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HANDBOOK OF DATA ON SELECTED ENGINE COMPONENTS FOR SOLAR THERMAL APPLICATIONS

National Aeronautics and Space Administration
Lewis Research Center

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U.S. DEPARTMENT OF ENERGY
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NOTICE

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Washington, D. C. 20545
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CONTENTS

1.0 INTRODUCTION

by Thaddeus S. Mroz

Up-to-date and specific component data are needed to realistically predict the performance and cost of projected solar thermal power-conversion systems in planning and design studies. Information is needed to determine if new components must be developed for efficient systems or if the design and performance characteristics of developed and commercially available components adequately meet the system design requirements. Information is also needed on current component development in industry, including performance improvements, product upgrading of existing component designs, and cost and size reductions.

This handbook provides a data base on developed and commercially available power-conversion-system components for Rankine- and Brayton-cycle engines, which have potential application to solar thermal power-generating systems. The status of the Stirling engine is also discussed. Although this prime mover is still in the development stage, it is a viable candidate for this application.

The handbook is not intended as a vehicle for comparing or evaluating manufacturers, their products, or their pricing structures. The information presented is based on data received from the manufacturers. The data encompass a broad power range of 5 to 50 000 kilowatts electric and the principal power-conversion-system components of the organic Rankine, the steam Rankine, the Brayton, and the Stirling cycles. (These cycles are described thermodynamically in appendix A.) The components discussed are the prime movers (steam and organic turbines, gas turbines, reciprocating expansion engines, and the Stirling engine), pumps, condensers, speed reducers-increasers, and alternating-current generators. These components are needed for the conversion of solar thermal power to electric power. The solar boiler-receiver, the controls, and auxiliary equipment external to the power-conversion components are not discussed.

For the purpose of the handbook, the overall power range (5 to 50 000 kWe) was divided into three subranges: low power (5 to 100 kWe), medium power (100 to 1000 kWe), and high power (1000 to 50 000 kWe). Specific base power levels were selected for each subrange. Components were then selected and sized, depending on component availability, to these power levels for each power conversion system.

Component data sheets were prepared for each type of prime mover. These sheets provide the criteria to select and size components and to identify the required technical and cost data. The required technical data include component flow parameters, temperature, pressure, speed, and efficiency. Cost data are based on single- and multiple-unit procurements, as available from the manufacturers. The data sheets and the selection criteria were submitted to the component manufacturers, and they were asked to supply information for the engine handbook on existing components and on their current in-house component development programs. Technical and cost data presented in this handbook are not identified with any particular manufacturer but are included in a range for a particular component and power level. The handbook does list the component manufacturers with the type and power range of the components that they manufacture in appendix B.

Although not all manufacturers may be included in this survey, it was not the intent to exclude any manufacturer with applicable components. Those manufacturers wishing to include their component data in future revisions of this handbook can do so by submitting their information on the sample data sheets in appendix C.

The handbook is structured into eight sections. Solar thermal power-conversion systems are described in section 2.0. Typical applications, operating characteristics, and reliability requirements are discussed, including the operating constraints resulting from solar input characteristics. Component descriptions and the technical and cost data received from the manufacturers are given in sections 3.0 to 7.0. Design, operating characteristics, availability, production status, cost, and experience factors are discussed. Component data are presented in tables and graphs after each section. The component technical and cost data presented in the handbook are summarized in section 8.0.

Much of the information received from the manufacturers was classed as sensitive data (e.g., cost, efficiency, steam flow, etc.) and could not be shown individually as plot points or identified in tabular form. For this reason the data were presented collectively as a range for the particular parameters.

This handbook documents the results of a survey performed by the NASA Lewis Research Center for the Department of Energy (DOE), Division of Central Solar Technology. This survey was conducted under Interagency Agreement EX-76-A-29-1060. In addition to the authors given for each section, contributions were made by the following people: Lloyd W. Ream and Thomas P. Moffitt - Rankine cycle; Richard W. Niedzwiecki, Thomas P. Moffitt, Arthur J. Glassman, and Calvin L. Ball - Brayton cycle; Charles S. Corcoran, Jr., and Richard R. Secunde - alternating-current generators; and Robert E. Hyland and Lloyd S. Shure - overall.

2.0 SOLAR THERMAL POWER-CONVERSION SYSTEM

by Harvey S. Bloomfield

2.1 DESCRIPTION

A solar thermal power-conversion system consists of a heat engine that converts solar energy to usable mechanical or electric power, a collector-concentrator (reflecting mirror), and a receiver (solar boiler).[1] A generator or alternator can be added for electric power output. The receiver converts concentrated solar radiation to thermal energy by using a working fluid (liquid or gas). The heat engine converts the thermal energy in the working fluid to shaft power by using either a Rankine, Brayton, or Stirling cycle to extract the energy.

Several methods are used to collect and concentrate solar energy and convert it to mechanical or electric energy. One method, called the central-power concept, involves collecting the solar energy in a large field of flat mirrors and concentrating it on a single large receiver (boiler). A second method, called the distributed-power concept, involves collecting the solar energy on a single curved mirror with either a linear focus (trough collector) or a point focus (dish collector) and converting it to thermal energy in a receiver located at the focal point of the collector. In this method, the thermal energy can be converted to electric energy by an engine connected to each dish or trough collector, or it can be sent to a central point for conversion to electric power. Solar thermal electric power systems are more completely described in reference 1.

Both the central-power system and the distributed-power system collect the direct (rather than the diffuse) component of sunlight and use it to heat a working fluid, thereby converting solar energy to thermal energy. In the central-power system (fig. 2-1) a large field of dual-axis tracking mirrors (heliostats) intercepts and redirects solar radiation toward a single large receiver mounted atop a tower. The redirected solar radiation strikes the receiver and heats an internally circulating working fluid. Working fluids under consideration for this type of system include high-pressure water, superheated steam, hydrocarbon oils, molten salts, liquid metals, and various gases (including air).

Unlike the central-power system, which focuses incident radiation on a single receiver, the distributed-power system (fig. 2-2) concentrates sunlight and converts it to heat and/or electric power at multiple collector modules. The collector module consists of a cylindrical or circular, tracking reflecting mirror; a receiver; and either a heat engine or an energy-transport loop connected to a central power-conversion system. The mirrors can be either troughs or dishes. Troughs are single-axis, cylindrical mirrors that provide a linear focus. Dishes are circular, paraboloidal, two-axis

[1]Heat exchanger.

tracking reflectors that provide a point focus. The mirror redirects and concentrates the Sun's rays onto a receiver located at the focus of each module. The working fluid is circulated through the receiver and into the heat engine.

At present, three types of thermodynamic cycle are being investigated for the heat engine: Rankine, Brayton, and Stirling. A Rankine cycle uses vaporized two-phase fluids, such as superheated steam, to power a turbine or a reciprocating engine. The vapor used in the engine either can be heated in boiler-superheater tubes within the receiver or can extract energy from the working fluid in an intermediate heat exchanger. The Brayton and Stirling cycles use gaseous working fluids. The Rankine cycle uses steam and organic working fluids. The Rankine cycle and the open-cycle Brayton engines are being investigated for solar thermal power-conversion applications similar to utility applications. Also under consideration is a closed-cycle Brayton engine that can use monatomic inert gases as well as air and nitrogen as the working fluid and a Stirling-cycle engine that can use helium or hydrogen as the working fluid.

Like conventional Rankine-cycle powerplants, solar thermal powerplants would use condensers and cooling towers to remove and reject excess turbine heat. Either wet or dry cooling could be used. For more efficient operation, thermal storage is required for most solar thermal power-conversion systems. In solar total-energy systems, the waste heat from the heat engine is stored in a low-temperature storage unit - an insulated tank containing various media. The waste heat heats a working fluid, usually water, which is pumped through a network of pipes to individual buildings where it is then used for heating, cooling, and hot water. Solar total-energy systems are described more completely in reference 2. A high-temperature storage unit, similar to the low-temperature unit in solar total-energy systems, may be required in solar thermal central-receiver powerplants. Thermal energy generated in excess of immediate requirements can be stored in the unit. During periods when direct solar radiation is unavailable, the storage unit can be tapped for necessary energy. Energy storage technology may employ the sensible-heat-transfer and/or latent-heat-of-fusion properties of various storage media - including rock, hydrocarbon oils, and eutectic salts. A conventionally fueled auxiliary power system may also be required to ensure power availability in the event of solar power system outage due to excessive cloud cover or inclement weather. This auxiliary system may be located on site and integrated with the solar powerplant. This integration may involve mixing solar energy and fossil energy before they enter the engine. These hybrid solar - fossil fuel systems are being investigated for both new and repowered applications.

In summary, solar thermal power-conversion systems use solar radiation to heat a working fluid to a temperature high enough to operate a power-conversion system. Mechanical output can be used to drive an electric power generator. Figure 2-3 schematically diagrams a basic solar thermal power system, including those features unique to total-energy systems.

2.2 APPLICATIONS

Three major applications of solar thermal power-conversion systems are being considered:

(1) Solar powerplants in small- to medium-scale dispersed-power applications or integrated into large-scale utility applications

(2) Solar total-energy systems that provide both electric and thermal power

(3) Solar irrigation pumping systems that use power output to operate the pumps by generator drive or direct pump drive

These applications have been subdivided into central or dispersed power according to the output power level required.

2.2.1 Central Power Applications

Central power applications involve solar powerplants of relatively large capacity (10 MWe or greater) that are designed for use in utility networks. Three potential utility applications have been identified:

(1) Solar storage systems: These solar powerplants are characterized by the use of energy-storage subsystems. The operations being developed are a baseline system (a central-power-system powerplant using steam conversion), near-term alternatives to the baseline system (e.g., central-power systems using a sodium or salt heat receiver with a steam Rankine cycle), and advanced systems for longer-term development (e.g., central- or distributed-power systems using the high-temperature Brayton engines).

(2) Solar - fossil fuel hybrid systems: This application involves the combined use of solar thermal and fossil-fuel technology in new powerplants.

(3) Repowering of existing fossil-fueled powerplants: This application involves the retrofitting of existing fossil-fueled powerplants with solar thermal technology.

These central power applications are based on solar thermal systems that use a field of dual-axis flat tracking mirrors (heliostats) to redirect sunlight to a heat receiver mounted on a centrally located tower. Systems currently being developed by DOE through its field centers can produce temperatures greater than 1000° F - suitable for heat engines operating on Rankine, Stirling, or Brayton cycles. Current contract activities are compiled in reference 3.

2.2.2 Dispersed Power Applications

Dispersed-power applications generally involve solar powerplants of lesser capacity than those used in central power applications and are characterized by close geographic proximity to the point of energy use. Three systems have been identified for dispersed power applications:

(1) Solar total-energy systems: These systems are designed for "cascading" energy use (i.e., the thermal energy rejected from the electricity-generating subsystem is used to provide either space heating and cooling and domestic water heating or process heat as required in some industrial applications. Applications in the 0.5- to 10-megawatt-electric range are likely.

(2) Small solar power systems: These solar powerplants will be in the 1- to 5-megawatt-electric range and will probably be used in municipally owned power systems. Both point-focusing distributed-power systems and small central-power systems are being considered.

(3) Irrigation pumping systems: This application involves the replacement of fossil-fueled mechanical irrigation pumps with small (under 500 kWe) linear- or point-focusing solar-powered systems. Electrically powered pumps are longer range candidates for replacement.

These dispersed-power applications are based on solar thermal systems that use a field of single-axis, cylindrical trough or dual-axis parabolic dish collectors that concentrate sunlight on a linear (trough) or point (dish) heat receiver mounted at the focus of the collector. Linear-focus systems currently being developed by DOE through its field centers can produce temperatures of about 600° F - suitable for heat engines operating on a Rankine cycle (ref. 3). Point-focus systems currently being developed by DOE through its field centers can produce temperatures of over 1000° F - suitable for heat engines operating on Rankine, Stirling, or Brayton cycles.

For any solar thermal power application, knowledge of the amount and nature of the solar energy available at the application site is required. The direct, normal component of available solar energy, which is the quantity of interest for most power applications, has been tabulated in reference 4. Additional solar and climatic data are given in reference 5.

2.3 OPERATING CONCEPTS

The intermittent and limited availability of sunlight poses an operating constraint on solar thermal system design. Three operating concepts have been identified that will provide continuous and reliable power during solar outages.

National Aeronautics and
Space Administration

Lewis Research Center
Cleveland, Ohio
44135

NASA

Need Foster Miller "Linear Stirling" - 159588

Reply to Attn of: 4521

June 10, 1980

Dr. Douglas Elliott
San Francisco Operations Office
1333 Broadway
Oakland, CA 94612

Dear Doug:

Enclosed are copies of each of the reports that we at NASA-LeRC on the Solar Thermal Power group have either produced or were contract managers for. The copy of the final report by United Stirling of Sweden is due at LeRC sometime this month. At that time, I will send you a copy of their report.

The following reports are enclosed for your convenience:

179027 1. DOE/NASA/1060-78/1, Handbook of Data on Selected Engine Components for Solar Thermal Applications

159590 2. DOE/NASA/0062-79/1, 15 KWe (Nominal) Solar Thermal Electric Power Conversion Concept Definition Study - Steam Rankine Reheat Reciprocator System

159591 3. DOE/NASA/0063-79/1, 15 KWe (Nominal) Solar Thermal Electric Power Conversion Concept Definition Study - Steam Rankine Reciprocator System

159592 4. DOE/NASA/0069-79/1, Concept Definition Study of Small Brayton Cycle Engines for Dispersed Solar Electric Power Systems

159589 5. DOE/NASA/0061-79/1, 15 KWe (Nominal) Solar Thermal Electric Power Conversion Concept Definition Study - Steam Rankine Turbine System

159587 6. DOE/NASA/0056-79/1, Design Study of a 15 KW Free Piston Stirling Engine - Linear Alternator for Dispersed Solar Electric Power Systems.

Sincerely,

Robert E. Hyland

Robert E. Hyland

6 enclosures

2.3.1 Thermal Storage

An additional solar-collector field is required to charge a thermal-storage subsystem during sunlight periods. This charged subsystem then provides thermal input to the selected heat engine during solar outages. Ideally, the storage subsystem should provide input at the same temperature and enthalpy as did the solar heat receiver. A schematic diagram of a current-technology steam-Rankine-cycle solar thermal power system using thermal storage is shown in figure 2-4. The design operating turbine-inlet temperatures cannot be attained with current thermal-storage technology. Therefore, a second, lower temperature turbine port is required for energy from storage and results in lower power output during operation with stored heat. This constraint can be eliminated by developing advanced thermal-storage concepts to operate at design conditions or by using fossil-fuel energy (a hybrid concept) during solar outages. The reliability of all power systems must be comparable to that of conventional central power stations before widespread commercialization can be realized.

2.3.2 Parallel Solar - Fossil Fuel Hybrid Concept

The parallel solar - fossil fuel hybrid concept functions like a thermal-storage subsystem, but an additional solar collector field is not required. During solar outages, fossil-fuel energy is used to provide thermal input to a heat engine. The use of fossil fuel as a "storage media" places no constraint on the temperatures and enthalpy of the working fluid because of the very high combustion temperatures of fossil fuel. However, during periods of operation with only sunlight the peak cycle temperature is constrained to the limits of the solar heat receiver. A schematic diagram of a steam-Rankine-cycle solar thermal power system using the parallel solar - fossil fuel hybrid concept is shown in figure 2-5.

2.3.3 Series Solar - Fossil Fuel Hybrid Concept

The series solar - fossil fuel hybrid concept eliminates the peak-cycle-temperature constraint of the parallel-hybrid and thermal-storage concepts by using fossil fuel to augment, or boost, solar receiver outlet temperatures. Therefore, heat engines with higher peak cycle temperatures (the Brayton cycle or the combined Brayton-Rankine cycle) can be used. This series-hybrid concept also satisfies the basic storage requirements for solar outage. A schematic diagram of a simple Brayton-cycle solar thermal power system using the series solar - fossil fuel hybrid concept is shown in figure 2-6. In addition, any fuel, including residual oils and coal-derived synthetics, can be used in the indirectly fired combustors.

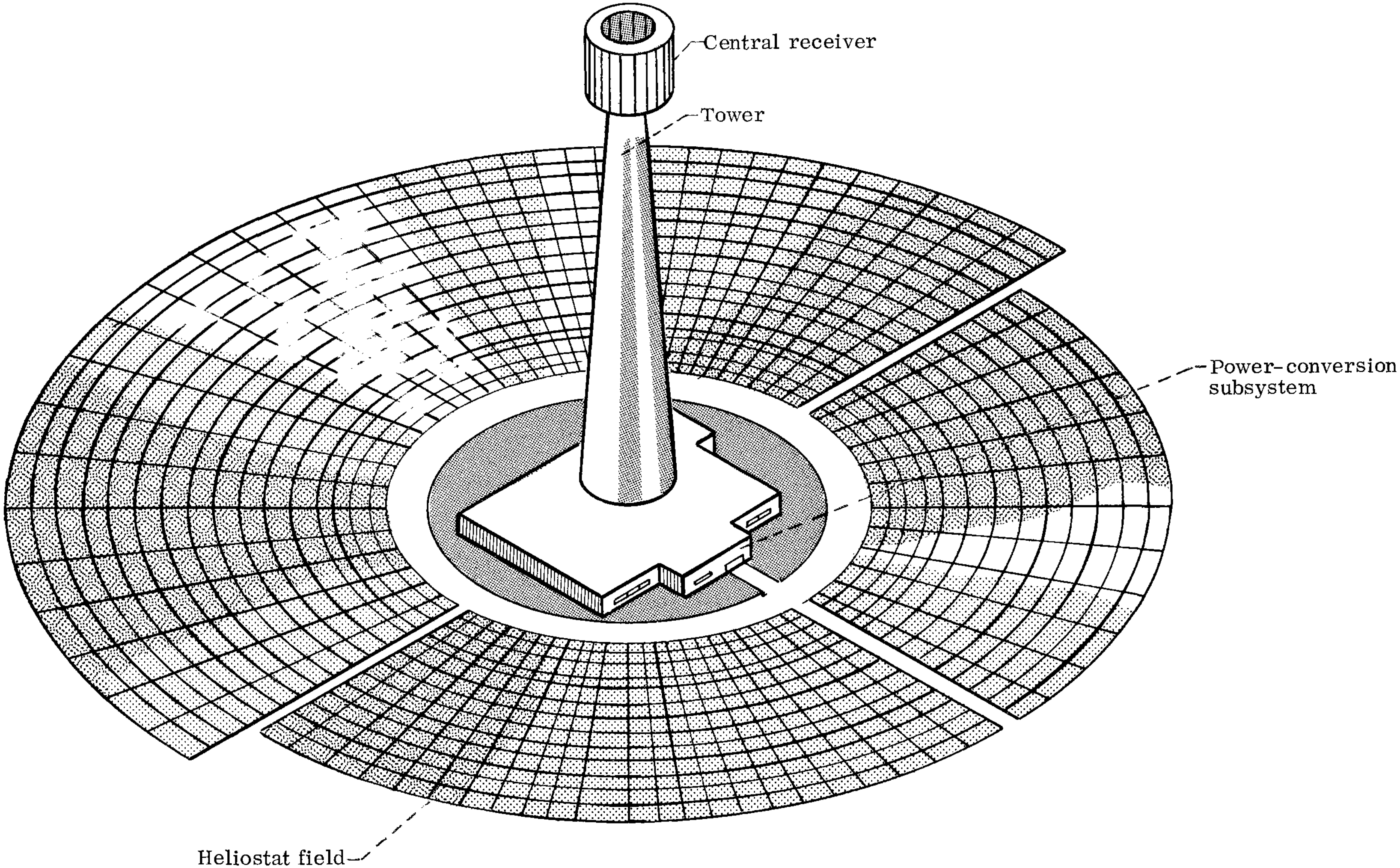


Figure 2-1. - Central-power solar thermal power-conversion system.

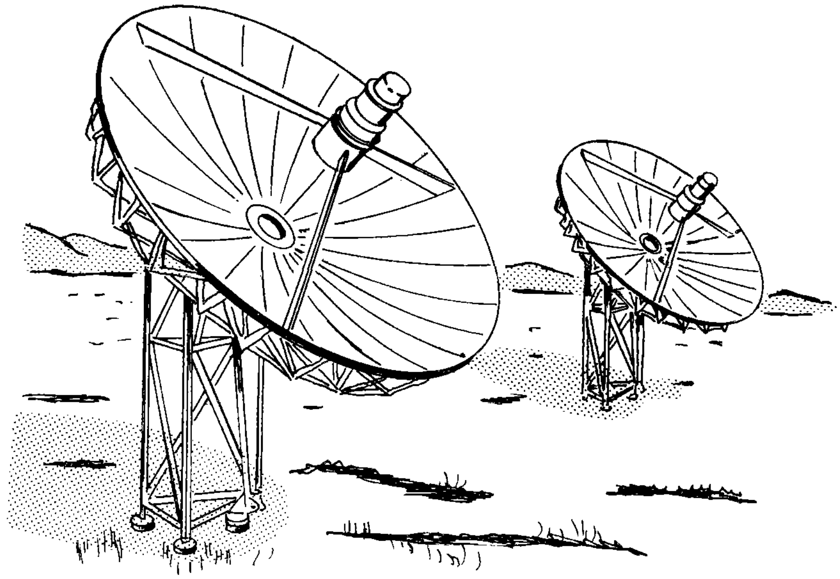


Figure 2-2. - Dispersed-power solar thermal power-conversion systems.

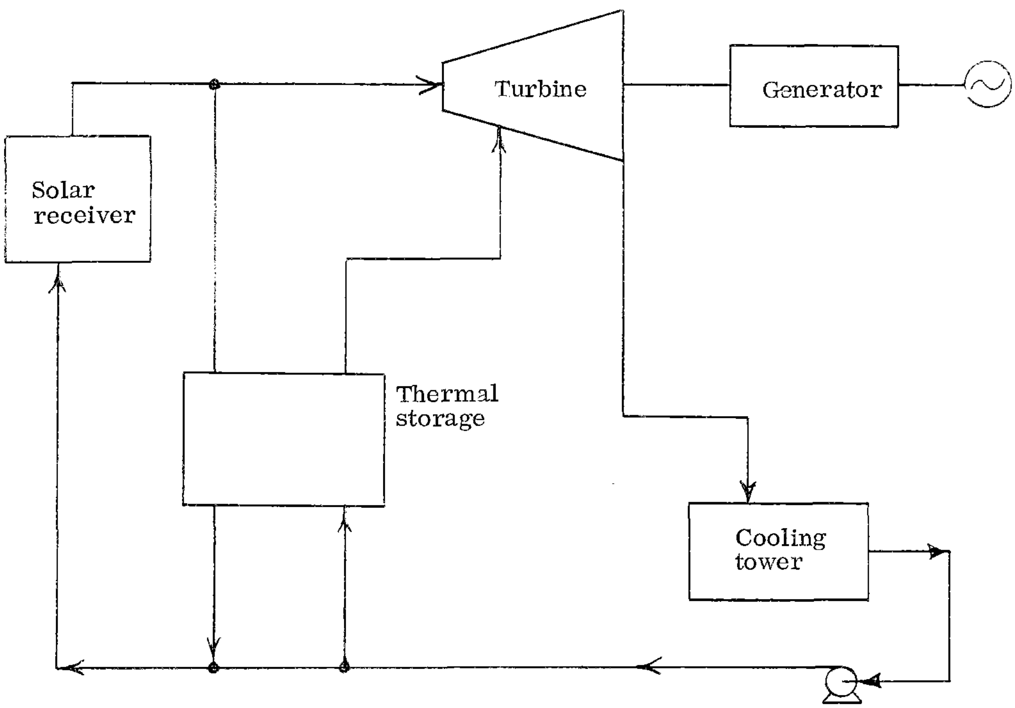


Figure 2-3. - Schematic of a basic solar thermal power-conversion system with thermal storage.

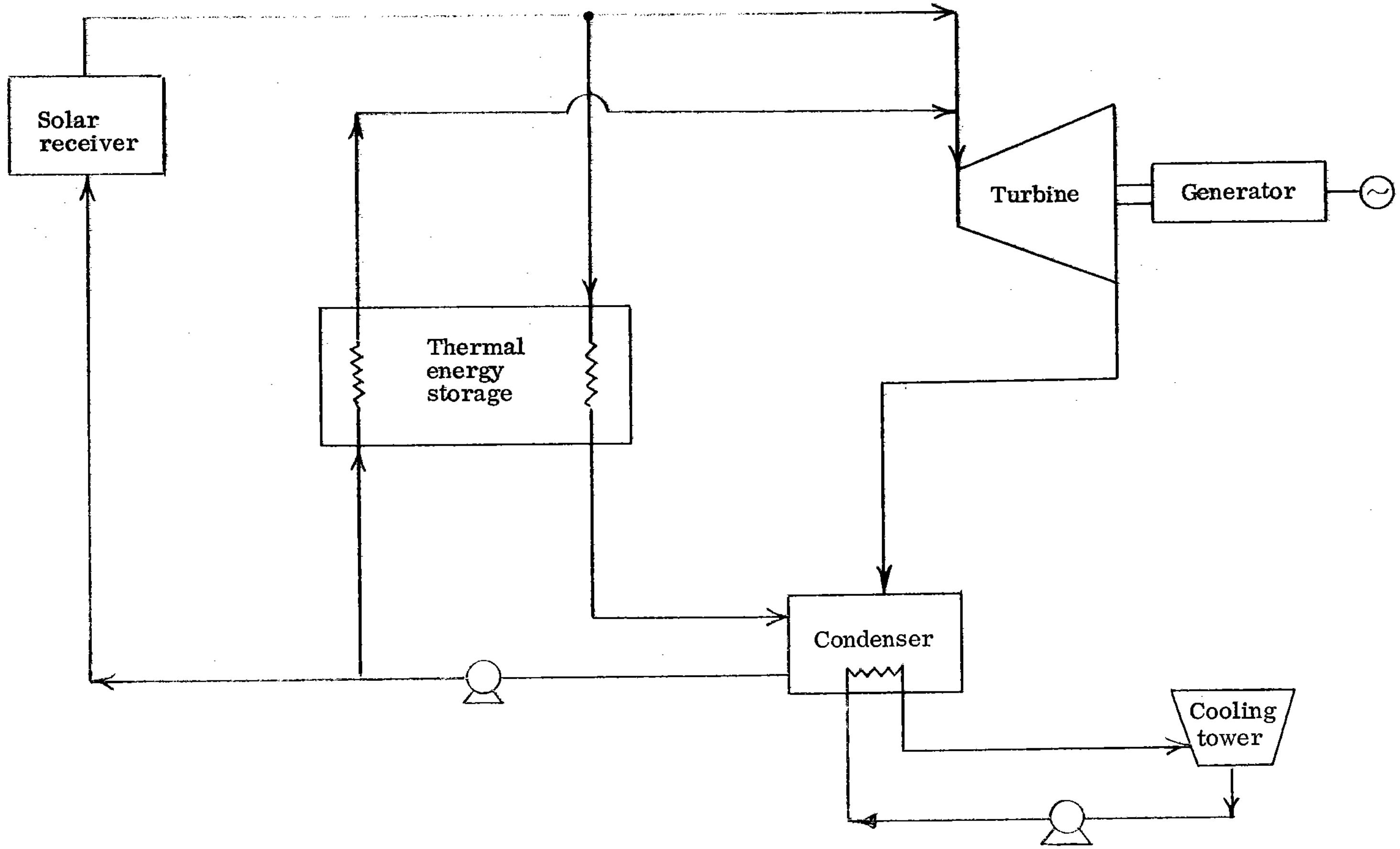


Figure 2-4. - Schematic of steam Rankine-cycle solar thermal power-conversion system with thermal storage.

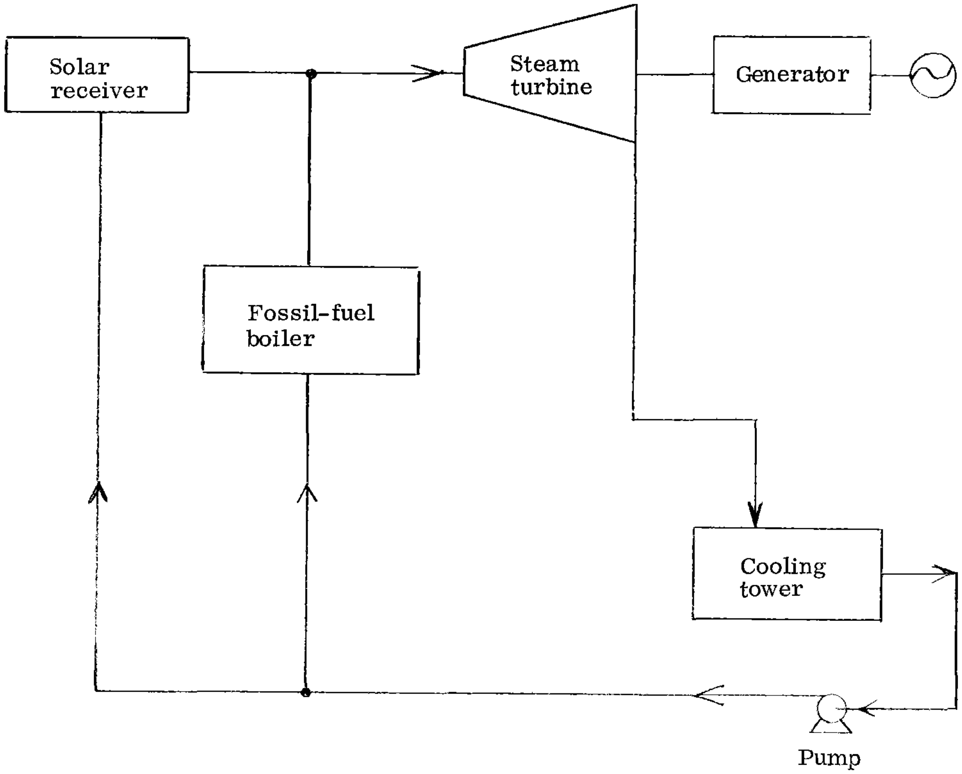


Figure 2-5. - Schematic of parallel solar - fossil fuel hybrid steam Rankine-cycle solar thermal power-conversion system.

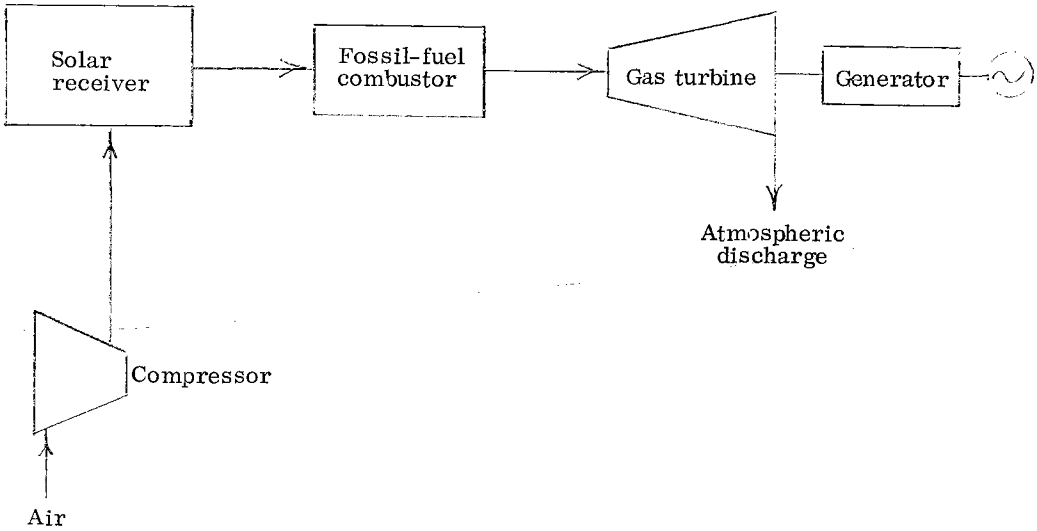


Figure 2-6. - Schematic of series solar - fossil fuel hybrid open-cycle Brayton power-conversion system.

3.0 RANKINE-CYCLE COMPONENT CHARACTERISTICS

by Thaddeus S. Mroz and M. Murray Bailey

In this section, performance and cost data are given for the major components of steam and organic Rankine-cycle power-conversion systems. The components are sized for specific system output power levels (in kWe) within an overall electric output power range of 5 to 50 000 kilowatts electric. This range encompasses components for small (5 to 100 kWe), intermediate (100 to 1000 kWe), and large (1000 to 50 000 kWe) power systems. Although this handbook is directed primarily to components that are commercially available production items, information is also presented on new components that are still in the hardware development stage. These developmental components have been designed, fabricated, and extensively tested for specific new applications, such as organic Rankine-cycle power-conversion systems.

The discussion of Rankine-cycle components includes steam and organic turbines, reciprocating engines, surface condensers, and boiler feed and condensate pumps. The data are based on information supplied by the component manufacturers. Components were selected for specific power levels by the manufacturers.

A technical description is provided of existing component designs, component development status, operating characteristics, availability, cost (pressure, temperature, etc.), and component experience factors (reliability, service, life, etc.). Data are presented by component for each specific power level to provide a parameter map through the overall power spectrum. The information provided in this section is based only on existing designs and does not reflect potential improvement in capability and performance through current and future research and development. Improvements in component efficiency for near-term and future designs will be discussed in the final version of this handbook.

3.1 STEAM TURBINES

3.1.1 Description

The steam turbine is a dynamic heat engine that converts available steam energy to shaft work. It is the prime mover most commonly used in the Rankine-cycle power system. The two fundamental elements of the steam turbine are the stationary nozzle and the rotating blades. Some of the thermal energy in the steam is converted to kinetic energy by expanding it through a stationary nozzle into a high-speed jet that is directed at the rotating blades. As the steam passes through the rotating blades, both its direction and its momentum are changed. The momentum change is translated into a turning force on the shaft (mechanical work).

Steam turbines can be designed for an infinite number of operating conditions and applications and have been built in sizes from 1 to 100 000 shaft horsepower and higher. Applications range from mechanical drives and electric power generation to ground and marine propulsion. Some applications use directly coupled systems, where the turbine shaft is directly coupled to the shaft of the driven machine. Others use a separately mounted speed reducer-increaser between the turbine and the driven machine to provide the required input speed to the driven machine. Single- and multistage turbines are available for various system output requirements. The steam turbine can have a design speed from 1200 to over 22 000 rpm depending on the size and application. The trend has been to design new turbines for higher speeds and with increased inlet pressure and temperature capability for increased efficiency. Generally, higher speed turbines (over 3600 rpm) are more compact and more efficient.

Different types of steam turbine are available commercially. The two basic types are the condensing and noncondensing steam turbines. Condensing turbines are designed for operation at an exhaust steam pressure below atmospheric pressure (14.7 psia), as measured at the exhaust connection of the turbine. All the expanded steam from the last stage is exhausted to the condenser. Noncondensing turbines are designed to operate at an exhaust pressure equal to, or greater than, atmospheric. They are generally used to supply exhaust steam at selected pressures to a process or to a low-pressure condensing turbine. Turbine variations within these two types are

(1) Straight flow (single- and double-flow exhaust)

(2) Reheat

(3) Extraction

A schematic of each turbine variation is shown in figure 3-1.

In the straight-flow turbine, all the full-throttle steam originating at the inlet is exhausted at the discharge connection. The straight-flow design includes single-flow- and double-flow-exhaust turbines. In the single-flow-exhaust turbine, the most widely used, the steam flows parallel to the shaft axis and emerges at the exhaust end. In the double-flow-exhaust turbine, used in multistage turbines, the steam flows parallel to the shaft axis but, before it enters the final stage, is divided into two equal parts. One part continues in the same direction through a set of low-pressure blades, and the other part flows through a duplicate set of blades. The double-flow-exhaust design permits low backpressures even though the steam volume grows enormously and calls for very large passages between the last-stage stationary nozzle and the rotating blades.

In the reheat design, used in multistage turbines, steam flow is exhausted to an intermediate stage, is returned to the boiler for resuperheating, and is then returned to the next lower pressure stage for expansion and exhaust. Reheat improves steam quality and overall efficiency and minimizes potential erosion problems (ref. 6). Single- and double-reheat turbines are commercially available. Turbines are also available for interstage steam extraction in the automatic and nonautomatic modes. Steam can be extracted between stages at the required pressure and flow rate for process use or feedwater heating. Selective design features are required for either application. Feedwater heating is normally used in large turbines.

Only fully condensing turbines are considered in this handbook. Data are not provided for reheat or extraction turbines because sizing turbines with these two functions requires specific system designs that are beyond the scope of this handbook.

The steam turbine is an assembly of a casing, a steam chest, a rotor, bearings, seals, nozzle rings, stationary vanes, a lubricating system, diaphragms, a governor, and valving (refs. 7 and 8). Single- and multistage commercial turbines are shown in cross section in figures 3-2 and 3-3.

3.1.1.1 Turbine casing. - A turbine casing is a pressure enclosure that surrounds the rotating element of the turbine and the steam component parts. Depending on the turbine design, the casing can be horizontally split (parallel to the shaft) or vertically split (perpendicular to the shaft). The casing can be divided into two (or more) sections: the steam inlet, and the exhaust end. The steam-inlet section contains high-pressure steam. The exhaust-end section contains the exhaust connection and the steam at exhaust conditions. The split casing designs permit the upper half-casing to be removed without disturbing the piping, the rotor alinement, or the components in the lower half-casing. The lower half-casing houses the steam inlet and exhaust connections. This arrangement provides easy access for rotor inspection and repair. The steam chest houses the inlet connection and governor-controlled valves and can either be cast integral with the lower half-casing or be bolted to it. Multistage turbines have also an intermediate section, which is that portion of the casing between the steam-inlet and exhaust-end sections. Casing materials vary with the design pressure and temperature.

3.1.1.2 Rotor. - The rotor consists of a shaft and the machined disks that hold the blades. Rotor design will vary, depending on speed, output power, and manufacturer's design and construction techniques. Two designs are available: solid and built-up. In solid rotors, used in high-speed applications, the shaft and disks are machined from a single forging. In built-up rotors the shaft and disks are machined separately. Figure 3-4 illustrates both constructions. The turbine blades can be machined into the disks (solid rotor) or installed on the disks and secured with shrouds (built-up rotor). The rotational speed of the built-up rotor is generally limited by the shrink fit, which is required to overcome the centrifugal growth of the disk. The disks are designed to minimize centrifugal stresses, thermal gradients, and blade loading at the disk rims. The rotor assembly is balanced to ensure smooth operation.

3.1.1.3 Nozzle. - Nozzles are stationary, machined, or formed openings that expand the steam and direct and organize its flow against the revolving turbine blades.

Converging or converging-diverging nozzles are used. Steam flow to the first stage originates at the steam chest and flows to a nozzle ring or nozzle subassembly, depending on the turbine size. In the succeeding stages the nozzles are machined and assembled on stationary disks, called diaphragms, which are stationary elements that contain a set of nozzles and are used between stages in a multistage turbine to expand steam and direct it against rotating blades. The diaphragms are split horizontally. One is installed in the upper half-casing and the other is installed in the lower half-casing.

3.1.1.4 Blades. - Blades are curved vane elements mounted on, or machined into, a rotating disk and used to convert the internal energy of the steam to the mechanical energy of the rotor. In built-up rotors, blades are installed in slots in the disks and are secured by a shroud band. In solid rotors, blades are integrally machined into a turbine disk. Two types of blades are used in turbine designs: impulse blades and reaction blades. With impulse blades, steam expansion and pressure drop occur only across the stationary vanes. The steam flow is directed to the rotating blades, where the working torque is developed. With reaction blades, expansion and pressure drop occur across both the stationary vanes and the rotating blades. A combination of impulse and reaction blades can be used in many large multistage turbines. Both types of blades are illustrated in figure 3-5. Sealing strips are normally used at the tops of the blades to minimize pressure loss and steam leakage past the blade tips into the adjacent stationary nozzle.

3.1.1.5 Bearings. - Radial and thrust bearings carry the full weight of the turbine rotor and enable it to spin free with a minimum of friction. Pressure-lubricated journal bearings are used for large turbines and ball bearings and ring-oiled journal bearings for small turbines. During operation the steam pressure differential across the shaft creates a net thrust. This net thrust is counterbalanced by the thrust bearing to maintain the rotor in the proper axial position.

3.1.1.6 Seals. - Leakage of steam from a turbine, which generally occurs at connections and at the casing-shaft interface, represents a loss. Shaft seals minimize this leakage. The type of seal selected depends on turbine design, size, speed, and inlet and outlet pressures. In a condensing turbine, steam leakage out of the turbine must be restricted at the high-pressure inlet end and air leakage into the turbine must be restricted at the low-pressure exhaust end. In a noncondensing turbine, steam leakage must be restricted at both the inlet and exhaust ends.

Two basic types of seal arrangement are used: segmented carbon-ring packing, and the labyrinth gland seal. Generally, carbon-ring packing, which consists of carbon rings held in place with a spring, is used for small turbines. The labyrinth gland seal, which is designed to provide a long tortuous path with a very high flow resistance to steam or air, is generally used in large turbines. In some cases, labyrinth seals are used in series with a set of carbon-ring seals at the shaft end. Seals are also provided on the diaphragms to restrict steam leakage between stages.

3.1.1.7 Governor. - A governor system is provided on a turbine to control steam flow to maintain a constant speed or output load. During operation, turbine speed remains constant as long as both the load and setting of the steam throttle valve do not change. If load decreases at the constant throttle valve setting, turbine speed increases. With the decreased load, steam flow must be decreased to maintain constant speed. If the load increases at a constant steam flow, turbine speed decreases. With the increased load the steam throttle valve must be reset to match the load and readjust the speed to the operational level. The governor senses the shaft speed and automatically controls the steam flow for the required load. Governor designs vary with the turbine size, shaft speed, required degree of control, and accuracy.

The governing system includes the speed governor, the control mechanism, the governor-controlled throttle valve, and the speed changer. A basic governor arrangement is shown in figure 3-6.

3.1.2 Development Status

A sound technical data base exists in the commercial steam turbine industry. A number of companies have developed and offer a complete line of single- and multistage production steam turbines that meet all utility requirements for mechanical drive and electric power generation.

Turbine development has been directed to the competitive market demands and such operating requirements as output ranges and inlet conditions. The technical data base is excellent for development of new units tailored for specific near-term and future applications. Commercially available production turbines have a very wide range of design ratings:

(1) Inlet steam pressure, 250 to 4200 psi

(2) Inlet temperature, 700° to 1200° F

(3) Output shaft power, 1 to over 100 000 hp

(4) Exhaust pressure, 2 in. Hg to 175 psig

(5) Speed, 1700 to 22 000 rpm

(6) Number of stages, 1 to 14 or more

Turbine designs and construction can be tailored to almost an infinite number of performance requirements. In developing a line of turbines, a number of frame sizes can be designed for a horsepower range to provide application flexibility. In some cases, rotating elements can be interchanged with specific frames for maximum efficiency and lowest steam flow rate. The ranges of inlet pressure, inlet temperature, and turbine shaft speed for specific output power levels of turbines selected for maximum efficiency and optimum steam flow rate rather than for low initial cost are shown in figures 3-7 to 3-9. Single-stage turbines are commercially available with output shaft power to 2000 horsepower and speeds to 22 000 rpm. Multistage turbines are available at higher horsepowers. Steam turbine materials are selected for the specific temperature, pressure, and output power requirements of the various applications.

3.1.3 Availability

Typical availability of steam turbines for specific power levels is shown in figure 3-10. Availability will vary with the manufacturer. The delivery schedules for turbines are influenced by such factors as

(1) Standard design or design with special modifications
(2) Standard materials of construction or special materials
(3) Turbine size (mass production item or long-lead item)
(4) Quantity (staggered delivery)
(5) Stocking practice for component parts
(6) Existing shop loads

Large steam turbines are not mass production or stock items but are made on an order basis.

3.1.4 Cost

Cost as a function of power level is shown in figure 3-11 for steam turbines in the 5- to 50 000-kilowatt-electric range. Costs are for a single turbine assembly complete with a speed reducer-increaser and a baseplate to accommodate the turbine, the speed reducer-increaser (as applicable), and the alternating-current generator. They were provided by the manufacturers as list or estimated costs and are subject to changes with market conditions and quantity procurements. Costs based on quantity procurements are not available.

Turbine weight as a function of power level is shown in figure 3-12.

3.1.5 Operating Characteristics

The performance characteristics of the steam turbine are a function of the inlet and outlet conditions, degree of superheat, speed, and specific turbine design selected for the application. High steam pressures provide more available energy. Increasing the steam temperature and superheat can increase the efficiency.

For maximum efficiency and low steam rate, the design characteristics of the selected turbine must be matched to the operating conditions, output power, and efficiency requirements of the specific application. A low-cost turbine with low nozzle and blade efficiency and high exit losses may be acceptable for an industrial application where high efficiency and low steam rate are not the overriding criteria. However, steam turbines with high nozzle and blade efficiency and low exit losses would be preferred and would have to be considered for a high-efficiency application.

Performance capabilities of selected commercially available steam turbines are shown in figures 3-13 to 3-16. Figure 3-13 shows turbine efficiency as a function of power level at full power. Figures 3-14 to 3-16 show steam rate as a function of power level for 100-, 75-, and 50-percent power, respectively.

3.1.6 Operating Constraints

Steam turbines have been designed and fabricated in sizes dictated by application requirements. Operationally, steam turbines are constrained to the specific design component ratings of pressure, temperature, and speed determined by the stress levels in the casing and rotor and the loading on the seals and bearings.

3.1.7 Experience Factors

The steam turbine is one of the most developed prime movers and energy-conversion devices. Steam turbines have been used in electric power production and mechanical drive applications for over 80 years and have established a record of high reliability. Commercial turbines are designed as heavy-duty equipment for operating lives of over 10 to 20 years with minimal scheduled maintenance. Operating histories and maintenance schedules are well established and documented in the industrial, utility, and marine applications.

3.2 ORGANIC TURBINE

3.2.1 Description

The organic turbine, like the steam turbine, converts the available energy in a working fluid to mechanical energy. In this process the available energy of the organic working fluid (a function of pressure and temperature) is converted to mechanical energy by directing a jet of organic vapor at a proper angle against curved blades mounted on a rotating disk.

The design and fabrication methods used for condensing steam turbines are also applicable for organic turbines. The inherent difference between these two turbines is that the organic turbine is designed specifically for the properties of the organic fluid and the steam turbine is designed specifically for the properties of steam. Unlike steam, the selected organic fluids have high molecular weights (e. g., 84 or greater as compared with 18 for steam). Like steam, the selected organic fluids have low specific heats and high vapor pressures at the turbine exhaust. Some organic fluids have a saturated vapor line with a positive slope. Vapor with this property becomes superheated as it expands through the turbine and is in a superheated state at the turbine exhaust. This feature avoids moisture formation and turbine-blade erosion and simplifies the material design requirements. Steam has a negative saturated vapor line, and therefore large steam turbines require superheat and interstage reheat to maintain the required quality in the final stages.

The use of high-molecular-weight working fluids results in higher turbine efficiencies than those of steam turbines with fewer stages. Therefore, organic turbines are smaller and less complex than steam turbines.

The organic turbines uses the same basic turbine technology and component parts as the condensing steam turbine. Depending on the application and working fluid, single and multistage organic turbines can be designed.

3.2.2 Development Status

Organic turbines have been developed for a limited number of aerospace and commercial applications and for developmental-prototype solar thermal power systems for mechanical drive (e.g., pumps) and electric power generation. Information received from organic turbine manufacturers suggests that present organic turbines are developmental or prototype (i.e., only one or two units of a kind are made). A commercial, production line of organic turbines has not been marketed to date, as steam turbines have been, because of limited applications and lack of market demand. Development of different size turbines would be required for operating conditions beyond present coverage.

The development of organic turbines has been limited to

(1) Developmental component test turbines

(2) Preproduction-prototype turbines for field testing of developmental power-generating systems (mechanical or electrical output)

(3) Prototype turbines for commercial field application (e.g., bottoming cycles and combined cycles)

The design operating ratings of the turbines developed to date include

(1) Inlet pressure, 300 to 900 psia

(2) Inlet temperature, 170° to 825° F

(3) Output power range, 1 to 600 kilowatts

(4) Number of stages, 1 to 6

(5) Operating speeds, 9300 to 60 000 rpm

(6) Working fluids, toluene, Dowtherm, CP-34, FL-50, FL-85, CP-34, R-12, R-22, R-112, etc.

The design and construction of present organic turbines are tailored to the properties of the working fluid and to the specific performance requirements.

A technology base, including design and fabrication practice, does exist for the organic turbine. The application range developed to date is shown in figures 3-17 to 3-19, where the design inlet pressures, inlet temperatures, and shaft speeds for developed turbines are plotted as a function of power level. These turbines were developed for maximum efficiency and optimum working-fluid flow rate. The information in the figures is based on data supplied by manufacturers.

3.2.3 Availability

Although organic turbines have not yet attained production status, prototype and preproduction turbines have been extensively tested. The hard tooling required for production turbines is not available because of the limited demand.

3.2.4 Operating Characteristics

Performance characteristics of the organic turbine are a function of the properties of the working fluid, the turbine-inlet conditions (pressure and temperature), the backpressure, and the design selected for the application. Experience is based on the li limited number of developed units. Performance data are available for 5- to 600-kilowatt turbines for a variety of working fluids, for inlet pressures from 86 to 800 psig, for backpressures from 3 to 20 psig, for inlet temperatures from 187° to 600° F, and speeds from 5500 to 60 000 rpm. The efficiency of developed turbines is shown as a function of actual turbine output power in figure 3-20. These data are based on full-power operation. Efficiency data at part-power operation are not available.

3.2.5 Cost

Since organic turbines are not yet production items, cost data were not available from the manufacturers.

3.2.6 Operating Constraints

The organic turbine is constrained to the maximum allowable temperature of the working fluid, above which temperature thermal decomposition can occur. There are no physical constraints on the components. Turbine design requirements are well within established design practices.

3.2.7 Experience Factors

Experience factors such as design life, reliability, and maintenance history have not been established for organic turbines because operating experience has been limited.

Because their design is similar to that of the steam turbine, their design life and reliability should parallel those of the steam turbine.

3.3 STEAM AND ORGANIC RECIPROCATING EXPANSION ENGINES

3.3.1 Description

The reciprocating expansion engine is a dynamic heat engine that converts the available energy in a working fluid (steam or organic) to mechanical energy. Linear motion in the cylinder is converted to rotary motion at the output shaft. The reciprocating expansion engine dates back to 1768 and since that time has been in use for electric power generation, pump and compressor drive, ship propulsion, automotive propulsion, and various other mechanical power applications. However, through the years the reciprocating steam expansion engine has been replaced to a large extent by steam turbines and internal combustion engines.

The reciprocating engine is a fixed-displacement machine in which work is performed by the pressure of the working fluid acting on a moving piston. The expansion is carried out in a given space (cylinder), the walls of which are exposed alternately to high and low temperatures. Various engine configurations have been manufactured, including vertical, horizontal, and angle. Depending on the application, single-acting engines (working fluid acts on one side of the piston) and double-acting engines (working fluid acts alternately on both sides of the piston) have been placed in service.

The reciprocating engine can be designed for condensing or noncondensing service. It can be a single-stage engine, where the complete working-fluid expansion occurs in a single cylinder, or a multistage engine, where the expansion is divided into a number of cylinders or stages. It is classified as a compound engine (two stages), a triple expansion engine (three stages), or a quadruple expansion engine (four stages).

The engines are further classified as being either uniflow or duoflow (counterflow). In the uniflow engine the working fluid is admitted into the top of the cylinder through a steam inlet or admission valve (or valves) and flows only in one direction, from the cylinder head to the center of the cylinder during the expansion stroke. The steam is then exhausted through ports in the cylinder wall. These exhaust ports are uncovered by the piston at the end of the expansion stroke (bottom dead center). In the duoflow (counterflow) engine the working fluid is admitted through a steam inlet valve (or valves) at the top of the cylinder and flows downward to the center of the cylinder during the expansion stroke. During the compression stroke the steam flows in the opposite direction and is exhausted at the top of the cylinder through the exhaust valve (or valves).

The two basic elements of the reciprocating engine are the steam or organic expansion end and the power-conversion end (fig. 3-21). The expansion end includes the cylinder, pistons, valves, seals, timing mechanisms, and manifolding for converting the available energy in the working fluid to mechanical energy. The power-conversion end includes the support frame and housing, crosshead, connecting rod, crankshaft, and lubrication system for converting linear motion to the rotary motion required for mechanical drive.

The component parts of a typical vertical, uniflow industrial steam expansion engine are shown in figure 3-22. This type of engine is rated for 300- to 3000-kilowatt service at operating speeds of 300 to 400 rpm. Newer engines for non-heavy-duty industrial applications, such as automotive and general-purpose power applications, have different design features, are more efficient at higher speeds, and are more compact.

In general, reciprocating engine design and application requires the consideration of a number of parameters that influence performance, including

(1) Friction
(2) Cylinder condensation
(3) Leakage past pistons and valves and valve timing
(4) Piston and rotative speed
(5) Clearance volume at top dead center
(6) Expansion ratio
(7) Inlet conditions including superheat
(8) Backpressure
(9) Reliability, weight, and cost

Friction, cylinder condensation, leakage past pistons and valves, and poor valve timing represent losses in engine performance. Friction losses are a fixed loss inherent in the design of a particular size engine at a specific loading. They affect both the engine economy and capacity. Cylinder condensation can occur on the cooler cylinder walls during admission (intake) and potentially in the early part of expansion. Condensation is followed by evaporation during late expansion on exhaust and increases working-fluid consumption. Leakage past pistons and valves is a direct loss and increases working-fluid consumption. Poor valve timing affects steam admission and exhaust and decreases output power. Higher piston and rotative speeds affect cylinder size, weight, and cost and the degree of cylinder condensation. The selection of piston and rotative speeds is based on valve timing, valve response, and engine size. Clearance volume (the volume between the piston at top dead center and the cylinder head) must be kept to a minimum in condensing engines. Heat losses increase as the clearance volume increases and result in an increase in working-fluid consumption. Superheating increases engine efficiency and minimizes the effect of cylinder-wall condensation. High inlet pressures and high expansion ratios have a marked influence on efficiency, since efficiency varies directly with expansion ratio.

The application of a steam or organic reciprocating engine, as in the case of steam turbines, requires careful consideration and matching of the application requirements with the engine design parameters for maximum efficiency and long service life.

3.3.2 Development Status

Demand for reciprocating engines has been replaced to a certain extent in the past 20 to 30 years by demand for high-speed steam turbines. Although the industrial steam reciprocating engine has demonstrated high reliability and a service life of 20 to 30 years, its size, weight, and low speed have contributed to the decreased demand. Currently, a few companies are marketing a line of industrial production engines with output ratings from 5 to 3000 kilowatts. These engines are fully developed and commercially available. Heavy-duty industrial machines operate with low-pressure (125 to 300 psig) steam and at speeds of 300 to 600 rpm. They are designed for service lives of 10 years or more.

In the past 10 years, several manufacturers have begun a number of small engine developments and proceeded to field installation and field demonstration. These small engines are more compact and efficient than larger engines and have higher rotative speeds (1800 to 3600 rpm) for automotive applications and as small power systems for military applications. They include both steam and a few small (1 to $1\frac{1}{2}$ kW) organic reciprocating engines. Developed engines are available in certain sizes. Further developments, including other sizes, are contingent on demand and funding.

In addition to their higher rotative speeds, the newer engine designs can operate at inlet pressures of 600 to 2500 psia and at inlet temperatures of 700° to 1050° F. Engine configurations vary from a four-cylinder "V" engine to four-cylinder vertical in-line engines with an expansion ratio of 12. The inlet pressure, inlet temperature, and rotative speed of a developed steam engine are shown in figures 3-23 to 3-25. Data on other engines were not available. Marketed industrial engines and other engines that are developed but have not been marketed are identified in these figures.

3.3.3 Availability

Availability data have not been received from the manufacturers. Data will be included as they become available.

3.3.4 Operating Characteristics

The performance characteristics of the reciprocating engine are a function of the inlet and exhaust conditions, the type of application, the specific engine design, and the type of working fluid (steam or organic). As indicated previously, increasing the inlet pressure, the temperature (and superheat), and the expansion ratio should increase efficiency. Engine selection that emphasizes efficiency and steam flow rate rather than initial cost requires a careful matching of system requirements with engine design.

The rate at which work is performed on the engine piston is called indicated horsepower. It is obtained from a pressure-volume indicator diagram (fig. 3-26) that shows the amount of work extracted from the working fluid. The ideal work can be calculated from the steam inlet conditions and the exhaust pressure. Engine mechanical efficiency is the ratio of actual shaft work to indicator-diagram work. Engine adiabatic efficiency is the ratio of indicator-diagram work to ideal work. Overall engine efficiency is the product of the mechanical efficiency and the adiabatic efficiency. Since the reciprocating engine is a fixed-displacement engine, output power and speed can be varied by changing the inlet conditions.

The indicated horsepower depends on initial pressure, backpressure compression, clearance volume, parts area, piston speed, valve assembly design, valve and piston tightness, superheat, and length of steam admission.

Overall engine efficiency and steam or organic fluid flow rate, both at full power, are shown as a function of actual engine output in figures 3-27 and 3-28, respectively. The data are plotted at actual engine ratings.

3.3.5 Cost

The cost of commercially available reciprocating engines is shown in figure 3-29. This figure includes only data made available by the manufacturers.

3.3.6 Operating Constraints

The operating constraints of the steam engine are the inlet pressure and temperature and the exhaust backpressure. A particular engine design is constrained to the thermal stress limitation of the expansion head and the bearing loading at the crankshaft and crosshead. Off-design backpressure affects expansion ratio and efficiency. The principal constraint of organic reciprocating expansion engines is the upper allowable operational pressure and temperature for the particular working fluid. Thermal decomposition parameters would be an overriding consideration.

3.3.7 Service Factor

Heavy-duty industrial reciprocating engines have established remarkably long service lives, over 20 years. With scheduled routine maintenance, these engines are highly reliable.

Service factors have not been established for more recently developed smaller and higher speed engines.

3.4 CONDENSERS

3.4.1 Description

The common forms of condensers can be broadly classified as water-cooled condensers, air-cooled condensers, and evaporative condensers. This handbook includes only the water-cooled and air-cooled condensers.

The condenser is one of the major components in the steam Rankine cycle. Its functions are

(1) To condense the vapors exhausting from the turbine or the reciprocating engine

(2) To produce an economical, continuous vacuum at the turbine or reciprocating engine exhaust to improve heat rate

(3) To deaerate the condensate

(4) To conserve the condensate for reuse as boiler feed

(5) To maintain the design turbine exhaust pressure and temperature

The condenser removes waste heat from the primary system and rejects this heat to the cooling system or heat sink.

In designing a power system the lowest condensing temperature is desirable to achieve optimum efficiency without incurring a high cost penalty. The condenser type and the condensing temperature are selected on the basis of the cooling method used and the available cooling-fluid temperature to the condenser. Three condenser cooling methods are presently used.

(1) Once-through cooling, in which the flow of cooling water originates directly from rivers, lakes, seas, etc.

(2) Wet cooling, in which cooling water is supplied to the condenser from a evaporative wet cooling tower

(3) Dry cooling, in which heat is dissipated directly to the atmosphere

Once-through cooling is preferred by the system designer since it can provide the lowest condenser temperature. However, this cooling method may not be available at many geographical locations either because water is unavailable or because environmental legislation limits the use of natural waters to avoid thermal pollution. Evaporative wet cooling is another cooling method. This cooling method depends on the wet-bulb temperature and normally cannot provide cooling-water temperatures as low as once-through cooling during the summer months. Selecting a higher cooling-water temperature affects the condensing temperature and therefore the prime-mover exhaust pressure. In areas where a water shortage exists, the use of a evaporative wet cooling tower can become a problem because of the required makeup water. This problem intensifies with water scarcity. Direct dry cooling permits powerplant siting without regard for supplies of cooling water, environmental legislation concerning water thermal effects, and evaporative loss makeup. A dry cooling system is designed for the dry-bulb temperature that is prevalent in the particular application area during the summer months (worst conditions). The increased heat-sink temperature translates into a higher condensing temperature and lower system efficiency. In addition to acceptable system efficiency, selection of the condenser type and cooling method is influenced by the size of the power system as well as consideration of initial capital equipment costs, land acquisition costs, construction costs, operating costs, environmental factors, degree of required support systems (e.g., makeup and purification), maintenance costs, reliability, etc.

3.4.1.1 Water-cooled condensers. - The principal element of the water-cooled condenser is the shell-and-tube assembly. The shell assembly consists of the outer structural shell with the shell-side and tube-side inlet and outlet connections, the hot well, and the inlet and outlet tube-side heads (water boxes). The tube assembly consists of the tube bundles (or tube nests), the tube sheets, and the tube spacers.

Figure 3-30 is a schematic of a typical water-cooled condenser. Exhaust steam enters at the top of the shell and flows downward over and between the cooling tubes. The condensate formed on the surface of the tubes drops into the condensate-collecting chamber, called the hot well, located at the bottom of the shell. Cooling water is pumped to the inlet header, called the water box, and flows through the tubes and into the outlet header to condense the steam.

Most water-cooled condensers are single or multipass with the tubes arranged in a horizontal configuration. The outlet water connection is normally at a higher elevation than the inlet connection to ensure complete filling of the tubes. The single-pass condenser has a single water box at each end of the tube bundle and the cooling water passes in only one direction.

In the multipass condenser the cooling water passes through half of the tubes in one direction, is reversed in the water box, and returns in the opposite direction through the other half of the tubes. Water boxes are designed to reduce erosion by minimizing turbulence and equalizing water velocities at the tube inlets. Depending on the manufacturer's design (large condensers), the locations of the cooling-water inlet and outlet can vary, as shown in figure 3-31.

Multipass condensers are designed with straight tubes. Condenser design is tailored to specific component and system operating requirements. Application and design of a surface condenser must consider steam flow rate, steam temperature, required turbine or engine backpressure, cooling-water flow rate, ratio of cooling-water flow to steam flow, cooling-water cleanliness, part-power operation, system arrangement, air leakage into the condenser, required net positive suction head (NPSH) of the condensate hot-well pump, materials, and pressure drop (pumping power) on the coolant side of the condenser.

In the physical design of the condenser, the shell (or steam) side must be designed for minimum resistance to steam penetration into the tube bundle. This design must incorporate a selective pressure differential in the steam flow in order to prevent over-penetration in one section and starving in other sections. The tube arrangement and layout must be designed for uniform steam velocity and distribution. To avoid dead pockets, the steam inlet and distribution passages must direct the steam flow across the entire tube bundle for uniform steam distribution and heat transfer. In cylindrical-shell condensers this is accomplished by closely packed tubes in the bottom of the tube bundle and restrictive venting into the cooler sections.

Tube spacing is determined by steam flow, area requirements, and entrance velocity. Tube length is determined by power-system space limitations, initial cost, and pumping power. A long condenser is usually less expensive than a shorter condenser but requires more heat-transfer surface. If water supply is limited, multipass condensers are normally used. Tube diameter is determined by thermal and space requirements. Although smaller diameter tubes have higher heat-transfer coefficients, larger diameter tubes carry more circulating cooling water at a given velocity with less pressure drop. Cooling-water velocities are normally limited to 6.5 to 8 feet per second. Lower velocities are normally uneconomical. Combinations of linear and triangular tube-spacing can be used to get the desired steam distribution. (Tubes can occupy 20 to 25 percent of the full cross-sectional area.) A controlled longitudinal distribution is incorporated into the design so that surfaces will be useful. The hot well (or condensate-collecting point) should be designed for adequate NPSH for the hot-well pump.

3.4.1.2 Air-cooled condenser. - The air-cooled condenser considered in this section is the direct-transfer type where the two fluids (air and steam) are separated by the metallic heat-transfer surface (tube wall). The tube section is directly exposed to the atmosphere. In air-cooled condensers, steam flows from the exhaust of the prime mover (turbine or reciprocating expander) to the inlet header and through the finned-tube condensing section. The condensate flows to the outlet header and to a collection point, such as a drum or hot well, which provides condensate to the pump suction. A fan provides cooling airflow across and between the tube fins to remove the waste heat and condense the steam.

Induced-draft (fan downstream of condensing tubes) and forced-draft (fan upstream of condensing tubes) condenser configurations are used. Figure 3-32 illustrates these configurations. The design and configuration of air-cooled condensers will vary with the application, the size of the power system, and the manufacturer's design practice. The basic elements of the air-cooled condenser are the finned-tube - header assembly, the fan drive assembly, the plenum chamber with fan enclosure, and the support structure. The principal element is the finned-tube - header assembly. With a multiple-tube condensing section the header is designed to evenly distribute the incoming steam flow to the condenser tubes. Since air is not an efficient cooling medium, such as water, fins are added to the tubes to augment the heat-transfer capability. The finned-tube design is selected for optimum air-side effectiveness and minimum resistance to airflow. Design selection considers the type of fin, the method of bonding the fins to the tube, the resistance to airflow, the thickness profile, the fin-to-bare-tube ratio, and the frontal area. Selecting a tube arrangement requires an evaluation of the airflow temperature distribution and pressure drop to minimize fan horsepower, which is a parasitic loss. Aluminum and copper fins are used for metal temperatures to 650° F. Steel and stainless-steel fins are used for higher temperatures. Tubes are fabricated in steel, stainless steel, chromium-molybdenum steel, and nonferrous metals. Material selection is determined by design requirements (temperature level, pressure, etc.).

Cooling air to the finned condenser tubes is provided by a propeller fan or a centrifugal fan. Fan selection is determined by the airflow requirements and the pressure differential across the flow path. A propeller fan is more efficient for condensers with low internal pressure drop and free air discharge. A centrifugal fan can be used for condensers with a higher pressure drop. Fan type and size, horsepower requirements, and fan drive are selected on the basis of heat-load and cooling-air requirements and environmental constraints. These constraints and requirements consider air resistance, noise level, unit shape, and operating speeds for design and off-design operation. Single- and multispeed fan drives are available. The condenser design and fan selection determine the operating cost, which is an important criterion in selecting a condenser.

3.4.2 Development Status

The technical data base is well developed and established for water-cooled and air-cooled surface condensers in the 5- to 50 000-kilowatt power-system output range. The technical base includes design techniques, experience factors, materials technology, and fabrication techniques for factory-assembled and site-erected condensers. A number of manufacturers offer a commercial line of developed condensers. Commercial condenser size as a function of power level on the basis of heat-transfer area is shown in figure 3-33. Larger condensers are considered to be engineered items and are made to order.

3.4.3 Availability

The availability of commercial water-cooled and air-cooled condensers is shown as a function of power level in figure 3-34. Availability is influenced by the size and the design requirements.

3.4.4 Cost

The cost of commercially available air-cooled and water-cooled condensers is shown as a function of power level in figure 3-35.

3.4.5 Operating Characteristics

The condenser is designed to remove the waste heat from a power-generating system at different operating levels and to maintain the prime-mover exhaust conditions at particular steam flow rates and coolant-inlet conditions. Design performance is contingent on maintaining the required heat transfer throughout the condenser, removing noncondensible gases, and maintaining cooling-water flow rate for the required design conditions.

Noncondensible gases in the condenser are the result of air leakage into the prime mover and into the condenser. Noncondensibles have a detrimental effect on attainable vacuum, as well as on the heat-transfer coefficient in the condenser. Provisions must be made for continuous removal of air from the steam space. An example of the effect of air leakage on attainable vacuum is shown in the following table:

Size of condenser, kW	Leakage, ft^3 of free air/min				
	10	20	30	40	50
	Vacuum, in. Hg				
8700	28.25	27.85	27.4	27.0	26.6
8500	28.4	28.1	27.8	27.5	27.2
4000	28.7	28.35	28.0	27.65	----

Condenser ratings are based on the heat transfer for a certain level of heat-transfer-surface cleanliness. For the water-cooled condenser, capacity will progressively decrease from the fouling of the heat-transfer surfaces on the water side of the tubes. Condensers must have sufficient excess tube-surface to offset the effects of fouling and to maintain satisfactory performance with a reasonable period of service between cleanings. Therefore, fouling factors are used to predict performance in sizing a condenser. The fouling (cleanliness) factor is defined as the ratio of heat transfer of fouled tubes to that of clean tubes. The air-side fouling of the air-cooled condenser is negligible when compared with the water-cooling problems of scale formation, corrosion, and biological growth in tubes.

The air-cooled condenser rating (heat-removal capability) is based on the difference between the dry-bulb temperature of the cooling air and the design condensing temperature. Air has a specific volume 830 times as great as that of water, but its heat capacity is only about one-fourth that of water. Consequently, for a given amount of cooling, the air-cooled condenser requires large volumes of air. During operation, steam flows from the inlet header into the finned condensing tubes and cooling air passing over the finned tubes condenses the steam. Because steam and condensate flow in the same direction, the pressure loss is minimized and the heat-transfer coefficient is increased. Design performance is affected by changes in flow and cooling-air temperature, changes in static pressure, and changes in air-side effectiveness due to flow restrictions at the tube fins and fouling. Shutters and fan speed control are used for off-design operation.

3.4.6 Operating Constraints

Operating constraints on a condenser are the coolant temperature, coolant flow rate, level of noncondensibles on the steam side, and cleanliness of the heat-transfer surfaces. Deviating from design operating parameters results in reduced performance.

3.4.7 Experience Factors

Condensers are well-developed, passive components with established operating and maintenance histories. Established designs meet power-system life and reliability requirements.

3.5 BOILER FEED PUMPS

3.5.1 Description

Pumping is defined as the transport of fluids from one point to another by adding energy to the fluid with a pump. Several types of pumps have been developed in the industrial pump sector and are commercially available for the range of industrial and boiler feed pump applications. These pump types are

(1) Positive-displacement pumps

(2) Centrifugal pumps

(3) Turbine regenerative pumps

Selection of a particular type of pump for a boiler feed application depends on the required system discharge pressure, the flow requirements (high pressure and low capacity, high pressure and intermediate capacity, etc.), and the temperature. Other factors considered in pump selection include pump efficiency, size, weight, reliability, design life, required net positive suction head (NPSH), and the known maintenance history for a particular type of pump. Selecting a pump requires careful matching of pump performance characteristics with system requirements at full load and at off-design conditions (e.g., 75 and 50 percent of system output operation). The pump driver should be selected for the maximum operational brake horsepower. Overpowering should be avoided.

3.5.1.1 Reciprocating positive-displacement pumps. - Commercial reciprocating positive-displacement pumps are normally characterized as fixed-speed, fixed-displacement (capacity) pumps designed for high-pressure applications. In this type of pump the liquid flows from the inlet (suction) into a cylinder with a fixed volume and is forced out of this chamber (displaced) by a moving piston or plunger. Pumping action is by displacement. The power end is an assembly consisting of the pump frame, bearings, crankshaft, connecting rods, and crossheads. The liquid end is an assembly consisting of the liquid cylinders, pistons or plungers, suction and discharge valves, inlet and outlet connections, and pressure seals. The pistons or plungers are directly connected to the crossheads. The liquid end is bolted to the power end. Figure 3-38 illustrates a typical pump construction. Designs will vary with size, speed, and manufacturer's design practices. The pump is generally driven through speed-reduction gears or a V-belt drive arrangement. The rotating motion of the driver and crankshaft is converted to linear motion at the pistons or plungers through the connecting rods and crossheads. During operation, the suction (inlet) valve opens on the backstroke of the piston or plunger and fills the displacement volume in the liquid cylinder. The suction valve closes, and on the forward stroke the discharge valve opens as the liquid is forced out of the cylinder into the system piping. The total capacity (in gallons per minute) is a function of the cylinder displacement volume, the number of pistons or plungers, and the speed. The discharge pressure is a function of system resistance. Commercial pumps are available in vertical and horizontal configurations. In some cases, changes in capacity can be achieved with variable-displacement pumps or variable-speed drivers. These are nonstandard design features. Commercial pumps are available in ratings from less than 1 gallon per minute to over 200 gallons per minute at discharge pressures to 3000 psi and above. In the past the reciprocating positive-displacement pump was used to a large extent for boiler feed applications in the low- and intermediate-capacity ranges. Figure 3-39 illustrates these application ranges. The use of this pump has somewhat diminished with new developments in centrifugal pumps and turbine regenerative pumps.

3.5.1.2 Centrifugal pumps. - As the name implies, centrifugal force is used in centrifugal pumps to transport liquids. Liquid flows to the center of the impeller, is picked up by the impeller vanes, is accelerated to a high velocity by rotation of the impeller, and is thrown by centrifugal force into a spiral volute of gradually increasing size. The velocity head is converted to pressure head in the volute. Figure 3-40 illustrates the flow pattern and the impeller-volute arrangement. In some pump designs a diffuser is used in lieu of a volute to convert velocity head to pressure head.

The principal elements of the centrifugal pump are the pump casing and the rotating element. The pump casing includes the inlet and outlet connections, the volutes or diffusers, the bearing housings, the shaft-seal housings, and the wearing rings. The rotating element includes the impellers, the shaft, and the shaft-mounted bearings. Figure 3-41 illustrates a typical multistage centrifugal pump configuration.

Centrifugal pumps are designed for a wide range of ratings such as low head and low capacity, intermediate head and higher capacities, etc. Single and multistage pumps are available. The difference in pump design ratings and operating characteristics is due to the impeller design. The different types of impellers are

(1) Forced vortex
(2) Radial
(3) Francis
(4) Mixed flow
(5) Propeller

As indicated, pumps with these impellers have different operating characteristics, and these characteristics are classified by a dimensionless factor called a specific speed index N_S. This factor is a guide in determining the maximum head at which a pump can operate and is defined as the speed at which an exact model of the pump would have to operate to deliver 1 gallon per minute against 1-foot head per stage. The lower the specific speed, the higher the head that can be developed per stage (per impeller). Figure 3-42 illustrates the specific speed range.

The forced-vortex pump is a low-specific-speed pump that operates at high rotational speed and is applicable for low-flow, high-head boiler feed pump applications. Multistage pumps with radial impellers (N_S of 100 to 2000) are used for high-head applications (e.g., 100 to 1000 psi and higher). Pump designs will vary with the application requirements and manufacturer's design practices.

3.5.1.3 Regenerative turbine pumps. - The regenerative turbine pump is used for low- to intermediate-capacity (3 to 300 gal/min) and high-head (100 to 1500 psi) applications. The impeller and casing of this pump differ from those of a centrifugal pump. The centrifugal-pump impeller incorporates vanes that extend from the impeller eye to the impeller periphery (straight and curved vanes) and a casing with a spiral volute or diffuser section. The regenerative pump derives its name from the many blades that are machined into the periphery of the impeller. This construction is illustrated in figure 3-43. This pump develops a higher pressure per stage than a centrifugal pump, is considered to be self-venting, and requires close-running axial clearances. An annular flow chamber is incorporated in the casing at the periphery of the turbine wheel in lieu of the spiral volute or diffuser section used in a centrifugal pump.

In the regenerative turbine pump, liquid flows from suction to both sides of the impeller and to the annular chamber and is pumped by shearing action through the channel. The flow of liquid within the impeller blades is shown in figure 3-44. The pressure buildup is continuous and progressive along the annular path. Single and multistage pumps are commercially available for high-discharge-pressure applications. In multistage pumps the discharge of a stage is directed to the next stage, where the pressure is doubled and the process repeated.

The principal elements of this pump are the rotating element and the pump casing. The rotating element assembly includes the shaft, the impellers, and the shaft-mounted bearings. The casing includes the pressure-shell housing with inlet and outlet connections, shaft-seal housings, bearing housings, pump mounts, and liners or rings with flow channels.

3.5.2 Development Status

Pumps have been developed and are commercially available through the full range of boiler feed applications for the industrial and utility sectors. Forced-vortex centrifugal pumps, reciprocating positive-displacement pumps, and turbine regenerative pumps are developed and commercially available for low-capacity, high-pressure boiler feed applications (capacity, <1 to >250 gal/min; pressure, 100 to >1000 psi). Multistage volute and diffuser centrifugal pumps are commercially available for higher capacities and higher discharge pressures with horizontally and vertically split (barrel type) casings. A sound technology base exists in the commercial pump sector.

