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SECURITY CLASSIFICATION O P E N

SUBJECT THE WEIGHT AND SIZE ESTIMATION OF A TURBO GAS GENERATOR FOR THE POWERPLANT OF A VTOL AIRCRAFT

PREPARED BY N. Galitzine

ISSUED TO Internal

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THE WEIGHT AND SIZE ESTIMATION OF A TURBO GAS GENERATOR FOR THE POWERPLANT OF A VTOL AIRCRAFT

In Reference 1 various gas turbine powerplants were compared for a VTOL aircraft, in which lift was obtained by fans immersed in the wings, and horizontal propulsion by jet reaction.

The powerplant consisted of four turbo gas generators, eight fans with drives, and the ducting, valves, nozzles, etc., necessary to obtain either the vertical lift or the horizontal propulsion.

The criterion in the comparison was the total powerplant plus fuel weight required to fulfill a given flight mission. The weight of the fans, drives and accessories was obtained by sketchily designing them, and then calculating the component weights. The weight of the gas generators was established by reference to the current literature on turbojet engines, since the gas generators were in fact such, when used for the horizontal propulsion of the aircraft.

The present memorandum applies the sketchy design method, as used for the fans and drives, to the gas generators. The aim is threefold:-

a) To check the figures obtained for the gas generators from reference to the literature, and thus also to back check on the method used for the fans and drives.

b) To establish a rational connection between the thermodynamic parameters and the weights.

c) To determine the main dimensions of the gas generators, for use in an overall design of the VTOL aircraft, and for relation to the other components.

Reference 1 established that one of two suitable gas generator thermodynamic cycles was the hot single stream cycle, and that for an aircraft with an all-up weight of 20000 lb. each of the four gas generators should be of 4500 h.p.

Such a gas generator is now taken as an example, its sketchy design worked out, and the weights and dimensions calculated. The results are summarized in Figure 1. The details and particulars follow:-

BASIC DESIGN AND ASSUMPTIONS

The thermodynamic parameters of the cycle assumed for the gas generator were taken from Reference 2. The main figures for the hot single-stream cycle are:-

	Pressure Ratio	Temperature °K
Compressor inlet	1	288
Compressor outlet	9.20	588
Turbine inlet	8.74	1300
Gas delivery	2.86	1033

Specific power	127 h.p. per lb/sec
Specific fuel consumption	0.51 lb/hr per h.p.

The power of the gas generator is defined as that which is produced by the expansion of the gas in an 85% efficient turbine. With a power of 4500 h.p., the air mass flow

$$= \frac{4500}{127} = 35.4 \text{ lb/sec.}$$

One of the main assumptions in the design of the gas generator is that the tip speed of the compressor = 800 ft/sec. This is the same as for the lifting fans. Together with the assumption that the inlet hub/tip ratio = a minimum 0.5, the theory of Reference 3, as applied to compressors, gives the maximum inlet axial velocity = 560 ft/sec. From this and the air mass flow one derives the compressor inlet diameter, 14.2", which is then also taken as the maximum diameter of the combustion chamber and the turbine, giving a long tubular unit of uniform diameter.

Another basic assumption concerns the use of various materials. This is illustrated in Figure 2, which shows the range of stress against temperature of the four types of alloy used in the design. The shaft and bearings are assumed to be made of steel, and are not represented in the figure, but the compressor uses magnesium, aluminum and titanium, whilst the combustion chamber and turbine are of a nickel-chromium base alloy.

The curve for each alloy consists of two parts. The left-hand or lower-temperature side shows the yield stress, representing the elastic limit of the material, and the right-hand or higher-temperature side shows the stress which produces 0.1% creep in 1000 hours, representing arbitrarily the plastic limit of the material. The curves have been derived from many and various sources in the literature, with some few gaps filled in by extrapolation.

The stresses used in the design were referred to the limits as set out in the figure. In the elastic range, a maximum of 2/3 of the yield stress was aimed at, and in the plastic range a stress of a little lower than the arbitrary creep limit shown, or about 0.1% creep in 10,000 instead of 1000 hours.

