GAS-TURBINE ENGINE UNIT
FOR INSTRUCTIONAL PURPOSES

By

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ABSTRACT

As a result of the increasing importancee of the gas-turbine engine in the power-plant field, especially in aviation, the Mechanical Engineering Department of Michigan State University is faced with the need of a low cost gas-turbine unit for instructional purposes. The problem then that confronts the university and which this paper strives to solve, is to secure a small low-cost gas-turbine engine of sufficient size and range to enable students to study at first hand the necessary control equipment and instruments and to analyze its operation. To solve the problem or to provide for the need, two alternatives were explored. The first alternative or solution is to convert a General Electric aircraft turbosupercharger into a simple, open-cycle gas-turbine engine and provide it with proper control equipment and appropriate instrumentation for thorough testing of the gas-turbine unit. As a second alternative a survey was made of small gas-turbine units available at present and which might be suitable for practical instruction at the engineering laboratory.

The two main problems in the conversion of the aircraft turbosupercharger are the design of an efficient combustion chamber and the incorporation of design features which will provide for maximum safety to operating personnel. In the combustor problem two solutions are offered. One solution would be to fabricate the combustion chamber to conform

with the propul[illegible] which is used on effi-
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accept [illegible] ll-[illegible] manufac-
turer. [illegible] will [illegible] but
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sonnel, [illegible] controls, emergency switches, by-pass
valves [illegible] are installed. In addition,
special [illegible] are
[illegible] area.

The survey of the small gas turbine unit has re-
vealed that only two commercial models [illegible] at pre-
sent. They are the Solar Aircraft Co.' "Mars" 55 hp engine and
Fairchild 'J44 jet engine.

From the standpoint of economy, reliability and
flexibility, it [illegible] turbo-jet [illegible] will [illegible]. [illegible] of the
[illegible] in favor of the
[illegible] unit will be its reliability [illegible] for
[illegible]. In addition, it will promote the
study [illegible] for which Illinois
[illegible] present.

Introduction

The gas turbine is not a new machine, for as far back as 130 B.C. the fundamental principle of the gas turbine engine was used by Hero of Alexandria, whose air heated in a vertical tube induced air flow in several tangentially displaced tubes creating an impulse which resulted in the rotation of a circular platform. Some people believe the gas turbine to be of recent origin. This impression is, in part, due to the current large increase of publicity on the combustion gas turbine engine, and to the dominant role it is playing in aviation's jet age.

The first significant attempt at building a practical gas turbine was made by Armengaud and Lemale during the years 1903 to 1906. Notwithstanding the poor performance of at the units, the experiments were significant because they were probably the first combustion gas turbines to actually produce useful work. In the intervening period up to the years just before World War II little was accomplished in gas turbine development. However, just before World War II, Switzerland built successful gas-turbine plants for industrial purposes. Recognizing its tremendous possibilities in aviation, Great Britain under the pioneering of Air Marshall Whittle of the Royal Air Force, started research and development on the engine during World War II for aircraft propulsion. Its successful development is intimately associated with the advances made in metallurgy which made possible the utilization of the high turbine inlet temperatures and the progress towards a more efficient compressor. Today the gas turbine has earned its place as a proven heat engine, with a demonstrated record of reliability. The inherent simplicity of its design, together with its independence of water facilities and complex auxiliaries, ideally suits it for certain types of service. Applications of the gas turbine are numerous as improvements are incorporated into their design and construction.

As they become cognizant of the potentialities of the gas turbine, more and more companies are inaugurating intensive and extensive research and developmental projects on it. To meet the demand for more trained men to take part in the continuing research, and to take up where the older men will leave off, universities and colleges have a special obligation to train engineering students in the operating and design fundamentals of this engine. It is for this reason that Michigan State University felt the need of a gas-turbine engine in its laboratory.

The problem confronting the university is to secure a small low-cost turbine engine of sufficient size and range to enable students to study at first hand the necessary control equipment and instruments and to analyze its operation. This will allow the students to observe and record all the important factors that affect the performance of the engine. In other words, the problem is to determine the optimum size and type of turbine for instructional purposes.

This paper will strive to investigate the most practical way of providing for this need. This study includes the conversion of an aircraft turbo-supercharger of World War II vintage into an open-cycle gas turbine complete with accessories which will make it suitable for instructional purposes. A comparison is made between this converted engine and engines available in the small gas-turbine market, in order to determine which one will serve the needs of the University best.

GENERAL DISCUSSION

LABORATORY ENGINE REQUIREMENTS

In order to impart to engineering students the basic fundamentals involved in the operation of a gas-turbine engine and to give them a first-hand knowledge in the way various factors affect its operation, the engine and set-up that should be used for practical instruction must meet certain basic requirements. Before enumerating these requirements it would best serve our purpose of clarifying subsequent discussion to give a general picture of what a basic gas-turbine engine looks like. Figure 1 is a schematic of a simple open-cycle gas turbine. Basically the gas turbine is a simple machine and consists of three major components as follows: the compressor, which compresses the air and delivers it to the combustor or burner; the combustor where the compressed air is mixed with the atomized fuel and burned; and the turbine where the high pressure, high temperature combustion products expand and do work. In a single-shaft, simple engine the compressor is on the same shaft as the turbine. In a practical and efficient machine a considerable amount of torque energy in surplus over that used to operate the compressor is available to do various kinds of work such as driving a generator, a pump or other driven machines. Aside from the accessories such as fuel pumps, etc., used to operate the gas turbine, auxiliaries such as intercoolers, regenerators, and reheaters are installed in an effort to

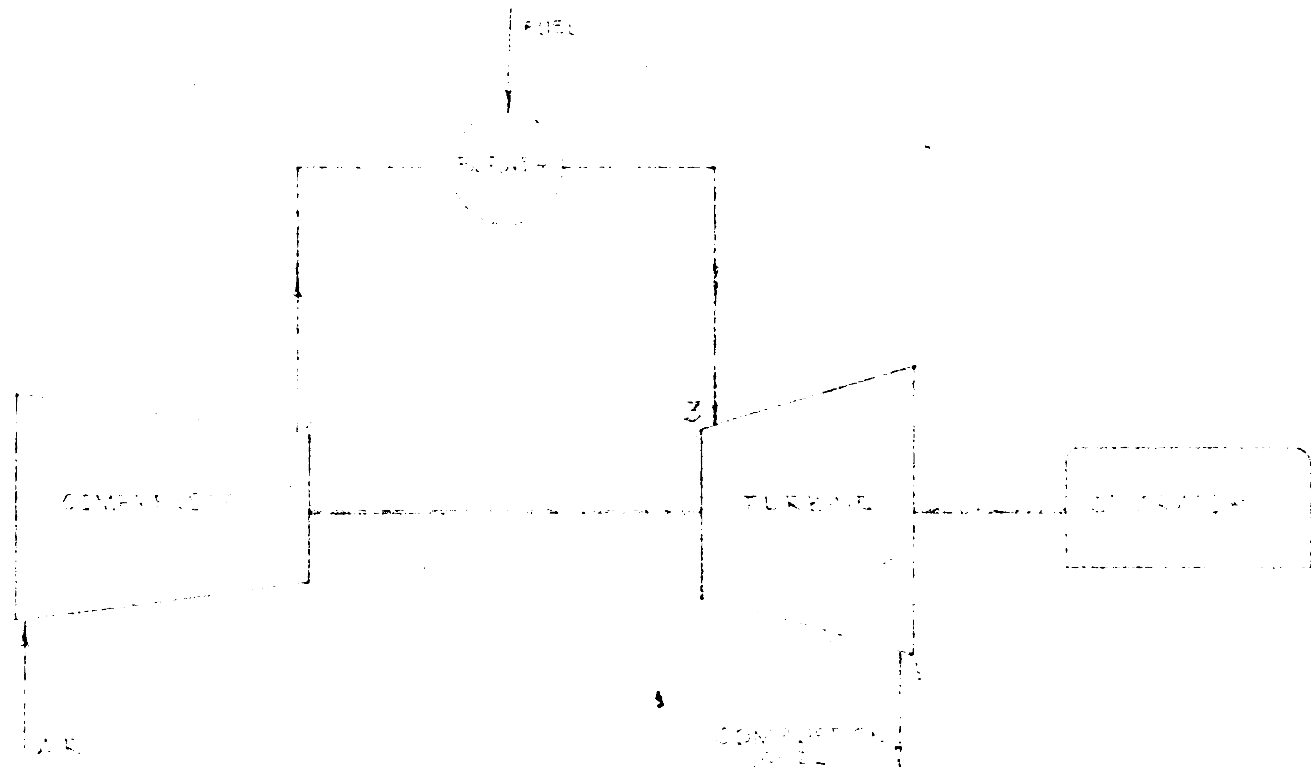

FIGURE 1 – SCHEMATIC OF A SIMPLE OPEN-CYCLE GAS-TURBINE

increase overall turbine efficiency. The engine can operate without these auxiliaries but at a lower efficiency and output.

There are certain operating characteristics of the gas turbine that are important and should definitely be demonstrated to students. For example, the ratio of the maximum pressure in the cycle (at the compressor discharge or the turbine inlet) to the compressor-inlet pressure (designated "pressure ratio") is one important item that must be determined in the operation of a gas-turbine engine. It is imperative therefore that manometers or other pressure-measuring devices be installed at the compressor outlet. In a stationary machine such as the one under consideration, the lower pressure level or compressor-inlet pressure can be taken from the barometric reading of the surrounding air. The selection of a suitable pressure ratio is of great importance in the successful design of a gas turbine and is controlled by many factors. Ambient temperature, turbine-inlet temperature, component efficiency and design, type and arrangement of the cycle all affect the optimum pressure ratio, which consequently varies from case to case.

Turbine-Inlet Temperature: The temperature of the combustion products as they leave the combustor and before they enter the turbine is commonly termed the turbine-inlet temperature. This temperature is a very important variable on which the design of the engine usually depends. In fact it is the primary variable when efficiency is the sole criterion. A thermocouple or a traversing thermocouple should be installed in order to obtain the average temperature of the combustion gases before they enter the turbine. This temperature is also useful in determining the amount of work done by the turbine and is used to calculate the amount of heat added in the combustor. Due to metallurgical considerations this will also become the limiting factor in the further improvement of engine efficiency.

Air-mass Flow: Air-mass flow expressed in pounds of air taken in by the compressor per unit is another variable. This variable is useful in the plotting of a compressor-performance curve. It is also of use in determining the amount of fuel that should be added in the burner to attain a desired temperature at the turbine inlet which will not exceed design temperatures for safe operation of the turbine blades. Air-mass flow together with the metered fuel will enable the calculation of the air-fuel ratio of a specific run of the engine. Air-mass flow can be measured by venturi meters or air flow meters attached to the compressor inlet.

Fuel Flow: The amount of fuel which is fed into the burner to obtain a given amount of work from the engine is one way of expressing the engine's operating economy and its efficiency. The ratio of the air to the fuel used is an important operating characteristic of a gas-turbine engine. For a given operating condition there is an air-fuel ratio that will give the optimum performance. Fuel regulation also controls the turbine-inlet temperature by regulating flow of fuel to the burners and for instructional purposes it is the best and most practical way of controlling not only turbine-inlet temperature but turbine speed as well. Aside from the fuel-flow measurements and control an over-speed tripping device should be installed to automatically stop flow of fuel to the burners when design speed is exceeded.

Turbine-Exhaust Temperature: The temperature of the gases at the exhaust of the turbine is useful in calculating the work accomplished by the turbine and would be useful in determining the efficiency of the turbine. Since the installation of temperature probes in this section is not as critical as at the inlet of the turbine the probes need not be traversing.

Ambient Temperature: Reading of ambient temperature must be provided as this is important in determining the work done by the compressor and consequently the efficiency of the engine unit.

Shaft Output: The actual work done by the engine or the torque energy available at the shaft over that used by the compressor should be measured. Shaft torque measurements may be made by strain gauges or spring couples or this may be done by installing reduction gearing and a generator or better yet a high-speed dynamometer which does not necessitate any reduction gearing. This will make possible the determination of the overall efficiency of the engine and the calculation of the mechanical efficiencies of the compressor and turbine.

In addition to the instruments necessary in obtaining the important variables and the accessories required in the smooth operation of the engine some control equipment must be incorporated in order to insure safe operation of the engine and to protect operating personnel. As mentioned above, there is a necessity of installing a governor-controlled tripping mechanism which cuts off automaticly the fuel to the burner the moment the speed of the turbine exceeds a predetermined limit necessary for the safe operation of the engine. In addition to this, a high-speed generator tachometer may be installed as a visual check of the turbine speed. Manual operation of the fuel regulating valve will enable the operator to regulate temperature at combustor and at turbine-inlet. Manual fuel control can also be used if pressure at the compressor outlet or at burner, (as shown by pressure indicator at the compressor outlet) exceeds design pressure necessary for optimum operation. Due to the nature of the usage of the engine where operation is for short duration only there is no cooling problem except for the correct lubrication of the compressor and turbine bearings.

OPTIMUM ENGINE SIZE

In the selection of a suitable gas-turbine engine for practical instruction purposes there is no definite optimum size. Optimum engine size for instructional purposes would mean the smallest possible engine size, and consequently a cheaper machine, which will permit the recording, with facility and with the least possible error, of all the necessary operating characteristics and variables enumerated above.

However, in sizes smaller than the selected optimum, instrumentation and mensuration will be quite difficult and accurate readings of variables would be hard to obtain. Though it would be worthwhile to show and demonstrate the effect of regeneration, intercooling, reheat, and in some instances precooling, on the efficiency and operating economy of the gas turbine, it would seem impractical and uneconomical for instructional purposes because it would inherently entail a large and costly engine. Manufacturers of small gas turbines in general do not incorporate heat exchangers as they become bulky and the increment of cost then rises more than the added advantage [illegible]. The optimum size varies with different purposes, [illegible] engine which will be required to give fully the basic fundamentals of the operating characteristics of a simple gas-turbine [illegible] will have a different optimum size than one which will be required to give the operating characteristics of a more complex gas-turbine cycle. With respect to what is available in the small gas turbine market, the small gas-turbine engine manufactured by Solar Aircraft Co., which is rated at 55 h.p. [illegible] for instructional purposes as the optimum size. It is [illegible] with or [illegible] provision for the installation of instruments and measuring devices which will [illegible] of the necessary operating characteristics [illegible] used above.

COMPRESSOR AND TURBINE TYPE

In the gas-turbine field today, all of the compressors used are either the axial-flow type or radial-flow type. A comparison of the two types will be made in order to determine which of the two will be more suitable for an instructional engine.

The axial-flow type is a more efficient type of compressor than the radial-flow type. The axial-flow compressor can handle larger volumes of air than can the radial-flow compressor. Due to its simplicity and rugged construction the radial compressor [illegible], is less liable to [illegible], and is less likely to foul up and lose efficiency if run in a dusty atmosphere. The other advantage of the radial compressor is that it has greater output flexibility, i.e. the machine will work satisfactorily over a greater range of pressure and rate of air flow at any given speed, thus making the performance of the whole engine less critical and less dependent on the accurate matching of the turbine and compressor characteristics.

From the foregoing discussion it should be apparent that the radial type compressor is a more suitable type of compressor in a gas turbine for instructional purposes. Efficiency is not an important consideration when the engine is used for practical instruction. For as long as it is efficient enough to have surplus torque energy that can be measured by a dynamometer or other devices, it will have served the purpose of demonstrating operational variables of the gas turbine. The inability of the radial compressor to handle large volumes of air will not affect the ability of the engine to demonstrate basic operating characteristics. The low first cost and low maintenance cost are important items in the selection of an instructional engine and for that matter any engine. In this respect the radial type has an advantage over the axial type. As a piece of laboratory equipment where the very nature of work performed precludes some minor mishaps the radial compressor's resistance to accidental damage and its non-fouling characteristics are great assets. Again, as a laboratory engine, where it is necessary to operate the engine through a great operational range, the greater stable range of the radial compressor and its consequent benefit as explained above is another great asset.

As to the type of turbines used, it makes little difference whether a reaction-type turbine is used or an impulse type turbine is used, except for the cost of manufacture. However, most gas turbines are of the reaction type including the small ones.

Regarding availability, so far no manufacturer, especially those of the small gas-turbine engine, has made available to the public turbines or compressors as separate items. They are sold as a whole unit. Due to previous commitments with the government some manufacturers of small engines are unable to make any sale to the public as yet. The low production rate of small gas turbines has kept the price exceptionally high. Mass production should bring prices to a more reasonable level.

TURBO-SUPERCHARGER CONVERSION INTO
A SIMPLE OPEN CYCLE GAS TURBINE

Due to the high price of small gas-turbine engines, a good alternative to provide for an inexpensive gas-turbine engine for instructional purposes would be to convert a turbo-supercharger used in aircraft engines into a simple cycle gas turbine.

The particular type of turbo-supercharger to be converted is a G. E. model-7B-B31-A1. The conversion would generally cover the design or selection of an appropriate combustion system, instrumentation, and control system. The design of the combustion system would include the design of the combustion chamber and proper selection of the fuel system, i.e, fuel pumps, fuel lines, fuel control valves, fuel nozzle, etc. The design or proper selection of the combustion chamber is of primary importance and will be treated first.

COMBUSTION CHAMBER DESIGN

The purpose of the combustion chamber is to supply heat to the stream of air passing from the compressor to the turbine in such a manner that a steady stream of gas at a uniform temperature is produced. In order to do this function a combustor in a simple open-cycle gas turbine burns fuel directly in the air stream.

In the design of the combustor certain important primary considerations must be taken into account to insure satisfactory performance. It must possess to some degree all of the following characteristics:

(a) Minimum pressure loss

(b) Positive ignition

(c) Flame stability with uniform outlet temperature

(d) Complete combustion or high combustion efficiency

(e) Absence of coke formation

(f) Durability and freedom from distortion

Since limited theoretical information is available and because of the complexity of the combustion processes the design of the combustor will depend to a great extent upon empirical information and upon the results of development and test programs. The combustor, therefore, for the turbo-jet engine will be designed to incorporate the primary design features that have been developed from highly-efficient [illegible] of similar design [illegible] experimental [illegible].

Relying upon the experience of [illegible] Company and [illegible] Corporation, two of the prominent manufacturers of combustion chambers for small gas turbines, it is decided to design a can-type combustion chamber. The can-type combustion chamber is the simplest and most flexible type of combustion chamber for a given design.

[illegible]:

As this combustor is to be used with the existing compressor of turbine, the work it must perform should conform to the optimum design conditions of the compressor and turbine in order to have an efficient and safe machine.

From manufacturers' rating and performance curves (See Figures 2 & 3) of the compressor and the turbine as published by the General Electric Co., manufacturer of the compressor and the turbine, the amount and condition of combustion gas the combustor must deliver, at optimum conditions, to the turbine may be found.

Turbo-supercharger rating:

Maximum turbine inlet temperature - 1500°F

Maximum Design RPM - 24,000 rpm

From the compressor and turbine performance curves (Figs. 2 and 3), optimum pressure ratio is found to be around 3.

From Figure 2, the flow factor, F = .1 when pressure ratio equals 3 and at maximum compressor efficiency. From Figure 3, the flow factor, F is given as:

$$F = \frac{\omega_1}{W_1 d^2}$$

where ω_1 = Compressor mass flow = #/min

d = Compressor diameter = 12.25"

$W_1 = 15.620\, P_1/\sqrt{T_1}$ (refer to figure 2)

P_1 = Compressor inlet total pressure ("Hg. abs)

= 29.05 (yearly average for Lake Landing)

T_1 = Compressor inlet total temperature

—°F abs.

= 47.3° F or 507.3° R (yearly average)

$$\therefore W_1 = 15.620 \times \frac{29.05}{\sqrt{507.3}} = 20.2$$

$$\therefore \omega_1 = F\,(d)^2\,(W_1)$$

$$= .1\,(12.25)^2\,(20.2)$$

= 304.5 pounds of air per minute

The total weight of combustion gases is equal to the weight of air per minute plus the weight of the fuel. But the minimum air-fuel ratio to maintain a stable combustion is around 50:1 as given by Jennings and Rogers in "Gas Turbine Analysis and Practice". The value 50:1 for air-fuel ratio is the same result obtained from Southampton experiments published by Horney in "Development and Testing of a Gas Turbine Combustor". Therefore, for an air-fuel ratio of 50:1 the fuel required per minute equals $\frac{304.5}{50}$ = 6.1 lbs.

Therefore, total weight of combustion gases = 304.5 + 6.1 = 310.6 pounds of gases per minute.

