UNCLASSIFIED

# D-420

## EDQP STUDY PAPER

# PROPULSION SYSTEM DESIGN

D

420

A

## CATEGORY:

DESIGN-D

CONCEPT/PRELIMINARY

DESIGN-4

REV. A

6/80

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ENGINEERING DUTY OFFICER
QUALIFICATION PROGRAM (EDQP)

UNCLASSIFIED

**Lesson Topic.** Propulsion System Design D-420 Rev. A

**TIme.** 2 hours

**Instructional Materials.**

1. References:
   a. NAVSEAINST 9060.1 (6 May 1975) "Top Level Specifications (TLS) for New Ship Design."
   b. Ship Design Division Technical Practices, dated 1 June 1965.
   c. George C. Sharp, Inc., "Recommended Practices for Preparing Heat Balances for U.S. Navy Combatant and Auxiliary Type Ships", Report No. 5353, dated 29 May 1969.
   d. SNAME, "Marine Steam Power Plant Heat Balance Practices", T&R Bulletin 3-11, Reprinted July 1975.
   e. Propulsion Systems Analysis Branch Technical Report No. 6144E-76-136, January 1976.
   f. Propulsion System Related Design Data Sheets.

**Terminal Objective.** Complete the EDQP

**Enabling Objective.** To obtain an understanding of the elements comprising the propulsion system of a Naval surface ship, the varying types of propulsion systems, and the design process that is presently used to develop the propulsion system for a surface ship.

**Study Method.**
The enclosed study paper contains the descriptive material required to obtain overall understanding of the propulsion system of a ship and the design process by which it is developed. The referenced source material also provides more detailed information and background on the subject. A lecture will be given to supplement the paper.

**Practical Factors.** See Attached.

**Questions.** See Attached.

## 1. Introduction

The propulsion system is the major component of the ship's MOBILITY system. This study paper will describe the design process and procedures used in developing the propulsion system for a conventional fossil fuel Naval surface ship. The remaining component of MOBILITY, ship control, is not discussed in this paper. The design development of nuclear powered surface ships is not specifically addressed, although the design procedure and engineering analyses conducted are quite similar to fossil fuel propulsion systems.

The propulsion system is designed to produce sufficient propulsive thrust to propel the ship at a maximum design speed. This design speed can be expressed in ship trial conditions, calm sea and clean hull bottom, or in varying sea state and years out of drydock (assumed hull fouling conditions). The propulsion system will permit continuous ship operation at all speeds throughout its complete operating range from design speed ahead to design speed astern. The design of the system will permit the ship to independently transit or escort other forces for defined distances (nautical miles) and speeds (knots). This is normally expressed as an endurance in miles at a prescribed endurance speed.

The propulsion system will provide the capability for the ship to accelerate, decelerate, and stop from varying speed conditions. At a minimum, these control requirements will be expressed as ship stopping distance (ship lengths) from full power design speed. In addition to its primary function, the propulsion system directly or indirectly provides energy to the combat/mission, ship and human support elements of the ship.

This paper will discuss and explain the design process from conceptual to contract design. Detail design, the initial phase of ship production conducted by the ship builder is not discussed in this paper.

## 2. Propulsion System Elements and Applications.

The propulsion system is composed of three basic sub-elements/systems; the heat energy source and prime mover, the transmission system, and the propulsor. These elements are supported by a propulsion control system, and a variety of propulsion auxiliaries, including air intake and exhaust, steam and feed piping (as applicable), lube oil, and S.W. cooling (sub)systems.

There are three basic marine propulsion systems used in naval and merchant ships; steam, diesel and gas turbine. Potentially there exist a multitude of combined plants incorporating any or all of these basic plants. In practice, only combinations of diesel and gas turbine have effectively been utilized to the present.

Steam propulsion systems were the primary plants used in all major Naval surface displacement ships through 1970. Since the early 1950's, there have essentially been two types of steam plants, one for combatant and the other auxiliary type ships. For combatants (DE, DD, DLG, CVA), a simple rankine cycle with steam conditions of 1200 psig and 950°F was used. The major propulsion auxiliaries were steam turbine driven because of their size (power) and to provide the rapid response rate and control required throughout the ship's operating speed range. These plants were designed to provide good partload efficiency and nominal loading of the main boilers at the actual ship speeds where the greatest percentage of operation occurs. This is nominally 25 percent of the installed power.

Auxiliary type ships, tenders, and the majority of the amphibious assault fleet, in general, use a regenerative rankine cycle with steam conditions of 600 psig and 850°F. These ships which operate close to their design speed are similar to merchant plants, using motor driven propulsion auxiliaries and two stages of feed heating. At present this is the only type of Naval surface ship that uses conventional steam propulsion systems, almost exclusively.

Diesel propulsion systems, while the major power plant world wide for merchant ships, has been limited to small HP applications in the U.S. Navy. These include a multitude of service craft, patrol craft, and minesweepers and high speed/performance ships, as secondary propulsion units. Diesel engines are now being seriously considered for the next class of amphibious assault ships. This undoubtedly will be true for all auxiliary type ships in the future, because of the inherent efficiency of the diesel engine.

Prior to 1970, gas turbine propulsion systems were primarily used in high speed ships and craft, including hydrofoils, ACV's, SES and planing craft. They are now the propulsion plant for all new Destroyer type ships (DD, FFG, DDG, DDH). The prime mover in all these applications is the marinized, second generation aircraft derivative engine.

Steam turbine prime movers and their associated steam generating boilers are available in the range from approximately 10,000 to 70,000 SHP per shaft. Steam plants can be designed at any discreet power in that range without normally requiring a development program. Available diesel engine plants, produced in this country would span the range from several hundred to approximately 30,000 BHP per shaft at the present time. The aircraft derivation gas turbine engines are available at very discreet BHP levels. AT present there is only one second generation marinized engine in service, at nominally 20,000 BHP. Two additional engines at 5000 and approximately 40,000 BHP are under development. By 1980, gas turbine propulsion engines would be available in the range from 5000 to 80,000 BHP per shaft.

The transmission system for surface ships normally consists of reduction gearing, shafting and bearings. The reduction gears are of the parallel axis type, normally double reduction locked train using through hardened helical gearing. Shaft reversal for gas turbine and existing diesel plants has been accomplished by the use of controllable pitch (CP) propellers. Future plants may well use reverse reduction gears which are now now under development.

The propulsor for surface ships consists of the conventional open water marine propeller, either fixed pitch or CP. High speed ships, hydrofoils and SES will normally use supercavitating propellers or waterjet propulsion systems. Air screws, either open or ducted will power ACV's. The transmissions for these ships and craft will normally be of the carburized or nitrided type, to reduce size and weight, and consist of spiral bevel, and epicyclic gearing as a function of the location of prime mover and propulsor. While these ship types will not be specifically discussed, in general the design procedures and engineering analyses conducted are quite similar to surface displacement ships.

### 3. Conceptual Design

It is during the conceptual design of the ship that an actual ship is defined. In this phase a feasibility study is conducted and a concept design is formulated. The feasibility study can be defined as an estimate of the ship system level physical characteristics and cost related data for a design which represents a feasible solution to a specific set of performance requirements. During this stage of design the basic overall characteristics of the propulsion system are defined.

Based on weapon threat and mission analyses, an operational requirement for a ship system will have been generated. The initial phase of feasibility study involves an iterative process, at the ship system level between CNO/OPNAV and NAVMAT/NAVSEA, to determine the trade-offs between ship performance capability and cost. The ship system performance requirements can be met by many feasible ship designs. To provide the range of performance capabilities and costs, in a reasonable time, a design procedure must be used that will permit the generation of many feasible designs in which the ship system level physical characteristics are systematically varied. This is accomplished through the use of a ship synthesis model, which converts a set of performance requirements into physical characteristics of a ship. The ship synthesis model is developed by providing estimating relationships of the physical characteristics (weight, space, location, efficiency, varying energy requirements) for the various subsystems that comprise the ship. These estimating relationships are developed from actual ships applications, design practices and procedures, and developed engineering design data. The DD07 Destroyer program is one example of a developed ship synthesis model that has been extensively used by the U.S. Navy in the conceptual stage of Destroyer type ships. For major ship programs, as many as 1000 feasibility studies may be developed before a conceptual design is selected.

The basic propulsion system is normally one of the major trade-offs that is extensively evaluated at this stage of design. For a given set of performance (design speed(s), endurance speed and range) and mission (survivability, vulnerability, detectability) requirements, a wide range of propulsion system alternatives will be investigated. These will normally include basic plant types, steam turbine, gas turbine, combined plants including diesels; the number of plants, single or multi-screw; plant location aft, amid ships and separation; and manning/maintenance policies. Based on input requirements, the program will provide physical characteristics of the ships and cost data. The propulsion system characteristics generated would then be criticially examined by an experienced design engineer to verify that they are feasible based on defined design procedures, and to determine if additional plant trade-offs or optimization studies should be conducted.

Based on the results provided, a ship and propulsion system configuration would then be selected to meet the basic requirements. At this stage, the draft of the Top Level Requirements (TLR), Reference (a), would be initiated and the concept design developed. The TLR will define the complete ship system operational requirements and overall design philosophy and constraints (costs, availability, etc.). In development of the concept design, the basic characteristics of the propulsive to meet system requirements, (speed, endurance, maneuvering) will be validated by additional engineering analysis of the major elements of the system.

### 4. Preliminary Design

At the preliminary design stage, the feasibility of the concept design is validated, and subsystems of the propulsion system developed and optimized. During preliminary design the TLR is completed, defining the complete requirements and definitive constraints, and the resultant ship system characteristics. This essentially produces a contract between CNO/OPNAV and NAVMAT/NAVSEA for the development of the ship system, subject to continuing verification and review (DSARC).

The engineering analyses conducted during this design phase for the propulsion system include the following:

- Overall Power Plant Analysis
- Machinery Arrangements
- Propeller Parametric Analysis
- Machinery and Hull Vibration Analysis
- Manning Studies
- Machinery Reliability Analysis
- Ship and Machinery Plant Maneuvering Analysis

The TLR delineates the operational requirements, speed time profiles (wartime and peacetime), logistic support (maintenance philosophy), readiness, and environment conditions that provide the basis for these engineering analysis.

In the overall power plant analysis, the initial rating and sizing of all elements of the propulsion system, including propulsion auxiliaries, are determined. For a steam turbine system, this takes the form of heat balance and flow diagrams which develop the capability of the propulsion system at design and off design conditions, and throughout the complete speed range of the ship. Reference (b) provides technical guidance on past design practices, as the baseline for propulsion system design. Heat balances are prepared in accordance with the procedures outlined in references (c) and (d). In considering all energy sources of the ship, they provide directly the basis for ship fuel consumption rates, and the sizing of supporting auxiliary systems. Based on the developed ratings and sizing of subsystems, weight of these elements are then developed.

For gas turbine and diesel prime movers, the overall analysis incorporates specific analyses of the propulsion system, ships service electric plant, and auxiliary energy requirements in developing ratings of equipment, and ship fuel consumption rates. Evaluation of energy sources for auxiliary subsystems will normally include waste heat rejection from both prime mover and ship service generator engines, as well as independent sources.

Machinery arrangement studies and drawings will be developed to determine space and total ship volume requirements for the propulsion system, propulsion auxiliaries and control spaces, and independent auxiliaries. The developed arrangements will provide for installation and operation of all machinery throughout its normal range, accessibility for inspection and maintenance of equipment, and to permit removal of machinery for overhaul. With the recent changes in logistic support and maintenance and manning policies, machinery arrangements must provide access routes and handling equipment for the rapid removal of most elements of the propulsion system.

The propeller parametric analysis determine the characteristics of the transmission and propulsor elements, optimizing between minimum acquisition and life cycle costs as noted in reference (e). The basic parameters selected are the propeller diameter and design RPM. Within the constraints of machinery space arrangements and hull/machinery vibration predictions, a range of propeller diameters and RPM's are evalauted. At the design (full power) speed the acceptable range of diameter and RPM are determined based on efficiency, propulsive coefficient (P.C.) and installed SHP, and cavitation limits, thrust breakdown and extent of cavitation. At the endurance speed, propeller performance (P.C.) is determined for the acceptable range of parameters, using propeller series data and lifting line calculations. In this range of parameters, the sizes and weights of transmission elements, reduction gear shafting and bearings are calculated, as well as the endurance fuel required. The condition of least total weight of machinery and fuel (LTWM&F) represents the optimum condition for the ship. Overall considerations of energy conservation, as well as the price of fuel will result in the future selection of basic parameters (lower RPM) approaching the minimum limits set by vibration and machinery space arrangements.

Preliminary hull and machinery vibration studies will be conducted based on the nominal propeller characteristics to determine that both the machinery and hull can operate throughout the ship's speed range without experiencing dynamic stresses that would damage these systems or deteriorate their future performance capabilities.

Manning studies for the propulsion system will determine the minimum personnel required to control and operate the plant as well as provide the organizational maintenance required to achieve system availability. Logistic support philosophy, planned maintenance procedures, crew training, and the degree of automation, ultimately determine the minimum required manning levels.

Using historical data from actual ship operations and engineering analysis for new equipments, the reliability data, mean time between failure (MTBF), for all elements of the machinery plant will be developed. Using the mission scenarios and overall system availabilities in the TLR, an assessment of machinery reliability and maintainability will be conducted. To meet specific mission requirements additional redundancy may well be considered. The achievement of overall system availablity requires the use and design of inherently reliable propulsion elements or the development of improved performance (product improvement programs).

Dynamic analysis of the ship/propulsion system is initially conducted to verify the maneuvering requirements and determine the transient loading of elements of the transmission. As the ship design proceeds the results of this analysis also provide the input and requirements for the design development of the propulsion control sub-system. Where unique maneuvering requirements are a necessary function of the ship, the results of dynamic analysis will determine the design requirements for the propulsion system as well as the control sub-system and the response rates of its elements.

At the conculsion of ship preliminary design a Top Level Specification (TLS), reference (a), will have been prepared. The TLS will describe and embody propulsion system and subsystem characteristics and capabilities that meet the defined requirements of the TLR.

## 5. Contract Design

Contract design is the last stage of system and subsystem development and performance verification, prior to the award of a contract for ship construction. The output of this design stage are complete comprehensive specifications and drawings, that define all elements of the propulsion system to enable a prospective shipbuilder to estimate machinery costs, and the requirements for detailed design verification and construction of the system as an integral part of the ship.

During the initial stage of contract design, ship model powering tests will be conducted with a stock propeller validating basic propulsion system performance. The results of these tests will provide the basis for the hydrodynamic design of the propeller. Depending on the changes that have occurred during the deisgn process, and the margin policy adopted at the initiation of the program, the tolerance on required propeller performance (efficiency and cavitation) may be reasonable or quite small. The selection of the hydrodynamic characteristics for the final design of the propeller will depend on this performance tolerance, plus the results of machinery and hull vibration analysis. Before contract design is completed, a model of the ship design propeller will be constructed and ship powering tests and cavitation tests conducted to validate ship powering and propeller performance.

Prime mover design performance throughout the ship's operating range will now be verified and these requirements specified. Overall power plant analysis (steam/heat balances) incorporating electric load analysis and auxiliary subsystem energy analyses, or separately for gas turbine and diesel plants, will be completed. Based on these analysis, performance requirements, as necessary, will be developed for incorporation into system, subsystem and component specifications.

Concurrent with the selection of final propeller hydrodynamic parameters the hull and machinery vibration analysis will be accomplished and the propulsion shafting design validated. The final configuration and arrangement of the reduction gear will complete the main elements of the propulsion system.

The major product of the design phase is the contract specification. All the engineering analyses accomplished and proposed solutions, to be meaningful, must be clearly expressed in the ship (contract) specifications. Ship specifications are presently formated in the Ship Work Breakdown Structure (SWBS) classification system. This is the old BuShips Classification Index (BSCI) used for weights and cost data. Group 200 is the propulsion system. These specifications cover all the components and supporting equipment. There are essentially two parts to a specification, invoking a Military Specifications (MIL SPEC) which contains the complete requirements for a generic equipment, and the ship specification which contains the specific requirements pertaining to the actual application. MIL SPEC's contain basic definitions, design guidance, material requirements, normal qualification testing, and quality control/assurance procedures. They essentially represent a state of the art definition of the equipment, embodying the practices and procedures necessary to meet the intended Naval application and requirements. The ship specification will contain specific ratings (HP, RPM, etc....), added ship or shop testing requirements, configuration constraints, plus all unique requirements necessary for the specific ship application. If minimum efficiencies of a particular equipment are necessary, they would be specified here, such as water rates for an auxiliary turbine. In evaluating system availability and reliability, it may well develop that one or several components could present problems in achieving the design requirements. For these equipments, the specification might contain a failure modes and effects analysis (FMEA) to improve reliability and demonstration testing to verify MTBF requirements. Shock, vibration, and noise requirements, in addition to the overall ship specification, may be included for critical equipments or systems.

During contract design many calculations are performed by the Navy to determine that systems are adequately sized and properly rated. Many of the procedures used in these engineering calculations are defined in Design Data Sheets (DDS), reference (f),and provide guidance for the shipbuilder during detail design and construction. These DDS procedures as with MIL SPEC's, are regularly updated as technological, or material improvements occur.

In addition to the ship specifications, machinery (and ship) drawings are prepared, most of them being guidance plans. These drawings show the arrangements and locations of all significant equipments, and incorporate the total weight and moment distribution intended for the ship. For supporting piping systems, individual diagrammatics will be provided as guidance to show the intended system functional arrangement.

It is most probably impossible, even with unlimited resources, and unlimited time to develop a ship specification and drawing package that would not contain discrepancies or conflicts and permit the shipbuilder to complete the detailed design; construct the ship; demonstrate required performance; and deliver it as scheduled, without incorporating change orders. However, this is still the ultimate goal with every ship during the contract design phase.

## Questions Related to Naval Surface Propulsion Systems

1. Diesel propulsion systems to the present have not been generally applied to major Naval surface ships, because
   - (a) The production and development technology for high powered (medium speed) diesel engines has not been developed due to a lack of commercial and marine applications.
   - (b) High weight and space requirements.
   - (c) High airborne and structure borne noise levels.
   - (d) High maintenance requirements.
   - (e) All of the above.

2. The most important product of the ship design process is:
   - (a) The multiple Feasibility Study Data.
   - (b) The Concept Design.
   - (c) Contract Design Specifications and Drawings.

3. The Mobility system of a ship consists of two basic parts, ______________, and ______________.

4. The three basic elements of the propulsion system are the ______________, ______________, and ______________________________.

5. The normal propulsor of a displacement surface is a conventional subcavitating ______________.

6. Waterjet propulsion systems are used on high speed ships and craft because they are more efficient than propellers. TRUE or FALSE.

7. Gas turbines have become the accepted prime movers for Destroyer typue ships, because of,
   - (a) Improved performance of second generation aircraft engines.
   - (b) Reduce manning and onboard maintenance of ship equipment.
   - (c) Elimination of residual based fuels from the Navy logistics system.
   - (d) All of the above.

8. Controllable pitch propellers are used on Naval surface ships because,
   - (a) They permit constant speed operation of the prime movers, and thus improved ship fuel consumption rates.
   - (b) They are more efficient than fixed pitch propellers.
   - (c) They provide thrust reversal capability that cannot effectively be provided by the prime mover.
   - (d) They have better cavitation characteristics than fixed pitch propellers.
   - (e) All of the above.

9. Successful prime movers for U.S. Naval Ships have a broad industrial technological and production base. TRUE or FALSE.

10. Propeller cavitation is a well understood phenomena and can be predicted full scale with great accuracy. TRUE or FALSE.