Other assumptions as applying to the unit as a whole were:-

Minimum blade chord	1/2"
Sheet Metal Thickness	0.050"
Casting Thickness	3/8"

COMPRESSOR

The design of the compressor was based on an approximate adaptation of the blade aerofoil theory with cascade correction, as summarized for fans in Reference 3.

The conditions at the hub being the most critical, the maximum pressure rise per stage was based on that point:-

$$\Delta p = \frac{\rho U^2}{2g} \cdot \frac{2}{\pi} \cdot C_L \cdot \frac{R_1}{R_2} \sqrt{\left(\frac{R_1}{R_2}\right)^2 + \phi^2}$$

where, all in consistent units,

Δp = maximum pressure rise per stage

ρ = mean air density in the stage

U = tip speed

C_L = maximum lift coefficient at the hub, assumed = about 1.75

R_1 = hub radius

R_2 = tip radius

ϕ = axial/tip velocity ratio

In the above is incorporated the cascade correction factor, K, which is assumed approximately

$$K = \frac{2}{\pi} \cdot \frac{2\pi R_1}{Nb} = \frac{4 R_1}{Nb}$$

where

N = number of blades

b = blade chord at hub

Both N and b disappear when K is applied to the basic equation in Reference 3, thus giving the above form of pressure rise equation.

The spacing/chord ratio at the hub, $2\pi R_1/Nb$, was assumed to be 0.8, which was used in the final stages of the compressor design, to find the number of blades, N, per stage.

The main particulars of the compressor are given in Table I.

Having established the maximum pressure rise per stage at the various points in the compression curve, the number of stages was then determined. The first stage shows no pressure rise above atmospheric, because in it is counteracted the depression which causes the velocity at the compressor inlet.

The thermodynamic design of the compressor is compromised to a certain extent by two factors: firstly, by the minimum height of blade considered feasible at the outlet, and, secondly, by the accommodation of the requirements of the dual-stream (bypass) cycle which, when used with heating, was considered as good as the hot single-stream cycle for providing gas to drive the lifting fans in Reference 1.

Had the inlet axial velocity of 560 ft/sec been continued throughout the compressor, the last stages would have had blades too short, particularly in the bypass case. For this reason the axial velocity was gradually reduced from 560 ft/sec at the inlet to 265 ft/sec at the outlet, thus giving a reasonable blade height of 1.43" at the outlet for the single-stream cycle. This dimension can then be reduced, for a bypass cycle, if a compressor design for it were required.

One advantage of the reduction in axial velocity is that no diffuser is deemed necessary between the compressor and the combustion chamber.

The dual-stream cycle chosen in Reference 1 has a compressor temperature rise of 200°C before the bypass, equivalent to a pressure ratio of 5.06. The design in Table I is so arranged that this pressure ratio is obtained in exactly 10 stages, leaving the last 4 stages to be bypassed, thus again accommodating any future design for a dual-stream cycle.

The blade chord was assumed to be the minimum 1/2" in the short blades of the last 4 stages, progressively increased to 5/8" in the middle 6 stages, and to 3/4" in the first 4 stages, thus matching the increasing length of the blades.

The compressor was visualized as being constructed of twelve different components or groups:-

Rotor, drum and blades, stages 1 to 4, magnesium alloy.

Rotor, drum and blades, stages 5 to 10, aluminum alloy.

Rotor, drum and blades, stages 11 to 14, titanium alloy.

Rotor, forward hub, aluminum alloy.

Rotor, rear hub (power transmitting), steel.

Stator, bullet, sheet metal, magnesium alloy.

Stator, inlet casting, magnesium alloy.

Stator, casing (cast in two halves), magnesium alloy.

Stator, outlet casting, aluminum alloy.

Stator, blades, stages 1 to 4, magnesium alloy.

Stator, blades, stages 5 to 10, aluminum alloy.

Stator, blades, stages 11 to 14, titanium alloy.

The stator inlet casting is shown in Figure 1 with a bell-mouth of 18" outside diameter. It is assumed that this diameter would cover any fuel and oil system components, pipes, controls, etc., mounted on the compressor casing, so that 18" represents the overall diameter of the gas generator unit as a whole.