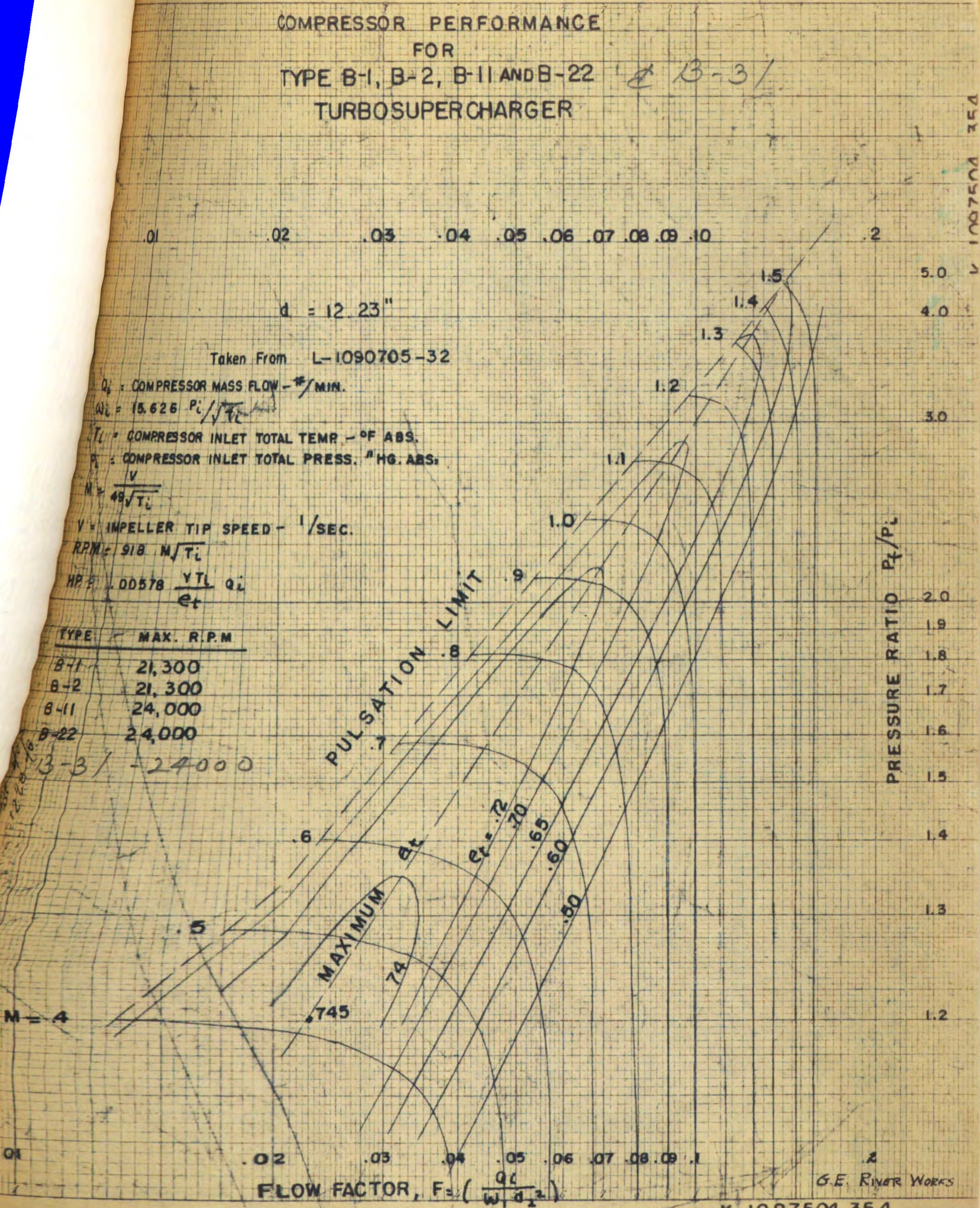

FIGURE 2 - COMPRESSOR PERFORMANCE CURVES

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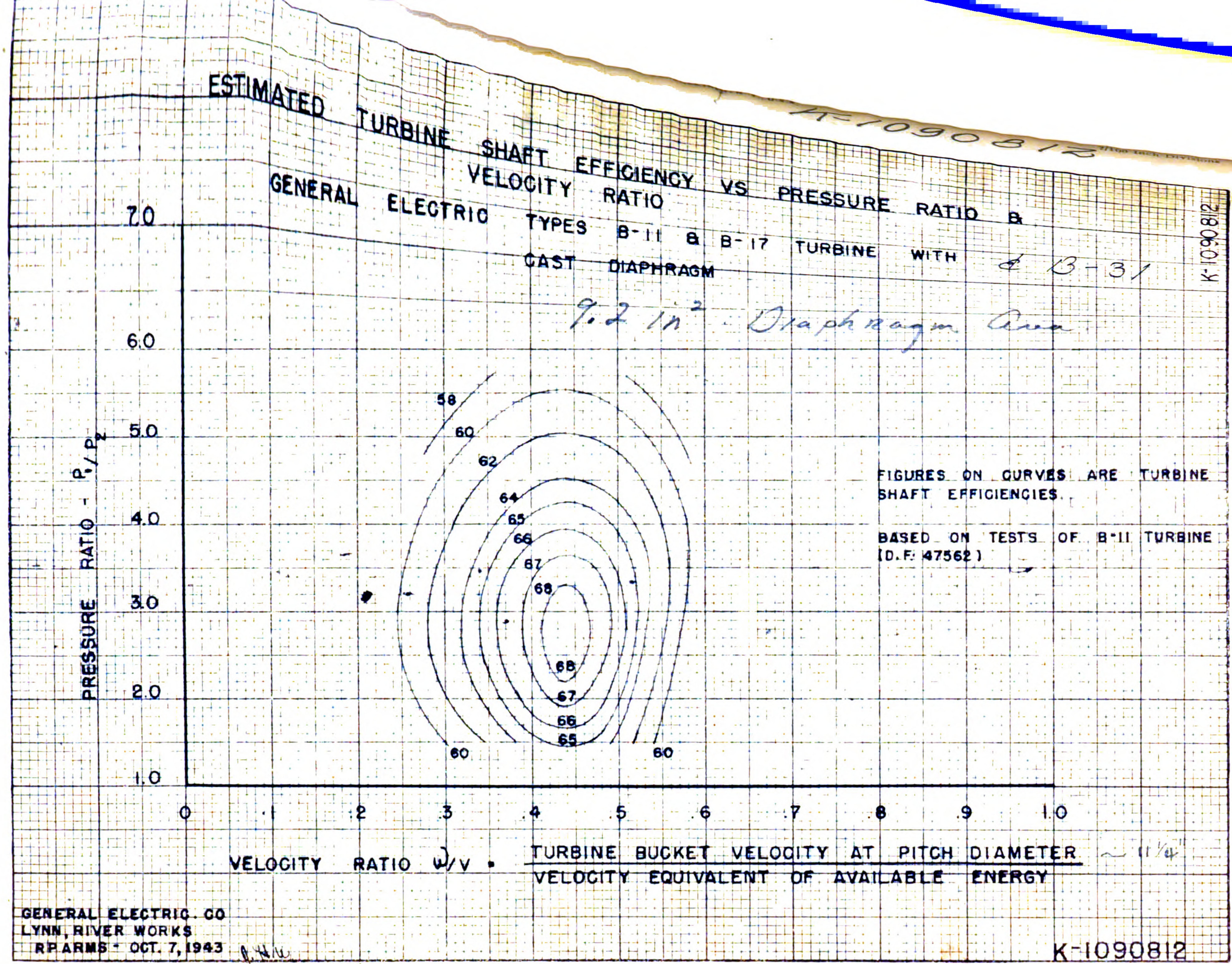

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FIGURE 3 – TURBINE PERFORMANCE CURVES

Fig 3

But the right air-fuel ratio required to obtain 1600°F at the inlet to the turbine can be found by the use of graphs developed by [illegible] and published in a paper entitled "Fuel-air Ratio for Constant Pressure Combustion of Hydrogen Fuels".

Before proceeding to find the required fuel-air ratio the compressor outlet temperature must be found first.

$T_{c.i}$ = Temperature of compressor inlet

= 47.3° or 507.3° R (yearly average of East Lansing given by U.S. Weather Bureau)

$T_{c.o.}$ = Compressor outlet temperature (uncorrected)

$T'_{c.o.}$ = Temperature of compressor outlet (corrected for efficiency)

P.R. = Pressure ratio

= 3 (taken from Performance curves - figures 2 and 3)

k = 1.4

η_c = compressor efficiency

= 74.5 o/o (taken from Compressor performance curves, fig. 2)

$$\frac{T_{c.o}}{T_{c.i}} = (P.R.)^{\frac{K-1}{K}} = (3)^{.286} = 1.368$$

$$\frac{T_{c.o.}}{507.3} = 1.368 \quad \therefore \quad T_{c.o.} = 507.3 \times 1.368 = 693^\circ F \text{ abs.}$$

$$\eta_c = \frac{T_{c.o.} - 507.3}{T'_{c.o.} - 507.3} = .745$$

$$.745 = \frac{693 - 507.3}{T'_{c.o.} - 507.3} = \frac{185.7}{T'_{c.o.} - 507.3}$$

$$\therefore T'_{c.o.} = \frac{195.7}{.745} + 507.3 = 240 + 507.3$$

$$= 756.3^\circ \text{ F abs} \approx 296.3^\circ F$$

Therefore combustor must raise temperature from 296.3 to 1600°F, or a temperature rise of 1304°F.

From Figure 4, fuel air ratio with combustion chamber temperature rise of 1304°F and inlet temperature to the combustor of 296.3°F or 756.3°R, **F/A** = .0161.

Since kerosene, the fuel to be used, has a hydrogen-carbon ratio of .167 and the fuel-air ratio taken from figure 4 is plotted from a hydrogen-carbon ratio of .167, so from figure 5, correction factor to be used is equal to .9956.

Since kerosene has a lower heating value of around 19,670 Btu/# and figure 4 is for a fuel of a lower heating value of 18600 Btu/#, so from figure 6 correction factor to be used is equal to .944.

Correction factors for presence of water vapor and for net products of combustion that may be included in the inlet air are neglected since they are negligible.

$\therefore$ Corrected fuel-air ratio = .0161 x .944 x .9956 = .01511 or air-fuel ratio is 66.2# of air per lb. of fuel.

Therefore in order to raise 304.5 pounds of air per minute from 196.3°F to 1600° the combustor must burn 1 pound of fuel for every 66.2 pounds of air. Total pounds of fuel every minute burned would then be equal to $\frac{305.4}{66.2}$ = 4.62 lbs.

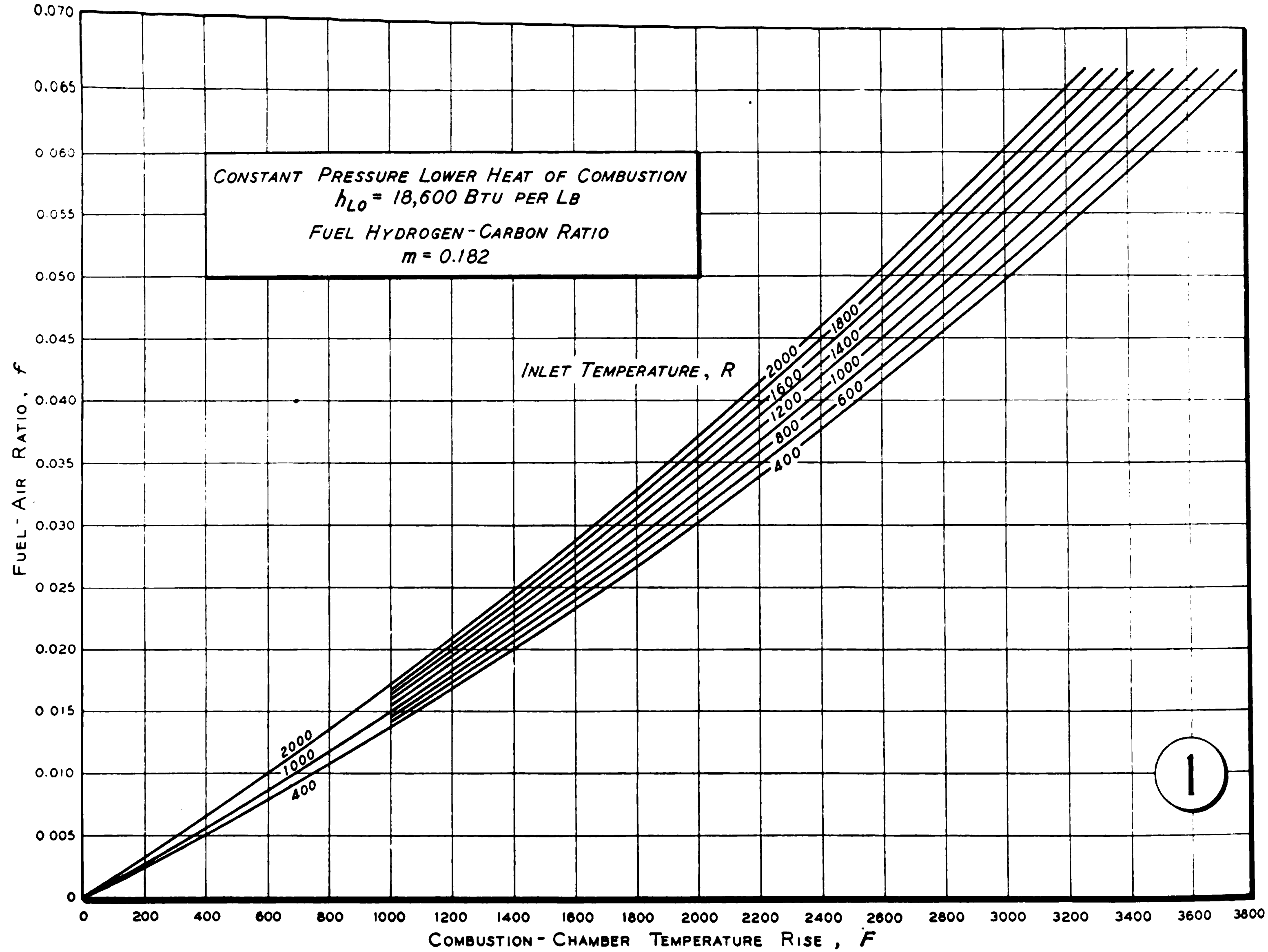


Figure 4 - Uncorrected fuel-air ratio for constant-pressure combustion of hydrogen fuels

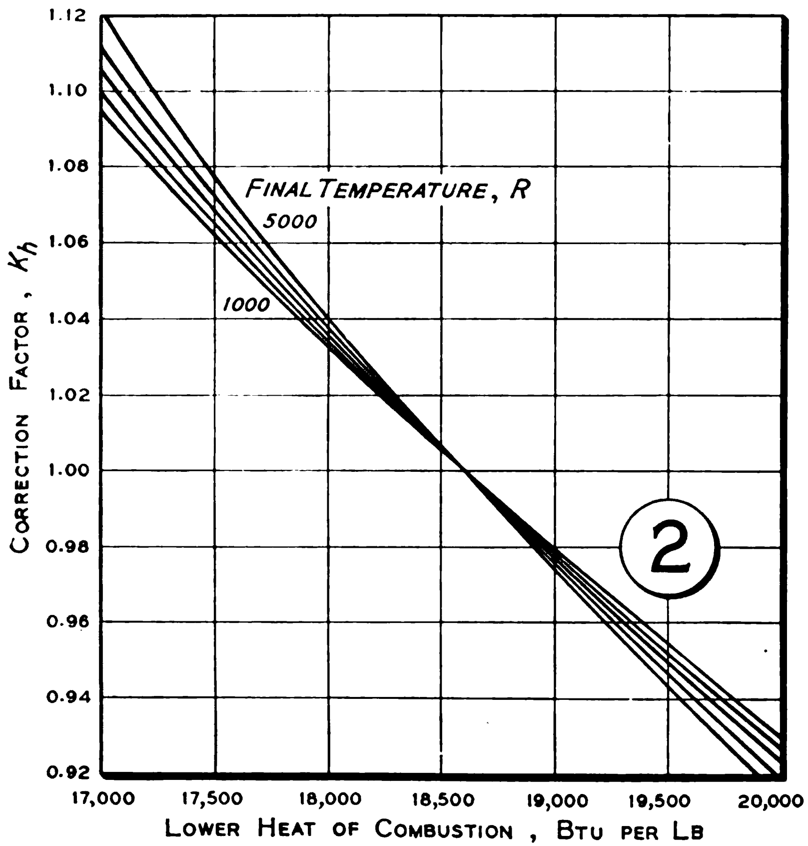


Figure 6 - Fuel-air ratio correction factor for heat of combustion

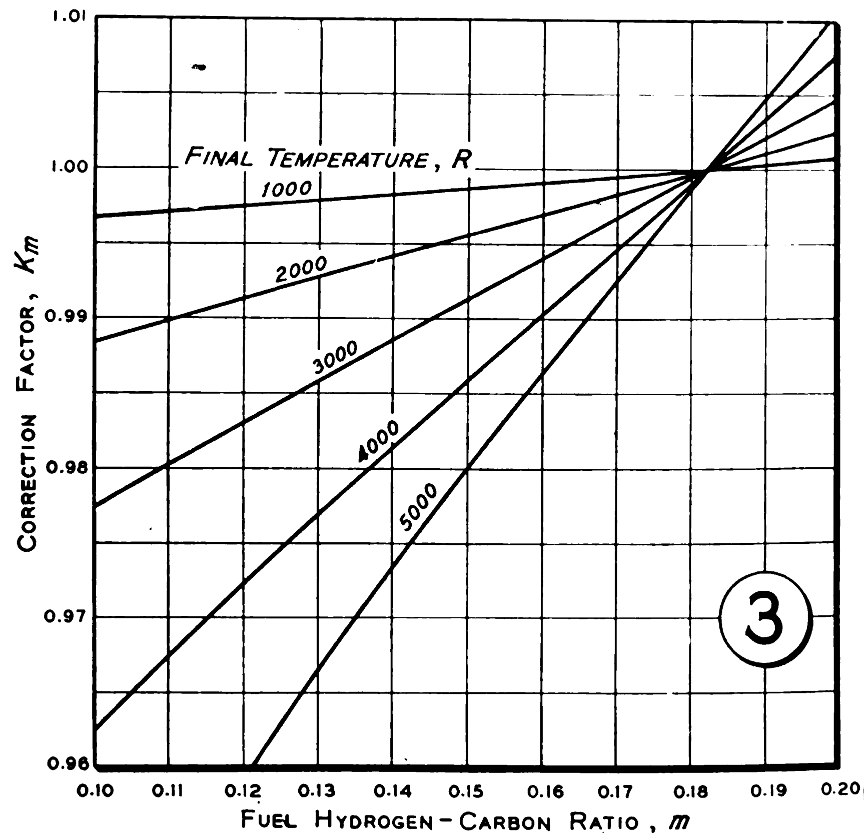


Figure 5 - Fuel-air ratio correction factor for fuel hydrogen-carbon ratio

Total combustion gases to be handled by the combustor = 305.4 + 4.62 = 310.02 #/min.

Next step would be to find the equivalent flow expressed in cfm.

(a) Quantity of air required for complete combustion: Fuel to be used is kerosene with 14.7% hydrogen and 85.3% carbon. Noting that the atomic weight of carbon is 12 and of hydrogen 1, we obtain from one pound of fuel:

$$\frac{.853 \times 44}{12} = 3.128 \text{ lb } CO_2$$

and $$\frac{.147 \times 18}{2} = 1.322 \text{ lb } H_2O$$

The oxygen of atomic weight 16 consumed is

$$\frac{3.128 \times 32}{44} + \frac{1.322 \times 16}{18} = 3.45 \text{ lbs.}$$

With 77% by weight of nitrogen and 23% oxygen in the air, the weight of nitrogen in the combustion gas per pound of fuel is

$$3.45 \times \frac{.77}{.23} = 11.55$$

and the weight of air consumed per pound of fuel is $3.45 + 11.55 = 15$ lbs.

