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TECHNICAL REPORT No. 365-45

PARTIAL ANALYSIS OF GERMAN AIRCRAFT GAS TURBINE PERFORMANCE AND DESIGN METHODS

October 1945

U·S·NAVAL·TECHNICAL·MISSION·IN·EUROPE

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TECHNICAL REPORT NO. 365-45

PARTIAL ANALYSIS OF GERMAN AIRCRAFT GAS TURBINE PERFORMANCE AND DESIGN METHODS

SUMMARY

The subject report is a partial analysis of German Aircraft jet engines and propeller drive gas turbine power plant performance and design methods. It is based on the results of interrogation of the leading gas turbine designers of BMW, Heinkel-Hirth, Brown-Boveri and MAN companies. It indicates that gas turbines designed by German aircraft engine manufacturers are not revolutionary in either their design or performance.

October 1945

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TABLE I

GERMAN JET ENGINE AND GAS TURBINE CHARACTERISTICS

Mfr. & Model Designation	S.l. Static Thrust (lb)(1)	Weight (lbs)	lbs wgt / lbs S.T. @ s.l.	lbs fuel/hr / lbs thrust	D.D. (in.)	Length (in.)	Comp. Ratio s.l. stat	Number Compressor Stages	Number Turbine Stages	Turb. Inlet Temp. oF	Airflow s.l. static lbs/sec	lb/thrust.	lb air/sec.
B.M.W. 003A2	1770	1345	0.758	1.35	22(2) 23.2	106	4.0	7 axial	1 axial	1436	41.6	42.6	
Junkers 004B4	1980	1590	0.802	1.35	(2) 22.0		3.5	8 axial	1 axial				
Heinkel Hirth 011	2430	1940	0.800	1.30	34.0	136	4.4	1 ind. 1 diag. 3 axial	2 axial	1382	66.0	37.0	
BMW 018	7730	5100	0.660	1.1	49.0	141	6.0	12axial	3 axial		176.0	44.0	
GERMAN GAS TURBINE PROPELLOR DRIVES													
HEINKEL Hirth 021		2700					4.4		3 axial				
B.M.W. 028	17,600 6800	7250			49.0	228	6.0	12 axial	4 axial		176.0		

(1) includes dual rotation props. (2) Compressor only.

TABLE II

GERMAN AXIAL COMPRESSOR CHARACTERISTICS

Manufacturer and Model Designation	Number of Stages	Press Ratio 68°F Inlet T	Ad. Eff. %	Axial Inlet Vel. ft/sec.	Rotor tip Speed ft/sec	First Stage Rotor Tip Mach No.	Static Press Rise Across Rotor %	(1 Stage) 2ε Ad Press Coeff.	D_{root}/D_{tip} Inlet	D_{root}/D_{tip} Dis.	Circ. Gap/Chord	ax. vel./Tip vel. Inlet	ax. vel./Tip vel. Disc.
B.M.W. 109-003	7 axial	4.0	80	460	860	0.73	80	0.5 1-5 0.6 5-7	0.725	0.875	1.0 tip 0.67 root	0.534	
Junkers 109-004	8 axial	3.5		460	840			0.66	0.66	0.82	tip 1.0	0.545	
Heinkel Hirth 109-011	1 ind 1 diag 7 axial	4.4	78	525	axial Stg. 1440	0.95	axial stg. 50	axial .475	ind. 0.19	axial 0.9			
Brown Boveri Hermso I	10 axial	4.3	85	460	820	360 0.68	50	0.48	0.70	0.835	0.8 root	0.56	0.44
Brown Boveri Hermso II	8 axial	3.5	86	460	820	360 0.68	50	0.48	0.70	0.82	0.8 root	0.56	0.44
Brown Boveri Hermso III	7 axial	3.5	86	490	360	330 0.75	50	Stg. 0.43 1 0.51 2	0.66	0.835	0.8 root	0.52	0.35
M.A.N. (Design Method)	-						80	0.40					
B.M.W 109-018	12 axial	5.0	82	395	955	0.81 (1)	80		0.61	0.85		0.41	

