THE FEASIBILITY OF ROTATING BOILERS FOR MARINE GAS TURBINE WASTE HEAT RECOVERY

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THESIS

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ABSTRACT

A theoretical study was conducted using existing correlations to examine the feasibility of using a small rotating boiler rather than a large conventional boiler to recover heat from a marine gas turbine and to generate steam without creating an excessive exhaust gas pressure drop.

The boiler studied was a 6 foot long, 8 inch diameter cylinder rotated directly by the turbine or by a separate electric motor. It was placed inside the exhaust gas duct of an Allison 501-K17 gas turbine engine, forming an annular gas passage. Assuming incompressible, fully developed, turbulent gas flow with constant properties, the effect of annular width upon heat transfer and gas pressure drop was calculated.

The analytical results indicate a maximum steam flow rate of 2550 lbm/hr and a minimum gas pressure drop of 11 inches of water for the boiler at a rotational speed of 3600 rpm. When these results are compared to the heat transfer and pressure drop characteristics of large, conventional boilers, it is apparent that the rotating boiler is not feasible for large naval vessels with large steam requirements. However, it does warrant consideration for use on small vessels with lower steam requirements, where size and weight limitations may be critical.

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NOMENCLATURE

a	Acceleration, ft/sec^2
A	Area, ft^2
A_f	Surface area of fins, ft^2
A_u	Unfinned area, ft^2
b	Fin spacing, ft
C_{fa}	Friction factor
C_l	Specific heat of the liquid, BTU/lbm-deg R
C_p	Specific heat, BTU/lbm-deg R
E	Modulus of elasticity, psi
f	Friction factor
g	Acceleration due to gravity, ft/sec^2
g_c	Gravitational Constant $\frac{ft-lbm}{lbf-sec^2}$
h_{fg}	Latent heat of vaporization, BTU/lbm
h_o	Outside heat transfer coefficient, BTU/hr-ft^2- deg F
h_i	Inside heat transfer coefficient, BTU/hr-ft^2- deg F
k	Thermal conductivity, BTU/hr-ft-deg F
l_g	Annulus width - feet
L	Length, feet
$\dot{m}_g$	Mass rate of exhaust gas flow, lbm/hr
$\dot{m}_s$	Mass rate of steam flow, lbm/hr
N	Number of fins
Nu	Nusselt number
p	Pressure, psia
p_i	Internal pressure, lbf/ft^2

p_v Vapor (steam) pressure, lbf/ft^2

P Perimeter of channel, feet

Pr Prandtl number

q Heat flow rate, BTU/hr

q_{act} Actual heat flow rate, BTU/hr

q_b Heat flow rate into the boiler, BTU/hr

r Mean radius, feet

r_c Radius of curvature, feet

r_i Inner radius of boiler, feet

r_o Outer radius of boiler, feet

r_2 Inner radius of two phase annulus, feet

$\bar{r}$ Mean radius of annulus, feet

Re_a Reynolds number for axial flow

Re_E Effective Reynolds number

R_h Hydraulic radius, feet

R_o Outer exhaust duct radius, feet

t Thickness of the boiler wall, feet

t_h Fin thickness, feet

t_w Thickness of the two phase annulus, feet

T_f Film temperature, deg F

T_G Exhaust gas temperature, deg F

T_i Temperature at inner wall, deg F

T_o Temperature at outer wall, deg F

T_{sat} Saturation temperature of vapor, deg F

T_{wi} Inside wall temperature, deg F

T_{wo} Outside wall temperature, deg F

U Overall heat transfer coefficient $\frac{\text{BTU}}{\text{hr-ft}^2\text{-deg F}}$

V_a	Axial velocity, ft/sec
V_E	Effective velocity ft/sec
x	Fin separation along axis, feet
β	Volume coefficient of expansion, 1/ deg R
θ	Fin angle, degrees
κ	Array efficiency
ϕ	Fin efficiency
ω	Angular velocity, rad/sec
ν	Kinematic viscosity, ft^2/sec
ρ_ℓ	Density of the liquid, lbm/ft^3
ρ_v	Density of the vapor, lbm/ft^3
$\rho_{2\phi}$	Density of the two phase mixture lbm/ft^3
σ_ℓ	Surface tension of the liquid
α	Coefficient of thermal expansion, $\frac{in}{in\text{-}deg\ F}$
γ	Density, lbm/ft^3
μ	Poisson's ratio
σ	Normal stress, psi
σ_o	Von Mises allowable stress, psi

I. INTRODUCTION

A. GAS TURBINES FOR MARINE APPLICATIONS

The development of gas turbines as propulsion and auxiliary power plants for marine applications is a relatively new field. Marine gas turbines have shown a tremendous growth during the period 1965-1971 in both propulsion and auxiliary uses [Ref. 1]. Gas turbines are utilized on various sizes of ships in which they are combined with either diesel or steam systems. Smaller gas turbines are also used as propulsion plants for smaller ships and as ship service power plants on larger ships.

These lightweight, compact, high-speed gas turbines offer many advantages over the conventional diesel or steam power plants. One of the major advantages of these gas turbines is the lightweight, small volume aspect of these units without reductions in available shaft horsepower. The resultant reduction in the machinery components necessary for these units increases the reliability of operation while decreasing the complexity of the piping and auxiliary machinery arrangements.

A major disadvantage of simple cycle gas turbine engines is the large heat loss of the high temperature exhaust to the atmosphere. These exhaust gas temperatures normally range from 950-1200 degrees F for normal operation. These losses represent as much as 75% [Ref. 2] of the total

chemical energy supplied to the system by the fuel combustion process. These heat losses greatly reduce the thermal efficiency of the gas turbine system. These high temperature exhaust gases also give the ship a large infra-red signature which is easily detected by heat-seeking missiles or infra-red spotting devices. The majority of the equipment mounted on the masts of a ship are the electronic antennas whose performance deteriorates rapidly at high temperatures. Therefore, reducing the temperature of these exhaust gases is essential.

B. WASTE HEAT RECOVERY BOILER

The reduction of these exhaust temperatures by heat removal to generate steam for uses on the ship, is the concept of the waste heat recovery boiler. The steam generated by this recovered heat is used to provide heating and hotel services, and to operate the distilling plants on the ship.

Sheldon and Todd [Ref. 3] discuss the design and physical characteristics of an operating heat recovery boiler. A simplified schematic of this system is presented in Figure 1. The boiler cycle begins with the feedwater as it enters the economizer and passes through tubes exposed to the exhaust gas flow. This process preheats the feedwater. The feedwater then enters the steam drum where the circulating pump removes the feedwater from the bottom of the steam drum. The water is then forced through the evaporator section where

a liquid-vapor mixture is formed as the temperature increases from the heat supplied by the exhaust gas flow. This mixture returns to the steam drum where the steam collects at the top portion and liquid collects at the bottom for another cycle. The steam is removed from the steam drum and passes through the superheater section to the steam outlet for distribution to the various steam systems.

These boilers may be unfired or separately fired [Ref. 3]. The unfired boiler uses the exhaust gas as the only source of heat, whereas the separately fired boiler uses a combustion process and the exhaust gas as heat sources.

C. TOTAL ENERGY CONCEPT

Greene and Warner [Ref. 2] discuss the concept of total energy applications to ship service power systems. The basis of this concept is to use as much of the combustion energy as possible. Any intermediate energy conversions or separate combustion processes have inherent losses which reduce the total amount of energy used from a single combustion process. The total energy application is presented schematically in Figure 2. For comparison purposes, the conventional system schematic is presented in Figure 3. A total energy application is reported to be capable of using as much as 80% of the energy provided by a single combustion energy addition [Ref. 2]. The primary advantages of the total energy application are stated to be [Ref. 2]:

1. Reduction of electrical power requirements; hence reduction in generator size and switch-gear.
2. Reduction in acquisition cost of the hardware.
3. Reduction in fuel consumption, since a greater proportion of the heating value of the fuel is utilized with waste heat recovery.
4. Elimination of hardware that is only infrequently utilized, such as air compressors and some motor-driven fire pumps.
5. Reduction in size (both volume and floor area) of the auxiliary machinery.

Disadvantages include:

1. Increase in cost of the ship plumbing as pneumatic and steam or hot water lines replace electrical.
2. Possibly an increase in maintenance costs compared to an all electric ship.

A theoretical cost, size, and weight comparison was conducted by Greene and Warner for approximately 2,800 kW ship service power loads. Three diesel-generator sets with fuel fired boilers were replaced by four gas turbine-generator sets with two waste heat recovery boilers. The results [Ref. 2] indicate a possible total savings of 53.4 long tons of weight, 33.6 percent in total costs, and 150 square feet of deck space with the total energy application. Further reductions in size and weight are possible with smaller components for the total energy systems.

Continuous efforts are being made to reduce the size and weight of gas turbines and components. Current waste heat boilers for marine applications are scaled down versions of industrial waste heat boilers [Ref. 3], and little effort has been expended in reducing the size of these boilers. The comparative sizes and weights of the gas turbine and the boiler are listed in Tables II and III, showing a significant weight difference of 1,280 vs. 16,200 pounds. Therefore, the waste heat boiler is a component for consideration in reducing the weight and volume of the system.

The boiler equipment which is external to the exhaust flow ducting in Figure 1 includes the steam drum, piping, and pumps. The primary purposes of the steam drum are to separate the steam from the liquid and store the heated water and steam. If either of these functions is performed in the evaporator (generating) section of the boiler, the size of these external boiler components can be reduced or the component eliminated, leading to an overall reduction in size and weight.

D. ROTATING BOILER

Recent studies conducted on a small, vertically oriented, rotating boiler, which was a 4 inch diameter, 2 inch long cylinder, indicated that stable boiling was obtained for heat fluxes as high as 818,000 BTU/hr ft^2 [Refs. 4 and 5]. The electrical input to the boiler heating coils [Fig. 4] was as high as 42 kilowatts.

Tests were conducted to determine the quality of the steam generated, boiling heat transfer coefficients, and the effects of inlet liquid subcooling. The outstanding results reported for this experimental boiler warrant its consideration as a steam generator.

In order to develop an understanding of the operation of this boiler unit a cross section view of the boiler is presented in Figure 4. The boiler cycle begins as feed water passes through the hollow rotating shaft and enters a rotating storage chamber through a float-valve arrangement. From this chamber, water is fed through small feed supply holes to the boiler cavity. The rotation of the boiler forces the liquid to form an annulus on the inside of the cylindrical cavity. The heat is supplied by electrical heaters embedded in the boiler wall. As the water is heated to the boiling temperature, the vapor "rises" from the wall and collects in the center portion of the cavity. At the liquid-vapor interface, the vapor breaks away from the liquid and carries some of the liquid with it into the vapor space. The liquid is returned to the interface by the force, due to rotation, on the heavier liquid particles. Excellent motion pictures of this separation process are available as indicated in Reference 4. Therefore, the rotating boiler is able to separate the liquid from the vapor in the steam generating cavity itself.

Tests for determining the quality of the steam were conducted using two calorimeters to test the steam samples.

The reported quality of the steam was always greater than 99 percent, and in some tests, several degrees of superheat were noted [Refs 4 and 5]. These results demonstrate the excellent steam separation ability within the generating section of the rotating boiler.

The rotation of the boiler provided accelerations which were up to 475 g's, for the experiments conducted on this unit. These high accelerations caused an exceptionally stable annulus to form, with no noticeable effect from the earth's gravitational field. The rotation also enhanced the boiling heat transfer coefficient on the inside surface of the boiler. Boiling coefficients were determined to be as large as 9000 BTU/hr-ft^2-°F, with heat fluxes up 505,000 BTU/hr-ft^2 [Ref. 4]. Further testing [Ref. 5] was conducted to study the effects of rotation on nucleate boiling in the rotating boiler. At very high accelerations and heat fluxes, the liquid vapor interface became stable and smooth. The results of the additional experiments led to the postulated theory that much of the total heat flux was transferred by direct evaporation at the liquid-vapor interface, with little bubble nucleation.

E. ROTATING BOILER ENGINE

In 1972, Doerner and Bechtold [Ref. 6] reported on a Rankine-cycle engine studied in the DuPont Laboratory which used a rotating steam generator section. This rotating boiler-nozzle-condenser unit produces the vapor used to

drive a single turbine wheel which develops approximately 20 horsepower of output power.

A cross section view of the engine is presented in Figure 5, with the following description of operation [Ref. 6]:

> External burner (A) boils fluid in annular chamber where liquid (B) is held on outer surface by centrifugal force. Hot vapor travels through tubes (C) to nozzle ring (D), expands through ring and drives turbine wheel (E). Engine power from turbine shaft drives external load (not shown). Vapor goes to condenser tubes (F). Condensed fluid flows back to channel (G). Centrifugal action forces liquid from channel through tubes (H) back to boiler (B). Cooling air is pulled into hollow center of condenser cylinder (I), then between condenser fins (J) by viscous drag rotating fin assembly.

The engine uses an external electric motor to rotate the boiler-nozzle-condenser unit at 2,500 rpm with the turbine wheel contra-rotating at 27,500 rpm. This rotation is reported to allow the nucleate boiling process to be replaced by smooth, stable boiling for high heat fluxes. The working fluid for this engine is an organic fluid, not water, since a dense vapor fluid allows the single stage turbine to operate with a good efficiency at low pressure.

F. HUETTNER ENGINE

The rotating boiler is not a recent innovation. As early as 1931, a patent application was filed with the German Patent Office for the principle of the Huettner engine [Ref. 7]. This engine was powered by the steam generated in a fuel fired, rotating boiler. The steam was passed through two nozzles to a single stage turbine which powered a 60 watt, direct-current generator. A small

electric motor started the boiler rotating and was disengaged when the boiler rotation could be sustained by the turbine. Unfortunately most of the Huettner engine test results were lost or destroyed during World War II. Designs were made for 30, 100, and 500 horsepower engines although none were built. In 1947, the Huettner family moved to the United States to conduct test and evaluation studies of the engine at the Naval Boiler and Turbine Laboratory. Personal correspondence [Ref. 8] from the NBTL representative assigned to the Huettner project states that shortly after arrival in the United States, Mr. Huettner became ill and died. His two sons continued the project for an additional 18 months before it was cancelled. During this time no new engines were built.

G. PROBLEM STATEMENT

The gas turbine provides power for many ships, but has a very high heat loss to the atmosphere through the exhaust gas. Waste heat recovery boilers have been designed to recover some of this heat to generate steam for other shipboard uses.

The excellent performance of the rotating boiler, with high quality steam generated, indicates the possible use of the rotating boiler unit inside the exhaust duct of a gas turbine to generate steam. If this is a feasible operation, it is anticipated that considerable size and weight can be saved in the boiler unit.

H. THESIS OBJECTIVE

The objective of this thesis is to determine the feasibility of developing a rotating boiler which can generate steam using only the waste heat from a gas turbine, while maintaining a small exhaust gas pressure drop.

II. CRITERIA FOR BOILER DESIGN

A. BOILER ARRANGEMENTS

The location and orientation of the rotating boiler can be selected to allow various modes of operation. The power to rotate the boiler can be supplied directly from the gas turbine or can be provided from a separate source such as an electric motor. If a separately driven mode is used, the boiler can be placed in either a cross flow or longitudinal flow position. Three basic flow situations are presented in Figures 6, 7, and 8 and a representative cross-section to illustrate the internal operation is presented in Figure 9.

Gas turbines deliver the power developed via the shaft which can extend through the exhaust flow (hot-end power takeoff) or back through the compressor (cold-end power takeoff). The cases presented in Figures 6 and 7 are for cold-end power takeoff gas turbines. This avoids complications in the feedwater inlet and steam outlet piping arrangements.

Each of these three systems has several advantages and disadvantages. Consideration is given to each situation in the following discussion.

1. The Direct Drive Annular Flow System

The system arrangement depicted in Figure 6 is the direct drive installation. Possible advantages of this arrangement include annular flow with heat transfer along

the entire length of the boiler. The boiler starts rotating as soon as the gas turbine starts. The high rotational speeds greatly increase the stability of the two-phase annulus.

Some of the disadvantages of this system include the necessity of securing the gas turbine in the event of a boiler failure. Seals between the rotating components and the stationary steam and feed piping are very important and must withstand severe wear without failure. The rotational speed can not be altered. The sensitivity of the gas turbine to vibrations requires a stringent dynamic balance of the attached boiler unit.

2. <u>The Separate Drive Annular Flow System</u>

The system depicted in Figure 7 is essentially the same as the previous case except an external motor provides the power to rotate the boiler unit. Some of the advantages of this system include the flexibility of the system in that the speed of rotation may be varied, a boiler casualty does not cause the gas turbine to be inoperable, and the dynamic stability requirements are not as stringent. The selection of lower speeds of rotation reduces the wear on the seals and bearings in the system.

Some of the disadvantages of this system are the addition of the electric motor to the system and the requirement to locate a support bearing directly in the exhaust gas flow.

3. The Separate Drive Cross-Flow System

The separate drive cross-flow system (Fig. 8) offers several advantages that are readily apparent. The boiler is supported by bearings external to the exhaust gas flow. The drive motor may be connected to one end and the steam taken out the other end to avoid the possibility of interference between these two functions. This system also offers independent operation, variable speed of rotation and reduced dynamic stability requirements as advantages.

The disadvantages of this system include the addition of the electric motor to the system, and the short path length presented to the exhaust gas flow. This reduces the pressure drop across the boiler but also reduces the amount of heat transferred by the exhaust gas.

B. FACTORS INFLUENCING THE BOILER DESIGN

From the above discussion it is observed that many factors enter into the selection of the system to be studied.

The determination of what factors must be studied stems from the objective of this thesis. The primary effects must be concerned with the heat transfer to the boiler and the ability to generate steam with this heat. The gas pressure drop across the boiler is important for estimating the adverse effect of the boiler on the gas turbine. This pressure drop acts as a back pressure for the gas turbine exhaust to overcome, and from thermodynamic considerations [Ref. 9], reduces the thermal efficiency of the gas turbine operation.

The addition of external fins to enhance the outside heat transfer to the boiler and the increase of the pressure drop across the external fins must also be included.

Basic stress computations must be made to determine the limitations on the size of the rotating boiler unit. The material for constructing the boiler should also be selected prior to any thermal, hydraulic analysis.

Several factors need not be considered here although they are important for a final analysis. Items such as bearings, seals, pumps, piping, external components, the feed system, controls, power drive connections, and exhaust ducting alterations are considered within the realm of modern technology. Other factors being neglected include friction losses in the bearings, the work required to rotate the boiler, dynamic loading conditions, and boiler fabrication problems.

C. GAS TURBINE SELECTION

In order to proceed in the analysis, the selection of a gas turbine system is considered at this time. This selection was greatly simplified by the recent decision of the United States Navy to construct the DD 963 class vessel with a conventional, non-rotating waste heat recovery boiler in the ship service power system [Ref. 10].

The gas turbine used for the ship service power system is the Allison 501-K17 engine [Refs. 11 and 12] which is rated at 3,860 horsepower and 13,820 rpm for continuous operation. This turbine offers the following features:

1. Single shaft

2. Compact size

3. Light weight

4. Exhaust heat recovery capability

A complete listing of the gas turbine characteristics is presented in Table I. A basic diagram of the gas turbine is presented in Figure 10.

Selection of this gas turbine not only defines the system parameters for the rotating boiler, but allows a comparison to be made with the conventional waste heat recovery boiler. No intention of offering the rotating boiler as a competitive design is intended however.

The conventional non-rotating boiler for the present installation is provided by the Condenser Service and Engineering Company, Inc. The only dimensional data obtained were the overall size and weight of the boiler in this installation [Ref. 13]. Details of the boiler construction and associated components were not available. Operational specifications [Ref. 13] for the boiler are presented in Table II.

D. BOILER SIZING DUE TO STRESS LIMITATIONS

The stresses on the rotating boiler are of major importance to the operation of the system. Developing a fairly accurate estimate of these stress values is essential for determining an allowable size for the boiler and for selecting a material which is able to withstand these stress loads.

