THE USE OF VARIABLE ENGINE GEOMETRY TO IMPROVE THE TRANSIENT PERFORMANCE OF A TWO-SPOOL TURBOFAN ENGINE

By

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SUMMARY

The prediction of the behaviour of a two-spool turbofan engine (Tay Mk610) under steady-running and during transient performances has been carried out in the present investigation. Four different variable geometry features have been investigated with a view to achieving surge-free acceleration and deceleration of the engine. These are: altering the scheduling of the Inlet Guide Vanes (IGVs) of the High Pressure (H.P.) compressor, altering the Handling Bleed Valve schedule in the H.P. compressor, bleeding some air from the I.P. compressor delivery into the Bypass Duct of the engine, and finally altering the throat area of the Final Nozzle of the engine. The method of the Inter Component Volumes has been adopted for the performance prediction. The simulation prediction, for the 'Design' engine, has been based upon the real characteristics of the engine components and engine data provided by the manufacturer, thus the simulation prediction is realistic. These characteristics used for this 'Design' engine are the characteristics used by the manufacturer in the original design of the engine - thus the H.P. compressor IGVs and Handling Bleed Valve move in accordance with the 'Design' schedules. With regard to the effects on steady-running and transient performances of heat transfer, for simplicity these have been ignored, i.e. the systems are regarded as being adiabatic.

The predicted transient trajectories, for reasonable thrust response rates, are satisfactory in all compressors with the comments that (a) in the acceleration there is a condition in the H.P. compressor near to surge at $\left(N_H/\sqrt{T_{26}}\right)$ of about 548, (b) in the deceleration there is an approach towards surge in the I.P. compressor over the band $\left(N_L/\sqrt{T_{24}}\right)$ 433 to 340.

Characteristics of the H.P. compressor (of 12 stages) are required for the situations with modified IGV and Handling Bleed schedules. Little information on this was available from the manufacturer. An existing programme, of the row by row type, for predicting axial flow compressor characteristics has been developed and used, along with the sparse engine manufacturer's data, to predict the changes in characteristics resulting from altered IGV and Handling Bleed schedules. These altered IGV and Handling Bleed schedules were introduced separately and then together. The effect of extending the IGV turning range so as to start at a lower non-dimensional speed $N_H/\sqrt{T_1}$, while maintaining the 'Design' $N_H/\sqrt{T_1}$ for completion of the opening, is predicted to be generally beneficial - higher mass flow rates, higher pressure ratios and slightly improved compressor efficiencies (about 0.3 per cent). This is almost equivalent to the compressor running at a higher rotational speed. The surge lines for the two cases are almost identical - there is a very slight rise but so small as almost to be ignored. However significant improvements in the position of the surge line, at a given inlet mass flow, or at a given $N_H/\sqrt{T_1}$, are predicted if the schedule for the Bleed Valve closing/opening is extended to lower compressor $N_H/\sqrt{T_1}$.

The changes in the characteristics of the H.P. compressor predicted by the above methods have been fitted to the characteristics of the 'Design' engine. These revised characteristics have then been used in the engine programme, introduced previously, for prediction of engine transient performance. In the first study, the scheduling alterations considered were

1. IGVs $\left(N_H/\sqrt{T_{26}}\right)$ range 493 - 593 compared with 'Design' 552 - 586
2. HP Handling Bleed Valve $\left(N_H/\sqrt{T_{26}}\right)$ range 493 - 520 compared with 'Design' 549 - 568

The engine operating with the altered schedules quoted above is referred to as the 'Production' engine. The alterations were implemented singly and together. Tests were run initially using the same acceleration and deceleration fuel schedules as for the 'Design' engine. However thrust responses were now changed, so to provide fairer comparisons, the fuel schedules were adjusted to give 'Equivalent' rates of thrust response.

In the acceleration, the alteration to the IGV schedule alone slightly eases the approach to surge in the H.P. compressor at, or immediately before, the start of the turning of the IGVs. However, the more forceful fuel schedule which is required leads to surge at $\left(N_H/\sqrt{T_{26}}\right)$ of 570. When only the Handling Bleed schedule is altered, a surge is encountered at $\left(N_H/\sqrt{T_{26}}\right)$ of 525 - at the end of the Handling Bleed Valve closure but before the IGVs begin to open. When both IGV and Handling Bleed schedules are changed, a very satisfactory H.P. compressor acceleration trajectory is obtained, free from surge tendency throughout. In the deceleration, moving the IGV schedule alone significantly eases the trajectory in the I.P. compressor. The improvement is reduced if the Handling Bleed schedule is also moved. The surge margin usage now being about 50 per cent, as compared with 60 per cent for the 'Design' engine.

The 'Production' engines studied above differed from the 'Design' engine by having the IGV schedule moved to a lower starting $\left(N_H/\sqrt{T_{26}}\right)$ and less steep rate of change and the Handling Bleed Valve closing/opening range also moved to a lower $\left(N_H/\sqrt{T_{26}}\right)$. To explore these effects more widely an investigation was then made for situations where the IGV schedule was again made less steep but moved towards higher $\left(N_H/\sqrt{T_{26}}\right)$ values and the Handling Bleed Valve range was also moved towards higher $\left(N_H/\sqrt{T_{26}}\right)$. The ranges of the schedules selected for this study were - for IGV turning, $\left(N_H/\sqrt{T_{26}}\right)$ from 550 to 620 and Handling Bleed Valve mid-opening at $\left(N_H/\sqrt{T_{26}}\right)$ of 573. The results have been presented.

It was found that using the 'Design' fuel schedules gave, for all of the above changes, both individual and combined, very similar thrust responses, both in acceleration and deceleration. So no adjustments were required for the transient fuel schedules. Regarding the transient trajectories in the compressors, in no case is the trajectory improved in the H.P. compressor during acceleration and in some cases worsened. This is attributed to moving the IGV and Handling Bleed Valve schedules to too high a speed range. In the deceleration, when only the Handling Bleed schedule is moved, there is a small easement of the trajectory in the I.P compressor. Significant improvement in this trajectory only happens when the IGV schedule is altered. This is attributed to the less steep schedule for turning the IGVs.

The conclusion from the studies of altering the IGV schedule and HP Handling Bleed schedule is (a) that it is best always to use the IGV schedule that starts at low speed and opens gradually, (b) for the acceleration, the HP Handling Bleed closure should occur at the lower speed range, but for the deceleration, the Handling Bleed Valve should open at the higher speed range. The above scheme is difficult to achieve with a hydro-mechanical control system, but can be much more easily achieved with an electronic controller.

The third change in engine geometry that has been considered has been the introduction of an air bleed from the I.P. compressor delivery into the Bypass Duct of the engine. The working lines in the I.P. compressor move favourably away from surge and the I.P. compressor has an adequate surge margin, hence presents no danger of surging particularly in the deceleration. An increase of about 60 per cent of surge margin of the I.P. compressor is obtained. The tendency of the I.P. compressor to surge in a deceleration is then relieved. Another advantage of the air bleed is a very fast deceleration of the engine, but it delays the thrust response in an acceleration.

The fourth variable geometry feature studied has been altering the throat area of the Final Nozzle. Closing the Final Nozzle of the engine improves the trajectories in the I.P. compressor both in rapid acceleration and deceleration. However, the surge margin of the Fan is deteriorated, due to the raising of the steady-running line in the fan. The reverse occurs when having the Final Nozzle open. Also, closing the Final Nozzle leads to a fast deceleration of the engine, but it slows the acceleration. The reverse effects occurs if the Final Nozzle throat area in increased. To obtain good transient performances of the engine, it is best to open the Final Nozzle in an acceleration and to close the Final Nozzle in a deceleration. For a two-spool turbofan engine, only the working lines in the Fan and in the I.P. compressor are affected. However, the working line in the H.P. compressor is not affected as long as both the LP and HP turbines are choked.

All tests for the prediction of the steady-running and transient performances of the engine have been performed at sea level and flight Mach number of 0.2. It has been shown elsewhere that the effects of changes of geometry, e.g. IGV schedule, which are observed at this flight condition, are also observed at other flight conditions.

Finally, the thesis ends with suggestions for further research.

UNITS

In the present investigation, the System International (S.I) units have generally been used. However the following exceptions have been made:

1. net thrust - lbf
2. mass flow rate - lbm/s
3. inlet non-dimensional mass flow rate in components (e.g. compressors) - $lbm \cdot K^{\frac{1}{2}} \cdot in^2 / s \cdot lbf$

The reason for this is the continuing use of Imperial units in aircraft engine industry, and the provision by the engine manufacturer of characteristics of components in these units.

STATION NUMBERING OF A TWO-SPOOL TURBOFAN ENGINE

The station numbering used for the Tay engine is shown in figure 5.3.

1	entry to the L.P. compressor (Fan)
13	Bypass Duct
24	entry to the I.P. compressor
26	inter-component-volume between the I.P. and H.P. compressors
3	inter-component-volume between the H.P. compressor and the burner
4	entry to the HP turbine
5	inter-component-volume between the HP and LP turbines
6	inter-component-volume between the LP turbine and the mixer
7	inter-component-volume between the mixer and the Final Nozzle
8	Final Nozzle exit plane

LIST OF FIGURES

NOMENCLATURE

NOTATION

a : distance of the point of maximum camber

c : blade chord

C_p : specific heat at constant pressure

h : enthalpy

i : incidence

i^* : nominal incidence

m : mass of gas in inter component volume

$\dot{m}$: mass flow rate

N : rotational speed of rotor/number of stages

P : pressure

P_a : atmospheric pressure

PR : pressure ratio

R : constant specific heat of gas

s : pitch

s : second

T : temperature

t : time

T_a : ambient temperature

U : blade tip velocity

V : volume

v : specific volume

W : work transfer

ΔP : static pressure rise

SUBSCRIPTS

C : compressor

ENG : engine

H : HP shaft

i : intake

in	: initial
Is	: isentropic
j	: exit plane
OF	: outer fan
P	: polytropic
S	: surge
t	: time
V	: volume
W	: working line
0	: stagnation
1	: inlet
2	: outlet
3	: turbine inlet
4	: turbine outlet

GREEK SYMBOLS

α	: air angle
β	: blade angle
γ	: specific heat ratio
η	: efficiency
δ	: deviation
δ^*	: boundary layer displacement thickness
ε	: deflection
ε^*	: nominal deflection
λ	: work done factor
ρ	: density
ν	: kinematic viscosity
σ	: cascade solidity
ω	: mean blade head pressure loss
Δ	: variation

ABBREVIATIONS

FCSP : factor of split
GEOM : fraction of total engine frontal area
HP : high pressure
IP : intermediate compressor
ICV : inter-component-volume
IGV : inlet guide vane
LCV : calorific value of fuel
rpm : revolution per minute
VTOL : vertical take off and landing

Chapter 1

Introduction

1.1 Introduction

In the past several years, major advances have been made to improve overall performance of aircraft and power generation gas turbine engines. These advances have been made possible largely by the increased reliability of calculated and experimental engine performance information and are very much the result of progress in analytical approaches, especially computer codes and math models, and by the improvement in the understanding of aerodynamics of turbomachines. While much progress has been made with respect to gas turbine engine performance evaluation, there are still many problems confronting the engineer. For example, improvement are still needed to make computational methods more usuable, reliable, and cost effective in calculating overall engine and engine component performance. Continued development of analytical and experimental methods for the evaluation of complex engine systems having variable cycles and geometries is also required.

Although there has been continuous improvement of the efficiency and performance of aircraft and power generation gas turbine engines, the transient behaviour of these engines remains one of the major problems facing the designer of these engines. The transient operation of a gas turbine engine is encountered when the inputs of the engine are altered, and the operation of the engine changes from one steady state condition to another. Rapid accelerations and decelerations are the most severe cases of transient operation. In certain applications the transient response of gas turbines following a demand for a change in output can be critical and aircraft engines are an obvious example of an application where the transient behaviour is critical. The prime requirement for civil aircraft is for rapid thrust response to cope with a balked landing when the aircraft is close to touchdown but it is forced to overshoot the runway. Lifting engines for VTOL aircraft provide another example, where an engine failure on a multi-engined installation may result in serious unbalanced forces which have to be adjusted rapidly.

Gas turbines are extensively used for the propulsion of aircraft, it is essential that these engines can be accelerated from idling speed to maximum speed, and vice versa, in as short times as possible. Cohen et al [5] pointed out that the response rate of the gas turbine itself from idle to maximum power must be less than ten seconds to allow time for the starting and synchronisation sequences. The behaviour of the engine during rapid accelerations and decelerations is of considerable interest, however it is important that these transients can be carried out rapidly without endangering the engine. The sensitivity and reaction of the engine during acceleration and deceleration operations can have significant implications on the transient behaviour of the engine. These include surge, stall, overspeeding, overheating of the components, etc. Since a surge phenomenon is physically destructive to the compressor and hence possibly the entire engine, this should be avoided at all costs.

The transient response of an aircraft gas turbine engine is difficult to investigate experimentally. Test of this nature cannot be carried out until late in the development program when the mechanical integrity of the design has been established by a considerable amount of engine running. However, accurate information on the behaviour of the engine during transient operations is needed at an early stage for an optimal design of the engine control system and also to ensure that customer requirements for rapid response can be satisfied.

Usually, aircraft gas turbine engine performance is presented for steady state running. This may be misleading and often it is far more important to know the behaviour within the components of the engine during transient operations. The most critical operations of aircraft gas turbine engine are clearly transient and its influence on the safety and the reliability, and, thus the operation of the engine cannot be underestimated. A good understanding of these transient mechanisms in the components is essential and neglecting transient effects cannot be justified. Therefore, accurate methods for predicting the transient behaviour of aircraft gas turbine engines are necessary to represent the actual engine transient performance characteristics. These methods could undoubtedly give an insight into the response of the characteristics of the engine and its components when given acceleration and deceleration fuel schedules are applied. Furthermore, the transient prediction methods would permit control schedules, or even totally new concepts of control, to be investigated without harming the engine. This should result in significant savings as a result of reduced engine test bed running and a reduction in the overall development time and cost.

1.2 Engine development implications

The tendency to improve the overall performance of gas turbine engines in terms of specific fuel consumption has increased sharply over recent years, and greater efforts are being made than ever before in this direction. Advances in specific fuel consumption for aero gas turbine engines have been achieved through gains in thermal efficiency and propulsive efficiency. In the latter, the specific fuel consumption has been improved by increasing bypass ratios. While further advances in fuel economy from the propulsive efficiency source are still to be expected, greater emphasis is now being placed on higher thermal efficiency in the race for the ultimate in good fuel consumption of these engines. The search for higher cycle thermal efficiency is leading towards increasing both the flame temperature and overall pressure ratio of the cycle. While high temperatures are thermodynamically desirable they mean the use of expensive alloys and cooled turbine blades leading to an increase in complexity and cost, or to acceptance of a decrease in engine life. Therefore, the thermodynamic gains of increased overall pressure ratio must be considered. Nowadays, aero gas turbine engines are operating at very high pressure ratios, and a typical civil engine of today may operate at cruise at a pressure ratio of over 30:1.

Although high performance of modern aero gas turbine engines has been obtained through the increasing of the overall pressure ratio of the engine cycle, the design of these engines is becoming very complex, and the stage matching characteristics of the engine compressor is remaining a very difficult task to obtain. This is due to the increasing number of stages in the compressor to achieve the now higher overall pressure ratio. Furthermore, the transient operations of these engines are difficult. The engine may encounter surge or stall difficulties when the engine starts to accelerate from an idle to maximum speed, or decelerate at high speeds. This means the transient operating line of the multistage high pressure axial flow compressor crosses the surge line during these off-design performance conditions. It is true to say that the more the pressure ratio of a compressor is increased the more difficult it becomes to ensure that it will operate efficiently over the full speed range. This is because the requirement for the ratio of inlet area to exit area, at the high speed case, results in an inlet area that becomes progressively too large relative to the exit area as the compressor speed and hence pressure ratio is reduced. The axial velocity of the inlet air in the front stages thus becomes low relative to the blade speed, this changes the incidence of the air onto the blades and a condition is reached where the flow separates and the compressor flow breaks down. This condition is referred to as stall of that compressor blade row.

If many, or all, of the blade rows enter partial or complete stall the compressor surges. This is a global oscillation of the mass flow through the compressor. Often complete reversal of the flow occurs, which in an engine means that burning gases from the combustion chamber are expelled through the front of the engine; this is sometimes called deep surge. Surge cycles generally have a frequency of the order 3 to 10 Hertz, and the large amplitude pressure and mass flow oscillations can cause physical damage to the engine. Also, combustor flame out may occur, causing loss of engine speed and loss of control. Subsequent relights may be hard due to an excess of unburnt fuel accumulation in the combustion chamber.

Referring again to compressor stall, when an engine encounters this a circumferentially distorted flow pattern is set up, with one or more regions of very low flow rotating around the compressor annulus at a speed of 30 to 70 per cent of rotor speed. The average flow through the compressor is nearly constant but at a much lower value than in unstalled operation. This condition is called 'rotating' stall. This presents a problem to the engine in that the engine can overheat extremely rapidly. This is caused by the fuel flow being excessive for the reduced air mass flow and thus a very high turbine entry temperature results which can quickly burn out the turbine nozzles. It is also more difficult to restore a compressor to stable operation from a rotating stall condition than from surge. Very rapid pilot action is required to shut down the engine before the turbine is seriously damaged. The stall and surge limit of the multistage H.P. compressor present a major problem with regard to design and off-design performance of gas turbine engines.

Stall and surge phenomena are internal aerodynamic instabilities and have two disadvantages; first the performance of the multistage H.P. compressor falls drastically, this performance degradation and resulting loss of thrust can be catastrophic. Second, flow instability may be a dangerous aerodynamic excitation resulting in blade vibration. Thus stall and surge are flow instabilities that not only affect the H.P. compressor performance and the performance of the engine but also limit the operating range of the H.P. compressor. Both phenomena have damaging consequences, therefore they must be avoided at all costs. During transient operations, a rapid acceleration, for example, pushes the compressor transient operating point towards the surge line, and the surge margin, distance between the equilibrium operating line and the surge line, is largely used. Large values of this margin will permit large increases in fuel flow and, therefore, rapid engine acceleration. On the other hand, if the surge margin is small, large increases in fuel flow will result in compressor stall or surge, and the engine acceleration will diminish, or be interrupted. In fact, in some cases the compressor performance may deteriorate so much that the engine will decelerate. However, the surge margin of the compressor is very crucial and some remedial actions must be taken to increase this margin in order to obtain satisfactory operations of the compressor and thus the engine during the transients.

The surge margin of a multistage H.P. compressor can be increased and a better transient performance of aero gas turbine engine might be achieved if the geometry of the engine is varied, such as adjustable H.P. compressor inlet guide vanes, compressor bleed and adjustable exhaust nozzles. To demonstrate the effects of variable geometry on the performance of gas turbine engines, these means have been investigated in the present study with the view to achieving surge free acceleration and deceleration of a twin spool turbofan engine.

1.3 Thesis layout

The scope of the present research is to predict the transient behaviour of a two-spool turbofan engine and how better transient performances of the engine might be achieved when varying the geometry of the engine. This technique could be very helpful to improve both steady-running performance and transient responses of the engine. The benefits to the engine will be improved acceleration and deceleration responses, thrust response, engine efficiency and surge free acceleration and deceleration of the engine.

In this thesis therefore there follows, in Chapter 2, a description of the variable engine geometry features that may be used. In Sections 2.3 and 2.4 there is a description of the engine in which these variable geometry features have been investigated in the present study.

Prediction procedures for the transient behaviour of gas turbine engines are reviewed in Chapter 3. As these prediction procedures require to use characteristics of compressors, a method of predicting these characteristics for an axial-flow compressor is described in Chapter 4. This prediction method can be used to forecast the effects of variable geometry, such as Inlet Guide Vanes.

Descriptions are then given in Chapter 5 of the programs which have been developed to predict the transient behaviour of two-spool turbofan engines, such as those investigated in this study. The application of the relevant prediction program to the Turbofan being studied is then reported in Chapters 6, 7 and 8, and the results discussed.

The general conclusions and suggestions for further work are given in Chapter 9.

Chapter 2

Variable Engine Geometry Features

2.1 Introduction

As explained in Chapter 1, the behaviour of aircraft gas turbine engines during rapid acceleration and deceleration is of considerable interest. It is very important that these transient operations can be carried out rapidly without endangering the operating stability of the engine, in particular encountering a serious stall or surge in the compressor(s). With most modern aircraft engine compressors, surge is likely to be encountered at low speeds and during acceleration. To overcome the difficulty of compressor to surge, the geometry of the engine can be altered and improvements to the surge margin can be made.

2.2 The variable engine geometry features to improve the transient response

The features of the variable engine geometry with the view to achieving surge free during transient operations of the engine are as follows:

1. Variable Inlet Guide Vanes for compressors.
2. Variable compressor stators.
3. Bleeds :
   - 3.1 Bleeds in compressors.
   - 3.2 Bleeds between compressors.
4. Variable Nozzle Guide Vanes for turbines.
5. Variable Area Nozzles.

The use of variable engine geometry is beneficial to the engine because it may provide a good performance of the engine and may improve the characteristics of the components of the engine during both steady-running and transient operations. A better optimisation could be obtained by altering engine schedules so that the engine can operate safely during flight envelope. In the actual application to the engine being studied in this work, this is available only for 1 and 3.1 of the above features. In the present theoretical transient prediction work, the variable geometry cases considered are extended to features 1, both 3.1 and 3.2 and 5 of the above.

2.3 The choice of the Rolls-Royce Tay engine (RB 163 Tay Mk 610)

A good understanding of the behaviour of aircraft gas turbine engines during transient conditions is of a great importance to optimise the overall performance of these engines. Therefore, accurate simulation techniques providing comprehensive information about transient operations are desirable. The degree of accuracy of these techniques and their reliability for the prediction of the gas turbine engine behaviour during these transient operations are highly dependent upon the accuracy of components characteristics. The engine model under the present investigation is the Rolls-Royce Tay engine. All information on engine characteristics and components characteristics of the engine have been provided by the manufacturer. Thus, the simulation code used can represent the actual engine transient performance characteristics. The major features which have been led to choose the Tay engine as the basis to carry out the investigation are that the engine is currently in operation and is powering different civil aircrafts such the Fokker 100, Gulfstream 4 and BAe 1-11 aircrafts. The most attractive features of the Tay engine are:

1. Reliability
2. Low specific fuel consumption (see figure 1.3)
3. Low noise output
4. Reduced harmful exhaust emission.

The Tay engine also provides a thrust increase of the order of 25 % compared with existing Spey installations, as illustrated in figure 1.4. It is therefore, an ideal power plant to provide development potential for existing Spey powered aircraft offering re-engining possibilities on existing airframes.

2.4 Description of the Tay engine

The Tay engine derives its pedigree from the Spey family of engines. The engine is developed partly from the Spey Mk555 engine. The main difference between these two engines is that the Spey engine has a low pressure (L.P.) compressor at its front while the Tay engine incorporates a wide chord Fan and intermediate pressure (I.P.) compressor at its front. The use of this split of compression in the L.P. system allows the Tay engine to have a bypass ratio of 3:1 as compared to the Spey engine's bypass ratio of 1:1. The Tay engine, as the Spey, mixes the bypass air with the exit core gases to give a mixed exhaust. This has an advantage because with the high air mass flow in the bypass duct, mixing this with the core gases reduces the noise otherwise caused by high jet velocity at the exit of the engine. The typical cross section of the Tay engine is shown in figure 1.2. The engine consists of a large Fan, one I.P. compressor, one H.P. compressor, ten combustion chambers, one H.P. turbine, one L.P. turbine, a twelve lobe forced mixer and finally an exhaust Nozzle.

Air is drawn in into the engine by the Fan where an amount of 25 per cent of the air flows to the engine core and the remaining of air bypasses the core into the bypass duct. The Fan is 44 inch in diameter and secured to the Fan shaft by a bolted up flange and curving coupling. The air in the core leaves the Fan and undergoes some further compression in the three stage I.P. compressor before it enters the high pressure compressor. Both the Fan and the I.P. compressor are driven by the three stage Low Pressure (LP) turbine. The LP turbine uses the latest proven technology. The main compression of the air in the core occurs in the H.P. compressor. The H.P. compressor consists of twelve separate rotor discs mounted on a two piece steel shaft coupled to the high pressure turbine. It provides a high compressive efficiency at the high speed at which the engine runs during the take off and cruise. An airflow control system is used to enable the engine to run at low engine speeds. It consists of variable guide vanes at the H.P. compressor inlet linked to an annular bleed valve in the casing surrounding the seventh stage of the H.P. compressor. This control system powered by a hydraulic actuator, responding to the H.P. compressor speed signal, limits the mass of air supplied to the combustion chamber at low speed, but reduces this control as the engine speed increases. The compressed air then passes through a diffuser to a combustion chamber where it mixes with the fuel. The combustion chamber is tubo annular with ten flame tubes, each with one duplex burner. A transply material is used in the combustion chamber. The chamber aerodynamics provides a good mixing of air with fuel, and hence efficient combustion. After combustion, hot gases leave the combustion chamber and expand through the two stage High Pressure (HP) turbine which extracts the power to drive the H.P. compressor, and then through the three stage LP turbine in which additional expansion occurs, providing sufficient power to drive the Fan and the I.P. compressor. After leaving the turbines, the hot gases combine with the cold bypass air in a forced mixer to form exhaust gases. The mixer is a forced deep chute type with twelve lobes. The mixture expands finally through the exhaust nozzle, leaving it with relative high velocity. The pressure at exhaust from the nozzle is approximately equal to the pressure of the surrounding atmosphere, and consequently the engine provides a corresponding thrust to push the aircraft forward.

The main performance parameters of the Tay engine are:

<u>Take Off performance at sea level ISA static</u>

Maximum thrust (lbf)	13550
Overall Pressure Ratio	15.5
Bypass Ratio	3.0
Turbine Entry Temperature (K)	1320

<u>Dimensions</u>

Overall length (in)	101.0
Fan tip diameter (in)	44.0

<u>Weight</u>

Basic definition (lb)	3000

Chapter 3

Literature Review of Prediction of Transient Performances of Gas Turbine Engines

3.1 Introduction

The first attempts to predict the transient performance of gas turbines were made on simple single-spool aero gas turbines or power generation gas turbines. Typical of this work was reported by Saravanamuttoo (1963) using an analogue computer. A major development was to use digital computation e.g. Fawke and Saravanamuttoo (1971a) which was applied both to single-spool and two-spool aero engines. The most complex of engine configurations can now be modelled, and many papers have been published, for example, Schobeiri, Attia and Lippke (1994).

The present work has been to investigate the use of variable geometry, or modified component performance, to improve the transient performance of a turbofan engine. These studies using transient prediction procedures of the types mentioned above are discussed below. Most of the theoretical studies have ignored heat transfer effects, and these are discussed first, and referred to as the adiabatic case. Then follows a review of the predictions when heat transfer effects are accounted for.

3.2 Transient performance predictions of gas turbine engines: case adiabatic

Much of the early development of transient prediction procedures was carried out by Saravanamuttoo and co-workers. They have used analogue, digital and hybrid computers for simulating the behaviour of gas turbine engines during transient and steady-running operations. In one study, Saravanamuttoo and Fawke (1970) have presented a mathematical model, with a digital computer, for predicting the transient trajectories in the compressors of a two-spool turbojet engine. To demonstrate the accuracy of the simulation model, the same authors (1971a) have carried out experimental tests on a high pressure ratio turbojet engine with variable nozzle. The test results have been compared directly with the simulation predictions. A satisfactory agreement between the simulation results and experimental data was obtained for both accelerations and decelerations.

In a later study, Fawke, Saravanamuttoo and Holmes (1972) have simulated the speed and thrust responses of a similar two-spool turbojet having a fully variable nozzle. The method of Inter-Component Volumes has been adopted for calculation procedures using a digital computer. The method of Inter-Component Volumes (ICV) is described in some detail later in this thesis - Chapter 5. A wide range of engine transients were performed experimentally to verify the validity of the simulation model. The results showed that the speed responses of the LP and HP shafts of the engine for all cases were in broad agreement with those predicted by the simulation model, both for acceleration and deceleration. They concluded that the model can be used with confidence to simulate the transient behaviour of aero gas turbine engines. However, it is worth noting that in all the predictions carried out and described above the transient schedule was controlled mainly by a schedule of fuel flow as a function of time. This kind of scheduling does not simulate a typical transient for a real engine but a forceful transient one. Therefore, scheduling the fuel flow as a function of time can be misleading and may give poor predictions of the performance of the engine during transients.

The increase demand for industrial gas turbine engines for power production, and health monitoring of these engines, has prompted a new surge in studying the transient behaviour of these engines. Blotenberg (1993) has described a simple model for simulating the dynamic response of power generation gas turbine engines. The simulation model is of a modular concept. The model was applied to a two-shaft gas turbine engine to predict the speed response of the engine during a load shedding of the gas generator, i.e. a sudden decrease of the electric output to zero load. The results showed that the speed response of the gas generator followed closely the fuel flow input and the speed decreased as the fuel flow was reduced. A good agreement was obtained between the transient response predictions of the model and the experimental results.

A series of extensive investigations on transient performance of industrial gas turbine engines has been reported by Schobeiri and co-workers. Schobeiri (1986) has developed a model called 'COTRAN' for simulating the dynamic behaviour of power generation gas turbine engines. The model uses one-dimensional conservation equations for different stages of the compressor and turbine. The thermal effects are also included in the code. The simulation capability of the code was however restricted to predict the transient behaviour of single-shaft power generation gas turbine engines only. This circumstance has motivated Schobeiri, Abouelkheir and Lippke (1993) to develop a new model for simulating the behaviour of more complex industrial gas turbine engines during transient operations. This simulation model called 'GETRAN' is of a modular concept and is based on solving a number of systems of partial differential equations governing each component of the engine. The simulation of the complete engine is accomplished by combining individual components that have been modelled mathematically. Different transient operations were performed on different engine configurations, and the simulation predictions were compared with test results to demonstrate the effectiveness of the model. They showed good agreement between the model and experimental data. More recently, Schobeiri, Attia and Lippke (1994a) and (1994b) have extended the simulation code 'GETRAN' to include single and multi-spool aero gas turbine engines.

