NOMENCLATURE

A	Area	m^2
A	Combustor constant	Joule K/Kg
B	Combustor constant	Joule K/Kg
C	Combustor constant	Joule K/Kg
C	Absolute velocity of flow	m/s
C_p	Specific heat at constant pressure	Joule/Kg. K
C_v	Specific heat at constant volume	Joule/Kg. K
C_x	Axial velocity of flow	m/s
D	Combustor constant	Joule/Kg
D	Diameter	m
ECV	Effective calorific value	Joule/Kg
f_a	Fuel to air ratio	
G	Torque	N.m
h	Blade height (ROGV blade row)	m
h	Specific enthalpy	Joule/Kg
Δh	Change in specific enthalpy	Joule/Kg
I	Rotor moment of inertia	$Kg.m^2$
I_f	Dynamometer field current (non-dimensional)	
J	Mechanical equivalent of heat = 1	
K	Design speed ratio = $(N_c/N_{GG})_{D.P.}$	
$K_{C.C}$	Combustor pressure loss factor	

1

52806

0-315-04912-X

NAME OF AUTHOR/NOM DE L'AUTEUR

M. R. S. OKELAH

TITLE OF THESIS/TITRE DE LA THÈSE

OPTIMIZATION OF THE DESIGN PARAMETERS
OF A CO-TURBOSHAFT GAS TURBINE ENGINE
AS A HEAVY EQUIPMENT POWER PLANT

UNIVERSITY/UNIVERSITÉ

CARLETON UNIVERSITY

DEGREE FOR WHICH THESIS WAS PRESENTED/

GRADE POUR LEQUEL CETTE THÈSE FUT PRÉSENTÉE

DOCTOR OF PHILOSOPHY

YEAR THIS DEGREE CONFERRED/ANNÉE D'OBTENTION DE CE GRADE

1981

NAME OF SUPERVISOR/NOM DU DIRECTEUR DE THÈSE

D. A. J. MILLAR

Permission is hereby granted to the NATIONAL LIBRARY OF
CANADA to microfilm this thesis and to lend or sell copies
of the film.

L'autorisation est, par la présente, accordée à la BIBLIOTHÈ-
QUE NATIONALE DU CANADA de microfilmer cette thèse et
de prêter ou de vendre des exemplaires du film.

The author reserves other publication rights, and neither the
thesis nor extensive extracts from it may be printed or other-
wise reproduced without the author's written permission.

L'auteur se réserve les autres droits de publication; ni la
thèse ni de longs extraits de celle-ci ne doivent être imprimés
ou autrement reproduits sans l'autorisation écrite de l'auteur.

DATED/DATE

22 JAN / 81

SIGNED/SIGNÉ

[Signature]
SUPERVISOR GRAD STUDIES AND RESEARCH
DEPT. MECH & AERO ENGINEERING

PERMANENT ADDRESS/RÉSIDENCE FIXE

c/o Mech & Aero Eng.
Carleton University
Ottawa, Canada

National Library of Canada
Collections Development Branch

Canadian Theses on
Microfiche Service

Bibliothèque nationale du Canada
Direction du développement des collections

Service des thèses canadiennes
sur microfiche

NOTICE

The quality of this microfiche is heavily dependent upon the quality of the original thesis submitted for microfilming. Every effort has been made to ensure the highest quality of reproduction possible.

If pages are missing, contact the university which granted the degree.

Some pages may have indistinct print especially if the original pages were typed with a poor typewriter ribbon or if the university sent us a poor photocopy.

Previously copyrighted materials (journal articles, published tests, etc.) are not filmed.

Reproduction in full or in part of this film is governed by the Canadian Copyright Act, R.S.C. 1970, c. C-30. Please read the authorization forms which accompany this thesis.

**THIS DISSERTATION
HAS BEEN MICROFILMED
EXACTLY AS RECEIVED**

AVIS

La qualité de cette microfiche dépend grandement de la qualité de la thèse soumise au microfilmage. Nous avons tout fait pour assurer une qualité supérieure de reproduction.

S'il manque des pages, veuillez communiquer avec l'université qui a conféré le grade.

La qualité d'impression de certaines pages peut laisser à désirer, surtout si les pages originales ont été dactylographiées à l'aide d'un ruban usé ou si l'université nous a fait parvenir une photocopie de mauvaise qualité.

Les documents qui font déjà l'objet d'un droit d'auteur (articles de revue, examens publiés, etc.) ne sont pas microfilmés.

La reproduction, même partielle, de ce microfilm est soumise à la Loi canadienne sur le droit d'auteur, SRC 1970, c. C-30. Veuillez prendre connaissance des formules d'autorisation qui accompagnent cette thèse.

**LA THÈSE A ÉTÉ
MICROFILMÉE TELLE QUE
NOUS L'AVONS REÇUE**

OPTIMIZATION OF THE DESIGN PARAMETERS OF A CO-TURBOSHAFT
GAS TURBINE ENGINE AS A HEAVY EQUIPMENT POWER PLANT

by

Mohamed Rafat Sayed Okelah

B.Sc., M.Sc. (Mech. Eng.)

A Thesis

Submitted to the School of Graduate Studies in
partial fulfillment of the requirements for the
degree

Doctor of Philosophy

Faculty of Engineering

Carleton University

Ottawa, Ontario

December 1980

The undersigned hereby recommend to the Faculty of
Graduate Studies, Carleton University, acceptance of this thesis,
" Optimization of the Design Parameters of a Co-Turboshaft Gas
Turbine Engine as a Heavy Equipment Power Plant", submitted by
Mohamed Rafat Sayed Okelah, in partial fulfillment of the
requirements for the degree of Doctor of Philosophy in
Mechanical Engineering.

Thesis Supervisor: D.A.J. Millar

Chairman, Mechanical and
Aeronautical Engineering

* * * In The Name of GOD

Most Gracious

Most Merciful * * *

"AND MY SUCCESS CAN ONLY COME FROM GOD.

IN HIM I TRUST, AND

UNTO HIM I LOOK"

ACKNOWLEDGEMENTS

It gives me great pleasure to acknowledge and thank my research supervisor, Dr. D.A.J. Millar for his invaluable advice throughout the period of this study. I would like to express my very deep gratitude to him for the continuous encouragement and support he rendered to me.

Special thanks are also due to Mr. M.S. Chappell my research supervisor. His experience in the field of gas turbines along with the advices which he offered me were of great value, particularly in the early stages of my research.

This work would not have been possible without the provision of the facilities of the National Research Council of Canada (NRC). I would like to thank all of the staff of the Analysis Laboratory of NRC for their sincere cooperation during the course of this study. In particular, I thank Mr. P. Hamil, Head of the Analysis Laboratory, for his support of the project, and Dr. R.E. Gagné, for his valuable assistance in dealing with problems encountered when using the hybrid computer.

I would like to express my gratitude to Dr. B.D. MacIsaac of GasTOPS Company, for his valuable assistance in building and developing the hybrid computer model.

I thank Dr. H.I.H. Saravanamuttoo and the staff of the Mechanical Engineering Department of Carleton University for their many helpful advices throughout the period of this study. Thanks are also due to

the staff of the Engine Laboratory of the NRC for their assistance and encouragement.

The typing of this thesis was most efficiently done by Mrs. F. Zaydan, for which I thank her.

ABSTRACT

The novel feature of the co-turboshaft engine is a rotating compressor casing, which co-rotates with the gas generator spool, hence, the name "co-turboshaft." This casing derives power from the compressor air stream and this power is transmitted to the output shaft, and is combined with the power turbine output in an epicyclic gear box. As load increases and output shaft speed decreases, the effective gas generator speed increases, with no increase in rotor speed, and the power output rises. This produces steeper torque-speed characteristics than conventional two-shaft gas turbine engines.

A study of the effects of the design parameters on the performance of the co-turboshaft engine is carried out. The basis of selection of these parameters to achieve the best performance, both mechanically and thermodynamically, is presented.

The thesis examines the potential of the co-turboshaft engine for use in vehicular applications. The basic characteristics of such an engine are being presented for assessment as a vehicle power plant. The co-turboshaft engine is compared to a conventional power plant, to assess the comparative performance of each. A comparison of the performance of a specific co-turboshaft engine and of a conventional engine, provided with sophisticated control systems, is also carried out.

A simulation program, utilizing well developed hybrid computer

techniques, has been built for a co-turboshaft engine. It provides a very flexible computer model that can be used and adapted to a wide variety of further studies. It became thus a simple task to extend the study to a regenerative co-turboshaft engine and to incorporate components of different characteristics into the model.

Results of the hybrid computer model, along with the analytical studies presented in this thesis, have provided a vast amount of insight into the "co-turboshaft" engine.

TABLE OF CONTENTS

	Page
ACKNOWLEDGEMENTS	iii
ABSTRACT	v
TABLE OF CONTENTS	vii
LIST OF FIGURES	xii
NOMENCLATURE	xvi
CHAPTER 1 INTRODUCTION	1
1.1 Gas Turbines in Vehicular Applications	2
1.1.1 The Free Shaft Gas Turbine Engine	2
1.1.2 Gas Turbine Engine as a Potential Power Source for Automotive Vehicles	3
1.2 Basic Features of a Co-Turboshaft Engine	4
1.3 Rationale for Co-Turboshaft Engine Selection	5
1.4 The Co-Turboshaft Engine - Comparison with Other Concepts	7
1.5 Reasons for Building Simulation Programs	8
1.5.1 Steady State Performance Predictions	8
1.5.2 Transient Performance Predictions	9
1.6 Objectives and Contribution of this thesis	9
CHAPTER 2 LITERATURE SURVEY	11
2.1 The Rotating Stator Concept	11
2.2 Alternatives to Improve Engine Performance	13

CHAPTER 2		Page
2.2.1	The GT-309 Power Transfer System	14
2.2.2	The Swedish KTT Engine	15
2.2.3	The ERDA/Chrysler Upgraded Gas Turbine Engine	16
2.3	The Simulation Program	19
2.3.1	Simulation Programs in Engine Development	19
2.3.2	Choice of Computer	20
2.3.3	Techniques Developed for Hybrid Computer Simulation Programs	22
CHAPTER 3	SYSTEM'S DESCRIPTION	23
3.1	Description of Co-Turboshaft Engine	23
3.2	Principles of Engine Operation	25
3.3	Gear Box Description	28
3.4	Matching Component Characteristics	36
3.5	Engine Modules	39
3.6	Information Flow Diagram	41
CHAPTER 4	THE MATHEMATICAL MODEL OF A CO-TURBOSHAFT ENGINE	43
4.1	Simplifying Assumptions	44
4.2	Mathematical Description of Engine Modules	46
4.2.1	The Inlet	46
4.2.2	The Inlet Volume	47
4.2.3	The Inlet Guide Vaves (IGV)	48
4.2.4	The Compressor	49

CHAPTER 4		Page
4.2.5	The Rotating Outlet Guide Vanes (ROGV)	52
4.2.6	The Intercompressor Volume (ROGV Volume)	53
4.2.7	The Combustor	54
4.2.8	The Gas Generator Turbine	56
4.2.9	The Interturbine Volume	60
4.2.10	The Power Turbine	61
4.2.11	The Exhaust Duct	65
4.2.12	Pressure Loss Due to Flow Swirl	67
4.2.13	The Gas Generator Rotor Dynamics	72
4.2.14	The Load	73
4.2.15	The Compressor Rotating Casing	74
4.2.16	The Output Shaft Rotor Dynamics	75
4.2.17	The Heat Exchanger	77
4.3	Scaling of Variables	80
CHAPTER 5	HYBRID COMPUTER MODELLING	81
5.1	The Operation of a Hybrid Computer - Compatibility of a Serial-Parallel Combination	81
5.2	The Program Structure	83
5.2.1	Work Division Between Analog and Digital Parts of the Hybrid Computer	83
5.2.2	Digital Program	84
5.2.2.1	Initialization	84
5.2.2.2	High Speed Loop	84
5.2.2.3	Off-Line Operations	85

CHAPTER 5		Page
5.2.3	Analog Program	85
5.3	Program Capabilities	88
5.3.1	Transformation into Conventional Engine	88
5.3.2	Dynamic Analysis	89
5.3.3	Program Extention	89
5.3.4	Engines of Different Designs	89
5.4	Model Implementation	90
5.4.1	Problem Control	90
5.4.2	Accuracy of Model Implementation	93
5.5	Model Verification and Validation	94
CHAPTER 6	PERFORMANCE CHARACTERISTICS OF THE BASIC CO- TURBOSHAFT ENGINE	96
6.1	Introduction	96
6.2	Cycle Analysis	97
6.3	Selection of Design Point Compressor Pressure Ratio and Maximum Cycle Temperature	108
6.4	Gear Box Speed Ratio K	111
6.5	Effect of Location of Design Point on Power Turbine Map	112
6.6	Effect of Location of Design Point on Compressor Map	118
6.7	Effect of Shape of the Compressor Map on Co-Turboshaft Engine Performance	125
6.8	Performance Comparison Between Co-Turboshaft Engine and Conventional Engine	129

CHAPTER 6		Page
6.9	Temperature Characteristics of Co-Turboshaft Engine	132
6.10	Power Turbine Flow Characteristics for a Constant TIT Operation	139
CHAPTER 7	THE CO-TURBOSHAFT ENGINE - FURTHER ASPECTS OF PERFORMANCE	147
7.1	Introduction	147
7.2	The Regenerative Co-Turboshaft Engine	147
7.3	Effect of Including Flow Swirl Pressure Loss on the Performance of a Co-Turboshaft Engine	150
7.4	Comparison Between the Performance of a Co-Turboshaft Engine and of a Conventional Engine with Sophisticated Control	153
7.5	Part-Load Performance of a Co-Turboshaft Engine	156
CHAPTER 8	CONCLUSIONS	166
REFERENCES		170
BIBLIOGRAPHY		174
APPENDICES		180
A	MATHEMATICAL DERIVATIONS	A-1
B	SCALED EQUATIONS	B-1
C	DESIGN POINT CALCULATIONS - VERIFICATION TASK	C-1
D	POWERFUL TECHNIQUES IN SUPPORT OF HYBRID COMPUTER MODELLING	D-1
E	ANALOG CIRCUITS	E-1
F	LIST OF DIGITAL PROGRAMS	F-1

LIST OF FIGURES

Figure	Page
1. Schematic Cross Section of a Co-Turboshaft Engine	24
2. Effect of Output Shaftspeed on Effective Compressor Speed of a Co-Turboshaft Engine as Function of Design Casing Rotor Speed Ratio	28
3. Schematic Representation of an Epicyclic Gear Box	29
4. Modified Design of Gear Box	33
5. Co-Turboshaft Engine Information Flow Diagram	42
6. Engine Layout	44
7. (a) Inlet Representation	47
(b) Inlet Information Flow Diagram	47
8. Inlet Volume Information Flow	48
9. Overall Performance Characteristics of NASA Eight-Stage Axial Flow Compressor	49
10. Compressor Information Flow Diagram	51
11. Compressor Representation	51
12. ROGV Model	53
13. ROGV Volume Information Flow	54
14. Combustion Chamber Model	55
15. Gas Generator Turbine Flow Characteristics	56
16. Gas Generator Turbine Efficiency Characteristics	57
17. Efficiency Characteristics of Gas Generator Turbine	57
18. Gas Generator Turbine Information Flow	59

Figure	Page
19. Gas Generator Turbine Representation	59
20. Interturbine Volume Module Representation	60
21. Power Turbine Flow Characteristics	61
22. Power Turbine Efficiency Characteristics	62
23. Power Turbine Information Flow	63
24. Power Turbine Representation	64
25. Exhaust Information Flow	65
26. Exhaust Pressure Loss	66
27. Conventional Mode-Power Turbine Velocity Triangles	67
28. Co-Turboshaft Engine-Power Turbine Velocity Triangle	69
29. Exhaust System Representation	72
30. Gas Generator Rotor Dynamics Model	73
31. Electric Dynamometer Characteristics	73
32. Dynamometer Model	74
33. Torques Contribution to Output Shaft Dynamics	75
34. Output Shaft Rotor Dynamics Model	76
35. Heat Exchanger Model	78
36. Heat Exchanger Information Flow Diagram	79
37. Digital Program Structure	86
38. Off-Line Service Operations	87
39. General Overview of Computer Model	91
40. Conceptual View of System Modelling Procedure	94

Figure	Page
41. Basic Co-Turboshaft Engine System	97
42. Co-Turboshaft Engine Operating Lines On Compressor Map	108
43. Efficiency of Basic Co-Turboshaft Engine	110
44. Specific Work Output of Basic Co-Turboshaft Engine	110
45. Effect of Design Speed Ratio (K) on Turbines Pressure Ratio (r_1 & r_2), I	113
46. Effect of Design Speed Ratio (K) on Turbines Pressure Ratio (r_1 & r_2), II	113
47. Location of Design Point on Power Turbine Map	114
48. Effect of Location of Design Point on Power Turbine Map on Co-Turboshaft Engine Performance, I	116
49. Effect of Location of Design Point on Power Turbine Map on Co-Turboshaft Engine Performance, II	117
50. Location of Design Point on Compressor Map	118
51. Comparison Between Different Design Point Locations	120
52. Effect of Design Point Location on Compressor Map on Co-Turboshaft Engine Performance, I	121
53. Effect of Design Point Location on Compressor Map on Co-Turboshaft Engine Performance, II	122
54. Effect of Design Point Compressor Pressure Ratio on Performance of Co-Turboshaft Engine, I	123
55. Effect of Design Point Compressor Pressure Ratio on Performance of Co-Turboshaft Engine, II	124
56. Orenda II Compressor Performance Map	126
57. Co-Turboshaft Engine Performance with Orenda Compressor, I	127
58. Co-Turboshaft Engine Performance with Orenda Compressor, II	128

Figure	Page
59. Performance Comparison Between Co-Turboshaft Engine and Conventional Engine, I	130
60. Performance Comparison Between Co-Turboshaft Engine and Conventional Engine, II	131
61. Operation of Turbines in Series	133
62. Co-Turboshaft Engine Performance - Constant Turbine Inlet Temperature, I	145
63. Co-Turboshaft Engine Performance - Constant Turbine Inlet Temperature, II	146
64. Performance of Regenerative Co-Turboshaft Engine, I	148
65. Performance of Regenerative Co-Turboshaft Engine, II	149
66. Effect of Including Flow Swirl Pressure Loss on the Performance of Co-Turboshaft Engine, I	151
67. Effect of Including Flow Swirl Pressure Loss on the Performance of Co-Turboshaft Engine, II	152
68. Performance of a Conventional Engine with Sophisticated Control, I	154
69. Performance of a Conventional Engine with Sophisticated Control, II	155
70. Co-Turboshaft Engine - Part-Load Performance, I	158
71. Conventional Engine - Part-Load Performance, I	159
72. Co-Turboshaft Engine - Part-Load Performance, II	161
73. Conventional Engine - Part-Load Performance, II	162
74. Co-Turboshaft Engine - Part-Load Performance, III	163
75. Conventional Engine - Part-Load Performance, III	164

NOMENCLATURE

A	Area	m^2
A	Combustor constant	Joule K/Kg
B	Combustor constant	Joule K/Kg
C	Combustor constant	Joule K/Kg
C	Absolute velocity of flow	m/s
C_p	Specific heat at constant pressure	Joule/Kg. K
C_v	Specific heat at constant volume	Joule/Kg. K
C_x	Axial velocity of flow	m/s
D	Combustor constant	Joule/Kg
D	Diameter	m
ECV	Effective calorific value	Joule/Kg
f_a	Fuel to air ratio	
G	Torque	N.m
h	Blade height (ROGV blade row)	m
h	Specific enthalpy	Joule/Kg
Δh	Change in specific enthalpy	Joule/Kg
I	Rotor moment of inertia	$Kg.m^2$
I_f	Dynamometer field current (non-dimensional)	
J	Mechanical equivalent of heat = 1	
K	Design speed ratio = $(N_c/N_{GG})_{D.P.}$	
$K_{C.C}$	Combustor pressure loss factor	

K_{exh}	Exhaust pressure loss factor	
K_{in}	Inlet pressure loss factor	
K_S	Effectiveness of deswirl exhaust stators	
K_θ	Power turbine blade velocity	m/s
K_1	Gear box ratio = N_{PT}/N_0	
K_2	Gear box ratio = N_C/N_0	
m	Engine mass flow	Kg
N_C	Speed of compressor casing	r.p.m.
N_{GG}	Speed of gas generator rotor	r.p.m.
N_{PT}	Speed of power turbine	r.p.m.
N_{rel}	Speed of compressor core relative to its casing	r.p.m.
N_0	Speed of output shaft	r.p.m.
P	Pressure	N/m^2
P	Power	Joule/s (Watt)
PR	Pressure ratio of compressor	
Q	Non-dimensional mass flow rate = $w\sqrt{\theta}/\delta$ (compressor) = $w\sqrt{T}/P$ (turbines)	
Q_{in}	Heat input	Joule
R	Gas constant = 287	Joule/Kg. K
r_1	Pressure ratio of gas generator turbine	
r_2	Pressure ratio of power turbine	
SFC	Specific fuel consumption = w_f/P_0	Kg/KJoule
T	Temperature	K
TIT	Turbine inlet temperature (maximum cycle temperature)	K

ΔT	Temperature change	K
t	Time	sec
U	Blade velocity	m/s
V	Volume	m^3
W	Work	Joule
w	Mass flow rate	Kg/s
w_f	Fuel mass flow rate	Kg/s
Z	Number of teeth of a gear	
α	Nozzle angle	
β	Rotor blade angle	
γ	Ratio of specific heats = C_p/C_v	
δ	Referred pressure = P/P_{ref}	
ϵ	Effectiveness of heat exchanger	
η	Efficiency	
θ	Referred temperature = T/T_{ref}	
ρ	Density	Kg/m^3
v	Ratio of the temperature equivalent of mean blade speed to turbine temperature drop = $U/\sqrt{2\Delta h}$	

Subscripts

a	Air; ambient conditions
C	Compressor casing
comp	Compressor
D.P.	Design point conditions
exh	Exhaust
g	Gases
GG	Gas generator
GGT	Gas generator turbine
H.Ex	Heat exchanger
HPT	High pressure turbine
IGV	Inlet guide vanes
in	Inlet
is	Isentropic
LPT	Low pressure turbine
mech	Mechanical
0	} Output
Out	
O.D.	Off-design conditions
rel	Compressor core relative to casing
ROGV	Rotating outlet guide vanes
1	} Engine stations (Figure 6)
2	
3	
4	
5	
6	
7	

CHAPTER 1

INTRODUCTION

The development of the gas turbine began with the military need for a power plant with a high power-to-weight ratio. The turbojet was able to supply this need, although at a cost in fuel consumption and efficiency which initially prohibited its use in any other application. Subsequent improvements in design, as well as in materials, allowed much higher pressure ratios and cycle temperatures to be achieved. The overall thermal efficiency has been so improved that the gas turbine has virtually taken over the aeronautical field, and is growing in popularity in other applications as well.

Thermodynamically, the gas turbine approximates the Brayton cycle. The processes involved are: an adiabatic compression, a heat addition with a small loss of total pressure, and an adiabatic expansion in a turbine. The remainder of the expansion can be effected either in a free power turbine, in which case shaft work is produced, or in a propelling nozzle which produces thrust. Thermodynamically, these two expansions are quite similar. These processes are continuous and since the spent gas is expelled to the atmosphere, the cycle is said to be "open". [1] *

* Number in square brackets designate REFERENCES at the end of the thesis.

1.1 Gas Turbines in Vehicular Applications

"The gas turbine engine has made tremendous progress since some of the earliest models were tested in vehicles less than 12 years ago. Those early engines were noisy, were characterized by large volumes of hot exhaust, and had a voracious appetite for fuel. Today's gas turbine engine is extremely quiet in operation, discharges its exhaust at 500°F or less, and has economy approaching that of the diesel engine. Much of this progress can be attributed to the development of the highly efficient rotating regenerator." [2]* Development continues until today with emphasis on engines for powering heavy vehicles.

1.1.1 The Free Shaft Gas Turbine Engine

The free shaft gas turbine engine is a promising candidate for powering heavy vehicles and construction equipment. One of the most attractive features of the free turbine engine is its torque-speed characteristic, which demonstrates rising torque as output speed drops. Because this behaviour is also characteristic of many such loads, the free turbine engine requires less torque amplification (and hence fewer gear ratios) in the transmission, than for reciprocating engines.[5]

The negative slope of the torque-speed curve arises from the fact that the "gas power" made available to the free turbine by the gas generator section of the engine is substantially constant at a fixed throttle setting. For modest power turbine speed changes, the turbine efficiency, which relates output power to gas power, is also essentially constant, so that the output power will also be fixed, and torque will

*Turunen, W.A. and Gollman, J.S., 1965.

vary inversely with speed, for small speed variations. For larger speed variations, power turbine efficiency falls off as the operating conditions depart substantially from the design values, and this drop in efficiency will modify the inverse relationship.

Obviously, this characteristic can be changed by modulating the gas generator output as shaft speed varies. However, the gas generator is normally speed-limited at full load conditions and additional gas power is not available to provide a burst of power and torque as output shaft speed drops under high load conditions.

1.1.2 Gas Turbine Engine as a Potential Power Source for Automotive Vehicles

"Development of the gas turbine engine as a potential power source for automotive vehicles began in the early 1950's. By 1976, three manufacturers were actively pursuing the development centered on the heavy-duty truck application because it represented the largest potential market in this engine power range and because the heavy-duty truck power requirements corresponded to gas turbine engine characteristics." [3] The gas turbine engine has shown multiple advantages and demonstrated success in heavy trucks. Under the Urban Mass Transportation Administration Transbus Program, a gas turbine engine was tested in one of the three different prototype transit coach designs. During the Program, this engine exhibited the following operational advantages over the conventional diesel engine in transit coach application:

- Reduction of installed weight and volume
- Elimination of cooling radiator, fan, and attendant piping
- Cleaner exhaust emissions
- Vibration-free operation
- Reduced oil consumption
- Improved serviceability
- Improved reliability
- Improved vehicle performance for given power rating
- Greater engine braking capability
- Superior cold weather starting. [3]

On the other hand one of the major disadvantages of using gas turbines in automotive applications is its unfavourable part load specific fuel consumption.

1.2 Basic Features of a Co-Turboshaft Engine

Among the gas turbine schemes proposed for vehicular application is the Co-Turboshaft Engine. This engine has a unique feature - the engine compressor casing co-rotates with the compressor rotor but at a lower speed, and is geared to the output shaft, together with the power turbine.

The co-turboshaft engine automatically increases gas power output from the gas generator, as the output shaft speed decreases,

without any increase in gas generator spool speed, that is, with a normal speed governor control system. The torque-speed product increases as speed falls, so that the actual output torque rises steeply as shaft speed drops. Although the gas generator spool speed is fixed, the air flow rate will vary in much the same way as it would for power modulation of a conventional engine.

1.3 Rationale for Co-Turboshaft Engine Selection

The objective of rotating the compressor casing is to extract power from the air stream within the compressor, thereby extracting power from the high pressure turbine. The high pressure turbine is thus assumed to provide the compressor air flow with power proportional to its enthalpy rise and the rotating casing with power proportional to its enthalpy drop. Simple conservation of angular momentum applied to the core compressor of this arrangement shows that, with axial flow at inlet and outlet, the torque transmitted to the rotating stator casing must equal the torque provided by the high pressure turbine via the high speed rotor shaft. Thus the fraction of the high pressure turbine power which reappears at the casing shaft is equal to the ratio of casing speed to high pressure turbine speed. This is the basic expression for the energy split between the rotating stator casing and the core compressor. It represents the major difficulty in applying the co-rotating stator concept to a turbofan engine* with the fan being mounted on, or driven by,

* This application was suggested by Rolls Royce (UK) and that company and the National Research Council of Canada carried out an experimental study of such a compressor.

the rotating stator. At high bypass ratios, where the need to use a co-rotating compressor concept had originated in order to optimize fan and turbine speeds, typical values for energy split implies that the ratio of fan to compressor speed should be relatively high, which is physically impractical.

When applied to a turboshaft engine, the energy split need not be specified, so that the speed ratio can be selected by other criteria. Hence a co-turboshaft engine provides flexibility for the designer in selecting casing to rotor speed ratio, to optimize the performance to suit the application. [4]

In a co-turboshaft engine arrangement, both the compressor stator casing and the free power turbine contribute, via a gear box, to the output of the engine. The gear box maintains a constant proportionality between output shaft speed, casing speed, and power turbine speed. With the gas generator rotor speed governed at 100%, decreasing output shaft speed will, therefore, simultaneously decrease both power turbine speed and compressor casing speed. The latter change will increase the relative speed of the core compressor and hence will increase the air mass flow rate through the machine, thereby increasing both power turbine and casing output. The result will be evidenced by a significantly increased torque multiplication at full throttle part speed, which may offer considerable advantages in vehicular applications. [4]

Thus the co-turboshaft engine appears to be a promising candidate as a prime mover for heavy equipment such as off-road transports, and construction and mining equipment. [5]

1.4 The Co-Turboshaft Engine - Comparison with Other Concepts

"The co-turboshaft configuration is not the only one which can achieve a steep torque speed characteristic. In fact, any configuration which produces extra power as shaft speed drops will have the same characteristics. If the gas generator speed is programmed to increase with decreasing output shaft speed, using a suitable controller, a similar effect will be achieved. However, a steeply increasing turbine inlet temperature, in combination with the increasing stresses in the rotating components, is likely to be a serious disadvantage. The temperature can be maintained constant by varying the turbine nozzle area, of course, but the rotor stresses remain as a serious difficulty. Aerodynamic overspeed at low ambient temperatures, with possible compressor surge, may also occur.

"The mass flow rate of a constant speed gas generator can be increased by using variable inlet guide vanes and stators, to achieve increased power under high load conditions, and the resulting turbine inlet temperature rise can be controlled by variable geometry turbine stators. The range of flow rates achievable by simple compressor variable geometry is limited, however, so that the power amplification attainable will also be limited." [5]

Other types of alternatives which depend mainly on the concept of power transfer, either through the engine or away from the engine, are presented briefly in chapter 2.

1.5 Reasons for Building Simulation Programs

Methods of predicting the behaviour of an engine would be clearly useful, and could lead to savings in development costs and time [1]. The need for such methods is more than elsewhere when dealing with engines of new designs or unknown performance such as the co-turboshaft engine. Building a simulation program to predict engine performance has proven to be one of the most successful methods in this respect.

In this thesis it was decided to develop a program which is capable of carrying out both transient and steady state performance analyses even though only steady state performance results are presented. Detailed analysis of the dynamic performance of the engine will be considered in future work.

1.5.1 Steady State Performance Predictions

Previous analysis has shown that the performance of the proposed co-turboshaft engine is affected by the engine design parameters, such as the ratios of component speeds, the cycle pressure and temperature.

A computer based simulation program is required to investigate the effect of these parameters on the engine performance. It provides the capability of changing the engine design parameters and consequently predicting the effects of changes in major control variables such as fuel flow and dynamometer torque as a function of output shaft load and inlet conditions, over the whole of the operating range, for a variety of engine designs.

Another reason for using a simulation program is that it has the ability to present, for comparison, a complete and immediate picture of both co-turboshaft engine and conventional engine performance, both engines having the same component characteristics.

1.5.2 Transient Performance Prediction

If the transient behaviour prediction could be made from the steady state component data, the value of the simulation would be greatly increased. The advantages of this kind of simulation are many. The controls engineer will require an accurate and flexible model as an essential tool in the design of the control system. This is particularly significant for small engines, for which the control system is a substantial part of the overall engine cost. The results of tests that are dangerous or impossible to perform on a prototype could be inferred from a model that performed well over the safe operating range. Design proposals could be evaluated in rapid succession, without actually building and testing prototypes. In addition, the understanding of the operation of an engine can be greatly enhanced by the careful study of its simulation. In this manner, much experience can be gained on new types of engines, rendering simulation a useful operator training tool. [1]

1.6 Objectives and Contribution of this Thesis

One of the objectives of this study is to define the design point characteristics of the proposed co-turboshaft engine which gives the best performance both mechanically and thermodynamically. Thus we can define a suitable mechanical layout for a prototype power plant. This plant is then compared to a conventional power plant to assess the comparative performance of each. An interesting comparison between the performance of a specific co-turboshaft engine and that of a conventional

engine, which is provided with sophisticated control systems, has also been carried out.

The thesis also examines the potential of the co-turboshaft engine for use in vehicular application. It provides possible users with the basic characteristics of such engines, for assessment as vehicle power plant. It has been made available, the performance characteristics of the co-turboshaft engine in the cases of: overloading, overspeeding, part-load, and constant turbine inlet temperature.

A simulation program, utilizing well developed hybrid computer techniques, has been built for a co-turboshaft engine. It provides a very flexible computer model that can be used and adapted to a wide variety of further studies. It became thus, a simple task to extend the study to a regenerative co-turboshaft engine and to incorporate components of different characteristics into the model.

Results of hybrid computer model plus analytical study presented in this thesis have provided a large amount of insight into the concept of a co-turboshaft engine.

CHAPTER 2

LITERATURE SURVEY

2.1 The Rotating Stator Concept

The concept of a co-rotating compressor casing was described by Howell in 1944 [6], but does not seem to have been implemented until the early seventies.

The concept of co-rotating blade rows within an axial compressor was conceived primarily as a means of overcoming the inherent rotational speed mismatch between a fan and its driving turbine in moderate to high bypass ratio turbofan engines. In essence, the arrangement serves as a torque converter or "aerodynamic gear box" which will permit both the fan and the turbine to operate closer to their individual optimum speeds.

A cooperative project was undertaken by the National Research Council of Canada and Rolls Royce (Canada) Ltd. to examine this concept of compressor spooling, called the "Fanstat" by Rolls Royce. In this collaborative program the National Research Council of Canada modified a three stage transonic research compressor provided by Rolls Royce to investigate the performance of the basic compressor and of the modified compressor having a rotational stator casing. The test facility, procedure and results are described by Chappell [7] and by Chappell, Millar and Swiderski [8], [9], [10]. Conclusions of this program were

presented in reference [9] as follows:

1. The Fanstat compressor arrangement, comprising an axial compressor with co-rotating stators, is aerodynamically feasible.
2. Aerodynamic performance of the co-rotating compressor, in the useful operating range, was not vastly different from that of the conventional compressor at the same relative (rotor to stator) speeds, after accounting for the difference in overall pressure ratio caused by the rotating inlet guide vanes and the stationary outlet guide vanes.
3. The flow range of the Fanstat compressor is considerably greater than that of a conventional compressor at low speeds, but is significantly reduced at high speeds.
4. The overall efficiency of the Fanstat compressor is not significantly different from the conventional compressor in the useful operating range and at rotor/casing speed ratios near the design value.
5. Surge characteristics of the Fanstat compressor were somewhat unusual at low speeds where stable operation was achieved at flows significantly below the maximum-pressure-ratio flow. At higher speeds, above about 85% of design relative speed, the surge points of the Fanstat compressor were clearly discernible and the surge characteristics were similar to those of the conventional compressor.
6. The co-rotating compressor, as tested using techniques derived from a previous study on a hybrid-computer model, did not exhibit any serious idiosyncracies during run-up or surge-exploration transients. However, inertias of the test rig drive and power absorption components may not be typical of potential applications, and independent control of

casing torque may not be available in certain engine embodiments of the concept.

7. Mechanical difficulties encountered on the particular Fanstat compressor tested did not appear characteristic of the configurational concept and seem soluble within current design technology.

8. With the exception of surge, the steady-state and transient behaviour of the Fanstat compressor conformed reasonably well to the predictions made from the hybrid-computer simulation, thus confirming the latter as a useful and powerful investigatory tool.

9. Further examination of the power transfer potential of the rotating stator case is required, especially relative to modern turbofan engine requirements as well as to other potential applications.

The significance of these facts lies in the demonstration of the physical and aerodynamic feasibility of the co-rotating compressor concept. The fact that this full size machine without either mechanical or aerodynamic development was able to run and operate successfully indicates that the concept is quite practical.

2.2 Alternatives to Improve Engine Performance

In what follows, some of the recent alternative proposals for development of gas turbine engine performance are presented. Proposals are presented mainly through quotations from researchers publications.

2.2.1 The GT-309 Power Transfer System

This power plant was proposed by GM Allison division in 1965, and is described in reference [2]. The following is extracted from that reference:

The most significant new feature of the GT-309 engine is its Power Transfer system. The primary function of Power Transfer is to improve the part-load fuel economy of the engine. In addition it provides or improves several important engine features.

Normally, the inlet temperature of a two shaft gas turbine engine decreases as its output is decreased. Since thermal efficiency is a function of the peak cycle temperature at any operating condition, the part-load fuel consumption of the two shaft gas turbine engine can be reduced if the cycle temperature is raised.

Several methods of raising the part-load cycle temperature were considered in the preliminary design phase of the GT-309 engine. In addition to Power Transfer, these included variable power turbine nozzles and supercharged turbine engine cycles. The final choice of Power Transfer was supported by several advantages, some of them unique.

Basically, Power Transfer is a hydromechanical system of transferring controlled amounts of power between the independent turbine shafts. One set of plates of a hydraulically actuated clutch is driven through suitable gearing by the gasifier turbine. The other set of clutch plates is geared to the power turbine. The clutch torque can be regulated by means of the hydraulic pressure. The gear ratios are chosen so that the

power turbine-driven plates turn slower than the gasifier-driven plates during all normal engine operating conditions.

Application of a slight amount of pressure to the hydraulic clutch cylinder results in a torque reaction between the clutch elements turning at different speeds. The torque on the faster turning gasifier turbine-connected elements acts to slow this component. The gasifier governor, however, increases the fuel flow to maintain constant speed. The result is an increase in turbine inlet temperature as well as a slight increase in engine output. By operating the engine at the maximum allowable temperature limit throughout its speed/power range, significant reductions in fuel consumption can be realized.

2.2.2 The Swedish KTT Engine

The specific feature of this engine is the use of a third stage turbine connected to both gas generator turbine and power turbine through a planetary gear system. Reference [11] has introduced the concept as follows:

"A conventional automotive turbine engine, such as the Chrysler gas turbine [PS, Sept., '73] consists of two stages. Kronogard has added a third stage. Through an ingenious system of planetary gears, the third-stage turbine supplies auxiliary power to the turbines in the first two stages. In addition, the third-stage turbine drives the accessories (fuel pump, oil pump, alternator, etc.) with energy that would otherwise be wasted.

For acceleration from low compressor speed, the power surplus from the third turbine is thrown toward the compressor shaft to help speed its rotation. And under high gas-flow conditions, the third turbine adds to the power available at the output shaft.

The planetary differential that distributes the power flow from the auxiliary turbine also serves to adjust the speed and torque relationship between the compressor shaft and the output shaft so that the output always corresponds to the vehicle requirement. This is done completely automatically and with such a fine degree of response that the need for a conventional gearbox - or even a torque converter - is eliminated." [11]

2.2.3 The ERDA/Chrysler Upgraded Gas Turbine Engine

Chrysler Corporation has been conducting an automotive gas turbine program for the Division of Transportation of the Energy Research and Development Administration since 1972. The final task of this program was to design, build, and demonstrate an Upgraded Engine. The following has been extracted from reference.[12].

"The two features of this engine which are new to the automotive gas turbine are the augmentation concept and the free rotor arrangement.

Power augmentation results from both water injection and variable inlet guide vanes (VIGV) at the compressor inlet. Water injection is simple to implement, requiring no more than a water fogging nozzle at the air intake bowl and a water tank pressurized from the compressor discharge. However, a water flow rate slightly larger than the fuel flow rate is needed in order to get a modest 10% boost in maximum

engine power. Therefore, to keep the water tank a reasonable size, the injection system is only activated at maximum power which is normally a small fraction of total operating time. Use of water injection only at ambients greater than 15 °C (60 °F) serves to conserve water, compensates for the gas turbine's characteristic temperature dependency, and obviates freezing protection. Commercial grade distilled water is used to minimize compressor performance deterioration from scale deposits. In addition, the forward, or inducer, section of the compressor is made from steel to minimize the effect of water droplet erosion.

Augmentation with VIGV is accomplished by imparting swirl to the air at the compressor inlet opposite to the direction of rotation. In this way, the change in angular momentum is increased for the fluid passing through the compressor. This results in increased pressure ratio and flow as though the compressor were running faster. Swirl in the plus or co-rotational direction is used at low vehicle speeds to regulate engine power while still maintaining engine temperature and efficiency. This works by reducing pressure rise and flow as though the compressor were running slower. A practical VIGV implementation consists of 13 uncambered airfoils (NACA 65-0010) with the front 45% non-moveable. This is capable of deflecting the incoming air through a range of angles from -30° to +60°.

The second novel feature of the Upgraded engine is the use of a free rotor. A "free rotor" configuration is a design in which all power takeoffs have been removed from the gas generator rotor. Thus all engine auxiliaries such as the oil pump, fuel pump, atomizing

air pump and regenerator are driven from the power turbine instead of the gas generator. Such a free rotor design has a number of advantages:

- a) It is conducive to the use of a gas bearing. By removing a fraction of the bearing load due to power take-off, the bearing requirements are eased.
- b) It is quieter. Previous power take-off drives have been the source of identifiable gear noise.
- c) It is potentially less costly. The high speed gear system of a geared rotor is a significant cost item.
- d) Improved vehicle driveability. Although tests and calculations show small improvements in rotor response, experience with a prototype system suggests improved vehicle "feel". In particular, the combination of a free rotor and gas bearing will have superior cold starting performance.

At the same time there are a number of disadvantages and consequences for the Upgraded Engine as a result of the free rotor choice:

- a) The accessory drive is complex. Some means had to be provided to supply lubricating oil to the gas generator and atomizing air to the burner during the start sequence prior to the power turbine assuming the drive. It is presumed that technology advances will eventually eliminate the need for these services during the initial start period and that this disadvantage will not exist for future free rotor engines.
- b) The fuel control philosophy is changed. Previous fuel controls have

been based on a mechanical governor driven from the gas generator. Removal of the accessory drive results in an electronic speed sensor as input to the fuel control. This is a consequence of the free rotor choice, but is not a disadvantage for the Upgraded Engine since other considerations resulted in the choice of an electronic fuel control. Under other circumstances, the lack of a convenient mechanical drive could be a disadvantage as a control limitation.

- c) The thermal loading distribution is altered. A gas bearing being cooled by compressor discharge air at 232°C (450°F) maximum as compared to the oil bearing running at a maximum oil temperature of 121°C (250°F) will cause higher temperatures in the shaft, bearing and support structure during both running and soakback conditions.

During the initial stages of the design layout, these factors were recognized and weighed as to relative importance. On balance the consensus was that the advantages outweighed the disadvantages for the purposes of the Upgraded Engine.

2.3 The Simulation Program

2.3.1 Simulation Programs in Engine Development

The prime requirement for any dynamic simulation is that it should accurately represent the behaviour of the engine over the range of operation of interest, for any form of transient disturbance. The

benefits of simulation are most likely to be realized if the simulation model is available early in the development program, and preferably at the design stage; this means that the simulation must be based on readily available data which can be updated as required during the development program. [13]

These requirements can all be met by approaching the problem from the viewpoint of the engineering thermodynamicist, making use of the normal compressor and turbine characteristics, estimates of which are usually available at the design stage for prediction of off-design performance.

The basis of the thermodynamic approach to engine dynamics is the calculation of the compressor and turbine torques from data obtained by matching of the component characteristics, followed by integration of the net torque applied to the rotor system to get the rotor speed. The matching of flow between the engine components determines the pressure ratio across each, and the resulting temperature changes determine the work transfers and torques. [13]

2.3.2 Computer Type Selection

Computation of the engine performance and dynamic response can be carried out using either digital, analog, or hybrid computers.

The digital computer is a serial machine, hence the computation time is dependent upon both the complexity of the engine to be studied and the efficacy of the numerical integration procedure. It is unlikely that

real time operation will be possible but perhaps the most significant disadvantage is the inherent remoteness of large computer systems, making it very difficult for the simulation engineer to experiment on-line with alternative philosophies and obtain a real "feel" for the engine being simulated.

Analog computers are parallel devices, thus the computation time is independent of the engine complexity. The analog computer cannot compete in accuracy with the digital computer, but the accuracy of modern analogs is quite adequate for engineering simulation purposes. A disadvantage of the analog is the need for a substantial amount of non-linear function generation particularly for the generation of functions of two variables for component characteristics. The major advantage of the analog method of simulation is that an engineer can sit at the console and operate the simulation just as he would operate an engine on the test bed; this is especially valuable for complex engines.

A balanced hybrid system combines an analog computer with a general purpose digital computer through an interface which provides for interchange of both logical and variable signals between the constituent machines. The integrating amplifiers of the analog provide full dynamic capability removing the need for time consuming integration routines on the digital computer. The hybrid computer offers the computing speed of the analog computer with the accuracy and flexibility of the digital. It should be noted, however, that a requirement of extensive digital computation may prevent the use of real time simulation, and every effort must be made to minimize digital computing. Component characteristics

can be stored digitally and can be readily updated as the development program proceeds. Control schedules can be readily changed to meet specific requirements and different control philosophies can be examined.

2.3.3 Techniques Developed for Hybrid Computer Modelling

A group of well developed techniques of hybrid computer modelling has been utilized in the co-turboshaft engine modelling. Most of these techniques were developed by the staff of the Analysis Laboratory of the National Research Council of Canada. Examples of these techniques are: The Function Generation Routines [14], The BUGOFF routine [15], The Hytran Operations Interpreter [16], The CIRBUG [17] and the DCASET [18]. A summary of each of these routines is given in Appendix (D).

It should be appreciated that, without the aid of such useful techniques, the task of building a hybrid computer model of a complex engine, such as the co-turboshaft engine, would have been extremely time-consuming and tedious.

CHAPTER 3

SYSTEM'S DESCRIPTION

3.1 Description of Co-Turboshaft Engine

The novel feature of the co-turboshaft engine is a rotating compressor casing, which co-rotates with the gas generator spool, hence the name "co-turboshaft". This casing derives power from the compressor air stream and this power is transmitted to the output shaft, and is combined with the power turbine output in an epicyclic gear box. As a result, the output shaft speed and the speeds of the power turbine and the compressor casing are directly proportional to one another, so that a reduction in shaft speed not only reduces power turbine speed, but also reduces the speed of the compressor casing, resulting in an increase in the compressor effective speed when the rotor speed is constant.

A schematic representation of the engine is shown below, "Figure 1 .

Figure 1. Schematic Cross Section of a Co-turboshaft Engine

The engine modelled, in the main part of this study, consists of an eight-stage compressor of design pressure ratio of 6.2, having characteristics shown in reference [21]; a combustion chamber having a constant 5 percent loss of total pressure; a two-stage gas generator turbine operating choked, and a single stage power turbine, also choked. A constant 2 and 5 percent pressure loss were assumed for inlet and exhaust ducts, respectively. The design efficiencies of the turbines were 88 and 87 percent, respectively; the compressor design point was taken on the 85 percent speed line to provide surge and overspeed margin.

Mechanical efficiency was taken as 97 percent for the power shaft and 99 percent for the gas generator spool.

3.2 Principles of Engine Operation

Aerodynamically, the compressor performance is determined by the speed difference between the rotor and casing, the "relative speed" N_{rel} , where

$$N_{rel} = N_{GG} - N_C$$

where N_{GG} is the speed of the gas generator rotor and N_C is the speed of the compressor casing. N_{rel} is equivalent to the compressor speed in a conventional engine, so that as N_{rel} increases, so will the engine airflow and power. Since the casing is geared to the output shaft, a reduction in the output shaft speed, as a result of an increase in load, will cause both the power turbine speed and the compressor casing speed N_C to drop. Hence, the relative speed will increase, if the gas generator spool speed N_{GG} is fixed (by the governor, say), and the gas generator output will increase correspondingly, producing the steep torque rise mentioned previously.