3.5.3 Availability

The availability of these pumps in terms of mass production varies with pump size and the production capability and workload of the manufacturer. Estimates of availability are given in figure 3-45.

3.5.4 Cost

Cost estimates obtained from the manufacturers of centrifugal pumps are shown in figure 3-46.

3.5.5 Operating Characteristics

3.5.5.1 Reciprocating positive-displacement pumps. The conventional reciprocating positive-displacement pump is a direct-acting, fixed-displacement, fixed-speed pump that develops a discharge pressure equal to the system resistance. Changes in output capacity can be achieved by bypassing a portion of the discharge flow or by a design provision that permits changes in crankshaft speed and total displacement per unit of time. Bypassing is not considered the optimum method of capacity control since the pump operates at full load and power but delivers a lower capacity to the system. Variable-displacement control is available in some commercial pumps. Pump efficiency is a function of volumetric efficiency and mechanical efficiency. Mechanical efficiency varies with the friction losses in the pump power-end and through the packed stuffing box at each plunger. Losses in volumetric efficiency are due to valve slip (back leakage across the valve) and direct leakage from each stuffing box. Pressure fluctuates at the discharge of the reciprocating piston-plunger pump because of the pump flow pattern. This characteristic varies with the pump design. Cushion chambers are used to dampen the pulsation effect on system operation.

3.5.5.2 Centrifugal pumps. - The theoretical performance characteristics of a centrifugal pump are a function of the impeller and the specific speed (pump type). The actual in-service pump performance characteristics are a function of how well the design characteristics of the specific pump were matched to the actual requirements during the pump selection process. When efficiency rather than initial cost is emphasized, the pump should be selected at the highest efficiency of a pump head-capacity curve. The slope of the curve should be selected for stable operation that matches the system operating cycle. With the centrifugal pump, the capacity and discharge pressure will vary with the system friction load. Oversizing or undersizing a pump results in a lower efficiency for the application. Head-capacity curves can be flat, drooping, rising, steep, stable, or unstable - depending on the shape of the impeller blades.

The different head-capacity curves are illustrated in figure 3-47 and are described as follows:

(1) A flat curve indicates little variation in discharge head between the design point and pump shutoff.

(2) A steep curve indicates a large increase in discharge head between the design point and pump shutoff.

(3) A rising curve indicates a continuous increase in discharge head and a decrease in capacity from the design point to pump shutoff.

(4) A drooping curve indicates a decrease in discharge head between the design point and pump shutoff.

The drooping curve is considered to be an unstable curve since the same discharge pressure can be developed at two different capacities. The flat, steep, and rising curves are considered to be stable curves since there is one capacity for a particular discharge pressure. In the selection of a pump, the required NPSH should always be less than the available system NPSH. Otherwise, performance can deteriorate.

Centrifugal-pump efficiency is affected by the following losses, which are a function of the design and the fabrication methods:

(1) Hydraulic losses - losses caused by shocks, eddy currents, and friction of the fluid through the impeller and casing

(2) Disk-friction losses - power required to rotate the impeller in the liquid

(3) Short-circuit losses - losses caused by leakage from the discharge to the suction side of the pump through clearances between the casing and the impeller wearing rings

(4) Mechanical losses - losses resulting from friction in the stuffing boxes and the pump bearings.

3.5.5.3 Regenerative turbine pumps. - The regenerative turbine pump has a characteristic steep head-capacity curve. Unlike the characteristics of the conventional centrifugal pump, the brake horsepower decreases with increasing capacity and decreasing head. Figure 3-48 illustrates a typical head-capacity performance curve. The regenerative turbine pump follows the same head-capacity speed law as the centrifugal pump. The capacity varies as the speed, and the head varies as the speed squared. These pumps are self priming and develop higher heads per stage than conventional centrifugal pumps and are particularly applicable for low capacities, where some centrifugal pumps would operate too close to shutoff (unstable region). The steep head-capacity curve provides stable performance over a wide range in head.

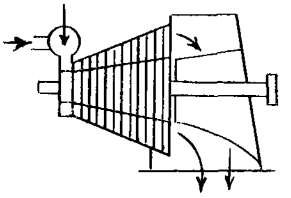
Pump efficiency and brake horsepower as a function of power level are shown in figures 3-49 and 3-50 for the pumps discussed.

3.5.6 Operating Constraints

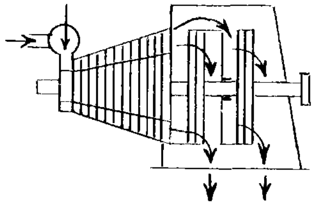
For successful application and operation of a pump for boiler feed service, the pump must be supplied with sufficient NPSH, and the electric motor or other driver must be sized for the operational pump brake horsepower range. A pump is suitable only for a specific pressure-capacity range.

3.5.7 Experience Factors

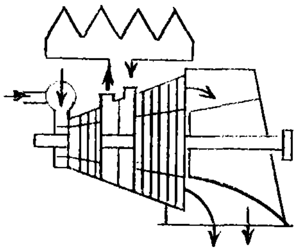
Commercial boiler feed pumps are fully developed powerplant components with an established design life, high reliability, and well-documented maintenance schedules. Equipment operating lives of 5 to 10 years are not uncommon.



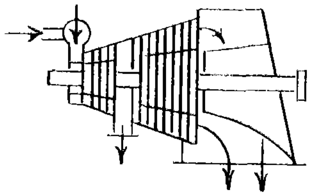
(a) Straight flow (single).



(b) Straight flow (double).

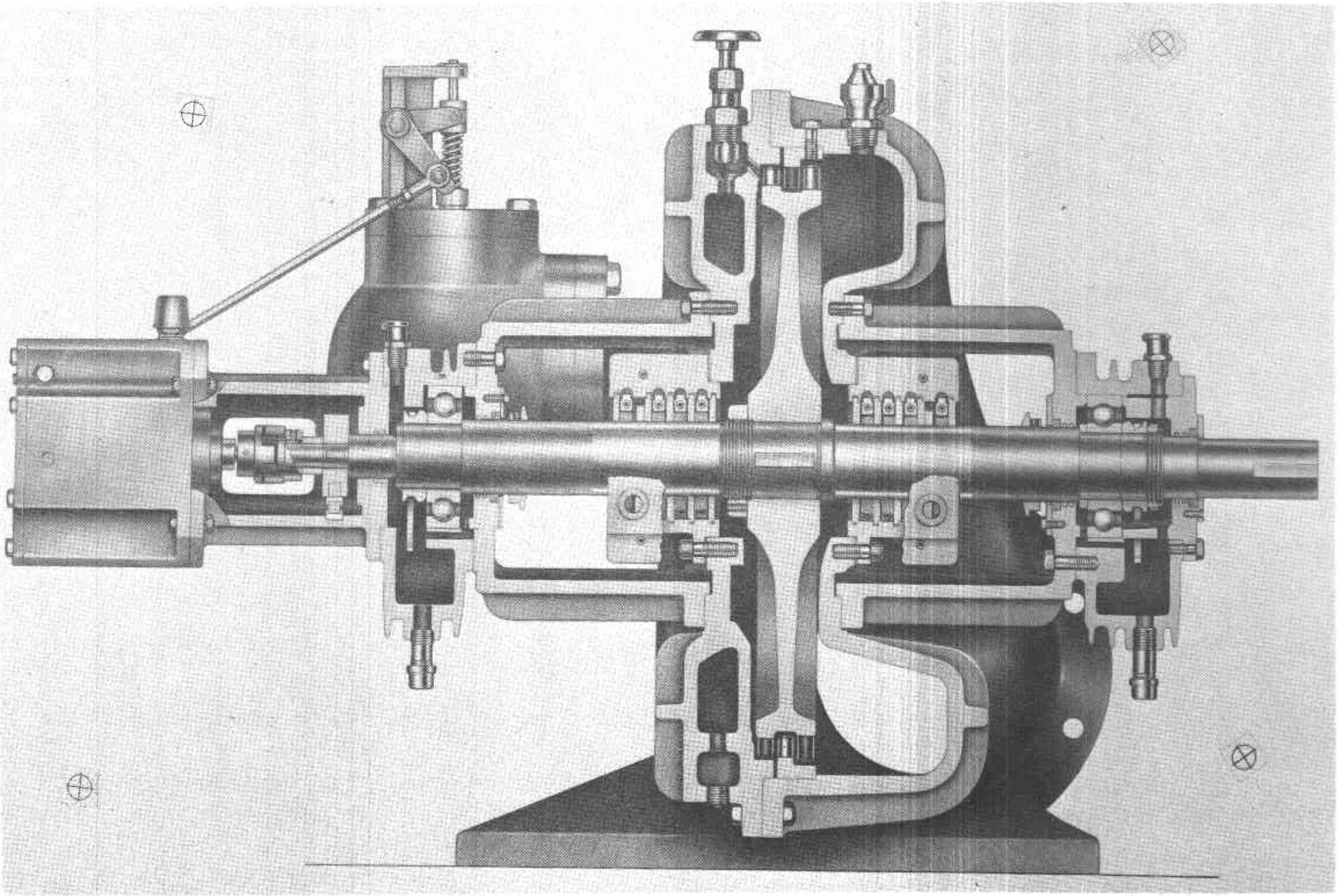


(c) Reheat.



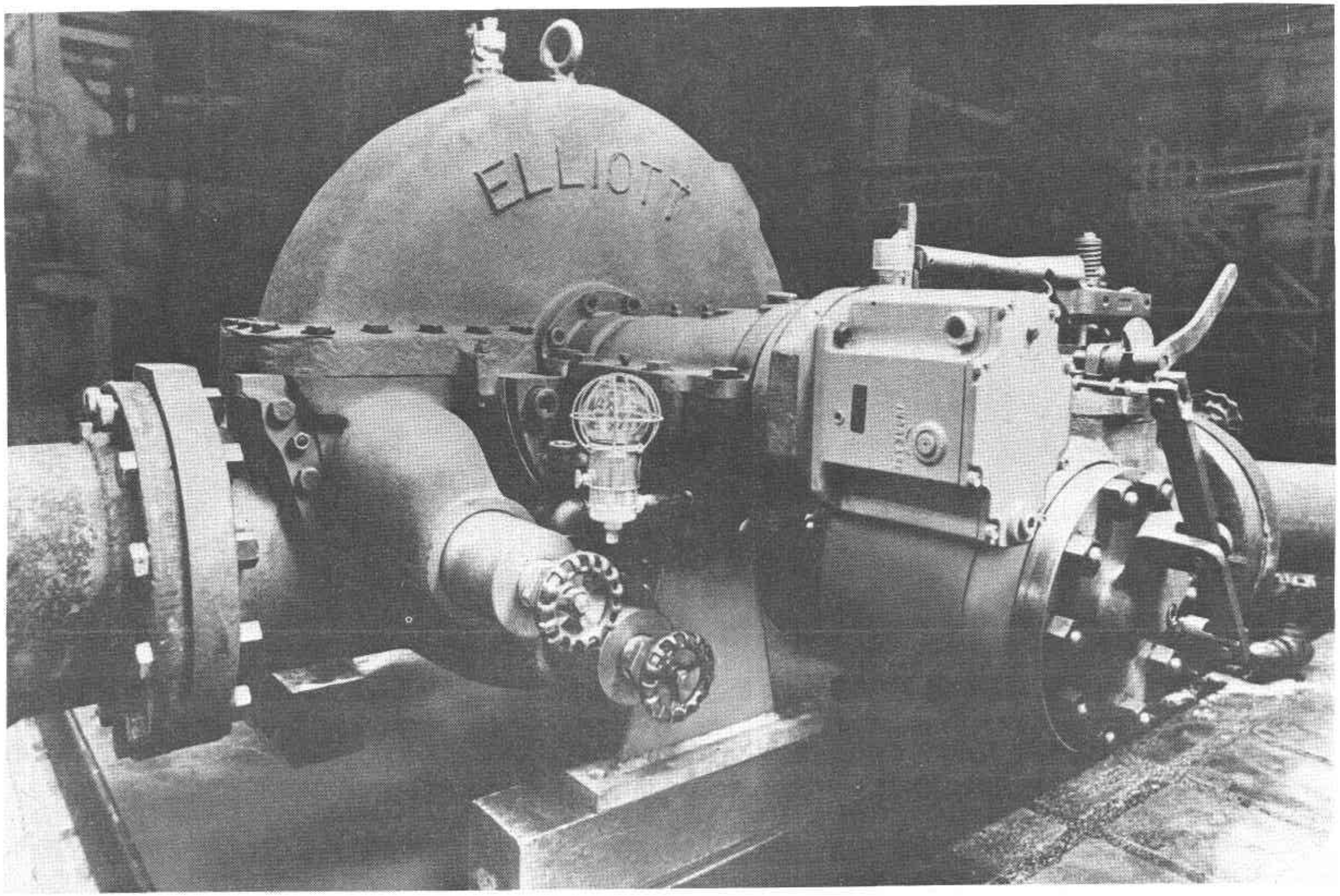
(d) Extraction.

Figure 3-1. - Steam turbine variations.



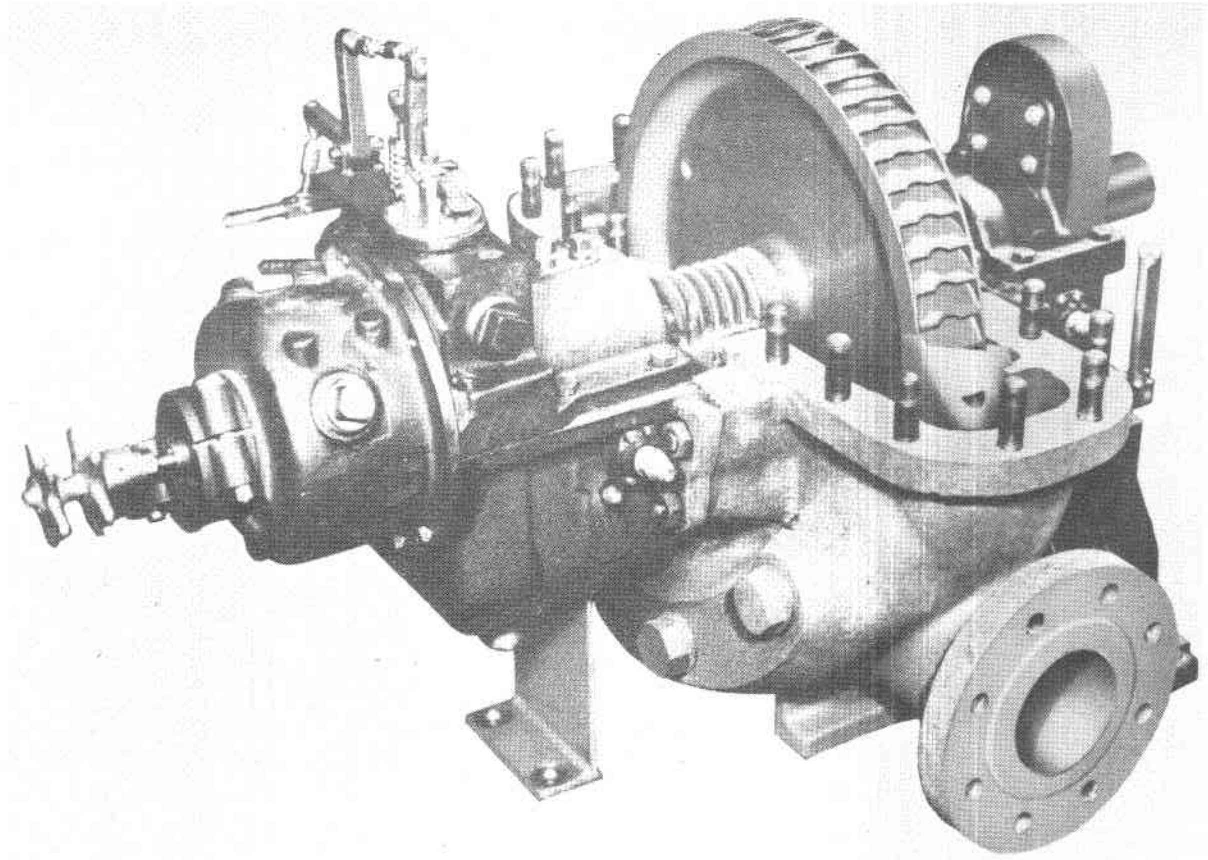
(a) Single-stage industrial steam turbine. (Courtesy of Coppus Engineering Corp.)

Figure 3-2. - Industrial steam turbines.



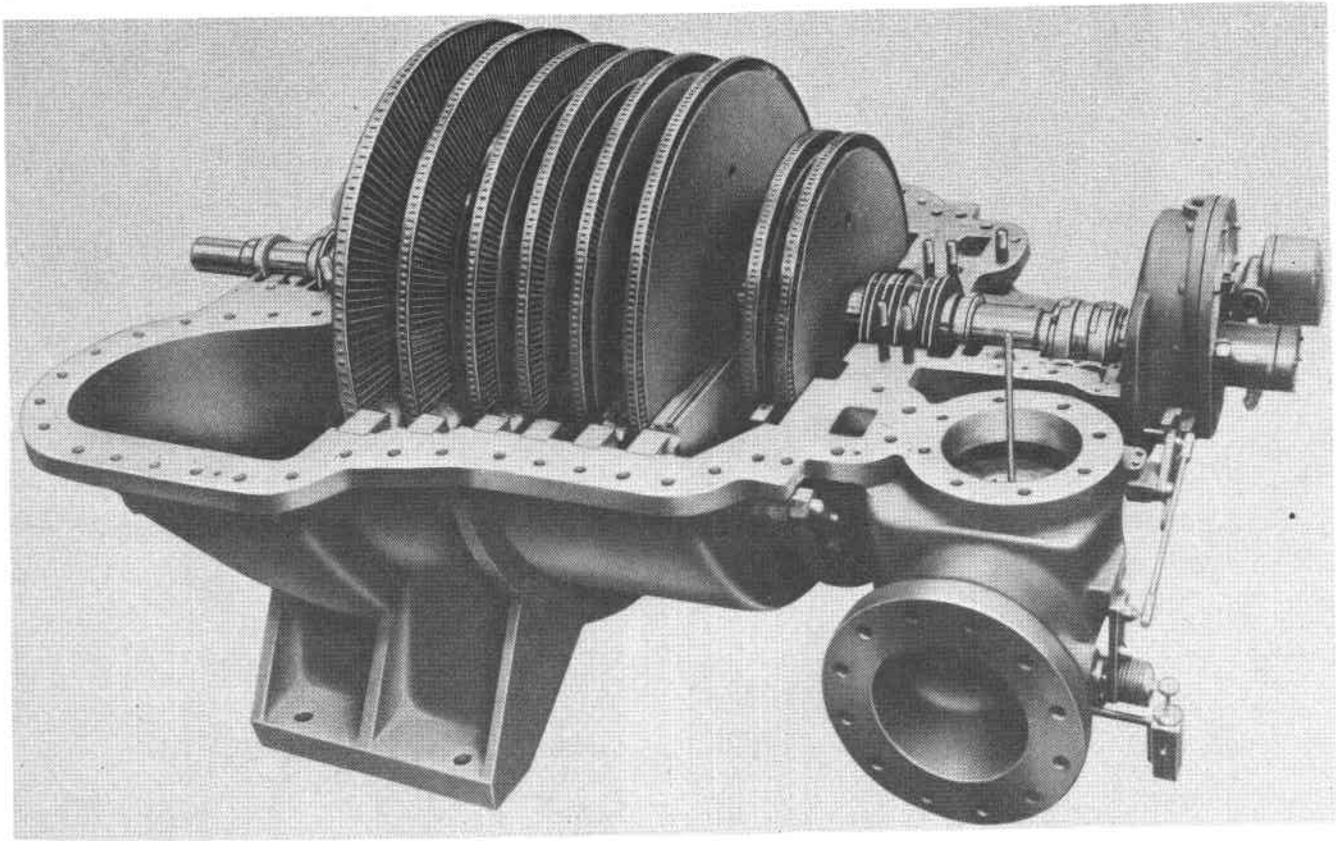
(b) Single-stage industrial steam turbine. (Courtesy of Elliot Co., Division of Carrier Corp.)

Figure 3-2. - Continued.



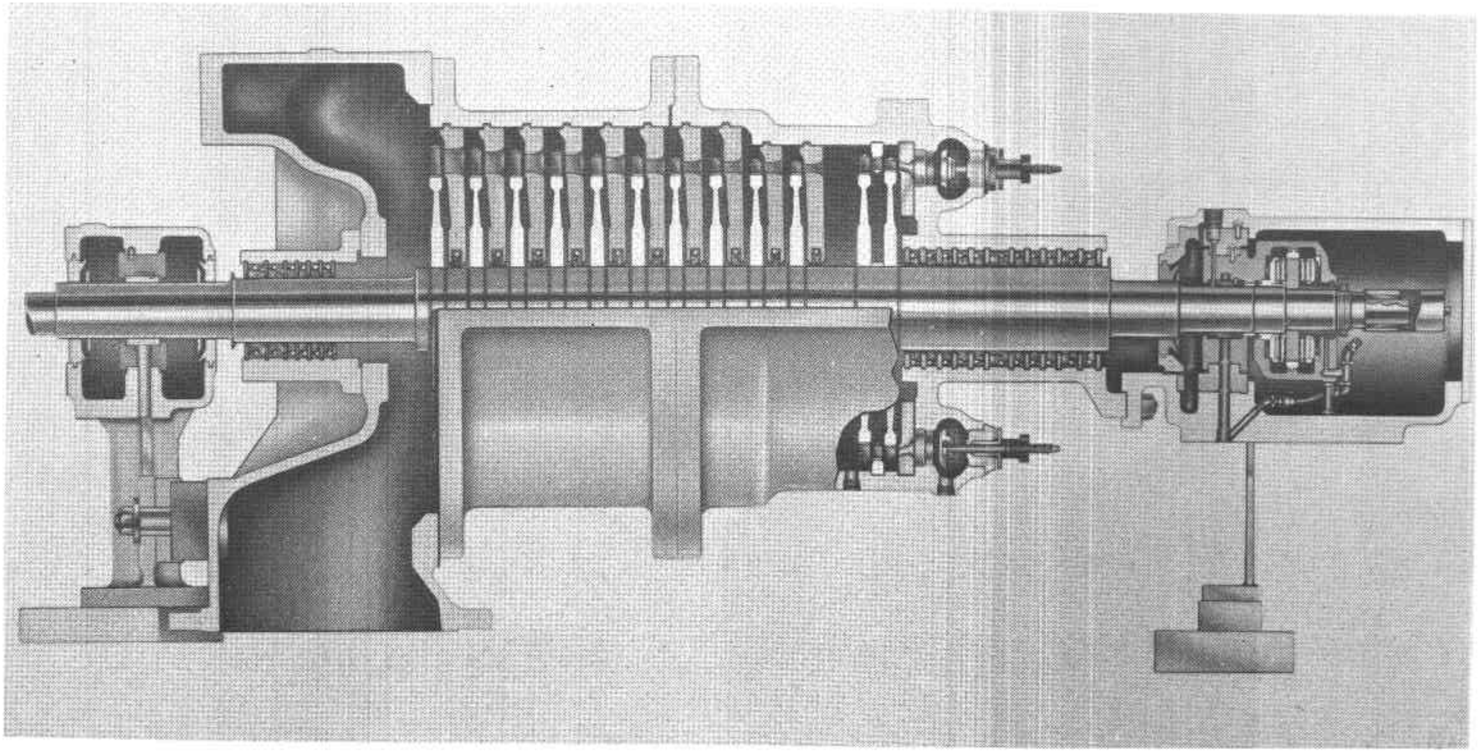
(c) Single-stage, helical-flow industrial steam turbine. (Courtesy of Terry Steam Turbine Co.)

Figure 3.2 - Concluded.



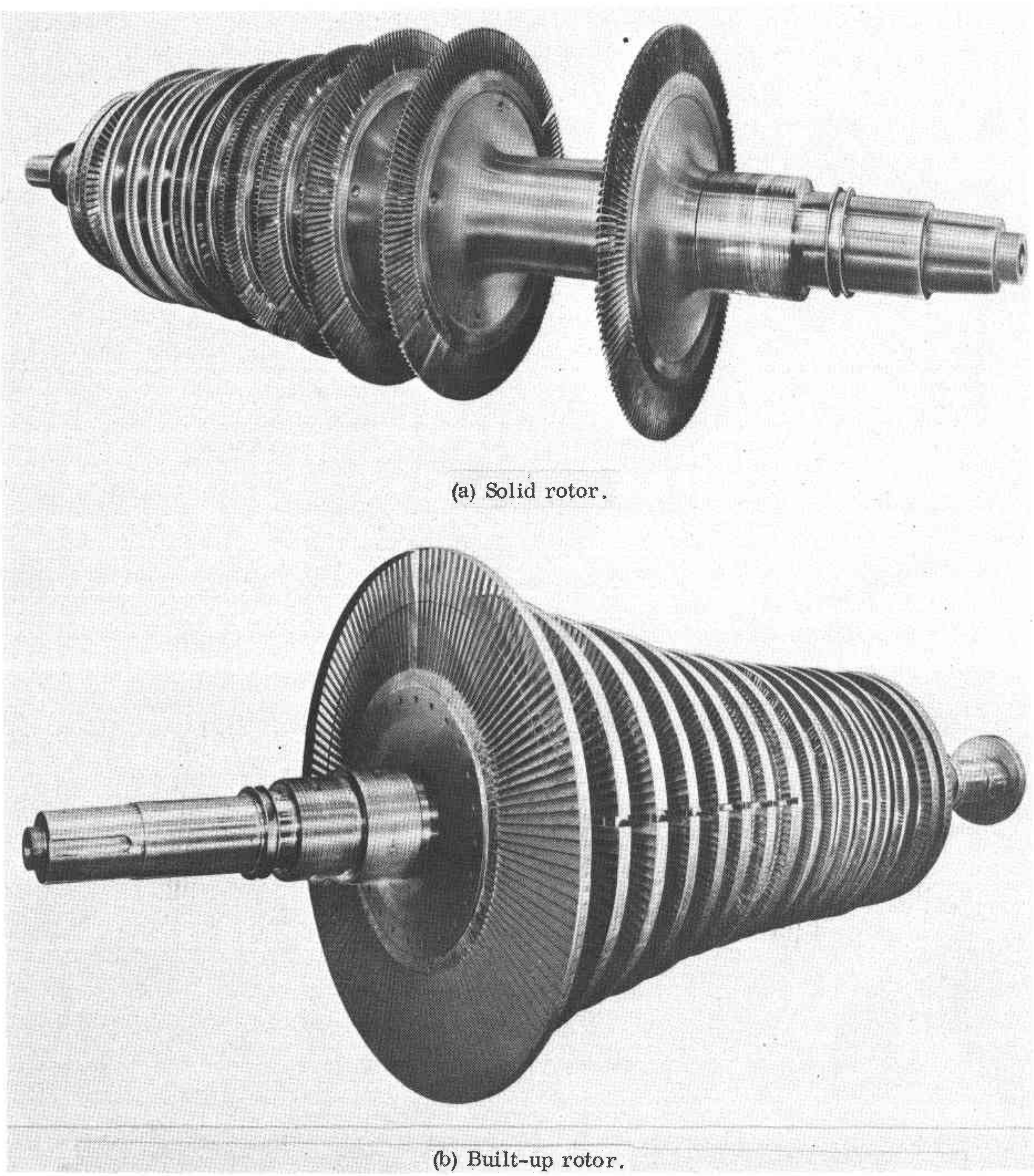
(a) Multistage industrial steam turbine. (Courtesy of Murray Steam Turbines, Division of Trane Co.)

Figure 3-3. - Industrial steam turbines.



(b) Multistage industrial steam turbine. (Courtesy of Turbodyne Corp., Steam Turbine Division.)

Figure 3-3. - Concluded.



(a) Solid rotor.

(b) Built-up rotor.

Figure 3-4. - Multistage steam turbine - rotor construction. (Courtesy of Elliott Co., Division of Carrier Corp.)

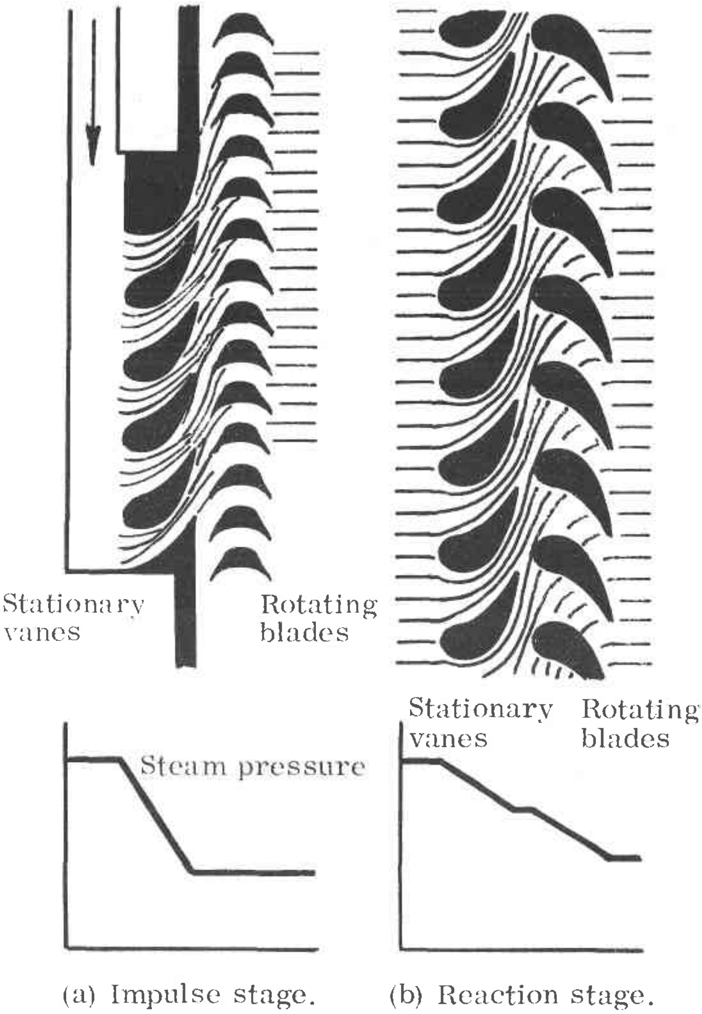


Figure 3-5. - Types of blades used in steam turbines.

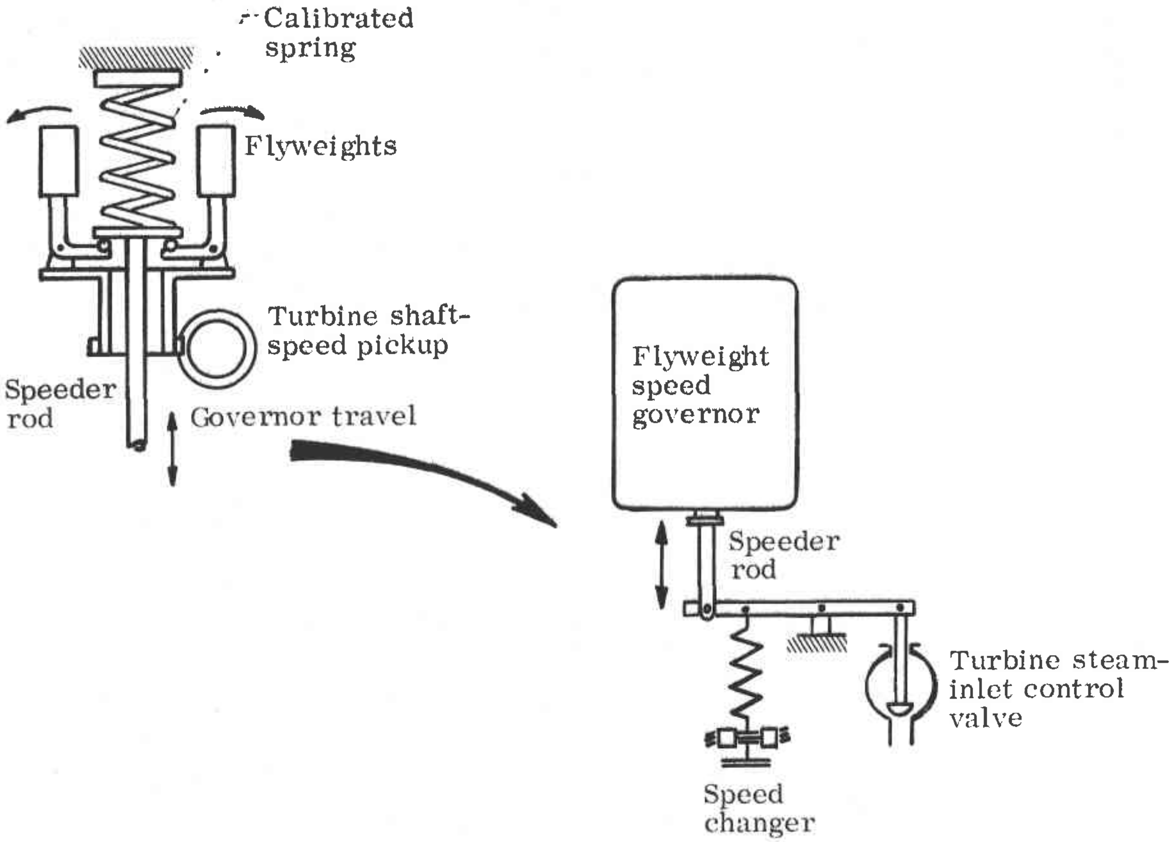


Figure 3-6. - Schematic of basic governor arrangement.

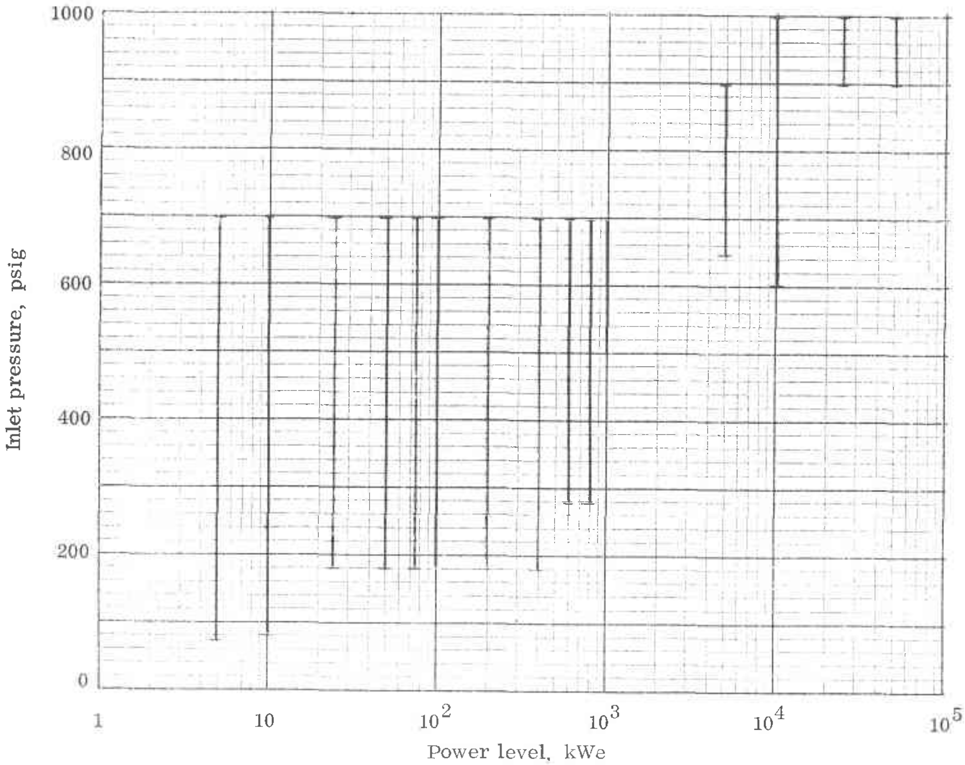


Figure 3-7. - Inlet pressure of Rankine-cycle steam turbines as function of power level.

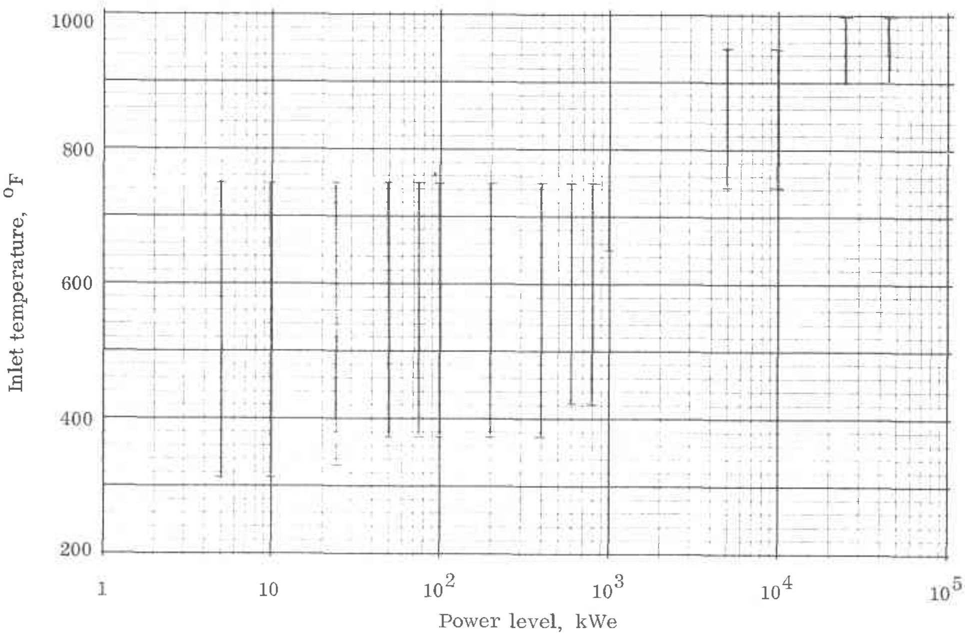


Figure 3-8. - Inlet temperature of Rankine-cycle steam turbines as function of power level.

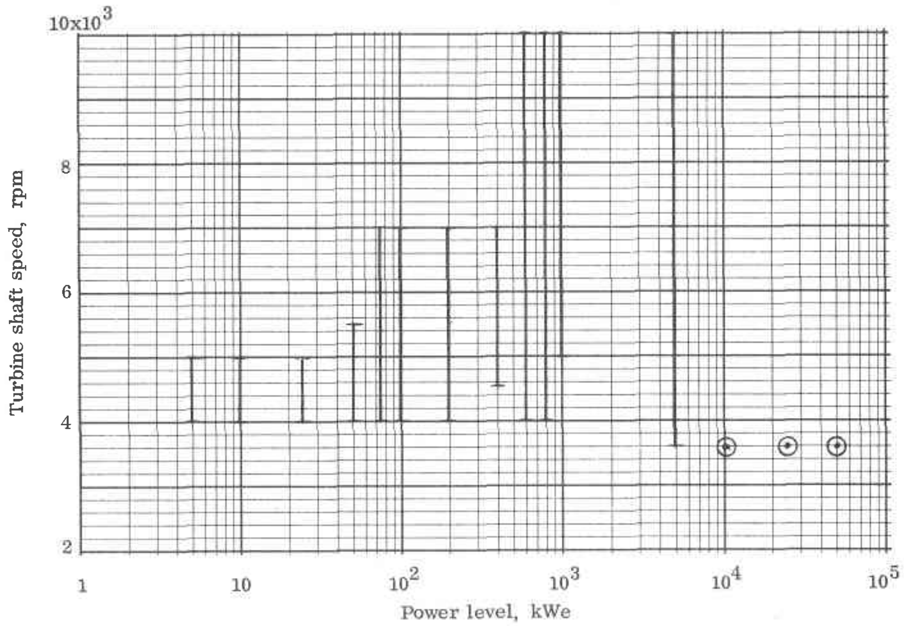


Figure 3-9. - Turbine shaft speed of Rankine-cycle steam turbines as function of power level.

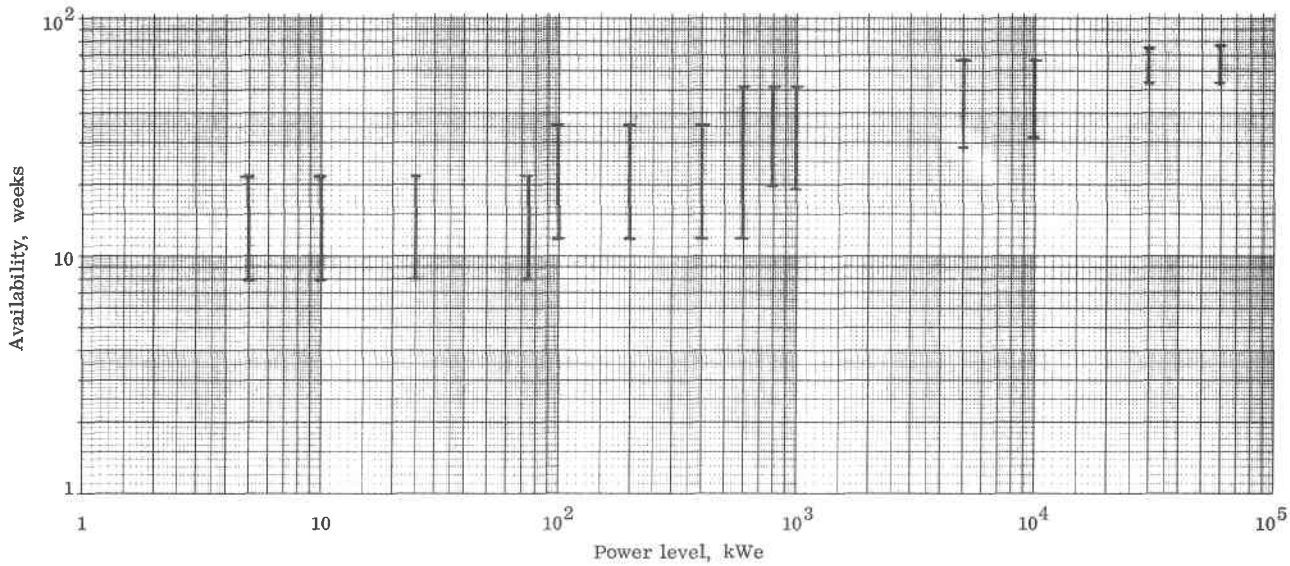


Figure 3-10. - Availability of Rankine-cycle steam turbines as function of power level.

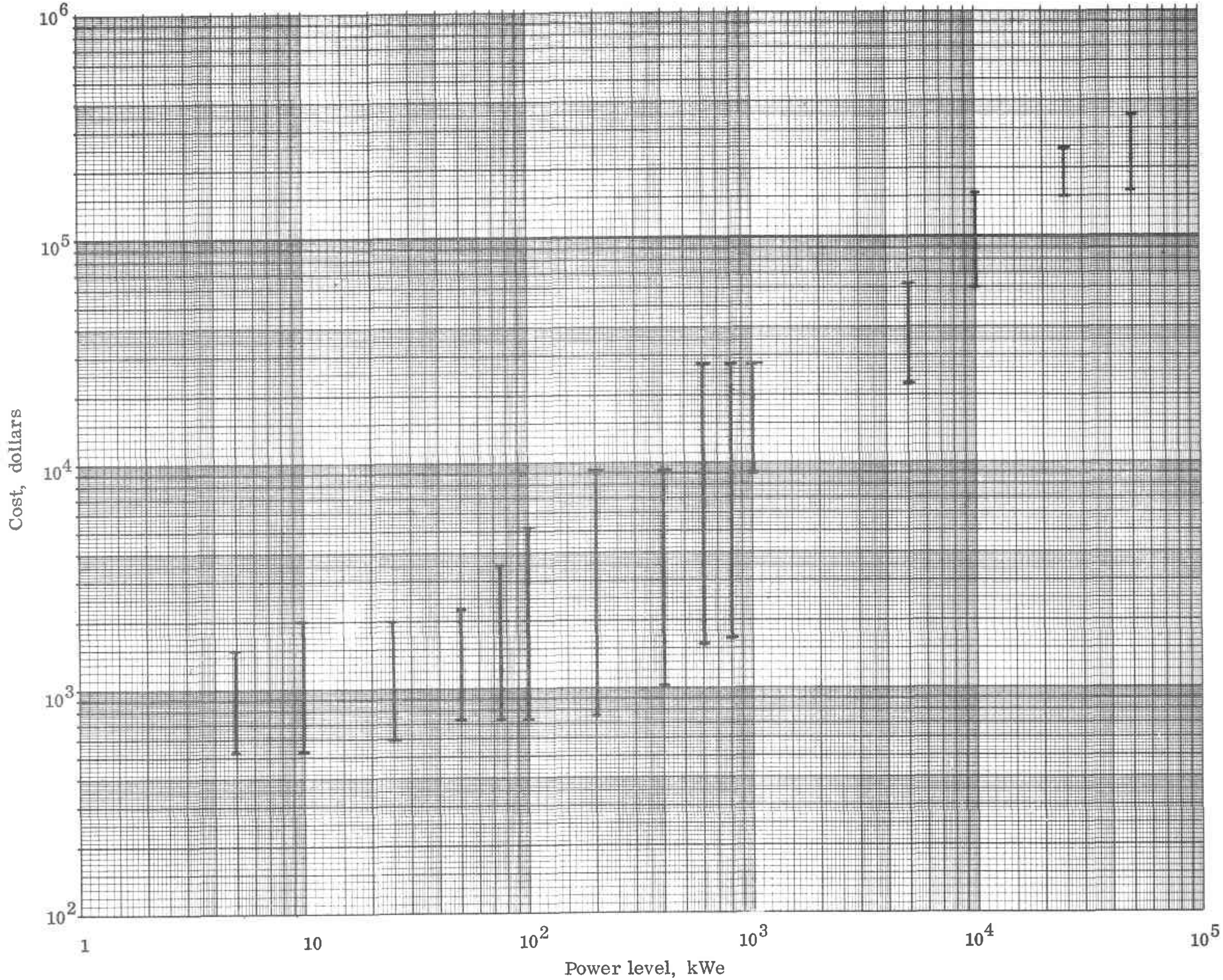


Figure 3-11. - Cost of Rankine-cycle steam turbines as function of power level.

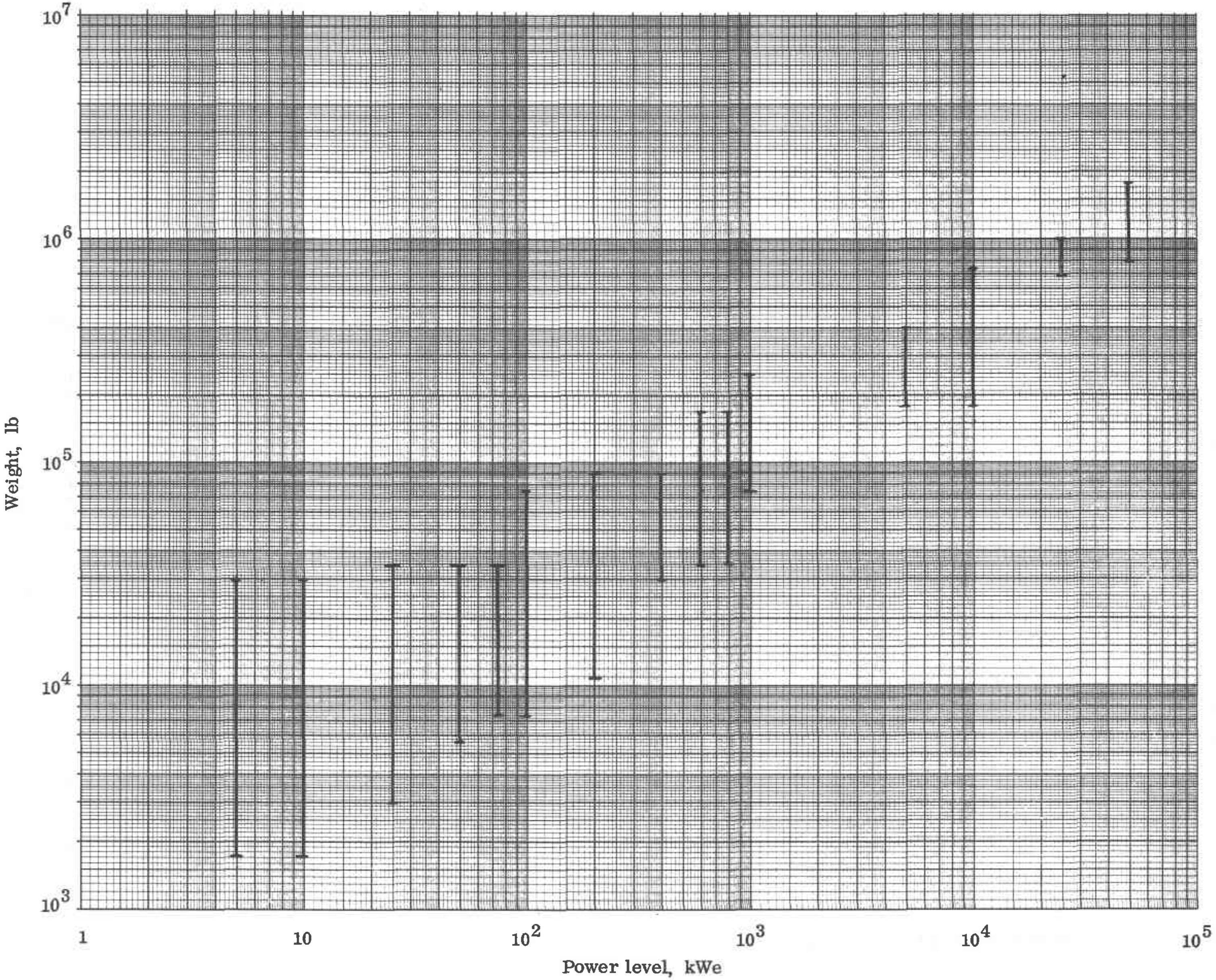


Figure 3-12. - Weight of Rankine-cycle steam turbines as function of power level.

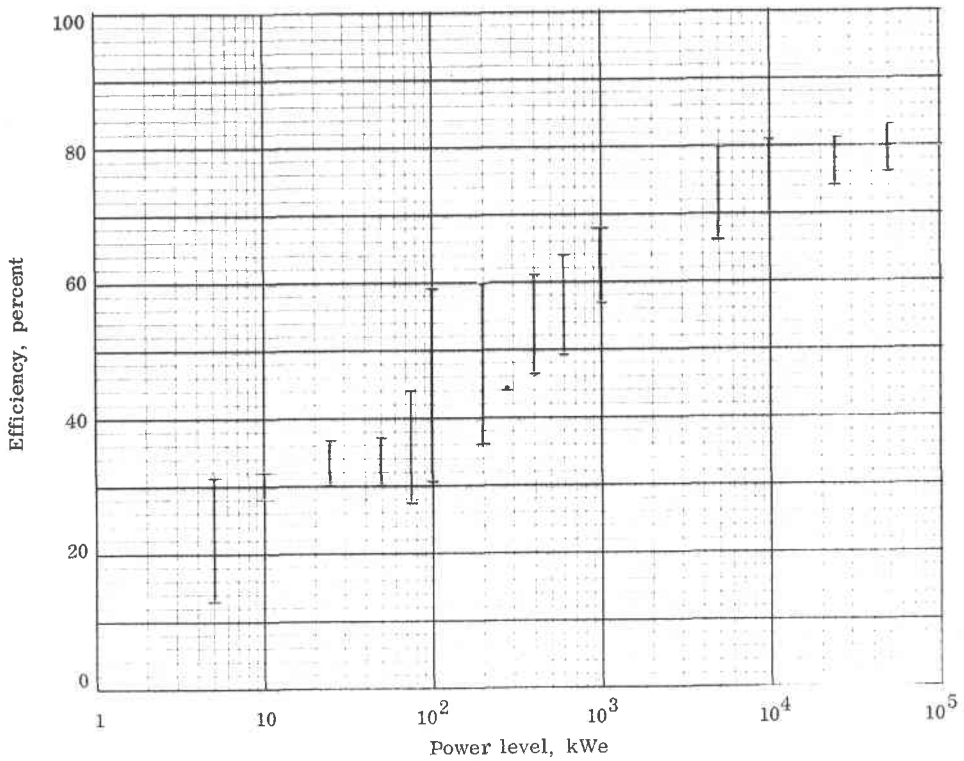


Figure 3-13. - Efficiency of Rankine-cycle steam turbines at full load as function of power level.

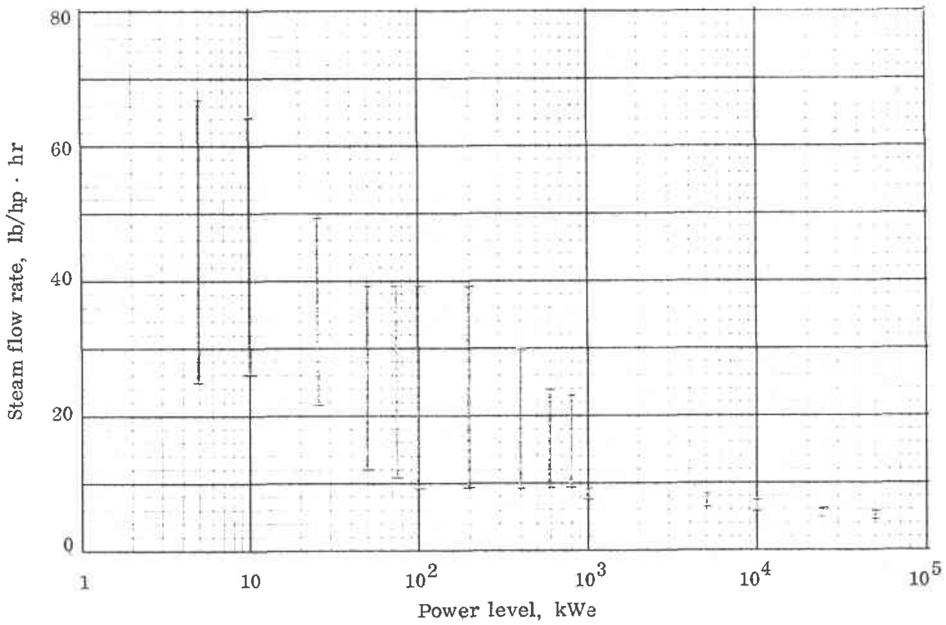


Figure 3-14. - Steam flow rate for Rankine-cycle steam turbines as function of power level - at full load.

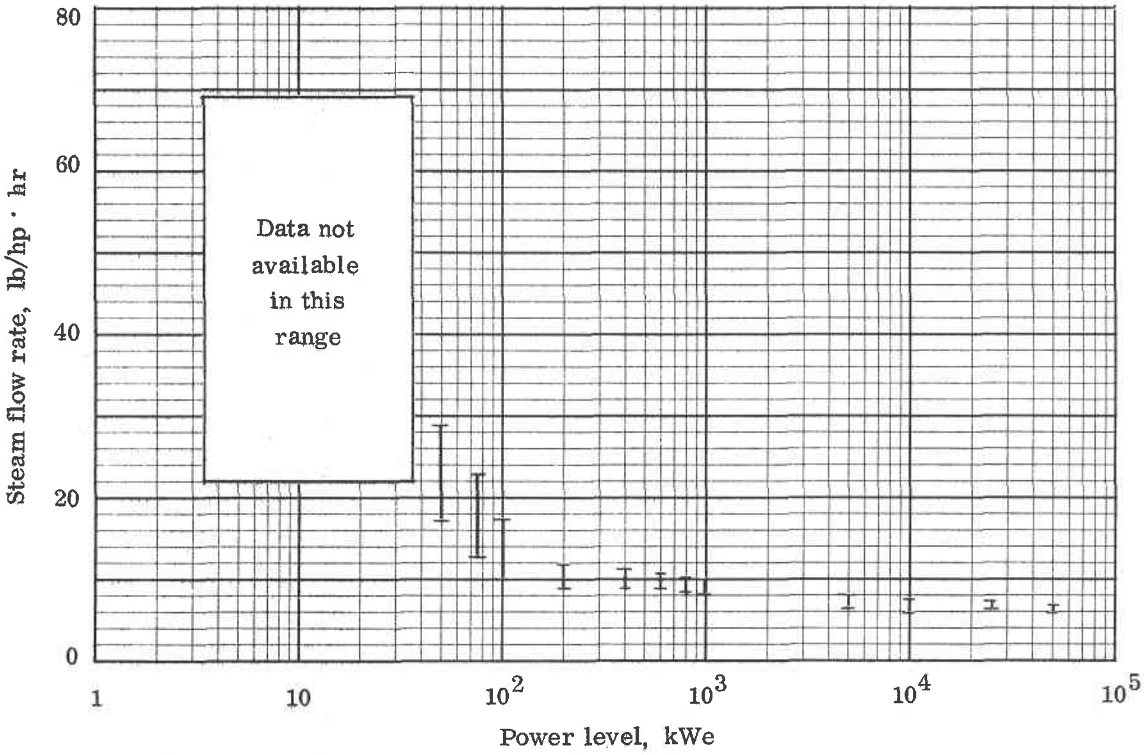


Figure 3-15. - Steam flow rate for Rankine-cycle steam turbines as function of power level - at 75-percent load.

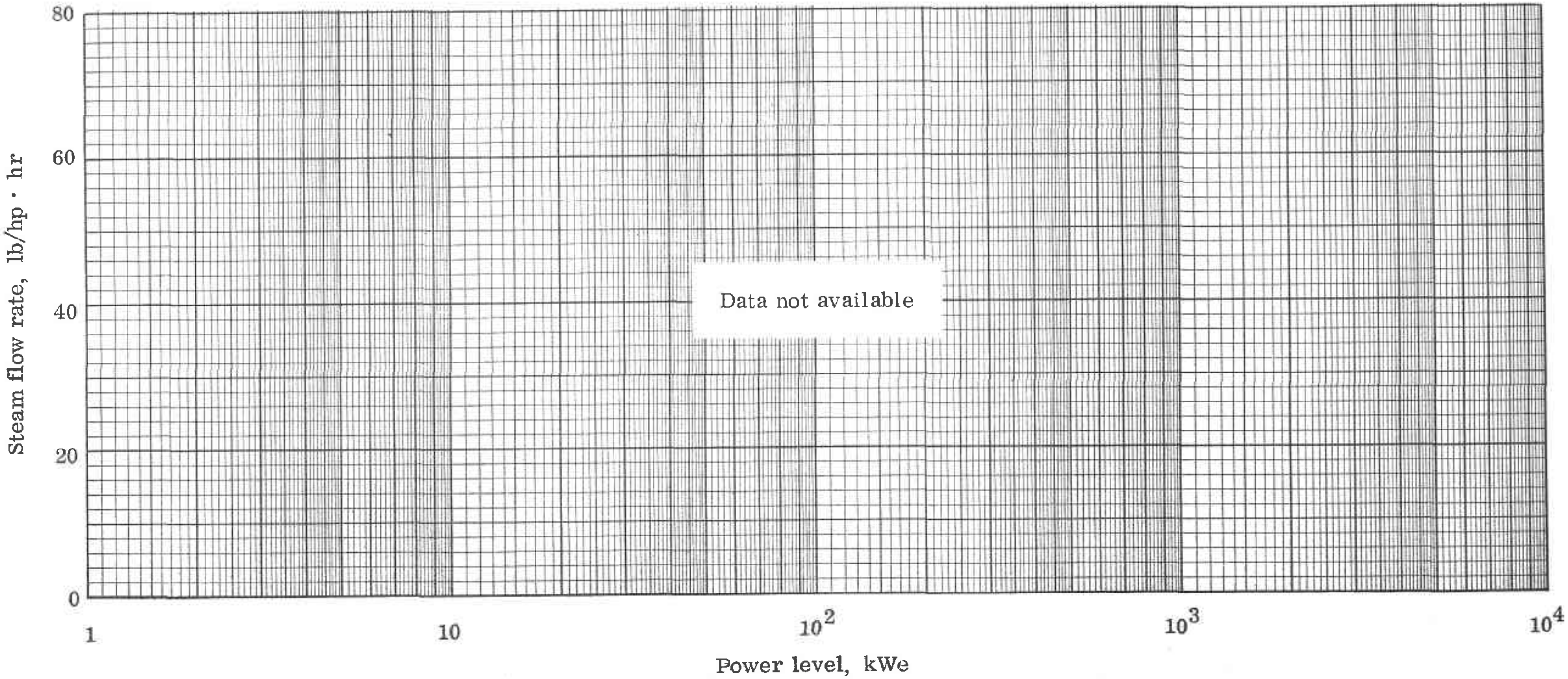


Figure 3-16. - Steam flow rate for Rankine-cycle steam turbines as function of power level - at 50-percent load.

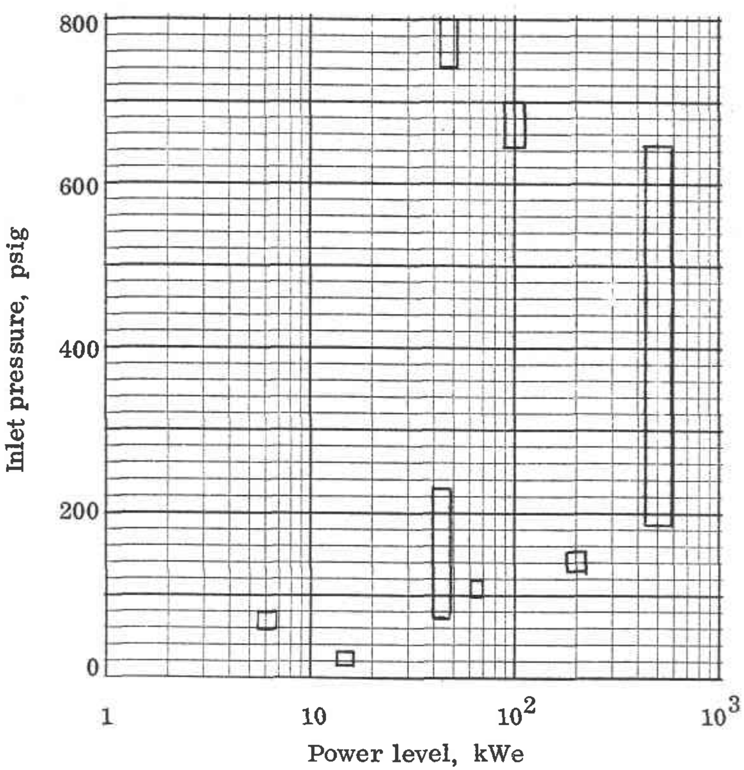


Figure 3-17. - Inlet pressure of Rankine-cycle organic turbines as function of power level.

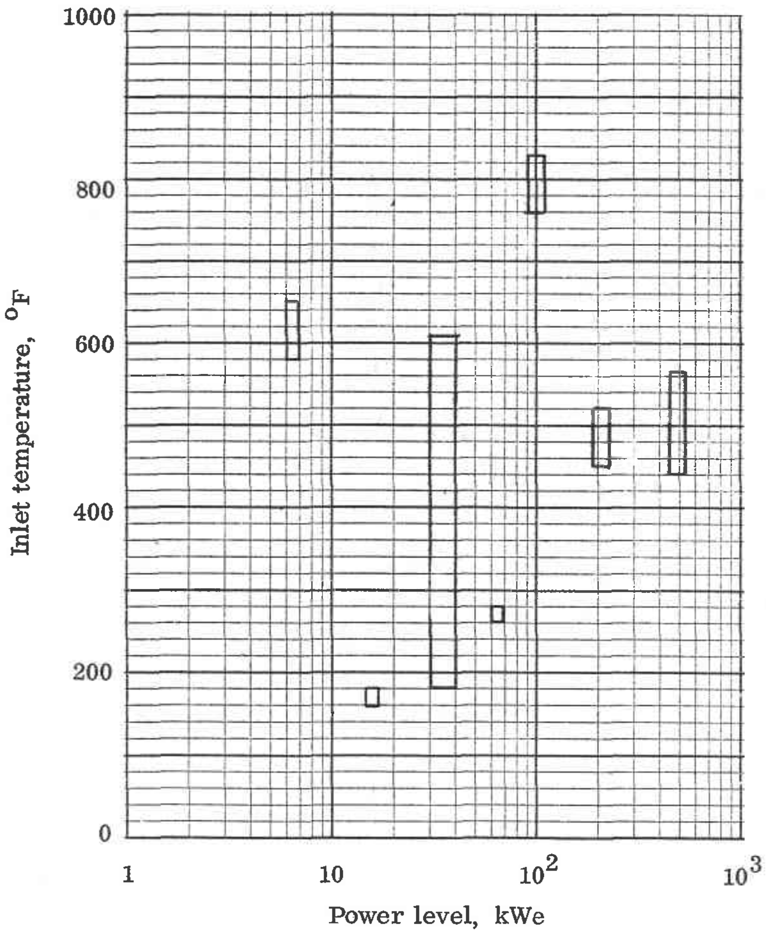


Figure 3-18. - Inlet temperature of Rankine-cycle organic turbines as function of power level.

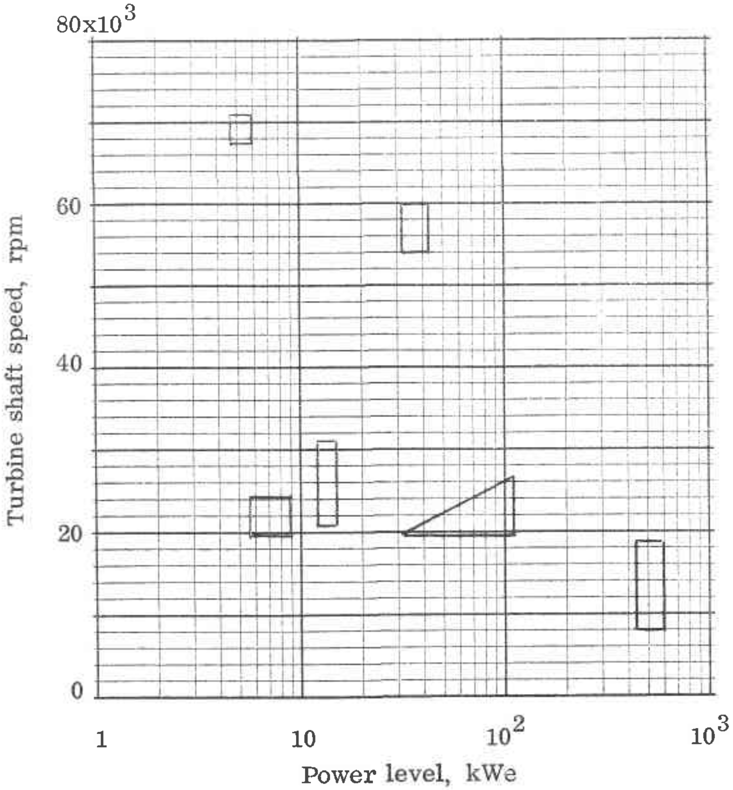


Figure 3-19. - Speed of Rankine-cycle organic turbines as function of power level.

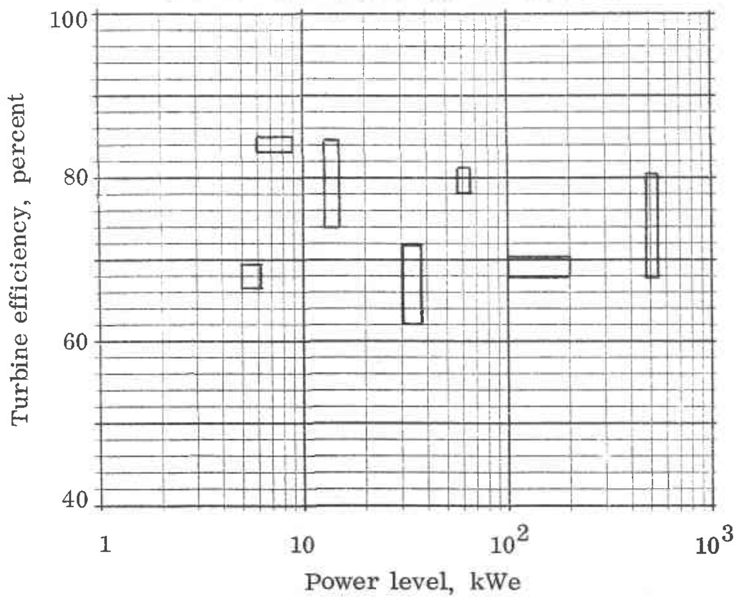


Figure 3-20. - Efficiency of Rankine-cycle organic turbines as function of power level. (Working fluids include Freon-11, FL-50, FL-85, and R-113. All units are developmental, prototype, or preproduction. Turbines are designed for specific operating condition. Turbine outputs do not include generator or gear losses.)

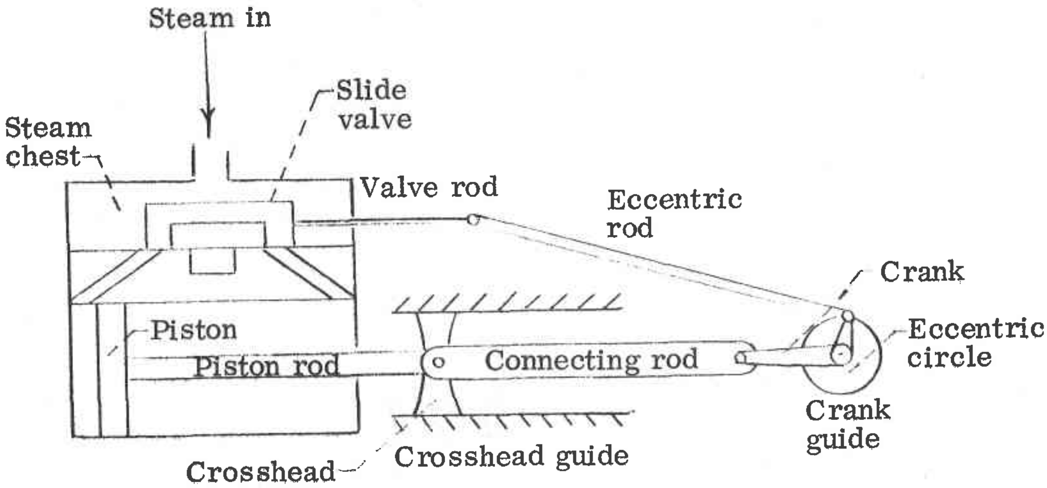
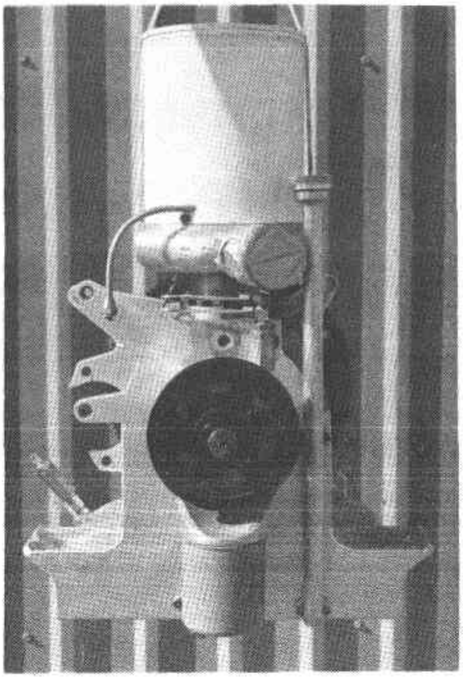
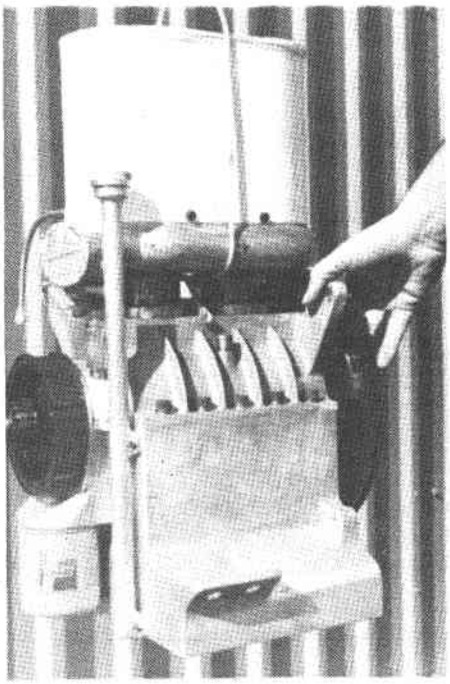


Figure 3-21. - Schematic of basic steam reciprocating expansion engine.



(a) Front view.



(b) Side view.

Figure 3-22. - Vertical uniflow steam turbine engine (15 to 74 shp). (Courtesy of Jay Carter Enterprises, Inc.)

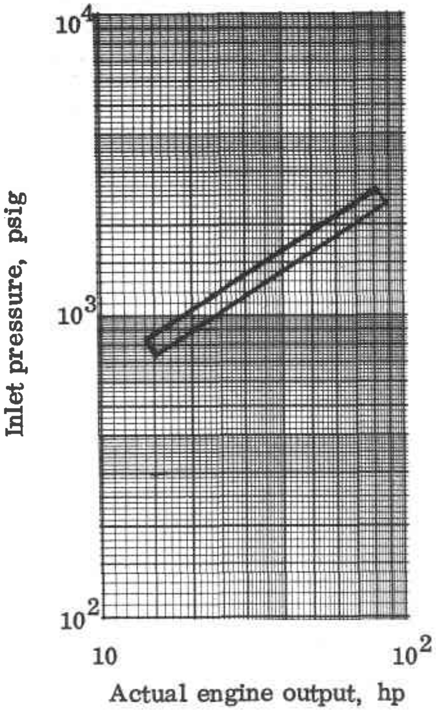


Figure 3-23. - Inlet pressure of steam and organic reciprocating expansion engines as function of actual engine output.

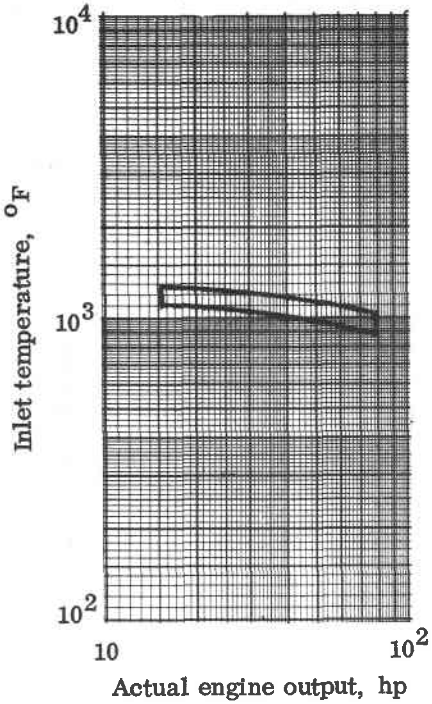


Figure 3-24. - Inlet temperature of steam and organic reciprocating expansion engines as function of actual engine output.

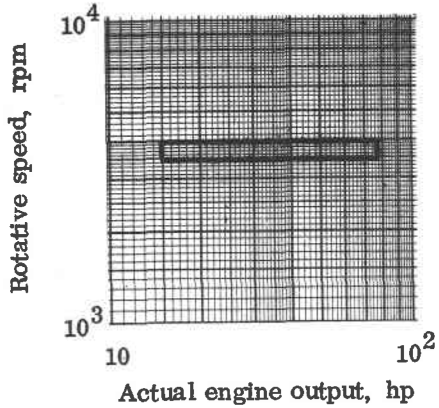


Figure 3-25. - Rotative speed of steam and organic reciprocating expansion engines as function of actual engine output.