11. Achievement of higher ship speed,
    - (a) Is always justified at any cost.
    - (b) Is justified if the cost is moderate.
    - (c) Is justified if it supports the overall mission effectiveness of the ship system.

12. Design practices, MIL SPEC's, Design Data Sheets, etc....inhibit innovative design and development of of improved propulsion systems. TRUE or FALSE.

# PRACTICAL FACTORS

Engineering Duty Officer
Qualification Program (EDQP)

## PROPULSION SYSTEM DESIGN

Engineering Analyses Conducted
During the Design Development
of the Propulsion System

## 1.0 Introduction

During the design of a ship's propulsion system there are many engineering analyses conducted. To provide a general background as to the types of analyses conducted, five typical problems have been prepared.

These analyses would normally be initially conducted during the preliminary design stage of the ship design. They provide the basis for engineering and RMA analyses, trade off studies and economic analyses conducted throughout all stages of ship design.

These analyses will be repeated throughout the ship design process as more detailed engineering analyses of the ship and machinery plant subsystems are conducted and component characteristics are selected.

The problems provided are generally applicable to all types of propulsion system prime movers, steam, gas turbine and diesel, with the exception of the first part of section 4, the actual heat balance calculations for steam plants. For internal combustion engine prime movers the overall power plant analyses does not include basic heat balances, since the prime movers are existing engines developed for marine service. The selection of auxiliary subsystems and their component ratings for I.C. engine plants, is similar to that conducted for steam plants; and in lieu of heat balances, separate load analyses are conducted for the auxiliary energy consuming subsystems.

## 2.0 Endurance Fuel Calculation

The endurance fuel calculation procedure for naval surface ships is essentially the same for all marine propulsion systems, steam, diesel and gas turbine. The only difference being the source of fuel consumption data. For steam turbine ships, the overall ship fuel consumption, including all energy consumption sources, is obtained from heat balance calculations. In the case of internal combustion engine plants, the individual fuel consumption rates of each element are determined separately for each energy consuming source, main propulsion, ships service electric, and auxiliary systems.

The calculation procedure for surface ships is contained in a Design Data Sheet, DDS 200-1, which is included in this section. For high speed, high performance ships, hydrofoils, ACV's and Surface Effect Ships, a variation in this procedure is made to allow for the change in ship resistance as fuel is consumed.

**2.1 Endurance Fuel Problem**—For this problem, assume we have a single screw gas turbine plant, consisting of two (2) LM 2500 gas turbine engines geared to a controllable pitch propeller (CPP) to provide astern operation, and overall ship propulsion system maneuverability, acceleration, deceleration and stopping capability. The ship's service electric plant consists of three (3) diesel generator sets each rated at 1000 KW. All ship auxiliary loads for all services are provided by a waste heat recovery system from the D-G's augmented by electric heating units.

**2.2 Ship Input Data and Requirements**—The following requirements and calculated powering and load data have been determined:

- Required Endurance in Nautical Miles = 4000
- Endurance Speed = 18 Knots
- Full Load Displacement = 4000 Tons
- Rated Full Power = 40,000 SHP
- Design Endurance Power = 10,600 SHP
- Transmission System Efficiency = 0.975
- 24 Hour Average Electric Load = 1200 KW
- SSDG Fuel Consumption Rate = 0.6 lbs/KW-hr.
- Tail Pipe Allowance Factor = 0.98
- Reduction Gear Ratio = 18
- Rated Full Power RPM's —
  - CP Propeller = 200
  - Engine Power Turbine = 3600
- Endurance Power Propeller RPM = 120

**2.3 Calculate the Endurance Fuel**—Enclosed is an LM 2500 engine fuel consumption map. Knowing the design endurance SHP, the average engine BHP can be calculated. With the power turbine RPM calculated at the endurance power propeller RPM, the specific fuel consumption rate can be taken from the engine fuel map, and the ship's endurance fuel calculation completed.

# DDS200-1
# CALCULATION OF SURFACE SHIP ENDURANCE FUEL REQUIREMENTS

1 August 1975

## 200-1-a. General

A major consideration in the design of any Naval ship or craft is its ability to meet the endurance requitements established by the Chief of Naval Operations. This Design Data Sheet outlines the procedure followed by NAVSEA to determine the necessary fuel tankage for conventionally powered steam, diesel, or gas turbine propelled ships or craft.

## 200-1-b. Definition of Major Items

1. **Endurance** is the theoretical distance which a ship can run utilizing all of its available fuel (excluding cargo), at a specific speed, ambient air and sea water conditions, in deep water, at full load displacement.
2. **Design endurance power** is the shaft horsepower endurance speed, as indicated by the latest available speed-power curve applicable to the ship or craft. This curve may be either one prepared in the early design stages and based on predicted performance of the ship or craft, or one based on actual self-propelled model basin test results. It normally includes a correlation allowance ($\Delta C_f$) of 0.0005, which is the equivalent of freshly applied vinyl paint on surface ships. While the 0.0005 value is a reasonable approximation for the majority of endurance calculations, this factor is not a constant applicable to all designs. Should a bottom paint such as hot plastic ($\Delta C_f = 0.0009$) be used, a correction must be applied to allow for the increased roughness. Appendix A contains an accepted method for determining this correction.
3. **Average endurance power** is the design endurance power increased by 10 percent. This increase is an allowance for adverse sea conditions and average bottom fouling over a two-year period.
4. **24 hour average electric load** is the average anticipated electrical load, without growth, over a 24 hour period when operating at the specific endurance speed, ambient air and sea water conditions.
5. **Calculated all-purpose fuel rate** is the specific fuel rate in lbs/SHP-hr. based on the total fuel consumption for propulsion machinery, ship service generators, and other services when operating at the specified endurance speed, ambient air and sea water conditions. In the case of steam plants, this is the figure resulting from the heat balance conditions. For a diesel or gas turbine propelled ship or craft, it is necessary to calculate the consumption of each service separately to arrive at the all-purpose fuel rate.
6. **Ambient conditions** to be utilized in determining the calculated all-purpose fuel rate are 100°F and 40 percent relative humidity air to the fuel consuming services.
7. **Specified fuel rate** is the calculated all-purpose fuel rate increased by a correction factor to allow a tolerance for instrumentation inaccuracy (torsionmeter and shaft modulus) during ship acceptance trials, and for minor machinery design changes made during the construction period. This factor, used as a multiplier, is 1.04 if the average endurance power is one-third or less of the rated full power of the propulsion plant, 1.03 if between one-third and two-thirds, and 1.02 if between two-thirds and full power.
8. **Average endurance fuel rate** is the specified rate increased by five percent. This is an additional increase which allows for plant deterioration over a two-year period.
9. **Endurance fuel (burnable)** is the actual fuel, in tons, required to meet the specified endurance.
10. **Tailpipe allowance** is a factor applied to the endurance fuel (burnable) to allow for the unavailable fuel remaining in the tank below the tailpipes. If the majority of tanks are broad and shallow the factor is 0.95; if narrow and deep it is 0.98.
11. **Endurance fuel load** is the fuel load in ton obtained by dividing the endurance fuel (burnable) by the tailpipe allowance. It is the full load of ship's fuel for which tankage must be provided to meet its endurance requirement. It does not include an additional five percent in equivalent tank volume which must be provided to allow for expansion of fuel. For a compensated system an allowance of less than five percent may be provided, however, this must be determined on a case basis.

## 200-1-c. Procedure

After calculating the average endurance power and average endurance fuel rate, fuel requirements are determined by the following formulae:

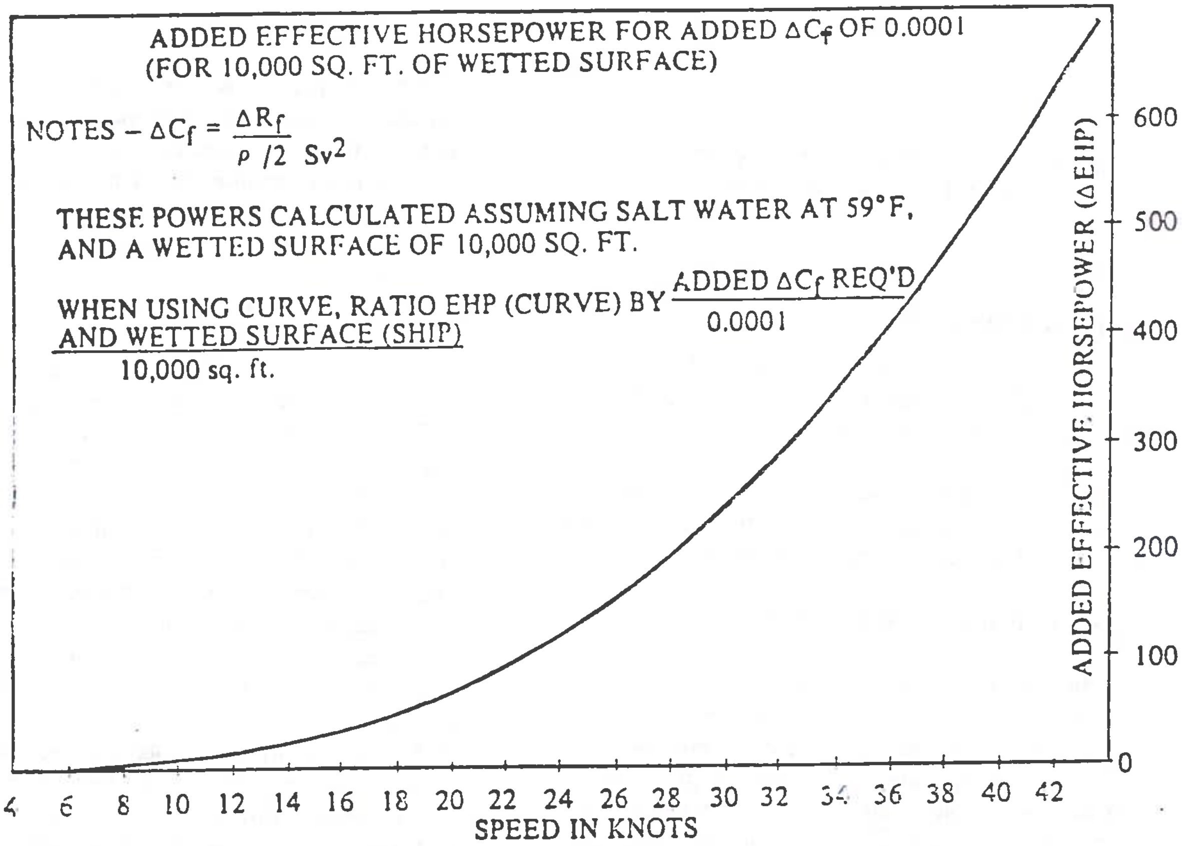
1. Endurance fuel (burnable), tons =

$$\frac{\text{Endurance} \times \text{Avg. End. Power} \times \text{Avg. End. Fuel Rate}}{\text{Endurance Speed} \times 2240}$$

2. Endurance fuel load, tons =

$$\frac{\text{Endurance Fuel (burnable)}}{\text{Tailpipe Allowance}}$$

Appendix B is a sample calculation form.



ADDED EFFECTIVE HORSEPOWER FOR ADDED $\Delta C_f$ OF 0.0001
(FOR 10,000 SQ. FT. OF WETTED SURFACE)

NOTES – $\Delta C_f = \dfrac{\Delta R_f}{\rho/2\ S v^2}$

THESE POWERS CALCULATED ASSUMING SALT WATER AT 59°F, AND A WETTED SURFACE OF 10,000 SQ. FT.

WHEN USING CURVE, RATIO EHP (CURVE) BY $\dfrac{\text{ADDED } \Delta C_f \text{ REQ'D}}{0.0001}$ AND $\dfrac{\text{WETTED SURFACE (SHIP)}}{10,000 \text{ sq. ft.}}$

SAMPLE CALCULATION:

TO FIND THE ADDED EHP FOR A DLG AT 30 KTS
WHEN INCREASING $\Delta C_f$ FROM 0.0006 to 0.0008;

THE WETTED SURFACE OF THE DLG IS 32,000 SQ. FT.
FROM CURVE, $\Delta$EHP=236 AT 30 KTS.

THEN ADDED EHP (@ 30 KTS)

$$= (236) \times \left(\frac{0.0002}{0.0001}\right) \times \left(\frac{32,000}{10,000}\right)$$

$$= (236)\ (2)\ (3.2)$$

$$= 1510.4 \quad \underline{\underline{\text{ANS.}}}$$

**CAUTION!**

NOTE THAT THIS CURVE GIVES ADDED EHP.
FOR ADDED SHP, RESULT MUST BE
DIVIDED BY P.C. AT SPEED IN QUESTION

**SYMBOLS USED ABOVE:**

$\Delta C_f$ = Correlation allowance, treated as an increase in the coefficient of frictional drag.
$\Delta R_f$ = Increased frictional drag, pounds.
$\rho$ = Density of sea water - pounds x sec.$^2$/ ft.$^4$ (=1.9905 at 59° F)
S = Wetted surface area of the ship - ft.$^2$
v = Speed of the ship - ft./sec.

# APPENDIX B
# SURFACE SHIP ENDURANCE CALCULATION FORM

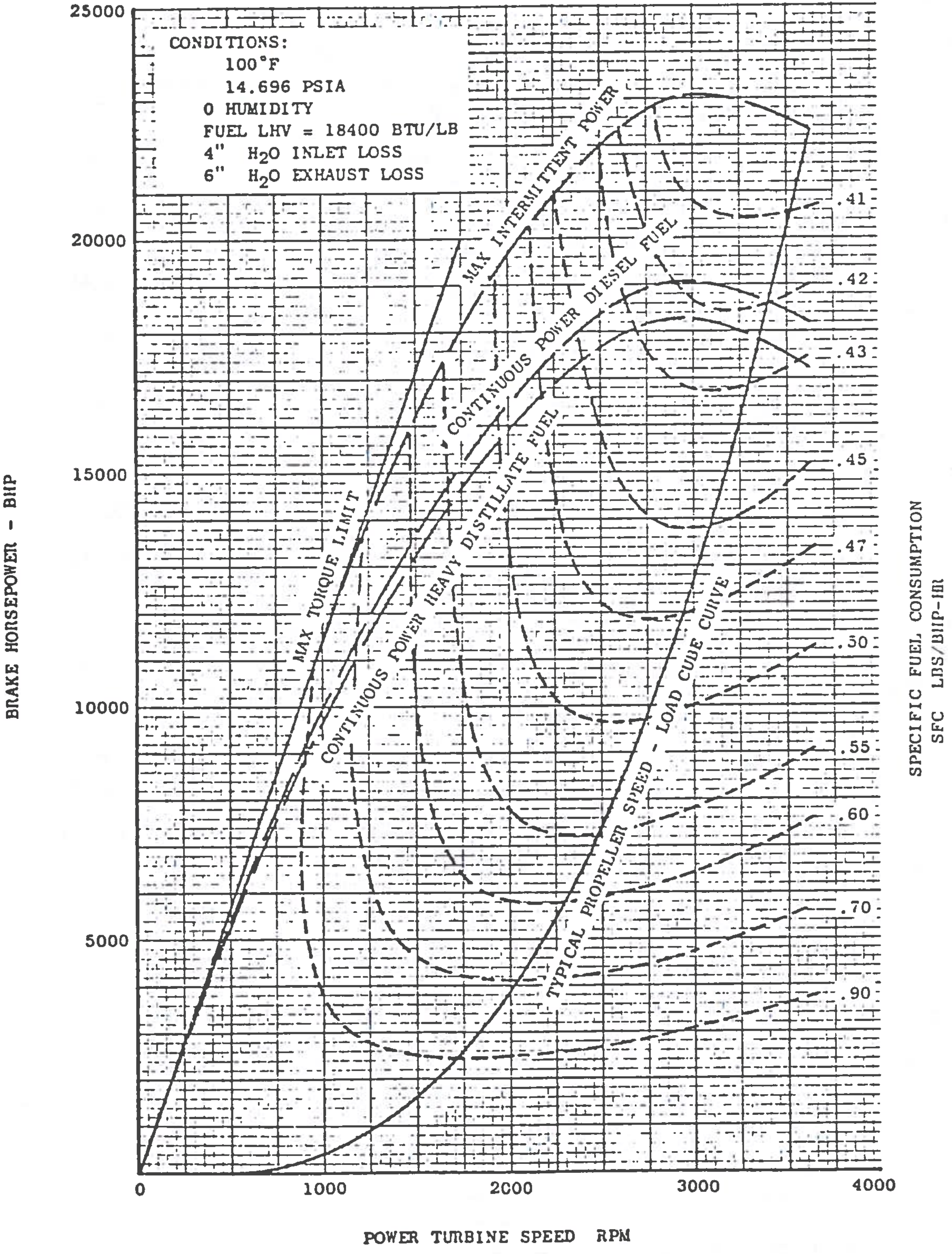
DESIGN ______________________
PREPARED BY__________________
CHECKED BY___________________

| | EXAMPLES | |
|---|---|---|
| | STEAM | DIESEL OR GAS TURBINE |
| (1) Endurance Required, Nautical Miles | 3,000 | 1,200 |
| (2) Endurance Speed, Knots | 15 | 6 |
| (3) Full Load Displacement, Tons | 3,000 | 400 |
| (4) Rated Full Power, SHP | 50,000 | 700 |
| (5) Design Endurance Power @(2) &(3), SHP | 3,000 | 150 |
| (6) Average Endurance Power, SHP: (5) x 1.10 | 3,000x1.10 = 3,300 | 150 x 1.10 = 165 |
| (7) Ratio, Avg. End SHP/rated F.P. SHP: (6)/(4) | 0.066 | 0.24 |
| (8) Average Endurance BHP: (6) / Transmission Efficiency | --- | 165 / 0.95 = 174 |
| (9) 24 Hour Average Electric Load, KW | 500 | 30 |
| (10) Calculated Propulsion Fuel Rate @(8), lbs/BHP-hr. | ---- | 0.479 |
| (11) Calc. Prop. Fuel Consumption, lbs./hr: (10)x(8) | ---- | 0.479 x 174 = 83.4 |
| (12) Calc. S.S. Gen. Fuel Rate @(9), lbs./KW-hr. | ---- | 0.690 |
| (13) Calc. S.S. Gen. Fuel Consumption, lbs./hr: (12)x(9) | ---- | 0.690 x 30 = 20.8 |
| (14) Calc. Fuel Consumption For Other Services, lbs/hr. | ---- | 15.0 (heating) |
| (15) Total Calc. All-Purpose Fuel Consumption lbs/hr: (11) + (13) + (14) | ---- | 83.4 + 20.8 + 15.0 = 119.2 |
| (16) Calc. All-Purpose Fuel Rate, lbs/SHP-hr: (15)/(6) or Heat Balance | 1.00 | 119.2/165 = 0.722 |
| (17) Fuel Rate Correction Factor Based on (7) | 1.04 | 1.04 |
| (18) Specified Fuel Rate, lbs/SHP-hr: (16) x (17) | 1.00 x 1.04 = 1.04 | 0.722 x 104 = 0.750 |
| (19) Avg. Endurance Fuel Rate, lbs/SHP-hr: (18) x 1.05 | 1.04 x 1.05 = 1.092 | 0.750 x 1.05 = 0.787 |
| (20) Endurance Fuel (burnable), Tons: (1) x (6) x (19)/(2) x 2240 | $\frac{3{,}000 \times 3{,}300 \times 1.092}{15 \times 2240}=322$ | $\frac{1{,}200 \times 165 \times 0.787}{6 \times 2240}=11.6$ |
| (21) Tailpipe Allowance Factor | 0.98 | 0.95 |
| (22) Endurance Fuel Load, tons: (20)/(21) | 322 /0.98 = 329 | 11.6 /0.95 = 12.2 |

REFERENCE FOR SOURCE DATA

Design Endurance Power ______________________
Transmission Efficiency ______________________
Calc. Prop. Fuel Rate ______________________
Calc. S.S. Gen. Fuel Rate ______________________
Calc. Fuel Consumption for
Other Services ______________________
Heat Balance ______________________
Full Load Displacement ______________________

LM2500 ESTIMATED ENGINE PERFORMANCE
(SUBJECT TO 5% PRODUCTION VARIATION)



### 3.0 Installed SHP Requirement

For a future single screw frigate type ship of approximately 4000 tons full load displacement, it is desired to know the acceptable range of propeller RPM at the full power condition. The propulsion plant will consist of two (2) LM 2500 gas turbine engines with a maximum full power rating of 40,000 SHP. A CP propeller will be provided for thrust reversal/ship maneuverability. The maximum propeller diameter that can be accomodated in the ship is 17 feet. Based on the calculated ship powering data (EHP) for the fully appended ship with air drag and power margin included, the approximate range of acceptable full power RPM from a propulsive efficiency, installed SHP is required.