While the torque characteristic is the most attractive feature of this engine, the power transfer from the gas generator turbine, through the compressor casing to the output gearbox, provides a degree of design flexibility which can also be an advantage. The casing torque is equal to the gas generator shaft torque, for axial flow into and out of the compressor, so that the rotating casing power is related to the gas generator turbine power by:

$$P_C = P_{GG} \frac{N_C}{N_{GG}}$$

However, the gas generator turbine output power provides the power to compress the air, plus the casing power, or $P_{GG} = P_{comp} + P_C$. Thus the casing power can be shown to be related to the compressor power by

$$P_C = \frac{N_C}{N_{rel}} P_{comp}$$

This power is developed by the high speed gas generator turbine, but is delivered to the output gearbox at a lower speed, using the compressor as an "aerodynamic gearbox". The rotating stator acts as a power transfer device to permit the gas generator turbine to assist the power turbine to drive the output shaft.

In a conventional free power turbine engine, the split between compressor power and output power usually is such that the turbine staging is inconvenient. If both gas generator and power turbine are single stages, the gas generator turbine will be over loaded. If a two-stage turbine is used for the gas generator, it tends to be lightly loaded (and hence efficient) so that the power turbine becomes heavily loaded. By being able to transfer power from the gas generator to the output, this imbalance can be rectified, and the overall turbine efficiency can be improved.

It is apparent that the feasibility of this concept is crucially dependent on the behaviour of the compressor, both aerodynamically and mechanically, when the casing is co-rotated.

From the description of the engine, it is apparent that the increase in relative speed caused by a given percentage reduction in shaft speed is directly related to the design selection of the casing-to-rotor speed ratio, if compressor rotor (gas generator) speed is fixed. That is,

$$\frac{N_{rel}}{(N_{rel})_{D.P.}} = \frac{N_{GG} - N_C}{(N_{GG} - N_C)_{D.P.}} = \frac{1 - \frac{N_C}{N_{C_{D.P.}}} \frac{N_{C_{D.P.}}}{N_{GG}}}{1 - \frac{N_C}{N_{C_{D.P.}}} \frac{N_{C_{D.P.}}}{N_{GG}}}$$

Calling the ratio $\frac{N_{C_{D.P.}}}{N_{GG}} = K$, the design speed ratio and noting that the gearing constrains $N_C/N_{C_{D.P.}}$ to equal $N_O/N_{O_{D.P.}}$ the output shaft speed ratio, then:

$$\frac{N_{rel}}{(N_{rel})_{D.P.}} = \frac{1 - K \frac{N_O}{N_{O_{D.P.}}}}{1 - K}$$

The larger K is chosen to be, the more sensitive is the compressor relative speed to the output shaft speed, as is shown in Figure 2. For example, if $K = 0.1$, the relative speed will increase by only 11 percent for a 100 percent reduction, in output shaft speed, but if $K = 0.2$, then stalling the output shaft will cause the compressor relative speed to increase by 25 percent, with no actual increase in rotor shaft speed.

Figure 2. Effect of Output Shaftspeed on Effective Compressor Speed of a Co-turboshaft Engine as Function of Design Casing Rotor Speed Ratio

3.3 Gear Box Description

An objective of the gear box of a co-turboshaft engine is to provide a value of the design speed ratio K , somewhere in the range from 0.1 to 0.4. A schematic representation of such gear box is shown in Figure 3. This design of gear box provides a value of K only greater than 0.25, as will be shown from the following analysis. However, a simple modification of the planetary arrangement of Figure 3, as shown in Figure 4, produces the required value of K . But for simplicity of the analysis let us consider first the epicyclic gear box of Figure 3.

A provision in the gear box design is made to allow the transformation from the co-turboshaft mode (rotating casing) into the conventional mode (stationary casing). This is done by tightening the brake B2 of Figure 3 and releasing B1. In such case there is no power transfer

from the engine to the output shaft through the casing. This is particularly important in the case of unloaded output shaft (minimum torque condition) where it is required to isolate the gas generator rotor from the output shaft.

Figure 3. Schematic Representation of an Epicyclic Gear Box

Considering Gear Train (1-2-3-4),

$$\frac{N_1 - N_4}{N_3 - N_4} = \frac{-Z_3}{Z_1}$$

where Z is the number of teeth of a gear

$$N_{PT} = N_0 \left(1 + \frac{Z_3}{Z_1} \right) = K_1 \cdot N_0 \quad \dots(3.1)$$

where:

$$K_1 = 1 + \frac{Z_3}{Z_1}$$

Considering Gear Train (6-5-3-4),

$$\frac{N_6 - N_4}{N_3 - N_4} = - \frac{Z_3}{Z_2} \cdot \frac{Z_5}{Z_6}$$

$$N_C = N_0 \left(1 + \frac{Z_3}{Z_2} \cdot \frac{Z_5}{Z_6} \right) = K_2 \cdot N_0 \quad \dots (3.2)$$

where

$$K_2 = 1 + \frac{Z_3}{Z_2} \cdot \frac{Z_5}{Z_6}$$

The compressor relative speed $N_{rel} = N_{GG} - N_C$, substituting from (3.2),

$$N_{rel} = N_{GG} - K_2 \cdot N_0$$

or in a non-dimensional form:

$$\begin{aligned} \frac{N_{rel}}{(N_{rel})_{D.P.}} &= \frac{N_{GG} - K_2 \cdot N_0}{(N_{GG})_{D.P.} - K_2 \cdot (N_0)_{D.P.}} \\ &= \frac{\frac{N_{GG}}{(N_0)_{D.P.}} - K_2 \frac{N_0}{(N_0)_{D.P.}}}{\frac{(N_{GG})_{D.P.}}{(N_0)_{D.P.}} - K_2} \end{aligned}$$

since

$$K = \left(\frac{N_C}{N_{GG}} \right)_{D.P.} = \frac{K_2}{(N_{GG}/N_0)_{D.P.}}, \text{ then:}$$

$$\frac{N_{rel}}{(N_{rel})_{D.P.}} = \frac{\frac{N_{GG}}{(N_{GG})_{D.P.}} - K \cdot \frac{N_0}{(N_0)_{D.P.}}}{1 - K} \quad \dots (3.3)$$

Equation (3.3) is the general equation relating variation in compressor relative speed to variation of both output shaft speed and gas generator rotor speed. Assuming that N_{GG} has a fixed value throughout the considered range of operation, then equation (3.3) reduces to:

$$\frac{N_{rel}}{(N_{rel})_{D.P.}} = \frac{1 - K N_0/N_0 \text{ D.P.}}{1 - K} \quad \dots(3.4)$$

The value of K , as shown above, depends on two factors:

- a- Gear box design factor K_2
- b- Selected design point speed ratio (N_{GG}/N_0)

Starting with $(N_{GG}/N_0)_{D.P.}$ N_{GG} will be determined by compressor-specific speed, compressor pressure ratio and mass flow. N_0 will be determined by vehicle requirements. N_0 is also related to power turbine speed N_{PT} through the relationship: $N_{PT} = K_1 \cdot N_0$. A reasonable selection of K_1 lies between 3 and 6, where the gear ratio Z_3/Z_1 lies between 2 and 5. Thus:

$$N_{PT} = (3 \rightarrow 6)N_0$$

Assuming for instance that N_{GG} is about equal to N_{PT} , which is generally the case, then:

$$N_{GG} = (3 \rightarrow 6)N_0$$

Thus the Design Point speed ratio N_{GG}/N_0 lies in the range from 3 to 6. Now consider the factor K_2 , where

$$K_2 = 1 + \frac{Z_3}{Z_2} \cdot \frac{Z_5}{Z_6}$$

since the objective of this gear box design is to provide $K = 0.1 \rightarrow 0.4$
then:

$$\frac{K_2}{3 \rightarrow 6} = 0.1 \rightarrow 0.4$$

Thus the range of values for K_2 is from 0.3 to 2.4

- Considering for example the case of $N_{GG} = 6N_0$, i.e., $K_1 = 6$:

$$\frac{Z_3}{Z_1} = 5, \quad R_3 = 5R_1 = R_1 + 2R_2$$

$$R_2 = 2R_1; \quad Z_2 = 2Z_1$$

Thus:

$$K_2 = 1 + \frac{Z_3}{Z_2} \cdot \frac{Z_5}{Z_6} = 1 + 2.5\left(\frac{Z_5}{Z_6}\right)$$

Keeping in mind that gear box design practice recommends that:

$$\left(\frac{Z_5}{Z_6}\right)_{\text{minimum}} = 0.2 \text{ for reasonable stressing of}$$

the pinion gear 5.

Thus Minimum $K_2 = 1 + 0.2 \times 2.5 = 1.5$. Consequently:

$$K_{\min} = \frac{1.5}{6} = 0.25$$

for any other value of $\frac{Z_5}{Z_6} > 0.2$, $K > 0.25$

- Considering the opposite case of $N_{GG} = 3N_0$, i.e., $K_1 = 3$, $\frac{Z_3}{Z_1} = 2$

$$R_3 = 2R_1 = R_1 + 2R_2, \quad R_2 = 0.5R_1;$$

$$Z_2 = 0.5Z_1, \quad \text{thus:}$$

$$K_2 = 1 + \frac{Z_3}{Z_2} \cdot \frac{Z_5}{Z_6} = 1 + 4 \times \frac{Z_5}{Z_6}$$

$$K_{2 \text{ minimum}} = 1 + 4 \times 0.2 = 1.8, \quad K_{\text{min}} = \frac{1.8}{6} = 0.3$$

Thus $K > 0.3$

The conclusion of the previous example is that the simplified design, shown in Figure 3, is only suitable for values of K in the range $K \geq 0.25$. A modification is needed to allow for smaller values of K . One of the possible modifications is shown in Figure 4.

Figure 4. Modified Design of Gear Box

The addition of gear 7 to the planet 2-5 has another advantage besides providing the required values of K . It allows the output shaft speed N_0 to be much smaller than the turbine shaft speed, which is suitable to vehicular applications.

In this case:

$$\frac{N_1 - N_4}{N_3 - N_4} = - \frac{Z_3}{Z_2} \cdot \frac{Z_7}{Z_1}$$

$$N_{PT} = (1 + \frac{Z_3}{Z_2} \cdot \frac{Z_7}{Z_1}) N_0 = K_1 \cdot N_0$$

where:

$$K_1 = 1 + \frac{Z_3}{Z_2} \cdot \frac{Z_7}{Z_1}$$

To achieve a large reduction ratio of 20 from the power turbine shaft speed to the output shaft speed, K_1 should be equal to 20. This results in:

$$\frac{Z_3}{Z_2} \cdot \frac{Z_7}{Z_1} = 19$$

If the two gear ratios are equal, then:

$$\frac{Z_3}{Z_2} = \frac{Z_7}{Z_1} = 4.36$$

Such gear ratios lie within the acceptable range of gear design.

A reduction ratio of 20 means that for an engine which has a power turbine shaft speed of 40,000 r.p.m. the output shaft speed is only 2,000 r.p.m.

This is quite suitable for vehicular applications, and no more reduction stages are needed.

To explore the possibilities of K in such a design, let $N_{GG} = N_{PT}$. Consequently, $(N_{GG}/N_0)_{D.P.} = 20$. The value of K_2 depends on the ratios Z_3/Z_2 and Z_5/Z_6 . Keeping Z_3/Z_2 equal to 4.36 and assuming the minimum value of Z_5/Z_6 to be 0.2, then:

$$K_{2\text{ minimum}} = 1 + \frac{Z_3}{Z_2} \cdot \frac{Z_5}{Z_6} = 1.872, \text{ consequently}$$

$$K_{\text{minimum}} = 0.0936$$

Since the value of Z_5/Z_6 is totally independent of other gear ratios, therefore:

for $K = 0.1$,	$Z_5/Z_6 = 0.229$	Acceptable Values ($0.2 \leq \text{gear ratios} \leq 5.0$)
$K = 0.15$,	$Z_5/Z_6 = 0.458$	
$K = 0.2$,	$Z_5/Z_6 = 0.688$	
$K = 0.25$,	$Z_5/Z_6 = 0.917$	
$K = 0.3$,	$Z_5/Z_6 = 1.146$	
$K = 0.4$,	$Z_5/Z_6 = 1.605$	

These values of (Z_5/Z_6) are achievable and produce acceptable stresses of pinion gears. Thus the Gear Box Configuration shown in Figure 4 provides $K = 0.1$ to 0.4 . This design has also proven to be mechanically feasible. The design, shown in Figure 4, provides also the possibility of having different values of K . This can be obtained by shifting connection (5-6) to 5'-6' or 5"-6". This means more flexibility.

in engine performance specially at off-design conditions.

3.4 Matching Component Characteristics

Steady state operation of a conventional gas turbine requires that the power developed by the gas generator turbine be equal to the power required by the compressor plus any mechanical loss, and that the power developed by the power turbine be equal to the power required by the load, including mechanical losses.

The steady state operation of a co-turboshaft engine requires that the power developed by the gas generator turbine be equal to the power required by the compressor, in order to compress air, plus the power delivered to the output shaft, via the compressor casing and gear box, plus any mechanical loss. Steady state operation requires also that the power developed by the power turbine plus the power transmitted through the casing be equal to the power required by the load, including mechanical losses. When this compatibility of work is satisfied and the engine is in equilibrium, compatibility of flow will also be satisfied, establishing the pressure and temperature level at each station.

Transient Behaviour

During transient operation, both compatibility of flow and work are upset. The incompatibility of flow is due to the fact that pressures cannot change instantaneously in practice. The length of time required to re-establish pressure levels is dependent on internal volumes within the engine, and is very much smaller than the rotor time constant

determined from the rotor moment of inertia [13]. The concept of inter-component volumes is used to deal with flow mis-match between components, and it is introduced as follows:

The mass storage property of the engine is assumed to be concentrated between the components in the form of small volumes. In general, it will be found that flow compatibility is not satisfied for the initial values of pressure and temperature following a disturbance. The discrepancy in mass flow between two adjacent components produces a rate of change of the mass of gas stored in the intercomponent volumes which, in turn, causes a change in the pressure level of this volume. Consider, for example, the interturbine space. The mass flow rate into this space w_{HPT} , and the mass flow rate out, w_{LPT} , are both known from component characteristics. Hence, if m represents the mass of gas stored in the volume:

$$\dot{m} = w_{HPT} - w_{LPT}$$

The rate of change of pressure is now established from the gas laws:

$$P = mRT/V \quad (P \text{ \& } T \text{ are interturbine pressure and temperature})$$

Thus

$$\dot{P} = \frac{R}{V} \cdot \frac{d}{dt} (mT)$$

It has been shown [22] that: $\dot{mT}$ is small compared with $\dot{mT}$, so that

$$\dot{P} \approx \frac{RT}{V} \dot{m} = \frac{RT}{V} (w_{HPT} - w_{LPT}) \quad \dots (3.5)$$

Effect of temperature variations on transient behaviour is investigated in detail in Appendix (A). The analysis has proven that the effect of $m\dot{T}$ in the $\dot{P}$ equation can be ignored without any noticeable effect on the dynamic behaviour of the engine.

Expressions similar to equation (3.5) for other intercomponent pressures can be obtained in terms of the components mass flow. Feeding an integrator with $\dot{P}$, results in a new value of intercomponent volume pressure P which will reduce the mismatch of flow and finally achieve flow compatibility at a stable operating point.

The mismatch of work during a transient is manifested by a difference between the torque required to drive the compressor and that delivered by the turbine, for the gas generator rotor, and by the difference between the torque applied to the output shaft by the load, and that delivered by the power turbine, for the power turbine rotor. The acceleration of each rotor is determined by these net unbalanced torques and the polar moment of inertia of the rotor systems. Consider, for example, the gas generator rotor. The accelerating torque (G_{acc}) = turbine torque (G_T) less Compressor torque (G_C^*). That is:

$$G_{acc} = G_T - G_C = I \cdot \frac{d\omega}{dt}$$

With initial values of rotor speed and thermodynamic conditions, and known compressor and turbine characteristics, the compressor and turbine torques can be determined. Since

$$G_C^* = P_{comp} / N_{rel}$$

in the case of a co-turboshaft engine where

P_{comp} : compressor power
 N_{rel} : compressor relative speed

$$\frac{d\omega}{dt} = \frac{G_{acc}}{I} , \text{ then}$$

$$\frac{dN}{dt} = A(G_T - G_C) , \text{ where } A \text{ is a constant.}$$

In the hybrid computer, an integrator is fed with a signal representing this acceleration, and will generate a new value of speed. This new speed, transmitted to the digital section of the computer, will result in updated values of the system torque, until eventually the net torque becomes zero and a steady state condition is achieved.

3.5 Engine Modules

To simulate the dynamic behaviour of the proposed engine, each component is modelled separately, both mechanically and thermodynamically. The interactions between components are modelled, and the resulting performance of each component and of the engine is calculated for an assumed design and given settings of the controls, as a function of time. The component models are shown in TABLE 1.

Thermodynamic Models	Mechanical Models	Interaction Models
<ul style="list-style-type: none"> . Inlet . Rotating Inlet Guide Vanes (RIGV) . Compressor . Rotating Outlet Guide Vanes (ROGV) . Heat Exchanger . Combustor . Gas Generator Turbine . Power Turbine . Exhaust 	<ul style="list-style-type: none"> . Gas Generator rotor . Output shaft, compressor casing and power turbine rotor . Gear box . Dynamometer . Torque relationships between compressor rotor and stator 	<ul style="list-style-type: none"> . Inlet Volume . RIGV and compressor inter Volume . Compressor and ROGV inter Volume . Combustor Volume . Interturbines Volume

TABLE 1

ENGINE MODULES

The proposed engine is assumed to produce 1000 shaft horse power at design point conditions. The initial design point overall pressure ratio and turbine inlet temperature are 6.2 and 1150 °K, respectively. The model has been designed, of course, to permit these design point parameters to be changed so that the performance of various designs can be investigated.

As the performance of the proposed engine is assumed to be the resultant of the performance of its specified models shown in TABLE 1, the thermodynamic and mechanical characteristics of each component are required. Also the information generation and flow network, within and between the components, are required to enable the designer to choose a calculation strategy. The method of calculation will also depend on the work division between the analog and digital computers.

3.6 Information Flow Diagram

The information generation and flow network within and between components is shown in Figure 5.

Figure 5. Co-Turboshaft Engine Information Flow Diagram

CHAPTER 4

THE MATHEMATICAL MODEL OF A CO-TURBOSHAFT ENGINE

The description given here is that of a mathematical model of the physical processes thought to be an adequate description of the engine under consideration. It is to be distinguished from the implementation of the equations which will be regarded as a computer model. The form of the computer model is to a large extent determined by the choice of computer to be used; however, the mathematical model depends only on the physical processes perceived by the analyst to be a description of the physical plant.

A basic co-turboshaft engine is used as a basis for developing the modelling procedure. It is a simple cycle-two shaft engine with fixed geometry turbines. Figure 6 shows a layout of this engine and the station numbering adopted for the study. Following the investigation of the basic co-turboshaft engine performance, the modelling procedures are extended to the regenerative-variable geometry co-turboshaft engine.

Figure 6. Engine Layout

4.1 Simplifying Assumptions

The underlying principle upon which this simulation is based is the continuous integration of a torque imbalance, determined by the thermodynamic relations describing the engine components. This integral defines the engine speed at all times. The exact description of some of the component operations is complicated and inconvenient to express analytically. In order that the simulation should not be unnecessarily complex, certain simplifying assumptions have been used. These assumptions do not significantly affect the accuracy of the solution but nevertheless should be justified.

- (1) The fluid specific heats are held constant through each

component. The ratio of specific heats (γ) along with the pressure ratio and efficiency determines the temperature rise across the compressor. While the value of specific heat (C_p) increases with increasing temperature, the quantity $(\gamma-1)/\gamma$ decreases with increasing temperature.² The net effect of these opposing phenomena is that the quantity $C_p \Delta T$ remains virtually constant over a wide variation in the value of C_p . It is from this quantity that the compressor torque is calculated and so little error is introduced by the assumption of constant specific heats. [1]

In this study a value of specific heat has been assigned to the mean temperature of each component, at design point condition, and is assumed to remain constant.

(ii) The relationship between fuel-air ratio and the combustor temperature rise has been treated as shown in section 4.2.7. Combustor temperature rise is assumed to vary parabolically with fuel-air ratio, linearly with combustor inlet temperature and inversely with the effective calorific value of the fuel.

(iii) The loss in total pressure in the combustor is regarded as a constant percentage of the inlet pressure. Part of this pressure loss is a function of the dynamic head at entry. A further pressure loss may be expected due to the thermodynamic effect of combustion. At the design point of a conventional engine, the loss in total pressure is quite small and experimental data indicate that this loss as a fraction of inlet pressure remains almost constant. [1]

(iv) Flow mismatch between adjacent components creates a change in pressure at each station separating the two components which is proportional to the difference of flows. Effect of temperature variation on rate of change of pressure is insignificant, since the selection of the intervolum is arbitrary.

4.2 Mathematical Description of Engine Modules

As indicated in Figure 5, the system was treated as a set of physical modules which interact with each other in the same way as they would in the actual engine; a set of functions or equations is used to describe each module. The approach used in modelling the co-turboshaft engine is similar to that used by MacIsaac [20], in his study of the JT15D gas turbine engine.

4.2.1 The Inlet

The inlet has inputs of ambient temperature and pressure. The characteristics of the inlet can be represented, as shown in Figure 7.a, by a screen and duct where the air flow suffers pressure drop proportional to the square of the rate of flow.

Figure 7.a. Inlet Representation

Figure 7.b Inlet Information Flow Diagram

Figure 7.b shows the information flow within the inlet module. Equations (4.1) and (4.2) determine the output flow and temperature w_0 and T_1 respectively

$$\frac{P_a - P_1}{P_a} = K_{in} \cdot \left(w_0 \frac{\sqrt{\theta_1}}{\delta_1} \right)^{2*} \quad [23] \quad \dots(4.1)$$

$$T_1 = T_a \quad \dots(4.2)$$

4.2.2 The Inlet Volume

Following the approachs explained in section 3.4, equation (3.5) is applied here to determine rate of pressure variation during transient periods. Information flow diagram of inlet volume is shown in Figure 8.

$$\frac{dP_1}{dt} = \frac{R}{V_{in}} \cdot T_1 \cdot (w_0 - w_1) \quad \dots(4.3)$$

*For derivation, refer to section 4.2.11

Figure 8. Inlet Volume Information Flow

4.2.3 The Inlet Guide Vanes (IGV)

A provision is made in the modelling for a row of inlet guide vanes, rotating with the casing, which affect the engine performance substantially. The power for this blade row comes from the casing of the compressor, and so reduces the transfer of power from the gas generator turbine to the output shaft. This inlet guide vanes, in fact, supercharges the engine, thus the mathematical representation of it is typically similar to a fan or compressor representation with rather flat characteristics [8]. In the absence of this supercharging effect the inlet guide vanes module and its following volume are by-passed by the model and values of variables at station 2 (compressor inlet, Figure 6) are set equal to their values at station 1 (outlet from Inlet module).

4.2.4 The Compressor

Figure 9 shows the performance map describing the compressor of the co-turboshaft engine. The compressor characteristics shown suggest the use of pressure ratio as the independent variable. This makes the task of modelling easier.

Figure 9. Overall Performance Characteristics of Nasa Eight-Stage Axial Flow Compressor [21]

The basic assumption here is that a compressor with co-rotating stator and aerodynamic speed N_{rel} (speed of core relative to casing) behaves like a conventional compressor of the same characteristics and aerodynamic speed equal to N_{rel} . Thus the compressor of Figure 9 is assumed to maintain the same characteristics while operating in the co-rotating stator mode. This assumption is justified by the work of Chappell, Millar and Swiderski [7, 8, 9 and 10]. Quoting from reference 10: "In fact, the authors were surprised to find so little deterioration of performance between the two modes. With a co-rotating compressor designed for that mode of operation, equal or even improved performance should be attained with the co-rotating machine."

The compressor temperature rise can be calculated from compressor pressure ratio and efficiency according to the equation:

$$T_3 - T_2 = \frac{T_2}{\eta} \left[\left(\frac{P_3}{P_2} \right)^{\gamma-1/\gamma} - 1 \right]$$

Thus a map similar to that of Figure 9 but with temperature rise as the vertical coordinate can be produced easily. Such a map seems more convenient than the more usual efficiency map for use in modelling since it provides temperature rise as a function of speed and pressure ratio directly.

The information flow diagram of the compressor is shown in Figure 10, and the compressor model is shown in Figure 11.

Figure 10. Information Flow Diagram

Figure 11. Compressor Representation

4.2.5 The Rotating Outlet Guide Vanes (ROGV)

A co-rotating compressor is assumed to have a row of outlet guide vanes mounted on the end of the casing to remove the swirl from the flow at compressor delivery. This row of rotating blades has the effect of little turbine. It extracts part of flow energy causing an enthalpy and pressure drop of flow. At the same time this energy is transferred to the output shaft through the compressor casing and gear box. The performance of the rotating outlet guide vanes is represented by equations (4.5) and (4.6). Detailed analysis and derivation of these equations are shown in Appendix A. Figure 12 shows the ROGV model representation. The temperature drop is:

$$T_3 - T_4 = K_{ROGV} \cdot \left(\frac{N_C \cdot w_3 \cdot T_3}{R_3} \right) \quad \dots(4.5)$$

where:

$$K_{ROGV} = \frac{\tan \beta_3 \cdot R}{60 \cdot h \cdot C_p}$$

The pressure ratio is:

$$P_4/P_3 = \left[1 - \left(\frac{K_{ROGV}}{\eta_{ROGV}} \cdot \frac{N_C \cdot w_3}{P_3} \right) \right]^{\gamma/\gamma-1} \quad \dots(4.6)$$

Figure 12. ROGV Model

4.2.6 The Intercompressor Volume (ROGV Volume)

The rate of change of compressor delivery pressure $\dot{p}_3$ is determined by equation (4.6). The information flow diagram is shown in Figure 13.

$$\frac{dP_3}{dt} = \frac{R}{V_{ROGV}} \cdot T_3 \cdot (w_2 - w_3) \quad \dots(4.7)$$

Figure 13. ROG Volume Information Flow

4.2.7 The Combustor

The combustion chamber is treated as a cold volume followed by an instantaneous heat release due to fuel addition which can be imagined to take place at the end of the chamber. Based on assumption (iii), section 4.1 the combustor pressure loss is determined by equation (4.8)

$$P_5 = K_{C.C.} \cdot P_4 \quad \dots(4.8)$$

The temperature rise is determined from equation (4.9)

$$T_5 - T_4 = (A + B.f + C.f^2 + D.f.T_4)/ECV \quad [19] \quad \dots(4.9)$$

The rate of change of combustor pressure is:

$$\frac{dP_4}{dt} = \frac{R}{V_{C.C.}} \cdot T_4(w_3 + w_f - w_4) \quad \dots(4.10)$$

(a)

(b)

Figure 14. Combustion Chamber Model

4.2.8 The Gas Generator Turbine

The flow characteristics of the Gas Generator Turbine (GGT) are shown in Figure 15. Turbine efficiency as a function of speed and pressure ratio is shown in Figure 16. However this kind of efficiency representation is not adopted for use in this model. As can be seen in Figure 16, the process of interpolation to estimate efficiencies at speeds other than these represented is an obvious source of inaccuracy. A more accurate method of estimating turbine efficiency is adopted in this study which is based on the $\eta - v$ characteristics shown in Figure 17. In this figure the pressure ratio has a weak secondary effect on turbine efficiency. The more significant factor is v , the velocity ratio. (Note that the denominator of this ratio is a strong function of turbine pressure ratio).

Figure 15. Gas Generator Turbine Flow Characteristics

Figure 16. Gas Generator Turbine Efficiency Characteristics [30]

Figure 17. Efficiency Characteristics of Gas Generator Turbine

$$v = \frac{U}{\sqrt{2\Delta h}} \quad \text{where:}$$

$$U = \frac{\pi D_m N_{GGT}}{60} \quad \dots (4.11)$$

$$\Delta h = \eta_{GGT} \cdot C_{P_{GGT}} \cdot T_5 \left[1 - \left(\frac{P_6}{P_5} \right)^{\gamma-1/\gamma} \right] \quad \dots (4.12)$$

Thus:

$$v = \frac{\pi D_m}{60} \cdot \frac{N_{GGT} / \sqrt{T_5}}{\sqrt{\eta_{GGT} \cdot C_{P_{GGT}} \cdot [1 - (1/PR)^{\gamma-1/\gamma}]}} \quad \dots (4.13)$$

Equation (4.13) shows that the value of v is not independent of η . If we denote the design point value of a variable by D.P. and by O.D. its value, off-design, then:

$$\frac{v_{O.D.}}{v_{D.P.}} = \frac{(N_{GGT} / \sqrt{T_5})_{O.D.}}{(N_{GGT} / \sqrt{T_5})_{D.P.}} \cdot \sqrt{\frac{\eta_{D.P.} [1 - (1/PR_{D.P.})^{\gamma-1/\gamma}]}{\eta_{O.D.} [1 - (1/PR_{O.D.})^{\gamma-1/\gamma}]}} \quad \dots (4.14)$$

A simple iteration procedure using equation (4.14) and Figure 17 leads to a very good estimation of efficiency within three trials or less. Once the efficiency is found, the turbine temperature drop is calculated from equation (4.15)

$$T_5 - T_6 = \eta_{GGT} \cdot T_5 \cdot [1 - (P_6/P_5)^{\gamma-1/\gamma}] \quad \dots (4.15)$$

The gas generator turbine information diagram is shown in Figure 18.

Figure 18. G.G. Turbine Information flow

Figure 19 shows the model of the gas generator turbine.

Figure 19. Gas Generator Turbine Representation

4.2.9 The Interturbine Volume

The interturbine volume calculation considers the conservation of mass in a small volume. It is used as a mechanism to generate a pressure which could be used as inputs to both of the turbine modules. The form of the model for this module is shown in Figure 20. The duct pressure loss was assumed to be included in the efficiency terms of the two turbines.

The rate of change of pressure P_6 is determined by equation (4.17)

$$\frac{dP_6}{dt} = \frac{R}{V_T} \cdot T_6 \cdot (w_4 - w_5) \quad \dots(4.17)$$

(a)

(b)

Figure 20. Interturbine Volume Module Representation

4.2.10 The Power Turbine

The mathematical modelling of the power turbine is similar to that of the gas generator turbine. The flow characteristics of the power turbine are shown in Figure 21 while the efficiency characteristics, are shown in Figure 22. The power turbine was assumed to be rotor-choked, whereas the gas generator turbine was stator choked [30].

Figure 21. Power Turbine Flow Characteristics [30]

Figure 22. Power Turbine Efficiency Characteristics (Schematic)

same procedure as in gas generator turbine is used here. Equations (4.13), (4.14), (4.15) and (4.16) respectively become:

$$v = \frac{\pi D_m}{60} \cdot \frac{N_{PT}/\sqrt{T_6}}{\sqrt{\eta_{PT} \cdot C_{P_{PT}} \cdot [1 - (1/PR)^{\gamma-1/\gamma}]}} \quad \dots (4.18)$$

$$\frac{v_{O.D.}}{v_{D.P.}} = \frac{(N_{PT}/\sqrt{T_6})_{O.D.}}{(N_{PT}/\sqrt{T_6})_{D.P.}} \cdot \sqrt{\frac{\eta_{D.P.} [1 - (P_7/P_6)_{D.P.}^{\gamma-1/\gamma}]}{\eta_{O.D.} [1 - (P_7/P_6)_{O.D.}^{\gamma-1/\gamma}]}} \quad \dots (4.19)$$

$$T_6 - T_7 = \eta_{PT} \cdot T_6 \cdot [1 - (P_7/P_6)^{\gamma-1/\gamma}] \quad \dots(4.20)$$

$$G_{PT} = J \cdot w_5 \cdot \frac{C_{P_{PT}} \cdot (T_6 - T_7)}{N_{PT}} \quad \dots(4.21)$$

The information flow diagram and model representation of the power turbine are shown in Figures 23 and 24 respectively.

Figure 23. Power Turbine Information Flow

Figure 24. Power Turbine Representation

4.2.11 The Exhaust Duct

The pressure loss for the exhaust duct varies with non-dimensional mass flow, and power turbine speed. The temperature is assumed to be constant through the exhaust duct which will normally be insulated in practice. The information flow diagram for the exhaust system is shown in Figure 25.

Figure 25. Exhaust Information Flow

The factor that affects cycle analysis is the ratio $(\Delta P_0/P_0)$. Thus the pressure loss (ΔP) in intake and exhaust ducts are more critical than at combustion chamber or heat exchanger because P_0 is essentially the atmospheric pressure in cases of intake or exhaust ducts while P_0 is compressor delivery pressure in cases of combustion chamber or heat exchanger [23]. The size of the exhaust ducting for a gas turbine is determined primarily by the requirement for low pressure losses at maximum power, which normally corresponds to maximum flow rate.

Velocities in components like intake or exhaust ducts are sufficiently low for the flow to be treated as incompressible [23], thus: ΔP_0 is proportional to Inlet Dynamic Head, or in a non-dimensional form:

$$\frac{\Delta P_0}{P_0} \propto \left(\frac{w \sqrt{T_0}}{P_0} \right)^2 \quad \dots (4.22)$$

For insignificant change in total temperature T_0 ..

$$\frac{\Delta P_0}{P_0} = \left(\frac{\Delta P_0}{P_0} \right)_{D.P.} \cdot \left[\left(\frac{w \sqrt{T_0}}{P_0} \right) / \left(\frac{w \sqrt{T_0}}{P_0} \right)_{D.P.} \right]^2 \quad [23] \quad \dots (4.23)$$

or simply for exhaust duct:

$$\frac{P_7 - P_a}{P_a} = K_{exh} \cdot \left(\frac{w_5 \sqrt{T_7}}{P_7} \right)^2 \quad \dots (4.24)$$

The exhaust model can be represented as shown in Figure 26.

Figure 26. Exhaust Pressure Loss

4.2.12 Pressure Loss Due to Flow Swirl

In a free shaft engine, where the power turbine shaft is mechanically coupled to the load shaft, the power turbine speed varies significantly over the range of operation of the engine. If turbine blades are designed so that flow leaves turbine with axial absolute velocity C_2 at design point conditions, Figure 27, then a change of turbine speed produces flow swirl (C'_2). This section presents a simple analysis for estimating pressure losses associated with flow swirl in both conventional and co-turboshaft modes of operation of the engine.

Figure 27. Conventional Mode - Power Turbine Velocity Triangles

A - Conventional Gas Turbine Engine

In the analysis of subsection A, the power turbine flow characteristics is assumed to be a single line for simplicity (stator choked). When the power turbine is rotor choked, a change in turbine speed results in a change in non-dimensional mass flow. Analysis of such case is deferred to subsection B.

Considered first is the case of a fixed gas power supplied to the power turbine (constant N_{GG}). The nozzle exit velocity C_1 remains close to design point value C_1 , as shown in Figure 27, as does the axial velocity. From Figure 27 it can be seen that $V_{\theta 2} = U_{D.P.} - U'$, thus flow swirl $V_{\theta 2}$ can be found from equation (4.25).

$$V_{\theta 2} = U_{D.P.} \cdot \left(1 - \frac{U'}{U_{D.P.}}\right) = U_{D.P.} \left(1 - \frac{N_{PT}}{N_{PT D.P.}}\right) \quad \dots (4.25)$$

or simply:

$$V_{\theta 2} = K_{\theta} \cdot \left(1 - \frac{N_0}{N_{0 D.P.}}\right), \quad K_{\theta} = U_{D.P.} \quad \dots (4.26)$$

Consider the swirl pressure loss is equal to $(1 - K_s) \cdot \frac{1}{2} \rho V_{\theta}^2$, where K_s is the effectiveness of the "deswirl stators". A good set of exhaust deswirl stators would recover perhaps 80 to 90 percent of the swirl dynamic pressure $(\frac{1}{2} \rho V_{\theta}^2)$; while a honeycomb, straightener would recover none of it. Thus:

$$\Delta P_{swirl} = (1 - K_s) \cdot \frac{1}{2} \frac{P_7}{RT_7} \cdot \left[K_{\theta} \left(1 - \frac{N_0}{N_{0 D.P.}}\right)\right]^2 \quad \dots (4.27)$$

The flow swirl $V_{\theta 2}$ is assumed to be a combination of the effect of reduction of speed (part a in Figure 28) and the effect of increase in mass flow (part b in Figure 28). Part a remains as determined by equation (4.25). Part b is equal to $(C'_{x2} - C_{x2})/\tan \beta_2$, thus

$$b = \frac{C'_{x2} - C_{x2}}{C_{x2}/U_{D.P.}} = U_{D.P.} \left(\frac{C'_{x2}}{C_{x2}} - 1 \right) \quad \dots (4.28)$$

The mass flow w is equal to ρAC_x , where ρ is the flow density and A is the turbine annular area. Thus:

$$C_x = \left(\frac{R}{A} \right) \cdot \frac{WT}{P}, \text{ and consequently:}$$

$$\frac{C'_{x2}}{C_{x2}} = \frac{w_5}{w_{5D.P.}} \cdot \frac{T_7}{T_{7D.P.}} \cdot \frac{P_{7D.P.}}{P_7}, \text{ thus:}$$

$$b = K_{\theta} \cdot \left(\frac{w_5}{w_{5D.P.}} \cdot \frac{T_7}{T_{7D.P.}} \cdot \frac{P_{7D.P.}}{P_7} - 1 \right) \quad \dots (4.28)$$

Adding a to b to determine V_{θ} ,

$$\begin{aligned} V_{\theta} &= K_{\theta} \left\{ \left(1 - \frac{N_0}{N_{0D.P.}} \right) + \left(\frac{w_5}{w_{5D.P.}} \cdot \frac{T_7}{T_{7D.P.}} \cdot \frac{P_{7D.P.}}{P_7} - 1 \right) \right\} \\ &= K_{\theta} \left(\frac{w_5}{w_{5D.P.}} \cdot \frac{T_7}{T_{7D.P.}} \cdot \frac{P_{7D.P.}}{P_7} - \frac{N_0}{N_{0D.P.}} \right) \quad \dots (4.29) \end{aligned}$$

The swirl pressure loss becomes:

$$\Delta P_{\text{swirl}} = (1 - K_s) \cdot \frac{1}{2} \frac{P_7}{RT_7} \cdot [K_\theta \cdot \left(\frac{w_5}{w_{5 \text{ D.P.}}} \cdot \frac{T_7}{T_{7 \text{ D.P.}}} \right) \dots \dots \dots \left(\frac{P_{7 \text{ D.P.}}}{P_7} - \frac{N_0}{N_{0 \text{ D.P.}}} \right)^2] \dots (4.30)$$

The turbine back pressure P_7 is equal to $(P_a - \Delta P_{\text{exh}} - \Delta P_{\text{swirl}})$. Thus total exhaust pressure loss, becomes:

$$\Delta P_{\text{tot}} = \Delta P_{\text{exh}} + \Delta P_{\text{swirl}}$$

$$P_7 - P_a = P_a \cdot K_{\text{exh}} \cdot \left(\frac{w_5 \sqrt{T_7}}{P_7} \right)^2 + (1 - K_s) \cdot \frac{1}{2} \frac{P_7}{RT_7} \dots \dots [K_\theta \left(\frac{w_5}{w_{5 \text{ D.P.}}} \cdot \frac{T_7}{T_{7 \text{ D.P.}}} \cdot \frac{P_{7 \text{ D.P.}}}{P_7} - \frac{N_0}{N_{0 \text{ D.P.}}} \right)^2] \dots (4.31)$$

Preliminary estimation of the order of ΔP_{swirl} relative to ΔP_{exh} based on model results shows that for $0.9 \leq (N_0/N_{0 \text{ D.P.}}) \leq 1$,

* It can be seen from equation 4.31 that P_7 appears in both sides of the equation, which is normally solved on the digital computer. Since the method used to obtain a new equilibrium operating point is to initialize first the design point on the computer model and then depart from design point condition to other equilibrium points, the digital computer faces no problem in dealing with such kinds of equations (equations 4.31, 4.24). The computer assumes a value of P_7 , in the right hand side of equation 4.31, equal to the D.P. value and calculates a new value of P_7 . In the next run of calculation, the computer takes the calculated value of P_7 and substitutes it in the right hand side of equation 4.31 to get a new value of P_7 and so on. It should be noted, however, that the change in P_7 between two successive runs of calculations is quite small so that the accuracy of model calculations is not affected.

ΔP_{swirl} can be safely ignored. ΔP_{swirl} is of the same order of ΔP_{exh} as $N_0/N_{0D.P.}$ approaches 0.6. The value of ΔP_{swirl} exceeds ΔP_{exh} and increases rapidly for further reduction of speed. The mathematical model of the exhaust system including swirl effect is shown in Figure 29.

Figure 29. Exhaust System Representation

4.2.13 The Gas Generator Rotor Dynamics.

The calculation of the GG rotor speed consists of integration of the net torque on the spool with respect to time. The net torque is determined by the difference between the torque produced by the GG turbine and that absorbed by the GG compressor. Calculation of these torques is an easy matter if the component operating points are known, see equations (4.4) and (4.16).

For the purpose of this study, the polar mass moment of inertia for the GG spool was deduced by comparison with a J.85 gas turbine engine, for which this inertia was known.

The form of the GG rotor dynamics is shown in Figure 30.

Figure 30. Gas Generator Rotor Dynamics Model

4.2.14 The Load

An electric dynamometer was selected to represent the driven load.

It has the characteristics shown in Figure 31.

Figure 31. Electric Dynamometer Characteristics [20]

This kind of characteristic is quite suitable to the purpose of the study. It also allows I_f to vary independently of speed, thus controlling the output torque. Figure 32 shows the dynamometer model

Figure 32. Dynamometer Model

4.2.15 The Compressor Rotating Casing

In a co-rotating compressor, the casing rotates in the same direction of the rotor at 10% to 40% of its speed (section 3.3). About this fraction of the power input to the compressor rotor is extracted by the casing, to supply the output shaft with power in addition to the power turbine.

Simple conservation of angular momentum applied to the core compressor of this arrangement shows that, with axial flow at inlet and outlet to the casing, the torque (G_c) transmitted to the rotating casing must equal the torque provided by the driving turbine via the high speed

rotor shaft. Thus the power (P) available from the rotating casing is the same fraction of the total power generated by the turbine as the observed rotational speed (N_C) of the casing is of the observed rotational speed of the turbine.

$$\begin{aligned} \text{i.e., } G_{\text{casing}} &= G_{\text{turbine}} \\ G_C &= G_{GT} \end{aligned} \quad \dots(4.32)$$

$$\text{Therefore } \frac{P_{\text{casing}}}{P_{\text{turbine}}} = \frac{N_{\text{casing}}}{N_{\text{turbine}}} = \frac{N_C}{N_{GG}} \quad \dots(4.33)$$

This relation is termed the torque constraint.

4.2.16 The Output Shaft Rotor Dynamics

Three torques are acting on the output shaft as shown in Figure 33.

Figure 33. Torques Contribution to Output Shaft Dynamics

2

The net result of these torques is the accelerating torque G_{acc} . However, it should be noticed that these torques are collected in the gear box at different speeds. The balance of energy throughout the gear box results in the following equation:

$$G_{PT} \cdot N_{PT} + G_C \cdot N_C + G_O \cdot N_O + G_{acc} \cdot N_O = 0 \quad \dots(4.34)$$

Thus, the net torque which affects the output shaft speed G_{acc} is:

$$G_{acc} = G_{PT} \cdot \frac{N_{PT}}{N_O} + G_C \cdot \frac{N_C}{N_O} + G_O \quad \dots(4.35)$$

where G_O has a negative sign. Speed ratios (N_{PT}/N_O) and (N_C/N_O) are of fixed values, K_1 and K_2 respectively. Thus

$$G_{acc} = K_1 G_{PT} + K_2 G_C + G_O \quad \dots(4.36)$$

Gears, shafts and other rotating masses can be reduced to a single equivalent value of moment of inertia at output shaft speed (I_O). The output shaft rotor dynamic model is represented in Figure 34.

Figure 34. Output Shaft Rotor Dynamics Model

4.2.17 The Heat Exchanger

The major factor which affects the performance of heat exchangers is the heat exchanger effectiveness ϵ . The effectiveness of a heat exchanger depends on the fixed parameters of heat transfer area and configuration (counter-flow, cross-flow, or parallel-flow), and the parameters varying with engine operating conditions such as the overall heat transfer coefficient U and the thermal capacities of fluids C 's. The effectiveness can be related to these variables as follows:

$$\epsilon = f\left(NTU, \frac{C_{\min}}{C_{\max}}\right) \quad \dots (4.37)$$

where:

$$NTU = U.A/C_{\min} \quad \text{: Number of Heat Exchanger Transfer Units}$$

$$C = m.C_p$$

For vehicular gas turbine-regenerator application the ratio $C_{\min}/C_{\max}$ is: $0.95 < C_{\min}/C_{\max} < 1.00$ [24]. $C_{\min} = w_{\text{air}} \cdot C_{p_{\text{air}}}$ and $C_{\max} = w_{\text{gases}} \cdot C_{p_{\text{gases}}}$. Typical values of ϵ lie in the range of 0.8 to 0.9. The ratio $\frac{C_{\min}}{C_{\max}}$ remains almost constant over the entire range of off-design performance. Cima and London [25] have shown that the variation of NTU as conditions depart from design point values is quite insignificant. To quote them,

"Changes in NTU encountered in practice are relatively small

because a decrease of C is largely compensated by a decrease in U in the definition $NTU = (A U / C_{\min})$. Consequently, at a high NTU level, e.g., $NTU = 3$, the change in the steady - state effectiveness ϵ is very small as previously shown".

Thus variation of steady state effectiveness is ignored for the purpose of this study and a value of $\epsilon = 0.9$ is assumed.

The heat exchanger pressure drop still follows the relation:

$$\frac{\Delta P_0}{P_0} = K \cdot \left(\frac{w \sqrt{T_0}}{P_0} \right)^2$$

This relation is applied to both air and gas sides of the heat exchanger. Losses of 3 percent are assumed at the air side and gas side at the design point conditions. The heat exchanger model is illustrated in Figures 35 and 36.

Figure 35. Heat Exchanger Model

Equations (4.37), (4.38), (4.39), and (4.40) relates model output variables to its input variables:

$$\frac{P'_4}{P_4} = 1 - 0.03 \left[\frac{(w_4 \sqrt{T_4/P_4})}{(w_4 \sqrt{T_4/P_4})_{D.P.}} \right]^2 \quad \dots (4.37)$$

$$\frac{P'_7}{P_7} = 1 - 0.03 \left[\frac{(w_7 \sqrt{T_7/P_7})}{(w_7 \sqrt{T_7/P_7})_{D.P.}} \right]^2 \quad \dots (4.38)$$

$$T'_4 = T_4 + \epsilon (T_7 - T_4) \quad \dots (4.39)$$

$$T'_7 = T_7 - \epsilon \cdot \left(\frac{C_{air}}{C_{gas}} \right) (T_7 - T_4) \quad \dots (4.40)$$

where $C_{air} = w_4 \cdot C_{p4}$ and $C_{gases} = w_7 \cdot C_{p7}$

Figure 36. Heat Exchanger Information Flow Diagram

4.3 Scaling of Variables

The scaling of a dynamic problem on a hybrid computer takes the form of both amplitude and time scaling. The necessity for amplitude scaling arises by virtue of the fixed point operation of the analog computer, wherein no voltage signal can exceed the machine limits. Time scaling is a device which permits dynamic analysis at speeds other than real time.