COMBUSTION CHAMBER

The particulars of the (sheet metal, annular) combustion chamber are shown in Table II.

The design was simply based on two criteria: first, the combustion intensity and, second, the mean or 'cold' velocity, values for which were assumed with reference to various recent reports on combustion chamber tests.

Mean or 'cold' velocity is defined as

$$U = \frac{W}{\rho A}$$

where, all in consistent units,

W = air mass flow

ρ = density at combustion chamber inlet

A = maximum cross-section area of the combustion chamber.

The value of U assumed (about 120 ft/sec) was deemed consistent with the pressure loss (5%) allowed for the combustion chamber in Reference 2.

Having derived the area $A = 0.84\ ft^2$, and assuming the outside diameter to be the same 14.2" as that of the compressor, the inner diameter of the combustion chamber then followed.

The combustion intensity is defined as

$$I = \frac{\text{Fuel flow x calorific value of the fuel}}{\text{Comb. ch. volume x Comb. ch. pressure}}$$

This gives either the overall or flame tube volume, depending on which volume the intensity value is based.

The overall volume and maximum cross-section area determine the length of the combustion chamber, and the flame tube is made to fit this, and also its own volume requirement.

The fuel spray nozzles, piping, igniters, etc., are not shown in Figure 1. Their weight is included in the fuel system.

FUEL AND OIL SYSTEMS

The weight of the fuel system was based on the rate of fuel flow and estimated by reference to the literature on the subject. A small amount of weight was added to take care of the oil system.

This is not entirely a rational procedure since the fuel system weight must also have some reference to the fuel pressure, and hence to the combustion chamber pressure, as well as to the number of controls required, etc.

Since the weight estimated for the two systems amounted to a considerable proportion (about 12%) of the total weight, a separate sketchy design should perhaps be made of the fuel system, in order to obtain a more accurate estimate of the weight.

TURBINE

The particulars of the (3-stage) turbine are shown in Table III.

It was designed as a pressure-compounded simple impulse type, since the degree of impulse or reaction was not thought to have any large effect on the weight and overall dimensions.

The number of stages was determined from considerations of the ideal or (velocity) 'diagram' efficiency. An overall figure of at least 90% was aimed at, to cover the overall actual efficiency of 85% required by Reference 2.

All the stages were limited by a gas deflection of 110° in the rotor blades, and the final stage was also limited by the allowable leaving axial velocity, which in turn depended on the leaving area, and hence the hub/tip diameter ratio. This last was permitted to be as low as 0.5.

As in the compressor, the blade chord was assumed to be the minimum 1/2" in the short blades of the first stage, progressively increased to 5/8" in the second and 3/4" in the third, to match the increasing length of the blades.

The number of rotor blades in each stage was determined from an allowable spacing/chord ratio at the tip = 0.6.

The spacing/chord ratio of the nozzles was allowed = 0.75.

The turbine was visualized as being constructed of six different components or groups, all made of nickel-chromium alloy:-

Rotor drum and blades, 3 stages.

Rotor hub.

Stator inlet plate (for rigidity).

Stator casing, sheet metal, in two halves.

Stator nozzles, 3 stages.

Stator cooling drum, sheet metal.

A drum type of construction was allowed for the rotor because of the relatively small diameter and low peripheral speed, and the cooling possible of its inside surface.

The purpose of the cooling drum is to constrain the cooling air to flow at a high velocity past this surface, to the turbine outlet. The cooling air is assumed derived from the compressor delivery.

EXHAUST AND NOZZLE

These are shown together in Figure 1, without any jet pipe between them, being the minimum necessary for the operation of the gas generator as a turbojet engine. There would be ducting and valves at the turbine outlet, if the gas from the generator were also used for driving the lifting fans.

SHAFT AND BEARINGS

The shaft was assumed to be made of the strongest carbon steel. The maximum diameter of 1 3/4" was based on the permissible shear stress resulting from the transmission of power from the turbine to the compressor, which is 6500 h.p. at 12900 r.p.m.

The power transmitting part of the shaft extends from the turbine hub to the steel hub at the high pressure end of the compressor.