(1) estimated

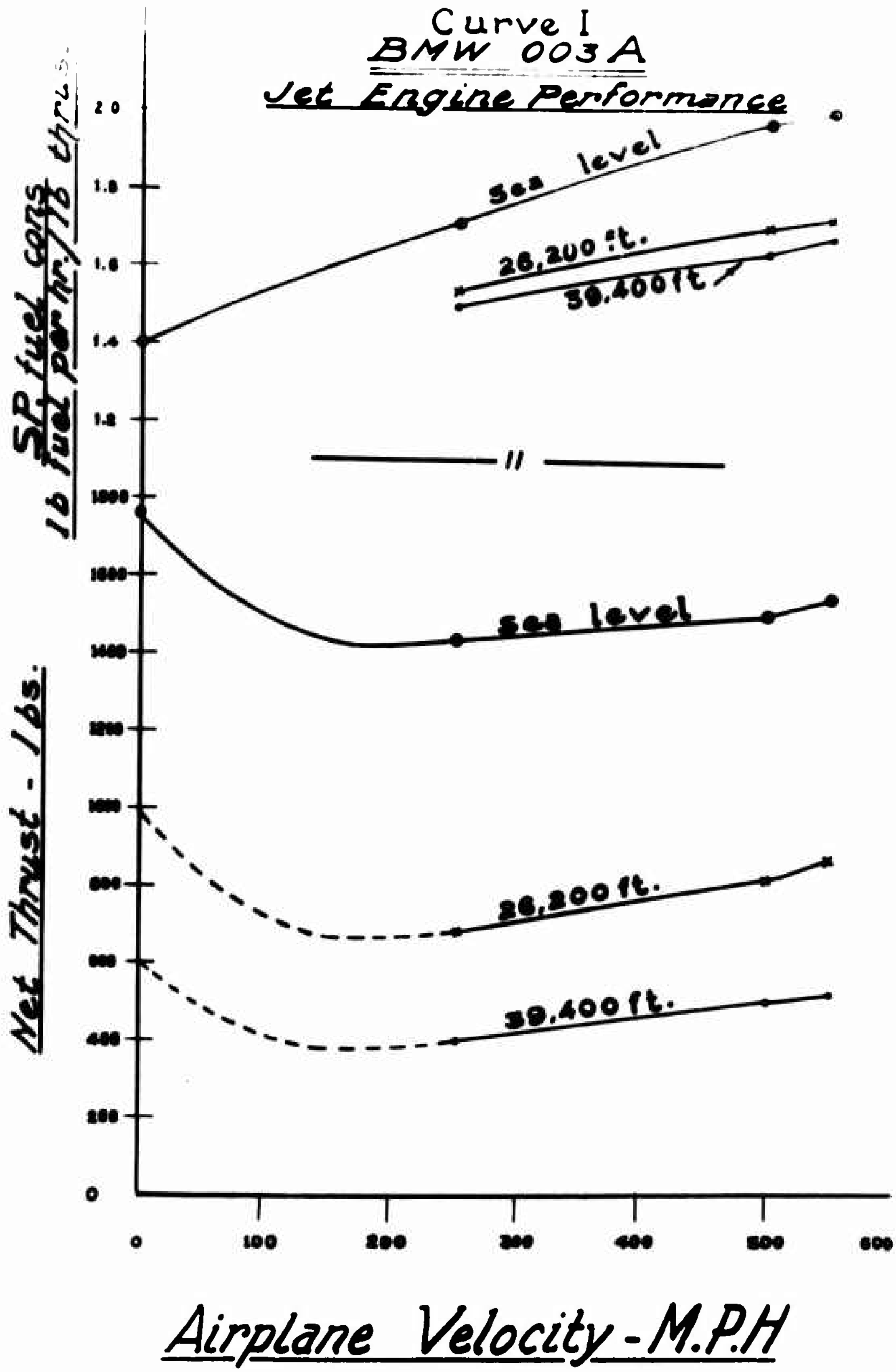
RESTRICTED

TABLE III

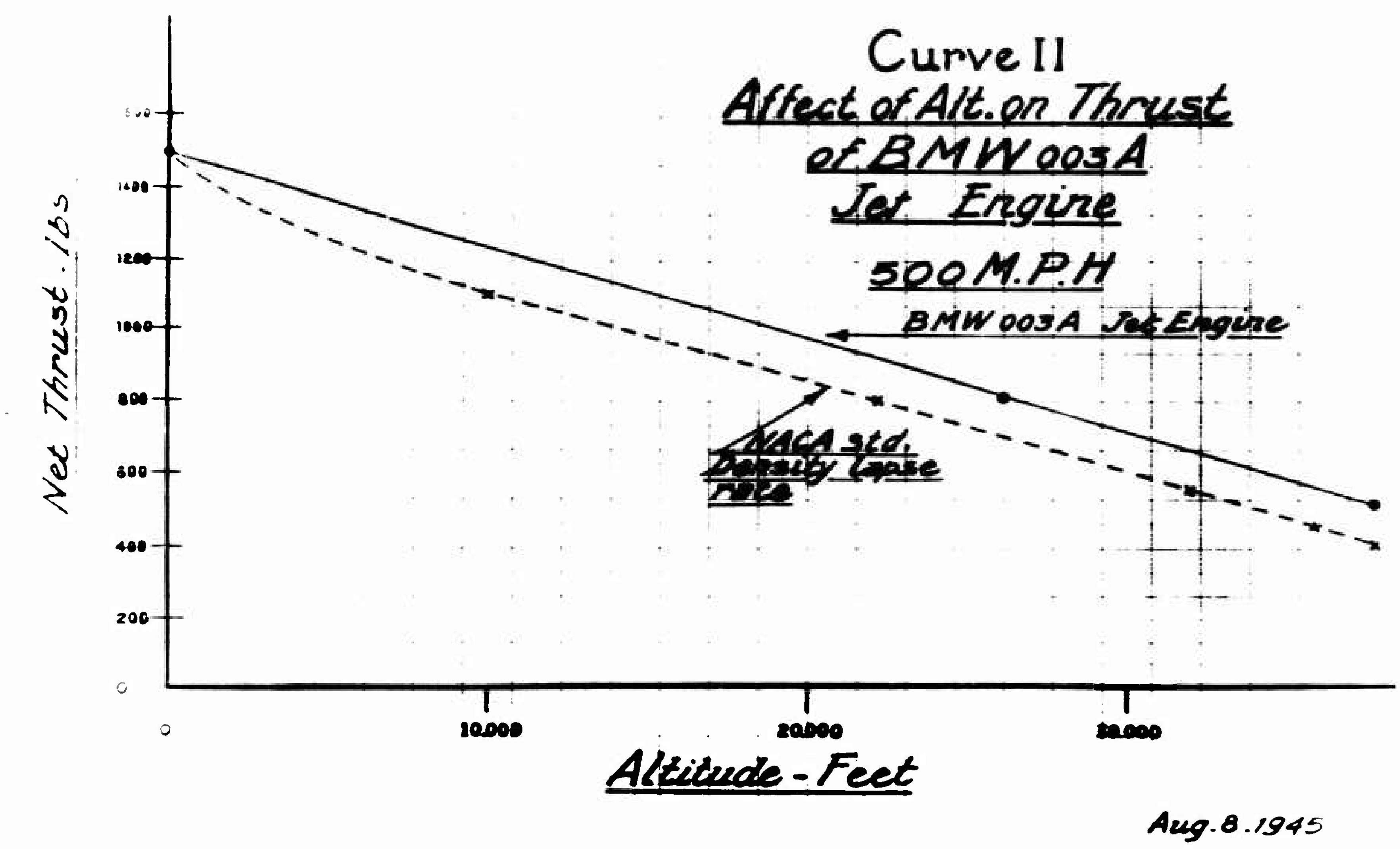
GERMAN GAS TURBINE CHARACTERISTICS

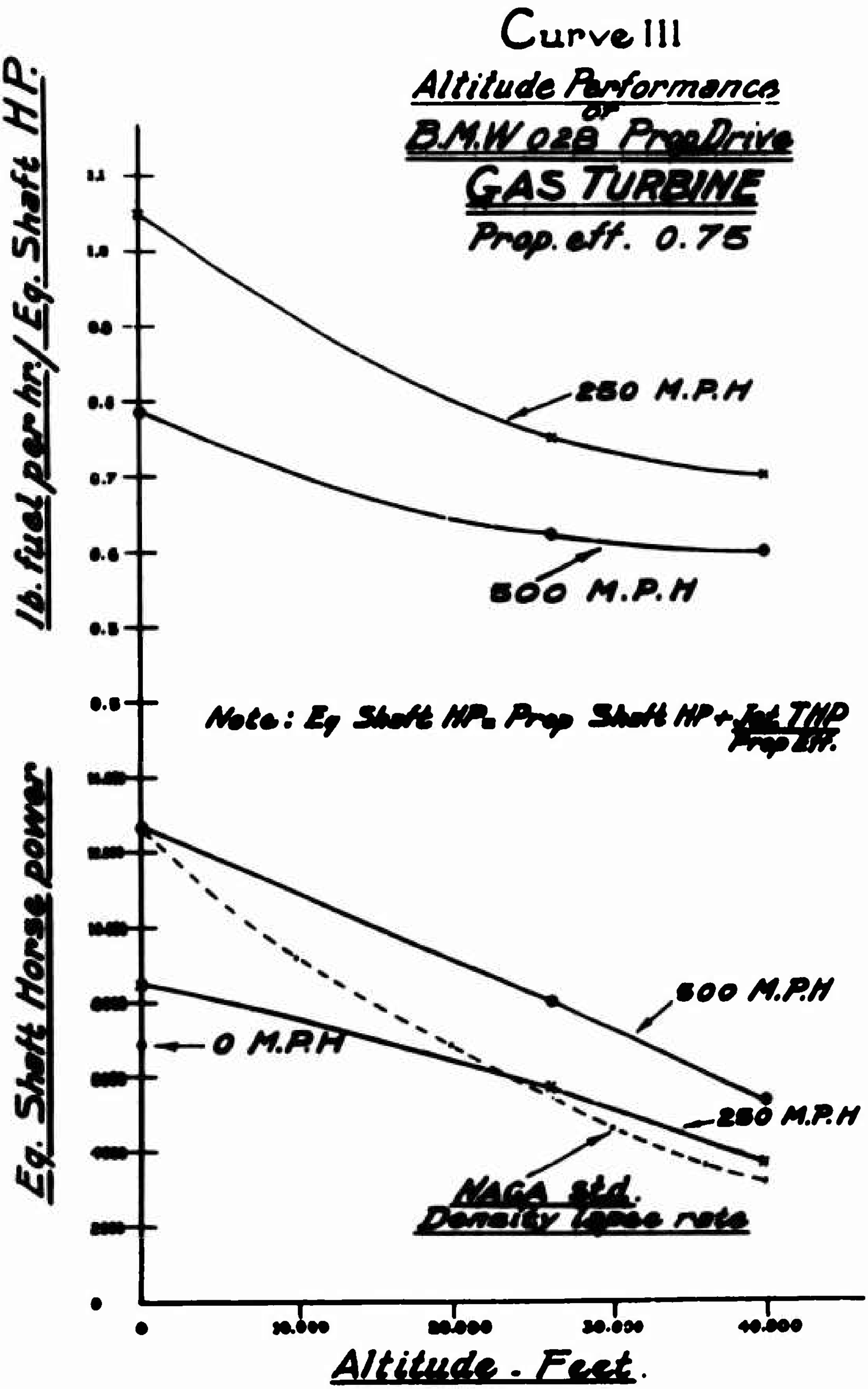
Manufacturer and Model Designation	Number of stages	Turb. Inlet Temp. °F	% Reactive at Mean Diameter	Rotor Tip Speed Ft/sec.	Int. Eff. %	Axial disc. Vel. ft/sec.	Gap / Chord	D_{root} / D_{tip}	Rotor Blade Aspect Ratio	Nozzle Aspect Ratio	Exp. Ratio (Approx) No ram
B.M.W. 109-003	1	1436	30	1040	82			0.60	2.65	1.15	2.0
Junkers 109-004	1										1.7
Heinkel Hirth 109-011	2	1392	30	1150	82	1300	0.69 Mean 0.83 Tip	0.60	3.1	1.14	2.2
B.M.W. 109-019	3			1040				0.65	(2) 3.6	(1) 1.3	3.0
Brown Bovri Sub. Turbo Super Chrg.	1	1160		960			0.77	0.89	2.25	1.0	2.0
B.M.W. 109-028	4			1040		850				-	6.0

(1) Projected chord
(2) First stage


Curve I
BMW 003A
Jet Engine Performance
SP. fuel cons.
lb fuel per hr./lb thrust
Net Thrust - lbs.
Sea level
26,200 ft.
39,400 ft.
Airplane Velocity - M.P.H
2.0
1.8
1.6
1.4
1.2
1800
1600
1400
1200
1000
800
600
400
200
0
100
200
300
400
500
600

Aug. 8. 1945


Curve II
Affect of Alt. on Thrust
of BMW 003A
Jet Engine
500 M.P.H
BMW 003A Jet Engine
NACA std.
Density Lapse
rate
Net Thrust . lbs
1400
1200
1000
800
600
400
200
0
10,000
20,000
30,000
Altitude - Feet
Aug. 8. 1945



Curve III
Altitude Performance of B.M.W 028 Prop Drive GAS TURBINE
Prop. eff. 0.75
lb. fuel per hr./Eq. Shaft HP.
250 M.P.H
500 M.P.H
Note: Eq Shaft HP = Prop Shaft HP + Jet THP / Prop Eff.
Eq. Shaft Horse power