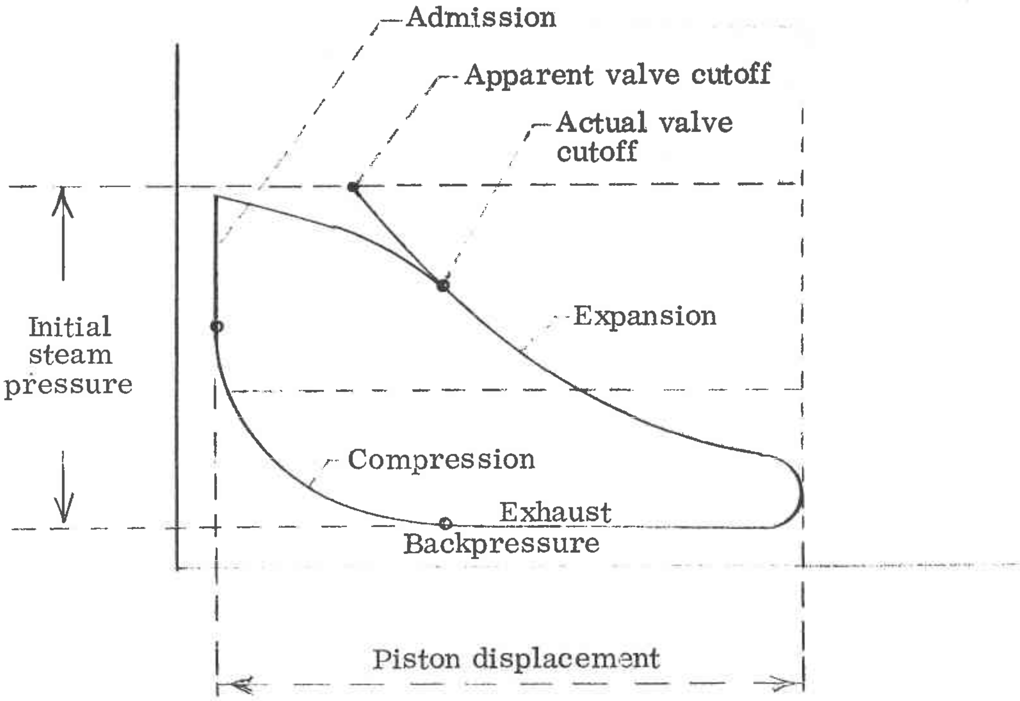


Figure 3-26. - Indicator diagram for steam and organic reciprocating expansion engines.

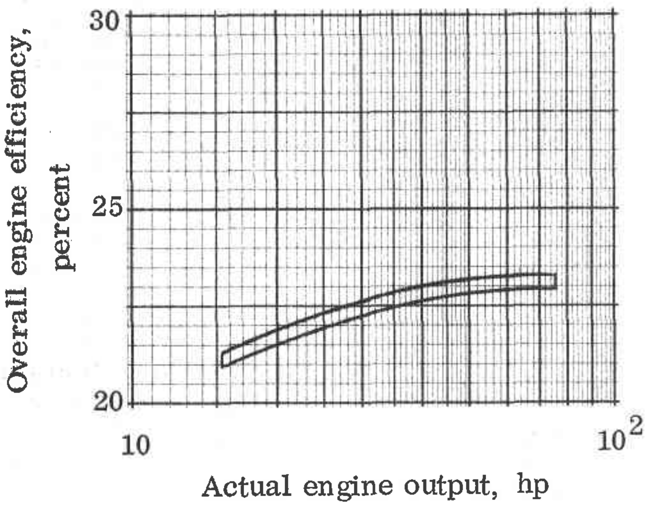


Figure 3-27. - Overall efficiency of steam and organic reciprocating expansion engines as function of actual engine output.

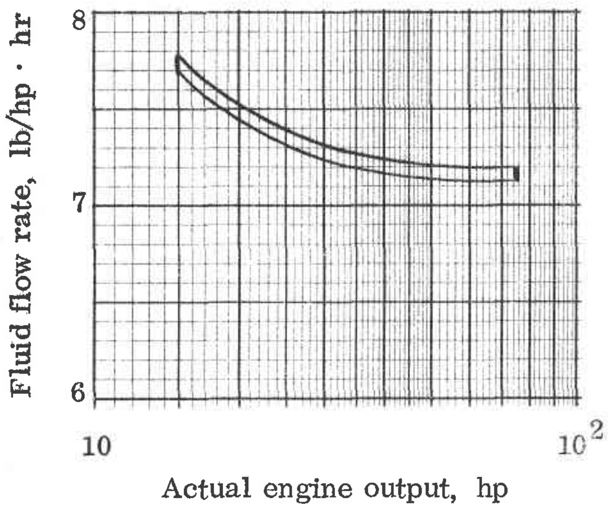


Figure 3-28. - Fluid flow rate of steam and organic reciprocating expansion engines as function of actual engine output.

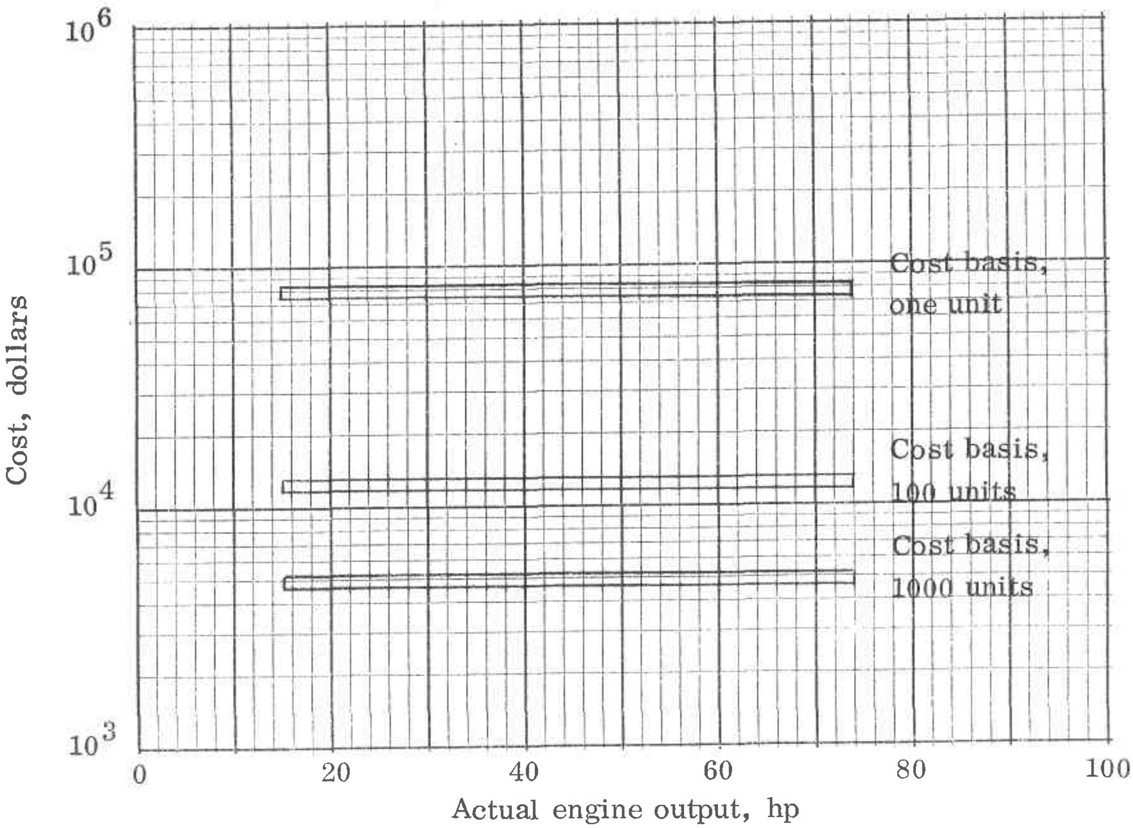


Figure 3-29. - Cost of steam and organic reciprocating engines as function of actual engine output.

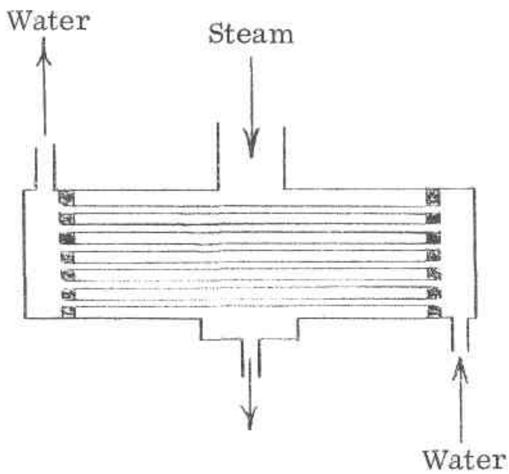


Figure 3-30. - Schematic of typical surface condenser.

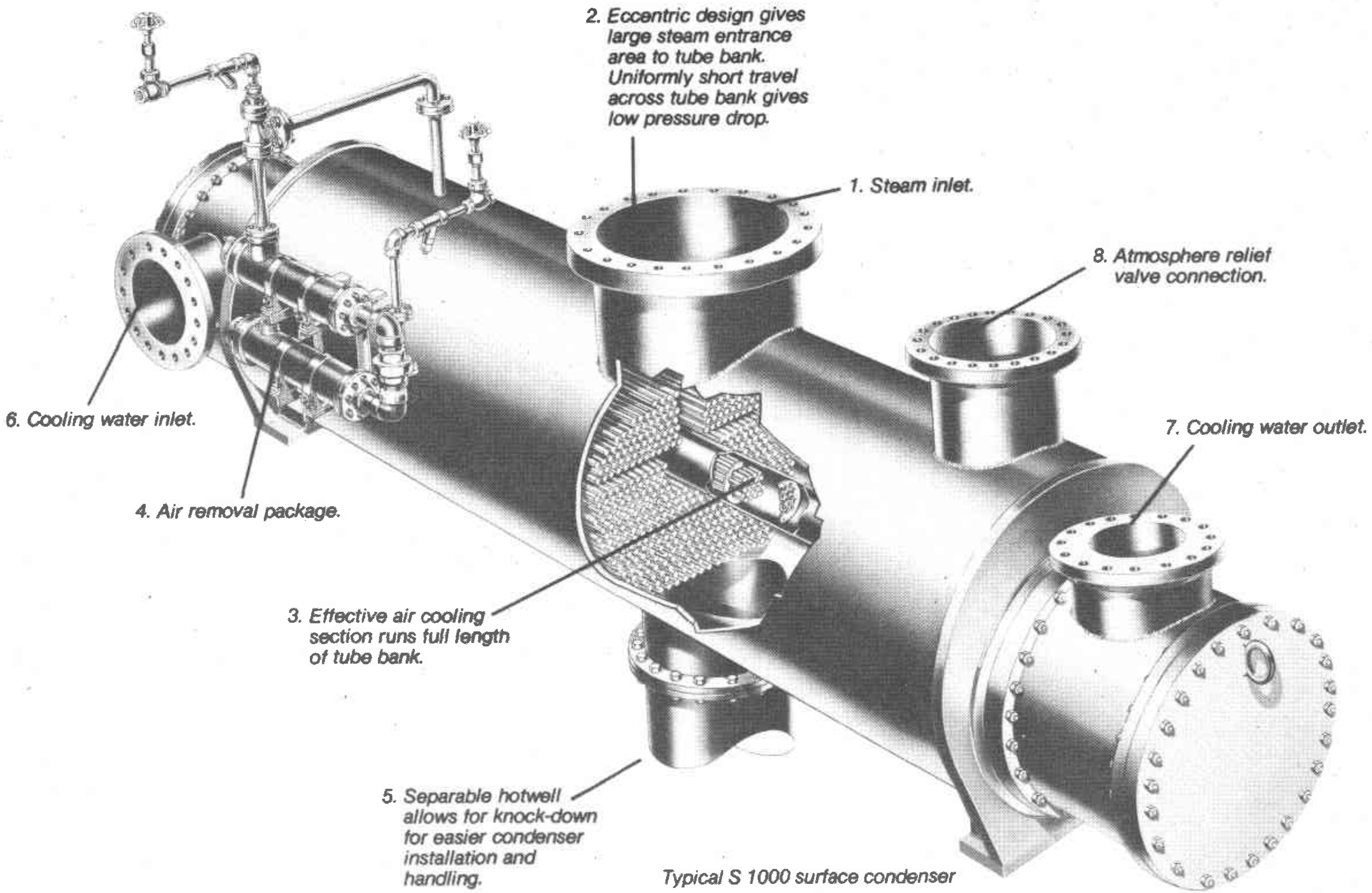
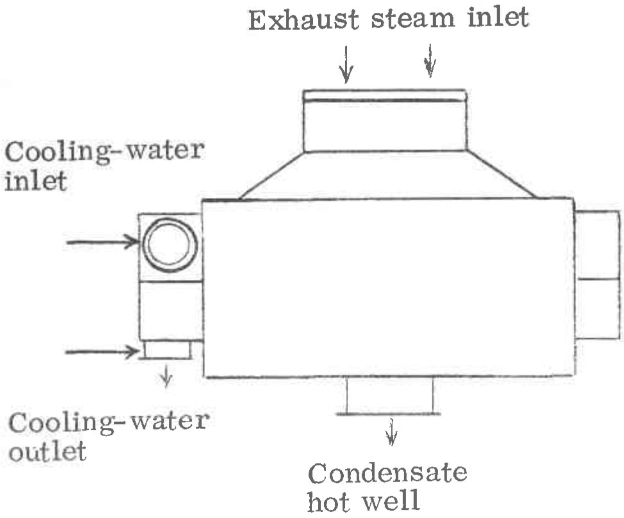
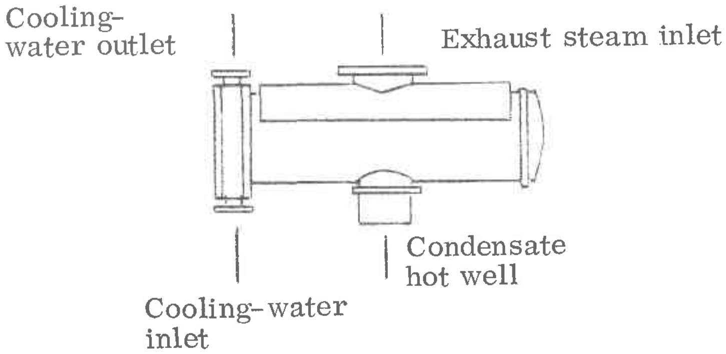


Figure 3-31. - Commercial surface steam condenser. (Courtesy of Heat Transfer Division, American-Standard.)

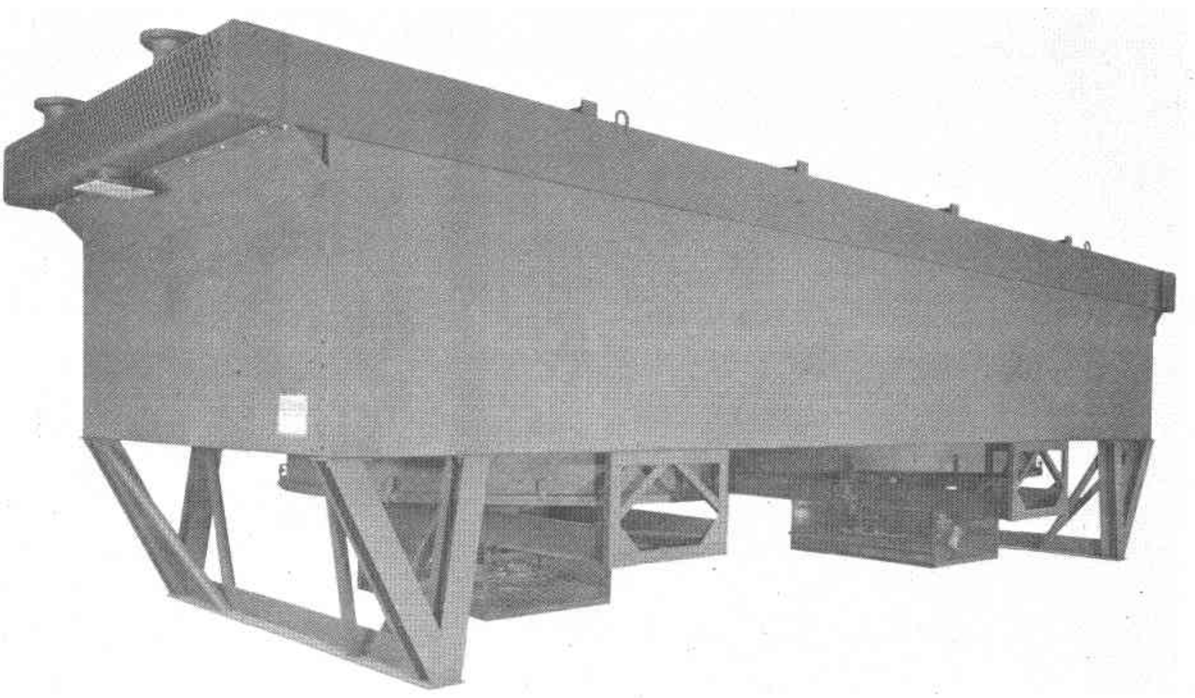


(a) With top cooling-water inlet.

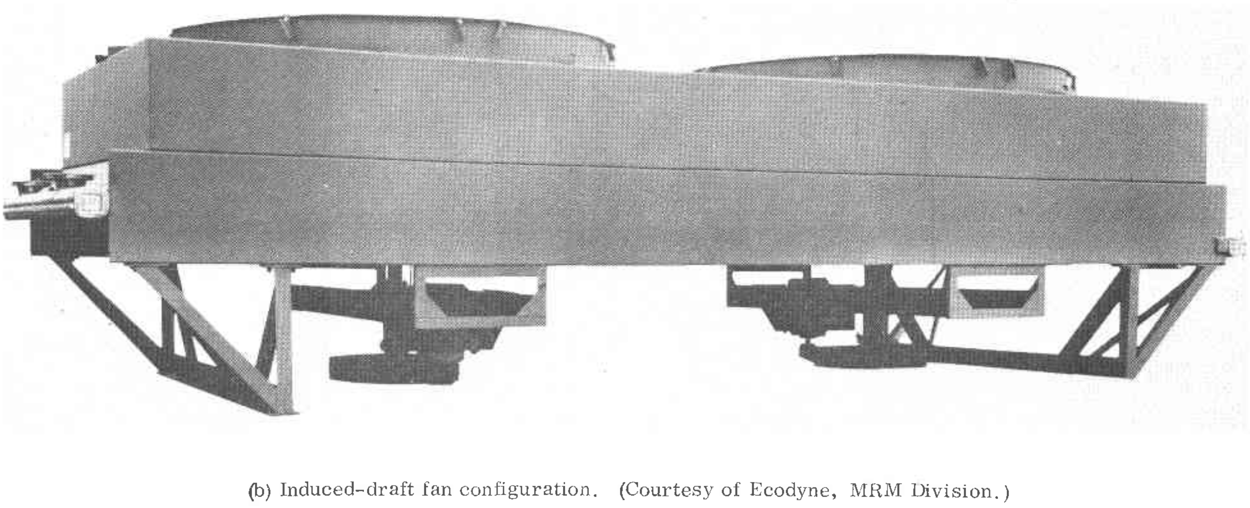


(b) With bottom cooling-water inlet.

Figure 3-32. - Multipass condensers.



(a) Forced-draft fan configuration. (Courtesy of Ecodyne, MRM Division.)



(b) Induced-draft fan configuration. (Courtesy of Ecodyne, MRM Division.)

Figure 3-33. - Industrial air-cooled heat-exchanger configurations.

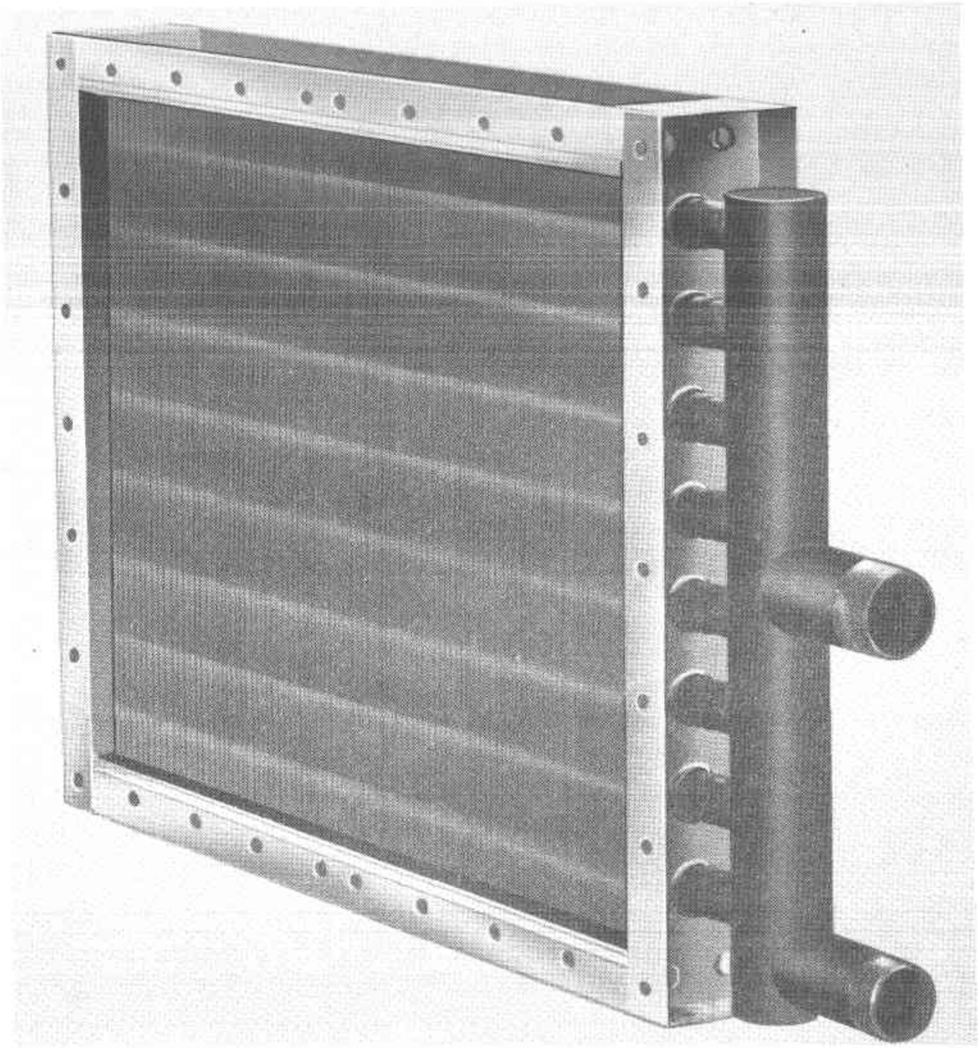
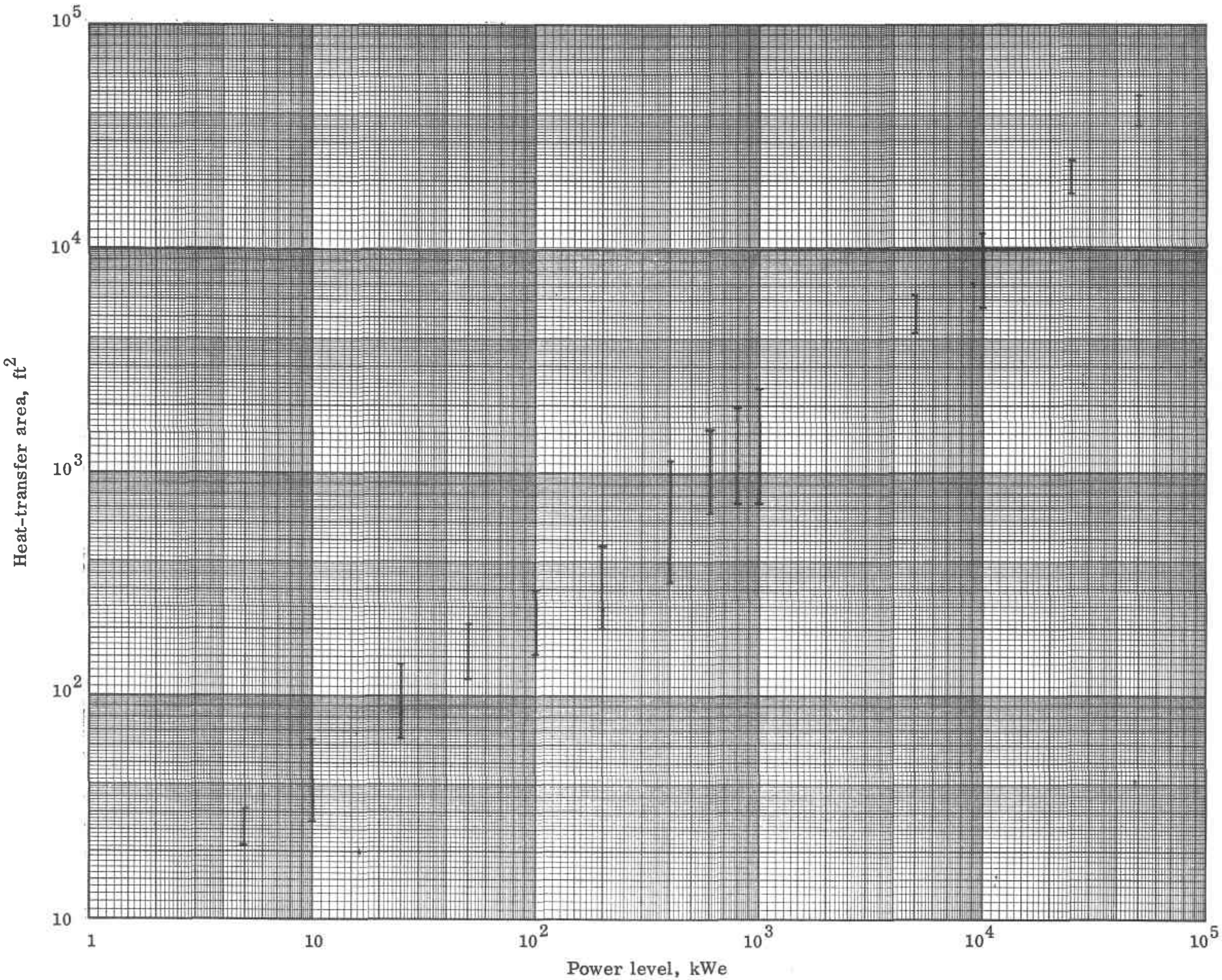
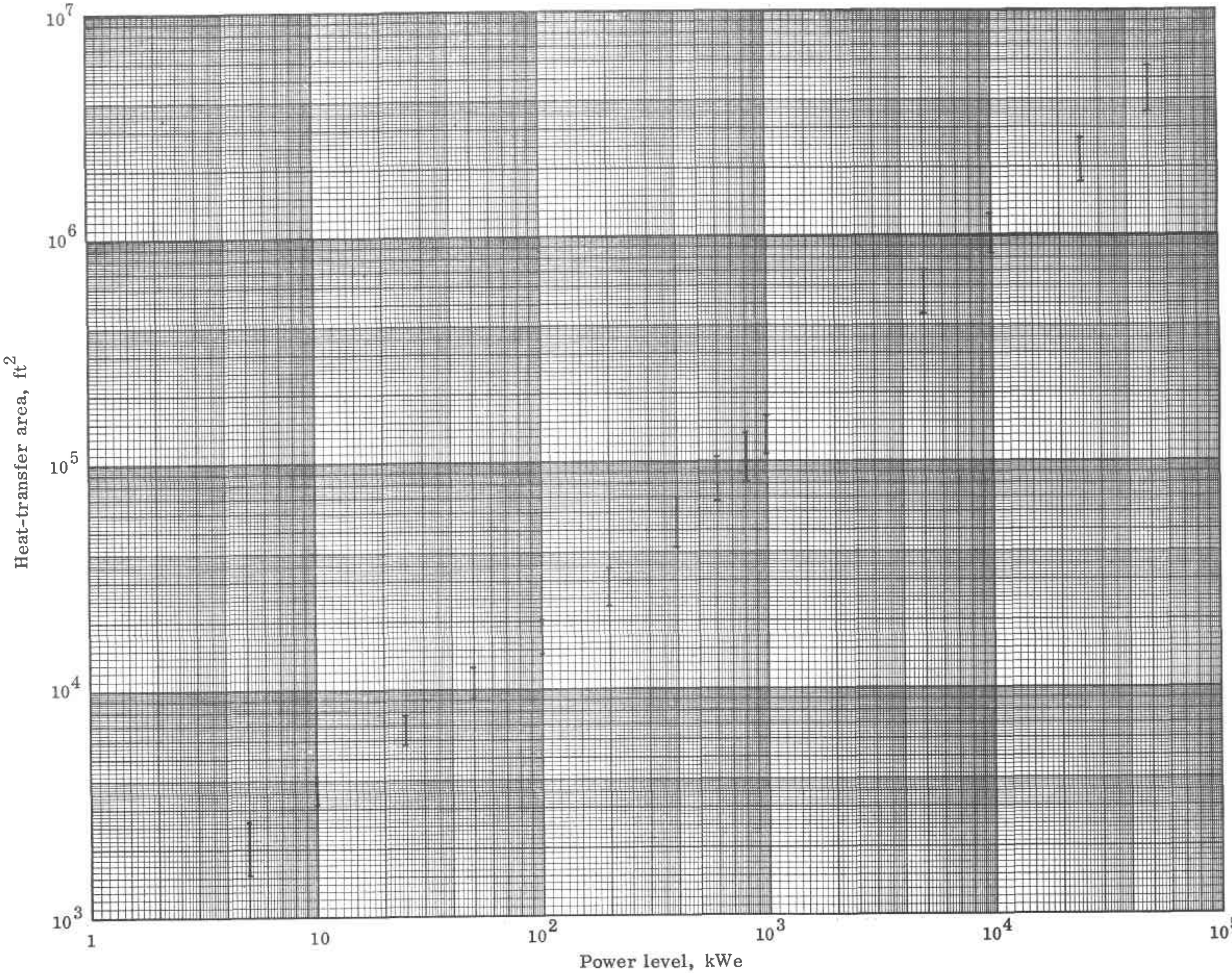


Figure 3-34. - Finned condensing-tube - header assembly with nonfreezing steam coils. (Courtesy of Dunham-Bush, Inc.)



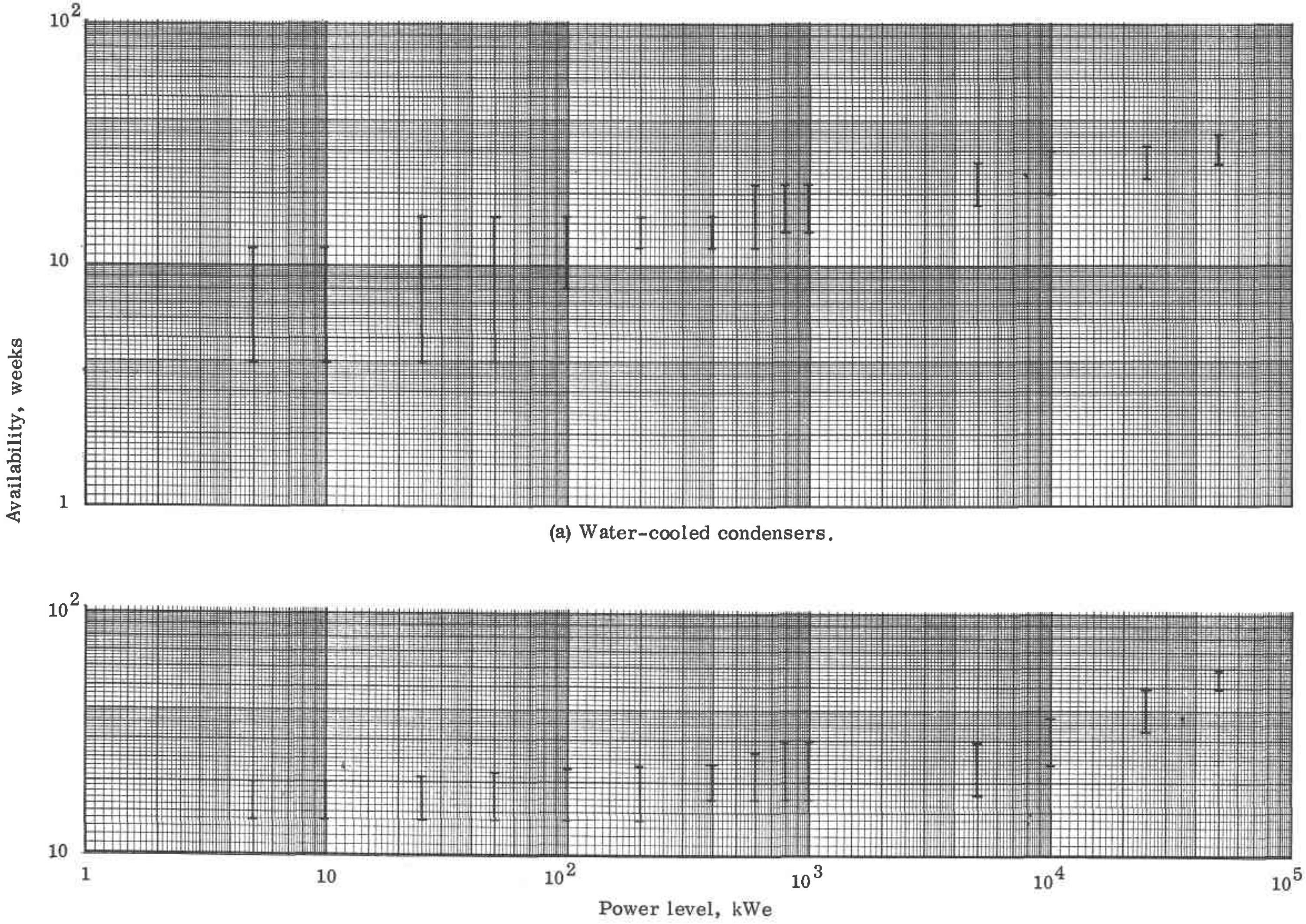
(a) Water-cooled condensers.

Figure 3-35. - Steam condenser size as function of power level.



(b) Air-cooled condensers.

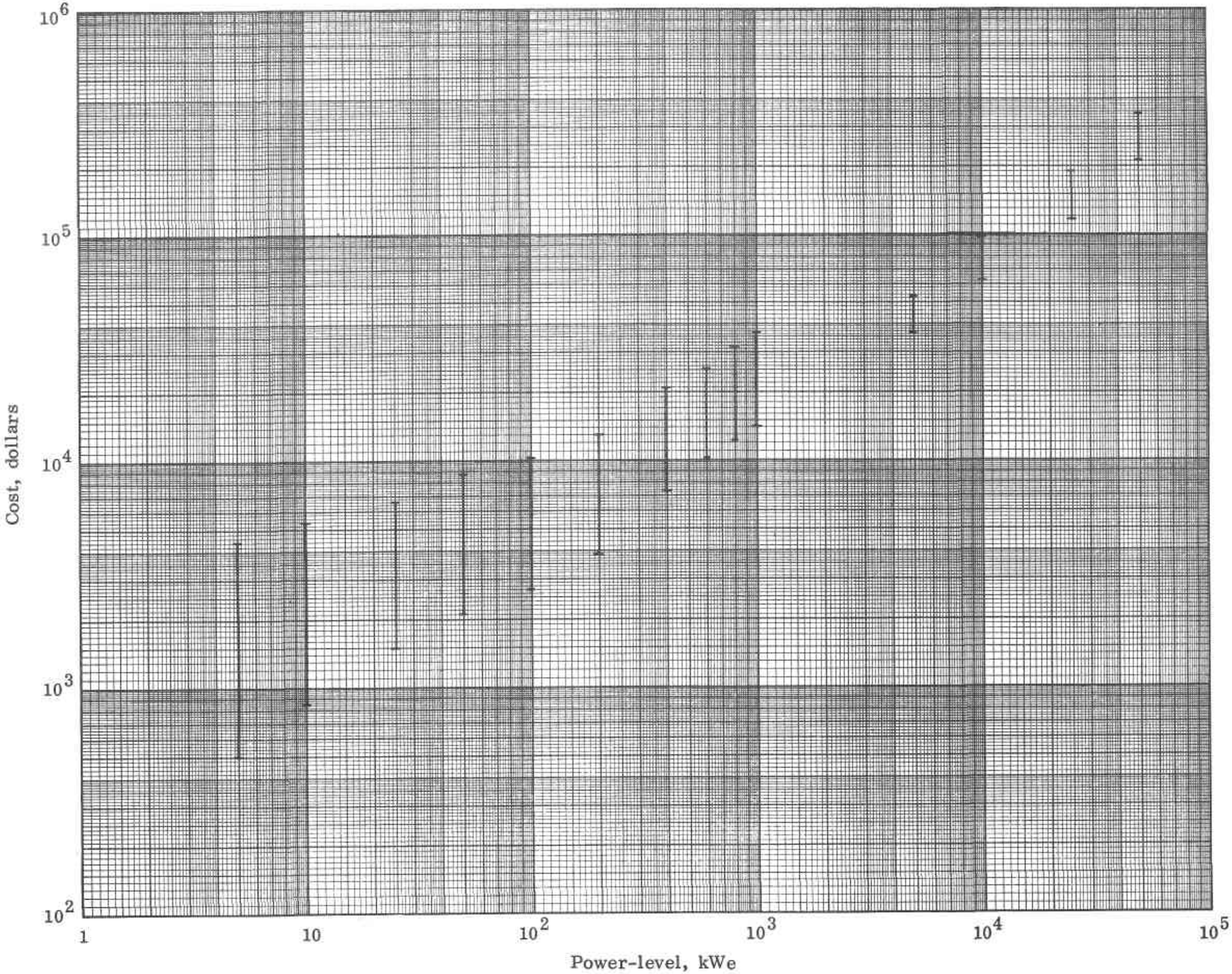
Figure 3-35. - Concluded.



(a) Water-cooled condensers.

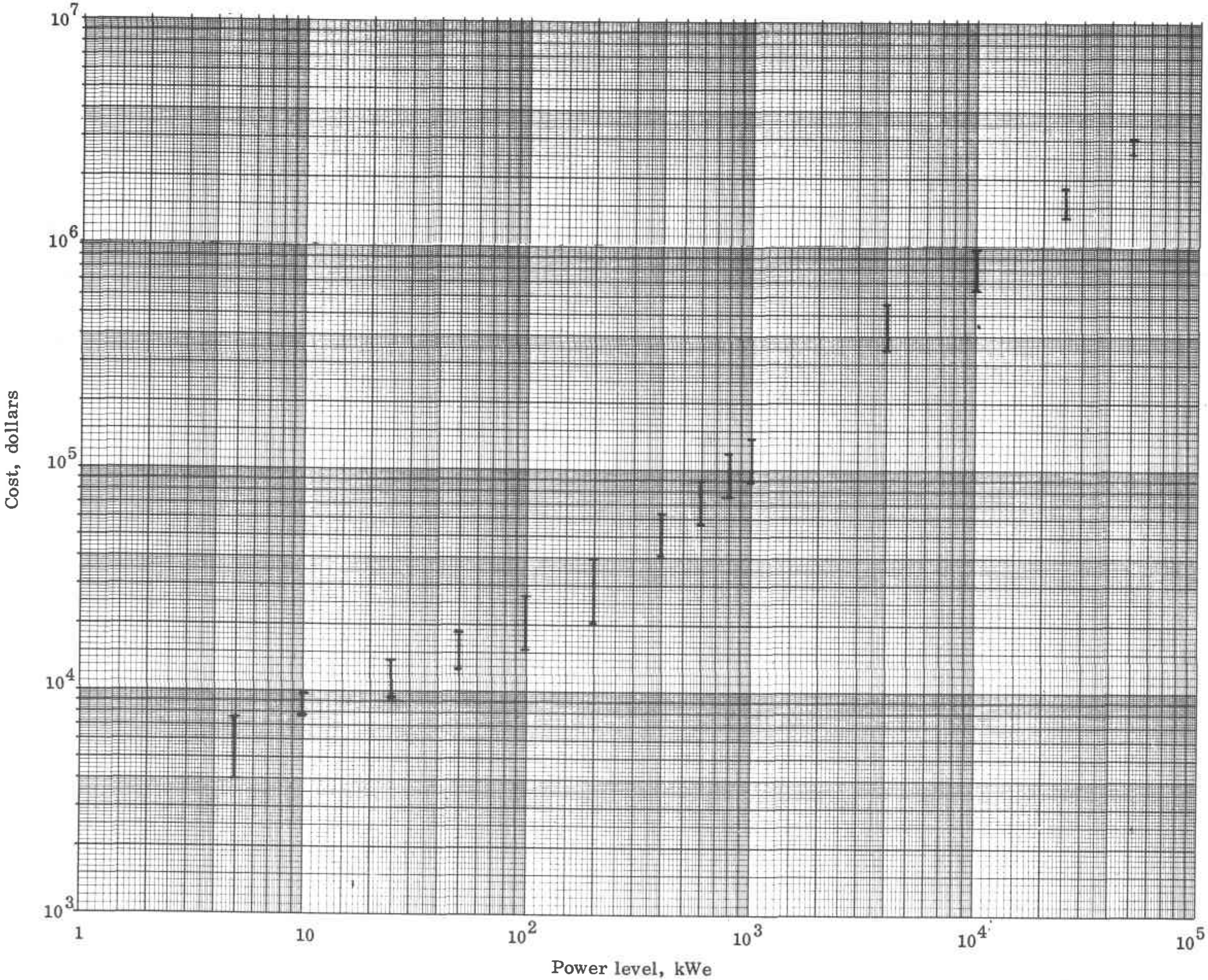
(b) Air-cooled condensers.

Figure 3-36. - Availability of steam condensers as function of power level.



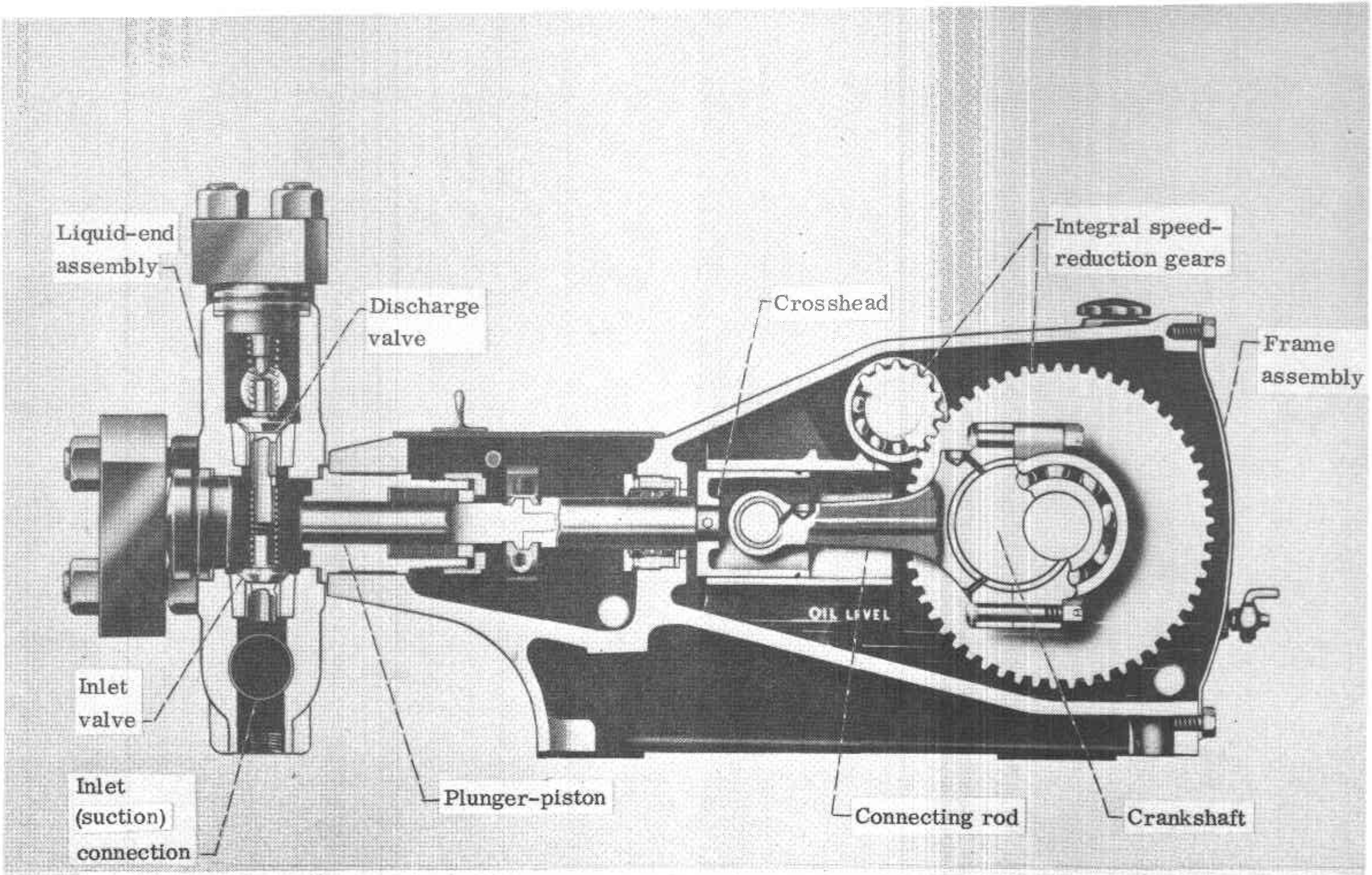
(a) Water-cooled condensers.

Figure 3-37. - Cost of steam condensers as function of power level.



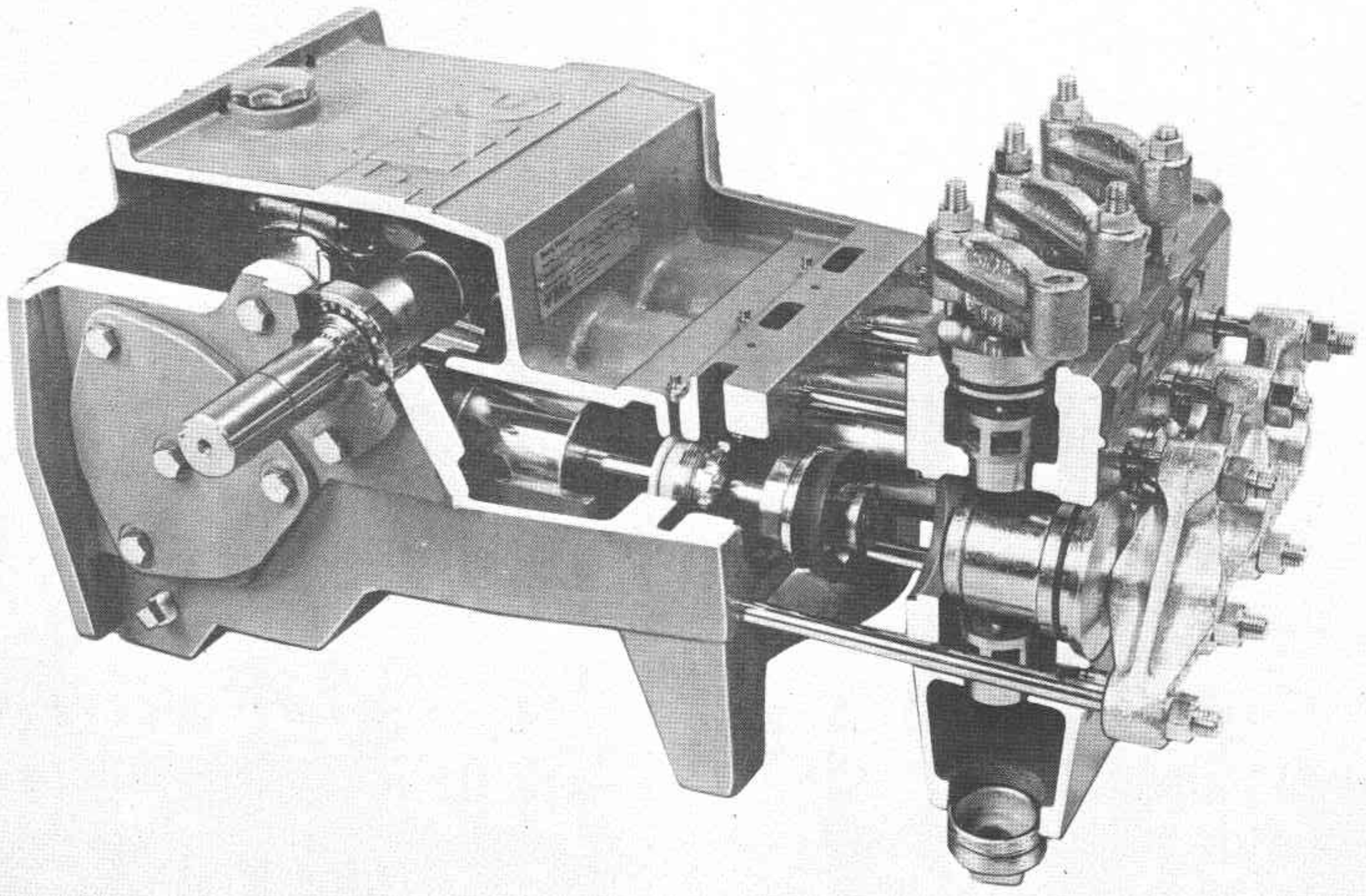
(b) Air-cooled condensers.

Figure 3-37. - Concluded.



(a) Cross section of high-pressure triplex reciprocating pump. (Courtesy FMC Corp., Agricultural Machinery Division.)

Figure 3-38. - High-pressure reciprocating pumps.



(b) Cutaway of high-pressure, triplex reciprocating pump, showing pump drive shaft. (Courtesy of FMC Corp., Agricultural Machinery Division.)

Figure 3-38. - Concluded.

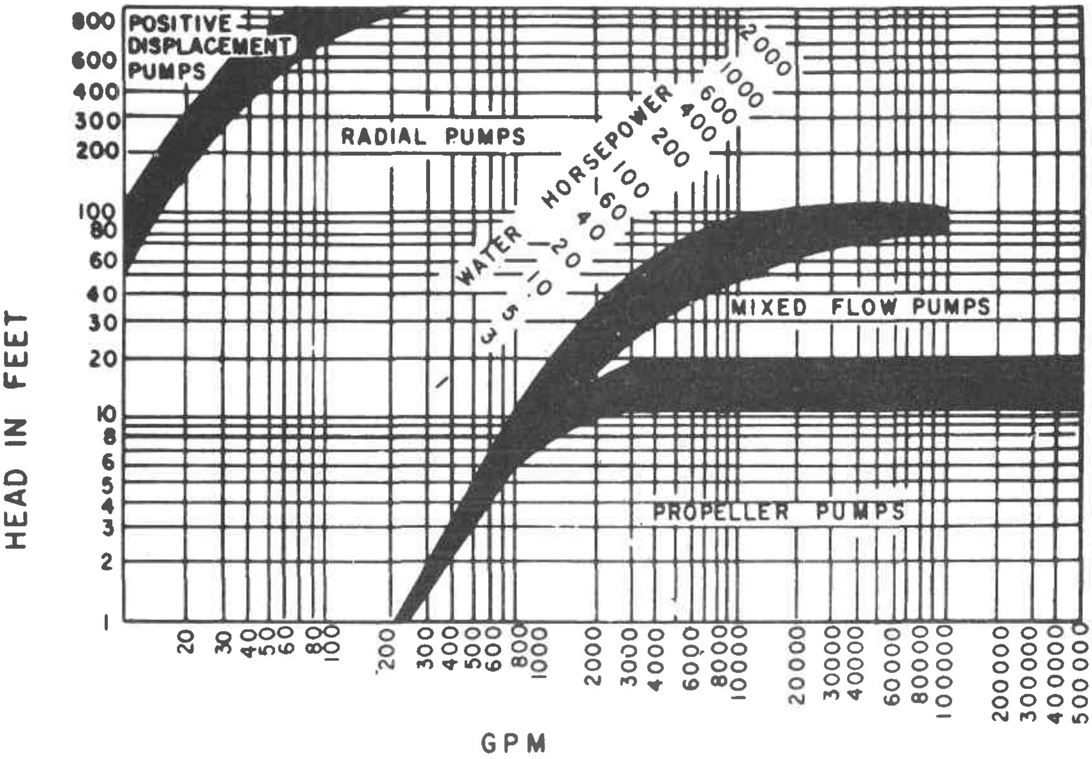


Figure 3-39. - Pump application range. (Courtesy of John Wiley & Sons, Inc.)

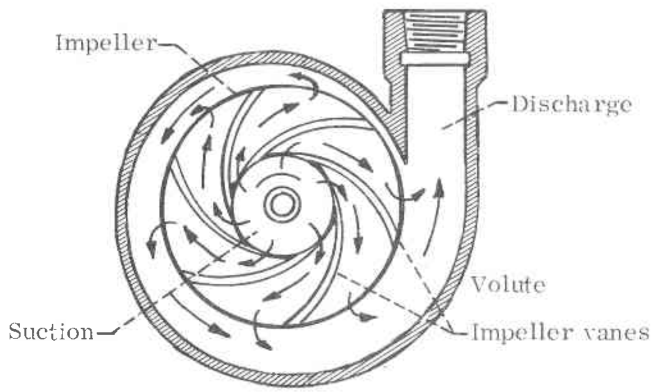
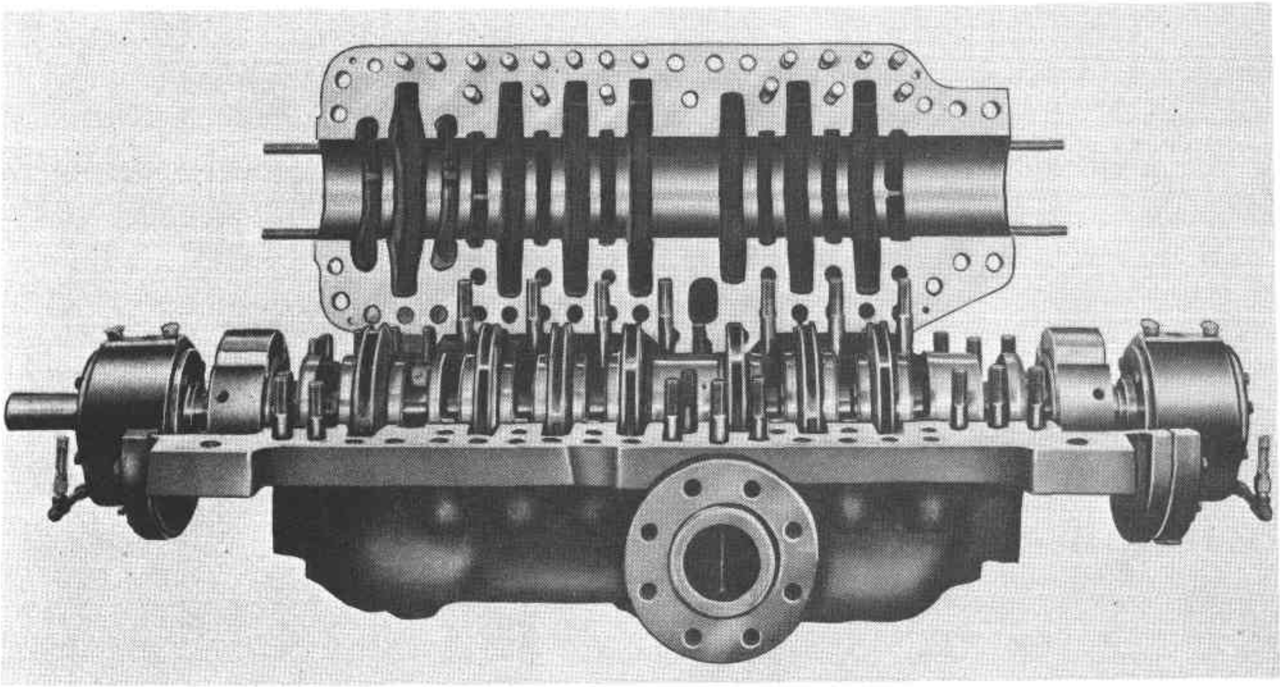
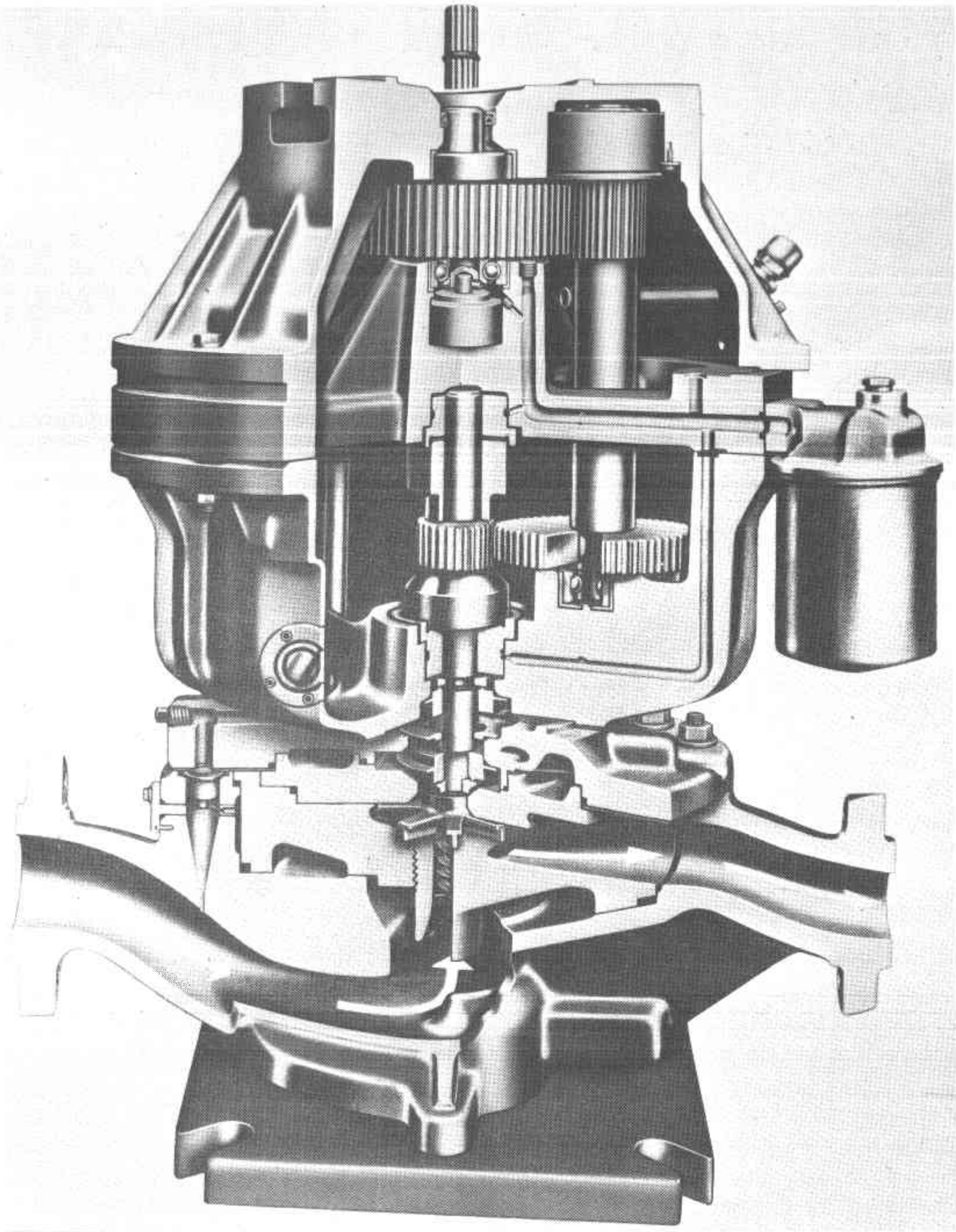


Figure 3-40. - Typical pump casing - impeller schematic.



(a) Industrial, multistage centrifugal pump. (Courtesy of Union Pump Co.)

Figure 3-41. - Centrifugal pumps.



(b) Industrial, single-stage, high-speed, forced-vortex centrifugal pump with speed increaser. (Courtesy of Sundstrand Fluid Handling, a unit of Sundstrand Corp.)

Figure 3-41. - Concluded.

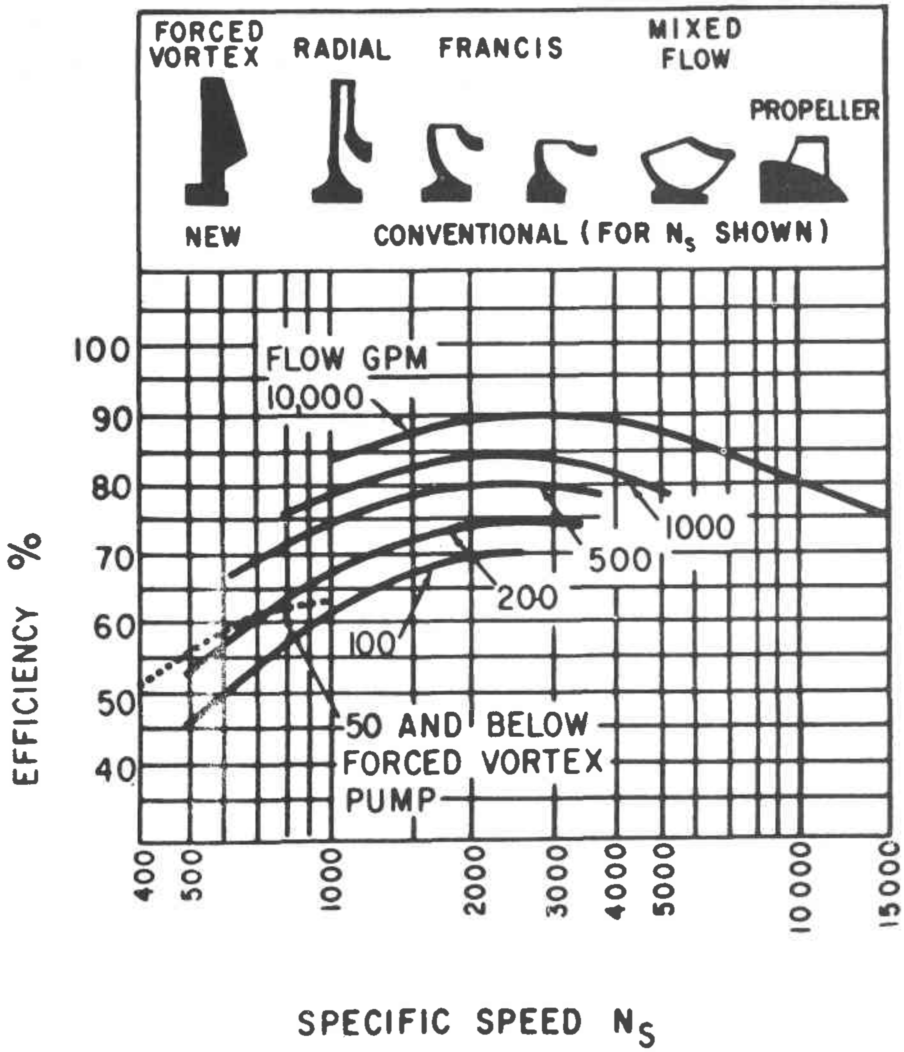


Figure 3-42. - Centrifugal pump specific-speed chart. (Courtesy of Product Engineering.)

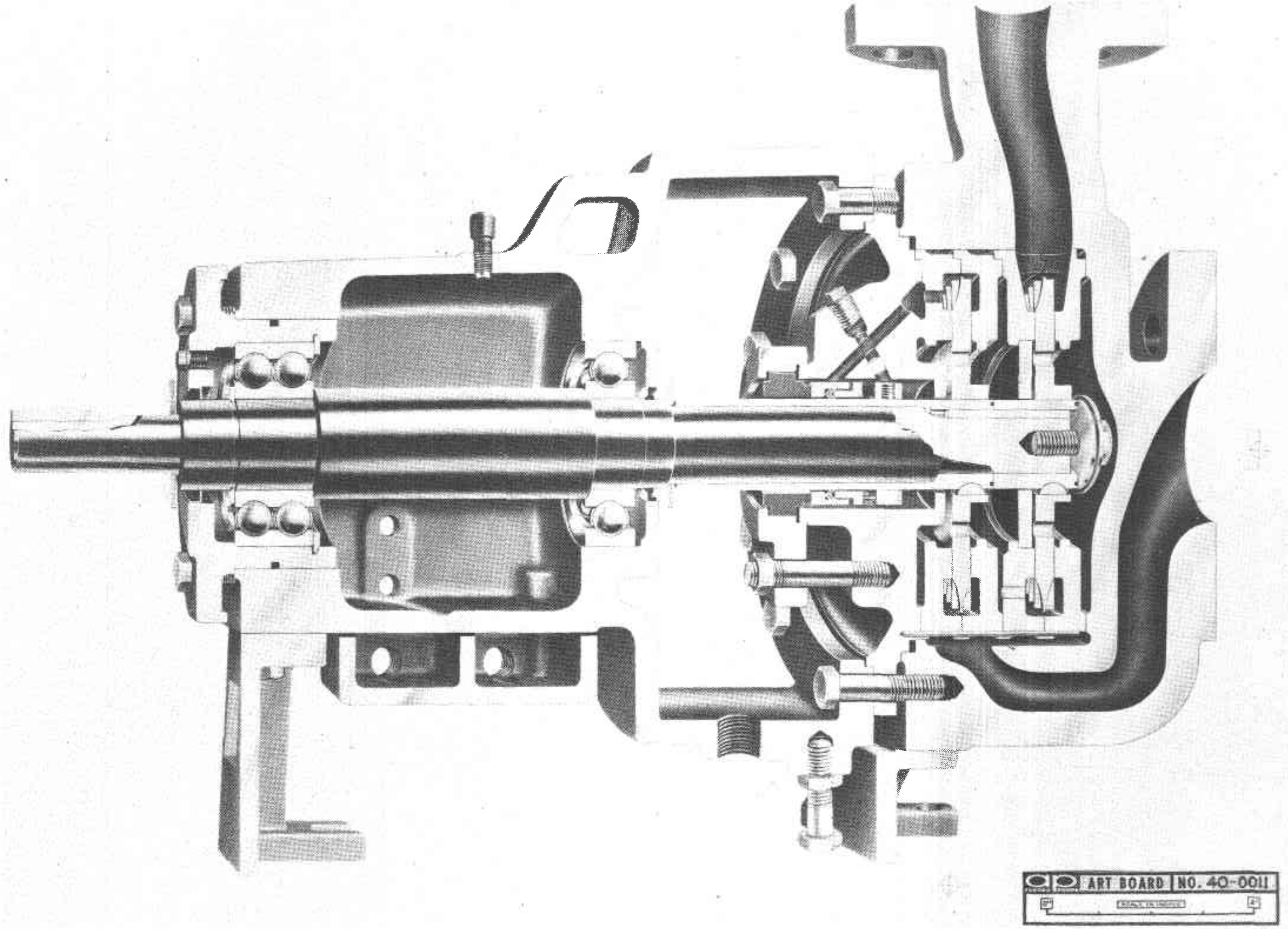


Figure 3-43. - Cross section of a two-stage industrial regenerative turbine pump. (Courtesy of Aurora Pump, a unit of General Signal.)

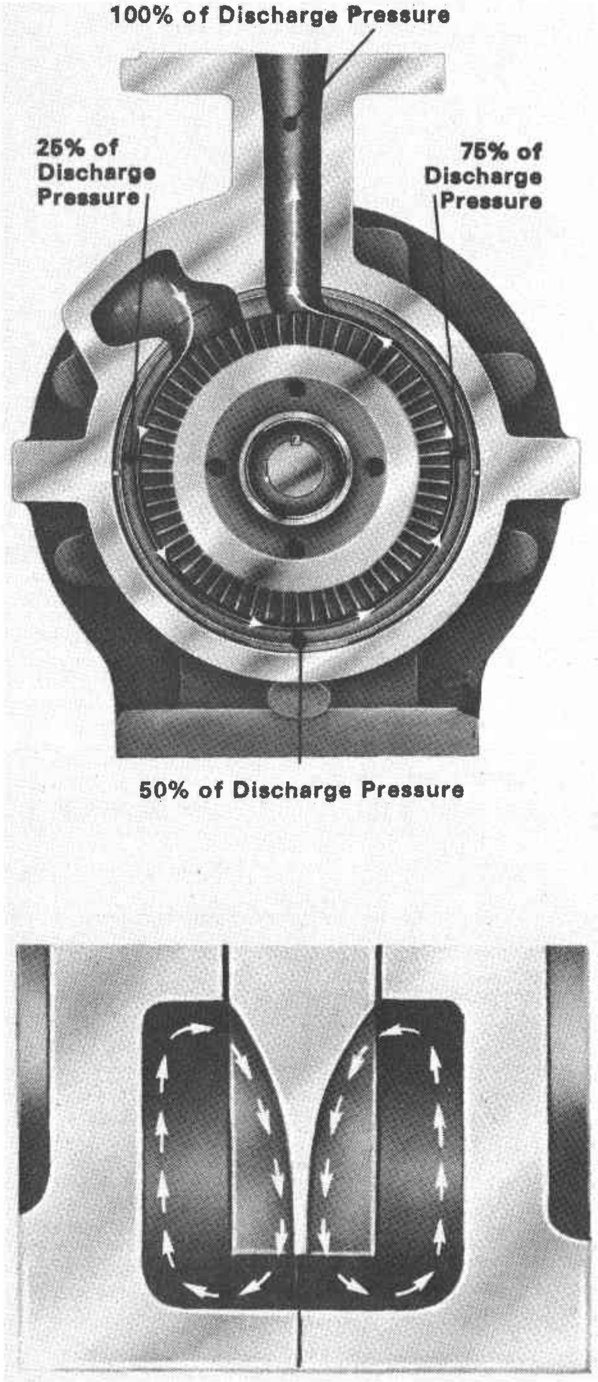


Figure 3-44. - Operating principle of regenerative turbine pump. (Courtesy of Aurora Pump, a unit of General Signal.)

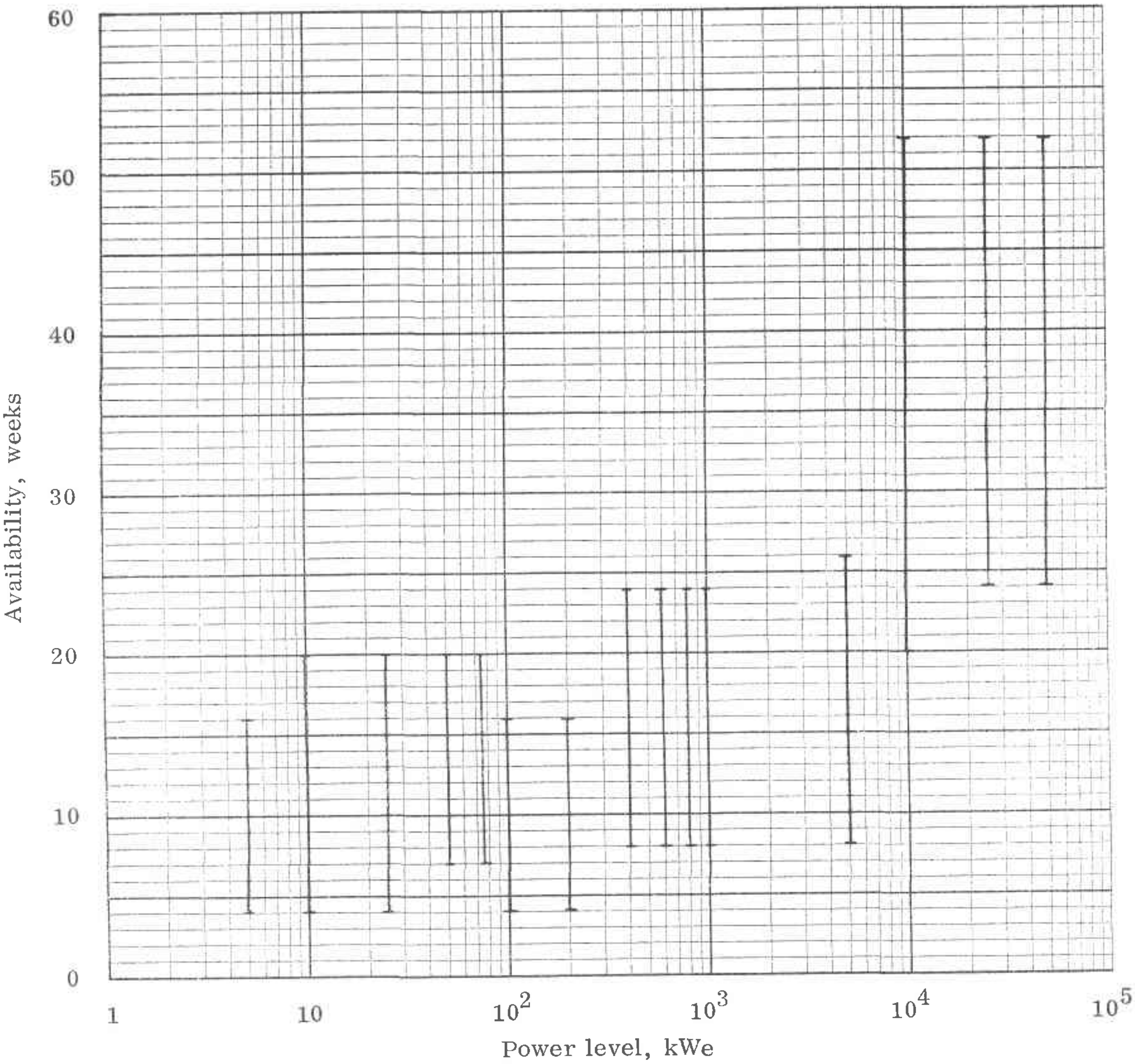


Figure 3-45. - Pump availability as function of power level.

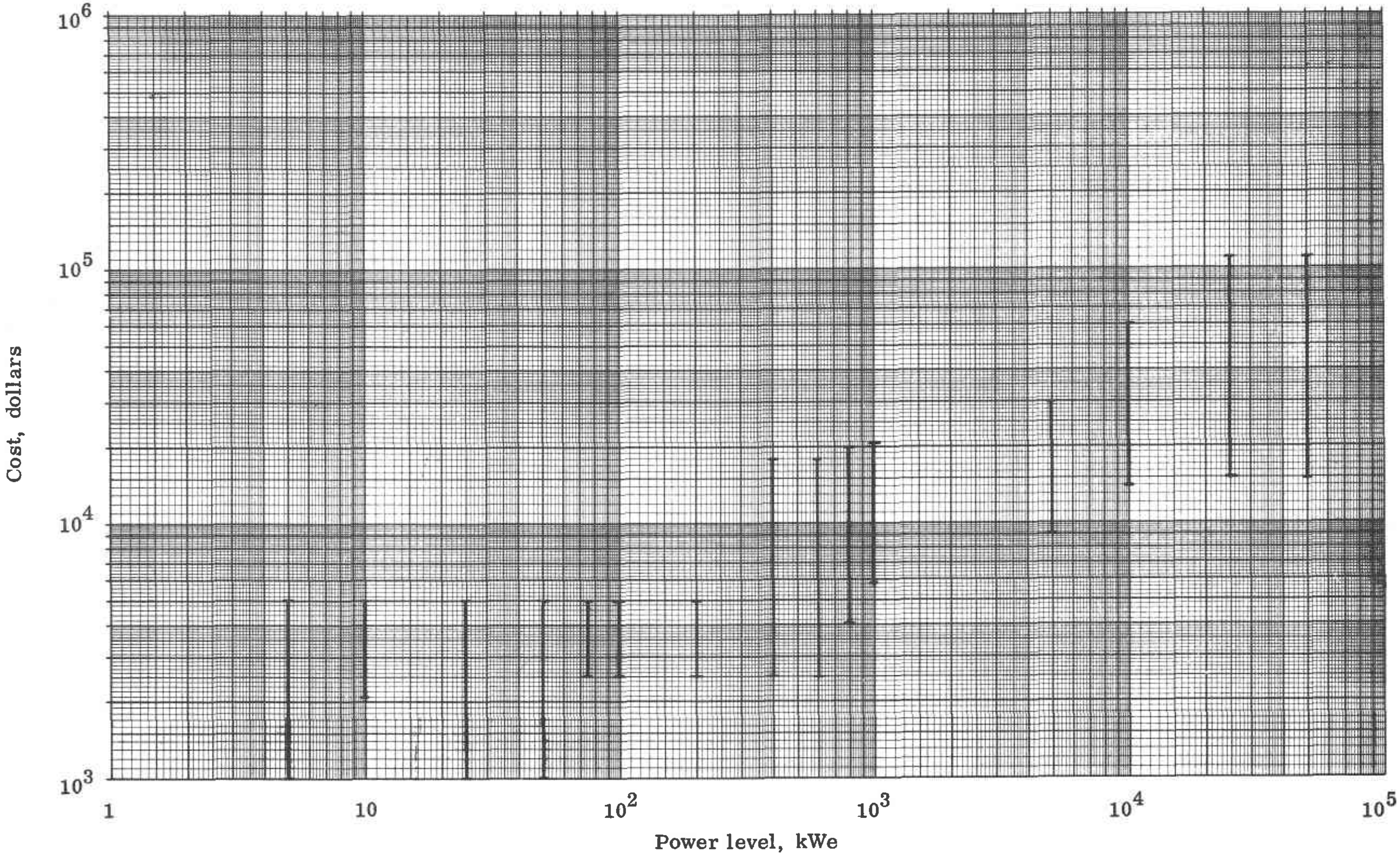


Figure 3-46. - Pump cost as function of power level.