Another work on predicting the transient performance of aero gas turbine engines has been reported by Pilidis and Maccallum (1985). In contrast to other simulation predictions described above, which have used the method of inter-component volumes, they have used the method of continuity of mass flow in their simulation. The method of Continuity of Mass Flow (CMF) is described in Chapter 5. The authors have also incorporated the effects of heat transfer in the model and these are explained in a later paper, Pilidis and Maccallum (1986). In their report, the authors have predicted the transient performances of two-spool engines with and without mixed exhausts. They observed that both engines had similar responses during the transients. To demonstrate the effect on engine performance of the mixing in the jet pipe between the mixer and the final nozzle, a comparison of the steady-running performance of mixed exhaust engines with zero and complete mixing was investigated. They showed that a better fuel consumption at higher thrust was obtained with the engine with complete mixing.

The choice of the method in the simulation model is very important for accurate predictions of the transient performance of the engine. Recently, Maccallum (1989) has compared the predictions of these two methods on transient performance of a two-spool turbofan with mixed exhausts. He found that the speeds of LP and HP shafts and thrust responses predicted by the inter-component volumes method were slower, about 4 per cent, than those predicted by the method of continuity of mass flow. He also showed that the transient trajectories in the fan and the I.P. compressor predicted by both methods were similar. For the H.P. compressor, a less severe departure of the transient trajectory from the steady-running was predicted by the inter-component volumes method than that predicted by the method of continuity of mass flow during the acceleration. He concluded that the predictions of the method of inter-component volumes are more realistic because the effect of the mass storage between the components are accounted for, and which is ignored in the second method. The calculation procedures adopted in these two methods are explained in details in the Chapter 5.

More recently, Ganji, Khadem and Khandani (1993) have developed a general methodology for simulating the sensitivity of a single-spool turbojet engine during transient conditions. Based on the method of inter-component volumes, the method is based on solving a set of stiff, time dependent non-linear differential equations using an Ordinary Differential equations (ODE) solver developed by Hindermarsh. Three cases of step, ramp and sinusoidal fuel flow rate into the combustion chamber were selected to study the response of the engine. The results showed that the response of the temperature at the turbine entry was much faster than the response of the turbine inlet pressure, and the temperature followed closely the input of fuel flow rate. These predictions are reasonable. However their further predictions on air mass flow and shaft speed are that the air mass flow closely follows the fuel flow but the shaft speed response is rather slower, yet still faster than the pressure response. These last predictions are dubious - one expects pressure response to lead the shaft speed response.

3.2.1 Cases of modified compressor performance

A number of studies have been conducted towards obtaining an insight into the conditions within gas turbine compressors during transients. Such investigations require a model for representation of surge, and/or stall within the compressor to be included in the prediction program. A model of this nature has been evolved by Schobeiri et al (1993), who have applied to a two- shaft power generation engine.

Aircraft gas turbine engines, being air breathing machines, ingest large quantities of air. The presence of water in this air will affect the performance of the engine, as the total gas mass flow will rise along the flow path as the water evaporates. Also the compressor will experience an additional drag due to the impact of droplets. Effectively, the compressor characteristics will be modified. Models to represent these effects during engine transients have been introduced by Haykin and Murthy (1988) and others. Typically, in an acceleration the working line in the H.P. compressor is raised and the risk of surge is increased.

If dust is ingested this, over a period of time may build deposits on blading surfaces in the compressor(s) and turbine(s), or it may erode the blading. Here again, the characteristics of the compressor (and turbine) are modified. Experimental observations have been made by Batcho et al (1987). The most serious effect was erosion of the H.P. compressor blades. Models were formed and, typically, a reduction of up to 50 per cent in the surge margin was predicted.

From the above review, the off-design performance of axial-flow compressors is critical for the efficient and stable operation of aircraft and industrial gas turbine engines. However, compressors are not beyond redemption. Improvements in the compressor performance can be obtained by the use of variable geometry. To demonstrate the effectiveness of this technique, Muir, Saravanamuttoo and Marshall (1989) have developed a general method for predicting the influence of variable geometry axial compressor on engine performance. The method has been applied to a single-spool shaft marine gas turbine engine, and uses the stage-stacking method. The results showed that a decrease in the stagger angle of the variable stator at a given speed is equivalent to opening the variable stator vanes, thus increasing the airflow in the core of the engine. For example, at the given speed of 9160 rpm, a 1 degree decrease in the stagger angle resulted in a 3 per cent increase in airflow, a 4 per cent increase in fuel flow, and a 5 per cent increase in power.

The use of variable geometry of the compressor can be beneficial since the characteristics of the compressor are altered. If a good scheduling of the stator or vane turning angles is achieved an improvement in surge margin of the compressor can be obtained, hence a good transient performance of the engine.

3.2.2 Scheduled changes of final nozzle area

Another alternative feature of engine variable geometry is the variable area of the propelling nozzle. The use of variable area of the jet engine nozzle might alleviate the tendency of surging in the compressors if appropriate scheduling of the nozzle is used. A number of studies have been conducted to investigate this effect. Fawke and Saravanamuttoo (1971b) have carried out experimental tests to study the effect on compressors trajectories of varying the nozzle area of a two-spool engine. They observed that in an acceleration, when the nozzle was open the trajectory in the H.P. compressor moved less close to the surge line, whereas the trajectory in the L.P. compressor moved nearer surge during the transient. The opposite situation happened when leaving the nozzle closed. The surge margin of the H.P. compressor was thus improved at the expense of the surge margin of the L.P. compressor by opening the final nozzle during the acceleration. They quoted that the surge margin in the L.P. compressor can be improved by closing the nozzle during decelerations.

The response of the gas turbine engine during a transient is also very important. Maccallum (1981) has predicted the effect on the transient responses of reducing the final nozzle area, by two per cent, of single-spool and two-spool turbofan engines during accelerations. He illustrated that the change of the final nozzle area of the two-spool engine had a very little effect and increased the acceleration times by just over 1 per cent, whereas the same reduction with the single-spool engine produced an increase in the acceleration times by 6 per cent.

More recently, Schobeiri, Abouelkheir and Lippke (1993) have simulated the effect on the transient performance of closing the exit nozzle area of a single-spool engine, single-shaft engine (this is effectively an aero engine configuration). The transient has been performed with constant fuel flow, according to the ramp variation in the nozzle exit area. They found a strong mismatch between the turbine and the compressor resulting in changes in turbine power until the completion of the transient operation. The matching of the compressor and turbine was resumed once the nozzle was open. The effect of closing the nozzle area of a single-spool engine is to increase the pressure at the exit from the turbine resulting in a reduction in the mass flow through the turbine, hence a loss of turbine power. Consequently the air mass flow in the compressor is reduced. Thus the fuel air ratio increases leading to a temperature increase at the entry to the turbine. Consequently, the result of this transient, as one would expect, is a strong mismatch between the turbine and compressor, resulting in extreme changes in turbine power.

3.3 Non-adiabatic effects on the engine performance during transients

In practice, there are some discrepancies between the ‘adiabatic’ predictions and the performance of real engines. These discrepancies are obviously due to the thermal effects. In a real engine, the heat transfer between gas streams and the metal during a transient is very significant and cannot be ignored. Neglecting the heat transfer in the simulation model may underestimate the performance of the engine during transients. Therefore, models for these thermal effects have been developed. In the present work these models for thermal effects have not been used, in order to achieve reasonably fast simulation. Therefore they are not discussed further here. For overall summaries of heat transfer effects in gas turbine engines during transients the reader is referred to too recent papers - Pilidis and Maccallum (1986) and Maccallum and Qi (1989).

3.4 Concluding remarks

Gas turbine engine performance computer models constitute a very important tool, useful for different aspects of engine study. During the early stages of gas turbine engine conception, they are used to evaluate the influence of different design choices on overall performance. During the service life of an engine, they offer the possibility of estimating performance parameters and cycle details for the condition encountered during operation. This possibility is of fundamental importance to any technique of engine performance monitoring. The building of performance models is based on dividing the gas turbine engine into component; e.g. compressor, combustion chamber, turbine, nozzle, according to the kind of thermodynamic process occurring in each of them. The determination of performance parameters and cycle details is achieved by solving a system of equations and expressing the state changes of the working gas and the compatibility conditions between the components. Today, the simulation models have greatly contributed in the understanding of the transient performance of gas turbine engines, and any configuration no matter how complex, can be modelled.

For accurate prediction, heat transfer effects need to be taken into consideration. Typically, the predictions of a procedure ignoring heat transfer effects ('adiabatic' prediction) underestimate response times of shaft speeds, or thrust, by about 25 to 35 per cent. The present work is a study of the benefits to be obtained by use during transients of variable geometry. The predictions of the present work have been made only for adiabatic cases. However the benefits, or otherwise, predicted for the adiabatic cases will read across to actual engines where heat transfer effects are accounted for.

Chapter 4

Predicted Effects of Altering IGVs and Bleed Valve on an Axial flow Compressor

4.1 Introduction

Axial flow compressors are, in the whole, amenable to analytical treatment, and usually a good prediction of their performance can be made before they are run. The performances are conveniently thought of in terms of the overall characteristics of pressure rises, temperature rises, and efficiency plotted against mass flows for various values of the shaft rotational speed. A knowledge of the overall characteristics is essential if one is to know the conditions under which a compressor is liable to surge so that these conditions may be avoided if at all possible. The predictions of the characteristics are also needed in preliminary design as arbitrary stop signs in setting up a geometry-flowfield condition and they are used in design analysis to estimate the degree of risk associated with a particular flow pattern. The forms of their surge lines have marked influences on the allowable starting, acceleration, and control qualities of gas turbine engines.

In the present investigation, the prediction of the overall characteristics of an axial compressor has been carried out on the 12 stage axial flow H.P. compressor of a two-spool bypass engine (the Rolls-Royce 'Tay' engine). The method adopted for the prediction of the H.P. compressor characteristics has been based on the row by row method. It is described in more detail in section 4.2 which follows. The blading and the flow cross sectional area for each blade row of the compressor is required for the appropriate design mass flow and rotational speed. The geometry blade angles are also needed for determining the angles of the flow entering and leaving each blade row to construct the velocity triangle diagrams. This is because it is the air flow angles into each blade row which really determine whether a blade row is operating correctly. The air flow angles depend on the ratio of the flow velocity to blade speed. The blade inlet and outlet angles for all rotors and stators are then provided. Other important geometric values of the blade height, blade mean diameter, blade pitch and chord are also known and incorporated in the prediction method for estimating the losses in each blade row of the compressor.

The compressor is designed with one row of variable inlet guide vanes located at the front of the first rotor to ensure the correct angle of incidence for the first row of rotor blades. As the flow varies with the speed of the compressor, the angle of incidence of the rotor blade will vary, therefore, the IGVs turning angles have been varied over a specified range of non-dimensional speeds to accommodate large variations of air mass flow rate and speed. The angle settings of the IGVs have been selected to vary with the highest angles at the lowest flows and the lowest angles at the highest flows accordingly to a schedule set to them. This means as the compressor speed is reduced, the angle of the variable vanes is increased to limit the amount of air flow and thus avoid unstable operating condition. The range of the IGVs angle settings, expressed in terms of the flow swirl angle into the first rotor, has been from 34^0 swirl to 0^0 swirl. This has been quantified by the parameter AINZ(1) in the prediction program, and which represents the absolute air inlet angle approaching the first row of rotor blades.

Since compressor is 'non-dimensional', then any inlet conditions can be used, if outputs are expressed in non-dimensional form - which of course is the normal.

4.2 Description of the method for predicting overall characteristics of an axial flow compressor

It is the flow past the individual blade elements which largely determines the overall characteristics given by an axial flow compressor. The flow in an axial compressor is very complex and is influenced by three-dimensional effects. Accurate method for predicting the overall characteristics of a compressor should be based on blade element data, with radial integration, followed by axial stacking. Such a procedure is complex and some of the information required is as yet incomplete. Furthermore, three-dimensional calculation methods are relatively new and most people find it difficult to think, let alone design, in three dimensions.

In the present investigation, a simple method has been adopted to predict the overall characteristics of a multistage axial flow compressor using calculations at the pitch-line with possibility of the existence of stall cells. The method uses row by row calculations at the mean blade height, and this is considered reasonable for hub/tip ratios greater than 0.6. The hub/tip ratios of the H.P. compressor under investigation are in the range 0.61 to 0.91. The flow has been taken as two-dimensional and the three-dimensional effects on the performance in the real axial compressor are correlated using empirical coefficients obtained from cascade data, (Howell and Bonham, 1950). Axial flow compressors in modern aero gas turbine engines are operating at very high speeds and Mach numbers, and the assumption of the flow being incompressible is not valid at these high speeds. The air flow within a blade row is transonic and even supersonic, thus the flow is compressible. The Mach number effects on the performance of the blade are therefore considered. Another very important aerodynamic input in the design of axial compressors is the ratio of the exit axial velocity to the inlet axial velocity of the blade. This was often not given the attention it deserved and it is apparent that changes in the axial velocity across the blade row have a direct effect on the blade boundary layers. The axial variation within a blade row into the exit velocity triangles and energy transfer calculations are incorporated in the present method as well.

The next step is to evaluate the air angles for each row. The air flow inlet angle (α_1) in the first row (or rotor row) is calculated from the velocity diagram, and the incidence angle is calculated by subtracting the inlet blade angle (β_1) of the rotor from the air flow angle (α_1). The blade notation is defined in figure 4.1. Naturally, for design purposes, it is desirable to use as large a deflection as possible in order to get the greatest pressure rise from the blade row with minimum loss. The mean deflection rises with increasing incidence, reaching a maximum value in the region of the positive stalling incidence. Stalling of an axial compressor is taken to occur when the blade loss is twice its minimum value. It is, however, not advisable to operate on the point of stalling, so a nominal deflection (ε^*) is used. This nominal deflection and its corresponding air flow outlet angle (α_2^*) are of a great importance to the design of an axial compressor and its performances. The nominal air flow outlet angle for the blade row is calculated by adding a deviation to the blade outlet angle (β_2) of the rotor. The angles (α_1) and (β_1) are fixed primarily by the design of the compressor; they are not strictly constant, for the flow does not exactly follow the blade outlet angle but deviates by a few degree and the deviation is a function of incidence. The correlation for deviation is given in Appendix I. The value of the nominal deflection is mainly dependent on the pitch/chord ratio and the nominal air flow outlet angle. The nominal deflection can be found from the graph given in the figure 4.2, (Howell and Bonham, 1950). Thus, the nominal incidence angle (i^*) can be easily determined by adding the nominal deflection to the nominal air flow outlet angle and subtracting the inlet blade angle (β_1) of the rotor. Knowing the nominal angles values, a suitable value of the deflection angle (ε) can be easily found from the diagram shown in the figure 4.3, (Howell, 1945a).

When the incidence angle of the flow into a particular row becomes too great, it is assumed that a stall cell is then formed in part of the annular area, and in the remaining area normal flow is restored. The quantitative definition adopted for this stalling of a blade row is that the profile drag coefficient rises to double its minimum value, and this occur when the numerical value of the dimensionless relative incidence group $(i - i^*/\varepsilon^*)$ reaches 0.4. When the flow through a particular row is reduced below that giving this stalling value of the incidence group, it is assumed that the flow redistributes itself in the annulus so that part of the annulus operates with a flow giving exactly this stalling value of the incidence group, and the remainder of the annulus is occupied by a stall cell with no significant flow.

After completion of the evaluation of the air flow angles for the bade row, it will now be necessary to check over the performance particularly in regard to the efficiency which for a given work input will completely govern the final pressure ratio. In the real axial compressor, the flow is strongly three-dimensional and losses are generated. That is the efficiency of the blade row can be calculated after the pressure rise coefficient being found from a correlation for loss. The work done factor is inserted to account for the reduced energy transfer. Due to the endwall boundary layers in the annulus area, the work done factors will vary through the compressor, and the corresponding value of the work done factor for every stage is taken from the diagram shown in the figure 4.4. The temperature rise in the stage is corrected by the corresponding value of the work done factor for this stage.

Efficiencies of the complete compressor have also been predicted. The efficiency is defined here as an 'aerodynamic efficiency' for the complete compression of the delivery air:

$$\eta = \frac{\Delta h_{is}}{\Delta h} \tag{4.1}$$

Note that the definition of compressor efficiency is not taking account of the energy lost in dumping the air bled, when at lower speeds, from midway along the compressor.

4.3 Performance prediction of H.P. compressor

The prediction of the overall characteristics of the H.P. compressor has been carried out at several values of rotational speed. At each rotational speed, the overall characteristics mass flow rate has been varied between two limits. The upper limit of mass flow has been close to the maximum obtainable at that speed. The lower limit has been fixed by the existence of an aerodynamic instability known as surge. It is recognised that the pressure ratio point on the characteristics at which a compressor surges is very difficult to determine. The surge point is therefore taken as the point of maximum pressure ratio that the compressor can deliver at a given rotational speed. The surge line shown on the H.P. compressor characteristics is found by joining the surge points together. The predicted overall characteristics and efficiency of the 'Design' H.P. compressor are shown in the figures 4.7 and 4.8, respectively.

In these figures, it can be seen that the pressure ratio of the compressor depends strongly on the rotational speed, increasing as the compressor operates at higher speeds. At a particular rotational speed, the pressure ratio increases rapidly as the air mass flow rate is reduced, reaching a maximum point, and any further decrease in the mass flow rate is accompanied by a fall of pressure ratio for that given non-dimensional speed of the compressor. This part of the characteristics has not been drawn since it represents a zone of flow instability.

With regard to efficiency, when the compressor operates in a stable zone at a given non-dimensional speed, as the mass flow is increased from the surge mass flow rate, the efficiency rises to a maximum and then falls off until the choking flow is reached (e.g. figure 4.8), the pressure ratio delivered by the compressor of course reducing in the process. These predicted peak efficiencies at the various non-dimensional speeds may be compared. It is seen that, on decreasing the speed from the maximum (and design) values, the peak efficiency progressively rises. This is partly due to the reduction in profile drag pressure loss. Also, importantly, at lower speeds, Mach number effects, and losses, vanish. There is a further influence in that, at lower speeds, the mid-compressor bleed allows more favourable incidence angles in the earlier stages. It is worth pointing out that the maximum efficiency of the compressor at a given rotational speed is obtained at a mass flow rate slightly higher than that giving maximum pressure ratio. This mass flow increase is small, thus, the locus of the operating point for a maximum efficiency would be close to the surge line of the compressor.

The most striking feature on the predicted overall characteristics of the H.P. compressor is that, at the beginning of closure of the Bleed Valve, a very rapid increase in the surge pressure ratio occurs, exhibiting a sharp 'kink' in the characteristics of the compressor as shown in figures 4.7 and 4.9. This sharp increase in the surge line pressure ratio, associated with the Bleed Valve closing, is explained and discussed in Section 4.4.2 below. It may be noted that when the Bleed Valve closes there will be an increase in mass flow to the later stages. If this cannot be fully accepted it may results in an increase in the incidence of the front stages, hence their unwanted stalling. The kink in the surge line, which is encountered with high performance axial flow compressors, can be a serious problem towards the operating stability of the compressor during transient operations. For example, the H.P. compressor of the 'Tay' engine has a kink which appears to be a serious problem, hindering a good acceleration of the engine.

4.4 Performance with altered IGV and Bleed Valve schedules

With the above reported method of predicting the characteristics of a compressor, it is very easy to study the predicted effects of altering the settings of IGVs and of changing the Bleed Valve schedule. Predictions have therefore been made for the H.P. compressor when using altered IGV and Bleed Valve schedules. These predictions, and the results, are discussed below.

4.4.1 Performance prediction with 'Altered' IGV schedule

To demonstrate the effects of altering the IGVs angle settings on the overall characteristics of the H.P. compressor, the prediction has been run by altering the operating range of speeds of IGVs set ahead of the first rotor of the H.P. compressor. Two different states of scheduling of operating IGVs have been selected for this purpose. For the first schedule, the IGVs begin to open at the $\left(N_H/\sqrt{T_1}\right)$ value of 549 $\left(rev/\min K^{1/2}\right)$, from the 34^0 'closed' position, to become and remain fully 'open' at the non-dimensional speed value of $589\left(rev/\min K^{1/2}\right)$. For the second schedule, the operating range of the IGVs has been extended to include lower speeds of the H.P. compressor. The IGVs begin to open progressively from the $\left(N_H/\sqrt{T_1}\right)$ value of 480

$\left(rev/\min K^{1/2}\right)$ until they are in the fully 'open' position for all speeds higher the non-dimensional speed of 589 $\left(rev/\min K^{1/2}\right)$. The operating schedules of IGVs in function of the H.P. compressor speeds are shown in figure 4.5. For the purpose of this study, the first schedule will be termed the 'Design' schedule and the second schedule will be termed the 'Production' schedule. As mentioned earlier, the engine under investigation is fitted with both IGVs and Bleed Valve, therefore the operating bleed schedule of the 'Design' Engine has been kept the same. This would enable one to estimate the performance brought about when altering the IGVs angle turnings on the overall characteristics of the H.P. compressor and its surge line. The predicted effects of altering the IGVs schedules on the H.P. compressor characteristics and efficiency are illustrated in figures 4.9 and 4.10, respectively.

The effects of moving the IGV schedule from the 'Design' setting to the setting defined above can be seen by comparing figures 4.9 with 4.7 and 4.10 with 4.8. The comparison is clearer on figure 4.15. Moving the IGV schedule to the lower $\left(N_H/\sqrt{T_1}\right)$ range i.e. making the IGVs more open at an $\left(N_H/\sqrt{T_1}\right)$, has the effect of moving the constant speed lines to the right thus to higher mass flow rates. The most marked change is for the constant speed line of 549 $\left(rev/\min K^{1/2}\right)$ where the IGVs have the largest change (from 34^0 to 11.6^0). Here, the increase in mass flow is about 8 per cent.

It is helpful to express the predicted change in mass flow, at an $\left(N_H/\sqrt{T_1}\right)$, in terms of the change in IGV setting. The predicted relationship is

$$\frac{\Delta\dot{m}}{\dot{m}} = 0.0036\cdot\Delta\alpha_1 \tag{4.2}$$

where $\left(\alpha_1\right)$ is the swirl angle in degrees. This relationship is only approximate. Engine data will be shown in Chapter 6 which leads to an experimental relationship equivalent to the above.

The reason for the effects noted above is that, at a particular non-dimensional mass flow and at a particular non-dimensional speed, opening the inlet guide vanes slightly increases the incidence to the first set of rotor blades. Provided stall is not encountered, this increases the pressure rise achieved in the first blade pair. The lower axial velocities at entry to the subsequent pairs also lead to increased pressure rises. Thus the overall pressure rise across the compressor, at that mass flow, is increased.

It is noted that the position of the surge line is not significantly altered.

With regard to efficiency, comparison of figures 4.10 and 4.8 shows a slight improvement of efficiency, of about 0.3 per cent, over much of the range. This is due to slightly better incidence angles.

4.4.2 Performance prediction with 'Altered' Bleed Valve schedule

In the actual engine, up to 14.7 per cent of the H.P. compressor inlet mass flow can be bled from the seventh stage to the Bypass Duct when the Bleed Valve is fully open. In the 'Design', the Bleed Valve operates over a specified range of the H.P. compressor speeds, and starts to close at the $\left(N_H/\sqrt{T_1}\right)$ value of 549 $\left(rev/\min K^{1/2}\right)$, from fully 'open', to become and remain fully closed at the value of $\left(N_H/\sqrt{T_1}\right)$ of 568 $\left(rev/\min K^{1/2}\right)$. To assess the effects of the Bleed Valve scheduling on the characteristics of the compressor, the 'Design' Bleed Valve schedule has been moved to operate at lower values of the H.P. compressor speeds. The Bleed Valve schedule begins to operate at the value of the non-dimensional speed 493 $\left(rev/\min K^{1/2}\right)$, from full bleed, to become and remain fully closed when the H.P. compressor reaches its non-dimensional speed 520 $\left(rev/\min K^{1/2}\right)$. For convenience, this operating Bleed Valve schedule will be termed the 'Production' schedule. It is actually the schedule that has been adopted for the 'Production' engine. The operating schedules of the Bleed Valve as function of the H.P. compressor speeds are shown in the figure 4.6. The prediction method has been run for both Bleed Valve schedules by keeping the operating range of the IGVs the same as the 'Design' IGV schedule portrayed above, enabling one to estimate the performance brought about by such alteration. The predicted effect of altering the Bleed Valve schedule on the H.P. compressor characteristics and efficiency are shown in figures 4.11 and 4.12, respectively.

The effects of moving the Bleed Valve in the H.P. compressor from the 'Design' schedule to the schedule defined above can be seen by comparing figures 4.11 with 4.7 and 4.12 with 4.8. The comparison is shown further in figure 4.16. Moving the Bleed Valve schedule from the 'Design' setting of 549 - 568 $\left(N_H/\sqrt{T_1}\right)$ to the lower speed range of 493 - 520 has the effect of moving the constant speed lines which lie in the range affected to the left, i.e. to lower compressor inlet mass flows. It is seen that, at the $\left(N_H/\sqrt{T_1}\right)$ value of 549 $\left(rev/\min K^{1/2}\right)$ the inlet mass flow reduces by about 5 per cent.

This can be explained by considering a particular inlet mass flow at a particular rotational speed (both non-dimensional), in the region where the Bleed Valve opening is decreased by the schedule movement. Up to the blade pair where the Bleed Valve is positioned, there is no change in the pressure rises. However in the later blade pairs when the Bleed Valve is more closed, the pairs operate with higher axial velocities, due to higher mass flows. Thus the pressure rises achieved in these pairs are decreased, the overall pressure rise decreases and the constant speed line on the characteristic moves to the left.

It is important to note that, in the range, the predicted surge points on these new non-dimensional speed lines lie above, on the pressure ratio - mass flow plot, the 'Design' surge line. For example, at the $\left(N_H/\sqrt{T_1}\right)$ value of 560 $\left(rev/\min K^{1/2}\right)$ the surge margin improves by about 5 per cent (the surge margin in this thesis is defined as the difference in pressure ratio between the surge condition and the steady-running condition at a given non-dimensional speed).

It is to be noted that the position of the steeper increase in the pressure ratio, i.e. the 'kink', has moved with the change of the Bleed Valve schedule, thus demonstrating that the 'kink' is associated with the closing of the Bleed Valve.

4.4.3 Effects of moving both the IGV schedule and the Bleed Valve schedule

The effects of moving the Bleed Valve schedule to operate at lower speeds and extending the IGV schedule to include these speeds on the overall performance of the H.P. compressor are additive. These modified schedules have been defined in Sections 4.4.1 and 4.4.2, and are referred to as 'Production' schedules. The 'Production' IGV schedule leads to an increase in the air mass flow in the H.P. compressor and a slight improvement in efficiency, and the 'Production' Bleed Valve schedule brings a great improvement in the surge line of the compressor as shown in the figures 4.13 and 4.14, for the same reasons given previously. Therefore a such combination of altering both IGV schedule and Bleed Valve schedule to the 'Production' schedules would seem like a good choice to improve the overall performance of the H.P. compressor and could be used with confidence to adjust the characteristics of high performance compressors to provide better transient operations of high performance gas turbine engines.

Chapter 5

Procedures for Predicting Transient Behaviour of Turbojets and Turbofans

5.1 Introduction

In order to achieve the continuous improvement of the efficiency and the performance of aero gas turbine engines, the design of these engines is now becoming very complex. The transient performance of these engines is very critical and remains one of the major problem facing the engine designer. These high performance engines are operating at very high pressure ratios for higher thermal cycle efficiency. Unfortunately, they are more prone to encounter surge difficulties than ever when operating at off-design performances, such as during transients. Therefore, as discussed in Chapter 3, a knowledge of performance during transient operations is a prerequisite for evaluating the overall performance of gas turbine engines. It should be recognised that the response of an aircraft gas turbine engine during a transient is difficult to investigate experimentally. Test of this nature cannot be carried out until late in the development program when the mechanical integrity of the design has been established by a considerable amount of engine running. However, accurate information on the behaviour of the engine during transient operations are needed at an early stage for the development of a suitable control system of the engine and also to ensure that customer requirements for rapid response can be satisfied. An alternative method for providing comprehensive information about the transient performance of an aircraft gas turbine engine is the use of the computer simulation.

The literature review in Chapter 3 showed that computer simulations have been in common use since the advent of the digital computer as a practical engineering tool in the late 1950 and, at the present time, they are extremely effective diagnostic tools in analytical determination of engine performance. The use of the computer simulation is great in that the investigations of the transient performance of a gas turbine engine can be carried out at low cost without endangering the engine. Furthermore, the computer simulation can give an insight to improve engine transient performance for a full flight envelope operation if appropriate control mechanisms are used. Computer simulations using mathematical models permit the investigation of the engine transient performance as early as the design stage, and can be continued in parallel with the development program. Of course, a successful simulation model must be capable of accurately predicting the engine transient behaviour under various dynamic conditions and be able to operate over the entire running range of the engine.

The information required to construct the simulation model must be available during the early stages of the engine design. These requirements can be met by making use of component characteristics of the engine. It is true to say that the accuracy of the simulation model is dependent upon the real characteristics of the components and overall engine performance data available to make the simulation model. If all components characteristics and the engine layout are known, then the dynamic behaviour of the gas turbine engine can be expressed mathematically. The advantage of this approach to gas turbine simulation are numerous. Estimates of component characteristics are available early in the design stage and can be revised as the development program proceeds. Effects such as variable engine geometry, blade tip and seal clearance changes, unsteady mass accumulation, heat storage and release in and between the engine structure and gas path, fuel schedules can easily be accommodated within the simulation model. Thus, a simulation model of this type is a true representation of the thermodynamic processes in the engine.

While analytical, component rig test, and full engine test results are progressively obtained as the engine is designed, developed, tested and put into the service, these results can be used to continuously improve and update the simulation model. The simulation model, therefore, represents the current best state of knowledge from all sources for a given gas turbine engine model at a current point in the design history.

5.2 Computational modelling of aero gas turbine engines

The function of the computer performance simulation is to satisfy gas mass and mechanical energy and momentum conservation equations throughout the gas turbine engine. In so doing, a unique match of the engine components operating characteristics can be established. To simulate the transient performance of aero gas turbine engines, the transient period is segmented into time intervals. At each time interval the thermodynamic parameters along the gas path are calculated. The power input, or output, for each component is then determined. From the power balance on each shaft, the accelerating torque is found. By integrating this torque, usually assumed constant, over the time interval, the change in shaft speed is obtained. This process of thermodynamic variable calculation and torque integration is repeated over as many time intervals as required.

