-3,600
Stewart & Stevenson Services, Inc.	30- 4,000	30- 675 hp	70- 1,500 hp	720-2,900
Teledyne Wisconsin Motor	3.5- 80	3- 69.5 hp	-	1,500-3,600
Volkswagen of America, Inc., Central Zone	-	20- 70 hp	-	2,000-4,000
Waukesha Engine Division of Dresser Industries, Inc.	21- 1,754	20- 2,845 hp	-	600-2,400
White Engines, Inc.	25.5- 180	26- 143 hp	-	1,200-3,600
White Superior Division, White Motor Corp.	220- 2,400	250- 2,650 hp	220- 2,400 hp	500-1,000
Witte Engine Corp.	9.5- 30	8- 30 hp	-	600-2,000

2 INTERNAL COMBUSTION PISTON ENGINE STANDARD PRACTICE

2.1 STANDARD RATINGS

The standard rating of an internal-combustion (IC) piston engine is the net brake horsepower (Bhp) delivered by that engine in good operating condition, with atmospheric temperature not over 90°F (32°C) and barometric pressure at 29.38 in. (∿ 1,500 ft above sea level) of mercury (SAE standard conditions). Engine manufacturers offer engines with sufficiently conservative ratings to permit delivery of an output 10% in excess of full-load rating at rated speed with safe operating temperatures for any two hours, but not to exceed a total of two hours out of any consecutive 24 hours of operation. Standard practices for low- and medium-speed stationary diesel and gas engines are given in Ref. 4.

To determine maximum usable output under non-standard operating conditions, it may be necessary to derate the standard horsepower outputs of the engines in accordance with Eq. 2.1 as follows:[5]

$$P_u = P_r \times F_r \qquad \text{(Eq.2.1)}$$

where:

P_u = usable shaft power, hp

P_r = rated engine power, hp

F_r = derating factor, fraction

The derating factor (Eq. 2.2) consists of an altitude correction, a temperature correction, a heating value correction, and a reserve to allow for unforeseen conditions such as dusty environment, poor maintenance, higher ambient temperature, and lowered cooling efficiency, that would reduce output. Unless otherwise specified by the manufacturer, derating factors can be determined as follows:

$$F_r = [100-(C_a + C_t + C_h + C_r)]/100 \qquad \text{(Eq.2.2)}$$

where:

C_a = 3%/1,000 ft>specified level for naturally aspirated engines; 2%/1,000 ft for turbocharged engines;

C_t = 1%/10°F rise>specified base temperature for air intake;

C_H = 2%/100 Btu/ft^3 decrease in fuel (gas) heating value below base value of 1,000 Btu/ft^3 (for gas engines); and

C_r = values of minimum engine reserves for air conditioning and refrigeration applications as given in Table 3 of Ref. 5.

Engine builders have approved various conventionally cooled models for ebullient cooling applications. However, the engine power outputs are generally derated by the manufacturer when the jackets are to be ebulliently cooled because plant designs using hot water for cooling often require higher than normal jacket water temperatures. Derating factors for ebulliently cooled engines usually are about 80% of the standard prime power ratings.

Ebulient cooling involves the natural circulation of the jacket water at or near saturation temperature, and engine cooling is accomplished through utilization of the heat of vaporization. Some benefits of evullient cooling are: (1) elimination of the jacket water circulating pump; (2) extended engine life because of uniform temperatures throughout the engine (normally 2-3°F differential between inlet and outlet); (3) recovered heat in the form of low pressure steam (up to 15 psig and 250°F); and (4) recoverability of all heat rejected to the jacket water.

Some engines have modified gasket and seal designs to ensure satisfactory operation with ebullient cooling systems. However, it is necessary to maintain a constant back pressure at the steam outlet of the ebullient cooling unit flash chamber to prevent sudden lowering of operating pressure. If the operating pressure should change rapidly, the steam bubbles in the engine could expand and interfere with fluid circulation, thereby permitting possible overheating at critical points in the engine. The engine coolant passages must be arranged for gravity circulation and free elimination of steam bubbles.

2.2 SAFETY REQUIREMENTS

Internal combustion engines are comparatively free from hazards to life and property. When accidents do occur, their effects generally are confined to engine damage only. The safety requirements specified in the codes and standards established by the authority having jurisdiction over the installation must be met. These codes usually are based on state, regional, and national codes; nevertheless, reasonable engineering judgment will be necessary in all cases.

Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines[4] provide generally accepted standards for nomenclature, installation, application, operation, and maintenance of engines and accessory equipment in various types of stationary engine installations.

NFPA Standard 37 indicates the environment and applications under which prime mover installations are considered to be in hazardous locations.

The National Electric Code specifies the type of equipment and wiring to be used in certain hazardous locations, but the Code does not define these conditions.

ANSI Standard Z21-40.2 covers the construction and performance of gas-engine-driven air conditioning appliances.

A comprehensive treatment of building design details, fuel oil storage, ventilation requirements, and other aspects of operation and maintenance pertaining to safety is beyond the scope of this evaluation. For standard practices regarding these aspects, refer to the Diesel Engine Manufacturers' Association (DEMA) publication, *Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines*.

2.3 EMPIRICAL PERFORMANCE EQUATIONS

A computerized version of the method of least squares was used to determine values of coefficients of a generalized empirical equation (Eq. 2.3) representing the part-load performance characteristics of the engines. The computer program fits least-squares polynomials to bivariate data, using an orthogonal polynomial method. The generalized empirical equation is of the form:

$$Y = A + BX + CX^2 + DX^3 \text{ ---} \qquad \text{(Eq.2.3)}$$

where the ordinate Y represents the value of a certain function corresponding to known, assigned, or observed values of its independent variable X. The value of Y may be any function that varies with engine load, such as brake horsepower, jacket water heat rejection, lubricating oil heat dissipation, or heat lost to the engine exhaust gas or radiated to the ambient atmosphere.

3 SPARK IGNITION GAS ENGINES

A spark-ignition gas engine is a prime mover whose piston is actuated by the combustion of a gaseous fuel in air. An evaluation of the gas engine from the standpoint of application in modular integrated utility systems (MIUS) for communities, with emphasis on thermodynamic evaluation and costs, is available in Ref. 6. Refer to Table 1.1 for the 16 major manufacturers of gas engines with headquarters in the United States, and with power outputs ranging up to 13,500 Bhp.[3]

3.1 DESCRIPTION

Spark ignition gas engines may be classified in the following general ways:

1. *by combustion cycle:* (a) four-cycle, (b) two-cycle.
2. *by power impulses:* (a) single-acting, (b) double-acting.
3. *by arrangement:* (a) vertical, (b) horizontal, (c) V-type, (d) opposed.
4. *by speed:* (a) low-speed (100 to 450 rpm), (b) intermediate-speed (450 to 900 rpm), (c) high-speed (900 to 1800 rpm).
5. *by air intake:* (a) naturally aspirated (b) turbocharged.

The engine components include a fuel input system, fuel-air mixing system, ignition system, combustion chamber, exhaust-gas collection and removal system, lubrication system, and power transmission with pistons, connecting rods, crankshaft, flywheels, etc.

Gas engines have had wide acceptance as drive units for electrical generators, reciprocating compressors, centrifugal compressors, and various types of pumps. Figure 1.1 shows a generalized energy distribution wherein a gas engine prime mover is assumed to drive the connected load, and rejected heat from the exhaust and jacket water is recovered for purposes of space and domestic hot water heating and for space cooling.

3.2 GAS ENGINE FUEL CONSUMPTION

Fuel consumption for gas engines customarily is expressed in terms of specific fuel consumption, or Btu (lower heat value) per brake horsepower hour. Manufacturers of gas engines usually publish the fuel consumption of their engines at standard rating conditions. These data can be presented graphically as "part-load" curves, and they provide fuel consumption data for loads less than rated output and usually at several speeds, as well as at rated speed and output.

Figure 3.1 shows comparative specific fuel-rate curves for large gas, dual-fuel, and diesel engines.[7] The curves in Fig. 3.1 indicate the improvement in economy obtained by turbocharging the 2-cycle gas engine, and of turbocharged 4-cycle engines, in general, over the unturbocharged 2-cycle gas engine. Characteristically, the diesel engine has the flattest fuel-rate curve, with performance at partial loads approaching that at full load. The 2-cycle spark-ignited gas engine has the poorest fuel-rate at full load and the fuel rate increases rapidly at loads below about 60%. In accordance with customary practice, the gas and dual-fuel engine fuel rates are based on the lower heating value of the fuel; whereas, the diesel-fuel rate is based on the higher heating value. Practically all high output IC engines are now turbocharged and intercooled to boost power output and improve efficiency.

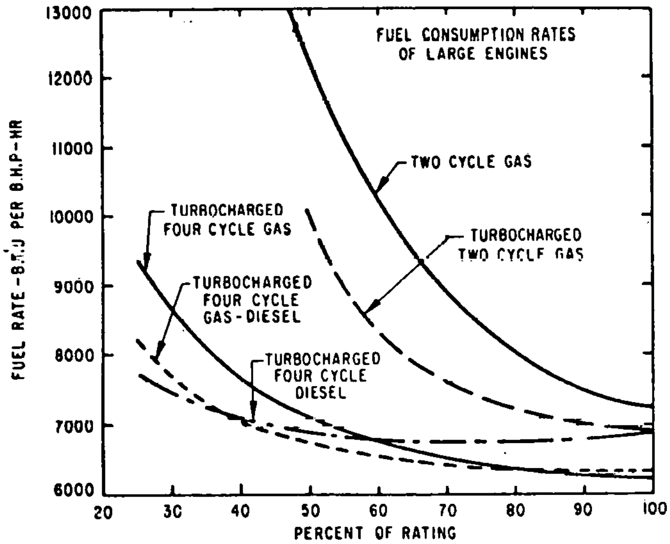

Fig. 3.1 Comparative Specific Fuel-Rates of Large Gas, Dual-Fuel and Diesel Engines

Source: Diesel Engineering Handbook (1976)

A mathematical model that approximates the percent of full-load fuel consumption as a function of the percent of full-load brake horsepower was given in Eq. 2.3, where X represents the percent of full load brake horsepower; Y represents the percent of full-load fuel consumption; and the values of the coefficients are as given in Table 3.1 for comparative fuel-rates of the large gas, dual-fuel and diesel engines shown in Fig. 3.1.

Table 3.1 Generalized Equation Coefficients - Percent (Y) of Specific Fuel Consumption at Full-Load for Gas, Dual-Fuel and Diesel Engines Vs Percent (X) of Rate Load ($25 < X < 100$)

Engine Type	Coefficients			
	A	B	C	D
2-cycle gas	506	-10.9	0.098	-2.96×10^{-4}
Turbocharged 2-cycle gas	558	-15.1	0.168	-6.30×10^{-4}
Turbocharged 4-cycle gas	219	-3.74	0.041	-1.54×10^{-4}
Turbocharged 4-cycle gas-diesel	176	-2.51	0.028	-1.09×10^{-4}
Turbocharged 4-cycle diesel	142	-1.61	0.019	-6.94×10^{-5}

3.3 SPARK IGNITION GAS ENGINE HEAT BALANCE

The energy input in the fuel appears as brake horsepower, as heat rejected to the jacket cooling water, as heat rejected to the lube oil, as heat rejected to the exhaust gas, and as lost by radiation and natural convection. Some representative heat balance curves for four-cycle, naturally aspirated and turbocharged gas engines are shown in Fig. 3.2 through 3.4.[8]

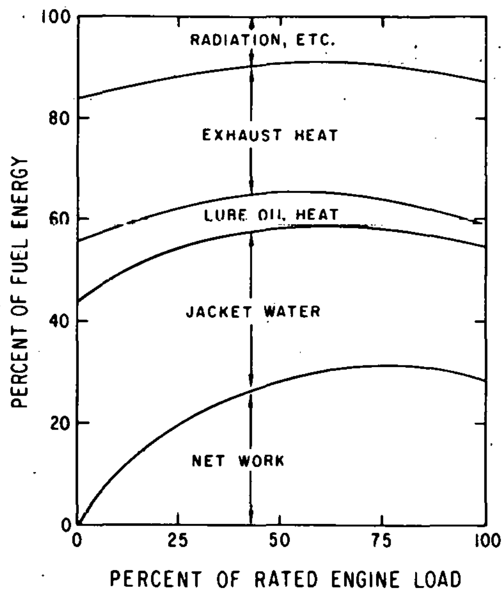

Fig. 3.2 Heat Balance - Naturally Aspirated Spark Ignition Gas Engine with Hot Exhaust Manifold

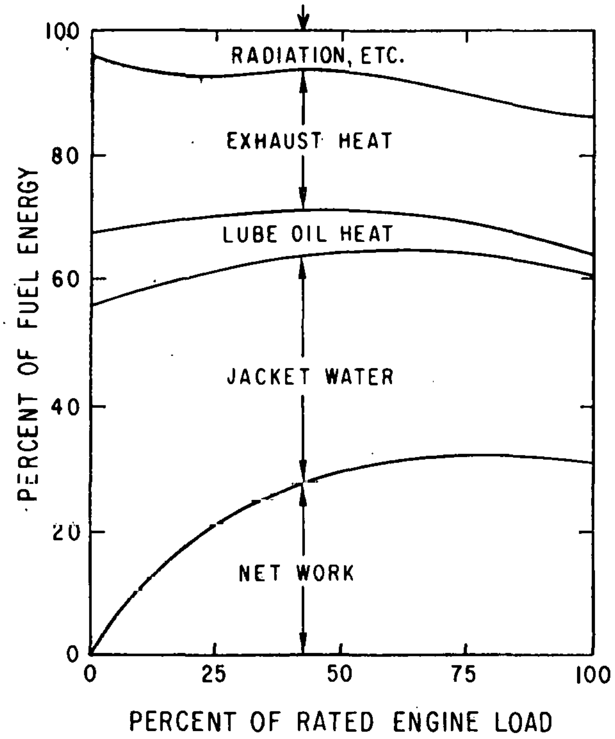

Fig. 3.3 Heat Balance - Naturally Aspirated Spark Ignition Gas Engine with Water-Cooled Exhaust Manifold