(b) Molecular weight of combustion gas (M_g)

$$M_g = \frac{16}{\frac{3.128}{44} + \frac{1.322}{18} + \frac{11.55}{28}} = \frac{16}{.071 + .0735 + .4125}$$

$= 28.7$ (molecular weight of mixture without excess air)

Excess air = 66.2 - 15 = 51.2 lbs of air

$$\therefore M_g = \frac{67.2}{\frac{15}{[illegible].7} + \frac{[illegible]}{28.95}} = \frac{67.2}{.[illegible]7 + [illegible]}$$

$$= [illegible]$$

$$\therefore V = \frac{WRT}{P}$$

where R = gas constant

$$= \frac{1544}{[illegible]} = 33.4$$

$$T = 1600 + 460 = 2060°R$$

$$W = 3[illegible] \;\#/min$$

$$P = 44.1 \text{ psia}$$

$$\therefore V = \frac{310 \times 33.4 \times 2060}{44.1 \times 144}$$

= [illegible] cubic feet per minute of combustion gas

MAJOR DIMENSIONS:

The major dimensions of the two can-type combustor designs of Thermo Projects Inc., [illegible] Corporation which are used in the DeLaval [illegible] at Trenton, New Jersey to fire [illegible] and that [illegible] figures are:

(1) For the [illegible],000 [illegible] combustor;

Outer shell length = [illegible]"

Outer shell diameter = [illegible]"

[illegible] section length = 1[illegible]"

(2) For the [illegible]000 [illegible] combustor:

Outer shell length = [illegible]"

Outer shell diameter = [illegible]"

[illegible] section length = 1[illegible]"

The combustor under development will be handling 1760 cfm which is quite similar to the 2000 cfm combustor given above. Since the two combustors designed by Thermal Projects Inc. were found to be efficient, the major dimensions of the [illegible] for all three dimensions are [illegible] will [illegible] the 2000 cfm combustor. The outer shell will [illegible] 10" [illegible] diameter and [illegible] to give the secondary air a better passage thru the orifice holes along the flame tube liner. The diameter of the flame tube will be made 1½" all throughout the length to give an annular clearance of 1½" at the [illegible] end and ½" at the [illegible].

In the design of the passages for the primary air the can-type combustor designed by AiResearch Manufacturing Company (Reference 1 Paper 52-SA-31) is used as a model. Primary air requirement in this case would be around 18 pounds of air per pound of fuel (stoichiometric quantity of 15:1 + 20% excess air). At design condition of operation primary air requirement is about 29% ([illegible]/(62.1)) of the total air delivered by the compressor. In order to provide for the passage of the primary air, 80-1 inch diameter orifices are made as shown in figure 7. The total area of the 80 - 1 inch orifices is roughly 33% of the annular passage of the secondary air. In addition to the circular orifices, 6 swirl chambers or swirl slots are incorporated in the primary-air passage design. The passage area of the swirl chambers is approximately 4.5 square inches. Therefore, the total passage area of the primary

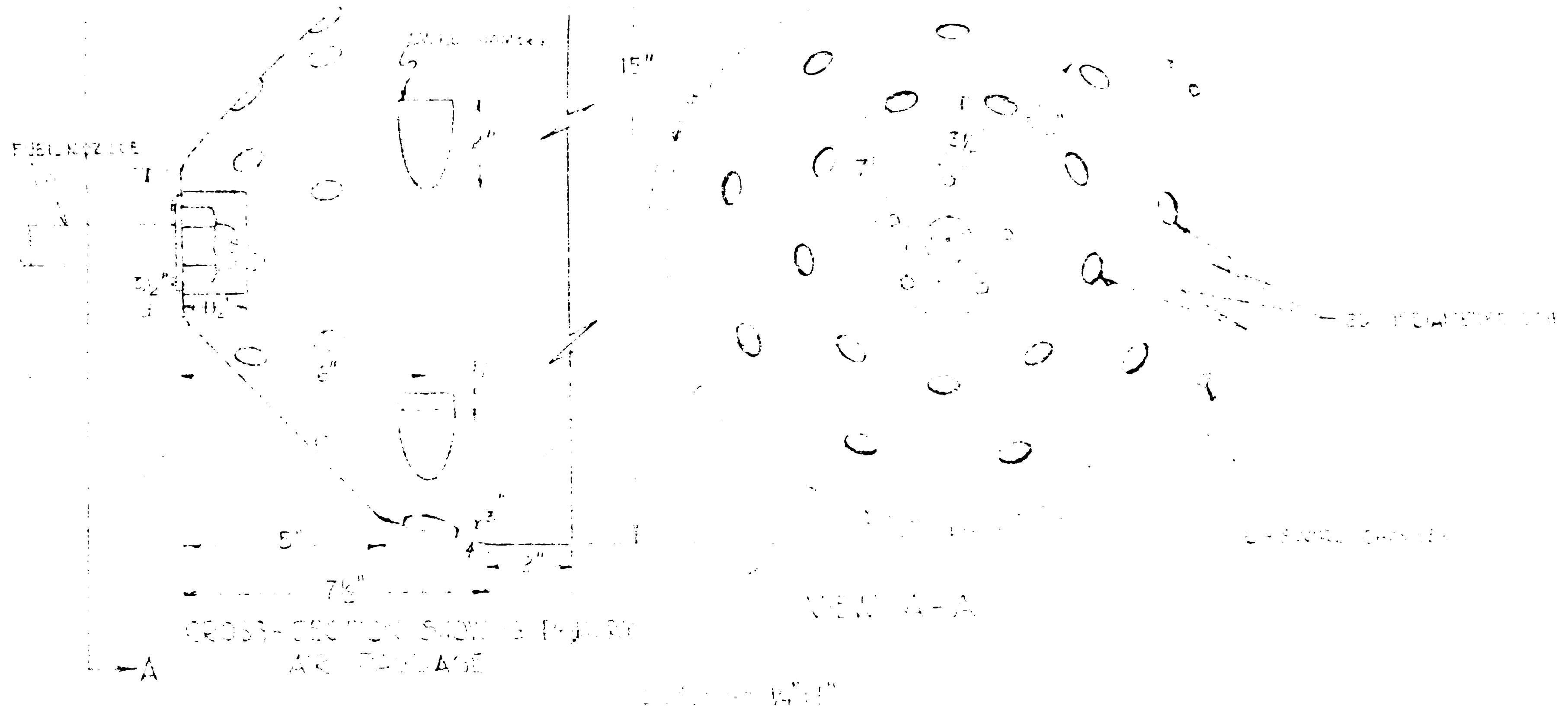


Figure 7 – Primary-air passage of combustion chamber with details of slots and orifices

air is 18.2 sq. in. or 27.4% of secondary-air annular passage. The function of the swirl slots is to give the primary air some turbulence so a good combustible mixture will result. For details of the swirl chamber or swirl slots, see Figure 7.

The secondary air shall be divided into three parts. The first portion shall be admitted into the "heating reaction zone" by 56 circumferential holes of 1/2 inch in diameter and are longitudinally spaced 1 1/2 inch apart. This portion shall extend from a circumferential line 6" from the nozzle end and up to 9" downstream. The second portion of the secondary air shall be admitted to the "complete combustion zone" thru 40-one inch diameter circumferential holes. The longitudinal spacing between holes shall be 1½ inches and shall extend 9" along the length of the tube. This would mean that there will be 8 rows with 5 holes to a row. The same is true with the reaction zone orifices where there shall be eight rows and shall have 7 holes to a row which makes a total of 56 holes.

The third portion shall be admitted to the "mixing zone" thru 40-3/4 inch diameter holes longitudinally spaced 1½ inches between holes and extends 9 inches of the tube length. Refer to Figure 8 for details of the secondary air orifices.

The total hole area or orifices area of the flame tube shall be 78.58 sq. in. while the cross sectional area of the flame tube shall be $.7854 \times (15)^2 = 176.6$ sq. in. This makes a ratio of tube cross-sectional area (A_1) to total

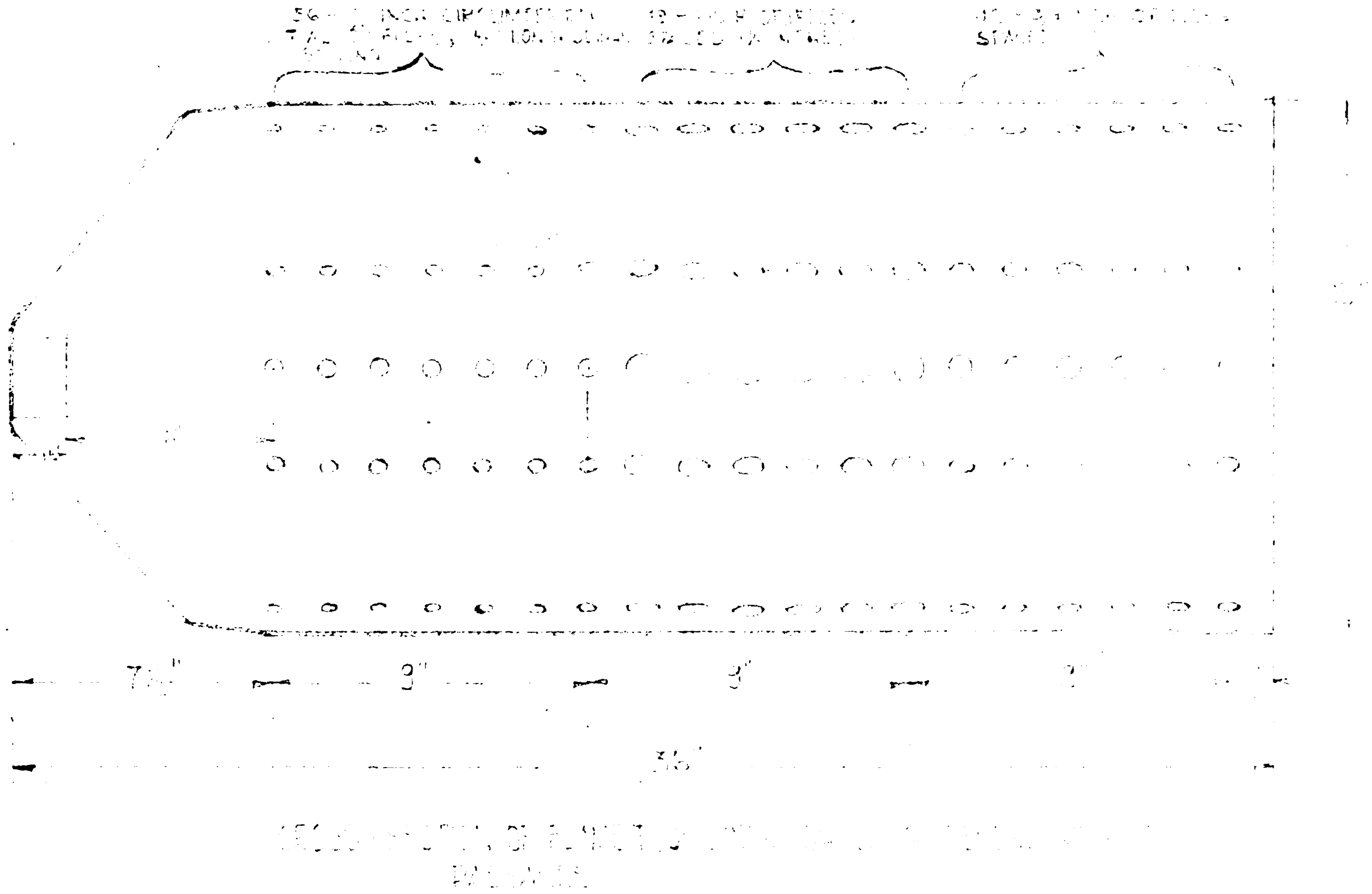


FIGURE 8 - SECONDARY-AIR PASSAGE OF COMBUSTION CHAMBER WITH DETAILS OF ORIFICES

orifice area (A_o) equal to = $\frac{176.3}{78.38}$ = 2.25. To reduce the pressure loss the area ratio should be decreased up to a minimum ratio of 1.4, where further decrease of the ratio would decrease the coefficient of discharge thus limit the gain in the reduction of pressure loss. This is clearly visualized with the aid of the formula developed by C. B. Stone of AiResearch Company

$$\frac{\Delta P_f}{P_{1st}} = \frac{k}{2} \times M_1^2 \times \left(\frac{A_1}{CA_o}\right)^2$$

where ΔP_f = Pressure loss due to friction

P_{1st} = Combustor inlet static pressure

k = 1.4

M_1 = Mach number of air at combustor inlet

A_1 = Cross sectional area of flame tube

A_o = Total orifice area

C = Coefficient of discharge

From the above formula the pressure loss due to friction and turbulence across the combustor expressed as a percentage pressure loss of the combustor inlet static pressure is found as follows:

Mach number, $M_1 = \frac{u}{\sqrt{kgRT}}$

where u = velocity of gases at combustor inlet = $\frac{CFS}{A_1}$

but $CFS = \frac{304.5 \times 53.35 \times 756}{44.1 \times 144 \times 60}$ = 32.2 cfs

$A_1 = \frac{176.3}{144} = 1.227$

$$\therefore u = \frac{32.2}{1.227} = 26.25 \text{ fps}$$

$$\therefore M_1 = \frac{26.25}{\sqrt{1.4 \times 32.15 \times 53.35 \times 755}} = \frac{26.25}{1340}$$

$$= .0195$$

Coefficient of discharge according to C. S. Stone for combustors of the size and type under consideration varies from .6 to .7 for area ratios greater than 1.5.

using C = .6

$$\therefore \frac{\Delta P_t}{P_{1st}} = \frac{1.4}{2} \times (.0195)^2 \times \left(\frac{176.8}{.6\,(76.08)}\right)^2$$

$$= .00400 \text{ or } .4\%$$

The results obtained herein are similar to those obtained by AiResearch from their can-type combustors which are quite within the allowable pressure loss due to friction and turbulence and will have negligible effect on performance of the combustor.

The second major component of pressure loss is due to momentum change. For a constant cross-section combustor the momentum pressure loss is a function only of the inlet Mach number and the temperature rise ratio across the combustor. From the relation derived by C. S. Stone the momentum pressure loss can be calculated as follows:

$$\frac{\Delta P_t}{P_{1st}} = \frac{k}{2} M_1^2 \left(\frac{T_2}{T_1} - 1\right)$$

$$= \frac{1.4}{2} (.0195)^2 \left(\frac{2060}{755.3} - 1\right)$$

$$= .000458 \text{ or } .0458\%$$

The momentum pressure loss is also small and will have negligible effect on efficiency. The total pressure loss would then be equal to

= .400 + .0[illegible] = .44[illegible] (expressed in % of combustor inlet pressure)

Total pressure loss across combustor is less than average obtained by [illegible] combustor of similar design and type.

<u>Fuel Nozzle</u>:

Kerosene is selected as the fuel for this particular combustor since it is the [illegible] in industry. It is low in [illegible] than [illegible] carbon formation. Carbon formation within the [illegible] is extremely [illegible] for it affects the performance [illegible] of the turbine assembly. Kerosene's [illegible] volatility is desirable from the standpoint of [illegible] vapor losses and trouble from vapor lock in the fuel lines. [illegible] kerosene is lighter than fuel oil the pressure required to [illegible] it is less, thus avoiding high pump [illegible]. [illegible] the fuel [illegible] in the [illegible] [illegible] negligible.

Oil pressure [illegible] fuel [illegible] Therefore, [illegible] nozzle will [illegible] in the combustor under [illegible]. [illegible] "Fuels and Combustion" for [illegible] minimum oil pressure of [illegible] psi is required for a fully [illegible]. [illegible] of kerosene, [illegible] oil pressure of [illegible] psi will be sufficient to get a good atomization of fuel [illegible] combustion chamber and will allow for variation in flow of fuel from pump for regulation. Taking the design condition of operation [illegible] for [illegible] fuel flow [illegible] in [illegible] the fuel nozzle, [illegible]:

Fuel Flow = [illegible] = [illegible] #/min.

or Fuel Flow = [illegible] x 60 = [illegible] #/hour

From Table I in [illegible] pounds per gallon of [illegible] = [illegible]

∴ Fuel Flow = [illegible] = [illegible] gph

For this particular purpose the spray angle of the spray will be [illegible] 60° which [illegible] the [illegible]. [illegible] spray angle is important in the [illegible] of the igniter. [illegible] will [illegible] of the fuel on the flame tube [illegible] carbon formation.

In summary nozzle specifications are:

Type = [illegible] Oil-Pressure Atomizing Nozzle

Capacity = [illegible] gph or [illegible]

Operating Pressure = [illegible] psi [illegible]

Spray Angle = 60°

The nozzle will be located on the center of the tube as shown in Figure [illegible]. Installation should be rigid and well sealed.

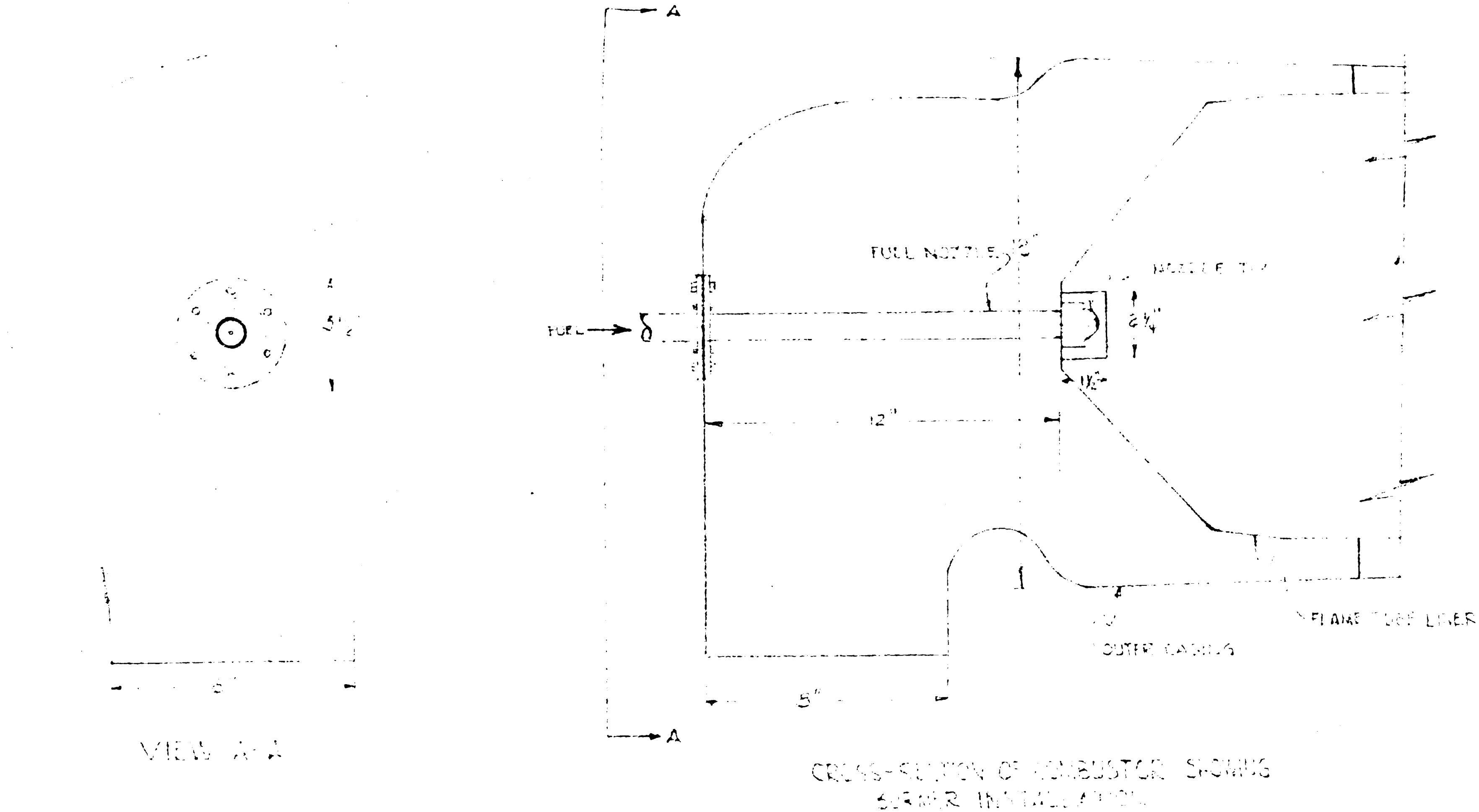


FIGURE 9 - LOCATION AND DETAIL OF FUEL NOZZLE

IGNITER:

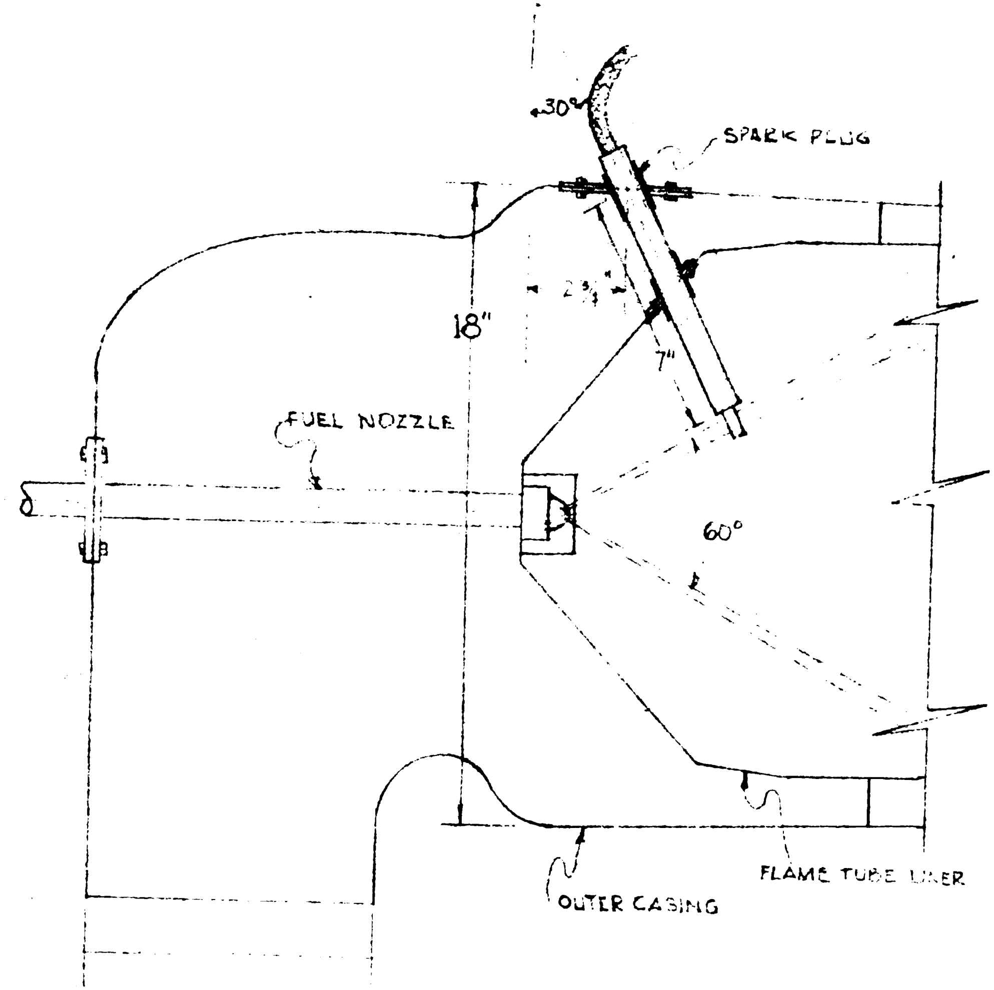
For the kind of service under consideration, tests conducted by [illegible] have shown that an ordinary C-10-[illegible] type of aircraft reciprocating-engine spark plug with an increased gap setting has been found adequate for a reasonable range of operating conditions. Since kerosene is used and operation is for short duration only, there will be no problem of plug fouling. The starting ability, when the fuel used is kerosene and when using around 15 psi oil pressure, is quite good with the C-10-[illegible] type spark plug.