Suitable scale factors are assigned to each variable or parameter to ensure that none of the scaled variables exceeds 1.0 during the entire range of operation, while providing reasonable computational accuracy.

Module equations in scaled form are presented in Appendix (B). Selected scale factors and engine constants are shown in Appendix (F).

CHAPTER 5

HYBRID COMPUTER MODELLING

5.1 The Operation of a Hybrid Computer - Compatibility of a Serial Parallel Combination

The concept of hybridization involves the interconnection of an analog with a digital computer such that the overall system retains the desirable characteristics of each component. Continuous processes such as the integration of the acceleration equations to provide the speeds of the two shafts, or of the volume dynamics equations to provide the pressures, are naturally handled on the analog computer. The calculation of the terms of these equations, such as the torques and mass flows, which involve characteristic maps and relatively complex algebraic equations, are most naturally done by a digital computer.

In view of the drastically different modes of operation of each, the interconnection and synchronization of the two computers is by no means an easy task. The digital machine is a discrete, sequential system, while the analog is parallel and continuous. This fact implies that information transmittal is not possible without some form of conversion.

To clarify this conversion process, the following has been extracted from reference 1.

During the course of data conversion and subsequent reconstruction, many sources of error may contribute to overall solution inaccuracy. Most of these errors have their foundation in the electronic circuitry of the computers and the interface and are beyond the control of the operator. For example, sampling errors due to a rapidly changing analog signal during the course of sampling can produce uncertainty in the quantized signal. Modern design has done much to minimize these errors, but it is worthwhile realizing their existence.

In practice the only source of error that can be controlled by the operator is the time delay associated with the serial operation of the digital computer forming an element in a dynamic loop. This error results from the fact that the output of the digital-analog converter is only updated when the digital computer completes a cycle. The analog computer operates in a continuous fashion on this signal, accumulating error as the solution progresses. The magnitude of this error is a function of the complexity of the digital computer operations and to some extent of the programming efficiency, both of which influence the computation time required. In the case of the gas turbine, engine time constants are sufficiently large that reasonable programming efficiency insures that this delay can be neglected, hence real time simulation is still possible.

5.2 The Program Structure

The results of the digital calculations are available at discrete intervals, spaced apart by the time required to access the speeds and pressures computed by the analog at the end of the previous interval and to carry out the component look-up calculations. Within any one time interval, the analog computer operates with a fixed set of inputs from the digital. In order to keep this interval short, for better accuracy and stability of operation, it is important to remove unnecessary calculations from the digital part, or at least from the high speed part of the digital program, which is the part exchanging information with the analog computer through A/D and D/A channels.

5.2.1 Work Division Between Analog and Digital Parts of the Hybrid Computer

In essence the computer has three sets of computations to carry out in parallel - the continuous integration of the acceleration equations of the rotors to determine their speeds, the integration of the pressure rates to determine the pressure level at each station, and the solution of the thermodynamic matching equations to determine the instantaneous mass flow rate and powers at those speeds and pressure levels. The results of each calculation are needed in the other calculations. In general the integrations and some of the associated calculations are carried out by the analog part of the computer and the results of integration (speeds and pressures) are supplied to the digital part through analog to digital

(A/D) converters. The thermodynamic calculations are carried out by the digital computer part and results like mass flow, temperatures, and torques are supplied to the analog computer part through digital to analog (D/A) converters. All the components maps are stored in the digital computer, while the controlling parameters, the design parameters, and ambient conditions are set on the analog computer.

5.2.2 Digital Program

The digital computer program carries out three main tasks.

5.2.2.1 Initialization

Initialization includes scaling maps and storing them, initiating the connection between analog and digital parts, reading system constants, component characteristics and specified initial steady state running point, converting data to scaled form for storage and later use, and setting the analog computer to the required initial conditions.

5.2.2.2 High Speed Loop

The high speed loop is the program which carries out the component data look-up and resulting calculations of torque. It consists of the thermodynamic models shown in TABLE 1, and each model receives input information through A/D converters. Part of the output of the model will be supplied to the analog computer through D/A converters and the rest will be used in the calculations of the next model.

5.2.2.3 Off-Line Operations

A group of subroutines is required for what might be called service operations. Some of these service operations are:

- Debugging of data
- Setting the analog computer potentiometers to the required values
- Changing the analog computer initial conditions to any current operating point selected
- Printing the output

The structure of the off-line operations are shown in Figure 38. The digital program structure is shown in Figure 37.

5.2.3 Analog Program

The analog computer program carries out the following tasks in a continuous fashion:

- a) integrates the gas generator dynamic equation for the compressor and gas generator turbine rotor to generate the rotor speed;
- b) integrates the output shaft dynamic equation and generates the output shaft speed, the power turbine speed and the compressor casing speed;
- c) integrates the equations of the pressure rates for each intervolumé to generate the pressures between engine components.

In addition to these computational tasks, the analog computer provides warning lights indicating some specific critical operating conditions. The engine control variables, such as fuel flow and dynamometer field current, design parameters such as gear box constants K_1

- ADC: Analog to Digital Converters
- DAC: Digital to Analog Converters
- OP: Operating Mode
- H: Hold mode
- IC: Initial Conditions mode
- ST: Static Test mode
- PC: Pot read
- SP: Pot setup
- Logical Variables:
 - . SSW: Sense Switch
 - . SLL: Sense Line

Figure 37. Digital Program Structure

Figure 38. Off-Line Service Operations

and K_2 , and ambient conditions, are set or adjusted by hand-set pots (Q-Pots) on the analog computer console.

Two X-Y plotters are available to plot such variables as the engine power, output torque, turbine inlet temperature, or specific fuel consumption versus output shaft speed or any other variable. These plotters can also be used to reproduce the component maps as stored in the digital computer for verification of input data. Also, a six-channel Brush strip-chart recorder permits any six analog variables to be plotted continuously against time.

Details of the Analog circuits are presented in Appendix (E).

5.3 Program Capabilities

5.3.1 Transformation into Conventional Engine

The hybrid computer program has been designed so as to provide the ability to calculate the performance of a conventional engine (stationary compressor casing) having generally the same component characteristics as the co-turboshaft engine. This is done by means of a logical flag which makes the digital program ignore certain models, and sets the compressor casing speed to zero. On the analog computer, the inter-volume between the compressor and ROGV is similarly bypassed.

5.3.2 Dynamic Analysis

The hybrid computer program is capable of carrying out engine dynamic analysis to permit different control strategies to be examined. However, the thesis work is mainly based on steady state results to evaluate the engine performance.

5.3.3 Program Extension

The present program is designed in such a way that it is easy to add or remove components without affecting the program. Thus it was relatively straightforward to incorporate a heat exchanger model in the engine model to examine its effect on engine performance. Similarly, the fixed geometry power turbine was replaced by a variable geometry one to study the effect of current engine technological advances.

5.3.4 Engines of Different Designs

The program provides the user with a powerful tool to examine different engine designs in a very simple way. Design point parameters and component characteristics can be varied, as well, to investigate their effects on the engine performance.

5.4 Model Implementation

The model, as defined in the previous chapter, was implemented on the hybrid computer in the Analysis Laboratory of the National Research Council of Canada. This computer consists of an EAI 680 analog computer and Pacer 100 digital computer. A general overview of the implementation is shown in Figure 39. All of the component maps were implemented on the digital computer using table "look-up" procedures; the thermodynamic data obtained from the maps were used to calculate the flows, pressures and temperatures for that component and these data were then transferred to the analog computer. The analog computer was assigned all of the dynamic portions of the problem and used the information provided by the digital computer to calculate the aerodynamic parameters in the associated volumes and the speeds of the rotors. Parallel, continuous integration of these derivatives of the analog computer, which is coupled to the digital computer, provided a closed loop, continuous solution to the specified equations. Detailed program listings and circuit diagrams are given in the appendices (E and F).

5.4.1 Problem Control

As is indicated in Figure 39, the centre for control of the problem is at the analog console. The digital computer is set to work at the start of the program and continues to execute the high speed loop until it is called upon to perform some other function such as printout of results. At this point it would set the analog computer into HOLD,

Figure 39. General Overview of Computer Model

perform the required task, return the analog computer to its original mode of operation, and restart the high speed loop. This feature permits a 'snapshot' of engine operation at any instant during a transient and also provides hard copy output of steady state performance when required.

A very important requirement of any model is that it be readily understandable and be able to provide the operator with any output that he desires. In a hybrid computer model all variables that appear somewhere on the analog computer are readily available for inspection or plotting; however, the digital computer variables, which do not appear at the interface are not nearly so accessible unless provision is made for this feature. This is particularly important when one wishes to trace the operation of the engine in terms of the non-dimensional description of one of its components, or during a debugging phase when one is trying to trace the cause of an error which may be propagating throughout the entire model. For this reason, all of the digital computer variables, which do not appear at the interface are equivalenced to elements of an array. It is then possible to use an analog potentiometer to address this array [27]. The operator is thus provided with the facility to 'dial up' any address in this variable array and have that particular element appear at the interface on a dedicated channel. In this model, nine such elements are brought to the interface simultaneously. Any nine variables can therefore be observed on a digital readout or can be plotted on the X-Y recorder.

5.4.2 Accuracy of Model Implementation

Accuracy of the implementation is a result of both the accuracy of the digital and analog computers and also a result of the dynamic lag in updating the digital outputs to the analog computer. The first of these factors, computer accuracy, is dependent, for the most part, on the number of analog components used and is therefore at best 0.01 percent. By careful scaling, overall accuracy of the order of 0.5 percent is achievable [27]. This is more than sufficient accuracy for an engine model and, in general, the component data inaccuracy will greatly outweigh any computer inaccuracy [27].

The effect of dynamic lag will be a function of both the rate of change of the signals and also the actual computational delay of the digital computer. Obviously, by time scaling, the "process" can be slowed down to permit more frequent updating from the digital high speed loop. For engine models of the type used in this study, the volume dynamics are several orders of magnitude faster than the rotor dynamics and hence it is these calculations which dictate the required sampling rate. In this study, of course, dynamic response was not the objective, and the sampling rate had to be such that stable transition from one steady state point to another would be conveniently obtained. Therefore, computer model accuracy is expected to be much more a function of the input data than any inaccuracies due to implementation [27].

5.5 Model Verification and Validation

The procedures involved in the production of a computer model are indicated in Figure 40. [27] These procedures are gradually becoming standardized and terms such as verification and validation have acquired quite specific meaning [27, 28 and 29].

Figure 40. Conceptual View of System Modelling Procedure

Once the analysis is complete, the resulting mathematical model is implemented to produce a computer model. The computer model requires verification to show that the computer is actually solving the specified

equations. Having verified that the computer model is indeed a true implementation of the mathematical model, the task of validation must begin. Validation, which requires information from actual engine tests, must show that the mathematical model is a reasonable representation of the real hardware. Ideally, this should be a continuing procedure during which further engine tests will point out to the analyst any errors or inadequacies in the mathematical model. These changes to the mathematical model must then be implemented, verified and finally validated to ensure that the model is a good representation of real life. [27]

The task of verification is carried out completely, and a sample verification is presented in Appendix (C). Validation is only possible if data from an actual engine are available, which is not the case in this study. However, most of the techniques used in this study have been used by MacIsaac [20] to simulate turbojet and turbofan engines, and the techniques have proven to be quite valid.

5.6 Method of Producing Model Results

The method used to obtain performance characteristics of the co-turboshaft engine at any equilibrium operating point can be summarized as follows:

- a - During the initialization stage of the program, variables values are set equal to design point values.
- b - The load is then adjusted to produce G_0 other than $G_{0,D.P.}$
- c - In response to the load change, the fuel flow is automatically adjusted to keep $N_{GG} = N_{GG,D.P.}$
- d - A new equilibrium operating point is achieved when all time derivatives of engine variables become zero. Selected engine variables are then recorded.

CHAPTER 6

PERFORMANCE CHARACTERISTICS OF THE BASIC CO-TURBOSHAFT ENGINE.

6.1 Introduction

Performance characteristics of the co-turboshaft engine are presented in this chapter and in the next one also. The term "Basic Co-Turboshaft Engine" means a simple cycle-fixed geometry co-turboshaft engine. Performance curves presented are the results of the model described in chapters 3 and 4 and is implemented on the hybrid computer described in chapter 5.

The selection of design point parameters, like compressor pressure ratio and maximum cycle temperature, is likely to follow the same rules of selection as in conventional gas turbine engines. However, the unique character of the co-turboshaft engine is very likely to have an impact on the selection of certain parameters. Of course there are parameters which, if increased, always produce favourable effects, such as maximum cycle temperature and component efficiencies. With respect to these parameters both the co-turboshaft engine and conventional engines have the same objective, that is to raise the value of these parameters as high as possible. However, mechanical and aerodynamic designs of engine components impose certain limits on the

selection.

As far as other design point parameters are concerned, they need to be selected in accordance with the co-turboshaft engine directions in order that the co-turboshaft engine performs at its best. Among these parameters are: compressor pressure ratio, location of the design point on compressor map and location of design point on power turbine map.

Apart from these parameters, the design speed ratio K should be selected quite carefully. K is the factor which distinguishes the co-turboshaft engine from other gas turbine engines. It simply indicates the rotation of the compressor casing in relation to the compressor core.

6.2 Cycle Analysis

The diagrammatic layout of the gas turbine cycle used in a co-turboshaft engine is shown in Figure 41.

Figure 41. Basic Co-Turboshaft Engine System

Following the same notations of Figure 6:

- T_5 : maximum cycle temperature (TIT)
 ΔT_{23} : compressor actual temperature rise
 PR : compressor pressure ratio (P_3/P_2)
 η_C, η_{GT} : isentropic efficiencies of compressor, gas generator turbine
 η_{PT} and power turbine respectively.

Then compressor work W_{comp} becomes:

$$W_{comp} = m_a \cdot C_{p_a} \cdot \Delta T_{23} \quad \dots(6.1)$$

since the casing is co-rotating with the compressor core, it derives its power from the compressor air stream and delivers it to the output shaft through the gear box. The torque constraint, as defined in section 4.2.15, results in:

$$W_C/W_{GGT} = N_C/N_{GGT}$$

where subscript C refers to casing and subscript GGT refers to gas generator turbine. Casing work is also related to compressor work, as shown in section 3.2, in the following form:

$$W_C = W_{comp} \cdot N_C/N_{rel}$$

where:

$$\begin{aligned}
 N_{rel} &: \text{compressor relative speed} \\
 &= N_{GG} - N_C
 \end{aligned}$$

Thus:

$$\begin{aligned}
 W_C &= W_{\text{comp}} \cdot \frac{N_C}{N_{GG} - N_C} \\
 &= W_{\text{comp}} \cdot \left(\frac{K \cdot \frac{N_C}{N_{C,D.P.}}}{1 - K \cdot \frac{N_C}{N_{C,D.P.}}} \right) \\
 &= W_{\text{comp}} \cdot \frac{K \cdot (N_0/N_{0,D.P.})}{1 - K \cdot (N_0/N_{0,D.P.})} \quad \dots(6.2)
 \end{aligned}$$

At the design point the above relationship becomes:

$$\begin{aligned}
 W_C &= W_{\text{comp}} \cdot \frac{K}{1 - K} \\
 &= \frac{K}{1 - K} \cdot m_a C_{P_a} \Delta T_{23} \quad \dots(6.3)
 \end{aligned}$$

The gas generator turbine work becomes:

$$W_{GGT} = (W_{\text{comp}} + W_C) / \eta_m$$

Thus W_{GGT} at the design point becomes:

$$W_{GGT} = (m_a C_{P_a} \Delta T_{23} \cdot \frac{1}{1 - K}) / \eta_m \quad \dots(6.4)$$

And at off-design points:

$$W_{GGT} = (m_a C_{p_a} \Delta T_{23} \cdot \frac{1}{1 - K \left(\frac{N_0}{N_{0,D.P.}} \right)}) / \eta_m \quad \dots (6.5)$$

This must be equal to: $m_g C_{p_g} \cdot \Delta T_{56}$, consequently:

$$\Delta T_{56} = \frac{m_a}{m_g} \cdot \frac{C_{p_a}}{C_{p_g}} \cdot \frac{\Delta T_{23}}{\eta_m} \cdot \frac{1}{1 - K \left(\frac{N_0}{N_{0,D.P.}} \right)} \quad \dots (6.6)$$

$$T_4 = T_5 - \Delta T_{56} \quad \dots (6.7)$$

The gas generator turbine pressure ratio (P/P) is given by:

$$P_5/P_6 = 1 / \left[1 - \frac{\Delta T_{56}}{\eta_{GT} \cdot T_5} \right]^{\gamma_1 / \gamma_1 - 1} \quad \dots (6.8)$$

Considering K_p as the total pressure loss factor, then:

$$K_p = K_{in} \cdot K_{C.C.} \cdot K_{exh} \quad , \text{ where}$$

K_{in} , $K_{C.C.}$, K_{exh} are the pressure loss factors at inlet ducts, combustion chamber and exhaust ducts, respectively. Hence, the power turbine pressure ratio (P_6/P_7) becomes:

$$P_6/P_7 = \frac{K_p \cdot PR}{(P_5/P_6)} \quad , \text{ where:}$$

$$PR = \left(1 + \frac{\eta_C \cdot \Delta T_{23}}{T_2}\right)^{\gamma/\gamma-1}, \text{ thus:}$$

$$P_6/P_7 = K_p \cdot PR \cdot \left[1 - \frac{\Delta T_{56}}{\eta_{GT} \cdot T_5}\right] \quad \dots(6.9)$$

The power turbine temperature drop is:

$$\Delta T_{67} = \eta_{PT} \cdot T_6 \cdot \left(1 - \left(\frac{P_6}{P_7}\right)^{\gamma_2-1/\gamma_2}\right) \quad \dots(6.10)$$

And consequently the power turbine work becomes:

$$W_{PT} = m_g \cdot C_{p_{PT}} \cdot \Delta T_{67}$$

The output power is the sum of the casing power and free turbine power, therefore:

$$\begin{aligned} P_{out} &= m_a \cdot C_{p_a} \cdot \frac{K(N_0/N_{0D.P.}) \Delta T_{23}}{1 - K(N_0/N_{0D.P.})} + m_g \cdot C_{p_{PT}} \cdot \eta_{PT} \dots \\ &\dots \cdot \left(T_5 - \frac{m_a}{m_g} \cdot \frac{C_{p_a}}{C_{p_C}} \cdot \frac{\Delta T_{23}}{\eta_m} \cdot \frac{1}{1 - K\left(\frac{N_0}{N_{0D.P.}}\right)}\right) \dots \\ &\dots \cdot \left[1 - \left\{K_p \cdot \left(1 + \frac{\eta_C \Delta T_{23}}{T_2}\right)^{\gamma/\gamma-1} \cdot \left[1 - \frac{m_a}{m_g} \cdot \frac{C_{p_a}}{C_{p_C}}\right] \dots \right. \right. \\ &\dots \cdot \left. \frac{\Delta T_{23}}{\eta_m} \cdot \frac{1}{1 - K\left(\frac{N_0}{N_{0D.P.}}\right)} \cdot \frac{1}{\eta_{GT} \cdot T_5} \right]^{\gamma_1/\gamma_1-1} \cdot \left[1 - \gamma_2/\gamma_2\right] \dots(6.11) \end{aligned}$$

The heat input $(Q_{in}) = (h_5 - h_3)_{m_a}$, thus

$$\begin{aligned} Q_{in} &= m_a (C_{p_5} \cdot T_5 - C_{p_3} \cdot T_3) \\ &= m_a [C_{p_5} \cdot T_5 - C_{p_3} (T_2 + \Delta T_{23})] \end{aligned} \quad \dots (6.12)$$

The thermal efficiency of the cycle is:

$$\eta_{th} = P_{out} / Q_{in}$$

to simplify equation (6.11), m_a and m_g may be assumed equal and variation of C_p and γ about its mean values at design point may be ignored.*

Equations (6.11) and (6.12) can be written in non-dimensional forms as:

*The reason why this does not lead to much inaccuracy is that C_p and γ vary in opposite senses with T . For cycle analysis we are interested in calculating compressor and turbine work from the product $(C_p \cdot \Delta T)$. Suppose that the temperature for which the assumed values of C_p and γ are true is lower than the actual mean temperature. γ is then higher than it should be and ΔT will be overestimated. This will be compensated in the product $(C_p \cdot \Delta T)$ by the fact that C_p will be lower than it should be.

$$\frac{P_{out}}{m_a C_{p_a} T_2} = \frac{\Delta T_{23}}{T_2} \cdot \frac{K \left(\frac{N_0}{N_{0D.P.}} \right)}{1 - K \left(\frac{N_0}{N_{0D.P.}} \right)} + C_1 \cdot \eta_{PT} \left(\frac{T_5}{T_2} \right) - C_2 \dots$$

$$\dots \cdot \frac{\Delta T_{23}}{T_2} \cdot \frac{1}{1 - K \frac{N_0}{N_{0D.P.}}} \cdot [1 - \{K_p \dots$$

$$\dots \cdot \left(1 + \frac{\eta_C \Delta T_{23}}{T_2}\right)^{\gamma/\gamma-1} \cdot \left(1 - \frac{C_2}{\eta_{GT}} \cdot \frac{\Delta T_{23}}{T_2} \dots$$

$$\dots \cdot \frac{T_2}{T_5} \cdot \frac{1}{1 - K \frac{N_0}{N_{0D.P.}}} \cdot \left(\gamma_1/\gamma_1-1 \right)^{1-\gamma_2/\gamma_2} \dots (6.13)$$

$$\frac{Q_{in}}{m_a C_{p_a} T_2} = C_3 \cdot \frac{T_5}{T_2} - C_4 \left(1 + \frac{\Delta T_{23}}{T_2}\right) \dots (6.14)$$

Where C_1, C_2, C_3 and C_4 are equal to: $C_{p_{PT}}/C_{p_a}, C_{p_a}/C_{p_C} \cdot \eta_m,$
 C_{p_5}/C_{p_a} and C_{p_3}/C_{p_a} respectively. P_{out} and Q_{in} can be
 written in terms of compressor pressure ratio PR instead of compressor

temperature rise ΔT_{23} as follows:

$$\begin{aligned}
 \frac{P_{out}}{m_a C_{p_a} T_2} &= \left(\frac{PR^{\gamma-1/\gamma}}{\eta_C} - 1 \right) \cdot \frac{K \left(\frac{N_0}{N_{0,D.P.}} \right)}{1 - K \left(\frac{N_0}{N_{0,D.P.}} \right)} + C_1 \cdot \eta_{PT} \dots \\
 \dots &\cdot \left(\frac{T_5}{T_2} - C_2 \cdot \left(\frac{PR^{\gamma-1/\gamma}}{\eta_C} - 1 \right) \cdot \frac{1}{1 - K \left(\frac{N_0}{N_{0,D.P.}} \right)} \right) \dots \\
 \dots &\cdot \left[1 - \left\{ k_p \cdot PR \cdot \left(1 - \frac{C_2}{\eta_{GT}} \cdot \left(\frac{PR^{\gamma-1/\gamma}}{\eta_C} - 1 \right) \frac{T_2}{T_5} \right) \right. \right. \\
 \dots &\cdot \left. \frac{1}{1 - K \left(\frac{N_0}{N_{0,D.P.}} \right)} \right\}^{\gamma_1/\gamma_1 - 1} \left. \right\}^{1 - \gamma_2/\gamma_2} \dots \dots (6.15)
 \end{aligned}$$

$$\frac{Q_{in}}{m_a C_{p_a} T_2} = C_3 \cdot \frac{T_5}{T_2} - C_4 \left(1 + \frac{PR^{\gamma-1/\gamma}}{\eta_C} - 1 \right) \dots (6.16)$$

As far as the specific work output is concerned, the addition of a heat exchanger merely causes a slight reduction due to the additional pressure losses. Therefore equation (6.15) still applies, taking into account that the additional pressure loss in the heat exchanger air and

gas sides is included in the value of K_p . The major task of a heat exchanger is to reduce the heat input of the cycle. This depends essentially on the heat exchanger effectiveness ϵ . The saving in heat input $Q_{H.Ex}$ is given by:

$$Q_{H.Ex} = m_a \cdot C_{p_m} \cdot \epsilon (T_7 - T_3) \quad \dots (6.17)$$

where: $T_3 = T_2 + \Delta T_{23}$ or $T_3 = T_2 \left(1 + \frac{PR^{\gamma-1/\gamma} - 1}{\eta_c}\right) \quad \dots (6.18)$

and : $T_7 = T_6 - \Delta T_{67}$

$$= \left(T_5 - \frac{m_a}{m_g} \cdot \frac{C_{p_a}}{C_{p_g}} \cdot \frac{T_2 (PR^{\gamma-1/\gamma} - 1)}{\eta_c \cdot \eta_m \cdot \left(1 - K \frac{N_0}{N_{0D.P.}}\right)} \right) \dots$$

$$\dots \cdot \left[1 - \eta_{pT} \cdot \left\{ 1 - \left[K_p \cdot PR \cdot \left(1 - \frac{m_a}{m_g} \cdot \frac{C_{p_c}}{C_{p_{GT}}} \right) \right] \right\} \right] \dots$$

$$\dots \cdot \frac{T_2}{T_5} \cdot \frac{1}{\eta_c \cdot \eta_{GT} \cdot \eta_m} \dots$$

$$\dots \cdot \frac{(PR^{\gamma-1/\gamma} - 1)}{1 - K \frac{N_0}{N_{0D.P.}}} \cdot \left(\frac{\gamma_1/\gamma_1 - 1}{1 - \gamma_2/\gamma_2} \right) \dots (6.19)$$

The heat input becomes:

$$Q_{in} = m_a [C_{p5} T_5 - C_{p3} (T_2 + \Delta T_{23})] - Q_{H.Ex} \quad \dots (6.20)$$

In practice most gas turbines utilize either a higher pressure ratio simple cycle or a low pressure ratio heat-exchange cycle [23]. Virtually all proponents of the vehicular gas turbine have now reverted to a low pressure ratio unit with a heat exchanger. [23]

The effect of heat exchanger in improving cycle efficiency is sound. It reduces the specific fuel consumption substantially. It was thus intended while discussing the selection of the design point parameters of the simple cycle co-turboshaft gas turbine to consider the transformation into a regenerative cycle.

It can be noticed from equations (6.13, 6.14, 6.15 and 6.16) that:

- 1 - As far as the heat input is concerned, there is not any difference between a co-turboshaft engine and a conventional gas turbine engine.
- 2 - A co-turboshaft engine and a conventional engine that have same component characteristics and design point parameters also have the same design point power and efficiency. Performance characteristics start to be different as the engines depart from design point speeds where the factor $(K \cdot \frac{N_0}{N_{0D.P.}})$ starts to produce its favourable effect on engine performance.
- 3 - A co-turboshaft engine performance again approaches the performance of a conventional engine at very low speeds where the effect of casing

rotation on compressor performance and on power transfer becomes insignificant.

- 4 - A co-turboshaft engine has a substantially different load sharing between gas generator turbine and power turbine at all operating points as shown from equations (6.4) and (6.5). This characteristic allows both turbines to have much better stage performances.
- 5 - Compressor pressure ratio and temperature rise are dependent on compressor effective speed (N_{rel}). Therefore both pressure ratio (PR) and temperature rise (ΔT_{23}) are function of, not only N_{GG} but also N_C . This relationship can be expressed as:

$$PR = f_1 \left\{ N_{GG} \cdot \left(\frac{1 - K \frac{N_0}{N_{0,D.P.}}}{1 - K} \right) \& w \right\} \quad \dots (6.21)$$

$$\Delta T_{23} = f_2 \left\{ N_{GG} \cdot \left(\frac{1 - K \frac{N_0}{N_{0,D.P.}}}{1 - K} \right) \& w \& \eta_C \right\}$$

This shows that, for a fixed gas generator speed and maximum cycle temperature, the output power is variable because of the variation of compressor pressure ratio and mass flow. In other words, the operating points on the compressor map are no more points but become lines, the whole forming an operating zone as shown in Figure 42.

Figure 42. Co-Turboshaft Engine Operating Lines on Compressor Map

6.3 Selection of Design Point Compressor Pressure Ratio and Maximum Cycle Temperature

A Regenerative-Low pressure ratio gas turbine engine has proven to perform successfully in many vehicular applications. No very high pressure ratio is ever required for a heat-exchange cycle, and the increase in weight and cost due to a heat-exchanger is partially offset by the reduction in size of the compressor [23].

In the case of the proposed application of the co-turboshaft engine, to power heavy duty vehicles, vehicle design can accommodate a heat-exchanger quite easily. Also, a low pressure ratio cycle does not need twin-spool or three-spool arrangement. It is therefore suitable for the

co-rotating compressor arrangement. Consequently, it is reasonable to consider for selection pressure ratios ranging from 4 to 8.

The advantage of using the highest possible value of maximum cycle temperature (T_5) is obvious. Cycle efficiency increases with T_5 because the component losses become relatively less important as the ratio of positive turbine work to negative compressor work increases. However, the gain in efficiency of simple cycles becomes marginal, as T_5 is increased beyond 1200 K [23], particularly if a higher temperature requires a complex turbine blade cooling system which incurs additional losses. When a heat-exchanger is used, the gain in efficiency is no longer merely marginal as T_5 is raised above 1200 K. This can be seen from Figure 43, which presents the cycle efficiency based on cycle analysis of section 6.2 using typical values for component losses and allowing PR, T_5 and K to vary independently. The other results are shown in Figures 44 to 46.

As mentioned before, the design limitations is the main factor which governs the selection of the maximum cycle temperature. A design point maximum cycle temperature of 1150 K is thought to be a reasonable selection since it is possible to achieve this in practice. It allows also for some increase in T_5 over 1150 K for overloading conditions.

Figure 43 shows that the optimum pressure ratio for maximum efficiency for a regenerative cycle is lying between 5 and 6 for T_5 ranging from 1150 K to 1300 K. The optimum pressure ratio for maximum specific output is remarkably higher than 6. Thus a design point pressure ratio of 6.2 is chosen to provide good specific output without

Figure 43. Efficiency of Basic Co-Turboshaft Engine

Figure 44. Specific Work Output of Basic Co-Turboshaft Engine

significant sacrifice in efficiency as shown in Figure 43 and 44. Results of the co-turboshaft engine performance based on the hybrid computer model confirms this selection. This will be shown later in section 6.6 , Figure 54.

6.4 Gear Box Speed Ratio K

As mentioned previously, the gear box speed ratio K indicates the rotation of the compressor casing in relation to its core at design point speeds. As K approaches zero, the co-turboshaft engine loses its special characteristics and turns out to be a conventional gas turbine engine. An excessive level of K means overspeeding of the gas generator rotor to a much greater speed than the speed of a conventional engine for same design point power output. This will result in increased mechanical problems. A value of $K = 0.2$ is considered to be the upper limit, for at such a value the gas generator rotor is 25% faster than the case of conventional engine. This in turn increases centrifugal effects, which is proportional to the square of the speed, to about 56%

The fraction of power transmitted through the co-rotating casing is shown to be equal to $(K/(1 - k))$ of the compressor power at design point conditions (equation 6.2). Thus for $K = 0.1$ the casing transmits only 11% of the compressor power, while at $K = 0.2$ this fraction rises to 25%. From Figure 2, a 20% reduction of output speed N_0 due to overloading causes the compressor relative speed N_{rel} to increase only by 2.2% for $K = 0.1$, while N_{rel} increases by 5.0% for $K = 0.2$. Since the performance of co-turboshaft engine is basically dependent on K , a

value of K of 0.1 or less is excluded. Marginal improvement in performance when using small values of K does not justify the additional complexity in the mechanical design of the engine. It is advisable then to have K as high as possible close to the chosen limit of 0.2.

Figures 45 and 46 show the effect of K on modifying the loading of turbine stages for different maximum temperatures. In the case of $T_5 = 1150$ K and $PR = 6.2$, Figure 45 shows that using $K = 0.2$ results in gas generator turbine pressure ratio r_1 of 3.0 and power turbine pressure ratio of 1.9. These values allow a superior loading of turbine stages when the gas generator turbine is a two-stage turbine. Thus K has been chosen to have the value of 0.2.

6.5 Effect of Location of Design Point on Power Turbine Map

The power turbine is assumed to be rotor-choked as shown in section 4.2.10. The design point turbine pressure ratio is sufficiently high to cause the power turbine to operate under choking conditions at almost all speeds. The effect of a reduction of speed of the power turbine on its flow characteristics in such a case is not sound, as is the case of power turbine efficiency which drops sharply as the speed decreases beyond certain region, irrespective of pressure ratio. This can be seen from Figure 22. It is therefore important to carefully locate the design point on the efficiency-speed curves to ensure the best performance, according to the expected variation of power turbine speed. In case of vehicular gas turbines, notably the co-turboshaft engine, the power turbine shaft that is mechanically coupled to the load, undergoes a considerable variation of speed. In addition it might be operating at low speeds for prolonged periods of its operating cycle.

Figure 45. Effect of Design Speed Ratio (K) on Turbines Pressure Ratio (r_1 & r_2)

Figure 46. Effect of Design Speed Ratio (K) on Turbines Pressure Ratio (r_1 & r_2)

In order to illustrate the effect of speed variation on turbine efficiency, the constant pressure ratio lines of Figure 22 can be replaced by a single line as shown in Figure 47. This does not cause a significant sacrifice of accuracy. It can be seen from Figure 47 that the power turbine efficiency curve is fairly flat over a fairly wide range of speed variation. Locating the design point at point 0, where η is a maximum, causes the efficiency to deteriorate very fast as power turbine speed decreases. Locating the design point at point 2, for example, although it reduces design point efficiency slightly, it maintains a fairly good efficiency over a wide range of speeds. This is primarily important where heavy duty vehicles have to perform the major part of their operating cycle at speeds less than the design point speed.

The effect of location of design point on power turbine map on

Figure 47. Location of Design Point on Power Turbine Map

engine performance is shown in Figures 48 and 49. These Figures are the results of the co-turboshaft engine hybrid computer model which assumes a compressor pressure ratio of 6.2, a maximum cycle temperature of 1150 °K and a design point speed ratio of 0.2. Figures 48 and 49 show that, with a little sacrifice of design point performance, a considerable gain in performance is obtained at lower output speeds if the location of design point on power turbine map is moved from say point 1 to point 2 in Figure 47. Thus point 2, which has an efficiency of 85%, was selected for the proposed engine to be the location of design point on the power turbine map.

Figures 48 and 49 present also the case of overspeeding of the output shaft. An increase of output speed reduces compressor relative speed and consequently reduces compressor pressure ratio and mass flow. This produces a steep torque-speed characteristics as observed from Figure 48.

Figure 48. Effect of Location of Design Point on Power Turbine Map on Co-Turboshaft Engine Performance, I

Figure 49. Effect of Location of Design Point on Power Turbine Map on Co-Turboshaft Engine Performance, II

6.6 Effect of Location of Design Point on Compressor Map

It can be seen from Figure 49 that the performance of the co-turboshaft engine is characterized by the remarkable increase in compressor mass flow w , as the output speed N_0 decreases. The change in maximum cycle temperature T_5 is very small for a considerable range of output speed. Since the operating point on compressor map, in the case of a conventional engine, now turns out to be a line, in the case of a co-turboshaft engine, then the proper selection of the design point location makes this line closely follow a constant TIT line as shown in Figure 50.

Figure 50. Location of Design Point on Compressor Map.

To take advantage of the steep increase in mass flow and pressure ratio, with an increase in compressor relative speed, the design point should be located on a relatively low speed line. In this case the advantage of maintaining a high compressor efficiency is also added. Locating the design point on a relatively low speed line means a bigger compressor for the same installed power. This is the main disadvantage of such a location. Therefore a compromise between better performance in one hand, and compressor oversizing on the other hand determines the final selection.

To examine the effect of the location of the design point on engine performance, the compressor map of Figure 9 was reproduced with different pressure ratios and mass flow scales to allow the moving of the design point, from one speed line to another, while keeping the same shape of compressor map and the same design point pressure ratio and mass flow. The resulting maps are shown diagrammatically in Figure 51. The design point locations 1, 2 and 3 are corresponding to locations 1, 2 and 3 of Figure 50 but all have the same pressure ratio of 6.2 and mass flow of 3.63 Kg/s.

The effects of design point locations 1, 2 and 3 on co-turboshaft engine performance is shown in Figures 52 and 53. The merits of location 3 are sound in all aspects of performance: higher values of power and torque, lower specific fuel consumption and perfectly constant maximum cycle temperature. However, the increase in compressor size remains a serious disadvantage. On the other hand, moving design point location

from 1 to 2 does not seem to have significant effect on power and torque characteristics as indicated in Figure 52. But obvious improvement in specific fuel consumption and much better variation of maximum cycle temperature are observed from Figure 53. Therefore locating the design point on location 2 ($85\% N/\sqrt{\theta}$) seems to be the best selection.

Figure 51. Comparison Between Different Design Point Locations

It is worth noting that the previous comparison is based on a fixed compressor pressure ratio of 6.2. However three values of pressure ratios are selected and their effects on engine performance are shown in Figures 54 and 55. It can be seen from Figure 54 that the gain in power and torque becomes marginal as compressor pressure ratio goes beyond 6.2.

Figure 52. Effect of Design Point Location on Compressor Map on Co-Turboshaft Engine Performance, I

Figure 53. Effect of Design Point Location on Compressor Map on Co-Turboshaft Engine Performance, II

Figure 54. Effect of Design Point Compressor Pressure Ratio on Performance of Co-Turboshaft Engine, I

Figure 55. Effect of Design Point Compressor Pressure Ratio on Performance of Co-Turboshaft Engine, II

Figure 55 shows a remarkable improvement in specific fuel consumption as pressure ratio goes from 6.2 to 6.9. But, it should always be remembered that a co-turboshaft engine is designed to be a regenerative cycle engine. Thus the effect of increasing pressure ratio will be less strong. 6.2 can be considered a reasonable maximum value for a single spool compressor to avoid surge. Incidentally, as output shaft speed is reduced, operating line moves away from surge. Hence when accelerating with low power turbine speed, the acceleration is expected to be "safer" than for a conventional engine.

6.7 Effect of Shape of the Compressor Map on Co-Turboshaft Engine

Performance

It might come to mind that the performance of the co-turboshaft engine is in the manner shown because of the particular shape of the compressor characteristics used (NASA-8 stage). To examine this as a feasible assumption an Orenda compressor, which has clearly different shape, was incorporated in the engine model.

Two different locations of design point on the Orenda compressor map are selected, as shown in Figure 56, and the compressor is rescaled to have same design point parameters as the NASA compressor. Results of engine performance are presented in Figures 57 and 58. In spite of the obvious change in shape of the compressor map, the performance curves maintains its characteristics to a great extent. This feature of the co-turboshaft engine, that is to adopt different shape compressor characteristics, adds to its merits.

Figure 56. Orenda II Compressor Performance Map

Figure 57. Co-Turboshaft Engine Performance with Orenda Compressor, I

Figure 58. Co-Turboshaft Engine Performance with Orenda Compressor, II

It is interesting to notice that using a compressor which has characteristics such as the Orenda compressor one, minimizes the problem of compressor oversizing of a co-turboshaft engine to a great extent. A design point located on the 95% compressor speed line (point 1) has produced a satisfactory performance. However, moving the design point to the 90% compressor speed line (point 2) results in a much better performance, as shown in Figures 57 and 58.

6.8 Performance Comparison Between Co-Turboshaft Engine and Conventional Engine

From cycle analysis presented in section 6.2, it can be seen that when setting the speed ratio K equal to zero the co-turboshaft engine loses its distinguishing feature and turns out to be a conventional gas turbine engine. This characteristic helps very much in holding a comparison between the performance of the two kinds of engines with the same component characteristics and the same design parameters.

Figure 59 and 60 present the results of this comparison. Figure 59 shows the substantial gain in engine output power and torque as the engine operates in the co-turboshaft mode (curve 1).^{*} The slight increase of turbine inlet temperature, as shown in Figure 60 is not likely to cause engine overheating within a considerable range of speed. It is also observed from Figure 60 that in the case of conventional engine there is a decrease of turbine inlet temperature along with a decrease

^{*}It may be noticed, from Figure 59, that the torque-speed characteristics of the conventional engine (curve 2) looks steeper than it is in most free-turbine gas turbine engines where the torque-speed curve approximates a straight line. This is mainly because the design point of the modelled conventional engine is located on point 2, Figure 47, rather than the best efficiency point 0, to allow for fair comparison with the co-turboshaft engine which has its design point located on point 2.

Figure 59. Performance Comparison Between Co-Turboshaft Engine and Conventional Engine, I

Figure 60. Performance Comparison Between Co-Turboshaft Engine and Conventional Engine, II

of output speed. This is mainly due to the characteristics of the power turbine which is assumed to be rotor choked and consequently the non-dimensional mass flow $w\sqrt{T/P}$ increases as output speed decreases causing a drop of turbine inlet temperature.

The slight increase in output power along with a reduction of output speed in the case of a stationary casing mode (curve 2), is due to the increase in power turbine efficiency as illustrated in Figure 47.

A slight increase in specific fuel consumption is noticed in the case of a co-turboshaft engine as output speed decreases beyond 50%. This is mainly due to a reduction of compressor efficiency as the operating point departs from the design point.

6.9. Temperature Characteristics of Co-Turboshaft Engine

As shown in section 6.2,

$$\begin{aligned} W_{\text{GGT}} &= (W_{\text{comp}} + W_{\text{casing}})/\eta_m \\ &= W_{\text{comp}} \left(1 + \frac{W_{\text{casing}}}{W_{\text{comp}}} \right) / \eta_m \end{aligned}$$

and from equation (6.2):

$$W_{\text{GGT}} = W_{\text{comp}} \left(1 + \frac{K \cdot \left(\frac{N_0}{N_{0\text{D.P.}}} \right)}{1 - K \left(\frac{N_0}{N_{0\text{D.P.}}} \right)} \right) / \eta_m \quad \dots (6.22)$$

Also:

$$\begin{aligned} W_{\text{GGT}} &= m_g C_{p_g} \Delta T_{56} \\ &= m_g C_{p_g} T_5 \cdot \left(\frac{\Delta T_{56}}{T_5} \right), \text{ thus:} \end{aligned}$$

$$m_g C_{p_g} T_5 \left(\frac{\Delta T_{56}}{T_5} \right) = m_a C_{p_a} \Delta T_{23} \left(1 + \frac{K \left(\frac{N_0}{N_{0,D.P.}} \right)}{1 - K \left(\frac{N_0}{N_{0,D.P.}} \right)} \right) / \eta_m \quad \dots (6.23)$$

Now, considering the operation of two turbines in series, shown in Figure 61, it can be seen that the requirement for flow compatibility between the two turbines places a major restriction on the operation of the gas generator turbine.

Figure 61. Operation of Turbines in Series

As long as the power turbine is choked the gas generator turbine will operate at a fixed non-dimensional point, i.e., at the pressure ratio marked (a). This in turn fixes the values of $w\sqrt{T_5}/P_5$ and $(\Delta T_{56}/T_5)$.

Since the power turbine is assumed to be choked, the value of $\frac{\Delta T_{56}}{T_5}$ is essentially constant, thus equation (6.23) can be written in the form:

$$T_5 \propto \Delta T_{23} \left(1 + \frac{K \left(\frac{N_0}{N_{0D.P.}} \right)}{1 - K \left(\frac{N_0}{N_{0D.P.}} \right)} \right) \quad \dots (6.24)$$

or in the form:

$$T_5 \propto \frac{1}{\eta_c} (PR^{\gamma-1/\gamma} - 1) \cdot \left[1 + \frac{K \left(\frac{N_0}{N_{0D.P.}} \right)}{1 - K \left(\frac{N_0}{N_{0D.P.}} \right)} \right] \quad \dots (6.25)$$

The gas generator turbine pressure P_5 can be written in the form:

$$P_5 = P_2 \times \frac{P_3}{P_2} \times \frac{P_5}{P_3}$$

where $\frac{P_5}{P_3}$ is the combustion chamber pressure loss factor $K_{C.C.}$ which can be assumed constant, thus:

$$P_5 = P_2 \times PR \times K_{C.C.} \quad \text{or}$$

$$PR = \frac{P_5}{P_2} \cdot \frac{1}{K_{C.C.}}$$

$$= A \cdot P_5, \quad \text{where } A \text{ is a constant.}$$

Since the non-dimensional mass flow $w\sqrt{T_5}/P_5$ is constant, then:

$$\frac{PR}{PR_{D.P.}} = \frac{P_5}{P_{5,D.P.}} = \frac{w\sqrt{T_5}}{(w\sqrt{T_5})_{D.P.}} \quad \dots(6.26)$$

Thus:

$$PR = PR_{D.P.} \cdot w\sqrt{T_5} / (w\sqrt{T_5})_{D.P.} \quad \dots(6.27)$$

Substituting into equation (6.25):

$$T_5 \propto \frac{1}{\eta_C} \left((PR_{D.P.} \cdot \frac{w\sqrt{T_5}}{(w\sqrt{T_5})_{D.P.}})^{\gamma-1/\gamma} - 1 \right) \cdot \left[1 + \frac{K(\frac{N_0}{N_{0,D.P.}})}{1 - K(\frac{N_0}{N_{0,D.P.}})} \right] \dots(6.28)$$

or in another form:

$$\begin{aligned} \frac{T_5}{T_{5,D.P.}} &= \frac{\eta_{C,D.P.}}{\eta_C} \cdot \frac{\left((PR_{D.P.} \cdot \frac{w}{w_{D.P.}} \cdot \sqrt{\frac{T_5}{T_{5,D.P.}}})^{\gamma-1/\gamma} - 1 \right)}{(PR_{D.P.})^{\gamma-1/\gamma} - 1} \dots \\ &\quad \cdot \frac{\left[1 + \frac{K(\frac{N_0}{N_{0,D.P.}})}{1 - K(\frac{N_0}{N_{0,D.P.}})} \right]}{\left[1 + \frac{K}{1 - K} \right]} \dots(6.29) \end{aligned}$$

Equation 6.29 shows that the turbine inlet temperature T_5 is governed by three factors in the following ways:

- i. T_5 is inversely proportional to compressor efficiency η_c
- ii. T_5 is proportional to airflow w
- iii. T_5 is proportional to output speed N_0 (or inversely proportional to output speed reduction).

The third factor is a special character of the co-turboshaft engine. Setting $K = 0$ in equation (6.29) cancels the effect of output speed completely, which is the case of a conventional engine.