The remaining part, of 1 3/8" diameter, is assumed to carry no power, but is used to transmit the compressor thrust to the thrust bearing, and to support the low pressure end of the compressor.

Only the forward bearing was assumed to carry any load, and that due to the compressor thrust, which was estimated to be about 7000 lb. Three angular contact ball bearings in series were necessary for this, according to data from a manufacturer's handbook.

The middle and turbine bearings were arbitrarily assumed to be of about the same size as one of the thrust bearings.

NUTS AND BOLTS

This item of 6 lb. covers a small amount for internal piping (such as cooling air pipes, if necessary) and any other unforeseen parts.

ACCESSORIES

No provision was made in the weight estimate for a starter, installation lugs, jet pipe and other accessories, so that only the 'dry' or 'basic' weight is given.

CONCLUSIONS

1. The basic weight of a 4500 h.p. hot single-stream-cycle gas generator, without accessories or installation, is estimated from a sketchy design to be 340 lb.

   This compares with 475 lb. allowed for in Reference 1, by derivation from the literature, thus leaving 135 lb. for accessories and installation, a figure considered adequate and appropriate.

2. The overall dimensions of the gas generator are estimated to be about 18" diameter x 4 ft. long, with a rotor diameter of about 14".

3. A rational method of weight and size estimation is established, related to the thermodynamic parameters, which may be used for various thermodynamic cycles and powers, other than the example taken in this memorandum, and a formula may perhaps be later derived, which would give quick approximate results.

REFERENCES

1. Laboratory Memorandum NAE-ENG-56, entitled "Dual-Operation Powerplants for Vertical Take-off Transport Aircraft: A Comparison of Several Drives for Wing-Immersed Fans." By E.P. Cockshutt and N. Galitzine. November 1957.

2. N.A.E. Report No. LR-201, entitled "Dual-Operation Powerplants for Vertical Take-off Transport Aircraft: A Preliminary Analysis of Several Gas-Turbine Cycles." By E.P. Cockshutt. July 1957.

3. Laboratory Memorandum NAE-ENG-54, entitled "Lifting Fans for Immersion in the Wings of a VTOL Aircraft." By N. Galitzine. August 1957.

TABLE I

PARTICULARS OF COMPRESSOR FOR HOT SINGLE-STREAM CYCLE

INLET TEMPERATURE = 288°K (15°C)

AIR MASS FLOW = 35.4 LB/SEC   TEMPERATURE RISE = 300°C

OVERALL EFFICIENCY = 85%

STAGE NUMBER	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Press. ratio per stage	-	1.190	1.192	1.194	1.196	1.198	1.200	1.202	1.204	1.206	1.160	1.160	1.160	1.160
Press.ratio at outlet	1.00	1.19	1.42	1.695	2.025	2.425	2.91	3.50	4.21	5.06	5.87	6.81	7.90	9.20
Temp. rise per stage °C	-	17	18	20	21	22	24	25	26	27	23	24	26	27
Temp. rise at outlet °C	-	17	35	55	76	98	122	147	173	200	223	247	273	300
Temp. at outlet °K	288	305	323	343	364	386	410	435	461	488	511	535	561	588
Density at outlet lb/ft^3	.0765	.086	.097	.109	.123	.138	.156	.177	.201	.229	.253	.280	.310	.344
Tip velocity ft/sec	800	800	800	800	800	800	800	800	800	800	800	800	800	800
Axial vel. at outlet ft/sec	560	543	525	508	490	473	455	438	420	385	350	315	280	265
Blade tip diameter inches	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2
Mean blade hub diameter ins.	7.1	7.54	8.28	8.96	9.56	9.96	10.28	10.70	11.04	11.28	11.34	11.34	11.34	11.34
Mean blade height inches	3.55	3.33	2.96	2.62	2.37	2.12	1.94	1.75	1.58	1.46	1.43	1.43	1.43	1.43
Blade section, thickness %	12	12	12	12	12	12	12	12	12	12	12	12	12	12
No. of blades per stage	37	40	43	47	60	63	65	67	69	71	88	88	88	88
Blade chord inches	3/4	3/4	3/4	3/4	5/8	5/8	5/8	5/8	5/8	5/8	1/2	1/2	1/2	1/2
Tip spacing/chord ratio	1.6	1.5	1.4	1.25	1.2	1.15	1.1	1.07	1.04	1.01	1.0	1.0	1.0	1.0