TABLE I

SPECIFICATIONS FOR THE ALLISON 501-K17 GAS TURBINE

SIZE:

length - 8 feet

width - 28 inches

height - 30 inches

weight - 1,280 lbs.

FUEL: uses liquid fuel with a minimum lower heating value of 18,200 BTU/lb.

POWER: 5500 horsepower

SPEED: 13,820 rpm

POWER-SHAFT: single shaft connecting turbine to compressor and on to the generator through a reduction gear

INLET AIR:

pressure - 1% of the ambient pressure

temperature - 40 deg F to 120 deg F

flow rate - 26,000 cfm

COMPRESSOR SECTION:

number of stages - 14 axial flow

pressure ratio - 9:1

coupled directly to the turbine

bleed air capability

COMBUSTION UNIT:

number of combustion chambers - 6 of the flow through type

TURBINE UNIT:

number of stages - 4

cooling - air cooled

inlet temperature - 1970 deg F max

EXHAUST GASES:

pressure - 10 inches of water above atmospheric

temperature - 1035 deg F max

velocity - 187 ft/sec (at exhaust tailcone outlet)

mass flow rate - 39.5 lbm/sec max

volume flow rate - 63,500 cfm max

exhaust duct size - 36 inches diameter

TABLE II

CONSECO NON-ROTATING WASTE HEAT RECOVERY BOILER SPECIFICATIONS AND PARAMETERS

SIZE:

length - 10 ft. 6 inches max

width - 10 ft. max

height - 7 ft. 2 inches max

weight - 16,200 lbs. max

INLET EXHAUST GAS CONDITIONS:

temperature - 1035 deg F max

mass flow rate - 39.5 lbm/sec max

volume flow rate - 63,500 cfm max

FEED WATER:

temperature - 130 deg F min

flow rate - 7.5-24.5 gpm

pressure - 150 psig @ 60 gph

purity - .15 epm chloride and .20 epm max

CONDENSER (sea water side)

inlet pressure - 50 psig

outlet pressure - 35.3 - 48.6 psig

inlet temperature - 28 - 95 deg F

average temperature rise - 55 deg F

maximum sea water outlet temperature - 140 deg F max

capable of condensing full steam rate

OUTLET EXHAUST GAS:

temperature - 405 deg F

PRESSURE DROP ACROSS BOILER: 4.5 inches of water max

STEAM REQUIREMENTS

output capacity - 7,000 - 12,000 lbm/hr of steam

design pressure - 120 psia

normal operating pressure - 100 psia

steam quality - dry saturated steam with no more than 1% moisture content

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The primary stresses considered are the normal stresses due to internal pressure, rotation (hoop), end-dislocation and thermal stresses on a rotating cylinder.

1. Pressure stresses

The pressure on the inside of the cylinder is assumed to be uniform as indicated in Figure 11a. The feed-water line is not shown for clarity. The resulting stress relations [Refs. 14 and 15] are,

$$\sigma_\theta = \frac{p_i r}{144t} \qquad \sigma_r = \frac{p_i}{144}$$

$$\sigma_z = \frac{\sigma_\theta}{2}$$

where σ is the normal (perpendicular) stress in the indicated direction using a cylindrical coordinate system. The term p_i is the internal pressure loading, r is the radius, and t is the thickness. The σ_z component is based on the assumption that the cylinder is closed on both ends. Although the boiler is not closed on both ends, the seals provide limitations on the movement of the boiler, and this condition is assumed to be the same as having both ends capped. Therefore these stress components are included.

The internal pressure consists of the vapor pressure, p_v, and a hydrostatic pressure loading produced by the two phase mixture in the annulus. Assuming the two phase mixture moves as a solid body and neglecting the earth's gravitational force on the mixture, then for uniform rotation about the axis [Ref. 16], the internal pressure is

$$p_i = p_v + \frac{\rho_{2\phi}\omega^2}{2g_c}(r_i^{\ 2} - r_2^{\ 2}).$$

In the above expression, $\rho_{2\phi}$ is the density of the two phase mixture, g_c is the gravitational constant, r_2 is the inner radius of the mixture annulus, r_i is the outer radius of the mixture and ω is the angular velocity. Since the density of the two phase mixture is not known, the density of water is used in later calculations.

2. Thermal Stresses

The thermal stresses are developed because of a temperature gradient across the wall of the cylinder. Assuming that the thickness of the wall is small in comparison to the outer radius, the thermal stress relationships [Ref. 17] are

$$\sigma_{\theta o} = \sigma_{zo} = \frac{\alpha E(T_i - T_o)}{2(1-\mu)}\left(1 - \frac{m}{3}\right)$$

$$\sigma_{\theta i} = \sigma_{zi} = \frac{\alpha E(T_i - T_o}{2(1-\mu)}\left(1 + \frac{m}{3}\right)$$

$$\sigma_{r_o} = \sigma_{r_i} = 0$$

where

$$m = \frac{r_o}{r_i} - 1,$$

α is the coefficient of thermal expansion, μ is the Poisson's ratio, and E is the modulus of elasticity. A radial temperature gradient of 100 deg F is assumed for the preliminary

calculations and is compared to the resultant temperature gradient as determined later in the analysis.

3. Rotational Stresses

The rotational stresses developed in the wall are hoop stresses. The two phase annulus has already been included in the pressure stress relations. The governing relations for rotational stresses [Ref. 17] are

$$\sigma_\theta = \frac{\gamma \omega^2 r^2}{144\ g_c}$$

$$\sigma_z = \sigma_r = 0$$

where γ is the density of the metal.

4. End Dislocation Stresses

These stresses are local stresses which act on a small area of the boiler. They act near the junction of the cylinder and the closure disc or end plate (Fig. 11b). These stresses may cause local yielding, but by careful selection of the metal, rupture at this juncture is avoided. The possibility of this yielding however, must be included in the determination of the necessary clearances for the seals and bearings.

This stress developed is a combination of the effects of rotation on the cylinder and the closure disc [Ref. 18]. As the boiler rotates, both the cylinder and the closure disc tend to expand at different rates. The difference between the two deflections is the driving force for the stress at this location. Since these stresses are local

stresses and are not included in the yield criterion a full development is not presented. However an order of magnitude determination is conducted to insure that tensile strength is not exceeded by this local stress.

5. Von Mises Yield Condition

The yield strength of the material is determined from the Von Mises yield condition [Ref. 14] as

$$(\Sigma\sigma_r-\Sigma\sigma_\theta)^2 + (\Sigma\sigma_\theta-\Sigma\sigma_z)^2 + (\Sigma\sigma_z-\Sigma\sigma_r)^2 \leq 2\sigma_o^2$$

where the yield strength of the selected material must be less than σ_o determined by this condition. By selecting Waspalloy as the material (from Table III presented in the next section) and the boiler pressure from Table II, an evaluation of these stress loads was made which gave the maximum radius of the cylinder as 5.2 inches with a wall thickness of .25 inches at a rotational speed of 13,820 rpm (direct drive). Incorporating a factor of safety, $N = \sigma_y/\sigma_o = 1.9$, where σ_y is the yield strength of Waspalloy, it was determined that a 4 inch radius cylinder was an acceptable size. The factor of safety is used to offset the effects of those factors not considered in this analysis as presented earlier. The Von Mises yield criterion gave an allowable stress, σ_o, of 58,320 psi and the end dislocation stress (max) was σ_z = 60,150 psi, which are acceptable stress loadings for the boiler.

E. SUITABLE MATERIALS

The selection of a suitable material for the rotating boiler is based on the preceding stress analysis and the range of temperatures the materials must withstand. The temperature environment surrounding the boiler requires that it be able to withstand temperatures as high as 1035 deg F (Table I) and the stress loads at this temperature.

Several other characteristics of the material are desired. The material must be resistant to corrosion from the exhaust gas and the water. In order to reduce thermal stress effects, a low coefficient of thermal expansion is desirable. For a given heat flux, the thermal conductivity determines the temperature gradient required in the wall. Therefore a high thermal conductivity is required to lower this temperature gradient and consequently reduce the thermal stresses. Ease of fabrication, weldability, machinability, and a low acquisition cost are also desirable features.

A partial listing of the suitable materials is contained in Table III. A more complete listing is available from Reference 19, or other suitable sources. In general, these metals are all corrosion resistant, heat resistant, and capable of withstanding the required stress loads at the maximum temperature required for the boiler.

F. DESIGN INPUT DATA

The following operating parameters are therefore determined:

TABLE III

PARTIAL LISTING OF HEAT RESISTANT METALS WITH PROPERTIES EVALUATED AT 1,000 DEG F UNLESS OTHERWISE INDICATED

Alloy Name	Tensile Strength (psi)	Yield Strength (psi)	Thermal Conductivity (BTU-in/hr-ft^2°F)	Coefficient of Thermal Expansion (1/°F)	Young's Modulus (psi)
INCOLOY 901	142,000	101,000	-----	8.5×10^{-6}	-----
INCONEL "X"	138,000	83,000	236 @ 1470 deg F	8.4×10^{-6}	25×10^{6}
NIMONIC 90	149,000	96,000	-----	7.9×10^{-6}	28×10^{6}
WASPALLOY	165,000	104,000	123	7.7×10^{-6}	26.5×10^{6}
RENÉ 41	168,000	125,000	138 @ 1100 deg F	7.62×10^{-6}	27×10^{6}

1. Gas Turbine - 501-K17

Rotational speed - 13,820 rpm

Exhaust gas temperature (max) - 1035 deg F

Mass rate of exhaust flow - 31.5 lbm/sec

Design exhaust pressure drop - 10 inches of water

Exhaust gas velocity at turbine exit - 187 ft/sec

Compressor pressure ratio - 9 to 1

2. Rotating Boiler

Max radius - 4 inches for the direct-drive system. This is held constant in all systems studied.

Length - 6 feet (selected for annular flow)

3 feet (for cross-flow only)

Wall thickness - 1/4 inch

Rotation speed of separately driven units - As desired. 3,600 rpm selected.

3. Material Properties

Material selected - Waspalloy

Thermal Conductivity - 10.25 BTU/hr-ft-deg F

4. Liquid-Vapor Properties

Vapor pressure - 120 psia

Saturation Temperature - 342 deg F

Two phase annulus thickness (max) - 1/2 inch

These are the design inputs for the following thermal and hydraulic analyses.

III. THERMAL AND HYDRAULIC DESIGN ANALYSES

A. DESIGN PROCEDURE

The design procedure used in this development is for the direct drive system (Fig. 6). The development also applies to the separately driven annular flow system (Fig. 7) except that the angular velocity is no longer fixed by the turbine rpm. The annular width, l_g, is selected as the design variable. Changes in the annular width produce a change in the annular flow area for the exhaust gas. The increase in velocity due to an area reduction changes the outside heat transfer coefficient. Therefore a change in the annular width produces a change in the heat flow and the mass rate of steam flow. Associated with the annular width is a pressure drop across the boiler and each width change produces a change in the pressure drop for the exhaust gas flow. These pressure variations result in changes in the thermodynamic efficiency of the gas turbine cycle. The development of the required thermal, hydraulic relationships is presented in the next section of this thesis.

The results from the evaluation of these relationships give insight into the interdependence of the mass rate of steam flow and the exhaust gas pressure drop with the annular width. Using these results, an evaluation of the feasibility of the rotating boiler is determined.

The cross-flow rotating boiler (Fig. 8) is not considered in this design procedure. The two designs being considered

offer similar analyses and allow the effects of rotational speeds to be evaluated. From the results obtained for these two situations, a determination can be made concerning further analysis of the cross-flow rotating boiler.

B. THERMAL AND HYDRAULIC DEVELOPMENT

1. Assumptions

The thermal and hydraulic analyses are based on the following assumptions.

1. The exhaust gas flow is isentropic from the turbine to the boiler annulus.
2. The gas flow is incompressible with constant properties throughout. The exhaust gas obeys the perfect gas law.
3. The flow is one-dimensional.
4. The flow is axial entering the annulus, therefore there is no swirl effect from the gas turbine.
5. All properties of the exhaust gas are evaluated at the film temperature where

$$T_f = \frac{T_{wo}+T_G}{2}$$

6. The boiler wall is thin, and the inside and outside surface areas are considered equal.
7. The vapor in the boiler is at the saturation temperature, T_{sat}.
8. The liquid enters the boiler at T_{sat}.

9. The vapor pressure in the boiler is constant along the entire length; therefore there is no longitudinal thermal gradient in the boiler wall.

2. Heat Transfer Development

a. Basic Relationships

The heat transfer development is based on the system represented in Figure 12. The outer duct wall is insulated and an energy balance on an element of differential length of the annulus gives

$$\dot{m}_G h_{in} = \dot{m}_G h_{out} + dq_b \tag{1}$$

or

$$(\dot{m}c_p)_G T_G = (\dot{m}c_p)_G \left[T_G + \frac{\partial T_G}{\partial x} dx\right] + dq_b \tag{2}$$

Using the electric circuit analogy [Ref. 20], the rate of heat flow into the boiler can be expressed as:

$$dq_b = \frac{T_G - T_{sat}}{\dfrac{1}{h_o dA_o} + \dfrac{\ell n(r_o/r_i)}{2\pi k dx} + \dfrac{1}{h_i dA_i}} \tag{3}$$

as depicted by the series of resistances in Figure 12.

The wall resistance is

$$dR_w = \frac{\ell n(r_o/r_i)}{2\pi k dx}$$

and from Figure 12,

$$r_o = r_i + \bar{t} \tag{4}$$

or

$$r_o/r_i = 1 + t/r_i$$

and since the wall is thin [Ref. 21]

$$\ell n\left(\frac{r_o}{r_i}\right) = \ell n\left(1 + \frac{t}{r_i}\right) \simeq \frac{t}{r_i} \tag{5}$$

With this value for the natural logarithm, equation (3) becomes

$$dq_b = \frac{T_G - T_{sat}}{\frac{1}{h_o dA_o} + \frac{t}{kdA_i} + \frac{1}{h_i dA_i}} \tag{6}$$

and using the assumption that the inside and outside areas are equal, then

$$dq_b = \frac{T_G - T_{sat}}{\frac{1}{h_o} + \frac{t}{k} + \frac{1}{h_i}} dA \tag{7}$$

Considering an overall heat transfer coefficient of

$$U = \frac{1}{\frac{1}{h_o} + \frac{t}{k} + \frac{1}{h_i}} \tag{8}$$

then equation (7) reduces to

$$dq_b = UdA\ (T_G - T_{sat}) \tag{9}$$

In the above equation the gas temperature T_G is a local value, that is, T_G is a function of x, and without knowing this functional relationship between T_G and x, the heat flow into the boiler can not be determined from equation (9).

In order to determine the heat flow into the boiler another approach must be taken. Treating the heat transfer system as a parallel-flow heat exchanger [Ref. 20], the effectiveness of the heat exchanger is determined by the NTU (number of transfer units) method. The effectiveness is defined as [Ref. 22]

$$\varepsilon = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}} \qquad (10)$$

and the actual heat transfer is the energy given up by the exhaust gas in the duct which is

$$q = (\dot{m}c)_G \quad (T_{G_{in}} - T_{G_{out}}) = \dot{m}_s(h_{fg}) \qquad (11)$$

For a boiler, the capacity rate ($\dot{m}c_p$) of the liquid is infinite. Therefore the heating fluid capacity rate is the minimum capacity rate, and the maximum heat transfer that is thermodynamically possible is given by

$$q_{max} = (\dot{m}c)_G \; (T_{G_{in}} - T_{sat}) \qquad (12)$$

and the effectiveness of a parallel flow (boiler) heat exchanger [Ref. 20] is

$$\varepsilon = 1 - e^{-(UA/\dot{m}c)_G} \qquad (13)$$

Equation (13) may be rewritten as

$$\varepsilon = 1 - e^{-NTU} \qquad (14)$$

since the group $UA/(\dot{m}c)_G$ is the number of transfer units (NTU). Substituting equations (12) and (14) into equation (10), the actual heat flow becomes

$$q_{act} = (1 - e^{-NTU})((\dot{m}c)_G(T_{G_i} - T_{sat})) \qquad (15)$$

The unknowns in this equation are q_{act} and UA in the NTU expression. By evaluating UA, the actual heat flow into the boiler may therefore be determined.

To solve for UA, equation (8) gives

$$UA = \frac{1}{\frac{1}{h_i A} + \frac{t}{kA} + \frac{1}{h_o A}} \qquad (16)$$

In order to evaluate this expression only the values of h_o and h_i must be determined since the thermal conductivity, k, is given in Table III.

(1) The outside heat transfer coefficient. In order to evaluate the outside heat transfer coefficient, h_o, the system must be viewed as axial flow through an annulus with the inner cylinder rotating. A literature search indicated that for this flow situation the only investigations were concerned with the air cooling of electric motors. The most pertinent of these works was due to Gazley [Ref. 23] who studied the effects of varying the rotational speed and axial flow velocity on the heat transfer coefficient from both the rotor and stator to air. Gazley's experiments were conducted with two rotor diameters, 4.985 and 4.540 inches, giving annular widths of .017 and .240 inches respectively. His experiments covered a Reynolds number range of 200 - 15,000. He determined an effective velocity to be used in the Reynolds number for general

correlations of the rotor-air heat transfer rates. This effective velocity is a combination of axial and rotational velocity components as given by

$$V_E = \sqrt{V_a^2 + V_\omega^2} \tag{17}$$

where

$$V_\omega = \frac{\omega r_o}{2} .$$

This rotational velocity term developed by Gazley applies to small annular widths. His experimental data can be correlated by a relationship for the Nusselt number in terms of the Reynolds number [Ref. 23] as

$$Nu = 0.0298\ [Re_E]^{0.8} \tag{18}$$

where

$$Nu = \frac{2h_o \ell_g}{k} \tag{19}$$

and

$$Re_E = \frac{2\ell_g V_E}{\nu} . \tag{20}$$

The form of equation (18) is similar to the Dittus-Boelter correlation for turbulent flow in ducts [Ref. 20], which for air is given by

$$Nu = .021\ [Re]^{0.8} .$$

This correlation however evaluates the parameters at the bulk temperature whereas equation (18) uses the film temperature. Assuming these differences are acceptable for

feasibility study purposes, equation (18) is used to determine the outside heat transfer coefficient, h_o.

In order to evaluate the axial velocity as a function of the variable ℓ_g, the incompressible flow assumption and the continuity equation provide the relationship

$$V_a = \frac{V_1 A_1}{A_2} \tag{21}$$

where V_1 is the velocity of the exhaust gas at the turbine exit and A_1 and A_2 are the flow areas at the turbine exhaust and at the annulus respectively. From the geometry of the system

$$A_2 = \pi(R_o^2 - r_o^2) \tag{22}$$

or

$$A_2 = \pi \ell_g (R_o + r_o) \tag{23}$$

where

$$\ell_g = R_o - r_o \tag{24}$$

is the annular width.