It is of interest to see how airflow has to increase, when bringing the output speed to half its rated value, and to zero, (at constant N_{GG}), in order to maintain a constant value of T_5 . Assuming no change in compressor efficiency, and let $K = 0.2$, with $\frac{\gamma}{\gamma-1} = 3.55$, the airflow can be written as:

$$\frac{w}{w_{D.P.}} = \left\{ \frac{1.25 PR_{D.P.}^{\gamma-1/\gamma} - 1.25}{0.2 \left(\frac{N_0}{N_{0D.P.}} \right)} + 1.0 \right\}^{\gamma/\gamma-1} / PR_{D.P.} \quad \dots (6.30)$$

$$\left[1 + \frac{\frac{N_0}{N_{0D.P.}}}{1 - 0.2 \left(\frac{N_0}{N_{0D.P.}} \right)} \right]$$

For $\frac{N_0}{N_{0D.P.}} = 0.5$,

$$\frac{w}{w_{D.P.}} = \left(1.125 - \frac{0.125}{PR_{D.P.}^{\gamma-1/\gamma}} \right)^{\gamma/\gamma-1} \quad \dots (6.31)$$

Resulting in :

PR	4	5	6	7
$\frac{w}{w_{D.P.}}$	1.151	1.171	1.187	1.200

TABLE 2

And for $\frac{N_0}{N_{0D.P.}} = 0$

$$\frac{w}{w_{D.P.}} = \left(1.25 - \frac{0.25}{PR^{Y-1/Y}} \right)^{Y/Y-1} \quad \dots (6.32)$$

Resulting in:

PR	4	5	6	7
$\frac{w}{w_{D.P.}}$	1.318	1.363	1.398	1.428

TABLE 3

TABLE 2 shows the increase in airflow which is necessary to maintain TIT. This increase in airflow must be accomplished along with an increase of compressor relative speed $N_{rel} = 1.111 N_{rel D.P.}$. For example, if the full output speed pressure ratio was, say 7 but the increase in airflow in bringing the output shaft to 50% N_0 was only say 15%, i.e., 5% below what is required to maintain $T_5 = T_{5 D.P.}$, the consequent result is a reduction of T_5 as indicated by equation (6.29). On the other hand, any reduction in compressor efficiency accompanying the increased compressor relative speed tends to increase T_5 as seen from equation (6.29). Thus, to have the engine operating at constant TIT or so, the location of the design point on the compressor map has to be carefully selected. Variations of airflow and compressor efficiency over the range of variation of compressor relative speed have to follow specific courses. This is what governs the selection of design point location. A constant TIT operation is well demonstrated in Figure 53, curve 3; the effect of location of the design point on TIT variation is obvious.

In conclusion, whether TIT increases or decreases, as the output shaft speed is reduced depends very much on the nature of the compressor characteristics and the chosen location of the compressor design point.

As the power turbine is assumed choked, over the entire range of operation, its inlet temperature varies essentially in the same way that TIT varies. This is because of the fixed $\left(\frac{\Delta T_{56}}{T_5}\right)$ operation of the gas generator turbine. This fact is illustrated in Figure 49, where the similarity between the T_6 and T_5 curves is clear. A

constant TIT operation, whether due to proper selection of design point or to control of power turbine flow capacity, results in a constant power turbine inlet temperature over the considered range of operation.

6.10 Power Turbine Flow Characteristics For a Constant TIT Operation

If the turbine inlet temperature is to be maintained constant as output shaft speed varies using a variable geometry power turbine, it becomes important to know the variation in its flow characteristics. The performance of the power turbine will obviously be affected by the position of the stators. But with careful design, the drop in turbine efficiency can be more than offset in maintaining a high turbine inlet temperature. In case of a co-turboshaft engine, power turbine area variation might be in a positive or negative way, depending on the shape of TIT curve as shown in Figure 53.

A facility for increasing the stator area is also advantageous with respect to starting and accelerating the gas generator. If the stators are rotated still further, the gas generator flow can be directed against the direction of rotation so that the flow impinges on the back of the power turbine blades. This can result in a substantial degree of engine braking which is extremely important for heavy vehicles.

Area variations of ± 20 percent can be obtained with acceptable losses in turbine efficiency. However, it is of interest to know the variation of stators area of the power turbine in the case of a co-turboshaft engine. The compatibility of flow between the compressor

and gas generator turbine results in:

$$\begin{aligned}\frac{T_5}{T_2} &= \left(\frac{Q_5}{Q_2} \cdot \frac{P_5}{P_3} \cdot \frac{P_3}{P_2} \right)^2, \text{ where } Q = w\sqrt{T}/P \\ &= \left(\frac{Q_5}{Q_2} \cdot K_{C.C.} \cdot PR \right)^2 \quad \dots(6.33)\end{aligned}$$

Since the gas generator turbine is always choked, then Q_5 is constant because A_5 is constant and equation (6.33) can be written as:

$$\frac{PR}{Q_2} = C' \cdot \sqrt{\frac{T_5}{T_2}}, \text{ where } C' \text{ is a constant} \quad \dots(6.34)$$

and for constant T_5

$$\frac{PR}{Q_2} = C \quad \dots(6.35)$$

The compatibility of flow between gas generator turbine and power turbine gives:

$$Q_6 = Q_5 \cdot \frac{P_5}{P_6} \cdot \sqrt{\frac{T_6}{T_5}} \quad \dots(6.36)$$

And from equations (6.6) and (6.7)

$$\begin{aligned}\frac{T_6}{T_5} &= 1 - \left(\frac{m_a}{m_g} \cdot \frac{C_{p_a}}{C_{p_g} \cdot \eta_m} \right) \cdot \frac{1}{1 - K \left(\frac{N_0}{N_{0D.P.}} \right)} \cdot \frac{T_2}{T_5} \cdot \frac{1}{\eta_C} \dots \\ &\dots \cdot (PR^{\gamma-1/\gamma} - 1) \quad \dots(6.37)\end{aligned}$$

$$\frac{T_6}{T_5} = 1 - \frac{A}{\eta_c} \frac{T_2}{T_5} \left(\frac{(PR)^{\gamma-1/\gamma} - 1}{1 - K \left(\frac{N_0}{N_{0D.P.}} \right)} \right) \quad \dots(6.38)$$

where $A = \frac{m_a}{m_g} \cdot \frac{C_{p_a}}{C_{p_g} \cdot \eta_m}$, and D.P. refers to design point values.

P_5/P_6 is determined from equations (6.6) and (6.8) as:

$$\frac{P_5}{P_6} = \frac{1}{\left[1 - \frac{A}{\eta_c} \frac{T_2}{T_5} \frac{(PR)^{\gamma-1/\gamma} - 1}{\eta_{GGT} \cdot \left(1 - K \frac{N_0}{N_{0D.P.}} \right)} \right]^{\gamma_1/\gamma_1-1}} \quad \dots(6.39)$$

Substituting from equations (6.38) and (6.39) into equation (6.36),

$$Q_6 = Q_5 \cdot \frac{\sqrt{1 - \frac{A}{\eta_c} \frac{T_2}{T_5} \frac{(PR)^{\gamma-1/\gamma} - 1}{(1 - K \left(\frac{N_0}{N_{0D.P.}} \right))}}}{\left[1 - \frac{A}{\eta_c} \frac{T_2}{T_5} \frac{(PR)^{\gamma-1/\gamma} - 1}{\eta_{GGT} \cdot \left(1 - K \frac{N_0}{N_{0D.P.}} \right)} \right]^{\gamma_1/\gamma_1-1}} \quad \dots(6.40)$$

Equation (6.40) is the general equation relating Q_6 to: Q_5 , T_5 , PR , K and output speed N_0 . In the case of constant T_5 operation,

$$Q_6 = Q_5 \cdot \frac{\sqrt{1 - \frac{B}{\eta_c} \left(\frac{PR^{\gamma-1/\gamma} - 1}{1 - K \frac{N_0}{N_{0D.P.}}} \right)}}{\left[1 - \frac{B}{\eta_c \cdot \eta_{GGT}} \left(\frac{PR^{\gamma-1/\gamma} - 1}{1 - K \frac{N_0}{N_{0D.P.}}} \right) \right]^{\gamma_1/\gamma_1-1}} \quad \dots(6.41)$$

or in another form:

$$Q_6 = Q_5 \cdot \frac{\sqrt{1 - \frac{B}{\eta_c} \left(\frac{(CQ_2)^{\gamma-1/\gamma} - 1}{1 - K \frac{N_0}{N_{0D.P.}}} \right)}}{\left[1 - \frac{B}{\eta_c \cdot \eta_{GGT}} \left(\frac{(CQ_2)^{\gamma-1/\gamma} - 1}{1 - K \frac{N_0}{N_{0D.P.}}} \right) \right]^{\gamma_1/\gamma_1-1}} \quad \dots(6.42)$$

$$\text{where } B = \frac{m_a}{m_g} \cdot \frac{C_{p_a}}{C_{p_c} \cdot \eta_m} \cdot \frac{T_2}{T_3}$$

Equation (6.41) can be written in a non-dimensional form as follows:

$$\frac{Q_6}{Q_{6,D.P.}} = \frac{\sqrt{\left\{1 - \frac{B}{\eta_C} \left(\frac{PR^{\gamma-1/\gamma} - 1}{1 - K \frac{N_0}{N_{0,D.P.}}} \right)\right\} / \left\{1 - \frac{B}{\eta_{C,D.P.}} \left(\frac{PR_{D.P.}^{\gamma-1/\gamma} - 1}{1 - K} \right)\right\}}}{\left[\left\{1 - \frac{B}{\eta_C \cdot \eta_{GGT}} \left(\frac{PR^{\gamma-1/\gamma} - 1}{1 - K \frac{N_0}{N_{0,D.P.}}} \right)\right\} / \left\{1 - \frac{B}{\eta_{C,D.P.} \cdot \eta_{GGT,D.P.}} \left(\frac{PR_{D.P.}^{\gamma-1/\gamma} - 1}{1 - K} \right)\right\} \right]^{\gamma_1/\gamma_1-1}} \quad (6.43)$$

Equation (6.43) relates variation of power turbine flow to only one independent variable, that is N_0 . It should be noted that PR and η_C are dependent on N_0 , while η_{GGT} remains fairly constant over the entire range of operation. Considering, for example, the case of a 50% reduction in output speed, with $K = 0.2$ and assuming constant compressor efficiency, then:

$$\frac{Q_6}{Q_{6,D.P.}} = 99\%$$

As the output speed approaches zero the ratio $(Q_6/Q_{6,D.P.})$ varies in a much steeper way due to the considerable variation of η_C . However, it will still be well below the recommended range of (100 ± 20) percent.

Results of the co-turboshaft engine computer model are shown in Figures 62 and 63. Comparing these with the performance curves of a basic co-turboshaft engine (curves 1, Figures 59 and 60), it can be seen that there is no significant difference in any of the performance curves. This is expected as indicated by the analysis of section 6.9.

Figure 62. Co-Turboshaft Engine Performance - Constant Turbine Inlet Temperature, I

Figure 63. Co-Turboshaft Engine Performance - Constant Turbine Inlet Temperature, II

CHAPTER 7

THE CO-TURBOSHAFT ENGINE - FURTHER ASPECTS OF PERFORMANCE

7.1 Introduction

In this chapter, specific aspects of co-turboshaft engine performance are presented. Performance of the Regenerative co-turboshaft engine is one of these aspects. Other aspects cover the effects of including the exit pressure loss due to variable flow swirl, the overloading characteristics, and Part-Load performance. An important comparison between the performance of co-turboshaft engine and the performance of a conventional engine with a sophisticated control system is also presented.

7.2 The Regenerative Co-Turboshaft Engine

As far as specific output is concerned, the addition of a heat-exchanger merely causes a slight reduction due to the additional pressure losses. Thus the output power and torque curves retain essentially the same form as those of the basic co-turboshaft engine. This is clear from Figure 64. However, the specific fuel consumption curves are very different, as shown in Figure 65. Heat exchange decreases specific fuel consumption substantially. The effect of heat-exchanger effectiveness on engine performance is sound. Not only does an increase in effectiveness

Figure 64. Performance of Regenerative Co-Turboshaft Engine, I

Figure 65. Performance of Regenerative Co-Turboshaft Engine, II

decrease specific fuel consumption appreciably, but it also reduces the value of the optimum pressure ratio of the cycle [23]. With continuous development in heat-exchangers design, an effectiveness of 0.9 becomes available. With cross flow rotary type compact heat exchanger, the problem of size and many other problems of heat exchanger performance can be reduced to a great extent.

7.3 Effect of Including Flow Swirl Pressure Loss on the Performance of a Co-Turboshaft Engine

As the power turbine speed decreases beyond its rated value, gas flow at turbine outlet will not be axial. This in turn means additional pressure loss in the exhaust duct. This is called swirl pressure loss ΔP_{swirl} . An estimation of ΔP_{swirl} is given in equation (4.30), section 4.2.12. Two cases are considered. The first is when a set of exhaust "deswirl" stators is used which recovers, say 80% of the swirl dynamic pressure. The second is the case of a straightener which recovers none of the swirl dynamic pressure. The performance in both of these cases is compared with the performance in the case of a basic co-turboshaft engine. Results of the comparison are shown in Figures 66 and 67. The remarkable difference between performance curves with inclusion of swirl pressure loss, and without it, emphasizes the importance of considering swirl pressure loss into engine model. Also it can be seen from these figures that using "deswirl" stators is necessary. A pressure recovering efficiency of such

Figure 66. Effect of Including Flow Swirl Pressure Loss on the Performance of Co-Turboshaft Engine, I

Figure 67. Effect of Including Flow Swirl Pressure Loss on the Performance of Co-Turboshaft Engine, II

stators of 80% or more is quite adequate. Figure 67 shows that the effect of swirl pressure loss on engine behaviour does not go beyond power turbine. Compressor and gas generator turbine parameters remain almost unaffected. This is mainly because of the nature of the relation between the two turbines in series. If the power turbine is choked, which is the case, it isolates gas generator components from variation of parameters behind power turbine.

7.4 Comparison Between the Performance of a Co-Turboshaft Engine and of a Conventional Engine With Sophisticated Control

It is of interest to know how a co-turboshaft engine would compare with a conventional gas turbine engine, incorporating sophisticated control systems that controls gas generator speed and turbine inlet temperature. In order to obtain power and torque curves identical to those of a co-turboshaft engine, gas generator speed was adjusted to a specific value for each setting of output speed. Performance curves of such an engine are presented in Figures 68 and 69. Figure 68 shows the gradual increase in gas generator speed as output speed decreases. A 6.2% increase in gas generator speed is required to provide the same stalling torque as in the co-turboshaft engine.

The turbine inlet temperature, as shown in Figure 69, increases gradually as output speed decreases. The airflow increases also, but at a considerably lower rate than in the case of a co-turboshaft engine. Such a conventional engine shows an improved specific fuel consumption when compared to a co-turboshaft engine. This is mainly because of the higher

Figure 68. Performance of a Conventional Engine with Sophisticated Control, I

Figure 69. Performance of a Conventional Engine with Sophisticated Control, II

levels of turbine inlet temperature and compressor pressure ratio.

In conclusion, both the co-turboshaft engine and the sophisticated control - conventional engine can play essentially the same role as far as power and specific fuel consumption are concerned. The co-rotating concept adds complexity to the mechanical design of the engine, although partially compensated by the reduction in gear box size. On the other hand, the conventional engine requires additional sophisticated control systems. Preference of one concept to another depends mainly on feasibility and costs of design.

7.5 Part-Load Performance of a Co-Turboshaft Engine

The performance characteristics of the individual components of a gas turbine engine may be estimated on the basis of previous experience or obtained from actual tests. When the components are linked together in an engine, the range of possible operating conditions for each component is considerably reduced. The problem is to find corresponding operating points on the characteristics of each component when the engine is running at a steady speed, or in equilibrium as it is frequently termed. The equilibrium running points for a series of speeds may be plotted on the compressor characteristic and joined up to form an equilibrium running line (or zone, depending on the type of gas turbine and load), the whole forming an equilibrium running diagram. Once the operating conditions have been determined, it is a relatively simple matter to obtain performance curves of power output and specific fuel consumption.

The variation of specific fuel consumption with reduction in power, which is referred to as part-load performance, is of major importance in applications where considerable running at low power setting is required. This would be the case for any vehicular gas turbine.

It is of interest to see how a co-turboshaft engine would compare to a conventional engine at part-load conditions. Such a comparison is shown in Figures 70 to 75. The case of engine overloading is also presented for comparison in these figures. It should be appreciated that although, for convenience of presentation of performance, the output speed has been chosen as the independent variable, in practice the fuel flow is the independent variable. A chosen value of fuel flow (and hence TIT) determines the gas generator speed and ultimately the output power.

Comparing Figure 70 with Figure 71, it can be seen that the co-turboshaft engine retains its advantage of having a much better torque characteristics. The gain in output power is considerable even at low output speeds, in spite of the fact that a co-turboshaft engine turns out to be a conventional engine as output speed approaches zero (K always appears in the analysis connected with N_0 in the form:

$K \cdot N_0/N_{0D.P.}$).

Gas generator speeds for a conventional engine have been adjusted for each setting of a co-turboshaft engine gas generator speed so that both engines have the same operating point on the compressor map. This provides a better base for comparison than having the same gas generator

Figure 70. Co-Turboshaft Engine - Part Load Performance; I

Figure 71. Conventional Engine - Part Load Performance, I

rotor speed. The compressor effective speed in the case of a co-turboshaft engine is N_{rel} while it is N_{GG} in the case of conventional engines.

They are related in the case of a co-turboshaft engine as shown in equation (3.3). For $K = 0.2$ this relationship becomes:

$$\frac{N_{rel}}{N_{rel_{D.P.}}} = 1.25 \left(\frac{N_{GG}}{N_{GG_{D.P.}}} - 0.2 \frac{N_0}{N_{0_{D.P.}}} \right)$$

Figures 72 and 73 show the main reasons for the improved power and torque-output speed characteristics in the case of a co-turboshaft engine. One of these reasons is the steep increase in mass flow as output speed decreases. Another reason is the higher turbine inlet temperature. The turbine inlet temperature has a direct impact on specific fuel consumption, as shown from Figures 74 and 75. A considerable improvement in specific fuel consumption is observed in the case of a co-turboshaft engine. This is also an interesting characteristic of the co-turboshaft engine.

Since the fuel consumption depends only on the gas generator parameters there will be only one value for each compressor speed. When combined with the output power data to give the specific fuel consumption, however, it is clear that the SFC, like the power output, will be a function of both compressor speed and power turbine speed, as shown in Figures 74 and 75.

The incorporation of a heat-exchanger will have little effect on

Figure 72. Co-Turboshaft Engine - Part Load Performance, II

Figure 73. Conventional Engine - Part Load Performance, II

Figure 74. Co-Turboshaft Engine - Part Load Performance, III

Figure 75. Conventional Engine - Part Load Performance, III

the equilibrium running line. The part-load behaviour of engines with and without heat-exchanger will be similar. The rapid drop in TIT with decreasing power is the basic cause of the poor part-load performance of the gas turbine. Although the design point specific fuel consumption is significantly reduced by incorporating a heat-exchanger, the shape of the SFC curves, at part-load conditions, is not basically altered. This is fundamentally due to the fact that in each case there is a similar drop in turbine inlet temperature as power is reduced. It remains true that the regenerative-co-turboshaft engine will have a considerable improvement in specific fuel consumption compared to a regenerative conventional engine, due to the higher TIT levels as shown in Figures 72 and 73.

CHAPTER 8

CONCLUSIONS

The feasibility of the co-turboshaft engine as a power plant for heavy duty vehicles has been examined. Although the investigation of engine performance presented in this thesis was based on an engine of specific design, the method used in the study provided a very flexible computer model that can be used and adapted to a wide variety of further studies.

A hybrid computer model of thermodynamic performance has been assembled for both a co-turboshaft engine and a conventional gas turbine engine with and without sophisticated control systems. The model allows incorporation of a heat-exchanger, a variable geometry power turbine, and components of different characteristics. The model also provides a very convenient tool to investigate a variety of co-turboshaft engine designs. The use of hybrid computer models proved to be an extremely flexible and cost-effective method of investigating the co-turboshaft engine design and performance. It provided a large amount of insight into the concept of a co-turboshaft engine.

A co-turboshaft engine has been proven to have a superior performance when compared with a conventional gas turbine engine. Substantial increase in output power, and consequently torque, has been observed as

output (or load) speed decreases. The basis for selection of design point parameters and location on component maps to produce such steep torque-speed characteristics has been presented. It has been found that the increase in output power is fundamentally due to the increase in airflow as output speed decreases. The turbine inlet temperature, however, does not change significantly over a considerable range of output speed.

A study of temperature characteristics of a co-turboshaft engine has indicated that turbine inlet temperature may maintain its design point value or it may increase or decrease, depending mainly on the design speed ratio K and the location of design point on the compressor map. The results of a specific co-turboshaft engine used in this study have shown that turbine inlet temperature remains almost constant over 40% of output speed reduction before starting to increase slightly. Thus the operating line on the compressor map nearly follows a constant temperature line.

The co-turboshaft engine has maintained its superior performance over the conventional engine at part-load conditions. In addition to its steep torque-speed characteristics, the co-turboshaft engine has showed a considerable improvement in specific fuel consumption. This is basically due to the higher levels of turbine inlet temperature and pressure ratio caused by the increase in compressor effective speed. Superiority of performance has been noticed not only at part-load conditions, which covers output powers ranging from 30% up to 100% of the design point power, but also at overloading conditions. The normal

operating line on the compressor map has turned out to be an operating zone in the case of co-turboshaft engines.

A conventional gas turbine engine which is provided with sophisticated systems, to control gas generator speed and power turbine stators area, has been modelled for the purpose of comparison with the co-turboshaft engine. Such control systems have been designed to allow this conventional engine to produce the same power-speed trajectory as do the co-turboshaft engine. Both engines have been shown to perform essentially in the same manner when gas generator speed is allowed to vary in a specific way.

In this study several aspects of co-turboshaft engine design and performance have been covered. However, there are many other aspects which need to be investigated. The dynamic performance of the co-turboshaft engine is one area that should be pursued. The computer model presented in this thesis is fully capable of performing such a task. Transient behaviour of the engine, accelerating and decelerating fuel schedules are examples of proposed areas for future work.

It is also recommended to consider for further study the case of a co-turboshaft engine of variable design speed ratio. This is particularly interesting at part-load conditions where design speed ratio can increase considerably, hence magnifying the effect of co-rotating casing, without overspeeding the gas generator rotor. This might result in a further improvement in part-load performance.

Much more is needed to be known about mechanical feasibility and complexity of a co-turboshaft engine. These are important factors

in judging the preference of the co-turboshaft engine over a conventional gas turbine engine. Consequently, this is another proposed area for future work.

REFERENCES

- [1] MacIsaac, B.B. "Prediction of Gas Turbine Performance by Hybrid Computer Simulation"
NRC Memorandum, AC-132
July, 1972.
- [2] Turunen, W.A. "The General Motors Research
Gollman, J.S. GT-309 Gas Turbine Engine"
S.A.E. Trans., 74(iv), 1965,
357-77.
- [3] Tarkir, R.A. "Gas Turbine Application in Transit
Buckel, H.H. Vehicles", Society of Automotive
Engineers, Inc., paper No. 780059,
1979.
- [4] Rafat Okelah "Hybrid Computer Modeling of a
Co-Turboshaft Engine."
NRC Report, LTR-ENG-80
May 1978.
- [5] Millar, D.A.J. "The Co-Turboshaft - A Novel Gas Turbine
Chappell, M.S. Engine Powerplant for Heavy Equipment"
Okelah, R. ASME paper No. 79-GT-132, March 1979.
- [6] Power Jets Limited "Improvements in or Relating to Internal
Howell, A.R. Combustion Turbine Power Plants"
British patent specification,
587528, 1944.
- [7] Chappell, M.S. "The Rotating Stator Concept",
Preliminary Calibration of the
Compressor in the Conventional
Configuration, National Research
Council of Canada, Report ME-234,
July 1970.
- [8] Millar, D.A.J. "A Hybrid Computer Model for a Rotating
Swiderski, A.A. Stator Compressor Test Rig", National
Chappell, M.S. Research Council, ME-241,
March 1974.

- [9] Chappell, M.S.
 Millar, D.A.H.
 Swiderski, A.A. "The Rotating Stator Concept -
 Experimental Performance Characteristics
 Compared with the Conventional Compressor"
 National Research Council, ME-242,
 July 1975.

- [10] Chappell, M.S.
 Millar, D.A.J. "Analytical and Experimental Studies
 of an Axial Compressor with
 Co-rotating Stators"
 Paper presented at the 3rd International
 Symposium on Air-Breathing Engines
 Paper No. 76-014, March 1976.

- [11] Norbye, J. "Goodbye to problems with emissions,
 mileage, and drivability?
 Turbine with a new Twist", Popular
 science, Vol. 215 No. 4, Oct. 1979.

- [12] Ball, G.A.
 Gumaer, J.I.
 Sebestyen, T.M. "The ERDA/chrysler Upgraded Gas Turbine
 Engine Objective and Design",
 paper No. 760279, Presented at SAE
 Automotive Engineering Congress and
 Exposition, Detroit, Michigan,
 Feb. 26, 1976.

- [13] Saravanamuttoo,
 H.I.H.
 MacIsaac, B.D. "The use of a Hybrid Computer in the
 Optimization of Gas Turbine Control
 parameters"
 Transactions of the ASME
 Journal of Engineering for Power
 July 1973.

- [14] Stock, F.T.
 MacIsaac, B.D. "ENGLIB - Function Generation Routines"
 NRC Report, LTR-AN-19
 March 1975.

- [15] Gagné, R.E. "Bugoff - An On-Line Interactive
 Debugging Package for Digital and
 Hybrid Computers"
 NRC Report, LTR-AN-30
 Sept., 1976.

- [16] Gagné, R.E. "HOT - An Introduction NRC Report,
 LTR-AN-26
 Feb., 1976.

- [17] MacIsaac, B.D. "CIRBUG - An On-Line Interactive
 Package for Setup and Checkout of
 Analog Computer Circuits in a
 Hybrid Environment"
 NRC Report, LTR-AN-35
 Feb., 1979.

- [18] MacIsaac, B.D. "DCASET"
NRC program note S-078,
August 1976.
- [19] Evans, R. "Helicopter Propulsion Controls Computer
Langton, R. Model Specification for Twin Engine
MacIsaac, B.D. Helicopter Drive System"
Aviation Electric Limited AETR 36588,
April 1976.
- [20] MacIsaac, B.D. "JT15D Hybrid Computer Model"
NRC Notes, Aug. 1977.
- [21] Wiggins, J.O. "Some experiences in scaling the
Waltz, G.L. NASA 8 Stage Transonic Axial Flow
Compressor"
SAE Paper No. 720711, Sep. 1972.
- [22] Saravanamuttoo, "Simulation of Gas Turbine Dynamic
H.I.H. Performance"
Fawke, A.J. ASME Paper No. 70-GT-23
May 1970.
- [23] Cohen, H. "Gas Turbine Theory", 2nd Edition
Rogers, G.F.C. Longmans, 1972.
Saravanamuttoo,
H.I.H.
- [24] Mondt, J.R. "Vehicular Gas Turbine Periodic-Flow
Heat Exchanger Solid and Fluid
Temperature Distributions"
Transactions of the ASME
Journal of Engineering for Power
April 1964.
- [25] Cima, R.M. "The Transient Response of a Two-Fluid
London, A.L. Counterflow Heat Exchanger -
The Gas-Turbine Regenerator", Paper 57-A-135
Transactions of the ASME, Dec. 1957.
- [26] London, A.L. "The Transient Response of Gas Turbine
Sampsell, D.F. Plant Heat Exchangers Additional
McGowan, J.G. Solutions for Regenerators of the
periodic - Flow and Direct Transfer
Types"
Transactions of the ASME,
Journal of Engineering for Power
April 1964.

3

- [27] MacIsaac, B.D. "Techniques For Improving the Dynamic Response of the Gas Turbine - A Comparative Study Using A Hybrid Computer Model"
NRC Report, LTR-AN-13
Dec. 1974.

- [28] Gagné, R.E. "Computer Modeling and Simulation Handbook MKII"
NRC Report, LTR-AN-24
Oct., 1975.

- [29] Gagné, R.E. "Computer Modeling and Simulation Handbook"
Simulation, Vol. 26, No. 5, May 1976
pp. 147-161.

- [30] Millar, D.A.J. Personal Communication

- [31] Hodge, J. "Cycle and Performance Estimation"
Butterworths Scientific Publications,
London 1955.

BIBLIOGRAPHY

1. Ahlbeck, D.R. "Simulating a Jet Gas Turbine With an Analog Computer," Simulation, Sept. 1966.
2. Ainley, D.G.
Mathieson, G.C.R. "An Examination of the flow and pressure Losses in Blade Rows of Axial Flow Turbines," H.M.S.O., A.R.C., R. & M. 2891, 1955.
3. Ainley, D.G.
Mathieson, G.C.R. "A method of Performance Estimation for Axial-Flow Turbines," H.M.S.O., A.R.C., R & M. 2974, 1957.
4. Angell, P.R.
Golec, T. "Upgrading Automotive Gas Turbine Technology - An Experimental Evaluation of Improvement Concepts," paper No. 760280, SAE, Inc. 1976.
5. Bauerfeind, K. "Prac-A-New Aero Gas Turbine Engine Control Concept," AGARD 1974, pp. 13-1 - 13-14.
6. Bindon, J. Graduate Course in Turbomachinery Lecture Notes Carleton University, Canada Winter 1977.
7. Blajě, G.E. "A Study on Reynolds Number Effect in Turbomachines," Trans, ASME, Journal of Engineering for Power, July 1964.
8. Chappell, M.S.
Cockshutt, E.P. "Gas Turbine Cycle Calculations: Thermodynamic Data Tables for Air and Combustion Products" NRC Aeronautical Report LR-517, Jan. 1969.
9. Clarke, J.S.
Price, C.K.J.
Bottoms, C.H. "Fuel Control Systems for Small Gas Turbines", SAE Volume 9, 1965 Progress in Technology pp. 77-82.
10. Cockshutt, E.P. "Equilibrium Performance Analysis of Gas Turbine Engines using Influence Coefficient Techniques" AGARD 1974, pp. 22-1 - 22-10.

11. Cottington, R.V. "Total Powerplant Simulation", AGARD 1974 pp. 24-1 - 24D2.
12. Curran, J.J. "T700 Fuel and Control System", Journal of the American Helicopter Society, July 1974.
13. Curran, J.J.
Levin, A.M. "Development of A "No Adjustment" Turboshaft Engine Control System", American Helicopter Society, Preprint No. 79-42.
14. Doyle, G.N.
Wilkinson, T.S. "The Split Compressor Differential Gas Turbine Engine for Vehicle Propulsion" Journal of Engineering for Power January 1976.
15. Electronic Associates, Inc. "PACER 100 Digital Computing System, Application Programming Manual", Electronic Associates, Inc., Publ. No. 00827.0051-3 - May 1975.
16. Electronic Associates "EAI680 Scientific Computing System Reference Handbook, Publ. No. 00800.2048-3 - March 1970.
17. Fawke, A.J.
Saravanamuttoo, H.I.H. "Digital Computer Methods for Prediction of Gas Turbine Dynamic Response.", Society of Automotive Engineers, paper number 710550, 1971.
18. Fawke, A.J.
Saravanamuttoo, H.I.H. "Experimental Investigations of Methods of Improving the Dynamic Response of a Twin-Spool Turbojet Engine" Transaction ASME, Journal of Engineering for Power, pp. 418-424, Vol. 93, Oct. 1971.
19. Fawke, A.J.
Saravanamuttoo, H.I.H.
Homes, M.W. "Experimental Verification of a Digital Computer Simulation Method for Predicting Gas Turbine Dynamic Behaviour." Proc. I. Mech. E., Vol. 186 27/72, p. 323-329, 1972.
20. Gagné, R.E. "Computer Modeling and Simulation handbook ... MKIII" Simulation, May 1976.

21. Gardner, W.B.
Sample, F. "Dynamic Simulation - A Tool for Engine Component Design", Society of Automotive Engineers, Inc., paper No. 690386.
22. Glenny, R.J.E. "Advanced Materials for Gas Turbines" Proc. Instn. Mech. Engrs. 1968-69 Vol. 183 Pt 3N.
23. Horlock, J.H. "Axial Flow Compressors", Butterworth, 1958.
24. Horlock, J.H. "Axial Flow Turbines", Butterworth, 1966.
25. Howell, A.R. "Fluid Dynamics of Axial Compressors and Design of Axial Compressors", Proc. Instn mech. Engrs, 153 W.E.P. No. 12, 1945.
26. Heard, T.C.
Lang, R.P. "The Concept of Availability for Gas Turbine Application Evaluation.", Transactions of the ASME, Journal of Engineering for Power vol. 100, July 1978.
27. Itoh, M.
Ishigaki, T.
Sagiya, Y. "Simulation Study of Transient Performance Matching of Turbofan Engine Using an Analogue Computer to Evaluate Its Usefulness as Design Tool.", Trans. ASME, Journal of Engineering for Power, July 1975.
28. Jerie, J. "Part Load Specific Fuel Consumption of Gas Turbines"., Transactions of the ASME, Journal of Engineering for Power, April 1975.
29. Kuczynski, W.A.
Cooper, D.E.
Twomey, W.J.
Howlett, J.J. "The Influence of Engine/Fuel Control Design on Helicopter Dynamics and Handling Qualities". American Helicopter Society, Preprint 79-37.
30. Langton, R.
Evans, R. "Helicopter Propulsion Controls System Model for Twin-Engine Helicopter Power Plant". Aviation of Electric Limited AETR 36586 March 20, 1976.

31. Larrowe, V.L.
Spencer, M.M. "Analog Computer Simulation of Gas Turbine Engines for Control Study." SAE Volume 9, 1965 Progress in Technology pp. 3-10.
32. Leeson, P.J. "Gas Turbine Control Systems", AGARD 1974 pp. 10-1 - 10-12.
33. Leung, P.
Durning, R.F. "Power System Economics: On Selection of Engineering Alternatives.", Transaction of the ASME, Journal of Engineering for power, April 1978.
34. London, A.L.
Biancardi, F.R.
Mitchell "The Transient Response of Gas-Turbine Plant Heat Exchangers. Regenerators, Intercoolers, Precoolers, and Ducting".., Transactions of ASME, Journal of Engineering for Power, Oct. 1959.
35. MacIsaac, B.D. "Computer Model Specification for Twin Engine Helicopter Drive System.", AETR 36588, April 1976.
36. MacIsaac, B.D. "Techniques for Improving the Dynamic Response of Gas Turbine: A comparative Study Using A Hybrid Computer Model" NRC report No. LTR-AN-13, Dec 1974.
37. MacIsaac, B.D.
Saravanamuttoo, H.I.H. "A Comparison of Analog, Digital and Hybrid Computing Techniques for Simulation of Gas Turbine Performance". ASME paper No. 74-GT-127, April 1974.
38. MacIsaac, B.D.
Saravanamuttoo, H.I.H. "Aerothermodynamic Factors Governing the Response Rate of Gas Turbines" The Proceeding AGARD, Sept. 1974.
39. MacCallum, N.R.L. "Effect of 'Bulk' Heat Transfers in Aircraft Gas Turbines on Compressor Surge Margins," Instn Mech Engrs, Conference Publication 3, 1973.
40. MacCallum, N.R.L. "The Performance of Turbojet Engines During the "Thermal Soak" Transient.", Proc. Instn. Mech. Engrs. 1969-1970.
41. McLean, A.F. "Case for the Single-Shaft Vehicular Gas Turbine Engine" Proc. Instn. Mech. Engrs. 1968-69 Vol 183 Pt 3N.

42. Millar, D.A.J. "Compressible Flow Tables and Their Use", Carleton University, June 1976.
43. Millar, D.A.J. Graduate Course in Turbomachinery
Lecture Notes
Carleton University, Canada, Winter 1978.
44. Newburgh, F.W.
Ingold, B. "Industrial Gas Turbine Fuel Controls"
SAE volume 9, 1965 Progress in Technology pp. 70-76.
45. Nye, S.
Vickers, R.J. "Use of Simulation in the Design,
Development and Testing of Power Plant
Control Systems", AGARD 1974, pp. 25-1 -
25-13.
46. Pampreen, R.C. "The Use of Variable Inlet Guide Vanes
for Automotive Gas Turbine Engine
Augmentation and Load Control", P.
No. 7G0285, Society of Automotive
Engineers, Inc., 1976.
47. Rahnke, C.J. "The Variable - Geometry Power Turbine",
paper No. 690031, SAE.
48. Reed, W.E.
Murry, J.H. "Use of an Electronic Model to Evaluate
Control Concepts for a Regenerative
Gas Turbine", SAE vol 9, 1965 Progress
in Technology pp. 11-18.
49. Reilly, R.J. "All-Fluid Devices and Their Applications
to Gas Turbines.", SAE vol 9, 1965
progress in Technology pp. 99-106.
50. Rodgers, C. "A Cycle Analysis Technique for Small
Gas Turbines"
Proc. Instn. Mech. Engrs. 1968-69
Vol 183 Pt 3N.
51. Saravanamuttoo, H.I.H. "Analog Computer Study of the Transient
Performance of the Orinda 600-hp
Regenerative Gas Turbine"
ASME Publication, paper number
63-AHGT-38.
52. Saravanamuttoo, H.I.H. Graduate Course in Gas Turbines
Lecture Notes
Carleton University, Canada, Fall 1976.

53. Scott, J.E. "Performance Characteristics of Aero Gas Turbine Engines." AGARD 1974 pp. 1-1 - 1-13.
54. Shouman, A.R.
Anderson, J.R. "The Use of Compressor - Inlet Prewhirl for the Control of Small Gas Turbines.", Transactions of the ASME, Journal of Engineering for power, April 1964.
55. Shulhan, G.M.
Saravanamuttoo, H.I.H. "Variable Geometry Compressors for Improvement of Gas Turbine Part Load Performance.", Society of Automotive Engineers, paper number 760283, 1976.
56. Simonis, E.A.
Perks, M.P. "Helicopter Engine Control - The Past 20 Years and the Next," AGARD 1974, pp. 14-1 - 14-16.
57. Spinks, L.J. "Some Observations on the Selection of Gas Turbine Generating Plant," Gas Turbine. States and Prospects Symposium, 1976 pp. 147-156.
58. Stone, A. "Effects of Stage Characteristics and Matching on Axial-Flow-Compressor Performance." Trans. ASME. Paper No. 57-A-139 August 1958.
59. Sullivan, D.A. "Gas Turbine Combustor Analysis.", Transactions of the ASME, Journal of Engineering for Power, October 1975.
60. Tervo, W.K.
Tringali, J.M. "Control Design Considerations for Variable Geometry Engines", AGARD 1974 pp. 12-1 - 12-8.
61. Urban, L.A. "Parameter Selection of Multiple Fault Diagnostics of Gas Turbine Engines.", Trans ASME, Journal of Engineering for Power, April 1975.
62. Wong, J.Y. "Theory of Ground Vehicles" John Wiley & Sons, Inc., 1978.
63. Yerbury, P.A. "Gas Turbine Power for Large Hovercraft," Gas Turbine-Status and Prospects Symposium, 1976 pp. 117-124.

APPENDICES

- A MATHEMATICAL DERIVATIONS
- B SCALED EQUATIONS
- C DESIGN POINT CALCULATIONS - VERIFICATION TASK
- D POWERFUL TECHNIQUES IN SUPPORT OF HYBRID COMPUTER MODELLING
- E ANALOG CIRCUITS
- F LIST OF DIGITAL PROGRAMS

APPENDIX A

MATHEMATICAL DERIVATIONS

A.1 Inter Volumes Equation

From the gas law, $PV = mRT$, the rate of change of pressure of a gas contained in a volume V is:

$$\begin{aligned}\dot{P} &= \frac{R}{V} \frac{d}{dt} (mT) \\ &= \frac{R}{V} (\dot{m}T + m\dot{T})\end{aligned}\quad \dots (A.i)$$

The rate of change of the mass contained in a volume, $\dot{m}$, is equal to $(w_{in} - w_{out})$. Thus equation (A.i) becomes:

$$\dot{P} = \frac{R}{V} \cdot T(t) \cdot (w_{in} - w_{out}) + \frac{P}{T(t)} \cdot \frac{dT}{dt} \quad \dots (A.ii)$$

The change of state of the gas is most likely to follow a polytropic process. If such polytropic process has an index n , then:

$$T \propto P^{\frac{n-1}{n}}$$

or in other form: $T \propto P^{\left(\frac{\gamma-1}{\eta_p \cdot \gamma}\right)}$

where η_p is the polytropic efficiency. Thus:

$$T = C \cdot P^{\left(\frac{\gamma-1}{\eta_p \cdot \gamma}\right)} \quad \dots (A.iii)$$

consequently:

$$\begin{aligned}\frac{P}{T} \frac{dT}{dt} &= \frac{P}{T} \cdot C \cdot \frac{d}{dt} \left(P^{\left(\frac{\gamma-1}{\eta_p \cdot \gamma} \right)} \right) \\ &= \frac{P}{T} \cdot C \cdot \frac{\gamma-1}{\eta_p \cdot \gamma} \cdot P^{\left(\frac{\gamma-1}{\eta_p \cdot \gamma} \right)} \cdot P^{-1} \cdot \frac{dP}{dt}\end{aligned}$$

substituting from equation (A.iii),

$$\begin{aligned}\frac{P}{T} \cdot \frac{dT}{dt} &= \frac{P}{T} \cdot \frac{\gamma-1}{\eta_p \cdot \gamma} \cdot \frac{T}{P} \cdot \frac{dP}{dt} \\ &= \frac{\gamma-1}{\eta_p \cdot \gamma} \cdot \frac{dP}{dt}\end{aligned} \quad \dots (A.iv)$$

Substituting from equation (A.iv) into equation (A.ii), then:

$$\frac{dP}{dt} = \frac{RT}{V} (w_{in} - w_{out}) + \frac{\gamma-1}{\eta_p \cdot \gamma} \cdot \frac{dP}{dt}$$

which finally gives:

$$\frac{dP}{dt} = \frac{RT}{V \left(1 - \frac{\gamma-1}{\eta_p \cdot \gamma} \right)} \cdot (w_{in} - w_{out}) \quad \dots (A.v)$$

Equation (A.v) illustrates the effect of including the variation of gas temperature, in the inter-component volumes, during transient conditions. Including T in equation (A.i) results in reducing the effective volume from V to $V \left(1 - \frac{\gamma-1}{\eta_p \cdot \gamma} \right)$. For typical values of γ and η_p this reduction in V amounts to 30%. However, the value of V is chosen arbitrary. V might also be subject to changes of greater magnitude (than 30%) to achieve stability of transient operation. Therefore the term mT in equation (A.i) can be ignored without significant effect on engine transient performance.