**3.1 Calculation of Propulsive Coefficient/Required Ship SHP**—At this early stage of design, it is desirable to verify that for the calculated ship resistance and maximum selected propeller diameter an acceptable range of full power RPM's exists from an installed SHP/propulsive coefficient (P.C.) standpoint. Later analysis of propeller blade cavitation, machinery and hull vibration, machinery arrangements, total ship performance (propeller parametric analysis), propeller performance calculations, and initial ship model testing results will be used to select the final propeller characteristics.

**3.2 Ship Input Data and Requirements**—The following requirements and powering data have been determined:

- Required Full Power Design Speed = 28 Knots
- Maximum Installed Power = 40,000 SHP
- CP Propeller Characteristics
  - Maximum Diameter = 17 feet
  - Number of Blades = 5
  - Maximum Blade Area Ratio = 0.75
  - Hull/Propeller Interaction Coefficients—
    - $1\text{-}w = 0.98$
    - $1\text{-}t = 0.95$
    - $e_{rr} = 0.99$
- Design calculated Ship Resistance = 26,500 EHP

**3.3 Calculate Propeller Propulsive Performance**—The initial step is to determine the minimum acceptable propeller efficiency that will meet the required full power speed within the installed SHP limit. This is determined by calculating the required propulsive coefficient (P.C.).

$$\text{P.C.} = \frac{\text{EHP}}{\text{SHP}} = e_p \times e_h \times e_{rr}$$

where

$e_p$ = propeller efficiency

$e_h$ = hull efficiency $= \frac{1\text{-}t}{1\text{-}w}$

$e_{rr}$ = propeller relative rotative efficiency

$$e_p = \frac{\text{EHP}}{\text{SHP} \times e_h \times e_{rr}}$$

The next step will be to determine the actual prpoeller efficiencies achievable using propeller series data. For this we will use the TROOST propeller series, propeller B5.75 a five bladed propeller with a blade area ratio of 0.75, which is enclosed. The basic coefficients that define propeller performance are thrust, torque coefficients ($K_T$, $K_Q$) and propeller open water efficiency ($e_p/\eta_p$) versus the advance coefficient (J).

$$K_T = \frac{T}{\rho N^2 D^4}$$

$$K_Q = \frac{Q}{\rho N^2 D^5}$$

$$e_p = \frac{K_T}{K_Q} \times \frac{J}{2\pi}$$

$$J = \frac{V_a}{ND}$$

where

T = propeller thrust

$\rho$ = density of water (1.99)

N = propeller revolutions per second

D = propeller diameter

Q = propeller torque

$V_a$ = speed of advance (ft/sec) = $V_K \times (1\text{-}w) \times 1.689$

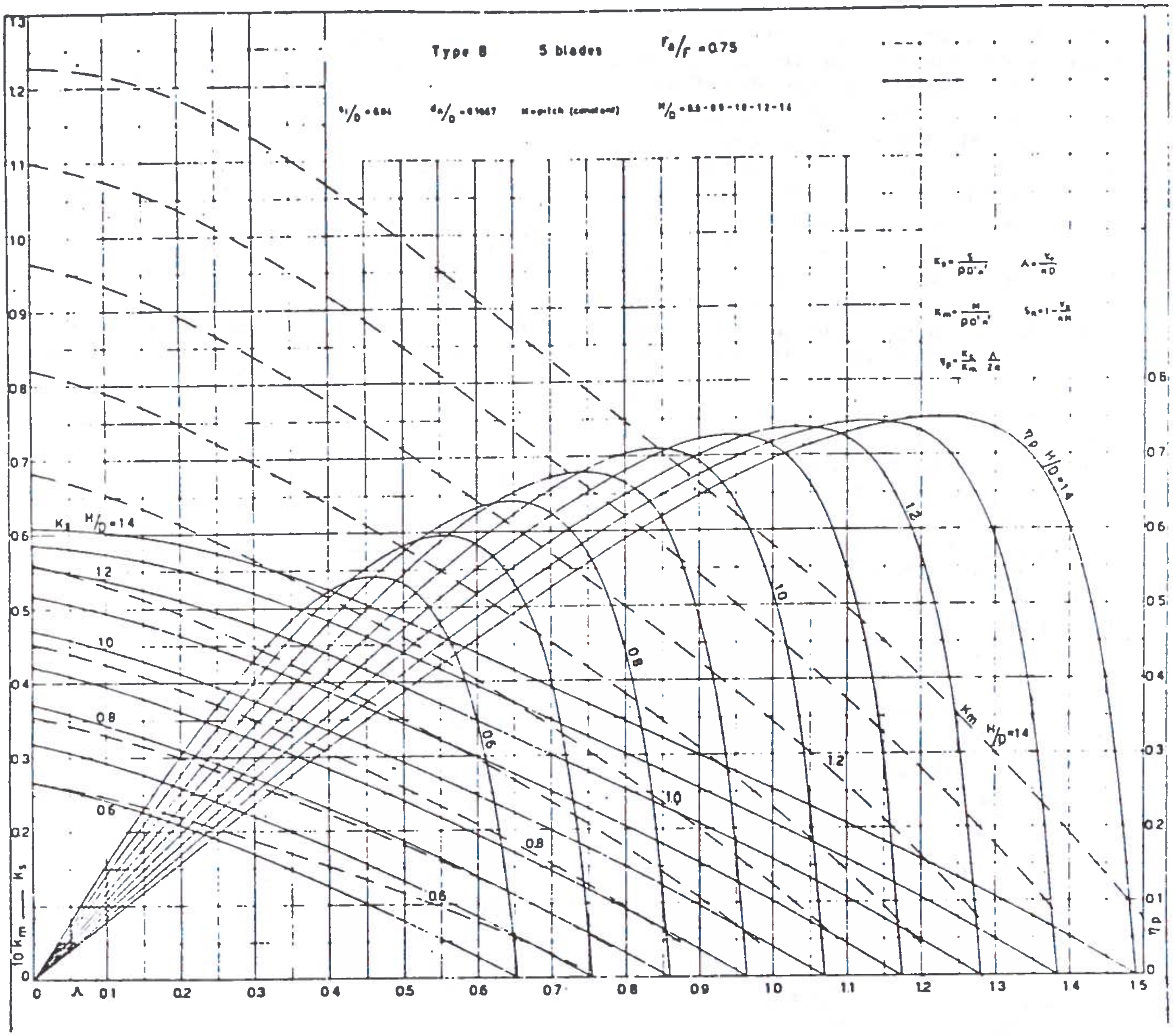
The calculated propeller thrust is equal to the ship calculated resistance including air drag and powering margin, plus thrust augmentation:

$$T = \frac{\text{EHP } 325.7}{V_K\,(1\text{-}t)}$$

Propeller performance can then be determined by calculating the thrust loading coefficient ($K_T/J^2$) and plotting it against J on the propeller curve.

$$K_T/J = \frac{T}{\rho D^2 V_a^2}$$

From the series propeller figure at the intersections of the $K_T/J^2$ curve with propeller $K_T$ lines, the propeller efficiency and speed of advance (J) are determined and a plot can be made of propeller efficiency versus propeller RPM. From this plot and the minimum calculated propeller efficiency the acceptable range of propeller RPM's is determined.



$K_T = K_s$

$K_Q = K_m$

$J = \Lambda$

## 4.0 Heat Balance Calculations

For steam turbine driven ships, the basic machinery plant analysis is contained in the heat balance calculations and flow diagrams prepared for the ship. These heat balances contain the developed requirements for the propulsion system and all energy consuming ship support systems. Heat balance calculations are conducted during all stages of ship design development to define the performance capability of the plant and all its elements and to provide the basis for determining the capability and size of all elements of the machinery plant.

The procedures for conducting heat balances are defined in the following documents:

- George G. Sharp, Inc. "Recommended Practices for Preparing Heat Balances for U.S. Navy Combatant and Auxiliary Type Ships," Report No. 5353, dated 29 May 1969.
- SNAME, "Marine Steam Power Plant Heat Balance Practices," T&R Bulletin 3-11, Reprinted July 1975.

Criteria for the selection of the number of components and their rated capacity is contained in:

- Ship Design Division Technical Practices, dated 1 June 1965.

Using heat balance flow diagrams developed for a new Carrier Design, the CVV, being considered in the shipbuilding program, two problems will be evaluated. The first is a heat balance calculation around the Deaerating Feed Tank, and the second is a calculation of the rated capacity of major components of the steam plant.

**4.1 Heat Balance Calculation**—Enclosed are two heat balance diagrams for a CVV at rated full power, and at a 20 knot condition. The proposed CVV machinery plant is 50 percent of the existing CVA's, with two 70,000 SHP steam turbines driving fixed pitch propellers. In lieu of the normal two boilers per turbine, three will be provided to insure maximum aircraft launch speeds when any one boiler is off the line. The electric plant consists of six ships service turbine generator sets each rated at 2500 KW. The conditions shown are steady state steaming, with no air operations.

**4.2 DFT Heat Balance**—The DFT range of operating pressure is from a minimum of 10 psig to 18 psig, corresponding to an allowable range of 12 to 20 psig in the auxiliary exhaust system. The nominal DFT design operating pressure is 13 psig, 15 psig in the auxiliary exhaust line. A high pressure turbine, extraction bleed line is connected to the auxiliary exhaust line through a reducing valve set at 15 psig. Thus at most steady state steaming conditions ahead, the nominal pressure in the auxiliary exhaust line will be 15 psig and 13 psig in the DFT. The auxiliary exhaust line receives the discharge of all steam driven auxiliary pumps. It is provided with a live steam augmenting line to maintain minimum pressure as well as the H.P. turbine bleed, for operation without the main turbine on the line or at low turbine powers. The four (4) distilling plants obtain their motivating steam from the auxiliary exhaust line.

In the 20 knot heat balance provided, the flow conditions into the DFT are all defined excluding the H.P. Turbine bleed extraction flow. Input flow and enthalpy are provided for the following:

- Main condensate line
- Steam driven auxiliary exhausts
- Catapult through heating drains
- Feed booster pump recirculation

The auxiliary exhaust flow to the distilling plants is shown and the total heat required is 37.9 × $10^6$ BTU/hr.

The problem is to determine the flow required from the H.P. bleed at the enthalpy given on the diagram to achieve a flow balance around the DFT and the output enthalpy of the feed, 214 BTU/hr., corresponding to the 13 psig operating pressure of the DFT.

This flow is determined by conducting a heat balance around the DFT.

**4.3 Sizing/Rating of Machinery Components**—Based on the complete heat balances conducted for a steam plant, the rating capacity of each component is determined. The general criteria for determining component capacities of steam plant machinery is provided as an enclosure to this section. These general criteria provide the baseline for all steam plants, however, for individual ship types and known operational requirements, the specific rating criteria will be varied to suit the ship's missions requirements.

Using the CVV full power heat balance provided and the general criteria, determine the capacities of the following machinery:

- Main Boiler (evaporation rate at boiler overload—lbs/hr.)
- Main Feed Pump (rated flow—GPM)
- Forced Draft Blowers (rated flow—CFM, and head-inches of water)

Using the general rating criteria, calculate the above rated capacities, reflecting that the CVV has three boilers and three main feed pumps per engine room (the specific weight of main feed is 7.88 lbs/gallon).

For the modern CVA's thre main forced blowers are provided for each boiler. The rated capacity of each blower is normally determined by a specific required air operations launch condition, where the required capacity exceeds the boiler overload blower capacity. The required blower capacity has to be determined at both conditions though for a specific CV to determine which is the controlling condition. The blower capacity at boiler overload is calculated as follows:

$$\text{CFM / blower} = 1.20 \times \frac{e_b \text{ @ F.P.}}{3_b \text{ @ B.O.}} \times$$

$$\frac{\text{Fuel Consumption @ F.P.}}{\text{No. of blowers}} \times$$

CF of Std. Air per pound of fuel

$$= 121 \times \frac{80261}{18 \times 60} \times 260$$

The total head of the blowers is then approximately determined as follows:

Total Head (inches of water) = head @ Full Power ×

$$\left(\frac{\text{CFM @ B.O.}}{\text{CFM @ F.P.}}\right)^2$$

## OPERATING CONDITIONS

| | |
|---|---|
| SHP | 43000 |
| PROPELLER RPM | |
| DRUM PRESS | 1220 PSIG |
| SHO PRESS | 1200 PSIG |
| DESHO PRESS | 1197 PSIG |
| SHO TEMP | 936 °F |
| DESHO TEMP | 630 °F |
| MN TURB THROTTLE PRESS | 1185 PSIG |
| MN TURB THROTTLE TEMP | 931 °F |
| MN TURB STEAM RATE (NON-EXTRACTION) | 5.908 LB/SHP-HR |
| MN COND VAC | 28.4 IN HG |
| ELEC LOAD | 7813 KW |
| SSTG THROTTLE PRESS | 1140 PSIG |
| SSTG THROTTLE TEMP | 931 °F |
| SSTG COND VAC | 28.2 IN HG |
| AUX EXH PRESS | 15 PSIG |
| DIST PLANT RATED CAP | 280,000 GPD |
| DIST PLANT OPER RATE | 280,000 GPD |
| FDB TOT HEAD | 14.4 IN WG |
| FW TEMP | °F |
| BOILER EFF | 86.8 % |
| FUEL OIL HNV | 19500 B/LB |
| FUEL CONSUMPTION | 30,694 LB HR |
| SPECIFIC FUEL RATE | 0.7138 LB/SHP-HR |

## MOTOR DRIVEN PUMPS

| | Installed | Operating |
|---|---|---|
| MN FD BOOSTER | 6 | 2 |
| MN CONDENSATE | 4 | 2 |
| MN CONDENSER VAC | 4 | 2 |
| AUX CONDENSATE | 6 | 4 |
| AUX CONDENSATE VAC | 6 | 4 |
| AUX CONDENSER CIRC | 6 | 4 |
| FIRE | | 0 |
| FW DRAIN | 2 | 2 |
| DIST PLANT DRAIN | 4 | 4 |
| LO SERVICE | 2 | 0 |
| FO SERVICE | 2 | 0 |

Air-Fuel Ratio ........ 240 Cf air/lb. fuel

## CVV
## HEAT BALANCE & FLOW DIAGRAM
## 20 KTS

---

## OPERATING CONDITIONS

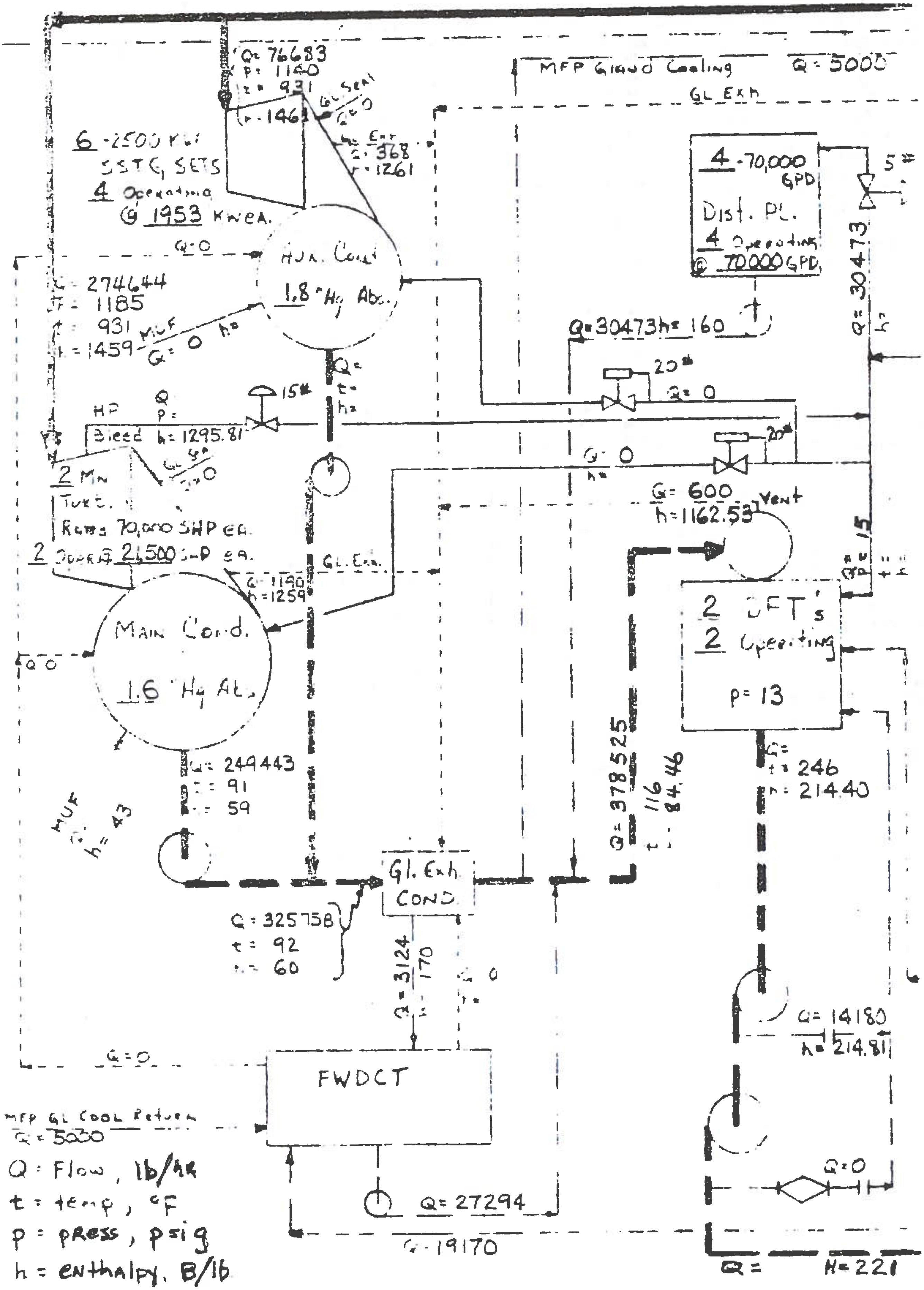
| | |
|---|---|
| SHP | 140.000 |
| PROPELLER RPM | |
| DRUM PRESS | 1257 PSIG |
| SHO PRESS | 1200 PSIG |
| DESHO PRESS | 1194 PSIG |
| SHO TEMP | 949 °F |
| DESHO TEMP | 647 °F |
| MN TURB THROTTLE PRESS | 1100 PSIG |
| MN TURB THROTTLE TEMP | 944 °F |
| MN TURB STEAM RATE (EXTRACTION) | 5.995LB/SHP-HR |
| MN COND VAC | 26.0 IN HG |
| ELEC LOAD | 7957 KW |
| SSTG THROTTLE PRESS | 1105 PSIG |
| SSTG THROTTLE TEMP | 944 °F |
| SSTG COND VAC | 28.2 |
| AUX EXH PRESS | 15 PSIG |
| DIST PLANT RATED CAP | 280.000 GPD |
| DIST PLANT OPER RATE | 280,000 GPD |
| FDB TOT HEAD | 39.2 IM WG |
| TW TEMP | °F |
| BOILER EFF | 85 % |
| FUEL OIL MHV | 19.500 B/LB |
| FUEL CONSUMPTION | 80261LB/HR |
| SPECIFIC FUEL RATE | 0.5733LB/SHP-HR |