500 M.P.H
0 M.P.H
250 M.P.H
NACA std. Density lapse rate
0
10.000
20.000
30.000
40.000
Altitude - Feet.

Summary of Results (3)(Cont'd).

and III show the altitude performance of a jet (gas turbine) engine in comparison with that of a propeller drive gas turbine engine.

4. Cycle Analysis.

The work done by Dr. Munsberg of BMW to determine analytically the optimum compressor pressure ratio at different compressor and turbine efficiencies, airplane speeds, and turbine inlet temperatures for both the jet engine and the propeller drive gas turbine power plant agreed in general with American results. For example, a sea level static compressor pressure ration of approximately 3.5 with turbine inlet temperature at 1436° Fahrenheit, compressor and turbine efficiency of 80%, and airplane speed of 560 miles per hour was selected for the BMW 109-003 jet engine. A compressor pressure ration of 6.0 was selected for the propeller drive gas turbine designated as the BMW 109-028 corresponding to propeller efficiencies at 26,000 ft of 75% at 500 miles per hour, and 65 to 70% at 575 miles per hour.

5. Axial Compressors.

Table II shows some of the design parameters for German axial compressors. Dipl. Ing. Hermann Reuter of Brown-Boveri Cie (Mannheim) designed an axial flow compressor for the Junkers 109-004 jet engine which was designated as Hermso II. A limiting rotor Mach number of 0.68 based on relative air speed at the rotor tip of the first stage and standard sea level conditions was used in this design.