A.2 The Rotating Outlet Guide Vanes Equations (ROGV)

The specific work of the ROGV row, W_s , is:

$$\begin{aligned}
 W_s &= U_C \cdot \Delta C_\theta \\
 &= U_C \cdot C_x \tan B_3 \\
 &= U_C \cdot \frac{\dot{m}}{\rho A} \cdot \tan B_3 \quad \dots (A.vi)
 \end{aligned}$$

where, ρ is the air density at compressor exit.

$$\rho = \frac{P_3}{RT_3}$$

$$\begin{aligned}
 A &= \frac{\pi}{4} (D_{tip}^2 - D_{hub}^2) \\
 &= \pi D_{mean} \cdot h
 \end{aligned}$$

$$U_C = \frac{\pi D_{mean} \cdot N_C}{60}$$

Substituting these values into equation (A.vi), then:

$$W_s = \left(\frac{R \cdot \tan \beta_3}{60 \cdot h} \right) \cdot \frac{w_3 \cdot T_3 \cdot N_C}{P_3} \quad \dots (A.vii)$$

Thus temperature drop in ROGV, ΔT_{34} can be determined to be:

$$\Delta T_{34} = \left(\frac{R \cdot \tan \beta_3}{60 h \cdot C_{P_{ROGV}}} \right) \cdot \frac{w_3 \cdot T_3 \cdot N_C}{P_3} \quad \dots (A.viii)$$

The isentropic temperature drop, ΔT_{34is} , therefore becomes:

$$\Delta T_{34is} = \frac{\Delta T_{34}}{\eta_{is}}$$

The pressure ratio (P_3/P_4) can be found from the relation:

$$\frac{P_3}{P_4} = \left(\frac{T_3}{T_{4is}} \right)^{\gamma/\gamma-1}, \text{ where}$$

$$\begin{aligned} T_{4is} &= T_3 - \Delta T_{34is} \\ &= T_3 - \frac{\Delta T_{34}}{\eta_{is}} \end{aligned}$$

Thus:

$$\frac{P_3}{P_4} = 1 / \left(1 - \frac{\Delta T_{34}}{\eta_{is} \cdot T_3} \right)^{\gamma/\gamma-1}$$

$$= 1 / \left\{ 1 - \left(\frac{R \cdot \tan \beta_3}{60 h \cdot C_{P_{ROGV}} \cdot \eta_{is}} \right) \dots \right.$$

$$\dots \left. \cdot \frac{w_3 \cdot N_C}{P_3} \right\}^{\gamma/\gamma-1} \quad \dots (A.ix)$$

APPENDIX B

SCALED EQUATIONS

B.1 Inlet System

$$\frac{P_a - P_l}{P_a} = K_{in} \left(\frac{w_0 \sqrt{\theta_a}}{\delta_a} \right) \quad , \text{ or}$$

$$w_0 = \frac{P_a - P_l}{P_a} \cdot \frac{1}{K_{in}} \cdot \frac{\delta_a}{\sqrt{\theta_a}}$$

In scaled form:

$$\left(\frac{w_0}{w_m} \right)^* = \left(\frac{\delta_a}{\delta_{am}} \right) \cdot \left(\frac{1}{\sqrt{\theta_a/\theta_{am}}} \right) \cdot \frac{(P_a/P_{am}) - (P_l/P_{lm}) \cdot [P_{lm}/P_{am}]}{[P_a/P_{am}]} \dots$$

$$\dots \times \left[\frac{\delta_{am}}{w_{om}} \cdot \frac{1}{\sqrt{\theta_{am}}} \cdot \frac{1}{\sqrt{K_{in}}} \right]^{**} \quad \dots(B.1)$$

where subscript m refers to maximum value (scale factor).

$$T_l = T_a$$

In scaled form:

$$\left(\frac{T_l}{T_m} \right) = \left(\frac{T_a}{T_{am}} \right) \cdot \left[\frac{T_{am}}{T_{lm}} \right] \quad \dots(B.2)$$

* Values between brackets () are scaled variables

** Values between square brackets [] are scaled constants

B.2 Inlet Volume

$$\frac{dP_1}{dt} = \frac{RT_1}{V_{in}} (w_0 - w_1)$$

- In scaled form:

$$\frac{d(P_1/P_{1m})}{d(t/t_m)} = \left(\frac{T_1}{T_{1m}}\right) \cdot \left\{ \left(\frac{w_0}{w_m}\right) - \left(\frac{w_1}{w_m}\right) \right\} \cdot \left[\frac{t_m}{P_{1m}} \cdot \frac{R}{V_{in}} \cdot T_{1m} \cdot w_m \right] \quad \dots (B.3)$$

B.3 Compressor

$$\theta_2 = \frac{T_2}{T_a} \quad , \quad \text{in scaled form}$$

$$\left(\frac{\theta_2}{\theta_{2m}}\right) = \frac{(T_2/T_{2m})}{(T_a/T_{am})} \cdot \left[\frac{T_{2m}}{T_{am}}\right] \cdot \left[\frac{1}{\theta_{2m}}\right] \quad \dots (B.4)$$

$$\delta_2 = \frac{P_2}{P_a} \quad , \quad \text{in scaled form}$$

$$\left(\frac{\delta_2}{\delta_{2m}}\right) = \frac{(P_2/P_{2m})}{(P_a/P_{am})} \cdot \left[\frac{P_{2m}}{P_{am}}\right] \cdot \left[\frac{1}{\delta_{2m}}\right] \quad \dots (B.5)$$

$$N_{map} = N_{rel}/\sqrt{\theta} \quad , \quad \text{in scaled form}$$

$$\left(\frac{N_{map}}{N_{map_m}}\right) = \frac{(N_{rel}/N_{rel_m})}{\sqrt{\theta_2/\theta_{2m}}} \cdot \left[\frac{N_{rel_m}}{N_{map_m} \cdot \sqrt{\theta_{2m}}}\right] \quad \dots (B.6)$$

$$Pr = P_3/P_2 \quad , \quad \text{in scaled form}$$

$$\left(\frac{Pr}{Pr_m}\right) = \frac{(P_3/P_{3m})}{(P_2/P_{2m})} \cdot \left[\frac{P_{3m}}{Pr_m \cdot P_{2m}}\right] \quad \dots (B.7)$$

The compressor mass flow and temperature rise can be obtained from compressor map using equations (B.6 and B.7). From compressor map, (Q_2/Q_{2m}) and $(\frac{\Delta T_{23}/T_2}{(\Delta T_{23}/T_2)_m})$ can be obtained. Thus

$$\left(\frac{w_2}{w_m}\right) = \left(\frac{Q_2}{Q_{2m}}\right) \cdot \left(\frac{\delta_2}{\delta_{2m}}\right) / \sqrt{\left(\frac{\theta_2}{\theta_{2m}}\right)} \cdot \left[\frac{Q_{2m} \cdot \delta_{2m}}{w_m \cdot \sqrt{\theta_{2m}}}\right] \quad \dots (B.8)$$

$$T_3 - T_2 = \Delta T_{23}$$

$$\left(\frac{\Delta T_{23}}{T_{3m}}\right) = \left(\frac{T_2}{T_{2m}}\right) \cdot \left(\frac{\Delta T_{23}/T_2}{(\Delta T_{23}/T_2)_m}\right) \cdot \left[\frac{(\Delta T_{23}/T_2)_m \cdot T_{2m}}{T_{3m}}\right] \quad \dots (B.9)$$

$$\left(\frac{T_3}{T_{3m}}\right) = \left(\frac{T_2}{T_{2m}}\right) \cdot \left[\frac{T_{2m}}{T_{3m}}\right] + \left(\frac{\Delta T_{23}}{T_{3m}}\right) \quad \dots (B.10)$$

Compressor power $P_{comp} = w_2 \cdot C_{p2} \cdot \Delta T_{23}$, thus

$$\left(\frac{P_{comp}}{P_m}\right) = \left(\frac{w_2}{w_m}\right) \cdot \left(\frac{\Delta T_{23}}{T_{3m}}\right) \cdot \left[\frac{w_m \cdot T_{3m} \cdot C_p}{P_m}\right] \quad \dots (B.11)$$

B.4 Compressor Volume

$$\frac{dP_3}{dt} = \frac{RT_3}{V_{comp}} (w_2 - w_3) \quad , \text{ in scaled form}$$

$$\frac{d(P_3/P_{3m})}{d(t/t_m)} = \left(\frac{T_3}{T_{3m}}\right) \cdot \left\{\left(\frac{w_2}{w_m}\right) - \left(\frac{w_3}{w_m}\right)\right\} \cdot \left[\frac{t_m}{P_{3m}} \cdot \frac{R}{V_{comp}} \cdot T_{3m} \cdot w_m\right] \quad \dots (B.12)$$

B.5 Combustor

$$P_5 = K_{C.C} \times P_4$$

$$\left(\frac{P_5}{P_{5m}}\right) = \left(\frac{P_4}{P_{4m}}\right) \cdot [K_{C.C} \cdot \frac{P_{4m}}{P_{5m}}] \quad \dots(B.13)$$

$$T_5 = T_4 + (A + B \cdot f_a + C \cdot f_a^2 + D \cdot f_a \cdot T_4)/ECV, \text{ thus:}$$

$$\begin{aligned} \left(\frac{T_5}{T_{5m}}\right) &= \left(\frac{T_4}{T_{4m}}\right) \cdot \left[\frac{T_{4m}}{T_{5m}}\right] + \left[\frac{A}{ECV \cdot T_{5m}}\right] + \left[\frac{3600 \times B}{ECV \cdot T_{5m}}\right] \cdot \left(\frac{w_f}{w_3}\right) \dots \\ &\dots + \left[\frac{(3600)^2 \cdot C}{ECV \cdot T_{5m}}\right] \cdot \left(\frac{w_f}{w_3}\right)^2 + \left[\frac{3600 \times D}{ECV \cdot T_{5m}}\right] \cdot \left(\frac{w_f}{w_3}\right) \cdot \left(\frac{T_4}{T_{4m}}\right) \dots(B.14) \end{aligned}$$

B.6 Combustor Volume

$$\frac{dP_4}{dt} = \frac{RT_4}{V_{C.C.}} (w_3 + w_f - w_4), \text{ in scaled form}$$

$$\frac{d(P_4/P_{4m})}{d(t/t_m)} = \left(\frac{T_4}{T_{4m}}\right) \cdot \left\{\left(\frac{w_3}{w_m}\right) + \left(\frac{w_f}{w_m}\right) - \left(\frac{w_4}{w_m}\right)\right\} \cdot \left[\frac{t_m}{P_{4m}} \cdot \frac{R}{V_{C.C.}} \cdot T_{4m} \cdot w_m\right] \dots(B.15)$$

B.7 Gas Generator Turbine

$$\text{Pressure ratio} = P_5/P_6, \text{ in scaled form:}$$

$$\left(\frac{PR}{PR_m}\right) = \frac{(P_5/P_{5m})}{(P_6/P_{6m})} \cdot \left[\frac{P_{5m}}{PR_m \cdot P_{6m}}\right] \quad \dots(B.16)$$

$$N_{map} = N_{GG}/\sqrt{T_5}$$

thus

$$\left(\frac{N_{map}}{N_{map_m}}\right) = \frac{(N_{GG}/N_{GG_m})}{\sqrt{(T_5/T_{5m})}} \cdot \left[\frac{N_{GG_m}}{N_{map_m} \cdot \sqrt{T_{5m}}}\right] \quad \dots(B.17)$$

The turbine mass flow and temperature drop can be obtained from gas generator turbine map using equations (B.16 and B.17). From turbine map, (Q_5/Q_{5m}) and $((\Delta T_{56}/T_5)/(\Delta T_{56}/T_5)_m)$ can be obtained. Thus:

$$\left(\frac{w_4}{w_m}\right) = \left(\frac{Q_5}{Q_{5m}}\right) \cdot \left(\frac{P_5}{P_{5m}}\right) / \sqrt{\left(\frac{T_5}{T_{5m}}\right)} \cdot \left[\frac{Q_{5m} \cdot P_{5m}}{w_m \cdot \sqrt{T_{5m}}}\right] \quad \dots(B.18)$$

$$T_5 - T_6 = \Delta T_{56}$$

$$\left(\frac{\Delta T_{56}}{T_{6m}}\right) = \left(\frac{T_5}{T_{5m}}\right) \cdot \left(\frac{\Delta T_{56}/T_5}{(\Delta T_{56}/T_5)_m}\right) \cdot \left[\frac{(\Delta T_{56}/T_5)_m \cdot T_{5m}}{T_{6m}}\right] \quad \dots(B.19)$$

$$\left(\frac{T_6}{T_{6m}}\right) = \left(\frac{T_5}{T_{5m}}\right) \cdot \left[\frac{T_{5m}}{T_{6m}}\right] - \left(\frac{\Delta T_{56}}{T_{6m}}\right) \quad \dots(B.20)$$

Gas generator turbine power $P_{GGT} = w_4 \cdot C_{p_{GGT}} \cdot \Delta T_{56}$, thus

$$\left(\frac{P_{GGT}}{P_m}\right) = \left(\frac{w_4}{w_m}\right) \cdot \left(\frac{\Delta T_{56}}{T_{6m}}\right) \cdot \left[\frac{w_m \cdot T_{6m} \cdot C_{p_{GGT}}}{P_m}\right] \quad \dots(B.21)$$

B.8 The Interturbine Volume

$$\frac{dP_6}{dt} = \frac{RT_6}{V_T} (w_4 - w_5) \quad , \text{ in scaled form}$$

$$\frac{d(P_6/P_{6m})}{d(t/t_m)} = \left(\frac{T_6}{T_{6m}}\right) \cdot \left\{\left(\frac{w_4}{w_m}\right) - \left(\frac{w_5}{w_m}\right)\right\} \cdot \left[\frac{t_m}{P_{6m}} \cdot \frac{R}{V_T} \cdot T_{6m} \cdot w_m\right] \dots(B.22)$$

B.9 Power Turbine

Pressure ratio = P_6/P_7 , in scaled form

$$\left(\frac{PR}{PR_m}\right) = \frac{(P_6/P_{6m})}{(P_7/P_{7m})} \cdot \left[\frac{P_{6m}}{PR_m \cdot P_{7m}}\right] \quad \dots (B.23)$$

$N_{map} = N_{PT}/\sqrt{T_6}$, thus

$$\left(\frac{N_{map}}{N_{map_m}}\right) = \frac{(N_{PT}/N_{PT_m})}{\sqrt{(T_6/T_{6m})}} \cdot \left[\frac{N_{PT_m}}{N_{map_m} \cdot \sqrt{T_{6m}}}\right] \quad \dots (B.24)$$

The turbine mass flow and temperature drop can be obtained from power turbine map using equations (B.23 and B.24). From turbine map,

(Q_6/Q_{6m}) and $((\Delta T_{67}/T_6)/(\Delta T_{67}/T_6)_m)$ can be obtained. Thus:

$$\left(\frac{w_5}{w_m}\right) = \left(\frac{Q_6}{Q_{6m}}\right) \cdot \left(\frac{P_6}{P_{6m}}\right) / \left(\frac{T_6}{T_{6m}}\right) \cdot \left[\frac{Q_{6m} \cdot P_{6m}}{w_m \cdot \sqrt{T_{6m}}}\right] \quad \dots (B.25)$$

$$T_6 - T_7 = \Delta T_{67}$$

$$\left(\frac{\Delta T_{67}}{T_{7m}}\right) = \left(\frac{T_6}{T_{6m}}\right) \cdot \left(\frac{\Delta T_{67}/T_6}{(\Delta T_{67}/T_6)_m}\right) \cdot \left[\frac{(\Delta T_{67}/T_6)_m \cdot T_{6m}}{T_{7m}}\right] \quad \dots (B.26)$$

$$\left(\frac{T_7}{T_{7m}}\right) = \left(\frac{T_6}{T_{6m}}\right) \cdot \left[\frac{T_{6m}}{T_{7m}}\right] - \left(\frac{\Delta T_{67}}{T_{7m}}\right) \quad \dots (B.27)$$

Power turbine power $P_{PT} = w_5 \cdot C_{p_{PT}} \cdot \Delta T_{67}$. Thus

$$\left(\frac{P_{PT}}{P_m}\right) = \left(\frac{w_5}{w_m}\right) \cdot \left(\frac{\Delta T_{67}}{T_{7m}}\right) \cdot \left[\frac{w_m \cdot T_{7m} \cdot C_{p_{PT}}}{P_m}\right] \quad \dots (B.28)$$

B.10 Exhaust

$$\frac{P_7 - P_a}{P_a} = K_{exh} \cdot (Q_6)^2 \quad , \text{ in scaled form}$$

$$\left(\frac{P_7}{P_{7m}}\right) = \left(\frac{P_a}{P_{am}}\right) \cdot \left(1 + \left(\frac{Q_6}{Q_{6m}}\right)^2 \cdot [Q_{6m}^2 \cdot K_{exh}]\right) \cdot \left[\frac{P_{am}}{P_{7m}}\right] \quad \dots (B.29)$$

B.11 Load (Dynamometer)

$$N_{map} = N_0 \quad \text{and} \quad I_{map} = I_f$$

$$\left(\frac{N_{map}}{N_{map_m}}\right) = \left(\frac{N_0}{N_{0m}}\right) \cdot \left[\frac{N_{0m}}{N_{map_m}}\right] \quad \dots (B.30)$$

$$\left(\frac{I_{map}}{I_{map_m}}\right) = \left(\frac{I_f}{I_{fm}}\right) \cdot \left[\frac{I_{fm}}{I_{map_m}}\right] \quad \dots (B.31)$$

Using equations (B.30) and (B.31) the map torque, $\left(\frac{G_{map}}{G_{map_m}}\right)$, can be obtained from Dynamometer map. Thus:

$$\left(\frac{G_{out}}{G_{out_m}}\right) = \left(\frac{G_{map}}{G_{map_m}}\right) \cdot \left[\frac{G_{map_m}}{G_{out_m}}\right] \quad \dots (B.32)$$

B.12 Gear Box - Output Shaft Dynamics

The balance of power of the gear box results in

$$P_{PT} + P_C + P_{out} + P_{accl} = 0 \quad , \text{ in other form}$$

$$G_{PT} \cdot N_{PT} + G_C \cdot N_C + G_{out} \cdot N_O + G_{accel} \cdot N_O = 0$$

$$G_{accel} = K_1 \cdot G_{PT} + K_2 G_C - 1G_{out} \quad , \text{ thus}$$

$$\left(\frac{G_{accel}}{G_m}\right) = \left(\frac{G_{PT}}{G_m}\right) \cdot \left[\frac{K_1}{K_m}\right] + \left(\frac{G_C}{G_m}\right) \cdot \left[\frac{K_2}{K_m}\right] \dots$$

$$\dots - \left(\frac{G_{out}}{G_{out_m}}\right) \cdot \left[\frac{G_{out_m}}{G_m}\right] \quad \dots (B.33)$$

$$\frac{dN_O}{dt} = \frac{60}{2\pi} \cdot \frac{1}{I_O} \cdot G_{accel} \quad , \text{ in scaled form}$$

$$\frac{d(N_O/N_{Om})}{d(t/t_m)} = \left(\frac{G_{accel}}{G_m}\right) \cdot \left[\frac{t_m}{N_{Om}} \cdot \frac{60 \cdot G_m}{2\pi \cdot I_O}\right] \quad \dots (B.34)$$

B.13 Gas Generator Rotor Dynamics

$$G_{GGT} + G_{comp} + G_{accel} = 0 \quad , \text{ in scaled form}$$

$$\left(\frac{G_{accel}}{G_m}\right) = \left(\frac{G_{GGT}}{G_m}\right) - \left(\frac{G_{comp}}{G_m}\right) \quad \dots (B.35)$$

$$\frac{dN_{GG}}{dt} = \frac{60}{2\pi} \cdot \frac{1}{I_{GG}} \cdot G_{accel} \quad , \text{ in scaled form}$$

$$\frac{d(N_{GG}/N_{GGm})}{d(t/t_m)} = \left(\frac{G_{accel}}{G_m}\right) \cdot \left[\frac{t_m}{N_{GGm}} \cdot \frac{60}{2\pi} \cdot \frac{G_m}{I_{GG}}\right] \quad \dots (B.36)$$

APPENDIX C

DESIGN POINT CALCULATION - VERIFICATION TASK

C.1 Design Point Calculations for the Basic Co-Turboshaft Engine

Design Point Parameters:

$$PR = 6.2$$

$$TIT = 1150^\circ K$$

$$K = 0.2$$

$$N_{rel}/\sqrt{\theta} = 85\%$$

$$\eta_{pT} = 0.85$$

Assumptions are as explained in section 4.1.

C.1.1 Inlet

$$P_a = 10.135 \times 10^4 \text{ N/m}^2$$

for 2% pressure loss in inlet duct,

$$P_1 = 0.98 P_a = 9.9326 \times 10^4 \text{ N/m}^2$$

Scale factor of P_1 , $P_{1m} = 15 \times 10^4 \text{ N/m}^2$, thus the scaled value of P_1 , $SVPI$, is

$$(SVPI = 0.6622)$$

$$T_a = 288^\circ K$$

 T_1 is assumed equal to T_a . The scale factor of T_1 , T_{1m} is equal to $320^\circ K$, thus the scaled temperature is

$$(SVT1 = 0.9000)$$

The rate of mass flow $w_1 = 3.63 \text{ Kg/s}$. The scale factor for mass flow $w_m = 6.00 \text{ Kg/s}$. Thus:

$$(\text{SVW1} = 0.6050)$$

C.1.2 Compressor

The scaled compressor relative speed ($\text{SVNREL} = 0.85$).

$T_2 = T_1 = T_a$. Thus $\theta = 1$. $\delta = 0.98$, hence

$$\begin{aligned} Q_2 &= w\sqrt{\theta}/\delta \\ &= 3.63\sqrt{1}/0.98 = 3.7041 \text{ Kg/s} \end{aligned}$$

$\text{SVQ2} = Q_2/Q_{2m}$, where $Q_{2m} = 6.00 \text{ Kg/s}$

$$(\text{SVQ2} = 0.6173)$$

The scale factor for pressure ratio $\text{PR}_m = 15$, thus

$$\text{SVPR23} = 6.2/15$$

$$(\text{SVPR23} = 0.4133)$$

From compressor map, at $\text{SVQ1} = 0.6173$ and $\text{SVPR23} = 0.4133$, the scaled temperature rise ratio $\text{SVDT23} = 0.4358$. As scale factor of temperature rise ratio is equal to 1.8, then:

$$\Delta T_{23}/T_2 = 1.8 \times 0.4358 = 0.7844, \text{ thus}$$

$$\Delta T_{23} = 0.7844 \times T_2 = 225.9^\circ \text{K},$$

$T_4 = 513.9^\circ \text{K}$. Selecting T_{3m} to be 1000°K ,

$$(\text{SVT3} = 0.5139)$$

$$P_4 = \text{PR} \times P_2 = 6.2 \times 9.9326 \times 10^4 = 61.5821 \times 10^4 \text{ N/m}^2$$

Using $P_{3m} = 150 \times 10^4 \text{ N/m}^2$

$$(\text{SVP3} = 0.4105)$$

C.1.3 Combustor

For 5% combustor pressure loss,

$$P_5 = 0.95 \times P_4$$

$$= 58.5029 \times 10^4 \text{ N/m}^2$$

$$(SVP_5 = 0.3900)$$

$$T_5 = 1150^\circ \text{K. The scale factor for } T_5 \text{ is } 1500^\circ \text{K.}$$

Thus:

$$(SVT_5 = 0.7667)$$

The fuel air ratio f_a can be found from:

$$f_a = \frac{h_5 - h_4}{(ECV)_5}$$

$$= \frac{1219.25 - 517.5}{40857.4} = 0.017175$$

$$= 1.717\%$$

The fuel consumption $w_f = w \times f_a$

$$w_f = 0.01717 \times 3.63$$

$$= 0.06235 \text{ Kg/s}$$

The scaled fuel consumption is

$$(SVWF = 0.6235)$$

Check on combustor model equation:

$$T_5 - T_4 = \{A + 3600 B.f_a + (3600)^2 C.f_a^2 + 3600 D.f_a.T_4\} / (ECV)_5$$

$$= \{156130 + 3600 \times 474129.23 \times f_a - (3600)^2 \times 513.35 \dots$$

$$\dots \times (0.017175)^2 - 3600 \times 47.416 \times 0.017175 \times 513.9\} / 40857.4$$

$$= 636.4^\circ \text{K}$$

$$\begin{aligned}\text{Thus } T_5 &= T_4 + 636.4 \\ &= 1150.3 \text{ }^\circ\text{K}\end{aligned}$$

C.1.4 Gas Generator Turbine

The non-dimensional mass flow $Q_5 = w_3 \sqrt{T_5} / P_5$

$$\begin{aligned}w_3 &= w_a + w_f = w_a (1 + f_a) \\ &= 3.63(1.017175)\end{aligned}$$

$$= 3.6923 \text{ Kg/s. Thus}$$

$$\begin{aligned}Q_5 &= \frac{3.6923 \sqrt{1150}}{58.5029} \\ &= 2.1403 \times 10^{-4}\end{aligned}$$

The scale factor for Q_5 is 2.4628×10^{-4} . Thus

$$(\text{SVQ5} = 0.8690)$$

The compatibility of work between gas generator turbine and compressor and casing results in the following equation

$$\begin{aligned}\Delta T_{56} &= \Delta T_{23} \cdot \frac{w_a}{w_g} \cdot \frac{C_{p \text{ comp}}}{C_{p \text{ GGT}}} \cdot \frac{N_{\text{GG}}}{N_{\text{rel}}} \cdot \frac{1}{\eta_{\text{mech}}} \\ &= 225.9 \times \frac{1}{1.01717} \times \frac{1015}{1147.1} \times \frac{46575}{37260} \times \frac{1}{0.99} \\ &= 248.1 \text{ }^\circ\text{K}\end{aligned}$$

$$\frac{\Delta T_{56}}{T_5} = \frac{248.1}{1150} = 0.21577$$

The scale factor of $(\Delta T_{56}/T_5)$ is 0.3. Thus

$$(\text{SVDT56} = 0.7192)$$

The interturbine temperature $T_6 = 1150 - \Delta T_{56}$

$$T_6 = 901.9 \text{ }^\circ\text{K. Since the scale factor is } 1200 \text{ }^\circ\text{K,}$$

$$(SVT6 = 0.7516)$$

The gas generator turbine efficiency is 0.89, hence:

$$\begin{aligned} P_5/P_6 &= 1/\left(1 - \frac{\Delta T_{56}}{T_5 \cdot \eta_{GGT}}\right)^{\gamma/\gamma-1} \\ &= 1/\left(1 - \frac{0.21577}{0.89}\right)^{3.99688} \\ &= 3.0336 \end{aligned}$$

The pressure ratio scale factor is 4.0, thus

$$(SVPR56 = 0.7584)$$

The interturbine pressure $P_6 = \frac{P_5}{(P_5/P_6)}$

$$P_6 = 19.285 \times 10^4 \text{ N/m}^2$$

The scale factor of P_6 is $75 \times 10^4 \text{ N/m}^2$. Thus

$$(SVP6 = 0.2571)$$

C.1.5 Power Turbine

The non-dimensional mass flow $Q_6 = w_4 \sqrt{T_6}/P_6$.

$$\begin{aligned} Q_6 &= \frac{3.6923 \sqrt{901.9}}{19.285 \times 10^4} \\ &= 5.7498 \times 10^{-4} \end{aligned}$$

The scale factor of Q_6 is 7.7858×10^{-4} , thus

$$(SVQ6 = 0.7385)$$

If the pressure loss factor of the exhaust duct is assumed to be 0.95, then the back pressure of the power turbine (P_7) is:

$$P_7 = \frac{P_a}{0.95} = 10.6684 \times 10^4 \text{ N/m}^2$$

Consequently, the pressure ratio of the power turbine becomes:

$$\begin{aligned} P_6/P_7 &= 19.285 \times 10^4 / 10.6684 \times 10^4 \\ &= 1.8077 \end{aligned}$$

Using a pressure ratio scale factor of 3.0,

$$(SVPR_{67} = 0.6026)$$

The temperature drop ΔT_{67} can be found from the relation:

$$\begin{aligned} \Delta T_{67} &= \eta_{PT} \cdot T_6 \cdot \left[1 - \left(\frac{1}{P_6/P_7} \right)^{\gamma-1/\gamma} \right] \\ &= 0.85 \times 901.9 \times \left[1 - \left(\frac{1}{1.8077} \right)^{0.25925} \right] \\ &= 109.1 \text{ } ^\circ\text{K} \end{aligned}$$

$$\frac{\Delta T_{67}}{T_6} = 0.1210$$

Using a scale factor equal to 0.22

$$(SVDT_{67} = 0.5498)$$

The power turbine exhaust temperature T_7 equals $T_6 - \Delta T_{67}$,

thus

$$\begin{aligned} T_7 &= 901.9 - 109.1 \\ &= 792.8 \text{ } ^\circ\text{K} \end{aligned}$$

Using a scale factor of 1100 $^\circ\text{K}$

$$(SVT_7 = 0.7207)$$

C.1.6 Output Power and Torque

The power turbine power $P_{PT} = w_g \cdot C_{p_{PT}} \cdot \Delta T_{67}$

$$\begin{aligned} P_{PT} &= 3.6923 \times 1112.6 \times 109.1 \\ &= 448.19 \text{ Kw} \end{aligned}$$

The gas generator turbine power $P_{GGT} = w_g \cdot C_{p_{GGT}} \cdot \Delta T_{56}$

$$\begin{aligned} P_{GGT} &= 3.6923 \times 1147.1 \times 248.1 \\ &= 1050.99 \text{ Kw} \end{aligned}$$

The compressor power $P_{comp} = w_a \cdot C_{p_{comp}} \cdot \Delta T_{23}$

$$\begin{aligned} P_{comp} &= 3.63 \times 1015 \times 225.9 \\ &= 832.39 \text{ Kw} \end{aligned}$$

The casing power $P_C = P_{comp} \cdot N_C/N_{rel}$, or

$$P_C = P_{GGT} \times \eta_{mech} \times N_C/N_{GG}, \text{ or}$$

$$P_C = P_{GGT} \times \eta_{mech} - P_{comp} \text{ . Thus}$$

$$P_C = 208.1 \text{ Kw}$$

The output power $P_{out} = P_{PT} + P_C$,

$$\begin{aligned} P_{out} &= 448.19 + 208.10 \\ &= 656.29 \text{ Kw} \end{aligned}$$

The scale factor for P_{out} is 2000 Kw, thus

$$(\text{SVPWR} = 0.3281)$$

$$P_C/P_{out} = 46.4\%$$

The output torque $G_{out} = P_{out}/w_{out}$

$$G_{out} = \frac{P_{out}}{2\pi N_o/60}$$

$$\begin{aligned} &= \frac{656.29 \times 10^3 \times 60}{2\pi \times 10000} \\ &= 626.7 \text{ N.m} \end{aligned}$$

The scale factor of G_{out} is 2000 N.m. Thus

(SVGOUT = 0.3134)

C.2 Verification of Computer Model Results

A sample of the results of the mathematical model, section C.1, is compared with the corresponding set of computer model results. This comparison is shown in TABLE C.1. It can be seen from TABLE C.1, that the difference between results is always less than 1 percent of VM.

Variable	Scaled Variable Name	Mathematical Model Scaled Value (VM)	Computer Model Scaled Value (VC)	Percentage Difference (VC-VM)/VM %
P_1	SVPI	0.6622	0.6621	-0.015 %
Q_2	SVQ2	0.6173	0.6180	+0.11 %
PR	SVPR23	0.4133	0.4139	+0.14 %
T_3	SVT3	0.5139	0.5147	+0.15 %
P_3	SVP3	0.4105	0.4110	+0.12 %
w_f	SVWF	0.6235	0.6268	+0.53 %
Q_5	SVQ5	0.8690	0.8690	0.0 %
T_5	SVT5	0.7667	0.7686	+0.25 %
T_6	SVT6	0.7516	0.7531	+0.20 %
P_6	SVP6	0.2571	0.2572	+0.039 %
Q_6	SVQ6	0.7385	0.7386	+0.013 %
T_7	SVT7	0.7207	0.7226	+0.2 %
P_{out}	SVPWR	0.3281	0.3269	-0.36 %
G_{out}	SVGOUT	0.3134	0.3157	+0.73 %

TABLE C.1

APPENDIX D

POWERFUL TECHNIQUES IN SUPPORT OF HYBRID
COMPUTER MODELLING

Many available and powerful techniques of hybrid computer modelling were used in this study. Most of these techniques were developed by the staff of the Analysis Laboratory of the National Research Council of Canada. Example of these techniques are the following:

D.1 The Function Generation Routines (FNLIB), [14]

This program package provides the ability for:

- generation of one, two and three dimensional functions,
- function generation of two and three variable maps,
- one, two, and three variable function look-ups.

The package contains three Fortran-callable subroutines, where each subroutine, designed for a specific number of independent variables, has separate entry points for the different operations. Each subroutine is independent of the other two. The use of an argument transfer and a variable normalization routine are shared by all three. All routines are written for scaled fraction variables.

D.2 The BUGOFF - An On-line Interactive Debugging Package for Digital and Hybrid Computers, [15]

This program package allows on-line access of variables by name in a Fortran environment. The package consists of a number of Fortran callable subroutines which provide to the user the ability to define, change and display variables during the running of his program. It is especially useful in computer systems which have A/D and D/A converters, such as with hybrid computers, although this is not a requirement.

D.3 The Hytran Operations Interpreter (HOI), [16]

If the analog computer is part of a Hybrid computing facility, where a digital computer can gain access to the analog components, then a digital computer programming language called the Hytran Operations Interpreter (HOI) can be used as a service aid to help with both the design and checkout phases of analog computer programming. It serves to answer the two crucial questions which the programmer must answer:

- Does the proposed design indeed solve the unscaled equations of the problem (off-line check)
- Has the checked design been properly patched and set up on the analog computer (on-line check)

HOI, as a programming language, is essentially an interpretive programming language, similar to the many others available (Basic, Focal, etc...) to which have been added those additional features which make it useful as an analog programming aid.

D.4 The CIRBUG - An On-line Interactive Package for Setup and Checkout of Analog Computer Circuits in a Hybrid Environment, [17]

The CIRBUG is a program package which provides the user of a hybrid computer, or an analog computer in a hybrid environment, with the ability to rapidly setup and checkout an analog circuit. The package is completely interactive and requires no coding by the user. Communication with the package is via the analog patching itself as directed by the user at the terminal of the digital computer.

This package can be used to capture a current state of the analog circuit at any time, which includes all settable components as well as other component outputs. A copy of this state can then be saved as a file on a cartridge tape. At any subsequent time the cartridge file can be used to reset the components to these captured values, and verify that the current state of the circuit is the same as it was at the time of the original capture.

D.5 The DCASET [18]

A program designed to enable the user to set any arbitrary DCA from the analog console. Set up is to be via the push buttons that are used normally to set the potentiometers with the exception of a separate "go" button which has to be provided by the user.

The routine reads the analog component address and then reads the output of the DVM in order to obtain the information used in setting The DCA. For this reason the user should set up the conditions required prior to pushing his programmed "GO" button.

APPENDIX E

ANALOG CIRCUITS

Volumes CircuitsInlet Volume Compressor Volume Interturbine Volumes

Combustor Volume

Gas Generator Rotor Dynamics

Gear Box and Output Shaft Dynamics

Controlling Variables

Fuel Flow

Dynamometer Field Current

Ambient Conditions

Pressure

Temperature

$$\delta_a = T_a / T_{\text{standard}}$$

$$\delta_a = P_a / P_{\text{standard}}$$

APPENDIX F

DIGITAL PROGRAM LISTINGS

The listings that are contained in this appendix are a complete description of the working digital program. The structure is completely modular following closely the information flow diagram of Figure 5 of the main text.

The symbols used follow the specified nomenclature but have a prefix before it to identify the type of variable that the computer must treat. The significance of the prefix is as follows:

<u>1st Character</u>	S - scaled fraction
	R - real
	I - integer
	L - logical
<u>2nd Character</u>	V - variable
	C - constant
	D - input data
	S - scale factor

The remaining four characters may be used to describe the physical meaning of the variable. For example, the digital symbol SVDT23 is a scaled fraction problem variable defining the temperature rise from station 2 to 3; i.e., across the compressor.

C ***** CO-TURBOSHAFT ENGINE MODEL ----- DECEMBER , 1980 *****
 C =====

LOGICAL LVSNS
 LOGICAL LVSENS(9)
 LOGICAL LVSSW
 LOGICAL SENSW, LERR, SENSE

SCALED FRACTION SVIC(40), SDIC(40)
 SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
 REAL RDAMP(30), RDPOT(30)

SCALED FRACTION SVPA , SVTA , SVTHEA , SVDELA , SVW0 , SVQ0
 SCALED FRACTION SVP1 , SVT1 , SVW1 , SVTHE1 , SVDEL1 , SVQ1
 SCALED FRACTION SVP2 , SVT2 , SVW2 , SVTHE2 , SVDEL2 , SVQ2
 SCALED FRACTION SVPR12 , SVDT12 , SVNC , SVNGG , SVNREL , SVGCOM
 SCALED FRACTION SVP3 , SVT3 , SVW3 , SVPR23 , SVDT23 , SVP4
 SCALED FRACTION SVT4 , SVW4 , SVWF , SVT34 , SVP5 , SVT5
 SCALED FRACTION SVW5 , SVRTT5 , SVQ5 , SVPR56 , SVDT56 , SVGGGT
 SCALED FRACTION SVP6 , SVT6 , SVRTT6 , SVQ6 , SVPR67 , SVDT67
 SCALED FRACTION SVNPT , SVGPT , SVP7 , SVT7 , SVOUT , SVGACC
 SCALED FRACTION SVNOUT , SVGC , SVGIGV , SVGOGV , SVGACL , SVDNGG
 SCALED FRACTION SVPWR1 , SVPWR2 , SVPWR3 , SVPWR4 , SVPWR5 , SVPWR
 SCALED FRACTION SVDNO , SVGOMP , SVNOMP , SVIF
 SCALED FRACTION SVGCAS , SVSFC , SVPSHR , SVFLWF
 SCALED FRACTION SVROW3 , SVT4S , SVE , SVT8
 SCALED FRACTION SVNMPF , SVNMPG , SVNMPG , SVNMPG , SVNMPD
 SCALED FRACTION SVNEW , SVNEW1 , SVEFF , SVEFF1 , SVNWI , SVEFFP

C DEFINITION OF ENGINE VARIABLE ARRAY

C -----

C

EQUIVALENCE (SVVAR(1) , SVQ0)
 EQUIVALENCE (SVVAR(2) , SVTHE1)
 EQUIVALENCE (SVVAR(3) , SVDEL1)
 EQUIVALENCE (SVVAR(4) , SVQ1)
 EQUIVALENCE (SVVAR(5) , SVTHE2)
 EQUIVALENCE (SVVAR(6) , SVDEL2)
 EQUIVALENCE (SVVAR(7) , SVQ2)
 EQUIVALENCE (SVVAR(8) , SVPR12)
 EQUIVALENCE (SVVAR(9) , SVDT12)
 EQUIVALENCE (SVVAR(10) , SVGCOM)
 EQUIVALENCE (SVVAR(11) , SVPR23)
 EQUIVALENCE (SVVAR(12) , SVDT23)
 EQUIVALENCE (SVVAR(13) , SVP4)
 EQUIVALENCE (SVVAR(14) , SVT34)
 EQUIVALENCE (SVVAR(15) , SVP5)
 EQUIVALENCE (SVVAR(16) , SVRTT5)
 EQUIVALENCE (SVVAR(17) , SVQ5)
 EQUIVALENCE (SVVAR(18) , SVPR56)
 EQUIVALENCE (SVVAR(19) , SVDT56)
 EQUIVALENCE (SVVAR(20) , SVGGGT)
 EQUIVALENCE (SVVAR(21) , SVRTT6)
 EQUIVALENCE (SVVAR(22) , SVQ6)

EQUIVALENCE (SVVAR(23), SVPR67)
EQUIVALENCE (SVVAR(24), SVDT67)
EQUIVALENCE (SVVAR(25), SVGPT)
EQUIVALENCE (SVVAR(26), SVP7)
EQUIVALENCE (SVVAR(27), SVGOUT)
EQUIVALENCE (SVVAR(28), SVGACC)
EQUIVALENCE (SVVAR(29), SVPWR1)
EQUIVALENCE (SVVAR(30), SVPWR2)
EQUIVALENCE (SVVAR(31), SVPWR3)
EQUIVALENCE (SVVAR(32), SVPWR4)
EQUIVALENCE (SVVAR(33), SVPWR5)
EQUIVALENCE (SVVAR(34), SVROW3)
EQUIVALENCE (SVVAR(35), SVT4S)
EQUIVALENCE (SVVAR(36), SVGCAS)
EQUIVALENCE (SVVAR(37), SVGACL)
EQUIVALENCE (SVVAR(38), SVNMPF)
EQUIVALENCE (SVVAR(39), SVNMPG)
EQUIVALENCE (SVVAR(40), SVNMPG)
EQUIVALENCE (SVVAR(41), SVNMPD)
EQUIVALENCE (SVVAR(42), SVNMPD)
EQUIVALENCE (SVVAR(43), SVGOMP)
EQUIVALENCE (SVVAR(44), SVGIGV)
EQUIVALENCE (SVVAR(45), SVPWR)
EQUIVALENCE (SVVAR(46), SVNEW)
EQUIVALENCE (SVVAR(47), SVNEW1)
EQUIVALENCE (SVVAR(48), SVEFF)
EQUIVALENCE (SVVAR(49), SVEFF1)
EQUIVALENCE (SVVAR(50), SVNWI)
EQUIVALENCE (SVVAR(51), SVEFFP)
EQUIVALENCE (SVVAR(61), SVE)

C

C DEFINITION OF A-D CONVERTER ARRAY

C

EQUIVALENCE (SVADC(1), SVPA)
EQUIVALENCE (SVADC(2), SVTA)
EQUIVALENCE (SVADC(3), SVTHEA)
EQUIVALENCE (SVADC(4), SVDELA)
EQUIVALENCE (SVADC(5), SVP1)
EQUIVALENCE (SVADC(6), SVP2)
EQUIVALENCE (SVADC(7), SVP3)
EQUIVALENCE (SVADC(8), SVP6)
EQUIVALENCE (SVADC(10), SVNNGG)
EQUIVALENCE (SVADC(11), SVNC)
EQUIVALENCE (SVADC(12), SVNREL)
EQUIVALENCE (SVADC(13), SVNOUT)
EQUIVALENCE (SVADC(14), SVNPT)
EQUIVALENCE (SVADC(15), SVWF)
EQUIVALENCE (SVADC(16), SVIF)
EQUIVALENCE (SVADC(17), SVFLWF)

C

C DEFINITION OF D-A CONVERTER ARRAY

C

EQUIVALENCE (SVDAC(1), SVW0)
EQUIVALENCE (SVDAC(2), SVW1)
EQUIVALENCE (SVDAC(3), SVW2)
EQUIVALENCE (SVDAC(4), SVW3)
EQUIVALENCE (SVDAC(5), SVW4)

```

EQUIVALENCE (SVDAC(6) , SVW5 )
EQUIVALENCE (SVDAC(7) , SVT1 )
EQUIVALENCE (SVDAC(8) , SVT2 )
EQUIVALENCE (SVDAC(9) , SVT3 )
EQUIVALENCE (SVDAC(10) , SVT4 )
EQUIVALENCE (SVDAC(11) , SVT5 )
EQUIVALENCE (SVDAC(12) , SVT6 )
EQUIVALENCE (SVDAC(13) , SVT7 )
EQUIVALENCE (SVDAC(14) , SVDNGG)
EQUIVALENCE (SVDAC(15) , SVDNO )
EQUIVALENCE (SVDAC(16) , SVSEC )
EQUIVALENCE (SVDAC(17) , SVPSHR)
EQUIVALENCE (SVDAC(18) , SVT8 )

C
C DECLARATION OF COMMON VARIABLE BLOCKS
C -----
COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBSLL/LVSENS

C
C ADRESS ANALOG
C
CALL QSHYIN(IER, 680, 680)

C
C GET MODEL DATA
C
CALL READ(IDV)

C
C CALCULATE MODEL PARAMETERS
C
CALL PARAM

C
C SET UP MODEL INFORMATION
C
CALL SETUP

C
C===== START HIGH SPEED LOOP =====
C -----
C CALCULATE TOTAL CONDITION AT INLET
100 CALL QRAMI(IMODE)
IF(IMODE.EQ.1.OR.IMODE.EQ.7) GO TO 160
CALL QRBADS(SVADC, 0, 24, IER)
CALL QRBSLL(LVSENS)

C
CCALCULATE ENGINE MODULES
C
CALL INLET
CALL RIGV
CALL COMP
CALL ROGV
CALL HTEXGR
CALL COMBST
CALL GGTURB
CALL PTURB
CALL LOAD
CALL GRBOX
CALL EXHST
CALL EXAMN

```

```
CALL QWBDCS(SVDAC, 0, 24, IER)
CALL QSTDA
CALL QWBDCS(90, SVDAC(29), 10, IER)
IF(LVSSW('377')) GO TO 140
IF(LVSENS(9)) GO TO 140
GO TO 100
```

```
140 CONTINUE
```

```
C
C OFF LINE OPERATION
```

```
C
CALL QSH(IER)
CALL OFFLIN
```

```
C
IF(IMODE EQ. 4) GO TO 150
GO TO 100
150 CALL QSOP(IER)
GO TO 100
160 CALL QRSLL(6, LVSNS, IER)
CALL QRSLL(6, LVSNS, IER)
IF(LVSNS) CALL DCASET
GO TO 100
END
```

```
PROGRAM SIZE = '376
```

C SUBROUTINE TO READ DATA

C
C SUBROUTINE READ(IDV)

C
C INTEGER WDTOT

C STORAGE DECLARATION

C 1- FILE NAMES

C -----
C DIMENSION NAME1(3)
C DIMENSION NAME2(3)
C DIMENSION NAME3(3)
C DIMENSION NAME4(3)
C DIMENSION NAME5(3)
C DIMENSION NAME6(3)
C DIMENSION NAME7(3)
C DIMENSION NAME8(3)

C 2- DATA ARRAYS

C -----
C LOGICAL LVFLAG(25)
C INTEGER INDEX(20)
C INTEGER IDDCA(30)
C REAL RDAMP(30), RDPOT(30)
C REAL RDSF(100), RDPAR(50)

SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50).

SCALED FRACTION SVIC(40), SDIC(40)

SCALED FRACTION SDRIGV(1)

SCALED FRACTION SDCOM4(1)

SCALED FRACTION SDGGT(1)

SCALED FRACTION SDGNEW(1)

SCALED FRACTION SDPT(1)

SCALED FRACTION SDPNEW(1)

SCALED FRACTION SDLOAD(1)

SCALED FRACTION SDNC(7), SDPR12(7, 2), SDQ1(7, 15), SDDT12(7, 15)

SCALED FRACTION SDNREL(8), SDPR23(8, 2), SDQ2(8, 15), SDDT23(8, 15)

SCALED FRACTION SDNGG(4), SDPR56(14), SDQ5(14, 4), SDDT56(14, 4)

SCALED FRACTION SDPR5 (4), SDNEW(12), SDEFF(12, 4)

SCALED FRACTION SDNPT(5), SDPR67(11), SDQ6(11, 5), SDDT67(11, 5)

SCALED FRACTION SDPR6 (4), SDNWI(12), SDEFFP(12, 4)

SCALED FRACTION SDNOUT(12), SDIF(11), SDGOUT(12, 11)

C EQUIVALENCE DECLARATIONS

EQUIVALENCE (INDEX(3), ICRIGV)

EQUIVALENCE (INDEX(4), ICCOM4)

EQUIVALENCE (INDEX(5), ICGGT)

EQUIVALENCE (INDEX(6), ICPT)

EQUIVALENCE (INDEX(7), ICLOAD)

EQUIVALENCE (INDEX(12), ICSF)

```

EQUIVALENCE (INDEX(13), ICPAR )
EQUIVALENCE (INDEX(14), ICIC)
EQUIVALENCE (INDEX(15), ICID)
EQUIVALENCE (INDEX(17), ICAMP)
EQUIVALENCE (INDEX(18), ICGNEW)
EQUIVALENCE (INDEX(19), ICPNEW)

```

```

EQUIVALENCE (SDRIGV(1), ICNC)
EQUIVALENCE (SDCOM4(1), ICNREL)
EQUIVALENCE (SDGGT(1), ICNGG)
EQUIVALENCE (SDGNEW(1), ICPR5 )
EQUIVALENCE (SDPT(1), ICNPT)
EQUIVALENCE (SDPNEW(1), ICPR6 )
EQUIVALENCE (SDLOAD(1), ICNN2)

```

C..... COMMON DECLARATIONS

```

COMMON/CBGEN/INDEX, RDSF, RDPAR
COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBRIGV/ICNC, ICPR12, SDQ1, SDNC, SDPR12, SDDT12
COMMON/CBCOM4/ICNREL, ICPR23, SDQ2, SDNREL, SDPR23, SDDT23
COMMON/CBGGT/ICNGG, ICPR56, SDNGG, SDPR56, SDQ5, SDDT56
COMMON/CBGNEW/ICPR5 , ICNEW, SDPR5 , SDNEW, SDEFF
COMMON/CBPT/ICNPT, ICPR67, SDNPT, SDPR67, SDQ6, SDDT67
COMMON/CBRNEW/ICPR6 , ICNWI, SDPR6 , SDNWI, SDEFFP
COMMON/CBLOAD/ICNN2, ICIF, WDTOT, SDNOUT, SDIF, SDGOUT
COMMON/CBANLG/RDAMP, RDPOT, IDDCA
COMMON/CBFLAG/LVFLAG

```

C..... NAMES OF DATA FILES

```

DATA NAME1(1), NAME1(2), NAME1(3)/2HR0, 2HIN, 2HLT/
DATA NAME2(1), NAME2(2), NAME2(3)/2HR0, 2HRI, 2HGV/
DATA NAME3(1), NAME3(2), NAME3(3)/2HR0, 2HCO, 2HM4/
DATA NAME4(1), NAME4(2), NAME4(3)/2HR0, 2HKG, 2HGT/
DATA NAME5(1), NAME5(2), NAME5(3)/2HR0, 2HPT, 2HK1/
DATA NAME6(1), NAME6(2), NAME6(3)/2HR0, 2HLO, 2HAD/
DATA NAME7(1), NAME7(2), NAME7(3)/2HR0, 2HGN, 2HEW/
DATA NAME8(1), NAME8(2), NAME8(3)/2HR0, 2HPN, 2HEW/

```

C..... FORMATS

```

100  FORMAT(20I4)
110  FORMAT(8F10.4)
120  FORMAT((8(3X, S7)))
130  FORMAT(5E15.5)
140  FORMAT((8(A4, 6X)))
170  FORMAT(20L4)

```

C

C..... READ ALL DATA

C LOGICAL FLAGS

```

READ(6, 170) (LVFLAG(I), I=1, 25)

```

C

C..... DATA ARRAY SIZES

```

READ(6, 100) (INDEX(I), I=1, 20)

```

C

C..... ENGINE PARAMETERS

```

READ(6, 130) (RDPAR(I), I=1, ICPAR)

```

C

C..... SCALE FACTORS

```

READ(6, 130) (RDSF(I), I=1, ICSF)

```



```
C..... READ INTEGRATOR ADDRESSES
      READ(6,160) (RDAMP(I), I=1, ICAMP)

C..... READ IC POT ADDRESSES
      READ(6,100) (IDDCA(I), I=1, ICAMP)
C
C..... INLET DATA

C..... RIGN DATA
      CALL QMOND(21, '22, NAME2)
      CALL QMOND(17, '22, SDRIGV(1), SDRIGV(ICRIGV))
C
C..... COMPRESSOR DATA
      CALL QMOND(21, '22, NAME3)
      CALL QMOND(17, '22, SDCOM4(1), SDCOM4(ICCOM4))
C
C..... G. G. TURBINE DATA
      CALL QMOND(21, '22, NAME4)
      CALL QMOND(17, '22, SDGGT(1), SDGGT(ICGGT))
C
C..... POWER TURBINE DATA
      CALL QMOND(21, '22, NAME5)
      CALL QMOND(17, '22, SDPT(1), SDPT(ICPT))
C
C..... LOAD(DYNAMOMETER) DATA
      CALL QMOND(21, '22, NAME6)
      CALL QMOND(17, '22, SDLOAD(1), SDLOAD(ICLOAD))
C
      CALL QMOND(21, '22, NAME7)
      CALL QMOND(17, '22, SDGNEW(1), SDGNEW(ICGNEW))

      CALL QMOND(21, '22, NAMES)
      CALL QMOND(17, '22, SDPNEW(1), SDPNEW(ICPNEW))

      RETURN
      END
```

PROGRAM SIZE = 1531

C PARAM . . .