Two different approaches can be used to accomplish the calculation procedures for simulating the transient performance of aero gas turbine engines. One is the method of continuity of mass flow and the other is the method of inter-component-volumes. In the former, the basic assumption is that flow continuity is satisfied at all times even when the engine is operating transiently. This means the mass flow rate is the same at that instant throughout the engine. In the latter, it is assumed that flow mismatch occurs in the engine during transient operation and this flow mismatch can be used to calculate the rates of change of pressure at various stations in the engine. For clarity, a single spool turbojet has been selected to describe the calculation procedures to be accomplished by these two methods when predicting the transient performance of an aero gas turbine engine. The calculation procedures can be expanded to include more complex gas turbine engine configurations.

5.2.1 Method of Continuity of Mass Flow (CMF)

The continuity of mass flow method is iterative and based on the assumption that flow continuity is maintained at all times even if the engine is operating transiently. The calculation procedures (illustrated in figure 5.1) to predict the transient performance of the engine is as follows: from the flight conditions, intake efficiency and the rotational shaft speed of the engine, the inlet pressure and temperature at the entry of the compressor, and the non-dimensional speed of the engine can easily be calculated. An initial guess for the total pressure at the exit of the compressor is made, hence the pressure ratio of the compressor is obtained. The compressor operating point, which is fixed by the pressure ratio and the non-dimensional speed of the engine, can then be found and the air mass flow rate and the compressor efficiency can be read off the compressor characteristics. This enables the exit temperature and the temperature rise in the compressor to be determined. All thermodynamic variables of the compressor are now known, and the compressor exit thermodynamic variables become the inlet conditions for the combustion chamber. The temperature rise in the combustion chamber is a function of the ratio of fuel flow, given by engine controller, to compressor air mass flow and the temperature at the exit of the compressor, and is found from the combustion temperature rise chart enabling the combustion exit temperature to be calculated. The combustion outlet pressure is calculated by applying a pressure loss factor to the exit pressure of the compressor. Now, the turbine entry conditions of mass flow rate, inlet pressure and temperature and shaft speed are known. By using the turbine characteristic, the turbine exit pressure and efficiency can be obtained enabling the temperature at the turbine exit to be calculated. If the value of the mass flow group for entering the turbine is outside the characteristics, or is on the 'turbine choked' line , the above step cannot be followed. The procedure then is to guess a turbine pressure ratio and iterate back to the compressor pressure ratio to achieve matching of the turbine inlet mass flow group. The guessed turbine pressure ratio is adjusted following test of the mass flow group into the final nozzle. Finally, a mass flow through the nozzle is determined from the ratio of turbine exit pressure to ambient pressure, the temperature at the exit of the turbine and the final nozzle area. In general, this value mismatches the turbine mass flow rate obtained previously. This means the initially guessed value of the compressor exit pressure was incorrect. A new value of this pressure is guessed and the above calculations are repeated. This iterative procedure is continued until the flow mismatch diminishes; thus satisfying the mass continuity condition. Finally, the power imbalance between the compressor and turbine is calculated and the net torque acting on the shaft is determined. Knowing the inertia of the shaft, the instantaneous acceleration of the engine is obtained. The acceleration of the compressor rotor and the torque are related by Newton's second law of motion. The new acceleration is used to calculate the new rotational shaft speed which forms the starting condition for the next time interval. This process of calculation is carried out until the complete transient of the engine is performed.

5.2.2 Method of Inter-Component-Volumes (ICV)

In the method of inter-component-volumes, volumes are allocated to the spaces between the components (also incorporating associated component volumes). The storage of mass flow during any transient process is assumed to occur in these inter-component-volumes. The calculation procedure at each time step is once through and requires no iterations to obtain flow match; each pass through the engine has a physical significance. The size of an inter-component-volume is the volume of the space between any two components plus the half of the volume of each adjacent component. The number of inter-component-volumes in aero gas turbine engines depends on the geometric complexity of the engine. For the example of a single turbojet engine, only two inter-component-volumes are required. The first volume is located between the combustion chamber and the turbine, and the second volume is between the turbine and the final nozzle.

The calculation procedure is outlined below and uses the characteristics illustrated in figure 5.1. Initial guesses of the pressures within the inter-component-volumes must be made, and should be as accurate as possible. These initial pressures will give mass flows into and out of these inter-component-volumes. The procedures for predicting the transient performance of the single turbojet are as follows: from the flight conditions, intake efficiency and engine speed, the temperature and pressure at the inlet of the compressor and the non-dimensional speed of the engine can be calculated. The pressure ratio of the compressor is then obtained from the initially selected pressure. Knowing the pressure ratio of the compressor and the non-dimensional speed of the engine, the air mass flow rate and efficiency are found from the compressor characteristic enabling the temperature at the exit of the compressor to be determined. The exit temperature and pressure of the combustion chamber are obtained from the combustion temperature rise chart and the pressure at the exit of the compressor. The inlet temperature and pressure of the turbine are now known. The initially guessed pressure at the outlet of the turbine is used to obtain the expansion pressure ratio of the turbine. The turbine operating point, which is fixed by the pressure ratio and the engine non-dimensional speed, can then be found and the turbine gas mass flow rate and efficiency are read from the turbine characteristic enabling the turbine outlet temperature to be calculated. Finally, the gas mass flow rate through the final nozzle can be found from the ambient and turbine exit temperatures and the final nozzle area. There are now three mass flow rates throughout the engine. In general, these mass flow rates will not be consistent and mass will either accumulate or diminish in the various inter-component-volumes during the subsequent short time increment. These flow mismatches are used to calculate new masses, temperatures and pressures in the inter-component-volumes of the engine. For a particular inter component volume, the calculation is as follows:

$$m_t = \frac{P_t \cdot V_t}{R \cdot T_t} \tag{5.1}$$

$$m_{t+\Delta t} = m_t + \left(\dot{m}_i - \dot{m}_j\right) \cdot \Delta t \tag{5.2}$$

$$T_{t+\Delta t} = \frac{\left(\dot{m} \cdot C_P \cdot \Delta t \cdot T\right)_i + m_V \cdot C_V \cdot T_V - \left(\dot{m} \cdot C_p \cdot \Delta t \cdot T\right)_j}{\left(\dot{m} \cdot C_V \cdot \Delta t\right)_i + \left(m_V - \dot{m}_j \cdot \Delta t \cdot\right) \cdot C_V} \tag{5.3}$$

$$P_{t+\Delta t} = \frac{R \cdot m_{t+\Delta t} \cdot T_{t+\Delta t}}{V} \tag{5.4}$$

where the subscripts i, j are the components before and after the volume under consideration. Finally, the net torque acting on the shaft is determined from the power imbalance between the compressor and the turbine, and given the shaft inertia, the acceleration of the engine is obtained. This new acceleration is used to calculate the new shaft speed for the next pass through the component calculations at the next time step. This process of calculation is carried out until the engine transient is completed. The procedure, which is a 'once through' calculation at each time step, requires a very short time intervals otherwise instabilities occur due the small values of the inter-component-volumes compared to large values of the air/gas mass flow rates.

5.3 Continuity of Mass Flow and Inter Component Volume comparison

By comparison, the method of continuity of mass flow (CMF) assumes that flow compatibility is maintained at all times even if the engine is performing a transient. This assumption is physically unrealistic. In fact, a mass storage in the engine occurs and cannot be ignored when the engine is operating at transient conditions. On the other hand, the method of inter-component-volumes (ICV) takes into consideration this effect since it includes allowance for a build up or diminution of mass flow in the various inter-component-volumes when the engine is running transiently. The CMF method is an iterative method and more computing time may be required, as several passes through the engine calculation are required to achieve the mass flow continuity, especially when the configuration of the engine is very complex and more iteration loops are required. The ICV method is a 'once through' method and no iterative loop is required to obtain the mass flow continuity. Nevertheless, short time increments are not necessary needed in the CMF method but this is a must in the ICV method to avoid instabilities. Although both methods yield to the same results when the engine is operating at steady-running performance, the predictions of the compressor trajectory lines by these two methods during transient operation are different during the first few instants of the transient but remain similar afterwards. When comparing the prediction of the transient compressor trajectory line of a single-shaft turbojet engine by using both methods, Fawke and Saravanamuttoo (1971b) showed that at the start of the acceleration, the method of continuity of mass flow predicted an instantaneous change in the compressor pressure ratio whilst remaining on the same constant non-dimensional speed line, whereas the ICV method predicted a more stable start to the transient with no discontinuities. This is illustrated in figure 5.2. Regarding the proximities of the predicted trajectories to the surge line, there was negligible difference. Nor was there significant difference in shaft speed or thrust responses. A comparison for a more complex engine configuration has been made by Maccallum (1989). This was for a two-spool turbofan engine with mixed exhausts. He found that the speed and thrust responses as predicted by the CMF method were about 4 per cent faster than those predicted by the ICV method during acceleration and deceleration performances. He also found that the trajectories in the H.P. compressor as predicted by the CMF method to have more severe departures from the steady-running line than those predicted by the ICV method. Computational times for the IGV procedure were however higher by a factor of 5 to 10, when compared with CMF, for this more complex engine.

It can be said that the ICV method is more realistic and its transient predictions are the most valid ones than the CMF method. This is because the mass storage in the inter-component-volumes are considered in the method. In conclusion, the ICV method can be used with confidence to simulate the transient performance of various aero and stationary gas turbine engines. A summary of the two methods is given below:

<u>The method of Continuity of Mass Flow</u>

1. Iterative loop is required to satisfy the mass flow continuity.
2. Mass storage in the inter-component-volumes of the engine is ignored.
3. More computing time may be required as the engine configuration becomes complex.
4. Small time increments are not necessary required.
5. Useful for large speed changes.

The method of Inter-component-volumes

1. No iterative loop is required.
2. Mass storage in the inter-component-volumes of the engine is considered.
3. Requires very short time increments.
4. Useful for large and small speed changes.
5. Computational time may be large, compared with CMF

5.4 Description of the model used for predicting Tay engine performances

The prediction of steady-state and transient performances of the Tay engine will be based upon the method of inter-component-volumes in the present investigation. The choice of this method has been considered on the basis that the ICV method is more realistic and gives a better representation of the transient behaviour of a gas turbine engine. Furthermore, the ICV method is the most widely used at the present time. The size of the inter-component-volumes used in the simulation model corresponds to the actual Tay engine volumes. By using the actual inter-component-volumes of the engine, the simulation model will be kept physically realistic.

As pointed out earlier in this chapter, accurate information on the transient performance of a gas turbine engine can be achieved if adequate components maps of the engine are known. Therefore, all components characteristics, fuel schedules, handling bleed valve schedules, bleed and cooling flows, inter-component-volumes, spools inertia, etc.., of the Tay engine have been supplied by the manufacturer of the engine. The characteristics provided for this engine include separate representations for the inner and outer sections of the fan and treated as two components. The outer fan delivers air to the bypass duct while the inner fan delivers air to the core of the engine. This may be a satisfactory way of treating the fan under steady running conditions but this procedure might be too rigid for transient performance, and some allowance is made for interchange of air mass flow between the inner fan and outer fan.

The allowance for the interchange of air mass flow between the inner and outer sections of the fan is achieved in two steps. Firstly, the characteristics initially provided are based on a frontal area split in the ratio 1 to 3 between the Inner Fan and Outer Fan. This is quantified by the parameter GEOM which represents the fraction of the total frontal flow area allocated to the Inner Fan in the initial characteristics. Thus, the parameter GEOM is assumed to have a value of 0.25. The second step to allow for interchange of air mass flow between components is to assume that the axial velocity of the air into the Fan is constant, at any instant, over the whole annular area. However, the fraction of this air passing through the Inner Fan is not necessarily equal to GEOM, but is some fraction of it. This fraction is named the fraction of split and labelled FCSP. An initial value for the factor of split, however, must be estimated at the start for the transient calculation. The value of this variable parameter is assumed to be unity in the simulation model, but re-calculated in the program at each time interval.

One computational problem arises when employing the inter-component-volumes method to the Inner Fan. The corresponding inlet air mass flow rate of the Inner Fan is not possible to determine for a specific pressure ratio of the Inner Fan when the linear interpolation is used due to the flatness of the Inner Fan characteristics. This means that different air mass flow rates can be read off the Inner Fan characteristics for a given pressure ratio of the Inner Fan, and the logic cannot decide which air mass flow rate is the correct one. The way to overcome this difficulty is that the air mass flow through the Outer Fan must be known. Once the inlet air mass flow rate of the Outer Fan is known, and in conjunction with GEOM and FCSP parameters, the air mass flow rate to the core of the engine can be obtained. Only six inter-component-volumes can be incorporated in the simulation for the steady-running and transient performances of the Tay engine. Seven inter-component-volumes could have been represented if it were possible to apply the input of pressure ratio to the Inner Fan. The position of these inter-component-volumes of the engine can be seen in figure 5.3. The sizes and guessed typical pressure values corresponding to these volumes are also specified in Appendix (II).

As altitude increases, the ratio of air density to absolute viscosity falls, and at certain altitude the Reynolds number will fall below a critical value of about 10^5, and the air flow will start to separate from the blades. Consequently, the compressor and turbine power will fall off resulting in a loss of thrust, since the air flow is not deflected as much as before, and the losses associated with the turbulent wake will cause the compressor and turbine efficiencies to diminish resulting in an increase of specific fuel consumption. Adjustments to compressor mass flow capacities and efficiencies are, therefore, incorporated in the simulation program to take into consideration Reynolds number effects. For the I.P. and H.P. compressors the simulation program uses the adjustments given in verses 24051 and 24001 for the I.P. compressor and 26051 and 26001 for the H.P. compressor.

The mass flow rates through the engine components which are obtained from the characteristics, are modified for the various cooling and bleed that exist within the engine. Cooling and bleed mass flow rates are specified as percentages of the total mass flow rate of the component with the exception of the H.P. compressor handling bleed valve. The magnitude of the H.P. compressor bleed flow is not a fixed percentage of the air flow in the engine core but varies accordingly to the non-dimensional HP spool speed. This Handling Bleed Valve schedule is illustrated in figure 6.4.

The magnitude of the step time increment in the simulation model is very important when predicting transient performances of a gas turbine engine. The simulation model based on the inter-component-volumes requires small time increments otherwise instabilities will occur leading to erroneous pressure values which in turn would adversely affect the mass flow rate calculation. Short step time increments of 0.1 to 0.5 milliseconds have been found by Maccallum (1984), to produce stable transients predictions of the Tay engine. The computation requires to use double precision to avoid oscillations due to the very short time increments. A short initialising time, e.g. from -0.2 to 0.0 second is required in which speed is held constant but gas pressures in the inter-component-volumes of the engine are allowed to adjust themselves from initial guesses. The block diagram for the calculation procedure is given in figure 5.4.

Chapter 6

Two-Spool Turbofan Engine - Variable Scheduling of IGVs and Bleed in H.P. Compressor

6.1 Adjustable inlet guide vanes

One means of achieving a wide speed range for a multi-stage axial flow compressor is to have a row of adjustable inlet guide vanes so that, by pivoting these in their sockets, the incoming air flow may be whirled in the direction of rotation and the relative velocity at the rotor blade is reduced. As the speed of the engine is built up, the inlet guide vanes are progressively closed to maintain an acceptable angle of the air flow entering the first row of rotor blades. The effect is to decrease the axial velocity and mass flow for a given rotational speed. This delays stalling of the first stages and the rear stages can swallow the air flow without choking. Almost all modern high performance gas turbine engines have variable inlet guide vanes and often from one to eight additional stages of variable stators are linked together and driven by a hydraulic ram via rings around the multistage H.P. compressor casings.

The use of inlet guide vanes is very beneficial to the multistage H.P. compressor and can prevent it from stalling and surging during transients. However, a great care should be taken in designing them and their operating range is also very important and must be carefully scheduled. On the other hand, their mal-scheduling during transient operations can be catastrophic to the operation safety of the compressor, and thus the performance of the engine.

6.2 Bleed valves

Another way of easing the problems of achieving a wide speed range of a compressor is to fit it with bleed valves. As the basic problem is that the air coming in at the front cannot get out at the back it is sensible to provide holes in the casing through which a proportion of the air can escape. The size and position of these holes are very important, and they have to be fitted with valves which shut them off as the engine runs up to speed and the compressor begins operating normally. These bleed valves can be arranged to open at reduced speed, which increases the mass flow through the front stages of the compressor, reducing the tendency to stall, and at the same time reduces the mass flow and pressure loss in the rear stages.

In practice, variable inlet guide vanes alone may be inadequate at very low speeds, while the use of bleed valves at medium speeds may lead to poor fuel consumption and high turbine entry temperatures. In consequence, it is normal to find both bleed valves and variable inlet guide vanes on modern high performance gas turbine engines.

6.3 Surge margin

There are many uncertainties concerned with stall and surge; uncertainties of prediction for a given design and uncertainties associated with the conditions of operation. The uncertainties of prediction are when a multistage axial compressor operates at some conditions with the stages mismatched so that some stages are at a non-dimensional flow rate below and some stages above that for which they were designed. The stages at the low flow rate will often have large regions of separated flow and would be unable to operate on their own without entering rotating stall. The coupling of stages together can maintain the compressor unstalled until the entire compressor crosses the criterion of instability. Uncertainties of operation include inlet flow distortion, which may be transient for example during aircraft manoeuvres, transient throttle changes, such as occur when a gas turbine engine is accelerated, transient geometry changes, such as tip and axial clearance changes following speed changes, and compressor mechanical damage including blade erosion and the effects of large foreign body ingestion.

To cope with these it is vital to increase the surge margin of the compressor in order that the steady operating line of the compressor does not intersect the surge line, thus removing the operating safety of the engine. The surge margin of a compressor is defined as a distance between the surge line and the running line, and the most common definition of the surge margin used is given by the relation below as:

$$SM = \frac{PR_S}{PR_W} - 1 \tag{6.1}$$

Where (PR_w) is the pressure ratio on the working line for a given non-dimensional rotational speed and (PR_s) is the pressure ratio on the surge line for the same mass flow rate as the condition on the working line.

According to this definition the non-dimensional speed will be higher for the points on the surge line than the working line. If operation is at a single non-dimensional speed it is more appropriate to define a surge margin in terms of the inlet mass flow on the working line and on the surge line for that one non-dimensional speed. The more logical definition considers the change in outlet flow function between the working line and the surge line for the same non-dimensional rotational speed. A suitable definition of surge margin would be:

$$SM = 1 - \frac{(F_{OUT})_S}{(F_{OUT})_W} \tag{6.2}$$

Where the outlet function is given by

$$F_{OUT} = \frac{\dot{m} \cdot \sqrt{T_{02}}}{P_{02}} \tag{6.3}$$

Compressor performance is usually quoted in terms of inlet non-dimensional flow rate using inlet variable, thus the outlet flow function can be written as:

$$\frac{\dot{m} \cdot \sqrt{T_{02}}}{P_{02}} = \frac{\dot{m} \cdot \sqrt{T_{01}}}{P_{01}} \cdot \frac{P_{01}}{P_{02}} \cdot \sqrt{\frac{T_{02}}{T_{01}}} \tag{6.4}$$

In fact, the temperature ratio of the compressor is much smaller than the pressure ratio. The temperature ratio, derived for a given polytropic efficiency, is expressed by the equation below as:

$$\frac{T_{02}}{T_{01}} = \left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\eta_P \cdot \gamma}} \tag{6.5}$$

It can be seen from the above equation (6.4) that the temperature ratio is further reduced to the half power in evaluating the flow function so the changes in temperature between the design and surge points can be neglected. Therefore, for stages of reasonably high efficiency

$$\left(\frac{P_{02}}{P_{01}}\right)^{\frac{1}{2}\frac{\gamma-1}{\eta_P \cdot \gamma}-1} \approx \left(\frac{P_{02}}{P_{01}}\right)^{-1} \tag{6.6}$$

Finally, a definition of surge margin which retains the physical significance of the outlet mass flow function but uses more easily measured inlet mass flow is:

$$SM = 1 - \frac{\left(P_{02}/P_{01}\right)_W}{\left(P_{02}/P_{01}\right)_S} \cdot \frac{\left(\dot{m} \cdot \sqrt{T_{01}}/P_{01}\right)_S}{\left(\dot{m} \cdot \sqrt{T_{01}}/P_{01}\right)_W} \tag{6.7}$$

For a given compressor the surge margin may be increased by lowering the working line; this means reducing the pressure rise in steady undistorted flow. In practice, this is unattractive because it is reducing the quantity most wanted from the compressor, namely pressure rise. Another alternative to increase the surge margin is to incorporate devices in the compressor to control the air flow.

6.4 Transient fuel scheduling

The control of aero gas turbine engines is of necessity a complex business, particularly when there are several of them and their performances have to be synchronised.

Because of the need to make every pound of metal justify its existence, aero gas turbine engines must work very close to their limits, and a system of control on the fuel flow must be provided to make sure the safety of the engine is maintained regardless of changing atmospheric conditions. The control system must ensure that the engine operates safely and is not endangered by surge in a compressor, and the critical operating limits of engine shaft speeds and turbine inlet temperature are never exceeded. The sensing of engine shaft speeds is by no means difficult and can be measured by sensing devices. On the other hand, it is very difficult to measure the turbine inlet temperature. It is not practicable to locate thermocouples at the inlet to the turbine because of the high temperatures of the gases involved at the turbine entry, but it is fairly simple to measure the high temperature in the jet pipe.

In high performance gas turbine engines, the multistage H.P. compressor is the most prone to surge at off-design conditions because of the number of the stages in the component and the difficulty of their matching. The situation is worse when the engine accelerates from ground idle speed to maximum speed or decelerates from cruise flight speed to ground speed. In order to avoid surge of the compressor during engine operations, a control system is required which controls the non-dimensional fuel flow according to a schedule based on one of the non-dimensional parameters. A fuel schedule which has been used in the present investigation relates a non-dimensional fuel group to the pressure ratio across the H.P. compressor and is expressed by the following relation as:

$$\frac{\dot{f}}{N_H \cdot P_{01}} = f\left(\frac{P_{02}}{P_{01}}\right) \tag{6.8}$$

where $(\dot{f})$ is the fuel flow rate, and (N_H), (P_{01}) and (P_{02}) are shaft speed and pressures at inlet and outlet of the H.P. compressor.

This type of control function has been achieved in the hydromechanical systems used in aero engines such as Rolls Royce Spey (Tay), RB211 (up to -524), and Pratt &

Whitney *JT8D/JT9D*. The fuel schedule is very important. The engine designer provides the control designer with information about the fuel flow required for steady running operation over the entire range of operating conditions, and the maximum fuel flow which can be used for transients without encountering surge or exceeding temperature limits. The designer provides the numbers for the Fuel flow which the controller then delivers. It is important to realise that if the geometry of the engine is fixed, the steady-running performance cannot be altered in any way by the control system. If, however, the engine has variable geometry, such as bleed valve and variable inlet guide vanes, the performance of the engine can be modified by including these devices to the control system.

The transients that have been used in the present study and in which the fuel flow is limited by the fuel schedule as described above, are acceleration and deceleration schedules as well as steady-running conditions. The engine which has been studied in this work is the Rolls-Royce 'Tay' Engine. This is a two spool turbofan of bypass ratio 3, having mixed exhausts. The starting speeds for the acceleration schedule are 1805 *rpm* for the LP shaft speed and 6030 *rpm* for the HP shaft speed. For the deceleration schedule, the starting speeds are 8387 *rpm* for the LP shaft speed and 12083 *rpm* for the HP shaft speed. A maximum and minimum fuel flow limits of 0.84 kg/s and 0.08 kg/s have been selected as the terminating conditions for both acceleration and deceleration, respectively, as well as for steady running conditions.

6.5 Design Engine performance

The output power from a gas turbine engine depends at its most basic level on the air flow through the power turbine. The temperature and pressure of this air are obviously very important, but to obtain a significant and sustained change in power it is necessary to increase the flow through the engine and therefore through the compressor. The relationship between flow and power is direct: if the temperatures and pressures remain equal, a small change in flow will give the same change in power. Thus the most important single factor in determining the ability to change power quickly is the ability to change the air mass flow quickly. Since flow depends largely on engine speed, the rate of change of power is primarily governed by the rate of change of flow with speed and the rate of change of speed with time. The former is governed by the flow-speed characteristic of the compressor and the latter by the spool inertia, the surge margin, and the temperature capability of the turbine. When a gas turbine engine is accelerated the fuel flow and thus the turbine inlet temperature is increased, thus providing an increase in turbine torque, resulting in an increase in spool speed and hence air flow. However, due to the inertia of the spool, instantaneously the speed will not change. Since the turbine is choked and its capacity measured as $\left(\dot{m}\sqrt{T}/P\right)$, is therefore fixed, the rise in turbine inlet temperature must be matched by either a fall in mass flow or a rise in compressor pressure ratio. Either results in a movement on the compressor characteristic towards surge. Thus, the surge margin limits the ability of the engine to accept a rapid increase in turbine inlet temperature, assuming that the limiting turbine inlet temperature is not reached first.

It is true to say that an engine with a poor surge margin will not accelerate well, but the determination of what is an acceptable surge margin will depend upon other aspects of the compressor characteristics, most notably the air mass flow-speed relationship. Most compressors with variable geometry exhibit a marked change in the slope of the air mass flow-speed line near the lower end of the operating range as illustrated in figure 6.1. The air flow-speed characteristic of the H.P. compressor of the Tay engine, under investigation, has the same pattern as in the figure, and is illustrated as the solid line in figure 6.2. (Also shown are the flow-speed lines for both 'Production' and 'Revised' schemes which are discussed later in sections 6.6 and 6.8, respectively). Since the surge margin in the compressor is very crucial for the efficient and stable operation of the gas turbine engine, it is very important to increase this margin so that the engine can operate safely and without encountering surge in the compressor during transients and steady-running conditions. Improvements to the surge margin can be made in several ways. A suitable scheduling of IGVs and bleed valves may also provide better transient performance of the engine.

The H.P. compressor of the Tay engine is fitted with both Inlet Guide Vanes and Handling Bleed Valve. The IGVs at the H.P. compressor are linked to the Handling Bleed Valve in the casing surrounding the seventh stage of the H.P. compressor by a control system. This control system, powered by a hydraulic actuator responding to non-dimensional H.P. compressor shaft speed signal, limits the mass of air supplied to the combustion chamber at low speeds, but reduces this control as the engine speed increases. In the Design Engine, the IGV schedule is set in the manner that the IGVs begin to turn in the non-dimensional speed range of 552 $\left(rev/\min K^{1/2}\right)$ from fully closed to become and remain fully open at the non-dimensional speed value of 586 $\left(rev/\min K^{1/2}\right)$. For the Handling Bleed schedule, the valve starts to close at the non-dimensional speed value of 549 $\left(rev/\min K^{1/2}\right)$ from full bleed to become and remain fully closed when the engine reaches its non-dimensional speed value of 568 $\left(rev/\min K^{1/2}\right)$. The numerical values are given in Table 6.1. When fully open, 14.7 per cent of the H.P. compressor inlet air mass flow can be bled from the seventh stage to the Bypass Duct. The IGV and Handling Bleed schedules are illustrated in figures 6.3 and 6.4, respectively. In this thesis, these IGV and Handling Bleed Valve schedules are labelled 'Design' schedules. The prediction of the transient performance of the 'Design' Engine is used as a reference for comparison with the performance of the engine when revised schedules are used for the IGVs and Handling Bleed Valve. The Bleed fraction in the 'Design' Engine is retained the same for all revised IGV and Handling Bleed schedules.

Predictions of steady-running and transient performances have been made for the Tay engine. The engine is of the 'Design' standard and the characteristics of the Fan, I.P. compressor, H.P. compressor, HP turbine, LP turbine, combustor, Mixer and Final Nozzle are those provided by Rolls-Royce. These characteristics are based on experimental test results. The prediction program used is the ICV program described in Chapter 5.

In the predictions reported in this thesis, the flight conditions were ISA day, sea level, Mach number 0.2 (a near take-off condition). Since all parameters, including fuel schedule, IGV schedule and Handling Bleed schedule are non-dimensional, predictions should read across to other flight conditions. The effects on transient performances of heat transfer are neglected, and only adiabatic case is considered, although the prediction program is capable to take into consideration of such effects. The performance of the engine during steady-running and transient operations is discussed below.

The steady-running line and transient trajectories in the different compressors of the 'Design' engine are illustrated in figures 6.5, 6.6, 6.7 and 6.8. The fuel schedules used in the acceleration and deceleration are those shown in figure 6.9.

Fan

The trajectory lines in the Outer Fan, as for the Inner Fan, during a rapid acceleration and deceleration are very close and very similar to the steady-running line, which as expected, lies between the transient trajectories as shown in figures 6.5 and 6.6. It can be seen that the sections of the Fan are not affected by the transients and the transient deviations from the steady-running line are insignificant. The reason is that the Outer Fan is feeding into a fixed exiting area so if compression effects in the Bypass Duct can be ignored one should expect the transient trajectories to coincide with steady-running line. Furthermore, the LP shaft speed accelerates/decelerates slowly due to high inertia of the LP shaft. For the Inner Fan, this slow acceleration/deceleration of the LP shaft effectively makes transient trajectories coincide with steady-running line. Thus, both Outer Fan and Inner Fan have large surge margins and are in no danger of encountering surge difficulties when the engine accelerates or decelerates.

I.P. compressor

The steady-running and transient working lines in the I.P. compressor are shown in figure 6.7. The transient trajectories in the I.P. compressor are by far the most interesting and they deviate from the steady-running line by substantial amounts. When the engine performs a rapid acceleration from a condition of minimum fuel to high fuel, the trajectory moves, after about 5 s, away below the steady-running line with no danger of surging the I.P. compressor. This is because the IGVs in the H.P. compressor begin to open after 4.9 s have elapsed of the acceleration time causing the air flow requirement of the H.P. compressor to increase more quickly than is available from the I.P. compressor. The I.P. compressor cannot immediately satisfy this increase demand, therefore, the pressure in the inter-component volume between the I.P. and H.P. compressors is reduced causing the pressure ratio across the I.P. compressor to fall, thereby meeting the non-dimensional air flow requirement of the H.P. compressor.

In the deceleration, the opposite situation occurs. The early start of closure of the IGVs (at 0.4 s) result in a rapid reduction in the non-dimensional mass flow rate into the H.P. compressor. This is slightly checked when the Handling Bleed Valve begins to open at 0.9 s. However the restricting effect of the IGVs movement dominates. The reduction in non-dimensional mass flow has to be matched by a rise in the pressure delivery of the I.P. compressor. Consequently, the pressure ratio across the I.P. compressor is higher. Thus, the transient trajectory departs from the steady-running line and moves towards the surge line of the compressor until the IGV schedule is completed. Hence the I.P. compressor has a poor surge margin.