Another axial flow compressor designated as the Hermso IV was also designed by Mr. Reuter for the BMW 109-003 engine. In this design a limiting rotor Mach. number of 0.75 was used. A thickness-to-chord ratio of 9.5% was used at the tip and 12.5% at the root of the first stage of this design.

The first stage inducer of the Heinkel-Hirth 109-011 compressor was designed by Dr. von Ohain. It was designed for a rotor Mach Number of 0.95 and a tip thickness-to-chord ratio of 4.6%.

In most of the compressors investigated, an axial velocity of approximately one half the tangential tip speed of the first stage rotor was used for all stages except the last in which case 70-80% of this value was used.

Axial Compressors (5)(Cont'd).

The distribution of work between stages and the amount of work per stage were briefly as follows:

(a) The Hermso II was designed to attain a pressure coefficient.

$$\left(\frac{2g\ \mathrm{Had}}{V^2}\right) \text{ of } 0.48$$

(b) The Hermso III was designed to have a pressure coefficient of 0.43 for the last stage increasing linearly to 0.51 for the first stage. Thereby reducing the possibility of blade failures in the last stage due to stalling.

Mr. Reuter was of the opinion that in spite of the lower stage efficiency resulting from using reltively high pressure coefficients, the resulting reduction in wetted surface and number of stages required resulted in a net increase in efficiency. For example, he compared the performance of the Hermso II having eight stages with an obsolete Brown-Boveri design having fourteen stages. The eight stage unit developed the same pressure ratio at only a slightly lower efficiency, but was only half as long and two thirds the diameter.

Prof. Sorenson of MAN (Augsburg) indicated that they were using pressure coefficients of about 0.4 for all stages in their design.

Dr. von Ohain of Heinkel-Hirth based all of his designs of axial stages on a lift coefficient of 0.68.

Considerable time was spent in discussing the problem of influence of boundary layer on compressor size and rotor tip blade angles. In no case did any of the designers correct the root or tip diameters to obtain the required flow and no trouble had been apparently experienced in matching compressors and turbines. It had not been found necessary to correct for an increase in boundary layer thickness as the flow progressed toward the exhaust end of the compressor. Apparently there is no cumulative build up of boundary layer axially along the compressor walls. There was some concern expressed about the possibility of excessive boundary layer thickness occuring at the entrance of the compressor because of poor inlet duct design. In most cases axial compressor tests were conducted with rapidly

Axial Compressors (5)(Cont'd).

converging rounded approach orifices at the compressor inlet. Prof. Sorenson said that MAN design practice was to correct the outer 10% of the rotor blade for the velocity distribution estimated to exist in the boundary layer. He was not certain as to the exact methods used nor did he mention similar corrections being applied to the inner end of the stator blade.

The most controversial problem discussed concerned the distribution of static pressure rise between the stator and rotor. However, it was generally agreed that 50% of the static pressure rise per stage across the rotor was the theoretical optimum. Dr. Mansburg of BMW, on the basis of single stage test results obtained by Prof. Eckert at Braunschweig, preferred to design for 80% of the static pressure rise occuring across the rotor at the mean radius, 90% at the tip and 70% at the root. It was his opinion that the secondary losses resulting from the poor axial velocity distribution along the radius of the blade with 50% reaction blading more than compensated for the higher rotor tip leakage and boundary layer losses, if any, which might possibly result from the higher static pressure drop across the rotor. Prof. Sorenson's original axial compressor designs were unsatisfactory because of the high losses in the stator when the stator was designed on the basis of no static pressure increase across the rotor (Schicht design). The latest MAN designs are based on 80% of the static pressure rise occuring across the rotor at the mean radius. One reason given for this choice was that the low static pressure drop permitted the use of stator blades stamped from sheet metal.

The radial distribution of work along the rotor blade was in most cases on the basis of constant static pressure rise at all radial stations. That is, the rotor blades were twisted to give constant circulation which was obtained by varying the tangential component of the difference between the absolute leaving and entering velocities inversely as the radius. No correction to the blade angle was made for the increase in static pressure at the outer diameter caused by centrifugal forces.

The blade angles were in most cases calculated from air angles, using the theoretical deviation angles calculated by Fritz Weining (see Curve IV). It was found that single stage tests were necessary to determine the correct blade angles, which, especially in the case of the large stagger angles, differed considerably from the calculated angles.

Axial Compressors (5)(Cont'd).

The results shown in Curve V were widely used in estimating the effect of Reynolds number of efficiency. The minimum permissable chord was usually selected to give a Reynolds number of greater than 1×10^5.

The thickness distribution along the chord was usually bases on NACA airfoils as specified in NACA Report #460. Brown-Boveri (Reuter) used the method proposed by Weinig. The thickness chord ratio was decreased usually at the tip to as low as 5% to reduce shock losses and root stresses and was increased to 12-15% at the root to provide adequate strength and high natural blade frequency.

During the first stages of development of the BMW 003 engine numerous first stage rotor blade failures were exprienced as a result of the blades being excited by the interference effects of the front compressor bearing supports. The trouble was corrected by reducing the number of supports and increasing the blade thickness at the root.

The Hermso II compressor experienced several last stage rotor blade failures that were attributed to running the last stage stalled. In order to correct this condition, the Hermso III was designed to operate at a lower lift coefficient for the last stages.

6. Resume of Method Used by Herr Reuter of Brown-Boveri to Design Axial Flow Compressor.

(a) Assume a pressure coefficient (Ψ) for each stage.

$$X = \frac{2g\ H_{ad}}{V^2} = \frac{2g\ Jc_p\ T_{inlet}}{V^2}\left[\left(\frac{P_{disc.}}{P_{inlet}}\right)^{0.283} - 1\right]$$

Here g = Adiabatic Head (ft. lbs./lb. mass)
V = Tip speed (ft./sec).
T_{inlet} = Inlet air temperature (°R.).
J = Mechanical equivalent of heat = 778 ft.lbs./B.T.U.
c_p = Specific heat of fluid being compressed at constant pressure (B.T.U. per lb./degree R.).
$P_{disc.}$ = Discharge pressure.
P_{inlet} = Inlet pressure.

For the Hermso III design, a value of $\bar{\Psi} = 0.51$ was assumed for the

Resume of Method Used by Herr Reuter of Brown-Boveri to Design Axial Flow Compressor. (6)(Cont'd).

first stage and a value of $\Psi = 0.43$ was assumed for the last stage in order to increase the range of the last stage. A linear variation of Ψ between these two values was assumed for the design of the intermediate stages.