## MOTOR DRIVEN PUMPS

| | Installed | Operating |
|---|---|---|
| MN FD BOOSTER | 6 | 4 |
| MN CONDENSATE | 4 | 4 |
| MN CONDENSATE VAC | 4 | 2 |
| AUX CONDENSATE | 6 | 4 |
| AUX CONDENSER-VAC | 6 | 4 |
| AUX CONDENSER CIRC | 6 | 4 |
| FIRE | | 0 |
| FW DRAIN | 2 | 2 |
| DIST PLANT DRAIN | 4 | 4 |
| LO SERVICE | 2 | 0 |
| FO SERVICE | 2 | 0 |

AIR-FUEL RATIO ........ 240 CF AIR/LB FUEL

## CVV
## HEAT BALANCE & FLOW DIAGRAM
## 6 BOILERS – 140,000 SHP
## 20 KTS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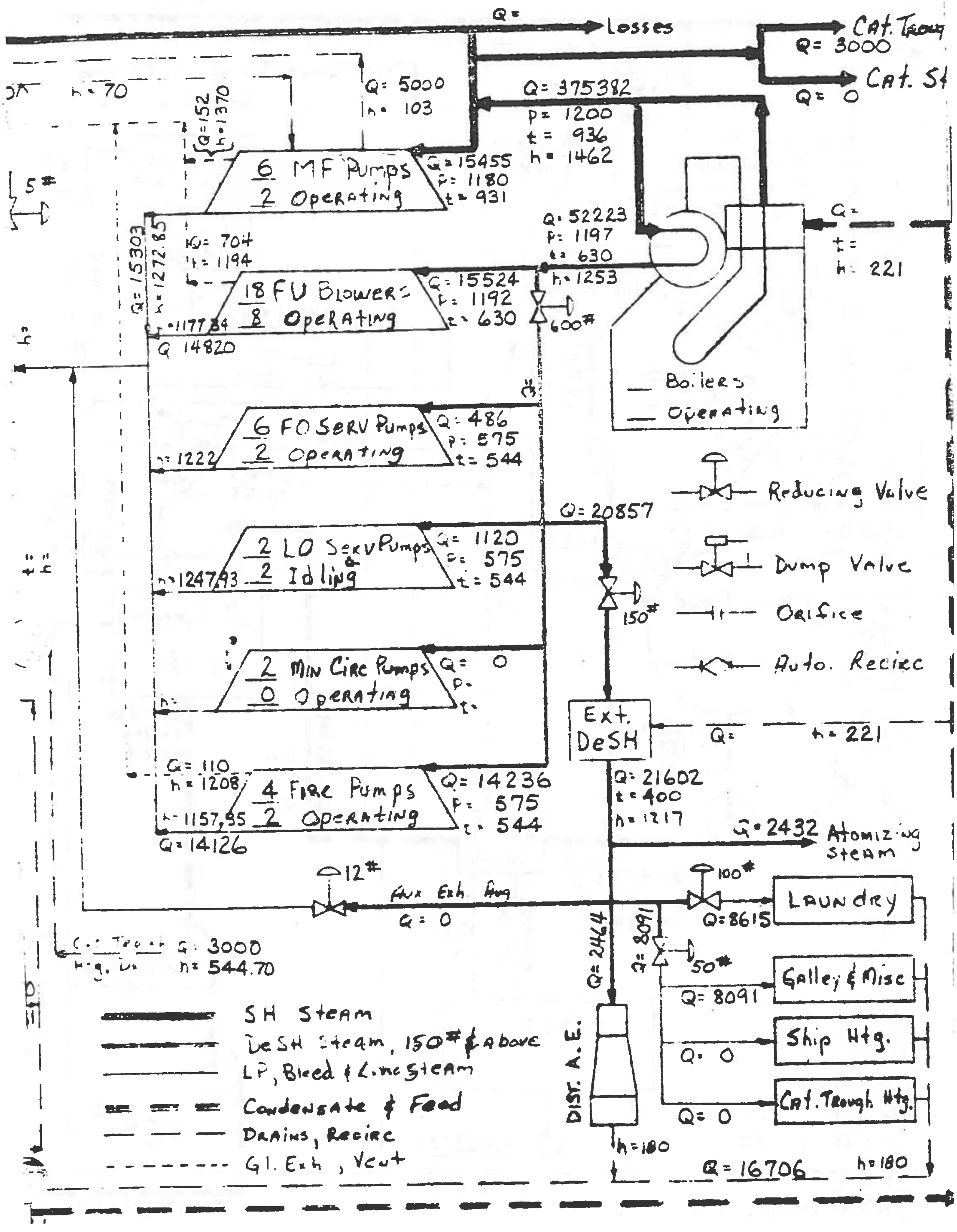
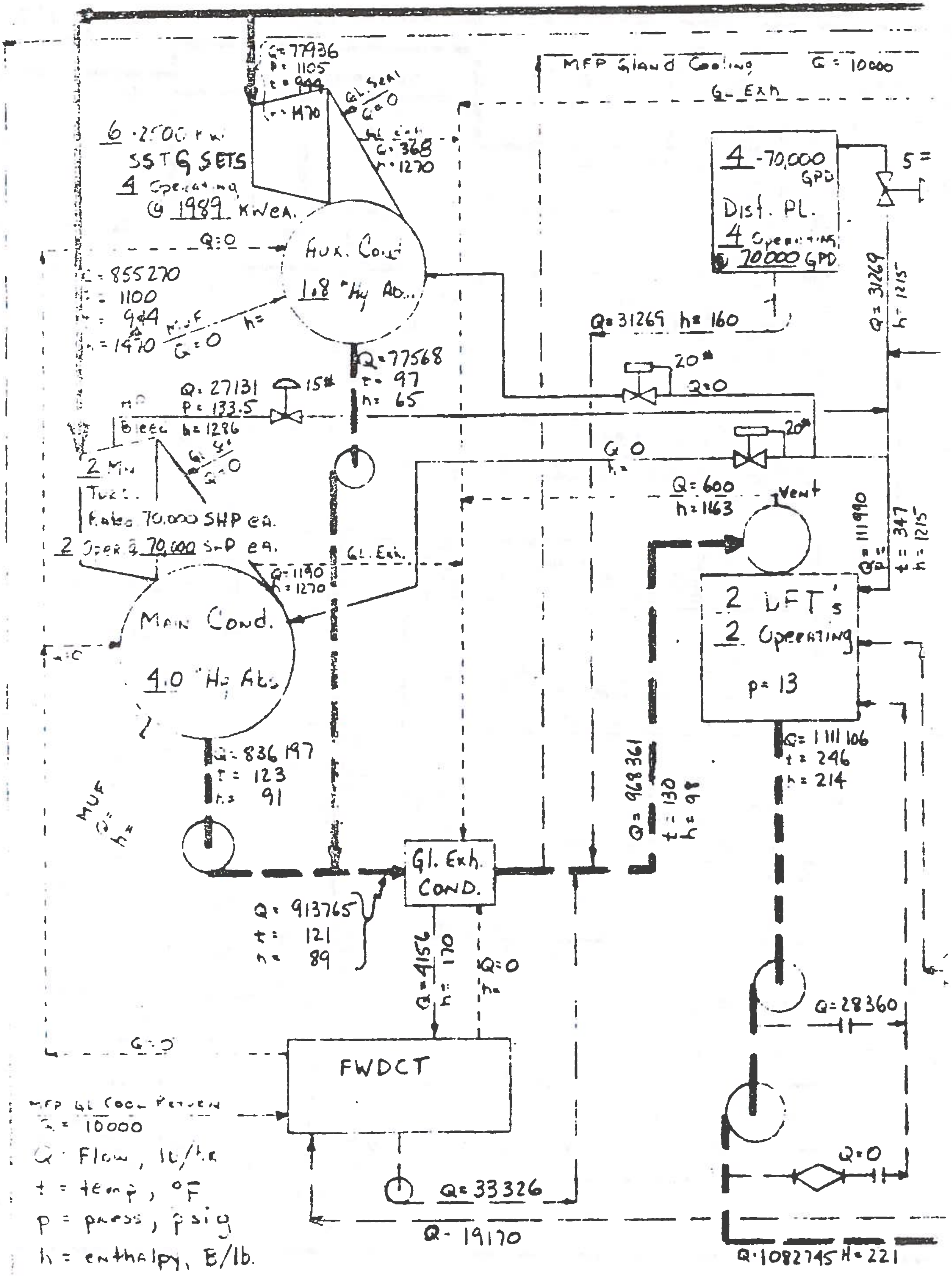
P= 1140
t= 931
h= 1461
Gl Seal Q= 0
Gl Exh
Q= 368
h= 1261
6 - 2500 KW SSTG SETS
4 Operating @ 1953 KW EA.
MFP Gland Cooling
Q= 5000
GL EXH
4 - 70,000 GPD
Dist. PL.
4 Operating @ 70000 GPD
5#
Q= 30473
h=
Q= 0
Aux. Cond
1.8" Hg Abs.
Q= 274644
P= 1185
t= 931
h= 1459
MUF Q= 0 h=
Q= 30473 h= 160
20#
Q= 0
Q=
t=
h=
HP Bleed
Q
P=
h= 1295.81
15#
20#
Q= 0
h=
Gl Sp Q= 0
2 Main Turb.
Rated 70,000 SHP EA.
2 Operat. 21,500 SHP EA.
Q= 600
h= 1162.53
Vent
Gl. Exh.
Q= 1190
h= 1259
15
2 DFT's
2 Operating
p= 13
Main Cond.
1.6" Hg Abs.
Q 0
Q= 249443
t= 91
h= 59
MUF
Q
h= 43
Q= 378525
t= 116
h= 84.46
Q=
t= 246
h= 214.40
Gl. Exh Cond.
Q= 325758
t= 92
h= 60
Q= 3124
h= 170
Q= 0
h=
Q= 14180
h= 214.81
Q= 0
FWDCT
MFP GL COOL Return
Q= 5030
Q= 27294
Q= 19170
Q=
H= 221
Q: Flow, lb/hr
t = temp, °F
p = press, psig
h = enthalpy, B/lb


Q =
Losses
Cat. Troug
Q = 3000
Cat. St
Q = 0
h = 70
Q = 5000
h = 103
Q = 375382
p = 1200
t = 936
h = 1462
Q = 152
h = 1370
5#
6 MF Pumps
2 Operating
Q = 15455
p = 1180
t = 931
Q = 52223
p = 1197
t = 630
h = 1253
Q =
t =
h = 221
Q = 15303
h = 1272.85
Q = 704
h = 1194
18 FU Blowers
8 Operating
Q = 15524
p = 1192
t = 630
600#
h = 1177.84
Q 14820
h =
Boilers
Operating
6 FO Serv Pumps
2 Operating
Q = 486
p = 575
t = 544
h = 1222
Reducing Valve
Dump Valve
Orifice
Auto. Recirc
Q = 20857
2 LO Serv Pumps
2 Idling
Q = 1120
p = 575
t = 544
h = 1247.93
t =
h =
150#
2 Min Circ Pumps
0 Operating
Q = 0
p =
t =
h =
Ext. DeSH
Q =
h = 221
Q = 110
h = 1208
4 Fire Pumps
2 Operating
Q = 14236
p = 575
t = 544
h = 1157.35
Q = 14126
Q = 21602
t = 400
h = 1217
Q = 2432 Atomizing steam
12#
Aux Exh Hdr
Q = 0
100#
Laundry
Q = 8615
Cat Trough Htg. Dr.
Q = 3000
h = 544.70
Q = 2464
Q = 8091
50#
Galley & Misc
Q = 8091
Ship Htg.
Q = 0
Cat. Trough Htg.
Q = 0
DIST. A. E.
h = 180
Q = 16706
h = 180
SH Steam
DeSH Steam, 150# & above
LP, Bleed & Line steam
Condensate & Feed
Drains, Recirc
Gl. Exh, Vent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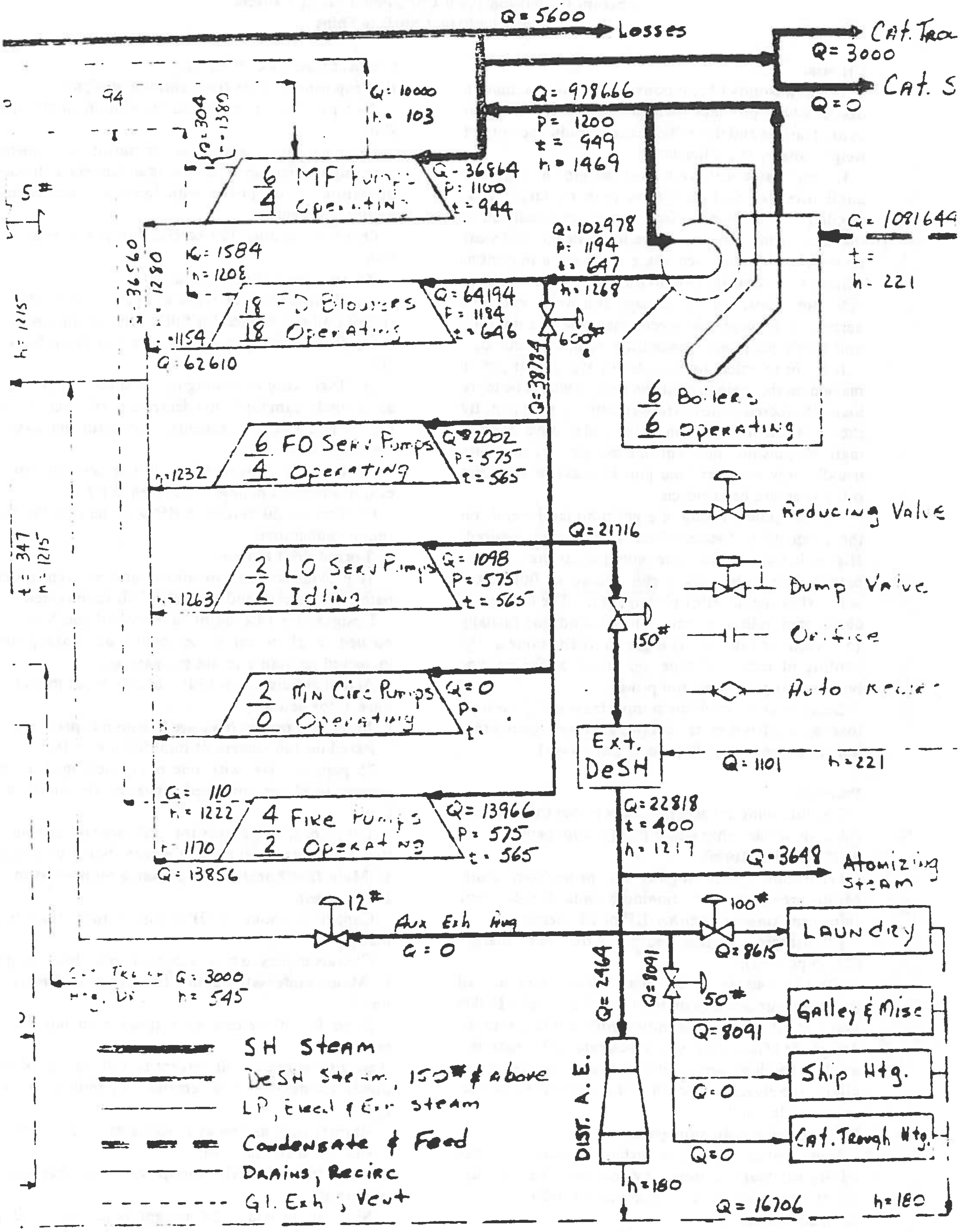
P= 1105
t= 944
h= 1470
Gl. Seal
Q=0
Gl. Exh
Q= 368
h= 1270
6 - 2500 kw SSTG SETS
4 Operating
@ 1989 KW ea.
MFP Gland Cooling
Q = 10000
Gl. Exh
4 - 70,000 GPD
Dist. Pl.
4 Operating
@ 70000 GPD
5
Q=0
Aux. Cond
1.8 "Hg Abs.
Q = 855270
P = 1100
t = 944
h = 1470
MUF
Q=0
h=
Q= 31269 h= 160
Q= 31269
h= 1215
Q=77568
t= 97
h= 65
20#
Q=0
Q= 27131
P= 133.5
h= 1286
15#
h=
Bleed
Gl. Seal
Q=0
20#
Q 0
h=
2 Main Turb.
Rated 70,000 SHP ea.
2 Oper @ 70,000 SHP ea.
Q= 600
h= 1163
Vent
Q= 111990
t= 347
h= 1215
Q=1190
h= 1270
Gl. Exh.
Main Cond.
4.0 "Hg Abs
Q=0
2 DFT's
2 Operating
p= 13
Q= 836197
T= 123
h= 91
MUF
Q=
h=
Q= 968361
t= 130
h= 98
Q= 1111106
t= 246
h= 214
Gl. Exh. COND.
Q= 913765
t= 121
h= 89
Q= 4156
h= 170
Q=0
h=
Q=28360
Q=0
FWDCT
MFP Gl Cool Return
Q= 10000
Q= 33326
Q= 19170
Q=1082745 H=221
Q = Flow, lb/hr
t = temp, °F
p = press, psig
h = enthalpy, B/lb.



Q=5600
Losses
Cat. Tra
Q=3000
Cat. S
Q=0
Q=978666
P=1200
t=949
h=1469
Q=10000
h=103
94
Q=304
h=1380
5#
6 MF Pumps
4 Operating
Q=36864
P=1100
t=944
Q=102978
P=1194
t=647
h=1268
Q=1081644
t=
h=221
Q=1584
h=1208
Q=36560
h=1280
18 FD Blowers
18 Operating
Q=64194
P=1184
t=646
600#
Q=1154
Q=62610
h=1215
Q=38784
6 Boilers
6 Operating
6 FO Serv Pumps
4 Operating
Q=2002
P=575
t=565
h=1232
Reducing Valve
Q=21716
t=347
h=1215
2 LO Serv Pumps
2 Idling
Q=1098
P=575
t=565
h=1263
Dump Valve
150#
Orifice
Auto Recirc
2 Main Circ Pumps
0 Operating
Q=0
P=
t=
Ext. DeSH
Q=1101
h=221
Q=110
h=1222
4 Fire Pumps
2 Operating
Q=13966
P=575
t=565
h=1170
Q=13856
Q=22818
t=400
h=1217
Q=3648
Atomizing Steam
12#
100#
Aux Exh Hdr
Q=0
Laundry
Q=8615
Cat Trough Htg. Dr.
Q=3000
h=545
Q=2464
Q=8091
50#
Galley & Misc
Q=8091
Ship Htg.
Q=0
Cat. Trough Htg.
Q=0
DIST. A. E.
h=180
Q=16706
h=180
SH Steam
DeSH Steam, 150# & above
LP, Elect & Exh Steam
Condensate & Feed
Drains, Recirc
Gl. Exh, Vent

## Steam Propulsion Plant Component Sizing Criteria
## For Combatant Surface Ships

**Criteria:**

The philosophy of component sizing is to achieve a design which provides such safeguards and margins as are feasible and desirable without undue penalty of weight, space, and efficiency.