It can be seen that the trajectory line in the I.P. compressor during a rapid deceleration is very critical, particularly at the start of the deceleration, and may endanger the transient performance of the engine. A simple way for avoiding the tendency of the I.P. compressor to surge during decelerations would be to employ a less severe deceleration fuel schedule. This is examined later in this Chapter, at Section 6.6. Another alternative to ease up the surge problem of the I.P. compressor is to bleed some air from the I.P. compressor delivery into the Bypass Duct. This is studied in the present investigation and the results of this effect are discussed in Chapter 7.

H.P. compressor

The steady-running and transient working lines in the H.P. compressor are shown in figure 6.8. The most striking feature in the H.P. compressor characteristics is the rapid increase in the pressure ratio across the compressor exhibiting a kink in the surge line. This kink had also been indicated in the predictions of Chapter 4 (Figs 4.23, 4.25, 4.27 and 4.29). The kink is associated with the partial ability of the earlier stages, when the compressor is running at lower speeds, to maintain a steady flow even when partially stalled (A. Stone, 1958). It was predicted in Chapter 4 (section 4.14) that moving the IGV schedule had a little effect on the surge line, but closing the compressor Bleed Valve could raise the surge line. Considering the transient condition, the transient trajectories in the H.P. compressor deviate from the steady-running line by significant amounts.

In the acceleration, progressively more fuel flow is admitted to the combustion chamber. This increase causes the temperature of the gases at entry to the H.P. turbine to rise. The non-dimensional flow capacity of the turbine is still roughly the same so the pressure at entry to the turbine has to rise, hence the pressure ratio of the compressor has to rise causing the trajectory to move closer to the surge line, particularly in the operating range of the Handling Bleed schedule. Consequently, the surge margin of the compressor is reduced.

In the deceleration, by the reverse of the argument above, the trajectory moves below the steady-running line. The air mass flow in the H.P. compressor is reduced as a result of the early closure of the IGVs. The trajectory thus presents no danger of surging the compressor until the termination of the transient.

Fuel flow control schedules

The fuel flow during the transient operations is controlled by a fuel schedule in which the non-dimensional fuel flow is scheduled as a function of the H.P compressor pressure ratio. This type of scheduling is used for both acceleration and deceleration, and compared with the non-dimensional steady running fuel flow values as shown in figure 6.9. The position of the accelerating fuel schedule dictates the proximity of the accelerating trajectory line to the surge line. If the fuel flow schedule line is raised the engine will accelerate more rapidly and the trajectory line in the H.P. compressor moves closer towards the surge line, and the opposite occurs if the fuel schedule line is lowered. This type of fuel control system is very important to guard the engine against surge during an acceleration and also to prevent a flameout during a rapid deceleration.

Shaft Speed, Thrust and Fuel Responses

Figures 6.13 and 6.17 show the fuel flow response when the engine is subjected to rapid acceleration and deceleration, respectively. It can be seen that the fuel flow line movement is very smooth during a deceleration, whereas a very sharp movement occurs when the engine accelerates. The sharp movement is caused by the late operation of the IGV and Handling Bleed schedules which rapidly pushes up the H.P. compressor pressure ratio.

The speeds of the LP and HP shafts and thrust responses of the engine during a rapid acceleration from an idle speed to maximum speed are shown in figures 6.10, 6.11 and 6.12, respectively. It can be seen that the thrust response of the engine is very slow at the start of the acceleration. The time required to reach 20 per cent of the maximum thrust is about 4.7 s. During this time, the Handling Bleed Valve is still open and the IGVs are still in their closing position, hence the thrust response of the engine is expected to be slow. When the Handling Bleed Valve begins to close the engine acceleration becomes more efficient. This causes more rapid increase in rotors speeds acceleration, hence fast thrust response of the engine. The engine reaches 90 per cent of the maximum thrust in a short time, at about 5.8 s of the acceleration time.

It should be noted that in the acceleration there is a temporary overspeed on the HP shaft before the engine stabilises. This is because, during the acceleration, the HP shaft speed leads the LP shaft due to the differences of shaft inertias. When the fuel flow reaches its limiting value of 0.84 Kg/s the shaft speed changes have to be redistributed to the new steady-running values. This requires a slight reduction of HP shaft speed.

For the deceleration, figures 6.14, 6.15 and 6.16 illustrate the shafts speeds and thrust response rates of the engine. It can be seen that the speeds and thrust responses of the engine are very smooth, following a smooth movement of the fuel flow during the transient. It is noted that the HP shaft decelerates ahead of the LP shaft, due to the relative inertias. Both decelerations are of course rapid. When the engine decelerates the operation of the IGVs and the Handling Bleed Valve is not as marked a feature on these speed and thrust responses as it is in the trajectories. The engine takes about 1.6 s to drop to 20 per cent of maximum thrust.

An important point to note is that the path of the thrust response rate of the engine is very identical to the path of the LP rotor speed during rapid acceleration and deceleration. Therefore, the thrust response is a function of the LP shaft speed.

A clearer confirmation of this relationship is seen by plotting the thrust to a base of, in turn, LP shaft speed and HP shaft speed. This is done in figures 6.18 and 6.19 respectively for the acceleration and 6.20 and 6.21 respectively for the deceleration. It can be seen that figures 6.18 and 6.20 are quite similar, and at, say, an LP shaft speed of 6100 rpm the thrusts are 6100 lbf in the acceleration and 5600 lbf in the deceleration. These values are quite close together. However, for the HP shaft plots, figures 6.19 and 6.21, these are quite dissimilar. For example, at an HP shaft speed of 11000 rpm, the thrust in the acceleration is 4400 lbf and 7200 lbf in the deceleration - values that are widely apart.

From the above discussion, it follows that, during the acceleration, the response of the engine is very slow. This is attributed to the late turning of the IGV schedule and the late closing of the Handling Bleed Valve on the H.P. compressor. At lower speeds, the H.P. compressor may encounter surge difficulties, particularly near the kink in the surge line of the H.P. compressor, if the engine performs a rapid acceleration. For the deceleration, there is a risk of the I.P. compressor to encounter surge especially at the start of the transient. A short deceleration followed by a rapid acceleration could endanger the trajectories in the both compressors, hence the transient performance of the engine. To alleviate these anomalies the air mass flow-speed line characteristic of the H.P. compressor needs therefore to be revised.

6.6 Modification of IGV and Bleed schedules to 'Production' engine specifications

In the 'Design' Engine, the major problem with the H.P. compressor of the engine is the kink in the surge line which hindered successful acceleration of the engine. The problem is associated with stalling of one or more of the early stages of the compressor and with the scheduling of the Handling Bleed Valve (Chapter 4, Section 4.4.2). The engine is fitted with both IGVs and Handling Bleed Valve. Near surge problems have been found in Prototype engines. It has been found (Nawrocki, 1989) that they can be relieved by adjusting the schedules of the IGVs and the Handling Bleed Valve. This also had been suggested previously by Maccallum (1984).

In this thesis it has been shown in Chapter 4 that altering the IGV and Handling Bleed schedules on the multistage H.P. axial flow compressor can be beneficial and help bring great improvements on the overall performance characteristics of the compressor. By considering this technique, it has, therefore, been decided to study the effect of changing the IGV and Handling Bleed scheduling of the Tay engine and thus modifying the actual air mass flow-speed characteristic of the H.P. compressor. The IGV schedule in the 'Design' Engine is altered from its 'Design' setting to operate over a wider range so the rate of change of air flow with non-dimensional speed is less steep. The Handling Bleed schedule is also altered from its 'Design' scheduling and moves along with the IGV schedule.

In the first part of this investigation, the IGV schedule is extended to operate at much lower non-dimensional speeds in which the IGVs start to open at the value of the non-dimensional speed of 493 $\left(rev/\min K^{1/2}\right)$ but finish their turning at the value of 593

$\left(rev/\min K^{1/2}\right)$. For the Handling Bleed schedule, this schedule is moved to an operating range of lower speeds. The Handling Bleed schedule operates in the non-dimensional speed range of 493 $\left(rev/\min K^{1/2}\right)$ from fully open to become and remain fully closed at the non-dimensional speed of 520 $\left(rev/\min K^{1/2}\right)$. The numerical values are given in Table 6.1. These changes are illustrated by chain-dotted lines in figures 6.3 and 6.4, respectively. Changes of this nature had been proposed by Maccallum (1984). Rolls-Royce indeed implemented the specific changes listed above in their production engines post 1985, although they did not quote the work of Maccallum as being the reason for the change. These altered schedules listed above are identified in this thesis as the 'Production' schedules.

The effects of these changes on the mass flow (steady-running)-speed characteristic of the H.P. compressor are shown in figure 6.2. For the 'Design' case they were taken as the mass flow at the appropriate pressure ratios for the 'Design' Engine, and for the 'Production' points they are taken from simulated line-ups based on the 'Production' Engine test data (Stoddart, 1991). It is seen that it is in the range where the IGVs are opening that the mass flow increases most rapidly when speed is raised. This rate of increase for the 'Design' Engine is at least three times as great as the rate of increase when the IGVs are stationary (either fully closed or fully open). Maccallum (1984) considered this as being at least a major contributor to the difficulties faced in the I.P. compressor in a deceleration. It also probably had an adverse effect on the surge line in the H.P. compressor, making accelerations having to be carefully controlled.

From the 'Production' Engine data, the actual changes in mass flow capacity are in the same direction as those predicted in Chapter 4 of this thesis, but almost double in magnitude those indicated in Equation 4.70. The appropriate equation for the actual engine behaviour is then

$$\frac{\Delta \dot{m}}{\dot{m}} = 0.0071 \cdot \Delta\alpha_1 \tag{6.9}$$

The actual engine changes are used in the investigation reported now. The movement of the IGV schedule, and also the Handling Bleed schedule, have their effects because they alter the pressure ratio - mass flow characteristics of the H.P. compressor. For the IGV movement, it was predicted in Chapter 4 that opening the IGVs had the effect of moving a constant speed $\left(N/\sqrt{T}\right)$ line to the right, i.e. to higher mass flow rate. Alternatively it could be regarded as the pressure ratio - mass flow points on an existing speed line being obtained at a lower non-dimensional speed. This new speed can be evaluated from diagrams such as figure 6.2. Where the Handling Bleed Valve remains either fully open or fully closed, it was predicted that the surge line was unaltered. In intermediate positions, the Handling Bleed Valve, for a compressor inlet mass flow, will be in a different position. This was predicted in Chapter 4 to alter the surge line, illustrated in figure 4.32. This alteration has to be incorporated in these intermediate positions. The alterations required when only the Handling Bleed schedule is changed can also be accounted for using the results such as shown in figure 4.32.

The characteristics, including surge line, of the 'Design' engine (used in figure 6.8) have been modified in the above ways to produce the characteristics shown in figures 6.25, 6.29 and 6.33.

The effects on transient performance of altering the scheduling of the IGVs and Handling Bleed Valve on the H.P. compressor are discussed below. The results are presented in a way that each state of scheduling is treated separately, i.e. by altering each 'Production' schedule and keeping the other 'Design', the contribution brought about on the transient performance of the engine is evaluated. The transient performance predictions of the 'Design' engine are also used for comparison with the performance of the 'Production' engines.

6.6.1 Engine performance with 'Production' IGV and 'Design' Bleed schedules

To evaluate the effect on the transient performance of altering only the IGV schedule from its 'Design' setting to the 'Production' setting, the Handling Bleed schedule is kept the same as in the 'Design' engine, (see Table 6.1). For the acceleration, the IGVs start to turn earlier before the beginning of closure of the Handling Bleed Valve. In the deceleration, the IGVs remain still closing after the Handling Bleed schedule has finished its opening. The effects on the performance of the engine of thus extending the IGV schedule to operate at lower non-dimensional speeds during the steady-running and transient conditions follow.

The steady-running and transient working lines in the various compressors of the engine when the engine is subjected to rapid acceleration, deceleration and steady operating conditions, are illustrated in figures 6.22, 6.23, 6.24 and 6.25.

Fan

The trajectories in the Outer Fan, as for the Inner Fan, during transients are very close and coincide with the steady-running line and, as expected, present no danger of surging as shown in figures 6.22 and 6.23, respectively, for the same reasons as explained in Section 6.5. Thus, adequate surge margins exist in the Fan. The working lines in the Fan when the IGV schedule is extended to operate at lower non-dimensional speeds alone are similar to those predicted in the 'Design' engine. It can, therefore, be said that extending the IGV schedule alone to include lower speeds has no effect on the trajectories in the Fan of the engine during the transients. However, there are significant changes in the trajectories, in both I.P. and H.P. compressors as shown in figures 6.24 and 6.25, respectively.

I.P. compressor

In the I.P. compressor, illustrated in figure 6.24, for the acceleration case, the IGVs in the H.P. compressor start to operate at lower non-dimensional speeds, beginning to open at $\left(N_H/\sqrt{T_{26}}\right)$ of 493, after 3.5 s have elapsed from the start of the acceleration of the engine. This permits an increase in air flow into the H.P. compressor. At that instant, the increase is relatively small, the Handling Bleed Valve is still open and 14.7 per cent of air flow is being bled from the seventh stage of the H.P. compressor. It will be seen later in figure 6.34 and 6.35 that at this stage, for an LP shaft speed, the HP shaft is rotating slightly slower than in the 'Design' engine. Thus, relatively, it now requires less mass flow, so the transient trajectory line in the I.P. compressor is slightly raised, and the departure from the steady-running line is less marked. It is worth saying that the effect on the acceleration trajectory in the I.P. compressor of moving alone the IGV schedule to a lower speed range is not so marked as long as the Handling Bleed schedule starts to operate late after the turning of the IGV schedule.

When the engine performs the deceleration, the opposite situation happens. As the engine starts to decelerate, the IGVs turn to close relatively slowly, at $\left(N_H/\sqrt{T_{26}}\right)$ of 593, at 0.4 s of deceleration time, and, in the new IGV turning range, the air flow in the H.P. compressor is higher compared to the 'Design' engine. As a result, the pressure delivery of the I.P. compressor is reduced to match the higher non-dimensional mass flow rate into the H.P. compressor. The pressure ratio across the I.P. compressor is thus dropped causing the trajectory in the I.P. compressor not to move so much towards the surge line. Hence the compressor has a greater surge margin than the 'Design' engine during the transient. It can be said therefore that extending the range of the IGV schedule from the 'Design' schedule to include lower non-dimensional speeds is beneficial and the engine may perform a good deceleration without endangering the trajectory in the I.P. compressor.

H.P. compressor

The predicted working lines in the H.P. compressor during steady-running and transient are illustrated in figure 6.25. The main interest in the H.P. compressor is that the characteristics at an $\left(N_H/\sqrt{T_{26}}\right)$of the compressor are altered and move to the right. The surge line is also modified, as explained above in Section 6.6. These movements are accompanied by an increase in the inlet non-dimensional flow rate in the compressor at a given constant speed line during the turning range of the IGVs. For example, an increase of about 15 and 8 per cent of air flow rate in the compressor have been obtained (from 'Production' Engine data) at the given non-dimensional speed values of 549 and 568, respectively. The trajectories are also affected, but relative to the steady-running line and surge line the effects are small.

When the engine starts to accelerate, the IGV schedule begins to turn earlier, at 3.5 s of the acceleration time, $\left(N_H/\sqrt{T_{26}}\right)$ of 493, permitting an increase in air flow in the H.P. compressor. Thus the non-dimensional mass flow rate in the compressor is slightly higher compared to the 'Design' engine. The rise in the air flow is matched by an increase in the pressure ratio across the H.P. compressor at a given speed. However at a given air flow, there is a decrease in the pressure ratio. It is seen that the trajectory in the vicinity of the kink in the surge line is slightly eased. At speeds higher than the IGV fully open position, the two engines have similar trajectories in this compressor.

For the deceleration, the IGVs begin to close much earlier, as with the 'Design' engine, at 0.4 s but they do not complete their turning until after 2.6 s have elapsed of the deceleration time, then inhibiting the air flow in the H.P. compressor. The pressure ratio across the H.P. compressor plotted, to a base of air flow, is thus reduced resulting in the trajectory to move much lower compared to the 'Design' engine. In the latter, the IGV schedule completes its operating range at 1.5 s of the deceleration time. Therefore, the air flow in the H.P. compressor is less than in the 'Production' engine during the operating range of the IGV schedule.

Shaft Speed, Thrust and Fuel Responses

The speeds of the LP and HP shafts, thrust and fuel response rates of the engine during a rapid acceleration are illustrated in figures 6.34, 6.35, 6.36, and 6.37, respectively, along with the predicted responses of the 'Design' engine. It should first be noted that, in the IGV turning range, the steady-running HP shaft speeds, at an LP shaft speed, are found to be lower than the corresponding values for the 'Design' engine. This is because of the higher air flow capacity of the H.P. compressor at an $\left(N_H/\sqrt{T_{26}}\right)$. When considering this acceleration, the HP shaft speed will lie lower than for the 'Design' engine. The closure of the Handling Bleed Valve is a function of $\left(N_H/\sqrt{T_{26}}\right)$ and this is reached later, starting at about 4.9 s as compared with 4.5 s for the 'Design' engine. This closure of the Handling Bleed Valve causes the rapid acceleration and rapid thrust increase, and thus these occur about 0.4 s later in this engine than in the 'Design' engine. The time required for this engine to reach 90 per cent of the maximum thrust is 6.3 s, 0.5 s slower than the 'Design' engine.

Figures 6.38, 6.39, 6.40 and 6.41 show the LP and HP shafts speeds, thrust and fuel responses of the engine during a rapid deceleration. It can be seen that the engine decelerates faster than the 'Design' engine. The decay of thrust response is slightly more rapid, dropping to 20 per cent of the maximum thrust in about 1.5 s, 0.1 s faster than the 'Design' engine. This is partly due, in the deceleration, the IGVs beginning to close slightly earlier - at 593 $\left(N_H/\sqrt{T_{26}}\right)$ as compared with 586 $\left(N_H/\sqrt{T_{26}}\right)$ for the 'Design' engine. Also, it was noted above that, in the vast bulk of the IGV turning range, at an LP shaft speed, the HP shaft speed under steady-running is slightly lower in this engine than in the 'Design' engine. Thus, in the deceleration, the opening point of the Handling Bleed Valve is reached slightly earlier (0.7 s, as against 0.9 s). This produces a 'braking' effect on this engine and the engine decelerates more rapidly.

The relationship of the thrust response and LP and HP shafts speeds of the engine are illustrated in figures 6.42 and 6.43, respectively for the acceleration, and 6.44 and 6.45, respectively for the deceleration, along with the predicted response of the 'Design' engine. It can be seen that the thrust response with the LP shaft speed of the engine 'Production' IGV coincide entirely with the 'Design' engine both for acceleration and deceleration, showing that the thrust is very closely a function of the LP shaft speed, but not of the HP shaft speed.

Summary

The effect of altering the IGV schedule alone from the 'Design' setting to the 'Production' scheme is beneficial with regard to the transient trajectories in the compressors. In the acceleration, the trajectory in the H.P. compressor keeps away from surge, including in the operating range of the IGV schedule. The thrust response of the engine in the acceleration is slower. For the deceleration, the trajectory in the I.P. compressor is less severe, hence the compressor has an improved surge margin than the 'Design' engine. The decay of the thrust response of the engine is slightly more rapid than the 'Design' engine. In general, significant improvement in performance is achieved, with the exception of the thrust response during the acceleration. The slow acceleration of the engine is attributed to the late closing of the Handling Bleed Valve. However, the thrust response of the engine may improve if the Handling Bleed schedule is moved to operate at lower non-dimensional speeds.

6.6.2 Engine performance with 'Design' IGV and 'Production' Bleed schedules

To examine the effect on the engine transient performance of altering the Handling Bleed schedule alone, the Handling Bleed schedule is moved from the 'Design' setting to the 'Production' setting. The IGV schedule is retained the same as in the 'Design' engine, (see Table 6.1). For the acceleration, the IGVs do not begin to turn until the Handling Bleed schedule is completed. In the deceleration, the Handling Bleed Valve begins to open late, after the IGVs finish their closing.. The effects on the steady-running and transient performance of moving the Handling Bleed schedule to operate at lower speed range are described below.

The working lines in the various compressors of the engine under steady-running conditions and during a rapid acceleration and deceleration are illustrated in figures 6.26, 6.27, 6.28 and 6.29.

Fan

The transient trajectories in the Outer Fan, as for the Inner Fan, are very close and similar to the steady-running line as shown in figures 6.26 and 6.27, respectively, for the same reasons as explained previously in Section 6.5. Both sections of the Fan have good surge margins, and similar to the 'Design' engine, and are in no danger of surging. Therefore, there is no effect on the trajectories in the Fan when altering the Handling Bleed schedule to the 'Production' scheme during the transients.

I.P. compressor

In the I.P. compressor, during the range where the positions of the Handling Bleed Valve differ in this engine from those in the 'Design' engine, the steady-running line and the transient trajectories are moved towards higher pressure ratios - figure 6.28 compared with 6.8. During this period of difference, when the Handling Bleed Valve in this engine is partially or totally closed, the air mass flow (non-dimensional), at $\left(N_H/\sqrt{T_{26}}\right)$, is reduced (Chapter 4). This leads to the working lines in the I.P. compressor being raised in this range, relative to the 'Design' engine. Outwith this range the steady-running and trajectory lines (acceleration and deceleration) are identical to those in the 'Design' engine. The shift of times of starting and finishing the closing of the Handling Bleed Valve can be illustrated by considering the acceleration. In this engine, the Handling Bleed Valve closing begins at 3.3 s and finishes at 4.0 s. The corresponding times in the 'Design' engine are 4.5 s and 4.8 s.

H.P compressor

The working lines in the H.P. compressor during the steady-running and transient operations are illustrated in figure 6.29. It can be observed that the characteristics of the H.P. compressor are altered and moved to the left when the Handling Bleed Valve is moved to operate at a lower $\left(N_H/\sqrt{T_{26}}\right)$ range (Chapter 4). The kink in the surge line is also moved to lower speeds following the operation of the Handling Bleed Valve. The surge line of the compressor is altered and raised significantly to the left during the transition from full bleed to zero bleed. This raise in the surge line has been discussed and explained in Chapter 4 (Section 4.14). The movement in the characteristics is equivalent to a decrease in the inlet non-dimensional mass flow rate in the compressor at a given constant speed line. For example, a decrease of about 2.4 and 4.6 per cent of the compressor inlet mass flow rate have been obtained (from

'Production' Engine data) at the given non-dimensional speeds of 500 and 549, respectively. The steady-running in the compressor is also affected by a such movement, and the kink in it moves to lower air mass flow rates. In the speed range when the Handling Bleed Valve is more closed, the pressure ratio of the compressor is higher for a particular value of the inlet mass flow rate causing the working line to move more closer to the surge line.

During the acceleration, the Handling Bleed Valve begins to close earlier at 3.3 s, at $\left(N_H/\sqrt{T_{26}}\right)$ of 493, and becomes fully closed at 4.0 s of the acceleration, at $\left(N_H/\sqrt{T_{26}}\right)$ of 520. During this time interval for the transition from full bleed to zero bleed, the air mass flow to the combustion chambers increases more rapidly. The nozzle guide vanes at inlet to the HP turbine are of almost constant capacity, being choked or nearly choked. Thus the pressure ratio across the H.P. compressor is pushed up. As a result, the trajectory runs closer to surge. It is seen that in this range in this engine there is a poor surge margin of the H.P. compressor.

During the deceleration, the Handling Bleed schedule does not begin to operate until late, after the IGVs have finished their closing. The Handling Bleed Valve begins to open at 3.5 s to become and remain fully open after 4.1 s of the deceleration. Consequently, a higher air mass flow exists through the H.P. compressor for a longer period of time during the transition from full bleed to zero bleed. Thus, the pressure ratio of the compressor is higher compared to the 'design' engine. It can be said that moving the Handling Bleed schedule to operate at a lower speed range causes the working lines to move towards surge, hence reducing the surge margin of the H.P. compressor.

Shaft Speed, Thrust and Fuel Responses

The speeds of the LP and HP shafts, thrust and fuel responses of the engine during the rapid acceleration and deceleration are shown in figures 6.34 to 6.37 and 6.38 to 6.41, respectively, alongside the predicted responses for the 'Design' engine. It can be seen that moving the Handling Bleed schedule to lower speeds leads to a faster acceleration but a slower deceleration compared to the 'Design' engine. During the rapid acceleration, the Handling Bleed schedule completes its operating range earlier at 4.0 s of the acceleration time, resulting in no more air flow being allowed to escape through the Valve, thus a more efficient engine acceleration. Hence there is a fast response of the engine. The engine reaches 90 per cent of the maximum thrust in 5.4 s, 0.4 s faster than the 'Design' engine. On the other hand, the loss of thrust takes longer in the deceleration, dropping to 20 per cent of the maximum thrust in about 1.7 s, 0.1 s longer than the 'Design' engine, as shown in figure 6.40. The reason for the slow deceleration of the engine is attributed to the late opening of the Handling Bleed Valve - The Handling Bleed 'Production' schedule does not begin to operate until 3.5 s have elapsed of the deceleration time, whereas the Handling Bleed Valve in the 'Design' engine is fully open after 2.0 s of the deceleration tine. Thus a higher air mass flow exists in the H.P. compressor for a longer period of time. The Handling Bleed schedule in this engine completes its operating range at about 4.1 s.

Summary

For the acceleration, the effect of altering the Handling Bleed schedule alone to the 'Production' scheme improves the thrust response of the engine but worsens the trajectory in the H.P. compressor, particularly during the movement of the Handling Bleed schedule. For the deceleration, the situation is worse. The transient trajectory in the I.P. compressor moves close to the surge line for a longer period, hence reducing the surge margin of the compressor from the start until near end of the transient. Also, the thrust response of the engine is slowed during the deceleration. Therefore, it is not beneficial to move the Handling Bleed schedule alone to operate at lower non-dimensional speeds. It seems improvements in the transient performance of the engine can be obtained only if both IGV and Handling Bleed schedules on the H.P. compressor are moved together.

6.6.3 Engine performance with 'Production' IGV and 'Production' Bleed schedules

The 'Production' engine has both IGV and Handling Bleed schedules at the 'Production' settings. The Handling Bleed Valve schedule operates at lower values of non-dimensional speed and the IGV schedule has been extended to include these values (figure 6.2), enabling the Handling Bleed Valve to operate at the early stage of IGV operation, similar to the 'Design' Engine.

It has been shown from the previous Sections 6.6.1 and 6.6.2 that extending the IGV schedule from its 'Design' schedule to an operating range of lower non-dimensional speeds leads to a slower acceleration and a faster response of the engine during a deceleration. On the other hand, an early Bleed scheduling leads to a faster acceleration and a slower deceleration of the engine. By considering this, the combination of the IGV and Handling Bleed schedules would suggest that the response of the engine would be maintained during the acceleration and deceleration of the engine. Therefore, it has been decided to operate both 'Production' IGV and 'Production' Handling Bleed schedules together as in the 'Production' Engine and examine how the engine would perform with such schedules (see Table 6.1). In the acceleration, both IGV and Handling Bleed schedules are set to turn earlier and simultaneously during the transient. For the deceleration, the Handling Bleed Valve does not open until late after the IGV schedule starts to close. The effects on the performance of the engine of the 'Production' IGV schedule and the 'Production' Handling Bleed Valve are discussed below.

The working lines in the various compressors when the engine is subjected to steady-running operations and during rapid acceleration and deceleration are illustrated in figures 6.30, 6.31, 6.32 and 6.33. These predictions have used the same transient fuel schedules as had been used with the 'Design' engine (figure 6.9).

Fan

It can be seen that the transient trajectories in the Outer Fan, as in the Inner Fan, are virtually close and similar to the steady-running line as shown in figures 6.30 and 6.31, respectively. The trajectories are identical to the 'Design' engine for the same reasons given in Section 6.5. Both sections of the Fan have large surge margins, and hence present no danger of encountering surge difficulties during the rapid acceleration or deceleration of the engine. Therefore, the effects of altering both IGV and Handling Bleed schedules to the lower speed range have no significant influence on the trajectories in the Fan during the transients.

I.P. compressor

It is perhaps simplest to compare the trajectories in the 'Production' engine with those in the 'Design' engine - figures 6.32 and 6.7

In the acceleration, for the 'Production' engine the downward movement of the trajectory in the I.P. compressor from the steady-running line begins earlier, due to the earlier start of opening of the IGVs (of the H.P. compressor), at $\left(N_H/\sqrt{T_{26}}\right)$ of 493, hence the earlier demand for additional air into the H.P. compressor. However since the subsequent rate of opening of the IGVs is less rapid in the 'Production' engine, the rate at which the air demand into the H.P. compressor rises is less severe, and so the trajectory in the I.P. compressor is not drawn so far below the steady-running line. In the range where the IGVs have completed their opening, the trajectories in the I.P. compressor for the two engines ('Production' and 'Design') are almost identical.

For the deceleration, the closing of the IGVs of the H.P. compressor starts at almost the same H.P. compressor $\left(N_H/\sqrt{T_{26}}\right)$. As discussed earlier, it is this closure of the IGVs in the deceleration that pushes the trajectory in the I.P. compressor upwards from the steady-running line and towards the surge line. The rate of closing the IGVs in the 'Production' engine is much more gradual than in the 'Design' engine and so the movement of the trajectory from the steady-running line in the I.P. compressor of the 'Production' engine is predicted to be significantly less than in the 'Design' engine. As the deceleration continues, the trajectories in both cases run almost parallel to the surge line and then return towards the steady-running line after the IGV and Handling Bleed schedules are completed.

The conclusion about the effect on the deceleration trajectory of how quickly the IGVs are scheduled to close is important. It means that, the tendency towards surge in the I.P. compressor can be eased by using a more gradual schedule for closing the IGVs of the H.P. compressor. This had been done in the development of the design from the 'Design' engine to the 'Production' Tay engine. It had previously suggested by Maccallum (1984) that this could be done, and that suggestion has been commented upon earlier in this Chapter, in Section 6.6.

H.P. compressor

The characteristics of the H.P. compressor have been adjusted for the altered IGV and Handling Bleed schedules, using the methods described in Sections 6.6.1 and 6.6.2. The predicted steady-running line and the predicted trajectories during rapid acceleration and deceleration of the 'Production' engine are shown in figure 6.33. These can be compared with the steady-running and transient working lines of the 'Design' engine's H.P compressor, shown in figure 6.8.

In the acceleration, for the 'Production' engine the IGVs begin to turn earlier, at the value of 493 $\left(N_H/\sqrt{T_{26}}\right)$, and not at such a fast rate, and the Handling Bleed Valve begins to close earlier. The modified IGV movement, particularly, eases the movement of the transient trajectory, in the IGVs turning range, from approaching so closely to surge, as compared with the 'Design' engine.