(b) 50% reaction at the mean radius was assumed for all stages
(c) Calculate the optimum inlet velocity ratio $\left(\frac{W_m}{u}\right)$ from:

$$N_{ad} = f(N_{pol}, K)$$

$$N_{pol} = \frac{1 - 2^E \frac{W_m}{u}}{1 + \frac{E}{2} \frac{W_m}{u}}$$

$N_{ad} = f(N_{pol}, K)$

Nad having been plotted as a function of Wm/u for various values of K. here

N_{ad} = adiabatic efficiency.
W_m = Axial inlet air velocity (ft./sec.).
u = Rotor tip speed (ft./sec.).
Npol = Polytropic efficiency. (Npol assumed to be 0.875 when $\frac{W_m}{u} = 0.5$ and $K = \infty$)

e = Drag coefficient/Lift coefficient (e assumed to be 0.066)
K = Polytropic exponent.

$$W_o = \frac{g\,H_{ad}}{u^2} = \frac{\Psi_{ad}}{2N_{ad}}$$

$$K = \frac{S\,W_o}{j}$$

where

S = No. of stages. J = Leaving loss coefficient.
J = 0 when the diffusion efficiency = 100% (based on tangential velocity)

(d) The Mach number (based on tangential velocity) for the first stage rotor tip was assumed to be 0.83 and, knowing the design angular velocity and the temperature of the air, the rotor O.D. was computed.
(e) The air angles at the mean rotor radius are determined from the formula.

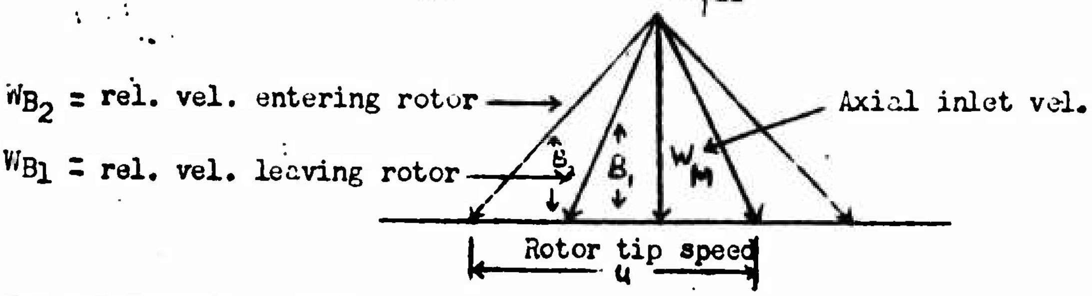
Resume of Method Used by Herr Reuter of Brown-Boveri to Design Axial Flow Compressor (6) (Cont'd).

$$\mathrm{Tan}B_2 = \frac{Wm/u}{1 + w_o/2}$$

$$\mathrm{Tan}B_1 = \frac{Wm/u}{1 - w_o/2}$$

For symmetrical blading only (50% reaction)