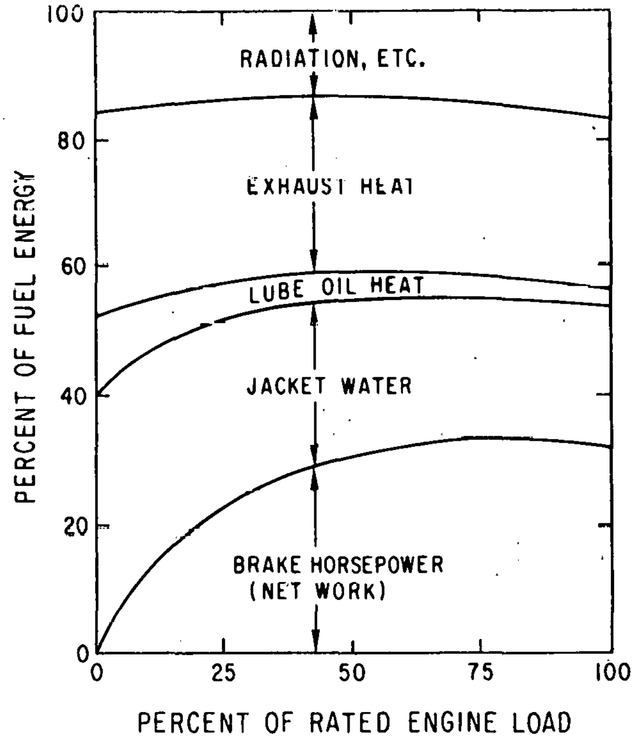

Fig. 3.4 Heat Balance - Turbocharged Spark-Ignition Gas Engine

The exhaust gas temperatures for these engines are about 1,200°F at full load and 1,000°F at 60% load. Figures 3.2 and 3.3 show heat balance curves for unsupercharged, naturally aspirated gas engines with hot exhaust manifold and water-cooled exhaust manifold, respectively.[8] Figure 3.4 is a heat balance for a turbocharged gas engine.[8]

As shown in Figs. 3.5 through 3.8, the percent of fuel energy input converted to work, or rejected to the cooling water, lube oil and exhaust of spark ignited gas engines, can be affected by whether or not the engine is turbocharged and/or exhaust manifold cooled.

The indicated horsepower (ihp) of an engine varies with the mass of air trapped in the cylinders per cycle, with a corresponding increase in brake horsepower (bhp). An increase of 30% in the mass of trapped air can produce an increase of up to 37.5 to 41.6% in bhp of the engine. Increased power output of an engine by turbocharging also results in a corresponding increase in the amount of heat liberated in the engine per cycle.

Exhaust manifold after cooling can provide an additional improvement in the performance of a turbocharged engine. When air at 90°F is compressed to greater than twice atmospheric pressure by turbocharging, the temperature of the air rises to about 300°F. After cooling, using engine jacket water, lowers the air temperature to very near the jacket-water temperature, and as a result more and cooler air enters the engine, resulting in more power. As added bonus, engine parts operate at lower temperatures, and peak cylinder pressures and exhaust temperatures are lower resulting in longer engine life and greater reliability.

3.3.1 Brake Thermal Efficiency

The brake thermal efficiency of spark ignition gas engines at rated load and speed will vary by several percent depending on: (1) the model; (2) whether the engine is supercharged or naturally aspirated; and (3) if supercharged, on the intercooler temperature. Brake thermal efficiencies

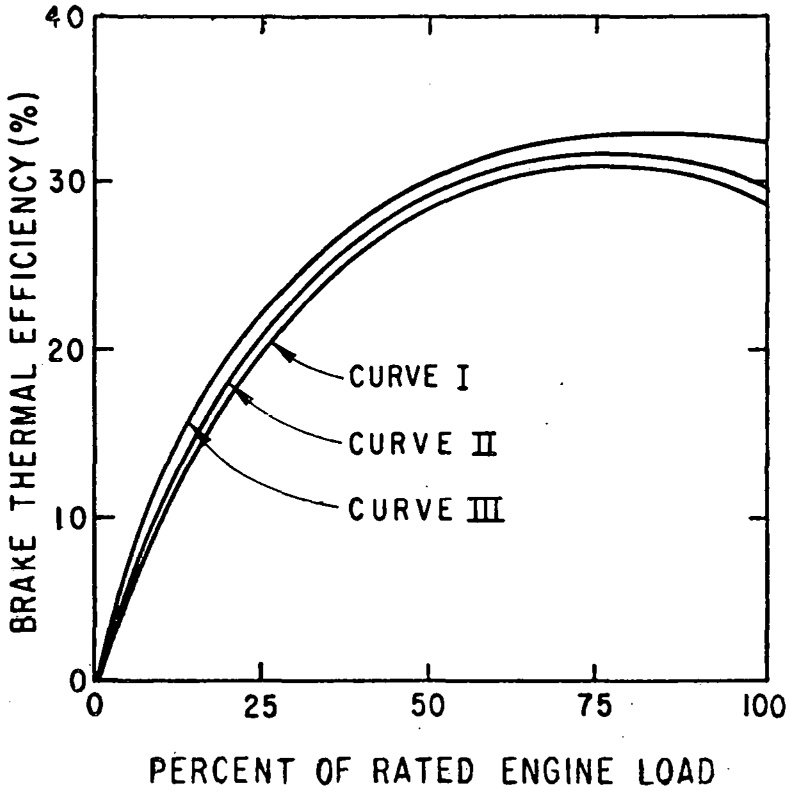

Fig. 3.5 Thermal Efficiency of Typical Spark-Ignition Gas Engines
Source: ASHRAE Guide & Data Book, Systems[8]

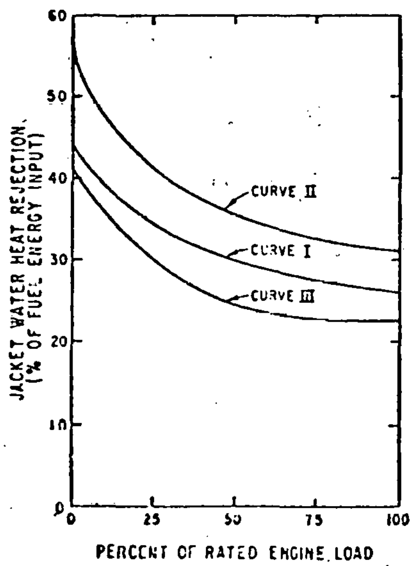

Fig. 3.6 Jacket Water Heat Rejection - Typical Spark Ignition Gas Engine
Source: ASHRAE Guide & Data Book, Systems[8]

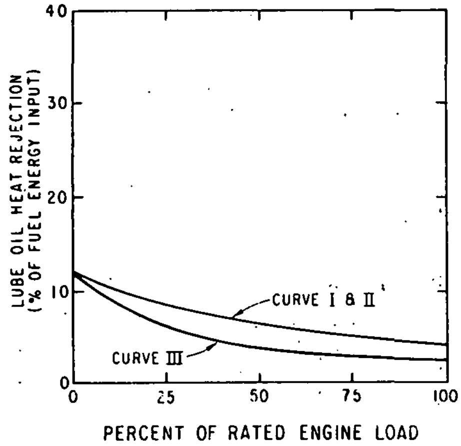

Fig. 3.7 Lube Oil Heat Rejection — Typical Spark Ignition Gas Engine
Source: ASHRAE Guide & Data Book, Systems[8]

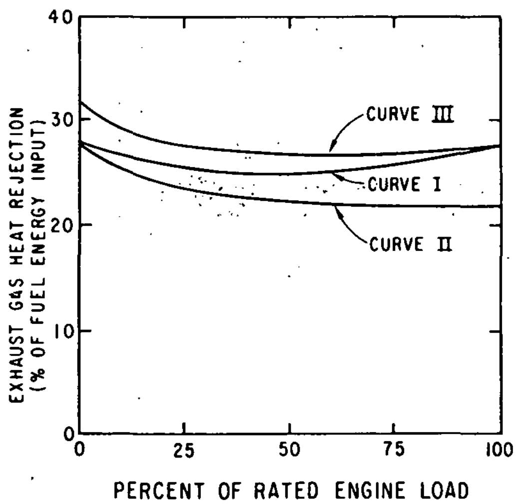

Fig. 3.8 Exhaust Gas Heat Rejection — Typical Spark Ignition Gas Engines
Source: ASHRAE Guide & Data Book, Systems[8]

of spark-ignition gas engines are shown in Fig. 3.5, and Table 3.2 gives values of coefficients to use in the generalized formula (Eq. 2.3) equating brake thermal efficiency to part-load performance.

Table 3.2 Generalized Equation Coefficients — Brake Thermal Efficiency of Spark-Ignited Gas Engines (Y) Vs Percentage of Rated Load (X)[a]

Gas Engine Type		Coefficients			
		A	B	C	D
I	Naturally aspirated with hot exhaust manifold	0.0	1.0364	-1.1052×10^{-2}	3.5880×10^{-5}
II	Naturally aspirated with water cooled exhaust manifold	0.0	1.0976	-1.2143×10^{-2}	4.1667×10^{-5}
III	Turbocharged	0.0	1.2670	-1.6334×10^{-2}	6.8866×10^{-5}

[a] $0 \leq X \leq 100$

3.3.2 Jacket Water Heat Rejection

Jacket water temperature varies from 180°-250°F, depending on whether the engine is cooled by forced circulation heat exchanger or ebullient systems. The heat rejected to the jackets ranges from >10% on low-speed (<600 rpm) engines to 30% for high-speed (1200 rpm) engines of the heat input at full loads, but may vary with the engine model. Specific jacket cooling water heat rejection rates at full load are available from engine manufacturers' data. Figure 3.6 shows the percent of fuel energy input typically rejected to the jacket cooling water as a function of the percent of full load. Table 3.3 gives the coefficients for use in the empirical equation (Eq. 2.3) relating the jacket water heat rejection to the part-load performance of the engine.

3.3.3 Lube Oil Heat Rejection

The amount of heat rejected to the lube oil of a 4-cycle gas engine with high temperature (310-350°F) jacket water coolant is about 420 to

Table 3.3 Generalized Equation Coefficients — Jacket Water Heat Rejection from Spark Ignition Gas Engines (Y) Vs Percentage of Rated Load (X)[a]

Engine Type	Coefficients			
	A	B	C	D
I Naturally aspirated with hot exhaust manifold	43.754	-0.503	5.784×10^{-3}	-2.546×10^{-5}
II Naturally aspirated with water cooled exhaust manifold	55.730	-0.842	1.090×10^{-2}	-4.977×10^{-5}
III Turbocharged	40.925	-0.613	6.940×10^{-2}	-2.662×10^{-5}

[a] $0 \leq X \leq 100$

480 Btu/Bhp-hr and 180 to 240 Btu/Bhp-hr at lower (180°F) lube oil and jacket water temperatures. Therefore, between 5 and 15% of the total fuel input will result in heat that must be extracted from the lube oil, and this may warrant operation of oil coolant at a high enough temperature to permit economic utilization in a process such as domestic hot water heating. Figure 3.7 shows the variations in total heat input to the engine rejected to the lube oil as a function of the percent part-load operation of the typical spark ignition gas engine, and Table 3.4 gives the value of coefficients to be used in the empirical equation relating lube oil heat rejection to part-load operation of the engine.

Table 3.4 Generalized Equation Coefficients — Lube Oil Heat Rejection from Spark-Ignition Gas Engines (Y) Vs Percentage of Rated Load (X)[a]

Engine Type	Coefficients			
	A	B	C	D
I Naturally aspirated with hot exhaust manifold	12.0	-0.189	2.014×10^{-3}	-9.259×10^{-6}
II Naturally aspirated with water-cooled exhaust manifold	12.0	-0.189	2.014×10^{-3}	-9.259×10^{-6}
III Turbocharged	12.0	-0.3418	4.489×10^{-3}	-2.026×10^{-5}

[a] $0 \leq X \leq 100$

3.3.4 Exhaust Gas Heat Rejection

A considerable portion (∿ 30%) of the toal heat input to a gas engine is rejected to the exhaust, but only about 60-65% of this heat can be recovered because it is necessary to maintain the exhaust gas at a temperature greater than 325°F ± 25°F to prevent corrosion of the heat recovery equipment. The exhaust gas temperature of gas engines varies from approximately 800° to 1,350°F depending on the size of the engine, its efficiency, and whether it is turbocharged. Figure 3.8 shows the percent of heat input rejected to the exhaust gas of a typical spark ignition gas engine as a function of the percent part-load operation of the engine. Table 3.5 gives the value of the coefficients in the generalized equation relating exhaust gas heat rejection to part-load performance of the engine.