In the deceleration, the opposite effect occurs and the transient trajectory does not move so far below the steady-running line, as in the 'Design' engine. A higher air mass flow exists in the H.P. compressor due to the slow closing of the IGVs and the late opening of the Handling Bleed Valve (at 3.2 s of the deceleration time).

Shaft Speed, Thrust and Fuel Responses

The speeds of the LP and HP shafts, thrust and fuel response rates of the engine during the rapid acceleration and deceleration are illustrated in figures 6.34 to 6.37 and 6.38 to 6.41, respectively, and compared with the equivalent predictions of the 'Design' engine. The fuel flow in all the cases was controlled by the fuel schedule function shown in figure 6.9. It can be seen that the response of the 'Production' engine is faster during a rapid acceleration, whereas the response of the engine is slower during the deceleration, compared to the 'Design' engine. For the acceleration, there is early closure of the Handling Bleed Valve and early start to the turning of the IGV schedule, at 3.4 s of the acceleration time. As already indicated, in Section 6.6.2, closure of the Handling Bleed Valve in an acceleration allows the engine cycle to become more efficient, thus giving more rapid acceleration of the shafts. The engine attains 90 per cent of the maximum thrust in 5.5 s, 0.3 s faster than the 'Design' engine.

For the deceleration, the late operation of the Handling Bleed schedule and the slow closing of the IGV schedule cause a slower deceleration of the engine. The loss of thrust takes slightly longer, dropping to 20 per cent of thrust in about 1.8 s of the deceleration time, 0.2 s slower than the 'Design' engine. The Handling Bleed Valve in this engine does not begin to open until late, at 1.8 s, and completes its operating range at 3.2 s of the deceleration time, compared with 0.9 s and 2.1 s of the start of closure and full bleed, respectively, in the 'Design' engine. Therefore higher air mass flow exists in this engine for a longer period of time.

Figures 6.42, 6.43 and 6.44, 6.45 show the variation of the thrust response with speed of the LP and HP shafts during rapid acceleration and deceleration, respectively. They confirm the finding from the previous Sections 6.5, 6.6.1 and 6.6.2 that for a turbofan engine with a fixed nozzle the thrust is mainly a function of the LP spool speed only. Obviously, the thrust of a turbofan engine can be increased by increasing the total air flow through the engine.

Summary

The effects of altering the IGV and Handling Bleed schedules on the H.P. compressor from the 'Design' settings to the 'Production' IGV and 'Production' Handling Bleed schemes are beneficial and great improvements on the transient performance of the engine are achieved. For the acceleration, the trajectory in the H.P. compressor is very satisfactory. Improvement in the surge margin of the compressor is thus obtained particularly during the movement of the IGV turning range, in which the kink in the surge line of the compressor hindered a rapid acceleration of the 'Design' engine. The thrust response of the engine is now more rapid than the 'Design' engine. For the deceleration, the usage of the surge margin of the I.P. compressor is less, compared to 'Design' engine. The thrust response of the engine is, however, slightly less rapid during the deceleration. The engine takes 0.2 s longer to drop to 20 per cent of thrust than the 'Design' engine. This can still be satisfactory since the difference is very small.

6.7 Engine performance with 'Equivalent' Thrust Response

The Tay engine, as described earlier in the Chapter one, is a two-spool turbofan engine having mixed exhausts and powers different civil aircraft currently in operation. The engine produces a maximum take-off thrust of 13550 lbf at sea level ISA standards. The predictions of the transient performance of the engine carried out and reported to this stage had all used the same functional relationships between non-dimensional fuel flow and H.P. compressor pressure ratio for the fuel schedules which controlled the transients.

In the previous Sections 6.6.1 to 6.6.3, it has been shown that altering the IGV turning schedule and Handling Bleed schedule from their 'Design' schedules is advantageous with respect to the trajectory lines in the compressors during rapid acceleration and deceleration of the engine. The compressors have improved surge margins and the engine may perform the transients with no danger of encountering surge problems in the compressors. Unfortunately, the gain in the surge margins of the compressors may be obtained at the expense of the response rates of the engine. For example, extending the IGV schedule alone to operate at much lower non-dimensional speeds leads to a slower thrust response of the engine during a rapid acceleration. Also, by moving the Handling Bleed schedule alone to operate at these speeds the decay of thrust takes longer during a deceleration

Altering the IGV schedule and Handling Bleed schedule from their 'Design' schedules give different thrust responses, for both acceleration and deceleration. A logical move is to adjust the transient fuel schedules so that, for example, 'Equivalent' thrust responses are obtained. This would enable one to estimate the improvements brought about on the transient trajectories, thus surge margins, of the compressors. This adjustment has been done for all the 'Production' IGV and 'Production' Handling Bleed schedules described above. The transient fuel schedule is adjusted accordingly to the thrust response of the engine, i.e. more fuel is added to the combustion chambers if the thrust response of the engine is slower, or reducing the fuel flow rate if the thrust response of the engine is faster.

In the prediction program, the adjustment of fuel flow schedule is quantified by the parameter FCTFSC (FaCTor of Fuel SChedule). To obtain 'Equivalent' thrust rate responses, different runs of the engine with different values of this parameter have been tried until all the thrust response trajectories of the 'Production' IGV and 'Production' Handling Bleed schedules have the same thrust response trajectory as of the 'Design' Engine. These runs have been carried out for both acceleration and deceleration. the 'Equivalent' thrust responses obtained are shown in figures 6.46 and 6.47, respectively. The corresponding values of the parameter FCTFSC obtained during the transient runs are also shown on the figures. In the present investigation, these engines will be termed as engines with 'Equivalent' thrust response. The transient performances of these engines are compared with the transient performance of the 'Design' engine to enable one to examine the effect on the trajectories in the compressors of the engine during the transients.

The working lines under steady-running and during rapid acceleration and deceleration operations in the different compressors of the engine are illustrated in figures 6.48 to 6.59. The trajectories in the Outer Fan, as in the Inner Fan, are virtually close to the steady-running line, and are perfectly similar to the trajectories in the 'Design' engine, for all engines with individual and combined 'Production' IGV and 'Production' Handling Bleed schedules. The Fans still have large surge margins and present no danger of encountering surge difficulties during rapid acceleration and deceleration operations. It can be said that a forceful fuel schedule has no effect on the trajectories in the Fan of a two-spool turbofan engine during the transients for the same reasons as given in the section 6.5. The Fans need not be discussed further in this section.

6.7.1 'Production' IGV and 'Design' Bleed schedules

When altering the IGV schedule alone to the 'Production' scheme, the acceleration trajectory in the I.P. compressor stays close to the steady-running line before the IGV schedule starts to operate. The trajectory then moves away below the steady-running line once the IGVs start to turn until their complete opening. The trajectory in the I.P. compressor of the engine is almost identical to that of the 'Design' engine. In the H.P. compressor, the acceleration trajectory worsens. The trajectory hits the surge line of the compressor after the complete turning range of the IGV schedule, after 5 s have elapsed of the acceleration time. For this transient, the scheduled fuel flow rate in the engine is relatively higher, by 2.4 per cent, than in the 'Design' engine. The temperature at the entry to the HP turbine is, as expected, higher resulting in a very fast acceleration of the HP shaft. The temperature increase into the choked HP turbine and the speed increase of the HP shaft thus result in an increase in the pressure ratio across the H.P. compressor. The trajectory thus moves towards surge of the H.P. compressor.

For the deceleration, the trajectory in the I.P. compressor moves favourably less close to the surge line while the I.G.V schedule moves through its operating range, and then remains close to the steady-running until the termination of the transient. Hence, the compressor has a significantly improved surge margin usage compared to the 'Design' engine. In the H.P. compressor, a slight improvement in the trajectory line occurs during the turning of the IGV schedule. The reason is due to the existence of higher air mass flow in the H.P. compressor, and which is caused by the late closure of the IGV schedule, at 2.6 s of the deceleration, and relatively slow deceleration of the engine.

To summarise, the effect of altering the IGV schedule alone from the 'Design' schedule to the 'Production' schedule with a forceful fuel schedule endangers the trajectory in the H.P. compressor during the acceleration. In deceleration it improves the trajectory in the I.P. compressor, hence surge margin of the I.P. compressor.

6.7.2 'Design' IGV and 'Production' Bleed schedules

When the Handling Bleed schedule alone is changed to the 'Production' scheme, there is no significant alteration in the trajectory in the I.P. compressor during the acceleration. In the H.P. compressor, there is still a danger of surging the compressor in the transition from full to zero bleed during the acceleration. However, the trajectory moves slightly away from surge after the Handling Bleed schedule finishes its operating range until the end of the transient. This is because it is possible to use a value of 0.974 for the factor on the fuel schedule for this engine to give 'Equivalent' thrust response. This reduction of the scheduled fuel flow eases the trajectory in the H.P. compressor and it is no longer so close to the surge line. However it is noted that a danger of surge still remains around the speed where the Handling Bleed Valve just finishes closing. In this engine, the Handling Bleed schedule now completes its operating range at about 4.3 s of the acceleration time, 0.4 s earlier than on the 'Design' engine, due to the earlier scheduling of the Handling Bleed Valve.

For the deceleration, the trajectory in the I.P. compressor moves towards surge for a longer period until the operating range of the Handling Bleed schedule is reached.

During the transient, the Handling Bleed Valve does not open until late, after 3.8 s have elapsed of the deceleration time, 1.8 s slower compared to the 'Design' engine. The supply of air flow from the I.P. compressor to the H.P. compressor is thus higher for a longer period. As a result, the pressure ratio across the I.P. compressor is higher causing the trajectory to move towards surge until the Handling Bleed schedule completes its opening. The trajectory moves away from surge once the Handling Bleed Valve is fully open. No significant change occurs on the trajectory in the H.P. compressor.

To summarise, the effect of altering the Handling Bleed schedule alone to the 'Production' scheme with less fuel flow on the engine, therefore worsens the trajectories in both I.P and H.P compressors.

6.7.3 'Production' IGV and 'Production' Bleed schedules

When both IGV and Handling Bleed schedules are altered to the 'Production' schemes together, the acceleration fuel flow has to be reduced by 2.1 per cent for 'Equivalent' thrust response. The acceleration trajectory in the I.P. compressor stays near the steady-running line from the start until the complete turning of the IGV schedule, similar to the 'Design' engine. At this stage, the air flow requirement of the H.P. compressor is less due to the slower initial acceleration of the engine as a result of the reduction of fuel flow on the engine. The trajectory then moves away below the steady-running line, again similar to the 'Design' engine. In the H.P compressor, the acceleration trajectory moves favourably, not so close to surge until the termination of the transient, hence a reduction in surge margin usage of the compressor, particularly at the kink. This is partly due to the lower fuel flow, hence lower temperature at entry to the choked turbine, and partly due to the early opening of the IGVs and the early closure of the Handling Bleed Valve.

For the deceleration, the trajectory in the I.P. compressor moves less close to the surge line of the compressor until the IGV schedule finishes its operating speed range, hence less usage of surge margin of the I.P. compressor compared to the 'Design' engine. This is attributed to the gradual turning of the IGV schedule on the H.P. compressor. Therefore, higher air mass flow exists in the I.P. compressor for a longer period. As a result, the pressure ratio of the compressor is reduced causing the trajectory to move favourably, less close to surge, for a longer period compared to the 'Design' engine. However, this improvement is not as large as had been obtained when using the 'Production' IGV schedule with the 'Design' Handling Bleed schedule. There is no significant change on the trajectory in the H.P. compressor.

To summarise, it appears that altering both IGV and Handling Bleed schedules together from their 'Design' schedules to the 'Production' schemes is beneficial to trajectories in compressors. For the acceleration, the trajectory in the H.P. compressor is satisfactory and the surge margin of the H.P. compressor is much improved, particularly at lower speeds where the kink in the characteristics of the compressor is the main handicap for a good acceleration of the engine. For the deceleration, the trajectory in the I.P. compressor also moves favourably from the surge line of the compressor. The surge margin of the I.P. compressor is now improved. The usage of surge margin is being about 50 per cent as against 60 per cent for the 'Design' engine. Also, but if less importance, the alterations give a slightly lower fuel flows for the same thrust responses compared to the 'Design' engine, both in accelerations and decelerations.

6.7.4 Possible Future Development

In the current control systems used in the Tay engine, the IGVs and Handling Bleed Valve are inter-linked. The limits of their respective operating ranges are preset and these remain the same for both accelerations and decelerations. From the results shown above, using the 'Equivalent' thrust method, it is seen that the most favourable combination for an acceleration is 'Production' IGVs and 'Production' Bleed. For a deceleration the most desirable combination (to minimise surge risk in I.P. compressor) is 'Production' IGVs and 'Design' Bleed. (See Table 6.1 for $\left(N_H/\sqrt{T_{26}}\right)$ values).

It therefore would be very desirable to separate the controls to the IGVs from those to the Handling Bleed Valve. The IGVs should for all transients operate on the 'Production' schedule, i.e. between $\left(N_H/\sqrt{T_{26}}\right)$ values of 493 and 593. The Handling Bleed Valve should close in the $\left(N_H/\sqrt{T_{26}}\right)$ range 493 to 520 in an acceleration, and should open in the range 568 to 549 in a deceleration. This could be done with an Electronic Controller.

6.8 An alternative IGV and Bleed schedule scheme-'Revised'

As was discussed earlier, the surge problems of the H.P. compressor of a two-spool engine when running at low speeds, are established by the inability of the rear stages to pass the large volume of air flow causing the front stages of the compressor to operate in a stalled condition. At high speeds, the H.P. compressor can also encounter surge difficulties due to the stalling of the rears stages of the compressor. Stable operation for the H.P. compressor, hence surge free acceleration of the engine, is only possible if the behaviour of the rear stages is stabilised. Consequently, surge problems in the H.P. compressor impose a severe limitation on the range of operation of any engine, particularly at high pressure ratio where the H.P. compressor characteristics are steep and limited by choking at increased flow rates. Modern high performance aircraft gas turbine engines experience the same difficulties when running at off-design conditions.

The performance of the Tay engine is very much dependent upon the performance of the H.P. compressor of the engine. The 'Production' engines studied above have shown that altering the IGV schedule from the 'Design' setting to the 'Production' setting with its less steep rate of change, and altering the Handling Bleed schedule to move it to operate at lower values of non-dimensional speed, are helpful with regard to transient performance of the engine. To explore these effects more widely it has been decided to study the effect of modifying the flow-speed characteristic of the H.P. compressor by having the IGV schedule moved towards higher non-dimensional speeds with less steep rate of change as compared to 'Design'. The Handling Bleed schedule is also moved to operate at higher non-dimensional speeds. With the transient performances obtained for the 'Production' engines, such an investigation would be able to optimise the operating schedules of the engine and set rules which schedules for a two-spool turbofan engine could follow.

For the present investigation, the IGV schedule is extended from the 'Design' schedule to operate over a wider range of higher non-dimensional speeds. The IGVs are set to start to open progressively at the value of the non-dimensional speed above 549 $\left(rev/\min K^{1/2}\right)$ until they are in the fully open position for all speeds above the non-dimensional speed of 620 $\left(rev/\min K^{1/2}\right)$, (figure 6.3). The Handling Bleed Valve is also arranged to close progressively at the non-dimensional speed value of 559 $\left(rev/\min K^{1/2}\right)$, and remains fully closed beyond the value of the non-dimensional speed of 589 $\left(rev/\min K^{1/2}\right)$, (figure 6.4). These altered schedules are identified in this thesis as the 'Revised' schedules. These numerical values are given in Table 6.1.

The effects of these alterations on the steady-running air flow-compressor speed characteristic (H.P. compressor) are illustrated in figure 6.2. The overall pressure ratio-air mass flow characteristics of the H.P. compressor have been modified by the methods described in Chapter 4 and in Section 6.6 of this Chapter.

The effects on the engine transient performance of the 'Revised' IGV schedule and 'Revised' Handling Bleed schedule have been predicted. Again, each of the 'Revised' schedules is treated separately to assess the contribution brought about on the transient performance of the engine. The transient control fuel schedules used are the same as used for the 'Design' engine, with the Factor on the Fuel Schedule of unity (figure 6.9). It is found that the engine thrust responses in accelerations and decelerations for all the alternatives with 'Revised' schedules are almost identical to the responses of the

'Design' engine. It has therefore not been necessary to adjust the Factor on the Fuel Schedule as 'Equivalent' thrust conditions are already satisfied. The reason for the similarity in thrust responses is because the Handling Bleed schedule is always close, in $\left(N_H/\sqrt{T_{26}}\right)$, to the 'Design' values.

The transient performance of the 'Revised' engines is compared with that of the 'Design' engine.

6.8.1 Engine performance with 'Revised' IGV and 'Design' Bleed schedules

Changing only the IGV turning range from the 'Design' schedule to the 'Revised' setting will enable its effects on the transient performance of the engine to be examined. The IGV schedule has been extended from the 'Design' schedule to operate at higher speeds of the engine, (Table 6.1). The Handling Bleed schedule remained the same as in the 'Design' setting. During the progress of the acceleration, both IGV and Handling Bleed schedules start to operate at the same time but the IGVs do not finish their turning until the Handling Bleed schedule is first completed. For the deceleration, the IGVs start to close earlier and finish their turning just as the Handling Bleed Valve is fully open. The effects are discussed below.

The working lines in the various compressors of the engine under steady-running and during rapid acceleration and deceleration operations are illustrated in figures 6.60, 6.61, 6.62 and 6.63.

Fan

The transient trajectories in the Outer Fan, as in the Inner Fan, are virtually close and similar to the steady-running line as shown in figures 6.60 and 6.61, respectively. Both sections of the Fan have large surge margins, and hence they are in no danger of surging during the transients. The trajectories are, as expected, identical to those predicted in the 'Design' engine for the same reasons as given in Section 6.5.

Extending the IGV schedule to higher speed range has no significant effect on the trajectories in the Fan during rapid acceleration and deceleration of the engine.

I.P. compressor

In the I.P. compressor, the trajectories are displaced favourably away from the surge line of the compressor by substantial amounts as shown in figure 6.62. During the acceleration of this engine, the air mass flow requirement of the H.P. compressor is lower for a longer period compared to the 'Design' engine as a consequence of the late completion of opening of the IGVs in the H.P. compressor, at 6.6 s of the acceleration time, at $\left(N_H/\sqrt{T_{26}}\right)$ of 620. In the 'Design' engine, the IGVs opening is completed at 4.9 s of the acceleration time, at $\left(N_H/\sqrt{T_{26}}\right)$ of 586, 1.7 s earlier than in this 'Revised' engine. In this 'Revised' engine the rotation of the IGVs, and hence the rate of increase of air flow into the H.P. compressor, is less rapid than for the 'Design' engine, hence the acceleration trajectory is not, initially, pulled so far below the steady-running line. This causes the inter-shaft speed relationship to be slightly altered, the lead of the HP shaft in the 'Revised' engine being slightly more pronounced (figures 6.72 and 6.73). In the last part of the acceleration, when the IGVs in both 'Revised' and 'Design' engines are both more open, this slightly increased HP shaft speed lead causes the trajectory in the I.P. compressor to be slightly depressed.

When the engine decelerates, the more gradual rate of closing of the IGVs with its 'Revised' schedule means that the trajectory is not displaced so far above the steady-running line (figure 6.62 cf. Figure 6.7). Thus the danger of surge is eased - an improvement of more than 10 per cent of surge margin usage is obtained.

H.P. compressor

The working lines in the H.P. compressor during steady-running, rapid acceleration and rapid deceleration conditions are illustrated in figure 6.63. The main drawback of extending the IGVs to higher speed range is that the characteristics of the compressor are altered and the constant speed lines move to the left along the surge line in the operating range of the IGV schedule. The leftward shift in speed lines illustrates a decrease in the inlet mass flow rate and total pressure rise at a given $\left(N_H/\sqrt{T_{26}}\right)$. The loss in the inlet mass flow rate at the constant non-dimensional speed values of 568 and 589 are predicted to be about 4 and 11 per cent, respectively, of the expected inlet mass flow rate of the 'design' engine at the steady-running line. The trajectories are also affected and move to higher pressure ratios.

The acceleration trajectory is very critical. The trajectory starts to depart severely from the steady-running line at the start of the turning of the IGV schedule and crosses the surge line of the compressor until the IGV schedule finishes its operating speed range. This departure of the trajectory in this engine is more pronounced than in the case of the 'Design' engine (figure 6.8). This is because, at a particular non-dimensional mass flow into the H.P. compressor, in the IGVs turning range, the non-dimensional speed has to be higher in this engine than in the 'Design' engine. For the trajectories to be the same, then the pressure ratios across the H.P. compressor would be the same, conditions at the same non-dimensional mass flow being compared. Thus the fuel controller would indicate the same values of the group $\left(\dot{f}/N_H \cdot P_{26}\right)$. But N_H will be higher in this engine, so fuel flow will be higher, leading to higher turbine entry temperatures. Thermodynamic balance of flow capacity in the HP turbine nozzle NGVs will not have been achieved in this iteration. The compressor delivery pressure will have to rise in order to achieve convergence. Thus the acceleration trajectory in the H.P. compressor in this engine will be higher, i.e. nearer or crossing the surge line, compared to that in the 'Design' engine. A simple way to avoid the surge in the H.P. compressor is to employ a less severe acceleration fuel schedule, but equivalent thrust response performance will not then be achieved.

For the deceleration, the downward movement of the trajectory in the 'Revised' IGV schedule case is generally similar to the downward movement in the 'Design' engine, although there is a small departure in the region where the 'Revised' IGV schedule is more altered from the 'Design' schedule. The change there to the trajectory is due to the different rotational speed for a given air mass flow - the inverse of what had happened in the acceleration, discussed above.

Shaft Speed, Thrust and Fuel Responses

The speeds of the LP and HP shafts, thrust and fuel responses of the engine during the acceleration from a condition of minimum fuel to maximum fuel are shown in figures 6.72, 6.73, 6.74 and 6.75, respectively, alongside with the predicted responses of the 'Design' engine. The thrust response of this engine is marginally slower than the 'Design' engine. When the engine performs the acceleration, the IGVs do not complete their opening until late, at 6.6 s of the acceleration time, causing a less air flow in the H.P. compressor than in the 'Design' engine. Making a comparison at the same $\left(N_H/\sqrt{T_{26}}\right)$, when in this IGV turning range, the air flow into the H.P. compressor in this 'Revised' engine is less, as the constant speed lines are moved to lower mass flow rates. Thus the compressor pressure to meet the choking flow capacity at the HP turbine nozzle guide vanes is slightly reduced. Hence slightly lower acceleration. (Note that, at an $\left(N_H/\sqrt{T_{26}}\right)$, although the H.P. compressor pressure ratio is reduced, at a mass flow rate it is higher and can enter surge, as explained above). The engine reaches 90 per cent of the maximum thrust in 5.9 s, 0.1 s slower than the 'Design' engine.

It appears that the contribution brought about on the response rate of the engine when extending the IGV schedule to higher speed range is almost insignificant as long as the Handling Bleed schedule completes its operating range before the IGVs finish their turning. The Handling Bleed Valve becomes closed at 4.7 s, about the same time as in the 'Design' engine. Thus, there is only a very slight difference in the thrust response trajectory between this engine and the 'Design' engine.

It should be noted from figure 6.73 that the equilibrium HP shaft speed for this engine at the end of the acceleration is about 12300 rpm as compared with 12100 rpm for the 'Design' engine. This is because, at that steady fuel flow of $0.84\,Kg/s$, the

$\left(N_H/\sqrt{T_{26}}\right)$ is about 612 in the 'Design' engine and 614 in this 'Revised' IGV schedule engine. In the former case this point lies well outwith the IGVs turning range, and the IGVs are fully open. In the latter engine, however, the point lies inside the IGVs turning range, and the IGVs are thus not completely open. Thus, for a given LP shaft speed (and thrust), one would expect the HP shaft speed to be higher.

For the deceleration case, the responses of the engine, particularly in the latter part of the deceleration, are slightly slower compared to the 'Design' engine, as shown in figures 6.76, 6.77, 6.78 and 6.79. This is primarily because, in the earlier part of the deceleration, there is slightly less mass flow through the core of this 'Revised' engine, at $\left(N_H/\sqrt{T_{26}}\right)$, than through the core of the 'Design' engine. Thus the braking power (hence braking torque) available to decelerate the engine is slightly reduced. The loss of thrust takes slightly longer, about 1.8 s to drop to 20 per cent of thrust, 0.2 s slower than the 'Design' engine.

Figures 6.80, 6.81 and 6.82, 6.83 show the thrust response plotted against speed of the LP and HP shafts during rapid acceleration and rapid deceleration of the engine, respectively, along with those predicted in the 'Design' engine. It can be seen again that the thrust of the engine is dependent more on the LP shaft speed (figures 6.80 and 6.82 are similar) rather than the HP shaft speed (figures 6.81 and 6.83 are dissimilar) during the transients.

Summary

The effect of altering alone the IGV schedule to operate towards higher speed range is not favourable with regard to the performance of the engine during the acceleration. The response of the engine is slightly slower and the trajectory in the H.P. compressor worsens. There is a gain, however, in the surge margin of the I.P. compressor during the deceleration. The trajectory in the I.P. compressor stays closer with less departure from the steady-running line for a longer period compared to the 'Design' engine. This is attributed to the less steep mass flow-speed line of the H.P. compressor. The response of the engine is slightly slower during the deceleration. In general, there are no great improvements in the engine performance during the transients. The situation remains critical only in the H.P. compressor in which surge is predicted during the acceleration. However, the trajectory in the H.P. compressor may improve if the Handling Bleed schedule is also moved to operate at a higher speed range.

6.8.2 Engine performance with 'Design' IGV and 'Revised' Bleed schedules

The Handling Bleed schedule is moved from the 'Design' schedule to operate at higher speed range, shown in figure 6.4. The IGV schedule is kept unchanged, the same as the 'Design' schedule (see Table 6.1). The Handling Bleed operating schedule is thus almost similar to the operating range of the IGV schedule. The results of such alteration on the performance of engine are discussed below.

The working lines in the various compressors of the engine under steady-running and during transient operations are illustrated in figures 6.64, 6.65, 6.66 and 6.67.

Fan

The trajectories in the Outer Fan, as in the Inner Fan, are very close and similar to the steady-running line, as expected, and present no danger of surging - this is shown in figures 6.64 and 6.65, respectively. Both sections of the Fan have sufficient surge margins. The trajectories in the Fan of this engine are perfectly identical to those predicted in the 'Design' engine for the same reasons given in Section 6.5. Therefore, altering the Handling Bleed schedule to the 'Revised' schedule has no effect on the trajectories in the fan during the transients.

I.P. compressor

In the I.P. compressor, the trajectory in the maximum acceleration remains almost identical to that in the 'Design' engine. This is mainly because the time for the complete turning range of both IGV schedule and Handling Bleed schedule is almost similar as on the 'Design' engine.

In the deceleration, the Handling Bleed schedule begins to operate earlier, at 0.5 s of the acceleration time, at $\left(N_H/\sqrt{T_{26}}\right)$ of 589, against 0.9 s on the 'Design' engine, at the value of 568 $\left(N_H/\sqrt{T_{26}}\right)$. Thus, the non-dimensional mass flow rate in the H.P. compressor is slightly higher (see Chapter 4). This rise is matched by a drop in the pressure delivery of the I.P. compressor. Consequently, the trajectory moves slightly less towards the surge line of the compressor. Hence, the I.P. compressor has an improved surge margin usage compared to the 'Design' engine. An average decrease of about 3 per cent of usage of the surge margin of the I.P. compressor is noted during the critical period of the deceleration, and it is in this period that the opening of the Handling Bleed Valve has been rescheduled.

H.P. compressor

The working lines in the H.P. compressor during the steady-running and transient conditions are illustrated in figure 6.67. It can be seen that the characteristics of the compressor are altered. The constant speed lines move to the right in the $\left(N_H/\sqrt{T_{26}}\right)$ where the Handling Bleed Valve is still partially open, but had been closed with the 'Design' schedule. The surge line is also altered and lowered in this range. The rightward displacement of the speed lines illustrates an increase in the inlet air mass flow rate and pressure ratio of the compressor at a given constant speed line. A gain of about 2.1 and 1 per cent of the inlet air mass flow rate at the steady-running line of the compressor is predicted at the given $\left(N_H/\sqrt{T_{26}}\right)$ values of 560 and 576, respectively. Considering the trajectories, they are altered and move slightly downwards, particularly in the operating range of the Handling Bleed schedule.

In the acceleration, the Handling Bleed Valve does not begin to close until late, at $\left(N_H/\sqrt{T_{26}}\right)$ of 559. During the period of the movement of the Handling Bleed schedule, more air flow is allowed to escape from the H.P. compressor to the Bypass Duct. Also, as stated above, the constant speed line characteristics have been moved to the right when the Handling Bleed Valve is more open. Thus, at an $\left(N_H/\sqrt{T_{26}}\right)$, there is 1 or 2 per cent more air going into the H.P. compressor, but a typical bleed of 5 to 10 per cent more. So there is significantly less air going into the combustion chambers and hence to the HP turbine nozzles. These will be essentially choked. There may be some alteration in the temperature at turbine entry, but the lower mass flow will be the dominating influence, allowing the compressor delivery pressure to drop, thus lowering the transient trajectory in the H.P. compressor. However, the significant lowering of the surge line, mentioned above, means that the proximity of the trajectory to the surge is similar to that of the 'Design' engine.

During the deceleration of the engine, the earliest opening of the Handling Bleed Valve allows a dropping of the transient trajectory in the $\left(N_H/\sqrt{T_{26}}\right)$ range 580 to 550.

Shaft Speed, Thrust and Fuel Responses

The speeds of the LP and HP shafts, thrust and fuel responses of the engine during the rapid acceleration are illustrated in figures 6.72, 6.73, 6.74 and 6.75, respectively, along with the predicted responses of the 'Design' engine. It can be seen that the responses of this engine are very slightly slower compared to the 'Design' engine. During the acceleration, the late closure of the Handling Bleed Valve, at 4.8 s of the acceleration time, compared with 4.7 s for the 'Design' engine, causes more air flow to escape from the H.P. compressor. The engine is thus less efficient in achieving the excess shaft torques to enable acceleration. The thrust response of the engine takes slightly longer, about 5.9 s to obtain 90 per cent of the maximum thrust, less than 0.1 s longer than the 'Design' engine.