$$w_o = \frac{g\ Had}{u^2} = \frac{\Psi\ ad}{2\eta\ ad}$$



For unsymmetrical blading

$$(u_o)\ 50\%\ \text{reaction} = u_o \pm \left(1 - \frac{\rho_o}{100}\right)$$

+ for B_1

- for B_2

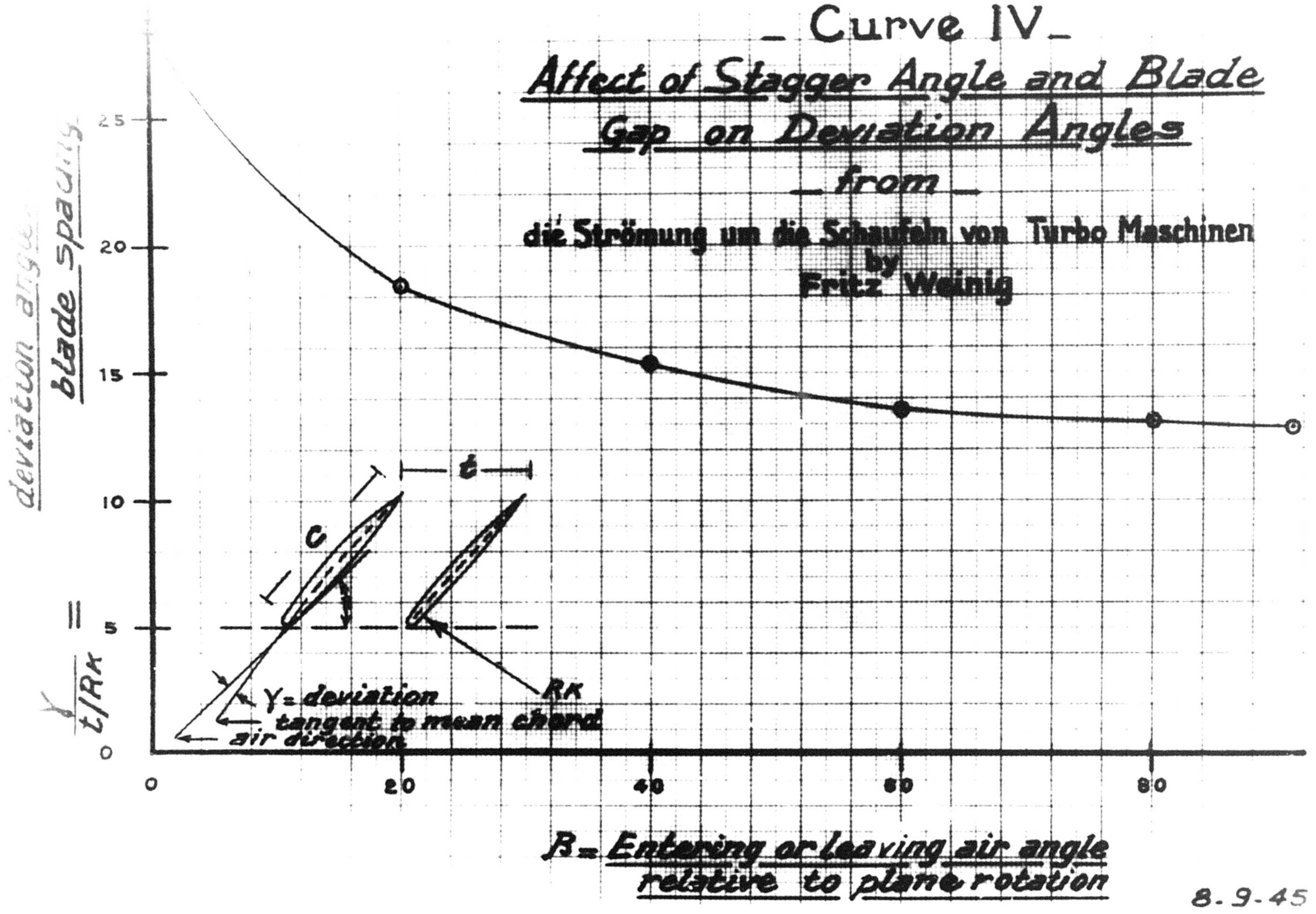
ρ_o = % reaction

(f) Calculate theoretical lift coefficient (C_a) and chord/gap ratio (c/t) from

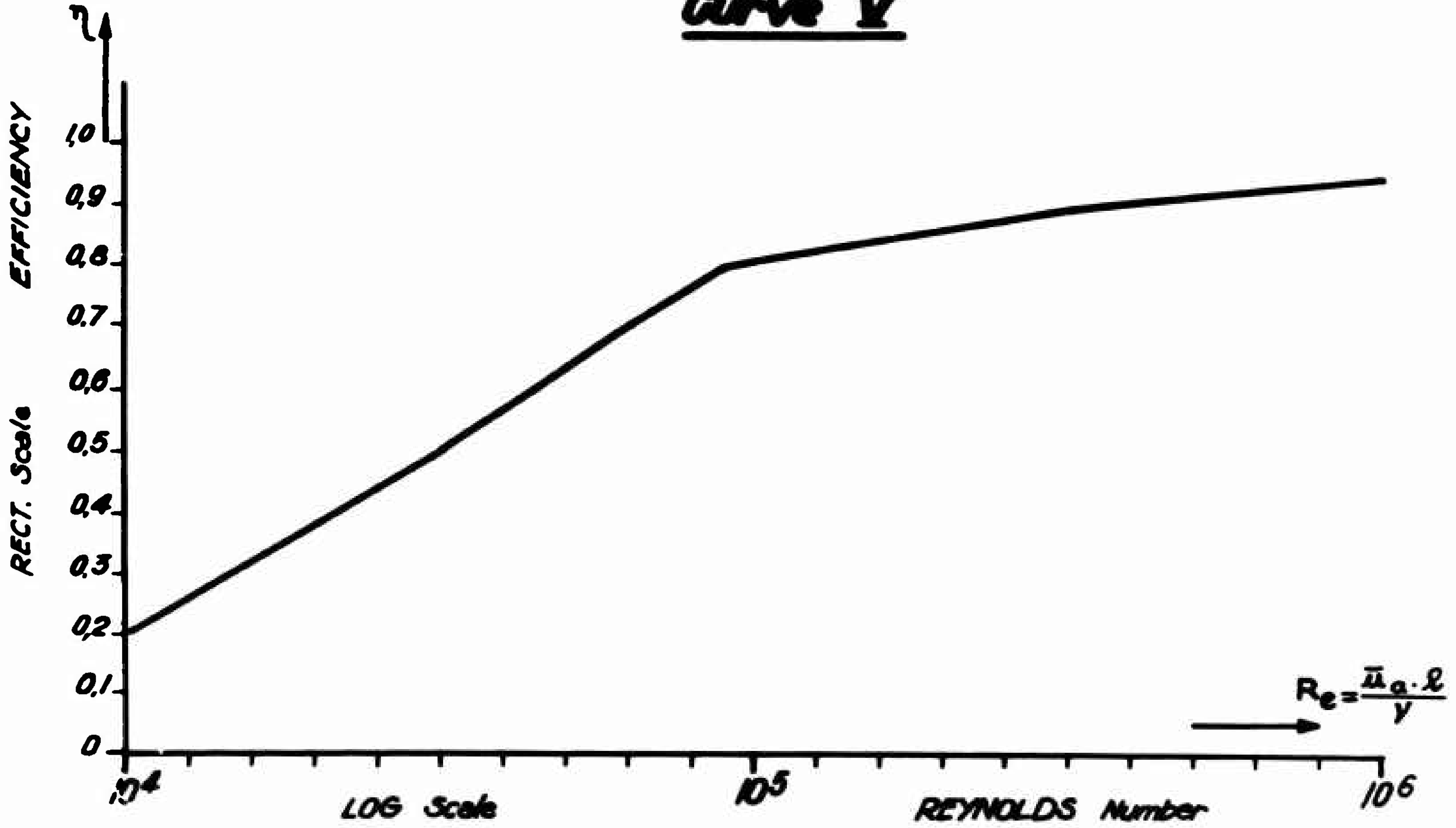
$$c/t \times C_a = \frac{4uo}{\sqrt{1 + \left(\frac{2\ Wm}{u}\right)^2}}$$

(for 50% reaction only)
use correction under para. (e) for other blading.

(g) Calculate $W_{B1}/u = \sqrt{\left(\frac{W_m}{u}\right)^2 + \left(\frac{1 + u_o}{2}\right)^2}$


_ Curve IV _
Affect of Stagger Angle and Blade
Gap on Deviation Angles
_ from _
die Strömung um die Schaufeln von Turbo Maschinen
by
Fritz Weinig
deviation angle / blade spacing
$\frac{\gamma}{t/R_K}$ =
0
5
10
15
20
25
20
40
60
80
C
t
R_K
γ = deviation
tangent to mean chord
air direction
β = Entering or leaving air angle relative to plane rotation
8-9-45

Curve V



Single Stage Tests

Resume of Method Used by Herr Reuter of Brown-Boveri to Design Axial Flow Compressor (6)(Cont'd).

$$\frac{W_{B_2}}{u} = \sqrt{\left(\frac{W_m}{u}\right)^2 + \left(\frac{1}{2} + uo\right)^2}$$

w_o is for 50% reaction only
Use correction under para. (e), but with signs reversed for unsymmetrical staging.

(h) Calculate blade angles from air angles using Weinig's Method of determining deviation angles (i.e. angle between tangent to mean chord and air velocity vector). See Curve IV.

(i) Tip and root air angles are calculated on the assumption of constant static pressure rise radially along the blade (i.e. constant circulation or u x Wu = constant). Assume $\frac{t}{l}$ = 0.8 at root.

(j) Minimum chord is limited by a Reynolds number of 1 x 10^5 (see Curve V) and by the natural frequency of rotor blade.