From these relationships, the outside heat transfer coefficient is given by:

$$h_o = \frac{0.0298k}{2\ell_g} \left[\frac{2\ell_g \sqrt{V_a^2 + (\frac{1}{2}\omega r_o)^2}}{\nu} \right]^{0.8} \tag{25}$$

(2) The inside heat transfer coefficient. This heat transfer coefficient is the boiling heat transfer coefficient. The acceleration due to the boiler rotation

has a definite effect on this coefficient. Judd and Merte [Ref. 24] investigated this effect and developed the following empirical relationship, which is valid for subcooling:

$$q/A = 0.16\, P_r^{1/3} \left(\frac{g\beta k_\ell^{\,3}}{\nu_\ell^{\,2}}\right)^{1/3} \left(\frac{a}{g}\right)^{1/3} [Tw_i - T_{sat}]^{4/3}$$

$$+\ 2.39 \times 10^6\, Pr^{-4.1} \left[\frac{k_\ell (T_w - T_{sat})}{\sqrt{g_c \sigma_\ell / g(\rho_\ell - \rho_v)}}\right] \left(\frac{C_\ell (Tw_i - T_{sat})}{h_{fg}}\right)^2 \qquad (26)$$

where

Pr = Prandtl number

g = local acceleration of gravity

a = angular acceleration of the cylinder

β = volume coefficient of expansion

k_ℓ = thermal conductivity of the liquid

ν_ℓ = kinematic viscosity of the liquid

σ_ℓ = surface tension of the liquid

C_ℓ = specific heat of the liquid

h_{fg} = heat of vaporization

From Newton's law of cooling [Ref. 20], the heat transfer is

$$q/A = h_i (Tw_i - T_{sat}) \ . \qquad (27)$$

Substituting for q/A in Judd and Merte's relationship yields:

$$h_i = 0.16\ Pr^{1/3} \left(\frac{g\beta k_\ell^{\,3}}{\nu_\ell^{\,2}}\right)^{1/3} \left(\frac{a}{g}\right)^{1/3} (Tw_i - T_{sat})^{1/3}$$

$$+ 2.39 \times 10^6\ Pr^{-4.1} \left[\frac{k_\ell}{\sqrt{g_c \sigma_\ell / g(\rho_\ell - \rho_v)}}\right] \left(\frac{C_\ell (Tw_i - T_{sat})}{h_{fg}}\right)^2 \tag{28}$$

The value of the inside wall temperature is necessary to evaluate this expression and this value is not known. In reality this temperature is also a function of x, or location. However, the inside heat transfer coefficient is very large, as reported in Reference 5, which means that the temperature variation along the boiler wall is small since T_{sat} is a constant. Therefore assuming the inside wall temperature is constant, and assuming a value for Tw_i, the inside heat transfer coefficient is determined.

Using this result and the other known quantities, equation (16) can be evaluated for UA. With this value, the effectiveness and actual heat flow are determined from equations (14) and (15) respectively. Then using equation (27) the inside wall temperature is evaluated for this actual heat flow. If this value of Tw_i is different from the assumed value, the above steps are repeated using the calculated temperature as the new assumed temperature. This iteration is continued until the assumed inside wall temperature is the same as the calculated temperature.

The actual heat flow, calculated above, yields the mass rate of steam flow from the relation [Ref. 20]

$$\dot{m}_s = \frac{q_{act}}{h_{fg}} \tag{29}$$

and the exhaust gas temperature leaving the end of the annulus is

$$T_{G_o} = T_{G_i} - \frac{q_{act}}{(\dot{m}c)_G} \, . \tag{30}$$

b. External Fins

The addition of external fins increases the surface area for heat transfer on the outside. This decreases the outside resistance to heat flow. The fins are placed on the cylinder along the direction of the effective velocity vector (Figure 13) where the average tangential velocity is $\omega\bar{r}$ and

$$\bar{r} = r_i + \frac{\ell_g}{2} \, . \tag{31}$$

This fin arrangement gives channel flow between the fins with the effective velocity normal to the channel cross-section. This condition exists only at the design rotational speed and axial velocity. Off-design operation will create a different flow situation, but is not anticipated to generate severe problems.

Assuming fully developed, turbulent flow in the channel the friction factor, f, [Ref. 25] is

$$\frac{1}{\sqrt{4f}} = -0.8 + 2.0 \log \left(Re_E \sqrt{4f}\right) \, . \tag{32}$$

To include the effects of curvature of the channel around the boiler unit, the relation [Ref. 25]

$$4f' = 4f\left[1 + 0.075 \, (Re_E)^{\frac{1}{4}} \left(\frac{R_h}{r_c}\right)^{\frac{1}{2}}\right] \tag{33}$$

is used although approximate, where

$R_h = A/P$, hydraulic radius

$A = \ell_g \times b$, cross sectional area of channel

$p = 2(\ell_g + b)$, perimeter of the channel

b = fin spacing

r_c = radius of curvature of the spiral channel

$$= \frac{V_a^{\,2} + \bar{r}^{\,2}\omega^2}{\bar{r}\,\omega^2} \qquad (34)$$

Using the Reynolds analogy and this friction factor, with the assumption that the Prandtl number of air is nearly one, the heat transfer coefficient in the channel is [Ref. 20]:

$$h_o' = \frac{f'}{2}\, Re_E \frac{k_g}{dH} \qquad (35)$$

where

$$Re_E = \frac{D_h\, V_E}{\nu} \qquad (36)$$

$$D_h = \frac{4A}{p}$$

Assuming the tips of the fins are insulated, the efficiency of a simple longitudinal fin is [Ref. 22]

$$\phi = \frac{\tanh \mu_b}{\mu_b} \qquad (37)$$

where

$$\mu_b = \ell_g \sqrt{h_o'/ky_b} \qquad (38)$$

and y_b is one-half the fin thickness. The value of thermal conductivity is based on the fin material. Copper fins with a very thin, protective coating of nickel are used for the analysis.

The efficiency of an array of fins is [Ref. 20]:

$$\kappa = 1 - \frac{A_f}{A_t}(1-\phi) \tag{39}$$

where

A_f = surface area of the fins = NS

N = number of fins in the array

S = surface area of each fin

$A_t = A_f + A_u$ = total surface area exposed for heat transfer

and

A_u = the boiler surface area which is exposed (no fins) for heat transfer.

To evaluate the effect of changing the fin thickness, the fin spacing, b, is fixed and the fin thickness is varied. The number of fins in the array is determined from the fin spacing and the fin angle. Referring to Figure 13, the fin angle is

$$\theta = \arctan \frac{\omega\bar{r}}{V_a}$$

and

$$x = \frac{b + th}{\sin\theta} \tag{41}$$

This value of x is for adjacent fins. For each continuous fin, the separation between adjacent points on the surface, for one complete wrap is

$$X = 2\pi \frac{V_a}{\omega} \tag{42}$$

and the number of continuous fins is

$$N = \frac{X}{x} \ . \tag{43}$$

Using this array, the new outside resistance to heat transfer is

$$R_{th_o} = \frac{1}{h'_o A_t \kappa} \tag{44}$$

and the overall conductance for the boiler becomes

$$UA = \frac{1}{\frac{1}{h'_o A_t \kappa} + \frac{t}{kA} + \frac{1}{h_i A}} \ . \tag{45}$$

Using this new value of overall conductance, the same iterative procedure as used previously is used to determine the actual heat flow, mass rate of steam flow, and exhaust gas temperature at the annulus exit.

3. Hydraulic Analysis

The exhaust gas pressure drop for the annular flow situations depends on the geometry of the outer surface of the boiler.

a. Unfinned Boiler

The pressure drop across the exterior of the smooth cylinder, as developed by Gazley [Ref. 23], is

$$\Delta p = \frac{C_{fa} \, \Delta L \, \rho V_a^{\,2}}{\ell_g} \, . \tag{46}$$

The relationship for the friction factor, C_{fa}, is

$$\frac{1}{\sqrt{C_{fa}}} = -0.40 + 4.0 \log_{10} \left(Re_a \sqrt{C_{fa}} \right) \tag{47}$$

with

$$Re_a = \frac{2 \, \ell_g \, V_a}{\nu} \, . \tag{48}$$

The term ΔL in equation (46) is the distance travelled along the cylinder by a gas particle travelling at the effective velocity. Thus, this distance is [Ref. 23]:

$$\Delta L = \frac{V_E}{V_a} \, L \, . \tag{49}$$

Notice that the friction factor is independent of the effective velocity, but the pressure drop depends on this effective velocity through the ΔL term.

b. Externally Finned Boiler

The friction factor for the curved channel flow is given by equations (32) and (33). The pressure drop in each channel is

$$\Delta p_f = \frac{f' \, L_f \, \rho V_a^{\,2}}{D_h} \tag{50}$$

where

$$L_f = \frac{A_f}{2\ell_g}$$

is assumed to be the average length of each longitudinal fin.

IV. RESULTS AND CONCLUSIONS

The computations of the equations developed in the preceding section were done on an IBM-360 digital computer. The basic independent variable was the annular width, ℓ_g, in all cases. However, with the addition of the external fins, the additional variables of fin spacing and fin thickness were included. The fin thickness was varied from one-eighth inch to three-fourths inch for each fin spacing. The fin spacings ranged from one inch to three inches in one-half inch increments. The annular width, ℓ_g, was varied from 14 inches to 4 inches unless the Mach number became greater than one.

All properties of the exhaust gas were taken from References 26 and 27 and evaluated at the film temperature. The properties of the liquid and vapor in the boiler were taken from Reference 28 and evaluated at the saturation pressure, 120 psia. The fin material properties were taken from References 20 and 27 and evaluated at 1000 °F. The values for all properties used in the computations are given in Table IV.

A. UNFINNED ROTATING BOILER

The results of the calculated mass rate of steam flow and the exhaust gas pressure drop for the unfinned rotating boiler are presented in Figures 14 and 15 respectively. The steam flow rate (Fig. 14) is very low for all annular widths.

TABLE IV

VALUE OF PROPERTIES USED IN COMPUTATION

Property	Value
ν_G	.000615 ft^2/sec
k_G	.02829 BTU/hr-ft-°F
$(C_p)_G$	.263 BTU/lbm-°R
ρ_G	.0344 lbm/ft^3
σ_ℓ	.00295 lbf/ft
β	.000612 1/°R
k_ℓ	.3947 BTU/hr-ft-°F
T_{sat}	341.5 °F
ρ_ℓ	61.7 lbm/ft^3
ρ_v	.00492 lbm/ft^3
$(C_p)_\ell$	1.039 BTU/lbm-°R
h_{fg}	887.8 BTU/lbm

The marked increase at the lower widths is attributed to the increase in the outside heat transfer coefficient as a result of the increased axial and relative velocities. The pressure drop (Fig. 15) also displays a sharp increase at the smaller annular widths. This is likewise due to the velocity increase.

These results indicate the necessity for external fins to enhance the outside heat transfer coefficient in order to increase the mass rate of steam flow.

B. FINNED DIRECT DRIVE BOILER SYSTEM

The results for the finned direct drive boiler are presented in Figures 16 and 17. The mass rate of steam flow (Fig. 16) is considerably larger than before as expected. Contrary to the unfinned case, a decrease in annular width does not immediately increase the steam flow rate. The initial decrease in steam flow rate for a decrease in annular width is due to the large decrease in surface area of the fin array. Although the outside heat transfer coefficient from equation (35) does increase with an increase in velocity due to a smaller annular width, the large reduction of surface area creates an overall reduction in heat transfer and in the steam flow.

The pressure drop (Fig. 17) is presented using only two fin thicknesses. The other fin thicknesses are basically the same and including all thicknesses on the figure makes it cluttered and unreadable. The pressure drop also decreases until the velocities become extremely high at a width of six inches. The reason for this decrease is the reduced effective length of the fin channel as the axial velocity increases. Reviewing Figure 13, for the direct drive case, the $\omega\bar{r}$ is very large for the large annular widths. As the annular widths decrease the axial velocity component increases and the mean radius, $\bar{r}$, from equation (31) decreases. This causes the rotational velocity term to decrease which results in a decrease of the angle, θ, and a shorter effective length. As the axial velocity gets much

larger than the rotational velocity, the pressure drop increases again as expected from equation (50).

A check of the Mach number in the channel indicated that $M \geq .58$ for all fin arrays. This is greater than allowed in incompressible flow, and the results are considered approximations, at best, since the development assumed incompressible flow in the system.

C. FINNED SEPARATELY DRIVEN BOILER SYSTEM

The separately driven longitudinal boiler permits a much lower speed of rotation. Using this slower rotational velocity, the Mach number based on the effective velocity is reduced below $M = .3$ where the compressibility effects are assumed negligible. As the annular width decreases, the axial velocity component will increase and cause the compressibility effects to be significant for the lower annular widths. In all cases the flow is below $M = .3$ for annular widths greater than 8 inches. For the larger (2.5 and 3 inch) fin spacings, $M = .3$ is not exceeded until ℓ_g reaches 6 inches. Therefore, the results for the mass rate of steam flow and exhaust gas pressure drop which are presented in Figure 18-27, satisfy the incompressible flow assumption for annular widths greater than 8 inches in all cases.

As discussed briefly in the preceding section, the mass rate of steam flow decreases for an initial annular width decrease. This is due to the reduction of surface area of the fins. For example, a 1/4 inch thick fin with 2 in. fin spacing (Fig. 22) in going from an ℓ_g of 13 inches to

12 inches the fin surface area decreases from 395 to 350 square feet while the outside heat transfer coefficient only increases from 56.0 to 56.2 BTU/hr-ft^2-°F due to a velocity change from 405 to 414 feet per second. The effectiveness of the fin array is .20 and .21 for the two respective annular widths. The overall conductance, equation (45) decreases from 2570 to 2518 BTU/hr-°F.

For this same case the pressure drop results (Fig. 23) also indicate a decrease as a result of the effective length decreasing from 11.4 to 10.3 feet while the axial velocity increases from 236 to 270 feet per second and the corrected friction factor, equation (33), decreases from .024 to .023. The hydraulic diameter also decreases from .288 to .286 feet for this annular width change. These result in a net pressure drop change from 16 to 15 inches of water.

As the effective velocities increase due to the large increase in axial velocity for the smaller annular widths these trends are reversed and the velocity changes are the dominant factor. However, as stated above, the compressibility effects become dominant when these reversals occur.

In addition, Figures 18, 20, 22, 24, and 26 show that an increased fin spacing gives a reduced mass rate of steam flow. This is as expected since an ever increasing fin spacing would reduce to no fins at the extreme, and this situation has already been presented. The pressure drop

also decreases for increasing fin spacings, Figures 19, 21, 23, 25, and 27, which also follows the above reasoning.

Using Figures 22 and 23 for specific discussion, it is noted that the increased fin thickness gives larger heat flows. This is due primarily to the increased efficiency of the fin array as the wider fins are used. For example, for the two inch fin spacing the average fin array efficiencies are 17.8%, 23.3%, 27.3%, 30.7% and 35.8% for fin thicknesses of 1/8, 1/4, 3/8, 1/2, and 3/4 inch respectively. These fin array efficiencies are relatively low due to the large values of the argument μ_b (equation (38)) for determining the single fin effectiveness, but do increase for an increased fin thickness.

For exhaust gas flows with negligible compressibility effects, the exhaust gas temperature decreases by 40 to 60 deg F in passing over the boiler. The radial ΔT for the boiler wall is less than 75°F which indicates the stress design values presented earlier are safe.

The major resistance to the heat flow is the outside resistance due to the low h_o.

The three resistances

$$R_{th_o} = \frac{1}{h_o' A_f K}, \quad R_w = \frac{t}{kA}, \quad \text{and } R_i = \frac{1}{h_i A}$$

for the 2 inch fin spacing at an annular width of 12 inches for a 3/8 inch thick fin were .017, .00203, and .000276 respectively. These values show that the outside resistance

is still more than eight times larger than the wall resistance, and more than 61 times larger than the inside resistance even with external fins. From these computations, it is concluded that even with the external fins the controlling factor for the heat transfer is still the outside surface.

D. FEASIBILITY DETERMINATION

In order to evaluate the feasibility of the rotating boiler, a review of what has transpired in this analysis is necessary.

1. Designs Considered

Three basic configurations were presented, however only two of these were analyzed. The results of these analyses indicate that the rotating boiler is not capable of producing a high mass rate of steam flow. The primary cause of this poor performance is the outside heat transfer coefficient. It is assumed that the same limitation would exist for the cross-flow orientation.

The possibility of multiple rotating boilers is recognized, especially using the cross-flow orientation, but has not been discussed or studied.

2. Benefits of the Rotating Boiler

Laboratory testing has demonstrated an excellent steam separation ability with the rotating boiler. The rotating boiler can reduce the weight and size required for installing a waste heat recovery boiler. Although there have been no calculations presented to verify this statement,

it is still considered a benefit of the rotating system.

3. Disadvantages of the Rotating Boiler

The disadvantages of the rotating boiler were initially considered to be the increased pressure drop in the exhaust gas flow, and the complexities induced into the system by rotation of the boiler.

Analysis indicates that additional disadvantages can be incurred. First the high stress loads due to rotation can severely limit the size of the boiler. These stresses also require the use of high temperature alloys to withstand these loads, which can increase the cost of the boiler unit. The primary disadvantage is the low heat transfer which is due to the low outside heat transfer coefficient. This results in a low steam flow rate and a very small change in the exhaust gas temperature.

From these considerations and the results of the calculations, it appears that the rotating boiler is not feasible on large Naval vessels where size and weight savings are generally not critical. However, for small vessels, where size and weight are critical and the steam flow rate required is not excessive, the possibility of using the rotating boiler warrants consideration.

E. PRESSURE DROP EFFECTS

The effects of the pressure drop, or back pressure on the gas turbine, were not studied in detail. However, a single computation was made using a simple gas turbine cycle with the same operating characteristics as the Allison 501-K17.

Compressor and turbine efficiencies were assumed to be 0.85. From this computation, it was determined that a back pressure of 19 inches of water decreased the thermodynamic efficiency from .295 for zero back pressure to .285. This computation neglected the amount of work used to rotate the boiler, which would further decrease the efficiency, and lead to larger turbine operating cost.

V. RECOMMENDATIONS FOR FURTHER STUDY

1. As determined in this analysis, the external fin array efficiency is low for the arrangement studied. Further study of the external fins is recommended to develop possible methods to increase the array efficiency.

2. Further study to be conducted on the rotating boiler as presented in this thesis should use a compressible flow analysis in order to evaluate the direct drive boiler unit, and to determine the compressibility effects at all flow conditions.

3. As indicated earlier, Gazley's experiments used very small annular widths with relatively large rotors. The ratio of annular width to diameter is .0034 and .052 for the smallest and largest widths respectively in his study. In the flow situation for the rotating boiler, this ratio is 1 and 3.5 for the smallest and largest widths. This is an increase of two orders of magnitude. Therefore, it is recommended that an experimental study be conducted on rotating cylinders in axial flow fields to determine the effect of changing the annular widths on the heat transfer.

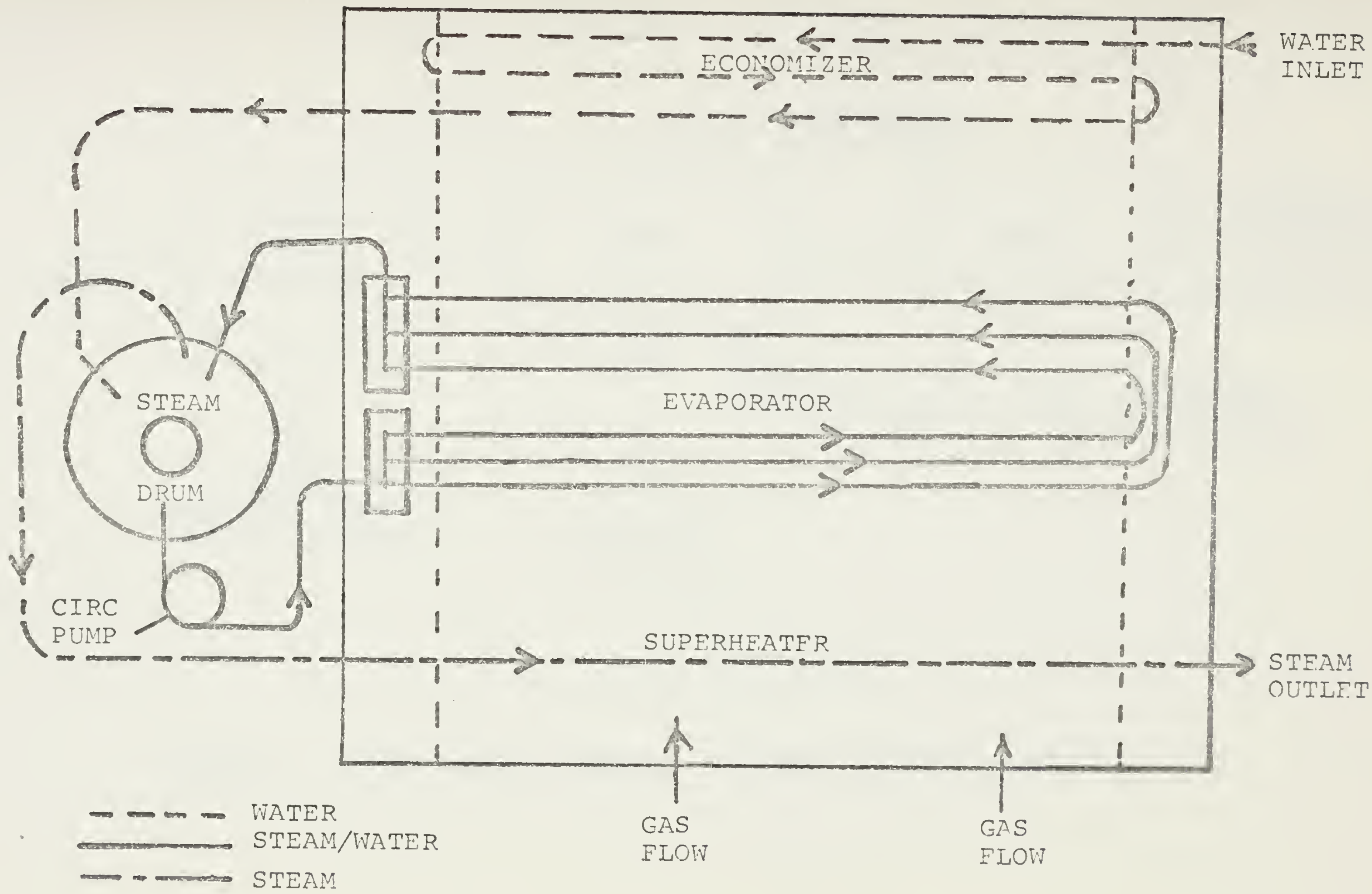


Figure 1. Heat-Recovery Waste Heat Boiler Circuit Schematic [Ref. 3].