When the engine decelerates, altering the Handling Bleed schedule towards a higher speed range leads to marginally faster responses of the engine during the transient, as shown in figures 6.76, 6.77, 6.78 and 6.79. The Handling Bleed Valve begins to open earlier, at 0.5 s from the start of the deceleration, allowing more air flow to escape from the H.P. compressor. Hence the net torques on the shafts are more negative, resulting in a very slightly faster deceleration of the engine. The loss of thrust is very slightly faster, dropping to 20 per cent thrust in 1.5 s, less than 0.1 s faster compared to the 'Design' engine.

Figures 6.80, 6.81 and 6.82, 6.83 show the response of thrust plotted against the speed of the LP and HP shafts during rapid acceleration and deceleration, respectively, alongside with those in the 'Design' engine. Again, it can be seen that the thrust of the engine is primarily function of the LP rotor speed during the transients.

Summary

The effects on the performance of the engine of altering the Handling Bleed schedule alone to operate at a higher speed range are mixed. For the acceleration, the trajectory in the H.P. compressor slightly moves away from surge in the zone of transition from full to zero bleed but the response of the engine is slightly slower. In the deceleration, the trajectory in the I.P. compressor moves significantly less towards surge for a longer period in the operating range of the Handling Bleed schedule. The decay of the thrust is very slightly faster during the transient.

It may be helpful to alter both the IGV schedule and the Handling Bleed schedule to the 'Revised' schemes. This is now examined.

6.8.3 Engine performance with 'Revised' IGV and 'Revised' Bleed schedules

The results obtained in the previous Sections 6.8.1 and 6.8.2 have showed that moving the IGV schedule to more gradual rate of turning, and in a high speed range, aids the surge margin in the I.P. compressor during a deceleration but worsens the surge situation in the H.P. compressor in an acceleration. It also slows up the thrust responses in both transients. Moving the Handling Bleed schedule to operate at higher speeds slightly improves the surge tendency in the H.P. compressor in an acceleration. It was therefore decided to study how the engine would perform with both IGV and

Handling Bleed schedules modified as described above - (IGV, 549 t0 620 $\left(N_H/\sqrt{T_{26}}\right)$; Handling Bleed, 559 to 589 $\left(N_H/\sqrt{T_{26}}\right)$, see Table 6.1).

The working lines in the compressors when the engine is under steady-running and transient conditions are illustrated in figures 6.68, 6.69, 6.70 and 6.71.

Fan

The trajectories in the Outer Fan, as in the Inner Fan, are very close and similar to the steady-running line, and present no danger of surging, as shown in figures 6.68 and 6.69, respectively. Thus, both sections of the Fan have adequate surge margins, similar to the 'Design' engine, for the same reasons as given in Section 6.5. Therefore, altering both IGV and Handling Bleed schedules to a higher speed range has no effect on the trajectories in the Fan during the rapid acceleration and deceleration of the engine.

I.P. compressor

In the I.P. compressor, the trajectories are moved downwards (slightly to lower pressure ratios) - relative to the 'Design' engine both in the acceleration and deceleration of the engine, as shown in figure 6.70. In the acceleration, it will be seen later that the HP shaft responds as with the 'Design' engine, whereas the LP shaft lags the 'Design' engine by about 0.1 s, (figures 6.72, 6.73). This influences the acceleration trajectory in the I.P. compressor particularly in the IGVs turning range. At the start of the acceleration, the trajectory stays close to the steady-running line before the turning of the IGV schedule, but it then moves further away in the operating range of the IGV schedule. The IGV schedule starts to operate at $\left(N_H/\sqrt{T_{26}}\right)$ of 549, at 4.5 s, and complete its turning late at $\left(N_H/\sqrt{T_{26}}\right)$ of 620, after a time of 6.7 s has elapsed of the acceleration time. Compared to the 'Design' engine, at an LP shaft speed the air flow requirement of the H.P. compressor is higher due to the higher H.P. shaft speed. The I.P. compressor cannot immediately satisfy this high demand. Thus, the pressure rise across the I.P. compressor is reduced. Hence the trajectory departs severely away from the steady-running line until the near end of the transient, more so than had been the case for the 'Design' engine.

During the deceleration, the air flow in the I.P. compressor is slightly higher than the 'Design' engine because of the late closure of the IGVs, at 2.0 s of the deceleration time. This high air flow in the compressor is thus matched by a decrease in the pressure ratio across the compressor at a given non-dimensional speed. Hence the trajectory moves further away from surge and remains much closer to the steady-running line of the compressor for a longer period of time. The I.P. compressor therefore has a greater surge margin and presents no danger of surging during the transient. An average of more than 8 per cent increase of surge margin of the compressor is obtained during the deceleration. It appears that the operation of the IGV schedule towards a less steep rate of movement, extending to a higher speed range is dominant for a longer period both in the acceleration and deceleration.

H.P. compressor

The working lines in the H.P. compressor during steady-running and rapid acceleration, and deceleration conditions are illustrated in figure 6.71. For the acceleration, the trajectory moves dangerously towards the surge. The first danger point is at the $\left(N_H/\sqrt{T_{26}}\right)$ value of 560, at the kink, where the Handling Bleed Valve is just beginning to close. The proximity in this zone is very similar to that for the 'Design' engine, the only difference is that the proximity at the kink now extends to a higher $\left(N_H/\sqrt{T_{26}}\right)$ value - 560, as compared with 549 for the 'Design' engine. For the next part of the acceleration, the trajectory runs along the surge line, or just enters surge. This movement towards and into surge is more marked than for the 'Design' engine (figure 6.8), but not quite as marked as for the engine with the 'Revised' IGV schedule but 'Design' Handling Bleed schedule (figure 6.63). The reason for the trajectory movement being large in the latter case was explained in Section 6.8.1 above. The slight easing of this trajectory for the present case is associated with the engine having spent longer in the acceleration with the Handling Bleed Valve open, and the consequent slight alterations of the two shaft speeds.

During the deceleration, the trajectory follows fairly closely the trajectory in the 'Design' engine. The only slight difference is during the period when the Handling Bleed Valve is more open. - then the compressor pressure ratio, and trajectory, are lower.

Shaft Speed, Thrust and Fuel Responses

The shaft speeds, thrust and fuel response rates of the engine during the acceleration from ground idle speed to maximum speed are illustrated in figures 6.72, 6.73, 6.74 and 6.75, respectively, and compared with those predicted in the 'Design' engine. During the transient, the response rates of the engine are slightly slower by less than 0.2 s, compared to the 'Design' engine. The explanations for this are similar to those given in Section 6.8.1 for the engine with the 'Revised' IGV schedule but 'Design' Handling Bleed schedule. These were based on this engine's lower core mass flow rate, relative to the 'Design' engine, at an $\left(N_H/\sqrt{T_{26}}\right)$, hence lower H.P. compressor pressure ratio, hence lower fuel flow. There is the additional factor that the Handling Bleed Valve closes later in the acceleration, at 4.8 s of the acceleration time. This further slightly delays the acceleration response. The time required for the engine to attain 90 per cent of the maximum thrust is 5.9 s, 0.1 s slower than the 'Design' engine.

For the deceleration case, the responses of the engine are slower than those in the 'Design' engine from the start to the end of the transient as shown in figures 6.76, 6.77, 6.78 and 6.79. The reason is attributed to the fact that, in the earlier part of the deceleration, the core air flow in this engine, is less than in the 'Design' engine, due to the earlier closing of the IGVs. Thus the 'braking' effect of the H.P. compressor is reduced. This explanation is as had been given earlier in Section 6.8.1 for the engine with the 'Revised' IGV schedule but 'Design' Handling Bleed schedule. The engine takes slightly longer about 1.8 s to drop to 20 per cent of thrust, 0.2 s slower than the 'Design' engine.

Figures 6.80, 6.81 and 6.82, 6.83 show the response of thrust plotted against speed of the LP and HP shafts during the rapid acceleration and deceleration, respectively, along with those of the 'Design' engine. They confirm the finding from the previous Sections that the thrust is dependent primarily on the speed of the LP shaft of the engine.

Summary

Generally, altering the IGV and Handling Bleed schedules together to the 'Revised' schemes brings no great improvements on the performance of the engine during the transients. For the acceleration, the response of the engine is slightly slower and the trajectory in the H.P. compressor worsens in the turning of the IGV schedule. In the deceleration, the response of the engine is slower. The loss of thrust takes slightly longer than in the 'Design' engine. However, an improvement in the deceleration trajectory in the I.P. compressor is obtained. The compressor has a greater surge margin for a longer period compared to the 'Design' engine. This is attributed to a less steep flow-speed line on the H.P. compressor in the IGV turning, and also the earlier opening of the H.P. compressor Handling Bleed Valve. By considering these results, it is not beneficial to alter the IGV schedule and the Handling Bleed schedule together towards a higher speed range.

6.8.4 Possible Future Development

It was previously pointed out (Section 6.7.4) that there could be scope for improvement if the controls to the IGVs and the Handling Bleed Valve were separated, and could differ in a deceleration from an acceleration. From studying the results of the studies of the 'Design' schedules, the 'Production' schedules and the 'Revised' schedules, it is considered that, for an acceleration, the best schemes would be 'Production' schedule for IGVs, i.e. 493 to 593 $\left(N_H/\sqrt{T_{26}}\right)$, and 'Production' schedule for Handling Bleed Valve, i.e. 493 to 520 $\left(N_H/\sqrt{T_{26}}\right)$. For a deceleration, the most favourable schemes would be 'Production' schedule (as above) for IGVs and 'Revised' schedule for Handling Bleed Valve, i.e.559 to 589 $\left(N_H/\sqrt{T_{26}}\right)$.

Chapter 7

Two-Spool Turbofan Engine - Use of Bleed from I.P. Compressor Delivery

7.1 Introduction

It is known that the concept of a two shaft compressor system is to improve the surge problem. In the simplest terms it can be said that by having two independent compressors, with only aerodynamic coupling, the surge line of each compressor should present no great problems. However, when operating in series driven by two turbines in series it can be shown that if the turbines operate with choked nozzle guide vanes and the final nozzle is also choked the compressors are thermodynamically locked and the operating line of the low pressure compressor tends to converge rapidly on the surge line as the engine is throttled back at high forward speeds. In the Tay engine, the two-spool arrangement comprises a H.P. compressor driven by a two-stage HP turbine and a much larger L.P. fan and Intermediate Pressure (I.P.) compressor driven by a three-stage LP turbine via a shaft passing down the centre of the tubular HP shaft.

The main anxiety in the engine is that the I.P. compressor has a difficult trajectory during decelerations, i.e. the trajectory in the I.P. compressor moves towards the surge line when the engine decelerates. This is attributed to the very steep schedule for turning the IGVs, as had been previously discussed in Chapter 6. However, in the interest of safety of the engine, it is clearly desirable to keep the trajectory in the I.P. compressor as far as possible from the surge line during decelerations. One method of relieving the tendency of the I.P. compressor to surge during decelerations is to bleed some fraction of air at the exit of the I.P. compressor, thus giving a more favourable slope to the transient trajectory in the compressor. Therefore, it has been suggested to provide the engine with a transient bleed between the I.P. and H.P. compressors.

7.2 Engine performance with I.P. compressor bleed

To examine any performance contribution brought about by the inter component bleed on the transient performance of the engine, a fraction of 10 per cent of air bleed at the exit from the I.P. compressor delivery has been selected to bypass the core of the engine into the Bypass Duct. At high speed end of the running range, the speed line characteristics in the I.P. compressor of the engine are almost vertical. By bleeding a small amount of air flow at the exit from the I.P. compressor, a reduction in pressure ratio can be produced and hence the surge margin of the I.P. compressor can be improved. It is hoped that this bleed fraction of delivery air will be adequate to enable one to assess its effect on the transient performance of the engine and would also contribute to relieve the tendency of the I.P. compressor from surge, thus a good transient performance of the engine during decelerations. In the present investigation, the IGV schedule and the Handling Bleed schedule are kept unchanged, as in the 'Design' engine. The acceleration and deceleration fuel flow schedules are also the same as in the 'Design' engine. The performance of the engine with the I.P. compressor bleed are discussed below.

The working lines in the different compressors of the engine under steady-running and during the rapid acceleration and deceleration operations are shown in figures 7.1, 7.2, 7.3 and 7.4.

Fan

The trajectories in the Outer Fan, as in the Inner Fan, follow the steady-running line very closely, as expected, both in the acceleration and deceleration, as shown in figures 7.1 and 7.2, respectively. The trajectories are similar to the 'Design' engine. Both sections of the Fan have adequate surge margins and present no danger of encountering surge difficulties during the transients, for the same reasons as given previously in Chapter 6, (Section 6.5). Therefore, bleeding 10 per cent of the air from the I.P.

compressor delivery has no effect on the trajectories in the Fan, either in the acceleration or deceleration.

I.P. compressor

The working lines in the I.P. compressor are illustrated in figure 7.3. The main feature in the I.P. compressor is that the steady-running line and the transient trajectories drop significantly below the corresponding working lines for the 'Design' engine, and hence away from the surge line of the compressor. This is because of the now less resistant route which is available to the air leaving the I.P. compressor.

In the acceleration, the trajectory departs away by a substantial amount below the steady-running line in the operating range of the IGV schedule, but it then moves closer to the steady-running line towards the end of the transient. The effect of the 10 per cent of the air bleeding from the I.P. compressor delivery indeed causes the air flow requirement from the I.P. compressor to be much higher than for the 'Design' engine. The I.P. compressor cannot immediately supply the increase demand of air flow to satisfy the requirement of air flow of the H.P. compressor. Thus, the pressure ratio of the I.P. compressor is much reduced. Hence, the trajectory in the I.P. compressor departs severely below the steady-running line until the IGV schedule completes its turning.

For the deceleration, the non-dimensional mass flow rate of the I.P. compressor is increased, by 10 per cent, compared to the 'Design' engine, at a given H.P. compressor $\left(N_H/\sqrt{T_{26}}\right)$. The non-dimensional mass flow rate at inlet to the I.P. compressor will be almost unchanged, hence the dimensional mass flow rate will be about the same. To achieve the correct match of $\left(\dot{m}_{IP}\cdot\sqrt{T_{26}}/P_{26}\right)$, the pressure $\left(P_{26}\right)$ drops significantly. Therefore, the I.P. compressor deceleration trajectory moves less close to the surge line. An increase of about 20 per cent of the 'Design' engine surge margin is obtained during the transient. Thus, the I.P. compressor has a larger surge margin than in the 'Design' engine, and hence presents no danger of surging during the deceleration. The tendency of the I.P. compressor to surge in decelerations is thus greatly relieved by bleeding some air from the I.P. compressor delivery into the Bypass Duct of the engine. It should be noted that the deceleration trajectory stays slightly away from the steady-running line at the start of the transient and before the turning of the IGV schedule. The reason is due to the mismatch between the HP and LP spools.

H.P. compressor

The working lines in the H.P. compressor under steady-running and during the rapid acceleration and deceleration conditions are illustrated in figure 7.4. In the acceleration, it can be seen that the influence of the 10 per cent air bleed from the I.P. compressor delivery on the trajectory is not noticeable until the Handling Bleed schedule completes its operating speed range, at $\left(N_H/\sqrt{T_{26}}\right)$ value of 568, at 5.0 s of the acceleration. After that time, the trajectory moves slightly nearer the surge line of the H.P. compressor in the remainder of the turning of the IGV schedule. This is attributed to the reduction in the pressure delivery of the I.P. compressor. Thus, the pressure ratio across the H.P. compressor is slightly higher than in the 'Design' engine, until the IGV schedule completes its turning (fully open). For the deceleration, the trajectory is virtually unaffected due to the fast deceleration of the engine.

Shaft Speed, Thrust and Fuel Responses

The speeds of the LP and HP shafts, thrust and fuel responses of this engine with IP bleed during a rapid acceleration from a condition of minimum fuel to maximum fuel are illustrated in figures 7.6, 7.7, 7.8, and 7.9, respectively, alongside with those predicted in the 'Design' engine. It can be seen that the effect of bleeding 10 per cent of the air at the exit of the I.P. compressor, leads, as expected, to a slower response of the engine than for the 'Design' engine. The Handling Bleed Valve in the H.P. compressor of this engine does not close until late, at 5.0 s, and the IGVs do not finish their turning until 5.2 s of the acceleration time, resulting in less air flow in the H.P. compressor compared to the 'Design' engine. Consequently, a less fuel is admitted into the combustion chambers, resulting in a lesser increase in the temperatures at the entry to turbines. The net torque provided by the turbines is reduced causing a slower acceleration of the shafts, and hence a slower response of the engine. The maximum thrust developed by the engine is, 1.7 per cent less, at the same fuel flow, than the 'Design' engine due to the effect of I.P. compressor bleed. This is due simply to the engine working on a less efficient cycle, energy being dissipated in bleeding air from the I.P. compressor delivery into the Bypass Duct. The engine takes about 0.7 s longer to attain 90 per cent of maximum thrust, which it reaches in about 6.5 s.

It should be noted that in the acceleration there is an overspeed on the HP shaft when the fuel flow stabilises at the end of the transient, the HP shaft speed then being 12700 rpm as compared with 12200 rpm for the engine without the IP bleed. The reason for this is that, at the maximum fuel flow of 0.84 kg/s, the I.P. compressor pressure ratio is about 1.67 for the case with no IP bleed, but this drops to about 1.4 when there is the 10 per cent bleed. There is little change in the mass flow through the I.P. compressor. Thus, considering the non-dimensional mass flow group at entry to the H.P. compressor, the numerator drops by 10 per cent, but the denominator drops by about 20 per cent (neglecting temperature change effects, which are to the power one half). Thus the H.P. compressor has to take in a non-dimensional mass flow which is about 10 per cent higher, hence the HP shaft speed has to be higher.

In the deceleration, the response rate trajectories of the engine are, as expected, more rapid than those predicted in the 'Design' engine, as shown in figures 7.10, 7.11, 7.12 and 7.13. The transient air bleed at the exit from the I.P. compressor effectively causes the IGVs on the H.P. compressor to finish their closing earlier, at 1.5 s and the Handling Bleed Valve to close earlier at 2 s of the deceleration time, resulting in air flow in the core of the engine to decrease more rapidly, and hence a faster deceleration of the engine. The decay of thrust is slightly more rapid than the 'Design' engine, dropping to 20 per cent of thrust in about 1.5 s, as compared with 1.6 s for the 'Design' engine.

Figures 7.14, 7.15 and 7.16, 7.17 show the thrust response plotted against speed of the LP and HP shafts, respectively, for a rapid acceleration, and, respectively, for a rapid deceleration, along with those predicted in the 'Design' engine. It can be seen that the thrust of the engine is again principally dependent on the speed of the LP shaft, both in acceleration and deceleration.

Summary

The effect of bleeding 10 per cent of the air from the I.P. compressor delivery into the Bypass Duct is very beneficial very regard to the working line in the I.P. compressor. The working line in the H.P. compressor is virtually unaffected. In deceleration, the trajectory in the I.P. compressor drops favourably away from surge by a substantial amount, hence the I.P. compressor has an improved surge margin. The tendency of the I.P. compressor to surge in a deceleration is thus relieved. Also, the effect leads to a fast deceleration, the decay of thrust is slightly more rapid dropping to 20 per cent of thrust in about 1.5 s, 0.1 s faster than with zero IP bleed. However, the 10 per cent air bleed is not favourable in the acceleration. The engine is less efficient and the acceleration response of the engine is slow. The engine takes longer, in about 6.5 s, to reach 90 per of the maximum thrust, as against 4.8 s with zero IP bleed.

The use of a transient air bleed at the exit from the I.P. compressor during an acceleration, is not a practical means for three reasons. First, a transient air bleed slows the acceleration of the engine, and which is undesirable. Second, the acceleration trajectory in the I.P. compressor is well displaced away from the surge line and the risk of surge is thus far removed. Finally, air bleeding itself will involve a waste of work of the turbine, and hence a poor fuel consumption of the engine. Therefore, one would suggest that the transient IP bleed would be used in decelerations only.

Chapter 8

Two-Spool Turbofan Engine - Use of Variable Throat Area of the Final Nozzle

8.1 Introduction

It has been stated earlier that the transient operations of a high performance aero gas turbine engine are difficult, and the compressors may encounter surge or stall difficulties when the engine accelerates or decelerates. At steady-running conditions, the steady-running line of the compressors of the engine may move towards surge at lower or at higher speeds and the surge margin on the compressor is reduced. The situation worsens when the engine performs a critical transient, such as a rapid acceleration from ground idle speed to maximum speed or a deceleration followed by a rapid acceleration. It is clear, therefore, that the surge margin of the compressor is very crucial and presents a major problem with regard to engine acceleration and off-design performance of the engine. To obtain satisfactory operations of the engine during transients and steady running conditions it is, however, very important to increase the surge margin of the compressor. Large values of this margin permits large increase in fuel flow and, therefore, rapid acceleration of the engine. Among other possible methods of improving the surge margin in the compressor is to use variable propulsive nozzle area of the engine.

The use of variable area jet nozzle can alleviate the tendency of the compressor(s) to surge by permitting the compressor to operate at a lower pressure ratio at a given speed, hence lowering the working line of the compressor resulting in an increase in the surge margin of the compressor. When investigating the effects on transient trajectories of nozzle area schedules of a twin-spool turbojet engine, Fawke and Saravanamuttoo (1971a) have showed that the steady-running line in the H.P. compressor can be improved by opening the nozzle area during accelerations, hence improving the surge margin of the H.P. compressor, while the surge margin in the L.P.

compressor can be improved by closing the nozzle area during decelerations. When Maccallum (1973) has investigated the effects of bulk heat transfer on the compressor surge lines and on the steady running conditions of a two-spool bypass engine, he found that the raising of the steady-running line due to heat transfer reduced the surge margin in the L.P. compressor by 18 per cent during altitude decelerations and 4 per cent during sea level decelerations. In the H.P. compressor the surge margin was reduced by 35 per cent when attempting an acceleration immediately following a rapid deceleration. He suggested that the situation in the L.P. compressor during the decelerations could be improved by using a variable area final nozzle with the nozzle area reduced during deceleration. Therefore, the propelling nozzle size is extremely important and must be carefully designed to obtain the correct balance of pressure, temperature and thrust. It should be recognised that if the area of the nozzle is too small, the jet pressure will rise, the total air flow will fall and the L.P. compressor moves towards surge. If the area is too large, the reverse will happen. It is sometimes advantageous for thrust to schedule the nozzle too small in cases where the higher pressure wins out over the lower air flow. This is known as over-restoration and is practised at low nozzle pressure ratio. At high nozzle pressure ratio $(M = 2.0)$ it is better to open the nozzle because air flow wins. This is called under-restoration.

Because of the proximity of the trajectory in the H.P. compressor of the Tay engine to surge during the acceleration particularly near the kink in the compressor surge line, which tends to hinder a rapid acceleration of the engine, and also the movement of the transient trajectory in the I.P. compressor towards surge during deceleration, it has been decided to investigate the effect on the transient performance of the engine of altering the Final Nozzle area of the engine. This means by opening or closing the Final Nozzle of the engine. The Design area of the Final Nozzle of the engine is $0.6541\,m^2$. To open the engine nozzle, a fraction of 10 per cent increase in the area of the Final Nozzle is selected to allow more gas flow to be discharged to the atmosphere without any build up in pressure to upset the running of the engine. The same fraction is used for reducing the Design area of the Final Nozzle of the engine.

8.2 Transient engine performance

By altering the geometry of the Final Nozzle of the engine, it is hoped that the fraction of 10 per cent increase/decrease in the Final Nozzle area of the engine would be adequate to demonstrate the effects of varying the nozzle area on the performance of the engine during the steady-running and during transient conditions. It is also hoped that encountering surge difficulties in the compressors can be avoided, particularly at low speeds and at high speeds, thus a better performance of the engine can be achieved during the steady-running and transient operations. For this investigation, both IGV and Handling Bleed schedules on the H.P. compressor are kept similar to the Design engine. The fuel flow rate in the engine is also limited to the value of 0.70 kg/s as the terminating condition. The results of the performance of the engine with the variable nozzle area are presented below.

The working lines in the various compressors of the engine under the steady-running and during the transient operations for different Final Nozzle areas are shown in figures 8.1 to 8.4 and 8.6 to 8.9.

Fan

The trajectories in the Outer Fan, as in the Inner Fan, follow the steady-running line very closely both in the accelerations and decelerations, as illustrated in figures 8.1, 8.2, respectively, for the Final Nozzle open, and 8.6, 8.7, respectively, for the Final Nozzle shut. The trajectories in both sections of the Fan are far removed from the surge lines when the Nozzle is open, hence a good surge margin of the Fan. By contrast, the steady-running line and the transient trajectories in both sections of the Fan lie very much nearer the surge lines when the nozzle is closed, thus reducing the surge margin of the Fan.

These movements can be explained by considering the case when the Final Nozzle is opened. This allows, at an LP shaft speed, a reduction in the pressure ratio required for the Outer Fan. Thus the working line moves away from surge. For the Inner Fan, the restriction on the core engine air flow caused by the Final Nozzle is reduced. Thus there is a higher air mass flow through the Fan and the working line moves to the right, along the constant speed line, and away from surge.

The opposite situation occurs for the steady state and transient performances when the Nozzle is closed.

I.P. compressor

The working lines in the I.P. compressor during steady-running and transient operations both for open and closed Nozzles are shown in figures 8.3 and 8.8, respectively. It can be seen that the effect of having the Nozzle open results in the working lines lying nearer the surge line of the compressor, i.e. at higher pressure ratios. The reasons for this are (a) the pressure rise required across the Outer Fan, for the same LP shaft speed (and $\left(N_L/\sqrt{T_{24}}\right)$), is reduced due to the easing of the restriction caused by the Final Nozzle and (b) there will now be a larger pressure drop across the LP turbine, due again to the lesser restriction of the Final Nozzle. Assuming for the moment that the LP shaft speed remains, thus the Fan absorbs less power, and the LP turbine delivers more power, so the LP shaft will stabilises at a higher speed. Thus the speed of LP shaft, for the same fuel flow, is raised, as seen in figure 8.11 from 7700 rpm to 8500 rpm when the Final Nozzle is opened by 10 per cent (fuel flow remaining constant at $0.70\ kg/s$). With this higher I.P. compressor shaft speed, but virtually the same H.P. compressor speed (figure 8.12) and $\left(N_H/\sqrt{T_{26}}\right)$ the pressure ratio across the I.P. compressor has to be raised, i.e. a movement of the steady-running line towards the surge line. Thus the transient trajectories in the I.P. compressor will also be moved to higher pressure ratios. Only the deceleration is critical, but is obviously now in much more danger of surge.

However, there is a great advantage to be gained in the deceleration trajectory by closing the Nozzle during the transient. The reasons being the inverse of those given above when the Final Nozzle is opened. The compressor then has a greater surge margin.

It is worth noting that the surge margin of the Fan is improved at the expense of the surge margin of the I.P. compressor when the engine is performing a steady-running or a transient operation with the Nozzle open. The opposite occurs when the Nozzle of the engine is closed.

H.P. compressor

The working lines in the H.P. compressor under the steady-running and during the rapid acceleration and deceleration conditions, both for open and closed Final Nozzles are illustrated in figures 8.4 and 8.9, respectively. It can be seen that opening or closing the Final Nozzle of the engine has no significant effect on the working lines in the H.P. compressor during the steady-running and transient operations as long as the LP turbine is choked. Furthermore, any effect resulting from variation of Nozzle area will directly influence the LP turbine, and hence the LP shaft. The HP turbine, however, is separated from the Final Nozzle by the LP turbine and, if the LP turbine is choked, the HP shaft is shielded from disturbances caused by the variable nozzle.

It is worth noting that the non-dimensional speed of the H.P. compressor at higher values of the non-dimensional mass flow rate of the H.P. compressor is reduced when the Final Nozzle is open. This is because of the high rise temperature of the I.P. compressor, and which is resulted from the high speed of the LP shaft of the engine. The opposite situation occurs when the nozzle is closed.

Shaft Speed, Thrust and Fuel Responses

The shafts speeds, thrust and fuel responses of the engine during the rapid acceleration for the different Final Nozzle throat areas are illustrated in figures 8.11, 8.12, 8.13 and 8.14, respectively, and compared with those predicted in the 'Design' engine. The transient responses of the 'Design' engine are also predicted to that range of fuel flow rates. It can be seen that the effect of having the Final Nozzle open leads to a faster acceleration of the engine. The engine, however, accelerates slower when the Final Nozzle is closed.

Increasing the Final Nozzle throat area causes an increase in the net torque of the LP turbine because of the redistribution of pressure ratio between the LP turbine and the Final Nozzle. The air flow in the engine is increased more rapidly because of the improved LP shaft speed. The increase of air flow permits a more rapid increase in fuel in the combustion chambers, and hence the engine performs a faster acceleration. It is to be noted that the thrusts obtained at the ends of the accelerations in both increased and decreased Final Nozzle areas, and at the same fuel flows of $0.70\,kg/s$, are slightly below (1 to 2 per cent) the thrust for the 'Design engine case. It is considered that, for the case with the reduced Final Nozzle area, this is due to the reduction in air flow rate more than compensating for the increased jet velocity. However, when the Final Nozzle area is increased, while the opposite effects do occur, there is also an increased loss of power in the increased power dissipated in the transmission inefficiency. It should be noted that, as before, there are slight overspeeds on the HP shaft when the fuel flow stabilises.

The responses of the LP and HP shafts speeds, thrust and fuel of the engines during the decelerations for the different Final Nozzle areas are shown in figures 8.15, 8.16, 8.17 and 8.18, respectively. It is seen that the decelerations are very rapid in all cases. There may appear a slight speed in the predictions, but this is partly due to the initial speeds from which the decelerations started being the same numerical values in all cases, and not the stabilised values. The choice of the same shaft speeds was adopted so that in each case there was the same kinetic energy in the system. It will also be noted that there are slight, but not significant, differences in the stabilised shaft speeds at the end of the transients.

The variation of the response of thrust with speeds of the LP and HP shafts for different Final Nozzle areas of the engine during a rapid acceleration and deceleration are shown in figures 8.19, 8.20, 8.21 and 8.22, respectively. A most interesting feature of using a variable Final Nozzle is that it permits selection of air flow and thrust independently, or LP rotor speed and fuel flow.

Summary

Probably the greatest benefit for the transient behaviour of the engine is to reduce the area of the Final Nozzle by an amount during a deceleration. This eases the trend towards surge in the I.P. compressor, but tends to push the trajectory closer to surge in the Outer Fan. A reduction by 5 per cent in the area of the Final Nozzle is probably a good compromise.