Table 3.5 Generalized Equation Coefficients — Exhaust Gas Heat Rejection from Spark-Ignition Gas Engines (Y) Vs Percentage of Rated Load (X)[a]

	Coefficients			
Engine Type	A	B	C	D
I Naturally aspirated with hot exhaust manifold	28.056	-0.167	2.937×10^{-3}	-1.273×10^{-5}
II Naturally aspirated with water-cooled exhaust manifold	27.905	-0.236	3.155×10^{-3}	-1.389×10^{-5}
III Turbo-supercharged	31.857	-0.232	3.438×10^{-3}	-1.563×10^{-5}

[a] $0 \leq X \leq 100$

3.3.5 Intercooler Heat

Practically all high output engines are now supercharged, usually with exhuast-gas-driven turbines to drive the compressors. Intercooling downstream of the turbine to reduce air temperature serves to increase air density and mass rate of air flow. Turbochargers on gas engines require a relatively low intercooler water temperature (90°F or less for high-compression ratios and best fuel economy). The amount of heat removed by the intercooler water is relatively low (from 200 to 400 Btu/Bhp-h, depending on the engine size).

3.3.6 Radiation and Other Losses

A certain amount of heat (approximately 3-16% of the heat input) will be radiated to the environment. The sum of all the losses plus heat rejections and heat converted to useful work must always add up to the total heat input at any engine load.

3.4 PREFEASIBILITY EVALUATION

Full-load brake horsepower efficiencies of 34% for spark ignition gas and dual-fuel engines, and 30% for high-speed, spark ignition gas engines have been recommended for initial prefeasibility studies.[9] However the data represented by the heat balance curves for the turbocharged engines (Figs. 3.4 through 3.8) are suggested for prefeasibility evaluation of spark-ignition gas engines and dual-fuel engines in the ICES Program.

4 COMPRESSION IGNITION DIESEL ENGINES

4.1 DESCRIPTION

"Diesel" is the generic name of a type of prime mover in which air, heated to the ignition temperature of the fuel by compression, is the sole means of igniting the charge. Table 4.1 presents full-load application data from various manufacturers[10-13] of diesel engines ranging from 480 to 13,540 rated Bhp.

Engine capacities for stationary power plants and total energy applications range up to 40,000 hp (30,000 kW) in a single engine. The very high output engines are mostly two-stroke/cycle European products characterized by large displacements and low-speed operation with piston diameters of 40 in. (100 cm) or more, developing up to 4,000 hp (3,000 kW) per cylinder at speeds between 100 and 150 rpm and bmep (brake mean effective pressure) values over 150 psi (1,000 kPa), and the trend to higher specific outputs continues. Two Enterprise RV20, medium-speed, diesel engines, rated at 12,200 hp each, have been designed by the DeLaval Engine and Compressor Division for a Florida municipal system. A 58-MW diesel plant is scheduled to begin operation soon at the Twin Buttes (AZ) copper mine of Anamax Mining Company. This plant will contain nine RV16 Enterprise diesel engine-generators, each rated at 6,415 kW. The engines will burn heavy fuels to 3,500 sec (Redwood No. 1) and lighter grades of diesel oil.

4.2 FUEL CONSUMPTION

Standard practice in the engine industry is to publish fuel consumption data for diesel engines based on the high heat value (HHV) of the fuel, usually expressed in pounds per brake horsepower-hour (lb/Bhp-h) or pounds per kWh for diesel generator sets. Figure 4.1 shows the diesel cycle air standard efficiency, and Fig. 4.2 presents specific fuel consumption curves (heat rates) in Btu/h for low-speed (450 rpm), medium-speed (900 rpm), and high-speed (1200 rpm) stationary diesel engines.[14]

4.1 Representative Diesel Engine Application Data

Reference	(10)	(11)	(12)	(13)	(13)	(13)
General Specifications						
Rated power[a], Bhp	480	1,505	6,420	8,125	10,833	13,540
Rated speed, rpm	1,800	1,200	514	450	450	450
Strokes per cycle	4	4	4	4	4	4
Number of cylinders	V-8	V-12	V-12	V-12	V-16	V-20
Displacement, $in.^3$	1,190	5,788	42,322	57,199	76,265	95,332
Aspiration	Turbo	Turbo	Turbo	Turbo	Turbo	Turbo
Compression ratio	16.5:1	13:1	11.5:1	NA	NA	NA
Rated BMEP, psi	NA	173	234	NA	NA	NA
Engine weight, lb	(3,100)	(9,790)	133,200	190,000	225,000	288,000
Overall length, in.	(195)	(109)	271	232	280	328
Overall width, in.	60	80	138	133.5	133.5	133.5
Overall height, in.	68	116	139	152	152	196.5
Fuel consumption data (full load)	(b)	(c)	(d)	(c)	(e)	(e)
Fuel rate[f], lb/hphr	0.395	0.392	0.364	0.369	0.369	0.369
Heat rate[g], Btu/hphr	7,703	7,631	6,621	6.712	6,712	6,712
Thermal efficiency[h], %	33.0	33.4	38.4	37.9	37.9	37.9
Air standard efficiency[i], %	66.2	62.8	62.0	NA	NA	NA
Relative efficiency[j], %	49.8	53.2	62.0	NA	NA	NA
Heat rejection data (full load)						
Jacket water, Btu/min	17,100	45,438	104,325	152,350	203,133	253,900
Lube oil, Btu/min	4,087	8,135	25,466	36,567	48,750	60,933
Intercooler water, Btu/min	NA	5,450	51,253	NA	NA	NA
Radiated, Btu/min	NA	11,210	22,470	NA	NA	NA
Exhaust system data (full load)						
Exhaust gas temp., °F	875	771	900	855	855	855
Exhaust gas flow, cfm	2,800	7,912	64,478	49,857	66,678	82,710
Max. allowable back pressure, in. H_2O	20	12	NA	NA	NA	NA

[a] $Bhp = \frac{(BMEP)(L)(A)(N)}{33,000}$ where: BMEP - Brake mean effective pressure, psi; L - Stroke of piston, ft; A - Net piston area, $in.^2$; N = Number of power stokes/min in all cyclinders; 33,000 = ft-lb of work/min/hp.

[b] Based on fuel oil having a gross heat value of 19,500 Btu/lb and weighing 7.12 lb/U.S. gal.

[c] Based heat value of fuel = 19,450 Btu/lb and weighing 7.29 lb/U.S. gal.

[d] Based on heat value of 18,190 Btu/lb.

[e] Based on heat value of 18,190 Btu/lb (LHV).

[f] Fuel rate = fuel consumption (gal/hr) x 7.29 (lb/gal) ÷ rated full load output (Bhp).

[g] Heat rate = fuel rate (lb/hp-hr) x (heating value of fuel, Btu/lb).

[h] Thermal efficiency = 2,545/heat rate.

[i] The air standard efficiency is the thermal efficiency in which the working medium is assumed to be air, and it has a constant specific heat. The air standard efficiency of a diesel engine can be estimated from Fig. 4.1 (Ref. 15), assuming a cutoff ratio of 1.2 (arbitrary).

[j] The relative efficiency of the engine is the ratio of the thermal efficiency (conversion) actually obtained to the air standard efficiency (theoretical maximum).

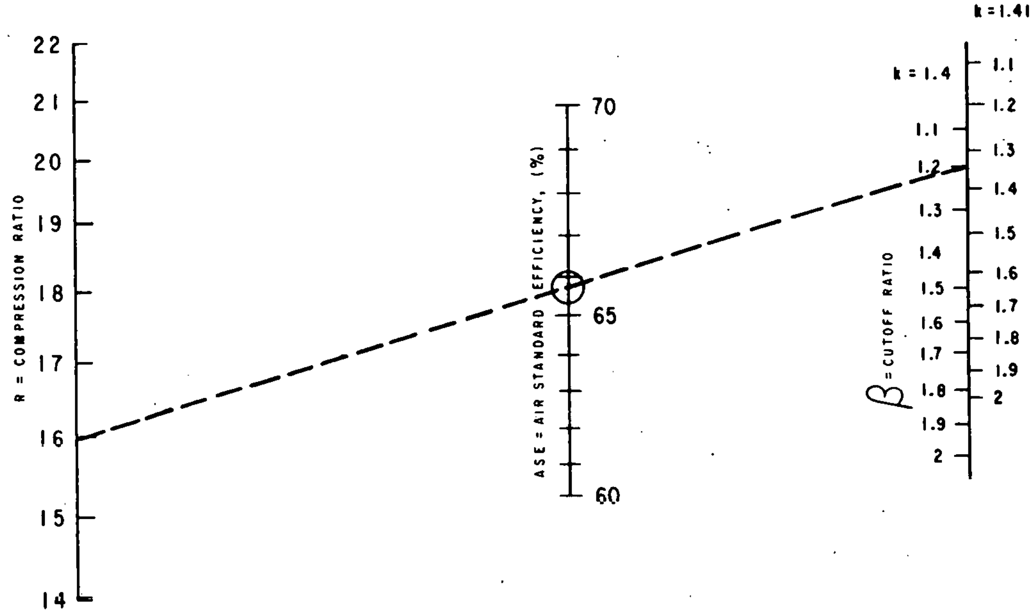

Fig. 4.1 Diesel Cycle Air Standard Efficiency

Source: *Diesel Cycle Efficiency,* Power[15]

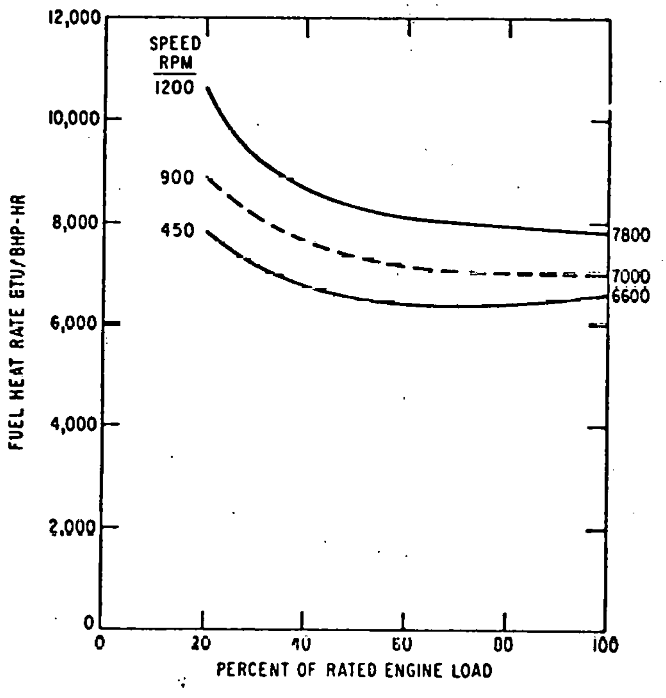

Fig. 4.2 Specific Fuel Consumption Curve for Low-Speed (450 RPM), Medium-Speed (900 RPM) and High-Speed (1200 RPM) Stationary Diesel Engines

Source: *Advanced Coal Using Community Systems - Preliminary Final Draft Report, Task IA*[14]

The fuel consumption curves shown in Fig. 4.2 have been given down to 25% load to permit estimating fuel consumption at this lower load; however, fuel consumption guarantees are not usually made at less than 50% of rated full load. The rates shown in Fig. 4.2 are based on operation at altitudes up to 1,500 ft above sea level, temperatures not to exceed 90°F, and a fuel heating value of not less than 19,350 Btu (HHV) per pound. Multiplying the specific fuel consumption by the horsepower load will provide the total fuel consumption in pounds per hour of a given gravity fuel for the respective load.

A mathematical model to approximate the percent of full-load fuel consumption as a function of the percent of full-load brake horsepower is given by Eq. 2.3, where X represents the percent full-load brake horsepower and Y represents the percent full-load fuel consumption. The values of the coefficients are given in Table 4.2.

Table 4.2 Generalized Equation Coefficients - Percent (Y) of Full-Load Fuel Consumption Vs Percent (X) of Rated Load for the Low-Speed (450 RPM), Medium-Speed (900 RPM) and High-Speed (1200 RPM) Diesel Engines of Fig. 4.2 (@ $25 \leq x \leq 100$)

Engine Category	Coefficients			
	A	B	C	D
High-speed (1200 RPM)	184	-2.885	0.035	-1.458×10^{-4}
Medium-speed (900 RPM)	157	-2.036	0.024	-9.375×10^{-5}
Low-speed (450 RPM)	146	-2.020	0.025	-9.375×10^{-5}

4.3 DIESEL ENGINE HEAT BALANCE

The general range of energy distribution in a commercially available, large four-cycle turbocharged diesel engine at full rated power[7] is given in Table 4.3.

Table 4.3 Representative Heat Balance, Turbocharged and Aftercooled 4 Cycle, Low-Speed Diesel Engine

	Total Heat Supplied, Per Cent	
	General Range	Average
Brake horsepower	33 to 38	36
Exhaust & Radiation	37 to 40	38
Jacket Water	16 to 20	18
Lube Oil Cooler	2 to 6	4
Aftercooler	2 to 5	4

Figure 4.3 shows a representative variable load heat balance diagram for a four-cycle slow-speed turbocharged diesel engine operating at constant speed. The percentage of heat that is converted to work (bhp) is fairly constant above 60% of full power. Below 60% the percentage decreases at an increasing rate. The proportion of total heat that is dissipated to the exhaust remains approximately constant over the whole power range. Engines of different makes will show considerable variation. At any load, however, the summation of heat distributed as net effective work, jacket water heat rejection, lube oil heat rejection, exhaust heat rejection, and heat lost as radiation must always add up to the total fuel energy input.