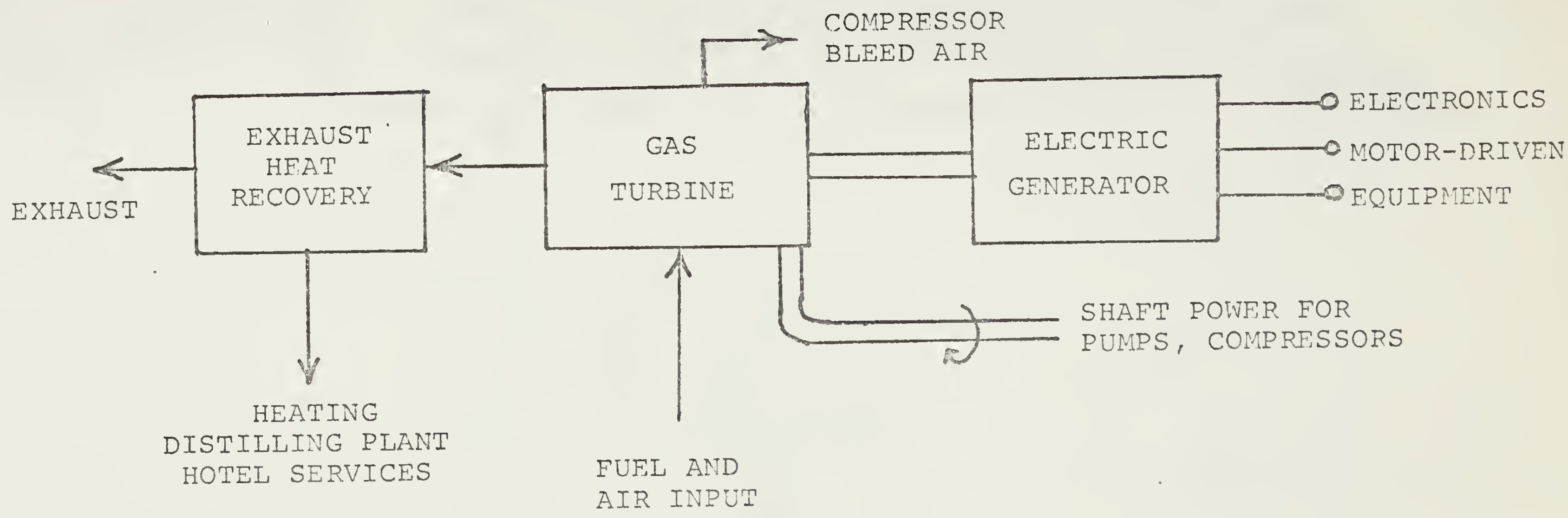


Figure 2. Total Energy Concept for Ship Service Power [Ref. 2].

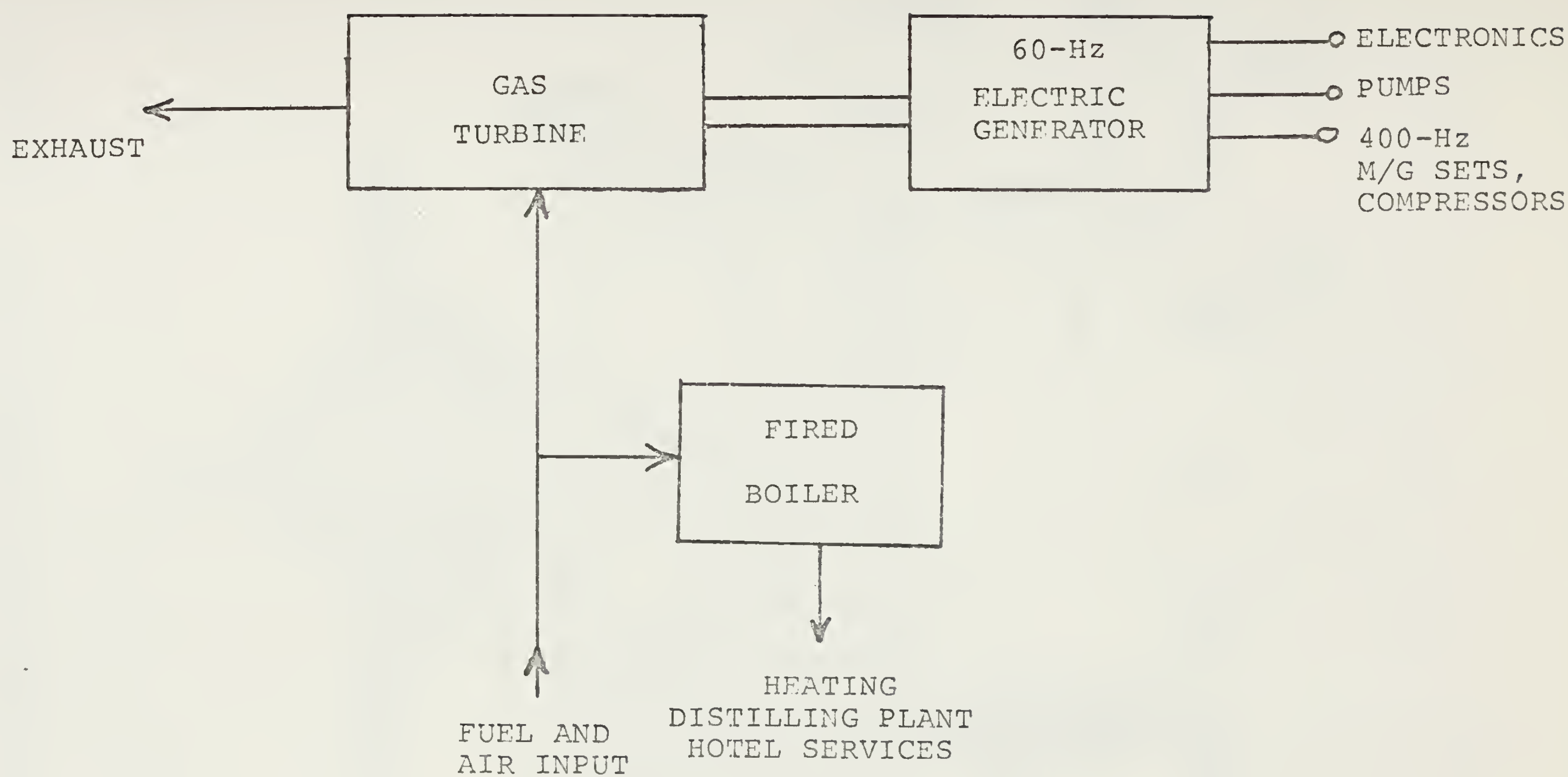


Figure 3. Conventional Ship Service Generator and Heating System [Ref. 2].

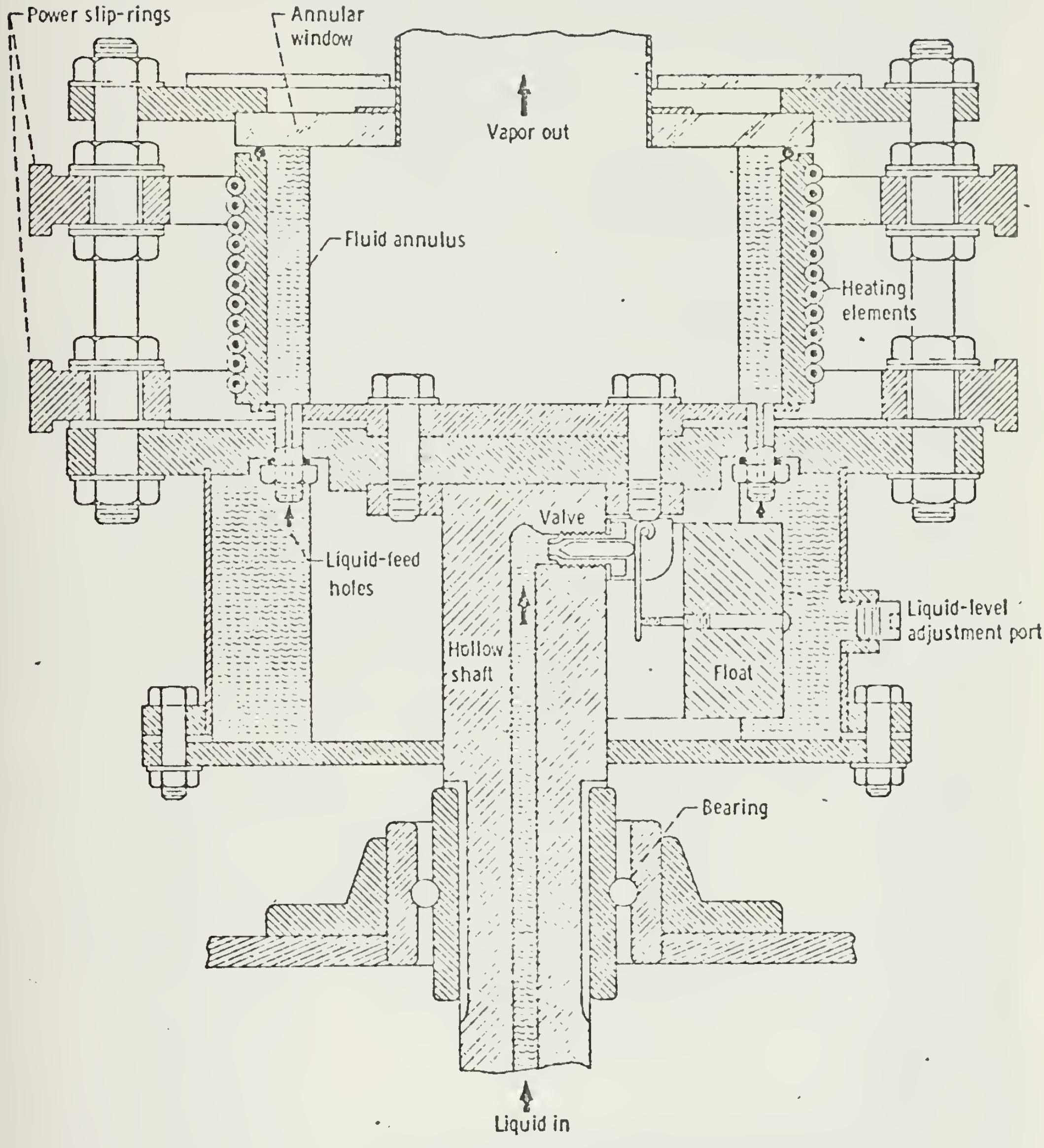


Figure 4. Lewis Research Center Experimental Rotating Boiler [Ref. 4,5].

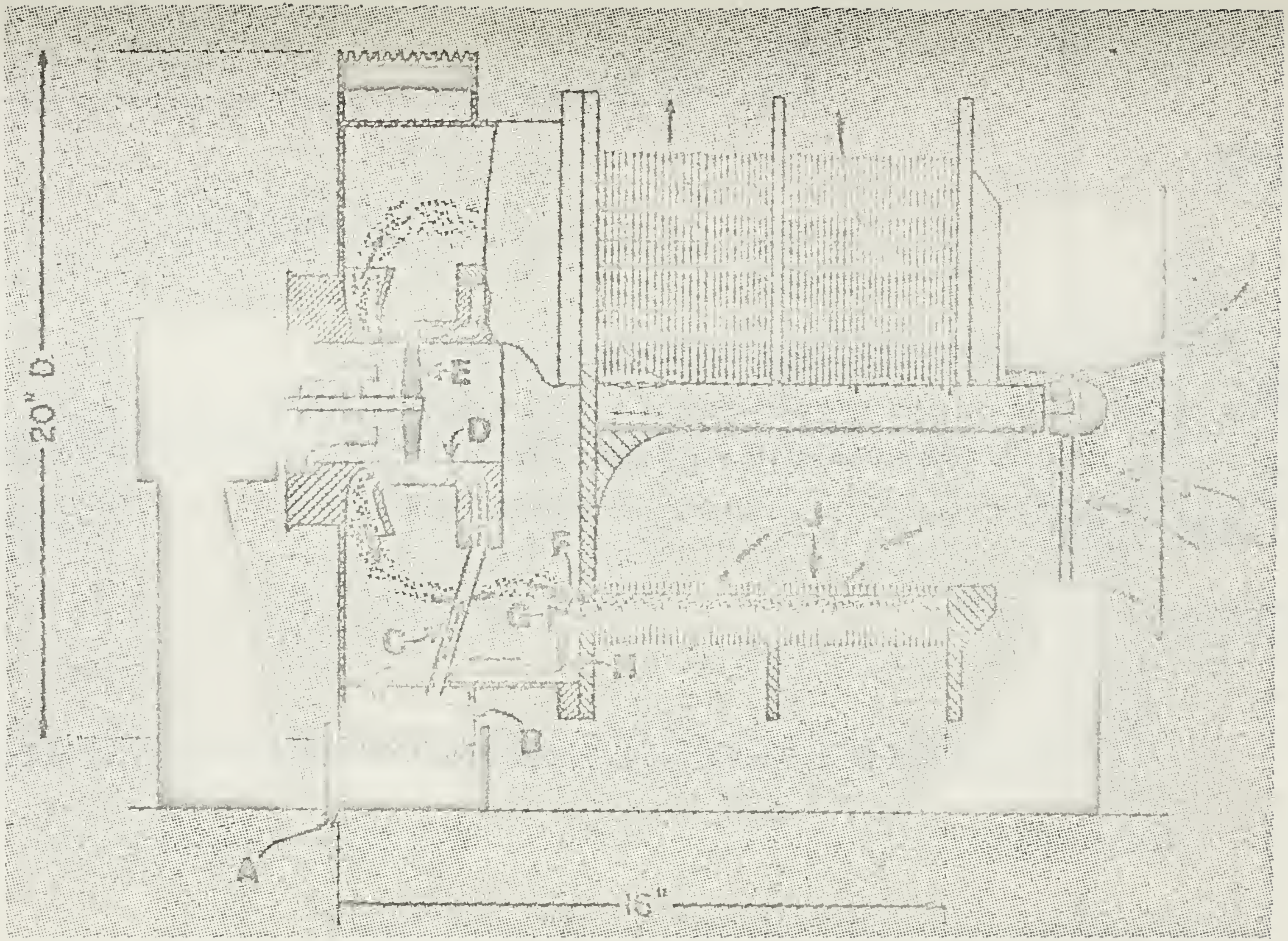


Figure 5. DuPont Engine Schematic Cross Section Indicating Cycle Phases [Ref. 6].

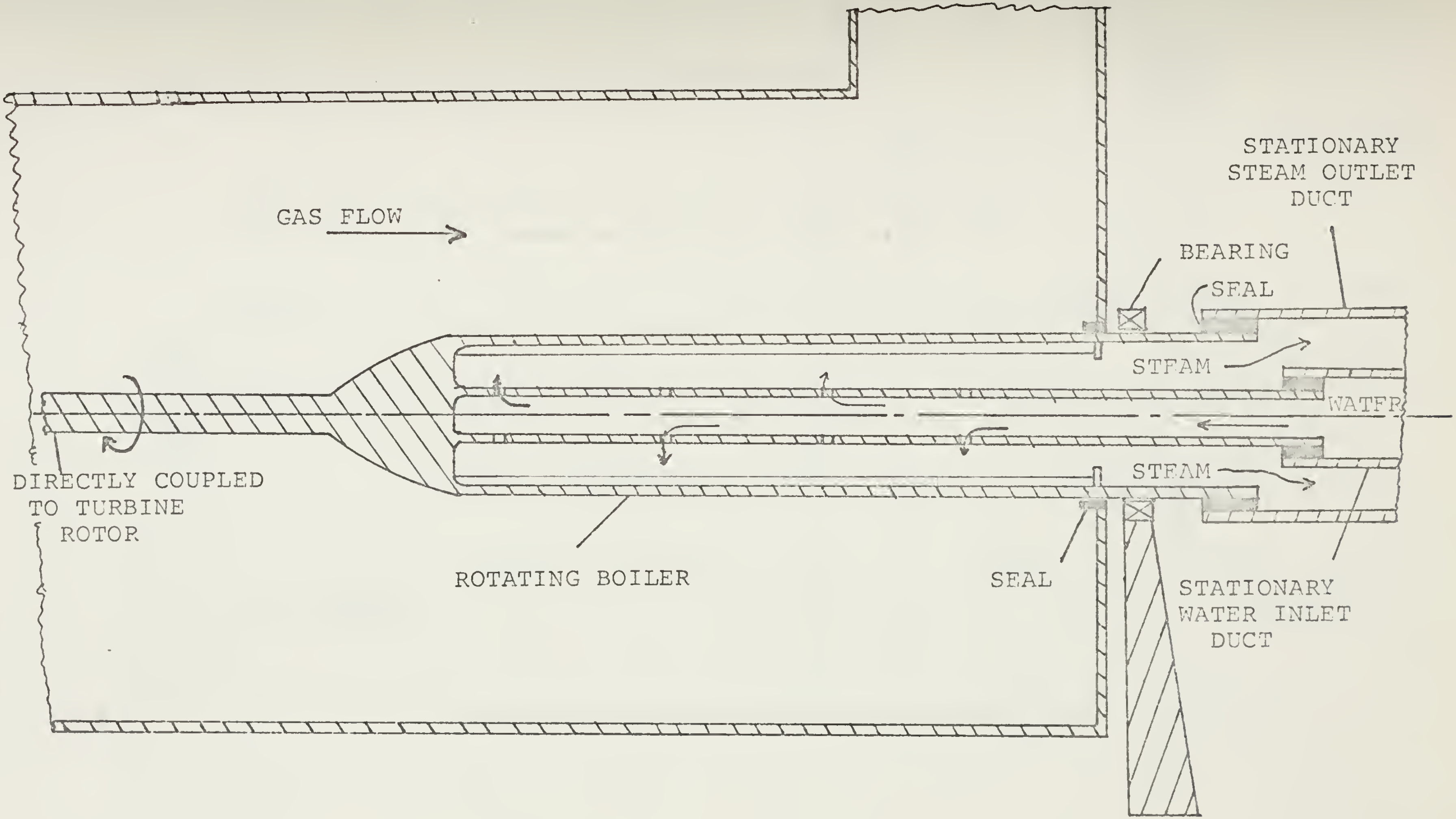


Figure 6. Direct Drive Annular Flow Rotating Boiler Representation.