Chapter 9

Conclusions and Suggestions for Further Work

9.1 Conclusions

The prediction effects on the performance of varying the geometry of a two-spool turbofan engine (with common exhaust stream) under steady-running and during rapid accelerations and decelerations operations have been carried out in the present investigation. The variable geometry features that have been examined on how better transient performance of the engine can be achieved are:

1. Variable scheduling of the Inlet Guide Vanes (IGVs) of the H.P. compressor.
2. Variable scheduling of the Handling Bleed on the H.P. compressor.
3. Introduction of Bleed from the air delivered from I.P. compressor.
4. Variable Final Nozzle throat area.

To assist the understanding of the first two of the above features, a row by row programme has been used for the prediction of the performance of the axial-flow H.P. compressor.

9.2 Predictions of H.P. compressor characteristics

From the above mentioned compressor programme it was predicted that when IGVs are opened, the constant speed lines on the pressure ratio - air mass flow characteristics are moved to higher air mass flows and the surge line is almost unaltered. The predicted effects of closing the Handling Bleed Valve are that the constant speed lines are moved slightly to lower inlet mass flows, and with a significantly higher surge line. This effect, as compressor speed is raised, is a major contributor to the kink in the compressor surge line. The peak efficiency of the compressor for a given rotational speed approaches its maximum value at a pressure ratio just below its maximum pressure, and at a mass flow slightly lower than the surge point mass flow.

9.3 Engine Predictions - Effects of changes in IGV schedule

The H.P. compressor characteristics used in this work were modified for IGV and Handling Bleed changes in terms of the results reported above, scaled to match observed changes in engine performance. The critical transients were the acceleration - possibility of surge in H.P. compressor - and deceleration - possibility of surge in I.P. compressor. The transient trajectories in both Inner and Outer sections of the Fan always lay close to the steady running lines, and so no danger of surge. Two rates of change of IGV angle with $\left(N_H/\sqrt{T}_{26}\right)$ were examined, covering three non-dimensional speed ranges. It was found that the more gradual rate of change of IGV angle considerably eased the trajectory in the I.P. compressor in the deceleration. However it slightly worsens the acceleration trajectory in the H.P. compressor (but see section 9.5 below). The lowest of the non-dimensional speed ranges suited the engine best - $\left(N_H/\sqrt{T}_{26}\right)$ 493 - 593.

These comparisons were made using fuel schedules adjusted to give 'Equivalent' rates of thrust response.

9.4 Engine Predictions - Effects of changing Handling Bleed schedule

Three H.P. compressor Handling Bleed ranges were considered: 493 to 520 $\left(N_H/\sqrt{T}_{26}\right)$, 549 to 568 $\left(N_H/\sqrt{T}_{26}\right)$ and 559 to 589 $\left(N/\sqrt{T}\right)$. In the acceleration, the trajectory in the H.P. compressor is improved when the Handling Bleed Valve operates over the lowest speed range (using 'Equivalent' thrust comparison). In the deceleration, the trajectory in the I.P. compressor is helped when the Handling Bleed Valve operates over the highest speed range. To summarise, the Handling Bleed Valve should operate at the earliest range in the transient, whether acceleration or deceleration.

9.5 Engine Predictions - Combination of changes to IGV and Bleed schedules

For the acceleration, the lowest speed range closing of the Handling Bleed Valve and the lowest speed range for turning the IGVs gives the best trajectory in the H.P. compressor (IGVs 493 to 593 $\left(N_H/\sqrt{T}_{26}\right)$ and Handling Bleed Valve 493 to 520 $\left(N_H/\sqrt{T}_{26}\right)$). This in fact is the combination used by the 'Production' Engine (see Table 6.1).

For the deceleration, the important factors are a gradual rate of turning of the H.P. compressor IGVs (493 to 593 $\left(N_H/\sqrt{T}_{26}\right)$) and an early opening (in the deceleration) of the Handling Bleed Valve in the H.P. compressor (559 to 589 $\left(N/\sqrt{T}\right)$). This set of schedules is the best for avoidance of surge in the I.P. compressor.

In many engine designs, both the IGV movement and the Bleed Valve movement are actuated in common by a hydromechanical control system operating on an $\left(N_H/\sqrt{T}_{26}\right)$ sensor. In such schemes it would be very difficult to have different schedules in accelerations and decelerations for the IGVs and the Handling Bleed Valve. However, if an Electronic Controller is used, then much more flexible arrangements of schedules can be employed. It is recommended that with such an Electronic Controller, the combinations of scheduling reported above be used.

9.6 Engine Predictions - Use of IP compressor Delivery Bleed

This is helpful in decelerations in relieving the trajectory movement in the I.P. compressor from approaching surge. A bleed of 10 per cent from the I.P. compressor Delivery typically increases the unused surge margin in the I.P. compressor in a deceleration from 33 per cent to 80 per cent. The air bleed is of no help in an acceleration and actually it delays the thrust response.

9.7 Engine Predictions - Alteration to Final Nozzle area

Opening the Final Nozzle lowers the working line in the Fan but raises it in the I.P. compressor. The reverse effects occur if the Final Nozzle is reduced in area. There is a negligible effect on the working line in the H.P. compressor (engine working with choked HP and LP turbines).

In an acceleration, it therefore is worth opening the Final Nozzle, the trajectory in the I.P. compressor otherwise being below the normal steady-running line. The opening of the Final Nozzle shortens the acceleration time, without risking surge in any compressor. Typically, a 5 per cent increase in the Final Nozzle throat area reduces the acceleration by about 0.2 s in 5.6 s.

In a deceleration, if there already is adequate surge margin in the Fan, it could be helpful to close the Final Nozzle slightly. While this raises the trajectory in the Fan, it would ease the trajectory in the I.P. compressor. There could also be a very slight reduction in the deceleration time, although this may be negligible. In the engine considered in this study, a 5 per cent reduction in Final Nozzle area would reduce the surge margin of the Fan to about 70 per cent of its value with the 'Design' Nozzle, but ease the usage of surge margin in the I.P. compressor from about 60 per cent to about 50 per cent. The time for deceleration to 20 per cent thrust was reduced from 1.6 s to 1.5 s.

9.8 Deductions from Conclusions

The major problem areas encountered in transient operations of two-spool turbofan aircraft gas turbines are:

(a) Risk of surge in H.P. compressors during accelerations.

(b) Risk of surge in I.P. compressors during decelerations.

The first problem above, case (a), can be tackled by designing to have reasonable placing of the steady-running line relative to the surge line, by using a reasonable acceleration fuel schedule and by having the H.P. compressor Handling Bleed Valve(s) closing reasonably early (i.e. at lower $\left(N_H/\sqrt{T_{26}}\right)$) in the acceleration. This problem has been encountered in engines of all configurations for many years, and has been resolved by the methods mentioned above (additionally, for a single-spool engine, opening the Final Nozzle at lower speeds can be helpful).

Considering the situation in the I.P. compressor during a deceleration, case (b), this can be eased by:

(i) Closure of the H.P. compressor IGVs over a wider speed range. This makes less severe the shutting-off of the air flow into the H.P. compressor as the HP shaft speed drops.

(ii) Opening of the H.P compressor Handling Bleed Valve(s) early in the deceleration (i.e. at high $\left(N_H/\sqrt{T_{26}}\right)$). This allows the non-dimensional air flow out from the I.P. compressor to be higher, hence the I.P. compressor delivery pressure can drop (i.e. away from surge).

(iii) Introducing a bleed from the I.P. compressor delivery out into the Bypass Duct. This 'drops' the working line in the I.P. compressor (in same way as happens at (ii) above).

(iv) Closing the Final Nozzle during a deceleration. This lowers the LP shaft speed (for a particular HP shaft speed) due to the higher resisting torque at the Fan. This lower LP speed allows the working line in the I.P compressor to drop.

Schemes to allow solutions (iii) and (iv) above both require additional cost and add weight. Solutions using (i) and (ii), which are required anyway for satisfactory steady running, are therefore preferable. The best acceleration and deceleration schedules for IGVs and Handling Bleed Valve in the Tay engine are given in Section 9.5 above. However, it was there explained that the combination cannot be achieved with a conventional hydromechanical controller. It can be achieved by using an electronic controller.

9.9 Suggestions for further work

It was observed that, when the conventional fuel controller - with its schedule (figure 6.9) is used, there are considerable zones where, for example in the acceleration, the trajectory is significantly clear of surge. Further research might be directed towards finding an adaptive fuel controller that would allow these zones to be utilised. The variable geometry features might also be inter-linked with this adaptive controller.

Another form of research on the fuel flow scheduling might be considered. The engine designer has to provide the information about the fuel flow required under steady-running and during rapid acceleration and deceleration conditions without endangering the operating stability of the engine. In the Tay engine, the acceleration and deceleration fuel flow schedules are based on the pressure ratio and the speed of the H.P. compressor only, so the acceleration trajectory in the H.P. compressor can be prevented from surge. This type of fuel system, unfortunately, lacks of information about the proximity of the trajectory in the I.P. compressor to the surge line in a deceleration. An additional fuel control system would be advantageous to control the speed of the I.P. compressor, thus to prevent the deceleration trajectory in the I.P. compressor approaching surge. A Full Authority Digital Electronic Controller (FADEC) can be used since it allows greater sophistication in control, thus assisting the surge problems of the compressors.

When a gas turbine engine performs a transient, such an acceleration, a heat exchange will occur between the engine materials and gas streams. This exchange of heat will have an influence on the response rate of the engine as well as on the performance of engine components. Thermal effects are very important and, however, cannot be neglected. Inclusion of thermal effects in a simulation model is necessary to obtain a good prediction of the performance of the engine during transient operations. This will definitely give accurate information on the transient behaviour of the engine so an appropriate engine control system can be designed.

An aircraft gas turbine engine is designed for flight at high speeds and high altitudes. The performance of the engine is affected as the flight speed increases. The effect of flight speed is important and it is necessary to take into consideration this effect on the trajectories of the compressors of the engine. Usually, the flight speed is expressed by flight Mach number. The influence of Mach number on the engine components and engine performance are various and cannot be ignored. Therefore, the prediction of Mach number effects on the performance of the engine both in rapid acceleration and deceleration is required.

Surge problem in a high performance gas turbine engine has been and still remains one of the major barriers with regard to engine acceleration and off-design performance of the engine. The surge margin in the multistage high pressure compressor of the engine is small and limits the ability of the engine to accept large increases in fuel flow. One of the methods to prevent the compressor from surge is the casing treatment. This has been proved to be effective for improving surge margins of axial flow compressors. The use of the casing treatment, therefore, could be of a great help to improve the transient performance of the engine.

Most of the time, a two-spool gas turbine engine operates with the LP turbine choked during take-off, climb and cruise. This means the LP turbine operates at a fixed non-dimensional mass flow rate and pressure ratio. The choke of the LP turbine, however, seriously restricts the operating range of the turbine, thus the performance of the H.P. compressor. This effect will produce an increase in the pressure ratio at a given speed of the H.P. compressor, and hence will raise the operating line towards the surge line of the H.P. compressor. The situation may worsen if the engine performs a rapid acceleration. The tendency of the H.P. compressor to surge in an acceleration may be alleviated if the capacity of the LP turbine is increased. An increase of 4 per cent of the LP turbine capacity may be adequate to improve the acceleration trajectory in the H.P. compressor, hence a good transient performance of the engine.

The most exciting new concept in propulsion technology is the variable cycle engine. Providing sufficient flexibility to flow path geometry will enable the variable cycle engine to continuously match its characteristics to the needs of the aircraft while in flight. An essential ingredient of the variable cycle engine is the variable geometry turbine. This means the replacement of one or more rows of stators with rows of stators that can be adjusted by rotation about a radial axis. In this manner, the turbine operating map can be continuously altered to suit the immediate needs of the engine.

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APPENDIX I

Flow Through Axial Compressor Blading

The prediction of the fluid deviation

The fluid deviation is conventionally defined as the difference between the flow outlet angle (α_2) and the blade outlet angle (β_2). The advantage of working with the fluid deviation is that it is normally a reasonably small number. When a blade stalls the most serious effect is usually that the deviation rises to a high level, far outweighing the effect of the rise in loss.

There is in fact no reason to suppose that the flow would leave the cascade in the direction of the blade outlet and the discrepancy is predominantly a potential flow effect for which the explanation is as follows. Around mid-chord the potential flow may reasonably be thought of as following the camber line and the centripetal acceleration requires a pressure difference across the passage. At the trailing edge, however, the blade loading must go to zero so that the Kutta-Jukowski condition can be satisfied there and as the trailing edge is approached there must be a gradual reduction in the pressure gradient across the passage to make this possible. The chordwise distance over which the effect of the unloading will be felt upstream of the trailing edge will be proportional to the blade pitch. If the blade camber line remains curved right up to the trailing edge, as it normally does with the simple forms used with conventional profile families, it is clear that the flow can no longer follow the camber line because the loading is reduced below the level needed to turn it and balance the centrifugal acceleration. This means that the flow, averaged across the passage, can no longer have the outlet direction of the blades, in other words a deviation is produced.

Naturally, for design purposes it is desirable to use as large a deflection as possible in order to get the greatest pressure rise from the stage, provided the losses or drag coefficients do not increase too much. This nominal deflection and its corresponding fluid outlet angle are of the greatest importance to axial compressor performance and design. To obtain the actual fluid outlet angle a knowledge of the deviation is required. The correlation of the fluid deviation which is still most often used, is known as Howell's rule and takes the form

$$\delta = m \cdot \left(\frac{s}{c}\right)^{n} \tag{A.1}$$

where

$$m = 0.23 \cdot \left(\frac{2 \cdot a}{c}\right)^{2} + 0.1 \cdot \left(\frac{\alpha_2^*}{50}\right) \tag{A.2}$$

(a) is the distance of the point of maximum camber from the leading edge of the blade, and (α_2^*) is the nominal flow outlet angle given in degrees. Frequently a circular arc camber line is chosen so that $(2 \cdot a/c) = 1$, thereby simplifying the formula form, but its general form as given above embraces all shapes including a parabolic arc which is sometimes used. The cascade tests on compressor blades agreed better with a half power and $n = 0.5$ is recommended for decelerating cascades. For accelerating cascades, such as inlet guide vanes, a first power is recommended. Finally, the nominal flow outlet angle can be obtained by adding the flow deviation and the blade outlet angle together.

$$\alpha_2^* = \delta + \beta_2 \tag{A.3}$$

The above method was used in the present study for the prediction of the overall characteristics of the H.P. compressor.

APPENDIX II

A.1 Inter-component-volume size

The size of the inter-component-volumes used in this model correspond to the actual engine volumes. The sizes of these volumes were estimated from engine drawings and are listed below:

Vol BP = $1.50\,m^3$
Vol 26 = $0.15\,m^3$
Vol 3 = $0.12\,m^3$
Vol 5 = $0.09\,m^3$
Vol 6 = $0.3\,m^3$
Vol 78 = $5.00\,m^3$

The positions of these volumes relative to the engine components are portrayed in figure 5.3. By using the actual engine volumes, the simulation model has been kept physically realistic.

A.2 Pressure estimates

The pressure estimates in the inter-component-volumes used for the predictions of the performance of the engine during steady-running and transient operations are given below.

These initial estimates are used to allow the prediction program to start. The program then runs, typically from time $-0.2\,s$ to time $zero\,s$, with the shaft speeds and the fuel flow being held constant at the starting values, but the pressures in the inter-component-volumes being allowed to adjust by the small amounts required to bring thermodynamic balance through the engine before the transient starts. This balance was always satisfactorily obtained by the time $zero\,s$ was reached. The transient was then allowed to begin.

Initial pressure estimates for acceleration and steady-running predictions

P13in	=	$106.50\, KN/m^2$
P26in	=	$111.50\, KN/m^2$
P3in	=	$248.20\, KN/m^2$
P5in	=	$122.10\, KN/m^2$
P6in	=	$103.30\, KN/m^2$
P7Sin	=	$102.80\, KN/m^2$

Initial pressure estimates for deceleration predictions

P13in	=	$175.55\, KN/m^2$
P26in	=	$290.81\, KN/m^2$
P3in	=	$1893.94\, KN/m^2$
P5in	=	$558.25\, KN/m^2$
P6in	=	$160.61\, KN/m^2$
P7Sin	=	$143.93\, KN/m^2$

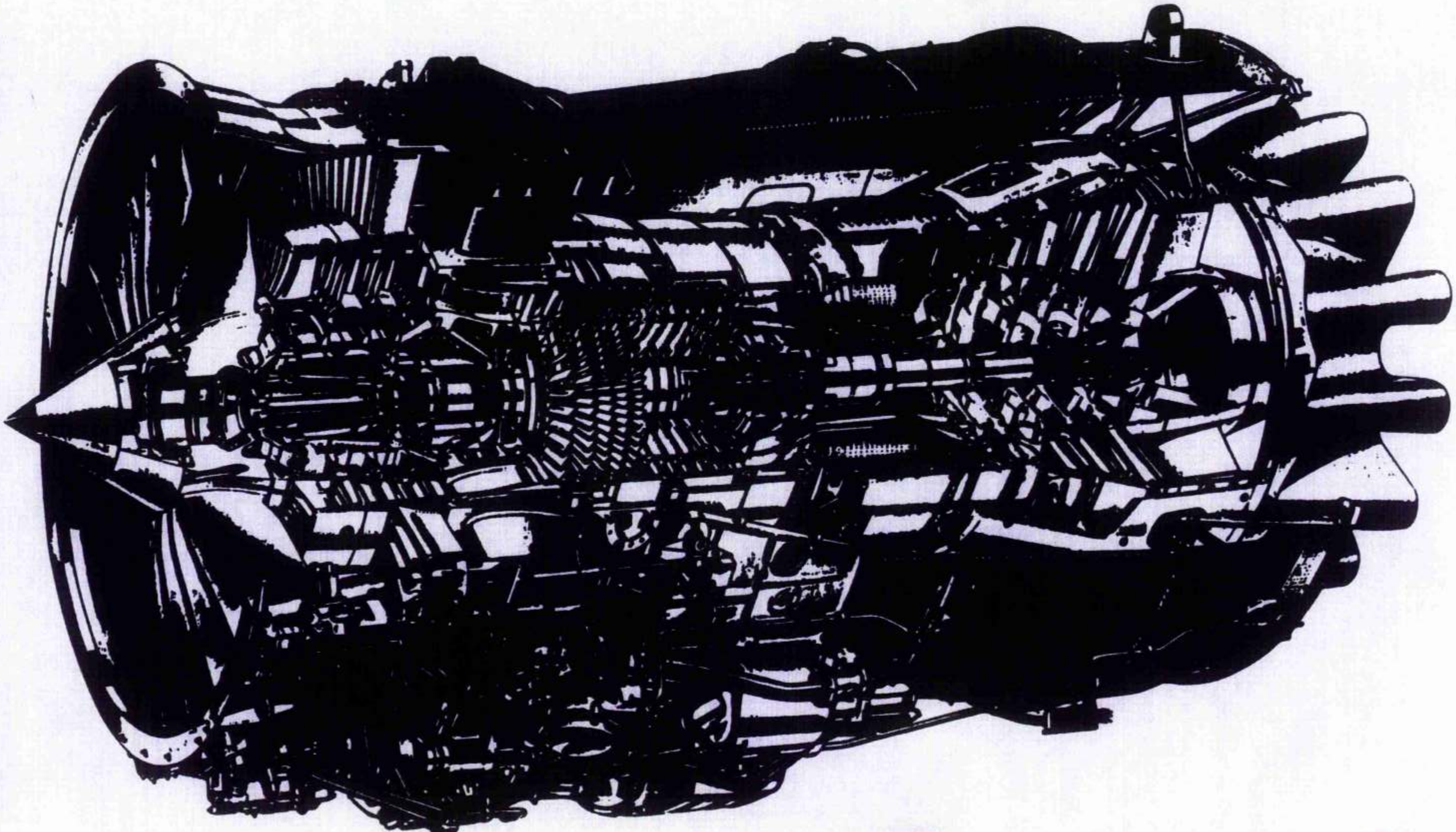
TABLE

Table 6.1. Operating IGV and Handling Bleed schedules of H.P. compressor for different schemes.

Scheme	Design	Production	Revised
IGVs $\left(N_H/\sqrt{T_{26}}\right)$	552 - 586	493 - 593	549 - 620
Handling Bleed $\left(N_H/\sqrt{T_{26}}\right)$	549 - 568	493 - 520	559 - 589

N_H in Rev/min

T_{26} in Kelvin



Rolls-Royce Tay

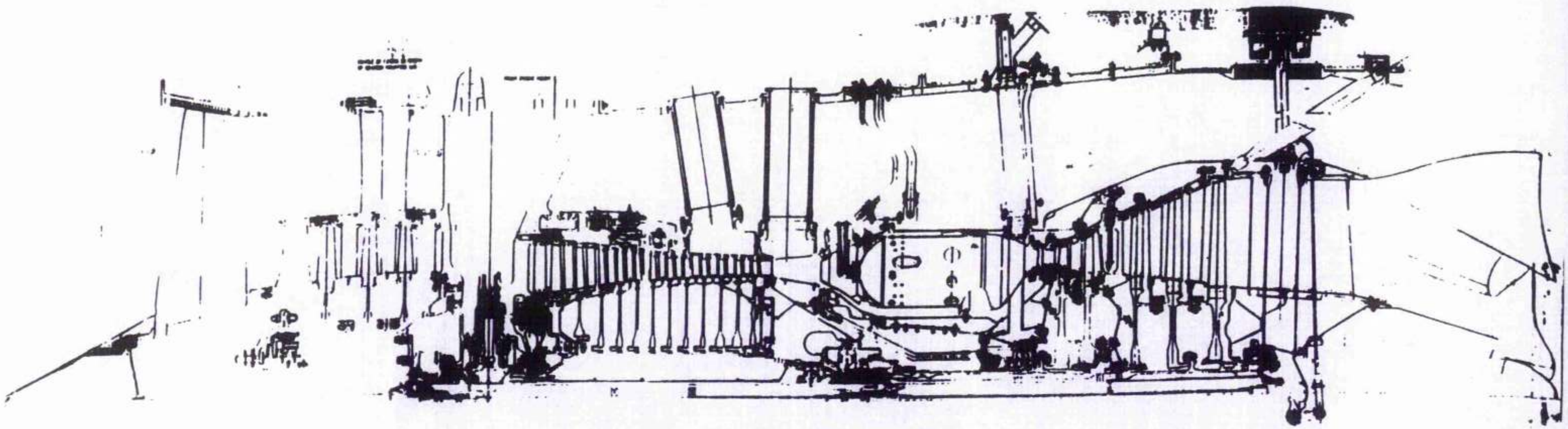


FIG. 1.1 General arrangement of the Tay turbofan engine

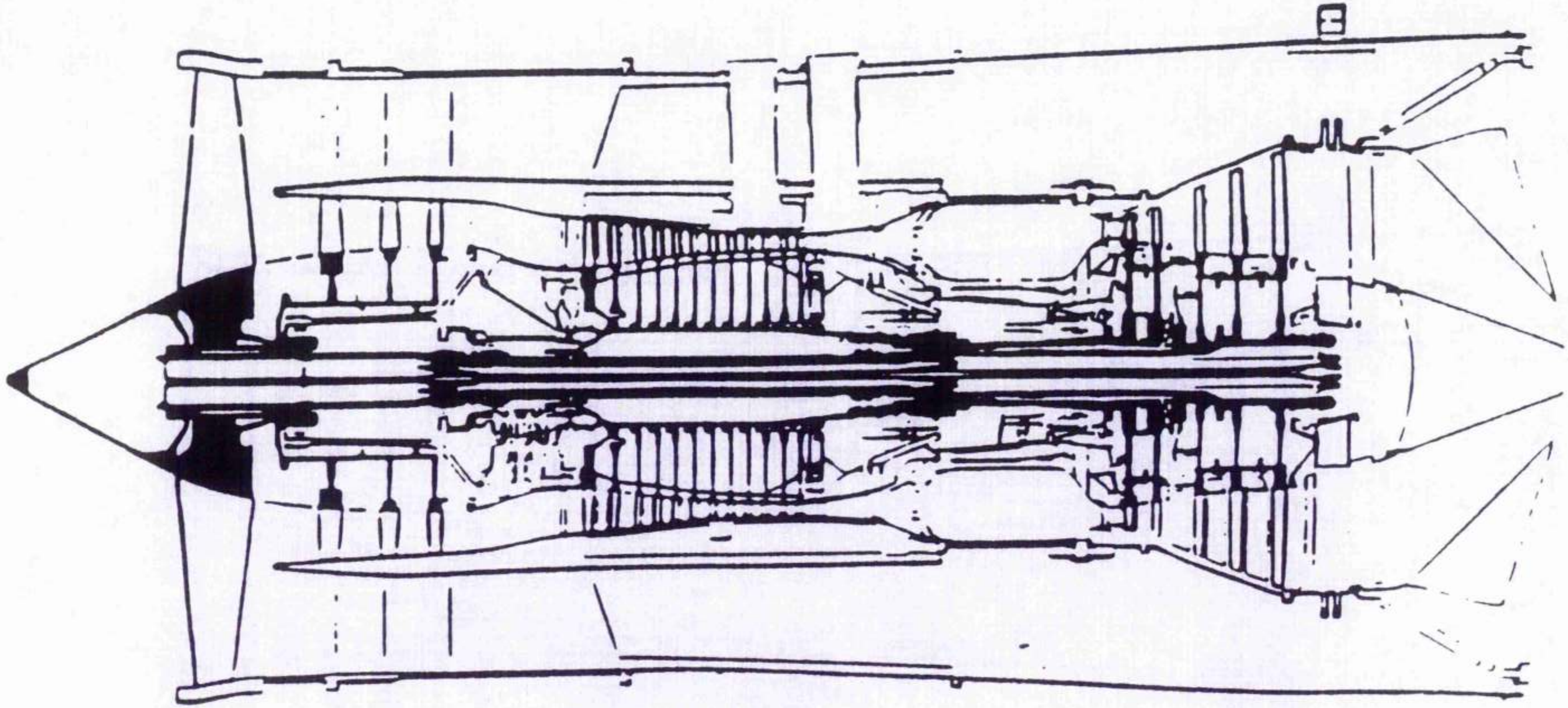
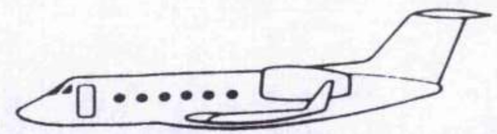
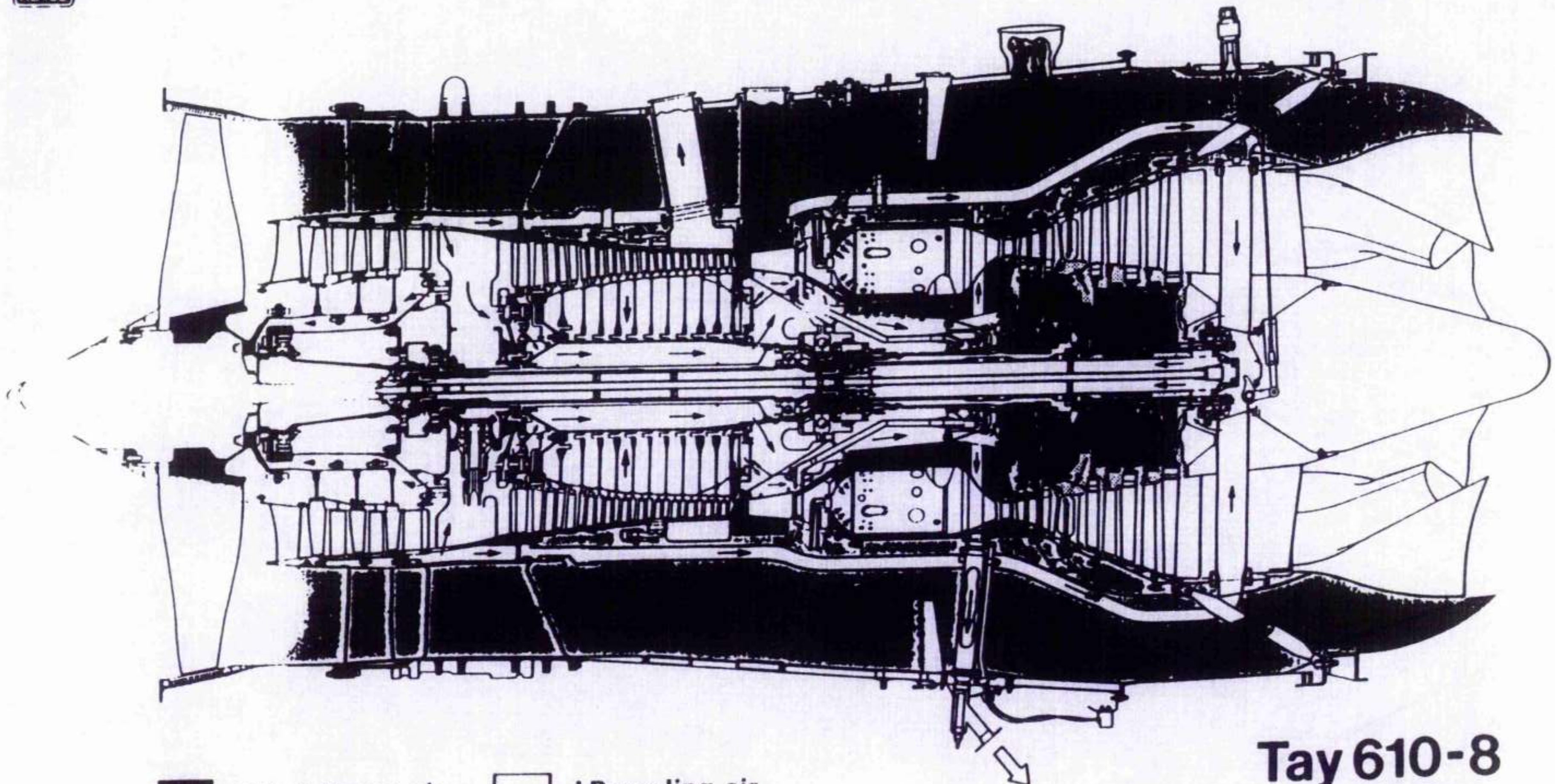


FIG. 1.2 Typical cross section of the Tay engine

Rolls-Royce Tay airflow diagram





- HP delivery air
- HP restricted air
- By-pass air
- IP cooling air
- HP air 5th stage
- HP air 7th stage