(k) Calculate natural frequency of rotor blade from:

$$f_e = \sqrt{\frac{1}{\pi^2}\frac{B_x^4}{L^4}\frac{TE}{FS} + 1.5\,N^2\frac{R}{L}}$$

(note: gives approximate value for straight blades)

f_e - frequency (cycles per second).
L - blade length (centimeters)
E - Mod. elasticity (kg/cm^2)
T - polor moment of inertia = $\int y^2\,dF$ (cm^4)
S - sp. wt/g ($kg\ sec^2/cm^4$)
N - angular velocity (rps)
R- - radius to C.G. (centimeters)
B_x - constant depending on order x
B_1 - 1.875 first order
B_2 - 4.694 second order
B_3 - 7.855 third order
B_4 - 10.996 fourth order

Resume of Method Used by Herr Reuter of Brown-Boveri to Design Axial Flow Compressor (6)(Cont'd).

The blade natural frequencies are selected to avoid the first and second harmonics of inlet strut excitiation frequencies and the first and second harmonics of the stator blade exitiation frequencies.

Fatigue failures of rotor blades have been experienced when compressors have been operated for short periods of time with some stages partially or completely stalled.

Axial velocity at the last stage is reduced 20-30% from value at inlet in order to reduce the length of diffusor between compressor and combustion chamber and to reduce tip losses due to low aspect ratio and leakage.

7. Gas turbines.

Table II on page 5 is a summary of some of the characteristics of several German gas turbines.

The turbine designs were greatly influenced by the lack of adequate high temperature alloys and the necessity of air cooling and turbine blades. This resulted in compromising the aerodynamic form of the blades to permit fabrication of the hallow blades. Although turbine inlet temperatures of 1382° F to 1436°F were employed, the turbine blade life was from 100 to 150 hours for the early models using 15% nickel, 15% chromium alloys and 20, to 80 hours for the later production models. The turbine blade life was quite inconsistant because of poor quality control of the steel at the mills, poor control of the turbine inlet temperature during acceleration, and inconsistencies in the fuel control regulations which would cause the turbine inlet temperature to exceed the maximum permissible valve.

Table III shows the decline in creep strength and high temperature alloy content as the final phase of the war was approached.

Gas Turbines (7)(Cont'd).

Table IV

High Temperature Turbine Blade and Disc Alloys.

Data obtained from Mr. W.H. Mayo, Carnegie Illinois Steel Corp.

Krupp Blade Alloys

Year	Alloy Name	Creep Strength lb/in	% Carb.	% Cr.	% Ni.	% Tit.	% Mo.	% tung.	% Van.	% Mn	% Nitrogen.	% Si.
1941	Trinidur	56,5000	0.15	15.0	30.0	1.6	-	-	-	-	-	-
1942	Vanidur	35,500	0.12	18.0	9.0	0.6	-	-	1.0	-	-	-
1943	Cromadur	42,500	0.10	12.0	-	-	-	-	0.6	18.0	0.20	-
1945	FKM-10	21,400	0.22	2.85	-	-	0.40	-	0.75	0.40	-	0.40

Krupp Disc Alloys.

									% Van.	% Mn.	% Nitrogen.	% Si.
1941	FKDM-10	21,400	0.20	2.8	-	-	0.4	0.40	0.75	-	-	-
1945	Crn 25	2,800	0.42	-	-	-	-	-	0.12	1.7	-	0.40

Gas Turbines (7)(Cont'd).

(a) Creep strength is stress at 1112°F at which creep rate is 0.001% per hour during the 25th to 35th hours of the test.
(b) Creep to rupture strength of Vanidur is 14,200 lb/in^2 at 1112°F and 28,400 lb/in^2 at 932°F. (5000) hours.

The practice of Brown-Boveri using Vanidur for both blades and discs was to design on the basis of 20,000 lb/in^2 root stress at 1112°F. turbine inlet temperature. The life was expected to exceed 2,000 hours and was estimated from test data which indicated the blade root temperature (solid blade) was 932° and the disc temperature (aircooled) was 752°F. The blade temperature was measured to be approximately 77°F, cooler than the gas stagnation temperature of 1112°F. at a turbine tip speed of 820 ft. sec. and an expansion ratio of approximately 1.8.

The effect of gap/chord ratio on efficiency has been experimentally investigated by several firms. A value of 0.7 to 0.8 is common practice. Brown-Boveri tests results indicated the optimum gap/chord ratio at the mean diameter was 0.4to 0.6 at a Parson's number of 2,000, and 1.0 to 1.2 at a Parson's number of 4,000.

According to Prof. Sorensen MAN was investigating ceramic turbine blades and disc materials. A 10" cast quartz integral blade and disc had been fabricated but was unsatisfactory because of its very law resistance to thermal shock. Carbonundum was though unsatisfactory because of the difficulties of machining. A porcelain consisting of silicon oxide, aluminum silicate and magnisium oxide (proportions unknown) and called "Ardastor" was considered as one of the more promising materials. It had a creep to rupture strength of 4,250 lb/in^2 at 1832°F and 70,000 hours. The ultimate strength cold was 6,000 to 16,000 lb/in^2. Hot fatigue had just been started but no results had been obtained.

Prof. Sorensen was of the opinion that it would be possible to develop ceramic turbine blades to run at 1832°F. and 500 ft/sec. tip speed in 5 years time.

8. Combustion.

Most of the following information was obtained from Messrs. Hagen and Bock of BMW. As a result of considerable testing on single cylindrical and segments of annular type burners as well as full scale units, it was

Combustion (8)(Cont'd).

concluded that it was necessary to prove a region of law velocity in which to establish the flame front, a certain minimum pressure loss through the combustion chamber was considered necessary to obtain a stable flame. Special provisions such as double fuel nozzles were considered necessary to insure adequate fuel penetration and dispersion at low values of full flow in order to prevent flame "blow-out" at high altitudes part load conditions.

Burner development was started in October 1939 on the BMW 003 engine, and occupied the full time of 6 engineers. Combustion chamber designs were evaluated on by the discharge gas temperature distribution by the calculated combustion efficiency by the total pressure drop across the chamber and by comparing the lean and rich limits of stability at different values of airflow. (See illustration on page 24).

Heat release rates of 2×10^7 B.T.U./hr/ft^3 have been experimentally obtained at 4 atmospheres pressure. However, in the case of the BMW 109-003 engine at sea-level and rated speed the heat release rate was 1.1×10^7 B.T.U./hr/ft^3 @ 4 atmospheres; and 0.84×10^7 B.T.U./hr/ft^3 in the case of the BMW 109-018 engine @ 6 atmospheres. Although no testing had been done to determine scale effects it was believed the larger combustion chambers required more fuel nozzles (16 for BMW 003 and 24 3.9 for BMW 018).