TABLE II

PARTICULARS OF COMBUSTION CHAMBER

FOR HOT SINGLE-STREAM CYCLE

Type	Annular
Air Mass Flow	35.4 lb/sec
Inlet Temperature	588°K
Outlet Temperature	1300°K
Temperature Rise	712°C
Inlet Pressure Ratio	9.2
Outlet Pressure Ratio	8.74
Pressure drop/inlet pressure	5%
Air Inlet Velocity	265 ft/sec
Air Inlet Density	0.344 lb/ft^3
Combustion Efficiency	100%
Fuel Flow	2300 lb/hr
Calorific Value of the Fuel	10,000 CHU/lb
Overall Air/Fuel Ratio	55.5
Primary Air/Fuel Ratio	18.5
Outside Diameter	14.2"
Inside Diameter	7.1"
Length	13 1/2"
Max. Cross-Section Area	0.84 ft^2
Overall Volume	0.89 ft^3
Flame Tube Volume	0.51 ft^3
Combustion intensity (based on overall volume)	3 x 10^6 CHU/hr/ft^3/at.
Combustion intensity (based on flame tube vol.)	5 x 10^6 CHU/hr/ft^3/at.
Mean or cold velocity (based on max. cross-sect. area)	120 ft/sec
Pressure loss factor	12

TABLE III

PARTICULARS OF TURBINE FOR HOT SINGLE-STREAM CYCLE

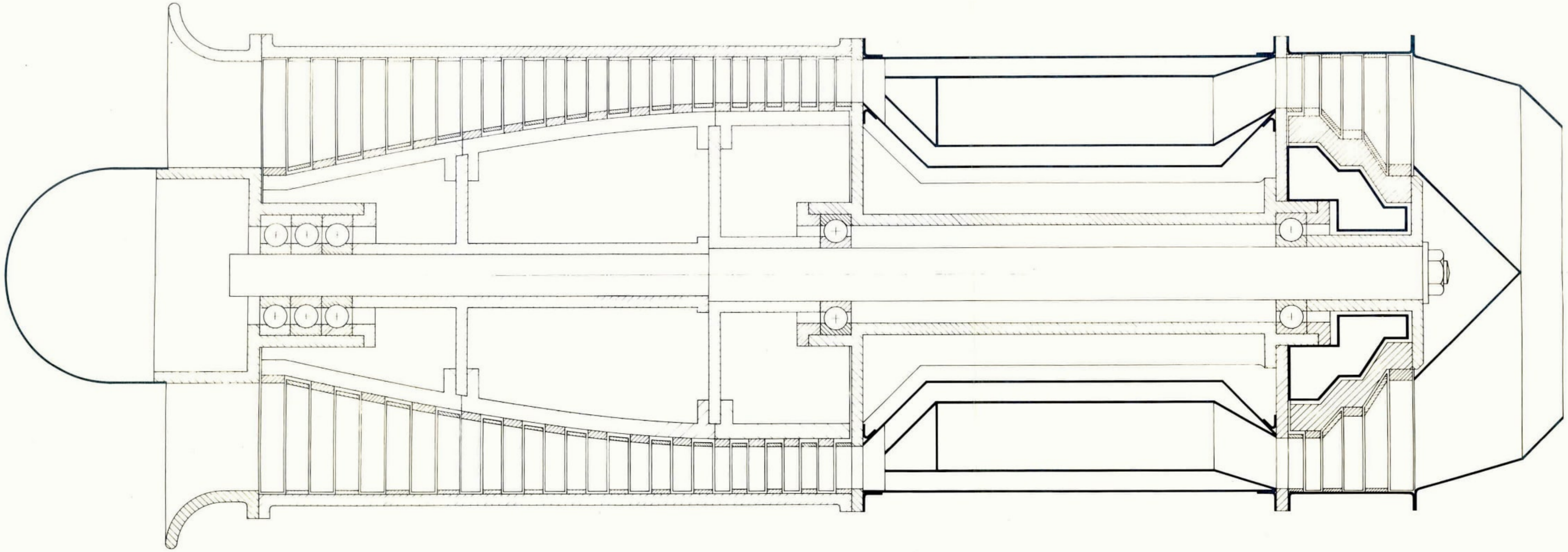
INLET TEMPERATURE = 1300°K (1027°C)