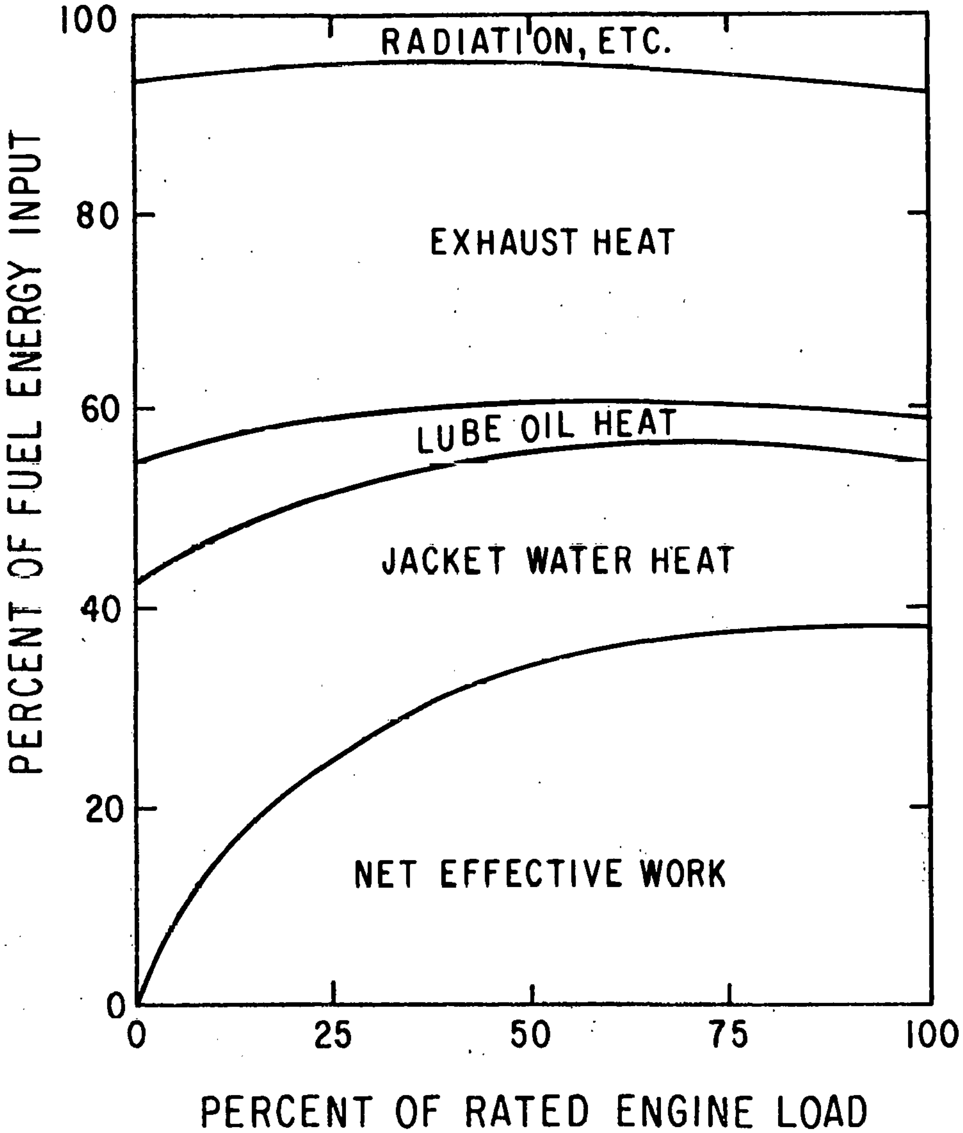

Fig. 4.3 Heat Balance — Turbocharged Low-Speed Compression Ignition Diesel Engine

The amount of power lost by friction increases only slightly with increases in power output at constant engine speed operation. According to Ref. 12, at no-load conditions, the total indicated developed horsepower is equal to the frictional horsepower, and may represent over 40% of the total heat supplied. At no-load, a small amount of fuel is required, and no heat is being converted to brake horsepower.

The curves given in Fig. 4.4 showing the percent of fuel energy input as a function of percent rated load for net work, exhaust, jacket water, lube oil and radiation are suggested for preliminary preevaluation studies of large (4,000 to 13,500 Bhp) stationary slow speed (450 rpm) diesel engines operating at constant speed. Table 4.4 gives the values of the coefficients for use in the generalized equation (Eq. 2.3) representing the part load performance characteristics of the engine.

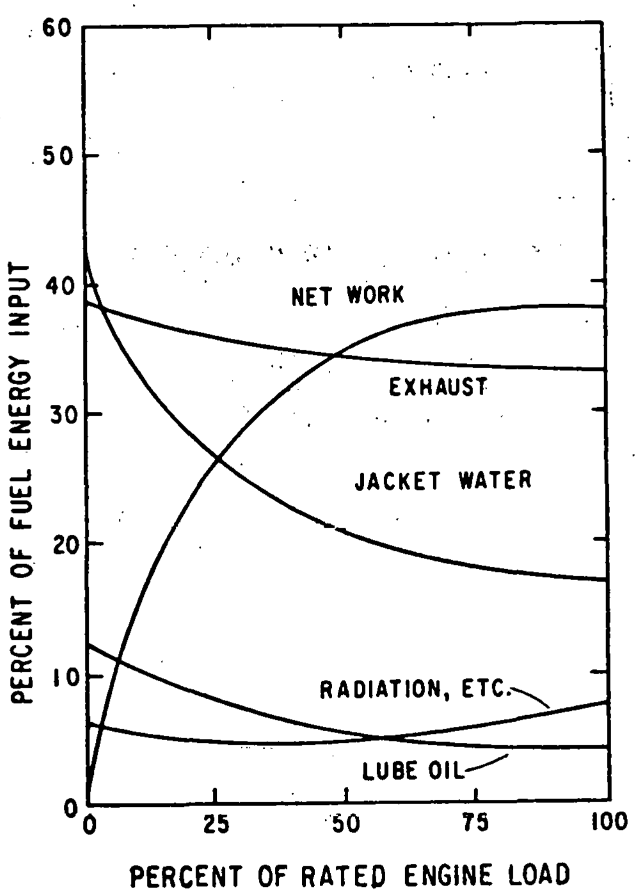

Fig. 4.4 Individual Heat Balance Curves - Turbocharged Compression Ignition Diesel Engine

Table 4.4 Generalized Equation Coefficients — Typical Diesel Engine Heat Balance (Y) Vs Percentage of Rated Load (X)[a]

Function	Coefficients A	B	C	D
Brake thermal efficiency	0.0	1.449	-1.869×10^{-2}	8.00×10^{-5}
Jacket water heat	43.0	-0.886	1.114×10^{-2}	-4.907×10^{-5}
Lube oil heat	12.0	-0.194	1.143×10^{-3}	0.0
Exhaust heat	39.0	-0.159	1.714×10^{-3}	-6.400×10^{-6}
Radiation, etc.	6.0	-0.090	1.314×10^{-3}	-2.667×10^{-6}

[a] $0 \leq X \leq 100$

4.3.1 Brake Thermal Efficiency

The brake thermal efficiency of a diesel engine is the ratio of the heat equivalent of 1 hp-h (2,545 Btu) to the heat units actually supplied per bhp-h, based on the higher heating value (HHV) of the fuel. The major advantages of the diesel engine are (1) its ability to operate with high efficiency over a wide range of engine loads (the efficiency of a diesel engine changes very little down to about 60% load, as shown in Fig. 4.4); and (2) its ability to cover, in a series of engines, a wide range of power outputs with little or no loss in efficiency.

4.3.2 Jacket Water Heat Rejection

In the full load heat balance given in Table 4.3, the general range of the heat rejected to the cooling water is between 16 and 20%. Heat rejection to the cooling water can be as high as 2,000 Btu/bhp-h (∿30% for turbocharged engines), and the quantities of heat evolved for naturally aspirated engines are some 600 Btu/bhp-h less.[16] If the engine is cooled by a forced-convection heat exchanger system, most manufacturers of large engines recommend engine jacket water temperature of 180°F and hold the temperature rise between 10 to 15°F with the lower value preferred. In ebullient-cooled engines, low pressure (15 psig) steam can be produced at a water outlet temperature of 250°F.

With an approximately constant percentage of heat dissipated to the exhaust (see Fig. 4.3), there is a correspondingly larger increase in the amount of heat dissipated to the cooling water from the jackets and oil cooler, and a smaller increase in the percentage to radiation and unaccounted for heat loss as the load decreases.

See Sect. 22, "Cooling System", of Ref. 7, for further discussion of the design of cooling systems for internal combustion engines. A chart (Fig. 22-1) to determine the amount of cooling water in gpm of water circulated per horsepower is presented in that section.

4.3.3 Lube Oil Heat Rejection

A full-load temperature of 160°F (71°C) can safely be maintained for lubricating oil leaving an engine; therefore, a secondary water temperature of about 135°F (57°C) or greater can be obtained at the oil cooler outlet.[15] The amount of heat rejected to the lube oil can vary from about 300 Btu/Bhp-h to about 500 Btu/Bhp-h.

4.3.4 Exhaust Gas Heat Rejection

The exhaust gas heat rejection for the engine heat balance given in Table 4.4 was based on a flowrate of 12 lb/Bhp-h and an exhaust gas temperature of 855°F at full load. For some highly turbocharged 4-stroke engines, the mass flow can be as high as 13 lb/Bhp-h. The temperature of the exhaust gas following the supercharger is variable, depending on the engine manufacturer, but generally will be from about 650°F to 850°F at full load. At part load, the exhaust gas temperature decreases. Figure 4.5 shows some representative exhaust temperatures of different types of diesel engines at various loadings.[17]

4.4 PREFEASIBILITY EVALUATION

In comparing the efficiencies of various internal combustion engines for initial evaluation, shaft efficiencies of 38% have been recommended for diesel engines.[8] The full load heat balance of Table 4.3 and the heat balance data represented by Figs. 4.3 and 4.4 and Table 4.4 are, however, suggested for initial prefeasibility analyses of ICES installations in which the use of diesel engines is considered. For analysis of specific engines, full-load, heat-balance data should be obtained from the manufacturer.

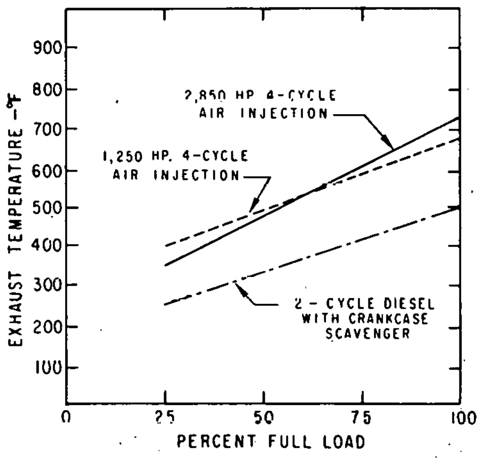

Fig. 4.5 Representative Exhaust Gas Temperature of Some Different Types of Diesel Engines

5 DUAL-FUEL ENGINES

5.1 DESCRIPTION

Various interpretations of the term "dual fuel," as applied to internal combustion piston engines, may be found in the literature.[5] For this discussion, "dual fuel" will refer to engines that are capable of changing automatically from gas to oil operation. The *1976 Diesel and Gas Turbine Catalog*[3] lists 119 worldwide manufacturers of dual-fuel engines ranging to 34,800 Bhp. The largest dual-fuel engine manufactured in the United States is rated at 13,540 Bhp. The 1975 Annual Plant Design Report[1] issued by *Power* indicates that only about 10% of the 128 engines covered in 1975 were of the dual-fuel type. Some users have evidently specified dual-fuel engines in the belief that enough advantageously priced gas will be available to justify the added cost of dual-fuel capability.

The dual-fuel engine can compete satisfactorily with oil-diesel engines because it operates either on fuel oil or gas and pilot oil. The pilot oil ignites the gas and is only a small percentage of the fuel requirements for full-load operation. The greatest advantage of the dual-fuel engine is its ability to be operated efficiently on whichever fuel is available and most economical. The engine can be easily and quickly converted from one fuel operation to the other because of the dual-purpose parts and the simplicity of controls.

During gas operation, the gas enters the cylinders through gas admission valves actuated by the camshaft. Pilot fuel is supplied from the fuel injection pumps and is injected into the cylinders through pintle-type nozzles. If the gas supply pressure decreases to a pre-determined value, a control valve can automatically shut off the gas supply and the engine will revert to full diesel operation. Some dual-fuel engines have an additional safety feature whereby the oil-actuated gas shutoff valve automatically closes if the lubrication or fuel oil pressure becomes low. Thus, the accumulation of gas within or around the engine can be prevented.

The dual-fuel engine operating in the gaseous fuel mode is essentially the same as a spark-ignition, gas engine except that the high-tension, spark-ignition system is eliminated and dual-purpose parts are added for gas and pilot oil admission.

5.2 FUEL CONSUMPTION

A fuel rate curve for a large turbocharged four-cycle, gas-diesel engine is shown by the dotted line in Fig. 3.1.[7] The rates are based on operation at altitudes up to 1,500 ft above sea level, temperatures not to exceed 90°F, with use of an approved pilot oil. Gas consumption is based on low heat value.

Table 5.1 lists the values for the coefficients to use in Eq. 2.3 representing the percent full load fuel consumption (variable Y) versus percent full load brake horsepower (variable X).