As observed in a series of experiments by Frank [illegible] of Bendix Aviation Corporation, favorable ignition conditions can be obtained when the spray cone was slightly outside of the spark gap. See figure 10 for location of ignitor. Installation of the ignitor should be made rigid.

COMBUSTION MATERIAL:

Although mechanical stresses in the flame tube are nominal, the yield strength of the material may be exceeded when high metal temperatures are in combination with the mechanical stresses. Oxidation and carburization due to highly oxidizing atmosphere in the inner surface of the flame tube plus possible corrosion effects of [illegible] and lead from fuel can lead to early formation of cracks in the flame tube.

There are several alloys which have been used successfully as flame-tube material. Notable among them are



CROSS-SECTION OF COMBUSTOR SHOWING
IGNITER INSTALLALLATION

FIGURE 10 – LOCATION AND DETAIL OF IGNITOR

sheet stock of the very high temperature alloys such as type 310 stabilized stainless steel, Haynes Stellite Alloy N-155, Inconel and Inconel-X. For high temperature service a Haynes Stellite alloy if available will be used. If H.S. #27 is used, which has the lowest strength value in the Haynes alloy group, at 1500°F the ultimate tensile strength is 41,400 psi (Taken from ASME Transactions, Vol. [illegible])

Assume factor of safety = 4

Pressure inside liner = 60 psi (equivalent to a pressure ratio of over 4)

Since circumferential stress is more than the longitudinal stress, thickness of liner will be calculated from circumferential stress formula.

$$S_1 = \frac{Pd}{2t}$$

where S_1 = circumferential stress (allowable)

$$= \frac{41400}{4} = 10,350 \text{ psi}$$

$P = 60$ psi

d = liner diameter = 15"

t = thickness of liner in inches

$$\therefore t = \frac{60 \times 15}{10,350 \times 2} = .0435 \text{ inches}$$

$\therefore$ use sheet of $\frac{1"}{16}$ thick.

For the outer casing of the combustor (has cooler surface than inner liner) a 18/8 stainless steel of 1/16" thick will be quite sufficient to withstand the stresses.

The outer casing shall have two corrugations with dimensions as given in figure 11. This corrugation shall take care of any metal expansions. The inner liner shall have a loose fit to the (downstream) end with the outer casing so it will be free to expand. The inner liner shall be suspended in the middle portion from the outer casing.

REMARKS:

A good alternative to provide for an efficient combustor for the turbo-supercharger would be to accept the offer of Thermo Projects Inc., Todd Shipyards Corporation to build one combustor which will conform to the requirements of the converted gas-turbine engine. This will cost the university approximately forty-five hundred dollars ($4500.00). The experience of Todd Shipyards Corporation in manufacturing combustors for firing gas turbines and for testing turbo-superchargers of the type the school has, is enough guarantee that it will get an efficient dependable combustor.

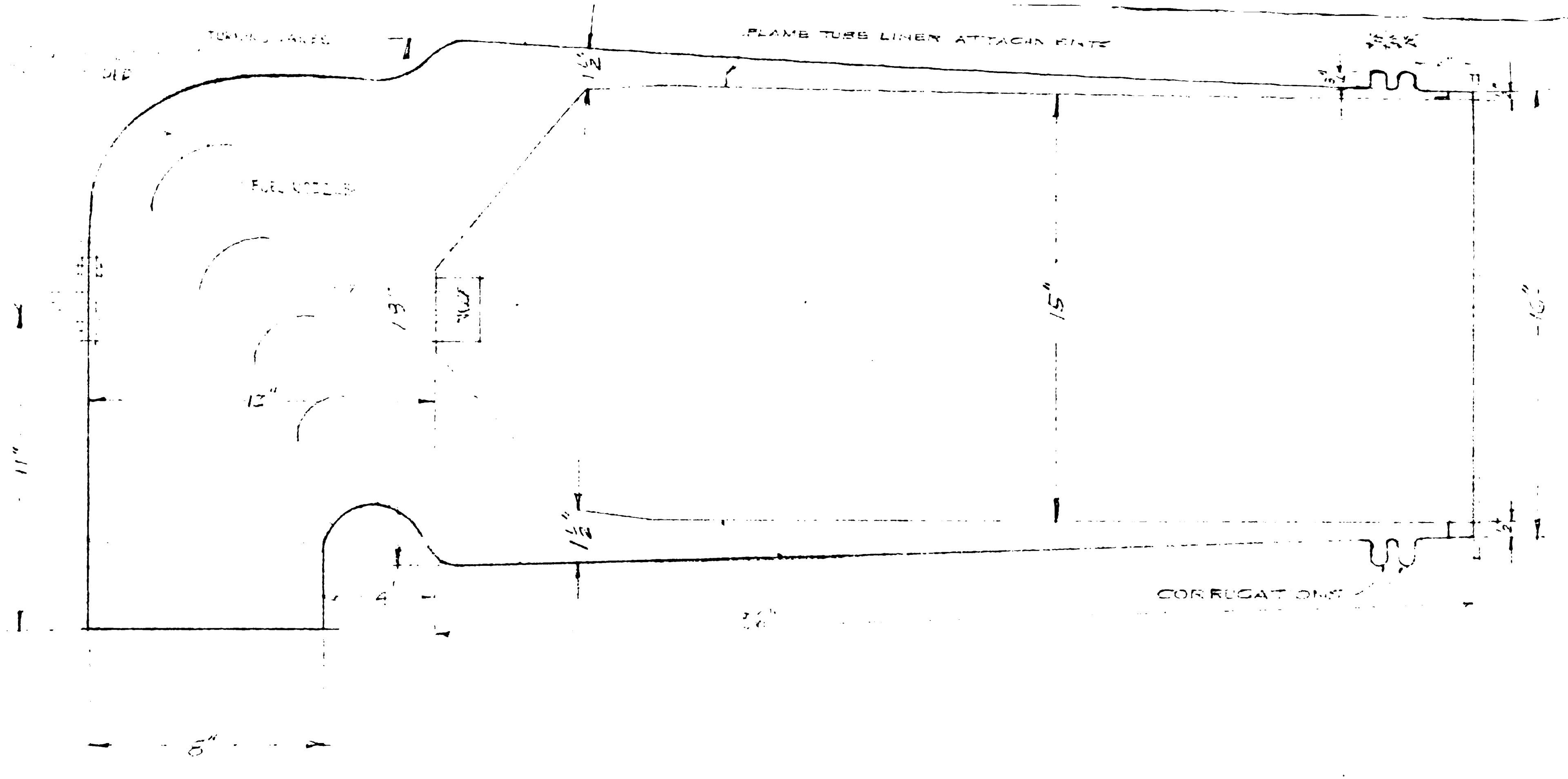


FIGURE 11 — DETAILS OF COMBUSTION CHAMBER'S OUTER CASING

INTAKE AND EXHAUST SILENCER DESIGN

Intake Silencer:

Following proposed design of a mine silencer company a silencer at the intake of the converted gas-turbine engine which is going to handle [illegible] lbs. per minute of air at 70°F and 14.7 psia will have specifications as shown in Figure 12.

As an alternative to making the silencer at the engineering shops, a mine silencer company offered to build one at a cost of two-hundred thirty dollars ($230.00), with a possible 25% discount.

Exhaust Silencer:

Following proposed design of a mine silencer company a silencer at the exhaust of the converted gas-turbine engine, which is going to handle approximately 310 lbs per minute of hot gases at slightly more than atmospheric pressure and at a temperature of 710°F, will have specifications as shown in the detailed drawing (Figure 13).

Exhaust temperature is found in the following manner:

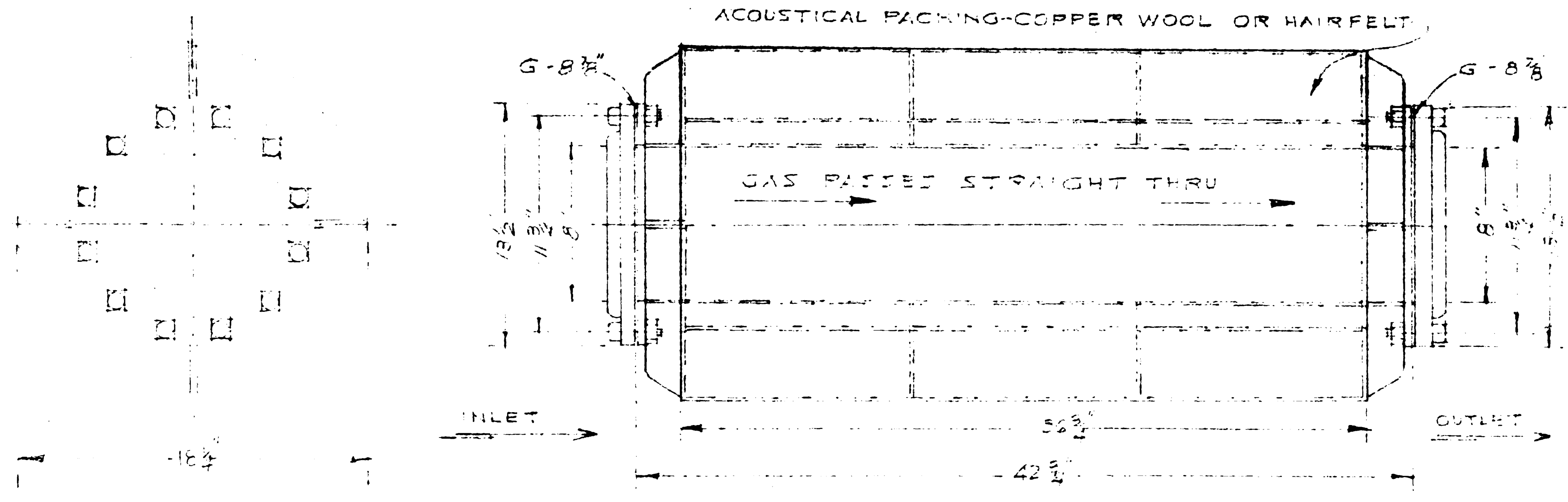
$$\frac{T_1}{T_2} = (P.R.)^{\frac{k-1}{1}}$$

where $T_1 = 1600 + 460 = 2060°R$

T_2 = Exhaust temperature

P.R. = Expansion ratio = 3

k = 1.324 (average value between 800°F and 1600°F)



ALL FLANGE DIMENSIONS ± 1/16"   ALL OTHER DIMENSIONS ± ¼" UNLESS OTHERWISE SPECIFIED

NOTE:

MATERIAL - STD PLATE STEEL - WELDED
125 LB. STD. COMPANION FLANGES.

(DESIGN TAKEN FROM THE MAXIM SILENCER SPECIFICATIONS).

Figure 12 - Intake silencer details

$$\therefore = \frac{T_1}{T_2} = (3)^{\frac{1.324-1}{1.324}} = (3)^{.245} = 1.30[illegible]$$

$$\therefore T_2 = \frac{2060}{1.30[illegible]} = 1575°R \text{ (uncorrected for turbine efficiency)}$$

η_T = .68 (Taken from turbine performance curves)

$$\eta_T = \frac{2060 - T_2}{2060 - T'_2} = .68 = \frac{2060-1575}{2060-T'_2} = \frac{485}{2060-T'_2}$$

$$\therefore T'_2 = 2060 - \frac{485}{.68} = 2060 - 714$$

= 1[illegible]°R or 7[illegible]° F (corrected for turbine efficiency)

Therefore exhaust silencer material should withstand temperature of about 7[illegible]°F.

If difficulty is encountered in the procurement of the material used in local manufacture to the engineering shops, a good alternative would be to accept the offer of Maxim Silencer Company to furnish said exhaust silencer at six-hundred forty-five dollars ($645.00) with a possible 25% discount. For details of exhaust silencer, refer to figure 13.

<u>Special Exhaust Fitting</u>:

The setting up of the exhaust silencer is made difficult by the installation of a dynamometer which has to be coupled to a shaft projecting from the turbine disk. In order to accommodate both, a 90° elbow shall be installed. The 90° elbow will be fabricated from a 18/8 stainless steel sheet of 1/8" thick. For dimensions and specifications, see figure 17. A hole will be drilled through the sides of the

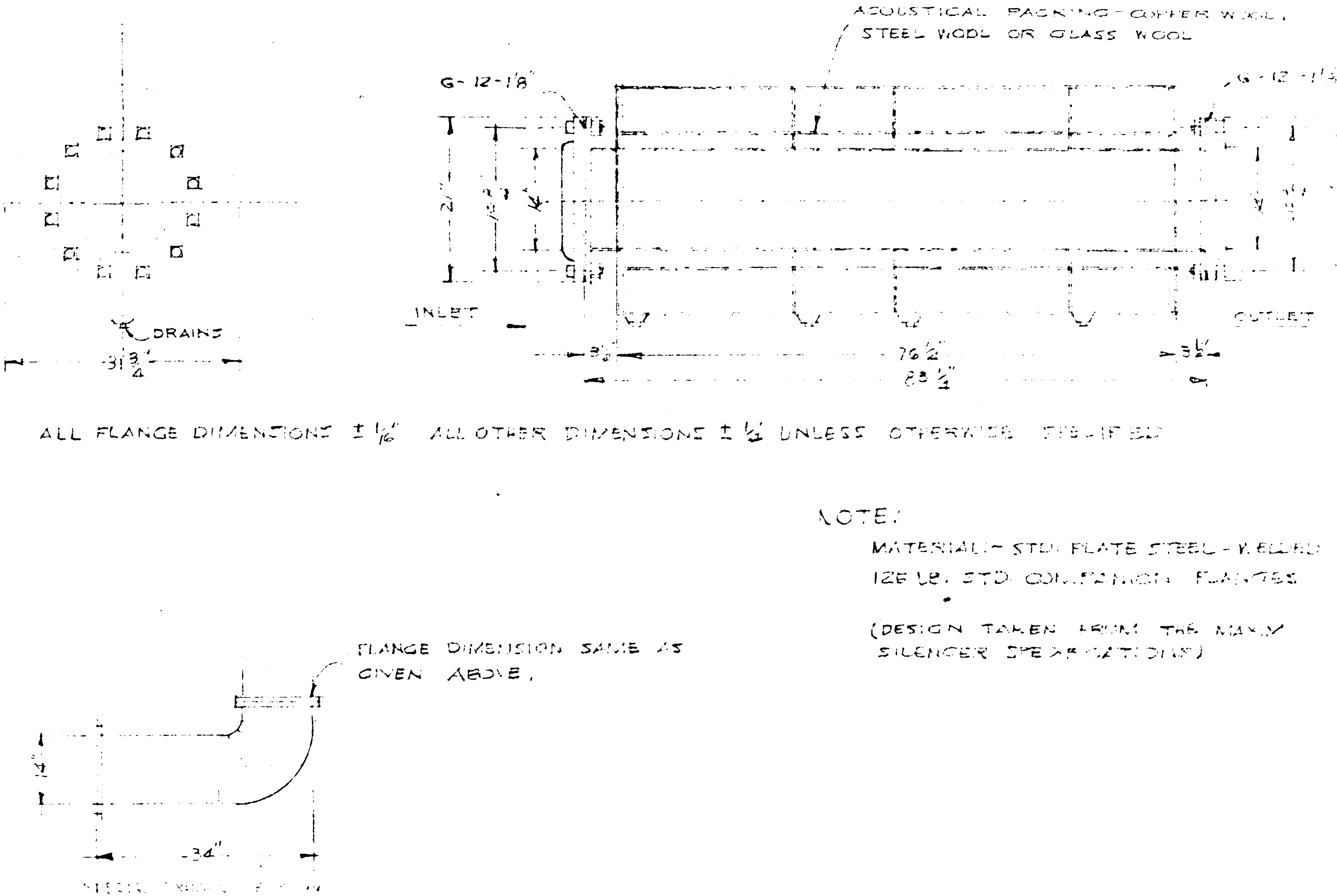


FIGURE 13 - EXHAUST SILENCER AND SPECIAL EXHAUST ELBOW DETAILS

elbow to give way to a flexible coupling which will be installed to couple the short turbine shaft and the dynamometer. Details of this installation are taken up under instrumentation. High temperature-resistant gaskets shall be installed between flanges.

Pages 46 - 49 inadvertently missed

INSTALLATION AND SET-UP

Turbo-Supercharger Mounting Frame:

The first item to be considered in the installation of the complete engine would be the design of the stand to hold the compressor-turbine component. The turbo-supercharger or the compressor-turbine component weighs 144 lbs. The piping which conducts the compressed air from the compressor outlet to the combustor will have negligible weight. There will be slight vibration due to the rotating parts but a stand made of a [illegible] structural channel, which has a depth of 4 inches, .247" web thickness, and 1.647" flange width will be able to support and make the installation rigid. In addition, General Electric advises that the turbo-supercharger should be isolated from its rigid mounting frame by means of 1/4" thick rubber pads. These pads 2" by [illegible]" should be inserted between the compressor mounting lugs and the mounting frame. Precaution should be taken in bolting the turbo-supercharger down to the mounting frame to prevent compressor distortion and excessive compression of the rubber pads. Details of the mounting frame or stand is shown in figure 14.