C

C SUBROUTINE TO GENERATE SCALED ENGINE CONSTANTS

C

SUBROUTINE PARAM

C

C STORAGE DECLARATIONS

INTEGER INDEX(20)

REAL RDSF(100), RDPAR(50)

SCALED FRACTION SVIC(40), SDIC(40)

SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)

SCALED FRACTION SCUNIT, SCINL1, SCINL2, SCINL3, SCINL4

SCALED FRACTION SCINL5, SCFAN1, SCFAN2, SCFAN3, SCFAN4

SCALED FRACTION SCFAN5, SCFAN6, SCCOM1, SCCOM2, SCCOM3

SCALED FRACTION SCCOM4, SCCOM5, SCCOM6, SCOGV1, SCOGV2

SCALED FRACTION SCOGV3, SCOGV4, SCOGV5, SCOGV6, SCGGT1

SCALED FRACTION SCGGT2, SCGGT3, SCGGT4, SCGGT5, SCGGT6

SCALED FRACTION SCPT1, SCPT2, SCPT3, SCPT4, SCPT5

SCALED FRACTION SCPT6, SCLD1, SCLD2, SCLD3, SCLD4

SCALED FRACTION SCLD5, SCLD6, SCGB1, SCGB2, SCGB3

SCALED FRACTION SCGB4, SCGB5, SCGB6, SCCC1, SCCC2

SCALED FRACTION SCCC3, SCCC4, SCCC5, SCCC6

SCALED FRACTION SCFANO, SCFAN7, SCFAN8, SCFAN9

SCALED FRACTION SCCOM0, SCCOM7, SCCOM8, SCCOM9, SCCOMG

SCALED FRACTION SCOGV0, SCOGV7, SCOGV8, SCOGV9

SCALED FRACTION SCGGT0, SCGGT7, SCGGT8, SCGGT9, SCGGTG, SCGB0

SCALED FRACTION SCPT0, SCPT7, SCPT8, SCPT9, SCEXH4, SCEXH5

SCALED FRACTION SCGB7, SCGB8, SCGB9, SCEXH1, SCEXH2, SCEXH3

SCALED FRACTION SCNEWD, SCEFFD, SCPRDG, SCNGGD, SCGGTP, SCGGTE

SCALED FRACTION SCNWID, SCPRDP, SCPTD, SCNPTD, SCPTP

SCALED FRACTION SCHEX1, SCHEX2, SCHEX3, SCHEX4, SCHEX5, SCHEX6

SCALED FRACTION SCEXH6

C EQUIVALENCE DECLARATIONS

C 1. ENGINE REAL CONSTANTS

EQUIVALENCE (RDPAR(1), RCPA)

EQUIVALENCE (RDPAR(2), RCTA)

EQUIVALENCE (RDPAR(3), RCGC)

EQUIVALENCE (RDPAR(4), RCJ)

EQUIVALENCE (RDPAR(5), RCR)

EQUIVALENCE (RDPAR(6), RCPI)

EQUIVALENCE (RDPAR(7), RCGAM1)

EQUIVALENCE (RDPAR(8), RCGAM2)

EQUIVALENCE (RDPAR(9), RCGAM3)

EQUIVALENCE (RDPAR(10), RCGAM4)

EQUIVALENCE (RDPAR(11), RCGAM5)

EQUIVALENCE (RDPAR(12), RCCP1)

EQUIVALENCE (RDPAR(13), RCCP2)

EQUIVALENCE (RDPAR(14),RCCP3)
EQUIVALENCE (RDPAR(15),RCCP4)
EQUIVALENCE (RDPAR(16),RCCP5)
EQUIVALENCE (RDPAR(17),RCVIN)
EQUIVALENCE (RDPAR(18),RCVCOM)
EQUIVALENCE (RDPAR(19),RCVRQV)
EQUIVALENCE (RDPAR(20),RCVCC)
EQUIVALENCE (RDPAR(21),RCVTUR)
EQUIVALENCE (RDPAR(22),RCKIN)
EQUIVALENCE (RDPAR(23),RCKCC)
EQUIVALENCE (RDPAR(24),RCKEXH)
EQUIVALENCE (RDPAR(25),RCA)
EQUIVALENCE (RDPAR(26),RCB)
EQUIVALENCE (RDPAR(27),RCC)
EQUIVALENCE (RDPAR(28),RCD)
EQUIVALENCE (RDPAR(29),RCLVH)
EQUIVALENCE (RDPAR(30),RCIGG)
EQUIVALENCE (RDPAR(31),RCIGB)
EQUIVALENCE (RDPAR(32),RCILD)
EQUIVALENCE (RDPAR(33),RCK1)
EQUIVALENCE (RDPAR(34),RCK2)
EQUIVALENCE (RDPAR(35),RCBETA)
EQUIVALENCE (RDPAR(36),RCH)
EQUIVALENCE (RDPAR(37),RCEFF)
EQUIVALENCE (RDPAR(38),RCNGDP)
EQUIVALENCE (RDPAR(39),RCNPDP)
EQUIVALENCE (RDPAR(40),RCT5DP)
EQUIVALENCE (RDPAR(41),RCT6DP)
EQUIVALENCE (RDPAR(42),RCA5)
EQUIVALENCE (RDPAR(43),RCA6)

C

C.....2. REAL SCALE FACTORS

EQUIVALENCE (RDSF(1),RSPA)
EQUIVALENCE (RDSF(2),RSTA)
EQUIVALENCE (RDSF(3),RSTHEA)
EQUIVALENCE (RDSF(4),RSDELA)
EQUIVALENCE (RDSF(5),RSW0)
EQUIVALENCE (RDSF(6),RSQ0)
EQUIVALENCE (RDSF(7),RSP1)
EQUIVALENCE (RDSF(8),RST1)
EQUIVALENCE (RDSF(9),RSW1)
EQUIVALENCE (RDSF(10),RSTHE1)
EQUIVALENCE (RDSF(11),RSDEL1)
EQUIVALENCE (RDSF(12),RSQ1)
EQUIVALENCE (RDSF(13),RSP2)
EQUIVALENCE (RDSF(14),RST2)
EQUIVALENCE (RDSF(15),RSW2)
EQUIVALENCE (RDSF(16),RSTHE2)
EQUIVALENCE (RDSF(17),RSDEL2)
EQUIVALENCE (RDSF(18),RSQ2)
EQUIVALENCE (RDSF(19),RSPR12)
EQUIVALENCE (RDSF(20),RSDT12)
EQUIVALENCE (RDSF(21),RSNC)
EQUIVALENCE (RDSF(22),RSNGG)
EQUIVALENCE (RDSF(23),RSNREL)

EQUIVALENCE (RDSF(24), RSGCOM)
EQUIVALENCE (RDSF(25), RSP3)
EQUIVALENCE (RDSF(26), RST3)
EQUIVALENCE (RDSF(27), RSW3)
EQUIVALENCE (RDSF(28), RSPR23)
EQUIVALENCE (RDSF(29), RSDT23)
EQUIVALENCE (RDSF(30), RSP4)
EQUIVALENCE (RDSF(31), RST4)
EQUIVALENCE (RDSF(32), RSW4)
EQUIVALENCE (RDSF(33), RSWF)
EQUIVALENCE (RDSF(34), RSP5)
EQUIVALENCE (RDSF(35), RST5)
EQUIVALENCE (RDSF(36), RSW5)
EQUIVALENCE (RDSF(37), RSQ5)
EQUIVALENCE (RDSF(38), RSPR56)
EQUIVALENCE (RDSF(39), RSDT56)
EQUIVALENCE (RDSF(40), RSGGGT)
EQUIVALENCE (RDSF(41), RSP6)
EQUIVALENCE (RDSF(42), RST6)
EQUIVALENCE (RDSF(43), RSQ6)
EQUIVALENCE (RDSF(44), RSPR67)
EQUIVALENCE (RDSF(45), RSDT67)
EQUIVALENCE (RDSF(46), RSNPT)
EQUIVALENCE (RDSF(47), RSGPT)
EQUIVALENCE (RDSF(48), RSP7)
EQUIVALENCE (RDSF(49), RST7)
EQUIVALENCE (RDSF(50), RSGOUT)
EQUIVALENCE (RDSF(51), RSGACC)
EQUIVALENCE (RDSF(52), RSNOUT)
EQUIVALENCE (RDSF(53), RSVOL)
EQUIVALENCE (RDSF(54), RSAREA)
EQUIVALENCE (RDSF(55), RSTIMP)
EQUIVALENCE (RDSF(56), RSTIMN)
EQUIVALENCE (RDSF(57), RSI)
EQUIVALENCE (RDSF(58), RSKIN)
EQUIVALENCE (RDSF(59), RSKEXH)
EQUIVALENCE (RDSF(60), RSGC)
EQUIVALENCE (RDSF(61), RSK)
EQUIVALENCE (RDSF(62), RSGIGV)
EQUIVALENCE (RDSF(63), RSGOGV)
EQUIVALENCE (RDSF(64), RSGACL)
EQUIVALENCE (RDSF(65), RSDNGG)
EQUIVALENCE (RDSF(66), RSDNO)
EQUIVALENCE (RDSF(67), RSNOMP)
EQUIVALENCE (RDSF(68), RSGOMP)
EQUIVALENCE (RDSF(69), RSPWR)
EQUIVALENCE (RDSF(70), RSROW3)

C. SCALED ENGINE PARAMETERS

EQUIVALENCE (SCPAR(1), SCUNIT)
EQUIVALENCE (SCPAR(2), SCINL1)
EQUIVALENCE (SCPAR(3), SCINL2)
EQUIVALENCE (SCPAR(4), SCINL3)
EQUIVALENCE (SCPAR(5), SCINL4)
EQUIVALENCE (SCPAR(6), SCINL5)
EQUIVALENCE (SCPAR(7), SCFAN1)

EQUIVALENCE (SCPAR(8), SCFAN2)
EQUIVALENCE (SCPAR(9), SCFAN3)
EQUIVALENCE (SCPAR(10), SCFAN4)
EQUIVALENCE (SCPAR(11), SCFAN5)
EQUIVALENCE (SCPAR(12), SCFAN6)
EQUIVALENCE (SCPAR(13), SCCOM1)
EQUIVALENCE (SCPAR(14), SCCOM2)
EQUIVALENCE (SCPAR(15), SCCOM3)
EQUIVALENCE (SCPAR(16), SCCOM4)
EQUIVALENCE (SCPAR(17), SCCOM5)
EQUIVALENCE (SCPAR(18), SCCOM6)
EQUIVALENCE (SCPAR(19), SCOGV1)
EQUIVALENCE (SCPAR(20), SCOGV2)
EQUIVALENCE (SCPAR(21), SCOGV3)
EQUIVALENCE (SCPAR(22), SCOGV4)
EQUIVALENCE (SCPAR(23), SCOGV5)
EQUIVALENCE (SCPAR(24), SCOGV6)
EQUIVALENCE (SCPAR(25), SCCC1)
EQUIVALENCE (SCPAR(26), SCCC2)
EQUIVALENCE (SCPAR(27), SCCC3)
EQUIVALENCE (SCPAR(28), SCCC4)
EQUIVALENCE (SCPAR(29), SCCC5)
EQUIVALENCE (SCPAR(30), SCCC6)
EQUIVALENCE (SCPAR(31), SCGGT1)
EQUIVALENCE (SCPAR(32), SCGGT2)
EQUIVALENCE (SCPAR(33), SCGGT3)
EQUIVALENCE (SCPAR(34), SCGGT4)
EQUIVALENCE (SCPAR(35), SCGGT5)
EQUIVALENCE (SCPAR(36), SCGGT6)
EQUIVALENCE (SCPAR(37), SCPT1)
EQUIVALENCE (SCPAR(38), SCPT2)
EQUIVALENCE (SCPAR(39), SCPT3)
EQUIVALENCE (SCPAR(40), SCPT4)
EQUIVALENCE (SCPAR(41), SCPT5)
EQUIVALENCE (SCPAR(42), SCPT6)
EQUIVALENCE (SCPAR(43), SCGB1)
EQUIVALENCE (SCPAR(44), SCGB2)
EQUIVALENCE (SCPAR(45), SCGB3)
EQUIVALENCE (SCPAR(46), SCGB4)
EQUIVALENCE (SCPAR(47), SCGB5)
EQUIVALENCE (SCPAR(48), SCGB6)
EQUIVALENCE (SCPAR(49), SCFAN0)
EQUIVALENCE (SCPAR(50), SCFAN7)
EQUIVALENCE (SCPAR(51), SCFAN8)
EQUIVALENCE (SCPAR(52), SCFAN9)
EQUIVALENCE (SCPAR(53), SCCOM0)
EQUIVALENCE (SCPAR(54), SCCOM7)
EQUIVALENCE (SCPAR(55), SCCOM8)
EQUIVALENCE (SCPAR(56), SCCOM9)
EQUIVALENCE (SCPAR(57), SCCOM6)
EQUIVALENCE (SCPAR(58), SCOGV0)
EQUIVALENCE (SCPAR(59), SCOGV7)
EQUIVALENCE (SCPAR(60), SCOGV8)
EQUIVALENCE (SCPAR(61), SCOGV9)
EQUIVALENCE (SCPAR(62), SCGGT0)
EQUIVALENCE (SCPAR(63), SCGGT7)
EQUIVALENCE (SCPAR(64), SCGGT8)

```

EQUIVALENCE (SCPAR(65), SCGGT9)
EQUIVALENCE (SCPAR(66), SCGGTG)
EQUIVALENCE (SCPAR(67), SCPT0 )
EQUIVALENCE (SCPAR(68), SCPT7 )
EQUIVALENCE (SCPAR(69), SCPT8 )
EQUIVALENCE (SCPAR(70), SCPT9 )
EQUIVALENCE (SCPAR(71), SCGB7 )
EQUIVALENCE (SCPAR(72), SCGB8 )
EQUIVALENCE (SCPAR(73), SCGB9 )
EQUIVALENCE (SCPAR(74), SCEXH1)
EQUIVALENCE (SCPAR(75), SCEXH2)
EQUIVALENCE (SCPAR(76), SCLD1 )
EQUIVALENCE (SCPAR(77), SCLD2 )
EQUIVALENCE (SCPAR(78), SCLD3 )
EQUIVALENCE (SCPAR(79), SCLD4 )
EQUIVALENCE (SCPAR(80), SCNEW)
EQUIVALENCE (SCPAR(81), SCEFFD)
EQUIVALENCE (SCPAR(82), SCPRDG)
EQUIVALENCE (SCPAR(83), SCNGGD)
EQUIVALENCE (SCPAR(84), SCGGTP)
EQUIVALENCE (SCPAR(85), SCGGTE)
EQUIVALENCE (SCPAR(86), SCNWID)
EQUIVALENCE (SCPAR(87), SCPRDP)
EQUIVALENCE (SCPAR(88), SCNPTD)
EQUIVALENCE (SCPAR(89), SCPTP )
EQUIVALENCE (SCPAR(90), SCPTC )
EQUIVALENCE (SCPAR(91), SCHEX1)
EQUIVALENCE (SCPAR(92), SCHEX2)
EQUIVALENCE (SCPAR(93), SCHEX3)
EQUIVALENCE (SCPAR(94), SCHEX4)
EQUIVALENCE (SCPAR(95), SCHEX5)
EQUIVALENCE (SCPAR(96), SCHEX6)
EQUIVALENCE (SCPAR(97), SCEXH3)
EQUIVALENCE (SCPAR(98), SCEXH4)
EQUIVALENCE (SCPAR(99), SCEXH5)
EQUIVALENCE (SCPAR(100), SCGB0)
EQUIVALENCE (SCPAR(101), SCEXH6)

```

C

C..... COMMON DECLARATIONS

COMMON/CBGEN/INDEX, RDSF, RDPAR

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC

C

C..... CALCULATE CONSTANTS FOR INLET

SCUNIT= 0.0

SCINL1=RSP1/RSPA

SCINL2=RSTA/RST1

SCINL3=1.0/(RSDLA/(RSW0*SQR(T(RSTHEA*RCKIN)))

SCINL4= 0.0

SCINL5= 0.0

C..... CALCULATE CONSTANTS FOR FAN

SCFAN0=RST1/RSTA

SCFAN1=RST1/(RSTA*RSTHEA)

SCFAN2=RSP1/(RSPA*RSDLA)

SCFAN3=RSP2/(RSP1*RSPR12)

SCFAN4=0.99999
 SCFAN5=(RSQ1*RSDEL1)/(RSW1*SQRT(RSTHE1))
 SCFAN6=RSDT12*RST1/RST2
 SCFAN7=RST1/RST2
 SCFAN8=RSPWR/(RSW1*RST2*RCCP1)
 SCFAN9=RSPWR*60.0/(2.0*RCPI*RSNC*RSGIGV)

C. CALCULATE CONSTANTS FOR HP COMPRESSOR

SCCOM0=RSTA/RST2
 SCCOM0=0.9999
 SCCOM1=RST2/(RSTA*RSTHE2)
 SCCOM2=RSP2/(RSPA*RSDEL2)
 SCCOM3=RSP3/(RSP2*RSPR23)
 SCCOM4=0.99999
 SCCOM5=RSW2*SQRT(RSTHE2)/(RSQ2*RSDEL2)
 SCCOM6=RSDT23*RST2/RST3
 SCCOM7=RST2/RST3
 SCCOM8=RSPWR/(RSW2*RST3*RCCP2)
 SCCOM9=RSPWR*60.0/(2.0*RCPI*RSNREL*RSGCOM)
 SCCOMG=RSPWR/(RSNREL*RSGACC)

C. CALCULATE CONSTANTS FOR THE ROGV

SCOGV0=1.0/RST3
 SCOGV1=SIN(RCBETA)/(COS(RCBETA)*60.0*RCH)
 SCOGV2=RCEFF
 SCOGV3=(RCGAM3-1.0)/RCGAM3
 SCOGV4=RSP3/(RST3*RCR*RSROW3)
 RCOGV1=SCOGV1
 SCOGV5=RCOGV1*(RSNC*RSW3*RSW3)/(RSROW3*RSPWR)
 SCOGV6=(RSW3*RCCP3)/RSPWR
 SCOGV7=1.0/RST4
 SCOGV8=0.99999

C. CALCULATE CONSTANTS FOR HEAT EXCHANGER

SCHEX1=0.9000
 SCHEX2=0.9091
 SCHEX3=0.9800
 SCHEX4=0.9700
 SCHEX5=0.9700

C. CALCULATE CONSTANTS FOR THE COMBUSTOR

SCCC1=RSP4*RCKCC/RSP5
 SCCC2=RST4/RST5
 SCCC3=RCA/(RST5*RCLVH)
 SCCC4=3600.0*RCB*RSWF/(RSW3*RST5*RCLVH)
 SCCC5=3600.0**2.0*RCC*RSWF**2.0/(RSW3**2.0*RST5*RCLVH)
 SCCC6=3600.0*RCO*RST4*RSWF/(RSW3*RST5*RCLVH)

C. CALCULATE CONSTANTS FOR GG TURBINE

SCGGT0=RST6/RST5

SCGGT1=RSP5/(RSP6*RSPR56)
 SCGGT2=0.91468
 SCGGT3=RSW4*SQRT(RST5)/(RCA5*RSQ5*RSP5)
 SCGGT4=1.0/(RSDT56*RST5)
 SCGGT5=RSDT56
 SCGGT6=RSDT56*RST5/RST6
 SCGGT7=RSPWR/(RSW4*RST5*RCCP4)
 SCGGT8=RSPWR*60.0/(2.0*RCPI*RSNGG*RSGGGT)
 SCGGT9=(2.0*RCPI*RCIGG*RSDNGG)/60.0*RSGACC
 SCGGT10=RSPWR/(RSNGG*RSGACC)
 SCNEWD=0.420
 SCEFFD=0.893
 SCPRDG=0.65955
 SCNGGD=0.800
 SCGGTP=0.25019
 SCGGTE=SSQRT(0.99999S-(0.25/SCPRDG)**SCGGTP)

C..... CALCULATE CONSTANTS FOR POWER TURBINE

SCPT0=RST7/RST6
 SCPT1=RSP6/(RSP7*RSPR67)
 SCPT2=0.86692
 SCPT3=RSW5*SQRT(RST6)/(RCA6*RSQ6*RSP6)
 SCPT4=RSDT67*RST6/RST7
 SCPT5=RSPWR/(RSW5*RST6*RCCP5)
 SCPT6=RSPWR*60.0/(2.0*RCPI*RSNPT*RSGPT)
 SCPT7=RSDT67
 SCNWID=0.750
 SCPRDP=0.69285
 SCPTP=0.25795
 SCNPTD=0.83331
 SCPTTE=SSQRT(0.99999S-(0.33333/SCPRDP)**SCPTP)

C..... CALCULATE CONSTANTS FOR LOAD SYSTEM

SCLD1=RSNOUT/RSNOMP
 SCLD2=RSGOMP/RSGOUT
 SCLD3=0.99999

C..... CALCULATE CONSTANTS FOR GEAR BOX

SCGB0=RSNC/RSNGG
 SCGB1=(RCK1*RSGPT)/(RSK*RSGACL)
 SCGB2=(RCK2*RSGCOM)/(RSK*RSGACL)
 SCGB3=RSGOUT/RSGACL
 SCGB4=1.0/(RSGACL/RSI)
 SCGB5=(60.0*RSPWR)/(2.0*RCPI*RSNOUT*RSGOUT)
 SCGB6=RSGCOM/RSGACL
 SCGB7=RSPWR/(RSGACL*RSNC)
 SCGB8=1.0/RSK
 SCGB9=(2.0*RCPI*(RCIGB+RCILD)*RSDNO)/(60.0*RSGACL)

C..... CALCULATE CONSTANTS FOR EXHAUST SYSTEM

SCEXH1=RSPA/RSP7

PAGE 8

E-16

SCEXH2= RCKEXH*RSQ6**2. 0
SCEXH3=0. 69032
SCEXH4=0. 7071
SCEXH5=0. 3597
SCEXH6=0. 4

RETURN
END

PROGRAM SIZE = 12055

C--SETUP

SUBROUTINE SETUP

INTEGER IVADR(20)

C-----SUBROUTINE TO SET INITIAL CONDITIONS ON THE ANALOG
C COMPUTER TO THAT CORRESPONDING TO VALUES SET BY HOI

INTEGER INDEX(20)

INTEGER IDDCA(30)

LOGICAL LVSENS(9)

LOGICAL LVFLAG(25)

SCALED FRACTION SABS

SCALED FRACTION SVVAR(150)

SCALED FRACTION SCPAR(150)

SCALED FRACTION SVADC(50)

SCALED FRACTION SVDAC(50)

SCALED FRACTION SDIC(40)

SCALED FRACTION SVIC(40)

REAL RDAMP(30), RDPOT(30)

REAL RDSF(100), RDPAR(50)

REAL RVVAR(250)

EQUIVALENCE (INDEX(17), ICAMP)

COMMON/CBFLAG/LVFLAG

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC

COMMON/CBSLL/LVSENS

COMMON/CBADR/IVADR

COMMON/CBGEN/INDEX, RDSF, RDPAR

COMMON/CBRVAR/RVVAR

COMMON/CBANLG/RDAMP, RDPOT, IDDCA

C INITIALIZE DATA TO BUGOFF

CALL BUGINT(6HSVVAR, SVVAR, 2HSA)

CALL BUGINT(6HSVADC, SVADC, 2HSA)

CALL BUGINT(6HSVDAC, SVDAC, 2HSA)

CALL BUGINT(6HSCPAR, SCPAR, 2HSA)

CALL BUGINT(6HSVIC, SVIC, 2HSA)

CALL BUGINT(6HSDIC, SDIC, 2HSA)

CALL BUGINT(6HRDPAR, RDPAR, 2HRA)

CALL BUGINT(6HRDSF, RDSF, 2HRA)

CALL BUGINT(6HIDDCA, IDDCA, 2HIA)

CALL BUGINT(6HINDEX, INDEX, 2HIA)

CALL BUGINT(6HLVSENS, LVSENS, 2HLA)

CALL BUGINT(6HLVFLAG, LVFLAG, 2HLA)

CALL BUGINT(6HRDAMP, RDAMP, 2HAA)

CALL BUGINT(6HRDPOT, RDPOT, 2HAA)

CALL BUGINT(6HIVADR, IVADR, 2HIA)

C INITIALIZE POINTERS

IVADR(1)=27

IVADR(2)=45

IVADR(3)=15

IVADR(4)=4

IVADR(5)=5
IVADR(6)=6
IVADR(7)=7
IVADR(8)=8
IVADR(9)=9

C PUT ANALOG INTO IC
CALL QSIC(IER)

C PUT ANALOG BACK IN PC MODE
CALL QSPC(IER)
RETURN
END

PROGRAM SIZE = 313

C EXAMN

C-----SUBROUTINE TO DISPLAY FOUR MODEL VARIABLES
C ON THE ANALOG COMPUTER

C-----PROGRAMMER B. D. MACISAAC

SUBROUTINE EXAMN

INTEGER IVADR(20)
SCALED FRACTION SVIC(40), SDIC(40)
SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)

EQUIVALENCE (IVADR(1), IVADR1)
EQUIVALENCE (IVADR(2), IVADR2)
EQUIVALENCE (IVADR(3), IVADR3)
EQUIVALENCE (IVADR(4), IVADR4)
EQUIVALENCE (IVADR(5), IVADR5)
EQUIVALENCE (IVADR(6), IVADR6)
EQUIVALENCE (IVADR(7), IVADR7)
EQUIVALENCE (IVADR(8), IVADR8)
EQUIVALENCE (IVADR(9), IVADR9)

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBADR/IVADR

C LOAD VARIABLES INTO POSITION FOR TRANSFER

SVDAC(29)=SVVAR(IVADR1)
SVDAC(30)=SVVAR(IVADR2)
SVDAC(31)=SVVAR(IVADR3)
SVDAC(32)=SVVAR(IVADR4)
SVDAC(33)=SVVAR(IVADR5)
SVDAC(34)=SVVAR(IVADR6)
SVDAC(35)=SVADC(IVADR7)
SVDAC(36)=SVADC(IVADR8)
SVDAC(37)=SVADC(IVADR9)

RETURN
END

PROGRAM SIZE = 147

C OFFLIN

C SUBROUTINE TO SERVICE ALL OPERATIONS THAT
C ARE NOT PART OF THE MODEL, I. E. NOT IN THE
C HIGH SPEED LOOP

C PROGRAMMER B. D. MACISAAC

SUBROUTINE OFFLIN
LOGICAL LVFLAG(25)
LOGICAL SENSW,LVSENS(9)
COMMON/CBSLL/LVSENS
COMMON/CBFLAG/LVFLAG

C CHECK FOR MISSION CHANGE
IF(LVSENS(3)) CALL INLET

C CHECK FOR PROBLEM PARAMETER CHANGE
IF(.NOT. SENSW(3)) GO TO 110
CALL BUGOFF(1)
CALL PARAM
110 CONTINUE

C CHECK FOR BUGOFF OPERATION
IF(SENSW(4)) CALL BUGOFF(1)

C CHECK FOR BUGOFF OPERATION WITH INPUT FROM CARD READER
IF(SENSW(7)) CALL BUGOFF(6)
115 IF (SENSW(7)) GO TO 115

C CHECK FOR MODEL OPERATION WITH ANALOG I/O UNDER BUGOFF CONTROL
IF(.NOT. SENSW(5)) GO TO 120
CALL BUGADC
IF(LVFLAG(1)) CALL RIGV
IF(LVFLAG(2)) CALL COMP.
IF(LVFLAG(3)) CALL ROGV
IF(LVFLAG(4)) CALL COMBST
IF(LVFLAG(5)) CALL GGTURB
IF(LVFLAG(6)) CALL PTURB
IF(LVFLAG(7)) CALL LOAD
CALL BUGDAC
IF(SENSW(6)) CALL BUGPRT(1,1)
GO TO 110
120 CONTINUE

C CHECK FOR PRINT UNDER BUGOFF CONTROL

C CHECK FOR TYPEOUT
IF(LVSENS(1)) CALL OUTPUT

C CHECK FOR IC CHANGE
IF(LVSENS(2)) CALL CHNGIC

```
C  CHECK FOR REQUEST FOR SETTING A DCA  
  IF(LVSENS(7)) CALL DCASET  
  RETURN  
  END
```

PROGRAM SIZE = 157

```
C  CHNGIC

C  SUBROUTINE TO SET INITIAL CONDITIONS ON THE ANALOG
C  COMPUTER TO ANY CURRENT OPERATING POINT OF THE
C  MODEL TO WHICH THE OPERATOR WISHES TO RETURN

C      PROGRAMMER  B. D. MAC ISAAC
C      DATE  16, SEPT 76
C      SUBROUTINE CHNGIC
C      LOGICAL LVSENS
C      INTEGER INDEX(20)
C      INTEGER IDDCA(30)
C      SCALED FRACTION SABS
C      SCALED FRACTION SVIC(40), SDIC(40)
C      SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
C      REAL RDAMP(30), RDPOT(30)

C      EQUIVALENCE (INDEX(17), ICAMP)
C      COMMON/CBGEN/INDEX, RDSF, RDPAR
C      COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC

C      PUT ANALOG INTO HOLD
C      CALL QSC(2, IER)
C      CALL QSH(1, IER)
C      CALL QSC(0, IER)
C      CALL QSC(1, IER)

C-----CHECK IF ORIGINAL IC'S ARE TO BE RECOVERED
C      CALL QRSLL(2, LVSENS, IER)
C      CALL QRSLL(2, LVSENS, IER)
C      IF(LVSENS) GO TO 130

C-----RESET MODEL IC'S TO CURRENT STEADY STATE DATA

C-----READ OUTPUT OF AMPLIFIER AND SET CORRESPONDING
C      POTENTIOMETER TO THIS VALUE
C      DO 100 I=1, ICAMP
C      CALL QRAS(RDAMP(I), SVIC(I), IER)
100  SVIC(I)=SABS(SVIC(I))

C  SET DCA'S TO VALUES READ
C      CALL QWDCS(IDDCA, SVIC, ICAMP, IER)
C      GO TO 160

C  CONSOLE 1
130  CALL QWDCS(IDDCA, SDIC, ICAMP, IER)

C      PUT ANALOG BACK IN IC

160  CALL QSC(1, IERR)
C      CALL QSIC(1, IER)
```

PAGE 2

F-23

```
CALL QSC(0, IERR)  
CALL QSC(1, IERR)  
RETURN  
END
```

PROGRAM SIZE = 404

C OUTPUT...

C

C SUBROUTINE TO OUTPUT MODEL VARIABLES

C

```
SUBROUTINE OUTPUT
LOGICAL SENSW
REAL RVAR(50)
DIMENSION NAME(3), NAM(3)
INTEGER INDEX(20), IFORM(40), IDDISC(28), ID(6)
REAL RDSF(100), RDPAR(50), RVVAR(250), RVOUT(3), RVDISC(14)
SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
SCALED FRACTION SVIC(40), SDIC(40)
```

C EQUIVALENCE OF DATA

C CONSTANTS

```
EQUIVALENCE (RDPAR(1), RCPA )
EQUIVALENCE (RDPAR(2), RCTA )
EQUIVALENCE (RDPAR(3), RCGC )
EQUIVALENCE (RDPAR(4), RCJ )
EQUIVALENCE (RDPAR(5), RCR )
EQUIVALENCE (RDPAR(6), RCPI )
EQUIVALENCE (RDPAR(7), RCGAM1)
EQUIVALENCE (RDPAR(8), RCGAM2)
EQUIVALENCE (RDPAR(9), RCGAM3)
EQUIVALENCE (RDPAR(10), RCGAM4)
EQUIVALENCE (RDPAR(11), RCGAM5)
EQUIVALENCE (RDPAR(12), RCCP1 )
EQUIVALENCE (RDPAR(13), RCCP2 )
EQUIVALENCE (RDPAR(14), RCCP3 )
EQUIVALENCE (RDPAR(15), RCCP4 )
EQUIVALENCE (RDPAR(16), RCCP5 )
EQUIVALENCE (RDPAR(17), RCVIN )
EQUIVALENCE (RDPAR(18), RCVCOM)
EQUIVALENCE (RDPAR(19), RCVROV)
EQUIVALENCE (RDPAR(20), RCVCC )
EQUIVALENCE (RDPAR(21), RCVTUR)
EQUIVALENCE (RDPAR(22), RCKIN )
EQUIVALENCE (RDPAR(23), RCKCC )
EQUIVALENCE (RDPAR(24), RCKEXH)
EQUIVALENCE (RDPAR(25), RCA )
EQUIVALENCE (RDPAR(26), RCB )
EQUIVALENCE (RDPAR(27), RCC )
EQUIVALENCE (RDPAR(28), RCD )
EQUIVALENCE (RDPAR(29), RCT5D )
EQUIVALENCE (RDPAR(30), RCIGG )
EQUIVALENCE (RDPAR(31), RCIGB )
EQUIVALENCE (RDPAR(32), RCILD )
EQUIVALENCE (RDPAR(33), RCK1 )
EQUIVALENCE (RDPAR(34), RCK2 )
EQUIVALENCE (RDPAR(35), RCBETA)
```

EQUIVALENCE (RDPAR(36),RCH)
 EQUIVALENCE (RDPAR(37),RCEFF)
 EQUIVALENCE (RDPAR(38),RCONGDP)
 EQUIVALENCE (RDPAR(39),RCONPDF)
 EQUIVALENCE (RDPAR(40),RCT5DP)
 EQUIVALENCE (RDPAR(41),RCT6DP)
 EQUIVALENCE (RDPAR(42),RCA5)
 EQUIVALENCE (RDPAR(43),RCA6)

EQUIVALENCE (RVVAR(1) ,RVPA)
 EQUIVALENCE (RVVAR(2) ,RVTA)
 EQUIVALENCE (RVVAR(5) ,RVP1)
 EQUIVALENCE (RVVAR(6) ,RVP2)
 EQUIVALENCE (RVVAR(7) ,RVP3)
 EQUIVALENCE (RVVAR(8) ,RVP4)
 EQUIVALENCE (RVVAR(9) ,RVP6)
 EQUIVALENCE (RVVAR(10),RVNGG)
 EQUIVALENCE (RVVAR(11),RVNC)
 EQUIVALENCE (RVVAR(12),RVNREL)
 EQUIVALENCE (RVVAR(13),RVNOUT)
 EQUIVALENCE (RVVAR(14),RVNPT)
 EQUIVALENCE (RVVAR(15),RVWF)
 EQUIVALENCE (RVVAR(16),RVIF)

EQUIVALENCE (RVVAR(51),RVW0)
 EQUIVALENCE (RVVAR(56),RVW5)
 EQUIVALENCE (RVVAR(57),RVT1)
 EQUIVALENCE (RVVAR(58),RVT2)
 EQUIVALENCE (RVVAR(59),RVT3)
 EQUIVALENCE (RVVAR(60),RVT4)
 EQUIVALENCE (RVVAR(61),RVT5)
 EQUIVALENCE (RVVAR(62),RVT6)
 EQUIVALENCE (RVVAR(63),RVT7)

C. VARIABLE BANK

EQUIVALENCE (RVVAR(101),RVQ0)
 EQUIVALENCE (RVVAR(108),RVPR12)
 EQUIVALENCE (RVVAR(109),RVDT12)
 EQUIVALENCE (RVVAR(110),RVGCOM)
 EQUIVALENCE (RVVAR(111),RVPR23)
 EQUIVALENCE (RVVAR(112),RVDT23)
 EQUIVALENCE (RVVAR(115),RVP5)
 EQUIVALENCE (RVVAR(118),RVPR56)
 EQUIVALENCE (RVVAR(119),RVDT56)
 EQUIVALENCE (RVVAR(120),RVGGGT)
 EQUIVALENCE (RVVAR(123),RVPR67)
 EQUIVALENCE (RVVAR(124),RVDT67)
 EQUIVALENCE (RVVAR(125),RVGPT)
 EQUIVALENCE (RVVAR(126),RVP7)
 EQUIVALENCE (RVVAR(127),RVGOUT)
 EQUIVALENCE (RVVAR(129),RVPWR1)
 EQUIVALENCE (RVVAR(130),RVPWR2)
 EQUIVALENCE (RVVAR(131),RVPWR3)
 EQUIVALENCE (RVVAR(132),RVPWR4)
 EQUIVALENCE (RVVAR(133),RVPWR5)

C. 2. REAL SCALE FACTORS

EQUIVALENCE (RDSF(1), RSPA)
EQUIVALENCE (RDSF(2), RSTA)
EQUIVALENCE (RDSF(3), RSTHEA)
EQUIVALENCE (RDSF(4), RSELA)
EQUIVALENCE (RDSF(5), RSWO)
EQUIVALENCE (RDSF(6), RSQO)
EQUIVALENCE (RDSF(7), RSP1)
EQUIVALENCE (RDSF(8), RST1)
EQUIVALENCE (RDSF(9), RSW1)
EQUIVALENCE (RDSF(10), RSTHE1)
EQUIVALENCE (RDSF(11), RSE1)
EQUIVALENCE (RDSF(12), RSQ1)
EQUIVALENCE (RDSF(13), RSP2)
EQUIVALENCE (RDSF(14), RST2)
EQUIVALENCE (RDSF(15), RSW2)
EQUIVALENCE (RDSF(16), RSTHE2)
EQUIVALENCE (RDSF(17), RSE2)
EQUIVALENCE (RDSF(18), RSQ2)
EQUIVALENCE (RDSF(19), RSPR12)
EQUIVALENCE (RDSF(20), RSDT12)
EQUIVALENCE (RDSF(21), RSNC)
EQUIVALENCE (RDSF(22), RSNNG)
EQUIVALENCE (RDSF(23), RSNREL)
EQUIVALENCE (RDSF(24), RSGCOM)
EQUIVALENCE (RDSF(25), RSP3)
EQUIVALENCE (RDSF(26), RST3)
EQUIVALENCE (RDSF(27), RSW3)
EQUIVALENCE (RDSF(28), RSPR23)
EQUIVALENCE (RDSF(29), RSDT23)
EQUIVALENCE (RDSF(30), RSP4)
EQUIVALENCE (RDSF(31), RST4)
EQUIVALENCE (RDSF(32), RSW4)
EQUIVALENCE (RDSF(33), RSWF)
EQUIVALENCE (RDSF(34), RSP5)
EQUIVALENCE (RDSF(35), RST5)
EQUIVALENCE (RDSF(36), RSW5)
EQUIVALENCE (RDSF(37), RSQ5)
EQUIVALENCE (RDSF(38), RSPR56)
EQUIVALENCE (RDSF(39), RSDT56)
EQUIVALENCE (RDSF(40), RSGGGT)
EQUIVALENCE (RDSF(41), RSP6)
EQUIVALENCE (RDSF(42), RST6)
EQUIVALENCE (RDSF(43), RSQ6)
EQUIVALENCE (RDSF(44), RSPR67)
EQUIVALENCE (RDSF(45), RSDT67)
EQUIVALENCE (RDSF(46), RSNPT)
EQUIVALENCE (RDSF(47), RSGPT)
EQUIVALENCE (RDSF(48), RSP7)
EQUIVALENCE (RDSF(49), RST7)
EQUIVALENCE (RDSF(50), RSGOUT)
EQUIVALENCE (RDSF(51), RSGACC)
EQUIVALENCE (RDSF(52), RSNOUT)
EQUIVALENCE (RDSF(53), RSVOL)
EQUIVALENCE (RDSF(54), RSAREA)
EQUIVALENCE (RDSF(55), RSTIMP)
EQUIVALENCE (RDSF(56), RSTIMN)
EQUIVALENCE (RDSF(57), RSI)

```

EQUIVALENCE (RDSF(58),RSKIN )
EQUIVALENCE (RDSF(59),RSKEXH)
EQUIVALENCE (RDSF(60),RSGC )
EQUIVALENCE (RDSF(61),RSK )
EQUIVALENCE (RDSF(62),RSGIGV)
EQUIVALENCE (RDSF(63),RSGOGV)
EQUIVALENCE (RDSF(64),RSGACL)
EQUIVALENCE (RDSF(65),RSDNGG)
EQUIVALENCE (RDSF(66),RSDNO )
EQUIVALENCE (RDSF(67),RSNOMP)
EQUIVALENCE (RDSF(68),RSGOMP)
EQUIVALENCE (RDSF(69),RSPWR )
EQUIVALENCE (RDSF(70),RSROW3)

```

C. DISC FILE

```

EQUIVALENCE (RVDISC(1),IDDISC(1))

```

```

COMMON/CBGEN/INDEX, RDSF, RDPAR

```

```

COMMON/CBRVAR/RVVAR

```

```

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC

```

```

DATA NAME(1),NAME(2),NAME(3)/ZHRF,ZHRM,ZHAT/

```

```

DATA NAM(1),NAM(2),NAM(3)/ZHRI,ZHND,ZHEX/

```

C. CONVERT ALL VARIABLES TO REAL

```

I=1

```

```

400 RVVAR(I)=SVADC(I)

```

```

IF(I.GE.50) GO TO 410

```

```

I=I+1

```

```

GO TO 400

```

```

410 J=1

```

```

I=I+1

```

```

420 RVVAR(I)=SVDAC(J)

```

```

IF(J.GE.50) GO TO 430

```

```

I=I+1

```

```

J=J+1

```

```

GO TO 420

```

```

430 J=1

```

```

I=I+1

```

```

440 RVVAR(I)=SVVAR(J)

```

```

IF(J.GE.150) GO TO 450

```

```

I=I+1

```

```

J=J+1

```

```

GO TO 440

```

```

450 CONTINUE

```

C. OUTPUT RESULTS

```

IL=1

```

```

L=0

```

```

CALL QMOND(21,22,NAME)

```

```

CALL QMOND(21,23,NAM)

```

C. SET UP WRITE FORMAT

```

200 CALL QMOND(17,22,IFORM(1),IFORM(40))

```

```

IF (IL.EQ.3) GO TO 728

```

C. GET CURRENT INDICES

```

CALL QMOND(17,23,ID(1),ID(6))

```

C..... SET UP DATA

```
I=1
K=1
230 IV=ID(I)
IS=ID(I+1)
RVOUT(K)=RVVAR(IV)*RDSF(IS)
IF(K. GE. 3) GO TO 250
I=I+2
K=K+1
GO TO 230
```

C..... OUTPUT DATA

```
250 WRITE(6,IFORM) (RVOUT(K),K=1,3)
GO TO 280
```

```
728 WRITE (6,IFORM)
280 IF(IL. GE. 14) GO TO 290
IL=IL+1
GO TO 200
290 CONTINUE
RETURN
END
```

PROGRAM SIZE = 1022

SUBROUTINE INLET

```

C ..... EXECUTIVE SUBROUTINE TO CALCULATE TOTAL CONDITIONS AT INLET
C .....
C ..... STORAGE DECLARATIONS
      LOGICAL LERR
      SCALED FRACTION SSQRT
      SCALED FRACTION SVIC(40),SDIC(40)
      SCALED FRACTION SCPAR(150),SVVAR(150),SVADC(50),SVDAC(50)
      SCALED FRACTION SVPA ,SVTA ,SVTHEA ,SVDELA ,SVW0 ,SVQ0
      SCALED FRACTION SVP1 ,SVT1 ,SVW1 ,SVTHE1 ,SVDEL1 ,SVQ1
      SCALED FRACTION SCUNIT, SCINL1, SCINL2, SCINL3, SCINL4
C ..... EQUIVALENCE DECLARATIONS
      EQUIVALENCE (SVVAR(1) ,SVQ0 )
      EQUIVALENCE (SVVAR(2) ,SVTHE1)
      EQUIVALENCE (SVVAR(3) ,SVDEL1)
      EQUIVALENCE (SVVAR(4) ,SVQ1 )
      EQUIVALENCE (SVADC(1) ,SVPA )
      EQUIVALENCE (SVADC(2) ,SVTA )
      EQUIVALENCE (SVADC(3) ,SVTHEA)
      EQUIVALENCE (SVADC(4) ,SVDELA)
      EQUIVALENCE (SVADC(5) ,SVP1 )
      EQUIVALENCE (SVDAC(1) ,SVW0 )
      EQUIVALENCE (SVDAC(2) ,SVW1 )
      EQUIVALENCE (SVDAC(7) ,SVT1 )
      EQUIVALENCE (SCPAR(1),SCUNIT)
      EQUIVALENCE (SCPAR(2),SCINL1)
      EQUIVALENCE (SCPAR(3),SCINL2)
      EQUIVALENCE (SCPAR(4),SCINL3)
      EQUIVALENCE (SCPAR(5),SCINL4)
C ..... COMMON DECLARATIONS
      COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
      DEFINE IERR('635)
C ..... INITIALIZE OVERFLOW CHECK
      IERR=0
      CALL QWCLL(1, .FALSE., IER)
C .....
C ..... CALCULATE TOTAL TEMPERATURE
      SVT1=SCINL2*SVTA

      SVW0=(SVDELA/SSQRT(SVTHEA))*SSQRT(0.999998-((SVP1*SCINL1)/SVPA))
      1/SCINL3
      SVQ0=SVW0

      IF(LERR(13)) CALL QWCLL(1, .TRUE., IER)
      RETURN

```

PAGE 2

END

F-30

PROGRAM SIZE = 105

SUBROUTINE RIGV

C
 C EXECUTIVE SUBROUTINE TO LOOK UP FAN DATA
 C

C STORAGE DECLARATIONS

LOGICAL LERR
 LOGICAL LVFLAG(25)
 SCALED FRACTION SSQRT
 SCALED FRACTION SVIC(40), SDIC(40)
 SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
 SCALED FRACTION SDNC(7), SDPR12(7,2), SDQ1(7,15), SDDT12(7,15)

SCALED FRACTION SVPA , SVTA , SVTHEA , SVDELA , SVWO , SVQO
 SCALED FRACTION SVP1 , SVT1 , SVW1 , SVTHE1 , SVDEL1 , SVQ1
 SCALED FRACTION SVP2 , SVT2 , SVW2 , SVTHE2 , SVDEL2 , SVQ2
 SCALED FRACTION SVPR12 , SVDT12 , SVNC , SVNCG , SVNREL , SVGCOM
 SCALED FRACTION SVPWR1 , SVPWR2 , SVPWR3 , SVPWR4 , SVPWR5
 SCALED FRACTION SVNOUT , SVGC , SVGIGV , SVGOGV , SVGACL , SVDNG6
 SCALED FRACTION SVNMPF
 SCALED FRACTION SCFAN0, SCFAN7, SCFAN8, SCFAN9
 SCALED FRACTION SCINL5, SCFAN1, SCFAN2, SCFAN3, SCFAN4
 SCALED FRACTION SCFAN5, SCFAN6, SCCOM1, SCCOM2, SCCOM3

C EQUIVALENCE DECLARATIONS

EQUIVALENCE (SVVAR(1) , SVQO)
 EQUIVALENCE (SVVAR(2) , SVTHE1)
 EQUIVALENCE (SVVAR(3) , SVDEL1)
 EQUIVALENCE (SVVAR(4) , SVQ1)
 EQUIVALENCE (SVVAR(5) , SVTHE2)
 EQUIVALENCE (SVVAR(6) , SVDEL2)
 EQUIVALENCE (SVVAR(7) , SVQ2)
 EQUIVALENCE (SVVAR(8) , SVPR12)
 EQUIVALENCE (SVVAR(9) , SVDT12)
 EQUIVALENCE (SVVAR(29) , SVPWR1)
 EQUIVALENCE (SVVAR(38) , SVNMPF)
 EQUIVALENCE (SVVAR(44) , SVGIGV)
 EQUIVALENCE (SVADC(1) , SVPA)
 EQUIVALENCE (SVADC(2) , SVTA)
 EQUIVALENCE (SVADC(3) , SVTHEA)
 EQUIVALENCE (SVADC(4) , SVDELA)
 EQUIVALENCE (SVADC(5) , SVP1)
 EQUIVALENCE (SVADC(6) , SVP2)
 EQUIVALENCE (SVADC(11) , SVNC)
 EQUIVALENCE (SVDAC(1) , SVWO)
 EQUIVALENCE (SVDAC(2) , SVW1)
 EQUIVALENCE (SVDAC(7) , SVT1)
 EQUIVALENCE (SVDAC(8) , SVT2)
 EQUIVALENCE (SCPAN(7) , SCFAN1)
 EQUIVALENCE (SCPAN(8) , SCFAN2)
 EQUIVALENCE (SCPAN(9) , SCFAN3)


```

EQUIVALENCE (SCPAR(10), SCFAN4)
EQUIVALENCE (SCPAR(11), SCFAN5)
EQUIVALENCE (SCPAR(12), SCFAN6)
EQUIVALENCE (SCPAR(49), SCFAN0)
EQUIVALENCE (SCPAR(50), SCFAN7)
EQUIVALENCE (SCPAR(51), SCFAN8)
EQUIVALENCE (SCPAR(52), SCFAN9)