Table 5.1 Generalized Equation Coefficients — Percent (Y) Fuel Consumption for Dual-Fuel Engines Vs Percent of Rated Load (X)

	Coefficients			
Engine	A	B	C	D
Turbocharged 4-cycle Gas-Diesel	176	-2.15	0.028	-1.09×10^{-4}

5.3 HEAT BALANCE

The efficiency of a dual-fuel engine at full power is about the same as that of a diesel engine; however, its efficiency decreases with load in the same manner as for a gas engine. Therefore, in making parametric studies of dual-fuel engines, the heat-balance data given in Sections 3 and 4, can be used.

5.4 PREFEASIBILITY EVALUATION

As pointed out in Section 3.4, full load brake horsepower efficiencies of 34% for slow-speed, spark ignition, gas, and dual-fuel engines have been recommended for initial prefeasibility studies. However, the heat balance curves shown in Sections 3 and 4 for slow-speed, turbocharged, gas or diesel (depending on mode of operation) engines are suggested for prefeasibility evaluation of dual-fuel engines.

6 RELIABILITY OF INTERNAL COMBUSTION PISTON ENGINES

This section presents pertinent data from an ORNL report[18] that can be applied to determine the reliability of multiple internal combustion engine installations in integrated community energy systems (ICES). The reliability of a system is one of the most important attributes contributing to its overall performance, and, as such, must be given adequate consideration at the planning stage.

An ICES power-generation system normally will consist of several parallel prime power units, and the overall system reliability or availability will depend on the availability of the individual units, the number of units, and the excess capacity installed. For a system consisting of N units, all with an availability, A, and the availability of the units independent of each other, the probability of x, and only x, units being out of service is:

$$P_x = \frac{N!}{(N - x)!\, x!} A^{(N - x)} (1 - A)^x \qquad \text{(Eq.6.1)}$$

The dependence of P_x on N, x, and A is illustrated in Fig. 6.1. Figure 6.1 can be used to determine the probability of having some number of engines (N - x) operating or, as shown in Fig. 6.2, the probability of having x or more engines out of service. The probability of occurrence of the various combinations — assuming the probability to be uniform with time — also represents the fraction of time that this combination occurs. Thus, the probability calculations shown in Fig. 6.2 are expressed as hours per year.

Before these procedures can be applied to a system, availability data on the individual units are needed. One comprehensive study[19] made for the U.S. Army in connection with deployment of the Nike-X missile system found an overall availability of about 0.96 for both piston engine and gas turbine prime movers. Of the 4% average downtime observed, about 1% was attributed to forced outages and 3% to scheduled maintenance actions. The surprising aspect of these data is that for the piston engine, the mean time between failures was only slightly over 500 hr, and mean time for repairs

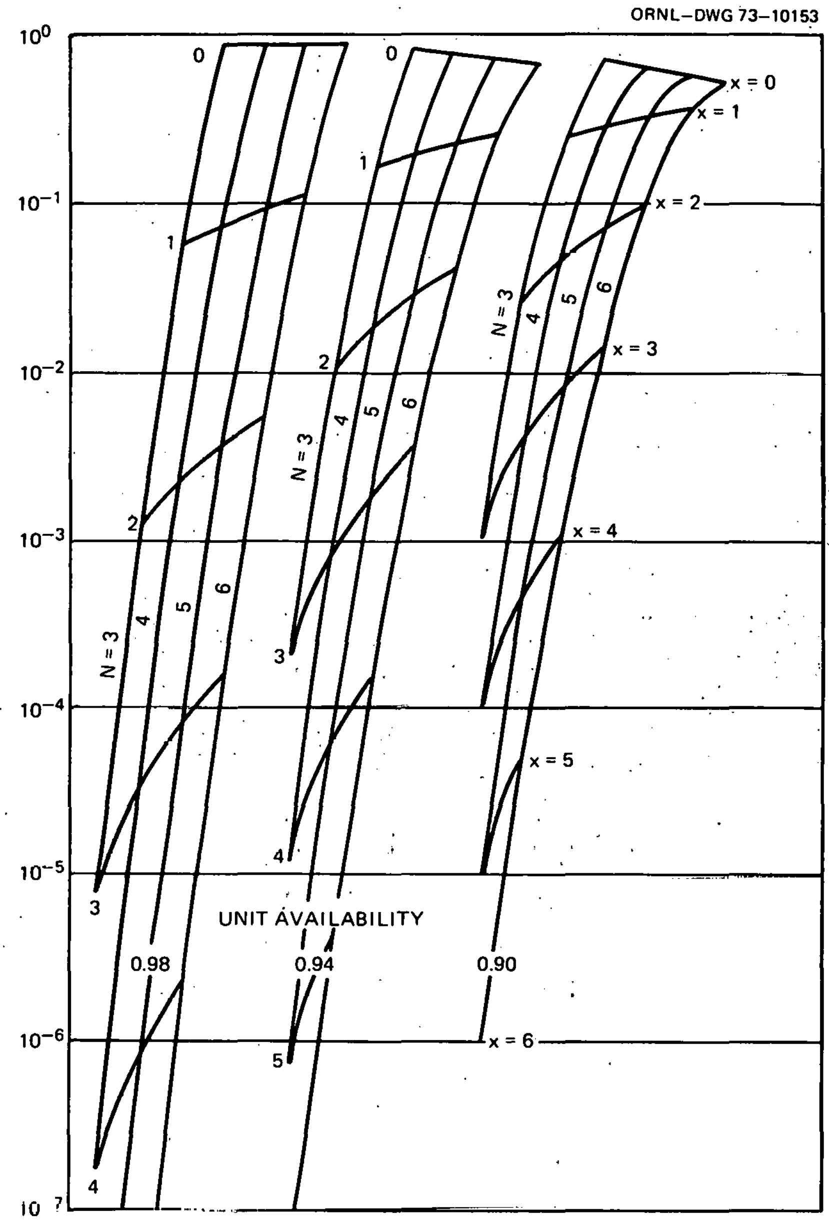

Fig. 6.1 Probability of Simultaneous Outages as a Function of Units Installed and Unit Availability. N = number of units; x = number of units out of service

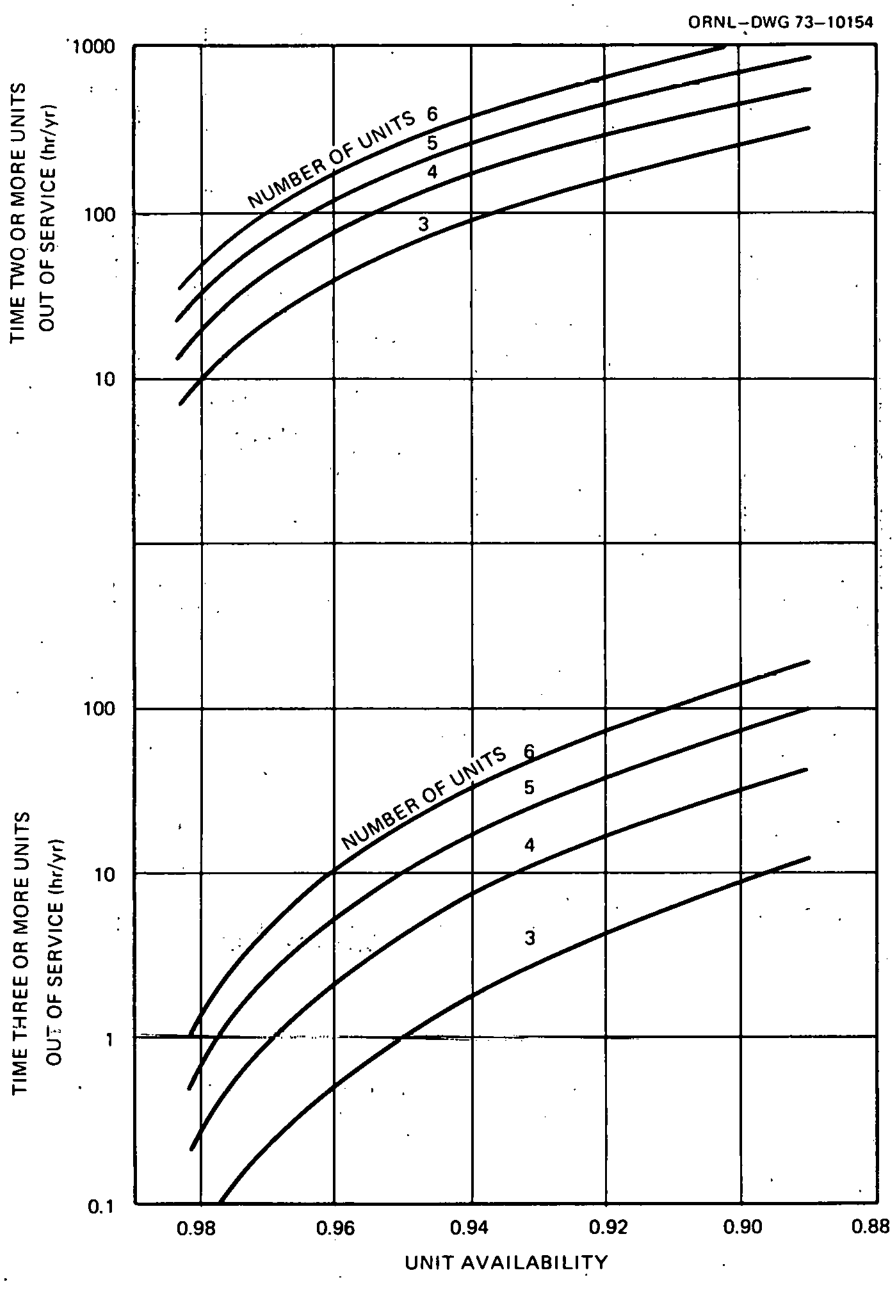

Fig. 6.2 Plant Downtime as a Function of Units Installed and Unit Availability

about 2.5 hr. Most of the outages were for items such as failures in water hose connections, lube oil and cooling water piping, and ignition systems.

The 1971 and 1974 ASME reports[20,21] on diesel and gas-engine power costs are another source of data on operating experience with diesel and gas-engine generator systems. The 1971 report lists 40 engines out of service for a total downtime of 10,255 hr for emergency repairs and 105 engines with no emergency outages. The average downtime for repairs was 226 hr for those engines requiring repair, and the average time out of service for all engines was 71 hr.

These data are for large, low-speed units and may be somewhat pessimistic. In addition to the problems of interpreting the data, 30% of the total outage time was traced to only three units, and it is not known whether these prolonged outages were necessary or caused by a lack of incentive to place the units back into operation. The total availability (including time out for maintenance) of over 96% has been found to be commonplace for large marine diesels.[19] A failure rate of 89 per 10^6 hr or 1 per 11,236 hr has been reported.[22] Data from a survey of small, high-speed piston engines used in total energy plants showed that with one manufacturer the failure rate was 1 per 9,640 hr and for a second manufacturer's engines, the failure rate was 1 per 3,020 hr.[23] Assuming an average repair time of 100 hr for these smaller engines, the availability, in terms of emergency or unscheduled outage, is between 0.99 and 0.967.

Most of the downtime for normal maintenance will be for minor and major overhauls. Generally, an engine will require one major and two minor overhauls every 30,000 to 40,000 running hours, or on the order of 50,000 actual hours. The total downtime during this 50,000-hr period will be between 1,000 and 1,500 hr, and the availability will be 0.98 to 0.97. However, there will probably be some overlapping of the above estimates of downtime, e.g., if an engine goes down for emergency repair about the time an overhaul is due, the two would be combined.

To illustrate the manner in which the above information can be applied, assume an installation having four 500-kW generators and a critical electrical load of 1,000 kW. The availability of the generators in relation to emergency outages will be taken as 0.98 and, for normal maintenance, as 0.97. With all four generators in service (or on standby) the critical load can be supplied unless three or more units become unavailable. The probability of three units being out of service simultaneously can be determined by Fig. 6.1. The value, 3.15×10^{-5}, corresponds to a plant downtime of 0.28 hr/yr as shown in Fig. 6.2. However, this value is low because it does not consider normal maintenance. Including the downtime for normal maintenance, the probability of not being able to meet the critical load is the probability of simultaneously losing two of the three engines in service (which is 1.18×10^{-3}) times the fraction of time that a generator is out of service for normal maintenance (which is 0.12). Thus the probability is 1.42×10^{-4} or 1.2 hr/year, which is less than the 5 hr/year experienced by electric utility customers in this country.

An ICES plant requires many combinations of equipment sizes, and the selection of both the number and size of the units will depend on the plant load characteristics. The above mentioned selection of four 500-kW units to meet a 1,000-kW critical load was based on a previous study for a 720-apartment complex which indicated that a peak load of 1,500 kW occurred in the summer. About 500 kW was used for compression air conditioning; however, because the compressors were located in the equipment building and under control of the plant operator, this load could be dropped in the event of a multiple-engine failure, and therefore it was not considered as part of the critical load.

Although the above discussion is for turbines and piston engines, the procedures can be applied to other equipment items for which availability data can be obtained. In other systems, such as heating or air conditioning, the probability of supplying part of the load might be of interest. The above type of analysis can be extended to include the mean time between multiple failures[24] and also the effects of having engines of different sizes[25] rather than of uniform size.