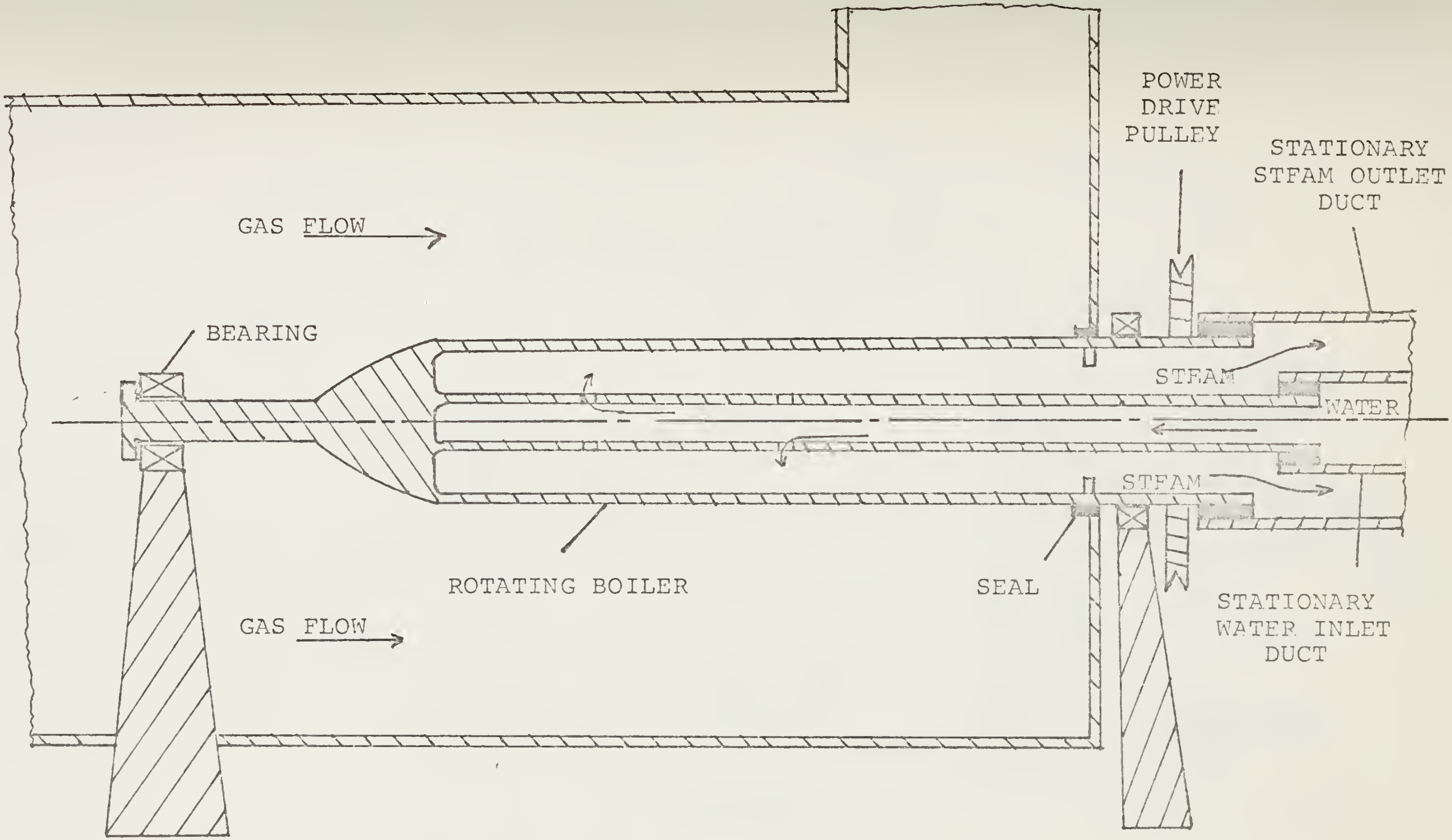


Figure 7. Separately Driven Annular Flow Rotating Boiler Representation.

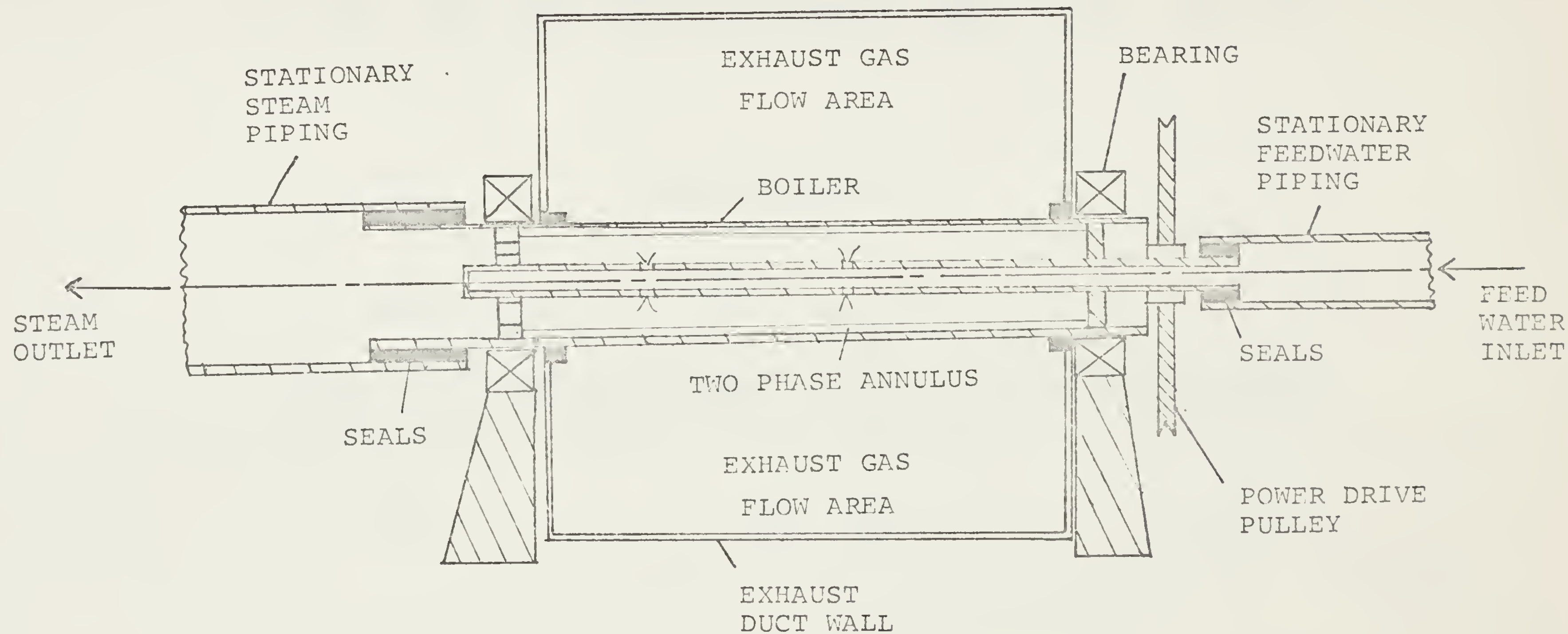


Figure 8. Schematic Representation of Cross Flow Boiler with Perpendicular Gas Flow.

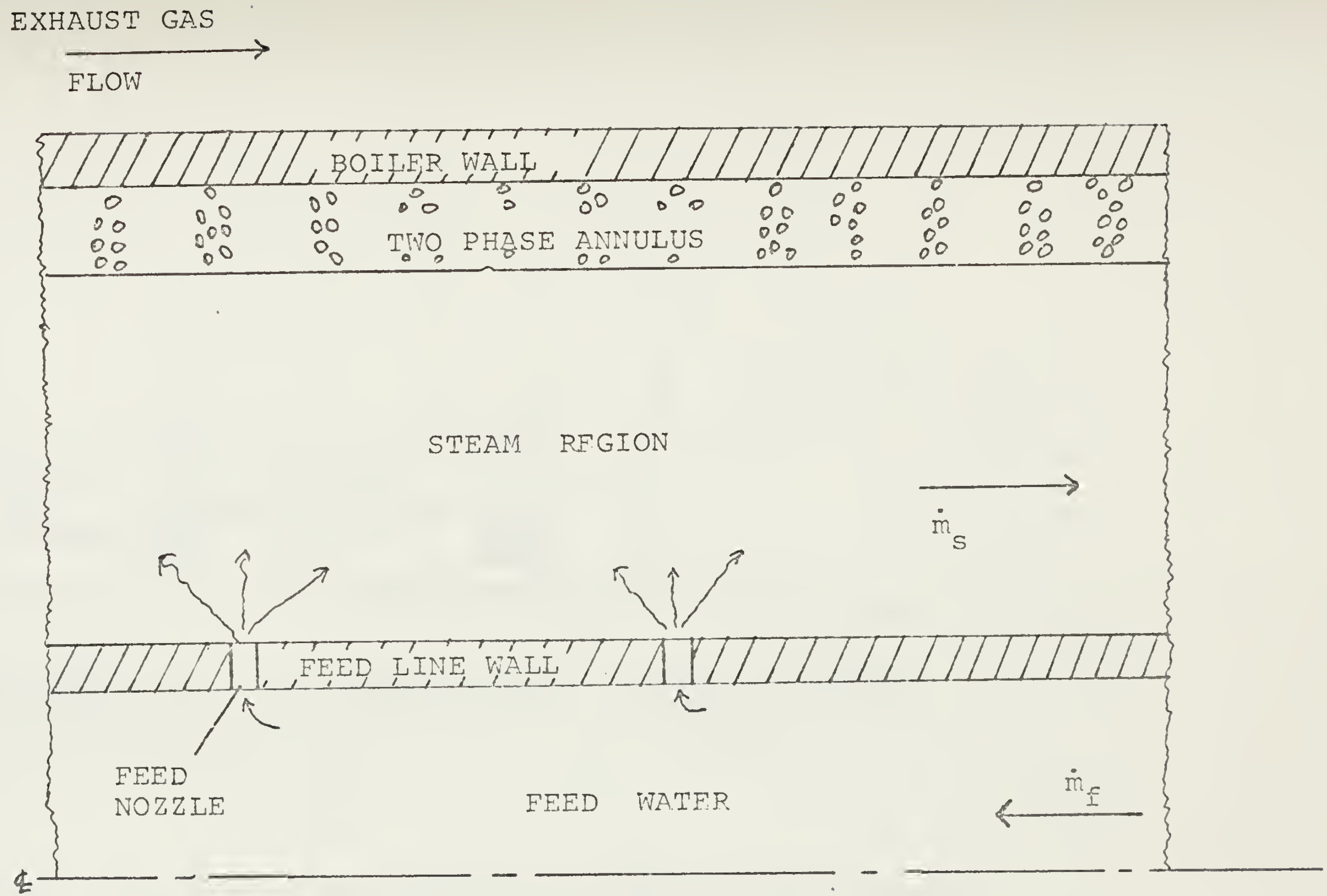
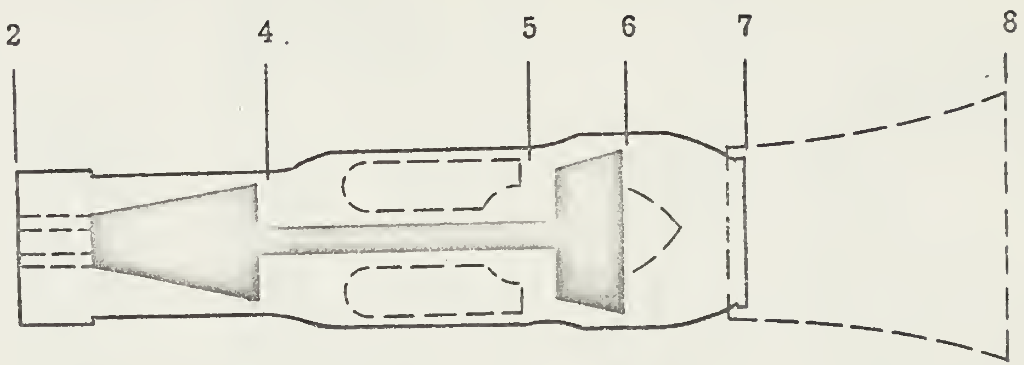


Figure 9. Boiler Cross Section Showing Location of Interface, Two Phase Annulus and Feedwater Inlet.



2. Engine Inlet
4. Compressor Outlet
5. Turbine Inlet
6. Turbine Outlet
7. Engine Exhaust
8. Exhaust Tailcone Outlet

Figure 10. Schematic Representation of the Allison 501-K17 Gas Turbine Showing Gas Flow Stations in the Turbine [Ref. 11].

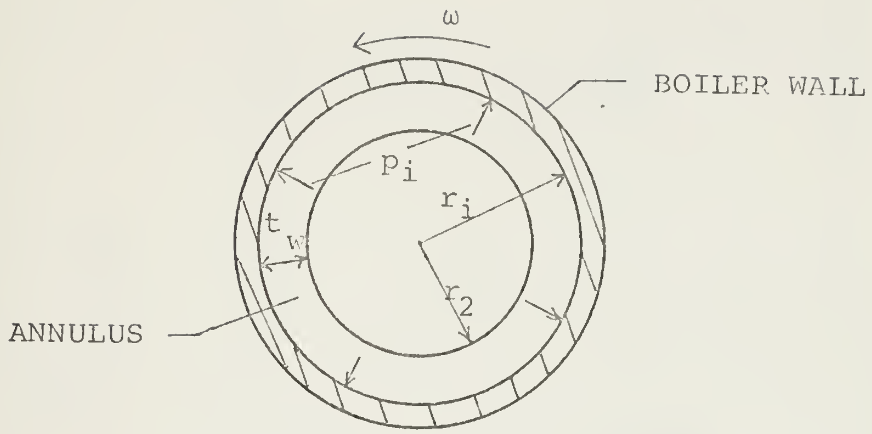


Figure 11a. Schematic Cross Section of the Rotating Boiler Showing Pressure Loadings and Dimensions with t_ω Defined as $r_i - r_2$.

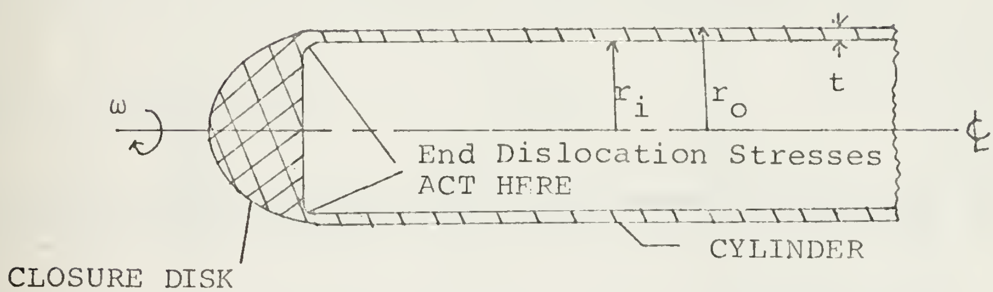


Figure 11b. Representation of Boiler Unit Showing Closure Disk (cross-hatched area) and Cylinder Indicating Point of End Dislocation Stresses.

Figure 11. Representative Sections Considered for the Stress Analysis of the Rotating Boiler Unit.

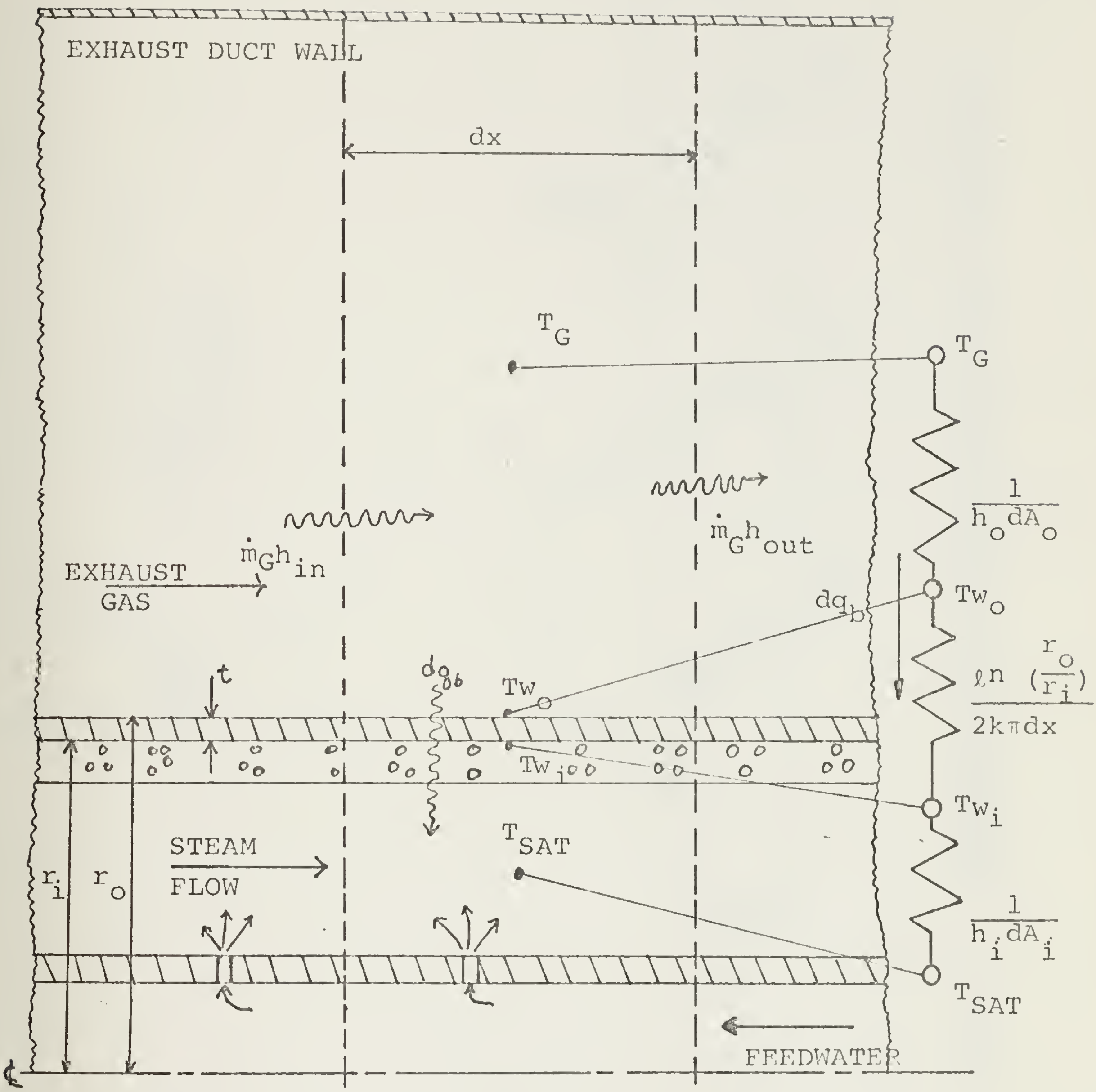


Figure 12. Cross-section Schematic Showing Exhaust Gas Flow, Heat Transfer, Steam Flow, Feed water Flow, and Location of Temperatures Used in Thermal Analysis.