```

C..... COMMON DECLARATIONS

```

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBRIGV/NCM, NPC, SDQ1, SDNC, SDPR12, SDDT12
COMMON/CBFLAG/LVFLAG
DEFINE IERR('635)

```

C..... INITIALIZE OVERFLOW CHECK

```

IERR=0
CALL QWCLL(2, FALSE, IER)

```

C

```

IF(.NOT. LVFLAG(25)) GO TO 500
SVTHE1=SCFAN1*SVT1/SVTA
SVDEL1=SCFAN2*SVP1/SVPA
SVPR12=SCFAN3*SVP2/SVP1

```

C..... CALCULATE AERODYNAMIC ROTOR SPEED

```

SVNMPF=(SVNC*SCFAN4)/SSQRT(SCFAN0*SVT1/SVTA)

```

C..... LOOK UP FAN DATA

```

CALL FNMAP(SVNMPF, SDNC, IX, NCM, SVPR12, SDPR12, IY, NPC, IER, SDQ1, SVQ1)
CALL FNLK2(SDDT12, SVDT12)

```

```

SVW1=(SVQ1*SVDEL1*SCFAN5)/SSQRT(SVTHE1)

```

C..... CALCULATE FAN EXIT TEMPERATURE

```

SVT2=SVT1*SCFAN7+SVDT12*SVT1*SCFAN6
SVPWR1=SVW1*(SVT2-SVT1*SCFAN7)/SCFAN8

```

C..... CALCULATE FAN TORQUE

```

SVGIGV=SCFAN9*SVPWR1/SVNC

```

```

GO TO 600

```

```

500 SVT2=SVT1*SCFAN7

```

```

SVQ1=SVQ0

```

```

SVDT12=0.0000

```

```

SVW1=SVW0

```

```

SVPWR1=0.0000

```

```

SVGIGV=0.0000

```

```

600 IF(IERR(13)) CALL QWCLL(2, TRUE, IER)

```

```

RETURN

```

```

END

```

PROGRAM SIZE = 266

SUBROUTINE COMP

```

C ..... EXECUTIVE SUBROUTINE TO LOOK UP HP COMPRESSOR DATA
C .....
C ..... STORAGE DECLARATIONS
C      LOGICAL LERR
C      LOGICAL SENSW
C      LOGICAL LVFLAG(25)

      SCALED FRACTION SSQRT
      SCALED FRACTION SVIC(40), SDIC(40)
      SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
      SCALED FRACTION SDNREL(8), SDPR23(8, 2), SDQ2(8, 15), SDDT23(8, 15)

C      SCALED FRACTION SVPA , SVTA , SVTHEA , SVDELA , SVWO , SVQ0
      SCALED FRACTION SVP1 , SVT1 , SVW1 , SVTHE1 , SVDEL1 , SVQ1
      SCALED FRACTION SVP2 , SVT2 , SVW2 , SVTHE2 , SVDEL2 , SVQ2
      SCALED FRACTION SVPR12 , SVDT12 , SVNC , SVNGG , SVNREL , SVGCOM
      SCALED FRACTION SVP3 , SVT3 , SVW3 , SVPR23 , SVDT23 , SVP4
      SCALED FRACTION SVNOUT , SVGC , SVGIGV , SVGOGV , SVGACL , SVDNGG
      SCALED FRACTION SVPWR1 , SVPWR2 , SVPWR3 , SVPWR4 , SVPWR5
      SCALED FRACTION SVNMPC
      SCALED FRACTION SCCOM0, SCCOM7, SCCOM8, SCCOM9, SCCOMG
      SCALED FRACTION SCFAN5, SCFAN6, SCCOM1, SCCOM2, SCCOM3
      SCALED FRACTION SCCOM4, SCCOM5, SCCOM6, SCOGV1, SCOGV2

C ..... EQUIVALENCE DECLARATIONS

      EQUIVALENCE (SVVAR(5) , SVTHE2)
      EQUIVALENCE (SVVAR(6) , SVDEL2)
      EQUIVALENCE (SVVAR(7) , SVQ2 )
      EQUIVALENCE (SVVAR(10), SVGCOM)
      EQUIVALENCE (SVVAR(11), SVPR23)
      EQUIVALENCE (SVVAR(12), SVDT23)
      EQUIVALENCE (SVVAR(30), SVPWR2)
      EQUIVALENCE (SVVAR(39) , SVNMPC)
      EQUIVALENCE (SVADC(1) , SVPA )
      EQUIVALENCE (SVADC(2) , SVTA )
      EQUIVALENCE (SVADC(6) , SVP2 )
      EQUIVALENCE (SVADC(7) , SVP3 )
      EQUIVALENCE (SVADC(10), SVNGG )
      EQUIVALENCE (SVADC(11), SVNC )
      EQUIVALENCE (SVADC(12), SVNREL)
      EQUIVALENCE (SVDAC(3) , SVW2 )
      EQUIVALENCE (SVDAC(8) , SVT2 )
      EQUIVALENCE (SVDAC(9) , SVT3 )
      EQUIVALENCE (SCPAR(13), SCCOM1)
      EQUIVALENCE (SCPAR(14), SCCOM2)
      EQUIVALENCE (SCPAR(15), SCCOM3)
      EQUIVALENCE (SCPAR(16), SCCOM4)
      EQUIVALENCE (SCPAR(17), SCCOM5)
      EQUIVALENCE (SCPAR(18), SCCOM6)

```

```

EQUIVALENCE (SCPAR(53), SCCOM0)
EQUIVALENCE (SCPAR(54), SCCOM7)
EQUIVALENCE (SCPAR(55), SCCOM8)
EQUIVALENCE (SCPAR(56), SCCOM9)
EQUIVALENCE (SCPAR(57), SCCOM0)

```

C..... COMMON DECLARATIONS

```

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBCOM4/NCM, NPC, SDQ2, SDNREL, SDPR23, SDDT23
COMMON/CBFLAG/LVFLAG
DEFINE IERR('635)

```

C..... INITIALIZE OVERFLOW CHECK

```

IERR=0
CALL QWCLL(3, .FALSE., IER)

```

C..... CALCULATE PRESSURE RATIO

```

400 SVTHE2=SCCOM1*SVT2/SVTA
SVDEL2=SCCOM2*SVP2/SVPA
SVPR23=(SCCOM3*SVP3)/SVP2

```

C..... CALCULATE AERODYNAMIC ROTOR SPEED

```

SVNMP2=SCCOM4*SVNREL*SSQRT((SVTA*SCCOM0)/SVT2)
IF(.NOT. LVFLAG(20)) GO TO 250
GO TO 260

```

```

250 SCCOM4=0.90094
SVNMP2= SVNREL*SSQRT((SVTA*SCCOM0)/SVT2)/SCCOM4

```

260 CONTINUE

C LOOK UP HP COMPRESSOR MAP DATA

```

CALL FNMAP(SVNMP2, SDNREL, IX, NCM, SVPR23, SDPR23, IY, NPC, IER, SDQ2, SVQ2
1)
CALL FNLK2(SDDT23, SVDT23)

```

C CALCULATE INLET MASS FLOW

```

SVW2=(SVQ2*SVDEL2)/(SCCOM5*SSQRT(SVTHE2))

```

C CALCULATE OUTLET TEMPERATURE

```

SVT3=SVT2*SCCOM7+SVDT23*SVT2*SCCOM6

```

C CALCULATE HIGH PRESSURE COMPRESSOR TORQUE

```

SVPWR2=SVW2*(SVT3-SVT2*SCCOM7)/SCCOM8
SVGCOM=SVPWR2*SCCOM9/SVNREL

```

```

*800 IF(IERR(13)) CALL QWCLL(3, .TRUE., IER)
RETURN
END

```

PROGRAM SIZE = 251

C. SUBROUTINE ROGV

C.

C. STORAGE DECLARATIONS

LOGICAL LERR

LOGICAL LVFLAG(24)

SCALED FRACTION SSORT

SCALED FRACTION SVIC(40), SDIC(40)

SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)

SCALED FRACTION SVPR12, SVDT12, SVNC, SVNGG, SVNREL, SVGCOM

SCALED FRACTION SVP3, SVT3, SVW3, SVPR23, SVDT23, SVP4

SCALED FRACTION SVT4, SVW4, SVWF, SVT34, SVP5, SVT5

SCALED FRACTION SVNOUT, SVGC, SVGIGV, SVGOGV, SVGACL, SVDNGG

SCALED FRACTION SVPWR1, SVPWR2, SVPWR3, SVPWR4, SVPWR5

SCALED FRACTION SVROW3, SVT4S, SVGACC, SVW2

SCALED FRACTION SCOGV0, SCOGV7, SCOGV8, SCOGV9

SCALED FRACTION SCCOM4, SCCOM5, SCCOM6, SCOGV1, SCOGV2

SCALED FRACTION SCOGV3, SCOGV4, SCOGV5, SCOGV6, SCGGT1

C. EQUIVALENCE DECLARATIONS

EQUIVALENCE (SVVAR(13), SVP4)

EQUIVALENCE (SVVAR(14), SVT34)

EQUIVALENCE (SVVAR(28), SVGACC)

EQUIVALENCE (SVVAR(29), SVPWR1)

EQUIVALENCE (SVVAR(30), SVPWR2)

EQUIVALENCE (SVVAR(31), SVPWR3)

EQUIVALENCE (SVVAR(34), SVROW3)

EQUIVALENCE (SVVAR(35), SVT4S)

EQUIVALENCE (SVADC(7), SVP3)

EQUIVALENCE (SVADC(11), SVNC)

EQUIVALENCE (SVDAC(3), SVW2)

EQUIVALENCE (SVDAC(4), SVW3)

EQUIVALENCE (SVDAC(5), SVW4)

EQUIVALENCE (SVDAC(9), SVT3)

EQUIVALENCE (SVDAC(10), SVT4)

EQUIVALENCE (SCPAR(19), SCOGV1)

EQUIVALENCE (SCPAR(20), SCOGV2)

EQUIVALENCE (SCPAR(21), SCOGV3)

EQUIVALENCE (SCPAR(22), SCOGV4)

EQUIVALENCE (SCPAR(23), SCOGV5)

EQUIVALENCE (SCPAR(24), SCOGV6)

EQUIVALENCE (SCPAR(58), SCOGV0)

EQUIVALENCE (SCPAR(59), SCOGV7)

EQUIVALENCE (SCPAR(60), SCOGV8)

EQUIVALENCE (SCPAR(61), SCOGV9)

C. COMMON DECLARATIONS

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC

COMMON/CBFLAG/LVFLAG

DEFINE IERR('635)

C... INITIALIZE OVERFLOW CHECK

IERR=0

CALL QWCLL(4, .FALSE., IER)

C CALCULATE TEMPERATURE RISE

IF(.NOT.LVFLAG(24)) GO TO 500

SVROW3=(SCOGV4*SVP3)/SVT3

SVW3=SVW2

SVPWR3=(SCOGV5*SVNC*SVW3*SVW3)/SVROW3

C CALCULATE EXIT TEMPERATURE

SVT34=(SCOGV0/SCOGV6)*((SVPWR3/SVW3)

SVT4=SVT3-SVT34

SVT4S=SVT3-SVT34/SCOGV2

SVP4=SVP3*((SVT4S/SVT3)**(3.59945))*SCOGV8

GO TO 800

500 SVT4=SVT3

SVW3=SVW2

SVP4=SVP3

SVPWR3=0.00000

C CALCULATE ROGV TORQUE

800 IF(IERR(13)) CALL QWCLL(4, .TRUE., IER)

RETURN

END

PROGRAM SIZE = 172

SUBROUTINE HTEXGR

C
C SUBROUTINE TO CALCULATE THE PERFORMANCE OF HEAT EXCHANGER
C

C STORAGE DECLARATIONS

LOGICAL LVFLAG(25)
LOGICAL LERR
SCALED FRACTION SSQRT
SCALED FRACTION SVIC(40), SDIC(40)
SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
SCALED FRACTION SCHEX1, SCHEX2, SCHEX3, SCHEX4, SCHEX5, SCHEX6
SCALED FRACTION SVE, SVT4, SVT7, SVT8, SVW3, SVP4, SVP7
C EQUIVALENCE DECLARATIONS

EQUIVALENCE (SVDAC(4), SVW3)
EQUIVALENCE (SVDAC(10), SVT4)
EQUIVALENCE (SVDAC(13), SVT7)
EQUIVALENCE (SVDAC(18), SVT8)
EQUIVALENCE (SVVAR(13), SVP4)
EQUIVALENCE (SVVAR(26), SVP7)
EQUIVALENCE (SVVAR(61), SVE)
EQUIVALENCE (SCPAR(91), SCHEX1)
EQUIVALENCE (SCPAR(92), SCHEX2)
EQUIVALENCE (SCPAR(93), SCHEX3)
EQUIVALENCE (SCPAR(94), SCHEX4)
EQUIVALENCE (SCPAR(95), SCHEX5)
EQUIVALENCE (SCPAR(96), SCHEX6)

C COMMON DECLARATIONS

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBFLAG/LVFLAG

DEFINE IERR(/635)

C INITIALIZE OVERFLOW CHECK

IERR=0
CALL QWCLL(11, .FALSE., IER)

IF(LVFLAG(21)) GO TO 100

C HEAT EXCHANGER EFFECTIVENESS = SVE
SVE=SCHEX1

C HEAT EXCHANGER OUTLET TEMPERATURE (GAS SIDE)
SVT8=SVT7-SCHEX3*SVE*(SVT7/SCHEX2-SVT4)

C HEAT EXCHANGER OUTLET TEMPERATURE (AIR SIDE)
SVT4=SVT4+SVE*(SVT7/SCHEX2-SVT4)

C HEAT EXCHANGER OUTLET PRESSURE (AIR SIDE)
SVP4=SVP4*SCHEX4

C HEAT EXCHANGER OUTLET PRESSURE (GAS SIDE)
SVP7=SVP7/SCHEX5

IF(LERR(13)) CALL QWCLL(11, .TRUE., IER)

100 CONTINUE
RETURN
END

PROGRAM SIZE = 113

SUBROUTINE COMBST

```

C .....
C     SUBROUTINE TO CALCULATE THE PERFORMANCE OF THE COMBUSTOR
C .....

C ..... STORAGE DECLARATIONS
LOGICAL LERR
SCALED FRACTION SSQRT...
SCALED FRACTION SVIC(40), SDIC(40)
SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
SCALED FRACTION SCGB4 , SCGB5 , SCGB6 , SCCC1 , SCCC2
SCALED FRACTION SCCC3 , SCCC4 , SCCC5 , SCCC6
SCALED FRACTION SVP3 , SVT3 , SVW3 , SVPR23 , SVDT23 , SVP4
SCALED FRACTION SVT4 , SVW4 , SVWF , SVT34 , SVP5 , SVT5

C ..... EQUIVALENCE DECLARATIONS

EQUIVALENCE (SVVAR(13), SVP4 )
EQUIVALENCE (SVVAR(15), SVP5 )
EQUIVALENCE (SVADC(15), SVWF )
EQUIVALENCE (SVDAC(4) , SVW3 )
EQUIVALENCE (SVDAC(10), SVT4 )
EQUIVALENCE (SVDAC(11), SVT5 )
EQUIVALENCE (SCPAR(25), SCCC1 )
EQUIVALENCE (SCPAR(26), SCCC2 )
EQUIVALENCE (SCPAR(27), SCCC3 )
EQUIVALENCE (SCPAR(28), SCCC4 )
EQUIVALENCE (SCPAR(29), SCCC5 )
EQUIVALENCE (SCPAR(30), SCCC6 )

C ..... COMMON DECLARATIONS
COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC

DEFINE IERR(635)

C     INITIALIZE OVERFLOW CHECK
IERR=0
CALL QWCLL(5, .FALSE., IER)

C     M CALCULATE PRESSURE LOSS IN COMBUSTOR
SVP5=SCCC1*SVP4

C     CALCULATE OUTLET TEMPERATURE
SVT5=SCCC2*SVT4+SCCC3+(SCCC4*SVWF)/SVW3+(SCCC5*SVWF**2)/SVW3**2
1+((SCCC6*SVWF)/SVW3)*SVT4

C     CHECK FOR OVERFLOW
IF(LERR(13)) CALL QWCLL(5, .TRUE., IER)
RETURN
END

```

PROGRAM SIZE = 125

SUBROUTINE GGTURB

C.....
 C EXECUTIVE SUBROUTINE TO LOOK UP HP TURBINE DATA
 C.....

LOGICAL LVFLAG(25)

LOGICAL LERR

SCALED FRACTION SSQRT

SCALED FRACTION SVIC(40), SDIC(40)

SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)

SCALED FRACTION SDNGG(4), SDPR56(14), SDQ5(14,4), SDDT56(14,4)

SCALED FRACTION SDPR5(4), SDNEW(12), SDEFF(12,4)

SCALED FRACTION SVPR12, SVDT12, SVNC, SVNGG, SVNREL, SVGCOM

SCALED FRACTION SVP3, SVT3, SVW3, SVPR23, SVDT23, SVP4

SCALED FRACTION SVT4, SVW4, SVWF, SVT34, SVP5, SVT5

SCALED FRACTION SVW5, SVRTT5, SVQ5, SVPR56, SVDT56, SVGGGT

SCALED FRACTION SVP6, SVT6, SVRTT6, SVQ6, SVPR67, SVDT67

SCALED FRACTION SVNPT, SVGPT, SVP7, SVT7, SVGOUT, SVGACC

SCALED FRACTION SVNOUT, CVCC, CVGICV, CVGQCV, CVGACL, CVDNCC

SCALED FRACTION SVPWR1, SVPWR2, SVPWR3, SVPWR4, SVPWR5

SCALED FRACTION SVNEW, SVNEW1, SVEFF, SVEFF1, SVNWI, SVEFFP

SCALED FRACTION SCGGT0, SCGGT7, SCGGT8, SCGGT9, SCGGT6

SCALED FRACTION SCCOMG

SCALED FRACTION SVNMPG

SCALED FRACTION SVGACC

SCALED FRACTION SCGGV3, SCGGV4, SCGGV5, SCGGV6, SCGGT1

SCALED FRACTION SCGGT2, SCGGT3, SCGGT4, SCGGT5, SCGGT6

SCALED FRACTION SCNEWD, SCEFFD, SCPRDG, SCNGGD, SCGGTP, SCGGTE

C..... EQUIVALENCE DECLARATIONS

EQUIVALENCE (SVVAR(15), SVP5)

EQUIVALENCE (SVVAR(16), SVRTT5)

EQUIVALENCE (SVVAR(17), SVQ5)

EQUIVALENCE (SVVAR(18), SVPR56)

EQUIVALENCE (SVVAR(19), SVDT56)

EQUIVALENCE (SVVAR(20), SVGGGT)

EQUIVALENCE (SVVAR(28), SVGACC)

EQUIVALENCE (SVVAR(29), SVPWR1)

EQUIVALENCE (SVVAR(30), SVPWR2)

EQUIVALENCE (SVVAR(31), SVPWR3)

EQUIVALENCE (SVVAR(32), SVPWR4)

EQUIVALENCE (SVVAR(40), SVNMPG)

EQUIVALENCE (SVVAR(46), SVNEW)

EQUIVALENCE (SVVAR(47), SVNEW1)

EQUIVALENCE (SVVAR(48), SVEFF)

EQUIVALENCE (SVVAR(49), SVEFF1)

EQUIVALENCE (SVADC(9), SVP6)

EQUIVALENCE (SVADC(10), SVNGG)

EQUIVALENCE (SVADC(12), SVNREL)

EQUIVALENCE (SVDAC(5), SVW4)

```

EQUIVALENCE (SVDAC(11),SVT5 )
EQUIVALENCE (SVDAC(12),SVT6 )
EQUIVALENCE (SVDAC(14),SVDNGG)
EQUIVALENCE (SCPAR(31),SCGGT1)
EQUIVALENCE (SCPAR(32),SCGGT2)
EQUIVALENCE (SCPAR(33),SCGGT3)
EQUIVALENCE (SCPAR(34),SCGGT4)
EQUIVALENCE (SCPAR(35),SCGGT5)
EQUIVALENCE (SCPAR(36),SCGGT6)
EQUIVALENCE (SCPAR(57),SCCOMG)
EQUIVALENCE (SCPAR(62),SCGGT0)
EQUIVALENCE (SCPAR(63),SCGGT7)
EQUIVALENCE (SCPAR(64),SCGGT8)
EQUIVALENCE (SCPAR(65),SCGGT9)
EQUIVALENCE (SCPAR(66),SCGGTG)
EQUIVALENCE (SCPAR(80),SCNEWD)
EQUIVALENCE (SCPAR(81),SCEFFD)
EQUIVALENCE (SCPAR(82),SCPRDG)
EQUIVALENCE (SCPAR(83),SCNGGD)
EQUIVALENCE (SCPAR(84),SCGGTP)
EQUIVALENCE (SCPAR(85),SCGGTE)

```

C..... COMMON DECLARATIONS

```

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBGGT/NCM, NPC, SDNGG, SDPR56, SDQ5, SDDT56
COMMON/CBGNW/NC , NP , SDPR5 , SDNEW, SDEFF
COMMON/CBFLAG/LVFLAG
DEFINE IERR('635)

```

C..... INITIALIZE OVERFLOW CHECK

```

IERR=0
CALL QWCLL(6, FALSE, IER)

```

C..... CALCULATE GG TURBINE PRESSURE RATIO

```

SVPR56=SCGGT1*SVPR5/SVR6

```

C..... CALCULATE DIMENSIONLESS SPEED

```

SVNMPG=(SVNGG/SSQRT(8VT5))*SCGGT2

```

C..... LOOK UP MAP DATA

```

CALL FNGN2(SVPR56, SDPR56, IR, NPC, SVNMPG, SDNGG, JR, NCM, IER, SDQ5, SVQ5)

```

```

SVNEW1=SCNEWD*((SVNMPG*SCGGTE)/SCNGGD)/SSQRT(0.99999S-(0.25/
1SVPR56)**SCGGTP)
SVNEW=SVNEW1

```

```

I=1

```

```

10 IF(I.GT.2) GO TO 20

```

```

CALL FNGN2(SVNEW, SDNEW, IR, NP , SVPR56, SDPR5 , JR, NC , IER, SDEFF, SVEFF

```

```

1)

```

```

SVEFF1=SVEFF

```

```

SVNEW=(SVNEW1*SSQRT(SCEFFD))/SSQRT(SVEFF1)

```

```

I=I+1

```

```

GO TO 10

```

```

20 CONTINUE

```

```

SVDT56=SVEFF*(0.99999S-(0.25/SVPR56)**SCGGTP)/SCGGT5

```

```
C..... CALCULATE HP TURBINE MASS FLOW
SVW4=(SVQ5*SVP5/SSQRT(SVT5))/SCGGT3

C      CALCULATE OUTLET TEMPERATURE
SVT6=(SVT5*(((0.999995-SVDT56*SCGGT6*SCGGT0)*0.80)/SCGGT0))/0.80

C      CALCULATE TURBINE TORQUE
SVPWR4=SVW4*(SVT5-SVT6*SCGGT0)/SCGGT7
SVGGGT=SVPWR4*SCGGT8/SVNGG

C..... CALCULATE ACCELERATING TORQUE

      IF (.NOT. LVFLAG(10)) GO TO 500

      SVGACC=SVPWR4*SCGGTG/SVNGG-SVPWR2*SCCOMG/SVNREL
      GO TO 150
500 SVGACC=((SVPWR4-SVPWR2)*SCGGTG)/SVNGG

C..... CALCULATE DN/DT
150 .SVDNGG=SVGACC/SCGGT9

      IF (LERR(13)) CALL QWCLL(6, .TRUE., IER)
      RETURN
      END
```

PROGRAM SIZE = 416

SUBROUTINE PTURB

C
 C EXECUTIVE SUBROUTINE TO LOOK UP LP TURBINE DATA
 C

C..... STORAGE DECLARATIONS

LOGICAL LERR
 LOGICAL LVFLAG(25)
 SCALED FRACTION SSQRT
 SCALED FRACTION SVIC(40), SDIC(40)
 SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
 SCALED FRACTION SDNPT(5), SDPR67(11), SDQ6(11, 5), SDDT67(11, 5)
 SCALED FRACTION SDPR6 (4), SDNWI(12), SDEFFP(12, 4)

 SCALED FRACTION SVW5 , SVRTT5 , SVQ5 , SVPR56 , SVDT56 , SVGGGT
 SCALED FRACTION SVP6 , SVT6 , SVRTT6 , SVQ6 , SVPR67 , SVDT67
 SCALED FRACTION SVNPT , SVGPT , SVP7 , SVT7 , SVGOUT , SVGACC
 SCALED FRACTION SVPWR1 , SVPWR2 , SVPWR3 , SVPWR4 , SVPWR5
 SCALED FRACTION SVNMP
 SCALED FRACTION SVNEW , SVNEW1 , SVEFF , SVEFF1 , SVNWI , SVEFFP
 SCALED FRACTION SCPT0 , SCPT7 , SCPT8 , SCPT9
 SCALED FRACTION SCPT1 , SCPT2 , SCPT3 , SCPT4 , SCPT5
 SCALED FRACTION SCPT6 , SCLD1 , SCLD2 , SCLD3 , SCLD4
 SCALED FRACTION SCNWD , SCPRDP , SCPT6 , SCNPTD , SCPTP , SVFLWF

C..... EQUIVALENCE DECLARATIONS

EQUIVALENCE (SVVAR(21), SVRTT6)
 EQUIVALENCE (SVVAR(22), SVQ6)
 EQUIVALENCE (SVVAR(23), SVPR67)
 EQUIVALENCE (SVVAR(24), SVDT67)
 EQUIVALENCE (SVVAR(25), SVGPT)
 EQUIVALENCE (SVVAR(26), SVP7)
 EQUIVALENCE (SVVAR(33), SVPWR5)
 EQUIVALENCE (SVVAR(41) , SVNMP)
 EQUIVALENCE (SVVAR(50) , SVNWI)
 EQUIVALENCE (SVVAR(51) , SVEFFP)
 EQUIVALENCE (SVADC(9) , SVP6)
 EQUIVALENCE (SVADC(14), SVNPT)
 EQUIVALENCE (SVADC(17), SVFLWF)
 EQUIVALENCE (SVDAC(6) , SVW5)
 EQUIVALENCE (SVDAC(12), SVT6)
 EQUIVALENCE (SVDAC(13), SVT7)
 EQUIVALENCE (SCPAR(37), SCPT1)
 EQUIVALENCE (SCPAR(38), SCPT2)
 EQUIVALENCE (SCPAR(39), SCPT3)
 EQUIVALENCE (SCPAR(40), SCPT4)
 EQUIVALENCE (SCPAR(41), SCPT5)
 EQUIVALENCE (SCPAR(42), SCPT6)
 EQUIVALENCE (SCPAR(67), SCPT0)
 EQUIVALENCE (SCPAR(68), SCPT7)
 EQUIVALENCE (SCPAR(69), SCPT8)
 EQUIVALENCE (SCPAR(70), SCPT9)

```

EQUIVALENCE (SCPAR(86), SCNWID)
EQUIVALENCE (SCPAR(87), SCPRDP)
EQUIVALENCE (SCPAR(88), SCNPTD)
EQUIVALENCE (SCPAR(89), SCPTP)
EQUIVALENCE (SCPAR(90), SCPTEN)

```

C..... COMMON DECLARATIONS

```

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBPT/NCM, NPC, SDNPT, SDPR67, SDQ6, SDDT67
COMMON/CBPNEW/NC, NP, SDPR6, SDNWI, SDEFFP
COMMON/CBFLAG/LVFLAG

```

```

DEFINE IERR(1635)

```

C..... INITIALIZE OVERFLOW CHECK

```

IERR=0
CALL QWCLL(7, FALSE, IER)

```

C

C..... CALCULATE P TURBINE PRESSURE RATIO

```

SVPR67=SCPT1*SVP6/SVP7

```

C CALCULATE DIMENSIONLESS SPEED

```

SVNMPT=((SVNPT*SCPT2)/SSQRT(SVT6))

```

C LOOK UP MAP DATA

```

CALL FNGN2(SVPR67, SDPR67, IR, NPC, SVNMPT, SDNPT, JR, NCM, IER, SDQ6, SVQ6)

```

```

SVNWI=SCNWID*((SVNMPT*SCPTEN)/SCNPTD)/SSQRT(0.99999S-(0.33333/
2SVPR67)**SCPTP)

```

```

CALL FNGN2(SVNWI, SDNWI, IR, NP, SVPR67, SDPR6, JR, NC, IER, SDEFFP,
3SVEFFP)

```

```

SVDT67=SVEFFP*(0.99999S-(0.33333/SVPR67)**SCPTP)/SCPT7

```

C..... CALCULATE P TURBINE MASS FLOW

```

SVW5=(SVQ6*SVP6/SSQRT(SVT6))/SCPT3

```

```

IF(LVFLAG(23)) GO TO 200

```

```

SVW5=SVW5/SVFLWF

```

```

GO TO 200

```

C CALCULATE OUTLET TEMPERATURE

```

200 SVT7=SVT6/SCPT0-SVDT67*SVT6*SCPT4

```

C CALCULATE TURBINE TORQUE

```

SVPWR5=SVW5*(SVT6-SVT7*SCPT0)/SCPT5

```

```

SVGPT=SVPWR5*SCPT6/SVNPT

```

```

IF(IERR(13)) CALL QWCLL(7, TRUE, IER)

```

```

RETURN

```

```

END

```

```

PROGRAM SIZE = 274

```

SUBROUTINE LOAD

C
 C SUBROUTINE TO CALCULATE OUTPUT TORQUE
 C

C STORAGE DECLARATIONS

LOGICAL LVFLAG(25)

LOGICAL LERR

INTEGER WDTOT

SCALED FRACTION SSQRT

SCALED FRACTION SVIC(40), SDIC(40)

SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)

SCALED FRACTION SDNOUT(12), SDIF(11), SDGOUT(12, 11)

SCALED FRACTION SVNPT , SVGPT , SVP7 , SVT7 , SVGOUT , SVGACC

SCALED FRACTION SVNOUT , SVGC , SVGIGV , SVGOGV , SVGACL , SVDNGG

SCALED FRACTION SVDNO , SVGOMP , SVNMPD , SVIF

SCALED FRACTION SVPWR1 , SVPWR2 , SVPWR3 , SVPWR4 , SVPWR5

SCALED FRACTION SVNMPD

SCALED FRACTION SCPT6 , SCLD1 , SCLD2 , SCLD3 , SCLD4

SCALED FRACTION SCLD5 , SCLD6 , SCGB1 , SCGB2 , SCGB3

EQUIVALENCE (SVVAR(27), SVGOUT)

EQUIVALENCE (SVVAR(42), SVNMPD)

EQUIVALENCE (SVVAR(43), SVGOMP)

EQUIVALENCE (SVADC(13), SVNOUT)

EQUIVALENCE (SVADC(16), SVIF)

EQUIVALENCE (SCPAR(76), SCLD1)

EQUIVALENCE (SCPAR(77), SCLD2)

EQUIVALENCE (SCPAR(78), SCLD3)

EQUIVALENCE (SCPAR(79), SCLD4)

C
 C COMMON DECLARATIONS

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC

COMMON/CBLOAD/NN2, NIF, WDTOT, SDNOUT, SDIF, SDGOUT

COMMON/CBFLAG/LVFLAG

DEFINE IERR(63)

C INITIALIZE OVERFLOW CHECK

IERR=0

CALL QWCLL(8, FALSE, IER)

SVNMPD=SVNOUT*SCLD1

C LOOK UP LOAD MAP

CALL FNGN2(SVNMPD, SDNOUT, IR, NN2, SVIF, SCLD1, JR, NIF, IER, SDGOUT, SVGOMP
 1)

SVGOUT=SVGOMP/SCLD2

```
SVGOUT=(SCLD3*SVGOMP)/SCLD2  
IF(.NOT.LVFLAG(24)) GO TO 500  
GO TO 150  
500 SCLD3=0.63264  
SVGOUT=(SCLD3*SVGOMP)/SCLD2  
150 IF(LERR(13)) CALL QWCLL(8,TRUE,IER)  
RETURN  
END
```

PROGRAM SIZE = 125

SUBROUTINE GRBOX

C
 C SUBROUTINE TO CALCULATE ACCELERATING TORQUE
 C

C STORAGE DECLARATIONS

LOGICAL LERR
 LOGICAL LVFLAG(25)
 SCALED FRACTION SSQRT
 SCALED FRACTION SVIC(40), SDIC(40)
 SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
 SCALED FRACTION SVPR12, SVDT12, SVNC, SVNGG, SVNREL, SVGCOM
 SCALED FRACTION SVNPT, SVGPT, SVP7, SVT7, SVGOUT, SVGACC
 SCALED FRACTION SVNOUT, SVGC, SVGIGV, SVGOGV, SVGACL, SVDNGG
 SCALED FRACTION SVPWR1, SVPWR2, SVPWR3, SVPWR4, SVPWR5, SVPWR
 SCALED FRACTION SVDNO, SVWF, SCGB0
 SCALED FRACTION SVGCAS, SVSFC, SVPSHR
 SCALED FRACTION SCGB7, SCGB8, SCGB9, SCEXH1, SCEXH2
 SCALED FRACTION SCLD5, SCLD6, SCGB1, SCGB2, SCGB3
 SCALED FRACTION SCGB4, SCGB5, SCGB6, SCCC1, SCCC2

EQUIVALENCE (SVVAR(10), SVGCOM)
 EQUIVALENCE (SVVAR(25), SVGPT)
 EQUIVALENCE (SVVAR(27), SVGOUT)
 EQUIVALENCE (SVVAR(29), SVPWR1)
 EQUIVALENCE (SVVAR(31), SVPWR3)
 EQUIVALENCE (SVVAR(32), SVPWR4)
 EQUIVALENCE (SVVAR(33), SVPWR5)
 EQUIVALENCE (SVVAR(36), SVGCAS)
 EQUIVALENCE (SVVAR(37), SVGACL)
 EQUIVALENCE (SVVAR(45), SVPWR)
 EQUIVALENCE (SVADC(10), SVNGG)
 EQUIVALENCE (SVADC(11), SVNC)
 EQUIVALENCE (SVADC(13), SVNOUT)
 EQUIVALENCE (SVADC(15), SVWF)
 EQUIVALENCE (SVDAC(15), SVDNO)
 EQUIVALENCE (SVDAC(16), SVSFC)
 EQUIVALENCE (SVDAC(17), SVPSHR)
 EQUIVALENCE (SCPAR(43), SCGB1)
 EQUIVALENCE (SCPAR(44), SCGB2)
 EQUIVALENCE (SCPAR(45), SCGB3)
 EQUIVALENCE (SCPAR(46), SCGB4)
 EQUIVALENCE (SCPAR(47), SCGB5)
 EQUIVALENCE (SCPAR(48), SCGB6)
 EQUIVALENCE (SCPAR(71), SCGB7)
 EQUIVALENCE (SCPAR(72), SCGB8)
 EQUIVALENCE (SCPAR(73), SCGB9)
 EQUIVALENCE (SCPAR(100), SCGB0)

C
 COMMON /CBVAR/ SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
 COMMON /CBFLAG/ LVFLAG

DEFINE IERR('635)

C..... INITIALIZE OVERFLOW CHECK

IERR=0

CALL QWCLL(9, FALSE, IER)

IF (.NOT. LVFLAG(20)) GO TO 100

SVGCAS=SVGCOM*SCGB6+(SVPWR3-SVPWR1)*SCGB7/SVNC

GO TO 150

100 SVGCAS=0.0000

GO TO 150

150 SVGACL=(SVGPT*SCGB1+SVGCAS*SCGB2)/SCGB8-SVGOUT*SCGB3

SVPWR=(SVNOUT*SVGOUT)/SCGB5

C..... CALCULATE DN/DT

SVDNQ=SVGACL/SCGB9

SVSFC=(SVWF*0.250)/SVPWR

SVPSHR=(SVPWR3+0.990*SVPWR4*(SCGB0*SVNC/SVNGG))/SVPWR5

IF(IERR(13)) CALL QWCLL(9, TRUE, IER)

RETURN

END

PROGRAM SIZE = '203

SUBROUTINE EXHST

```

C.....
C..... SUBROUTINE TO CALCULATE EXHAUST PRESSURE LOSS .....
C.....
LOGICAL LVFLAG(25)
LOGICAL LERR
SCALED FRACTION SSQRT
SCALED FRACTION SVIC(40), SDIC(40)
SCALED FRACTION SCPAR(150), SVVAR(150), SVADC(50), SVDAC(50)
SCALED FRACTION SVPA, SVTA, SVQ6, SVP7
SCALED FRACTION SCGB7, SCGB8, SCGB9, SCEXH1, SCEXH2
SCALED FRACTION SCEXH3, SCEXH4, SCEXH5, SVW5, SVT7, SVNOUT
SCALED FRACTION SCEXH6

EQUIVALENCE (SVADC(1), SVPA)
EQUIVALENCE (SVADC(13), SVNOUT)
EQUIVALENCE (SVDAC(6), SVW5)
EQUIVALENCE (SVDAC(13), SVT7)
EQUIVALENCE (SVVAR(22), SVQ6)
EQUIVALENCE (SVVAR(26), SVP7)
EQUIVALENCE (SCPAR(74), SCEXH1)
EQUIVALENCE (SCPAR(75), SCEXH2)
EQUIVALENCE (SCPAR(97), SCEXH3)
EQUIVALENCE (SCPAR(98), SCEXH4)
EQUIVALENCE (SCPAR(99), SCEXH5)
EQUIVALENCE (SCPAR(101), SCEXH6)

COMMON/CBVAR/SVADC, SVDAC, SVVAR, SCPAR, SVIC, SDIC
COMMON/CBFLAG/LVFLAG
DEFINE IERR('635)

C..... INITIALIZE OVERFLOW CHECK
IERR=0
CALL QWCLL(10, .FALSE., IER)

SVP7=SVPA*(SCEXH1/0.66666S)*(0.66666S+(SVQ6*SVQ6)*SCEXH2*0.66666S)

IF(LVFLAG(22)) GO TO 100

SVP7=SVP7+SCEXH5*(SVP7/SVT7)*(((SVW5*SVT7*SCEXH3*SCEXH6/SVP7))-
-(SVNOUT*SCEXH6/SCEXH41)**2.0/(SCEXH6)**2.0)

100 IF(LERR(13)) CALL QWCLL(10, .TRUE., IER)
RETURN
END

```

PROGRAM SIZE = 150

C PROGRAM TO PUT HP COMPRESSOR DATA ON DISC

LOGICAL SENSW

INTEGER NAME(3), WDTOT, NP(8)

REAL RDN(8), RDPR(8,8), RDQ(8,8), RDET(8,8), RDDT(8,8)

REAL RDX(8), RDY(8), RDZ(8)

REAL RDXE(15), RDYE(15), RDZE(15)

SCALED FRACTION SDN(8), SDPR(8,2), SDQ(8,15), SDDT(8,15)

SCALED FRACTION SDAT(330)

SCALED FRACTION SVADC(2), SVDAC(2), SVQ, SVN, SVPR, SVDT

EQUIVALENCE (SDAT(1), NCM)

EQUIVALENCE (SVADC(1), SVPR)

EQUIVALENCE (SVADC(2), SVN)

EQUIVALENCE (SVDAC(1), SVQ)

EQUIVALENCE (SVDAC(2), SVDT)

COMMON/CBDAT/ NCM, NPC, SDQ, SDN, SDPR, SDDT

C I/O FORMATS

100 FORMAT(3A2)

110 FORMAT(20I4)

120 FORMAT(8F10.4)

130 FORMAT(I4,F10.4)

C CONSOLE INITIALIZATION

CALL QSHYIN(IER,680,680)

C READ CARD DATA

C FILENAME

READ(6,100) (NAME(I), I=1,3)

IF (SENSW(1)) WRITE(6,100) (NAME(I), I=1,3)

C INDICES

READ(6,110) NCM, NPC

IF (SENSW(1)) WRITE(6,110) NCM, NPC

C SCALE FACTORS

READ(6,120) RSPR, RSN, RSQ, RSDT, RSPR1

IF (SENSW(1)) WRITE(6,120) RSPR, RSN, RSQ, RSDT, RSPR1

C CONSTANTS

READ(6,120) RCGAM

IF (SENSW(1)) WRITE(6,120) RCGAM

C CALCULATE DEL

RCDEL = (RCGAM-1.0)/RCGAM

C COMPONENT MAP DATA

I = 1

140 READ(6,130) NP(I), RDN(I)

IF (SENSW(1)) WRITE(6,130) NP(I), RDN(I)

J = 1

150 READ(6,120) RDPR(I,J), RDQ(I,J), RDET(I,J)

RDPR(I,J) = (RDPR(I,J)-1.0)*(RSPR/RSPR1)+1.0

C CALCULATE TEMPERATURE RISE

RDDT(I,J) = (RDPR(I,J)**RCDEL - 1.0)/RDET(I,J)

RSPR=15.0

```
      IF (SENSW(1)) WRITE(6,120) RDPR(I,J), RDQ(I,J), RDET(I,J), RDDT(I,J)
      IF (J.GE.NP(I)) GO TO 160
      J = J+1
      GO TO 150
160    IF (I.GE.NCM) GO TO 170
      I = I+1
      GO TO 140
170    CONTINUE
C      GENERATE EQUAL BREAKPOINT DATA AND SCALE
      I = 1
C      SCALE SPEED DATA
180    SDN(I) = RDN(I)/RSN
C      SET UP DUMMY ARRAYS FOR EACH SPEED LINE
      J = 1
190    RDX(J) = RDPR(I,J)
      RDY(J) = RDQ(I,J)
      RDZ(J) = RDDT(I,J)
      IF (J.GE.NP(I)) GO TO 200
      J = J+1
      GO TO 190
200    CONTINUE
C      GET EQUAL BREAKPOINTS THIS SPEED LINE
      CALL FIXBPT(NP(I), RDX, RDY, RDZ, NPC, RDXE, RDEY, RDZE)
C      SCALE AND SAVE NEW DATA
      J = 1
210    IF (J.EQ.1) SDPR(I,1) = RDXE(1)/RSPR
      IF (J.EQ.NPC) SDPR(I,2) = RDXE(2)/RSPR
      SDQ(I,J) = RDEY(J)/RSQ
      SDDT(I,J) = RDZE(J)/RSDT
      IF (J.GE.NPC) GO TO 220
      J = J+1
      GO TO 210
220    CONTINUE
      IF (I.GE.NCM) GO TO 230
      I = I+1
      GO TO 180
230    CONTINUE
C      SET UP DISC FILE
      WDTOT = (3 + 2*NPC)*NCM + 2
      CALL FLNAME(NAME, WDTOT)
C      WRITE DATA ON DISC
      CALL DISCIO(16, NAME, SDAT, WDTOT)
C      READ DATA BACK FROM DISC
      CALL DISCIO(17, NAME, SDAT, WDTOT)
C      WRITE ALL DATA FOR INSPECTION
      CALL COMPDA(2HPR, SDN, SDPR, SDQ, SDDT, NCM, NPC)
C      LOOP TO GET DATA FROM ANALOG AND LOOK UP MAP DATA
240    CALL QRBADS(SVADC, 0, 2, IER)
      CALL FNMAP(SVN, SDN, IX, NCM, SVPR, SDPR, IY, NPC, IER, SDQ, SVQ)
      CALL FNLK2(SDDT, SVDT)
      CALL QWBDS(SVDAC, 0, 2, IER)
      CALL QSTDA
      GO TO 240
      END
```

PAGE 1 C FIXBPT

C SUBROUTINE TO GENERATE FIXED BREAKPOINT
C DATA FROM DATA OF ARBITRARY BREAKPOINT
C INTERVAL

C PROGRAM SPECIFICATION: BMI
C PROGRAMMER: BMI DATE 13 JULY 1977

SUBROUTINE FIXBPT(NP, RDX, RDY, RDZ, NPC, RDXE, RDYE, RDZE)

REAL RDX(1), RDY(1), RDZ(1)
REAL RDXE(1), RDYE(1), RDZE(1)
REAL RNK(2)

C CALCULATE BREAKPOINT INTERVAL
RFX = (RDX(NP)-RDX(1))/FLOAT(NPC-1)
C SET UP TO CALCULATE EQUAL BREAKPOINTS
I = 1
RVX = RDX(1)
100 CALL VBNR(RVX, RNK, RDX, NP, IER)
CALL LOOK1(RNK, RVY, RDY)
CALL LOOK1(RNK, RVZ, RDZ)
C SAVE DATA
IF (I.EQ.1) RDXE(1) = RDX(1)
IF (I.EQ.NPC) RDXE(2) = RDX(NP)
RDYE(I) = RVY
RDZE(I) = RVZ
IF (I.EQ.NPC) GO TO 120
I = I+1
RVX = RVX + RFX
GO TO 100
120 CONTINUE
RETURN
END

PROGRAM SIZE = 320

PAGE 1

C FLNAME
C SUBROUTINE TO NAME AND SIZE A DATA FILE
C BLOCK COUNT (WORDS/RECORD) IS FIXED AT 44 (1 SECTOR)
C SUBSEQUENT EXECUTIONS OF THIS ROUTINE WILL BE IGNORED

C PROGRAM SPECIFICATION: BMI
C PROGRAMMER: BMI DATE 13 JULY 77

SUBROUTINE FLNAME(NAME, WDTOT)

INTEGER NAME(3), WDTOT, RCT
LOGICAL LST
DATA LST/.TRUE./

C CHECK IF 1ST ENTRY
C IF (.NOT. LST) GO TO 100
C CALCULATE BLOCK COUNT
C RCT = WDTOT/44 + 1
C CREATE FILE NAME RECORD
C CALL QMOND(23, '22, NAME, '3, 44)
C CALL QMOND(24, '22, NAME, 44, RCT)
C SET ENTRY FLAG FALSE
C LST = .FALSE.
100 RETURN
END

PROGRAM SIZE = '61

PAGE 1

C DISCIO
C SUBROUTINE TO READ OR WRITE AN INTEGER ARRAY ON DISC

C PROGRAM SPECIFICATION: BMI
C PROGRAMMER: BMI DATE 13 JULY 77

SUBROUTINE DISCIO(ID, NAME, SDX, WDTOT)

INTEGER ID, NAME(3), WDTOT
SCALED FRACTION SDX(1)

C POINT TO FILE
CALL QMOND(21, '22, NAME)
C READ/WRITE DATA
CALL QMOND(ID, '22, SDX(1), SDX(WDTOT))
RETURN
END

PROGRAM SIZE = '50

PAGE 1

C COMPDA
C SUBROUTINE TO WRITE ON LINE PRINTER THE DATA PREPARED
C FOR THE MAP LOOKUP SUBROUTINES.