7 FUELS FOR INTERNAL COMBUSTION PISTON ENGINES

State and local regulations governing fuel availability and fuel costs apparently have greatly influenced the design and selection of internal combustion piston engine plant types, as reported in the 1975 plant design report.[1] Although it was shown that diesel oil and No. 2 fuel oil powered most of the engines covered in the report, the gas and dual-fuel types showed fair strength. Apparently some users have gone back to the dual-fuel choice on the belief that enough advantageously-priced gas will be available to pay for the added cost of dual-fuel capability. Most of the spark-ignition gas engines listed in the report will serve in the natural-gas industry, in which gas-fuel availability will be less of a problem.

The materials inputs (fuels used), and energy inputs (heating value of fuels) of the fuels required for operation of oil and gas engines are given in Tables 7.1 and 7.2. Fuel use data may be in terms of either the higher heating value (HHV) or lower heating value (LHV) of the fuel used. Most natural gases have an HHV/LHV factor of 1.11. For gaseous fuels in general, however, this factor may range from 1.00 to 1.15. For fuel oils, the HHV/LHV factor ranges from 1.05 for heavy oils to 1.07 for light oils.

Table 7.1 Fuels for Use in Gas Engines

Gaseous Fuel	HHV (Btu/ft^3)	LHV (Btu/ft^3)
Dry, processed natural gas	1,000	900
Propane HD5 or equivalent	2,500	2,500-2,174
Butane	3,200	3,200-2,783
Natural gas w/propane-air	1,000	900
Sewage gas	600	600-522
Natural gas w/hydrogen	800	720

Table 7.2 Typical Fuel Parameters for Diesel Engines

	Distillates				Heavy Fuels			
	Kerosene No. 1 Fuel Oil	Diesel Oil	No. 2 Fuel	No. 4 Fuel	Navy Special	No. 5 Fuel	No. 6 Fuel	Ref.
Relative cost factors	1.00	0.92	0.87	0.65	-	0.55	0.44	2
High heat value (HHV), Btu/gal	-	134,700	141,800	-	-	-	-	1
Low heat value (LHV), Btu/gal	-	127,080	133,774	-	-	-	-	1
Specific weight, Lb/gal	-	6.79	7.29	-	-	-	-	1
Gravity, (API)	43.00	38.00	34.00	21.00	11.5	13-20	6-15	2
Viscosity, SSU @ 100°F	33.00	35.00	40.00	140.00	450.0	530.00	-	2
Viscosity, SSF @ 122°F	-	-	-	-	-	-	150	2
Flash Pt., °F(min)	115.00	150.00	130.00	150.00	150.00	150.00	150	2
Conrudson Carbon % wt	0	0.10	0.25	0.02-.06	0.15	0.07-0.16	10-20	2
Sulfur, % wt. (max)	0.15	0.50	0.01	1.90	1.90	-	-	2
Total ash % wt. (max)	0	0.01	0.01	0.02	0.10	0.09	0.12	2
Cetane No.	50-55	49-59	30-40	-	-	-	-	2,3
Diesel Index	60-65	50-65	30-40	-	-	-	-	2,4
End boiling point,°F	560.00	765.00	680.00	-	-	-	-	2

7.1 FUELS FOR USE IN SPARK-IGNITION GAS ENGINES

Fuel consumption for gas engines customarily is expressed in terms of the low heat value (LHV) of the fuel; whereas, manufacturers of oil (diesel) engines rate engines in terms of the high heat value (HHV) of the fuel oil. The difference between the high- and low-heat values of fuels containing hydrogen is the heat of vaporization of the water formed when the hydrogen burns. Diesel and gas engines exhaust the gases of combustion at temperatures well above those at which water vapor would condense; as a result, the heat represented by the difference between the two heat values is not available for conversion to useful work in the engine. Typical fuels[26] for use in gas engines are listed in Table 7.1.

7.2 DIESEL ENGINE FUELS

The properties of commercial grades of fuel oils depend on the refining practices used and the nature of the crude oils from which they are produced. The selection of a particular diesel fuel oil for use in a given engine requires consideration of the following factors:

- fuel price and availability,
- maintenance considerations,
- engine size and design,
- speed and load ranges,
- frequency of speed and load changes, and
- atmospheric conditions.

Typical parameters for diesel engine fuel oils[27] are given in Fig. 7.1.

Gravity (API), which varies from 6 to 15 for No. 6 fuel, and from 13 to 20 for No. 5, indicates to some degree the fuel quality. Fuels below 10 API gravity are heavier than water. Fuels lighter than water are recommended[28] because heavier fuels carry a higher concentration of carbon and metals.

Viscosity determines heating temperature. Less-viscous oils require less heating.

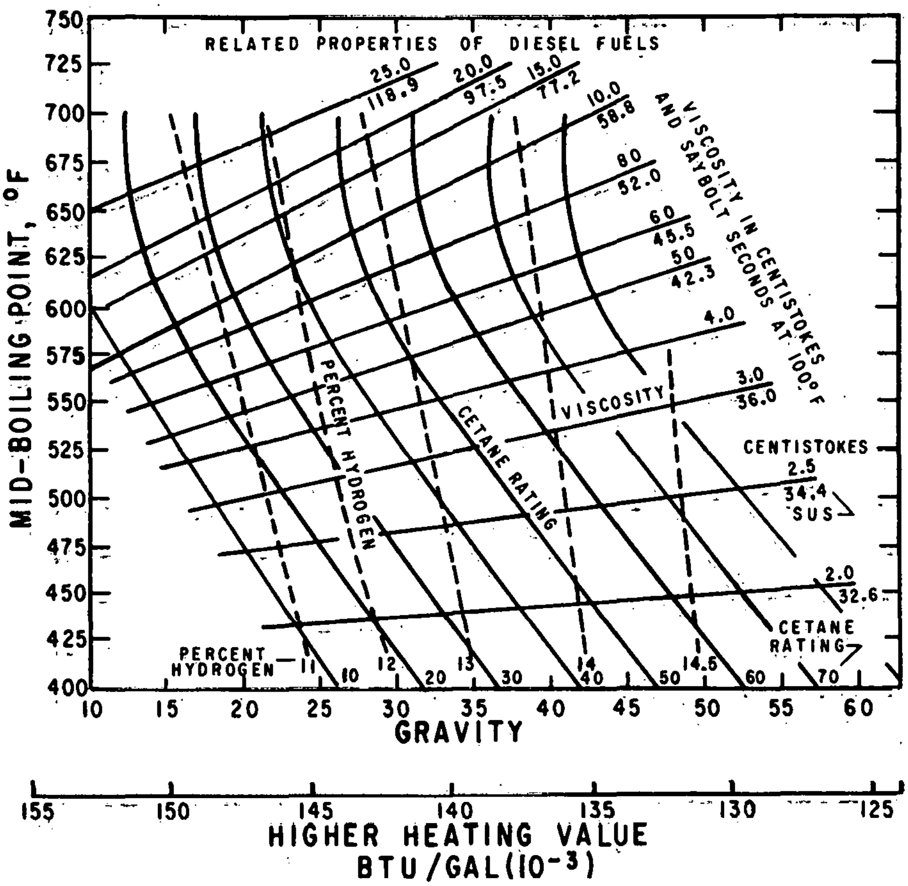

Fig. 7.1 Typical Parameters for Diesel Engine Fuel Oils (From Socony-Vacuum Oil Co.)*

*Reproduced by permission of Intex Educational Publishers, New York[27]

Conradson carbon can reveal excessive carbon which could impair injection efficiency. Depending on crude and refining process, the figure can vary from 0.1% to 18% by weight.

Sulfur and ash. Sulfur should not exceed 2% by weight to avoid excessive formation of corrosive acid. Bottom sediment and also water should be limited to 2%. Ash can be decreased to 0.01% by centrifuging.

Cetane number is determined by engine test (ASTM D613), or a Cetane index can be calculated for most fuels based on specific gravity and the midboiling point of the fuel (see Appendix II of Ref. 28). Cetane number, a measure of the ignition quality of the fuel, influences combustion roughness. The Cetane number requirements depend on engine design, size, nature of speed and load variations, and on starting and atmospheric conditions. Increase in Cetane number over values actually required does not materially improve engine performance; consequently, the Cetane number specified should be as low as possible to assure maximum fuel availability.

Diesel index number is one of three methods of specifying ignition quality proposed by the A.S.T.M. It is based on tests that indicate ignitability of an oil to vary in accordance with its aniline point and its gravity. The correlation between diesel index number and ignition quality has been found to be good only for certain types of oils; therefore, it is recommended only for quick and rough evaluation.

Higher heating value. To obtain reasonably close estimates of the higher heating value (HHV) of diesel fuel oils, the following formulas may be used[27] in which API represents the gravity of the fuel at 60°F:

HHV = 18,650 + 40(API - 10) Btu per lb for distillate fuel oil.

HHV = 18,320 + 40(API - 10) Btu per lb for heavy, cracked fuel oils.

8.1 NOISE ATTENUATION

The noise level emanating from internal combustion, reciprocating piston engines can be objectionable unless it is effectively silenced. Silencing, in simple terms, is reducing noise to an acceptable level for the location or working conditions surrounding the noise-producing equipment. Figure 8.1 illustrates typical reciprocating engine exhaust noise curves for silenced and unsilenced engines.[29]

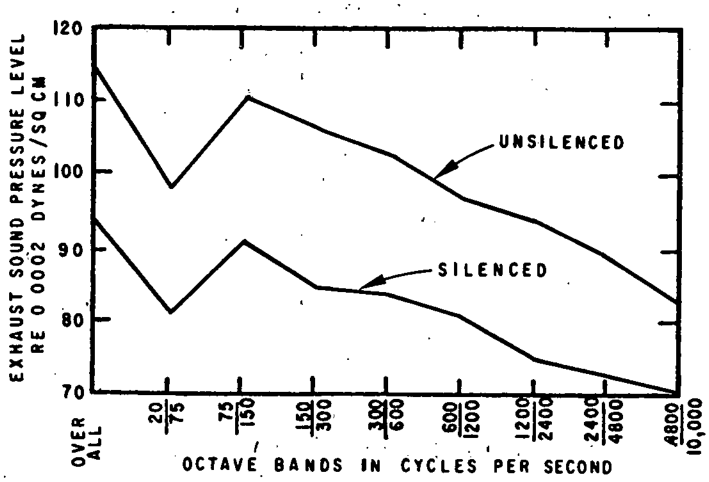

Fig. 8.1 Typical Reciprocating Engine Exhaust Noise Curve*

**Source:* *ASHRAE Guide and Data Book,* Systems[29]

Table 8.1 gives acceptable noise design criteria in decibels for various applications.[29]

Table 8.1 Noise Criteria (In Decibels) for Typical Areas[a]

Octave Bands in Cycles Per Second	37.5-75	75-150	150-300	300-600	600-1,200	1,200-2,400	2,400-4,800	4,800-10,000
Highly Critical Hospital or Residential Zone	70	49	38	35	34	33	33	33
Night, Residential	72	57	47	40	38	38	38	38
Day, Residential	75	62	52	45	43	43	43	43
Commercial	78	68	60	55	51	47	44	43
Industrial-Commercial	78	73	65	60	58	57	54	54
Industrial	85	82	76	72	70	68	66	66
Ear Damage Risk	110	102	96	94	94	94	94	94

(a) Reprinted by permission of ASHRAE.[29]

Several models of engine silencers are commercially available to reduce noise levels from engines. The cost of such units varies with the manufacturer and the size of the unit specified. Most exhaust heat recovery units also act as silencers. These units will be covered in the technology evaluation on heat recovery equipment.

Figure 8.2 compares typical attenuation curves[29] for individual silencer models and shows the relative capabilities of the different designs.

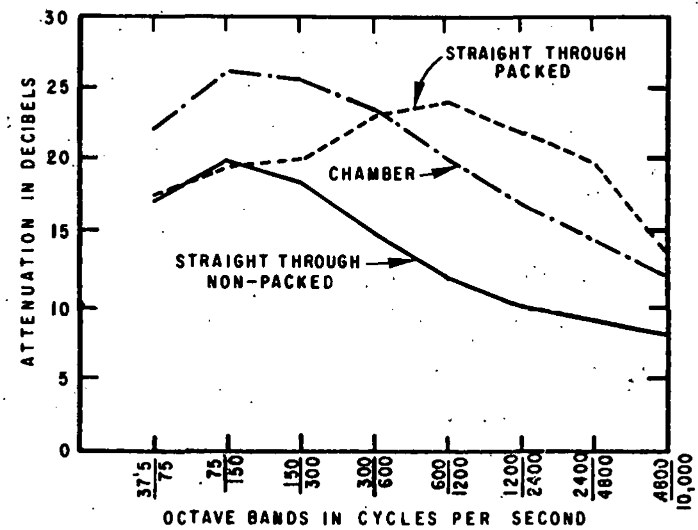

Fig. 8.2 Typical Attenuation Curves for Engine Silencers*

Source: ASHRAE Guide & Data Book, Systems[29]

8.1.1 Chamber-Type Silencers

Chamber-type silencers provide the best noise control across the entire audible range of the frequency spectrum from 63 through 8,000 Hz. The basic design incorporates nonresonant side tube arrangements to permit passage of the exhaust gases from one chamber to another. This creates a reversal of flow and develops a predictable amount of back pressure. Another important feature of chamber-type silencers is the availability of side inlet exhaust connections. Units of this type, except spark arrestor designs, can also be supplied with side outlet exhaust connections. These side connections, while having little effect on the pressure drop, will, in many instances, greatly facilitate installation of the silencer by eliminating the need for elbows in the piping systems.