Foundation:

The weight of the gas turbine is practically negligible when the determination of foundation dimension is concerned. Vibration problem in a turbine is not a serious problem. The use of rubber pads to isolate vibration brings

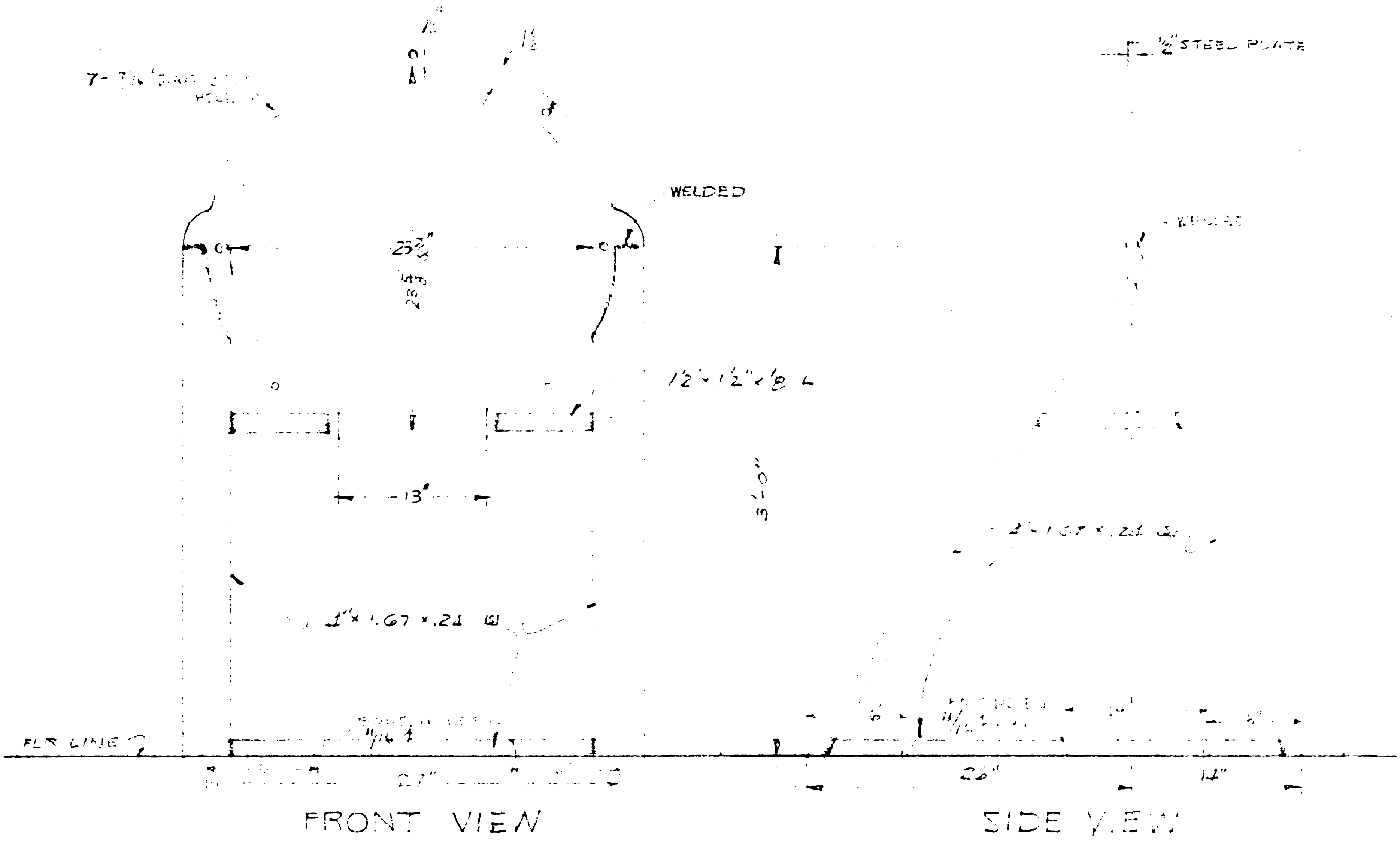


FIGURE 14 — TURBO-SUPERCHARGER MOUNTING FRAME DETAILS

to a minimum the vibration being transmitted to the frame and thence to the foundation. But for the sake of obtaining an approximate dimension of the foundation a very conservative estimate will be made.

As shown later in this paper the total horsepower developed by the turbine is 172hp.

Then from a formula given by Machinery Handbook, the equivalent rotating force acting tangent to the turbine wheel is:

$$F = \frac{5252 \text{ H.P.}}{n \times r}$$

where H.P. = 172 hp.

n = revolution per minute

= 24000

r = wheel radius in feet

$$= \frac{12.25}{12} = 1.02'$$

$$\therefore F = \frac{5252 \times 172.0}{24000 \times 1.02} = 36.9 \text{ lbs.}$$

According to Morse, "Power Plant Engineering" a mass of weight (foundation) equal to 10 to 20 times the rotating force causing the machine to vibrate, should be adequate to dampen vibration.

$\therefore$ Weight of foundation = 20 x 36.90 = 738#

Concrete weight = 137 pounds per cubic foot (Machinery Handbook)

$$\text{Volume} = \frac{738}{137} = 5.39 \text{ cu. ft.}$$

If thickness = 6" or .5 feet (sufficient to bear weight of unit)

FIGURE 15 - TURBO-SUPERCHARGER FOUNDATION DETAILS

Surface area = [illegible] = 10.77 sq. ft.

∴ Dimension of foundation = 4' x [illegible]' x .5'

The rest of the floor area will be made of 4" thick reinforced concrete. Refer to figure 15 for details of foundation and for details of the anchor bolts.

Condenser Installation and Attachment:

[illegible] tendency of the [illegible] to expand [illegible] to support the [illegible] figure 16. [illegible] figure 16. [illegible] location of the [illegible] with regards to the [illegible]. The piping [illegible] 1/2" [illegible] steel [illegible]. [illegible] Details of [illegible] figure 17. [illegible] flange 1/16" [illegible]. A [illegible] lining stainless steel [illegible] joint, [illegible] at 1500-F [illegible] between the condenser [illegible] piping connection to the [illegible], [illegible]. All [illegible] piping connections shall be [illegible] connections.

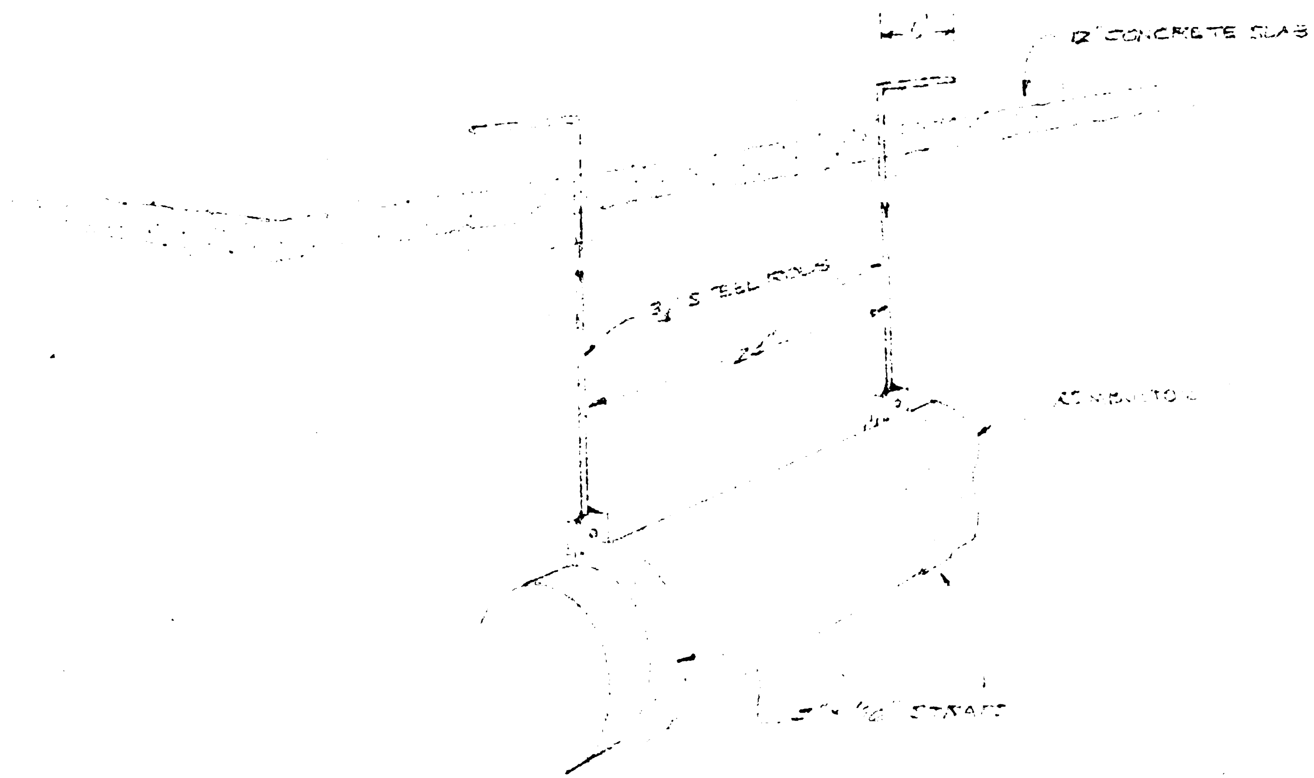


FIGURE 16 – COMBUSTOR HANGER ATTACHMENT DETAILS

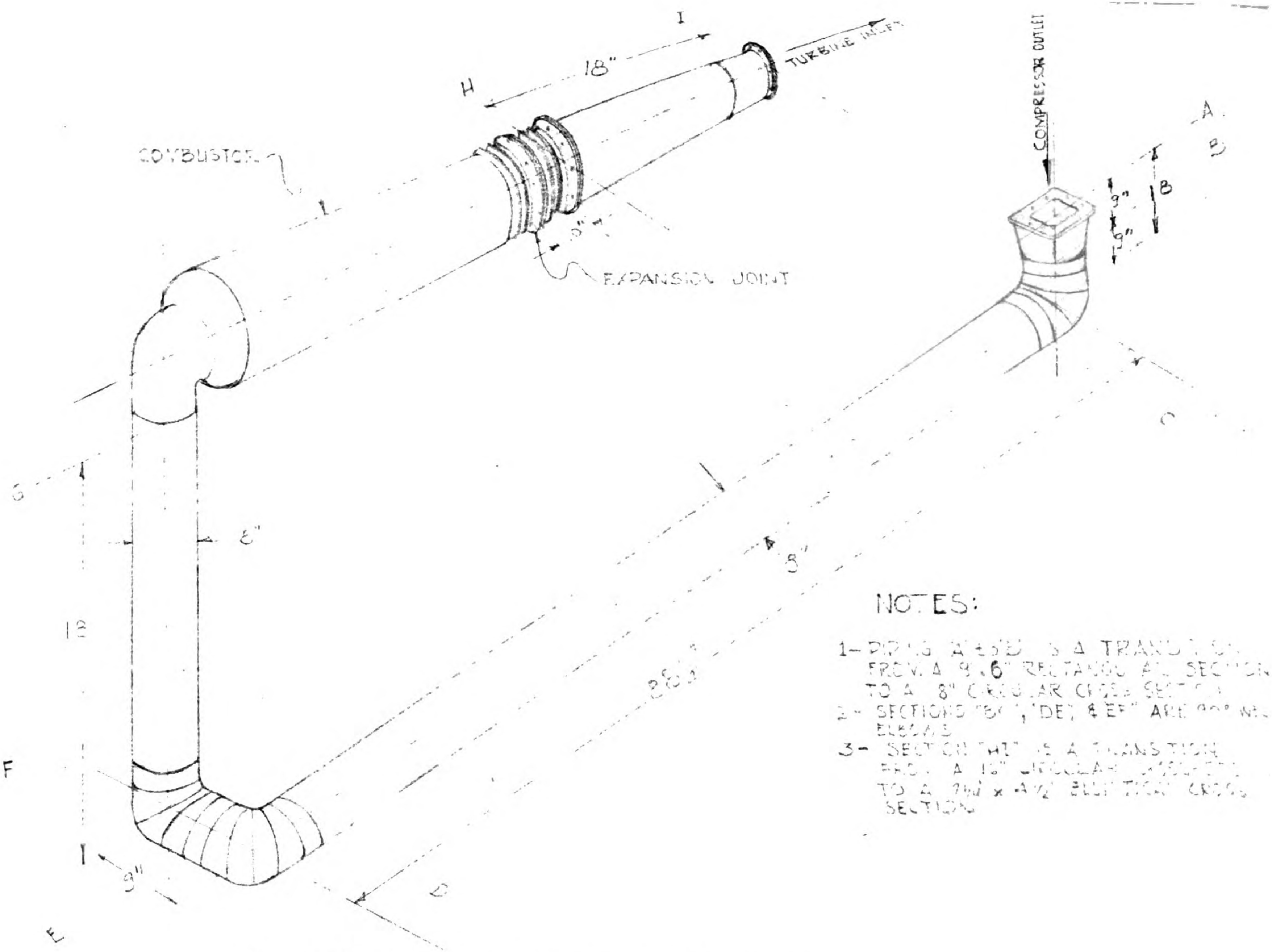


FIGURE 17 – DETAILS OF PIPING TO AND FROM COMBUSTOR

Intake and Exhaust Silencer Installation:

The intake silencer shall be installed [illegible] on top of the [illegible] as shown in Figure [illegible]. Details of [illegible] are shown in Figure [illegible].

The exhaust silencer shall be [illegible] vertically. [illegible]. [illegible] length may be installed [illegible] of the building. Pipe supports shall be made in such a manner as to allow for some movement of the silencer and connections. Detail of exhaust silencer [illegible] is shown in Figure 21.

Fuel System:

Pump:

A rotary pump directly coupled to a motor shall supply the fuel to the fuel nozzle under pressure. The total dynamic head the pump must operate against is found in the following manner:

$$H_T = \Delta H_{f_p} + Z_s + Z_d + H_p$$

where H_T = total dynamic or operating head

ΔH_{f_p} = Friction losses in pipe in ft-head of water

= 3 ft. (conservative estimate of losses due to friction in suction and [illegible] length is around 15 ft.)

Z_s = Static suction head

= 4 feet (approximate)

= -10 feet (approximate; minus sign means it is above pump level)

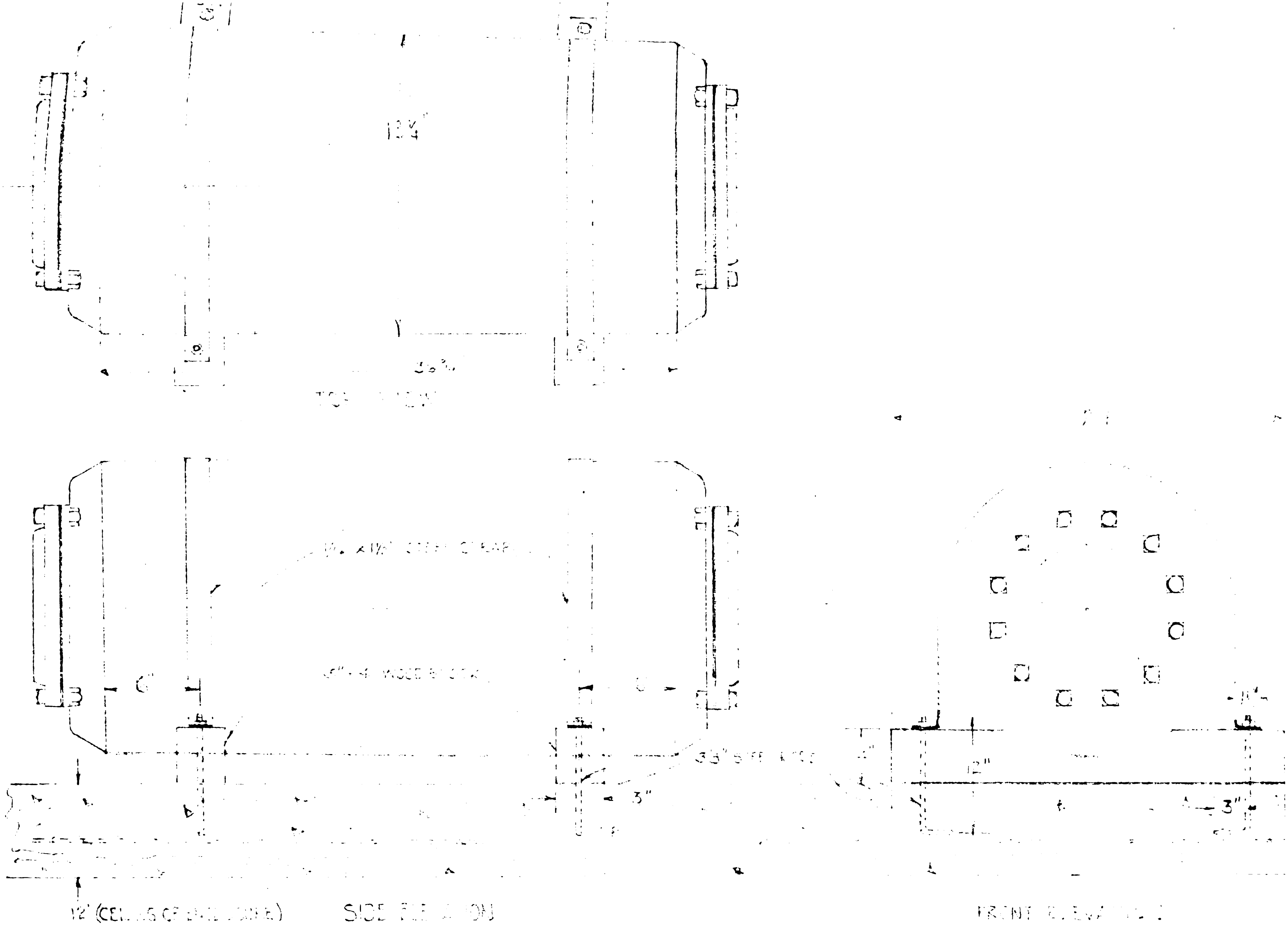


FIGURE [illegible] — [illegible] INSTALLATION [illegible]

H_p = pressure head at discharge end of pipe or at inlet of fuel nozzle which pressure is at 85 psi.

$$= \frac{85 \times 144}{.8 \times 62.4} = 246 \text{ feet}$$

Note: velocity head is neglected since it is very small.

Therefore H_T = 3-10 + 4 + 246

= 243 feet

Pump's rated capacity shall be based on the fuel requirement at optimum conditions which is equivalent to 41 gph, as computed earlier. To get a richer mixture the rpm of the motor and hence the pump should be increased.

Therefore, the fuel pump specification shall be:

Type: Rotary (gear or lobe)

Capacity: 41 gph

Operating head: 243 feet

Type of Drive: Motor Driven (directly coupled)

Liquid handled: Kerosene

Pump Motor:

The motor used shall have a constant speed and shall allow for a speed regulation through a field rheostat. A DC shunt, single phase motor may be used. Approximate drive horsepower is:

$$Hp = \frac{Gpm \times \gamma \times H_T}{7.48 \times 33000 \times \eta_p}$$

where Gpm = $\frac{31}{60}$ = .516 gpm

γ = specific weight of kerosene

= 49.9#/cu. ft.

H_T = operating head

$= 243$ feet

e_p = pump efficiency

$= .70$ (From Chemical Engineers Handbook - average value for gear pumps)

$$\therefore H_p = \frac{.115 \times 62.4 \times 243}{7.48 \times 33000 \times .7}$$

$= .$[illegible] hp

Therefore a fractional horsepower motor shall be used, rated at 1/8 hp.

<u>Fuel Lines</u>:

According to Gilley and Delany in "Fuel Oil and Steam Engineering" the velocity of light oils of moderate temperature (60° to 100° F) in fuel lines should not exceed 2 fps. Therefore, using fluid velocity of 2 fps, size of piping can be determined.

$Q = AV$

where Q = flow of fuel in cfs

A = cross-sectional area of pipe

V = velocity of fuel in fps

$= 2$ fps

but $Q = \dfrac{gps}{7.48} = \dfrac{41}{3600 \times 7.48} = .00152$ cfs

$\therefore A = \dfrac{Q}{V} = \dfrac{.00152}{2} = .00760$ sq. ft.

or $A = .1093$ sq. inches

$\therefore$ Pipe inside diameter $= \sqrt{\dfrac{.1093}{.7854}} = \sqrt{.1393}$

$= .373''$ use $3/8''$

For the suction piping use 1/2" diameter pipes. According to [illegible] Catalog, for use of this type, a standard 125 lb., black iron (wrought) pipe is usually used. Brass pipes of similar strength may also be used.

For sealing joints and connections use litharge or equivalent.

Consumption of fuel at optimum conditions is 41 gph. A 60-gallon service tank to store kerosene would be sufficient. Tank shall be placed at least 10 feet from floor level. Piping from tank to pump should run below floor level.