C PROGRAM SPECIFICATION: BMI
C PROGRAMMER: BMI DATE 13 JULY 77

SUBROUTINE COMPDA(ID, SDN, SDX, SDY, SDZ, NCM, NPC)

INTEGER ID, IDD, NCM, NPC
SCALED FRACTION SDN(1), SDX(1), SDY(1), SDZ(1)
DATA IDD/2HPR/

C I/O FORMATS
100 FORMAT(1H1, 10X, 60HSCALED FRACTION COMPRESSOR DATA//)
110 FORMAT(10X, 6HN/RTH=, S7, 8X, 7HMIN PR=, S7, 7X, 7HMAX PR=, S7//)
115 FORMAT(10X, 6HN/RTH=, S7, 9X, 6HMIN Q=, S7, 8X, 6HMAX Q=, S7//)
120 FORMAT(10X, 60HCORRECTED FLOW', 16X, 60HTEMPERATURE RISE')
125 FORMAT(10X, 60HPRESSURE RATIO', 16X, 60HTEMPERATURE RISE')
130 FORMAT(14X, S7, 24X, S7)

C WRITE HEADING
WRITE(6, 100)
C SET UP INDEX
I = 1
200 IMIN = I
IMAX = I+NCM
C WRITE ARGUMENTS
IF (ID.EQ.IDD) GO TO 210
C FLOW ARGUMENT
WRITE(6, 115) SDN(I), SDX(IMIN), SDX(IMAX)
WRITE(6, 125)
GO TO 220
C PRESSURE RATIO ARGUMENT
210 WRITE(6, 110) SDN(I), SDX(IMIN), SDX(IMAX)
WRITE(6, 120)
220 CONTINUE
C SET UP INDEX
J = 1
230 IJ = I + NCM*(J-1)
C WRITE DATA
WRITE(6, 130) SDY(IJ), SDZ(IJ)
C CHECK FOR END POINTS
IF (J.GE.NPC) GO TO 240
J = J+1
GO TO 230
240 CONTINUE
IF (I.GE.NCM) GO TO 250
I = I+1

PAGE 2

250 GO TO 200
CONTINUE
RETURN
END

PROGRAM SIZE = 1550

PAGE 1 C ... VBNR...

```

C .....
C      ARGUMENT NORMALIZATION, 1-D FUNCTION
C .....
C      SUBROUTINE  VBNR(RVX,RNX,RDX,NX,IER)
C      DIMENSION RDX(20),RNX(2)
C      J=RNX(1)

C .....
C ..... BEGIN SEARCH FOR BOUNDING BREAKPOINTS
C      IF(RVX-RDX(J)) 110,150,170

C .....
C ..... CHECK BOTTOM END
C      110  IF(RVX-RDX(1)) 120,140,130

C .....
C ..... POINT OFF BOTTOM END
C      120  IER=0
C           GO TO 140

C .....
C ..... POINT GREATER THAN MINIMUM
C      130  J=J-1
C           IF(RVX-RDX(J)) 130,150,160

C .....
C ..... POINT EXACTLY ON MINIMUM
C      140  J=1

C .....
C ..... POINT EXACTLY ON BREAKPOINT
C      150  RFX=0.0
C           GO TO 240

C .....
C ..... BOUNDING BREAKPOINTS FOUND
C      160  RFX=(RVX-RDX(J))/(RDX(J+1)-RDX(J))
C           GO TO 240

C .....
C ..... CHECK BREAKPOINT JUST ABOVE STARTING POINT
C      170  IF(RVX-RDX(J+1)) 160,220,180

C .....
C ..... POINT ABOVE STARTING BOUND, CHECK MAXIMUM
C      180  IF(RVX-RDX(NX))190,200,210

C .....
C ..... POINT BELOW MAXIMUM, COUNT UP
C      190  IF(RVX-RDX(J+1)) 160,220,230

C .....
C ..... POINT ON OR ABOVE MAXIMUM
C      200  J=NX-1
C           GO TO 220

C .....
C ..... POINT ABOVE MAXIMUM
C      210  IER=0
C           GO TO 200

C .....
C ..... POINT FOUND
C      220  RFX=1.0
C           GO TO 240

C .....
C ..... COUNT UP

```

PAGE 2 C ... VBNR. . .

230 J=J+1
GO TO 190

C

C. ARGUMENT NORMALIZED, RETURN

240 RNX(1)=J
RNX(2)=RFX
RETURN
END

PROGRAM SIZE = 350

44

OF / DE

PAGE 1 C ... LOOK1...

C FUNCTION LOOK UP, 1-D REAL ARGUMENTS

C.....

SUBROUTINE LOOK1(RNX, RVY, RDY)

DIMENSION RDY(20), RNX(2)

J=RNX(1)

RFX=RNX(2)

RVY=RDY(J)+RFX*(RDY(J+1)-RDY(J))

RETURN

END

PROGRAM SIZE = 174

OCCOM5

8 15

15. 0000 111. 1100 5. 8800 1. 8000 17. 1000

1. 3940

8 33. 3300

1. 0000 . 7940 . 6370 . 0000

1. 0254 . 7940 . 6470 . 0110

1. 0754 . 7840 . 6570 . 0316

1. 1193 . 7250 . 6660 . 0486

1. 1535 . 6950 . 6760 . 0609

1. 1939 . 6540 . 6860 . 0749

1. 2509 . 5630 . 6960 . 0938

1. 2904 . 4770 . 7060 . 1058

8 55. 5500

1. 4939 1. 3700 . 6660 . 1804

1. 6026 1. 3490 . 6860 . 2079

1. 6588 1. 3320 . 6960 . 2209

1. 7193 1. 2990 . 7060 . 2344

1. 7930 1. 2620 . 7150 . 2509

1. 8553 1. 2070 . 7250 . 2633

1. 8825 1. 1460 . 7250 . 2700

1. 9140 1. 0780 . 7150 . 2817

8 66. 6600

1. 9596 1. 8250 . 6860 . 3053

2. 0825 1. 8070 . 7150 . 3222

2. 1991 1. 7780 . 7550 . 3305

2. 3140 1. 7440 . 7840 . 3413

2. 3789 1. 7140 . 7940 . 3496

2. 4263 1. 6410 . 7940 . 3586

2. 4561 1. 5600 . 7840 . 3688

2. 4596 1. 5000 . 7740 . 3742

8 77. 7700

2. 3956 2. 3660 . 6860 . 4083

2. 8974 2. 3540 . 7840 . 4474

3. 1474 2. 3350 . 8180 . 4679

3. 2693 2. 3200 . 8330 . 4774

3. 3798 2. 2970 . 8380 . 4903

3. 5325 2. 2530 . 8410 . 5096

3. 6070 2. 2090 . 8430 . 5184

3. 6535 2. 1180 . 8410 . 5259

8 88. 8800

3. 0079 3. 0120 . 6860 . 5323

3. 6009 3. 0060 . 7840 . 5566

4. 1105 3. 0040 . 8330 . 5896

4. 5570 2. 9890 . 8530 . 6275

4. 8105 2. 9570 . 8580 . 6514

5. 0342 2. 9090 . 8580 . 6749

5. 2289 2. 8300 . 8530 . 6988

5. 3500 2. 7320 . 8430 . 7194

8 94. 4400

3. 4009 3. 3800 . 6860 . 6025

4. 0877 3. 3750 . 7840 . 6234

4. 7044 3. 3720 . 8330 . 6592

5. 2211 3. 3640 . 8530 . 6980

5. 5553 3. 3340 . 8580 . 7268

5. 8781 3. 2720 . 8580 . 7573

6. 1412 3. 1850 . 8530 . 7858

6. 3307 3. 0910 . 8430 . 8122

8 100. 0000

3. 7596 3. 6610 . 6860 . 6618

4. 5246 3. 6540 . 7840 . 6787

5. 2421	3. 6490	. 8330	. 7169
5. 8175	3. 6480	. 8530	. 7561
6. 2000	3. 6300	. 8550	. 7892
6. 5553	3. 6010	. 8550	. 8203
6. 9403	3. 5510	. 8530	. 8547
7. 3895	3. 4690	. 8330	. 9123
8. 111. 1100			
4. 7395	4. 0540	. 6860	. 8052
5. 5763	4. 0540	. 7440	. 8405
6. 1518	4. 0540	. 7840	. 8560
6. 7307	4. 0540	. 7840	. 9109
7. 1632	4. 0540	. 7840	. 9497
7. 5623	4. 0540	. 7940	. 9717
8. 2930	4. 0500	. 7940	1. 0306
8. 8596	4. 0080	. 7940	1. 0738

SCALED FRACTION COMPRESSOR DATA

N/RTH= .29996 MIN PR= .06665 MAX PR= .08600

CORRECTED FLOW

TEMPERATURE RISE

.13501	.00000
.13501	.00497
.13446	.00977
.13376	.01453
.13159	.01917
.12686	.02362
.12253	.02802
.11942	.03217
.11603	.03622
.11246	.04019
.10754	.04407
.10190	.04791
.09628	.05176
.08881	.05527
.08112	.05878

N/RTH= .49994 MIN PR= .09958 MAX PR= .12759

CORRECTED FLOW

TEMPERATURE RISE

.23297	.10019
.23199	.10440
.23102	.10861
.23001	.11282
.22882	.11691
.22729	.12082
.22510	.12460
.22232	.12833
.21967	.13205
.21710	.13580
.21445	.13950
.20996	.14279
.20544	.14609
.19431	.15033
.18332	.15646

N/RTH= .59995 MIN PR= .13062 MAX PR= .16397

CORRECTED FLOW

TEMPERATURE RISE

.31036	.16959
.30948	.17233
.30856	.17505
.30768	.17780
.30646	.17978
.30493	.18118
.30344	.18259
.30185	.18411
.30005	.18600
.29825	.18790
.29636	.18982
.29355	.19232
.28906	.19516
.27969	.19894
.25510	.20789

N/RTH= .69992 MIN PR= .15970 MAX PR= .24356

CORRECTED FLOW

TEMPERATURE RISE

.40237	.22681
.40701	.23071

40164	23459
40128	23849
40091	24237
40054	24628
39984	25024
39868	25433
39752	25842
39590	26239
39368	26682
39044	27267
38602	27899
38000	28516
36020	29214

N/RTH= .79993

MIN PR= .20050

MAX PR= .35666

CORRECTED FLOW

TEMPERATURE RISE

51224	29568
51193	29950
51166	30331
51135	30713
51117	31195
51105	31796
51093	32397
51047	33075
50952	33862
50858	34653
50565	35507
50150	36404
49539	37381
48447	38504
46460	39963

N/RTH= .84995

MIN PR= .22672

MAX PR= .42203

CORRECTED FLOW

TEMPERATURE RISE

57483	33475
57455	33826
57431	34180
57404	34534
57385	35117
57367	35791
57349	36465
57303	37292
57248	38165
57114	39078
56793	40082
56216	41150
55453	42276
54276	43536
52567	45120

N/RTH= .89999

MIN PR= .25064

MAX PR= .49261

CORRECTED FLOW

TEMPERATURE RISE

62259	36765
62219	37082
62180	37402
62140	37744
62109	38510
62079	39276
62054	40103
62045	41088
62027	42081

61819
61520
61115
60544
59799
58997

43329
44586
45853
47137
48834
50684

N/RTH= .99997

MIN PR= .31595

MAX PR= .59064

CORRECTED FLOW

68945
68945
68945
68945
68945
68945
68945
68945
68933
68906
68878
68533
68161

TEMPERATURE RISE

44730
45419
46112
46762
47202
47867
49414
50946
52414
53448
54520
55838
57156
58408
59653

C PROGRAM TO PREPARE THE HP TURBINE DATA
C FOR STORAGE ON DISC

LOGICAL SENSW
INTEGER NAME(3), WDTGT, NP(4)
REAL RDN(4), RDPR(14), RDQ(14,4), RDET(14,4), RDDT(14,4)
SCALED FRACTION SDN(4), SDPR(14), SDQ(14,4), SDDT(14,4)
SCALED FRACTION SDAT(1), SVADC(2), SVDAC(2)
SCALED FRACTION SVPR, SVN, SVQ, SVDT

EQUIVALENCE (SDAT(1), NCM)
EQUIVALENCE (SVADC(1), SVPR)
EQUIVALENCE (SVADC(2), SVN)
EQUIVALENCE (SVDAC(1), SVQ)
EQUIVALENCE (SVDAC(2), SVDT)

COMMON/CBDAT/ NCM, NPC, SDN, SDPR, SDQ, SDDT

C I/O FORMATS
100 FORMAT(3A2)
110 FORMAT(20I4)
120 FORMAT(9F10.4)
130 FORMAT(I4, F10.4)

C CONSOLE INITIALIZATION
CALL QSHYIN(1ER, 680, 680)

C READ CARD DATA

C FILENAME

READ(6, 100) (NAME(I), I=1,3)

C IF (SENSW(1)) WRITE(6, 100) (NAME(I), I=1,3)

C INDICES

READ(6, 110) NCM, NPC

IF (SENSW(1)) WRITE(6, 110) NCM, NPC

C SCALE FACTORS

READ(6, 120) RSPR, RSN, RSQ, RSDT

IF (SENSW(1)) WRITE(6, 120) RSPR, RSN, RSQ, RSDT

C CONSTANTS

READ(6, 120) RCGAM, RCAT

IF (SENSW(1)) WRITE(6, 120) RCGAM, RCAT

C CALCULATE DEL

RCDEL = (RCGAM-1.0)/RCGAM

C MAP DATA

J = 1

140 READ(6, 130) NP(J), RDN(J)

IF (SENSW(1)) WRITE(6, 130) NP(J), RDN(J)

I=1

150 READ(6, 120) RDPR(I), RDQ(I, J), RDET(I, J)

```
C      CALCULATE TEMPERATURE RISE
      RDDT(I,J)=RDET(I,J)
      IF (SENSW(1)) WRITE(6,120) RDPR(I) , RDQ(I,J), RDET(I,J), RDDT(I,J)
      IF (I.GE.NP(J)) GO TO 160
      I=I+1
      GO TO 150
160    IF (J.GE.NCM) GO TO 170
      J=J+1
      GO TO 140
170    CONTINUE
C      SCALE DATA
      J = 1
190    SDN(J) = RDN(J)/RSN
      I = 1
200    SDPR(I) = RDPR(I)/RSPR
      SDQ(I,J) = RDQ(I,J)/(RCAT*RSQ)
      SDDT(I,J)= RDDT(I,J)/RSDT
      IF (I.GE.NPC) GO TO 210
      I = I+1
      GO TO 200
210    IF (J.GE.NCM) GO TO 220
      J = J+1
      GO TO 190
220    CONTINUE
C      NAME AND SIZE DATA FILE
      WDTOT = 2 + NPC + NCM + 2*(NPC*NCM)
      CALL FLNAME(NAME,WDTOT)
C      WRITE DATA ON DISC
      CALL DISCIO(16,NAME,SDAT,WDTOT)
C      READ DATA BACK FROM DISC
      CALL DISCIO(17,NAME,SDAT,WDTOT)
C      WRITE PREPARED DATA ON LINE PRINTER
      CALL TURBDA(SDPR,SDN,SDQ,SDDT,NCM,NPC)
C      SET UP FOR DRAWING MAPS
230    CALL QRBADS(SVADC,0,2,IER)
      CALL FNGN2(SVPR,SDPR,IR,NPC,SVN,SDN,JR,NCM,IER,SDQ,SVQ)
      CALL FNLK2(SDDT,SVDT)
      CALL QWBDAS(SVDAC,0,2,IER)
      CALL QSTDA
      GO TO 230
      END
```

PROGRAM SIZE = 2053

C PROGRAM TO PREPARE THE LP TURBINE DATA
C FOR STORAGE ON DISC

LOGICAL SENSW
INTEGER NAME(3), WDTOT, NP(5)
REAL RDN(5), RDPR(11), RDQ(11,5), RDET(11,5), RDDT(11,5)
SCALED FRACTION SDN(5), SDPR(11), SDQ(11,5), SDDT(11,5)
SCALED FRACTION SDAT(1), SVADC(2), SVDAC(2)
SCALED FRACTION SVPR, SVN, SVQ, SVDT

EQUIVALENCE (SDAT(1), NCM)
EQUIVALENCE (SVADC(1), SVPR)
EQUIVALENCE (SVADC(2), SVN)
EQUIVALENCE (SVDAC(1), SVQ)
EQUIVALENCE (SVDAC(2), SVDT)

COMMON/CBDAT/ NCM, NPC, SDN, SDPR, SDQ, SDDT

C I/O FORMATS
100 FORMAT(3A2)
110 FORMAT(20I4)
120 FORMAT(8F10.4)
130 FORMAT(I4, F10.4)

C CONSOLE INITIALIZATION
CALL QSHYIN(IER, 680, 680)
C READ CARD DATA
C FILENAME
READ(6, 100) (NAME(I), I=1, 3)
IF (SENSW(1)) WRITE(6, 100) (NAME(I), I=1, 3)
C INDICES
READ(6, 110) NCM, NPC
IF (SENSW(1)) WRITE(6, 110) NCM, NPC
C SCALE FACTORS
READ(6, 120) RSPR, RSN, RSQ, RSDT
IF (SENSW(1)) WRITE(6, 120) RSPR, RSN, RSQ, RSDT
C CONSTANTS
READ(6, 120) RCGAM, RCAT
IF (SENSW(1)) WRITE(6, 120) RCGAM, RCAT
C CALCULATE DEL
RCDEL = (RCGAM-1.0)/RCGAM
C MAP DATA
J = 1
140 READ(6, 130) NP(J), RDN(J)
IF (SENSW(1)) WRITE(6, 130) NP(J), RDN(J)
I=1
150 READ(6, 120) RDPR(I), RDQ(I, J), RDET(I, J)

```
C   CALCULATE TEMPERATURE RISE
      RDDT(I,J)=RDET(I,J)
      IF (SENSW(1)) WRITE(6,120) RDPR(I) , RDQ(I,J), RDET(I,J), RDDT(I,J)
      IF (I.GE.NP(J)) GO TO 160
      I=I+1
      GO TO 150
160   IF (J.GE.NCM) GO TO 170
      J=J+1
      GO TO 140
170   CONTINUE
C   SCALE DATA
      J = 1
190   SDN(J) = RDN(J)/RSN
      I = 1
200   SDPR(I) = RDPR(I)/RSPR
      SDQ(I,J) = RDQ(I,J)/(RCAT*RSQ)
      SDDT(I,J)= RDDT(I,J)/RSDT
      IF (I.GE.NPC) GO TO 210
      I = I+1
      GO TO 200
210   IF (J.GE.NCM) GO TO 220
      J = J+1
      GO TO 190
220   CONTINUE
C   NAME AND SIZE DATA FILE
      WDTOT = 2 + NPC + NCM + 2*(NPC*NCM)
      CALL FLNAME(NAME,WDTOT)
C   WRITE DATA ON DISC
      CALL DISCIO(16,NAME,SDAT,WDTOT)
C   READ DATA BACK FROM DISC
      CALL DISCIO(17,NAME,SDAT,WDTOT)
C   WRITE PREPARED DATA ON LINE PRINTER
      CALL TURBDA(SDPR,SDN,SDQ,SDDT,NCM,NPC)
C   SET UP FOR DRAWING MAPS
230   CALL QRBADS(SVADC,0,2,IER)
      CALL FNGN2(SVPR,SDPR,IR,NPC,SVN,SDN,JR,NCM,IER,SDQ,SVQ)
      CALL FNLK2(SDDT,SVDT)
      CALL QWBDA(SVDAC,0,2,IER)
      CALL QSTDA
      GO TO 230
      END
```

PROGRAM SIZE = 2042

C---PROGRAM TO SCALE AND STORE ON DISC THE DATA DEFINING
C THE OPERATION OF THE DYNAMOMETER

```
INTEGER WDTOT
INTEGER NAME(3)
SCALED FRACTION SVADC(2)
SCALED FRACTION SVDAC(2)
SCALED FRACTION SVN2
SCALED FRACTION SVIF
SCALED FRACTION SVGL
EQUIVALENCE (SVADC(1), SVIF)
EQUIVALENCE (SVADC(2), SVN2)
EQUIVALENCE (SVDAC(1), SVGL)
REAL RDN2(12), RDIF(11), RDGL(12, 11)
SCALED FRACTION SDN2(12), SDIF(11), SDGL(12, 11), SDX(155)
COMMON/CBDAT/NN2, NIF, WDTOT, SDN2, SDIF, SDGL
EQUIVALENCE (SDX(1), NN2)
DATA NAME(1), NAME(2), NAME(3)/2HRO, 2HLO, 2HAD/
```

C-----I/O FORMATS

```
100 FORMAT(20I3)
110 FORMAT(8F10.4)
120 FORMAT(1H1, 60HDYNAMOMETER TORQUE SPEED DATA-UNSCALED', '/')
130 FORMAT(1H1, 60HDYNAMOMETER TORQUE SPEED DATA-SCALED', '/')
140 FORMAT(6X, 60HFIELD VOLTS=', F8.3),
150 FORMAT(20X, 60HROTOR SPEED', 12X, 60HTORQUE')
160 FORMAT(10X, 2F20.3)
170 FORMAT(10X, 2(13X, S7))
175 FORMAT(6X, 60HFIELD VOLTS=', S7)
CALL QSHYIN(1ER, 680)
```

C-----READ CARD DATA

```
READ(6, 100) NN2, NIF
READ(6, 110) RSN2, RSIF, RSGL
READ(6, 110) (RDN2(I), I=1, NN2)
READ(6, 110) (RDIF(I), I=1, NIF)
DO 180 J=1, NIF
180 READ(6, 110) (RDGL(I, J), I=1, NN2)
```

C-----SCALE DATA

```
DO 190 J=1, NIF
SDIF(J)=RDIF(J)/RSIF
DO 190 I=1, NN2
SDN2(I)=RDN2(I)/RSN2
190 SDGL(I, J)=RDGL(I, J)/RSGL
```

C-----WRITE DATA ON DISC

```
WDTOT=NN2+NIF+2*(NN2*NIF)+3
CALL FLNAME(NAME, WDTOT)
```

```
      CALL DISCIO(16, NAME, SDX, WDTOT)

C-----READ DATA BACK FROM DISC
      CALL DISCID(17, NAME, SDX, WDTOT)

C-----WRITE DATA FOR VISUAL INSPECTION
C      REAL DATA
      WRITE(6, 120)
      DO 200 J=1, NIF
      WRITE(6, 140) RDIF(J)
      WRITE(6, 150)
      DO 200 I=1, NN2

200  WRITE(6, 160) RDN2(I), RDGL(I, J)
C-----SCALED DATA
      WRITE(6, 130)
      DO 210 J=1, NIF
      WRITE(6, 175) SDIF(J)
      WRITE(6, 150)
      DO 210 I=1, NN2
210  WRITE(6, 170) SDN2(I), SDGL(I, J)
C DO TEST LOOK UP
220  CALL QRBADS(SVADC, 0, 2, IER)
      CALL FNGN2(SVN2, SDN2, IA, NN2, SVIF, SDIF, IB, NIF, IER, SDGL, SVGL)
      CALL QWB DAS(SVDAC, 0, 1, IER)
      CALL QSTDA
      GO TO 220
      END
```

PROGRAM SIZE = 1556

Engine Constants, Scale Factors and Design Point Scaled Variables

F-7

***BUGOFF PAGE 1 ***

1. ENGINE REAL CONSTANTS

RDPAR		
1) =	10135E+06	EQUIVALENCE (RDPAR(1), RCPA)
2) =	28800E+03	EQUIVALENCE (RDPAR(2), RCTA)
3) =	10000E+03	EQUIVALENCE (RDPAR(3), RCGC)
4) =	10000E+01	EQUIVALENCE (RDPAR(4), RCJ)
5) =	28700E+03	EQUIVALENCE (RDPAR(5), RCR)
6) =	31416E+01	EQUIVALENCE (RDPAR(6), RCPI)
7) =	14000E+01	EQUIVALENCE (RDPAR(7), RCGAM1)
8) =	13942E+01	EQUIVALENCE (RDPAR(8), RCGAM2)
9) =	13847E+01	EQUIVALENCE (RDPAR(9), RCGAM3)
10) =	13360E+01	EQUIVALENCE (RDPAR(10), RCGAM4)
11) =	13470E+01	EQUIVALENCE (RDPAR(11), RCGAM5)
12) =	10040E+04	EQUIVALENCE (RDPAR(12), RCCP1)
13) =	10150E+04	EQUIVALENCE (RDPAR(13), RCCP2)
14) =	10330E+04	EQUIVALENCE (RDPAR(14), RCCP3)
15) =	11410E+04	EQUIVALENCE (RDPAR(15), RCCP4)
16) =	11140E+04	EQUIVALENCE (RDPAR(16), RCCP5)
17) =	20000E-01	EQUIVALENCE (RDPAR(17), RCVIN)
18) =	30000E-01	EQUIVALENCE (RDPAR(18), RCVCOM)
19) =	20000E-01	EQUIVALENCE (RDPAR(19), RCVROV)
20) =	40000E-01	EQUIVALENCE (RDPAR(20), RCVCC)
21) =	30000E-01	EQUIVALENCE (RDPAR(21), RCVTUR)
22) =	15161E-02	EQUIVALENCE (RDPAR(22), RCKIN)
23) =	95000E+00	EQUIVALENCE (RDPAR(23), RCKCC)
24) =	16314E+06	EQUIVALENCE (RDPAR(24), RCKEXH)
25) =	15613E+06	EQUIVALENCE (RDPAR(25), RCA)
26) =	47413E+06	EQUIVALENCE (RDPAR(26), RCB)
27) =	51335E+03	EQUIVALENCE (RDPAR(27), RCC)
28) =	47416E+02	EQUIVALENCE (RDPAR(28), RCD)
29) =	40857E+05	EQUIVALENCE (RDPAR(29), RCLVH)
30) =	10000E+00	EQUIVALENCE (RDPAR(30), RCIGG)
31) =	15000E+00	EQUIVALENCE (RDPAR(31), RCIGB)
32) =	60000E+00	EQUIVALENCE (RDPAR(32), RCILD)
33) =	40000E+01	EQUIVALENCE (RDPAR(33), RCK1)
34) =	93150E+00	EQUIVALENCE (RDPAR(34), RCK2)
35) =	75000E+00	EQUIVALENCE (RDPAR(35), RCBETA)
36) =	20000E-01	EQUIVALENCE (RDPAR(36), RCH)
37) =	96000E+00	EQUIVALENCE (RDPAR(37), RCEFF)
38) =	50000E+05	EQUIVALENCE (RDPAR(38), RCNGDP)
39) =	50000E+05	EQUIVALENCE (RDPAR(39), RCNPDF)
40) =	12000E+04	EQUIVALENCE (RDPAR(40), RCT5DP)
41) =	93000E+03	EQUIVALENCE (RDPAR(41), RCT6DP)
42) =	10000E+01	EQUIVALENCE (RDPAR(42), RCA5)
43) =	10000E+01	EQUIVALENCE (RDPAR(43), RCA6)

***BUGOFF PAGE 1 *** ... 2. REAL SCALE FACTORS

RDSF

1) =	15000E+06	EQUIVALENCE (RDSF(1), RSPA)
2) =	32000E+03	EQUIVALENCE (RDSF(2), RSTA)
3) =	10000E+01	EQUIVALENCE (RDSF(3), RSTHEA)
4) =	10000E+01	EQUIVALENCE (RDSF(4), RSELA)
5) =	60000E+01	EQUIVALENCE (RDSF(5), RSWO)
6) =	60000E+01	EQUIVALENCE (RDSF(6), RSQO)
7) =	15000E+06	EQUIVALENCE (RDSF(7), RSP1)
8) =	32000E+03	EQUIVALENCE (RDSF(8), RST1)
9) =	60000E+01	EQUIVALENCE (RDSF(9), RSW1)
10) =	10000E+01	EQUIVALENCE (RDSF(10), RSTHE1)
11) =	10000E+01	EQUIVALENCE (RDSF(11), RSELE1)
12) =	60000E+01	EQUIVALENCE (RDSF(12), RSQ1)
13) =	15000E+06	EQUIVALENCE (RDSF(13), RSP2)
14) =	32000E+03	EQUIVALENCE (RDSF(14), RST2)
15) =	60000E+01	EQUIVALENCE (RDSF(15), RSW2)
16) =	10000E+01	EQUIVALENCE (RDSF(16), RSTHE2)
17) =	10000E+01	EQUIVALENCE (RDSF(17), RSELE2)
18) =	60000E+01	EQUIVALENCE (RDSF(18), RSQ2)
19) =	20000E+01	EQUIVALENCE (RDSF(19), RSPR12)
20) =	65000E-01	EQUIVALENCE (RDSF(20), RSDT12)
21) =	10647E+05	EQUIVALENCE (RDSF(21), RSNC)
22) =	60818E+05	EQUIVALENCE (RDSF(22), RSNNG)
23) =	43835E+05	EQUIVALENCE (RDSF(23), RSNREL)
24) =	20000E+04	EQUIVALENCE (RDSF(24), RSGCOM)
25) =	15000E+07	EQUIVALENCE (RDSF(25), RSP3)
26) =	10000E+04	EQUIVALENCE (RDSF(26), RST3)
27) =	60000E+01	EQUIVALENCE (RDSF(27), RSW3)
28) =	15000E+02	EQUIVALENCE (RDSF(28), RSPR23)
29) =	18000E+01	EQUIVALENCE (RDSF(29), RSDT23)
30) =	15000E+07	EQUIVALENCE (RDSF(30), RSP4)
31) =	10000E+04	EQUIVALENCE (RDSF(31), RST4)
32) =	60000E+01	EQUIVALENCE (RDSF(32), RSW4)
33) =	10000E+00	EQUIVALENCE (RDSF(33), RSWF)
34) =	15000E+07	EQUIVALENCE (RDSF(34), RSP5)
35) =	15000E+04	EQUIVALENCE (RDSF(35), RST5)
36) =	60000E+01	EQUIVALENCE (RDSF(36), RSW5)
37) =	24628E-03	EQUIVALENCE (RDSF(37), RSQ5)
38) =	40000E+01	EQUIVALENCE (RDSF(38), RSPR56)
39) =	30000E+00	EQUIVALENCE (RDSF(39), RSDT56)
40) =	20000E+04	EQUIVALENCE (RDSF(40), RSGGGT)
41) =	75000E+06	EQUIVALENCE (RDSF(41), RSP6)
42) =	12000E+04	EQUIVALENCE (RDSF(42), RST6)
43) =	77858E-03	EQUIVALENCE (RDSF(43), RSQ6)
44) =	30000E+01	EQUIVALENCE (RDSF(44), RSPR67)
45) =	22000E+00	EQUIVALENCE (RDSF(45), RSDT67)
46) =	48000E+05	EQUIVALENCE (RDSF(46), RSNPT)
47) =	20000E+04	EQUIVALENCE (RDSF(47), RSGPT)
48) =	25000E+06	EQUIVALENCE (RDSF(48), RSP7)
49) =	11000E+04	EQUIVALENCE (RDSF(49), RST7)
50) =	20000E+04	EQUIVALENCE (RDSF(50), RSGOUT)
51) =	20000E+04	EQUIVALENCE (RDSF(51), RSGACC)
52) =	20000E+05	EQUIVALENCE (RDSF(52), RSNOUT)
53) =	50000E+00	EQUIVALENCE (RDSF(53), RSVOL)
54) =	50000E-01	EQUIVALENCE (RDSF(54), RSAREA)
55) =	50000E-01	EQUIVALENCE (RDSF(55), RSTIMP)
56) =	10000E+02	EQUIVALENCE (RDSF(56), RSTIMN)
57) =	10000E+01	EQUIVALENCE (RDSF(57), RSI)
58) =	35000E-02	EQUIVALENCE (RDSF(58), RSKIN)
59) =	10000E-02	EQUIVALENCE (RDSF(59), RSKEXH)

60) = . 20000E+04 EQUIVALENCE (RDSF(60),RSGC)
61) = . 10000E+02 EQUIVALENCE (RDSF(61),RSK)
62) = . 20000E+04 EQUIVALENCE (RDSF(62),RSGIGV)
63) = . 20000E+04 EQUIVALENCE (RDSF(63),RSGOGV)
64) = . 20000E+04 EQUIVALENCE (RDSF(64),RSGACL)
65) = . 10000E+05 EQUIVALENCE (RDSF(65),RSDNGG)
66) = . 10000E+05 EQUIVALENCE (RDSF(66),RSDNO)
67) = . 20000E+05 EQUIVALENCE (RDSF(67),RSNOMP)
68) = . 16400E+04 EQUIVALENCE (RDSF(68),RSGOMP)
69) = . 20000E+07 EQUIVALENCE (RDSF(69),RSPWR)
70) = . 60000E+01 EQUIVALENCE (RDSF(70),RSROW3)

F-73

SVADC

1) =	67548	EQUIVALENCE (SVADC(1) , SVPA)
2) =	90009	EQUIVALENCE (SVADC(2) , SVTA)
3) =	99994	EQUIVALENCE (SVADC(3) , SVTHEA)
4) =	99994	EQUIVALENCE (SVADC(4) , SVDELA)
5) =	66211	EQUIVALENCE (SVADC(5) , SVP1)
6) =	66211	EQUIVALENCE (SVADC(6) , SVP2)
7) =	41113	EQUIVALENCE (SVADC(7) , SVP3)
9) =	25732	EQUIVALENCE (SVADC(9) , SVP6)
10) =	76575	EQUIVALENCE (SVADC(10) , SVNGG)
11) =	86475	EQUIVALENCE (SVADC(11) , SVNC)
12) =	85144	EQUIVALENCE (SVADC(12) , SVNREL)
13) =	49438	EQUIVALENCE (SVADC(13) , SVNOUT)
14) =	82343	EQUIVALENCE (SVADC(14) , SVNPT)
15) =	62689	EQUIVALENCE (SVADC(15) , SVWF)
16) =	29687	EQUIVALENCE (SVADC(16) , SVIF)
17) =	99982	EQUIVALENCE (SVADC(17) , SVFLWF)

SVDAC

DEFINITION OF D-A CONVERTER ARRAY

1) =	60233	EQUIVALENCE (SVDAC(1) , SVW0)
2) =	60233	EQUIVALENCE (SVDAC(2) , SVW1)
3) =	60620	EQUIVALENCE (SVDAC(3) , SVW2)
4) =	60620	EQUIVALENCE (SVDAC(4) , SVW3)
5) =	61542	EQUIVALENCE (SVDAC(5) , SVW4)
6) =	61526	EQUIVALENCE (SVDAC(6) , SVW5)
7) =	90005	EQUIVALENCE (SVDAC(7) , SVT1)
8) =	90002	EQUIVALENCE (SVDAC(8) , SVT2)
9) =	51477	EQUIVALENCE (SVDAC(9) , SVT3)
10) =	51477	EQUIVALENCE (SVDAC(10) , SVT4)
11) =	76868	EQUIVALENCE (SVDAC(11) , SVT5)
12) =	75314	EQUIVALENCE (SVDAC(12) , SVT6)
13) =	72263	EQUIVALENCE (SVDAC(13) , SVT7)
14) =	00006	EQUIVALENCE (SVDAC(14) , SVDNGG)
15) =	00052	EQUIVALENCE (SVDAC(15) , SVDNO)
16) =	47940	EQUIVALENCE (SVDAC(16) , SVSFC)
17) =	45825	EQUIVALENCE (SVDAC(17) , SVPSHR)
18) =	90265	EQUIVALENCE (SVDAC(18) , SVT8)

SVVAR

DEFINITION OF ENGINE VARIABLE ARRAY

1) =	60233	EQUIVALENCE (SVVAR(1) , SVQ0)
2) =	36737	EQUIVALENCE (SVVAR(2) , SVTHE1)
3) =	48404	EQUIVALENCE (SVVAR(3) , SVDEL1)
4) =	60233	EQUIVALENCE (SVVAR(4) , SVQ1)
5) =	99988	EQUIVALENCE (SVVAR(5) , SVTHE2)
6) =	98016	EQUIVALENCE (SVVAR(6) , SVDEL2)
7) =	61847	EQUIVALENCE (SVVAR(7) , SVQ2)
8) =	59305	EQUIVALENCE (SVVAR(8) , SVPR12)
9) =	00000	EQUIVALENCE (SVVAR(9) , SVDT12)
10) =	10709	EQUIVALENCE (SVVAR(10) , SVGCOM)
11) =	41394	EQUIVALENCE (SVVAR(11) , SVPR23)
12) =	43756	EQUIVALENCE (SVVAR(12) , SVDT23)
13) =	41113	EQUIVALENCE (SVVAR(13) , SVP4)
14) =	00488	EQUIVALENCE (SVVAR(14) , SVT34)
15) =	39056	EQUIVALENCE (SVVAR(15) , SVP5)
16) =	00018	EQUIVALENCE (SVVAR(16) , SVRTT5)
17) =	86905	EQUIVALENCE (SVVAR(17) , SVQ5)
18) =	75888	EQUIVALENCE (SVVAR(18) , SVPR56)
19) =	72073	EQUIVALENCE (SVVAR(19) , SVDT56)
20) =	10767	EQUIVALENCE (SVVAR(20) , SVGGGT)
21) =	00000	EQUIVALENCE (SVVAR(21) , SVRTT6)
22) =	73862	EQUIVALENCE (SVVAR(22) , SVQ6)
23) =	60245	EQUIVALENCE (SVVAR(23) , SVPR67)

24) =	54770	EQUIVALENCE (SVVAR(24), SVDT67)
25) =	05405	EQUIVALENCE (SVVAR(25), SVGPT)
26) =	42706	EQUIVALENCE (SVVAR(26), SVP7)
27) =	31573	EQUIVALENCE (SVVAR(27), SVGOUT)
28) =	00006	EQUIVALENCE (SVVAR(28), SVGACC)
29) =	00000	EQUIVALENCE (SVVAR(29), SVPWR1)
30) =	41873	EQUIVALENCE (SVVAR(30), SVPWR2)
31) =	00000	EQUIVALENCE (SVVAR(31), SVPWR3)
32) =	52530	EQUIVALENCE (SVVAR(32), SVPWR4)
33) =	22397	EQUIVALENCE (SVVAR(33), SVPWR5)
34) =	92325	EQUIVALENCE (SVVAR(34), SVROW3)
35) =	00119	EQUIVALENCE (SVVAR(35), SVT4S)
36) =	10706	EQUIVALENCE (SVVAR(36), SVGCAS)
37) =	00021	EQUIVALENCE (SVVAR(37), SVGACL)
38) =	23306	EQUIVALENCE (SVVAR(38), SVNMPF)
39) =	85135	EQUIVALENCE (SVVAR(39), SVNMPG)
40) =	79886	EQUIVALENCE (SVVAR(40), SVNMPB)
41) =	82257	EQUIVALENCE (SVVAR(41), SVNMPD)
42) =	49435	EQUIVALENCE (SVVAR(42), SVNMPD)
43) =	25894	EQUIVALENCE (SVVAR(43), SVGOMP)
44) =	00000	EQUIVALENCE (SVVAR(44), SVGIGV)
45) =	32687	EQUIVALENCE (SVVAR(45), SVPWR)
46) =	42239	EQUIVALENCE (SVVAR(46), SVNEW)
47) =	42203	EQUIVALENCE (SVVAR(47), SVNEW1)
48) =	89130	EQUIVALENCE (SVVAR(48), SVEFF)
49) =	89130	EQUIVALENCE (SVVAR(49), SVEFF1)
50) =	70316	EQUIVALENCE (SVVAR(50), SVNWI)
51) =	85104	EQUIVALENCE (SVVAR(51), SVEFFP)
61) =	58173	EQUIVALENCE (SVVAR(61), SVE)

SCPAR	SCALED ENGINE PARAMETERS	
1) =	00000	EQUIVALENCE (SCPAR(1), SCUNIT)
2) =	99997	EQUIVALENCE (SCPAR(2), SCINL1)
3) =	99997	EQUIVALENCE (SCPAR(3), SCINL2)
4) =	23361	EQUIVALENCE (SCPAR(4), SCINL3)
5) =	00000	EQUIVALENCE (SCPAR(5), SCINL4)
6) =	00000	EQUIVALENCE (SCPAR(6), SCINL5)
7) =	99997	EQUIVALENCE (SCPAR(7), SCFAN1)
8) =	99997	EQUIVALENCE (SCPAR(8), SCFAN2)
9) =	50000	EQUIVALENCE (SCPAR(9), SCFAN3)
10) =	99997	EQUIVALENCE (SCPAR(10), SCFAN4)
11) =	99997	EQUIVALENCE (SCPAR(11), SCFAN5)
12) =	06497	EQUIVALENCE (SCPAR(12), SCFAN6)
13) =	99997	EQUIVALENCE (SCPAR(13), SCCOM1)
14) =	99997	EQUIVALENCE (SCPAR(14), SCCOM2)
15) =	66666	EQUIVALENCE (SCPAR(15), SCCOM3)
16) =	99997	EQUIVALENCE (SCPAR(16), SCCOM4)
17) =	99997	EQUIVALENCE (SCPAR(17), SCCOM5)
18) =	57599	EQUIVALENCE (SCPAR(18), SCCOM6)
19) =	77631	EQUIVALENCE (SCPAR(19), SCOGV1)
20) =	95999	EQUIVALENCE (SCPAR(20), SCOGV2)
21) =	27780	EQUIVALENCE (SCPAR(21), SCOGV3)
22) =	87106	EQUIVALENCE (SCPAR(22), SCOGV4)
23) =	02478	EQUIVALENCE (SCPAR(23), SCOGV5)

24) = .00308	EQUIVALENCE (SCPAR(24), SCOGV6)
25) = .94998	EQUIVALENCE (SCPAR(25), SCCC1)
26) = .66666	EQUIVALENCE (SCPAR(26), SCCC2)
27) = .00253	EQUIVALENCE (SCPAR(27), SCCC3)
28) = .46417	EQUIVALENCE (SCPAR(28), SCCC4)
29) = -.03018	EQUIVALENCE (SCPAR(29), SCCC5)
30) = -.04645	EQUIVALENCE (SCPAR(30), SCCC6)
31) = .50000	EQUIVALENCE (SCPAR(31), SCGGT1)
32) = .91467	EQUIVALENCE (SCPAR(32), SCGGT2)
33) = .62903	EQUIVALENCE (SCPAR(33), SCGGT3)
34) = .00220	EQUIVALENCE (SCPAR(34), SCGGT4)
35) = .29999	EQUIVALENCE (SCPAR(35), SCGGT5)
36) = .37497	EQUIVALENCE (SCPAR(36), SCGGT6)
37) = .99997	EQUIVALENCE (SCPAR(37), SCPT1)
38) = .86691	EQUIVALENCE (SCPAR(38), SCPT2)
39) = .35593	EQUIVALENCE (SCPAR(39), SCPT3)
40) = .23999	EQUIVALENCE (SCPAR(40), SCPT4)
41) = .24933	EQUIVALENCE (SCPAR(41), SCPT5)
42) = .19891	EQUIVALENCE (SCPAR(42), SCPT6)
43) = .39999	EQUIVALENCE (SCPAR(43), SCGB1)
44) = .09314	EQUIVALENCE (SCPAR(44), SCGB2)
45) = .99997	EQUIVALENCE (SCPAR(45), SCGB3)
46) = .00049	EQUIVALENCE (SCPAR(46), SCGB4)
47) = .47745	EQUIVALENCE (SCPAR(47), SCGB5)
48) = .99997	EQUIVALENCE (SCPAR(48), SCGB6)
49) = .99997	EQUIVALENCE (SCPAR(49), SCFAN0)
50) = .99997	EQUIVALENCE (SCPAR(50), SCFAN7)
51) = .99997	EQUIVALENCE (SCPAR(51), SCFAN8)
52) = .89688	EQUIVALENCE (SCPAR(52), SCFAN9)
53) = .99988	EQUIVALENCE (SCPAR(53), SCCOM0)
54) = .31998	EQUIVALENCE (SCPAR(54), SCCOM7)
55) = .32840	EQUIVALENCE (SCPAR(55), SCCOM8)
56) = .21783	EQUIVALENCE (SCPAR(56), SCCOM9)
57) = .02280	EQUIVALENCE (SCPAR(57), SCCOM6)
58) = .00098	EQUIVALENCE (SCPAR(58), SCOGV0)
59) = .00098	EQUIVALENCE (SCPAR(59), SCOGV7)
60) = .99997	EQUIVALENCE (SCPAR(60), SCOGV8)
61) = .76837	EQUIVALENCE (SCPAR(61), SCOGV9)
62) = .79999	EQUIVALENCE (SCPAR(62), SCGGT0)
63) = .19473	EQUIVALENCE (SCPAR(63), SCGGT7)
64) = .15701	EQUIVALENCE (SCPAR(64), SCGGT8)
65) = .99997	EQUIVALENCE (SCPAR(65), SCGGT9)
66) = .01642	EQUIVALENCE (SCPAR(66), SCGGT6)
67) = .91666	EQUIVALENCE (SCPAR(67), SCPT0)
68) = .21997	EQUIVALENCE (SCPAR(68), SCPT7)
69) = -.47482	EQUIVALENCE (SCPAR(69), SCPT8)
70) = -.69171	EQUIVALENCE (SCPAR(70), SCPT9)
71) = .09390	EQUIVALENCE (SCPAR(71), SCGB7)
72) = .09998	EQUIVALENCE (SCPAR(72), SCGB8)
73) = .39267	EQUIVALENCE (SCPAR(73), SCGB9)
74) = .59998	EQUIVALENCE (SCPAR(74), SCEXH1)
75) = .09888	EQUIVALENCE (SCPAR(75), SCEXH2)
76) = .99997	EQUIVALENCE (SCPAR(76), SCLD1)
77) = .81998	EQUIVALENCE (SCPAR(77), SCLD2)
78) = .99997	EQUIVALENCE (SCPAR(78), SCLD3)
79) = -.21997	EQUIVALENCE (SCPAR(79), SCLD4)
80) = .41998	EQUIVALENCE (SCPAR(80), SCNEW0)
81) = .89297	EQUIVALENCE (SCPAR(81), SCEFFD)
82) = .77188	EQUIVALENCE (SCPAR(82), SCPRDG)
83) = .79999	EQUIVALENCE (SCPAR(83), SCNGGD)
84) = .25018	EQUIVALENCE (SCPAR(84), SCGGTP)
85) = .49570	EQUIVALENCE (SCPAR(85), SCGGT1)

86)	=	75003	EQUIVALENCE (SCPAR(86), SCNWID)
87)	=	56613	EQUIVALENCE (SCPAR(87), SCPRDP)
88)	=	83328	EQUIVALENCE (SCPAR(88), SCNPTD)
89)	=	25793	EQUIVALENCE (SCPAR(89), SCPTP)
90)	=	35733	EQUIVALENCE (SCPAR(90), SCPTD)
91)	=	89999	EQUIVALENCE (SCPAR(91), SCHEX1)
92)	=	90909	EQUIVALENCE (SCPAR(92), SCHEX2)
93)	=	97998	EQUIVALENCE (SCPAR(93), SCHEX3)
94)	=	96997	EQUIVALENCE (SCPAR(94), SCHEX4)
95)	=	96997	EQUIVALENCE (SCPAR(95), SCHEX5)
96)	=	51575	EQUIVALENCE (SCPAR(96), SCHEX6)
97)	=	85599	EQUIVALENCE (SCPAR(97), SCEXH3)
98)	=	70709	EQUIVALENCE (SCPAR(98), SCEXH4)
99)	=	35968	EQUIVALENCE (SCPAR(99), SCEXH5)
100)	=	17505	EQUIVALENCE (SCPAR(100), SCGB0)

F-77

END

0	2	H	0	9	H	8	2
---	---	---	---	---	---	---	---

FIN