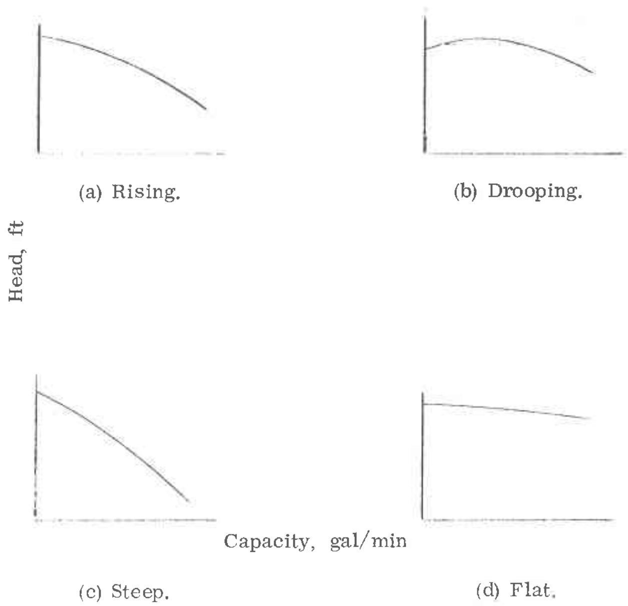


Figure 3-47. - Head-capacity curves for centrifugal condensate and boiler feed pumps.

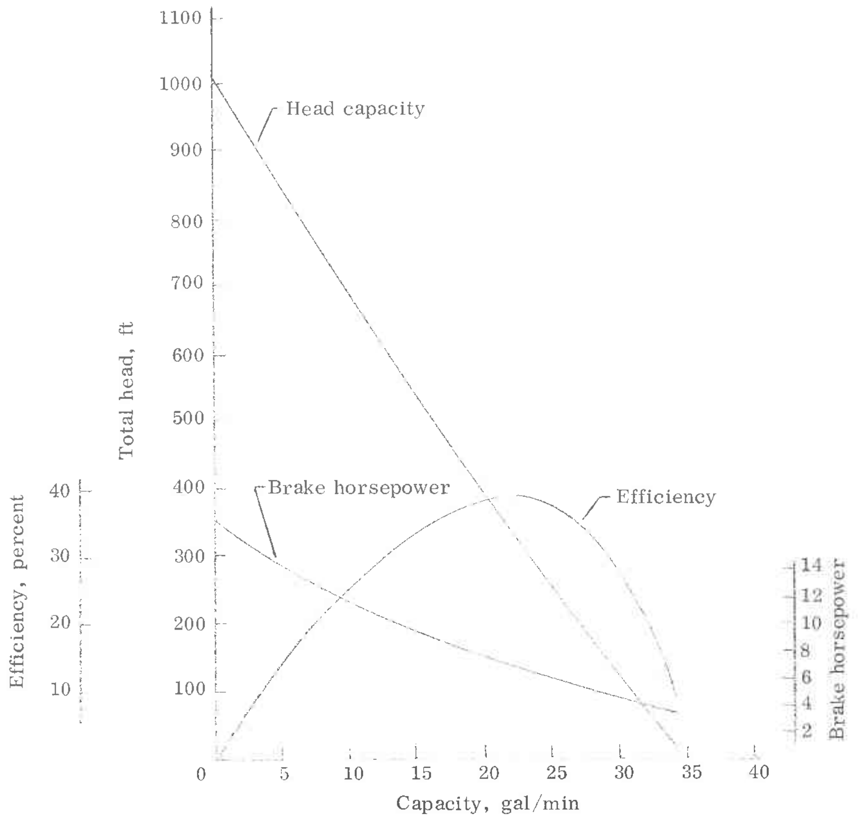


Figure 3-48. - Typical head-capacity curve for regenerative turbine pump.

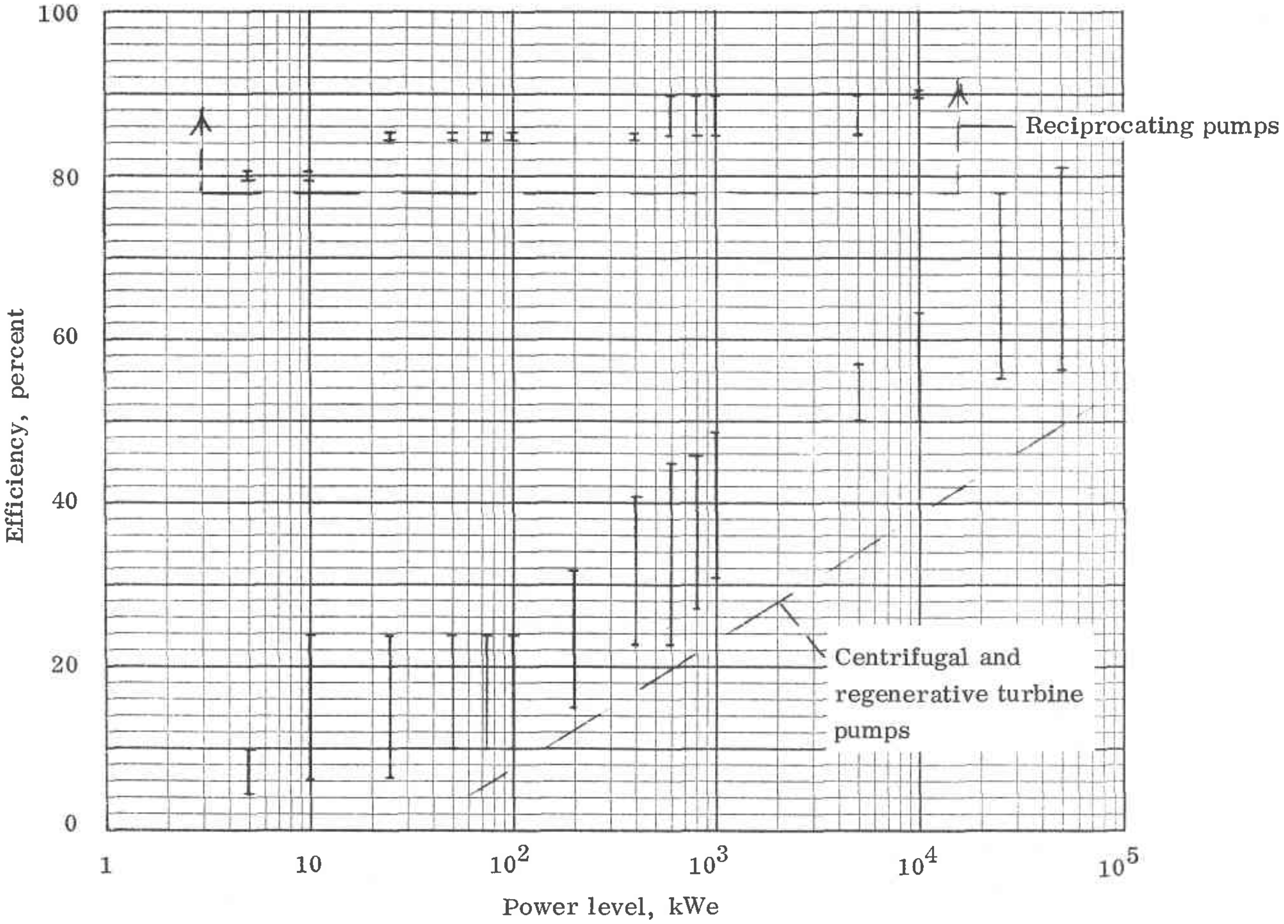


Figure 3-49. - Pump efficiency as function of power level.

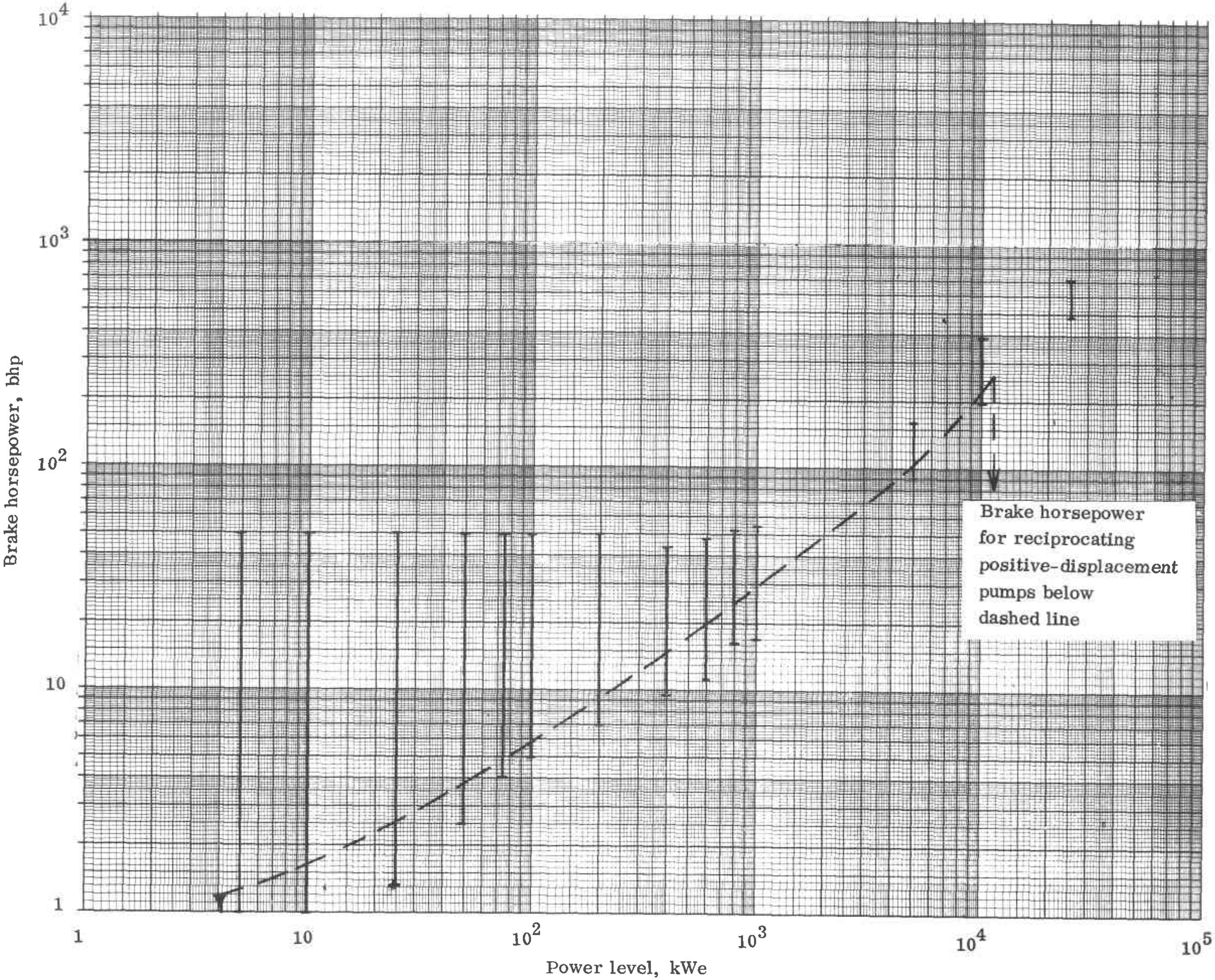


Figure 3-50. - Pump brake horsepower as function of power level.

4.0 BRAYTON-CYCLE COMPONENT CHARACTERISTICS

by Thaddeus S. Mroz, Jack A. Heller, and Harvey S. Bloomfield

The gas-turbine-engine power-generating system is defined as the overall assembly comprising the gas turbine engine (prime mover), the speed-reduction unit (if not integral with the prime mover), and the alternating-current (ac) electric generator. This section discusses only the gas turbine engine. The speed-reduction unit and the ac generator are discussed separately in sections 6.0 and 7.0.

The gas turbine engine operates on the Brayton-cycle principle. It is a dynamic energy-conversion device that converts chemical energy in the fuel or externally supplied heat energy to internal energy and finally to mechanical shaft energy. Depending on the engine configuration, the heat input to the engine can originate from combustion of coal or petroleum-derivative fuels, from a nuclear reactor, or from solar energy.

Unlike Otto-cycle engines and like Rankine- and Stirling-cycle engines, the Brayton engine is a continuous-combustion engine. It operates at high temperatures (1300° to 2300° F) and high speeds (3000 to 95 000 rpm) with a single-phase, gaseous working fluid. (The Rankine cycle uses a two-phase fluid.)

Two basic types of Brayton gas turbine engine have been developed and are presently in use: the closed-cycle engine and open-cycle engine. The closed-cycle Brayton engine is the natural cycle for the gas turbine since it can be used with any single-phase, gaseous working fluid (such as air or rare gases) and with any fuel. In this engine configuration, the working fluid operates in a closed-loop system (like the Rankine and Stirling cycles) and is not discharged to the atmosphere. This engine configuration has had limited development and application in this country. However, industrial units have been developed and are operational in Europe and Japan (ref. 9).

In the open-cycle Brayton gas turbine engine the working fluid (air) is exhausted to the atmosphere. Two configurations of this engine have been developed and are presently in use. In one configuration, called the simple open-cycle engine, waste heat is not recovered. In the other configuration, called the regenerated open-cycle engine, a portion of the exhaust waste heat is recovered to improve thermal efficiency.

As used in this handbook, a regenerated engine is one that uses a stationary heat exchanger or a rotating metallic or ceramic heat exchanger for waste-heat recovery. The stationary heat exchanger is sometimes referred to as a recuperator. The open-cycle engine is widely used in aircraft propulsion (simple open cycle) and in mechanical and electric power generation (simple and recuperated open cycles).

The gas turbine can be described as an assembly consisting of a rotating element contained in an engine housing that supports the rotating element and also serves as a pressure shell. Principal elements of this engine are the compressor, the turbine, and the combustor or the heat-source heat exchanger. The compressor and turbine are mounted on a shaft. In the open-cycle engine, the combustor is installed inside the engine housing. In some foreign designs, the combustor is installed outside the engine housing. In the closed-cycle engine, the heat-source heat exchanger is installed in the closed loop between the compressor and the turbine.

Engine operation involves four basic steps:

(1) The working fluid enters the compressor and is compressed to a design pressure for the operational pressure ratio.

(2) The compressed working fluid is heated to a design turbine-inlet temperature in the combustor or the heat-source heat exchanger.

(3) The working fluid is expanded in the turbine to provide mechanical energy.

(4) The expanded gases are exhausted to the atmosphere or to a heat exchanger (regenerator) for recovery or removal of waste heat.

Between 50 and 80 percent of the work developed by the turbine engine is used for compression. The actual value depends on the cycle and the system design. The remaining energy is available for shaft work (mechanical or electric power generation).

Single- and two-shaft open-cycle engines are manufactured. In the single-shaft engine, the compressor and turbine are mounted on a common shaft, which is supported on bearings. In this configuration, the compressor and turbine are mechanically connected. In the two-shaft engine, the compressor and the turbine that supplies the compressor work are mounted on a single shaft. This assembly is called the gas-generator assembly. The working fluid is only partially expanded in the gas-generator turbine. A second turbine is mounted on a second shaft to complete the expansion of the gases and to generate the power for shaft work. This turbine is called the low-pressure turbine, the power turbine, or the free turbine. Both assemblies are mounted along the horizontal engine axis and are connected by a gas coupling. Each shaft is supported by separate bearings. In the two-shaft configuration, the power turbine operates at a lower speed than the gas-generator turbine.

Heat addition to the gas turbine engine is provided by a combustor or a heat-source heat exchanger. Compressor discharge flow is ducted to the combustor or heat-source heat exchanger for heat addition and is then ducted through transition sections to the turbine.

The simple open-cycle gas turbine engine is illustrated schematically in figure 4-1. The regenerated open-cycle engine is illustrated schematically in figure 4-2. It incorporates the same features as the simple open-cycle engine except that a regenerator is used to tranfer waste heat from the exhaust to the gas flow between the compressor and the combustor or heat-source heat exchanger. The closed-cycle gas turbine engine is illustrated schematically in figure 4-3. It incorporates a regenerator for waste-heat recovery and a heat-rejection heat exchanger for removal of waste heat from the system.

Generally, the attributes of the gas turbine engine are

(1) Minimum number of moving parts

(2) Minimum friction

(3) Quick-starting characteristics even at very low ambient temperatures

(4) Rapid load-response characteristics

(5) Minimum vibration

(6) Low weight per horsepower

(7) Low installation costs

(8) Compactness

Unlike the Rankine-cycle power system, which is composed of separate and different components, the open-cycle gas turbine engine is a modular system. The characteristics of gas turbine engines are discussed in the following subsections.

4.1 SIMPLE OPEN-CYCLE GAS TURBINE ENGINE

4.1.1 Description

The simple open-cycle gas turbine engine is the most prevalent and developed gas turbine engine in this country. Depending on the application, turbine exhaust is discharged directly into the atmosphere or into a Rankine bottoming cycle for waste-heat recovery. Several generations of simple open-cycle gas turbine engines have been developed and are commercially available: the heavy-duty stationary engine (3700 to 95 000 kW), the lighter weight industrial engine (10 to 35 000 kW).

The heavy-duty stationary engine was first developed in the early 1900's and incorporated powerplant design and construction practices (ref. 6). More recent heavy-duty-engine designs incorporate both heavy-duty steam turbine and aircraft engine design and construction practices. This engine is used in utility, industrial, and marine applications. Figure 4-4 illustrates a typical design.

The lighter weight industrial engines are more compact and evolved from both aircraft-engine technology and design requirements for industrial, utility, and ground and marine transportation applications. The lighter weight industrial engine class includes aircraft engines modified for these applications as well as new engines designed in small, intermediate, and large sizes. Figure 4-5 illustrates typical industrial gas turbine engines.

Heavy-duty engines operate at low shaft speeds (3000 to 10 000 rpm) and low turbine-inlet temperatures (1400° to 2000° F). Lighter weight industrial engines operate at higher shaft speeds (6000 to 95 000 rpm) and higher turbine-inlet temperatures (1700° to 2300° F). Long life, high reliability, and high efficiency are common objectives for both engine types.

As noted previously, the basic engine consists of the rotating element and the engine housing assembly. The rotating element (rotor) is an assembly of components including the compressor, the turbine, and the shaft. This assembly is supported by radial and thrust bearings. The engine housing assembly incorporates the pressure-shell enclosure, the compressor stator vanes, the stationary turbine vanes, the combustor assembly, the inlet and outlet ducts, the diffusers, the transition sections for ducting airflow from the compressor to the combustor and from the combustor to the turbine section, the bearings, the fuel supply equipment, and the ancillary equipment. The engine designs vary depending on the application requirements, the operational mode (cyclic or steady state), and the manufacturer's design techniques. Typical design variations include

(1) Single- or two-shaft engine
(2) Single- or multistage compressor
(3) Centrifugal- or axial-flow compressor or combinations of a centrifugal stage and axial stages
(4) Single- or multistage turbine
(5) Radial- or axial-flow turbine
(6) Single-can, can-annular, or annular combustor
(7) Pressure ratios of 2.2 to 18
(8) Turbine-inlet temperatures of 1400° to 2300° F

Engine design and construction require close dimensional control to avoid local flow separations and stall and to reduce friction losses through the blading and duct passages. Accurate concentricity and balancing of the rotor are required to avoid damaging vibrations at high rotational speeds. Fabrication methods vary with the design, the type of engine, and the manufacturer's techniques. This is particularly true of the housing and the pressure containment. Materials used in these engines include ferrous and nonferrous metals or alloys, ceramics, cermets, superalloys, and refractory materials. Material selection is influenced by the engine's operational temperatures, pressure ratio, size, and construction and the manufacturer's design practices.

As a result of the trend to higher cycle-pressure ratios, higher turbine-inlet temperatures, longer service levels, and longer intervals between engine overhaul, it is necessary to cool the hot zones to maintain metal temperatures within allowable design limits. Cooling is therefore provided at the combustor walls (liner), in the transition section between the combustor and turbine, and in the turbine blades and nozzles. Cooling air originates from compressor discharge flow.

Operating temperatures will range from 3400° to 4000° F in the combustor and from 1400° to 2300° F at the turbine inlet. The degree of cooling and the selection of cooling techniques depend on the materials of construction, the impurities in the fuel, and the peak temperatures. Available cooling techniques include convection cooling, film cooling, impingement cooling, and transpiration cooling. The cooling requirements for high-temperature turbines (e.g., 2300° F turbine-inlet temperature) may significantly influence compressor size because of the quantity of compressor bleed flow required for cooling.

Since the output shaft speed of this engine is normally higher than the speed of the driven equipment, a speed-reduction unit is normally required for mechanical and electric generator drive applications. Integral speed-reduction gearing is incorporated in some engine designs. Generally, separately mounted speed-reduction units are used. Figures 4-6 and 4-7 illustrate a gas turbine engine with integral speed-reduction gearing and with a separately mounted speed-reduction unit, respectively.

4.1.2 Principal Components

4.1.2.1 Turbine. - The turbine converts the internal energy of the working fluid through a controlled expansion of the gas from the inlet pressure and temperature (combustor discharge flow) to lower pressure and temperature (turbine exhaust). The turbine design influences the design of both the compressor and the combustor.

Turbines are normally classified as either radial or axial. The radial-flow turbine (fig. 4-8) (ref. 10) is usually used in single-stage applications because of the complexity of the ducting required between the turbine stages. Typical applications are automotive gas engines; small auxiliary power units for aircraft; small and intermediate industrial engines; and advanced, small space-power systems such as the Brayton-cycle power unit. During operation, flow enters the turbine radially and leaves axially. Figure 4-9 shows the stator and rotor assembly of a typical radial-flow turbine (ref. 10). Radial-flow turbines can handle a higher pressure ratio per stage than the axial-flow turbine and are normally used where the overall turbine pressure ratio can be accommodated by one stage.

The axial-flow turbine (fig. 4-10) is used where higher pressure ratios are required. Axial-flow turbines can have one stage to 15 or 20 stages depending on the required overall pressure ratio. Typical applications are large ground-power generation systems, aircraft propulsion engines, and proposed space propulsion systems and industrial engines. During operation, flow enters and leaves the turbine axially. Figure 4-11 shows a four-stage turbine rotor used to drive the fan of an aircraft turbofan engine. Because the axial-flow turbine can have so many stages, it is usually found in systems that require very large output powers.

The radial-flow turbine is considered to be more rugged and may be cheaper to manufacture. The axial-flow turbine is smaller in diameter, requires simpler ducting, and has a lower inertia. Typical losses considered in turbine design include profile losses, tip clearance losses, disk friction losses, partial admission losses, and incidence losses (ref. 10).

4.1.2.2 Combustor. - Although combustion is not required in an all-solar energy-conversion system, a discussion of the combustor is presented here for reference with a hybrid (solar - fossil fuel) system.

In the open-cycle gas turbine engine, fuel is burned continuously in the combustor. During operation the temperature of the working fluid is raised by compression from ambient to a specific level depending on the engine pressure ratio. The temperature of the working fluid (compressor discharge flow) is then increased in the combustor to the required operating temperature at the turbine inlet. The combustor is the highest temperature zone in the engine.

Two major types of combustor are currently in use: the can, and the annular. Both have three operating zones:

(1) The primary zone - where the fuel-air ratio is highest (can be stoichiometric) and the temperature may be as high as 4000° F

(2) The secondary zone - where the reaction is completed

(3) The dilution zone - where relatively large volumes of air are introduced to reduce the average temperature of the combustion gases to the design turbine-inlet temperature

During operation of an open-cycle engine, a portion of the compressor discharge flow is ducted to the primary zone for combustion of the fuel. The remaining flow enters the combustor through openings in the combustor walls in the secondary and dilution zones. Since combustion temperatures are very high, the combustor walls are maintained within the allowable material temperature limit by film or convection cooling with dilution air.

The combustor, which is located between the compressor and turbine, can be the straight throughflow type or the reverse-flow type. A large number of combustor design configurations exist. Depending on engine design and size, the combustor assembly may consist of a single-can configuration (fig. 4-12); an annular configuration (fig. 4-13); or a can-annular configuration (fig. 4-14), where a number of combustor cans are installed radially around the compressor.

Combustor design is concerned with combustion efficiency, durability, pressure drop, outlet temperature profiles, combustion stability limits, fuel flexibility, and emissions. Combustion efficiency influences both specific fuel consumption and emissions. Pressure drop affects specific fuel consumption, power output, and weight. Outlet temperature profiles affect turbine life and engine efficiency. Combustion stability limits define the operating limits of the engine and its application. Considerable effort is being directed in combustor development to comply with Federal emission regulations and, at the same time, achieve fuel flexibility, that is, to achieve the combustion of light distillate and heavy fuels in an environmentally acceptable manner.

4.1.2.3 Compressor. - The compressor's function is to increase the pressure of the working fluid before it is heated and before the energy is extracted in the expansion process. The compressor may have a number of compression stages, depending on overall pressure ratio. Each stage consists of a rotating row of blades (the rotor), which adds energy to the air, and a stationary row of blades (the stator) which converts kinetic energy (velocity head) to static pressure. General design requirements include

(1) High efficiency
(2) Low compressor weight per unit of flow or output power (low inertia for fast response)
(3) Wide range of stable flow operations
(4) Low cost
(5) Ruggedness
(6) Reliability
(7) Simplicity

Two basic types of compressor stage are used in gas turbine engines: the centrifugal (radial)-flow compressor, and the axial-flow compressor. The centrifugal-flow compressor develops a higher pressure ratio per stage (fewer stages) and has shorter overall length than the axial-flow compressor. This results in fewer stages to achieve the design pressure ratio. The direction of inlet flow is axial and the direction of discharge flow is radial. A stationary diffuser converts the velocity head to static pressure. The centrifugal-flow compressor is normally used as a single-stage unit. Its efficiency depends on many factors, such as size and flow rate. It is characterized by short length and large frontal area. In general, the centrifugal-flow compressor is considered mechanically more rugged and more reliable than an axial-flow compressor.

The axial-flow compressor is based on the concept that gas can be compressed more efficiently if the pressure is raised in a series of small increments along the axial direction. The axial-flow compressor has a number of stages and is used for high-flow applications. The direction of flow is axial at the inlet and at the outlet. The axial-flow compressor has a lower pressure ratio per stage and requires more stages to achieve the design pressure ratio - but has a higher flow capacity per frontal area - than the centrifugal-flow compressor. The axial-flow compressor achieves high efficiencies in the larger sizes and is the more prevalent design in high-flow applications (larger engine). The number of stages required for a given overall pressure ratio results in a longer compressor configuration than for the centrifugal type. Because of the number of relatively fragile blades, the axial-flow compressor is generally more expensive and more difficult to manufacture than the centrifugal type. In some cases, a compressor design will include a matched combination of an axial- and a centrifugal-flow compressor.

4.1.3 Development Status

A sound and developed technology base exists for the simple open-cycle gas turbine engine in the 10- to 50 000-kilowatt output power range and above. Design and fabrication techniques are supported by substantial experience and operating history for heavy-duty stationary engines in the 3700- to 50 000-kilowatt range (and above) and for lighter weight industrial engines in the 10- to 35 000-kilowatt range. Gas-turbine-engine development is highly competitive and is oriented toward current and future technical requirements, market demands, and new applications in the industrial, utility, and marine transportation sectors. Current development is directed toward upgrading the performance and range of existing engine designs and toward generating new designs.

Development of simple open-cycle gas turbine engines in this country has been primarily in the output power range of 500 to 50 000 kilowatts because of market demands. More recent demands have resulted in development of engines in the 10- to 500-kilowatt output range and of larger engines to 75 megawatts. Although small engines are available, comprehensive product lines matching those for the steam turbine have not been developed. Existing designs use energy derived from the combustion of fossil fuel. Using a simple open-cycle gas turbine engine in a hybrid (solar - fossil fuel) system will require specific engine modifications to accommodate a solar-heat-source heat exchanger. The extent of the modifications will depend on the particular engine design, size, and operating requirements.

Development status is illustrated in figures 4-15 and 4-16, which show the design output power level of production, prototype, and developmental units as a function of turbine-inlet temperature and pressure ratio through the output power range of 10 to 50 000 kilowatts.

4.1.4 Production Status and Availability

Production and preproduction engines are available in the output power range of 8 to 95 000 kilowatts. The engines have gone through the development and prototype stages and are commercially available. Hard tooling and manufacturing techniques have been established, and the designs have been optimized for economical production. Production quantities depend on the engine size, the size and loading of production facilities, and the available tooling and manpower.

Current engine delivery schedules estimated by the manufacturers are shown in figure 4-17.

4.1.5 Operating Characteristics

The gas turbine engine is a continuous-combustion, high-temperature, high-speed engine. It is considered to be quiet, has a minimum of moving parts, and is practically free of vibration.

Performance of the gas turbine is measured in terms of specific fuel consumption (sfc) expressed in pounds per horsepower per hour or in terms of heat rate expressed in Btu per horsepower per hour. Both specific fuel consumption and heat rate are determined by the efficiency of the individual components (compressor, turbine, and combustor), the maximum turbine-inlet temperature, and the pressure ratio. The higher the turbine-inlet temperature, the higher will be the pressure ratio - which results in optimum engine efficiency. Engine performance is also affected by air temperature and air pressure at the compressor inlet. Because air density decreases with increasing ambient air temperature, a correction in engine mechanical speed is required. For this reason the engine designs are rated for a specific inlet temperature, and performance is mapped for a range of inlet temperature.

The overall efficiency of existing simple open-cycle gas turbine engines as a function of power level is shown in figure 4-18. The plot is based on information furnished by the manufacturers and includes stationary and industrial gas turbine engines.

4.1.6 Cost and Weight

Because insufficient cost data were provided by the manufacturers, unit cost could not be plotted as a function of power level. Engine weight as a function of power level is shown in figure 4-19.

4.1.7 Operating Constraints

The simple open-cycle gas turbines presently in use are limited in performance by fuel type, fuel quality, and the operating turbine-inlet temperature. Corrosion effects due to fuel impurities, allowable working stress levels, and high temperatures restrict the number of materials that can be used. In some cases, allowable temperatures are limited by the selection of a material because of its low initial cost.

4.1.8 Experience Factors

Simple open-cycle, heavy-duty gas turbine engines have been used for mechanical and electric power generation for over 50 years. Industrial engines evolved from aircraft technology have been commercially available for 10 years and have been used in the industrial and utility sectors. In the past few years these engines have also found application in marine and ground propulsion systems. Design life, reliability, maintenance schedules, and overhaul schedules have been established for the different engine designs on the basis of testing and field experience. Heavy-duty engines are commercially offered with a 10-year design life and 5 years between heavy maintenance or major overhaul. The designs incorporate such features as modular assembly and parts accessibility for ease of maintenance and to minimize downtime. The simple open-cycle gas turbine engine has established a record of very high reliability.

4.2 REGENERATED OPEN-CYCLE GAS TURBINE ENGINES

4.2.1 Description

The regenerated open-cycle gas turbine engine incorporates a heat exchanger to recover waste heat from the turbine exhaust for preheating of the compressor discharge flow. Adding regeneration or recuperation to the simple open-cycle engine is one way to improve engine efficiency. The ideal would be to preheat the compressor discharge flow to the turbine exhaust temperature. In practice, this is not possible because of heat-exchanger inefficiency. Experience with heavy-duty industrial machines has shown that fuel consumption can be reduced 20 to 25 percent by regeneration or recuperation.

The gas turbine engines developed for ground transportation (automobiles, trucks, buses, etc.) are compact, regenerated engines. They are one- and two-shaft engines designed with a ceramic or metallic rotating heat exchanger called a regenerator. Regenerated engines developed to date with rotating heat exchangers are in the 60- to 475-kilowatt range. This regenerator assembly includes high-temperature metallic seals with proprietary coatings, the ceramic or metallic heat-exchanger core, and the drive assembly (ref. 11). The engine housing, a casting designed with internal ducting for the regenerator assembly, incorporates two separate flow paths perpendicular to the regenerator core. Hot turbine exhaust flowing through one section of the regenerator core heats the core matrix. The cooler compressor discharge air flows in the opposite direction through the other section of the heated rotating core. Preheated compressor discharge air is ducted to the combustor inlet. One or two regenerators are used depending on the size of the engine. The regenerator assembly, which is driven off the power shaft, is installed within the engine housing and is an integral part of the engine. Figure 4-20 is a flow schematic of a regenerated engine.

A stationary heat exchanger called a regenerator is used with stationary heavy-duty gas turbine engines and lighter weight industrial engines. This heat exchanger is mounted outside the engine housing at the turbine exhaust duct. In this arrangement, compressor discharge flow is circulated through the high-pressure side of the heat exchanger, and turbine exhaust flow is circulated through the low-pressure side. Heat transfer is from the low-pressure side to the high-pressure side. The high-pressure side is manifolded across the compressor discharge and the combustor inlet. The low-pressure side receives flow from the turbine discharge duct. The cooled exhaust is then discharged to the atmosphere or to a heat-recovery boiler.

The degree of regeneration for an application depends on a trade-off between the cost of the incremental heat exchanger and fuel savings. The potential benefits must be evaluated for the specific application and fuel used. Figure 4-21 shows a regenerated engine arrangement and flow path, and figure 4-22 shows a typical regenerator.

4.2.2 Development Status

A well-developed technology base exists for regenerated open-cycle engines in the 10- to 50 000-kilowatt output power range, including heavy-duty stationary engines in the 3700- to 50 000-kilowatt range and industrial engines in the 10- to 7500-kilowatt range. This technology base includes design and fabrication techniques based on actual development, realistic experience, and operating history.

Development of regenerated engines has been directed to the 1120- to 50 000-kilowatt output power range and above and has focused on mechanical drive and electric power generation. Typical applications are in the industrial, utility, and marine transporation sectors.

Although a limited number of small regenerated engines have been developed in the low-power range (10 to 475 kW), a comprehensive, highly efficient line of engines is not presently available. However, technology is available for more efficient engines and development could provide the engines needed for such applications as the dish Brayton systems being considered.

The development status of regenerated engines is illustrated in figures 4-23 and 4-24, which are plots of actual engine design power level as a function of turbine-inlet temperature and pressure ratio.

4.2.3 Availability

A regenerated industrial engine rated for 310 to 475 kilowatts is classified as a production engine and is available in limited quantities. Regenerated open-cycle automotive gas turbine engines are not production items and are classified as developmental engines.

Regenerated heavy-duty stationary engines are commercially available in the 3700- to 50 000-kilowatt range and above. Regenerated lighter weight industrial engines are in production. These engines have gone through the development and prototype stages and are commercially available. Tooling and manufacturing techniques have been established, and the design has been optimized for fast and economical production. Production quantities of regenerated gas turbine engines depend on engine size, size of production facilities, available tooling and manpower, and production schedules. Engine delivery schedules estimated by the manufacturers are shown in figure 4-25.

4.2.4 Operating Characteristics

Incorporating an external or internal regenerator in a single- or two-shaft engine will increase both the pressure drop between the compressor discharge and the combustor inlet and the engine thermal efficiency. The net effect on engine operating characteristics will be a function of the specific engine design configuration, the engine output power, the pressure ratio, the turbine-inlet temperature, and the temperature of the compressor and turbine exhaust.

Using a regenerated engine requires an evaluation of the system performance and the operating requirements. Engine design and heat-exchanger performance must be matched for a particular operating cycle. As previously noted, pressure ratio and turbine exhaust temperature greatly influence the effectiveness of regeneration. The increased pressure drop and the net output power, pressure ratio, and effectiveness of the heat exchanger must be considered in this evaluation.

The operating characteristics of the simple open-cycle engine are applicable to the regenerated open-cycle engine.

The mechanical efficiency of existing regenerated gas turbine engines as a function of power level is shown in figure 4-26. The plot is based on information furnished by the manufacturers for stationary and lighter weight industrial gas turbine engines.

4.2.5 Cost and Weight

Because insufficient cost data were received from the manufacturers, unit cost could not be plotted as a function of power level. Engine weight as a function of power level is shown in figure 4-27.

4.2.6 Operating Constraints

The regenerated open-cycle gas turbine engines presently in use are limited in performance by fuel type, fuel quality, and operating turbine-inlet temperature. Corrosion effects due to fuel impurities, allowable working stress levels, and high temperatures restrict the number of materials that can be used. In some cases, temperatures are limited by the selection of a material because of its low initial cost.

4.2.7 Experience Factors

Regenerated open-cycle gas-turbine-engine development programs for the ground transportation sector resulted in a production industrial engine with an output power range of 310 to 475 kilowatts. Other engines are still in the development stage. Industrial regenerated engines have not accumulated an extensive history of use in industrial applications. These engines have been extensively field tested and operated in the development and prototype stages. Projected engine reliability and design life were established from test performance before the engine was put into commercial production. Routine maintenance schedules and time between major maintenance were based on operating history.

Regenerated, heavy-duty stationary open-cycle engines (ref. 12) have been in service for at least 10 years, enough to establish the reliability factor, the design life, and the required maintenance schedules. Today, regenerated engines are offered commercially with a design life of 10 years and above and a period between heavy maintenance of 5 years.

Designs include modular assembly and parts accessibility for ease of maintenance and to minimize downtime. Applications include power generation in such areas as the Middle East.

4.3 CLOSED-CYCLE GAS TURBINE ENGINE

4.3.1 Description

The closed-cycle gas turbine engine is a closed-loop system that operates with a single-phase working fluid. This engine comprises a turbine-compressor rotating element and housing, a regenerator, a waste-heat heat exchanger, ducting, and a heat-source heat exchanger that provides external heat input. Figure 4-3 schematically illustrates a basic closed-cycle gas turbine engine. Engines of this type have been developed in the output power range of 0.5 to 17 500 kilowatts.

The turbocompressor machinery used in the closed-cycle engine is the same type used in the open-cycle engine. The same component technology prevails. The compressor can be a centrifugal- or axial-flow type or a combination of both designs. The turbine can be a radial- or axial-flow type, as described for the open-cycle engine. The type and size of rotating machinery are determined by the output power range of the engine. Gas- or oil-lubricated bearings are used. Unlike the open-cycle engine, which uses an integral heat combustor for external heat input to the system, the closed-cycle engine employs a heat-source heat exchanger. The working fluid flows through one side of this heat exchanger and is heated by an external heat-source heat exchanger operated, for example, in a solar thermal power system. The solar heat is added on the same side of the exchanger. Heat transfer to the working fluid occurs through a metallic wall.

In the closed-loop system, the working fluid is clean. Thus the turbine is not subject to corrosion or scale formation, and the compressor is not subject to erosion from airborne particulate matter. Selection of the turbine-inlet temperature is dictated primarily by the heat-source characteristics and the material temperature limitations. Waste heat from this closed-loop system is rejected to the atmosphere by means of a gas-liquid or gas-gas heat exchanger, depending on the application.

The working fluid, cooled by the waste-heat heat exchanger, is compressed by the compressor and flows to the regenerator to be preheated by recoverable heat from the turbine exhaust. Preheated gas then flows to the heat-source exchanger for external energy input and on to the turbine for expansion and energy conversion to shaft power. Hot turbine exhaust then flows to the regenerator for heat recovery and on to the waste-heat heat exchanger for rejection of unusable heat to the environment.

Characteristics of the closed-cycle engine are as follows:

(1) A variety of single-phase working fluids can be used, including monatomic, inert gases (argon, krypton, helium, xenon-helium) as well as air and nitrogen. The turbomachinery is designed for the properties of the selected working fluid.

(2) A variety of heat sources can be used, including solar energy, stored heat, petroleum-derivative fuels, coal, or a nuclear reactor.

(3) Cycle efficiency is independent of atmospheric pressure changes.

(4) Part-load operation is achieved by changing the pressure level (changing the mass flow rate). Part-load efficiency is nearly constant down to 20-percent load, while turbine-inlet temperature is maintained constant.

In designing the closed-cycle engine, selecting the recuperator effectiveness (ref. 9) is second in importance only to selecting the turbine-inlet temperature, which is selected on the basis of the pressure-temperature capability of the heat-source heat exchanger. The regenerator can be the largest single component in the system; therefor, designing this component is a trade-off between required efficiency, weight, volume, and cost. Current low-cost materials can withstand a 1500° F turbine-inlet temperature.

Selection of cycle temperature and pressure is constrained by the allowable stress levels at the high-temperature components. High loop pressure is beneficial to the cycle since it minimizes flow area at the turbomachinery, the ducts, and the heat exchangers and improves heat transfer. Although cycle efficiency may be improved with higher operating pressures and temperatures, the increased stress levels and associated cost can constrain the system design.

The closed-cycle gas turbine engine incorporates three heat exchangers (a regenerator, a waste-heat heat exchanger, and a heat-source heat exchanger). Direct-transfer heat exchangers are used in this type of system. The most common is the shell-and-tube design. Plate-fin heat exchangers of counterflow design are becoming more prevalent in compact configurations. Pumping power for low-density working fluids is frequently as much a controlling factor in the design as heat-transfer requirements. Therefore, heat-transfer surfaces that would meet both heat-transfer pumping-power requirements should be investigated. Both dimpled-tube and extended-surface (fin) designs have been investigated extensively to improve performance.

4.3.2 Development Status

The closed Brayton cycle has been developed in this country only for space power (0.5 to 10 kW) and an automotive application (112 kW). Numerous studies and design efforts have been conducted for other applications - including industrial power, marine propulsion systems, and use with nuclear reactors. This engine has not been developed and used in this country because of the abundance of clean fuels such as natural gas, which favor the open-cycle engine. In Europe, where clean fuels (light distillates and natural gas) are scarce and expensive, the closed-cycle gas turbine engine was developed to operational status in the mid-1950's with output powers from 2000 to 17 500 kilowatts, turbine-inlet temperatures from 1220° to 1320° F, output shaft speeds from 6500 to 13 000 rpm, and pressure ratios from 3.7 to 4.3. These engines were developed for use with such fuels as bituminous coal, brown coal, blast-furnace gas, mine gas, and oil and can provide cycle efficiencies to 32 percent.

A sound technology base exists in this country and in Europe for design and fabrication of closed-cycle gas turbine engines with operational turbine-inlet temperatures to 1500° F. This includes turbomachinery, heat exchangers, bearing shaft seals, and materials technology. Higher operational temperatures may require such high temperature materials as ceramics for the turbine and heat-source heat exchanger or combustor. The component technology developed for the regenerated open-cycle engine is generic to the closed-cycle engine.

4.3.3 Availability

The closed-cycle engine is still in the development stage in this country for industrial applications. However, it is being sold commercially for industrial and utility applications in Europe and Japan at output power levels from 2000 to 17 500 kilowatts.

4.3.4 Operational Characteristics

The closed-cycle gas turbine engine is quiet and vibration free and operates at a high efficiency over a wide range of output loads. Closed-cycle part-load operation is achieved by maintaining the turbine-inlet temperature at design point and changing the pressure level (varying the mass flow rate of the working fluid). Heat input to the engine can be varied to maintain the desired turbine-inlet temperature. The engine can be operated to give constant or variable shaft speed, depending on the application.

4.4.5 Cost

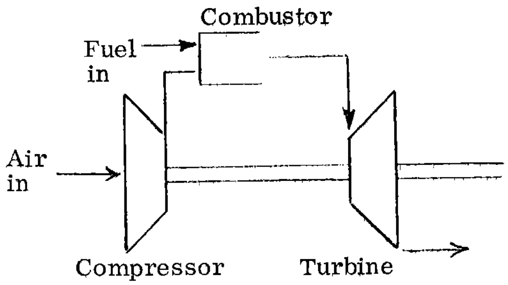
Cost estimates for a commercialized engine are not available.

4.4.6 Constraints

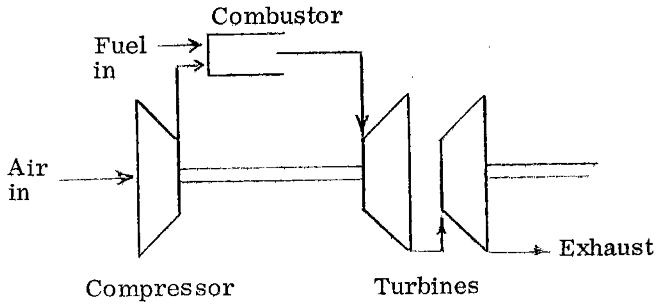
The operational temperature of the heat-source heat exchanger is considered the operational-design constraint on this system. It is imposed by the allowable operational stress levels of the heat-transfer materials.

4.4.7 Experience Factors

Experience factors - such as design life, reliability, and maintenance - have not been established for commercial closed-cycle engines in this country. However, closed-cycle engines in Europe have been operated for over 85 000 hours without major maintenance. Their reliability is considered very high, with low maintenance. A 10-kilowatt closed-cycle engine designed for space-power application demonstrated high reliability during a 38 000-hour unattended operational test conducted by the NASA Lewis Research Center.



(a) Single shaft.



(b) Two shaft.

Figure 4-1. - Schematic of simple open-cycle Brayton gas turbine engine.

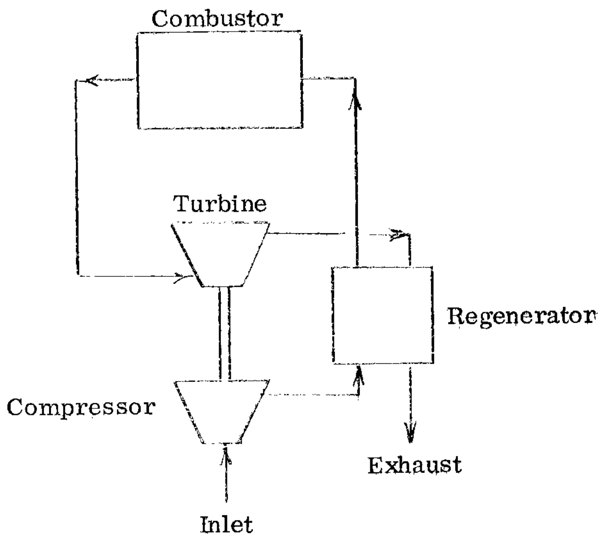


Figure 4-2. - Schematic of regenerated open-cycle Brayton gas turbine engine.

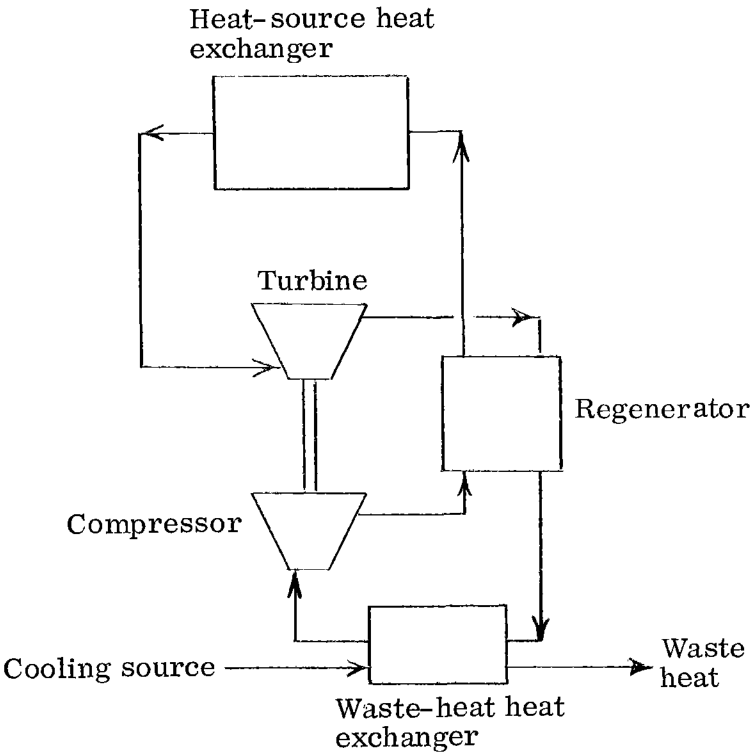


Figure 4-3. - Schematic of regenerated closed-cycle Brayton gas turbine engine.

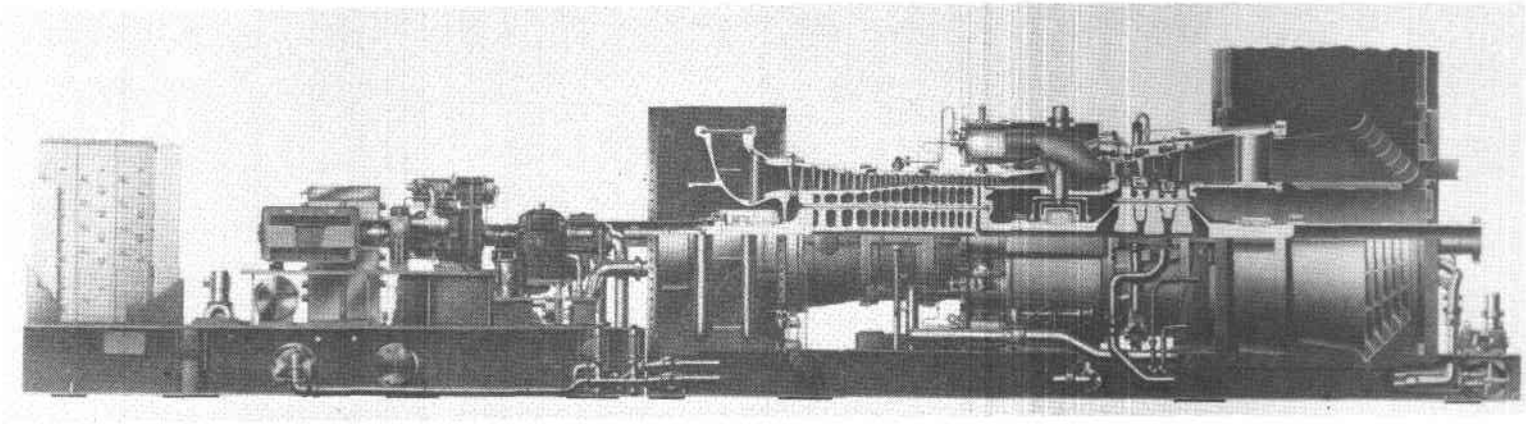
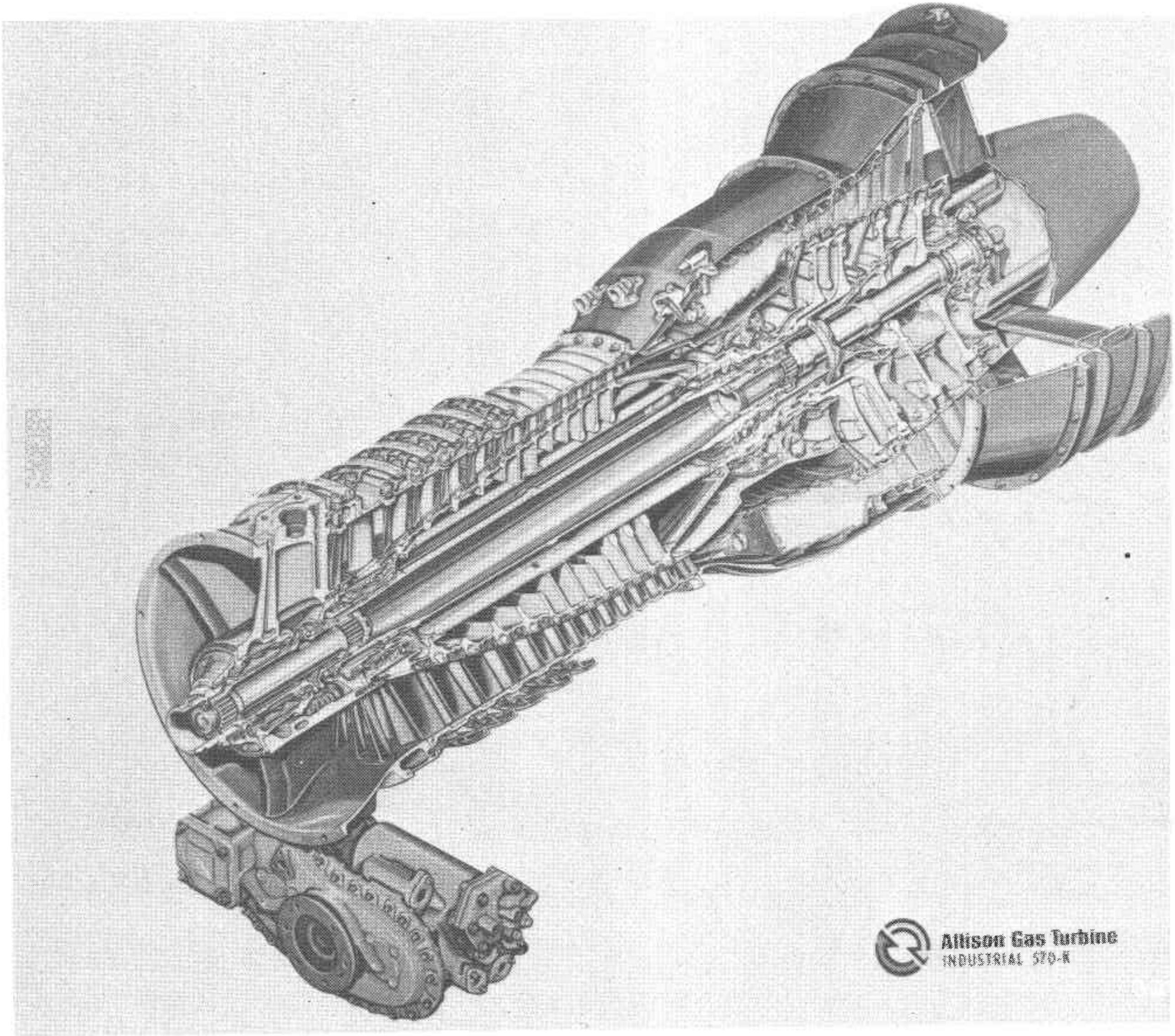
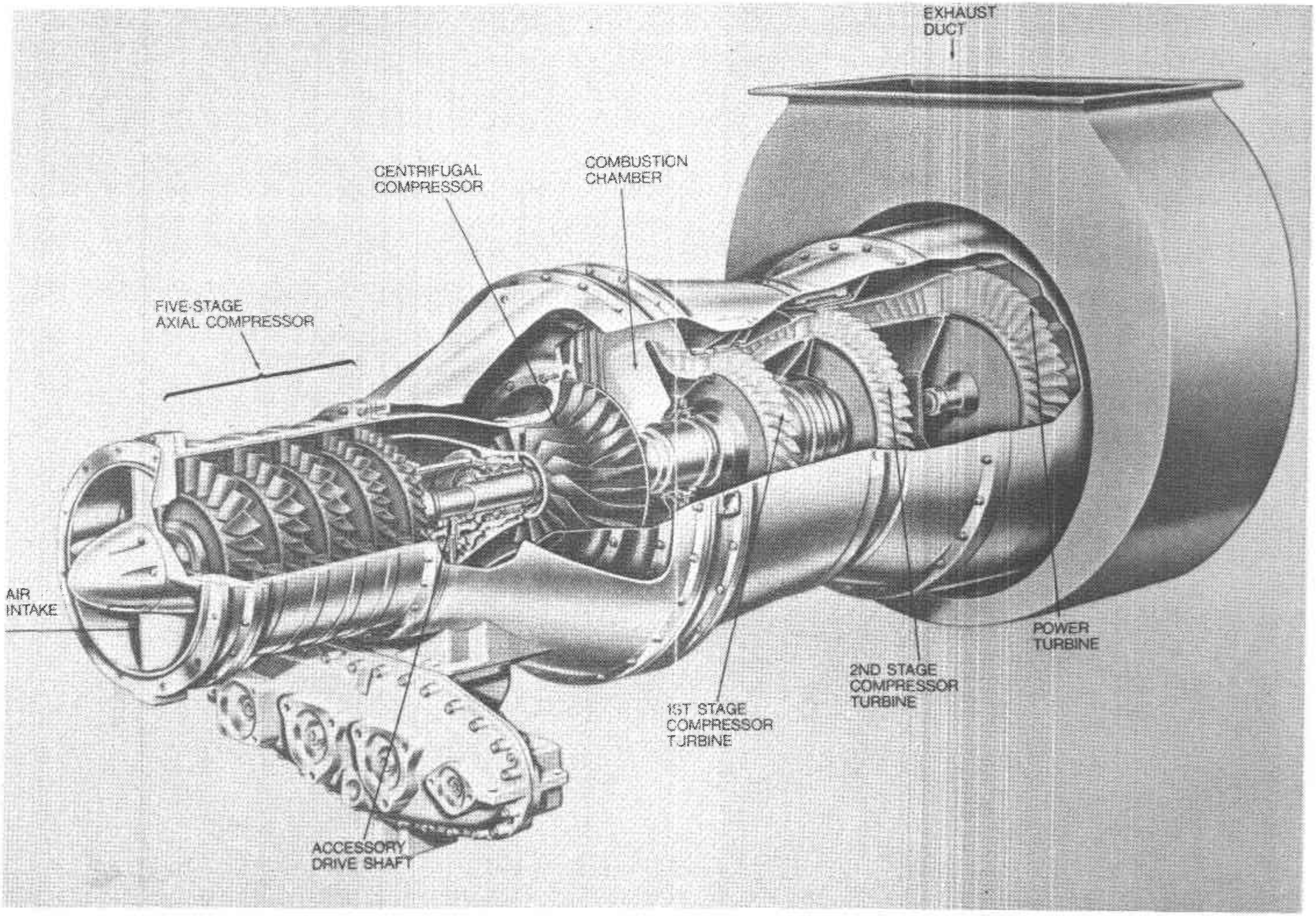


Figure 4-4. - Heavy-duty, industrial gas turbine engine. (Courtesy of General Electric Co.)



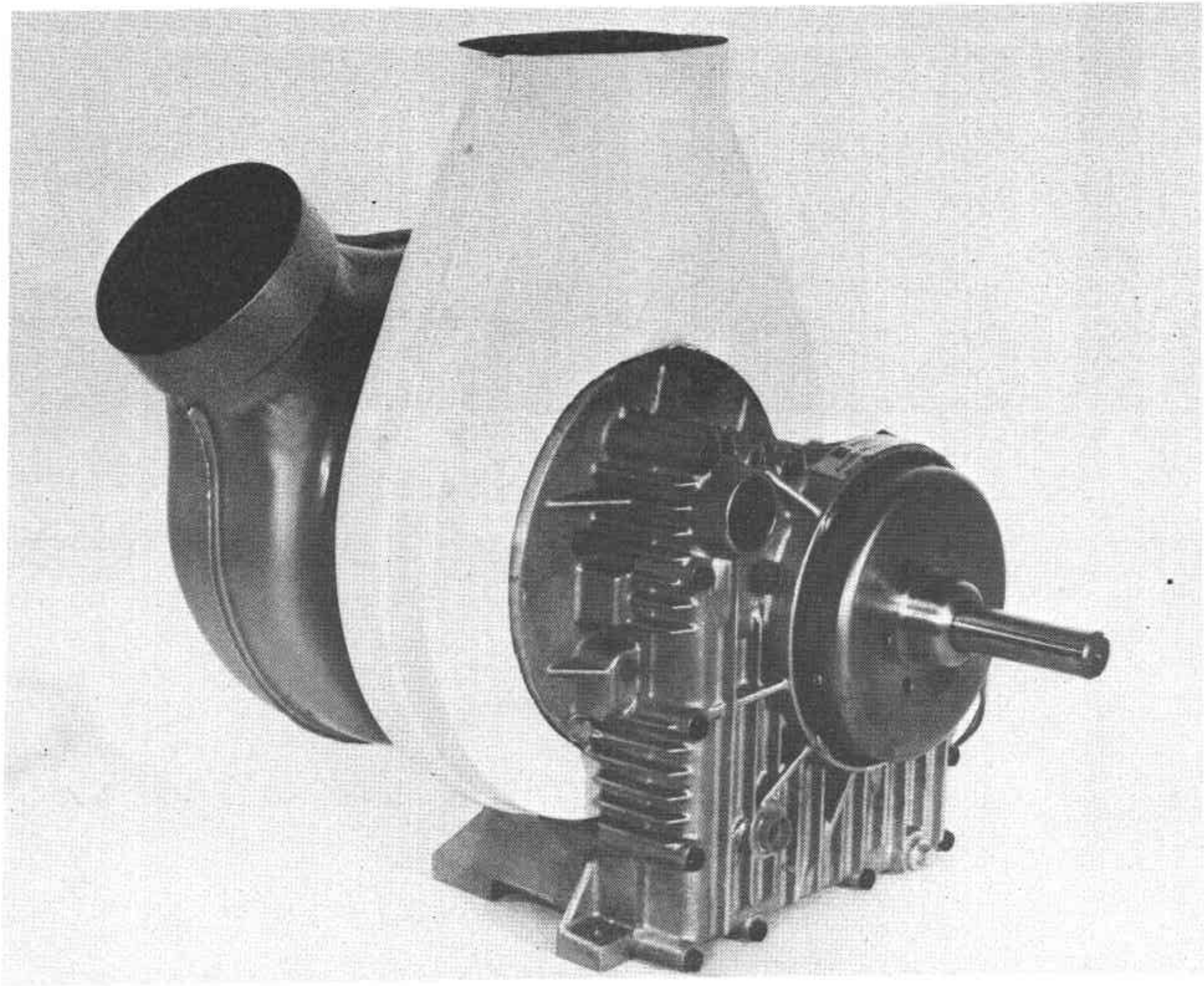
(a) Industrial gas turbine engine. (Courtesy of Detroit Diesel Allison, Division of General Motors Corp.)

Figure 4-5. - Typical industrial gas turbine engines.



(b) Simple open-cycle industrial gas turbine engine. (Courtesy of Williams Research Corp.)

Figure 4-5. - Continued.



(c) Simple open-cycle industrial gas turbine engine. (Courtesy of Williams Research Corp.)

Figure 4-5. - Concluded.

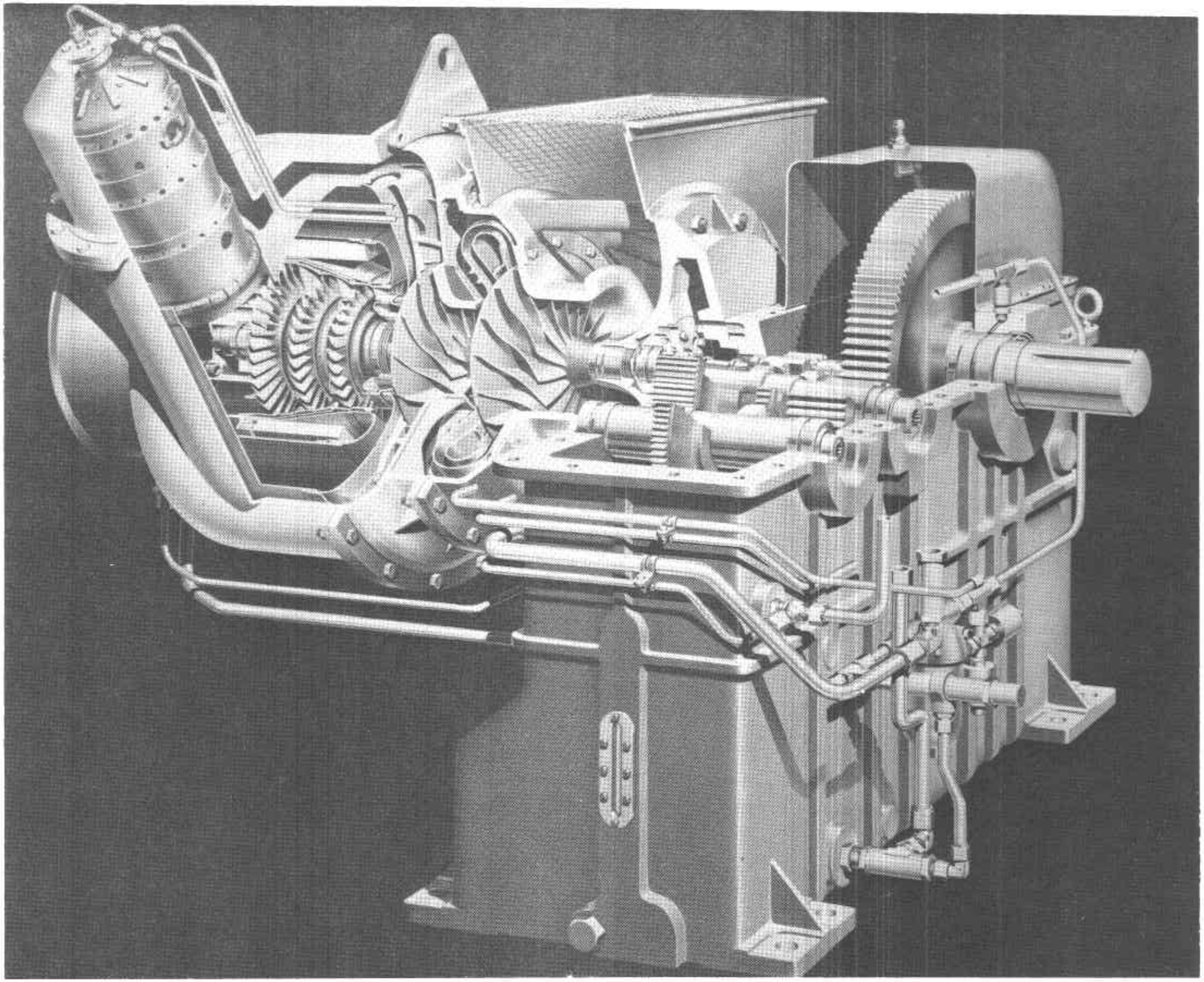


Figure 4-6. - Industrial gas turbine engine with integral speed reducer. (Courtesy of AiResearch Manufacturing Co. of America, Division of Garrett Corp.)

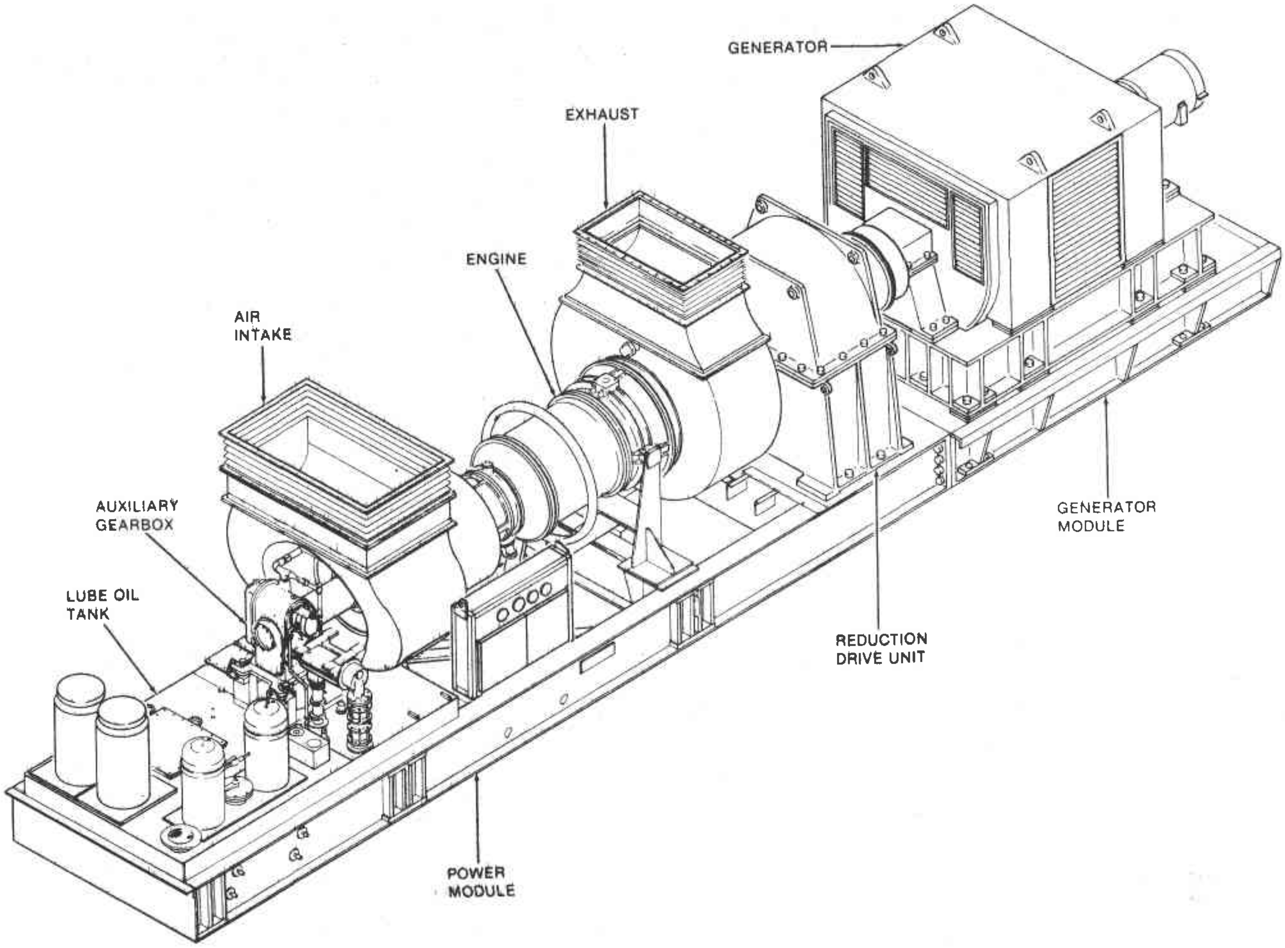


Figure 4-7. - Industrial gas turbine engine with separately mounted speed reducer. (Courtesy of Solar Turbine International.)

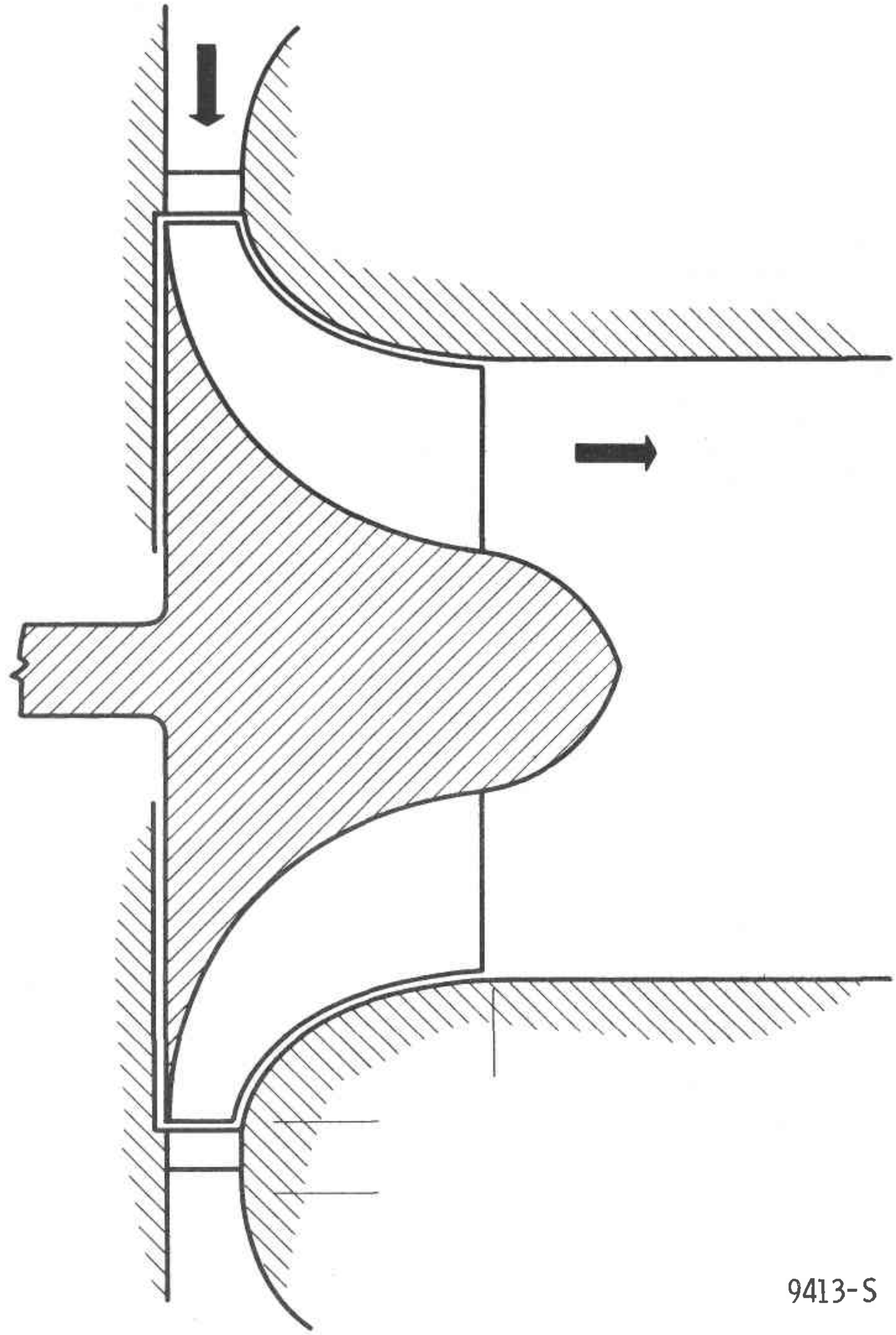


Figure 4-8. - Flow schematic of a radial-flow turbine.

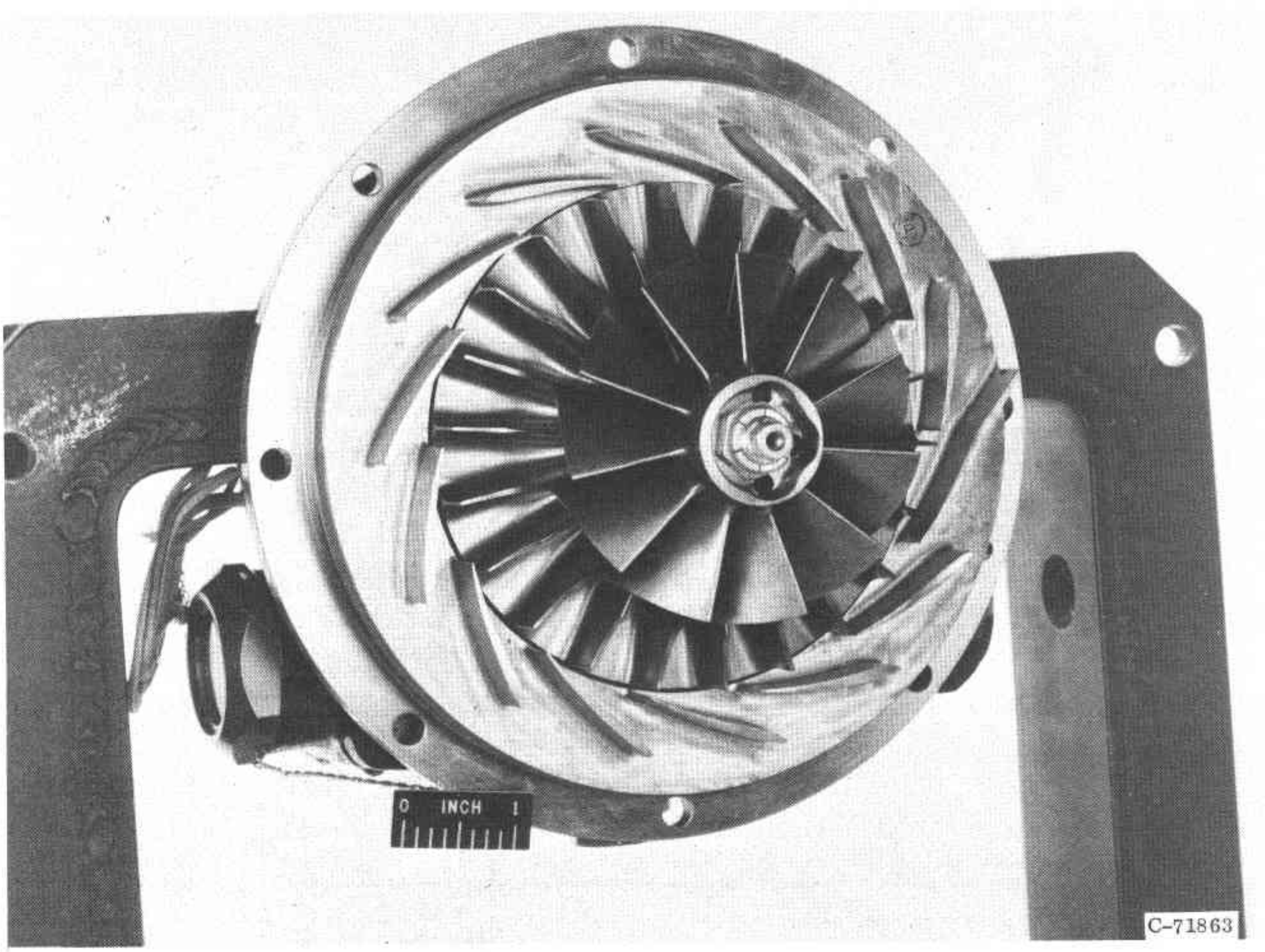


Figure 4-9. - Stator and rotor assembly for typical radial-flow turbine.