If anything goes wrong with the unit, the solenoid-operated shut-off valve is automatically closed. At times, however, when the switches fail the pump will keep on pumping fuel and with the solenoid shut-off valve closed, there is no place for the oil to go and excessive pressure builds up on the line. To prevent damage to the pump and equipment, a fuel return line is installed with a pressure relief valve placed on the line just right after the branch connection as shown in figure 1[illegible]. To prevent seepage of fuel through the return line and to give a few additional psi (10-20 psi) above the rated pressure in the nozzle for regulation purposes, the relief valve shall be set at 145 psi. The type and size of valve as taken from [illegible] Relief Valve Catalogue should be a 1/2 inch, screwed type valve, equivalent to [illegible] No. 403 pressure relief valves. For details of the fuel system, see figure 1[illegible].

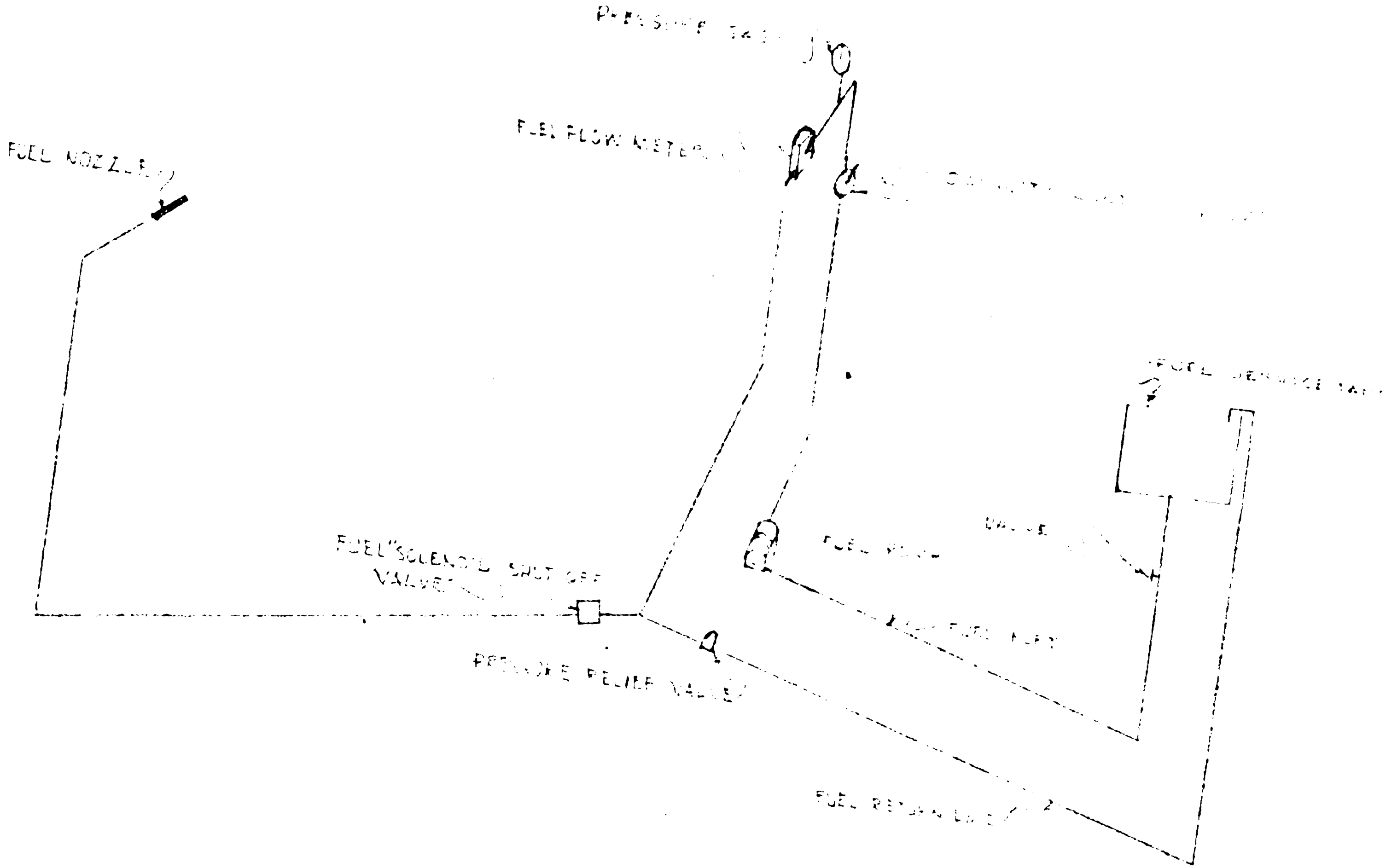


FIGURE 12 – SCHEMATIC OF FUEL SYSTEM

Ignition System:

As mentioned previously in this paper, a C-10-S type spark plug will be used for igniting the fuel mixture inside the combustor. The energy for the spark plug will be supplied by a 12,000-volt transformer which will be situated near the control panel. The selection of the 12,000-volt transformer is based on General Electric choice for a high-energy supply for a similar gas turbine converted from a turbo-supercharger.

Starting System:

A good way of starting the gas-turbine unit would be to direct a small air jet from the shop compressed air line on the exhaust side of the turbine wheel. This will serve to bring the unit up to around 3000 rpm, the approximate speed which Allis Chalmers' Engineers found to be the right point for a gas-turbine converted turbo-supercharger to ignite the fuel mixture in the combustor. When the combustion chamber is ignited the unit would then be able to accelerate up to a desired speed by controlling the kerosene fed to the burners. For details of the air nozzle attachments, see Figure 2[illegible].

Pipe size for compressed air is found in the following manner:

Compressor work is equal to:

$$W_c = .[illegible]\ C_p \times T_{[illegible]} \times \frac{1}{\eta_c}$$

Where $C_p = .[illegible]$

$T_{[illegible]} = 7[illegible].3$

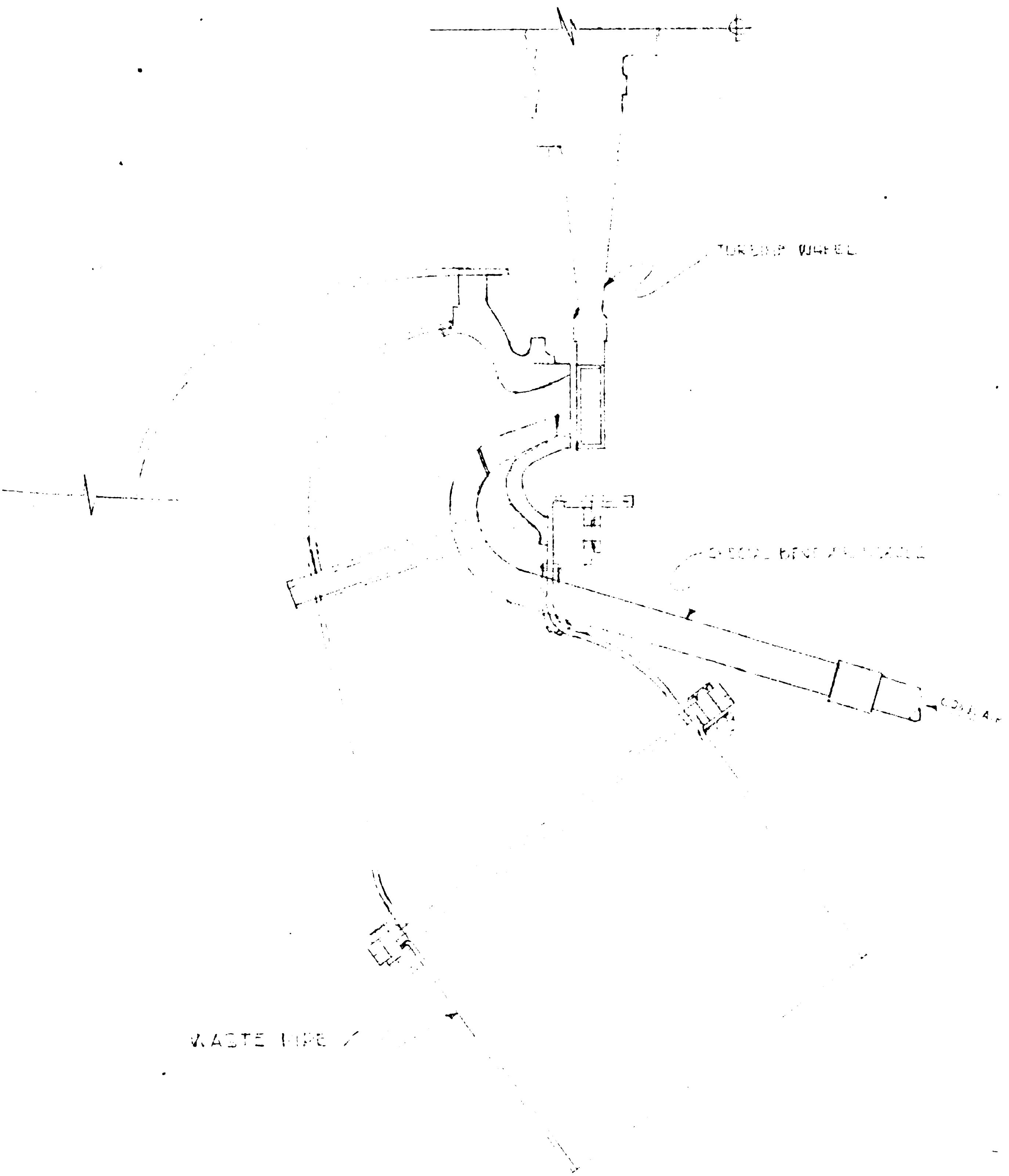


FIGURE 20 - STARTER NOZZLE ATTACHMENT DETAILS

η_v = [illegible] efficiency

= .[illegible]

$W_1 = \frac{.[illegible] \times .[illegible] \times [illegible].5}{.[illegible]}$

= 73.5 [illegible]

[illegible], i.e., [illegible] directly proportional to the [illegible], the new weight flow at 5000 rpm (at [illegible] of units) is equal to:

$$\frac{W_1}{W_2} = \frac{N_1}{N_2}$$

where W_1 = weight flow at 24000 rpm

W_1 = [illegible] lb/min

W_2 = weight flow at 5000 rpm

N_1 = 24000 rpm

N_2 = 5000 rpm

$\therefore W_2 = \frac{W_1 N_2}{N_1} = \frac{[illegible] \times 5000}{24000}$ =

= [illegible] lb/min

Therefore, the power required to operate the compressor at 5000 rpm is equal to:

$H.P._{(5000)} = \frac{[illegible] \times [illegible]}{[illegible]}$

where [illegible] = 1 hp

$\therefore H.P._{(5000)}$ = [illegible] hp [illegible] friction and wind [illegible] of [illegible], [illegible] overcome by [illegible]:

$$\ldots\ H(\text{in}) = [illegible] + [illegible] \times [illegible]$$
$$= [illegible] \times [illegible]$$
$$= 7[illegible].26\ [illegible]$$

Therefore, [illegible] pressure of 60 [illegible] following manner:

[illegible] adiabatic expansion [illegible] from 60 psig to 14.7 psia [illegible] .1[illegible] [illegible] of air per minute. Assuming [illegible] value [illegible] efficiency,

$$[illegible]/\text{CFM} = \frac{.1[illegible]}{.[illegible]} = .[illegible]$$

Therefore, [illegible] air [illegible] equals;

$$\text{CFM} = \frac{7[illegible].2[illegible]}{.1[illegible]} = [illegible]$$ ([illegible] of air [illegible] psig [illegible] [illegible] 3[illegible]0[illegible])

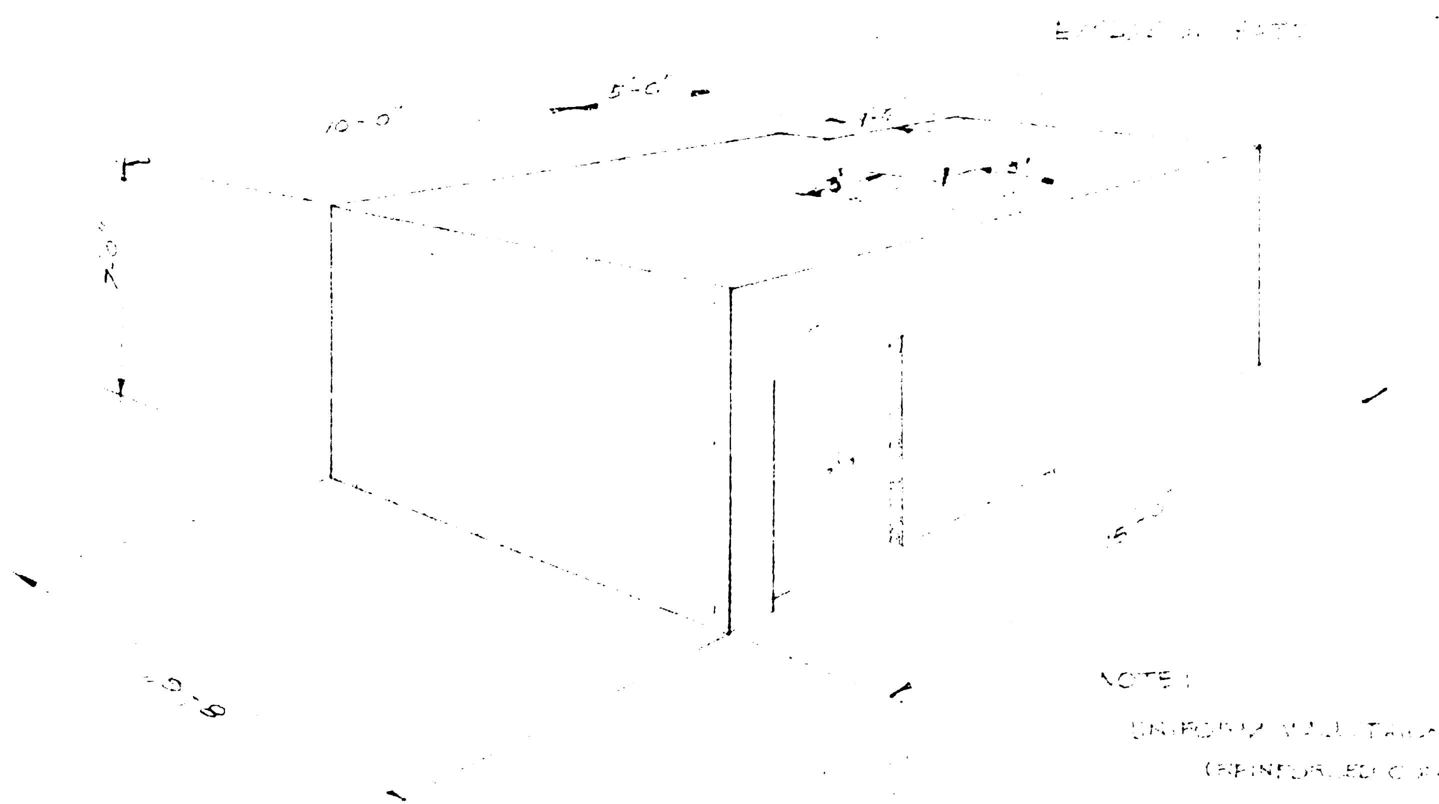
[illegible], [illegible] pipe to deliver [illegible] CFM of air [illegible] psig and [illegible] pressure loss of [illegible] for every 100 ft. of pipe is [illegible]". Therefore, if [illegible]" pipe is used and compressed air source is not more than [illegible] feet from air nozzle, pressure loss is negligible that [illegible] psig at the source will still enable [illegible] of air [illegible] 3[illegible]00 rpm at the [illegible]. If a smaller pipe is used [illegible] corresponding increase in pressure should be [illegible] to the source [illegible] losses.

Lubrication:

As recommended by General Electric, DTE-20 oil should be supplied to the built-in pump which supplies approximately 3 qt/min at 10 psig to the bearings. The oil inlet temperature should be between 70° and 110° F. The heat oil is returned to the supply tank by the scavenging element of this pump which has a capacity of 1.5 gpm.

Enclosure:

While the likelihood of a wheel failure is very remote, General Electric Company recommends that in cases where untrained personnel handle the equipment, sufficient protection be installed to stop a bursting wheel. In this respect a total enclosure shall be built around the unit as shown in Figure 21. General Electric recommends that a properly built reinforced concrete wall of 12 inches thick will give sufficient protection. An explosion hatch shall also be provided on top of the concrete enclosure as shown in Figure 21-A.



NOTE:

UNIFORM WALL THICKNESS = 12"

(REINFORCED CONCRETE)

FIGURE 21 A – DIMENSION DETAILS OF CONCRETE BLOCK ENCLOSURE

CONTROL AND INSTRUMENTATION

Temperature Limit Control:

Following the recommendation of General Electric, (manufacturer of the turbo-supercharger) the nozzle box inlet gas temperature should not at any time exceed 1600°F. For safety against excessive temperatures, a thermoswitch shall be located near the nozzle box inlet on the combustion chamber transfer section with a setting to trip out fuel supply at 1600°F. The fuel shut-off valve shall be solenoid-operated and shall be located near the combustor.

A manual capacity control valve shall be installed in the panel board to shut-off fuel supply in case automatic control fails and a reading of 1600°F is registered in any of the thermocouples installed at the inlet to the nozzle box section. The capacity control valve functions also as feed regulator and varies kerosene fed to the combustor in order to obtain desired performance. The variable-speed motor driving the fuel pump, will, in combination with the capacity control valve widen and improve fuel regulation to the combustor.

From the relay box of the temperature sensing element a solenoid-operated switch shall shut off motor of fuel pump simultaneously with the fuel shut-off valve. In addition to this, a toggle switch (emergency stop) shall be installed to shut pump motor if operator finds automatic control fails and gas-turbine is not functioning properly. An alarm horn may be installed in the panel board in addition to the other control equipment to warn of over-heating or excessive temperatures. Shut-off valves "open-close" buttons may also be installed in panel to show operator if shut-off valve is closed or opened.

For sketch of temperature limits control equipment see figure 22.

Over-Speed Control:

According to manufacturers' specification and guarantee the maximum rpm of the turbo-supercharger is 24000 rpm. In order to prevent distortion and wheel failure the rotor speed should be limited to a maximum speed of 24,000 rpm. To accomplish this a speed indicating instrument or tachometer indicator furnished with contacts for automatic operation of the solenoid-operated fuel shut-off valve be installed at the other end of the dynamometer. This type of dynamometer (Taylor Dynamometer) has short shafts protruding from both ends. When the speed reaches 24000 rpm and solenoid-operated shut-off valve closes, operator should immediately shut off motor by operating the "emergency stop" switch. Simultaneously, the operator should open the valve in the waste-pipe assembly to release high energy gases and thus prevent turbine wheel from over-speeding. The control wheel for the waste pipe valve shall be located in the control panel and connections to the valve are made thru a tachometer cable and thru a series of pulleys. See figure 23 for detail of waste pipe assembly.