The total pressure loss across the BMW 109-003 combustion chamber at sealevel and rated speed conditions was 2.25 lb/in^2 or approximately 4.3% of the compressor discharge pressure. The Heinkel-Hirth 109-011 engine combustion chamber "blow-out" conditions as determined in an altitude chamber were as follows:

Alt. Ft.	R.P.M.	Ram mi/h
42,600	8,000	560
16,400	5,000	560
36,000	35,000	300
16,400	2,500	300

BMW, Junkers, and Heinkel-Hirth were all providing double fuel nozzles to correct this trouble which was caused to a large extent by poor fuel dispersion and penetration at the low fuel flows and pressures required at high altitudes and part load. In addition Heinkel-Hirth were providing for additional primary air turbulence near the fuel nozzles.

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Combustion (8)(Cont'd).

The BMW 109-003 combustion chamber details are shown below: (See page 24).

Ignition is produced by two spark plugs mounted in the plane of the primary air cone. These spark plugs are insperative except during starting.

Two special fuel jets are provided for starting with gasoline, while six additional jets supply the heavier fuel to the combustion chamber while the engine is running.

The primary air cone is designed with a large angle of divergence (40°) to cause flow separation and reverse flow, thus stabilizing the falme front. Original designs using perforated ceramic plates were discarded because the fuel condensed on the plates. The secondary air chamber length of the BMW 109-003 was increased 8 inches to improve the combustion efficiency and discharge temperature distribution.

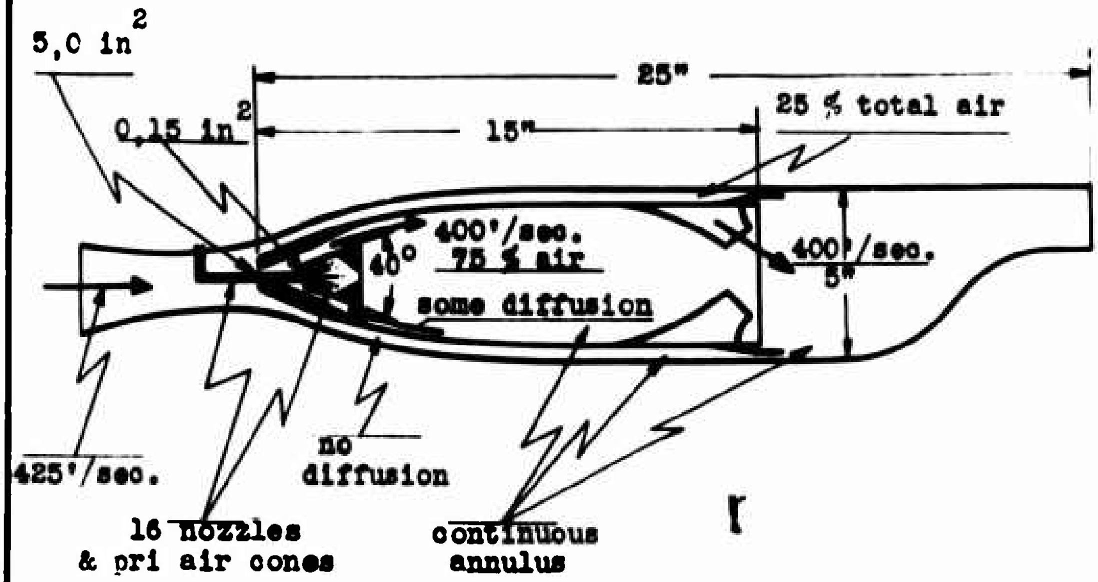
Experiments injecting fuel in the first stage rotor were unseccessful because the liquid fuel was thrown to the outer wall and would not mix with the air; besides burning was experienced in the tail pipe.

Burning of fuel in the tail pipe of jet engines was not possible below an exhaust temperature of 1660°F. However if the fuel was injected upstream and a primary air cone (turbalence cone) was installed, combustion was possible above 1120°F but the pressure loss in the thrust nozzle was excessive.

The project was dropped because of tail pipe burning and flame instability. The theoretical increase in thrust, of 330 lbs at 1832°F was never realized.

9. Control System.

The Junkers type of fuel control system was used on all the production jet engines. The fuel control system consisted of an idle or slow speed control which was effective below a throttle position corresponding to 5,000 rpm at sea-level. It consisted of a throttle controlled fuel pressure difference. Because it incorporated no altitude density compensation device the turbine speed increased with altitude for a given throttle position.



5.0 in^2
0.15 in^2
25"
15"
25 % total air
400'/sec.
75 % air
40°
some diffusion
400'/sec.
5"
425'/sec.
no
diffusion
16 nozzles
& pri air cones
continuous
annulus

[redacted] UNCLASSIFIED

Control System (9)(Cont'd).

The main system consisted of governor controlled fuel valve which maintained the turbine speed constant between 6000 and 9500 rpm for a given throttle setting and at any altitude or airplane speed. The idle system was rendered inoperative by a linkage connected to the throttle system.

The exhaust jet area was controlled by a servo-mechanism as a calculated function of indicated air speed at high turbine speeds and as a function of both indicated air speed and throttle position at low turbine speeds. The jet area is a maximum at starting and idling conditions and is a minimum at full throttle and high airplane speeds, (see Junkers 004 report no. 19).

The original BMW 109-003 fuel control system for gasoline consisted of 30,000 rpm centrifugal fuel pump having a 1.8" impeller which used the radial component of the centrifugal pressure (proportional to rpm) to directly control the fuel flow by means of a bypass valve at the discharge from the pump. The scheme was discarded when it became necessary to use Diesel and crude oils for fuel because the density and velocity tolerances which influence the radial pressure was so great that it caused prohibitive variation in turbine inlet temperature when these fuels were used.

The BMW proposal for a propellor drive gas turbine consisted of a constant speed propellor control coordinated with a variable turbine inlet temperature control which was biased by altitude and ram pressure. Both turbine speed and inlet temperature would be reduced at port load, the variation of power due to changing turbine inlet temperature being 2/3 of the total effect.

10. Starting.

It was estimated that a 50 h/p starter (2 cycle, air-cooled gasoline engine) would be required for the BMW 109-018 and 109-028 power plants. The starting speed would be about 20% of maximum speed for the BMW 109-018 and somewhere less in the case of the BMW 109-028. It was planned to use a propeller pitch for minimum torque for starting. No study had been made of the problem of providing minimum pitch stops for take-off.

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