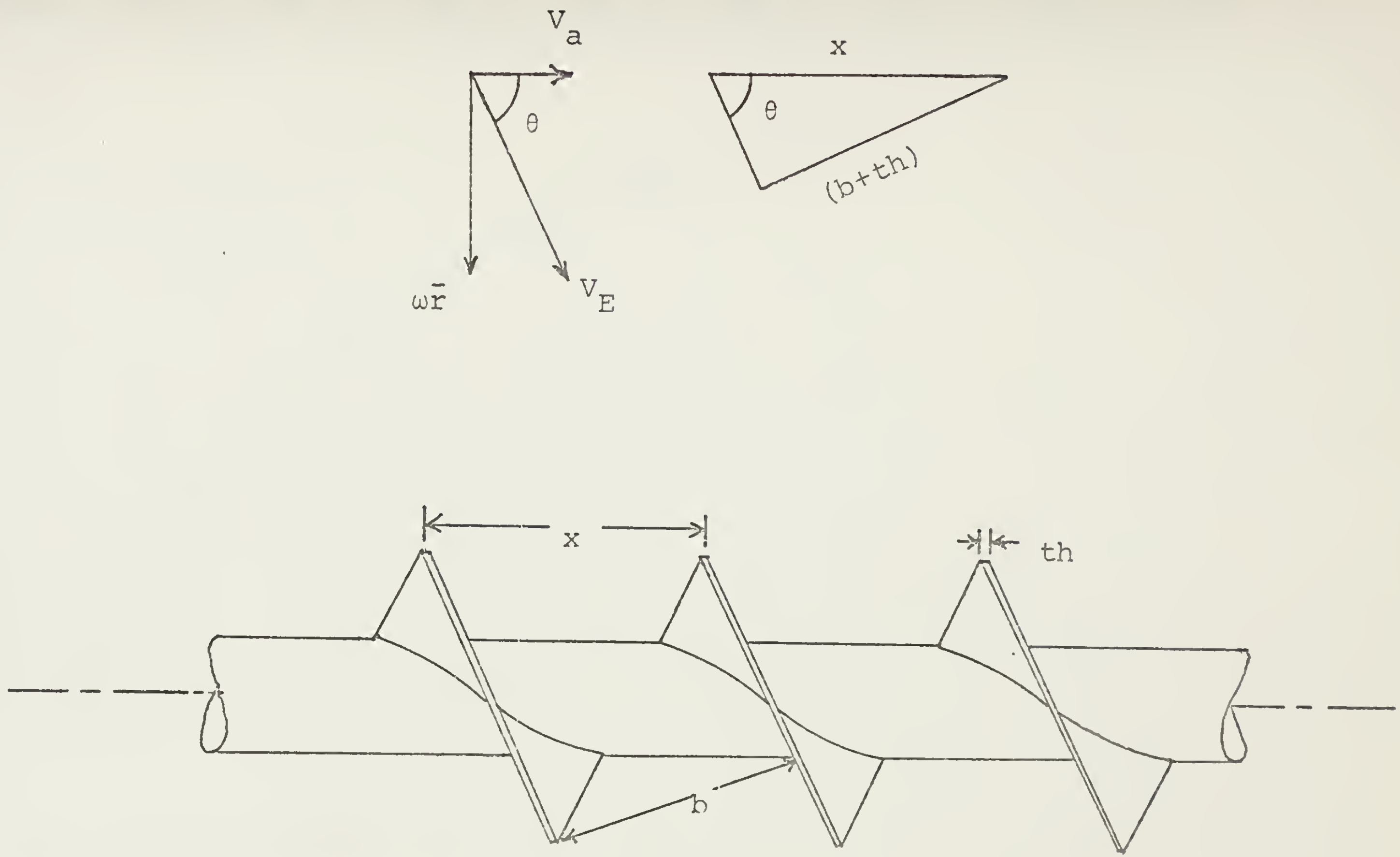


Figure 13. Representation of the Spiral Fins with Effective Velocity, Fin Spacing, and Fin Angle Indicated.

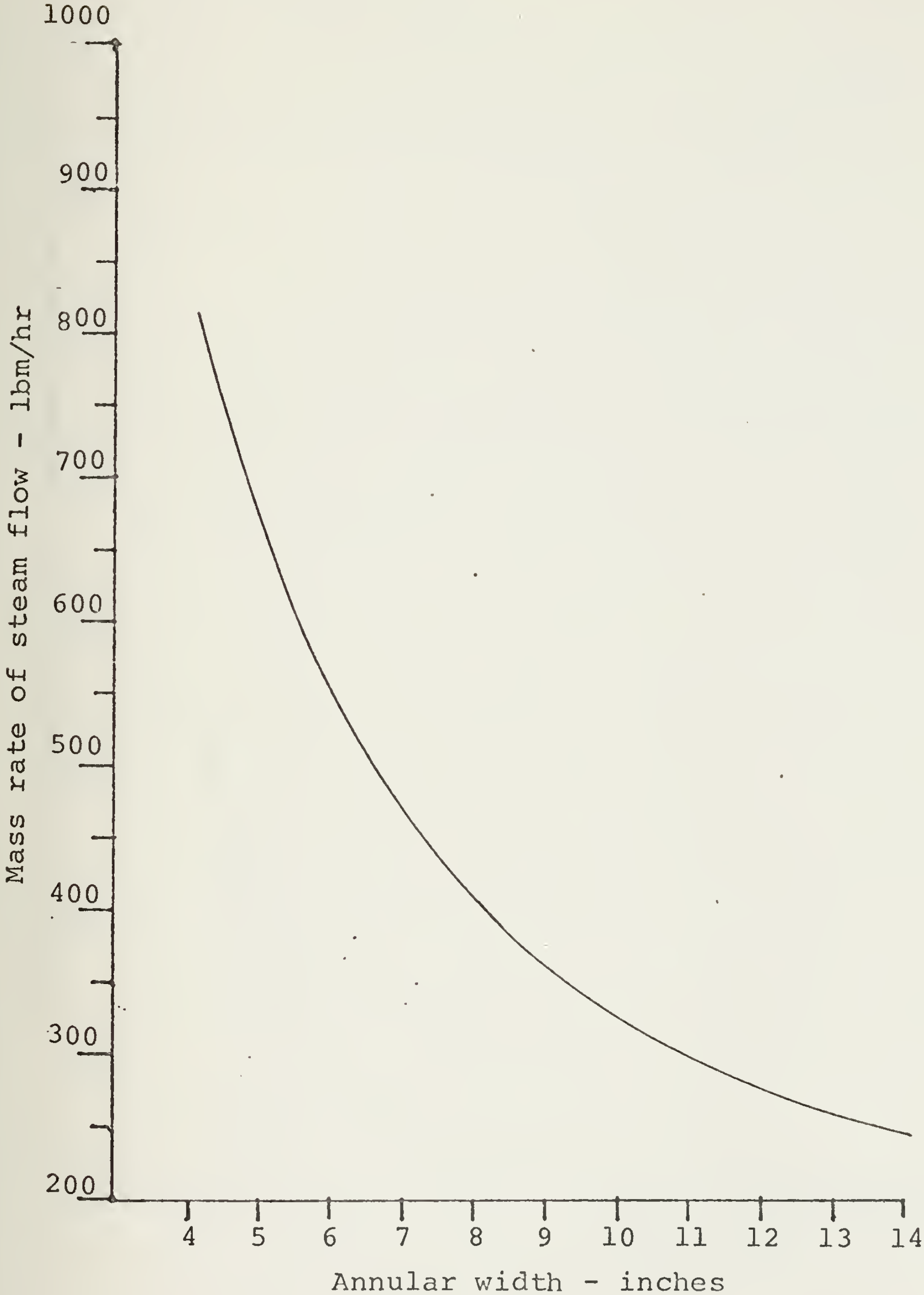


Figure 14. The Effect of Annular Width on the Mass Rate of Steam Flow for the Unfinned Boiler Rotating at 13,820 rpm.

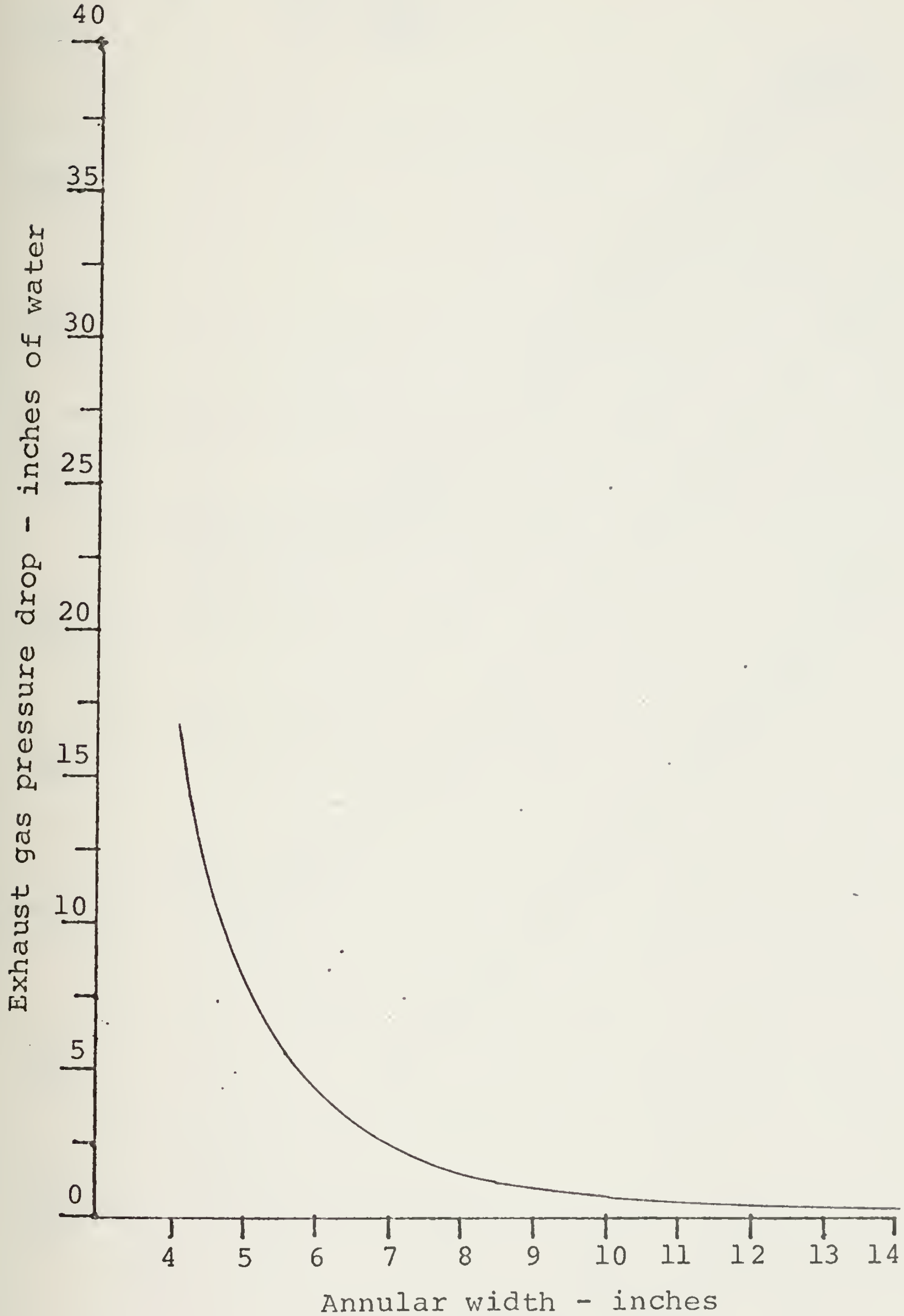


Figure 15. The Effect of Annular Width on the Exhaust Gas Pressure Drop Across the Boiler with a Rotation of 13,820 rpm.

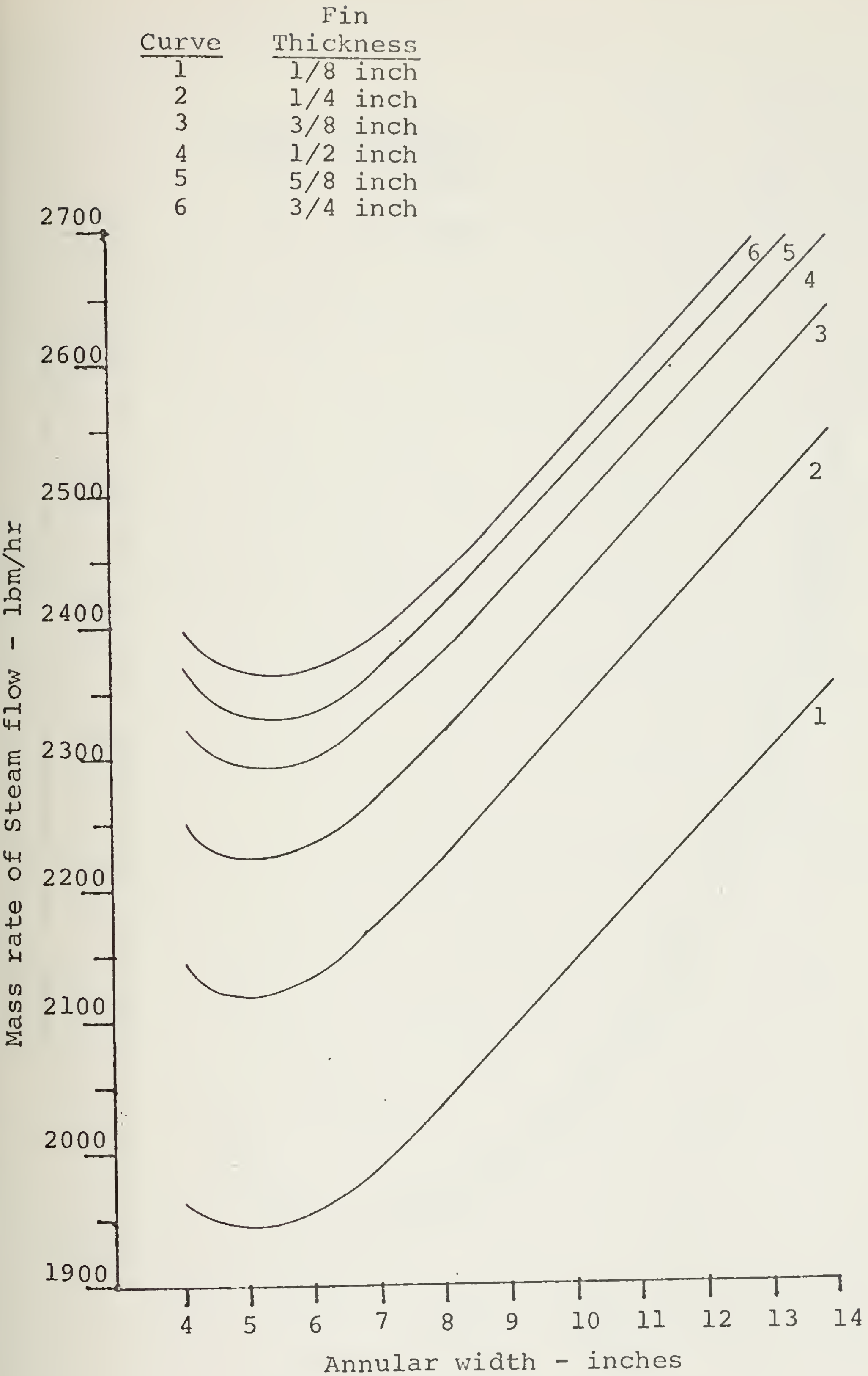


Figure 16. The Effect of Annular Width on the Mass Rate of Steam Flow for a Fin Spacing of 2.0 Inches with a Rotation of 13,820 rpm.

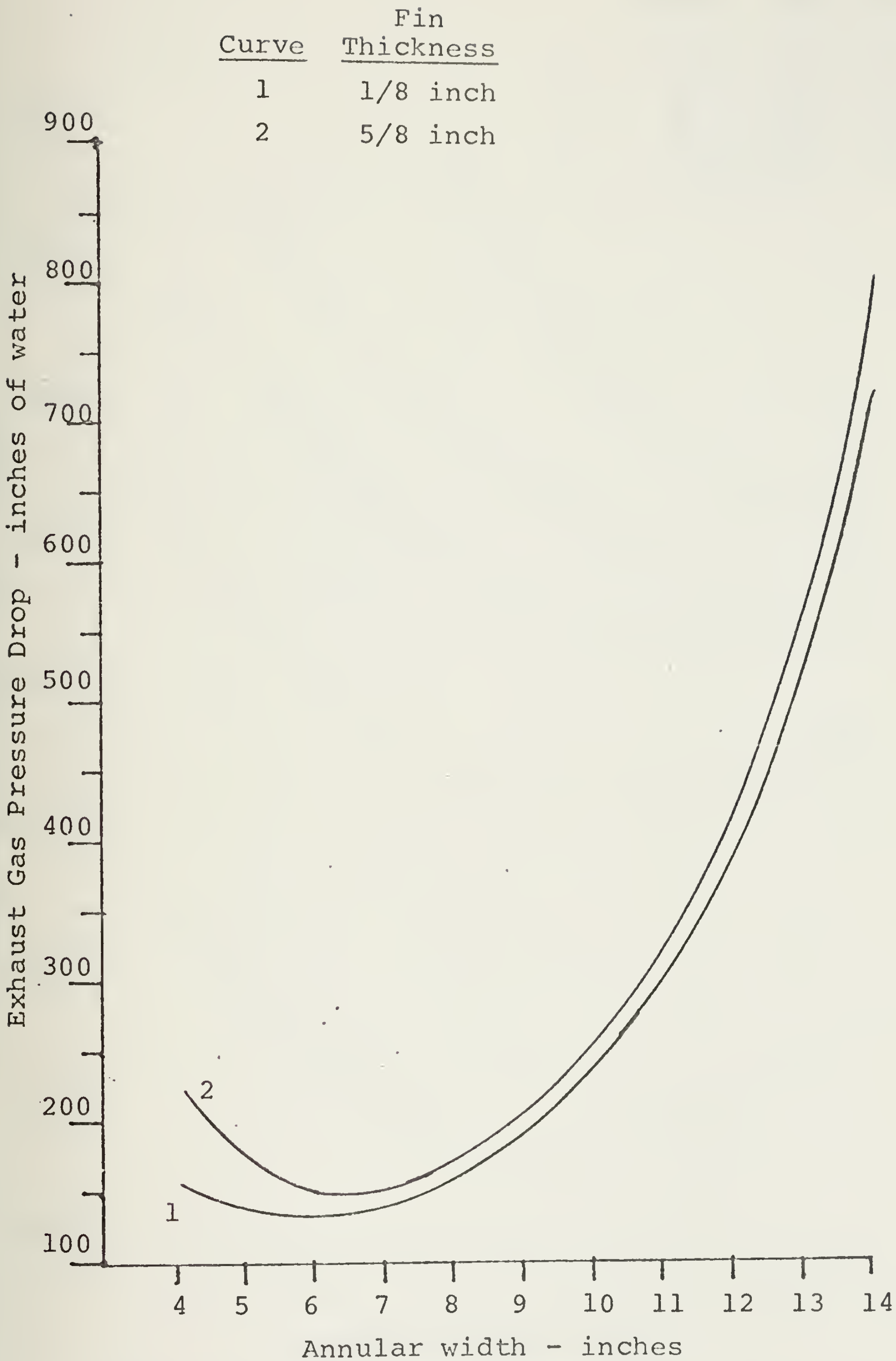


Figure 17. The Effect of Annular Width on the Exhaust Gas Pressure Drop Across the Boiler for a Fin Spacing of 2 Inches with a Rotation of 13,820 rpm.