It is more practical to put more margin on small vital auxiliaries like fuel oil pumps than on larger vital auxiliaries such as main feed pumps, especially since the latter, when over-sized, are less efficient and wear more severely. However, there is always a minimum margin for propulsion auxiliaries. This generally provides for allowance for leakage and wear, and also permits 90 percent ship speed when one of a multiple unit type (feed pump, condensate pump) is secured.

It is not practical to provide any significant power margin on the main propulsion unit. It would be more logical to increase the full power rating of the plant. By the same reasoning, it is not practical to have standby main propulsion units. Turbine margins are therefore usually only designers' margins to make certain that full power can be achieved.

The full power rating of a given boiler depends on the intensity of loading (heat release rate) desired. Higher loading will require more maintenance. Combatant type boilers are highly loaded at full power where they spend little operating time. The loading of combatant boilers at their cruising condition (usually 15 percent of full power) is generally the same as the loading of merchant type and naval auxiliary type boilers at their normal full power.

Because of the need for compactness and their high loading at full power, the overload rating of combatant boilers is kept low (120 percent full power).

**Practice:**

The following are general guides rather than specific rules for surface ship steam plant components:

**a. Propulsion Turbine.**

(1) Standby—not logical for propulsion units. Multi-screw units use remaining units. Single screw ships can "single-up" either HP or LP element.

(2) Margins—normal margin is designer's margin (2 to 5 percent).

(3) Life—40,000 hours with 1000 hours at full power for surface ship oil fired; nuclear usually 10,000 hours at full power. The most significant factor in the design life of a turbine is the creep rate of the materials used in the high temperature areas. This creep rate affects the clearances which in turn affect the performance of the unit.

**b. Propulsion reduction gears.**

Same power and life as turbine. Designed to take additional loads imposed by maneuvering. See applicable technical practice sheet under 9420.

**c. Main condenser.**

Design vacuum based on full power turbine rating and standard sea water temperature. Life to match turbine. Margin is in cleanliness factor and in conservative heat transfer coefficient.

**d. Propulsion boilers (conventional design)**

Two per shaft is preferred (minimum of two per ship).

Designed for—continuous operation at cruising with high efficiency and low maintenance-extended operation at full power with lower efficiencey and higher maintenance.

Overload rating—120 percent full power evaporation.

Reasons for 120 percent rating:

(1) Highest practical without adding appreciably to size of boiler designed for full power conditions.

(2) Higher rating would require too much blower HP.

(3) Takes care of fouling of fire sides and provides additional steam for future decrease in efficiency of the turbine and other components, or for future increase in electrical load.

(4) For new ship condition, 120 percent full power evaporation will handle 110 percent SHP.

(5) Permits 60 percent SHP with half of the fire rooms inoperative.

**e. Forced draft blowers**

To provide combustion air for most extreme anticipated 120 percent boiler overload air requirements.

Designed for 19.4 lbs. of 68°F free air per lb. of fuel burned to allow for boiler casing air leaking and increased resistance in hot gas passages.

Actual anticipated is 17 lbs. of 100°F per lb. of fuel burned for new ship.

**f. Main feed pumps (two per fireroom is preferred)**

Based on 150 percent of total full power feed.

75 percent SHP with one pump deranged, or 90 percent to 85 percent speed capability for combatant ship.

150 percent also allows for feed surge requirements (low boiler water) at the 120 percent boiler overload.

**g. Main feed booster pumps (same number as main feed pumps).**

Capacities about 30 GPM higher than main feed pumps.

Characteristics to match those of main feed pumps.

**h. Main condensate pumps (two per propulsion turbine).**

Sized for 90 percent ship speed with one pump secured.

Above gives about 130 percent to 150 percent of full power condensate requirements, depending on ship type.

Ratings kept as low as practicable to avoid pump cavitation at partial load.

**i. Main fuel oil service pumps (two per fireroom is preferred).**

Sized to provide 120 percent boiler overload in fireroom with one pump inoperative; about 240 percent of full power fuel for two-pump installation.

Large margins feasible for small size positive displacement pump.

Delivers lighter fuels at reduced capacity.

**j. Deareating Feed Tank and Heater (DFT) (one per shaft).**

Sized to deareate and heat total boiler overload requirement for one shaft.

Storage capacity—Approximately two minutes supply at full power—At least five minutes supply at cruising power—Volume of water between upper and lower extremes of gage glass level should be sufficient to handle change in total weight of boiler water between standby and overload condition—Above values vary with ship type; the weight and space criticality of the ship being a factor.

**k. Main propulsion shafting (see applicable Technical Practice Sheet under 9430).**

Shafting and thrust bearing sized for full power torque with an allowance (about 10-40 percent; ahead) for additional nontransient torque and thrust during maneuvering.

**l. Turbine-generator condensate pump.**

One per T.G. set, no standby. (Standby turbine generator set is normally available.)

**m. Turbine generator circulating pump.**

One per T.G. set, no standby. (Standby turbine generator is normally available.)

**n. Emergency feed pump (one per fireroom).**

Emergency cold feed to boiler on loss of DFT. Usually sized so that full power feed can be delivered when in parallel with one main feed pump.

**o. Emergency feed booster pump.**

Required with centrifugal emergency feed pump.

Used with vacuum priming pump for reserve feed transfer.

Capacity is 20-30 GPM more than emergency feed pump.

**p. Port use or lighting off blowers.**

One per boiler where required.

**q. Port use fuel oil service pumps.**

One per fireroom where required.

**r. Shaft driven main lube oil pump.**

One per shaft.

Designed to give adequate lubrication to main propulsion unit from full power down to a condition corresponding to one-third of full power RPM.

**s. Standby lube oil pump (normally turbine drive).**

One per shaft.

Arranged to operate when shaft driven pump is inadequate or inoperable. Designed to provide adequate lubrication (independently) at all ship speeds, ahead and astern.

**t. Emergency lube oil pump.**

One per shaft (normally electric drive).

Serves as emergency standby unit for low speeds and astern, and designed to provide adequate lubrication at those conditions. Normally rated at about 50 percent of capacity of shaft drives on turbine driven lube oil pumps.

**Background:**

The foregoing is a compendium of current Design Division practices in regard to component sizing and margins. These practices have evolved partly as a result of experience, and partly from studies and discussions within the Machinery Branch to obtain the most practical compact machinery plants. Because of the steepness of the upper range of the speed-power curve of most combatant ships, the 100 percent standby provided for merchant ship feed pumps and other auxiliaries is not necessary in this case. A large decrease in available combatant ship power will result in a small decrease in ship speed. The above criteria are in general as valid for nuclear powered as for oil fired combatant surface ships.

### 5.0 Propeller Parameteric Analysis

The propeller parametric analysis is an essential engineering analysis conducted during the preliminary design stage of all ship types. It validates the basic requirement of the propulsion system, its ability to propel the ship at the required sustained speed. Throughout all later stages of ship design, this is revalidated as changes in the ship occur and as more detailed analyses of the complete ship and machinery plant are conducted. The basic parameters of the propeller that are selected as the output of this analysis are the diameter and RPM. The complete analysis includes determining the acceptable range of propeller parameters at full power considering propeller propulsive performance and cavitation effects, acceptable machinery plant arrangements, satisfactory machinery and hull vibration characteristics; and then within this acceptable range selecting the parameters that provide the most effective performance for the ship throughout the complete operating speed range of the ship. This final optimization of the propeller parameters for total ship performance is conducted at the endurance speed of the ship. It consists of determining the least total weight of machinery and endurance fuel (LTWMF). This minimum value represents the nominal compromise between minimum ship acquisition costs and life cycle costs. As such it normally represents the actual propeller parameters selected for the ship, although if other constraints, technical and ship acquisition cost, permit it, a slightly lower condition with better propulsive efficiency will be selected.

**5.1 Overall Ship/Machinery Plant Characteristics—** For this problem, we again assume we have a single screw gas turbine plant consisting of two (2) gas turbines geared to provide 40,000 SHP to a CP propeller. Initial analyses at the required full power speed have determined that using propeller series data and actual lifting line propeller calculations, the required ship speed can be achieved with the installed SHP using 16 or 17 foot propellers within a range of full power propeller RPM from 160 to 220. The proposed ship's hull will accommodate a 17 foot propeller with a minimum tip clearance (25 percent of the propeller diameter). Calculations of propeller blade cavitation using the Gawn and Burrill loading criteria indicated that the 16 foot diameter would be marginal at full power from a thrust breakdown ($K_T$ breakdown) consideration.

The minimum propeller diameter satisfying cavitation criteria at the full power ship speed is 16.5 feet. Preliminary analysis of machinery and hull vibration using calculated unsteady propeller forces from a wake survey for a nominally similar hull form have determined that minimum full power PRPM should be greater than 160.

**5.2 Machinery Plant Calculations—**Based on full power design requirements the acceptable range of propeller parameters is:

| | |
|---|---|
| Diameter | 16½ to 17 feet |
| RPM | 160 to 220 |

Within this range, the predicted off-design propeller performance is calculated at the design endurance speed. Using the estimated hull interaction coefficients at the endurance speed from a similar hull form, the propulsion coefficients can be calculated and the required ship's endurance fuel determined (using DDS 200-1). Table 5.1 contains a tabulation of these calculated values.

Reduction gear sizes and weights are then calculated. For gas turbine ships there exist two basic design options that can be considered, design of the reduction gear to meet full power installed SHP requirements, or design of the high speed gear elements to accept the rated BHP input of one gas turbine engine. Where weight, space, and acquisition cost constraints impose severe constraints on the ship and propulsion system, design of the reduction gear to meet only the required full power condition will be selected. Design of the reduction gear to accept rated BHP of each prime at the PRPM associated with that SHP, will extend the speed range possible with one engine, significantly reducing the fuel consumption rate and ship life cycle costs. In this case, based on life cycle analyses, the rated engine power (BHP) option was selected, and the calculated results are given in Table 5.2.

Machinery arrangement studies will be conducted throughout the preliminary design and all later stages to determine the overall required sizes of all machinery spaces, and the proper location of all machinery elements. Table 5.3 contains a tabulation of the effects of the range of propeller parameters on engine room length and shafting rake.

Based on machinery arrangement studies, acceptable shafting arrangements from a machinery plant and overall ship hull consideration will have been developed. Using these shafting arrangements, the weight of propulsion shafting will then be calculated in accordance with DDS 243-1. Gas turbine ships using CPP's at present impose a unique requirement on the design of waterborne shafting, propeller, intermediate and stern tube shafting. The minimum size of waterborne shafting for CPP's is governed by the overhung moment of the CPP, to meet the maximum allowable shaft bending stress of 6000 psi. The results of shafting weight calculations are presented in Table 5.4.

**5.3 Selection of Basic Propeller Parameters—**Using the information and data provided, calculate the total weight of machinery and fuel, plot the net weight difference for 16½ and 17 foot propellers in the RPM range provided and select the recommended ship basic propeller parameters of diameter and RPM.

## TABLE 5.1

## PROPULSIVE COEFFICIENTS AT ENDURANCE SPEED AND ESTIMATED ENDURANCE FUEL LOAD

1-t = 0.94; 1-w = 0.99; $e_{rr}$ = 0.95; EHP = 5250

| Prop Dia. | FP PRPM | Endurance Speed P.C. | Endurance Fuel (tons) |
|---|---|---|---|
| 16.5′ | 160 | .658 | 671.6 |
| | 180 | .649 | 677.0 |
| | 200 | .632 | 688.0 |
| | 220 | .610 | 704.1 |
| 17′ | 160 | .664 | 667.6 |
| | 180 | .644 | 679.7 |
| | 200 | .620 | 696.5 |
| | 220 | .594 | 715.5 |

## TABLE 5.2

## REDUCTION GEAR SIZE ESTIMATES

| Design Full Power PRPM | Reduction Gear | | | Estimate PRPM @ Single Engine Rated BHP— (20,500) |
|---|---|---|---|---|
| | Weight (#) | LOA (in.) | Bull Gear Dia. (in.) | |
| 160 | 128,100 | 163.5 | 130 | 128 |
| 175 | 116,050 | 158.8 | 126 | 142 |
| 180 | 113,000 | 157.0 | 126 | 146 |
| 200 | 100,200 | 149.6 | 126 | 158 |
| 220 | 91,000 | 143.5 | 126 | 162 |

## TABLE 5.3

## MACHINERY ARRANGEMENT CONSIDERATIONS

| Prop. Dia. | PRPM | Shaft Rake (deg) | Engine Room Length (ft) |
|---|---|---|---|
| 16′ | 180 | 6 | 43 |
| | 200 | 6 | 42 |
| | 220 | 6 | 42 |
| 16.5′ | 175 | 6¼ | 43 |
| | 180 | 6¼ | 43 |
| 17′ | 160 | 6½ | 44 |

## TABLE 5.4

## PROPULSION SHAFTING WEIGHT ESTIMATES

| Prop Dia. | PRPM | Prop Weight | Prop CG | Prop Mom Arm | Prop Shaft OD | Weight #/in | Length Prop + ST | Weight # | Line Shaft OD | Weight #/in | Length | Weight | Total Shaft Weight | Total Est. Weight in Tons x1.25/2240 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 16.5 | 160 | 45,000 | 25.2 | 75.6 | 29.46 | 107.6 | 82' | 105878 | 18.41 | 42.0 | 10'6" | 5342 | 111220 | 62.1 |
| ↓ | 180 | ↓ | ↓ | ↓ | ↓ | ↓ | ↓ | ↓ | 17.70 | 38.8 | ↓ | 4935 | 110813 | 61.8 |
|  | 200 |  |  |  |  |  |  |  | 17.09 | 36.2 |  | 4605 | 110483 | 61.6 |
|  | 220 |  |  |  |  |  |  |  | 16.56 | 34.0 |  | 4325 | 110203 | 61.5 |
| 17 | 160 | 52,453 | 25.2 | 78.8 | 31.5 | 123.1 |  | 121130 | ↓ | ↓ |  | ↓ | 126472 | 70.6 |
| ↓ | 180 | ↓ | ↓ | ↓ | ↓ | ↓ |  | ↓ |  |  |  |  | 126065 | 70.3 |
|  | 200 |  |  |  |  |  |  |  |  |  |  |  | 125735 | 70.2 |
|  | 220 |  |  |  |  |  |  |  |  |  |  |  | 125455 | 70.0 |

## WORKED PROBLEMS

To be used by the student to verify his technique at solving the problems in Practical Factors.

Engineering Duty Officer
Qualification Program (EDQP)

## PROPULSION SYSTEM DESIGN

Calculation Procedure, Assumptions
and Solutions for the Engineering
Analyses Provided for Propulsion System Design

## 1.0 Introduction

The selection of engineering analysis problems for the practical factors section, represents typical problems conducted during the ship preliminary design stage. As stated these analyses are normally repeated or continually conducted during all stages of design as more detailed ship and machinery plant details are developed. These typical problems represent only a portionof the analyses and studies conducted in developing the propulsion system design for a Naval Surface Ship.

For each problem many simplifying assumptions were made and information provided to permit the student to actually conduct the analysis and derive a solution. The basic approach to the problems though was to provide the student with a background and understanding of what is required and how it is accomplished. It is not practical to develop a COOKBOOK for the design development of propulsion systems for Naval Surface Ships. First of all it is an iterative process with continuing development requiring modification and updating of the system as more detail is developed. During this process an interactive communication between the Naval Architects and Marine Engineers conducting the design is more important to its ultimate success than the actual calculation procedures conducted. Specific ship requirements, constraints, basic analysis techniques, simplifying computer processes, and actual technological state of the art, are constantly changing. As a guide to the instructor, a very limited amount of the reference material required in the design of the propulsion system has been provided.

The solutions calculated for each problem include further assumptions, alternative approaches, and in some cases further modifications of data and procedures provided as background material to the students.

## 2.0 Endurance Fuel Calculation

For this problem all the required information has been provided to conduct the calculation. Using the calculation form all the required data is available excluding the prime mover specific fuel consumption rate.

This is obtained by determining the PTRPM (18 × 120 = 2160) and entering the provided engine performance map. The estimated engine SFC = 0.494 lbs/BHP-hr.

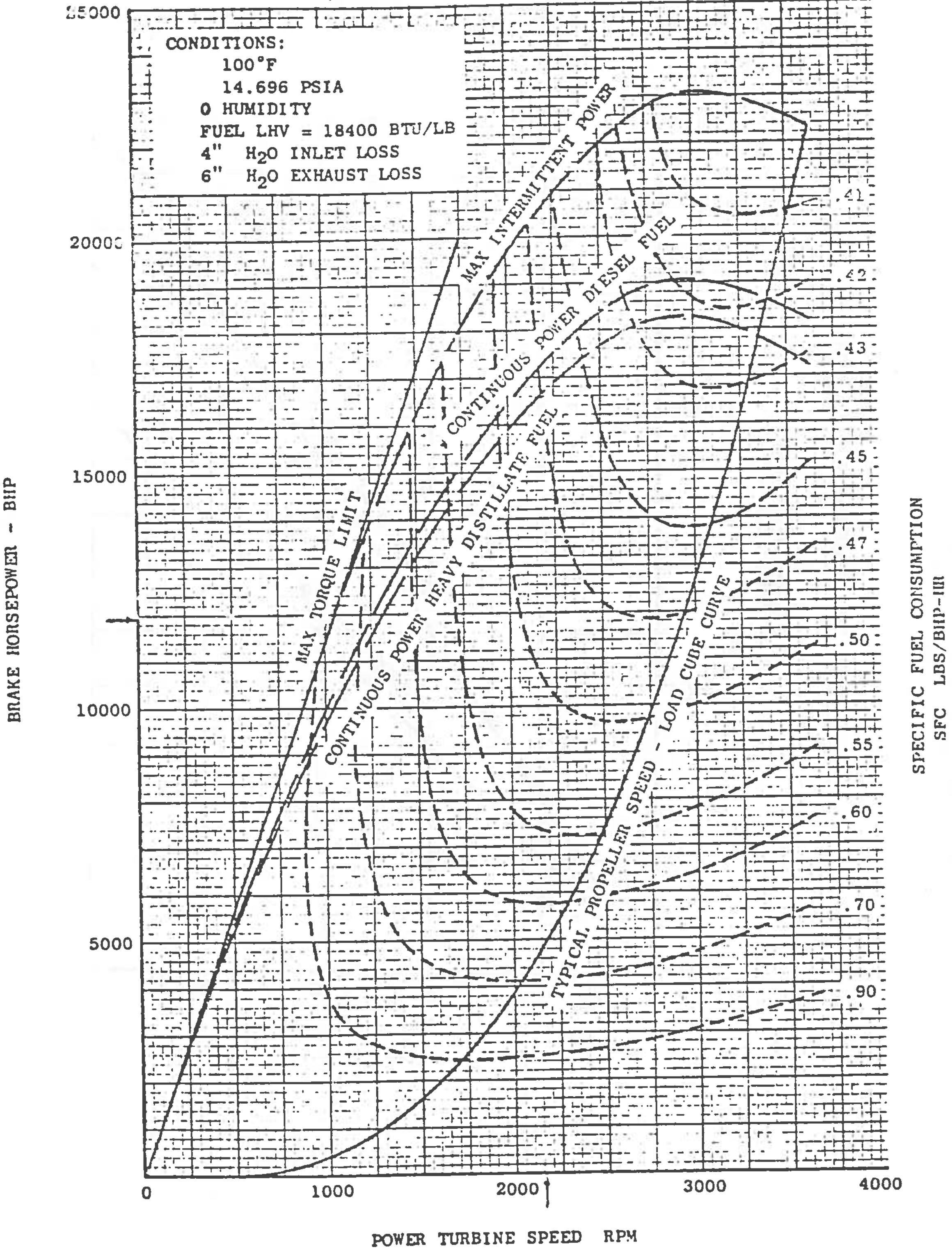
During the preliminary design stage an engine full map as provided will be used for determining SFC, recognizing that it is a conservative estimate. Normally the intake and exhaust systems for gas turbine ships are designed/sized to provide 4" and 6" $H_2O$ losses at full power (the DD 963 exhaust system being a unique solution evolved to meet a specific ship requirement). Thus at the actual endurance power condition the calculated losses will be lower.