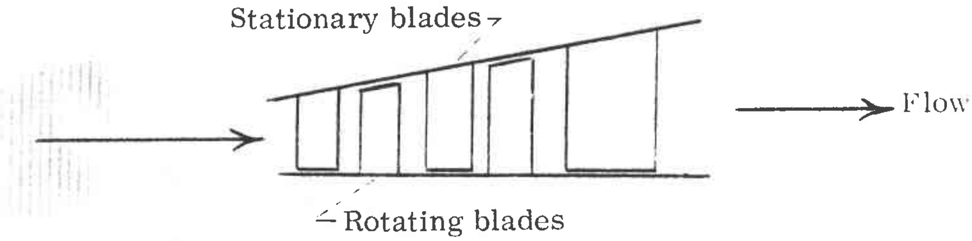


Figure 4-10. - Schematic of axial-flow turbine.

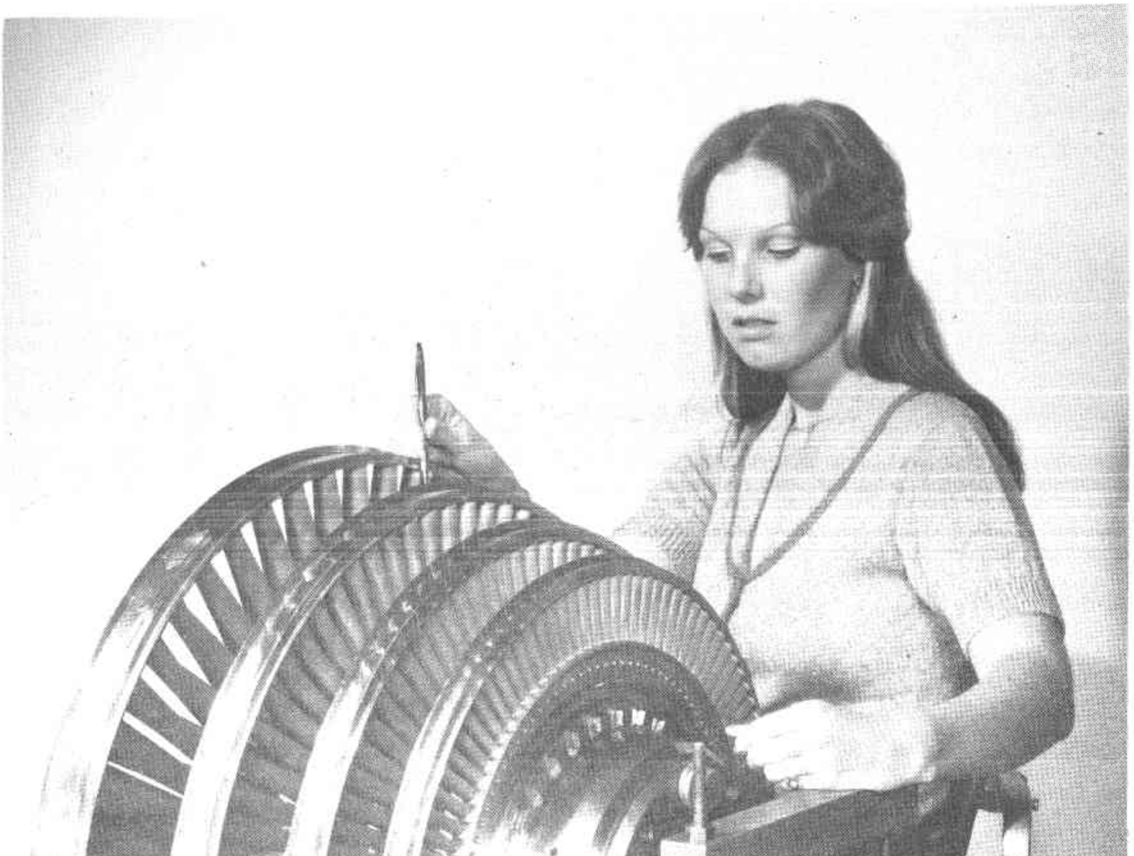


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Figure 4-11. - Four-stage axial-flow turbine.

1. CENTRIFUGAL COMPRESSOR
2. RADIAL TURBINE
3. COMPRESSOR DIFFUSER
4. SINGLE-CAN COMBUSTOR
5. TURBINE WHEEL CONTAINMENT
6. TURBINE PLENUM
7. TORUS
8. LOAD CONTROL VALVE
9. ATOMIZATION AIR PUMP
10. ELECTRIC START MOTOR
11. FUEL CONTROL
12. GENERATOR PAD
13. HYDRAULIC PUMP PAD
14. ACCESSORY GEAR BOX
15. IGNITOR
16. FUEL SOLENOID
17. AIR-OIL SEPARATOR VENT
18. INTEGRAL OIL TANK
19. AIR ASSIST ATOMIZER
20. PLANETARY GEAR SET
21. COMPRESSOR ROLLER BEARING
22. FORWARD MOUNT
23. REAR MOUNT
24. OIL LEVEL SIGHT GLASS
25. COMPRESSOR INLET HOUSING

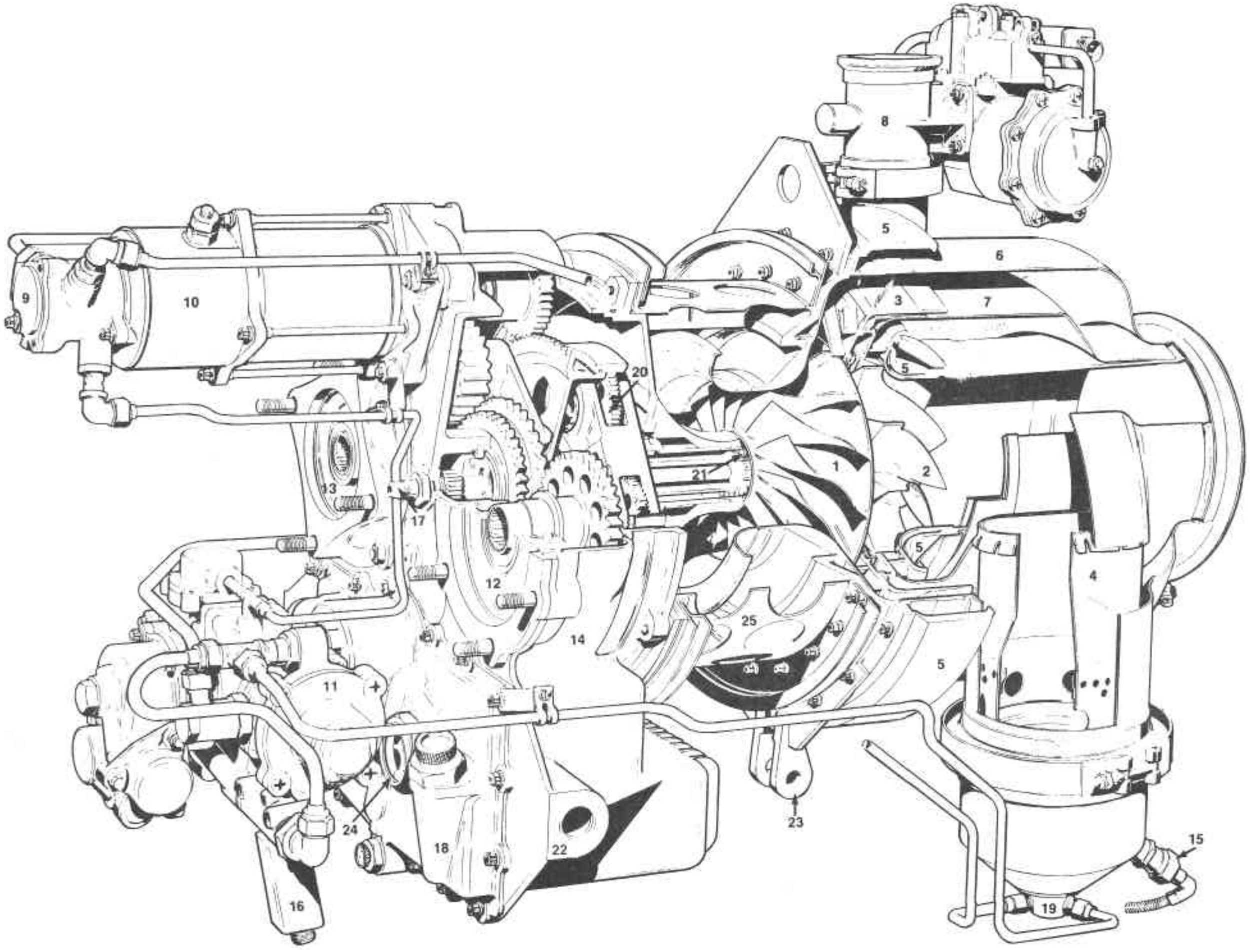


Figure 4-12. - Industrial gas turbine engine with single-can combustor. (Courtesy of AiResearch Manufacturing Co. of America, Division of Garrett Corp.)

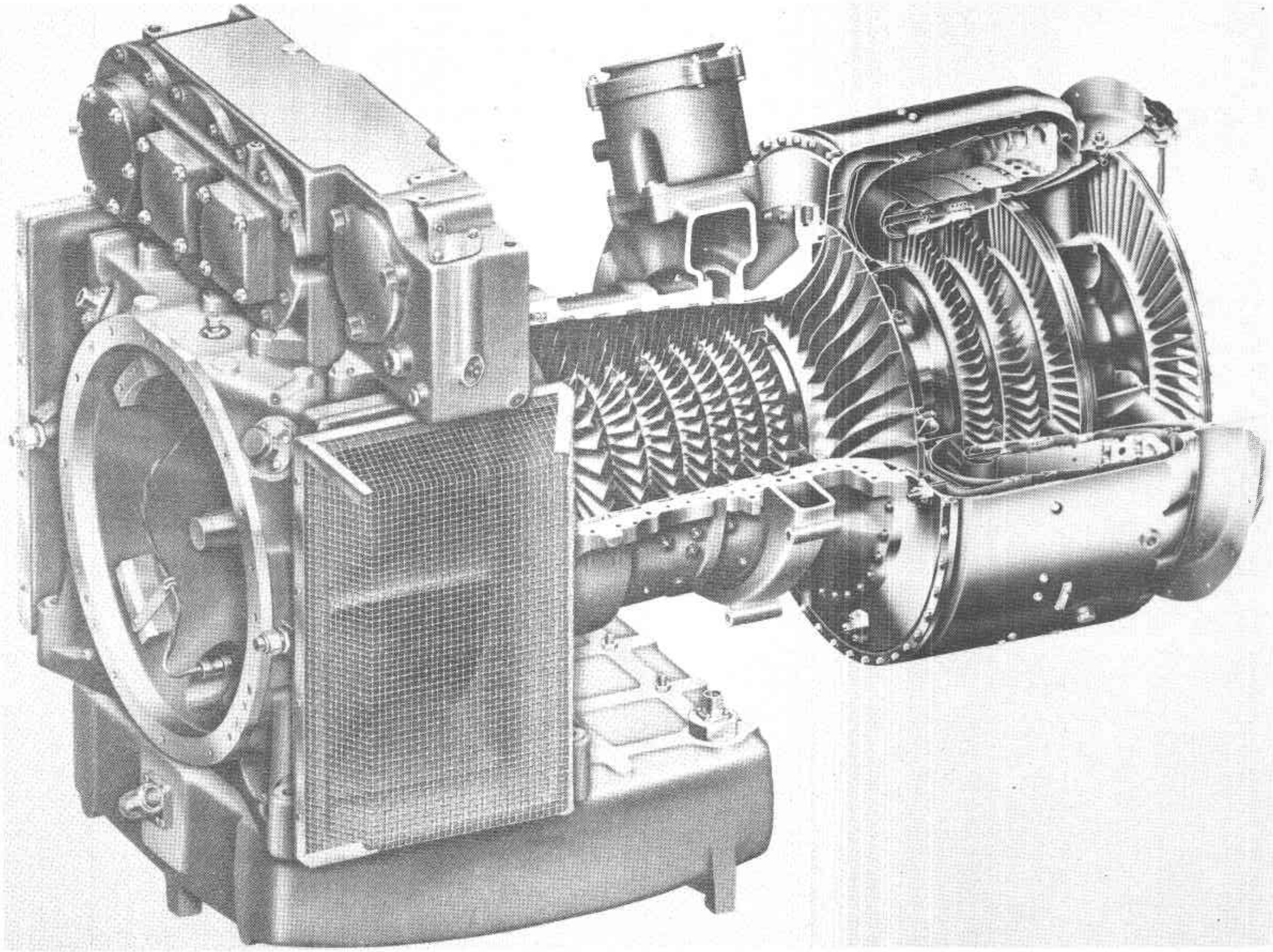


Figure 4-13. - Industrial gas turbine engine with annular combustor. (Courtesy of Avco Lycoming Division.)

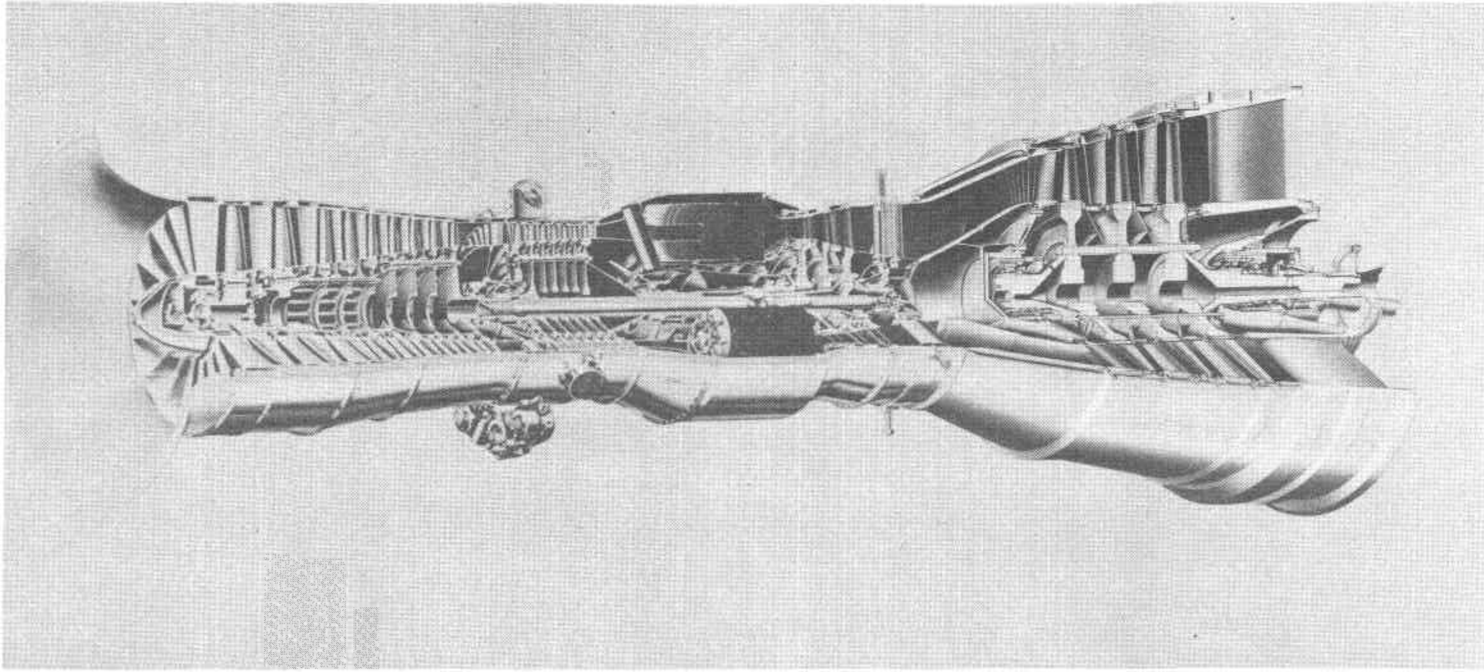


Figure 4-14. - Industrial modular gas turbine engine with can-annular combustor. (Courtesy of United Technologies, Power Systems Division.)

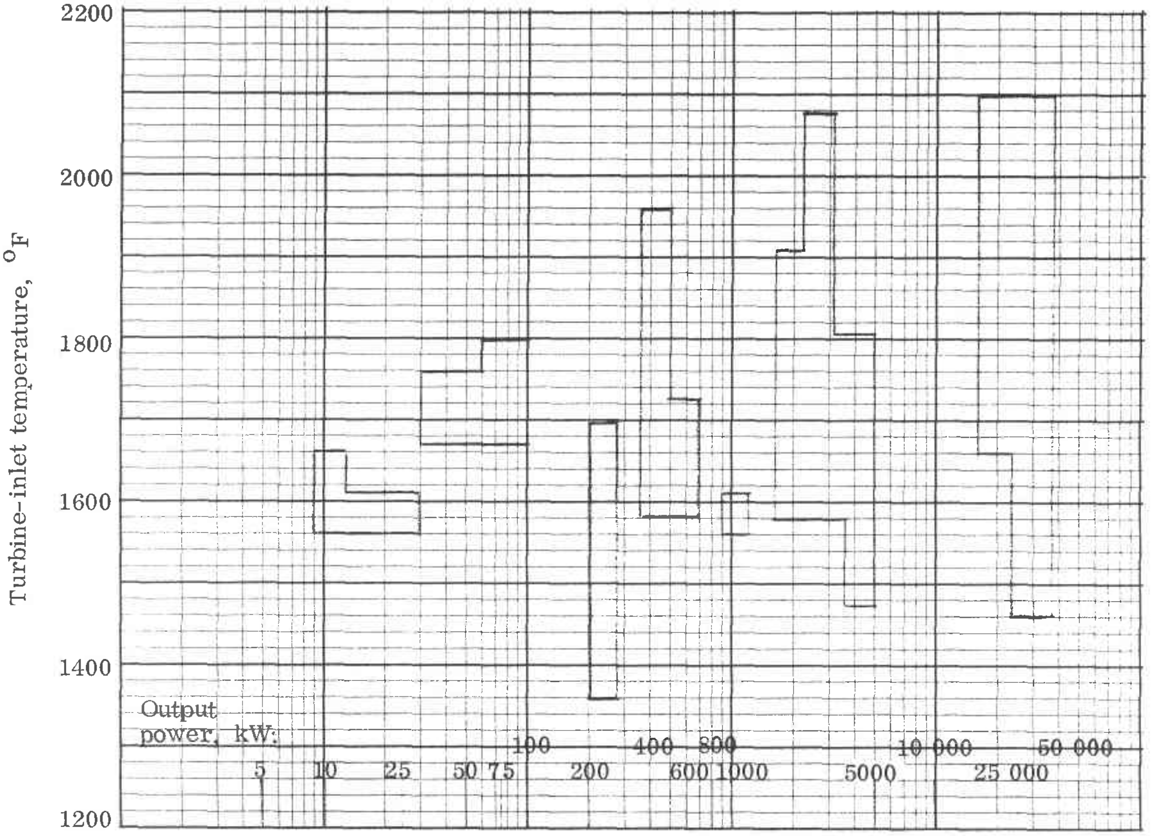


Figure 4-15. - Power level of simple open-cycle gas turbine engines using clean distillate fuels, as function of turbine-inlet temperature.

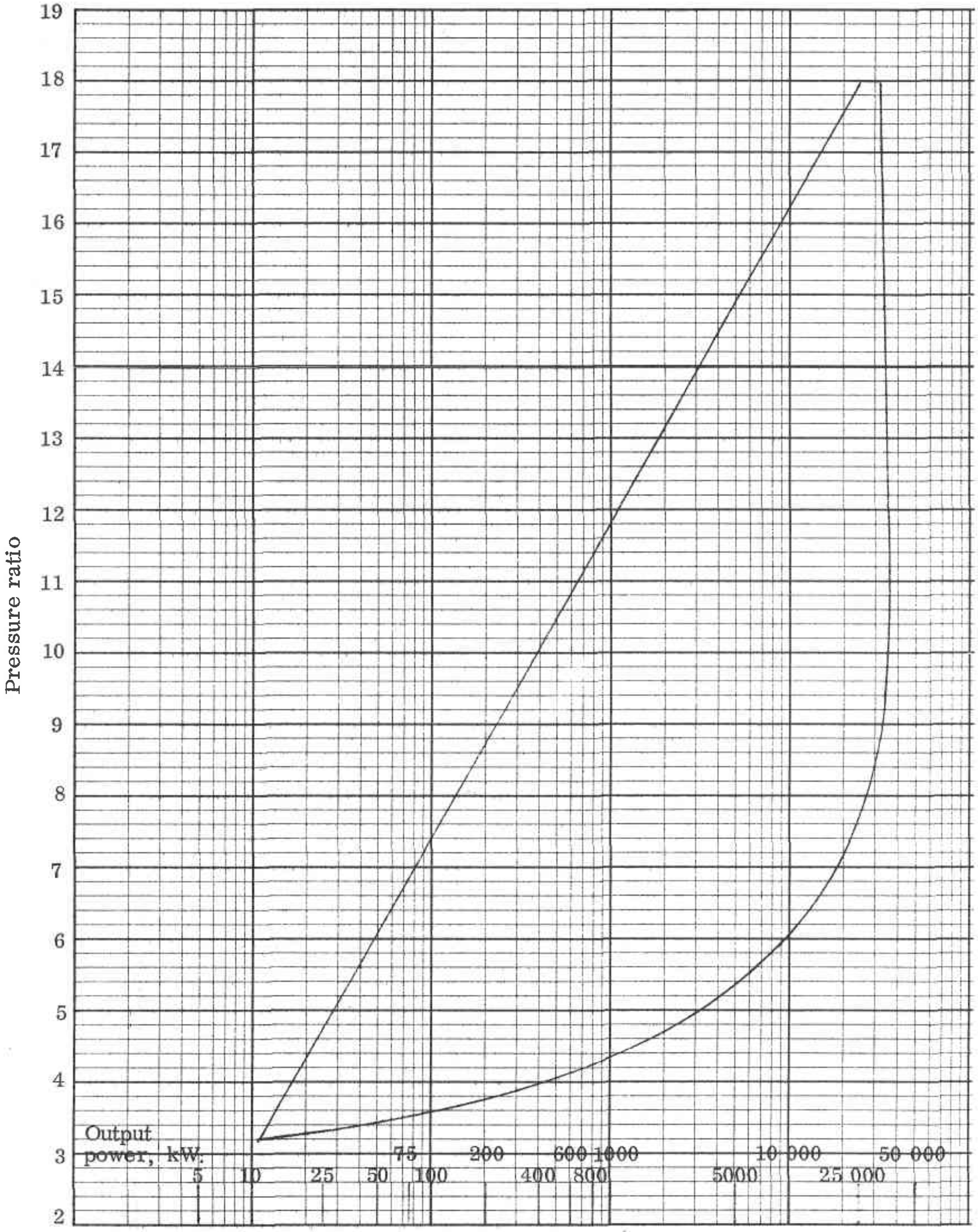


Figure 4-16. - Power level of simple open-cycle gas turbine engines as function of pressure ratio.

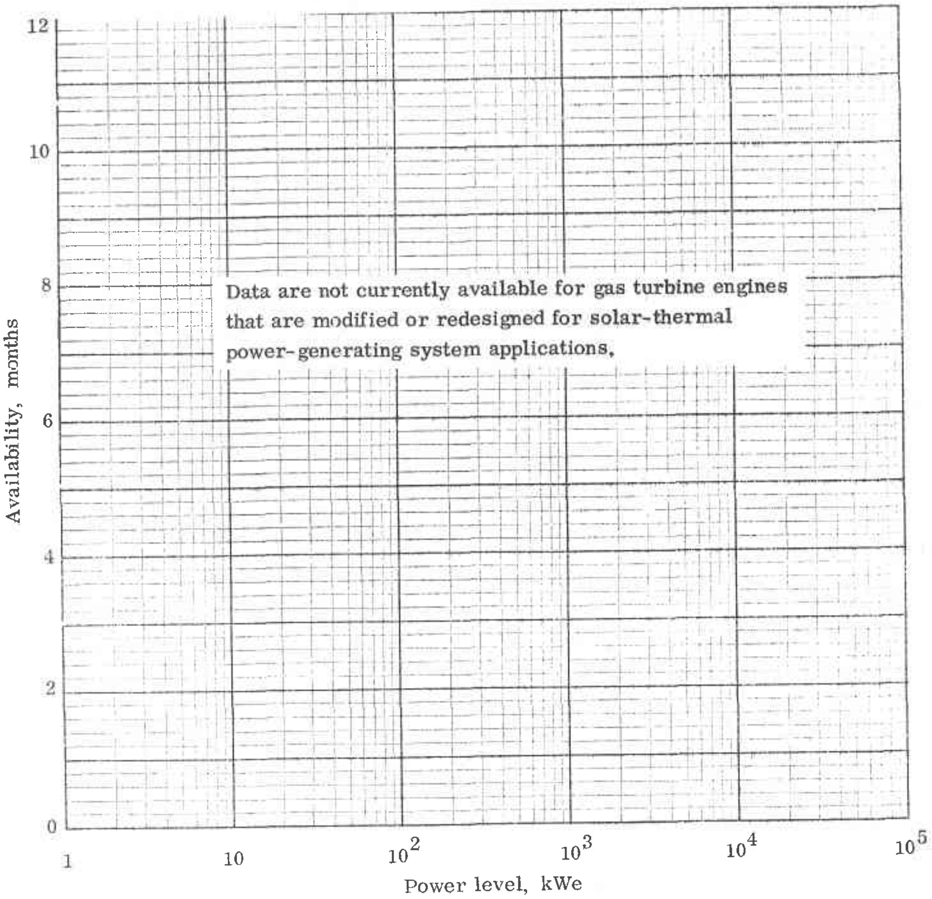


Figure 4-17. - Estimated availability of simple, open-cycle gas turbine engines as function of power level.

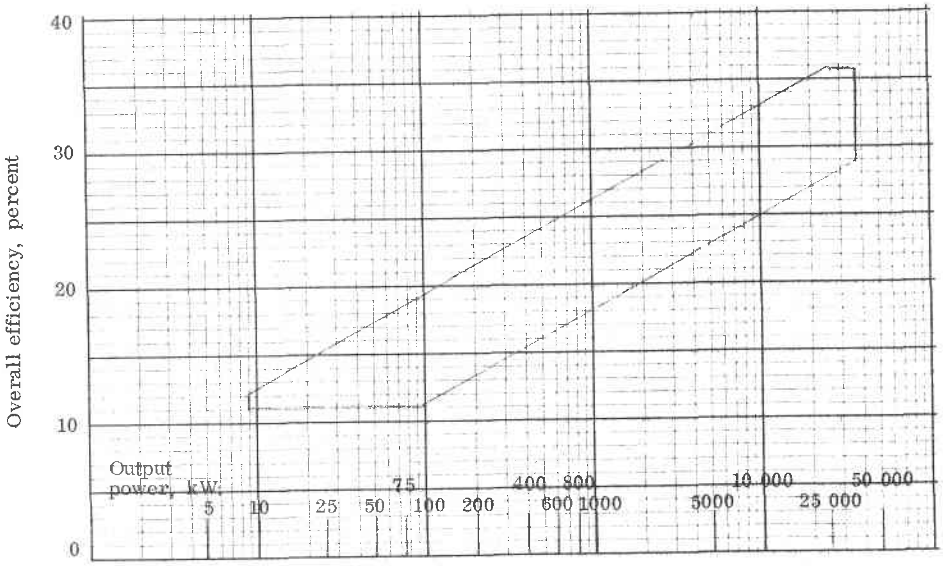


Figure 4-18. - Overall efficiency of simple open-cycle gas turbine engines using clean distillate fuels, as function of power level.

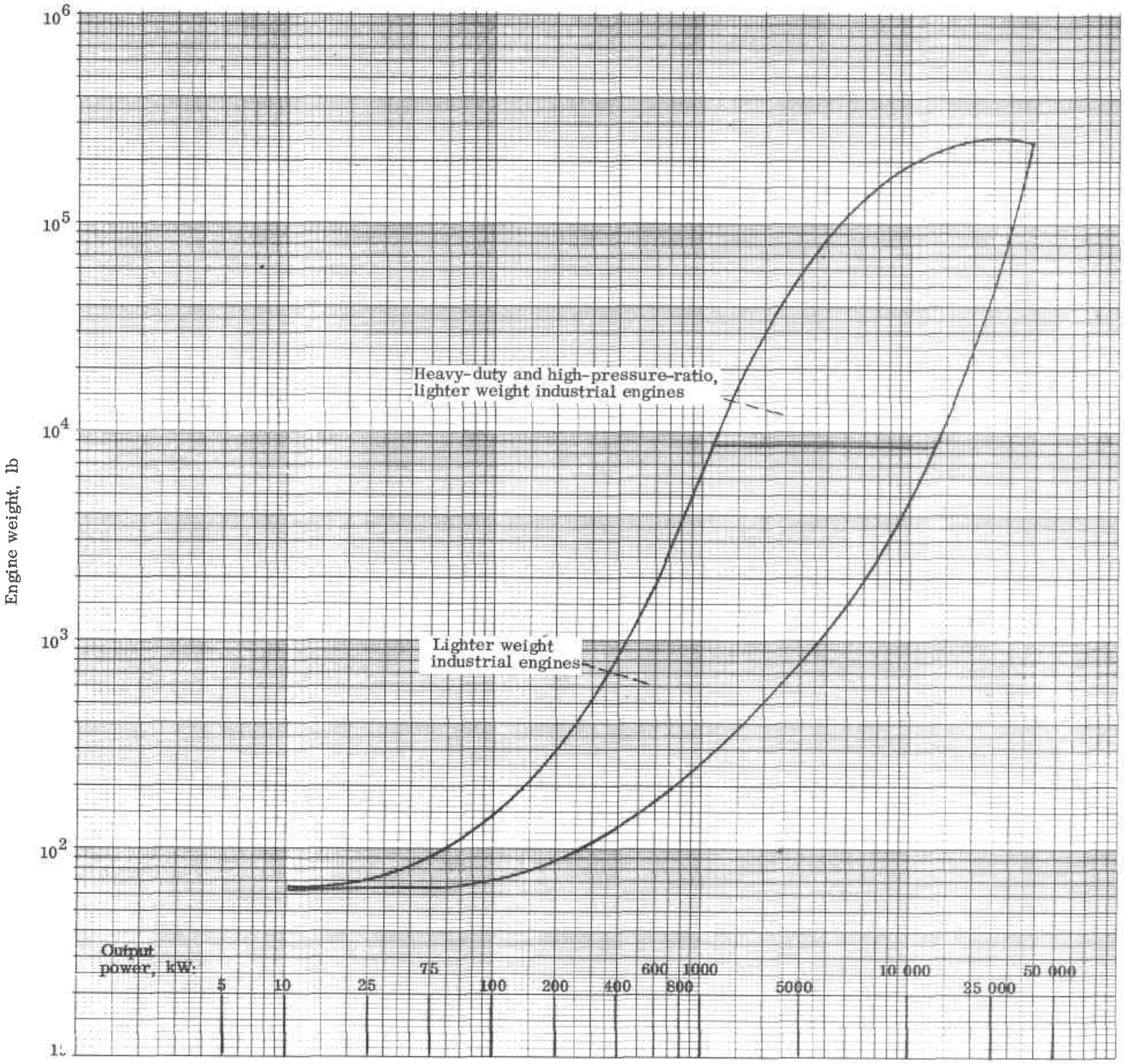


Figure 4-19. - Engine weight of simple open-cycle gas turbine engines as function of power level.

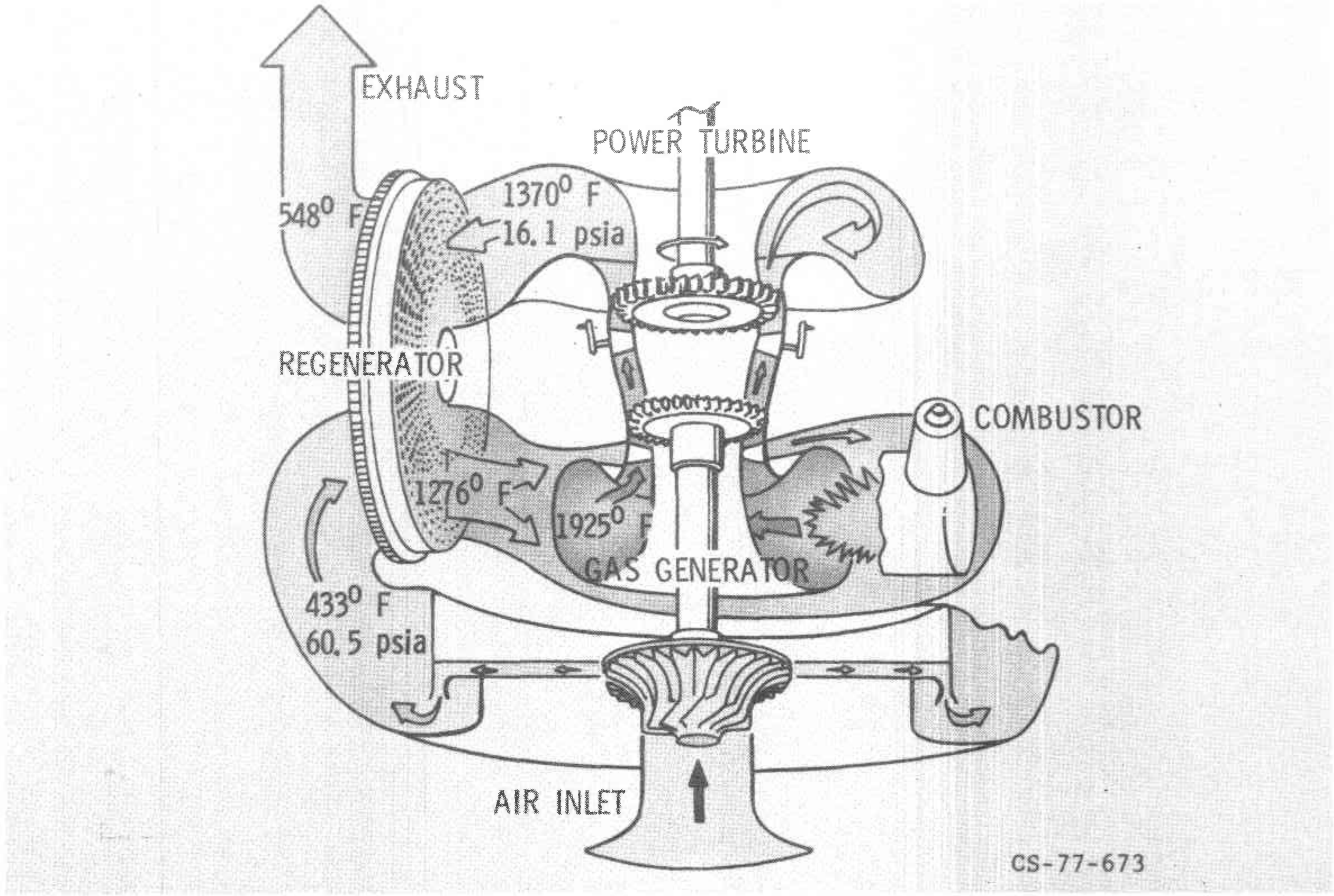
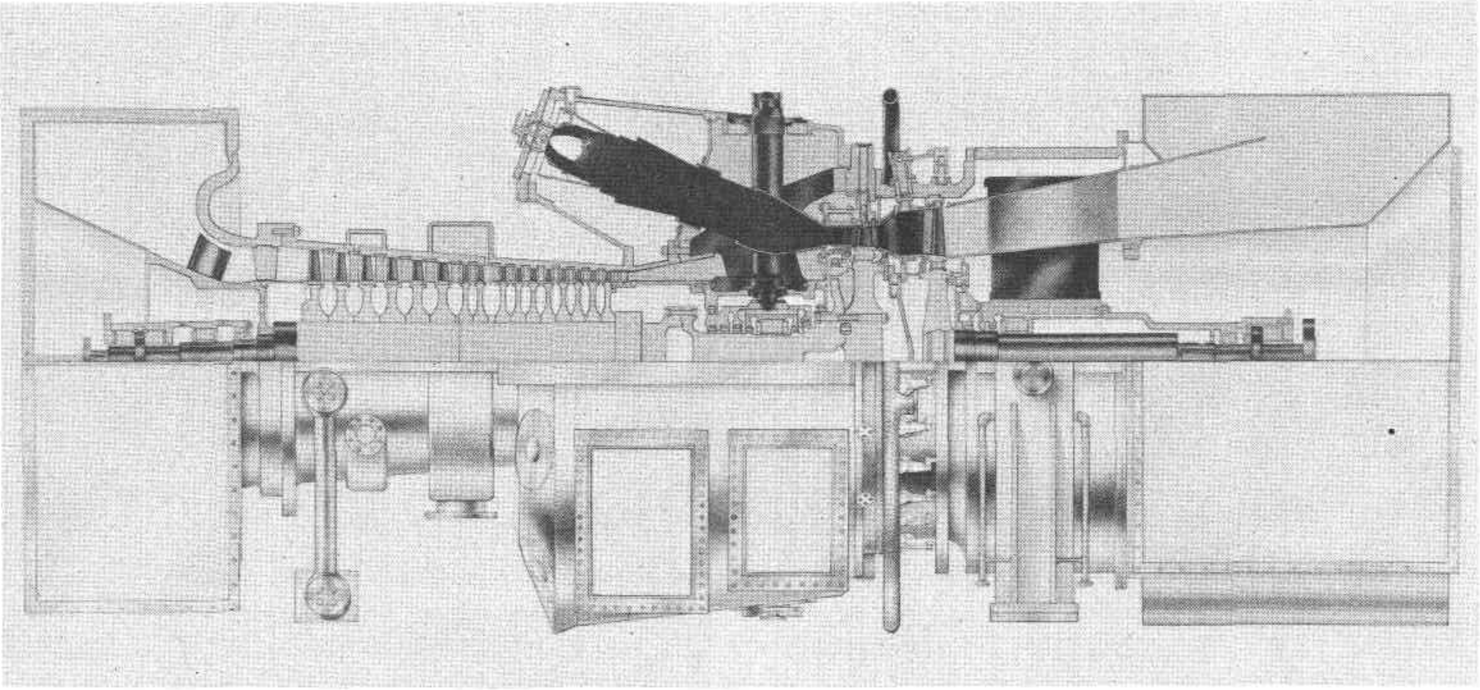
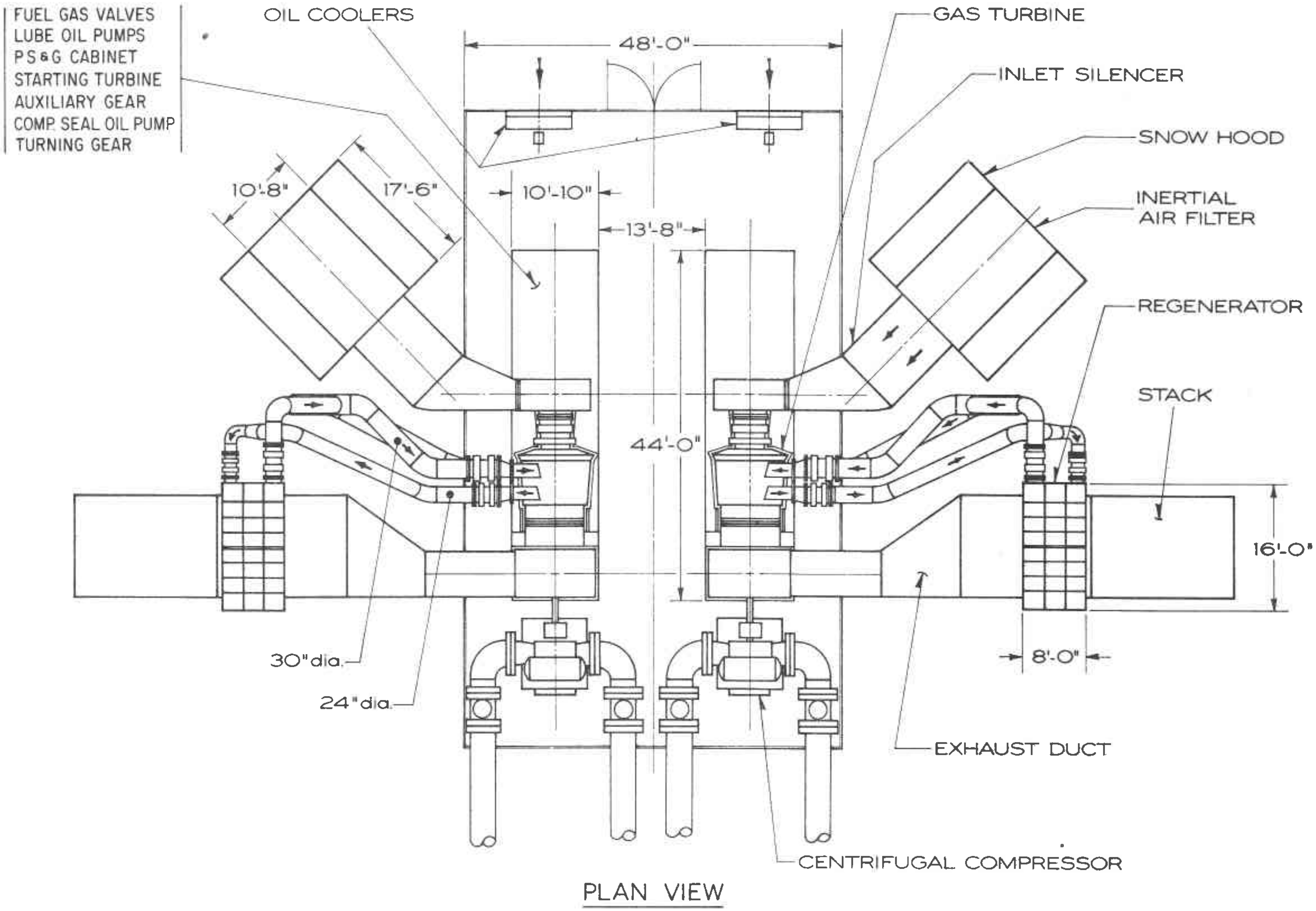


Figure 4-20. - Schematic of automotive gas turbine engine with rotating regenerator.



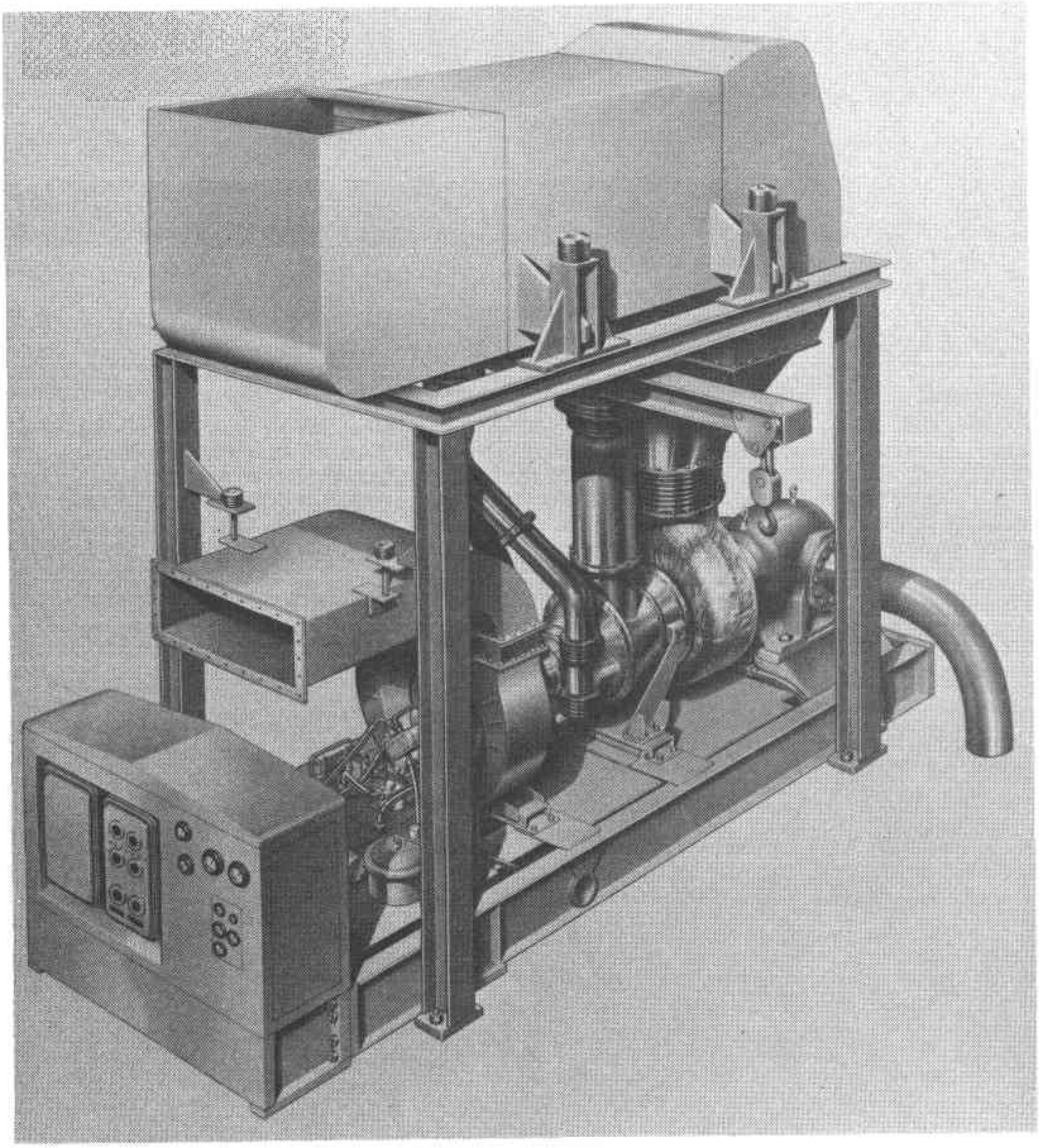
(a) Open-cycle gas turbine engine. (Courtesy of Westinghouse Canada, Ltd.)

Figure 4-21. - Regenerated gas turbine engine.



(b) Flow path and arrangement of open-cycle regenerated gas turbine engine. (Courtesy of Westinghouse Canada, Ltd.)

Figure 4-21. - Continued.



(c) Open-cycle gas turbine engine with overhead-mounted regenerator. (Courtesy of Solar Turbine International.)

Figure 4-21. - Concluded.

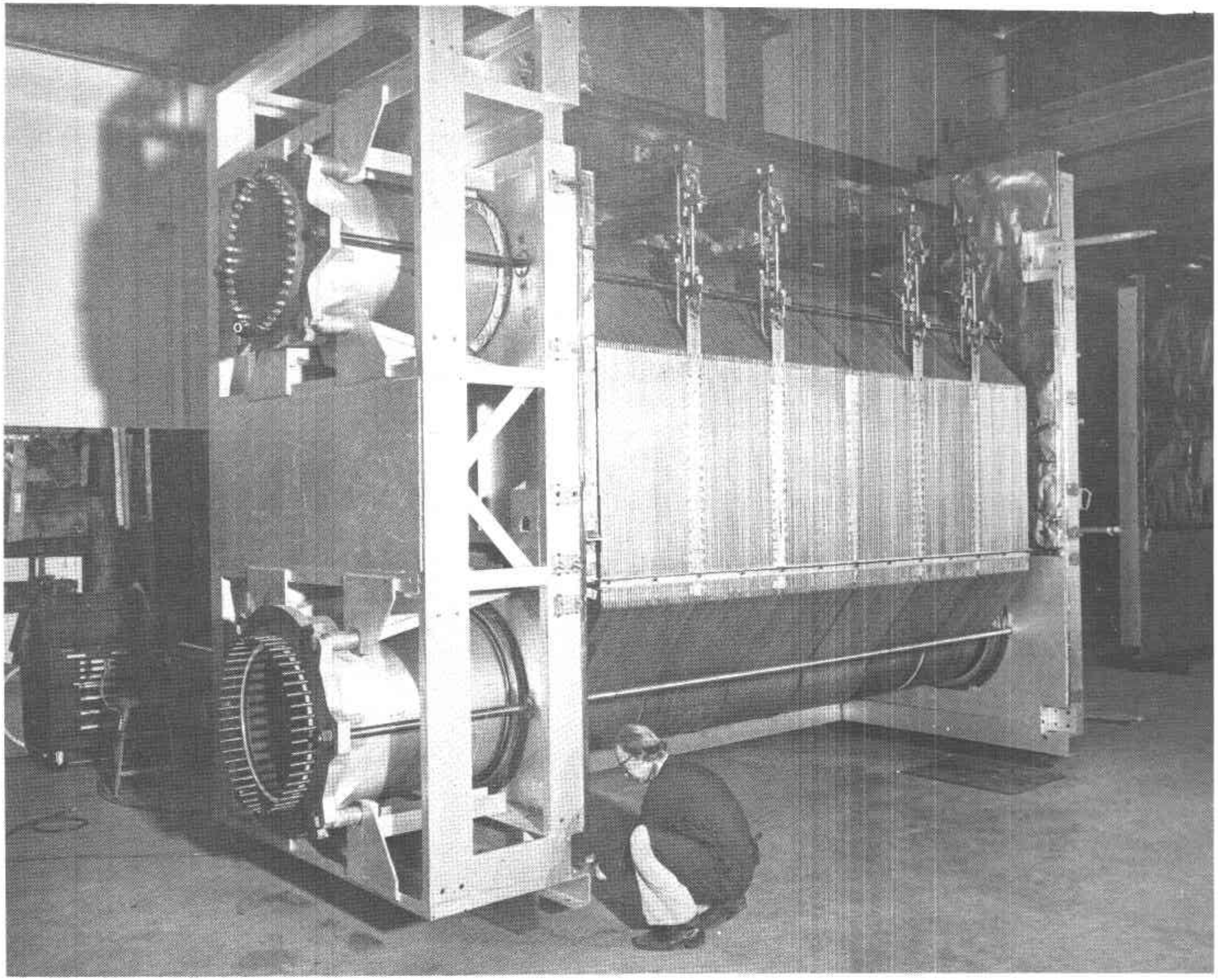


Figure 4-22. - Typical regenerator for gas turbine applications. (Courtesy of AiResearch Manufacturing Co. of California, Division of Garrett Corp.)

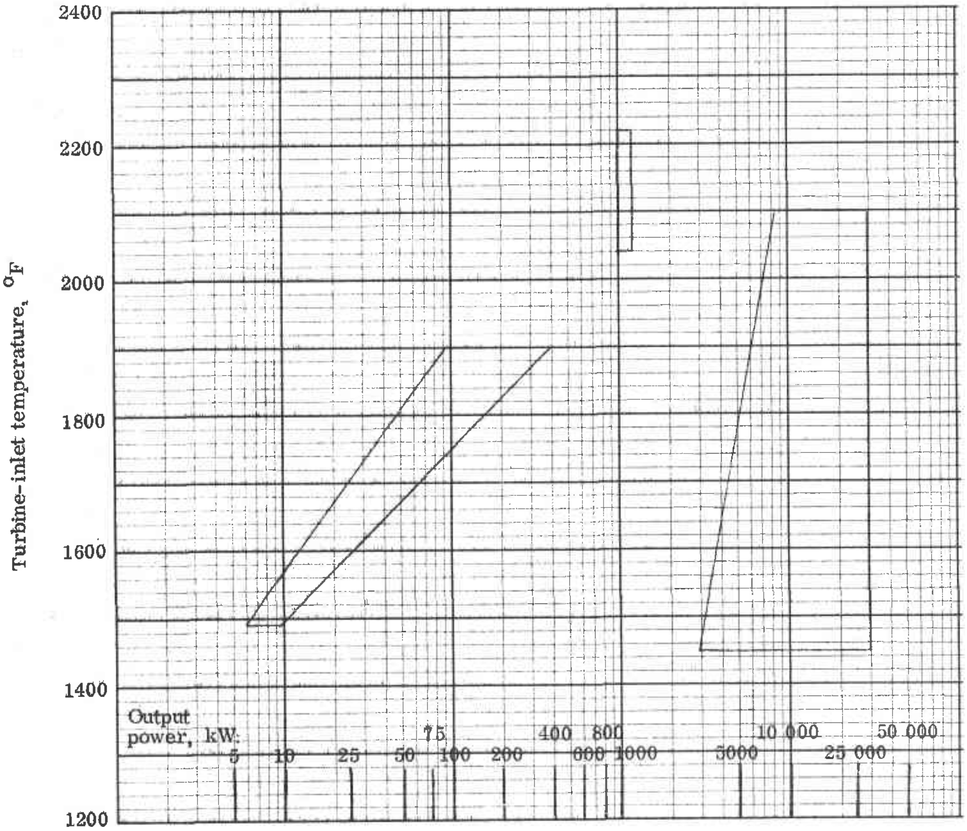


Figure 4-23. - Power level of regenerated open-cycle gas turbine engines using clean distillate fuels, as function of turbine-inlet temperature.

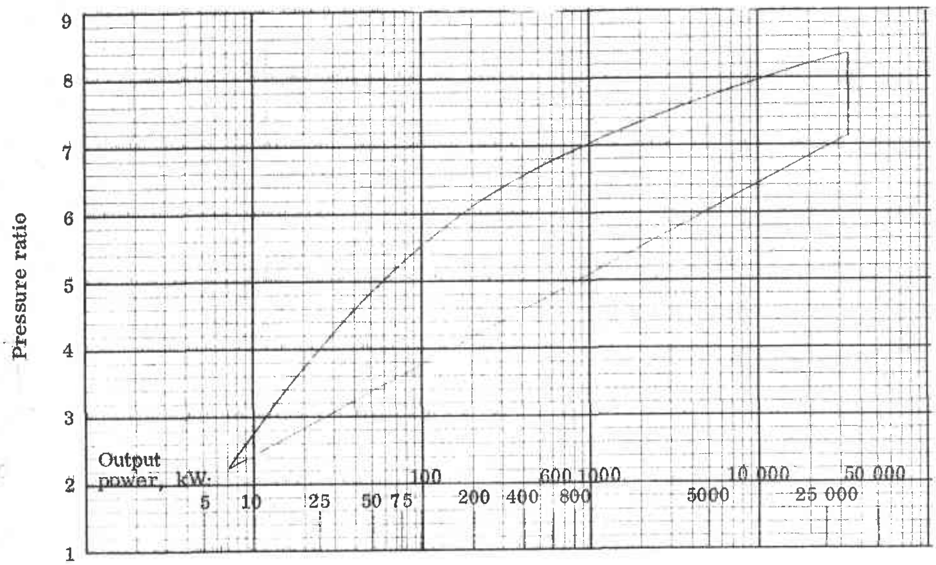


Figure 4-24. - Power level of regenerated gas turbine engines as function of pressure ratio.

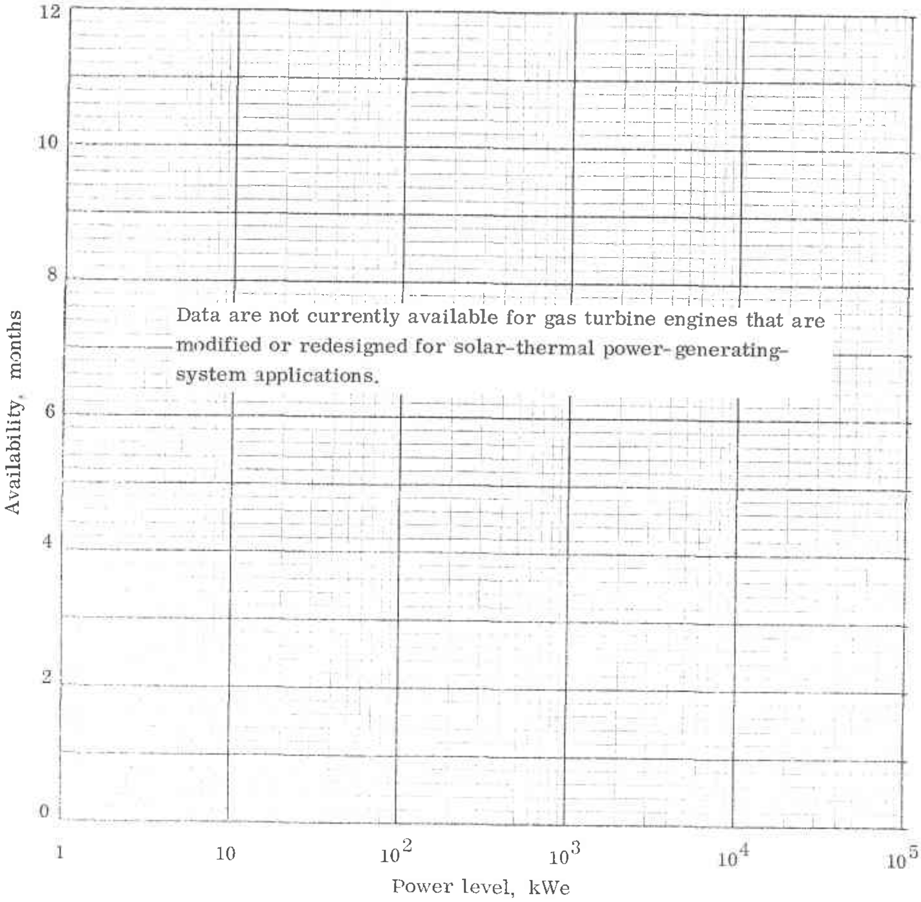


Figure 4-25. - Estimated availability of regenerated open-cycle gas turbine engines as function of power level.

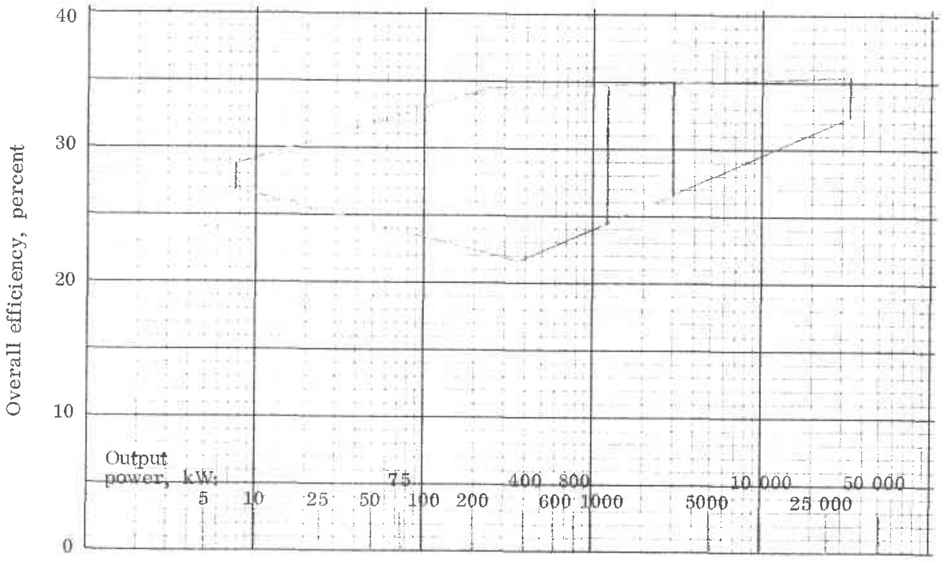


Figure 4-26. - Overall efficiency of regenerated open-cycle gas turbine engines using clean distillate fuels, as function of power level.

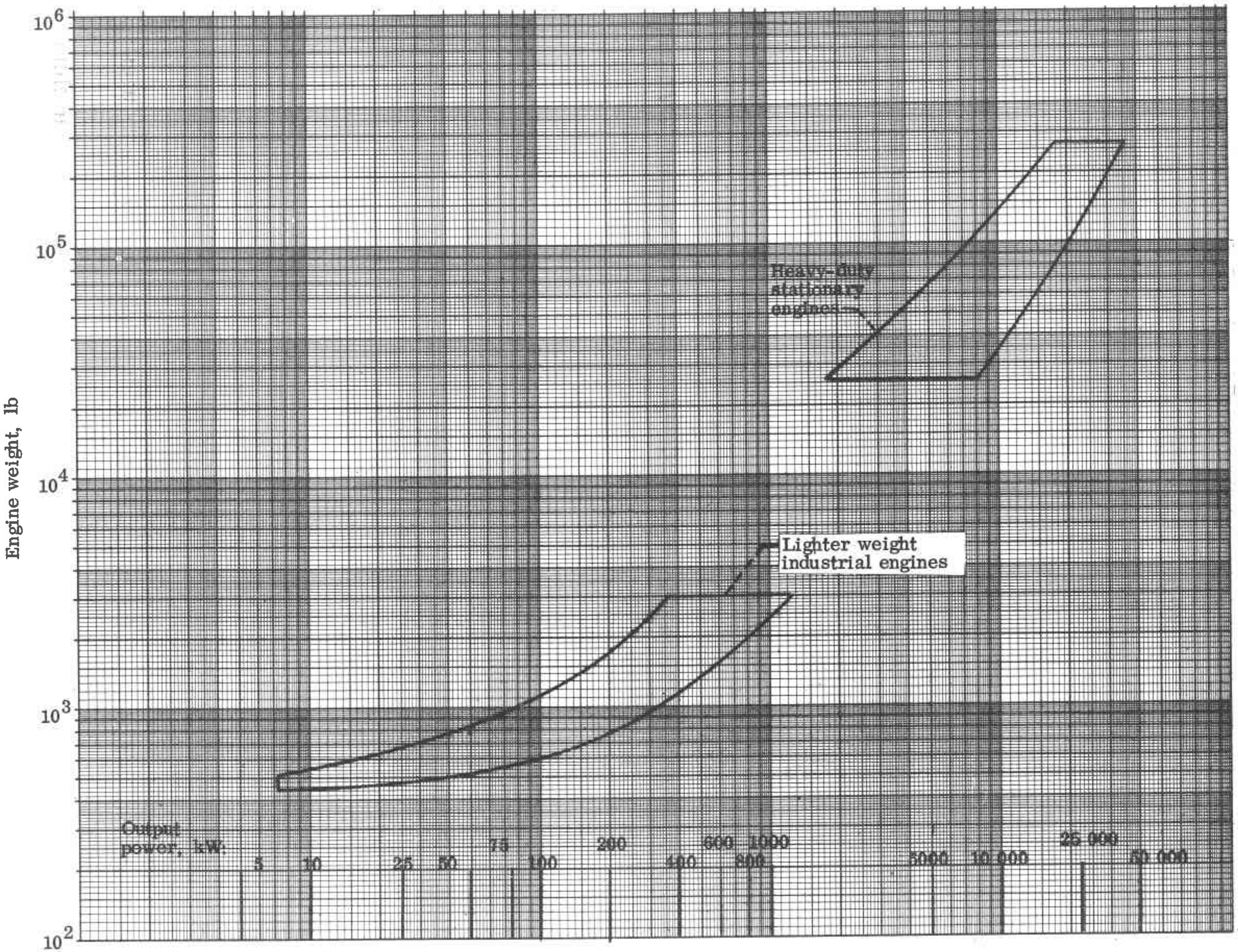


Figure 4-27. - Engine weight of regenerated open-cycle gas turbine engines as function of power level.

5.0 STIRLING ENGINE CHARACTERISTICS

by Harry M. Cameron

The Stirling engine is a closed-cycle, reciprocating, piston heat engine with continuous combustion - like a closed-cycle steam engine. Unlike the steam engine, which uses water, the modern Stirling engine uses a noncondensible, light-molecular-weight gas such as hydrogen or helium as the working fluid.

The Stirling engine, invented in the early 19th century, has been patented by the Philips Co. of Holland beginning in the late 1930's. Through patents and technology licensees, Philips still exercises a dominant worldwide role in the rapidly advancing state of the art of the Stirling engine. Today, the major developers and applications of Stirling engines are

Developer	Application
Philips Co., Holland	Automotive, electrical
Ford Motor Co., U. S.	Automotive
United Stirling, Sweden	Automotive, general
FFV Industrial Products, Sweden	General
Mechanical Technology, Inc., U. S.	Automotive, electrical
M. A. N., Germany	Trucks, tractors
General Electric Co., U. S.	Electric heat pumps
N. A. Philips, U. S.	Electrical
Stirling Power Systems, U. S.	Electrical

In addition, universities and smaller private establishments worldwide are conducting considerable research on the Stirling engine.

The accelerating interest in the Stirling engine stems principally from the following combination of factors:

(1) Excellent fuel economy. A 40-percent efficiency was demonstrated in the laboratory as early as 1960 (ref. 13).

(2) Low exhaust emissions. Continuous, low-pressure combustion produces low exhaust emissions that meet the most stringent EPA standards.

(3) Multifuel capability. Any fuel or heat source, including solar and nuclear, can be used.

(4) Flat torque curve. Torque is high at low speed.

(5) Low noise level. Continuous low-pressure exhaust and the absence of valves produce little noise.

The current state of the art as it relates to Stirling automotive powerplants can be summarized as follows:

(1) The basic technology for an efficient Stirling engine powerplant is well established. The Stirling engine is a complicated machine in which the best overall efficiency is achieved only when the engine components are properly matched. Design codes are well established for optimization.

(2) Laboratory engines have been built and operated for thousands of hours.

(3) Adaptation of the Stirling engine for commercial powerplants is in its infancy. Means for reducing material and mass-production fabrication costs have not been addressed in sufficient detail.

(4) The maintenance of high engine performance over an acceptable period of ownership cannot yet be assured.

5.1 THE STIRLING CYCLE

Pressure-volume and temperature-entropy diagrams of the ideal Stirling cycle are shown in figures 5-1 and 5-2. The cycle consists of isothermal compression (A-B), constant-volume heat addition (B-C), isothermal expansion (C-D), and constant-volume heat rejection (D-A). Heat is added to the working fluid during the constant-volume process (B-C) and the isothermal expansion (C-D); heat is rejected during the constant-volume process (D-A) and the isothermal compression (A-B).

The Stirling engine must employ a regenerator with an effectiveness in the upper-90-percent range to achieve the high engine efficiency for which it is noted. During the constant-volume cooling process (D-A), heat is removed from the working fluid and stored in the regenerator. During the constant-volume heating process (B-C), this heat is returned to the working fluid. The amount of heat involved in these two processes is equal in the ideal cycle. Heat is added to the system, from an external source, only during the isothermal expansion (C-D). Heat is rejected from the system only during the isothermal compression (A-B).

The ideal and textbook thermodynamic equations used to describe the individual processes are not very useful to calculate real-engine characteristics. Unlike the sharp-cornered ideal diagram, the real indicator diagram of a Stirling engine is oval because there cannot be a sharp delineation between the individual processes in a real engine. In 1861, Schmidt developed a theory that adapted single harmonic motion for the reciprocating elements but kept the assumption of isothermal compression and expansion as well as perfect regeneration (in real engines, expansion and compression are more nearly adiabatic). The Schmidt analysis (ref. 14) is still used as a first-order analysis in designing and sizing engine components. The set of basic equations is as follows:

Instantaneous volume of expansion:

$$V_e = \frac{1}{2} V_E (1 + \cos \varphi)$$

where V_E is the swept volume of the expansion space and φ is the crank angle.

Instantaneous volume of compression space:

$$V_C = \frac{1}{2} \kappa V_E [1 + \cos(\varphi - \alpha)]$$

where κ is the swept volume ratio V_C/V_E and α is the angle by which volume variations in the expansion space lead those in the compression space.

Instantaneous volume of total working space:

$$V_W = V_e + V_C + V_D$$

where V_D is the total internal volume of the heat exchangers and the volume of the regenerated and associated ducts and parts.

Instantaneous cycle pressure:

$$p = p_{max} \frac{1 - \delta}{1 + \delta \cos(\varphi - \theta)}$$

where p_{max} is the maximum cycle pressure

$$\delta = \frac{(\tau^2 + \kappa^2 + 2\tau\kappa \cos \alpha)^{1/2}}{\tau + \kappa + 2S}$$

and

$$\theta = \frac{\tan^{-1} (\kappa \sin \alpha)}{\kappa + \kappa \cos \alpha}$$

and τ is the ratio of the temperature of the working fluid in the compression space T_C to the temperature of the working fluid in the expansion space T_E, and S is the reduced volume.

Pressure ratio:

$$Pr = \frac{p_{max}}{p_{min}} = \frac{1 + \delta}{1 - \delta}$$

where p_{min} is the minimum cycle pressure.

Mean cycle pressure:

$$p_{mean} = p_{max}\left(\frac{1 - \delta}{1 + \delta}\right)^{1/2}$$

Net power per cycle:

$$P = (p_{max}V_T)\pi\left(\frac{\tau - 1}{\kappa + 1}\right)\left(\frac{1 - \delta}{1 + \delta}\right)^{1/2} \frac{\delta \sin \theta}{1 + (1 - \delta^2)^{1/2}}$$

where

$$V_T = V_E + V_C = (1 + \kappa)V_E$$

Power per unit mass of working fluid:

$$P_{mass} = \frac{RT_c\pi(\tau - 1)(1 + \delta \cos \theta)(\sin \theta)}{(1 - \delta^2)^{1/2}\left[1 + (1 - \delta^2)^{1/2}\right]\left[\tau + \frac{\kappa}{2}(1 + \cos \alpha) + S\right]}$$

where R is the characteristic gas constant of the working fluid.

Thermal efficiency:

$$\eta = \frac{T_E - T_C}{T_E} = 1 - \tau$$

Heat transferred in expansion space per cycle:

$$Q_E = \pi p_{mean} V_E \frac{\delta \sin \theta}{1 + (1 - \delta^2)^{1/2}}$$

Heat transferred in compression space per cycle:

$$Q_C = - \tau Q_E$$

5.2 STIRLING ENGINE CONFIGURATIONS

The engine that actualizes the thermodynamic cycle includes the following basic components: a heater for transferring heat into the working fluid; a cooler for rejecting heat after expansion; one or more power pistons, displacer pistons, and regenerators; and the mechanical drive subsystem. These are shown schematically in figure 5-3. Many engine types have been built and tested in both single- and multiple-cylinder configurations. In the single-cylinder displacer engine, shown in figures 5-3 and 5-4, both the displacer piston and the power piston reciprocate in a single cylinder. In figure 5-4 both the displacer piston and the power piston are shown in different operating positions, each one corresponding to the end point of the respective ideal thermodynamic process represented in figure 5-1. Although this type of configuration is also suitable for multicylinder arrangements, it has not recently been used.

A common multicylinder configuration is shown in figure 5-5. It does not require separate displacer pistons since the power pistons are double acting, each one performing displacer functions for the next cylinder. Each cylinder has its own separate heater tube, regenerator, and cooler. A cross section of a single-cylinder engine with rhombic drive is shown in figure 5-6. Many such engines have been built by the Philips Co. A rhombic drive consists of two counterrotating shafts, eccentrically mounted connecting rods, and an upper and lower yoke to which the power piston and the displacer piston, respectively, are fastened. The yokes achieve nearly perfect rectilinear motion. The entire drive can be balanced quite precisely. Cross sections of a "V" engine and a straight four-cylinder engine as produced by United Stirling Co. of Sweden are shown in figures 5-7 and 5-8. Both engines have crankshaft drives. Crossheads assure the absence of side loads on the pistons since the thermodynamic working areas of the cylinder must operate without oil lubrication. The pistons are generally fitted with Teflon piston seals. A cross section of a four-cylinder Philips engine configuration that is being developed by the Ford Motor Co. for automotive propulsion is shown in figure 5-9. This engine uses the double-acting-piston principle. A swashplate drive is used for mechanical output.

5.3 FREE-PISTON STIRLING ENGINES

The technology for free-piston Stirling engines - invented by William Beale of Sunpower Inc., Athens, Ohio - is in its early infancy. Such engines promise to give higher efficiency because of lower mechanical losses. However, limited experience with, and development of, these engines preclude a good insight into their true potential.

At the General Electric Co., a 3-kilowatt free-piston Stirling engine drives a free-piston Freon compressor that is part of a heat pump. Several 1-kilowatt engines that drive linear alternators have been designed, built, and operated by MTI. Although the concept of a free-piston engine driving a linear alternator appears attractive for a solar power system, the benefits over a rotary output engine driving a conventional alternator have not been established. Figure 5-10 is a cross section of a free-piston Stirling engine in which the electrical output is provided by a linear alternator.

5.4 STIRLING ENGINE CHARACTERISTICS

5.4.1 Size and Weight

Because Stirling engines are high-pressure engines, production models should be similar to diesel engines in size and weight. The Stirling engines currently being developed and tested are engineering prototypes and thus tend to be heavier than optimized production designs. Their weights vary between 5 and 10 pounds per horsepower. Weight reductions in each subsystem are expected as engine development continues.

5.4.2 Torque and Power

The torque-speed characteristic for a 40-horsepower Stirling engine under development is shown in figure 5-11. The torque curves are of special interest because they clearly illustrate the high torque of the Stirling engine at low engine speeds. This is typical of any closed-cycle engine in which the torque is modulated by changing the pressure of the working fluid. Increasing the mean charge pressure of the working fluid will result in higher power density, as will decreasing the dead volume ratio, which is the ratio of total internal volume to displacement. For high power density, peak operating pressure levels of 3000 psi or higher are being considered.

5.4.3 Efficiency

Both design and operating factors influence engine efficiency. Stirling engines have high efficiency at low speeds. If the engine design speed is at the low end, a lower power density will result. Conversely, higher power density can be obtained by sacrificing efficiency and designing at higher speeds. Other design factors that provide high efficiencies are higher heater temperatures, low cooler temperatures, a low-molecular-weight working fluid (e.g., hydrogen or helium, fig. 5-12), and good thermal isolation of the hot and cold regions of the engine. Typical efficiency plots are shown in figure 5-13 and table 5-1.

5.5 STIRLING-ENGINE-COMPONENT STATE OF THE ART

For commercialization of the Stirling engine, greater reliability, longer life, and lower costs must be achieved. Substantial development from the present prototypes will be required to arrive at a production engine. This description of engine components provides an overview of the requirements and problems for each major component. The shaft seal probably causes the most problems and will require the most serious attention.

5.5.1 Heater Head

The heater-head assembly acts as the heat exchanger for the transfer of heat from the external heat source to the enclosed working gas of the Stirling engine. This assembly is one of the most difficult to design and fabricate. Most heater designs consist of a series of small-diameter tubes with external finning through which the high-pressure working gas passes. If the selected working fluid is hydrogen, the tubes must be resistant to hydrogen embrittlement and diffusion. Philips is testing barrier coatings to reduce the hydrogen loss, but the results have not been published. The external tube surfaces are exposed to the products of combustion and, therefore, are subject to high-temperature oxidation and corrosion. Oxidation products do, however, appear to inhibit hydrogen diffusion. The heater tubes should be designed for minimum pressure drop. In a solar Stirling engine, the heater tubes may be integrated with the heat receiver or an intermediate heat-transport loop such as a heat pipe that uses a heat-transport fluid such as sodium or potassium. One end of each tube communicates with the expansion (hot) space, and the other end communicates with the regenerator. The heater-head assembly must be durable, reasonable to manufacture, and not too costly.