8.1.2 Straight-Through Silencers

Straight-through silencers have an unobstructed passage through the silencer, with no flow reversal. The result is a pressure drop across the silencer only slightly above that caused by an equivalent length of pipe. Side connections are not practical with these silencers because such connections would increase pressure drop and cost.

The silencing capability of straight-through designs will be less than that provided by a chamber-type design, particularly in the 1,000 and greater center frequency bands.

8.2 EMISSIONS

High-compression gas engines and diesel engines always operate with an excess of air over that required for theoretically-complete combustion of the fuel. In diesel engines the fuel/air ratio, by weight, may vary from 0.005 when idling to 0.06 for full power. Complete combustion of fuel with an excess of air will result in combustion products consisting of CO_2, H_2O vapor, O_2, and N_2.

Incomplete combustion occurs in a compression-ignition engine when (1) temperature at the end of compression is too low, (2) oxygen concentration is not sufficient to permit complete combustion, and (3) fuel concentration is too low for pre-flame reactions to produce sufficient heat to promote quick and complete combustion. Exhaust gas from a diesel, or compression-ignition, engine is composed of additional products — CO, aldehydes, unburned and partly burned fuel, carbon, and nitrogen oxides.[30]

In a general way, conditions favorable for good combustion, such as heavy load, high boost pressure, and high temperature, will act to reduce unburned fuel but will tend to increase the NO_x components to the exhaust.[30] Conversely, to reduce NO_x formation, combustion temperature must be lowered and oxygen content and residence time at high temperature reduced. This can be accomplished by late injection, reduced boost pressure, water injection, and exhaust-gas recirculation, all of which, however, degrade the performance of the engine.

8.2.1 Emission Data

Emission data obtained from source tests, material balance studies, engineering estimates, etc., have been compiled by the U.S. Environmental Protection Agency[30] for use by individuals and groups responsible for conducting air pollution emission inventories.

Heavy-duty, general utility gaseous - fueled engines. Emissions from heavy-duty, gaseous-fueled internal combustion engines are reported in Table 8.2. Test data were available for nitrogen oxides and hydrocarbons only; sulfur oxides are calculated from fuel sulfur content. Nitrogen oxides have been found to be extremely dependent on an engine's work output; hence, Figure 8.3 presents the relationship between nitrogen oxide emissions and horsepower.

Table 8.2 Emission Factors for Heavy-Duty, General-Utility, Stationary Engines Using Gaseous Fuels

Emission Factor Rating: C

Pollutant	Emissions[a]			
	$lb/10^6\ ft^3$	$kg/10^6\ m^3$	lb/hr	kg/hr
Sulfur oxides[b]	0.6	9.6	-	-
Nitrogen oxides[c]	-	-	-	-
Hydrocarbons[d]	1.2	19	4.2	1.9

[a]Values for $lb/10^6\ ft^3$ ($kg/10^6\ m^3$) based on 3.37 $10^6\ ft^3/hr$ heat input.

[b]Based on an average natural gas sulfur content of 2000 $gr/10^6\ ft^3$ (4600 $g/10^6\ m^3$).

[c]See Fig. 8.3

[d]Values were given as tons/day. In converting to lb/hr, 24-hour operation was assumed.

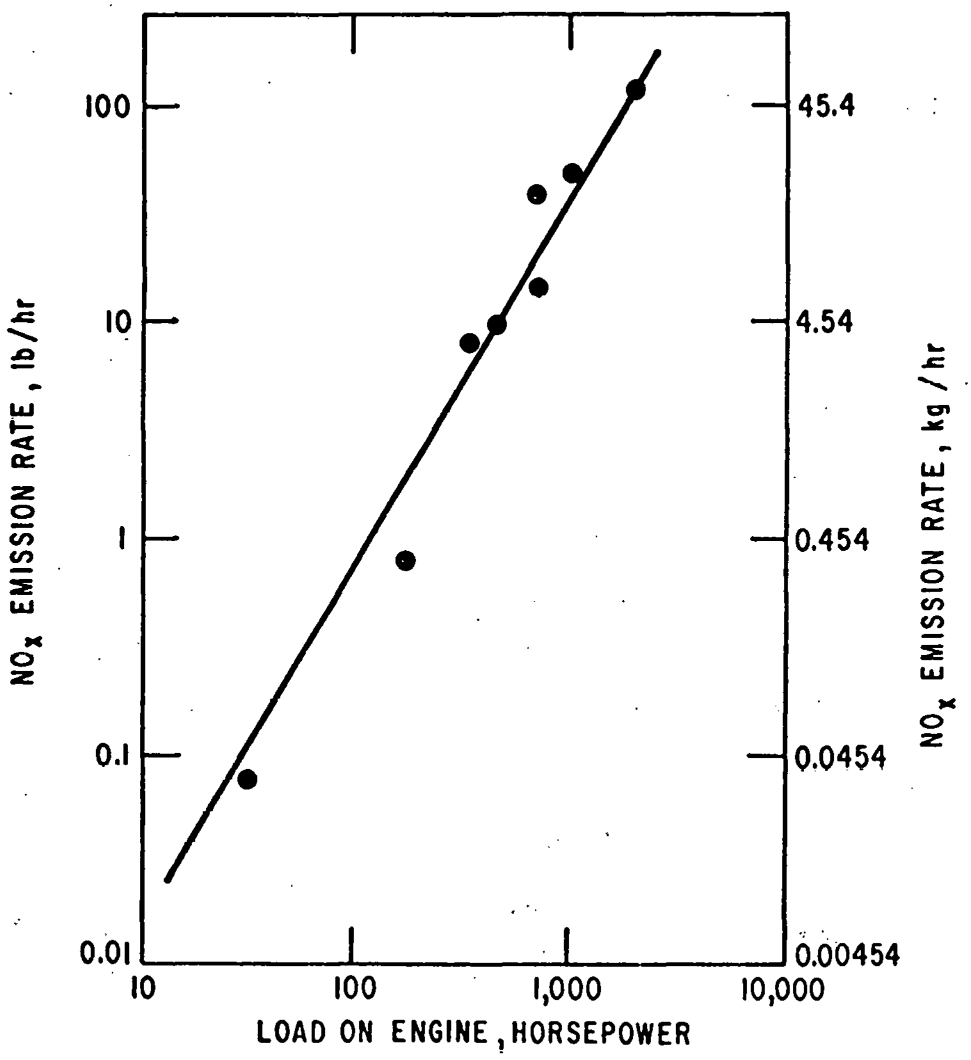

Fig. 8.3 Nitrogen Oxides Emissions from Stationary Internal Combustion Engines[30]

8.2.2 Gasoline and Diesel Industrial Engines

This engine category covers a wide variety of industrial applications of both gasoline and diesel internal combustion power plants, such as fork lift trucks, mobile refrigeration units, generators, pumps, and portable well-drilling equipment. The rated power of these engines covers a rather substantial range - from less than 15 kW to 186 kW (20 to 250 hp) for gasoline engines and from 34 kW to 447 kW (45 to 600 hp) for diesel engines. Understandably, substantial differences in both annual usage (hours per year) and engine duty cycles also exist. It was necessary, therefore, to make reasonable assumptions concerning usage in order to formulate emission factors.

Once reasonable usage and duty cycles for this category were ascertained, emission values from each of the test engines were aggregated (on the basis of nationwide engine population statistics) to arrive at the factors presented in Table 8.3. Because of their aggregate nature, data contained in this table must be applied to a population of industrial engines rather than to an individual power plant.

The best method for calculating emissions is on the basis of "brake specific" emission factors (g/kWh or lb/hphr). Emissions are calculated by taking the product of the brake specific emission factor, the usage in hours (that is, hours per year or hours per day), the power availabe (rated power), and the load factor (the power actually used divided by the power available).

8.2.3 Emission Standards

The Environmental Protection Agency has announced emission regulations for heavy-duty truck engines starting with the 1974 model year. These

Table 8.3 Emission Factors for Gasoline-and Diesel-Powered Industrial Equipment

Emission Factor Rating: C

Pollutant	Engine category[(a)]	
	Gasoline	Diesel
Carbon monoxide		
g/hr	5700.	197.
lb/hr	12.6	0.434
g/kWh	267.	4.06
g/hphr	199.	3.03
$kg/10^3$ liter	472.	12.2
$lb/10^3$ gal	2940.	102.
Exhaust hydrocarbons		
g/hr	191.	72.8
lb/hr	0.421	0.160
g/kWh	8.95	1.50
g/hphr	6.68	1.12
$kg/10^3$ liter	15.8	4.49
$lb/10^3$ gal	132.	37.5
Evaporative hydrocarbons		
g/hr	62.0	-
lb/hr	0.137	-
Crankcase hydrocarbons		
g/hr	38.3	-
lb/hr	0.084	-
Nitrogen oxides		
g/hr	148.	910.
lb/hr	0.326	2.01
g/kWh	6.92	18.8
g/hphr	5.16	14.0
$kg/10^3$ liter	12.2	56.2
$lb/10^3$ gal	102.	469.
Aldehydes		
g/hr	6.33	13.7
lb/hr	0.014	0.030
g/kWh	0.30	0.28
g/hphr	0.22	0.21
$kg/10^3$ liter	0.522	0.84
$lb/10^3$ gal	4.36	7.04
Sulfur oxides		
g/hr	7.67	60.5
lb/hr	0.017	0.133
g/kWh	0.359	1.25
g/hphr	0.268	0.931
$kg/10^3$ liter	0.636	3.74
$lb/10^3$ gal	5.31	31.2
Particulate		
g/hr	9.33	65.0
lb/hr	0.021	0.143
gk/Wh	0.439	1.34
g/hphr	0.327	1.00
$kg/10^3$ liter	0.775	4.01
$lb/10^3$ gal	6.47	33.5

[a] The engines used to determine the results in this table cover a wide range of uses and power. The listed values do not, however, necessarily apply to some very large stationary diesel engines.

regulations correspond closely to the California Air Resourced Board standards as given in Table 8.4. The EPA standards also specify a smoke limit that may be tightened in the future and made applicable to other classes of diesel engines.

Table 8.4 California Air Resources Board Emission Standards for 1973 to 1975 Gasoline and Diesel Vehicles Over 6001 lb Gross Weight, with Comparative Values for 1971 Engines.

	Hydrocarbons plus Nitrogen Oxides (as NO_2)	Carbon Monoxide
1971 Gasoline	30-40 grams/bhp-hr	150 grams/bhp-hr
1971 Diesel	6-14 grams/bhp-hr	10-20 grams/bhp-hr
1973-74 Standards	16 grams/bhp-hr	40 grams/bhp-hr
1975 Standards	5 grams/bhp-hr	25 grams/bhp-hr

Smoke limit.[7] As fuel/air ratio is increased above idling, to increase power output, causing higher temperatures, percentages of CO and aldehydes decrease rapidly. When the fuel/air ratio gets over 0.05, there is a tendency to form locally over-rich regions resulting in an increase in CO and smoke. This smoke is unburned carbon, formed by thermal decomposition, which did not find oxygen to complete combustion.

Diesel engines are usually rated at the brake horsepower developed at the smoke limit. At any definite engine speed, a certain amount of air enters the cylinder. This amount of air is sufficient to complete combustion of a certain quantity of fuel, depending upon the amount of turbulence present in the cylinder, the injection system, and design of the combustion chamber. If more fuel is injected, the output of the engine will be beyond the rated horsepower, or smoke limit. There will not be sufficient air present to burn all of the fuel and unburned fuel will be seen as smoke in the exhaust.

Federal standards have been established for *exhaust smoke* from high-speed heavy-duty diesel engines beginning with 1972 model year, – *"The opacity of smoke shall not exceed 40% during engine acceleration mode, or 20% during engine lugging mode"* when tested under specified conditions for idling, acceleration, and lugging modes.[7]

9.1 ESTIMATED CAPITAL COST

The 1976 uninstalled capital cost of large oil-fired diesel engines has been estimated to range from about $714,000 for a 5,416 rated Bhp engine, to about $1,526,000 for a 13,540 rated Bhp engine.[13] These costs include all components normally considered as standard equipment with the engine. They do not include the price of an electric generator.