The endurance fuel calculation during contract design will be based on actual corrections to the ambient conditions specified in DDS 200-1 (40 percent relative humidity) and actual calculated intake and exhaust losses. A typical correction curve for inlet and exhaust losses is provided for general background information.

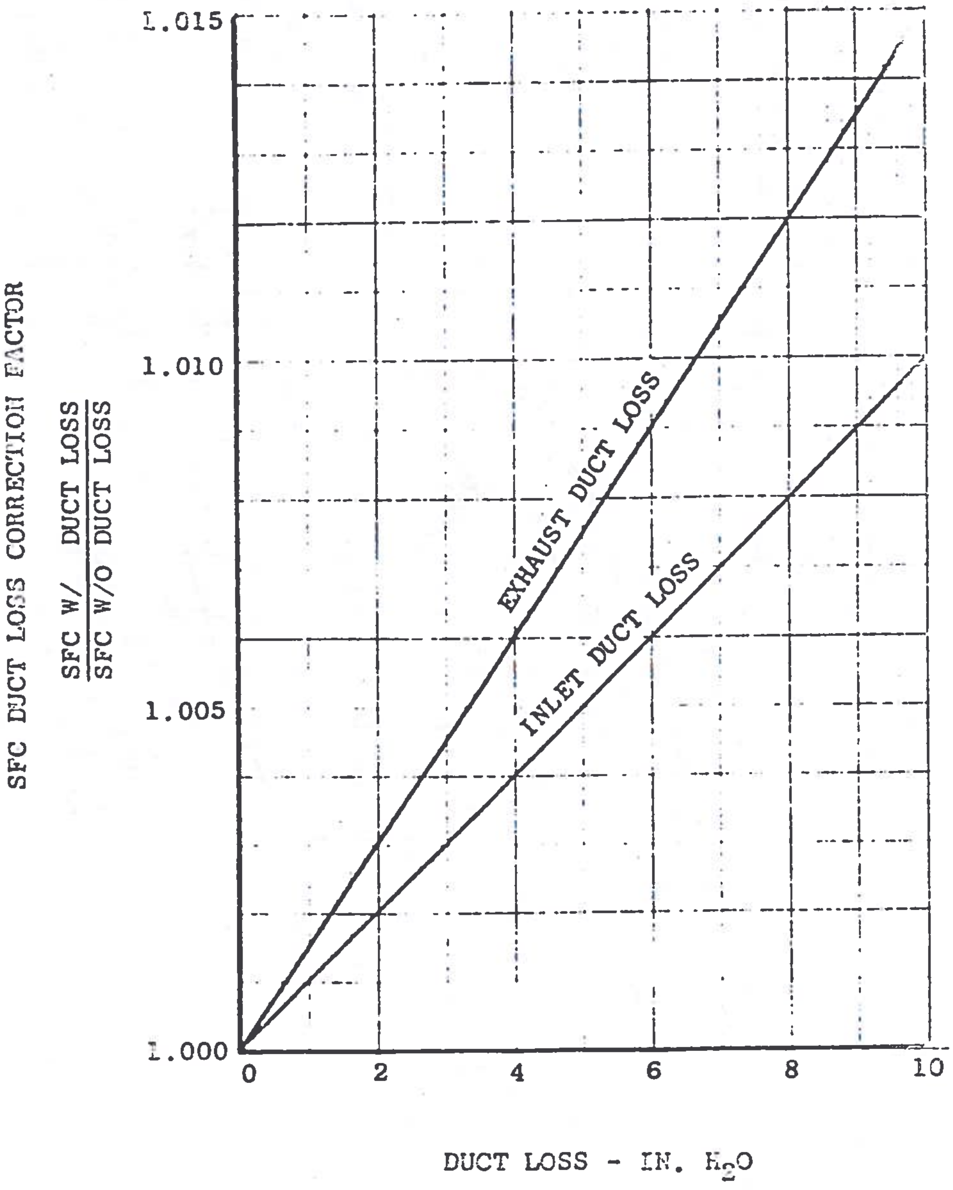
The completed calculation form is also provided.

# EDQP PROBLEM

LM2500 ESTIMATED ENGINE PERFORMANCE
(SUBJECT TO 5% PRODUCTION VARIATION)



## LM2500 ESTIMATED ENGINE PERFORMANCE
## SFC CORRECTION FOR DUCT LOSSES
### BHP = CONSTANT



# APPENDIX B
# SURFACE SHIP ENDURANCE CALCULATION FORM

DESIGN ____________________
PREPARED BY ____________________
CHECKED BY ____________________

|  | EXAMPLES<br>STEAM | DIESEL OR GAS TURBINE | SPECIFIC EDQP PROBLEM SOLUTION |
|---|---|---|---|
| (1) Endurance Required, Nautical Miles | 3,000 | 1,200 | 4000 |
| (2) Endurance Speed, Knots | 15 | 6 | 18 |
| (3) Full Load Displacement, Tons | 3,000 | 400 | 4000 |
| (4) Rated Full Power, SHP | 50,000 | 700 | 40,000 |
| (5) Design Endurance Power @(2) &(3), SHP | 3,000 | 150 | 10,600 |
| (6) Average Endurance Power, SHP: (5) x 1.10 | 3,000x1.10 = 3,300 | 150 x 1.10 = 165 | 11,660 |
| (7) Ratio, Avg. End SHP/rated F.P. SHP: (6)/(4) | 0.066 | 0.24 | .29 |
| (8) Average Endurance BHP: (6) / Transmission Efficiency | --- | 165 / 0.95 = 174 | 11,959 |
| (9) 24 Hour Average Electric Load, KW | 500 | 30 | 1,200 |
| (10) Calculated Propulsion Fuel Rate @(8), lbs/BHP-hr. | ---- | 0.479 | .494 |
| (11) Calc. Prop. Fuel Consumption, lbs./hr: (10)x(8) | ---- | 0.479 x 174 = 83.4 | 5908 |
| (12) Calc. S.S. Gen. Fuel Rate @(9), lbs./KW-hr. | ---- | 0.690 | .6 |
| (13) Calc. S.S. Gen. Fuel Consumption, lbs./hr: (12)x(9) | ---- | 0.690 x 30 = 20.8 | 720 |
| (14) Calc. Fuel Consumption For Other Services, lbs/hr. | ---- | 15.0 (heating) | — |
| (15) Total Calc. All-Purpose Fuel Consumption lbs/hr: (11) + (13) + (14) | ---- | 83.4 + 20.8 + 15.0 = 119.2 | 6628 |
| (16) Calc. All-Purpose Fuel Rate, lbs/SHP-hr: (15)/(6) or Heat Balance | 1.00 | 119.2/165 = 0.722 | .568 |
| (17) Fuel Rate Correction Factor Based on (7) | 1.04 | 1.04 | 1.04 |
| (18) Specified Fuel Rate, lbs/SHP-hr: (16) x (17) | 1.00 x 1.04 = 1.04 | 0.722 x 104 = 0.750 | .591 |
| (19) Avg. Endurance Fuel Rate, lbs/SHP-hr: (18) x 1.05 | 1.04 x 1.05 = 1.092 | 0.750 x 1.05 = 0.787 | .621 |
| (20) Endurance Fuel (burnable), Tons: (1) x (6) x (19)/(2) x 2240 | $\frac{3,000 \times 3,300 \times 1.092}{15 \times 2240} = 322$ | $\frac{1,200 \times 165 \times 0.787}{6 \times 2240} = 11.6$ | 736.8 |
| (21) Tailpipe Allowance Factor | 0.98 | 0.95 | .98 |
| (22) Endurance Fuel Load, tons: (20)/(21) | 322 /0.98 = 329 | 11.6 /0.95 = 12.2 | 752 |

REFERENCE FOR SOURCE DATA

Design Endurance Power ____________________
Transmission Efficiency ____________________
Calc. Prop. Fuel Rate ____________________
Calc. S.S. Gen. Fuel Rate ____________________
Calc. Fuel Consumption for Other Services ____________________
Heat Balance ____________________
Full Load Displacement ____________________

## 3.0 Installed SHP/Propellor Propulsive Efficiency Calculation

It is desirable that the Marine Engineer become familiar with the use of propeller series data to understand how to determine the probable propeller performance that can be achieved by the propulsion system. This should be done during the initial stage of preliminary design. After this point in the design most calculations of propeller performance will be based on computer program calculations using modifications of the basic (Lerbs) lifting line procedure.

For this problem all the information is provided as well as the procedure and a copy of the appropriate series propeller curve.

In the case of Naval Surface Ships, the actual requirements for the propeller to minimize cavitation inception at low speeds as well as CP hub mechanisms for gas turbine prime movers will result in lower actual propeller efficiencies than are predicted by propeller series data. This is quite satisfactory for the final range of acceptable propeller parameters will continually narrow as all the ship constraints are introduced during the complete propeller parametric analysis.

The actual calculations, constructed curves on the propeller series chart, and results are enclosed. The approximate range of acceptable PRPM for the 17 foot propeller is from 168 to 214.

While the TROOST Curve figure is quite convenient for this type of hand out problem, it is too small for great accuracy and it contains many values ($K_T$ and $K_Q$ lines). The curves should be explained to the students so that they understand what they are doing, as well as how to do it.

## 3.0 Installed SHP Requirement Problem Calculation Procedure

1. Minimum Acceptable Propeller Efficiency

$$e_p = \frac{EHP}{SHP \times e_r \times e_{rr}} = \frac{26{,}500}{40{,}000 \times \frac{.95}{.98} \times .99}$$

$e_p = .69$

2. Thrust Loading Coefficient @ Design $V_K$

$$T = \frac{EHP \times 325.7}{V_K\,(1\text{-}t)} = \frac{26{,}500 \times 325.7}{28(.95)}$$

$T = 324{,}476$ #'s

$D^2 = 17^2 = 289$ ft.$^2$

$V_a^2 = (V_K \times (1\text{-}w)\ 1.689)^2 = (28 \times .98 \times 1.684)^2$

$V_a^2 = 2148$ ft$^2$/sec$^2$

$$K_T/J^2 = \frac{T}{\rho\, D^2\, V_a^2} = \frac{324.476}{1.99 \times 289 \times 2148}$$

$K_T/J^2 = .2627$ (.263)

3. $K_T$ and J Calculated Values for TROOST CURVE

| J | $K_T$ |
|---|---|
| .8 | .168 |
| .9 | .213 |
| 1.0 | .263 |
| 1.1 | .318 |

- With these values a $K_T/J^2$ curve can be drawn on the enclosed TROOST 5.75 Propeller Curves.
- Where this curve intercepts the $K_T(K_S)$ lines for the P/D ratios from .9 to 1.4, the corresponding J ($\Lambda$) locations for each P/D ratio can be drawn on the propeller efficiency curve for that P/D ratio.
- A curve is then drawn through the efficency points.
- The points at which this curve is intercepted by the minimum efficiency line (.69) are the acceptable range of J (and RPM).

4. Calculation of RPM

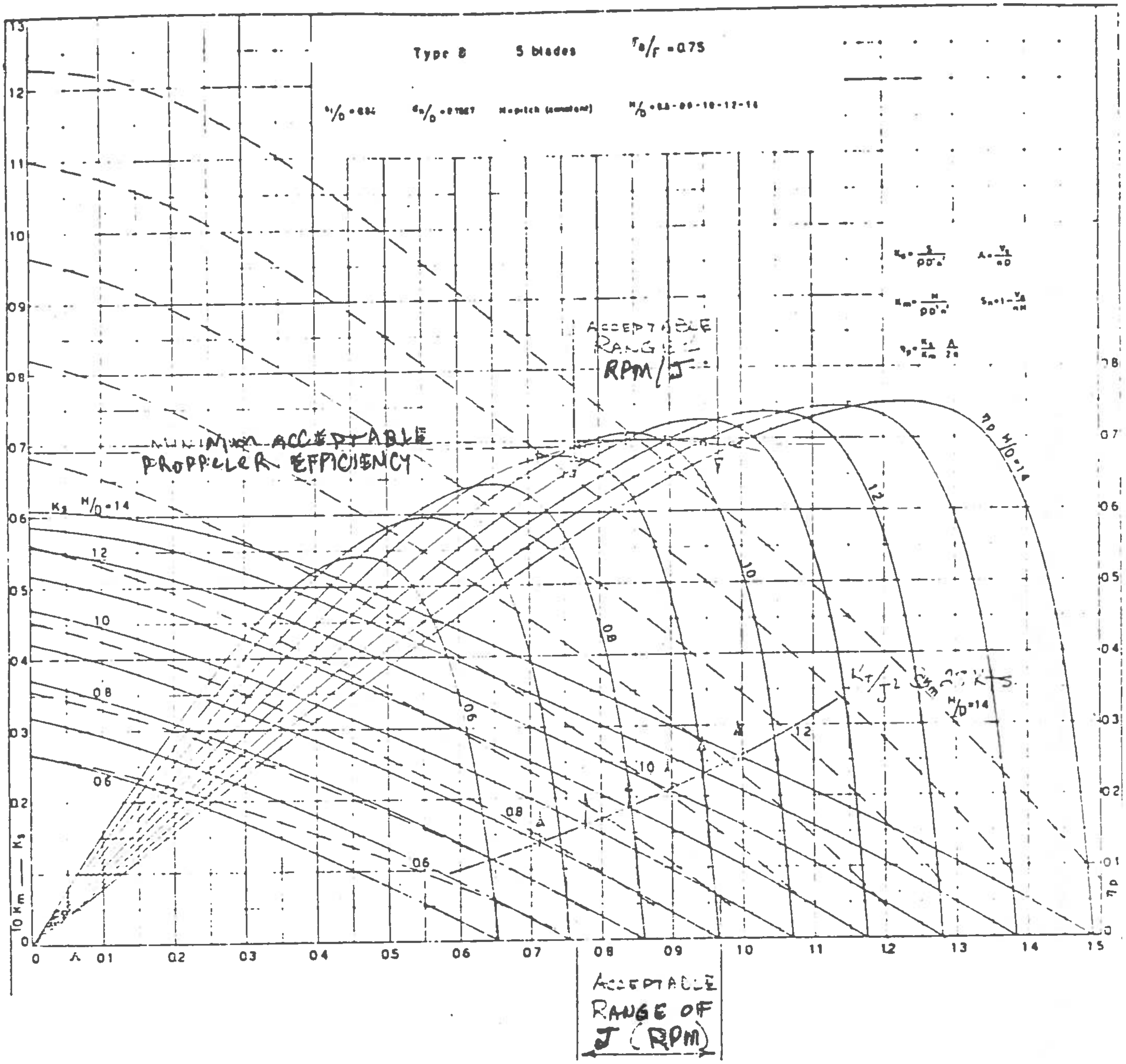
$$J = \frac{V_a}{ND}$$

$$N = \frac{V_a}{JD} = \frac{28 \times .98 \times 1.689}{J \times 17} \qquad \text{for } .765 \ \&\ .97$$

N = 3.56 & 2.81 RPS

N = 214 & 168 **RPM**

The nominal range of RPM is from ≈ 168 to 214



$$K_T = K_s, \quad K_Q = K_m, \quad J = \Lambda$$

## 4.0 Heat Balance Calculation Problems

The first problem is to conduct a heat balance around the DFT to determine the extraction steam required to balance the flows around the DFT. On the 20 knot heat balance flow diagram the required quantities and enthalpies that would permit solution of the problem without doing the DFT heat balance.

To simplify the problem, and avoid solving simultaneous equations, the student is to be given the actual heat extracted from the auxiliary heat line for the evaporators:

37,862,703 BTU/hr

On the calculation sheet a simplified diagram is provided, designating the flow quantities around the DFT, as well as the flow balance and heat balance calculations.

The completed instructor heat balance flow diagram contains the flows that are to be calculated, HP bleed extraction and combined feed flow from the DFT marked in red. Other related flows and enthalpies that were eliminated to prevent short circuiting the heat balance procedure are marked in with pencil.

The second problem is to determine the rated capacity of three machinery components, using the Full Power heat balance of the same ship (CVV). The actual calculations are quite simple. The intent of the problem is to acquaint the student with the background criteria for developing the equipment ratings for steam plants. It also highlights the manner in which the specific requirements for a given ship will dictate unique requirements for the propulsion system and the manner in which design practices are changed to provide improved operational plant characteristics.

The calculation contains the procedures used for rating the three machinery equipments based on the included design practices and the alternatives used for CVA's.

The calculated boiler overload rating of the boiler would be 216,329 #/hr. based on the 6 boiler full power evaporation rate of 180,274 #/hr. The ship as it is presently being designed is based on developing design full power with only 5 boilers in operation (one off the line). The required full power evaporation rate would then be approximately 220,000 #/hr., and the associated boiler overload rating would be 264,000 #/hr.

In the case of the main feed pumps, just considering the 6 boiler full power condition, the stated design practice is to provide two pumps rated at 150 percent of the design full power flow. One alternative that can be considered is providing three pumps rated at 150 percent of design full power flow. For CVA's, and nominally the CVV in this case, three main feed pumps are provided with a total capacity of 180 percent of design full power flow. With this capability two main feed pumps can supply the boilers up to the boiler overload rating with one feed pump in reserve. This third feed pump supplied from an elevated reserve feed tank provides the plant with a maximum evaporation rate of 60 percent of rated capacity if a casualty occurs to the DFT.

Based on the information provided and the design practices, the calculation of the main forced draft capacity is straight forward. Considering 5 boiler operation at full power, and actual air operations, the rated capacity of the forced draft blowers for the CVV would in fact be quite similar to the CVA's.