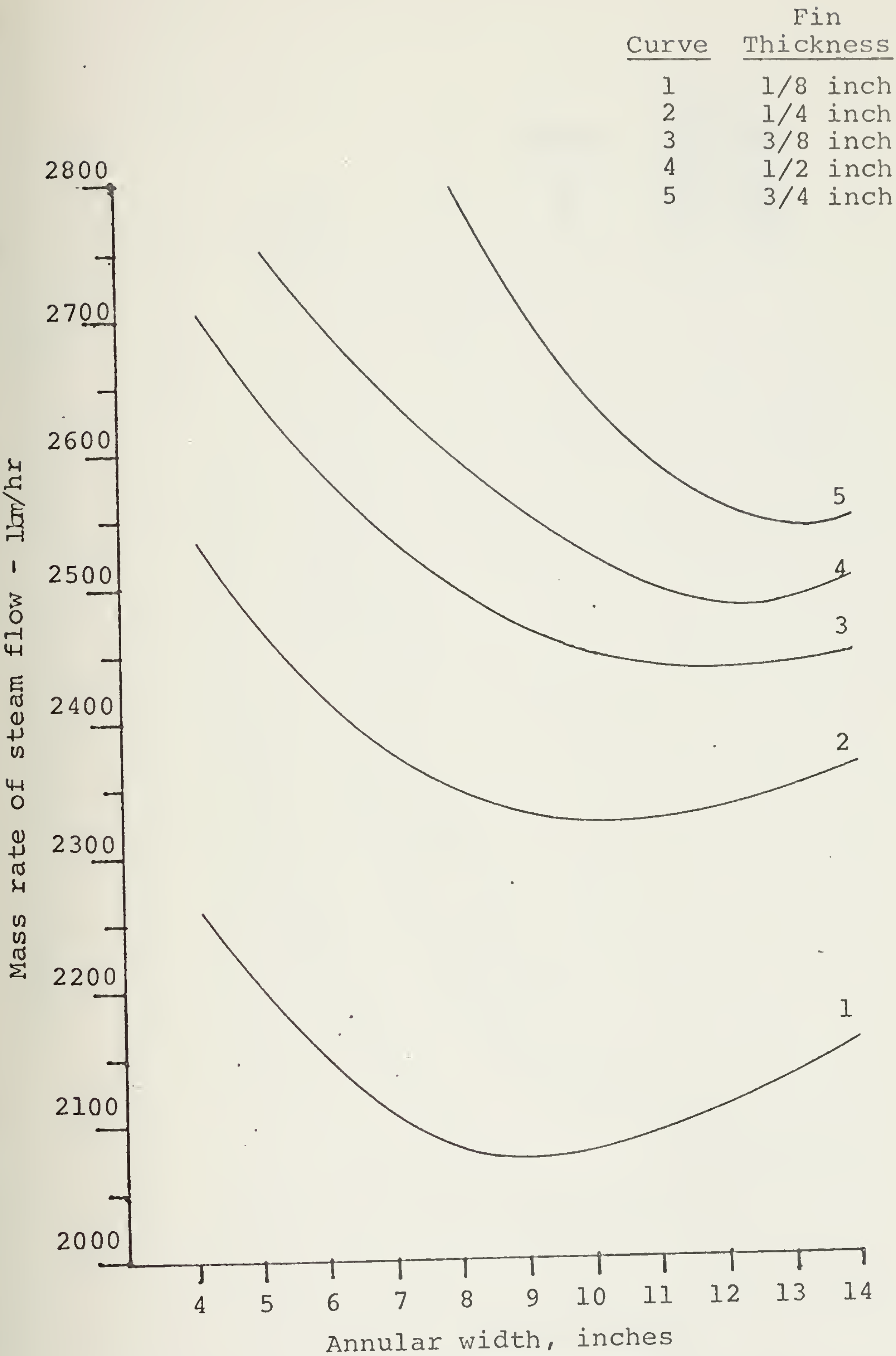


Figure 18. The Effect of Annular Width on the Mass Rate of Steam Flow for a Fin Spacing of 1 Inch with a Rotation of 3,600 rpm.

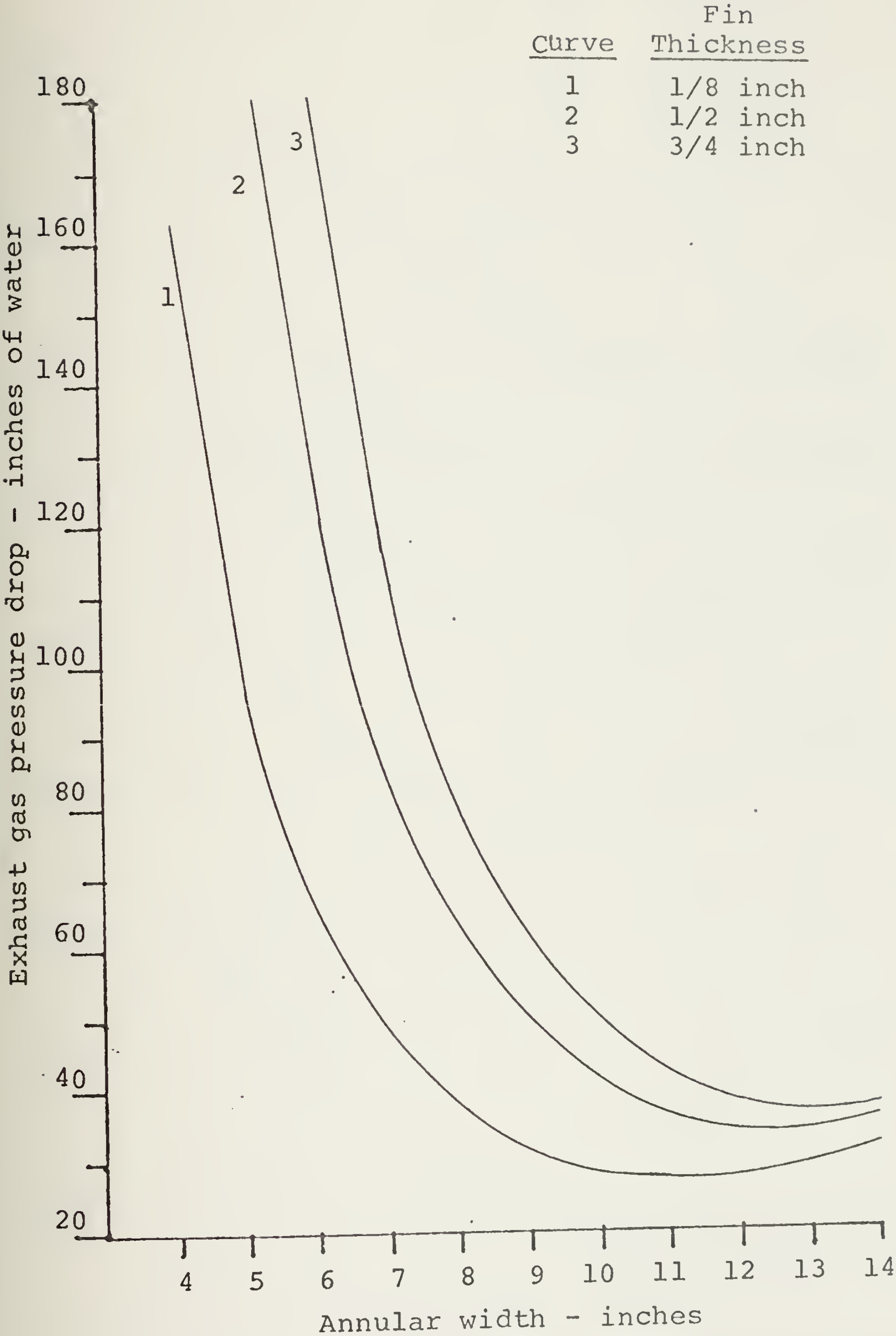


Figure 19. The Effect of Annular Width on the Exhaust Gas Pressure Drop Across the Boiler for a Fin Spacing of 1 Inch with a Rotation of 3,600 rpm.

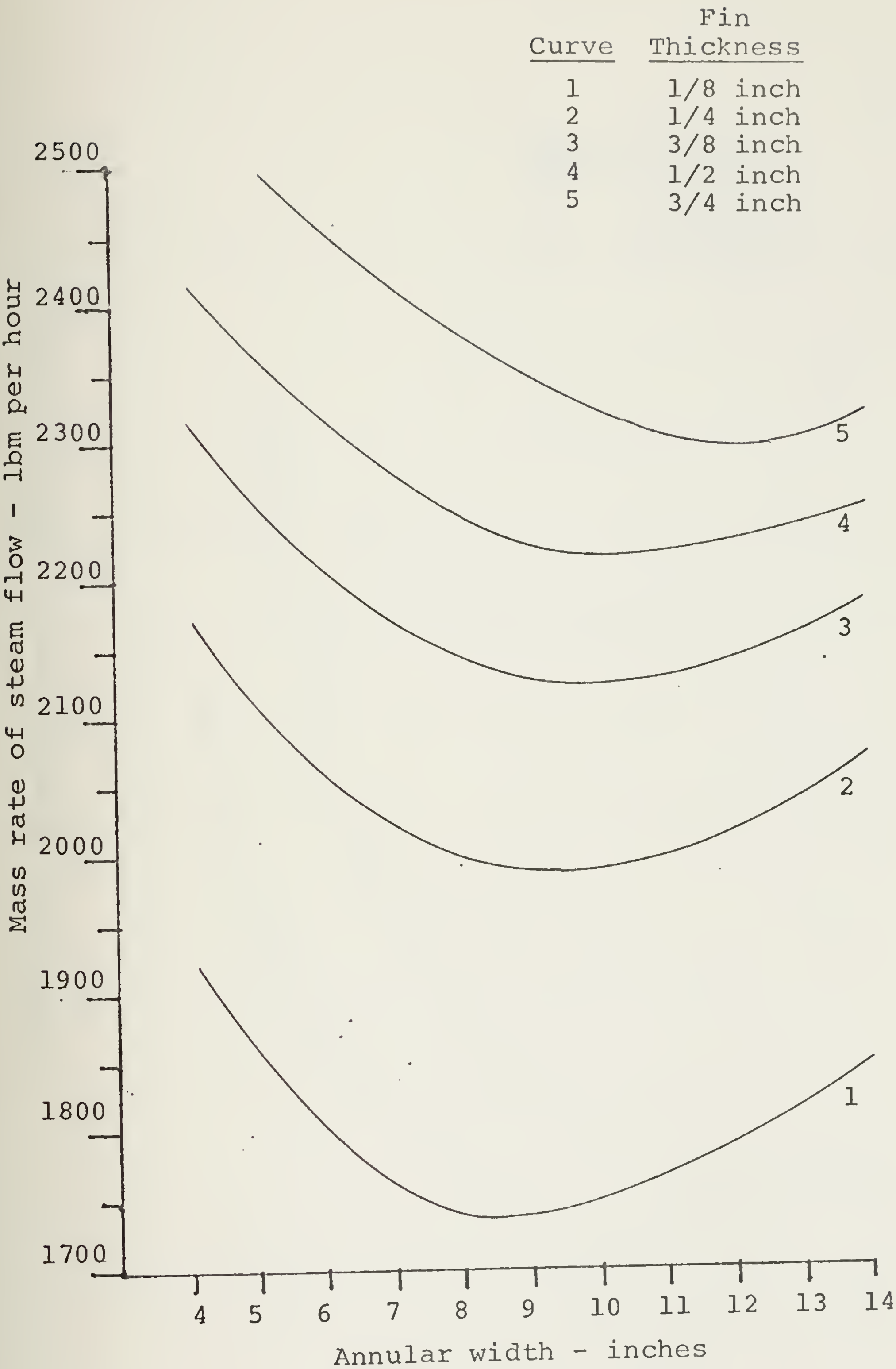


Figure 20. The Effect of Annular Width on the Mass Rate of Steam Flow for a Fin Spacing of 1.5 Inches with a Rotation of 3,600 rpm.

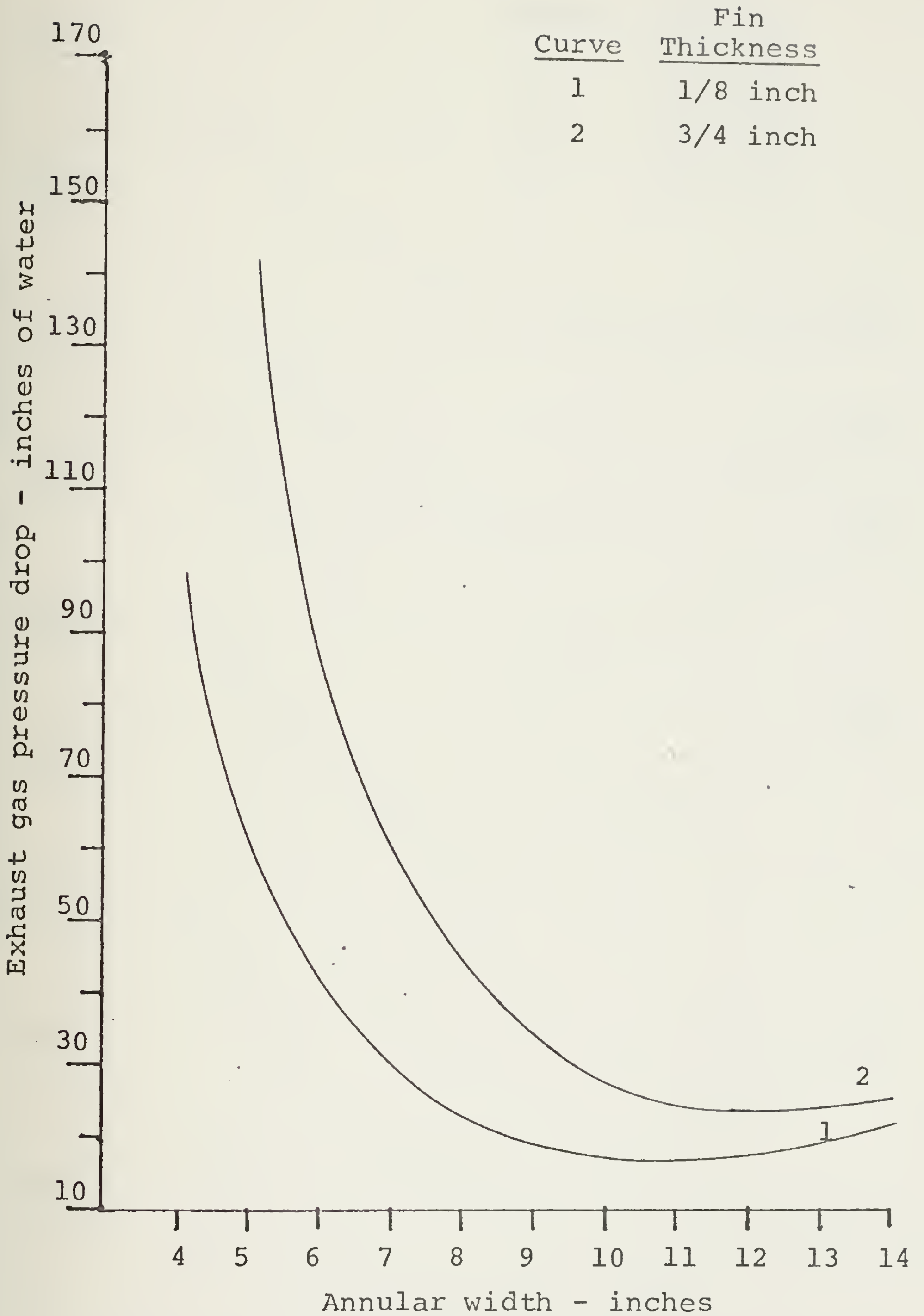


Figure 21. The Effect of Annular Width on the Exhaust Gas Pressure Drop Across the Rotating Boiler for a Fin Spacing of 1.5 Inches with a Rotation of 3,600 rpm.

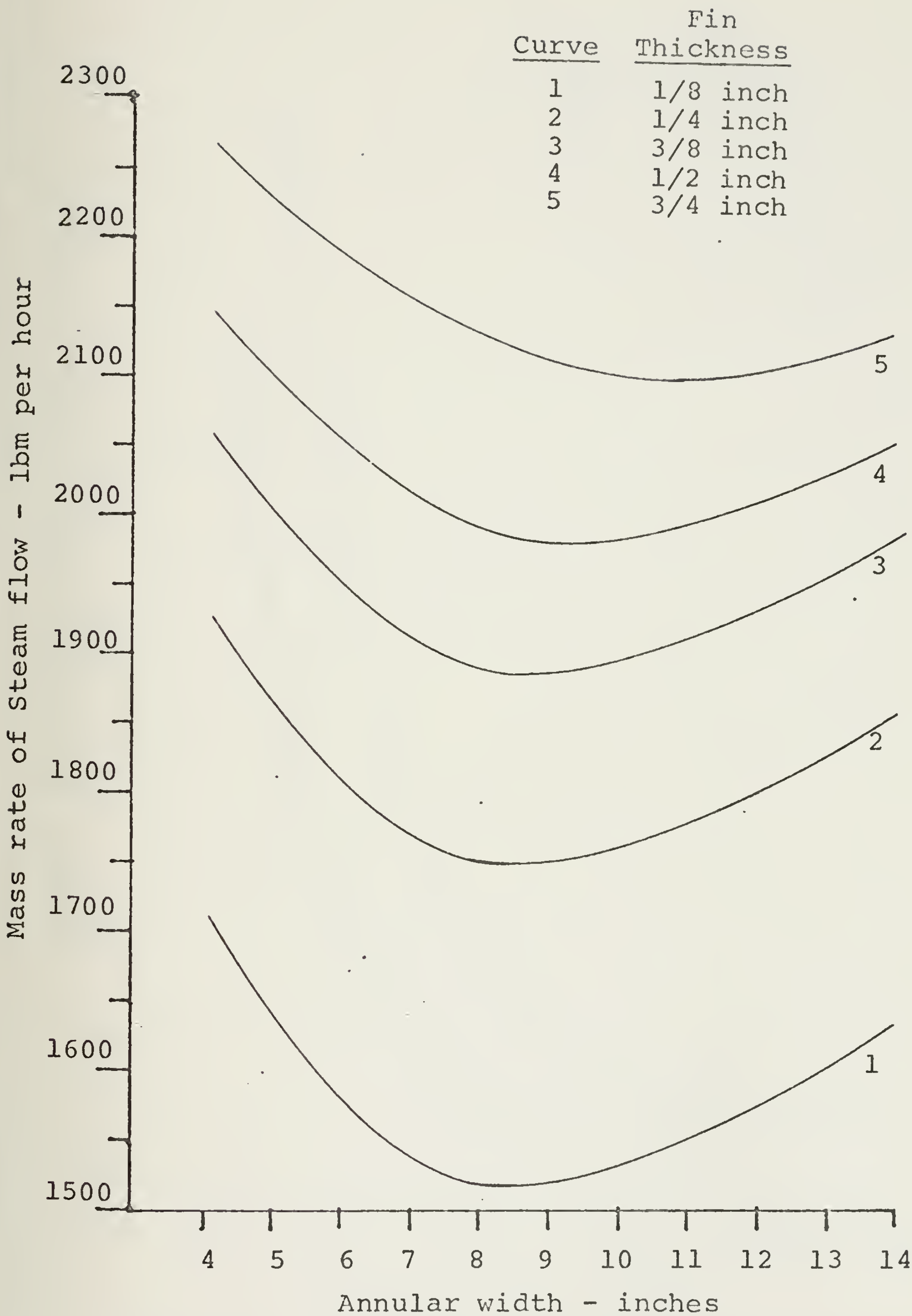


Figure 22. The Effect of Annular Width on the Mass Rate of Steam Flow for a Fin Spacing of 2 Inches with a Rotation of 3,600 rpm.