In the early 1960's (ref. 15) a 3-kilowatt-electric Stirling engine was designed, fabricated, and tested for space application. The engine incorporated a NaK (sodium-potassium eutectic mixture) loop to transport heat from the receiver to the heater head, which operated at approximately 677° C. This engine had been designed for Earth-orbital operation. The engine was solar powered and included a lithium-hydride heat-storage subsystem. A cross section of the engine heater head is shown in figure 5-14.

Present Stirling engine heater heads are metal, and thus their operating temperatures are limited by metallic technology to heater-tube temperatures of 720° to 750° C. The Ford/Philips 4-215 engine operates with a 750° C heater-tube temperature, and the United Stirling P-75 maintains a 720° C heater-tube temperature. Increases in heater-tube temperature to 770° to 820° C (expansion gas temperatures of 700° to 750° C) are considered to be possible with improved metallic materials. Gas temperatures of 1000° to 1100° C envisioned for advanced Stirling engines will require the use of ceramics, such as silicon carbide and silicon nitride. Development of ceramic heater heads is just beginning.

5.5.2 Combustors

Combustors for Stirling engines have, in general, been of the standard fixed-geometry type consisting of burner can, fuel atomizer, and igniter. Double-acting engines have normally been arranged so that a single combustor supplies energy to four cylinders. Essential operating requirements are to maintain constant heater-head temperature irrespective of load (variable flow rate, 20/1); to generate uniform heat flux; to minimize hydrocarbon, carbon monoxide, and oxides of nitrogen emissions; and to achieve high combustion efficiency. Some problems have been experienced in developing the uniform heat flux.

5.5.3 Air Preheater

The air preheater in the Stirling engine is a heat exchanger that removes heat from the exhaust gas and transfers it to the air stream going to the combustor. Effective heat transfer results in high engine efficiency. Two types of heat exchanger have been used: the rotary air preheater, and the recuperator. A preheater is not required for a solar Stirling engine.

5.5.4 Regenerator

The regenerator in a Stirling engine is an energy-storage heat exchanger located between the heater and cooler. Its functions are to remove heat from the working fluid as it moves from the expansion space to the compression space, to store the heat in the regenerator matrix, and to return the heat to the working fluid as it moves from the compression space back to the expansion space. Most regenerators have used closely packed fine-mesh screens or monolithic structures for energy storage. Regenerators need a large heat capacity for energy storage, high thermal conductivity, and small flow passages to ensure an adequate surface-film coefficient for heat transfer. The regenerator matrix should have a small void volume since this volume contributes to the engine dead volume. However, smaller flow passages produce larger flow pressure drops, which reduce engine efficiency. Since the regenerator is located between the heater and cooler, the regenerator matrix and container should function as a thermal barrier in the flow direction.

Limited, detailed information has been obtained on regenerator performance in actual Stirling engines. Difficulty in making meaningful measurements (because of the rapidly oscillating flow) has prevented any systematic evaluation of regenerator performance. One of the primary development goals is lower cost.

5.5.5 Pistons and Cylinders

Since lubricants foul the heat-exchanger surfaces, the pistons operate in a non-lubricated environment and are fitted with Teflon rings. Thin-wall construction, long piston skirts, and radiation shielding assure that the temperature in the piston-ring area does not exceed the thermal operating capability of the rings.

In a four-cylinder engine, the pistons are double acting; that is, the hot space is at the top of each cylinder and the cold space at the bottom. In single-acting engines, two pistons - a displacer piston and a power piston - are used. They may both be reciprocating in one cylinder, in which case a rhombic drive is used to achieve the appropriate phase relationships between the two pistons. The displacer piston leads the power piston by a 90^{o} to 120^{o} phase angle. In four-cylinder double-acting engines, phasing between adjacent pistons is usually 90^{o}.

5.5.6 Cooler and Cooling System

The cooler in a Stirling engine is a heat exchanger that removes heat from the working fluid. It consists of a number of small, high-pressure tubes surrounded by a cooling-water jacket. The heat is transferred to the water flowing in the cooler and is rejected to the atmosphere by the radiator. Small-diameter tubes are needed to keep the Reynolds number high enough for effective heat transfer from the working fluid. In double-acting engines, one cooler is required in each flow path connecting a compression space with an expansion space.

5.5.7 Power Control System

The function of the power control system in a Stirling engine is to adjust the engine power to fit the power needed at the load. The primary types of power control systems for Stirling engines are mean-pressure-level control, dead-volume control, stroke control, intermittent-short-circuit control, and phase control.

In the mean-pressure-level control method, the power output of the Stirling engine is controlled by varying the mean cycle pressure of the working gas. The system consists of a servoactuated control valve, a gas distributor, a storage bottle, and a compressor. When the engine load is increased and more horsepower is required, the control valve allows additional working gas to flow from the high-pressure storage bottle into the engine working volume. The distribution mechanism sequences the flow such that the additional working gas enters at the correct time when the cycle pressure is near its maximum. For reduced engine power, the control valve is positioned so that the compressor will pump some of the working gas back into the storage bottle. The position of the control valve depends on a feedback system that senses the mean cycle pressure. A rapid decrease in power is obtained by short-circuiting the hydrogen gas between cylinders. This, however, results in a fuel consumption penalty. The compressor that pumps working gas back to the storage bottle works more when there is leakage in the power control system. This increased compressor work adversely influences fuel economy. In tests, a free-running engine equipped with all auxiliaries has demonstrated a response time of 1 to 1.5 seconds from idle to full power using mean-pressure-level control.

In the dead-volume control method, engine power is controlled by increasing or decreasing the volumes that are available to the working gas and thus modulating the pressure amplitudes in the engine. The quantity of gas in the engine is generally held constant. Volume is increased or decreased in discrete increments through the use of valves. The use of discrete volume increments produces step changes in engine power output. The number of volume elements and valves required makes the dead-volume control approach complex and costly and, therefore, less attractive.

In the stroke control method, engine power is controlled by varying the piston stroke and thus modulating the pressure amplitude in the engine. One way of implementing this approach is by a variable-angle swashplate (VASP) drive. The VASP drive could be coupled directly to the accelerator pedal. Analyses of the VASP drive have been made, but no data from actual hardware tests have been reported. Philips plans to build and evaluate a small Stirling engine with a VASP drive.

In the intermittent-short-circuit control method, engine power is varied by intermittently short-circuiting the different cycles in a double-acting Stirling engine. This method is simple and gives good load-variation response. This power control technique is being pursued by M.A.N.

5.6 SEALS AND HYDROGEN CONTAINMENT

The Stirling engine requires a complex system of seals and surface barriers to contain the high-pressure working gas (hydrogen) and to prevent the lubricating oil from leaking into the gaseous working spaces. The critical seal in the Stirling engine is the piston-rod seal, which is located between the piston and the crosshead member in displacer engines. One seal is required for each piston rod. With an unpressurized crankcase, the seal must maintain positive isolation between the high-pressure working fluid and the ambient crankcase pressure. Two different types of seal are under development: the roll-sock seal, and the sliding-piston-rod seal.

The roll-sock seal system is an oil-supported diaphragm that has its own regulated pressure system to minimize the pressure differential across the diaphragm. The roll-sock seal, which is rigidly attached to both the piston rod and the adjacent wall provides a hermetic seal. The sealing system includes the hydraulic fluid and its regulating valve, which controls the pressure differential across the roll-sock diaphragm to a few atmospheres. The hydraulic system thus provides support for the roll-sock seal. The oil is provided by a metal pumping ring that also functions as a seal. The roll-sock seal does not stretch during its roll maneuver since the steps in the rod and housing are designed to maintain an equal oil volume throughout the piston stroke.

Some problems are associated with the roll-sock seal. It is easily damaged, especially during installation. Because it is made of elastomeric material, it degrades with temperature cycling. Therefore, it is incompatible with most hydraulic control systems to maintain the small differential pressure across the diaphragm. Failure of this support system represents a catastrophic engine failure since the engine has to be completely disassembled to clean the oil from the working gas space. The advantages of the roll-sock seal include its compact design, moderate cost, ability to function satisfactorily at high engine speeds, and reasonable mean lifetime.

The sliding-piston-rod seal is a multielement seal that includes a scraper ring and an oil-gas separator. One element converts the oscillating working pressure in the engine cycle to a constant pressure near the minimum cycle pressure. The minimum cycle pressure is maintained by a check valve that returns leaking hydrogen gas from the separator to the cycle. Lubricating oil that is deposited on the piston rod during its reciprocating motion is removed by the scraper ring, sent to the oil-gas separator, and then returned to the engine oil system. The main seal element maintains the difference between the minimum cycle pressure and the ambient crankcase pressure. The sliding-piston-rod seal concept just described was developed by United Stirling. Although much development work remains to be done, the sliding-piston-rod seal looks attractive because it has a more acceptable failure mode. It does, however, produce higher friction losses.

The hydrogen containment problem is not limited to the elastomeric seals. Some hydrogen is lost through the metal walls of the engine since hydrogen can diffuse through a metallic lattice, especially at elevated temperature. Proprietary surface coatings developed by Philips are reported to reduce the working-fluid loss through the metal parts of an engine by factors of 25 to 50.

5.7 EXPERIMENTAL ENGINES

5.7.1 The Philips 1-98 Engine

Philips has built about thirty 1-98 engines. The engine has one cylinder and a piston swept volume of 98 cubic centimeters. It operates with a heater temperature from 250° to 850° C and produces as much as 20 kilowatts of power. It is capable of delivering about 15 kilowatts at 3000 rpm and 220-atmosphere gas pressure. This type of engine has been successfully used by Philips and others in engine and component development.

5.7.2 The Philips-Ford 4-215 Engine

The 4-215 is a nominal 200-horsepower engine developed by Philips in 1972 (ref. 16). Philips has since collaborated with Ford to incorporate this engine in a Ford Torino automobile. About six 4-215 engines have been built, and several have been installed and tested in automobiles.

5.7.3 The 4-98 Engine

The 4-98 is a nominal 80- to 100-horsepower engine design that has not been built. Calculated engine weights are shown in table 5-2, and design information is given in table 5-3.

5.7.4 United Stirling of Sweden Engines

United Stirling has been the most prolific designer and builder of various types of Stirling engines. Their characteristics are listed here.

(1) P-75 - a four-cylinder engine
  (a) Total swept volume - 4 × 189 cubic centimeters
  (b) Performance - 67 kilowatts at 2400 rpm
  (c) Maximum efficiency - 35 percent
  (d) Applications - prototype engine for component development

(2) V4X - a four-cylinder V engine
  (a) Total swept volume - 4 × 90 cubic centimeters
  (b) Performance - 40 kilowatts at 4000 rpm
  (c) Maximum efficiency - 28 percent
  (d) Applications - demonstration of Stirling engine characteristics in cars, vans, and tractors and component development

(3) P-40 - a 4-cylinder engine with dual-crankshaft drive (convertible to swashplate drive)
  (a) Total swept volume - 4 × 95 cubic centimeters
  (b) Performance - 40 to 55 kilowatts at 3000 to 4800 rpm
  (c) Maximum efficiency - 35 percent
  (d) Applications - demonstration of Stirling engine characteristics in cars, vans, and tractors and development of more efficient engine by using ceramics in some heated components

(4) P-75 - a four-cylinder engine with dual-crankshaft drive (convertible to swashplate drive)
  (a) Total swept volume - 4 × 275 cubic centimeters
  (b) Performance - 75 kilowatts at 2200 rpm (production engine)
  (c) Maximum efficiency - 37 percent

(d) Applications - demonstration in trucks, buses, and construction equipment

(5) P-150 - an eight-cylinder engine with dual-crankshaft drive (two P-75 engines)

(a) Total swept volume - 8 × 275 cubic centimeters

(b) Performance - 150 kilowatts at 2200 rpm (production engine)

(c) Maximum efficiency - 37 percent

(d) Applications - demonstration in trucks, buses, and construction equipment

Other engines are listed in table 5-1.

5.7.5 Laboratory Engines for Stirling Cycle Investigations

The Jet Propulsion Laboratory recently completed the installation of a laboratory engine for the investigation of the fundamental Stirling process and components. The engine was designed with a high degree of flexibility to permit changes in operating characteristics and components without major engine modifications. The engine is illustrated in figure 5-15 (ref. 17), and the engine specifications are given in table 5-4. Another laboratory engine was recently designed for the same purpose by the General Electric Co. for the Lewis Research Center. A cross-sectional view of this engine is shown in figure 5-16, and the engine characteristics are listed in table 5-5. This engine has not yet been built.

TABLE 5-1. - ENGINE EFFICIENCIES

Manufacturer	Model	Number of cylinders	Performance, hp	Engine speed, rpm	Maximum pressure, psi	Heater temperature, ^{o}F	Working gas	Efficiency, percent	Inlet coolant temperature, ^{o}F	Drive
Philips	1-98	1	10	3000	1600	----	Helium	28	---	Rhombic
	1-365		40	2500		----	Hydrogen	35	---	
	1-400		30	1500		----	Helium	30	---	
	1-980		85	1500		----	Hydrogen	33	---	
	4-1080	4	360	1500		----	Hydrogen	26	---	
	4-215	--	175	4000	2700	----	Hydrogen	--	---	Swashplate
United Stirling	P-40	4	53	3000	2175	1325	---------	35	77	Crankshaft
	P-75	4	100	2200	2175	----	---------	37	---	
	P-150	8	200	2200	2175	----	---------	37	---	
	V4X	4	53	4000	----	1400	---------	28	100	
General Motors Research Laboratory	GPU-3	1	11	3600	1000	1400	---------	26.5	---	Rhombic
Detroit Diesel Allison	PD46	1	4	3000	1500	1250	---------	32	---	Rhombic

TABLE 5-2. - WEIGHT DISTRIBUTION IN

FORD 4-98 STIRLING ENGINE

Engine section	Weight, lb
Preheater (burner, recuperator, and water pump)	75
Heater head (base, tubes, regenerators, and cylinders)	115
High-pressure crankcase (reciprocating parts and power control)	70
Engine drive (swashplate, flywheel, oil pump, second block, and blower)	100
Accessory drives	40
Radiator (including water)	70
Total	470

TABLE 5-3. - DESIGN PARAMETERS FOR

FORD 4-98 STIRLING ENGINE

Dimensions, cm	$25.5 \times 20.6 \times 23$
Gross input power, kW	90
Gross indicated efficiency, percent	40.8
Design speed, rpm	5400
Bore, mm	56
Stroke, mm	40
Number of cylinders	4
Swept volume, cm^3	98.5
Number of heater tubes	72
Number of cooler tubes	2440
Working fluid	Hydrogen
Drive	Swashplate

TABLE 5-4. - DESIGN SPECIFICATIONS AND WEIGHTS FOR JPL LABORATORY RESEARCH STIRLING ENGINE

(a) Design specifications

Design power at 3000 rpm, hp (kW)	12.2 (9.0)
Maximum torque at 1700 rpm, ft-lb (N-m)	25 (34)
Cylinder bore, in. (mm)	2.875 (73.0)
Piston stroke, in. (mm)	2.125 (53.8)
Crosshead diameter, in. (mm)	2.500 (63.5)
Connecting rod length, in. (mm)	5.188 (131.8)
Number of crankshafts	2
Number of cylinders	2
Phase shift between crankshafts	Variable
Swept volume per cylinder, in^3 (cm^3)	13.79 (225.9)
Swept volume ratio	1.00
Design engine speed, rpm	4000
Design mean pressure, atm (kN/m^2)	68 (6900)
Working gas	Helium
Heater inside-wall temperature (design), ^oF (^oC)	1400 (760)
Cooler inside-wall temperature (design), ^oF (^oC)	100 (38)
Number of cooler tubes per cylinder	109
Cooler-tube length, in. (mm)	5.98 (151.9)
Cooler-tube inside diameter, in. (mm)	0.069 (1.75)
Cooler dead volume, in^3 (cm^3)	2.44 (39.9)
Number of heater tubes per cylinder	28
Heater-tube length, in. (mm)	11.61 (294.9)
Heater tube inside diameter, in. (mm)	0.142 (3.61)
Heater dead volume, in^3 (cm^3)	4.98 (81.6)
Regenerator matrix length, in. (mm)	3.75 (95.2)
Number of regenerators per cylinder	1
Regenerator matrix configuration	Parallel-plane type
Regenerator dead volume, in^3 (cm^3)	7.12 (116.7)
Clearance volume, in^3 (cm^3)	0.55 (9.01)
Miscellaneous internal volume, in^3 (cm^3)	1.5 (24.6)
Total dead volume, in^3 (cm^3)	16.59 (271.8)
Dead volume ratio	1.2

(b) Weights

Subassembly	Weight	
	lb	kg
Lubrication systems (dry)	122.6	55.6
Crankcases	96.2	43.6
Cylinder blocks	39.7	18.0
Crankshafts	20.8	9.4
Pistons and crossheads	4.5	2.0
Connecting rods and pins	3.5	1.6
Flywheels	70.4	31.9
Cooler (dry)	21.5	9.8
Regenerator	8.0	3.6
Heater (dry)	66.1	30.0
Miscellaneous	5.0	2.3
Total	458.3	(207.8)

TABLE 5-5. - CHARACTERISTICS OF LEWIS

GENERAL-PURPOSE STIRLING TEST ENGINE

Characteristic	Component or parameter involved
Operational flexibility	Regenerator-cooler housing Regenerator matrix Cooler Liner Piston seals Rod seals Heater head
Operating parameter adjustability	Phase angle Piston-head clearance Swept volume Dead volume
Ease of maintenance	------------------------
Uniformity of mass flow and heat transfer	------------------------ ------------------------
Ease of instrumentation	------------------------
Moderate cost	------------------------
State-of-the-art technology	------------------------
Low vibration	------------------------
Power, hp	24
Brake efficiency at maximum power, percent	33
Maximum speed, rpm.	5400
Output speed variation, percent	<1

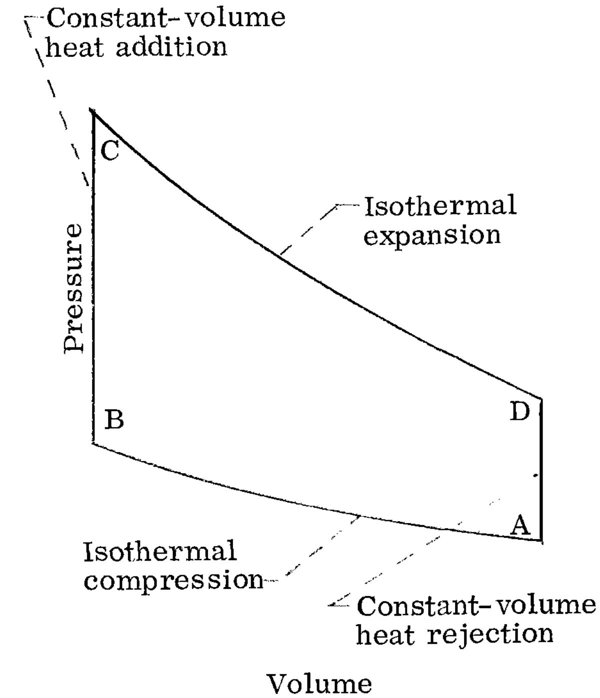


Figure 5-1. - Pressure-volume diagram for an ideal Stirling cycle.

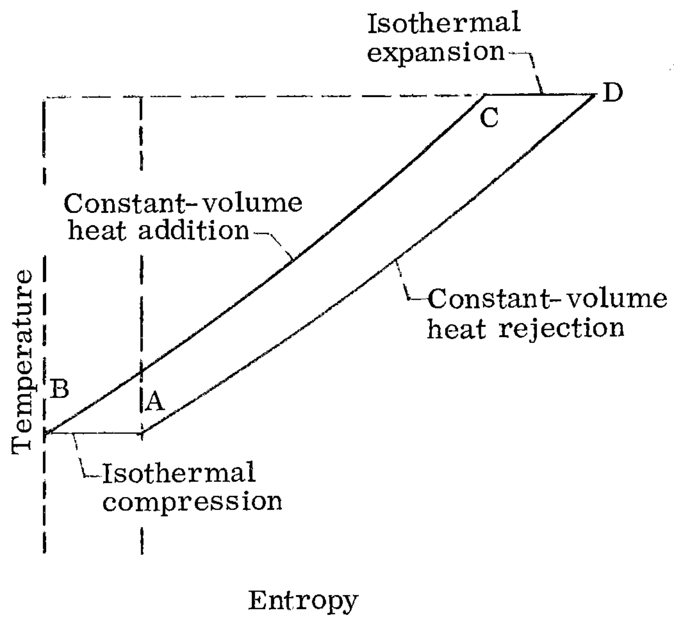


Figure 5-2. - Temperature-entropy diagram for an ideal Stirling cycle.

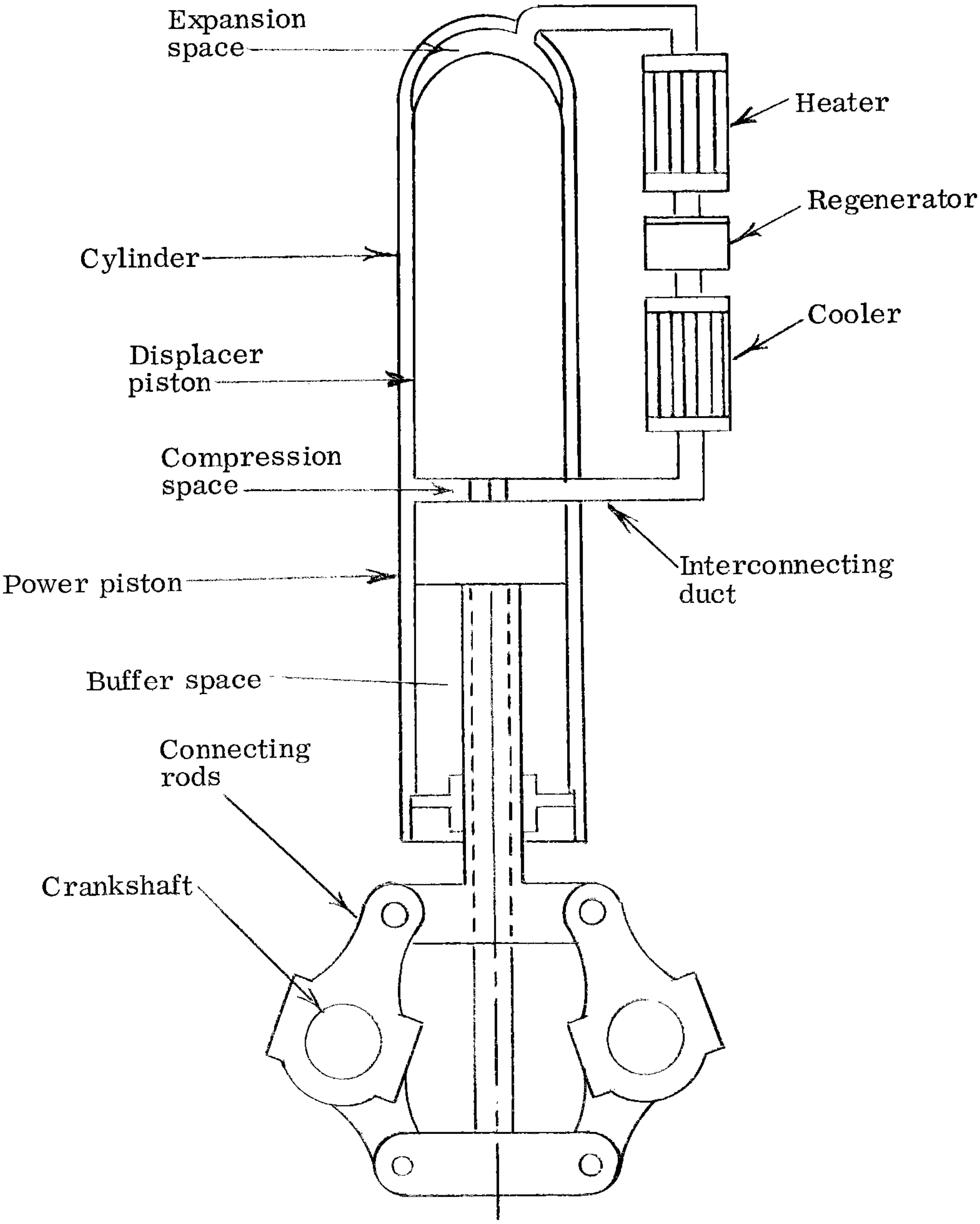
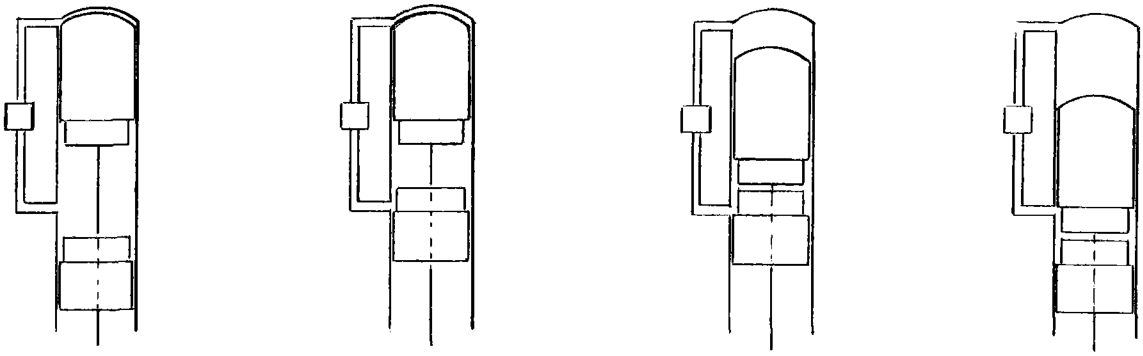


Figure 5-3. - Schematic of basic Stirling-cycle engine.



(a) Stage 1. Power piston in lowest position, displacer piston in highest position, all gas in cold space.

(b) Stage 2. Displacer piston still in highest position, power piston has compressed gas at low temperature.

(c) Stage 3. Power piston still in highest position, displacer piston has transferred gas from cold space to hot space.

(d) Stage 4. Hot gas has expanded, both power and displacer pistons in lowest positions, displacer piston will now return gas to cold space while power piston remains where it is to give stage 1 again.

Figure 5-4. - Four stages of Stirling cycle.

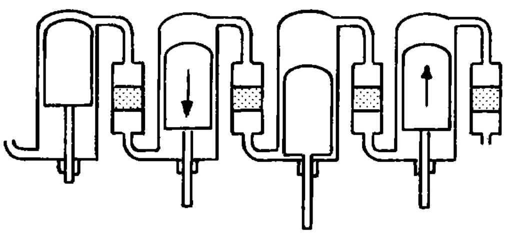


Figure 5-5. - Schematic of double-acting four-cylinder Stirling engine.

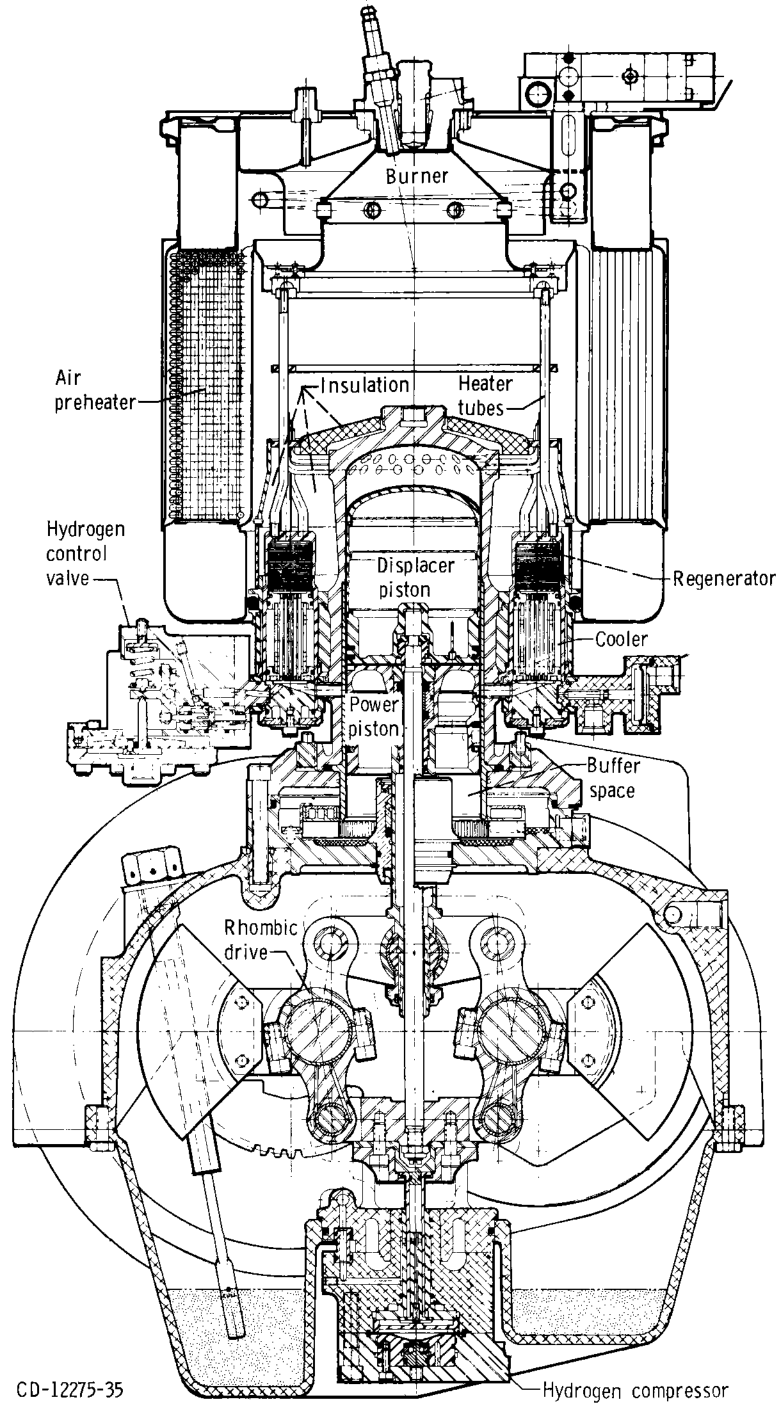


Figure 5-6. - Single-cylinder Stirling engine with rhombic drive.

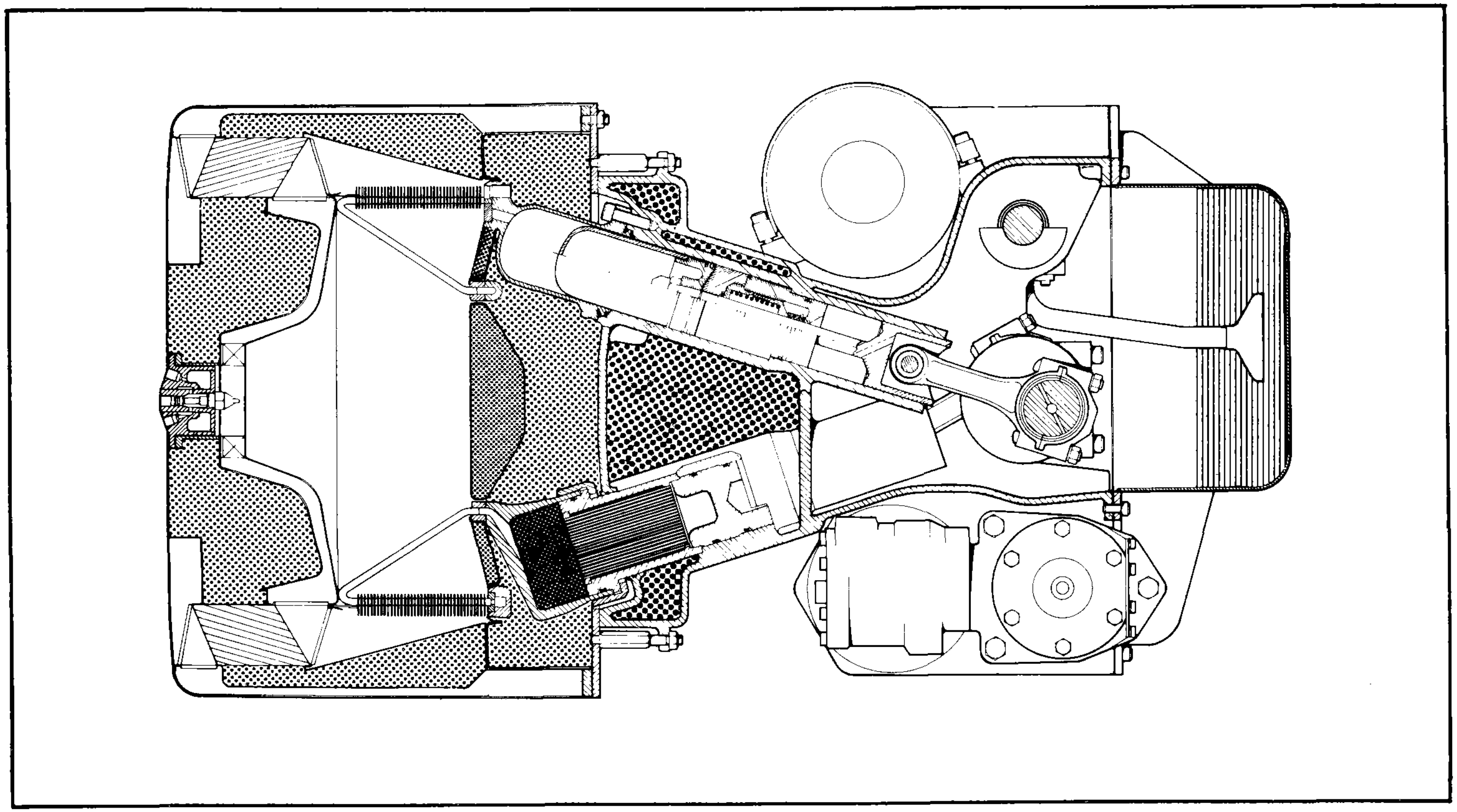


Figure 5-7. - Cross section of "V" Stirling engine. (Courtesy of United Stirling of Sweden.)

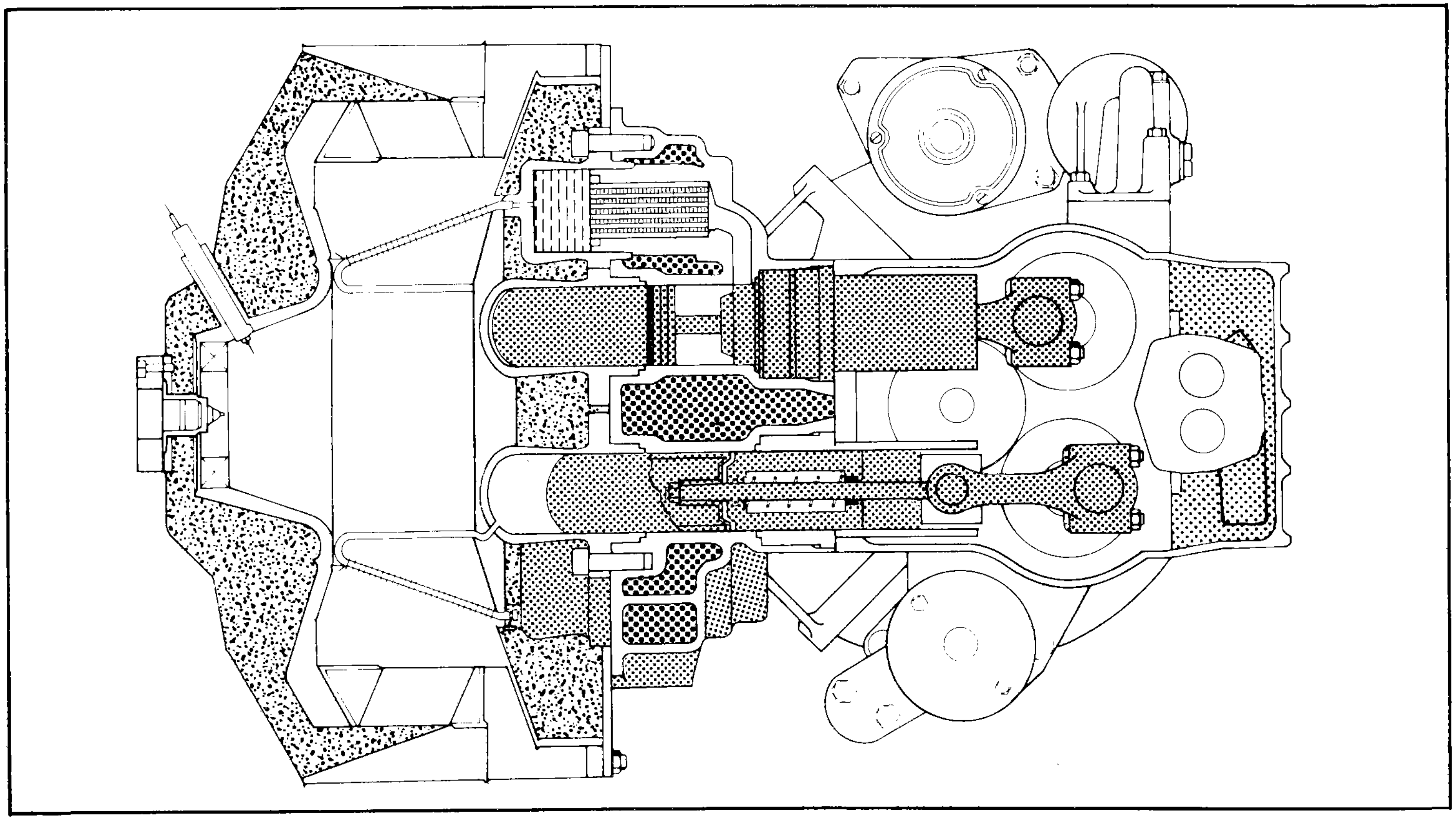


Figure 5-8. - Cross section of straight four-cylinder Stirling engine. (Courtesy of United Stirling of Sweden.)

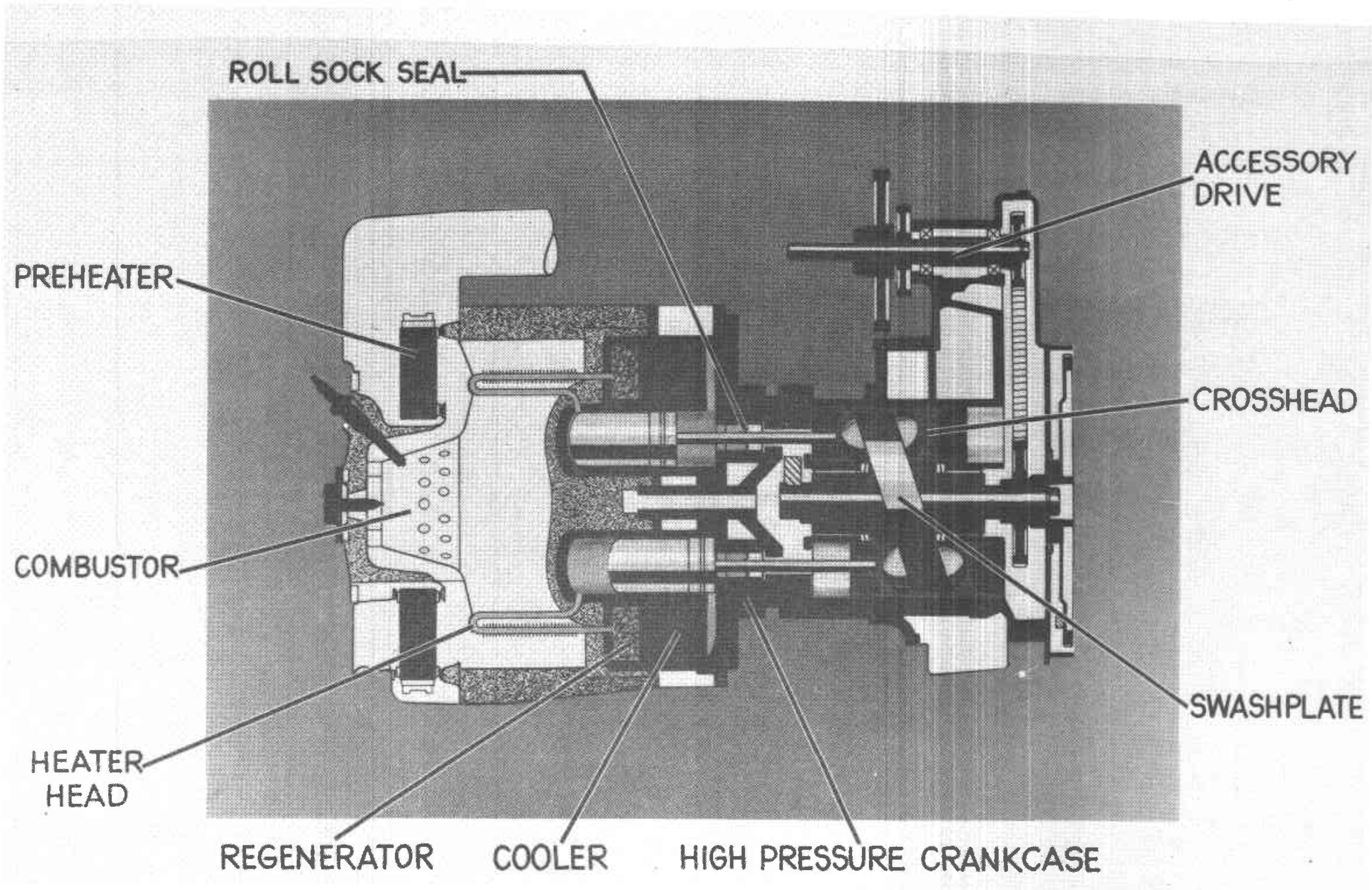


Figure 5-9. - Multicylinder Stirling engine with swashplate drive. (Courtesy of Ford Motor Co.)

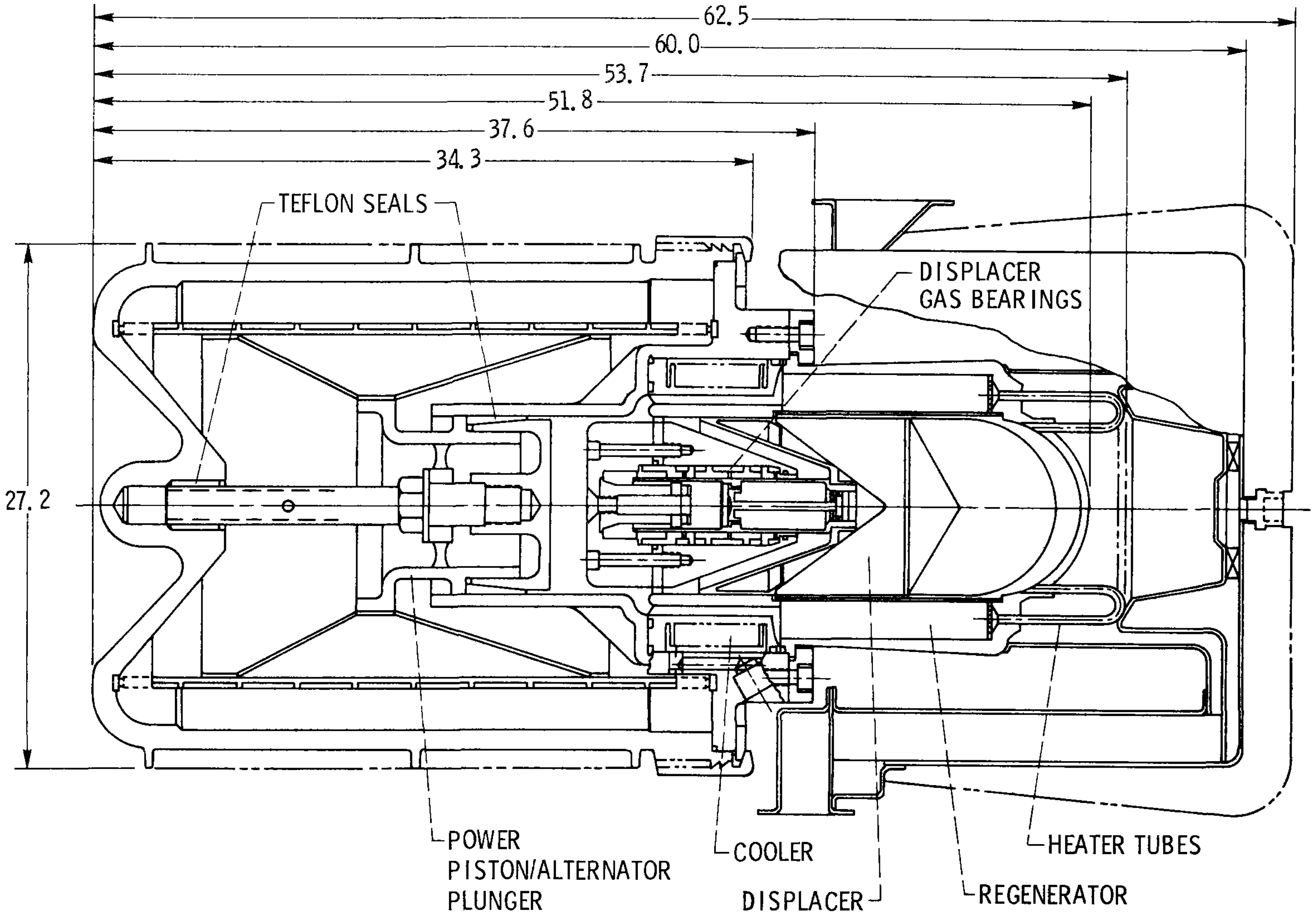


Figure 5.10. - Cross section of free-piston Stirling engine.

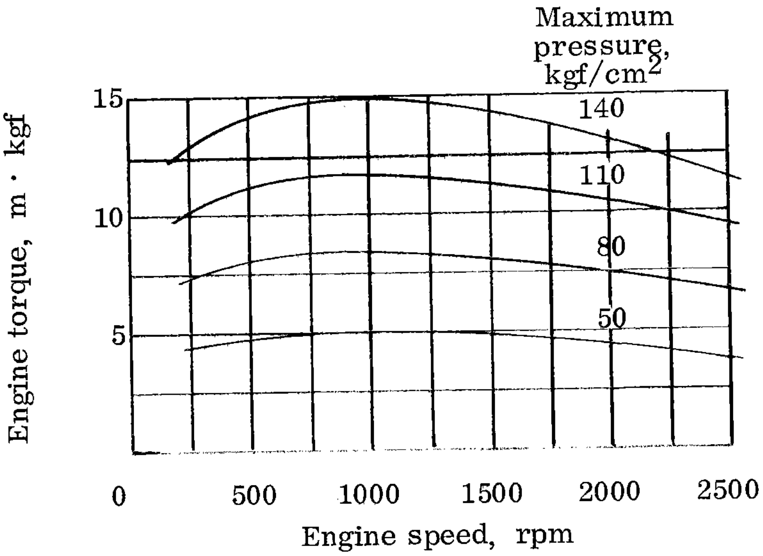


Figure 5-11. - Torque-speed characteristic of 40-horsepower Stirling engine for different maximum pressures.

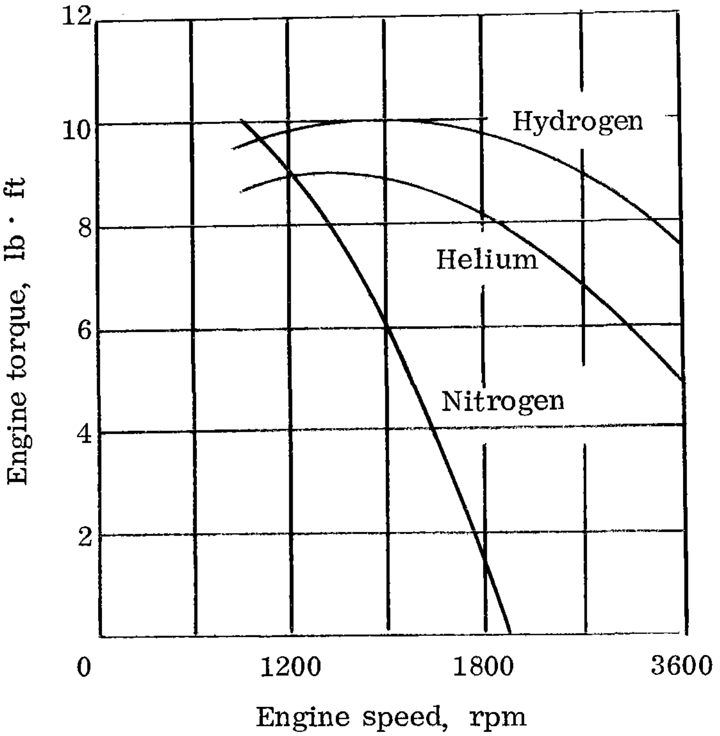


Figure 5-12. - Torque-speed characteristic of Stirling engine for different working fluids.

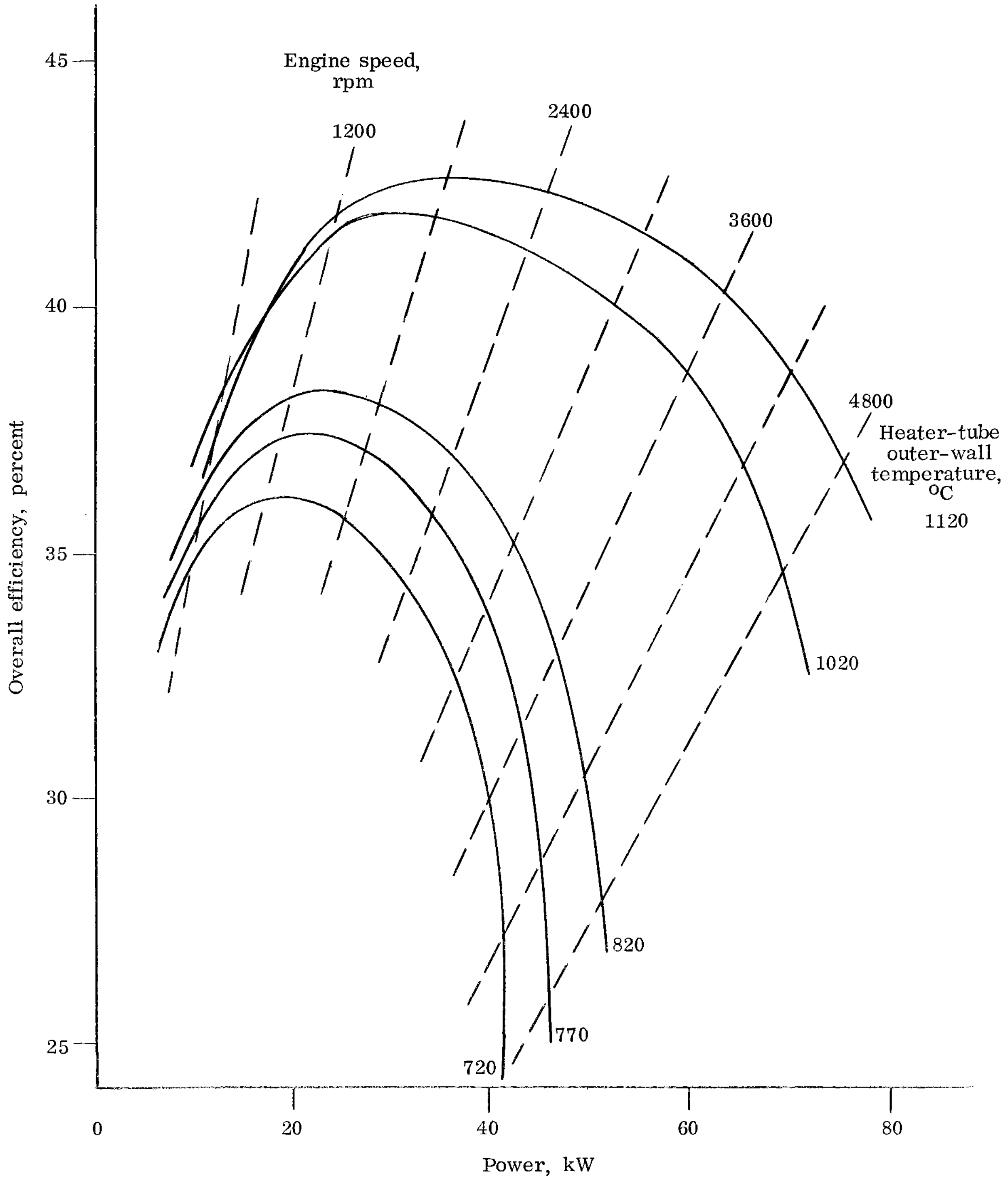


Figure 5-13. - Overall efficiency of United Stirling P-40 Stirling engine as function of load and heater temperature.

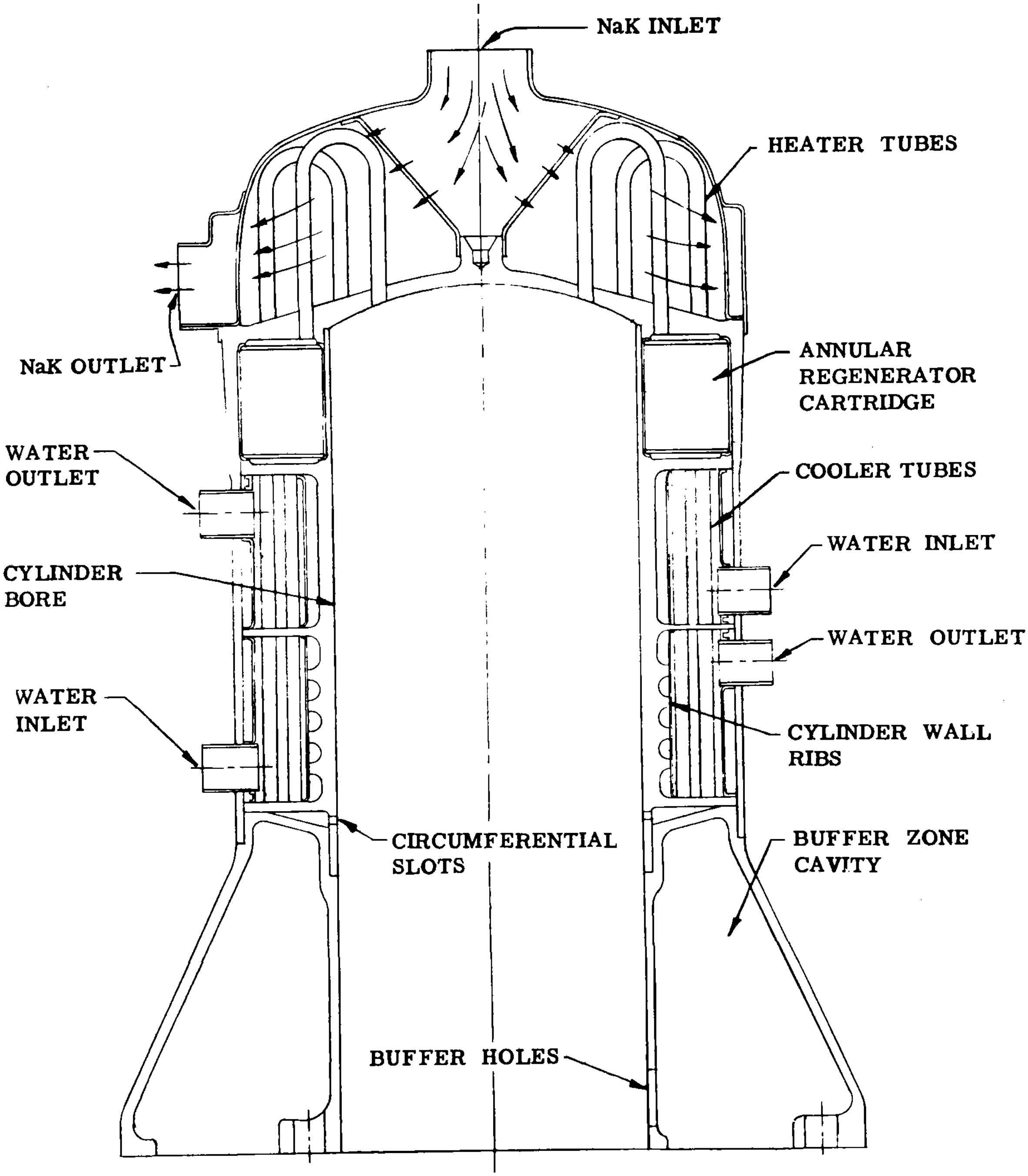


Figure 5-14. - Heater-head assembly for solar Stirling engine. (Courtesy of Detroit Diesel Allison, Division of General Motors Corp.)

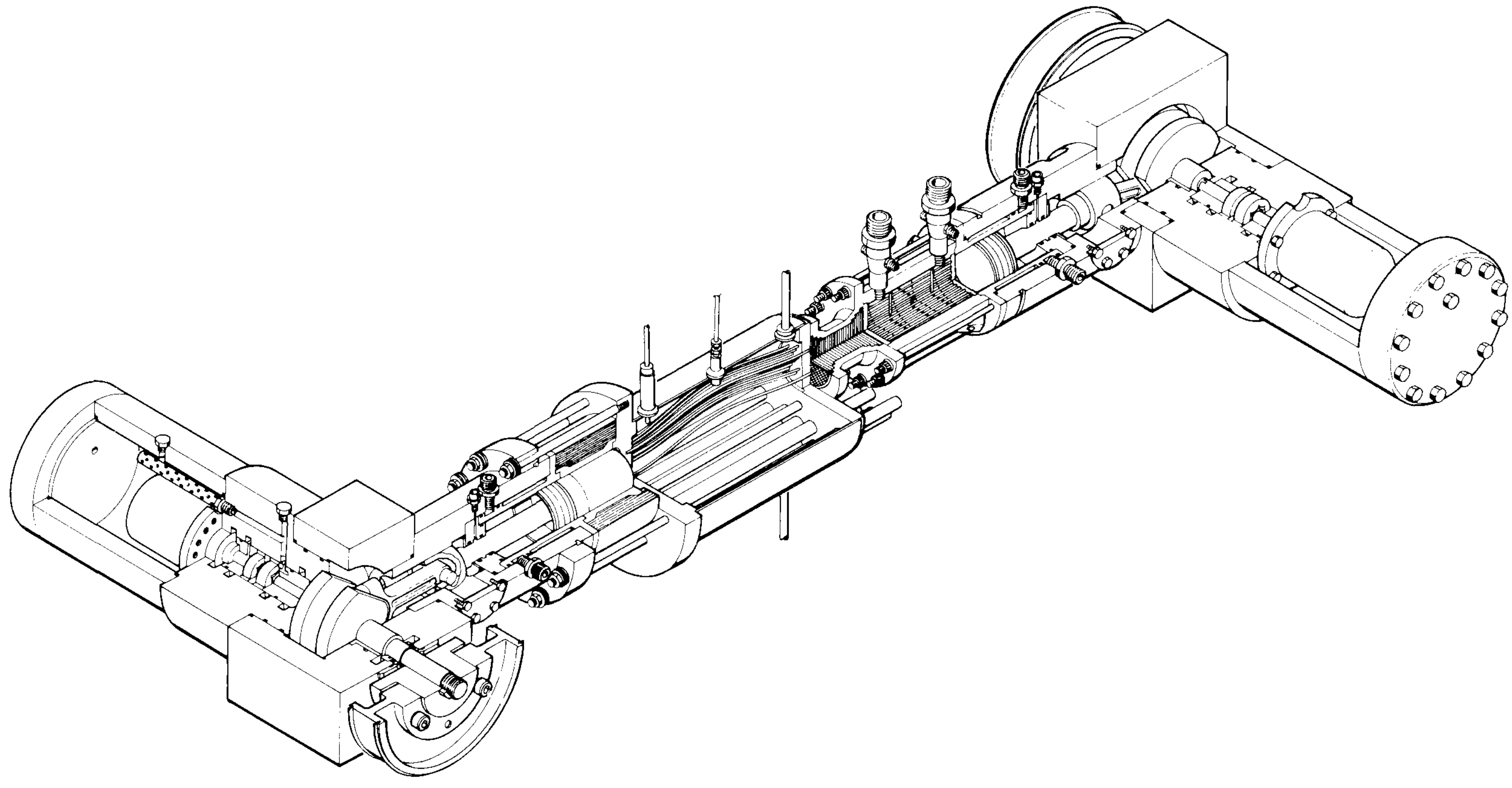


Figure 5-15. - Illustration of Jet Propulsion Laboratory Stirling research engine.

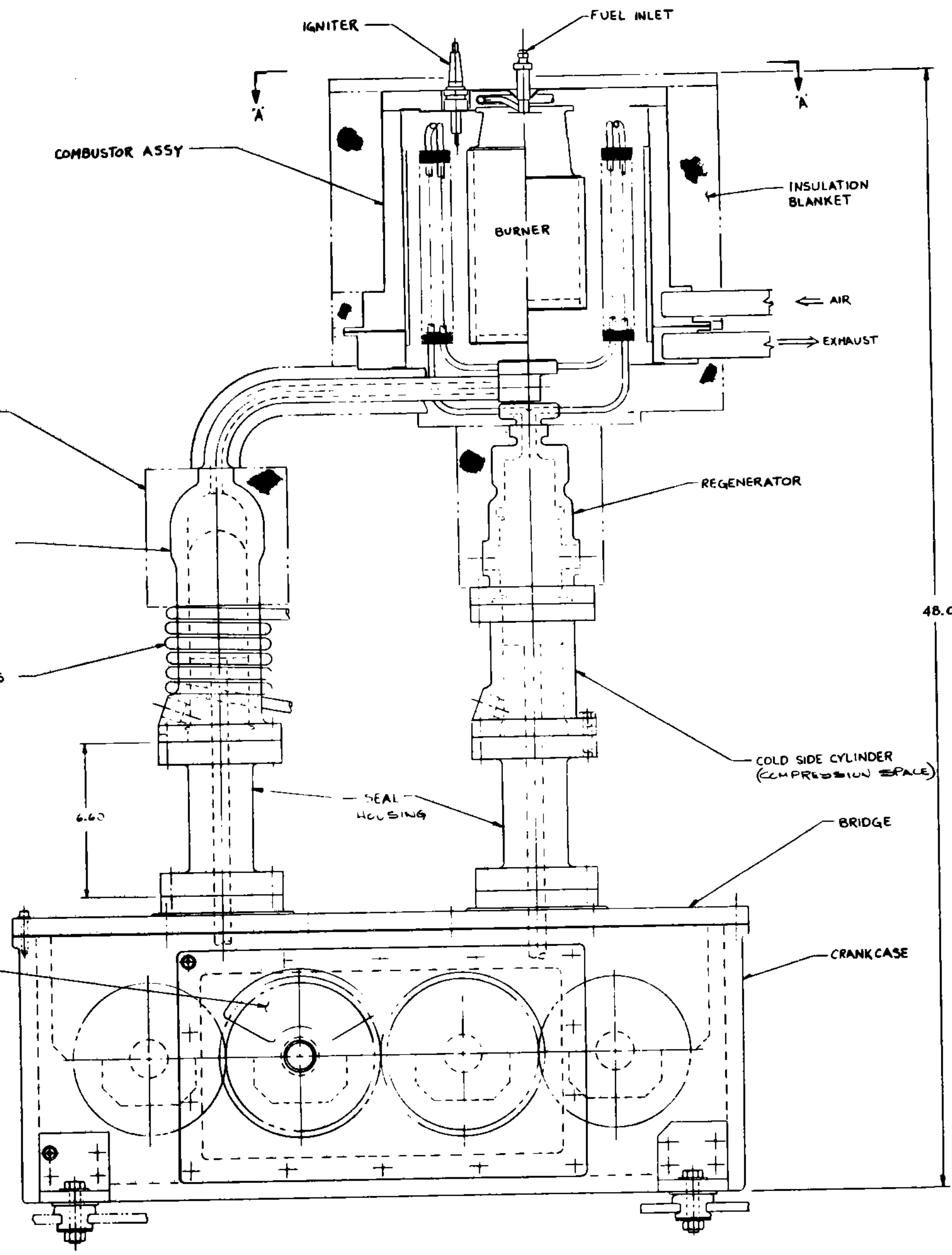


Figure 5-16. - NASA Lewis Research Center general-purpose Stirling test engine.

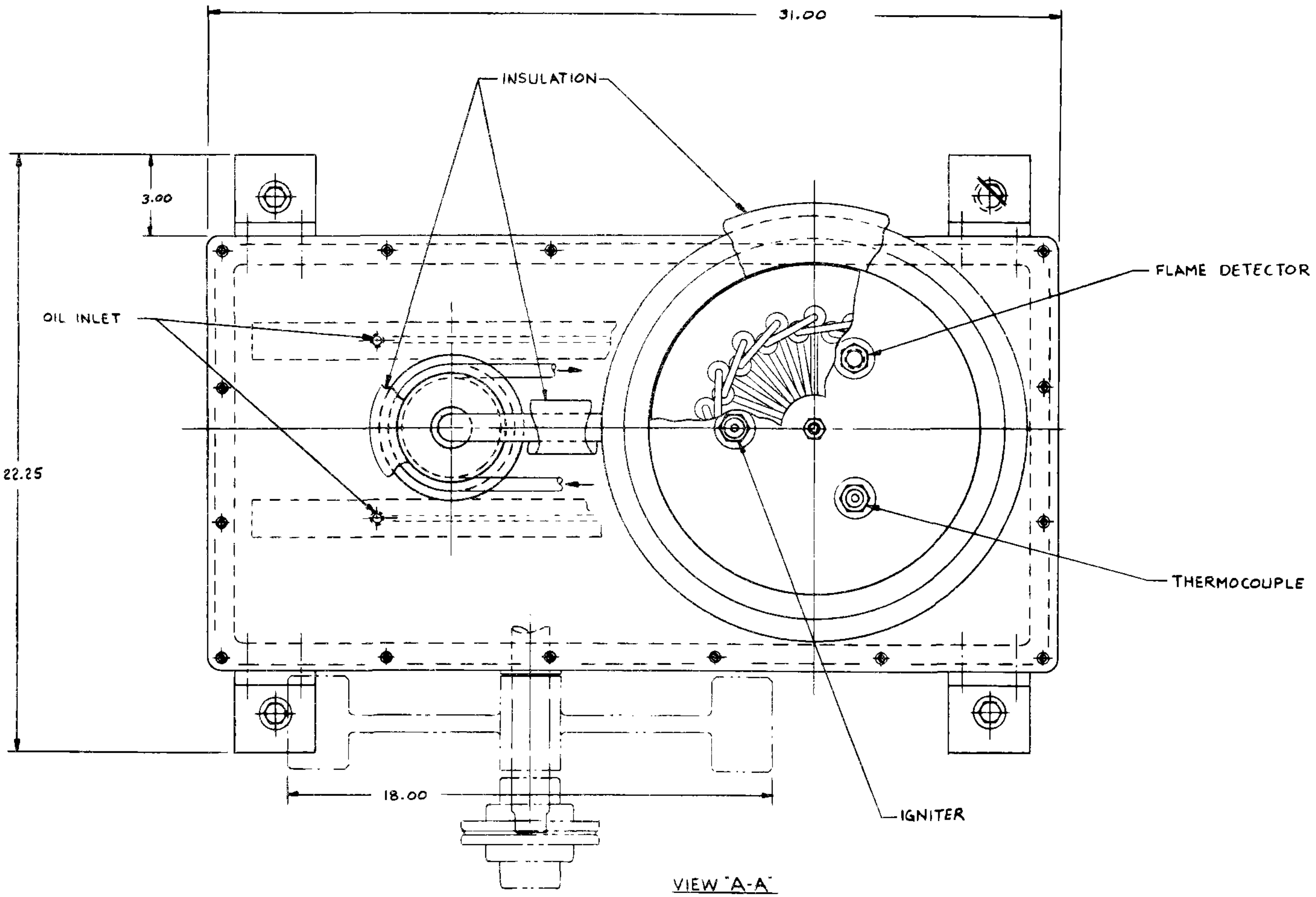


Figure 5-16. - Concluded.

6.0 SPEED REDUCERS-INCREASERS

by Thaddeus S. Mroz

6.1 DESCRIPTION

Prime movers such as steam-organic turbines, gas turbines, Stirling engines, and reciprocating engines are designed for peak efficiency at specific operating parameters. One parameter, output shaft speed, varies with the type and size of the prime mover and its specific design. The output shaft speed of steam-organic and gas turbines can vary from 3600 to 93 580 rpm. The output shaft speed of reciprocating engines can vary from 350 to 3600 rpm. Driven machines such as alternating-current generators and pumps operate at fixed speeds, depending on their design, from 720 to 3600 rpm.

Gear systems are used to reduce or increase the output speed of the prime mover to the speed requirement of the driven machine, at the operational load. In the prime movers previously mentioned, two types of speed reducer-increaser are used:

(1) A gear system integrated in the prime-mover housing

(2) A gear system in a separately mounted housing

A typical installation of each type of gear system is illustrated in figures 6-1 and 6-2, respectively. For prime movers, gear systems are generally of the following configurations:

(1) Horizontal parallel-shaft offset

(2) Horizontal parallel-shaft in-line

(3) Horizontal right angle

(4) Vertical right angle

(5) Epicyclic

These gear systems are illustrated in figure 6-3 (ref. 18). Only systems 1, 2, and 5 are discussed in this handbook.

In the horizontal parallel-shaft (offset or in-line) configuration (systems 1 and 2), the gear and shaft assemblies are mounted on fixed and parallel axes. Depending on the speed ratios, more than one set of gears operating in a train can be used. In this type of unit, the axes of the gears are fixed in one location. Horizontal parallel-shaft gear systems are used for moderate- and high-speed applications to loadings of 50 000 horsepower.

In the epicyclic configuration, various gear-axes rotate about one another to provide specialized output motions. The input-output shaft is coaxial for compactness. The gear system can be mounted directly on the prime mover or connected by a flexible coupling or torsion shaft. The epicyclic configuration is used for loadings of 150 to 50 000 horsepower. Bearings for the high-speed shaft, which are responsible for considerable losses in conventional designs, are smaller or are eliminated entirely.

The gear system is the most durable and rugged of all mechanical drives and transmits power to 99-percent efficiencies for long service lives. Gear systems are an assembly of a housing, gears, a bearing system, a lubrication system, oil seals, and shafts. A separate lubrication system and seals are not required for a gear system installed integrally in a prime-mover housing.

The housing of a separately mounted speed reducer-increaser is a heavy casting machined for mounting of the bearings, the gear-shaft system, the seals, and the lubrication system components. The design and machining provide precise alinement of the gear-shaft assemblies, bearings, and seals. The housing is reinforced to provide the rigidity for proper alinement. Generally, the housing material is a high-tensile cast iron alloy suitable for outdoor installation. Shafts are precision machined from heat-treated, high-quality alloy steel and are designed with an overload factor sufficient to rigidly maintain the required gear alinement and to keep stresses within allowable limits. Selection of the bearing system depends on the output level (horsepower-torque ratio) and type of service. Bearing systems can be double-row ball bearings; cylindrical-bore, spherical or tapered roller bearings; or precision journal bearings with thrust faces for axial loads. Generally, the bearings are sized larger than the industry standards. Oil seals are used on the input and output shafts. Grease seals are available for unusually abrasive or dusty applications.

The lubrication method depends on the design, loading, horsepower rating, and type of bearings used. In the splash type of oil lubrication method, the housing incorporates an oil reservoir (oil bath). One of the gears dips into the oil bath. As the gear rotates, it transfers the lubricant to the contacting teeth. Excess lubricant is thrown against the housing, where it is directed to the bearings and sump. Pressure or force-fed oil lubrication where the lubricant is pumped under pressure to the gear train, is required for vertical applications and for systems using split-sleeve bearings. In this type of lubrication, after the oil passes over the lubricated surfaces, it is returned to the reservoir, filtered, and repumped to the system. The oil is delivered as a stream running over the gearing or as a spray from nozzles.

Generally, speed reducers rated under 5000 horsepower can be supplied with an integral lubrication system of sufficient capacity to provide lubrication to the prime mover and/or the generator. A remote lubrication and cooling system is used for large speed reducers. In all cases the lubrication system is sized to provide adequate cooling.

The selection of gearing for speed reducers-increasers is based on the application and load requirements. The specific types of gear considered in this section are

(1) Spur gear

(2) Single helical gear

(3) Double helical gear

(4) Herringbone gear

These four gear types are illustrated in figure 6-4. The larger gear in a pair is called the gear and the smaller gear is called the pinion.

The spur gear is the most common type of gear and is used to transmit power between parallel shafts. The teeth are straight and parallel to the axis of rotation. Spur gears do not transmit end-thrust, are economical to manufacture, and can be used in moderate- and high-speed applications, but they are limited in loading. High-speed spur gears require precise manufacturing steps including care in tooth shaving, tooth grinding, and heat treatment.

The helical gear has a greater load-carrying capacity than comparable-size spur gears because of the larger contact surface area and the greater thickness in the direction of the load. The teeth are not parallel to the axis of rotation, as in the spur gear, but are cut at an angle known as the "helix angle." Teeth in mating gears have identical helix angles but have opposite hands of cut. A right-hand pinion meshes with a left-hand gear. In addition to the normal radial loads, the single helical gear transmits an end thrust along the axis of rotation. The end thrust is a function of the helix angle. Mounting assemblies and bearings are designed for this thrust load.

Double helical gears have two sets of opposed helical teeth. A groove is machined between the helices. Each set of teeth has the same helix angle, but the helices have opposite hands of cut. Double helical gears do not generate thrust loads. They can be used for moderate- and high-speed applications. Continuous-tooth herringbone gears are double helical gears cut without a groove between the two rows of teeth. This type of gear is used to transmit heavy loads at moderate speeds where continuous service is required, where shock and vibration are present, or where a high reduction ratio is necessary. The double helix and herringbone gears are suited for long service life under heavy loads.

There are two types of rating for speed reducers-increasers: a mechanical rating and a thermal rating. A mechanical rating is assigned on the basis of the durability, strength, and load-carrying capacity of the gears, shafts, and bolts. A thermal rating specifies the power that can be transmitted continuously without exceeding a specified rise in operating temperature above ambient. This rating varies with speed and ratio. Most American Gear Manufacturers Association (AGMA) ratings are based on a 100° F temperature rise above ambient with a maximum operating temperature of 200° F. At high ambient temperatures, a speed reducer-increaser's true thermal capacity is lower than that indicated by catalog rating. Cooling may be required.

Selecting a speed reducer-increaser for a specific application requires consideration of a number of factors:

(1) Driver- and driven-shaft arrangement
(2) Speed ratio and speed limitation
(3) Lubrication system
(4) Type of prime mover
(5) Required efficiency
(6) Space and weight limitations
(7) Required working room
(8) Quietness
(9) Required life
(10) Operating load and type of application
(11) Service factor

The gearing must have a service horsepower rating equal to the maximum continuous horsepower capacity of the prime mover multiplied by any overload capacity of the prime mover. The service and duty requirement must be defined to establish a load classification or numerical service factor for the application. Three load classifications are recognized: uniform load, moderate shock, and heavy shock. Numerical values based on field experience have been assigned to these classifications for intermittent service and for 10- and 24-hour service per day. These service factors depend on the type of prime mover. The load classifications and the resulting service factor for a specific application will vary for different types of gearing.

6.2 DEVELOPMENT STATUS

Geared speed reducers-increasers are fully developed for use in the steam turbine, the gas turbine, and the reciprocating engine. A sound technology base in gears, bearings, lubrication systems, and metallurgy exists for their use in industrial and special applications, including design and manufacturing techniques. Speed reducers-increasers in the 5- to 50 000-kilowatt range are fully developed production units.

6.3 AVAILABILITY

Commercial speed reducers-increasers are available in the 5- to 50 000-kilowatt range. Typical delivery schedules are shown in figure 6-5. These schedules are based on external, separately mounted gear systems and do not include prime-mover integral gear systems.