# CVV
# HEAT BALANCE & FLOW DIAGRAM
## 20 KTS

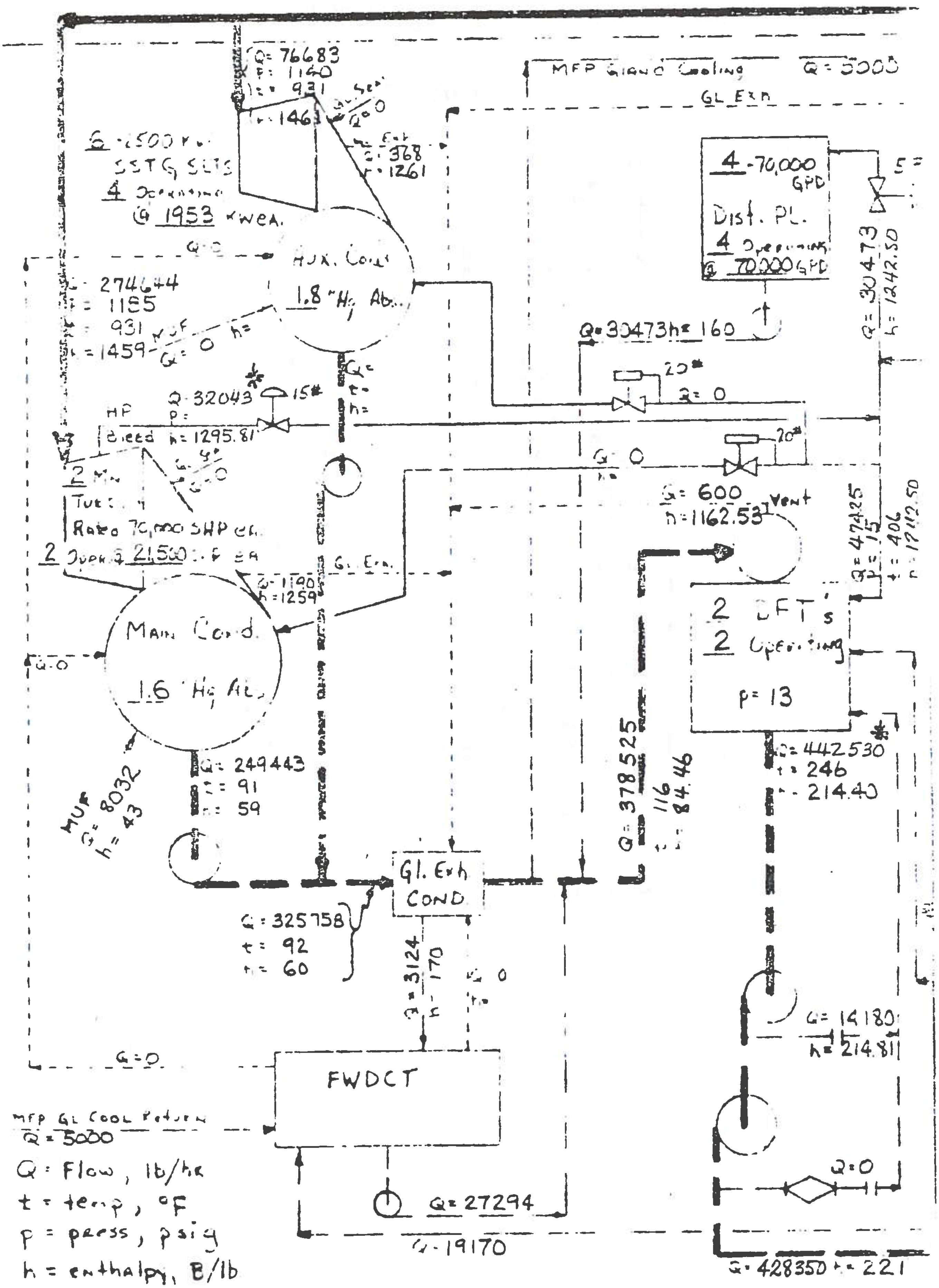
## OPERATING CONDITIONS

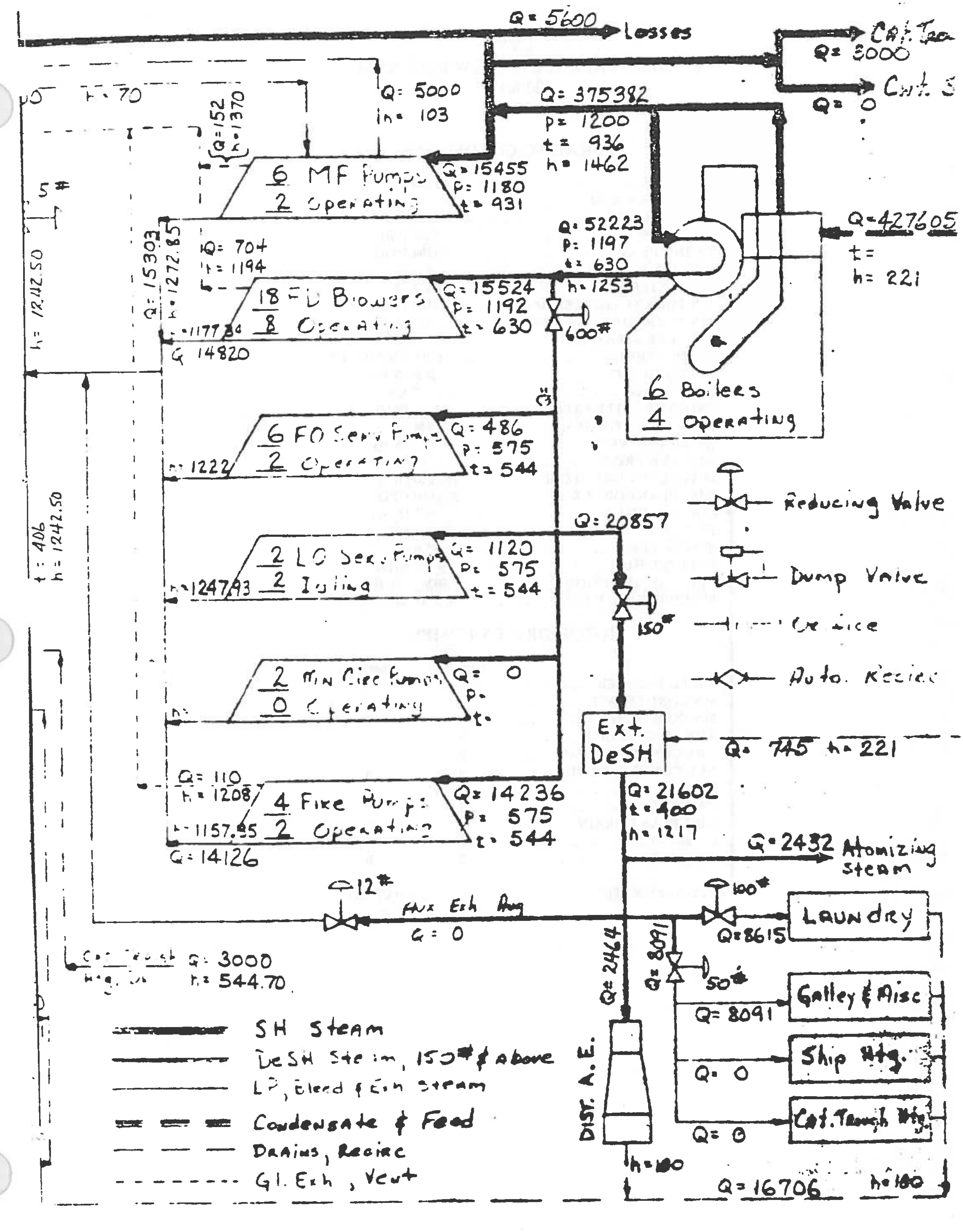
| | |
|---|---|
| SHP | 43000 |
| PROPELLER RPM | |
| DRUM PRESS | 1220 PSIG |
| SHO PRESS | 1200 PSIG |
| DESHO PRESS | 1197 PSIG |
| SHO TEMP | 936 °F |
| DESHO TEMP | 630 °F |
| MN TURB THROTTLE PRESS | 1185 PSIG |
| MN TURB THROTTLE TEMP | 931 °F |
| MN TURB STEAM RATE (NON-EXTRACTION) | 5.908 LB SHP-HR |
| MN COND VAC | 28.4 IN HG |
| ELEC LOAD | 7813 KW |
| SSTG THROTTLE PRESS | 1140 PSIG |
| SSTG THROTTLE TEMP | 931 °F |
| SSTG COND VAC | 28.2 IN HG |
| AUX EXH PRESS | 15 PSIG |
| DIST PLANT RATED CAP | 280,000 GPD |
| DIST PLANT OPER RATE | 280,000 GPD |
| FDB TOT HEAD | 14.4 IN WG |
| FW TEMP | °F |
| BOILER EFF | 86.8 % |
| FUEL OIL HNV | 19500 B/LB |
| FUEL CONSUMPTION | 30694LB/HR |
| SPECIFIC FUEL RATE | 0.7138 LB/SHP-HR |

## MOTOR DRIVEN PUMPS

| | Installed | Operating |
|---|---|---|
| MN FD BOOSTER | 6 | 2 |
| MN CONDENSATE | 4 | 2 |
| MN CONDENSER VAC | 4 | 2 |
| AUX CONDENSATE | 6 | 4 |
| AUX CONDENSATE VAC | 6 | 4 |
| AUX CONDENSER CIRC | 6 | 4 |
| FIRE | | 0 |
| FW DRAIN | 2 | 2 |
| DIST PLANT DRAIN | 4 | 4 |
| LO SERVICE | 2 | 0 |
| FO SERVICE | 2 | 0 |

| | |
|---|---|
| Air-Fuel Ratio | 240 Cf air<br>lb. fuel |


Q= 76683
P= 1140
t= 931
h= 1461
Gl. Seal Q= 0
Gl. Exh Q= 368 h= 1261
6 -2500 kw SSTG SETS
4 Operating
@ 1953 KWEA.
MFP Gland Cooling
Q = 5000
GL Exh
4 -70,000 GPD
Dist. PL.
4 Operating
@ 70,000 GPD
Q = 0
Aux. Cond
1.8" Hg Ab.
Q= 274644
P= 1155
t= 931
h= 1459
MUF Q= 0 h=
Q= 30473 h= 160
Q= 30473 h= 1242.50
20#
Q= 0
Q= 32043
P=
h= 1295.81
15#
HP Bleed
Q=
t=
h=
2 Main Turbines
Rated 70,000 SHP ea.
2 Oper @ 21,500 SHP EA
Gl. Seal Q= 0
20#
Q= 0
h=
Q= 600
h= 1162.53
Vent
Q= 47425
P= 15
t= 406
h= 1242.50
Gl. Exh.
Q= 1190
h= 1259
2 DFT's
2 Operating
p= 13
Main Cond.
1.6" Hg Ab.
Q= 0
Q= 249443
t= 91
h= 59
MUF Q= 8032 h= 43
Q= 378525
t= 116
h= 84.46
Q= 442530
t= 246
h= 214.40
Gl. Exh Cond
Q= 325758
t= 92
h= 60
Q= 3124
h= 170
Q= 0
h=
Q= 14180
h= 214.81
Q= 0
FWDCT
MFP GL COOL Return
Q= 5000
Q= 27294
Q= 19170
Q= 0
Q= 428350 h= 221
Q = Flow, lb/hr
t = temp, °F
p = press, psig
h = enthalpy, B/lb


Q= 5600 Losses
Cat. Trou
Q= 3000
Cnt. S
Q= 0
F- 70
Q: 5000
h= 103
Q= 152
h= 1370
Q= 375382
p= 1200
t= 936
h= 1462
6 MF Pumps
2 Operating
Q= 15455
p= 1180
t= 931
5#
Q: 15303
h= 1272.85
Q= 704
t= 1194
Q= 52223
p= 1197
t= 630
h= 1253
Q= 427605
t=
h= 221
h= 1242.50
18 FD Blowers
8 Operating
Q= 15524
p= 1192
t= 630
600#
h= 1177.34
Q 14820
6 Boilers
4 Operating
6 FO Serv Pumps
2 Operating
Q= 486
p= 575
t= 544
h= 1222
Reducing Valve
Q: 20857
Dump Valve
t: 406
h= 1242.50
2 LO Serv Pumps
2 Idling
Q= 1120
p= 575
t= 544
h= 1247.93
150#
Orifice
Auto. Recirc
2 Main Circ Pumps
0 Operating
Q= 0
p=
t=
h=
Ext. DeSH
Q= 745 h= 221
Q= 110
h= 1208
4 Fire Pumps
2 Operating
Q= 14236
p= 575
t= 544
h= 1157.95
Q= 14126
Q: 21602
t= 400
h= 1217
Q= 2432 Atomizing steam
12#
Aux Exh Hdr
G= 0
100#
Laundry
Q= 8615
Q= 2464
Q= 8091
50#
Cat Trough Htg.
Q: 3000
h= 544.70
Galley & Misc
Q= 8091
Ship Htg.
Q= 0
Cat. Trough Htg.
Q= 0
DIST. A.E.
h= 180
Q= 16706
h= 180
SH Steam
DeSH Steam, 150# & above
LP, Bleed & Exh Steam
Condensate & Feed
Drains, Recirc
Gl. Exh, Vent

# CVV
# HEAT BALANCE & FLOW DIAGRAM
# 20 KTS

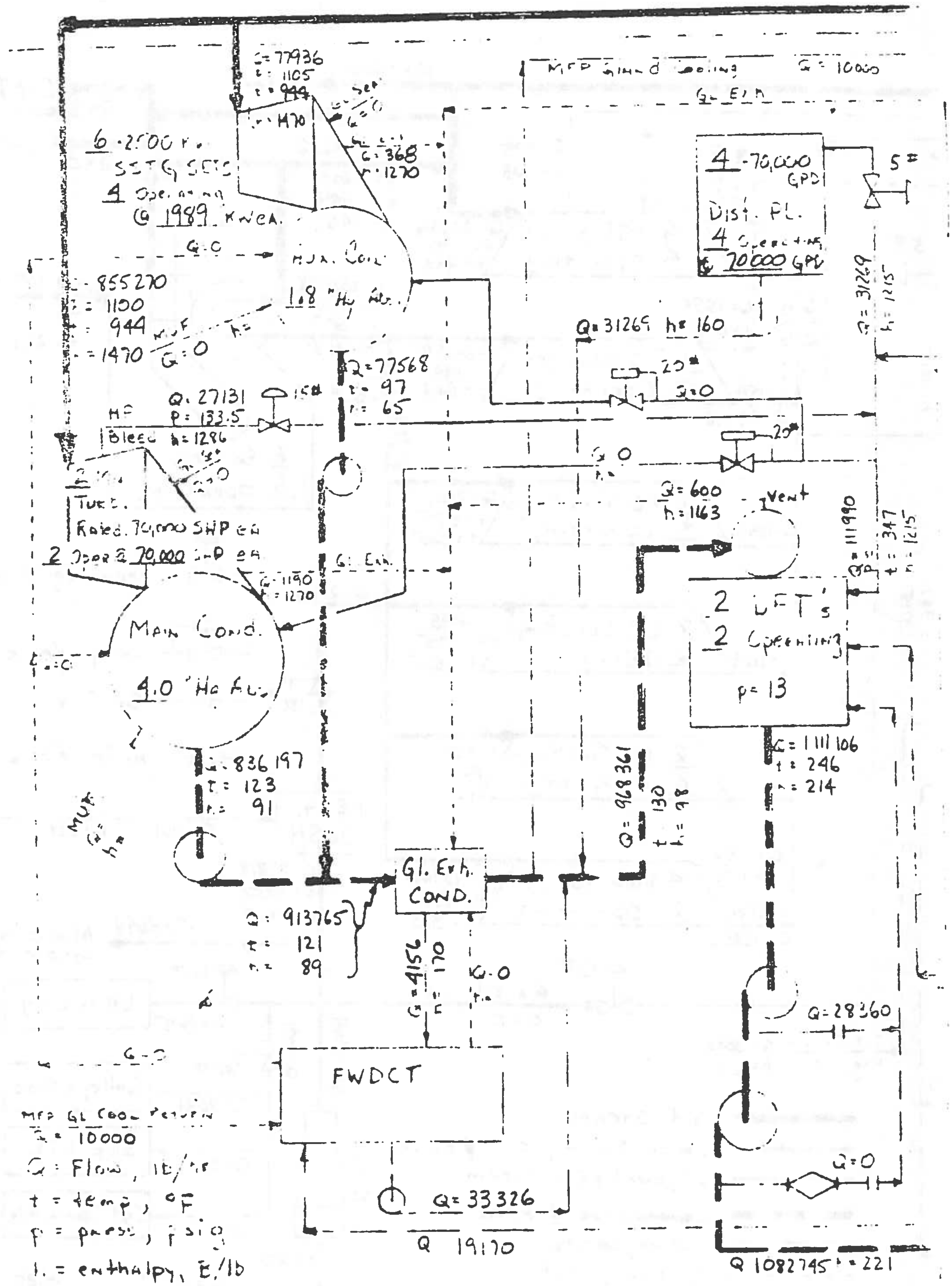
## OPERATING CONDITIONS

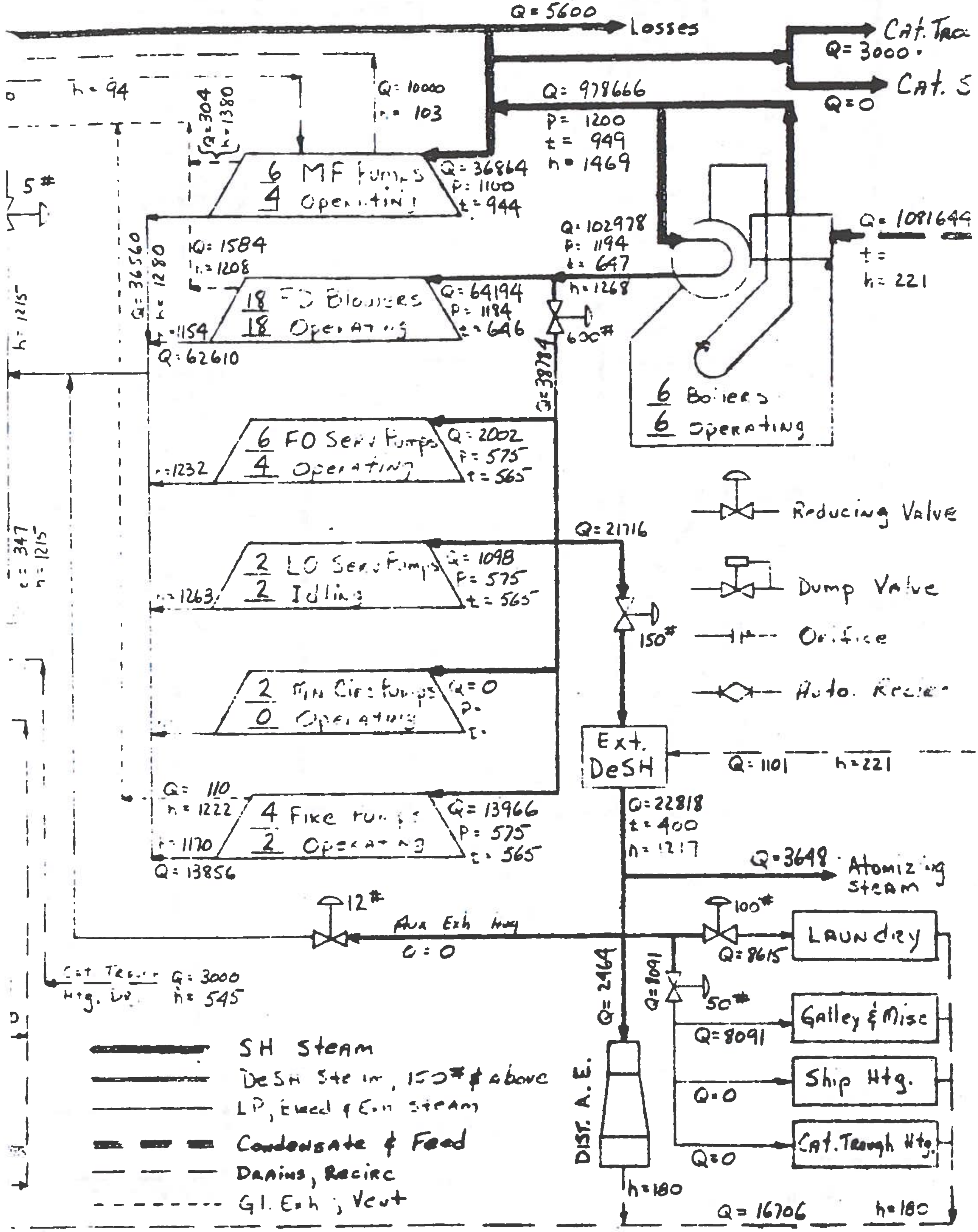
| | |
|---|---|
| SHP | 140,000 |
| PROPELLER RPM | |
| DRUM PRESS | 1257 PSIG |
| SHO PRESS | 1200 PSIG |
| DESHO PRESS | 1194 PSIG |
| SHO TEMP | 949 °F |
| DESHO TEMP | 647 °F |
| MN TURB THROTTLE PRESS | 1100 PSIG |
| MN TURB THROTTLE TEMP | 944 °F |
| MN TURB STEAM RATE (EXTRA TIGHT) | 5.995 LB/SHP-HR |
| MN COND VAC | 26.0 IN HG |
| ELEC LOAD | 7957 KW |
| SSTG THROTTLE PRESS | 1105 PSIG |
| SSTG THROTTLE TEMP | 944 °F |
| SSTG COND VAC | 28.2 IN HG |
| AUX EXH PRESS | 15 PSIG |
| DIST PLANT RATED CAP | 280,000 GPD |
| DIST PLANT OPER RATE | 280,000 GPD |
| FDB TOT HEAD | 39.2 IN WG |
| FW TEMP | °F |
| BOILER EFF | 85.0 % |
| FUEL OIL HHV | 19,500 B/LB |
| FUEL CONSUMPTION | 80261 LB/HR |
| SPECIFIC FUEL RATE | 0.5733 LB/SHP-HR |