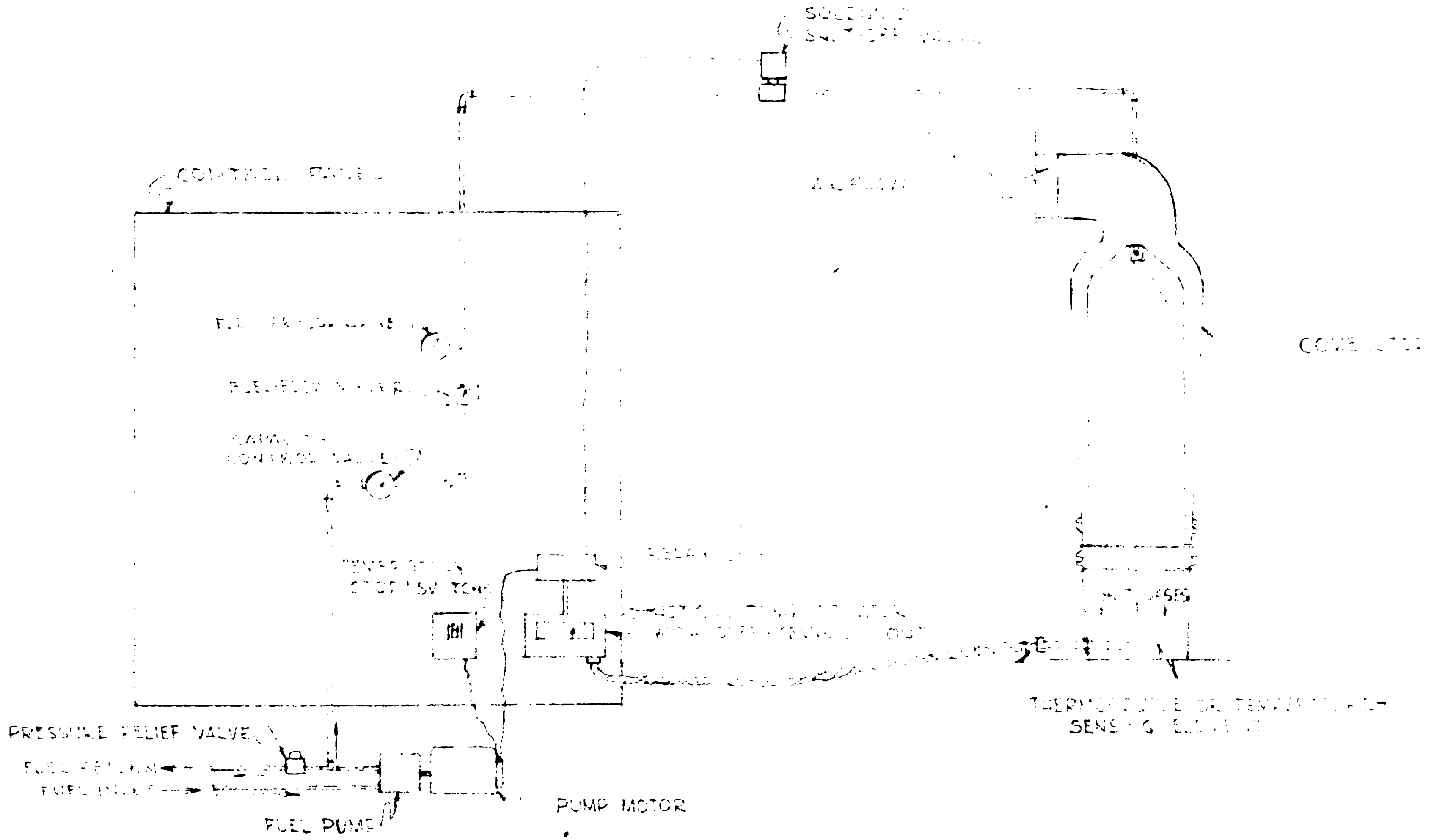


FIGURE 22 – LAYOUT OF TEMPERATURE LIMIT CONTROL

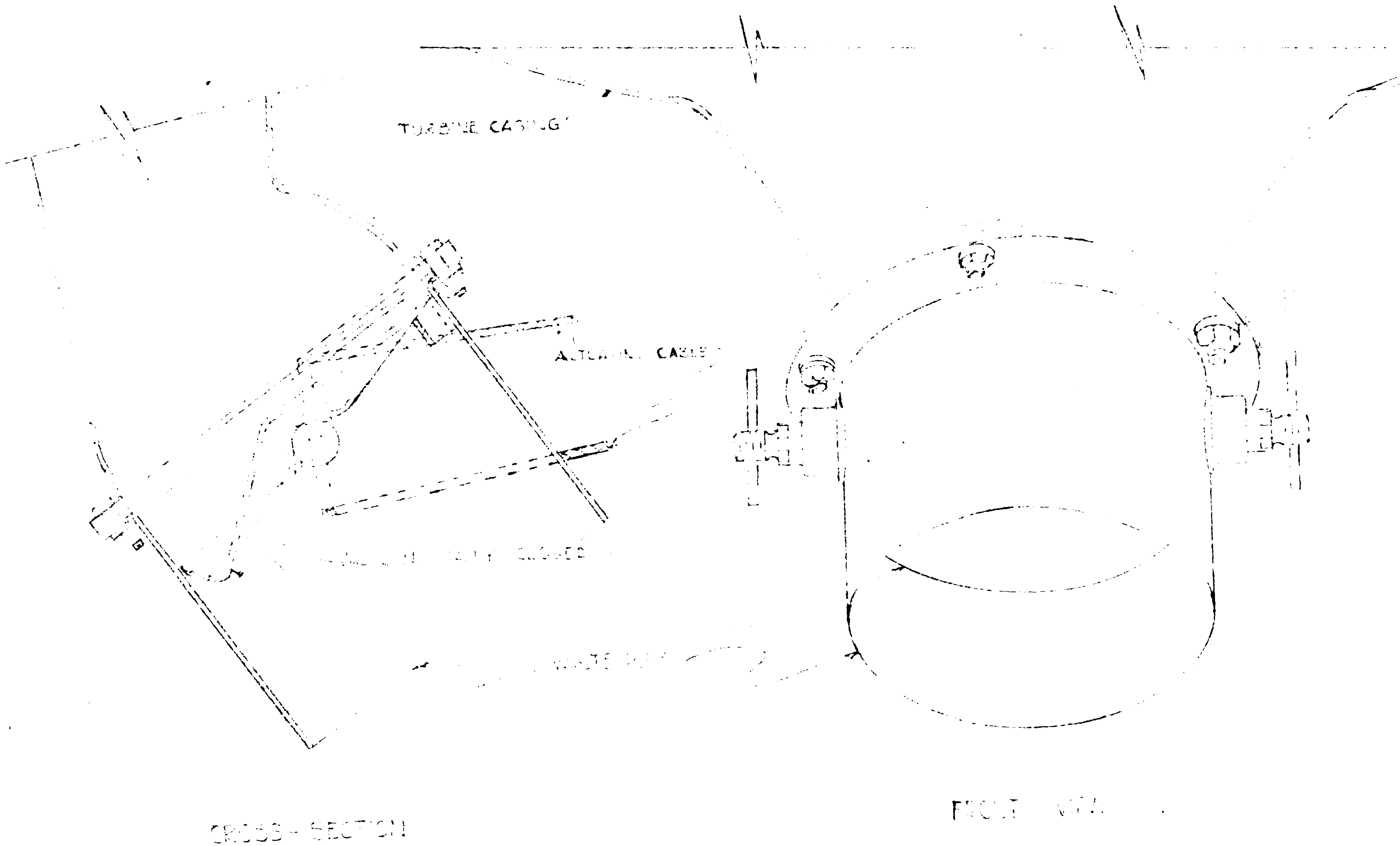


Figure 23 - Waste pipe assembly

The tachometer indicator shall be installed on the control panel.

Pressure Control:

As there is no immediate danger of pressure increase in the combustor, automatic control is not necessary. However, pressure measurements are taken in several points of the unit. These pressure readings will guide the operator regarding the unit operation.

Air Flow Measurements:

The inlet air to the combustor shall be measured by a venturi flow nozzle with a conventional manometer tube which is connected to the venturi tube. The manometer shall be calibrated to read flow in lbs per minute. A venturi flow meter is used here as greater accuracy in flow measurement and small pressure loss is required. Refer to figure 21 for location of venturi and manometer.

Fuel Flow Measurements:

In the measurement of the flow of fuel a good accurate method would be to install a 1/2" type "A" "[illegible]" oil meter or its equivalent. The oil meter range is 2 gpm capacity [illegible] psig and maximum oil temperature of 200°F. The meter shall [illegible]. For location of meter refer to figures 19 and 21.

Speed Measurement:

As described in section "Over-speed Control" a tach motor generator and [illegible] indicator shall be installed [illegible] dynamometer. [illegible]

[illegible]

[illegible]:

[illegible]

[illegible] = [illegible]

[illegible] = [illegible]

[illegible] = .[illegible] ([illegible])

$C_p - C_v = \frac{R}{J} = \frac{[illegible]}{[illegible]} = C_v \times (k-1)$

$C_p = C_v + .[illegible] = .[illegible]$

$\therefore k = \frac{.[illegible]}{.2115} + 1 = 1.[illegible]$

and $W_c = C_p T_{c.o.}\left[1-\frac{1}{(P.R)^{\frac{k-1}{k}}}\right]\times\frac{1}{\eta_c}$

where W_c = [illegible]

$k = 1.[illegible]$

$\therefore W_c = C_p T_{c.o.}\left[1-\frac{1}{(3)^{\frac{[illegible]}{[illegible]}}}\right]\times\frac{1}{\eta_c}$

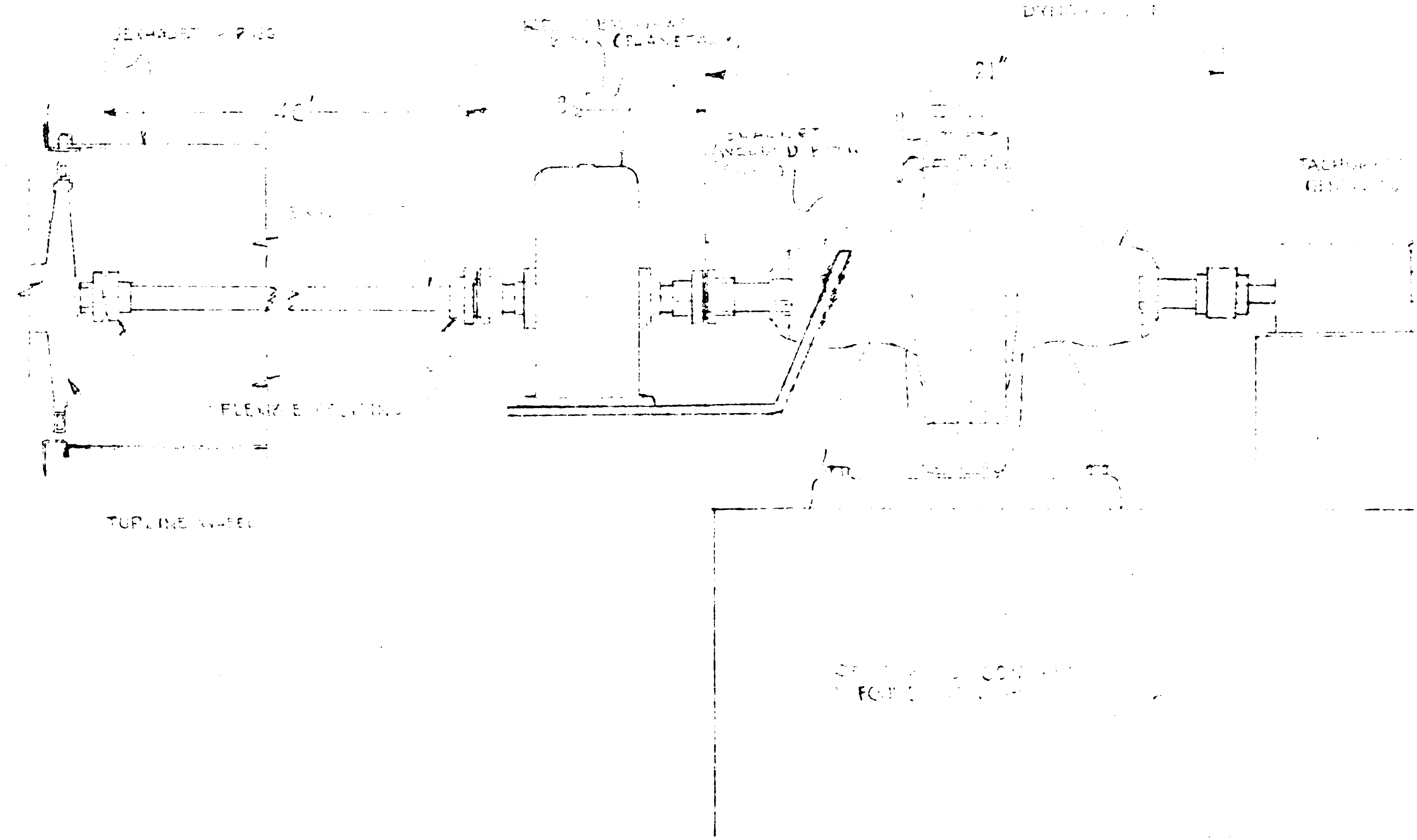


FIGURE 34 - DETAILS OF DYNAMOMETER ATTACHMENT TO TURBINE WHEEL AND TACHOMETER GENERATOR

but $\frac{W_t}{W_c} = \frac{T_{t.i}}{T_{c.o}}$ ([illegible]; for simplified [illegible] see [illegible] "Introduc-tion to [illegible] Turbine [illegible] Propulsion De-sign")

where:

W_c = [illegible]

$T_{t.i.}$ = Turbine inlet temperature

$= 2060°R$

$T_{c.o.}$ = 75[illegible] R

$\therefore W_t = \frac{2060}{75[illegible]} \times (.25 \times C_p T_{c.o}) \times N_T$

$= .[illegible]1\ C_p T_{c.o} \times N_T$

$\therefore$ Net work $= .[illegible]\ C_p T_{c.o.} \times N_T - .[illegible]\ \frac{C_p T_{c.o.}}{N_c}$

but $C_p = .25$

$T_{c.o.} = 75[illegible].7° R$

N_T = Turbine efficiency at optimum conditions

$= .[illegible]$

$\therefore$ Net work $= .[illegible]1 \times C_p T_{c.o.} \times .7[illegible] - \left[\frac{.[illegible] \times C_p \times T_{c.o.}}{.8[illegible]}\right]$

$= .13[illegible]\ C_p T_{c.o.}$

$= .13[illegible] \times .25 \times 755$

$= 27.1$ Btu/lb. of gas

or Net Work in Btu/sec. $= \frac{27.1 \times Q}{60}$

where $Q = 310$ lb/min. of gas

$\therefore$ Net work $= \frac{27.1 \times 310}{60} = 140$ Btu/sec.

and since 1.414 Btu/sec = 1 hp

$$\therefore HP = \frac{140}{1.414} = 99.5 \approx 100 \text{ hp}$$

Therefore, a dynamometer like the Taylor Dynamometer model D 13, with maximum rpm of 12000 rpm, maximum horsepower rating of 100 hp will be able to satisfy service requirements. The dynamometer shall be connected to the turbine shaft by a flexible coupling and a 2 to 1 reduction gear. Because of the peculiar situation in which it seems impossible for the dynamometer to be placed close to the turbine wheel shaft due to the exhaust piping which covers the turbine shaft, a type of Thomas Flexible coupling shall be installed. From the Thomas Flexible Coupling Catalog a Thomas 100 SN type flexible coupling with a maximum bore of 1" and 61" long will be able to transmit .22 hp per 100 rpm. Since the maximum horsepower developed by the turbine equals 100 hp at 24,000 rpm, therefore, the hp. per 100 rpm $= \frac{100}{240} = .416$. It would then be very safe to use such type of flexible coupling and at a little bit shorter length. Due to the very light loads that will be transmitted, there is no problem in the manner of attachment of the flexible coupling to the turbine shaft. A keyway shall be cut on the turbine shaft. From the Machinery Handbook for a 3/4" diameter shaft the dimensions for a flat, plain taper stock key are:

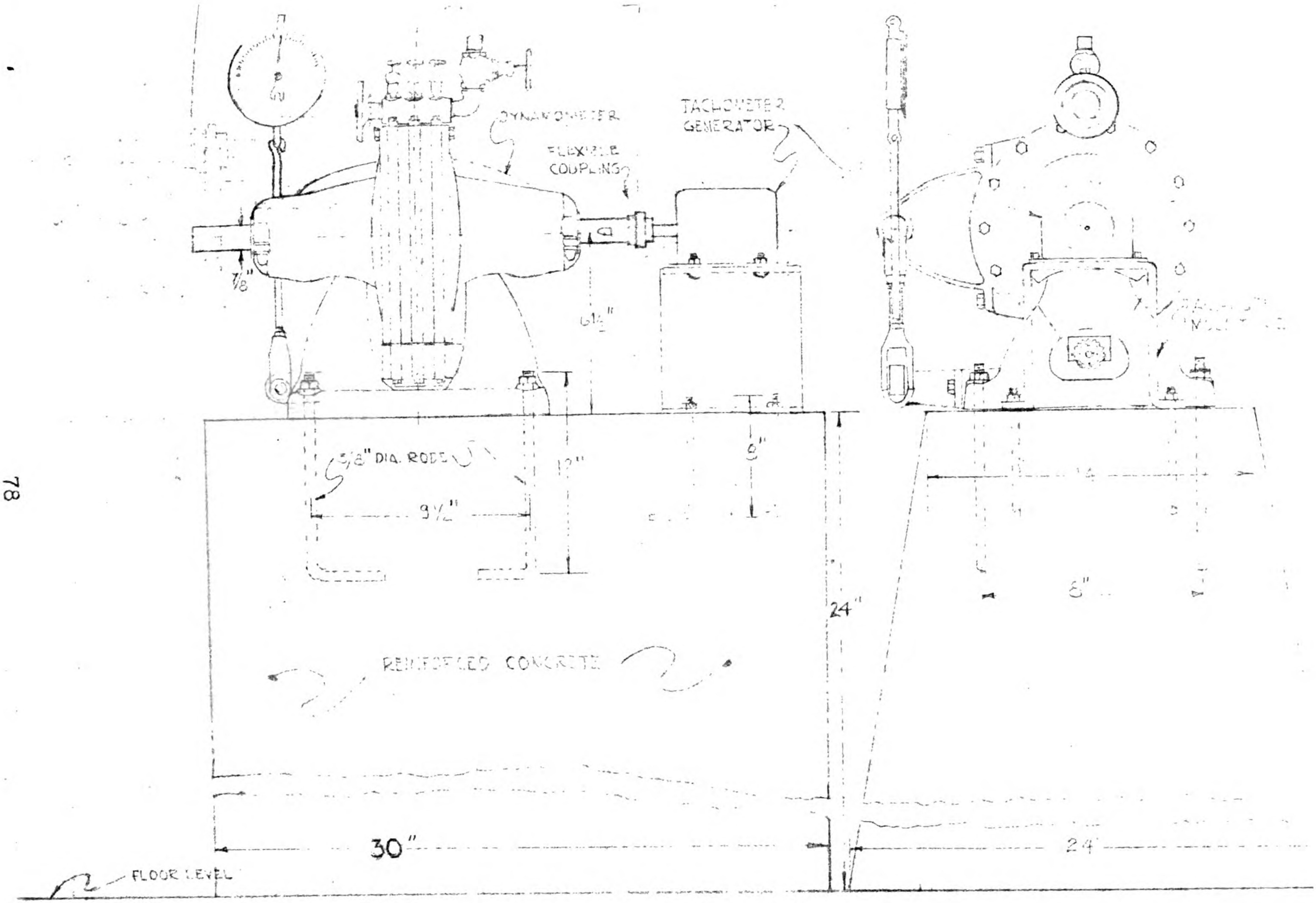


Figure 25 - Foundation details of dynamometer

Maximum width = 3/16"
Height = 1/8"
Length = 4 x 3/16 = 3/4"

An additional set screw shall be installed. The dynamometer end will have the same method of attachment except for the addition of a 2 to 1 reduction gearing.

The point, where the flexible coupling goes through the exhaust pipe bend, must be sealed to prevent leakage of exhaust gases.

For details of the dynamometer attachment and foundation see Figures 24 and 25.

Pressure Measurement:

Pressure measurements shall be made at the following points:

(a) At the compressor outlet and point of measurement shall be at the compressor discharge piping.

(b) At turbine inlet and point of measurement shall be located near the turbine nozzle box.

(c) At turbine exhaust and point of measurement shall be before exhaust silencer.

(d) At compressor inlet and point of measurement shall be made at venturi meter outlet.

(e) At fuel pump discharge piping.

The pressure measuring instrument that will be used for obtaining pressure readings at compressor inlet, compressor outlet, turbine inlet and turbine exhaust is the ordinary well-type manometer. The manometer scale shall be calibrated to read pounds per square inch. The method of attachment at point of measurement will be the conventional method using screwed union bushings. For locations of the different points where pressure measurements are to be taken, and location of the manometer plate refer to figure 21.

The fuel pressure gage will be supplied by the fuel pump manufacturer and range of gage shall be from 0 - 150 psig.

Temperature Measurements:

Accurate temperature readings are required at the following points:

(a) At compressor inlet and probe must be made near venturi meter outlet.

(b) At compressor outlet and location of probe shall be on compressor outlet piping near the pressure probe.

(c) At turbine inlet and thermocouple spider ring shall be placed at nozzle box inlet flange.

(d) At turbine exhaust and thermocouple shall be placed on exhaust piping before exhaust silencer and near the pressure probe.

(e) At combustor discharge and thermocouple shall be located on transfer section. Temperature readings at this point will enable operator to make a comparative study of temperature changes and errors in temperature readings that take place between combustor discharge and turbine inlet.

(f) A bulb thermometer shall be installed at the lubricating oil storage tank. This temperature reading will warn operator of excessive bearing temperatures.