6.4 COST

The cost of a speed reducer-increaser is based on the output size and the application requirements. Figure 6-6 shows cost as a function of output power level. The costs shown do not include prime-mover integral gear systems or a remote lubrication system for the large speed reducers. Weight is shown in figure 6-7.

6.5 OPERATING CHARACTERISTICS

Geared systems can be driven at constant or varying speeds. Although the unit is sized for maximum power output, operation at lower speeds requires careful selection of the lubrication system and evaluation of waste heat dissipation and dynamic characteristics. Figure 6-8 shows efficiency as a function of output power level. A geared unit is rated approximately as a constant-torque machine. The horsepower rating varies almost directly with input speed.

6.6 OPERATING CONSTRAINTS

The operating constraints of a speed reducer are the mechanical and thermal rating assigned for an application based on the load, speed, service factor, lubrication, and ambient conditions.

6.7 EXPERIENCE FACTORS

Speed reducers-increasers are production equipment with established and documented operational and maintenance histories. The design life and reliability of this equipment, with scheduled maintenance, match those of the prime mover.

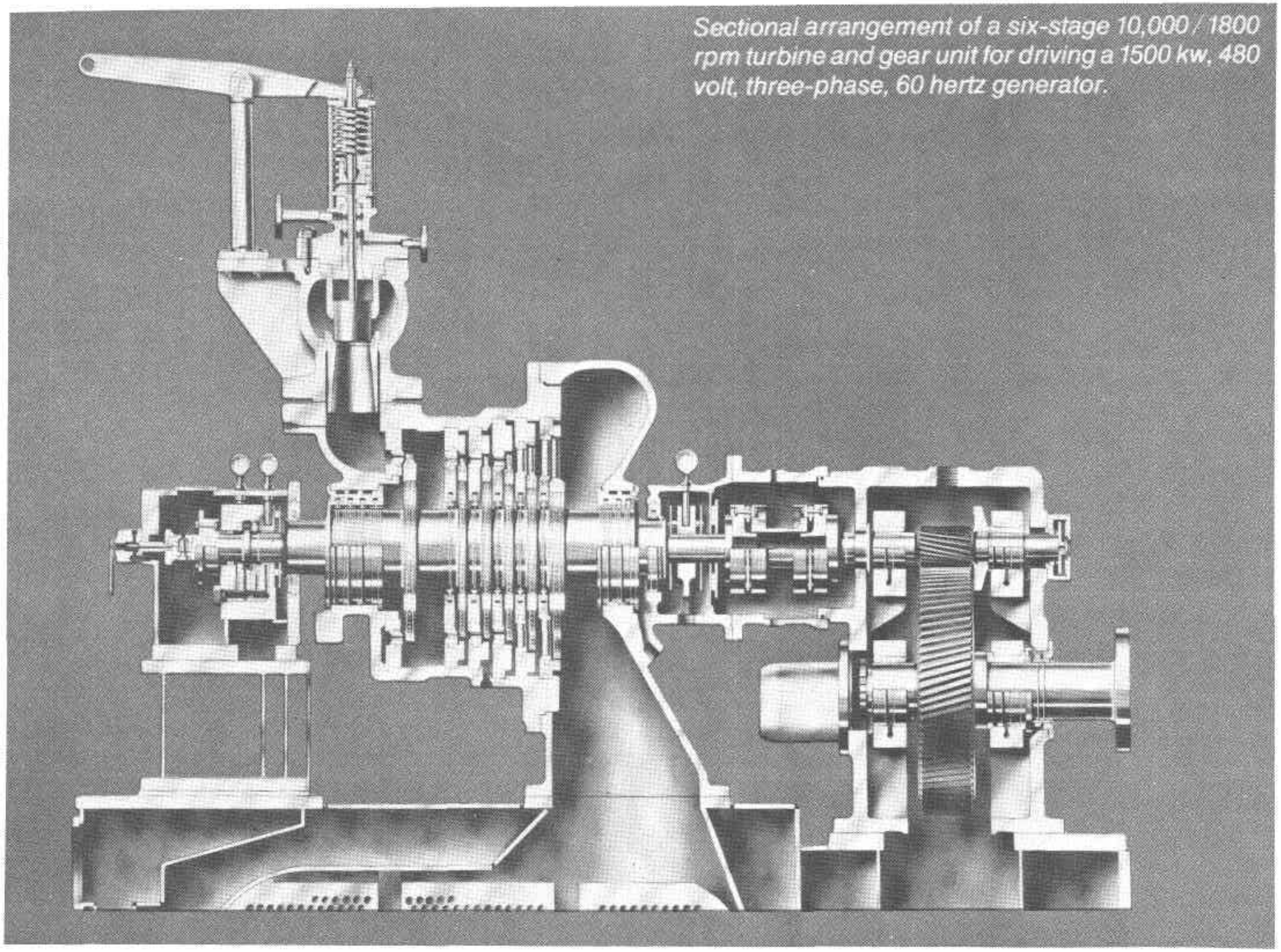


Figure 6-1. - Multistage steam turbine with integral speed reducer-increaser. (Courtesy of Thermo Electron Corp.)

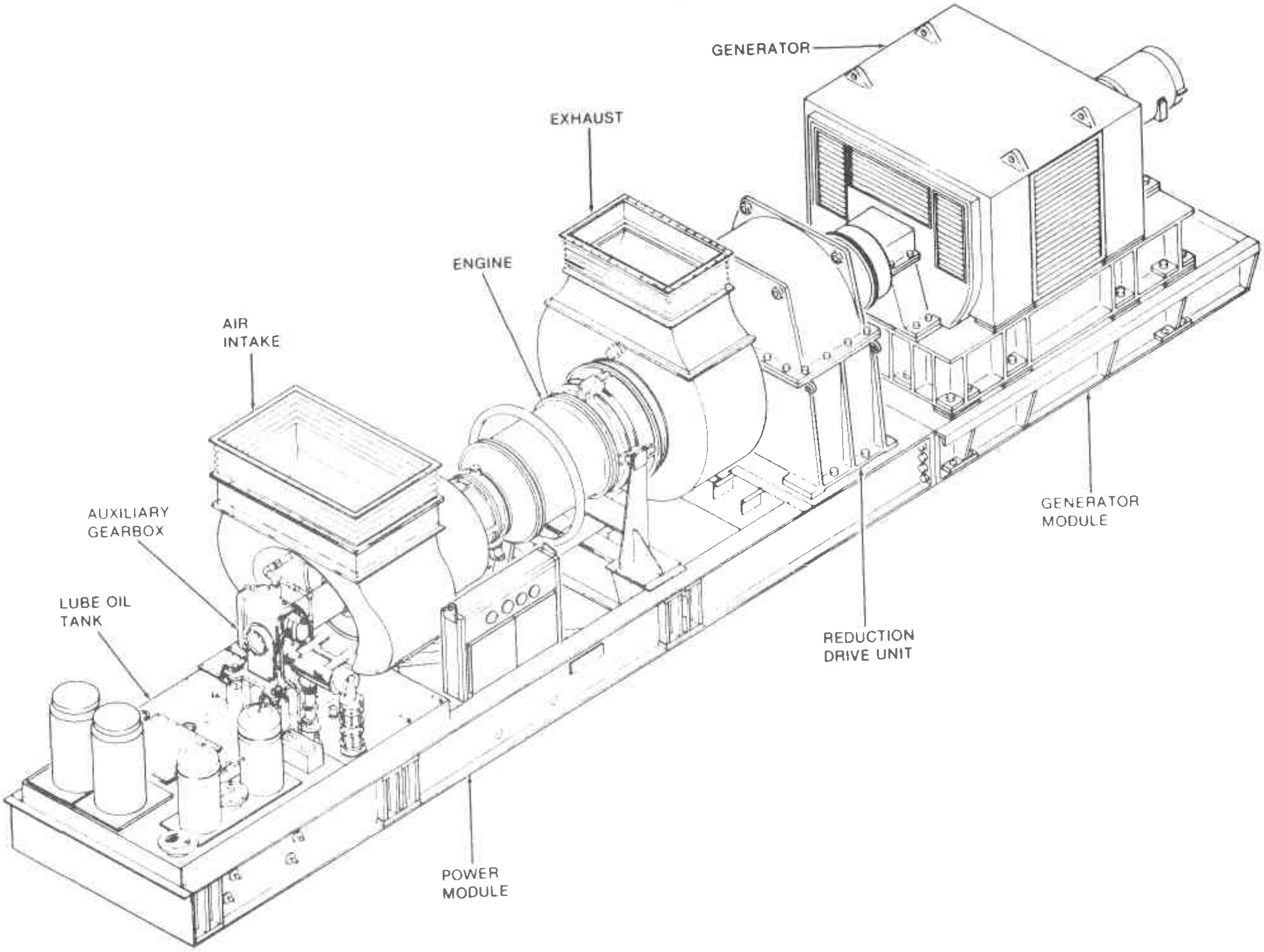
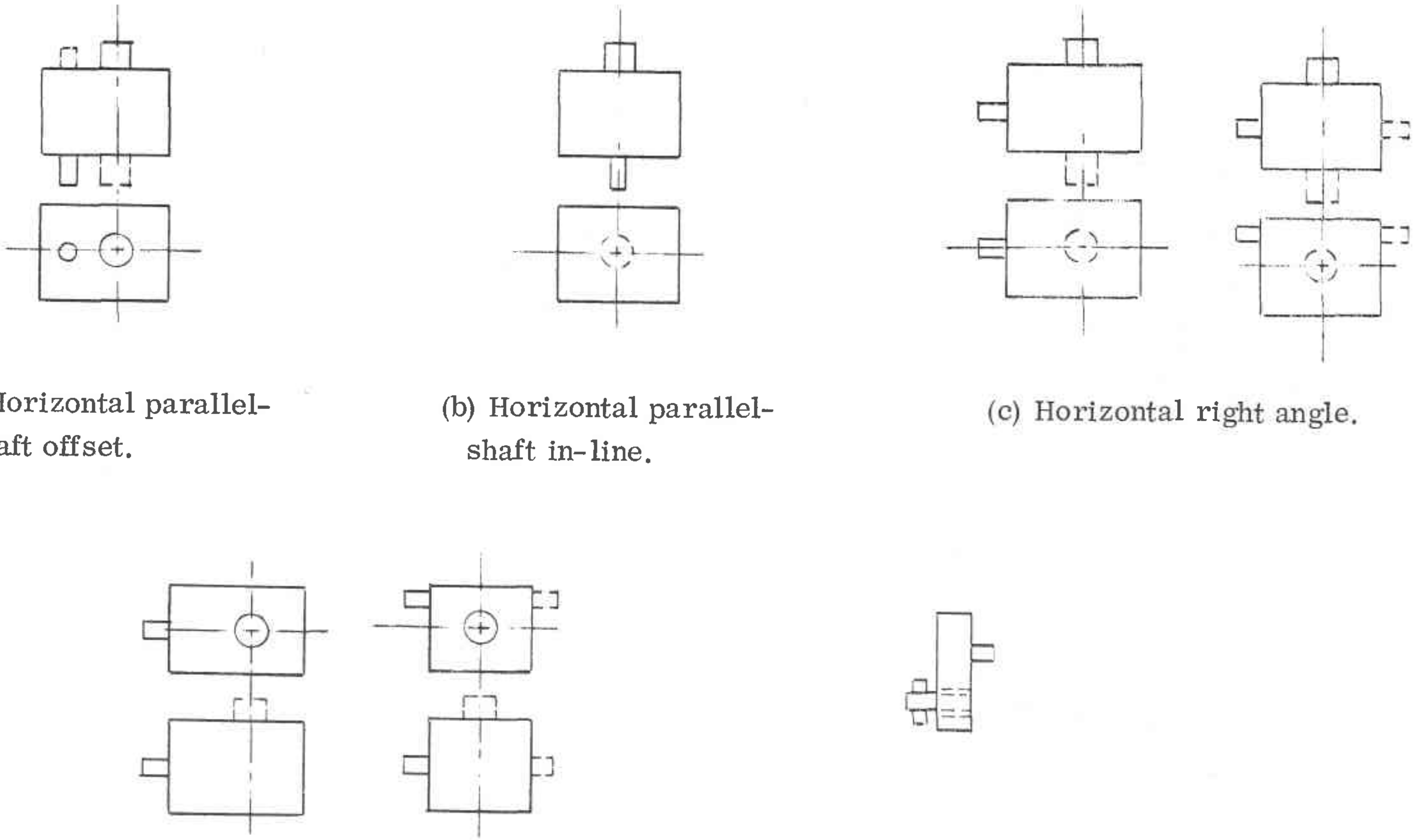


Figure 6-2. - Industrial gas turbine engine with separately mounted speed reducer. (Courtesy of Solar Turbine International.)



(a) Horizontal parallel-shaft offset.

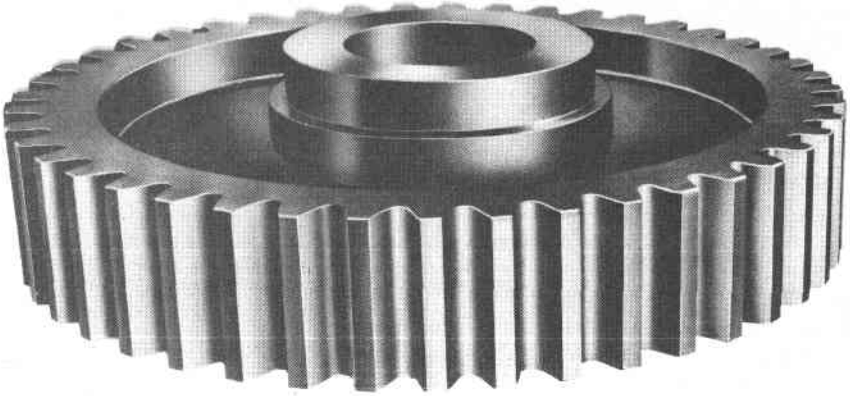
(b) Horizontal parallel-shaft in-line.

(c) Horizontal right angle.

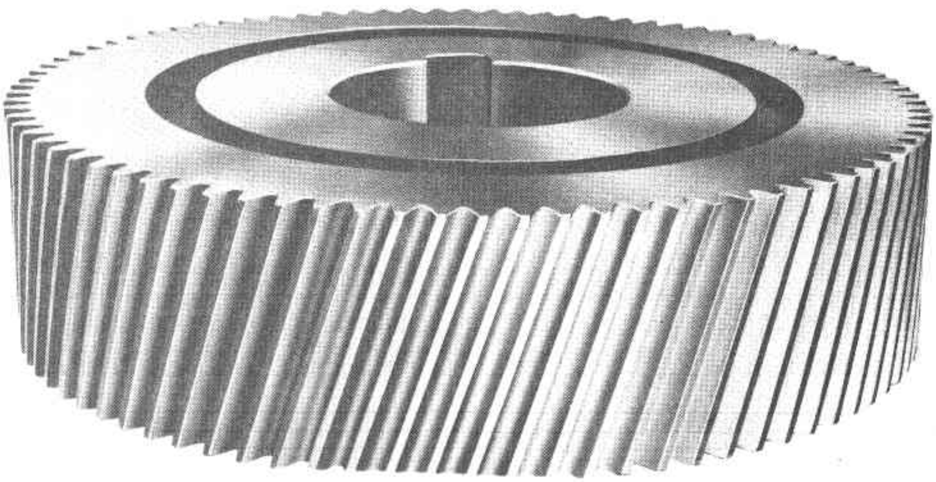
(d) Vertical right angle.

(e) Epicyclic.

Figure 6-3. - Speed reducer-increaser configurations.

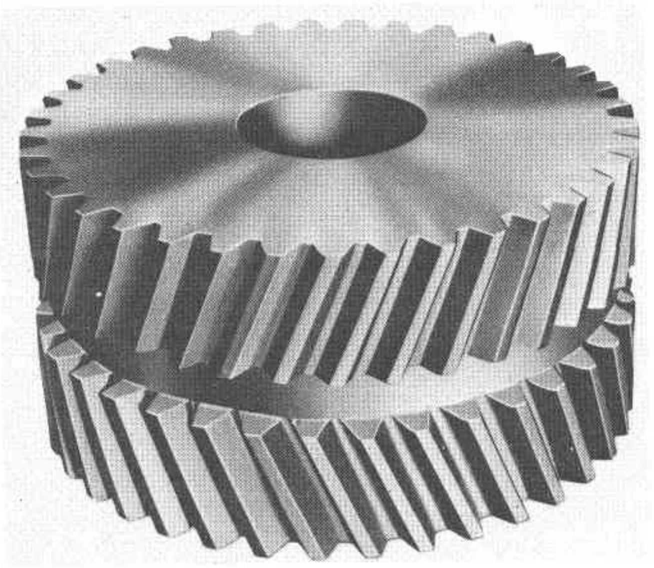


(a) Spur gear. (Courtesy of Philadelphia Gear Corp.)

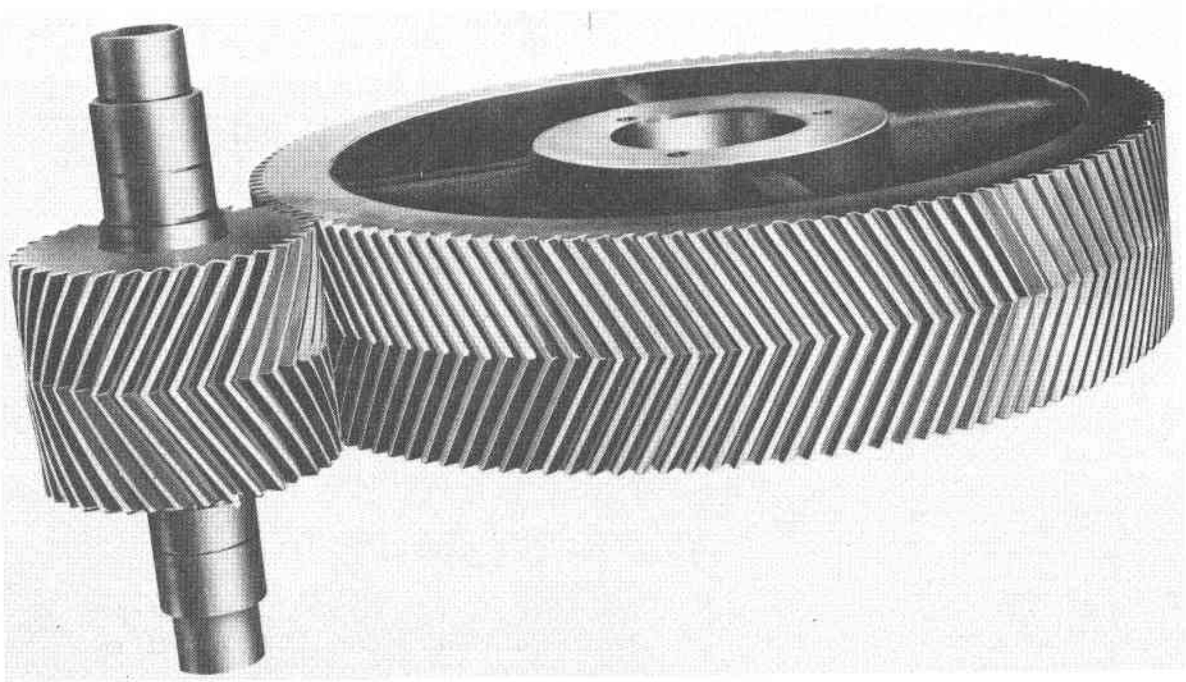


(b) Single helical gear. (Courtesy of Philadelphia Gear Corp.)

Figure 6-4. - Gear types.

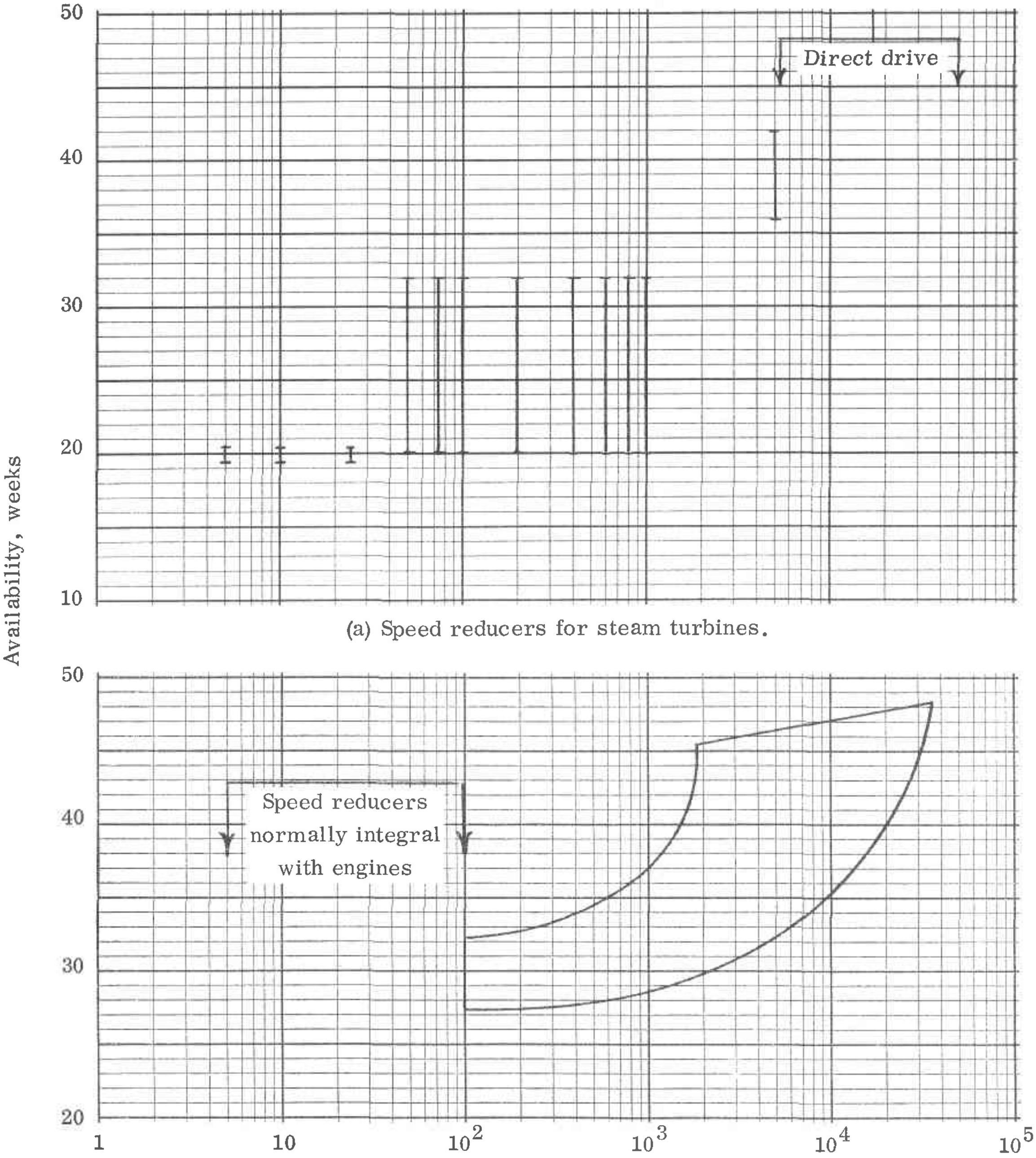


(c) Double helical gear. (Courtesy of Philadelphia Gear Corp.)



(d) Continuous herringbone gear. (Courtesy of Philadelphia Gear Corp.)

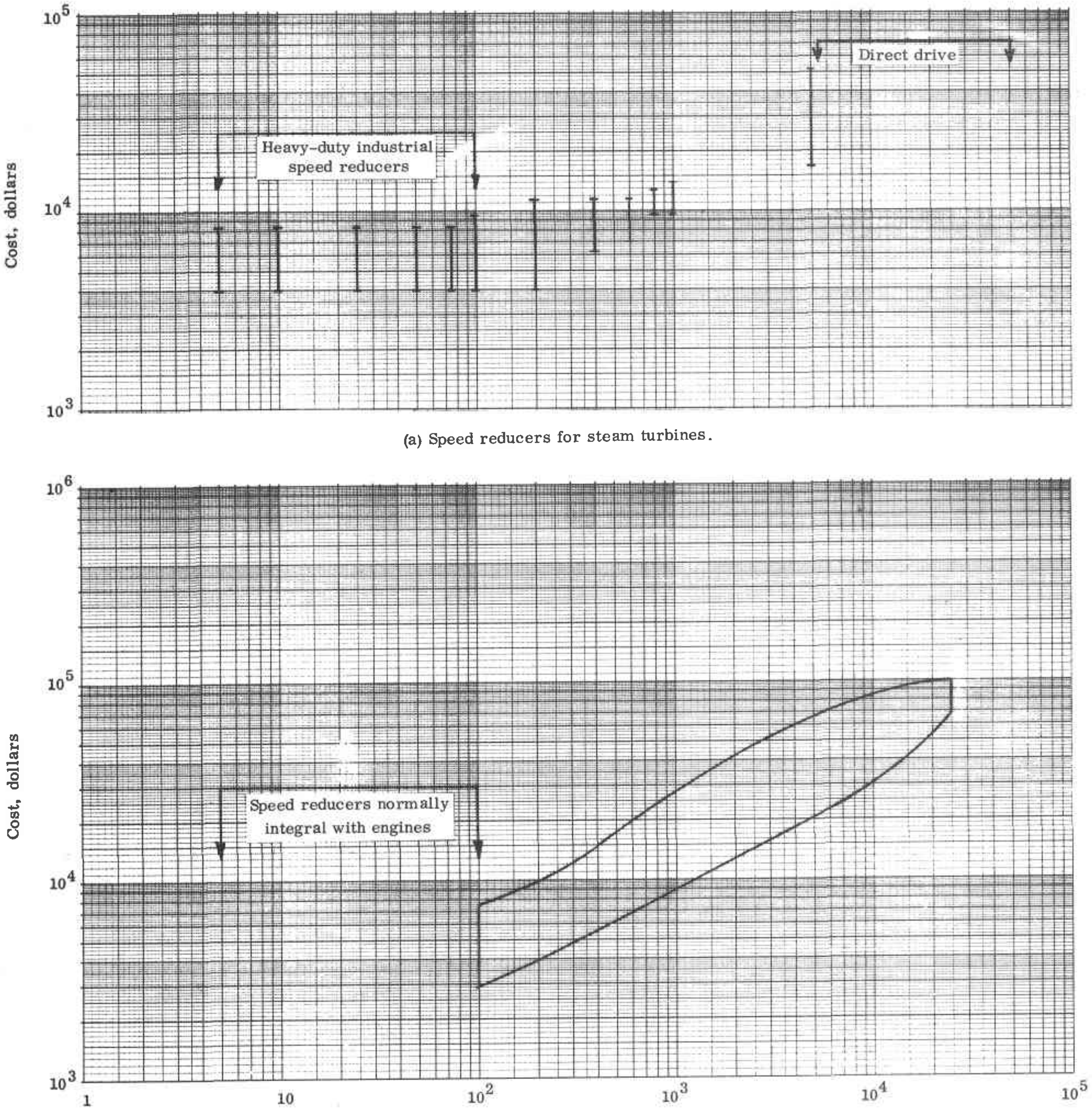
Figure 6-4. - Concluded.



(a) Speed reducers for steam turbines.

(b) Speed reducers for gas turbine engines. (Data are for externally mounted speed reducers for engines that require speed reduction but that do not incorporate an integral speed reducer.)

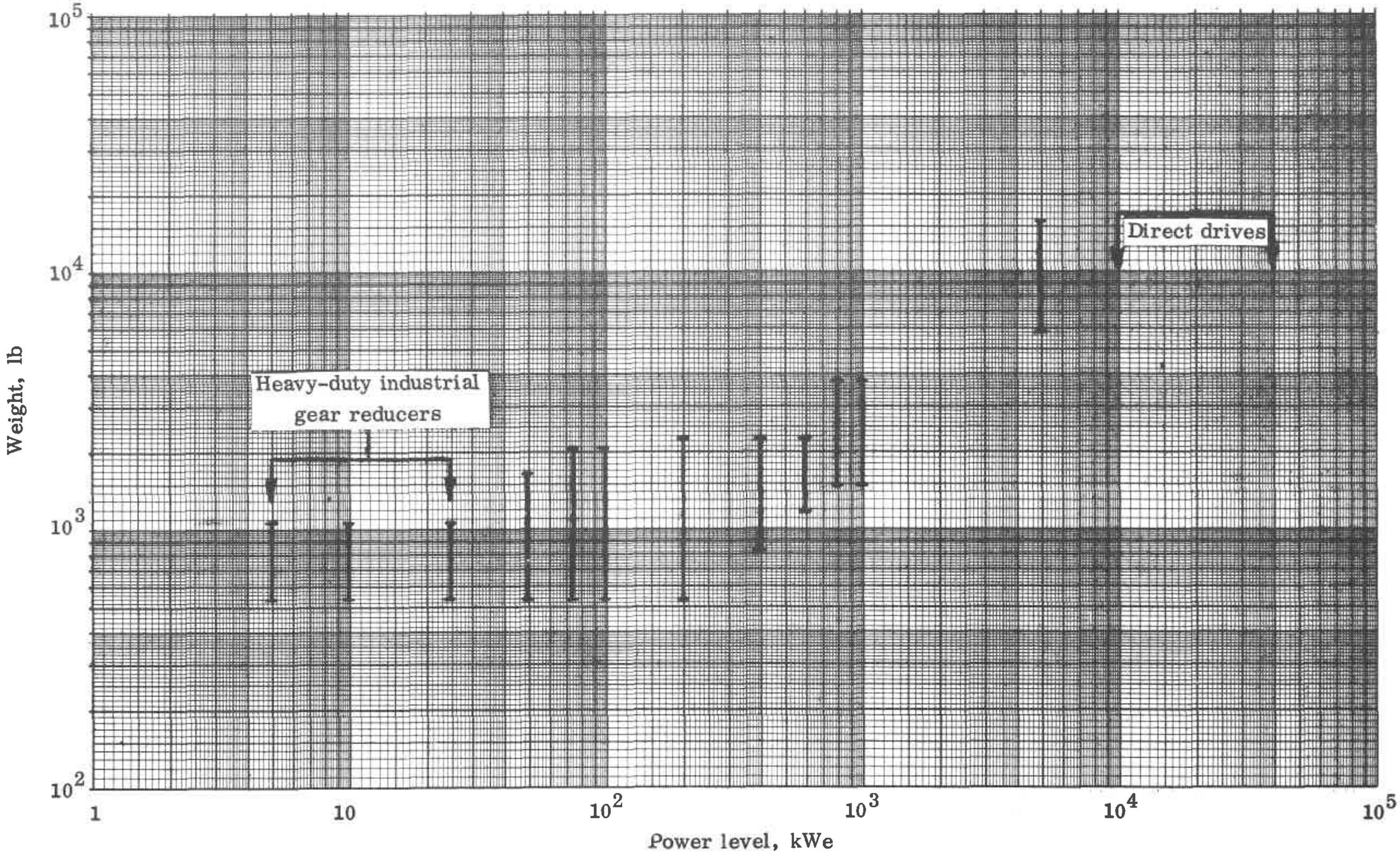
Figure 6-5. - Availability of speed reducers as function of power level.



(a) Speed reducers for steam turbines.

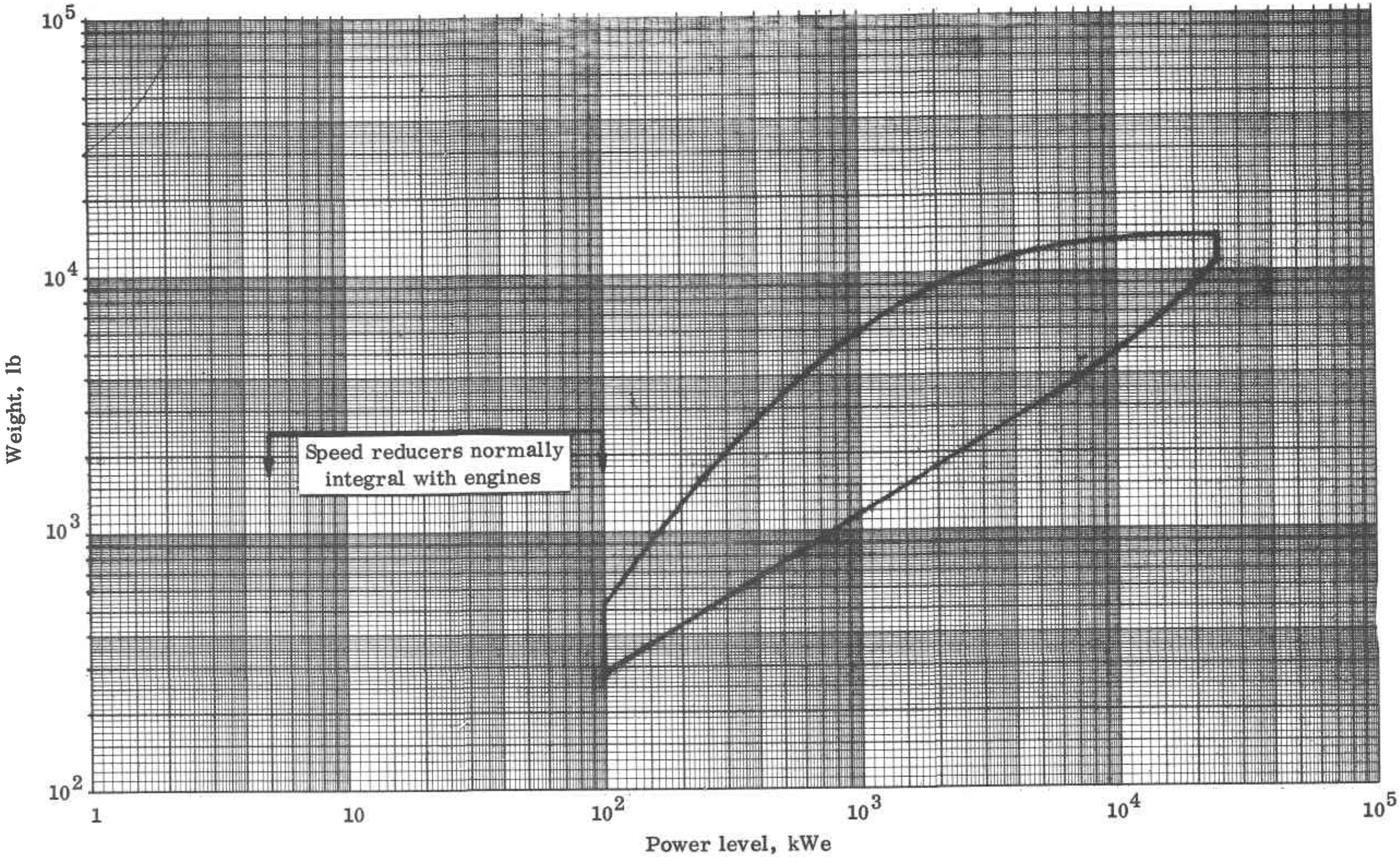
(b) Speed reducers for gas turbine engines. (Data are for externally mounted speed reducers for engines that require speed reduction but do not incorporate an integral speed reducer. Cost is contingent on speed ratio and power level.)

Figure 6-6. - Cost of speed reducers as function of power level.



(a) Speed reducers for steam turbines.

Figure 6-7. - Weight of speed reducers as function of power level.



(b) Speed reducers for gas turbine engines. (Data are for externally mounted speed reducers for engines that require speed reduction but do not incorporate an integral speed reducer.)

Figure 6-7. - Concluded.

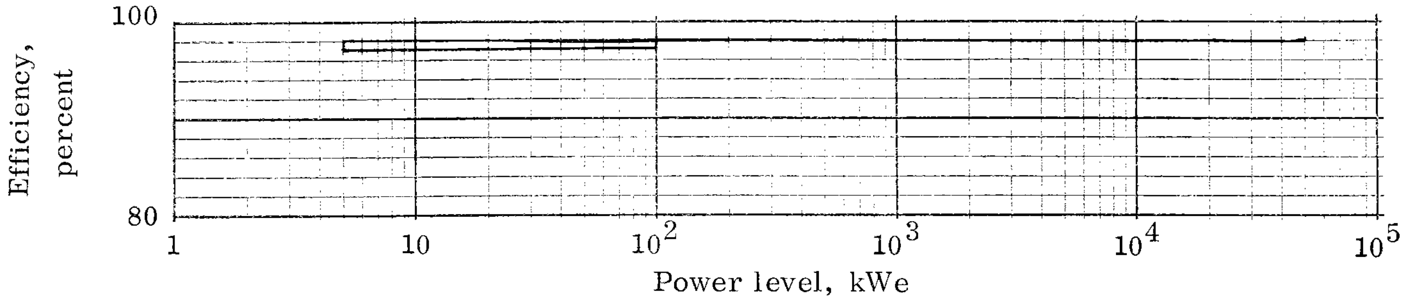


Figure 6-8. - Efficiency of speed reducers as function of power level.

7.0 COMMERCIAL SYNCHRONOUS ALTERNATING-CURRENT GENERATORS

by James H. Dunn

7.1 DESCRIPTION

The alternating-current (ac) generator is a driven, rotating machine that converts mechanical shaft power into electric power. Depending on the application and size, the generator can be directly or indirectly coupled to the prime-mover shaft. Generators can be driven by electric motors, reciprocating engines, and Brayton- or Rankine-cycle turbine engines. This handbook deals only with synchronous, direct-coupled ac generators.

The technology of generating electric power has advanced by many orders of magnitude to the sophisticated designs presently being marketed for low-power applications (1 to 2 kWe) to utility applications ($>10^6$ kWe). However, the basic principle of generating electric power is still the same.

The ac generator in its simplest form is a single turn of wire rotating in a magnetic field (fig. 7-1(a)). As the wire cuts through the magnetic flux, the voltage is generated in the direction indicated in the figure by the arrow. After the coil has rotated through 180° from the initial position, it is cutting down through the flux and the direction of the voltage is reversed. A complete revolution of the coil through 360° produces a sine wave of voltage if the field is constant. The same results are obtained if the coil is stationary and the magnetic field rotates.

Modern ac generators use a rotating electromagnetic field (ref. 19). The strength of the electromagnetic field is regulated to provide constant output voltage. Elements of a typical ac generator are shown in simplified form in figure 7-2. This generator has a single group of stator coils for each of three rotating poles (electromagnetic fields). Field excitation current is supplied by a separate generator called the exciter. Two types of exciter are available: a brushless exciter, and a direct-current dc generator. The brushless exciter is a small ac generator with its armature mounted on the main generator shaft. Alternating current produced by this exciter is rectified by silicon diodes on the rotor to direct current, which is provided to the armature field poles for field excitation. This design eliminates the need for brushes, a commutator, and slip rings. The dc generator incorporates a commutator and brushes. Direct current is then supplied to the ac generator field through slip rings. The exciter acts as an amplifier; that is, controlled low power applied to its field results in controlled higher power applied to the field of the main ac generator.

The stationary elements shown in figure 7-3 include the structural frame and housing, the insulated main-generator power windings, and the exciter field. The rotating elements include the shaft, the main field, and the exciter armature. A cooling-ventilation system is incorporated in the generator design to keep the temperatures of the windings and component parts within design limits. The generator cooling-ventilation requirements are determined by the application, the type of winding, the insulation, and the enclosure. Four classes of insulation are available (A, B, F, and H). Selecting the correct insulation is important and depends on the application. Thermal degradation or breakdown of the insulation affects generator life. Generators are available with two basic enclosures to suit the type of application and the environment: the open type, and the totally enclosed type. The open type includes drip-proof, splash-proof, and semiguarded enclosures. The totally enclosed type includes non-ventilated, fan-cooled, and explosion-proof enclosures. Very large generators are hydrogen cooled.

Generator efficiency is determined at rated output voltage, frequency, power factor, and balanced load conditions (ref. 6). The efficiency is defined as output power plus losses. Losses include copper losses, mechanical losses (friction), core losses, and stray load losses.

Generators are usually rated in terms of the kilowatt output divided by power factor (kVA) load at a specific voltage, speed, and power factor that can be carried continuously without overheating. The real (useful) power output (kW) of the generator is limited to a value within the kVA ratings by the capability of the prime mover.

Standards have been established by the National Electrical Manufacturers Association for the performance, safety, testing, construction, and manufacture of generators within certain product scopes (ref. 20). These standards relate to a product that is commercially standardized and subject to repetitive manufacture. Standards include output power ratings, speed ratings, and voltage ratings for 50- and 60-hertz generators (table 7-1).

Over the years, many different types of generator and generator system have been demonstrated. These include - but are not limited to - the inductor, Lundell, and flux-switch alternators. Various variable-speed, constant-frequency systems, including the field-modulated generator, have also been demonstrated. With the exception of the flux-switch alternator, which is used almost exclusively for induction hardening, these alternators have not been accepted to date in the industrial community. The major reasons for this lack of acceptance are higher cost, complexity, and low reliability.

7.2 DEVELOPMENT STATUS

A sound and well-developed technology base exists today in the commercial ac generator industry. A number of companies have developed and offer commercial ac generators in the 1-kilowatt to 50-megawatt range for industrial and utility use. Factors considered in applications include type of phase, frequency, output-line voltage, enclosure cooling, insulation temperature rating, and overload factors. Standard and engineered designs are available for the various applications. The ac generators are developed to established standards. Efficiency can be improved, in many cases, by using more select materials and by design changes - but these are nonstandard units.

7.3 AVAILABILITY

The typical availability of ac generators in the 5- to 50 000-kilowatt-electric power range is shown in figure 7-4. These data are based on units designed for a 0.8 power factor, three phases, a 60-hertz temperature rise, and class F insulation.

7.4 OPERATING CHARACTERISTICS

Generator operation is affected whenever the load increases or decreases. When an electric load is added, the generator draws more power from the prime mover to increase the electric power output. (The output voltage tends to drop because of internal impedance.) Conversely, removing or decreasing the load from the generator removes or decreases the load from the prime mover and raises the output voltage. Since electrical equipment on the line requires constant voltage for satisfactory operation, generator output power and voltage must be controlled to match the load requirement. A voltage regulator provided in the system senses the generator output voltage and controls the exciter to maintain a constant generator output voltage. With a load change, the regulator increases or decreases the current from the exciter to the generator field poles and thereby controls the magnetic field and the matching output with the load. A generator with a voltage regulator normally operates at a constant voltage with ±1 percent of rated voltage.

The frequency of the generated output is a function of the rotating speed of the generator. Generator speed, and hence output frequency, are regulated by controlling the speed of the direct-coupled prime mover.

The efficiency of commercial ac generators as a function of power level is shown in figure 7-5. The efficiency is shown as a range for each power level and is based on the different manufacturers' designs.

Efficiency increases with increasing size. At part-load operation the efficiency of the generator decreases slightly as shown in figures 7-5(b) and (c).

7.5 COST

Cost as a function of power level is shown in figure 7-6 for generators in the 5- to 50 000-kilowatt-electric range. Cost is shown as a range for each power level and is based on single-unit procurement and list or estimated cost data received from the manufacturers. The costs shown do not include the discount factor or the multiplier (proprietary data), which are based on the type of user (small or large) or multiple-unit procurements. These factors were unavailable as costs fluctuate periodically. The single-unit costs per kilowatt range from \$130 in the 10- to 100-kilowatt-electric size to \$30 to \$40 in the 1000-kilowatt-electric (or greater) size.

Generator weight as a function of power level is shown in figure 7-7.

7.6 OPERATING CONSTRAINTS

Operationally, the generator is constrained to the design electric load, voltage, and speed and the temperature rating of the insulation.

7.7 EXPERIENCE FACTORS

The ac generators are fully developed components with substantial operating history and documented maintenance schedules in industrial, utility, and marine applications. With correct scheduled maintenance the life will match that of the prime mover.

TABLE 7-1. - ALTERNATING-CURRENT GENERATOR RATINGS

(a) Speed

Number of poles	Frequency, Hz	
	60	50
	Speed, rpm	
4	1800	1500
6	1200	1000
8	900	750
10	720	600
12	600	500
14	514	429
16	450	375
18	400	333
20	360	300
22	327	273
24	300	250
26	277	231
28	257	214
30	240	200
32	225	188
36	200	167
40	180	150
44	164	136
48	150	---
52	138	---

(b) Voltage rating for 60-hertz circuit

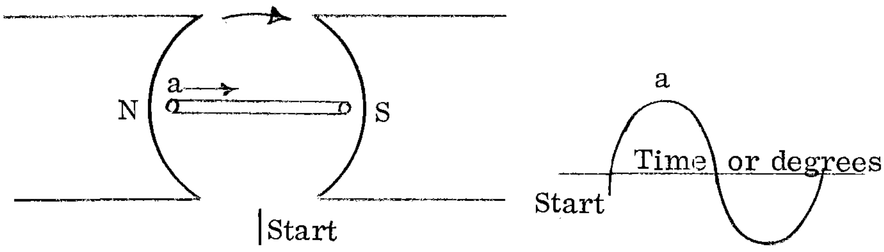
Number of phases	
3	1
Voltage rating, V	
208Y/120	125
240	125/250
480	250
600	
2400; [a]2500	
4160Y/2400; [a]4330Y/2500	
4800; [a]5000	
6900	
[b]11 500	
[b]12 500	
13 800	
14 400	

[a]Frequently desirable for servicing nominal 2400-, 4160Y/2400-, and 4800-volt distribution systems.

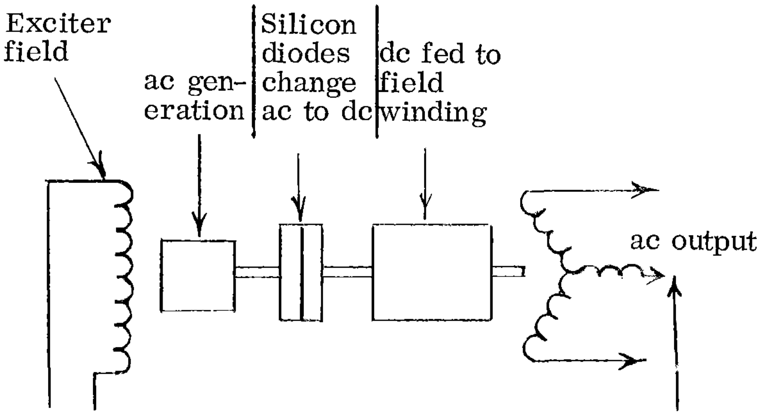
[b]Recognized for use on established systems but not preferred for new undertakings.

(c) Output power (kW) and output divided by power factor (kVA) for 60- and 50-hertz circuits and lagging power factor of 0.8

Output divided by power factor, kVA	Output power, kW	Output divided by power factor, kVA	Output power, kW
1.25	1	1 125	900
2.5	2	1 250	1 000
3.75	3	1 563	1 250
6.25	5	1 875	1 500
9.4	7.5	2 188	1 750
12.5	10	2 500	2 000
18.7	15	2 812	2 250
25	20	3 125	2 500
31.3	25	3 750	3 000
37.5	30	4 375	3 500
50	40	5 000	4 000
62.5	50	5 625	4 500
75	60	6 250	5 000
93.8	75	7 500	6 000
125	100	8 750	7 000
156	125	10 000	8 000
187	150	12 500	10 000
219	175	15 625	12 500
250	200	18 750	15 000
312	250	25 000	20 000
375	300	31 250	25 000
438	350	37 500	30 000
500	400	43 750	35 000
625	500	50 000	40 000
750	600	62 500	50 000
875	700	75 000	60 000
1000	800		



(a) Basic configuration.



(b) Brushless generator.

Figure 7-1. - Schematics of alternating-current generators.

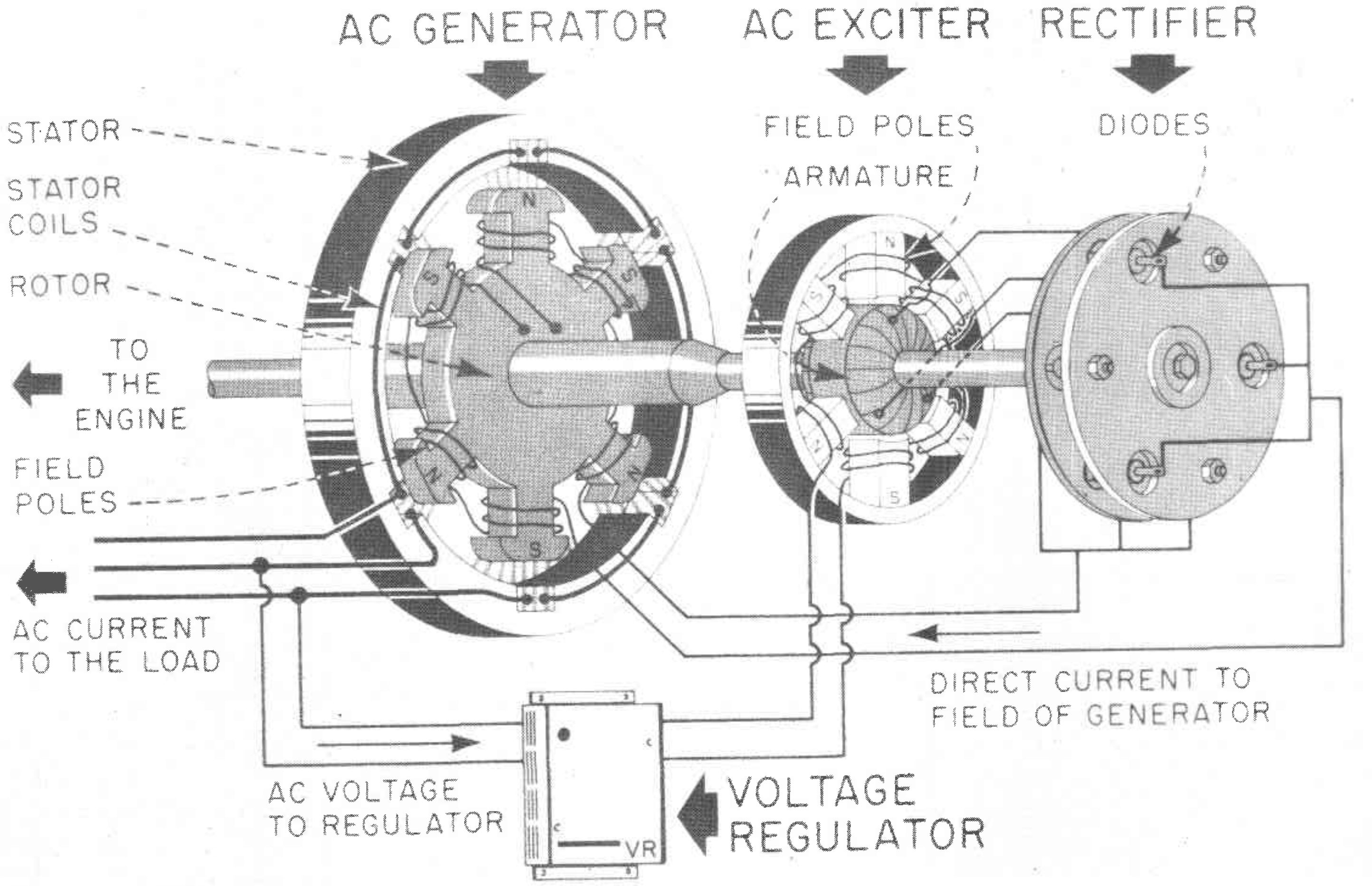


Figure 7-2. - Elements of typical alternating-current generator. (Courtesy of Electric Machinery Mfg Co., Division of Turbodyne Corp.)

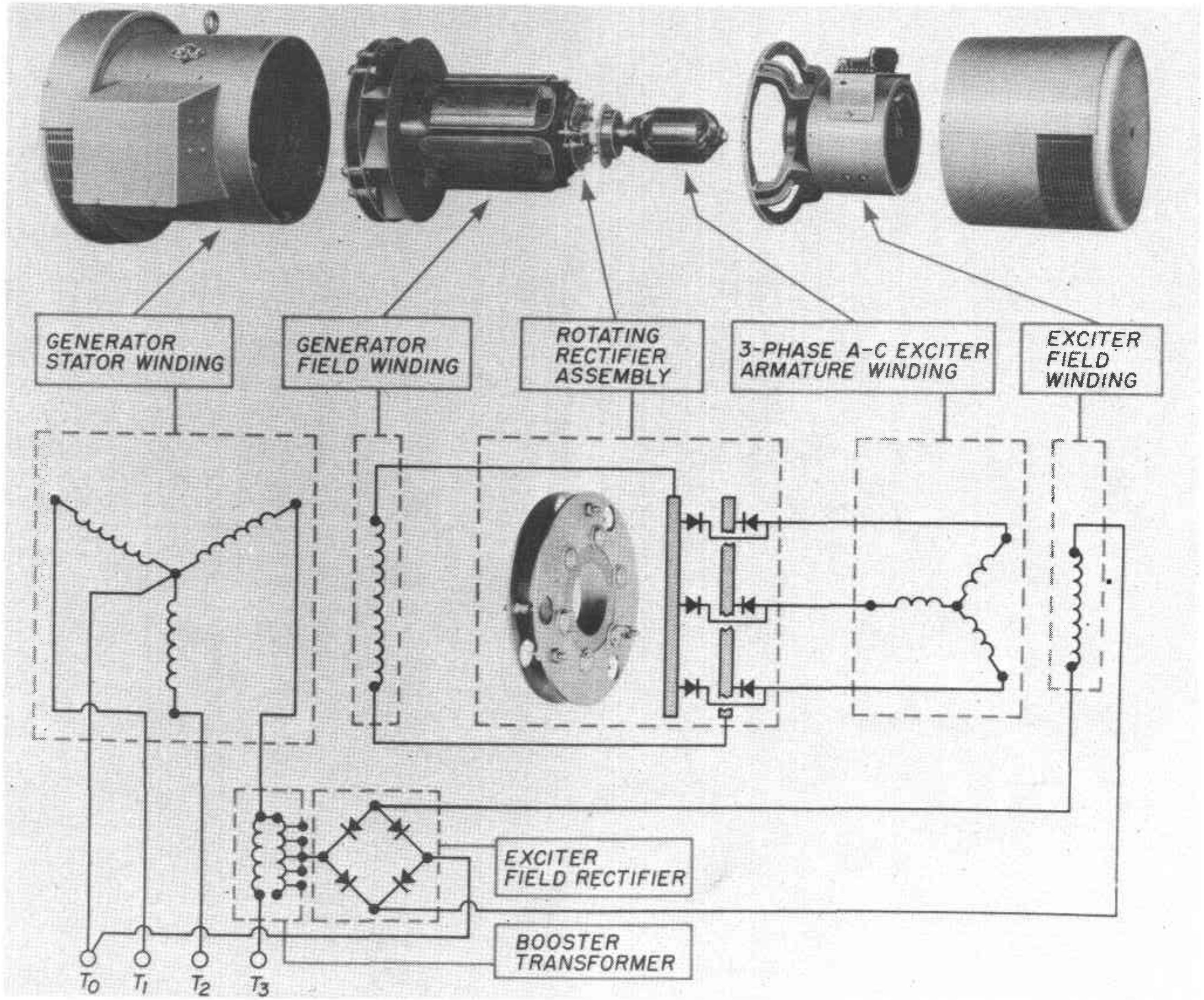


Figure 7-3. - Relationship of alternating current generator components to an electrical schematic. (Courtesy of Electric Machinery Mfg. Co., Division of Turbodyne Corp.)

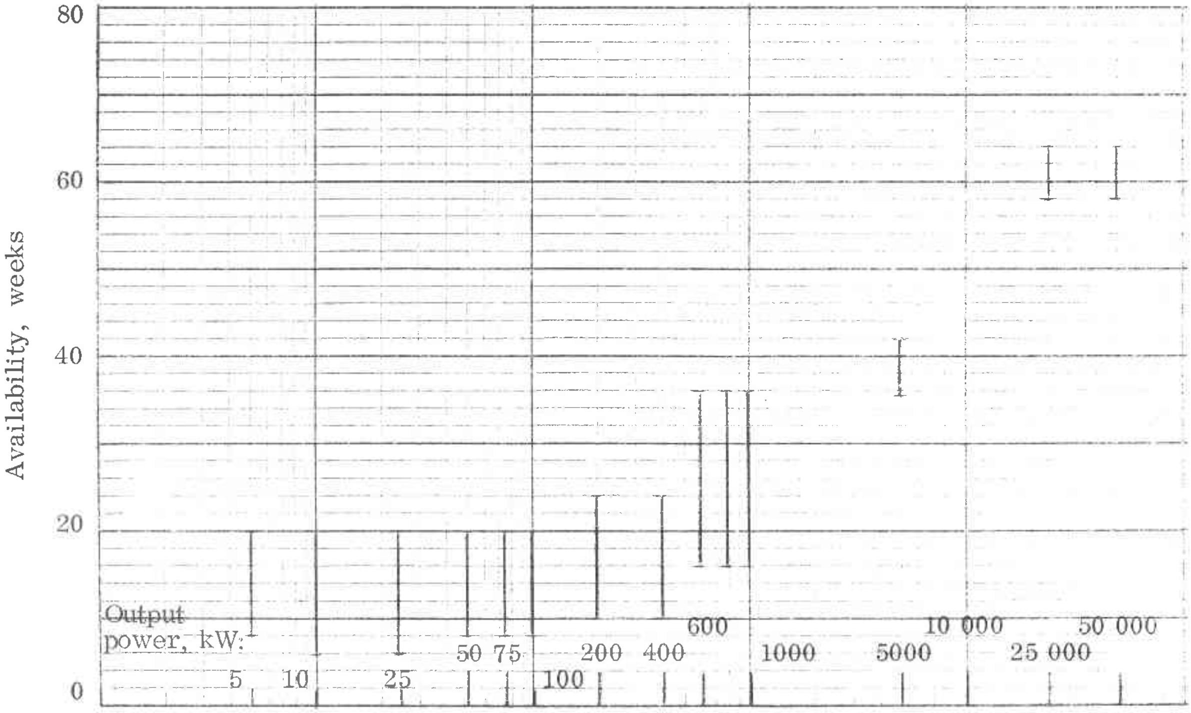
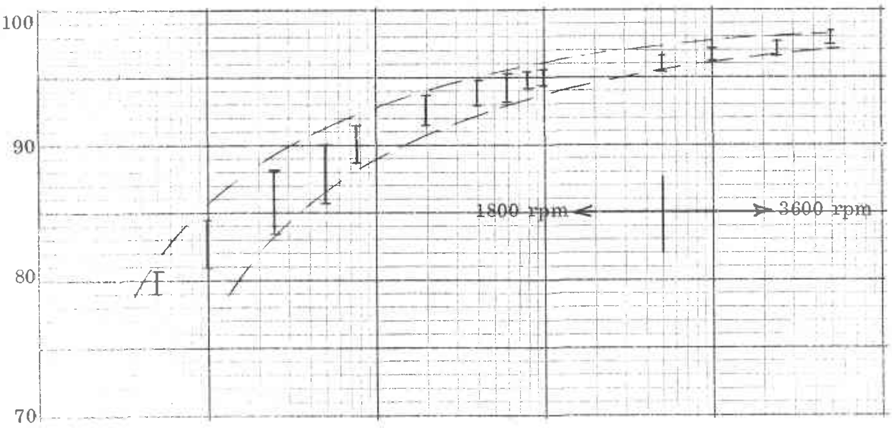
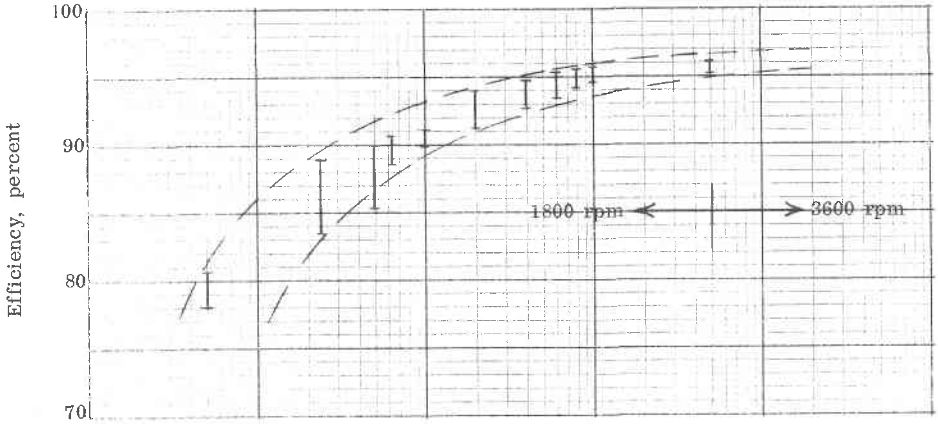


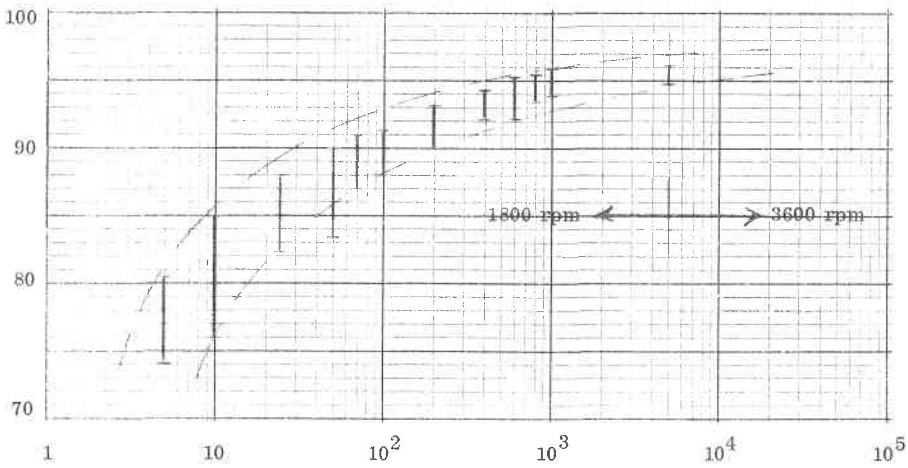
Figure 7-4. - Availability of alternating-current generators as function of power level.



(a) Load, 100 percent.



(b) Load, 75 percent.



(c) Load, 50 percent.

Figure 7-5. - Efficiency of alternating-current generators as function of power level and load. Power factor, 0.8.

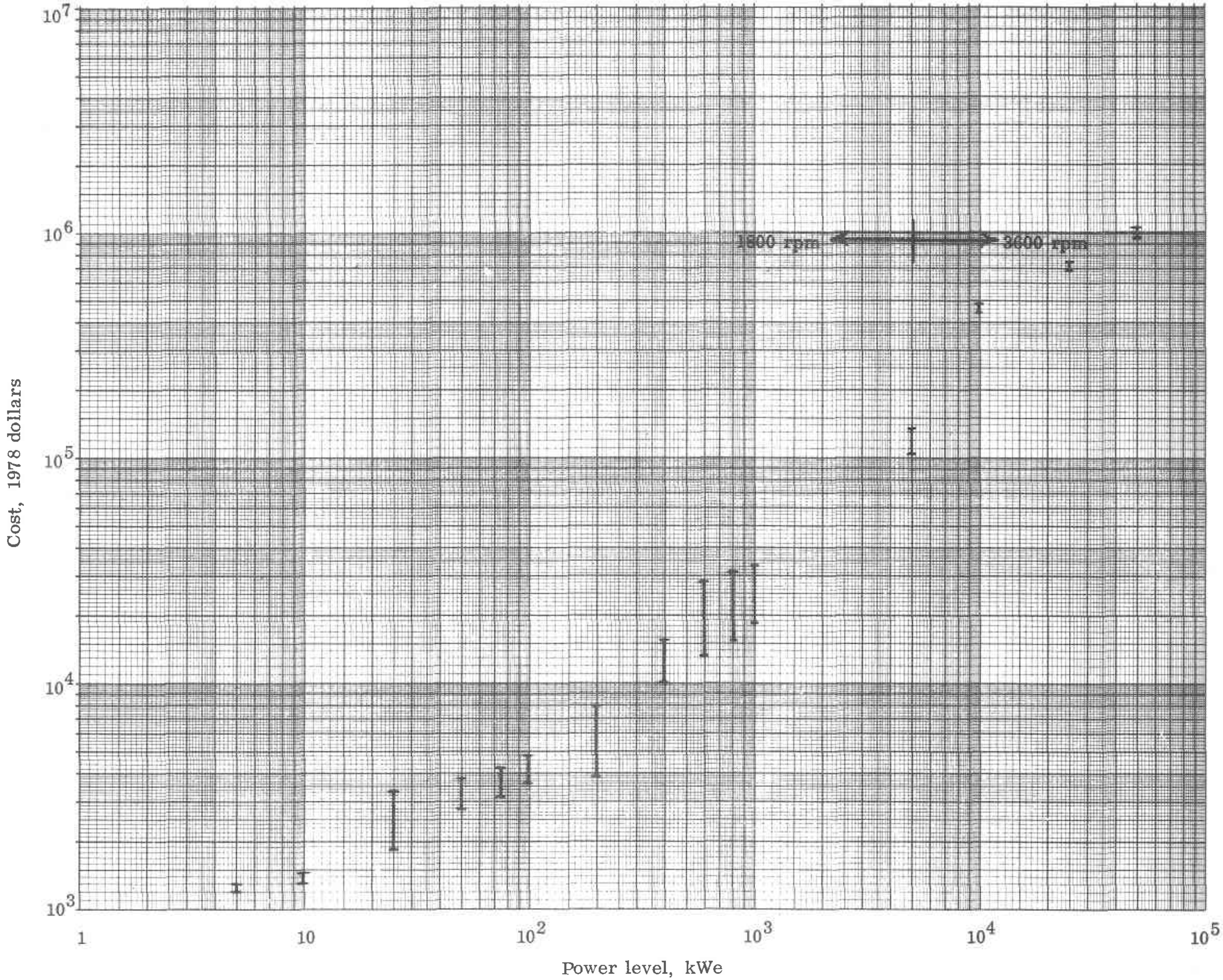


Figure 7-6. - Cost of alternating-current generators as function of power level. (Costs are manufacturers' list or estimated costs and do not include the discount or multiplier.)

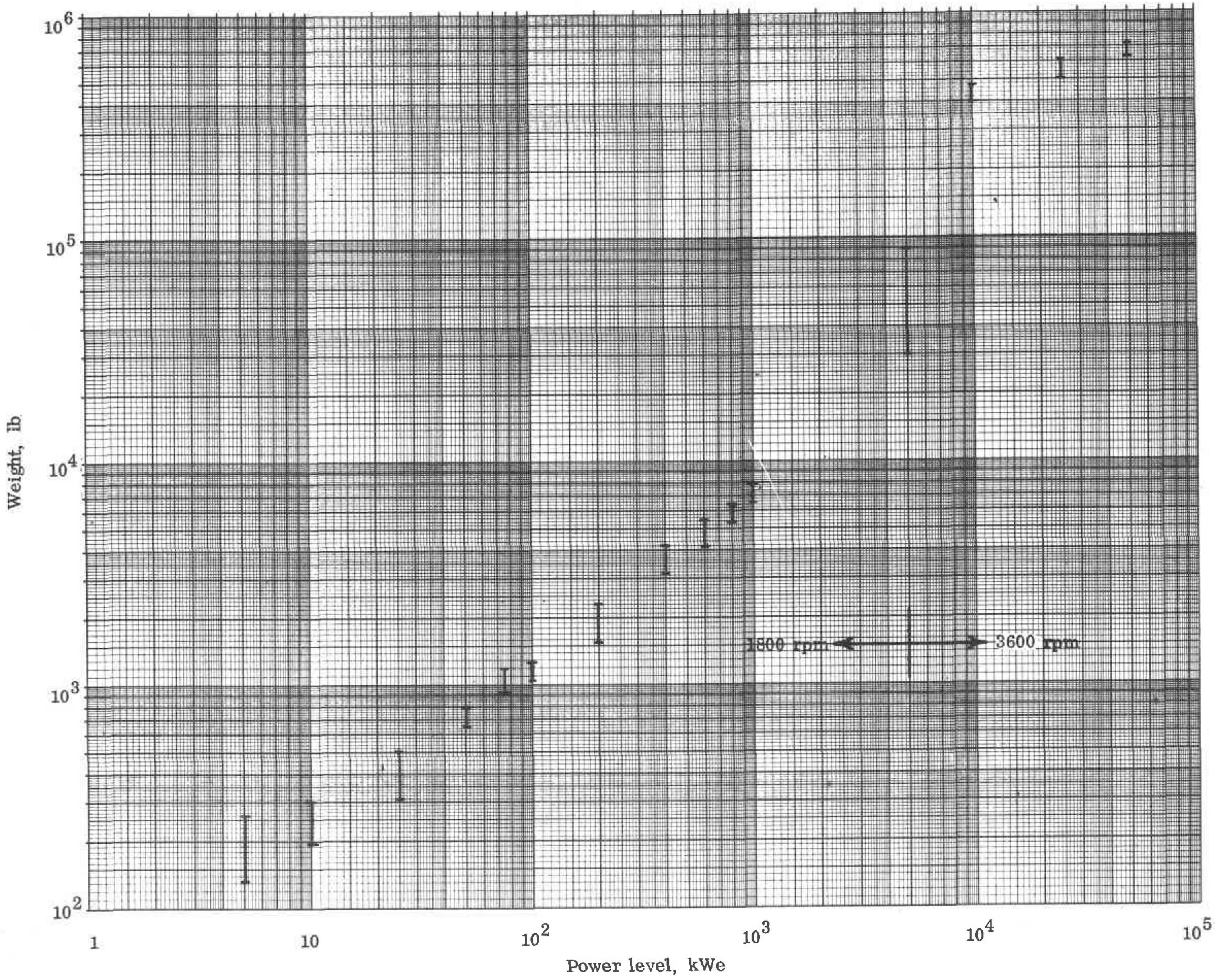


Figure 7-7. - Weight of alternating-current generators as function of power level.

8.0 POWER-CONVERSION SYSTEM COMPONENT SUMMATION

by Thaddeus S. Mroz, M. Murray Bailey, Jack A. Haller, Harvey S. Bloomfield, Robert J. Stochl, and Robert E. Hyland

This handbook presents the results of a survey of selected commercial components applicable to Rankine-cycle, Brayton-cycle, and Stirling-cycle solar thermal power-generating systems. The preceding sections described the solar thermal power-generating systems and their components. Selected data on these components are presented and include development status, availability, cost, operating constraints, operating characteristics, and experience factors. These data are provided for components sized for selected system power levels from 5 to 50 000 kWe.

The data presented are based on information provided by the respective component manufacturers. Information is provided for

(1) Rankine cycle
- (a) Steam turbines
- (b) Organic turbines
- (c) Reciprocating expansion engines
- (d) Condensers
- (e) Boiler feed pumps

(2) Brayton cycle
- (a) Simple, open-cycle gas turbine engines
- (b) Regenerated, open-cycle gas turbine engines
- (c) Closed-cycle engines

(3) Stirling cycle

(4) Components common to all cycles
- (a) Speed reducers
- (b) Alternating-current generators

In addition to the information in the preceding sections, the appendixes provide a description of each power-system cycle, a listing of commercial sources of power-system components, and the criteria provided to the manufacturers for component selection.

Data are provided in this handbook for developmental components that have been tested, components that are in the preproduction stage (hard tooling not yet available), components that can be mass produced (hard tooling available), and large components (10 000 to 50 000 kWe) that are designed and fabricated on order. This survey included component designs developed years ago, improved designs, and recently developed new designs that reflect higher efficiency and improved performance. Components developed for the commercial market may not be optimized for a specific operating point or narrow operating range, as is the case in space power applications. The designs cover a broader operating range.

Past development of these components has considered:

(1) Competitive operating range and rating, consistent with market demand

(2) Competitive cost

(3) Competitive performance characteristics and efficiency

(4) Long operational life

(5) High reliability

(6) Competitive size and weight

(7) Ease of maintenance

(8) Design consistent with multiple and mass-production techniques and tooling

(9) Cost-competitive materials of construction

Designs will vary among manufacturers because of their different design and fabrication techniques. In the small and intermediate sizes, and in some cases the large sizes, the manufacturers developed a line of components for a broad range of performance requirements.

The following sections summarize the individual components.

8.1 RANKINE-CYCLE COMPONENTS

8.1.1 Steam Turbines

Steam turbines marketed today fall into two basic categories, noncondensing and condensing turbines. The noncondensing turbine is the more prevalent design and is used for a variety of mechanical drive applications through an output range of 1 to over 50 000 shaft horsepower. The condensing turbine, which is of prime interest in this handbook, is designed for operational exhaust pressures below atmospheric and is used in electric power generation. Condensing turbines are available for the steam inlet conditions required for solar thermal power applications.

Condensing steam turbine designs are available for exhaust pressures to 2 inches of Hg absolute and, depending on application requirements, for single and multiple reheat and single and multiple extraction. Selecting a condensing steam turbine requires careful matching of system design requirements with the available turbine designs to select a component with an optimum efficiency and a low water flow rate for that application. Using an oversized turbine or a turbine with an incorrect backpressure rating will result in a lower efficiency and a higher steam flow rate.

Based on market demand and data inputs, commercial condensing turbines are designed for output power levels of over 100 shaft horsepower. Development of small condensing turbines optimized for efficiency in the lower power range for solar thermal applications is required. The current trend in new commercial turbine designs is toward higher rotating speeds, inlet pressures, and temperatures.

8.1.2 Organic Turbines

Organic turbines are turbines designed for use with an organic working fluid in a Rankine-cycle power system. A limited number of these turbines have been developed by different manufacturers for specific applications. In each case the performance was optimized for the specific system requirements. These organic turbines were designed for higher operational speeds and are classed as developmental or preproduction components. Cost data and availability were not included for these items since hard-tooling and production costs were not available.

8.1.3 Reciprocating Expansion Engines

The low-speed, heavy-duty reciprocating steam engines that were developed in the past were characterized by long life and high reliability. The engines were relatively large and heavy. Current reciprocating engine designs and development have been directed to compact and weight-competitive configurations and to higher operating speeds (36 000 rpm).

Presently, the availability of the reciprocating expansion engine is confined to the older, low-speed engines in the 12- to 3000-shaft-horsepower range and the more recently developed and compact high-speed engines (36 000 rpm) in the 5- to 75-shaft-horsepower range. The high-speed engines are in a preproduction status. Production status is contingent on market demand.

8.1.4 Condensers

Industrial condenser technology is fully established in the 5- to 50 000-kWe range for water-cooled and air-cooled condensers. Commercial condensers can be provided from a developed line of components or can be designed and tailored to specific application requirements.

8.1.5 Boiler Feed Pumps

Three types of pumps are commercially available for boiler feed service for solar thermal power applications in the system output range of 5 to 50 000 kWe. These are the reciprocating pumps, centrifugal pumps, and regenerative turbine pumps. A number of manufacturers offer a commercial line of these pumps that encompasses the pressure-flow requirements in this handbook. Selecting the type of pump for solar thermal application depends on the pressure-flow requirements. Pump development is not considered to be required.

8.2 BRAYTON CYCLE - GAS TURBINE ENGINES

Existing industrial gas turbine engine technology is derived from the early heavy-duty gas turbine engines developed for powerplant service and from aircraft engine technology. To date, industrial engine development includes the range from 11 to 100 000 shaft horsepower. This range includes both simple open-cycle engines and regenerative open-cycle engines in single- and double-shaft configurations. Currently, a number of manufacturers offer a line of gas turbine engines in a specific power range. However, turbine engine development through the output range of 10 to 50 000 kWe is not yet as extensive as for steam turbines.

Applying the gas turbine engine to solar thermal power-generating systems will require development of, or modifications to, existing designs for solar heat input to the engine. However, the technology base for industrial gas turbine engines is well established to accommodate application to solar thermal power-generating systems.

8.3 STIRLING CYCLE

The Stirling engine is still in the development stage for applications in ground and marine transportation and power generation using solar power or fossil derivative fuels. A number of companies are focusing current development on engines with outputs of 1 to 360 horsepower.

8.4 COMPONENTS COMMON TO ALL CYCLES

8.4.1 Speed Reducers

The industrial technology base is well established for separately mounted or integral speed-reduction gear systems at the speed ratios and torque requirements of steam-turbine and high-speed-gas-turbine alternating-current generator applications. A number of companies offer a line of speed reducers through the output range of this handbook. New development is not required.