## MOTOR DRIVEN PUMPS

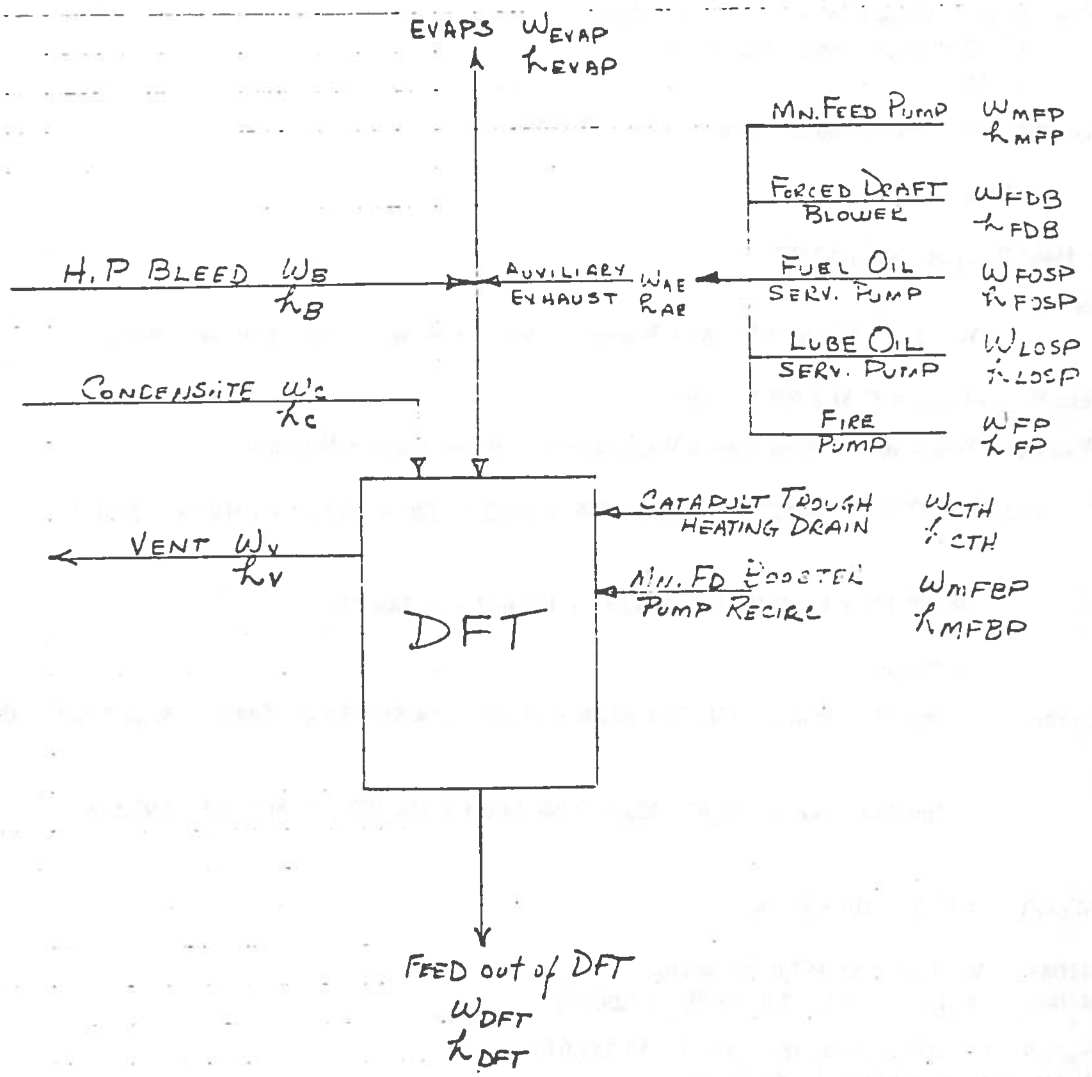
| | Installed | Operating |
|---|---|---|
| MN FD BOOSTER | 6 | 4 |
| MN CONDENSATE | 4 | 4 |
| MN CONDENSER VAC | 4 | 2 |
| AUX CONDENSATE | 6 | 4 |
| AUX CONDENSER VAC | 6 | 4 |
| AUX CONDENSER CIRC | 6 | 4 |
| FIRE | | 0 |
| FW DRAIN | 2 | 2 |
| DIST PLANT DRAIN | 4 | 4 |
| LO SERVICE | 2 | 0 |
| FO SERVICE | 2 | 0 |

| | |
|---|---|
| AIR-FUEL RATIO | 240 Cf Air/ lb. fuel |


Q = 77936
t = 1105
t = 944
h = 1470
Sep
Q = 0
Gl. Exh.
Q = 368
h = 1270
6 - 2500 KW
SSTG SETS
4 Operating
@ 1989 KW ea.
Q = 0
Aux. Cond.
1.8 "Hg Abs.
Q = 855270
t = 1100
t = 944
h = 1470
MUF
Q = 0
h =
Q = 77568
t = 97
h = 65
HP
Bleed
Q = 27131
p = 133.5
h = 1286
15#
Q = 0
2 Main Turb.
Rated 70,000 SHP ea
2 Oper @ 70,000 SHP ea
Gl. Exh.
Q = 1190
h = 1270
Main Cond.
4.0 "Hg Abs.
Q = 0
Q = 836197
t = 123
h = 91
MUF
Q =
h =
Gl. Exh. Cond.
Q = 913765
t = 121
h = 89
Q = 4156
h = 170
Q = 0
h =
FWDCT
Q = 0
MFP Gl Cool Return
Q = 10000
Q = Flow, lb/hr
t = temp, °F
p = press, psig
h = enthalpy, B/lb
Q = 33326
Q 19170
MFP Gland Cooling
Q = 10000
Gl. Exh.
4 - 70,000 GPD
Dist. Pl.
4 Operating
@ 70000 GPD
5#
Q = 31269
h = 1215
Q = 31269 h = 160
20#
Q = 0
20#
Q = 0
h =
Q = 600
h = 1163
Vent
Q = 111990
p = 347
t =
h = 1215
2 DFT's
2 Operating
p = 13
Q = 968361
t = 130
h = 98
Q = 1111106
t = 246
h = 214
Q = 28360
Q = 0
Q 1082745 t = 221


Q = 5600
Losses
Cat. Tra
Q = 3000
Cat. S
Q = 0
Q = 978666
P = 1200
t = 949
h = 1469
h = 94
Q = 10000
h = 103
Q = 304
h = 1380
5#
6 MF Pumps
4 Operating
Q = 36864
P = 1100
t = 944
Q = 102978
P = 1194
t = 647
h = 1268
Q = 1081644
t =
h = 221
Q = 36560
h = 1280
Q = 1584
h = 1208
18 FD Blowers
18 Operating
Q = 64194
P = 1184
t = 646
600#
h = 1154
Q = 62610
h = 1215
Q = 38784
6 Boilers
6 Operating
6 FO Serv Pumps
4 Operating
Q = 2002
P = 575
t = 565
h = 1232
Reducing Valve
Dump Valve
Orifice
Auto. Recirc
Q = 21716
2 LO Serv Pumps
2 Idling
Q = 1098
P = 575
t = 565
h = 1263
150#
t = 347
h = 1215
2 Min Circ Pumps
0 Operating
Q = 0
P.
T.
Ext. DeSH
Q = 1101
h = 221
Q = 110
h = 1222
4 Fire Pumps
2 Operating
Q = 13966
P = 575
t = 565
h = 1170
Q = 13856
Q = 22818
t = 400
h = 1217
Q = 3648
Atomizing Steam
12#
Aux Exh Hdr
Q = 0
100#
Laundry
Q = 8615
Cat Trough Htg. Dr.
Q = 3000
h = 545
Q = 2464
Q = 8091
50#
Galley & Misc
Q = 8091
Ship Htg.
Q = 0
Cat. Trough Htg.
Q = 0
DIST. A. E.
h = 180
Q = 16706
h = 180
SH Steam
DeSH Steam, 150# & above
LP, Bleed & Exh Steam
Condensate & Feed
Drains, Recirc
Gl. Exh., Vent

# 4.0 HEAT BALANCE CALCULATION AROUND the DFT



HEAT EXTRACTED FROM the AUXILIARY EXHAUST LINE to SUPPLY the DISTILLING PLANTS (EVAPS) is :

37, 862, 703 Btu/HR.

# CVV
# HEAT BALANCE & FLOW DIAGRAM
# 6 BOILERS — 140,000 SHP

## 1.1 Flow Balance Around the DFT

$W_{DFT} = W_{AE} + W_B + W_C + W_{CTH} + W_{MFBP} - W_{EVAP} - W_V$

$W_{AE} = W_{MFP} + W_{FDB} + W_{FOSP} + W_{LOSP} + W_{FP}$
$= 15303 + 14820 + 486 + 1120 + 14126$
$= 45855$

$W_{DFT} = 45855 + W_B + 378525 + 3000 + 14180 - 30473 - 600$

$= 410487 + W_B$

## 1.2 Heat Balance Around DFT

$W_{DFT}\, h_{DFT} =$
$W_{AE}\, h_{AE} + W_B\, h_B + W_C\, h_C + W_{MFBP} \quad h_{MFBP} + W_{ETRET} - W_{EVAP}\, h_{EVAP} - W_V h_V$

where $W_{EVAP}\, h_{EVAP} = 37{,}862{,}703$ Btu/hr

$W_{AE}\, h_{AE} = W_{MFP}\, h_{MFP} + W_{FDB}\, h_{FDB} + W_{FOSP}\, h_{FOSP} + W_{LOSP}\, h_{LOSP} + W_{FP}\, h_{FP}$

$= 15303 \times 1272.85 + 14820 \times 1177.34 + 486 \times 1222 + 1120 \times 1247.93 + 14126 \times 1157.35$

$= 19{,}478{,}424 + 17448179 + 593892 + 1{,}397{,}682 + 16{,}348{,}726$

$= 55{,}266{,}903$

$W_{DFT} h_{DFT} = 55{,}266{,}903 + W_B h_B + 378525 \times 84.46 + 14180 \times 214.81 + 3000 \times 544.7 - 37{,}862{,}703 - 600 \times 1162.53$

$= 55{,}266{,}903 + W_B h_B + 31{,}970{,}222 + 3{,}046{,}006 + 1{,}634{,}100 - 37{,}862{,}703 - 697{,}518$

$W_{DFT} h_{DFT} = 53{,}357{,}010 + W_B h_B$

$(410487 + W_B) h_{DFT} = 53{,}357{,}010 + W_B h_B$
$(410487 + W_B) 214.4 = 53{,}357{,}010 + W_B = 1295.81$

$W_B(1295.81 - 214.4) = 410487 \times 214.4 - 53{,}357{,}010$
$W_B(1081.41) = 88{,}008{,}413 - 53{,}357{,}010$

$$W_B = \frac{34{,}651{,}403}{1081.41}$$

$W_B = 32{,}043$ # HR.

## 2.1 Machinery Component Rating & Calculation

- Main Boiler

  Based on 6 Boiler-Full Power Heat Balance the total evaporation is 1,081,644 #/HR or 180,274 #/HR per boiler. Based on the information provided the boiler rating at the boiler overload condition is:

  Evap. Rate at Boiler Overload = 1.20 × F.P. Evap. Rate = 216329 #/HR

- Main Feed Pumps

  The design criteria states that two(2) main feed pumps are to be provided with a rating equal to 150% of full power flow. With three(3) main feed pumps that same total capacity could be provided;

  Full Power Flow = 1,082,745 #/HR
  F.P. Flow/Machy Plant = 541,373 #/HR
  Rated capacity for 3 pumps = 1.5 × 541,373
  = 812,060 #/HR

  $$\text{Rated capacity per feed pump, GPM} = \frac{812{,}060}{3 \times 7.88 \times 60} = 573 \text{ gpm.}$$

  For CVA's, the normal practice is to provide three main feed pumps rated at 180% of design full power rated flow.

  $$\text{GPM per pump} = \frac{1.8 \times 541{,}373}{3 \times 7.88 \times 60} = 687 \text{ gpm}$$

- Main Forced Draft Blowers

  Rated Capacity

  $$\text{CFM/Blower} = 1.21 \times \frac{80261 \times 260}{18 \times 60}$$

  $$= 23{,}380 \text{ Cfm}$$

  Rated total head

  $$\text{T.H., } {''}H_2O = 39.2 \times \left(\frac{1.21 \times 260}{240}\right)^2$$

  $$= 39.2 \times 1.72$$

  $$= 67.4\ {''}H_2O$$

### 5.0 Propeller Parametric Analysis

The background information and actual required data has been provided the student. Table 1 contains a tabulation of the net weight of the machinery and fuel. Figure 1 contains a plot of these net weights.

Based on these results the 16½ foot propeller would be selected. The full power design propeller RPM could range from 170 to 180. The nominal selection would be 180, at the present time. In the future the tendency would be to approach 170, assuming the machienry spaces can accommodate the equipment with adequate access for required onboard maintenance, casualty control, and removal of equipment for required maintenance; and that machinery vibration, logitudinal critical will not cause excess machinery plan vibration levels or large load variations in the main thrust bearing during steady state and maneuvering conditions near full power speed.

# TABLE 1

## NET WEIGHT OF MACHINERY AND FUEL

| Prop Dia. | FP PRPM | Prop Weight | Shafting Weight | Red. Gear Weight | Total Machy. Weight | Fuel Weight | Total Machy. + Fuel Wt. | Net Weight |
|---|---|---|---|---|---|---|---|---|
| 16.5 | 160 | 20.1 | 62.1 | 57.2 | 139.4 | 671.6 | 811.0 | + 1.7 |
| | 180 | | 61.8 | 50.4 | 132.3 | 677.0 | 809.3 | 0 |
| | 200 | | 61.6 | 44.7 | 126.4 | 688.0 | 814.4 | + 5.1 |
| | 220 | | 61.5 | 40.6 | 122.2 | 704.1 | 826.3 | + 17.0 |
| 17 | 160 | 23.4 | 70.6 | | 151.2 | 667.6 | 818.8 | + 9.5 |
| | 180 | | 70.3 | | 144.1 | 679.7 | 823.8 | + 14.5 |
| | 200 | | 70.2 | | 138.3 | 696.5 | 834.8 | + 25.5 |
| | 220 | | 70.0 | | 134.0 | 715.5 | 849.5 | + 40.2 |

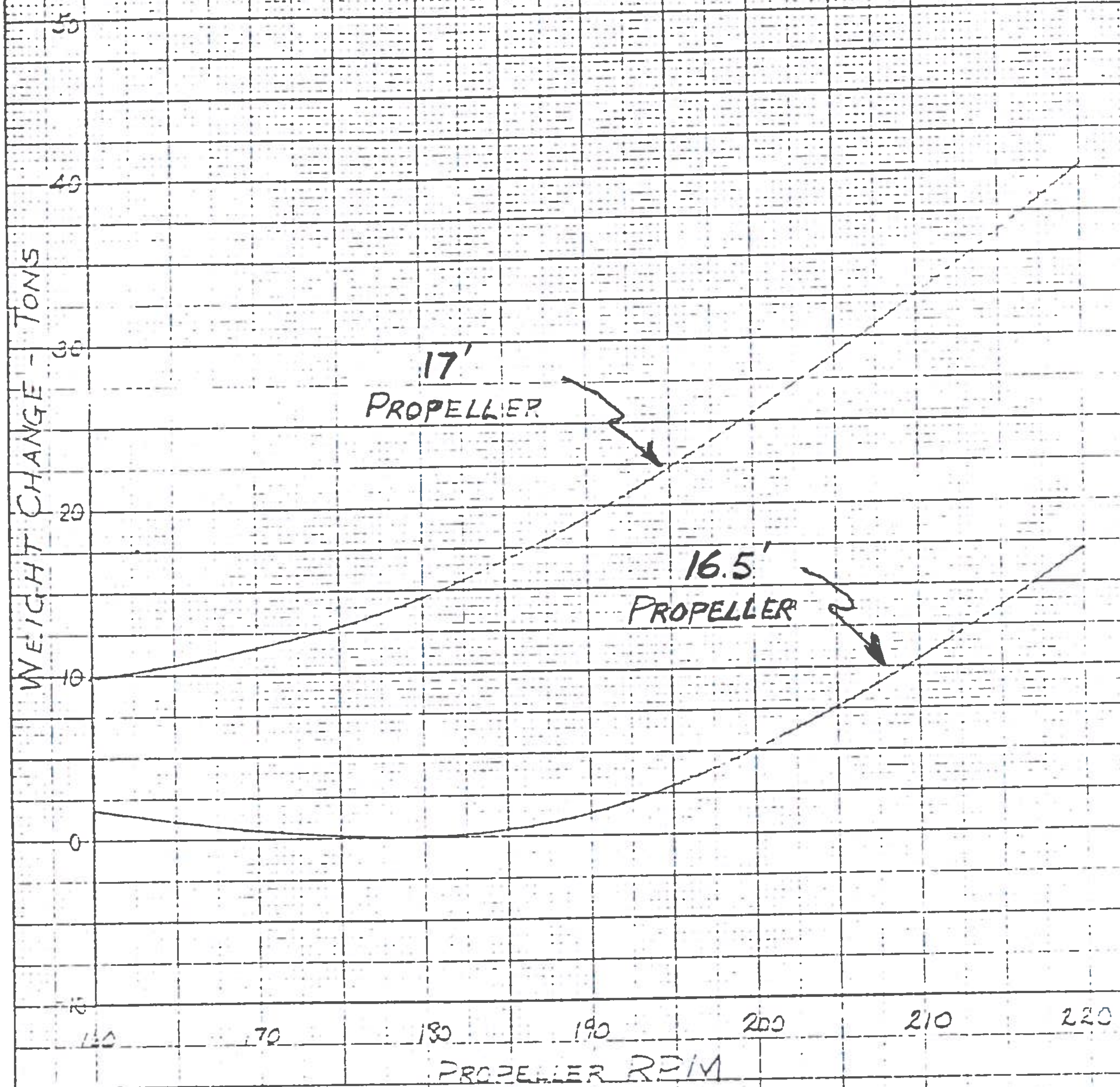


FIGURE 1

NET WEIGHT OF MACHINERY AND FUEL