Several techniques that correlate the cost of equipment with capacity have been published,[31] but the most frequently used procedure involves plotting, on log-log coordinates, the available cost values as a function of the capacity factor (Fig. 9.1). In most cases, these data

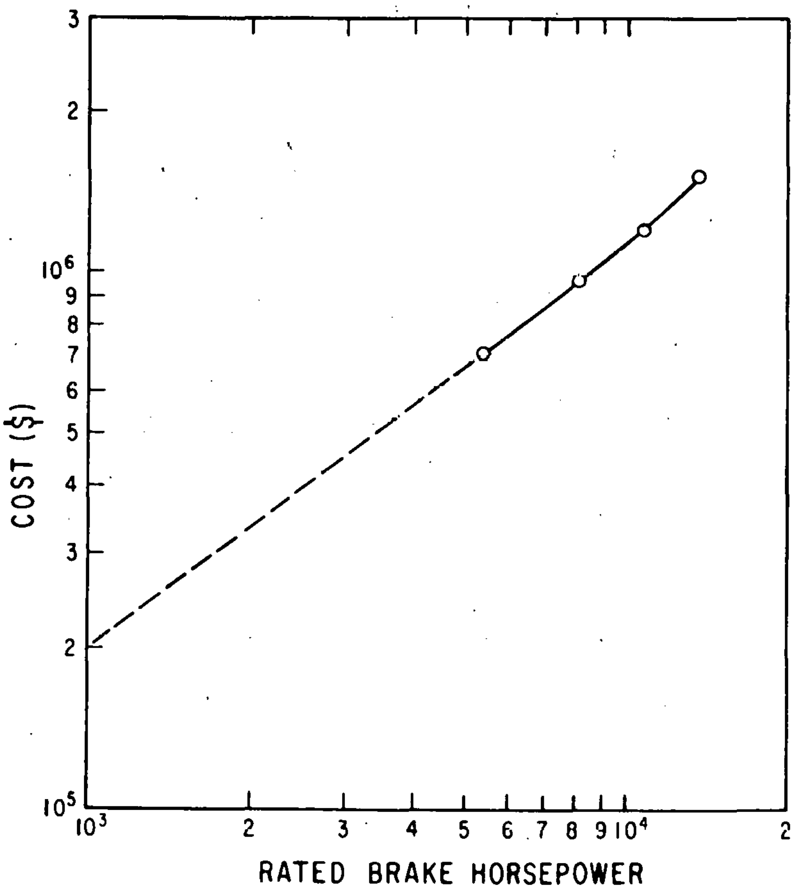

Fig. 9.1 1976 Uninstalled Purchase Price of Large Reciprocating Piston Engines

may be adequately correlated by a straight line and can be represented by the following analytical expression:

$$C_D = C_B \times \left(\frac{Q_D}{Q_B} \right)^p \qquad \text{(Eq.9.1)}$$

where: C_D = cost, desired capacity (dollars)

C_B = cost, base or nominal capacity ($714,000)

Q_D = desired capacity (Bhp)

Q_B = base or nominal capacity (5,416 Bhp)

p = exponent

The value of the exponent, p, in a power function that approximates the uninstalled purchase price of large internal combustion piston engines as a function of brake horsepower is p = 0.829. (See Fig. 9.1) However, the practice of some manufacturers of derating their engines when used in conjunction with heat-recovery applications can influence capital costs significantly on a dollar per rated Bhp basis.

The costs of internal combustion piston engine installations will vary with the engine type and with the speed of operation. The difference in the cost of diesel and gas engine installation lies mainly in the diesel fuel storage facilities, which include the storage tanks, transfer pumps, feed pumps, etc. for the fuel oil. The cost of dual-fuel engine installation is somewhat higher than a comparable gas engine because of the need for a separate fuel (and storage) system to supply the pilot charge. However, the purchase cost of the engines alone should be about the same for all types in a comparable capacity and speed range. These engines are for continuous duty and have an estimated service life of 20 to 30 years.

9.2 DELIVERY AND INSTALLATION

The delivery of the engine will be determined, in part, by the distance and method of transportation from the manufacturer to the project site. When the engine is received at the project site, it will be necessary to unload, inspect, move to the location of the installation, install, and perform any necessary cleanup work. The direct labor manhours required to install a diesel electric system are estimated[32] to be approximately 0.34 manhours per brake horsepower, assuming the moving distance will not exceed 200 ft and that site or other obstructions are negligible.

9.3 OPERATING COSTS

An ASME 1974 report[33] on diesel and gas engines power costs presents information on operating costs of 91 oil-diesel, dual fuel-diesel, and gas-engine power plants located within the United States. The engines listed in this report are vertical type, and all are direct-connected to generators. Operating costs, as relating to plants, are defined as consisting of the following items, *excluding fuel costs:* lubrication cost, labor cost for surveillance, cost of miscellaneous supplies, cost of maintenance and repairs, and cost of insurance as a separate item.

Approximate yearly internal combustion engine operating costs, as a function of rated brake horsepower, are shown in Fig. 9.2. The data are

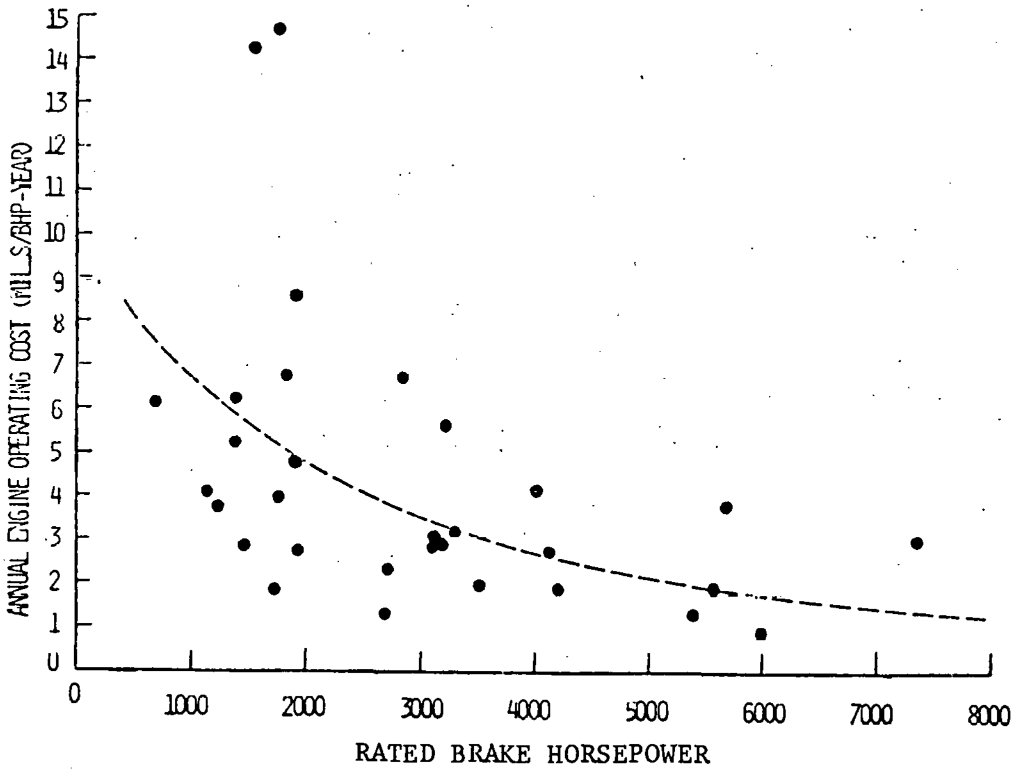

Fig. 9.2 Approximate Internal Combustion Engine Operating Cost
Source: ASME 1974 Report.

widely scattered but indicate a trend to higher production costs as the rated engine power decreases. The dashed line is an approximation of an average production cost, which can be represented by the empirical Eq. 2.3 using the following coefficients:

$$A = 10.644$$
$$B = -4.031 \times 10^{-3}$$
$$C = 6.659 \times 10^{-7}$$
$$D = -3.870 \times 10^{-11}$$

9.4 MAINTENANCE PRACTICES AND COSTS

Maintenance is necessary to provide continuous and economical engine operation. Of the two basic approaches to maintenance procedures, one is to retain a staff of trained mechanics to do all the maintenance of the power generating equipment, buying only the materials needed to keep the equipment in operable condition; the other is to purchase a full maintenance contract from an outside service in which, on a periodic basis, trained servicemen inspect all equipment and perform the required adjustments and overhauls.

The most frequent causes of failures have been analyzed in several types of IC engines, including liquid-fueled and dual-fueled diesel engines and spark-ignited gas engines.[19] These units drove electrical generators or centrifugal gas compressors, and they were of both two-cycle and four-cycle design and of low and medium speed. Diesel engines of different makes in continuous service appeared to perform equally well. On the average, availability was found to be 95-96%. The meantime between failures (MTBFs) was found to be 500-700 hr, a rate of about one forced outage per month.

The 96% availability factor is considered representative of diesel generators in continuous duty. Of the 4% average downtime observed, about 1% was attributed to forced outages (failure) and 3% to scheduled maintenance procedures.

Availability achieved in practice was seen to depend more often on the capability of the operating crew and availability of spare parts than on the inherent reliability of the engine and auxiliary systems.

Maintenance costs are composed of three basic items:

(1) *Miscellaneous maintenance and service costs*, including service manual recommendations plus makeup oil (excluding labor to perform this routine duty);

(2) *overhaul maintenance costs*, including all labor and parts necessary to perform major and minor overhauls at the recommended intervals;

(3) *labor costs* necessary to perform the miscellaneous service for Item (1).

Because of the many variables involved, 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.

Cost of engine maintenance in dollars per engine horsepower have been reported for a total of 91 different internal combustion engine plant types.[33] Some representative engine maintenance costs (1972 data), as a function of engine-rated power,[33] are shown in Fig. 9.3. Although the data

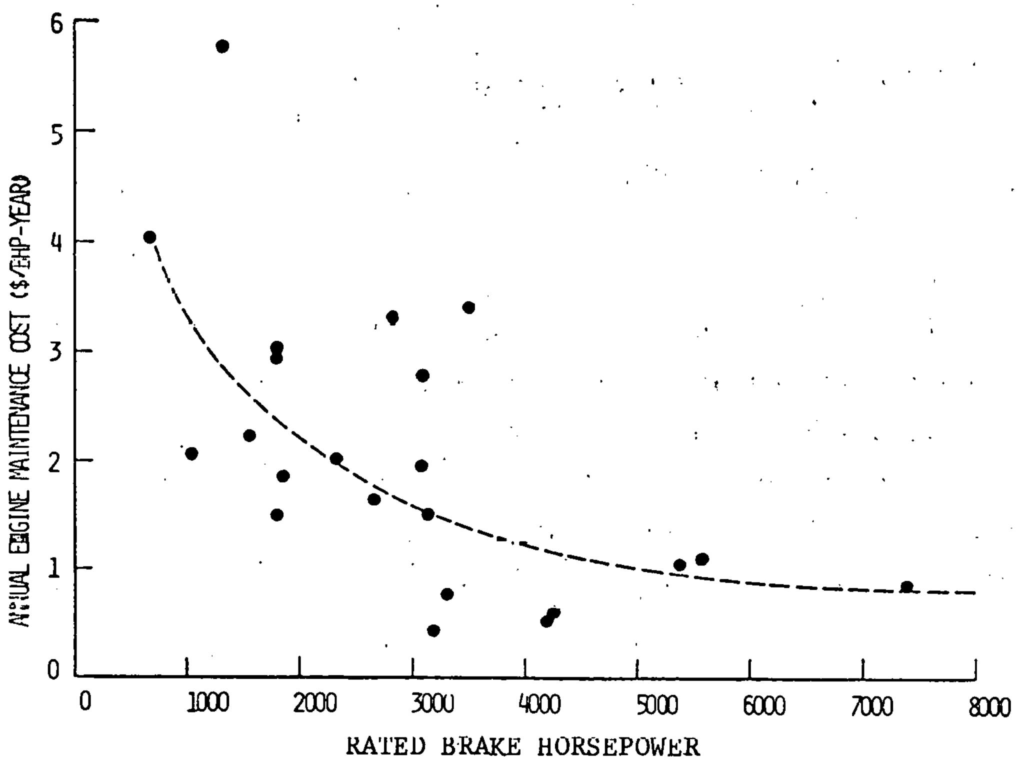

Fig. 9.3 Cost of Engine Maintenance as a Function of Engine Rated Power*

*1972 data

are widely scattered, Fig. 9.3 indicates that the cost of engine maintenance increases as the brake horsepower rating of the engine decreases. The dashed line represents an average engine maintenance cost from which a mathematical model can be expressed by use of the generalized empirical equation 2.3. The following coefficients are appropriate for use in the equation to model engine maintenance as a function of engine power, where the variable Y in Eq. 2.3 is annual cost in \$/Bhp and X is the rated engine brake horsepower:

$A = 4.9633$

$B = -1.9709 \times 10^{-3}$

$C = 3.2972 \times 10^{-7}$

$D = -1.8839 \times 10^{-11}$

10 STATUS OF DEVELOPMENT AND POTENTIAL FOR IMPROVEMENT

The major development effort on internal combustion engines over the past 30 years has been primarily devoted to engine types used in railroad locomotives, small ships, and stationary onsite power plants. The dual objectives of the development efforts have been to increase the power output -- both total and specific -- and to improve efficiency and reliability while minimizing maintenance problems and costs. These developments have resulted in improvements in fuel consumption to where large modern diesel engines have specific fuel consumptions under 0.35 lb/Bhp-h, and values as low as 0.32 lb/Bhp-h are guaranteed in some cases. These figures correspond to a heat rate on the order of 6,050 Btu/Bhp-h or a thermal efficiency of up to 42%.

The following future development possibilities are proposed[34] as potential areas for the improvement of internal combustion engines:

(1) increasing output by an increase in engine speed resulting from the availability of better lubricants and materials, and through employment of more sophisticated design methods.

(2) increasing output by raising Bmep (brake mean effective pressure) values. Experimental 4-stroke/cycle engines have been operated successfully with Bmep values exceeding 400 psi.

(3) increasing the amount of air available for combustion through improvements in supercharging devices and improved air intercoolers.

(4) improving fuel injector design (for diesel engines) to promote better mixing of air flow and fuel spray patterns to handle the increased fuel flow required at higher engine ratings. Injection pressures that are currently about 12,000 psi probably will be increased soon to 20,000 psi or higher.

It is unlikely that any dramatic increase in the efficiency of coversion of fuel energy to mechanical energy will occur. Specific fuel consumption of the best current production engines is very good, although an increase in output of an engine of a given displacement probably will yield a slight improvement. The development possibilities are expected to result in a 20-25% increase in engine output with no serious increase in weight or loss in economy, reliability, and life predicated on a simple extension of current development trends.

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