For measuring compressor inlet temperature and compressor outlet temperature a remote indicating bulb thermometer furnished with a rigid extension tube and screwed bushing and fitting shall be used. From Honeywell thermometer catalog, a lead covered bulb with a maximum temperature of 150°F and lead covered tubing with a maximum length of 25 feet will be able to satisfy service requirements. For compressor inlet temperature measurement, a Taylor [illegible] temperature indicator or equivalent with temperature range of 0 - 150°F shall be installed in the instrument panel to indicate temperature readings of the compressor inlet.

For the compressor outlet, a lead covered bulb with a maximum temperature of 450°F and lead covered tubing with a maximum length of 25 feet will be able to satisfy service requirements. [illegible] temperature indicator or its equivalent with a range of 0 - 450° F shall be installed in the instrument panel to indicate temperature readings of compressor outlet.

Since the gases in the gas-turbine are normally hotter than the surrounding walls, the heat radiation from the thermocouple hot junction to the walls may cause considerable error in the observation of gas temperatures, it is important therefore to shield the thermocouple. The construction of a shielded thermocouple for gas-turbine service was developed by [illegible]. [illegible] of the [illegible] of standard [illegible] is shown in Figure 25.

The thermocouples in the turbine exhaust piping and at the [illegible] section shall be made adjustable through the use of [illegible] as shown in Figure 26. The thermocouple [illegible] adjusted [illegible] by traversing it through the cross-section of the pipe.

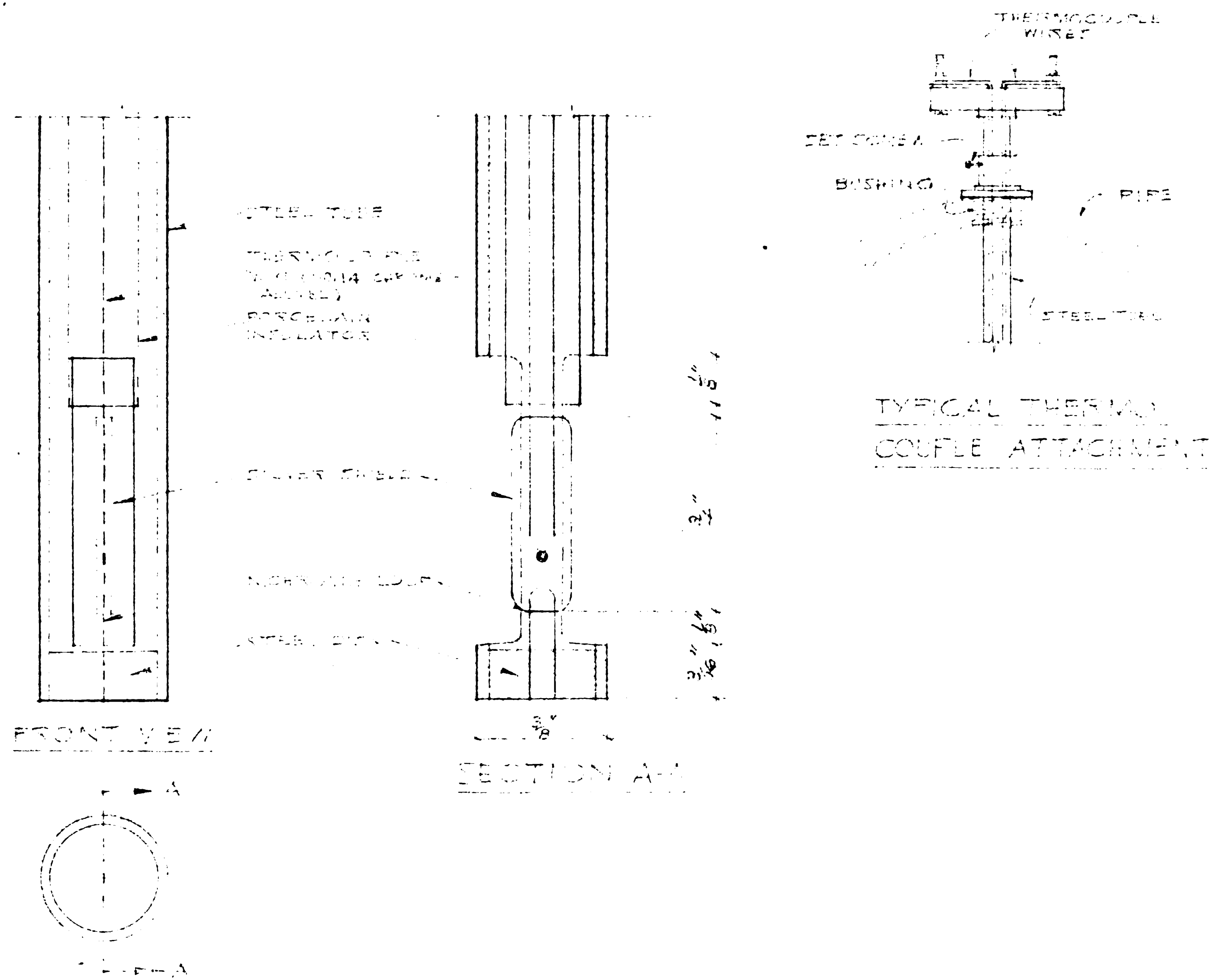


FIGURE 2[illegible] - THERMOCOUPLE CONSTRUCTION DETAIL AND METHOD OF ATTACHMENT

For the turbine inlet temperature measurement [illegible] "[illegible]" [illegible] thermocouples will give the [illegible]. [illegible] com- posed of a flange fitted with [illegible] placed between the flanges of the [illegible] piping and the [illegible] box inlet flange [illegible] shown in Figure 11. The [illegible] thermocouples shall be [illegible] the flange [illegible] placed in service. For details of construction see Figure 27.

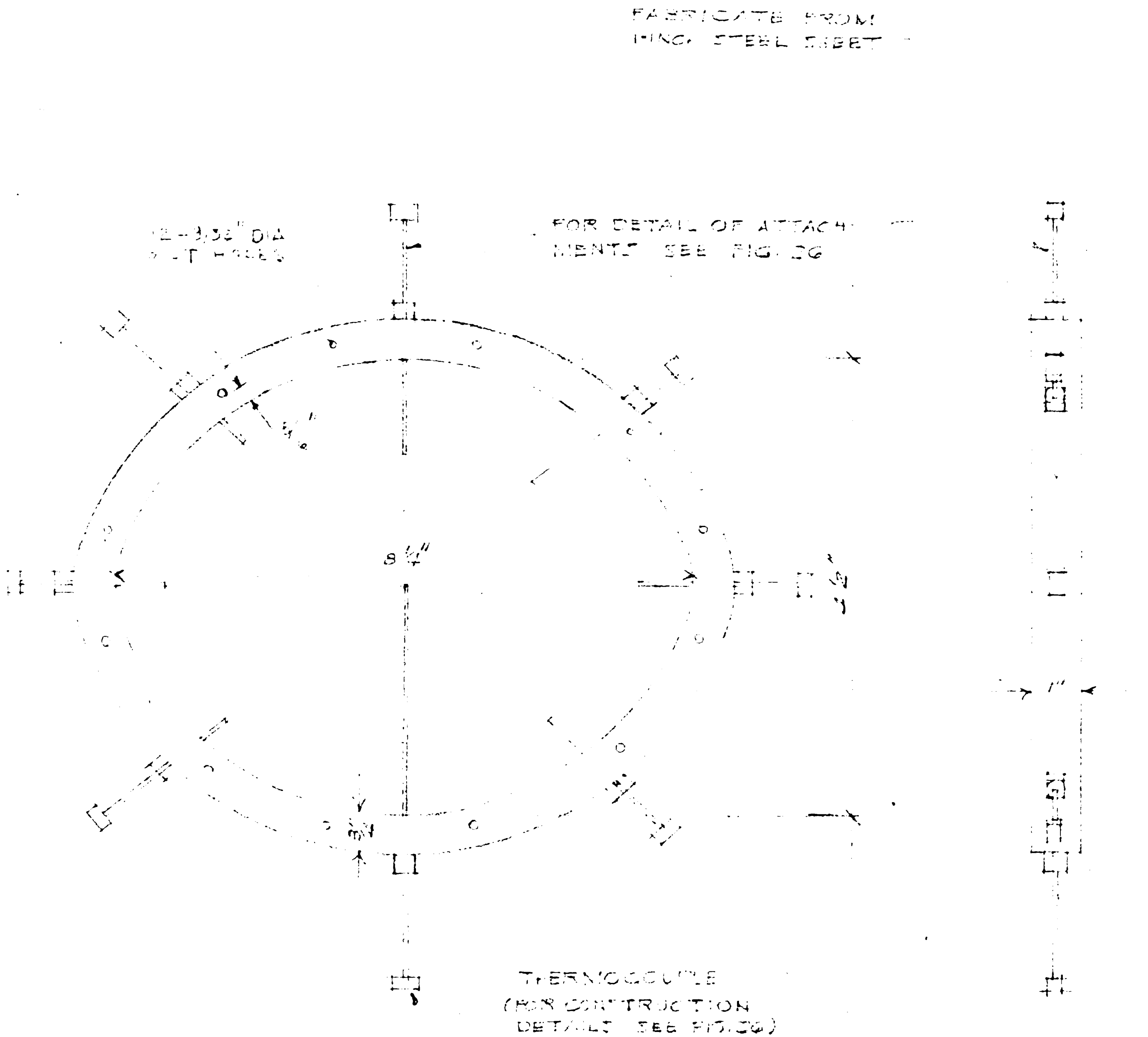


FIGURE 27 - THERMOCOUPLE SPIDER DETAILS

PRECAUTIONS AND OPERATING PROCEDURES

It is indeed very necessary that some precautionary measures and a set of operating procedures be made to guide operating personnel in the operation of the converted gas-turbine engines. Because of improper conditions, it is possible for the unit to overspeed and completely destroy itself by disintegration. The power differences are such that acceleration can be very rapid, and rapid overspeeding can occur from accumulated fuel burning in the combustion chamber, even after the main supply has been turned off. A case has been reported where a converted turbo supercharger has been destroyed by overspeeding due to the accumulation of fuel in the combustor which was sufficient to carry the unit to overspeed even after the main fuel supply was cut off.

The starting steps to be taken when operating the unit are the following:

(a) Before starting unit it is imperative that the unit be purged of any fuel that might have been left from previous operation as this would result in hot starts and probable damage to the turbine bucket. Purging can be done by just operating the starting air jet from the shop line with the fuel supply closed, fuel pump off and the ignition off.

(b) Inspect lines and valves for any leaks or defects and see that solenoid valves are opened. Check all switches and all should be off.

(c) Push "start" button allowing starting air to rotate unit up to proper firing speed (around 3000 rpm), then the quantity control valve should be opened to the proper amount (about 3/4 open) to allow the right amount of fuel for starting and then backed off to half-way mark.

simultaneously, ignition should be on and later turned off after combustion has been obtained. If unit starts, control of unit is dependent on the fuel hand-control or emergency-control valve and/or the emergency stop.

(d) If the unit does not start after first attempt, the unit must be purged of fuel and starting step repeated all over. Allow 10-second period during starting for fuel mixture to ignite before shutting off fuel and turning off ignition. Note: A flame detector instrument equipped with flame "on-off" buttons may be installed to aid operator in ascertaining a successful start.

Some of the precautions to be taken are the following:

(a) Unit must not be operated at full maximum speed (24000 rpm) and at maximum temperature (1500°F) for more than one minute. According to manufacturers recommendation, longer life for the turbine wheel can be achieved if unit will be operated at 1200°F.

(b) Radial cracks in the wheel blank extending beyond Title limit line (beginning of the raised section) are cause for removal. A 1/8" maximum overshoot of the line is the absolute limit. No more than 3/16" circumferential cracking is allowed in any section of the blank.

COMPARATIVE ANALYSIS BETWEEN CONVERTED ENGINE AND AVAILABLE SMALL GAS-TURBINES

Small Gas-Turbine Engine Survey:

In the "Gas-Turbine Progress Report for Small Gas-Turbines" in 1953 as published in A.S.M.E. Transactions, there were only six companies in the United States which were making small gas-turbines. These units range from 25 kw to 300 kw. The six companies are: Solar Aircraft Corporation, AiResearch Manufacturing Company, Boeing Airplane Company, General Motors Corporation, Continental Motors Company and Stratos Division of Fairchild Corporation. All of these companies except General Motors are producing limited quantities of small gas-turbine units. General Motors is experimenting on one model for possible vehicle application. Up to the present time due to previous commitments to the U.S. Government only the Fairchild Corporation and the Solar Aircraft Company can offer for sale to the public limited numbers of their small gas-turbine units. Fairchilds' J44 turbojet engine, which they will be able to offer has a thrust of 1000 lb. and is presently used in missile and target drone aircraft. Solar on the other hand has a unit with a maximum horsepower of 55 hp at 40,000 rpm to offer. The Solar unit is presently used to drive fire pumps. For reference see attached photographs of the commercial units.

Suitability:

For the accomplishment of a [illegible] laboratory unit, there are but three types or models of gas-turbine engines to select from, they are: the [illegible] supercharger unit, the Fairchild's J44 turbo-jet [illegible] the Solar Aircrafts' "Mars" gas-turbine engine. [illegible] result [illegible] the J44 [illegible] [illegible] unit [illegible] instructional purposes. [illegible] used in [illegible] from, [illegible] are unnecessary in [illegible] laboratory [illegible], which [illegible] for its purpose [illegible] flight, [illegible] [illegible], [illegible] would be quite [illegible] in [illegible] gas-turbine unit.

The [illegible] Solar Aircrafts' "Mars" gas-turbine engine, [illegible] that it [illegible] [illegible] power [illegible] [illegible] the [illegible] of the unit.

As for the [illegible] turbo-supercharger, the only [illegible] [illegible] [illegible] operation of the unit. [illegible] these [illegible] turbo-supercharger unit will [illegible] instructional purposes.

<u>[illegible]</u>:

[illegible] J44 [illegible] engine will [illegible] than [illegible]15,000. [illegible]

[illegible]
used by the Allison [illegible]. [illegible]
[illegible], [illegible] will [illegible] combustion
[illegible] for laboratory use.

[illegible] "[illegible]" [illegible] "[illegible]" [illegible] [illegible] profile [illegible] for [illegible] units [illegible] would result in a price of around $[illegible],000 each [illegible]. The cost of freight, instrumentation [illegible] the buying price will [illegible] around $[illegible],000.

The cost of conversion, in [illegible] 1-[illegible] of the turbosupercharger unit is estimated not to exceed $5,000 if most of the work will be done in the engineering shops.

Remarks:

As outlined above, the converted turbosupercharger comes out to be more [illegible] suitable for laboratory use. One further advantage it has is, it will enable engineering students to test several types of combustors. In this [illegible] of [illegible] turbine [illegible], the converted turbosupercharger will provide a [illegible] testing place for new combustion chamber designs.

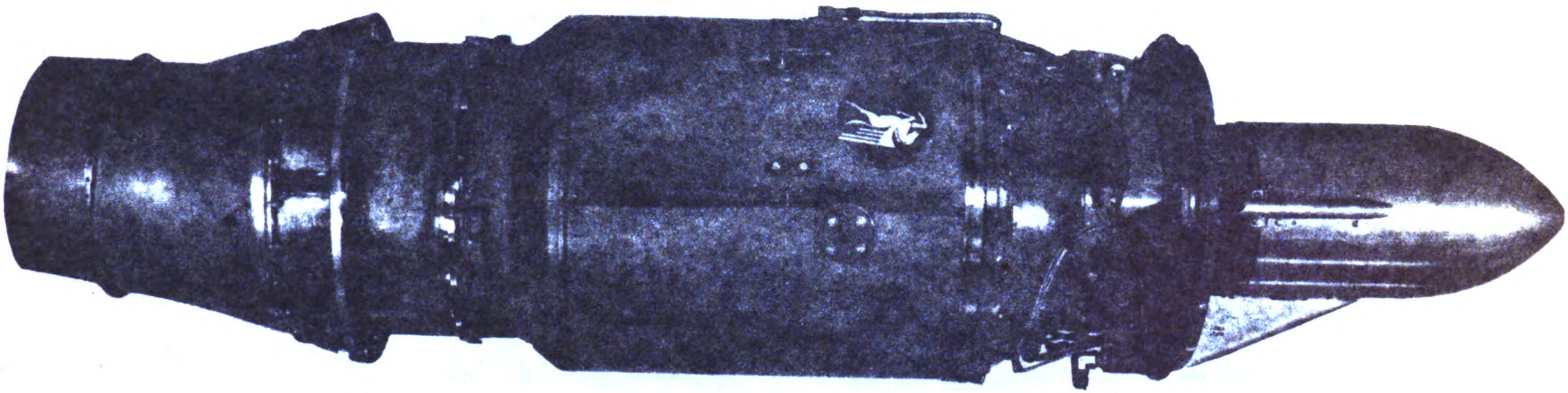
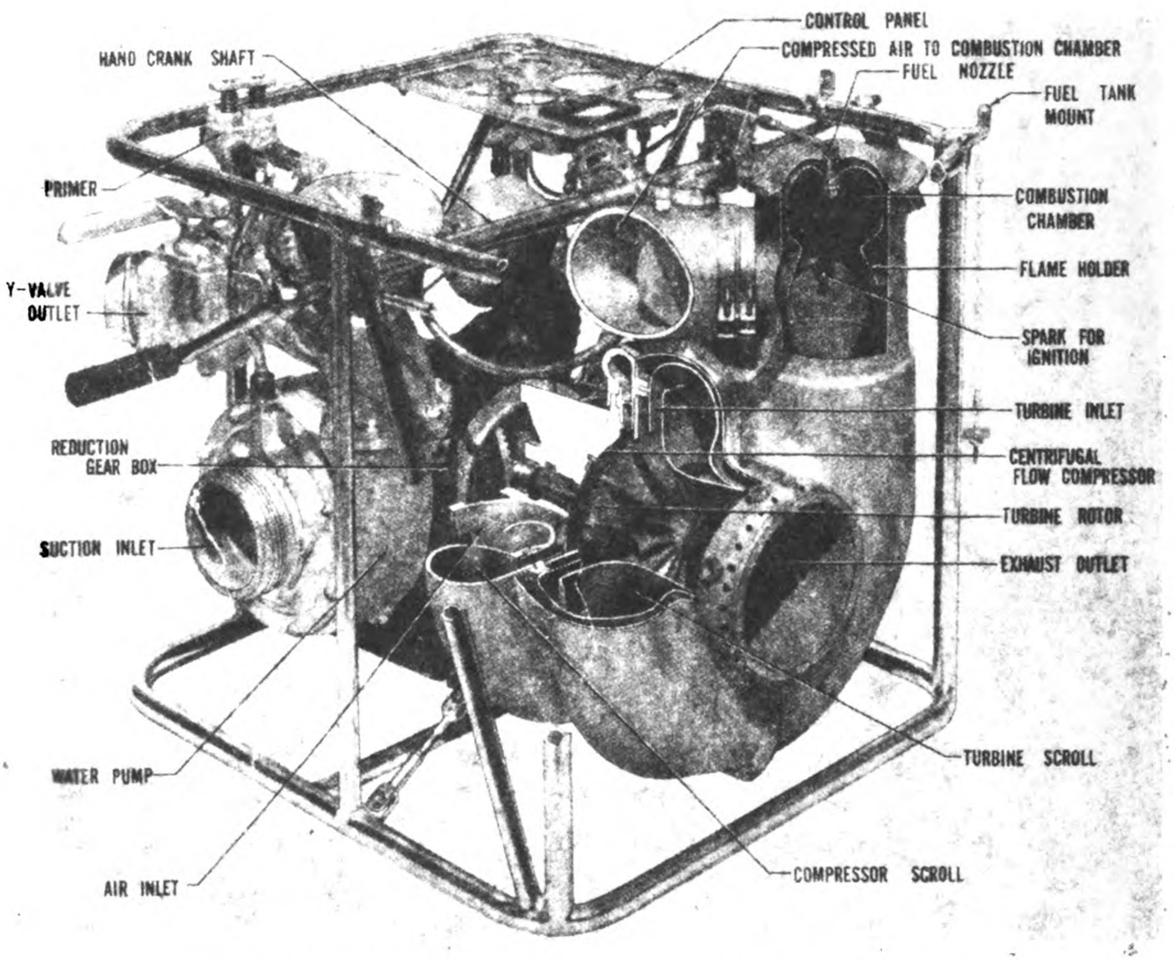


Figure 28 - Fairchild's J44 jet engine

GENERAL DESIGN DATA



SECTIONAL VIEW OF MARS ENGINE T-45M-2 FIRE PUMP ADAPTATION

Figure 29

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