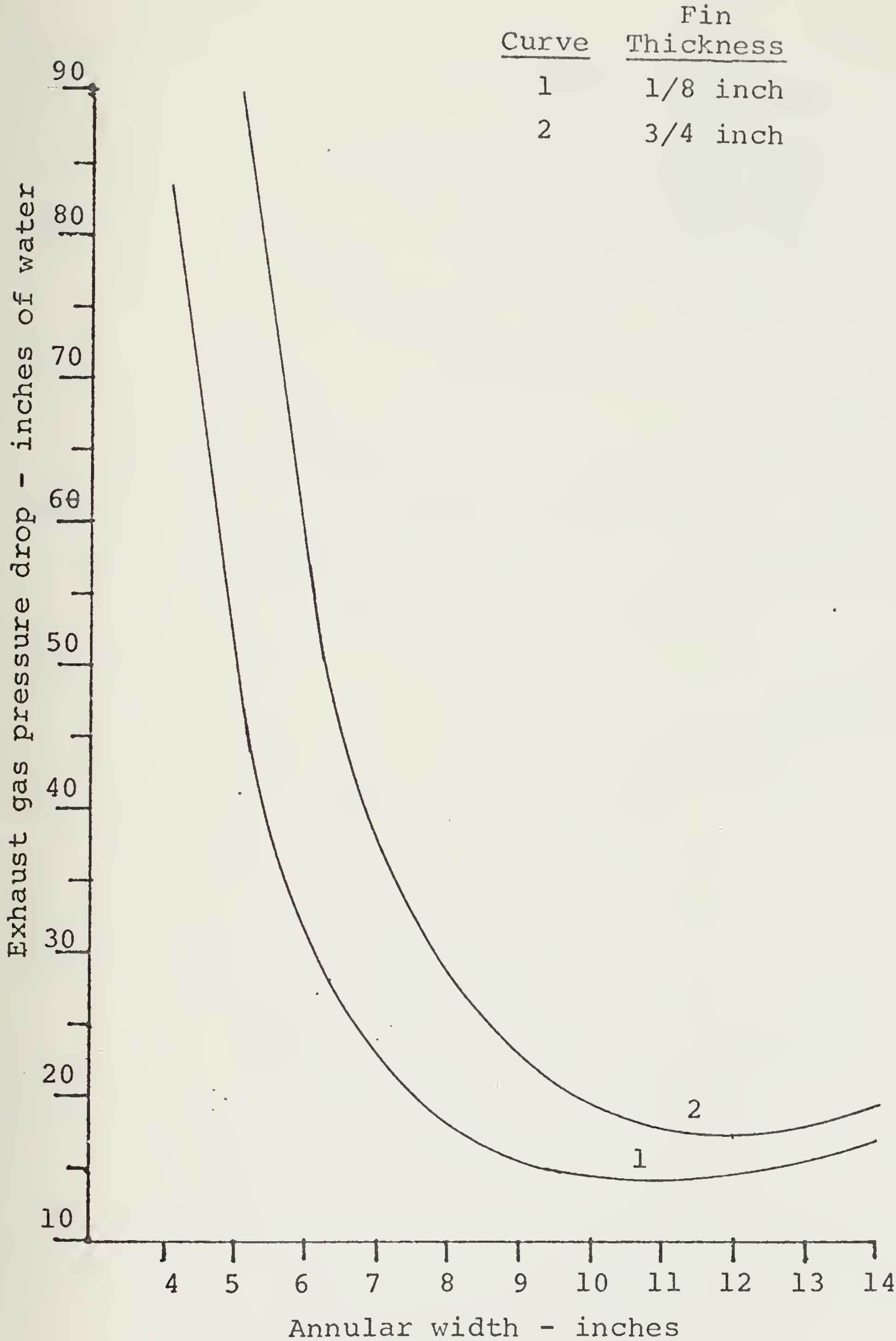


Figure 23. The Effect of Annular Width on the Exhaust Gas Pressure Drop Across the Rotating Boiler for a Fin Spacing of 2 Inches with a Rotation of 3,600 rpm.

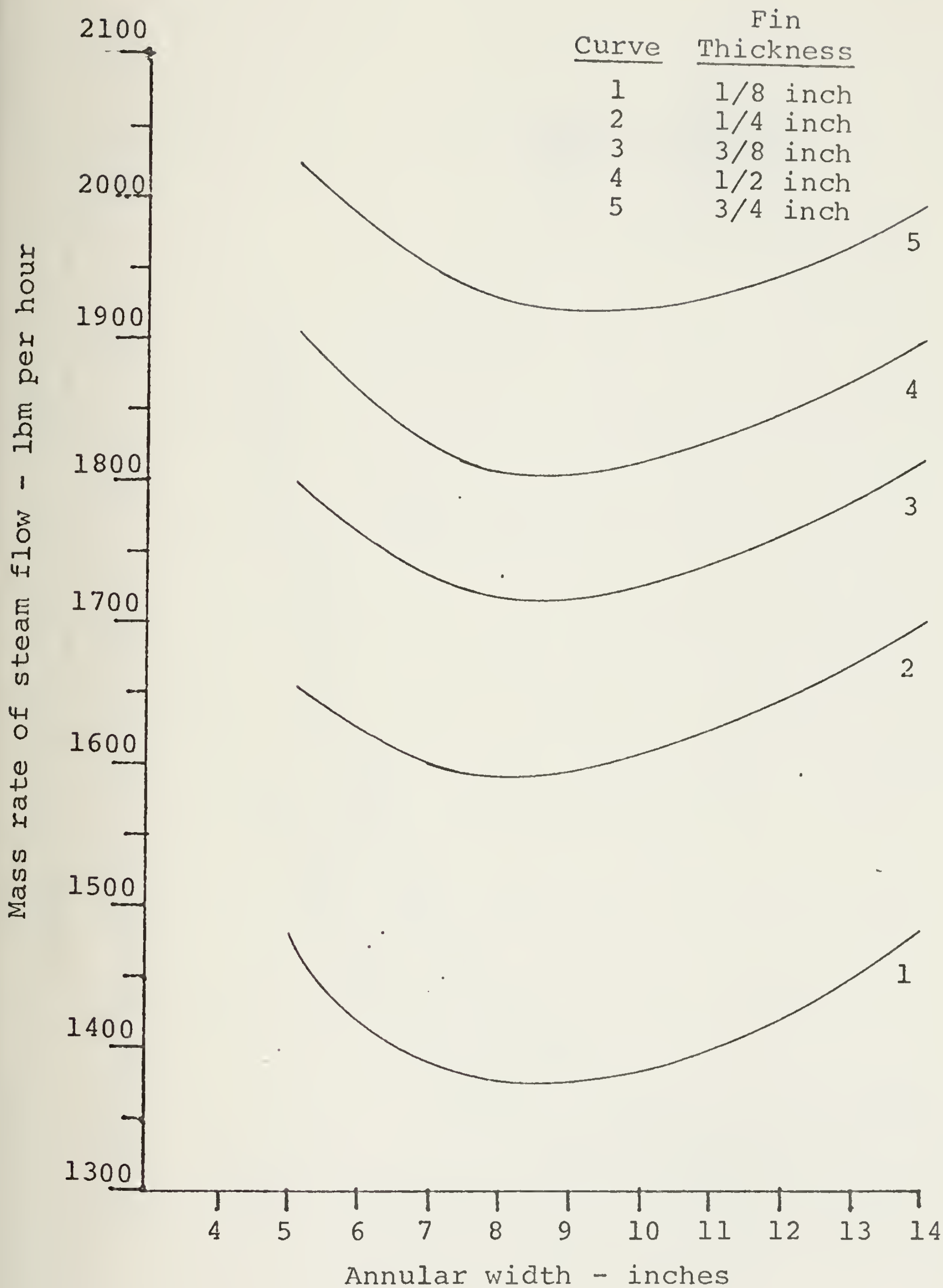


Figure 24. The Effect of Annular Width on the Mass Rate of Steam Flow for a Fin Spacing of 2.5 Inches with a Rotation of 3,600 rpm.

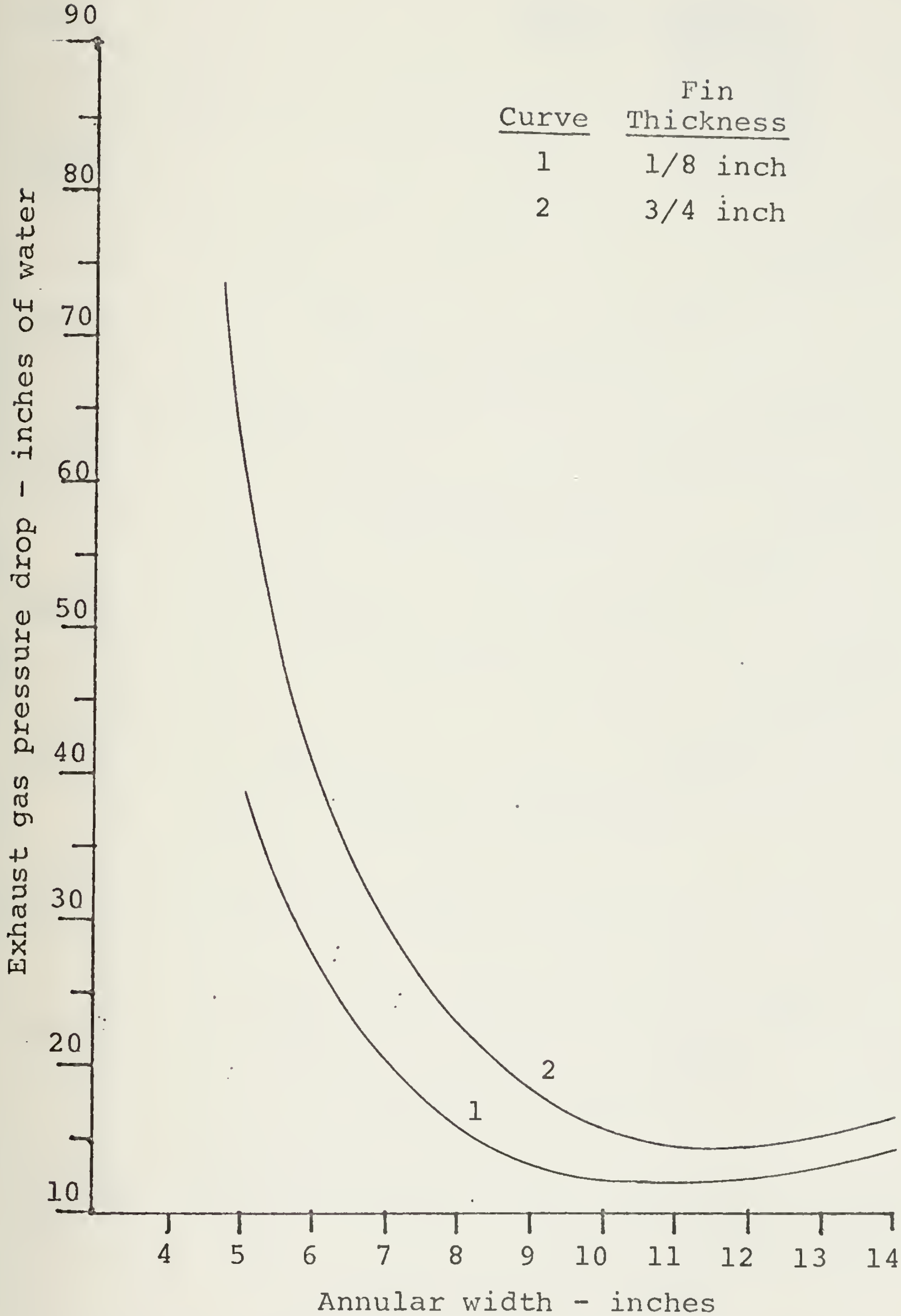


Figure 25. The Effect of Annular Width on the Exhaust Gas Pressure Drop Across the Rotating Boiler for a Fin Spacing of 2.5 Inches with a Rotation of 3,600 rpm.

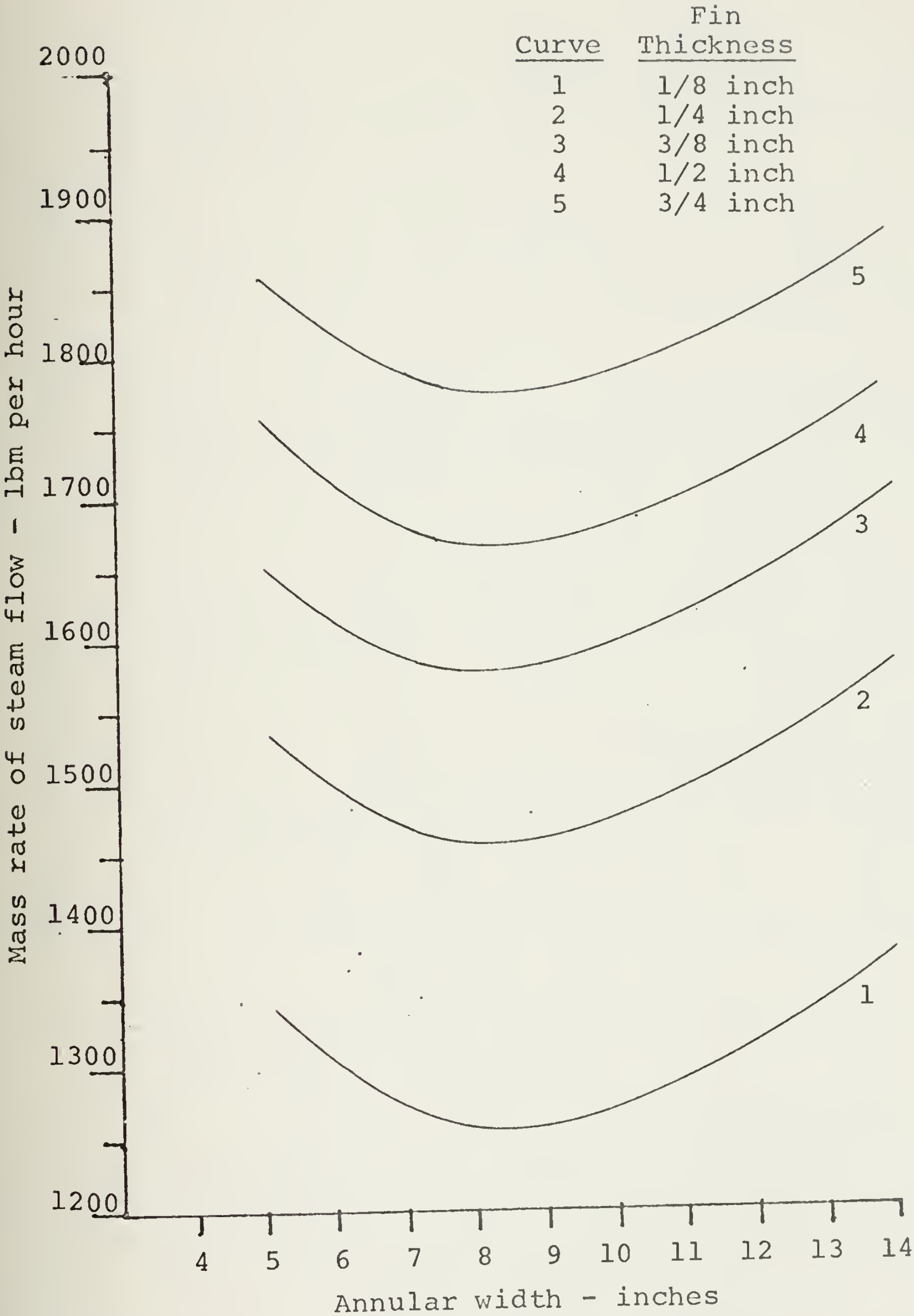


Figure 26. The Effect of Annular Width on the Mass Rate of Steam Flow for a Fin Spacing of 3 Inches with a Rotation of 3,600 rpm.

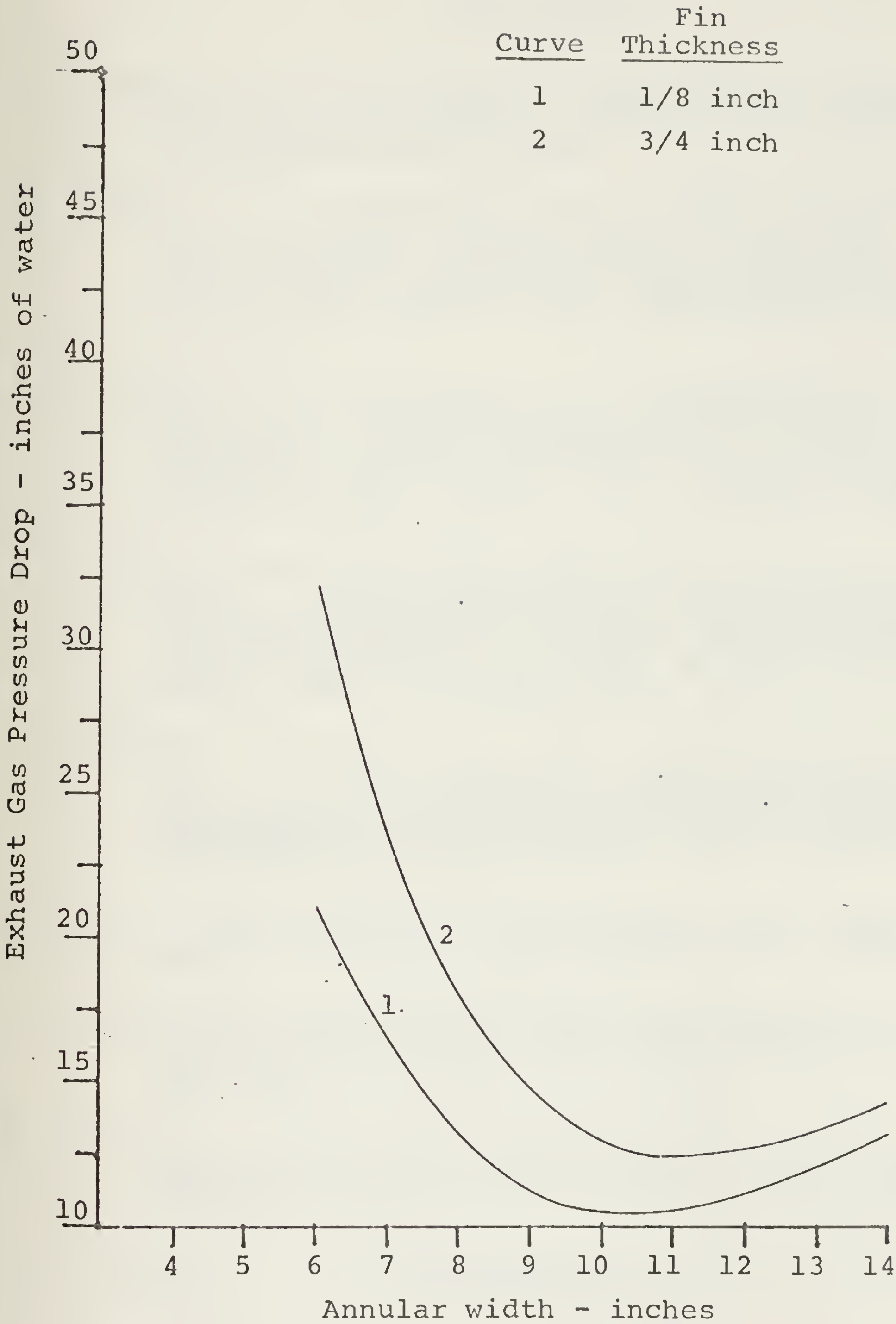


Figure 27. The Effect of Annular Width on the Exhaust Gas Pressure Drop Across the Boiler for a Fin Spacing of 3 inches with a Rotation of 3,600 rpm.

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ABSTRACT

A theoretical study was conducted using existing correlations to examine the feasibility of using a small rotating boiler rather than a large conventional boiler to recover heat from a marine gas turbine and to generate steam without creating an excessive exhaust gas pressure drop.

The boiler studied was a 6 foot long, 8 inch diameter cylinder rotated directly by the turbine or by a separate electric motor. It was placed inside the exhaust gas duct of an Allison 501-K17 gas turbine engine, forming an annular gas passage. Assuming incompressible, fully developed, turbulent gas flow with constant properties, the effect of annular width upon heat transfer and gas pressure drop was calculated.

The analytical results indicate a maximum steam flow rate of 2550 lbm/hr and a minimum gas pressure drop of 11 inches of water for the boiler at a rotational speed of 3600 rpm. When these results are compared to the heat transfer and pressure drop characteristics of large, conventional boilers, it is apparent that the rotating boiler is not feasible for large naval vessels with large steam requirements. However, it does warrant consideration for use on small vessels with lower steam requirements, where size and weight limitations may be critical.

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