AIR MASS FLOW = 35.4 lb/sec TEMPERATURE DROP = 267°C

OVERALL EFFICIENCY = 85%

Stage number	1	2	3
Press.ratio per stage	1.452	1.452	1.452
Press.ratio at inlet	8.74	6.01	4.14
Press.ratio at outlet	6.01	4.14	2.85
Temp. at inlet °K	1300	1205	1116
Temp. at outlet °K	1205	1116	1033
Temp. drop per stage °C	95	89	83
Density at outlet lb/in^3	0.110	0.082	0.061
Tip velocity ft/sec	800	800	800
Gas velocity ft/sec	1684	1715	1630
Axial velocity ft/sec	700	700	700
Mean periph.vel. ft/sec	706	666	600
Nozzle angle °	24½	24	25½
Gas deflection °	110	110	110
Blade tip diameter inches	14.2	14.2	14.2
Blade hub diameter inches	10.86	9.44	7.1
Blade height inches	1.67	2.38	3.55
Blade section, thickness %	15	15	15
No. of blades per stage	150	120	100
Blade chord inches	1/2	5/8	3/4
Tip spacing/chord ratio	0.6	0.6	0.6

FIG. 1.



4,500 H.P. GAS GENERATOR FOR THE POWER PLANT OF A V.T.O.L. AIRCRAFT

QUARTER SCALE SKETCH FOR WEIGHT ESTIMATION

THE GAS GENERATOR IS SHOWN ABOVE IN THE FORM OF A TURBOJET, WHERE THE GAS IS EXPANDED THROUGH A NOZZLE TO GIVE A STATIC SEA LEVEL THRUST OF 2,670 LB. OR THE GAS MAY BE USED IN AN 85% EFFICIENT TURBINE TO DEVELOP 4,500 H.P. FOR DRIVING LIFTING FANS (REFERENCE LAB. MEMO N.A.E.-ENG.-56)

THERMODYNAMICS

CYCLE	HOT SINGLE STREAM
TURBINE INLET TEMPERATURE	1,300 °K.
COMPRESSOR TEMPERATURE RISE	300°C.
COMPRESSOR PRESSURE RATIO	9.2
GAS DELIVERY PRESSURE RATIO	2.85
GAS DELIVERY TEMPERATURE	760°C.

MATERIALS AND WEIGHTS

COMPRESSOR ROTOR	Mg. AL.Ti.	59
COMPRESSOR STATOR	Mg. AL.	68
COMBUSTION CHAMBER	Ni.– Cr.	36
FUEL AND OIL SYSTEMS	—	40
TURBINE ROTOR	Ni.– Cr.	52
TURBINE STATOR	Ni.– Cr.	38
EXHAUST AND NOZZLE	Ni.– Cr.	7
SHAFT AND BEARINGS	STEEL	34
NUTS AND BOLTS	STEEL	6
TOTAL		340 LB.

SPEEDS AND DIMENSIONS

AIR MASS FLOW	35.4 LB./SEC.
TIP SPEED	800 FT./SEC.
SPEED	12,900 R.P.M.
COMPRESSOR INTAKE DIAMETER	14.2"
NUMBER OF COMPRESSOR STAGES	14
NUMBER OF TURBINE STAGES	3
OVERALL LENGTH AS TURBOJET	4'-3"

SPECIFIC WEIGHTS

PER LB/SEC AIR MASS FLOW 9.6 LB. PER LB. THRUST 0.127 LB. PER HP 0.076 LB.

RANGE OF APPLICABILITY OF VARIOUS ALLOYS IN GAS TURBINE SERVICE

THE SYMBOLS ON THE CURVES DENOTE THE METAL WHICH IS THE BASE OF THE ALLOY