8.4.2 Alternating-Current Synchronous Generators

The alternating-current synchronous generator is a fully developed and commercially available component through the 5- to 50 000-kWe range and above. Alternating-current generators are built to National Electric Manufacturers Association (NEMA) standards and thus conform to industrial requirements and a competitive industrial market. Generators with improved (over standard production units) efficiencies can be made available in certain sizes. These are special units with an upgraded design and premium-quality materials. This results in higher cost and longer delivery times.

APPENDIX A

CANDIDATE POWER-CONVERSION SYSTEM CYCLES

by Robert J. Stochl

The appropriate power-conversion system for a particular solar thermal power application depends on the temperature of the available heat source. A power-conversion system for a 3000° F heat source might, for example, be different from one for a 1000° F heat source. The type of available heat rejection also influences the choice of a power-conversion system. A Rankine cycle would not be selected if large amounts of cooling water were not available. A Brayton cycle might be more appropriate.

This section provides a thermodynamic description of the three basic power-conversion cycles that are candidates for use in near-term solar thermal power applications where the source temperature is between 1000° and 1500° F. These power-conversion cycles are the Rankine, Brayton, and Stirling cycles. Combined cycles, consisting of a Brayton topping cycle operating in conjunction with a Rankine bottoming cycle, are also described.

For each conversion system there are a number of possible cycle configurations. Some cycle configurations would not be applicable to systems designed for low power levels. For example, a reheated or regenerated (feedwater heaters) Rankine cycle would not be considered for a single-stage, low-power-level turbine. On the other hand, multiple reheats and several stages of feedwater heating would be considered for large-power-level turbines. There is a large variation in component performance with power level. The component characteristics discussed in sections 3.0 to 5.0 of this handbook indicate potential system performance and efficiency at different power levels.

The following discussion of each conversion system covers the basic cycle configuration, its operation, and the basic relations for calculating cycle efficiencies and work outputs. The system modifications that can be used to increase performance over that of the basic cycle are discussed. Generator and gearbox losses are not included in this discussion. These losses depend on power level and can be easily included by multiplying the quoted cycle efficiencies by the appropriate electric generator and gearbox efficiencies.

RANKINE CYCLE

The Rankine power-conversion cycle is perhaps the most widely used thermal cycle for electric energy generation. The basic Rankine cycle is shown in figure A-1. Heat is supplied to the vapor generator (solar boiler), where the working fluid is converted to vapor. The vapor is then expanded adiabatically in an expander to produce work.

Vapor leaving the turbine enters the condenser, where heat is removed until the vapor is condensed to a liquid. Saturated liquid is delivered to a boiler feed pump, where its pressure is raised to the saturation pressure (the pressure that corresponds to the boiling temperature in the vapor generator). The fluid then returns to the vapor generator and the cycle repeats itself.

The thermodynamic efficiency of the Rankine cycle is defined by

$$\eta_R = \frac{W_N}{Q_{in}}$$

where the net work output of the engine

$$W_N = W_T - W_P$$

the turbine specific work

$$W_T = \eta_T (h_1 - h_2)$$

the pump specific work

$$W_P = \frac{h_4 - h_3}{\eta_P}$$

the specific energy (heat) input into the vapor generator

$$Q_{in} = \frac{h_1 - h_4}{\eta_V}$$

and

η_T turbine efficiency
η_P pump efficiency
η_V vapor generator efficiency
h_i enthalpy corresponding to ideal cyclic points in figure A-1

Therefore,

$$\eta_R = \frac{\eta_T (h_1 - h_2) - \left(\dfrac{h_4 - h_3}{\eta_P}\right)}{\dfrac{h_1 - h_4}{\eta_V}}$$

One way to improve the efficiency of the basic Rankine cycle is by reheat, as shown in figure A-2. The vapor is expanded in the turbine until it reaches approximately the saturation line at point 2. It is then returned to the vapor generator and reheated at constant pressure to point 3. The resulting vapor is expanded in the remaining turbine stages to point 4. Such a heating process has two advantages: It increases the average temperature at which external heat is added and thereby increases cycle efficiency, and it results in an operating condition that involves less moisture in the expansion process. The thermodynamic efficiency of a reheated Rankine cycle is given by

$$\eta_R = \frac{\eta_T (h_1 - h_2) + \eta_T (h_3 - h_4) - \left(\dfrac{h_6 - h_5}{\eta_P}\right)}{\dfrac{(h_1 - h_6) + (h_3 - h_2)}{\eta_V}}$$

Another way to increase efficiency by increasing the average temperature at which external heat is added is regeneration (or feedwater heating). In a regenerated cycle, some hot vapor is extracted from an intermediate turbine stage and exchanges heat with cooler fluid before it enters the vapor generator. The effect of this heat exchange is shown in figure A-3. A fraction f of the total vapor flow $\dot{m}$ is extracted at point 2 and fed directly to the feedwater heater. The remainder $(1 - f)\,\dot{m}$ of the vapor flow passes through the condenser and is pumped to the feedwater heater, where it is heated by the extracted vapor. Thus, for the same maximum temperature, the average temperature at which external heat is added to the cycle is higher with the feedwater heater than without it. There are two types of feedwater heaters: the open, and the closed. In the open heater, vapor and liquid are intimately mixed at constant pressure. An additional pump is required for each open feedwater heater. In the closed heater, the fluids are kept separated. An additional pump is not required for the closed heater. The thermodynamic efficiency for the regenerated cycle is

$$\eta_R = \frac{\eta_T (h_1 - h_2) + \eta_T (1 - f)(h_2 - h_3) - \frac{1}{\eta_P} (1 - f)(h_5 - h_4) - \frac{1}{\eta_P} (h_7 - h_6)}{\frac{h_1 - h_7}{\eta_V}}$$

Because in the Rankine cycle the pump handles pure liquid, the pump work is usually only a small fraction (1 to 2 percent) of the turbine work output and can generally be neglected.

BRAYTON CYCLE

The Brayton and Rankine power cycles are thermodynamically similar except for the phase of the working fluid. The Brayton working fluid is a gas and the Rankine working fluid undergoes a phase change. As with the ideal Rankine cycle, the ideal Brayton cycle consists of two isentropic processes and two constant-pressure processes. There are two basic Brayton cycles: an open cycle, and a closed cycle. In an open Brayton cycle the working fluid (air) is not recirculated, in a closed cycle the working fluid is recirculated.

A simple open Brayton cycle is shown in figure A-4. A compressor raises the pressure of the air, heat is then added at high pressure in the heat-source heat exchanger. The high-pressure, high-temperature air is then expanded in the turbine to produce work. Part of this turbine work is used to drive the compressor, the remainder is available to produce useful work.

The thermal efficiency of the open Brayton cycle is defined the same as for the Rankine cycle; that is,

$$\eta_B = \frac{W_N}{Q_{in}} = \frac{W_T - W_C}{Q_{in}}$$

where the turbine specific work

$$W_T = \eta_T (h_3 - h_4)$$

the compressor specific work

$$W_C = \frac{h_2 - h_1}{\eta_C}$$

and the specific energy (heat) input to the vapor generator

$$Q_{in} = \frac{h_3 - h_2}{\eta_f}$$

where

η_C compressor efficiency
η_f efficiency of heat-source heat exchanger

For a gas

$$\Delta h = C_p \, \Delta T$$

where C_p is the specific heat at constant pressure. Therefore,

$$W_T = \eta_T C_p \, (T_3 - T_4)$$

$$W_C = \frac{C_p}{\eta_C} \, (T_2 - T_1)$$

$$Q_{in} = \frac{C_p}{\eta_f} \, (T_3 - T_2)$$

Finally, by using the relation for isentropic compression and expansion

$$\frac{T_{out}}{T_{in}} = \left(\frac{P_{out}}{P_{in}}\right)^{(\gamma-1)/\gamma}$$

and defining the compressor pressure ratio

$$\frac{P_2}{P_1} \equiv P_{RC}$$

and a loss pressure ratio

$$L \equiv \frac{\dfrac{P_3}{P_4}}{\dfrac{P_2}{P_1}}$$

the efficiency of a simple open Brayton cycle becomes

$$\eta_{B,S} = \eta_f \frac{\bar{C}_{p,T}\eta_T\left[1 - \left(\dfrac{1}{LP_{RC}}\right)^{(\gamma-1)/\gamma}\right] - \dfrac{\bar{C}_{p,C}}{\eta_C}\dfrac{T_1}{T_3}\left[P_{RC}^{(\gamma-1)/\gamma} - 1\right]}{\bar{C}_{p,f}\left(1 - \dfrac{T_1}{T_3}\left\{1 + \dfrac{1}{\eta_C}\left[P_{RC}^{(\gamma-1)/\gamma} - 1\right]\right\}\right)}$$

where $\bar{C}_{p,C}$, $\bar{C}_{p,T}$, and $\bar{C}_{p,f}$ are average heats for the compression, expansion, and heat-addition processes, respectively.

The closed Brayton cycle is thermodynamically identical to the open cycle just described, except that the closed cycle requires a heat-rejection heat exchanger. In the open cycle this function is performed by the atmosphere. A simple closed Brayton cycle is illustrated in figure A-5.

In both the open and closed simple Brayton cycles, the energy remaining in the turbine exhaust is either dissipated to the atmosphere (open cycle) or absorbed by the heat-rejection heat exchanger (closed cycle). Both methods represent a substantial loss of energy. In a regenerated open Brayton cycle, shown in figure A-6, some of the energy in the turbine exhaust is used to preheat the high-pressure gas from the compressor before it enters the heat-source heat exchanger. This regeneration reduces the energy that must be added to the cycle through the heat-source heat exchanger.

The term "regenerator effectiveness" E_r, where $E_r = (T_3 - T_2)/(T_5 - T_2)$, is a measure of how close this preheating process approaches the ideal process. With this term and the relations previously shown, the thermodynamic efficiency of a regenerated Brayton cycle is given by

$$\eta_{B,R} = \eta_f \frac{\bar{C}_{p,T}\eta_T\left[1 - \left(\dfrac{1}{LP_{RC}}\right)^{(\gamma-1)/\gamma}\right] - \dfrac{\bar{C}_{p,C}}{\eta_C}\dfrac{T_1}{T_4}\left[P_{RC}^{(\gamma-1)/\gamma} - 1\right]}{\bar{C}_{p,f}\left(1 - \dfrac{T_1}{T_4}(1 - E_r)\left\{1 + \dfrac{1}{\eta_C}\left[P_{RC}^{(\gamma-1)/\gamma} - 1\right]\right\} - E_r\left\{1 - \eta_T\left[1 - \left(\dfrac{1}{LP_{RC}}\right)^{(\gamma-1)/\gamma}\right]\right\}\right)}$$

It is also possible to improve the efficiency of the Brayton cycle by using multiple-stage compression with intercooling as well as multiple-stage turbines with reheat. However, these would only be used at high power levels and are not discussed further.

Using either an open or closed Brayton cycle has the following advantages: Normally, an open cycle has a combustor rather than a heat-source heat exchanger. The combustor eliminates much of the inefficiency associated with heat exchangers. However, in an application where solar energy supplies the heat input, a heat exchanger must be used. This negates one advantage of the open cycle. However, another advantage of the open cycle is that it does not require a heat-rejection heat exchanger, normally a costly item.

The closed Brayton cycle has a number of potential advantages that result from it being a closed thermodynamic cycle. The working-gas composition and pressure level are independent design parameters that can be chosen for the benefit of the turbomachinery and heat exchangers. Operation of the cycle at high-pressure levels would allow higher power density systems than could be obtained with open Brayton cycles. Also a working gas with a high thermal conductivity and hence a high heat-transfer coefficient can be used that, together with a high pressure level, could result in smaller heat exchangers.

The power level of a closed-cycle Brayton system can be controlled by changing system inventory and hence pressure level without changing system temperatures or volume flow rate. Therefore, component and system performance can be maintained at design-point levels for a wider range of power levels.

One disadvantage of the Brayton cycle in general is that, because it is a gas cycle, it requires a large amount of compressor work. Typically, the compressor requires 50 to 80 percent (depending on operating parameters) of the turbine output. This is in sharp contrast to the Rankine cycle, which requires only 1 or 2 percent of the turbine work to drive the pump.

STIRLING ENGINE

The Stirling engine is a reciprocating, external combustion, closed-cycle heat engine. It is gaining popularity because of its potential for high efficiency and its ability to use any source of heat.

Both a temperature-entropy and a pressure-volume diagram for the ideal Stirling cycle are shown in figure A-7. Heat is supplied to the working fluid during the constant-volume process (4-1) and during the isothermal expansion process (1-2). Heat is rejected during the constant-volume process (2-3) and during the isothermal compression process (3-4). During the constant-volume cooling process (2-3), heat is removed from the working fluid and stored in a regenerator. This heat is returned to the working fluid during the constant-volume heat process (4-1). In the ideal process all the heat rejected during the cooling process (2-3) is returned during the heating process (4-1).

The thermodynamic efficiency of the Stirling cycle is defined the same as for the Rankine and Brayton cycles; that is,

$$\eta_S = \frac{W_N}{Q_{in}} = \frac{W_{ex} - W_{comp}}{Q_{in}}$$

The expansion work (1-2) is given by

$$W_{ex} = \int_{V_1}^{V_2} P \, dV$$

for an isothermal expansion

$$PV = \text{Constant} = mRT_1$$

or

$$P = \frac{RT_1}{V}$$

Therefore,

$$W_{ex} = \int_{V_1}^{V_2} mRT_1 \frac{dV}{V} = mRT_1 \ln\left(\frac{V_2}{V_1}\right)$$

in the ideal cycle this quantity W_{ex} is also the heat input to the cycle Q_{in}. The compression work (3-4) is obtained in the same manner as the expansion work

$$W_{comp} = \int_{V_4}^{V_3} mRT_3 \frac{dV}{V} = mRT_3 \ln\left(\frac{V_3}{V_4}\right)$$

Therefore,

$$\eta_S = \frac{mRT_1 \ln\left(\frac{V_2}{V_1}\right) - mRT_3 \ln\left(\frac{V_3}{V_4}\right)}{mRT_1 \ln\left(\frac{V_2}{V_1}\right)}$$

since $V_3 = V_2$ and $V_4 = V_1$

$$\eta_S = 1 - \frac{T_3}{T_1}$$

So the ideal efficiency of the Stirling cycle is equal to the Carnot efficiency operating over the same temperature range. However, the ideal efficiency cannot be realized in practice because of the inefficiencies associated with the various processes.

One of the chief engine inefficiencies is that of the regenerator. With this inefficiency only a portion (4-5) of the heat rejected during the cooling process is returned during the heating process (4-1). This means that external heat must be added during the remaining portion of the constant-volume heat-addition process (5-1). The additional external heat that must be added is given by

$$Q' = mC_v (T_1 - T_5)$$

Therefore,

$$\eta_S = \frac{W_{ex} - W_{comp}}{Q_{in} + Q'} = \frac{mRT_1 \ln\left(\frac{V_2}{V_1}\right) - mRT_3 \ln\left(\frac{V_2}{V_1}\right)}{mRT_1 \ln\left(\frac{V_2}{V_1}\right) + mC_v (T_1 - T_5)}$$

If regenerator effectiveness is defined as

$$E_r = \frac{T_5 - T_3}{T_1 - T_3}$$

then

$$\eta_S = \frac{T_1 - T_3}{T_1 + \frac{C_v}{R} \frac{(T_1 - T_3)(1 - E_r)}{\ln\left(\frac{V_2}{V_1}\right)}}$$

The real Stirling engine processes differ significantly from the ideal cycle so that the basic performance equations just presented are useful for only the grossest approximations. The real and ideal cycles are shown on a pressure-volume diagram in figure A-8. Other factors, besides regenerator inefficiency, that lead to deviations from the ideal cycle are piston motion, dead air spaces, leakage, and external heat transfer. These factors are intimately related to one another and make strict analytical formulation difficult.

COMBINED CYCLES

The combined cycles considered herein are those wherein a Brayton and a Rankine cycle operate in series. Such a combination is thermodynamically advantageous and has a higher overall thermal efficiency than either the Brayton or Rankine cycle operating separately. The improvement in the combined thermal efficiency results from using energy that would otherwise be wasted.

A combined cycle using a regenerated closed Brayton cycle and a Rankine cycle is shown in figure A-9. Energy is normally lost from the regenerated Brayton cycle between points 6 and 7. In the combined cycle, a portion of this wasted energy is used as heat input to a lower temperature Rankine cycle.

The efficiency of the combined-cycle configuration shown in figure A-9 is given by

$$\eta_{CC} = \eta'_{B,R} + (1 - \eta_{B,R}) f \eta_R$$

where

$\eta'_{B,R}$ regenerated Brayton-cycle efficiency, including shaft loss

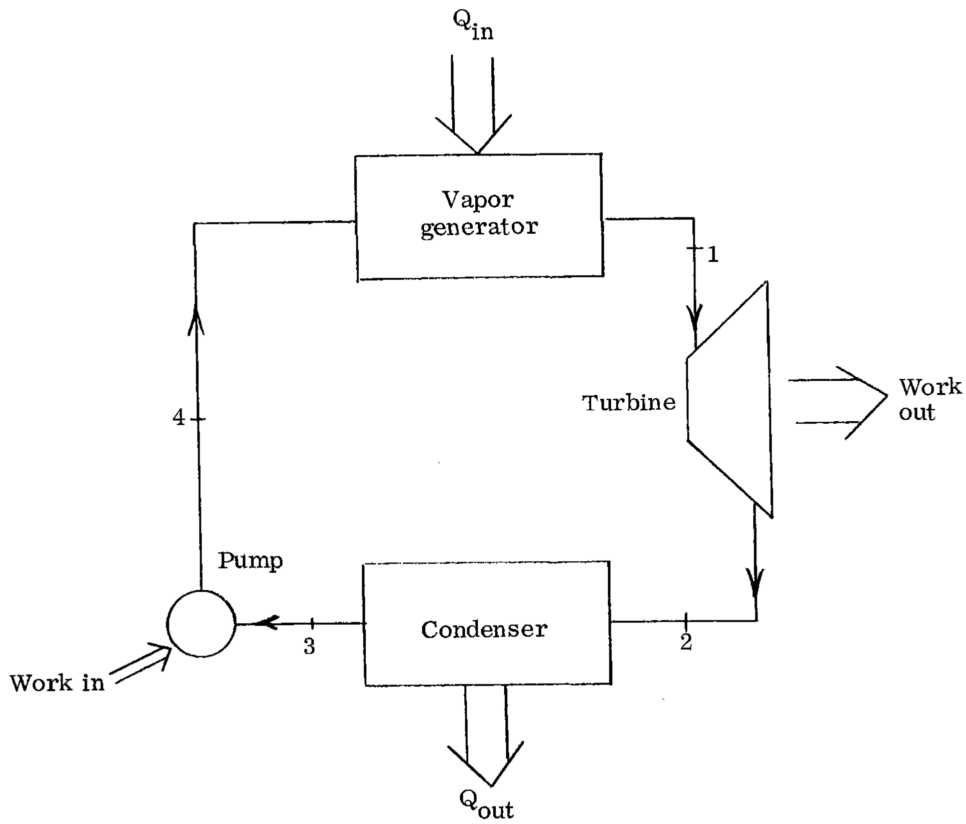
$\eta_{B,R}$ regenerated Brayton-cycle gross efficiency, not including shaft loss

f fraction of available exhaust heat from Brayton (topping cycle) that is transferred to Rankine (bottoming cycle)

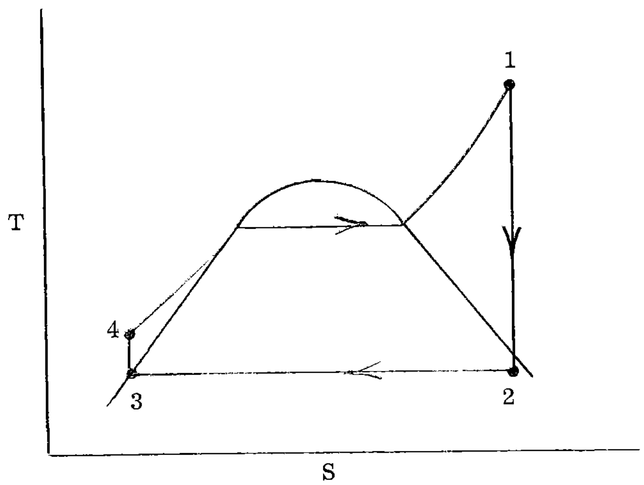
The factor $(1 - \eta_{B,R})$ is the fraction of the topping-cycle heat input that appears in the topping-cycle exhaust and is available for recovery. Values of $\eta'_{B,R}$, $\eta_{B,R}$, and η_R can be obtained from knowledge of the individual cycles and from the relations given earlier in this section in the Rankine and Brayton discussions. The factor f depends (1) on the topping-cycle exhaust temperature, (2) on the topping-cycle compressor inlet temperature, (3) on the pinchpoint in the vapor generator, and (4) on the thermodynamic properties of the topping and bottoming cycles.

This expression indicates that for a fixed topping cycle (i.e., fixed $\eta'_{B,R}$ and $\eta_{B,R}$) it is desirable from the standpoint of combined-cycle performance to choose bottoming-cycle fluid and operating parameters not to maximize just bottoming-cycle efficiency η_R but to maximize the product $f\eta_R$.

For combined cycles an indication of the power split between the Brayton topping cycle and the Rankine bottoming cycle can be obtained from the ratio of the topping-cycle efficiency to the combined-cycle efficiency. That is, the ratio of topping-cycle power to total power is equal to $\eta'_{B,R}/\eta_{CC}$.

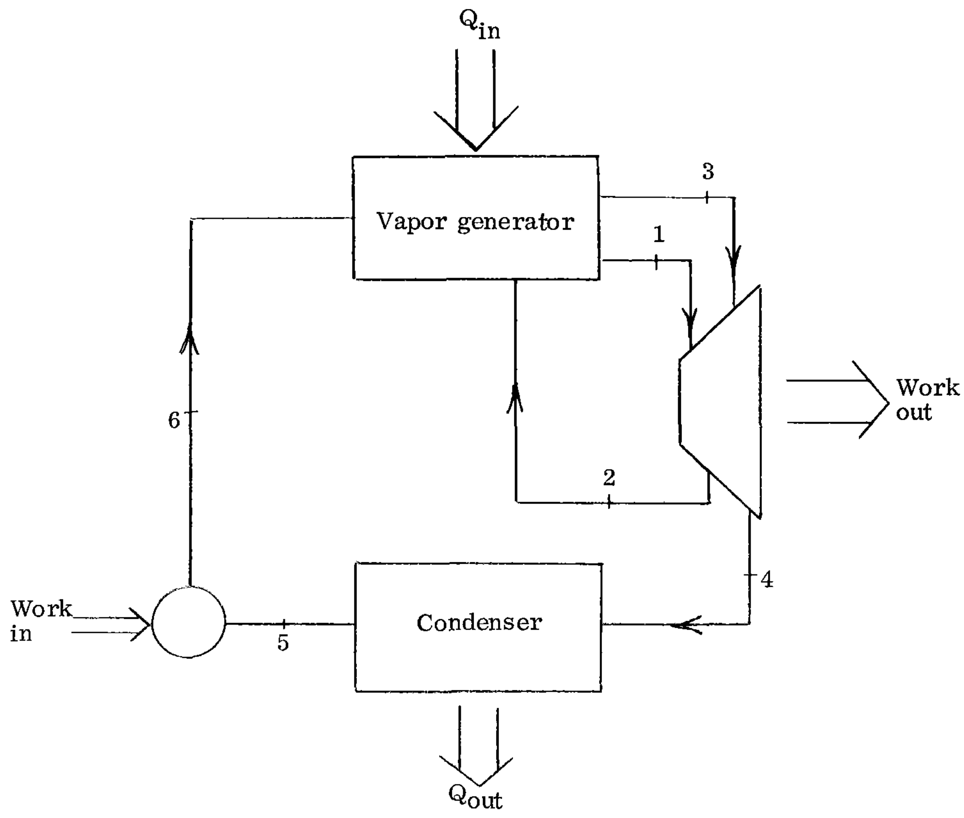


(a) Schematic.

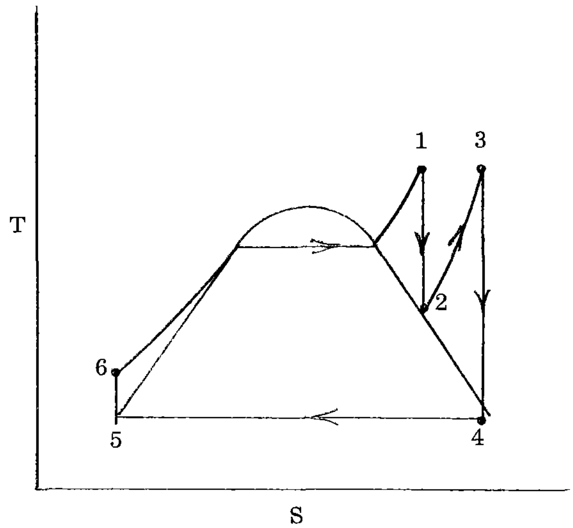


(b) Temperature-entropy diagram.

Figure A-1. - General Rankine cycle.

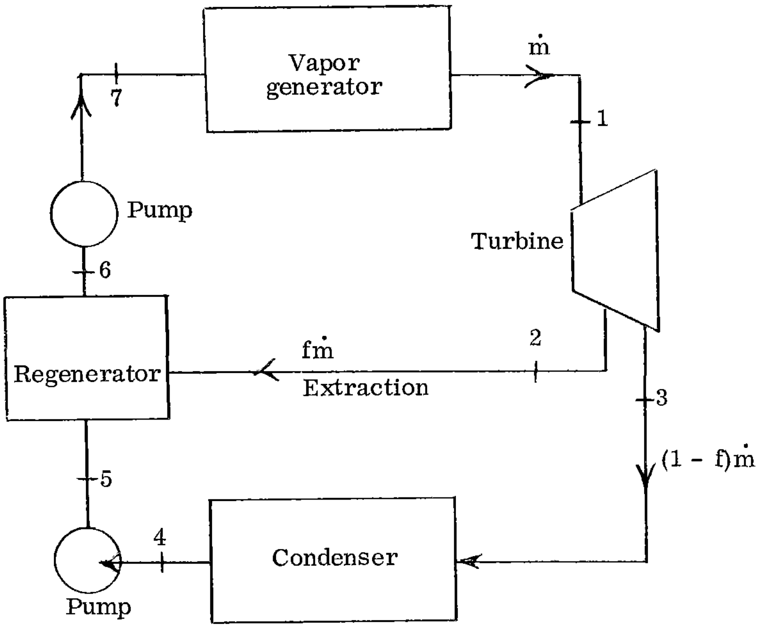


(a) Schematic.

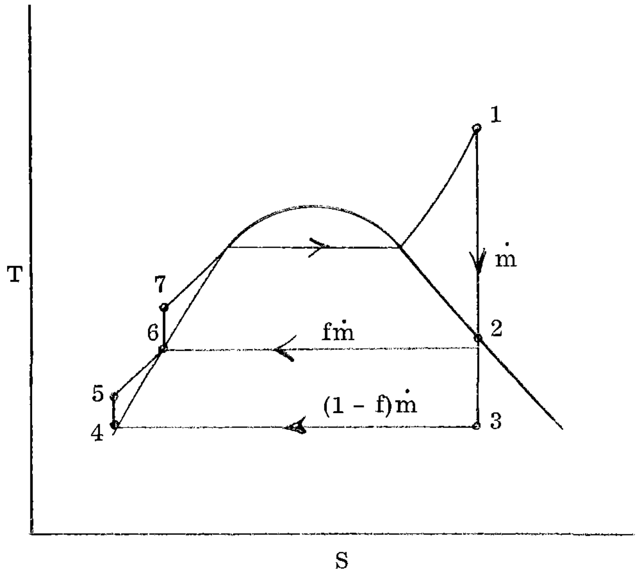


(b) Temperature-entropy diagram.

Figure A-2. - Reheated Rankine cycle.

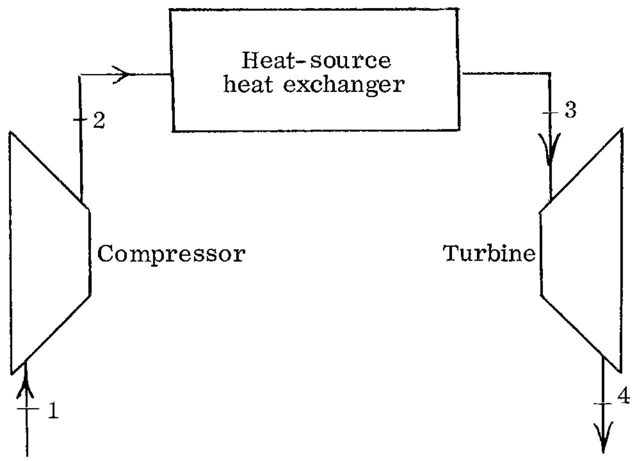


(a) Schematic.

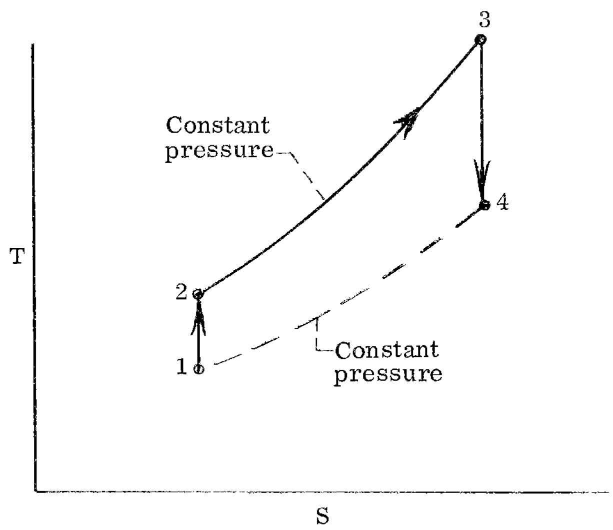


(b) Temperature-entropy diagram.

Figure A-3. - Regenerated Rankine cycle.

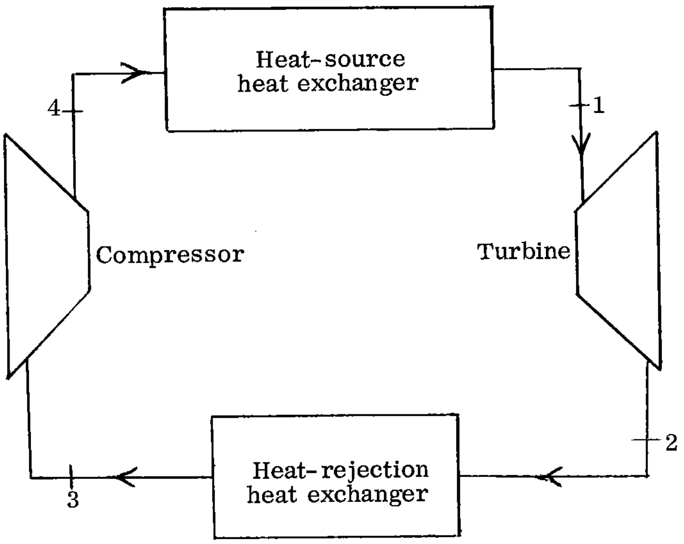


(a) Schematic.

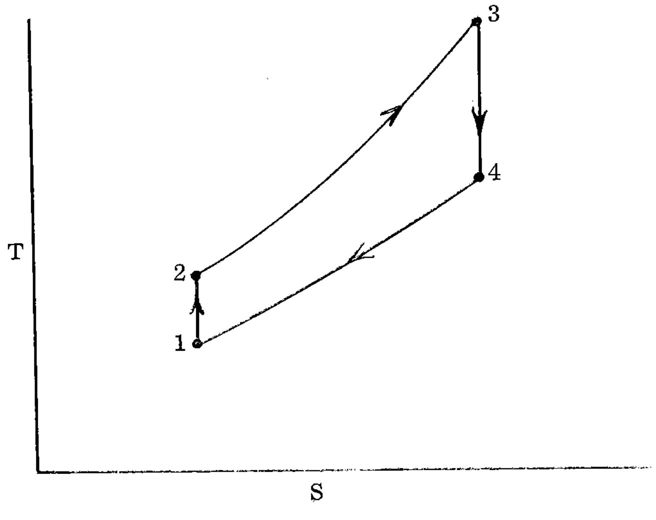


(b) Temperature-entropy diagram.

Figure A-4. - Simple open Brayton cycle.

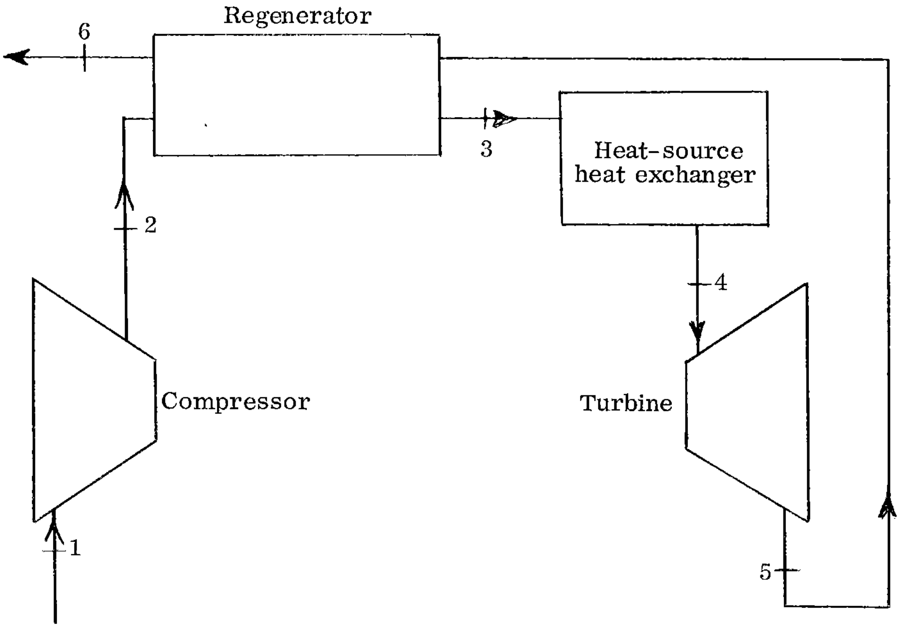


(a) Schematic.

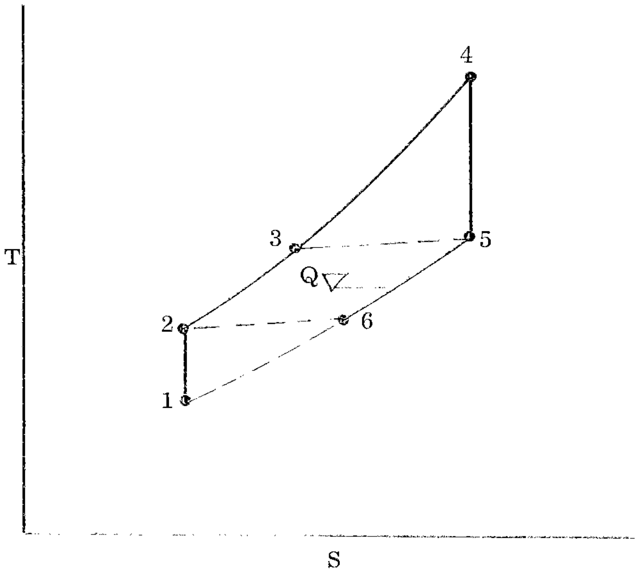


(b) Temperature-entropy diagram.

Figure A-5. - Closed Brayton cycle.

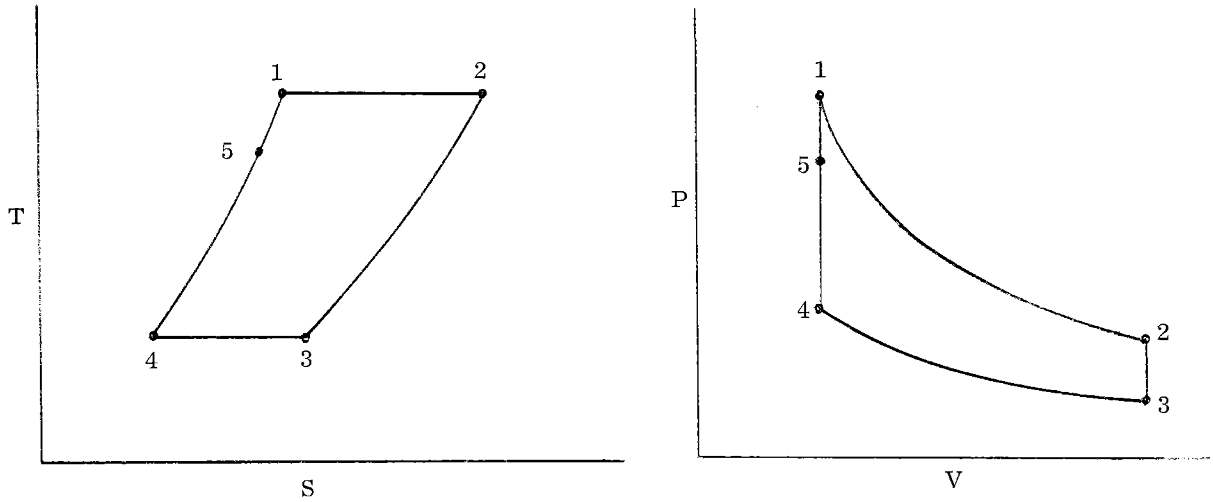


(a) Schematic.



(b) Temperature-entropy diagram.

Figure A-6. - Regenerated open Brayton cycle.



(a) Temperature-entropy diagram. (b) Pressure-volume diagram.

Figure A-7. - Ideal Stirling cycle.

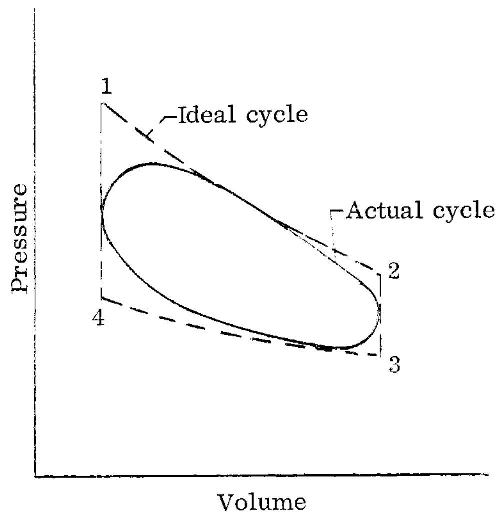
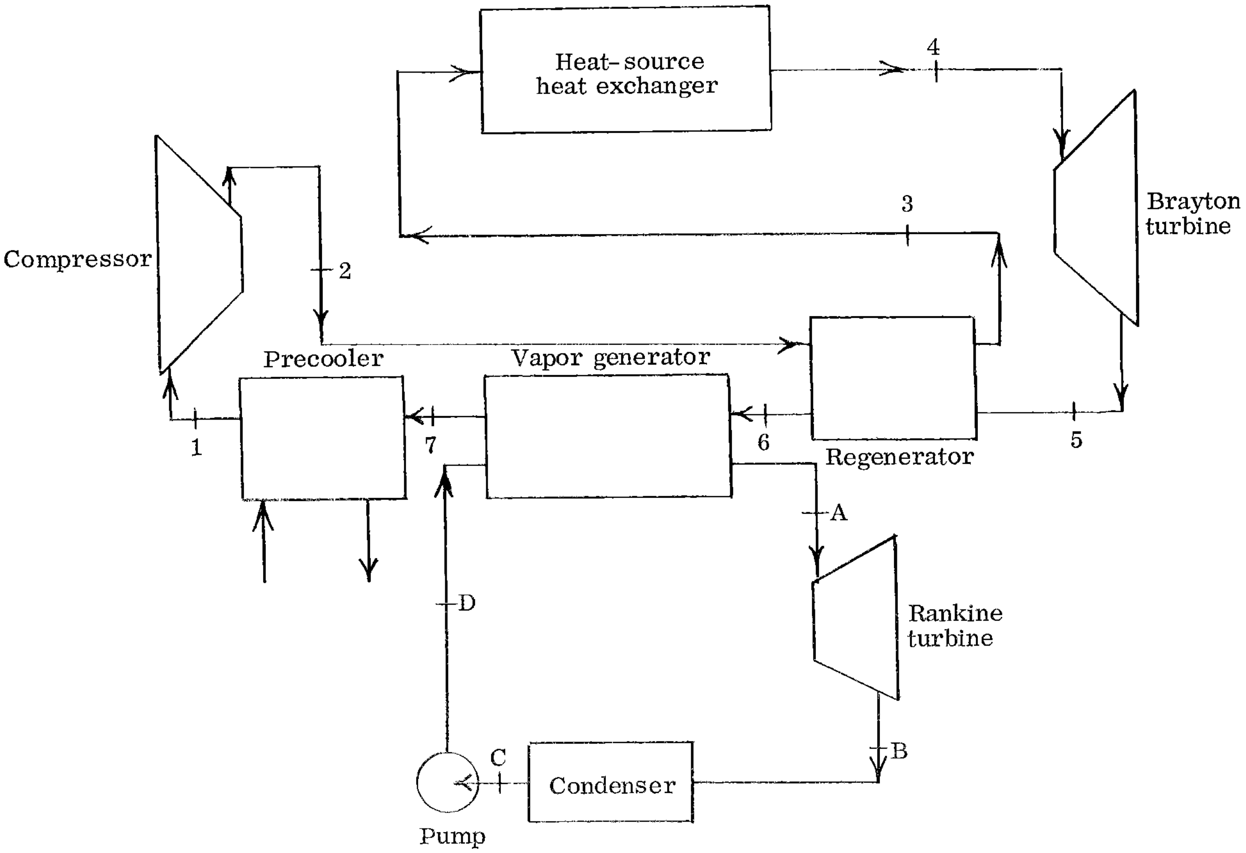
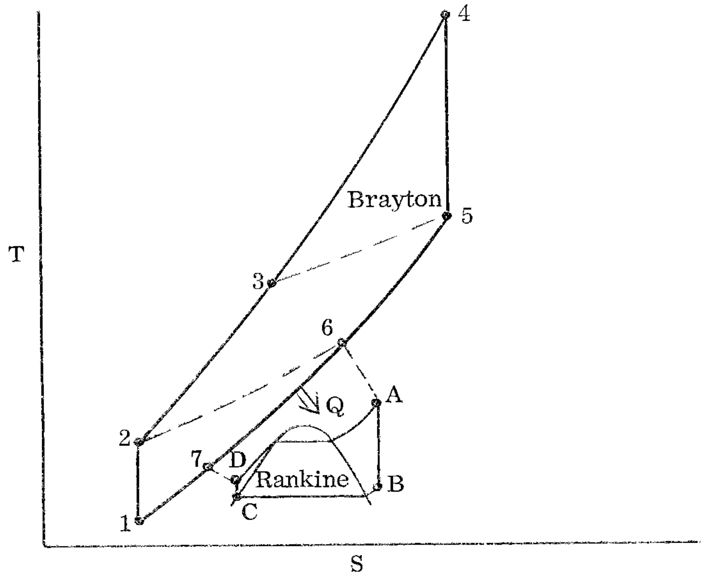


Figure A-8. - Real and ideal Stirling cycles.



(a) Schematic.



(b) Temperature-entropy diagram.

Figure A-9. - Combined Brayton-Rankine cycle.

APPENDIX B

COMMERCIAL SOURCES OF POWER-SYSTEM COMPONENTS

STEAM TURBINES

Carling Turbine Blower Co.
Worcester, Mass. 01610
(5 to 500 hp)

Coppus Engineering Corp.
Worcester, Mass. 01610
(1.1 to 4000 hp)

Elliot Co.
Div. of Carrier Corp.
Jeannette, Pa. 15644
(8 to 70 000 hp+)

General Electric Co.
Large Steam Turbine and Generator Dept.
Schenectady, N. Y. 12345
(68 000 to 134 000 hp)

General Electric Co.
Mechanical Drive Turbine Dept.
Fitchburg, Mass. 01420
(400 to 5000 hp)

General Electric Co.
Medium Steam Turbine Div.
Lynn, Mass. 01905
(5000 to 60 000 hp)

Skinner Engine Co.
Div. of Banner Industries
Erie, Pa. 16512
(10 to 5000 hp)

Terry Steam Turbine Co.
Lamberton Rd.
Windsor, Conn. 06095
(6 to 14 000 hp)

Thermo-Electron Corp.
101 First Ave.
Waltham, Mass. 02154
(350 to 14 000 hp)

Trane Co.
Murray Steam Turbine Div.
LaCrosse, Wis. 54601
(100 to 15 000 hp)

Turbodyne Corp.
Wellsville, N. Y. 14895
(8 to 69 000 hp)

Westinghouse Canada, Ltd.
Box 510
Hamilton, Ontario, Canada L8N3K2
(100 to 80 000 hp)

ORGANIC TURBINES

Barber-Nichols Engineering
6325 West 55th Ave.
Arvada, Colo. 80002
(2 to 86 hp)

Thermo-Electron Corp.
101 First Ave.
Waltham, Mass. 02154
(47 to 650 hp)

Sundstrand Energy Systems
4747 Harrison Ave.
Rockford, Ill. 61101
(10.7 to 1077 hp)

RECIPROCATING STEAM ENGINES

Jay Carter Enterprises, Inc.
P.O. Box 684
Burkburnett, Tex. 76354
(15 to 75 hp)

Skinner Engine Co.
Div. of Banner Industries
Erie, Pa. 16512
(8 to 3000 hp)

Roy P. Ferrier
5737 Venice Blvd.
Los Angeles, Calif. 90019
(5 to 25 hp)

WATER-COOLED CONDENSERS

Ambassador Standard Co.
A Division of Space Dynamics Corp.
P.O. Box 42334-T
Cincinnati, Ohio 45242

American Standard, Inc.
Heat Transfer Division
175 Standard Pkwy.
Buffalo, N. Y. 14227

Amdale Co.
143 E. Hancock Rd.
Lansdale, Pa. 19446

Atlas Industrial Manufacturing Co.
80 Somerset Pl.
Clifton, N. J. 07012

Berdell Industries, Inc.
8-18 43rd Ave.
Long Island City, N. Y. 11101

Engineers & Fabricators Co.
3501 W. 11th St.
P.O. Box 7395
Houston, Tex. 77008

Graham Manufacturing Co., Inc.
Dept. G
170 Great Neck Rd.
Great Neck, N. Y. 11201

Harris Thermal Transfer Products, Inc.
P.O. Box 339-T
Tualatin, Oreg. 97062

Industrial Process Engineers
428 French Rd.
Buffalo, N. Y. 14224

Industrial Systems Corp.
1023 Lake Rd.
Medina, Ohio 44256

ITT
Bell and Gossett
Fluid Handling Division
8200 N. Austin Ave.
Morton Grove, Ill. 60053

Joseph Oats and Sons, Inc.
236 Quarry St.
Philadelphia, Pa. 19106

KAM Products Corp.
98-21T 97th Ave.
Ozone Park, N. Y. 14416

KEMCO
Krueger Engineering & Manufacturing Co.
P.O. Box 11308
Houston, Tex. 77016

Manning & Lewis Engineering Co.
679 Rahway Ave.
Union, N. J. 07083

Modine Manufacturing Co.
1500 Dekoven Ave.
Racine, Wis. 53401

Nooter Corp.
1442 S. Third St.
St. Louis, Mo. 63166

Patterson Kelly Co.
Division of Harsco Corp.
115 Burson St.
East Stroudsburg, Pa. 18301

Pemco
A Subsidiary of Ecolaire, Inc.
York St. at Dowd Ave.
Elizabeth, N. J. 07201

Thermxchanger, Inc.
760-62 98th Ave.
Oakland, Calif. 94603

The Trane Co.
3600 Pammel Creek Rd.
LaCrosse, Wis. 54601

Vulcan Manufacturing Co.
P. O. Box 46465 T
Cincinnati, Ohio 45246

AIR-COOLED CONDENSERS

Aerofilm Corp.
4623 Murray Pl.
Lynchburg, Va. 24505

Air-X-Systems
Division of Harsco Corp.
1402 East Haskell St.
Tulsa, Okla 74016

Ambassador Standard Co.
A Division of Space Dynamics Corp.
P. O. Box 42334-T
Cincinnati, Ohio 45242

Armstrong Engineering Associates, Inc.
P. O. Box 566T
West Chester, Pa. 19380

Dunham-Bush, Inc.
176 South St.
West Hartford, Conn. 06110

Ecodyne
MRM Division
607 First St., S. W.
Massillon, Ohio 44646

Engineers & Fabricators Co.
3501 W. 11th St.
P. O. Box 7395
Houston, Tex. 77008

Graham Manufacturing Co., Inc.
Dept. G
170 Great Neck Rd.
Great Neck, N. Y. 11201

Hydro Dyne Co.
Wetmore & 3rd St.
Massillon, Ohio 44646

Modine Manufacturing Co.
1500 Dekoven Ave.
Racine, Wis. 53401

Therma Technology, Inc.
P. O. Box 2739
Tulsa, Okla. 74101

Vos Finned Tube Products
957 Lake Rd.
Medina, Ohio 44256

RECIPROCATING POSITIVE-DISPLACEMENT BOILER FEED PUMPS

FMC Corp.
Agricultural Machinery Division
5601 East Highland Dr.
Jonesboro, Ark. 72401

Ingersoll-Rand
Standard Pump-Aldrich Division
Allentown, Pa. 18105

Milton Roy Co.
203 Ivyland Rd.
Ivyland, Pa. 18974

Union Pump Co.
87 Capital Ave.
Battle Creek, Mich. 49016

Worthington Pump Corp.
401 Worthington Ave.
Harrison, N. J. 07029

CENTRIFUGAL BOILER FEED PUMPS

Aurora Pump
A Unit of General Signal
800 Airport Rd.
North Aurora, Ill. 60542

Byron Jackson Pump Division
P.O. Box 2017
2300 Vernon Ave.
Los Angeles, Calif. 90058

Carver Pump Co.
1056 Hershey Ave.
Muscatine, Iowa 52761

Chempump
Division Crane Co.
Dept. TR
Warrington Industrial Park
Warrington, Pa. 18976

Colt Industries
Fairbanks Morse Pump Division
3601 Kansas Ave.
Kansas City, Kans. 66110

Delaval Turbine, Inc.
Pump Division
P.O. Box 250
Trenton, N. J. 08602

Frederick Iron & Steel, Inc.
701 East St.
Frederick, Md. 21701

Goulds Pumps, Inc.
240 Fall St.
Seneca Falls, N. Y. 13148

Ingersoll-Rand
Centrifugal Pump Division
Woodcliff, N. J. 07675

Lawrence Pumps, Inc.
363 Market St.
Lawrence, Mass. 01843

Pacific Pump Division
Dresser Industries, Inc.
5715 Bickett St.
Huntington Park, Calif. 90255

Peerless Pump
An Indiana Head Co.
2005 Northwestern Ave.
Indianapolis, Ind. 46206

Sundstrand Fluid Handling Div.
14845 West 64th St.
P.O. Box FH
Arvada, Colo. 80004

Union Pump Co.
87 Capital Ave., S. W.
Battle Creek, Mich. 49016

Warren Pumps, Inc.
Subsidiary of Houdaille Industries, Inc.
Warren, Mass. 01083

Weinman Pump Co.
L.F.E. Fluids Control Division
110 Skiff St.
Hamden, Conn. 06514

Worthington Pumps Corp.
401 Worthington Ave.
Harrison, N. J. 07029

TURBINE REGENERATIVE BOILER FEED PUMPS

Aurora Pump
A Unit of General Signal
800 Airport Rd.
North Aurora, Ill. 60542

FMC Corp.
Coffin Turbo Pump
326 South Dean St.
Englewood, N. J. 07631

MTH Tool Co., Inc.
101 South Ben St.
Plano, Ill. 60545

Roth Pump Co.
Subsidiary Roy E. Roth Co.
P.O. Box 910
Rock Island, Ill. 61201

GAS TURBINE ENGINES

AiResearch Manufacturing Co. of Arizona
Div. of Garrett Corp.
111 South 34th St.
P.O. Box 5217
Phoenix, Ariz. 85010
(11 to 5800 hp)

AVCO Lycoming Division
Stratford Div.
550 South Main St.
Stratford, Conn. 06497
(590 to 3900 hp)

Detroit Diesel Allison
Div. of General Motors Corp.
Indianapolis Operations
P.O. Box 894
Indianapolis, Ind. 46206
(310 to 6445 hp)

General Electric Co.
Gas Turbine Div.
Schenectady, N. Y. 12345
(5050 to 127 000 hp)

Pratt & Whitney Aircraft of Canada, Ltd.
Industrial and Marine Div.
Box 10
Longueuil, Quebec, Canada J4K4X9
(569 to 1850 hp)

Solar
Div. of International Harvester
P.O. Box 80966
San Diego, Calif. 92138
(28 to 10 000 hp)

United Technologies
Power Systems Div.
Farmington, Conn. 06032
(31 000 to 46 000 hp)

Westinghouse Canada, Ltd.
Box 510
Hamilton, Ontario, Canada L8N3K2
(4480 to 39 000 hp)

Westinghouse Electric Corp.
Generation Systems Div.
Lester Branch Box 9175
Philadelphia, Pa. 19113
(51 000 to 125 000 hp)

Williams Research Corp.
2280 West Maple Rd.
Walled Lake, Mich. 48088
(14 to 1500 hp)

SPEED REDUCERS

Cincinnati Gear Co.
Wooster Pike and Mariemont Ave.
Cincinnati, Ohio 45227

Elliot Co.
Div. of Carrier Corp.
Jeannette, Pa. 15644

Lufkin Industries, Inc.
6610 Harwin
Lufkin, Tex. 75901

Philadelphia Gear Corp.
181 South Gulph Rd.
King of Prussia, Pa. 19406

Terry Steam Turbine Co.
Lamberton Rd.
Windsor, Conn. 06095

Turbodyne Corp.
Wellsville, N. Y. 14895

Western Gear Corp.
Power Transmission Div.
2600 East Imperial Highway
Lynwood, Calif. 90262

ALTERNATING-CURRENT GENERATORS

Beloit Power Systems, Inc.
555 Lawton Ave.
Beloit, Wis. 53511
(150 to 500 kW)

Delco Products Div.
Div. of General Motors Corp.
P.O. Box 1042
Dayton, Ohio 45401
(10 to 1250 kW)

Electric Machinery Mfg. Co.
Div. of Turbodyne Corp.
800 Central Ave.
Minneapolis, Minn. 55413
(50 to 50 000 kW)

Fidelity Electric Co.
Lancaster, Pa. 17604
(7.5 to 40 kW)

General Electric Co.
Large Generator and Motor Dept.
Schenectady, N. Y. 12345
(300 to 100 000 kW+)

General Electric Co.
Small A. C. Motor and Generator Dept.
Nashville, Tenn.
(10 to 300 kW)

Kato Engineering Co.
1415 First Ave.
Mankato, Minn. 56001
(5 to 5000 kW)

Lima Electric Co., Inc.
200 East Chapman Rd.
Lima, Ohio 45802
(7.5 to 400 kW)

Marathon Electric
Wausau, Wis. 54401
(50 to 100 kW)

Onan
1400 73rd Ave., N. E.
Minneapolis, Min. 55432
(6 to 20 kW)

Westinghouse Electric Corp.
Generator Sales, 3N46
700 Braddock Ave.
East Pittsburgh, Pa. 15112
(30 000 to 100 000 kW+)

APPENDIX C

CRITERIA FOR COMPONENT SELECTION AND SIZING

This handbook is intended to provide a data base for developed and commercially available power-conversion-system components in the system output power range of 5 to 50 000 kilowatts. Because this range is very broad and covers a large number of applications, it is divided into three subranges. Specific system output power levels were selected within each subrange. These subranges are shown in table C-1. Components were sized by the respective manufacturers for this output. Data sheets were provided for the component data and costs. The alternating-current generators were sized for each specific output power level. The prime movers were sized for the required input horsepower to the generator by taking into account the generator losses for that power level. Sizing of the prime mover also took into account the power requirement for the speed reducer (including the service factor and losses).

The criteria used for selecting and sizing the Rankine-cycle engines, the reciprocating expansion engines, and the gas turbine engines and related technical and cost data are shown in tables C-2 to C-4. The system support components were sized for the flow rate, pressure, and temperature requirements of the prime movers. The maximum available inlet conditions for the Rankine cycle turbines and the reciprocating expansion engines were 1000 psig and 1000° F. Components were selected on the basis of component pressure and temperature design capability. The highest efficiency (all prime movers) and the lowest steam flow rate (Rankine-cycle prime movers) were emphasized in component selection.

The criteria for selecting water-cooled and air-cooled condensers and boiler feed pumps, and related technical and cost data, are shown in tables C-5 to C-7.

The criteria for selecting speed reducers and related technical and cost data are shown in table C-8.

A four-pole, 1800-rpm alternating-current generator was considered for output powers of 5 to 5000 kilowatts. A direct turbine-driven, two-pole, 3600-rpm alternating-current generator was considered for output powers of 10 000 to 50 000 kilowatts. Technical and cost data on alternating-current generators are given in table C-9.

TABLE C-1. - SELECTED OUTPUT POWER LEVELS FOR CLASSIFICATION OF PRIME MOVERS

Required output from power system		Required output power corrected for ac generator efficiency, hp
kW	hp	
Low-power range, 5 to 100 kW		
5	6.7	8.4
10	13.4	15.8
25	33.5	37.8
50	67	74.2
75	100.5	110
100	134	146
Medium-power range, 100 to 1000 kW		
100	134	146
200	268	287.5
400	536	568
600	804	849
800	1072	1128
1000	1340	1408
High-power range, 1000 to 50 000 kW		
1 000	1 340	1 408
5 000	6 700	6 964
10 000	13 400	13 786
25 000	33 500	34 359
50 000	67 000	68 648

TABLE C-2. - CRITERIA FOR RANKINE-CYCLE TURBINE

SELECTION AND TECHNICAL AND COST DATA

(a) Selection criteria

Application	Electric power generation and mechanical drive
Operating mode, percent of total power	100, 75, 50
Turbine type	Fully condensing, mechanical drive (no reheat or extraction)
Backpressure, in. Hg abs	2
Ratio of steam flow rate to organic-fluid flow rate	Prime consideration
Turbine efficiency	Prime consideration
Turbine-inlet steam pressure, psig	1000 is available. For this application, use highest steam pressure consistent with selected turbine design rating
Turbine-inlet steam temperature, ^{o}F	1000 is available. For this application, use highest steam temperature within design range for best steam flow rate
Turbine speed,[a] rpm	Select on basis of steam-organic flow rate ratio and efficiency
Turbine cost	Show separately from cost of nonintegral speed reducer. Include bedplate for mounting speed reducer and generator. Also include standard accessories. If available, show cost of 1, 100, and 1000 turbines.
Weight	Show weight of turbine and bedplate
Superheat	Use superheat to fullest extent within turbine design temperature rating
Operating mode, hr/day	Up to 10 for 6.7 to 6700 hp; up to 16 for 13 400 to 67 000 hp

[a]For 13 400-, 35 500-, and 67 000-hp ratings show cost for 3600-rpm direct electric generator drive. If available, show cost for a higher speed mechanical component drive (turbine only).

TABLE C-2. - Concluded.

(b) Technical and cost data

Technical and cost data	Turbine output, hp														
	8.4	15.8	38	74	110	146	288	568	849	1128	1408	6964	13 786	34 359	68 648
Inlet steam pressure, psig															
Inlet steam temperature, ^{o}F															
Superheat, deg															
Backpressure, in. Hg abs	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2
Turbine speed, rpm													3600	3600	3600
Steam flow rate, lb/hp · hr:															
At full load															
At 3/4 load															
At 1/2 load															
Turbine efficiency, percent															
Turbine model															
Turbine cost, collars, with standard accessories and bedplate:															
For 1 unit															
For 100 units															
For 1000 units															
Turbine weight with bedplate, lb															
Application, gear-generator drive															

TABLE C-3. - CRITERIA FOR RECIPROCATING STEAM ENGINE SELECTION

AND TECHNICAL AND COST DATA

(a) Selection criteria

Application	Electric power generation and mechanical drive
Operating mode, percent of total power	100, 75, 50
Engine type	Fully condensing, mechanical drive (no reheat or extraction)
Backpressure, in. Hg abs	2
Engine efficiency	Prime consideration
Engine-inlet steam pressure, psig	1000 is available. For this application, use highest steam pressure consistent with selected turbine design rating
Engine-inlet steam temperature, ^{o}F	1000 is available. For this application, use highest steam temperature within design range for best steam and water flow rates
Engine speed,[a] rpm	Select on basis of steam-organic flow rate ratio and efficiency
Engine cost	Show separately from cost of nonintegral speed reducer. Include bedplate for mounting speed reducer and generator. Also include standard accessories.
Weight	Show weight of turbine and bedplate
Superheat	Use superheat to fullest extent within turbine design temperature rating
Operating mode, hr/day	Up to 10 in 6.7- to 6700-hp range

[a]For 13 400-, 35 500-, and 67 000-hp ratings show cost for 3600-rpm direct electric generator drive. If available, show cost for a higher speed mechanical component drive (turbine only).

TABLE C-3. - Concluded.

(b) Technical and cost data

Technical and cost data	Steam engine output, hp											
	8.4	15.8	38	74	110	146	288	568	849	1128	1408	6964
Inlet steam pressure, psig												
Inlet steam temperature, $^{\circ}F$												
Backpressure, in. Hg abs.	2	2	2	2	2	2	2	2	2	2	2	2
Engine shaft speed, rpm												
Mechanical efficiency, percent												
Steam-water flow rate, lb/hp · hr:												
At full load												
At 3/4 load												
At 1/2 load												
Overall engine efficiency, percent												
Engine model												
Engine cost, dollars, with standard accessories and bedplate:												
For 1 unit												
For 100 units												
For 1000 units												
Engine weight with bedplate, lb												
Application, ac generator drive												

TABLE C-4. - CRITERIA FOR GAS-TURBINE-ENGINE SELECTION

AND TECHNICAL AND COST DATA

(a) Selection criteria

Application	Electric power generation and mechanical drive
Operating mode, percent of total power	100, 75, 50
Type of gas turbine	Open simple cycle, open regenerated cycle, closed cycle
Operating mode, hr/day	10 continuous for 6.7 to 6700 hp (5 to 5000 kW); 16 continuous for 13 400 to 67 000 hp (10 000 to 50 000 kW)
Engine rating, hp	Based on natural gas or light distillate fuel at sea level and 59° F, or as otherwise noted by manufacturer
Efficiency, percent	Mechanical shaft output divided by heat input (lower heating value of fuel) for gas turbine with integral speed reducer or for gas turbine only (separate speed reducer). Show efficiency for rated power (100 percent); if available include efficiency at 75- and 50-percent power. If efficiency is available at 1500° F turbine-inlet temperature (derating for some machines), indicate in parentheses.
Listing of gas turbine by output	List gas turbine under design output shaft horsepower (in hp or kW) or under nearest horsepower on data sheet
Gas turbine cost, dollars	Include only cost of gas turbine, speed reducer, and baseplate to accommodate generator. Or only show cost of gas turbine and baseplate to accommodate speed reducer and generator. Do not include costs of external controls, fuel supply system, silencers, filters, external cooling-ventilation system or external engine starting system.
Gas turbine weight, lb	Show weight of gas turbine, speed reducer, and baseplate. Or show weight of gas turbine and baseplate only.
Airflow, lb/sec	Indicate flow rate for design full-power rating. If flow rate is available at 1500° F turbine-inlet temperature, indicate in parentheses.

TABLE C-4. - Concluded.

(b) Technical and cost data

Technical and cost data	Gas turbine output, hp														
	8.4	15.8	38	74	110	146	288	568	849	1128	1408	6964	13 786	34 359	68 648
Gas turbine actual rated shaft output horsepower, hp: For gas turbine with integral speed reducer For gas turbine only (not including losses for separately mounted speed reducer)															
Model															
Status (check one): Production Prototype Developmental															
Type of driven machine	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator	ac generator
Speed of driven machine, rpm	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800
Type of gas turbine (check): Simple open cycle Regenerated open cycle Closed cycle Single shaft Two shaft Lighter weight industrial Stationary Industrial															
Pressure ratio (full load)															

Type of combustor: Can Annular															
Number of combustors															
Turbine-inlet temperature, ^{o}F															
Shaft speed, rpm: Output speed from integral speed reducer Gas turbine speed - input to separately mounted speed reducer															
Efficiency, percent: At full load At 3/4 load At 1/2 load															
Airflow, lb/sec															
Weight, lb, for - Gas turbine with integral speed reducer and baseplate Gas turbine only with base-plate (separately mounted speed reducer)															
Cost, dollars: For gas turbine with integral speed reducer and baseplate 1 unit 100 units 1000 units For gas turbine only with baseplate (separately mounted speed reducer): 1 unit 100 units 1000 units															

TABLE C-5. - CRITERIA FOR WATER-COOLED CONDENSER SELECTION

AND TECHNICAL AND COST DATA

(a) Selection criteria

Application	Rankine-cycle electric-power-generating systems
Cooling source for system output of -	
5 to 200 kW	Lake or river
400 to 50 000 kW	Wet cooling tower
Basis of selection	Summer conditions shown in data sheets (Provisions for winter operation, such as control equipment, should not be considered because of the wide temperature variations with geographical location.)
Fluid being condensed	Steam (Treated water would be used in these power systems, as is commercial practice.)
Cooling fluid	Water
Condenser	Surface shell-and-tube, as is commercial practice, with steam on the shell side and water on the tube side
Steam turbine exhaust pressure, in. Hg abs	2.5 (Continuous vacuum will be maintained, with removal of noncondensibles as applicable.)
Steam turbine exhaust and condensate temperature, $^{\circ}F$	108.7
Subcooling	None
Heat-transfer area	Actual or estimated
Condenser materials and construction	Standard materials and construction, as are consistent with commercial practice, are suitable for this handbook
Cost	List or estimated (Do not include control or ancillary equipment.)
Weight	Actual or estimated
Envelope	Overall condenser dimensions, actual or estimated
Availability	Actual or estimated

TABLE C-5. - Concluded.

(b) Technical and cost data

Technical and cost data	Power system output, kW													
	5	10	25	50	100	200	400	600	800	1000	5000	10 000	25 000	50 000
Steam flow, lb/hr	185	374	817	1317	1723	2592	4998	7216	9362	11 264	47 355	89 609	192 410	377 564
Turbine exhaust steam temperature (at 2.5 in. Hg abs), ^{o}F	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7
Condensate temperature, ^{o}F	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7	108.7
Dry-bulb temperature (entering cooling tower), ^{o}F	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----
Wet-bulb temperature (entering cooling tower), ^{o}F	-----	-----	-----	-----	-----	-----	75	75	75	75	75	75	75	75
Condenser inlet cooling-water temperature, ^{o}F	65 (max)	65 (max)	65 (max)	65 (max)	65 (max)	65 (max)	85	85	85	85	85	85	85	85
Condenser outlet cooling-water temperature, ^{o}F	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Cooling-water flow rate through condenser, gal/min	11	22	48	78	102	153	688	993	1289	1550	6519	12 336	26 488	51 977
Condenser pressure drop (tube side), psi														
Heat-transfer area, ft^2														
Cost, dollars														
Weight, lb														
Envelope (length × height × width), ft														
Availability, weeks														

TABLE C-6. - CRITERIA FOR AIR-COOLED CONDENSER SELECTION AND

TECHNICAL AND COST DATA

(a) Selection criteria

Application	Rankine-cycle electric-power-generating systems
Application area	Sites where water is not available for condenser cooling
Basis of selection	Summer conditions shown in data sheets (Provisions for winter operation, such as control equipment, should not be considered because of the wide temperature variations with geographical location.)
Fluid being condensed	Steam (Treated water would be used in these power systems, as is commercial practice.)
Cooling fluid	Air
Condenser	Direct air cooled
Steam turbine exhaust pressure, in. Hg abs	3.5 (Continuous vacuum will be maintained, with removal of noncondensibles.)
Steam turbine exhaust and condensate temperature, ^{o}F . . .	120.5
Available approach temperature, ^{o}F . . .	25.5
Subcooling	None
Condenser materials and construction . . .	Standard materials and construction, as are consistent with commercial practice, are suitable
Cost	List or estimated (Include fan-motor assembly but do not include control equipment.)
Weight	Actual or estimated weight of heat exchanger and fan-motor assembly
Availability	Actual or estimated

TABLE C-6. - Concluded.

(b) Technical and cost data

Technical and cost data	Power system output, kW													
	5	10	25	50	100	200	400	600	800	1000	5000	10 000	25 000	50 000
Steam flow, lb/hr	185	374	817	1317	1723	2592	4998	7216	9362	11 264	47 355	89 609	192 410	377 564
Turbine exhaust steam temperature (at 3.5 in. Hg abs), $^{\circ}F$	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5
Condensate temperature, $^{\circ}F$	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5	120.5
Design dry-bulb temperature, $^{\circ}F$	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Heat-transfer area, ft^2														
Airflow, ft^3/min														
Required (total) fan horsepower, hp														
Model designation (as available)														
Cost (heat exchanger and fan-motor assembly), dollars														
Weight (heat exchanger and fan-motor assembly), lb														
Assembly envelope (length × width × height), ft														
Availability, weeks														

TABLE C-7. - CRITERIA FOR BOILER FEED PUMP SELECTION AND

TECHNICAL AND COST DATA

(a) Selection criteria

Application.	Boiler feed service for electric power generation
Liquid being pumped	Clean treated water
Inlet temperature, ^oF	110 to 150
Inlet conditions	Flooded suction (Consider that sufficient NPSH is available to meet pump requirements.)
Efficiency	Efficiency, rather than low cost, is overriding consideration
Total head shown in data sheets	Computed from steam turbine throttle pressure and predicted system pressure and corrected for specific gravity of water at 150^o F
Speed, rpm	Highest speed consistent with commercial practice is preferred (e.g., 3600 rpm)
Cost	List or estimated (Include pump, coupling, and baseplate but do not include motor unless pump and motor are an integral assembly.)
Weight	Actual or estimated (Include pump, coupling, and baseplate but do not include motor unless pump and motor are an integral assembly.)
Operation, hr/day, for pump capacity of -	
≤126 gal/min	10
≥243 gal/min	16
Pump life and reliability	Consistent with electric-power-generating plant practice

TABLE C-7. -

(b) Technical

Technical and cost data	Power system													
	5		10		25		50		75		100		200	
Capacity, gal/min	0.7		0.9		2		4		5.5		6		10.4	
Total head, ft	1767	2062	736	2062	736	2062	736	2062	736	2062	1767	2062	1915	2062
Required net positive suction head, ft														
Required brake horsepower, hp														
Efficiency, percent														
Speed, rpm														
Type: Regenerative Centrifugal Positive displacement														
Cost (pump, coupling, and baseplate), dollars														
Weight (pump, coupling, and baseplate), lb														
Estimated availability, weeks														

Concluded.

and cost data

output, kW															
400		600		800		1000		5000		10 000		25 000		50 000	
15.5		19		27		30		126		243		558		948	
1915	2062	1472	2062	1472	2062	1472	2062	1915	2651	1767	2948	2651	2948	2651	2948

TABLE C-8. - CRITERIA FOR SPEED REDUCER SELECTION AND TECHNICAL AND COST DATA

(a) Selection criteria

Application	Electric power generation
Type of prime mover	Steam and organic turbines (8 to 7000 hp); gas turbine (11 to 37 000 hp)
Operating mode	Uniform and continuous
Ambient air temperature, $^{\circ}F$	15 to 125
Operating mode, hr/day	Up to 10 for 8 to 7000 hp; up to 16 for 7000 to 38 000 hp
Output speed, rpm	1800 for 8 to 7000 hp; 3600 for 7000 to 38 000 hp
Type	Separately mounted between prime mover and ac generator
Construction and accessories	Standard commercial practice for horsepower range

TABLE C-8. - Continued.

(b) Technical and cost data for speed reducer used with steam turbines driving ac generators

Technical and cost data	Input power to speed reducer, hp																					
	8.5	8.5	15.8	38	38	74	74	110	110	146	146	288	288	568	568	849	849	1128	1128	1408	6964	6964
Input speed (turbine speed), rpm	4000	5000	5000	4000	5000	4000	5500	4000	7000	4000	7000	4000	7000	4500	7000	4000	5500	4000	5500	5000	3600	5000
Output speed, rpm	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800
Speed ratio	2.22	2.77	2.77	2.22	2.77	2.2	3.05	2.22	3.88	2.2	3.8	2.22	3.88	2.5	3.88	2.22	3.05	2.22	3.05	2.77	2	2.77
Service factor																						
Efficiency, percent																						
Model																						
Output torque, ft-lb																						
Weight, lb																						
Cost, dollars:																						
For 1 unit																						
For 100 units																						
For 1000 units																						
Availability, weeks																						
Operating mode, hr/day	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10

TABLE C-8. - Concluded.

(c) Technical and cost data for speed reducer used with gas turbines driving ac generator

Technical and cost data	Input power to speed reducer, hp																					
	85	145	150	200	370	590	1500	2500	2880	3450	3900	4380	5050	6445	13 750	14 600	22 158	25 000	25 200	26 250	30 576	33 550
Input speed (turbine speed), rpm	80 000	59 200	4800	4800	6584	9265	6000	13 730	18 000	14 630	14 630	13 820	10 290	11 500	6500	6500	4690	4860	4670	4670	4670	4670
Output speed, rpm	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	3600	3600	3600	3600	3600	3600	3600	3600
Speed ratio	44.4	32.9	2.66	2.66	3.66	5.15	3.33	7.63	10.0	8.14	8.12	7.67	5.77	3.58	1.80	1.8	1.3	1.35	1.29	1.29	1.29	1.29
Service factor																						
Efficiency, percent																						
Model																						
Output torque, ft-lb																						
Weight, lb																						
Cost, dollars:																						
For 1 unit																						
For 100 units																						
For 1000 units																						
Availability, weeks																						
Operating mode, hr/day	10	10	10	10	10	10	10	10	10	10	10	10	10	10	16	16	16	16	16	16	16	16

TABLE C-9. - TECHNICAL AND COST DATA FOR ALTERNATING-CURRENT GENERATORS

Technical and cost data	Generator output, kW														
	5	10	25	50	75	100	200	400	600	800	1000	5000	10 000	25 000	50 000
Speed, rpm	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	3600	3600	3600
Phase	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Number of cycles	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
Ambient temperature, °C	40	40	40	40	40	40	40	40	40	40	40	40	40	40	40
Temperature rise, °C	80	80	80	80	80	80	80	80	80	80	80	80	[a]80	[a]80	[a]80
Voltage, V	480/277	480/277	480/277	480/277	480/277	480/277	480/277	4160	4160	4160	4160	4160	13 800/4160	13 800	13 800
Power factor	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
Enclosure	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof	Drip proof[a]	Drip proof[a]	Drip proof[a]
Insulation class	F	F	F	F	F	F	F	F	F	F	F	F	[a]F	[a]F	[a]F
Efficiency, percent:															
At full load															
At 3/4 load															
At 1/2 load															
Weight, lb															
Cost,[b] dollars:															
For 1 unit															
For 100 units															
For 1000 units															
Frame size															

[a]Commercial practice.

[b]List or base price with standard equipment (e.g., with exciter, voltage regulator, and cooling).

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16. Abstract

A survey of developed and commercially available heat-engine components applicable for solar thermal power systems was conducted, the data were compiled, and a handbook was prepared for the Department of Energy, Division of Central Solar Technology, by the NASA Lewis Research Center. This work was conducted under Interagency Agreement EX-76-A-29-1060. Design, performance, and cost data were provided by the respective manufacturers on steam turbines, reciprocating expansion engines, condensers, pumps, gas turbines, speed reducers, and ac generators for use in Rankine- and Brayton-cycle power-generating systems. Components were selected for specific power levels from 5- to 50 000-kWe system output. Development status of the Stirling engine is included. The data presented in this handbook identify component design ranges and performance characteristics at specific power levels of developed and commercially available components.

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