Tay 610-8
in
Gulfstream IV

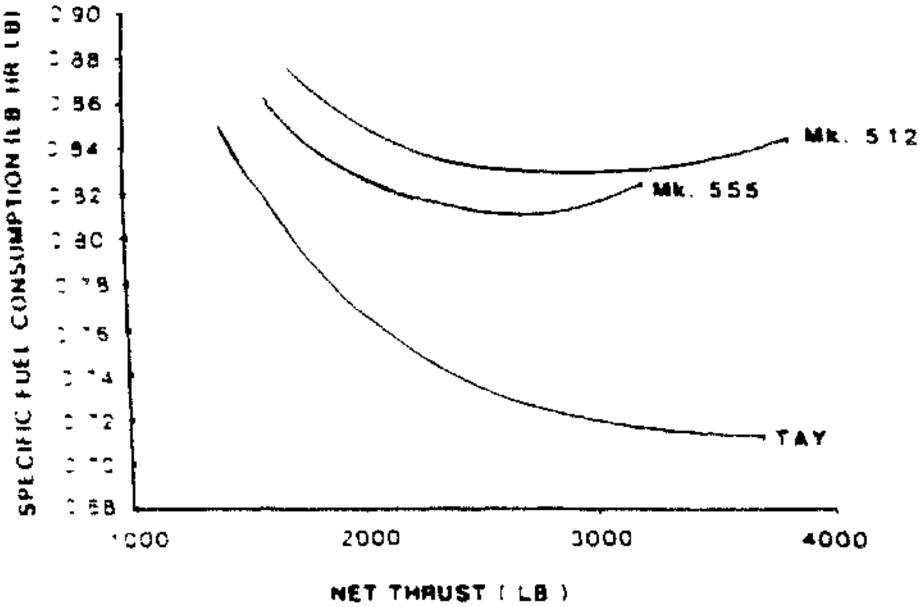


FIG. 1.3 Cruise performance 30,000 ft ISA 0.8 Mn

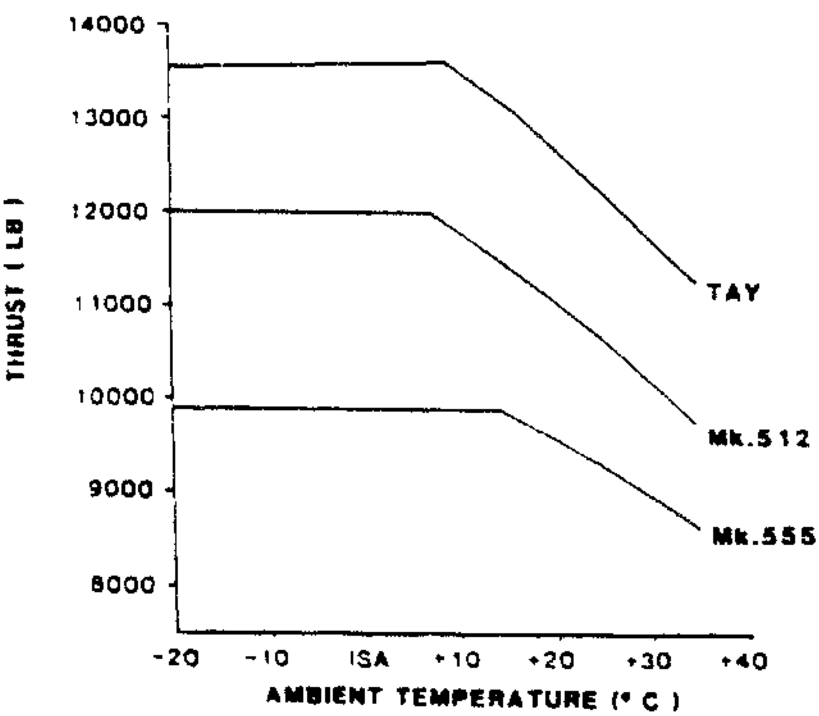
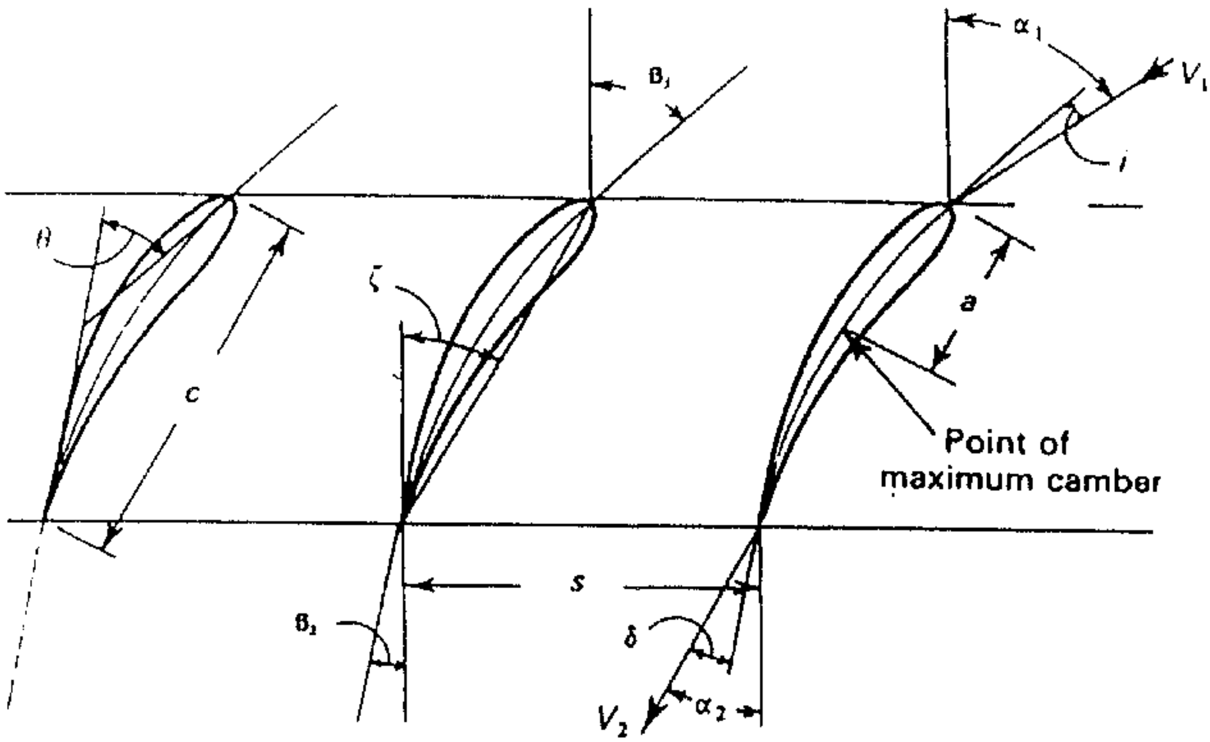


FIG. 1.4 Tay take-off thrust compared with
Spey 555 and Spey 512



C *chord*
θ *camber angle* $= \beta_1 - \beta_2$
α_1 *air inlet angle*
α_2 *air outlet angle*
β_1 *blade inlet angle*
β_2 *blade outlet angle*
V_1 *inlet velocity*
V_2 *outlet velocity*
i *incidence* $= \alpha_1 - \beta_1$
δ *deviation* $= \alpha_2 - \beta_2$
ε *deflexion* $= \alpha_1 - \alpha_2$
S *pitch*
ζ *stagger angle*

FIG. 4.1 Compressor cascade notation

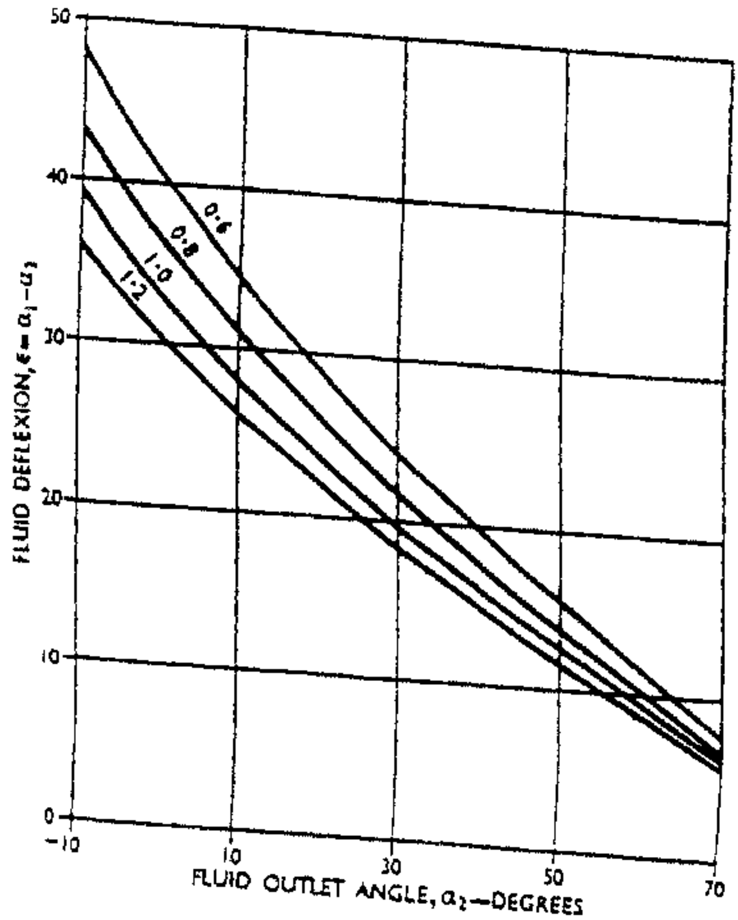


FIG 4.2 Nominal values of fluid deflexion for different for different pitch-chord ratios

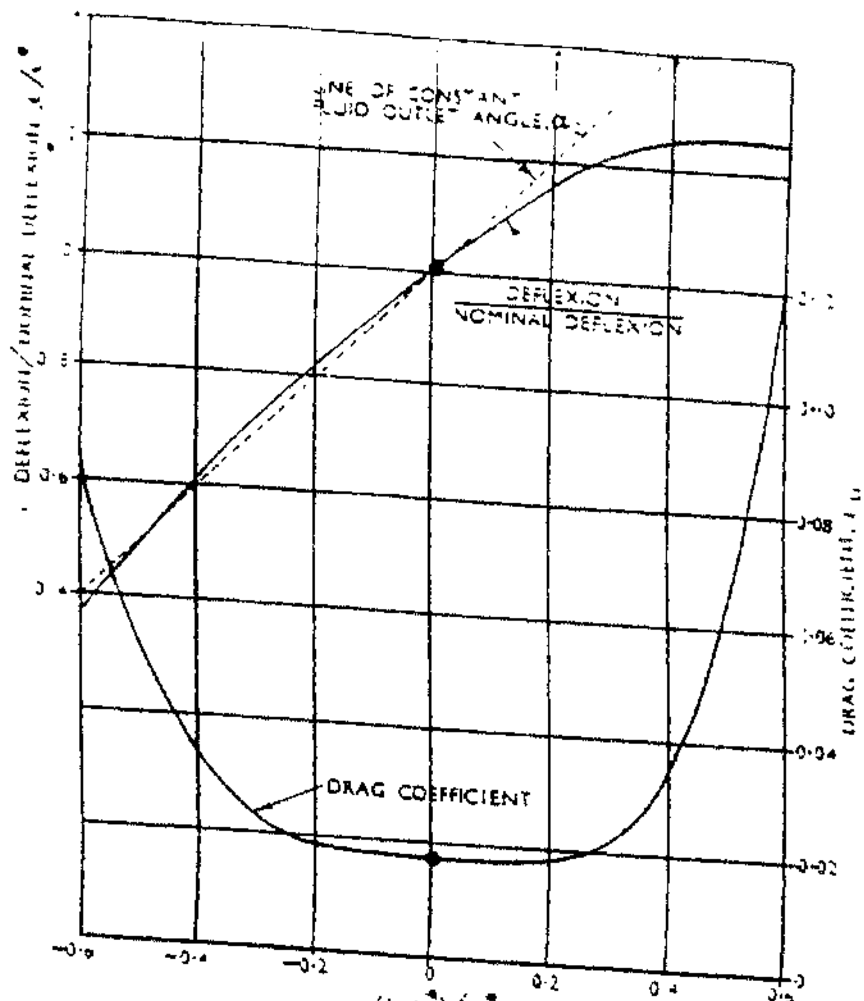


FIG 4.3 Deflexions and drag coefficients at other than nominal incidences

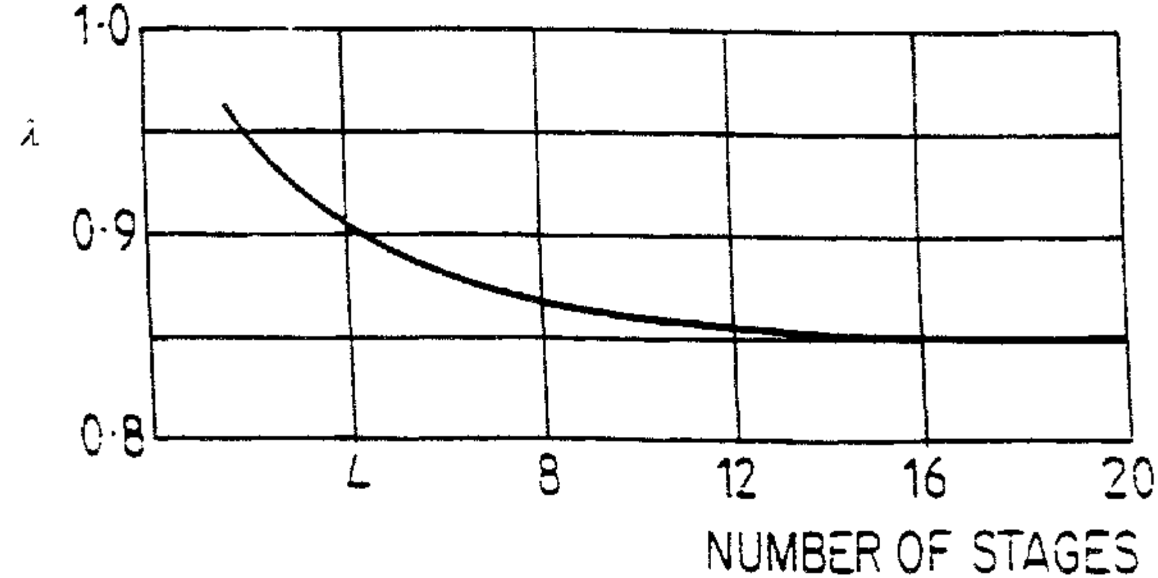


FIG 4.4 Variation of mean work done factor with number of stages

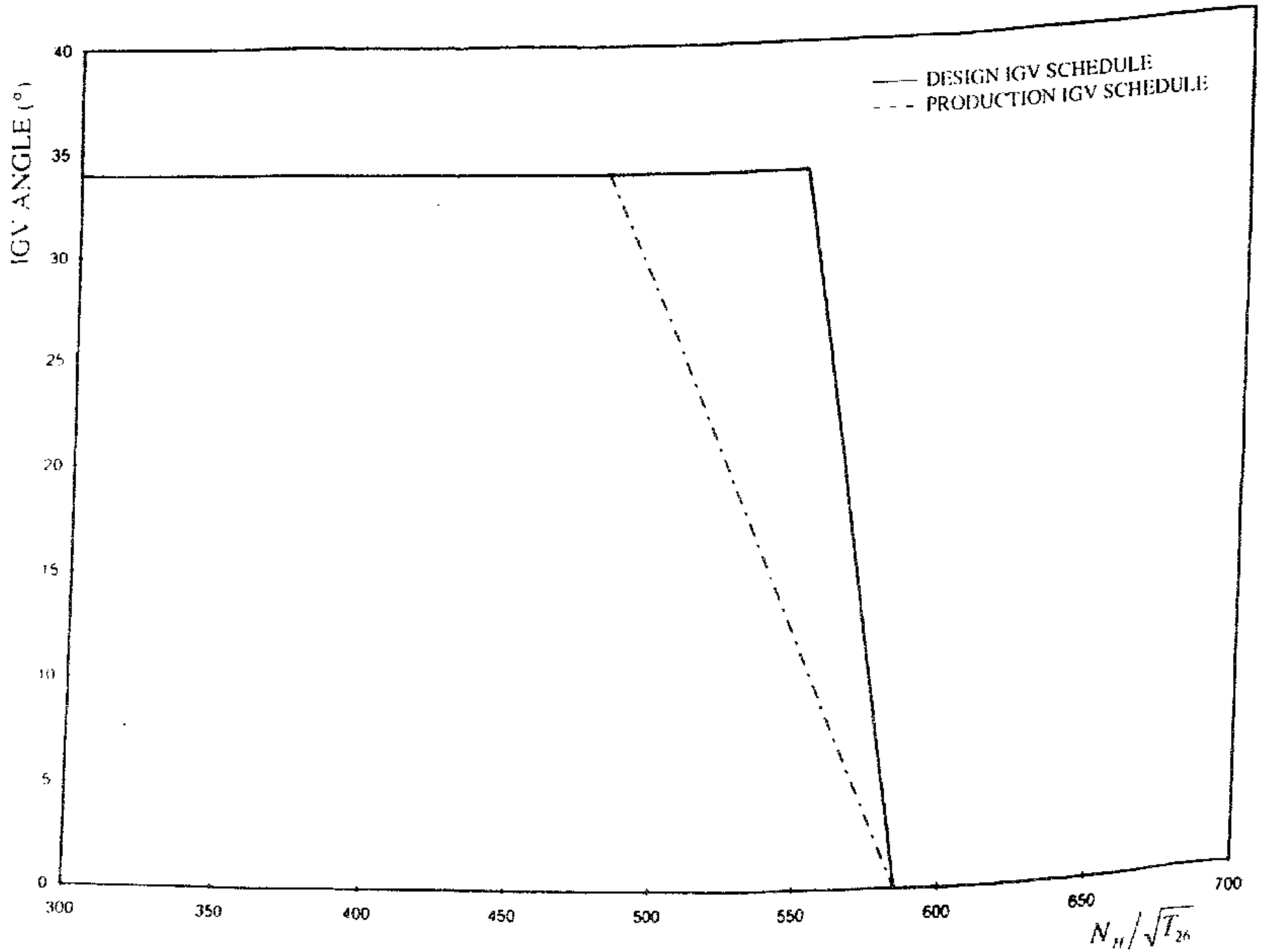


FIG 4.5 Operating schedules of IGVs in function of H.P. compressor speeds

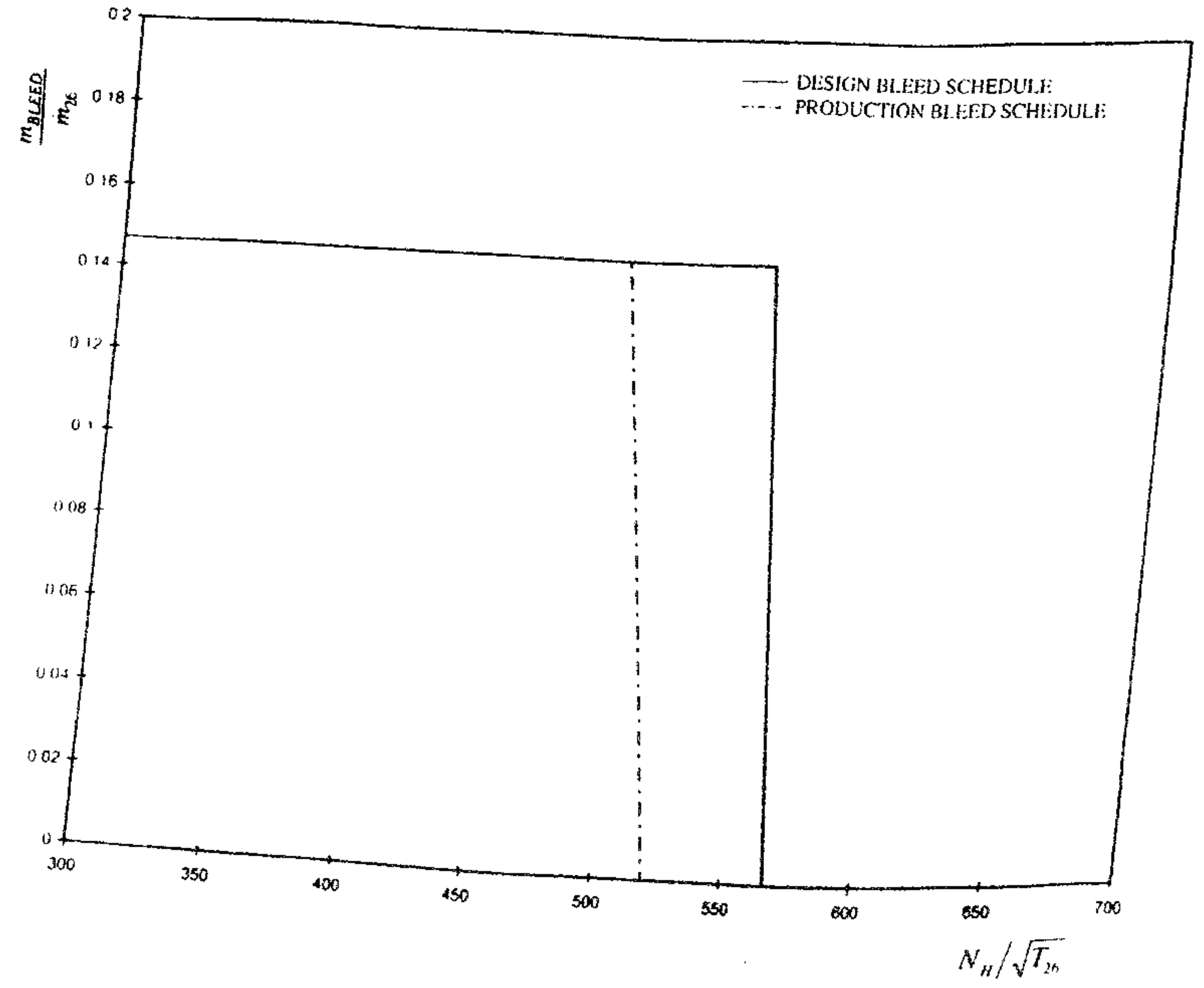


FIG 4.6 Operating schedules of bleed in function of H.P. compressor speeds

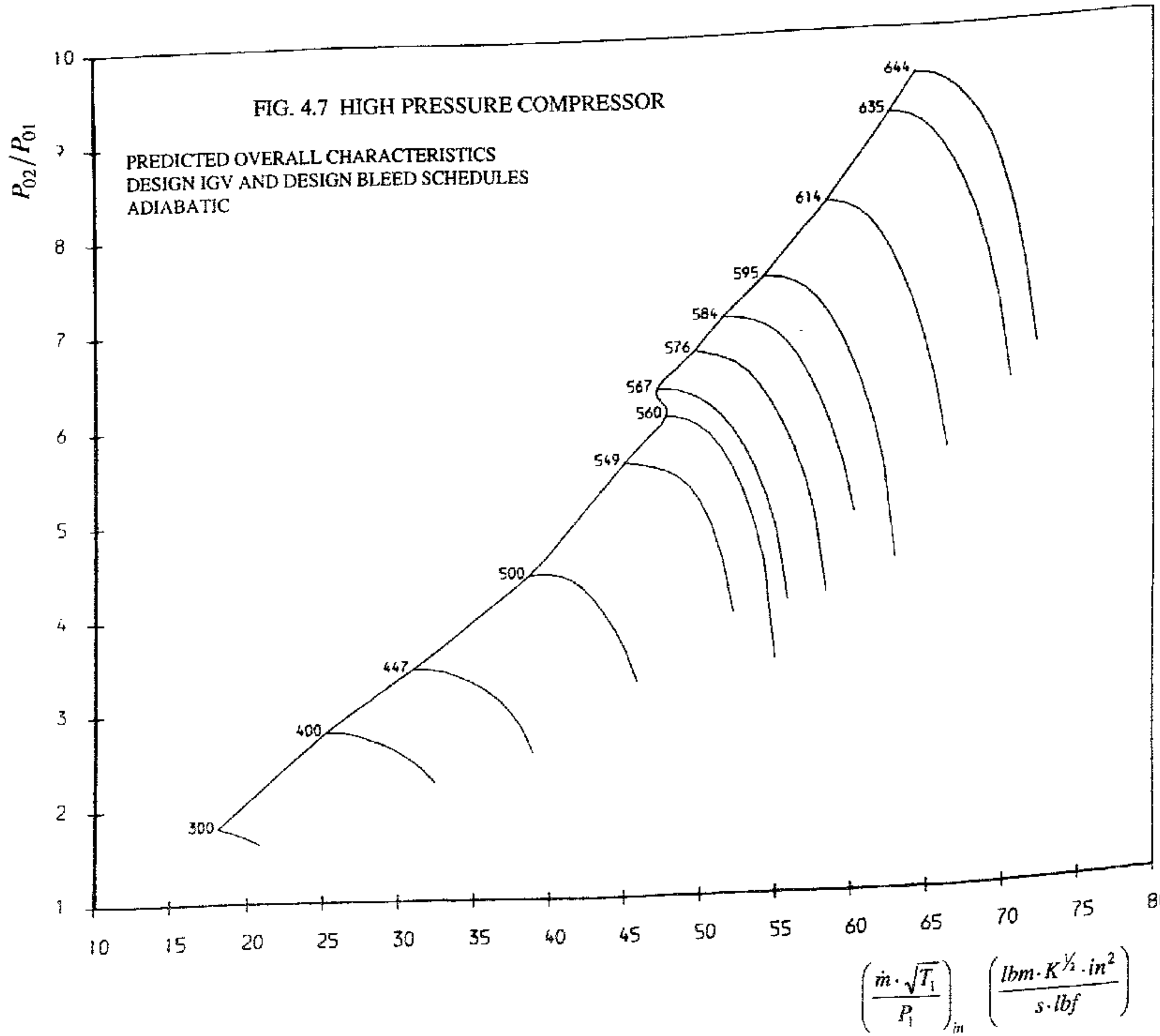


FIG. 4.7 HIGH PRESSURE COMPRESSOR

PREDICTED OVERALL CHARACTERISTICS
DESIGN IGV AND DESIGN BLEED SCHEDULES
ADIABATIC

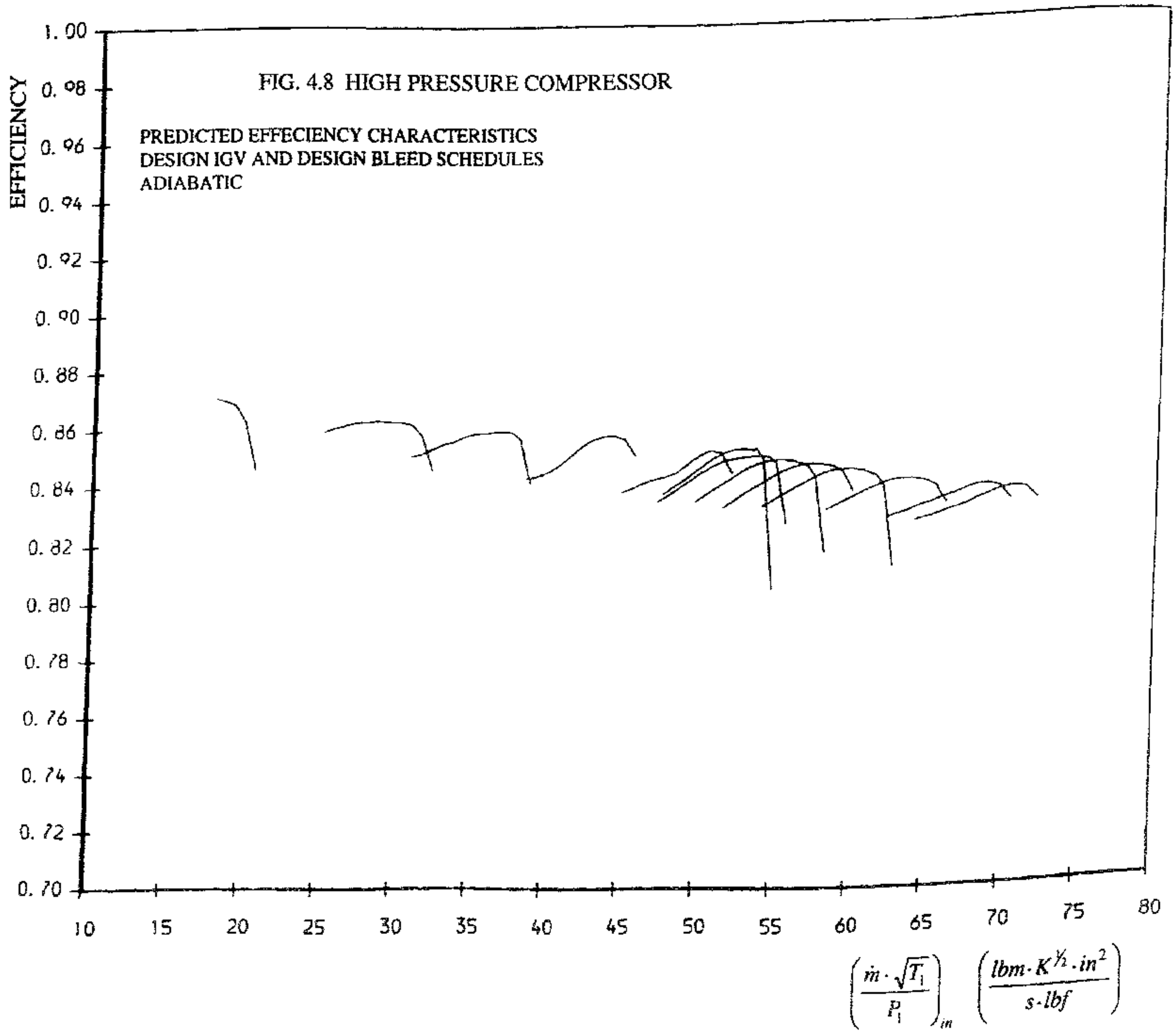


FIG. 4.8 HIGH PRESSURE COMPRESSOR
PREDICTED EFFECIENCY CHARACTERISTICS
DESIGN IGV AND DESIGN BLEED SCHEDULES
ADIABATIC
EFFICIENCY
1.00
0.98
0.96
0.94
0.92
0.90
0.88
0.86
0.84
0.82
0.80
0.78
0.76
0.74
0.72
0.70
10
15
20
25
30
35
40
45
50
55
60
65
70
75
80
$\left(\frac{\dot{m}\cdot\sqrt{T_1}}{P_1}\right)_{in}$ $\left(\frac{lbm\cdot K^{1/2}\cdot in^2}{s\cdot lbf}\right)$

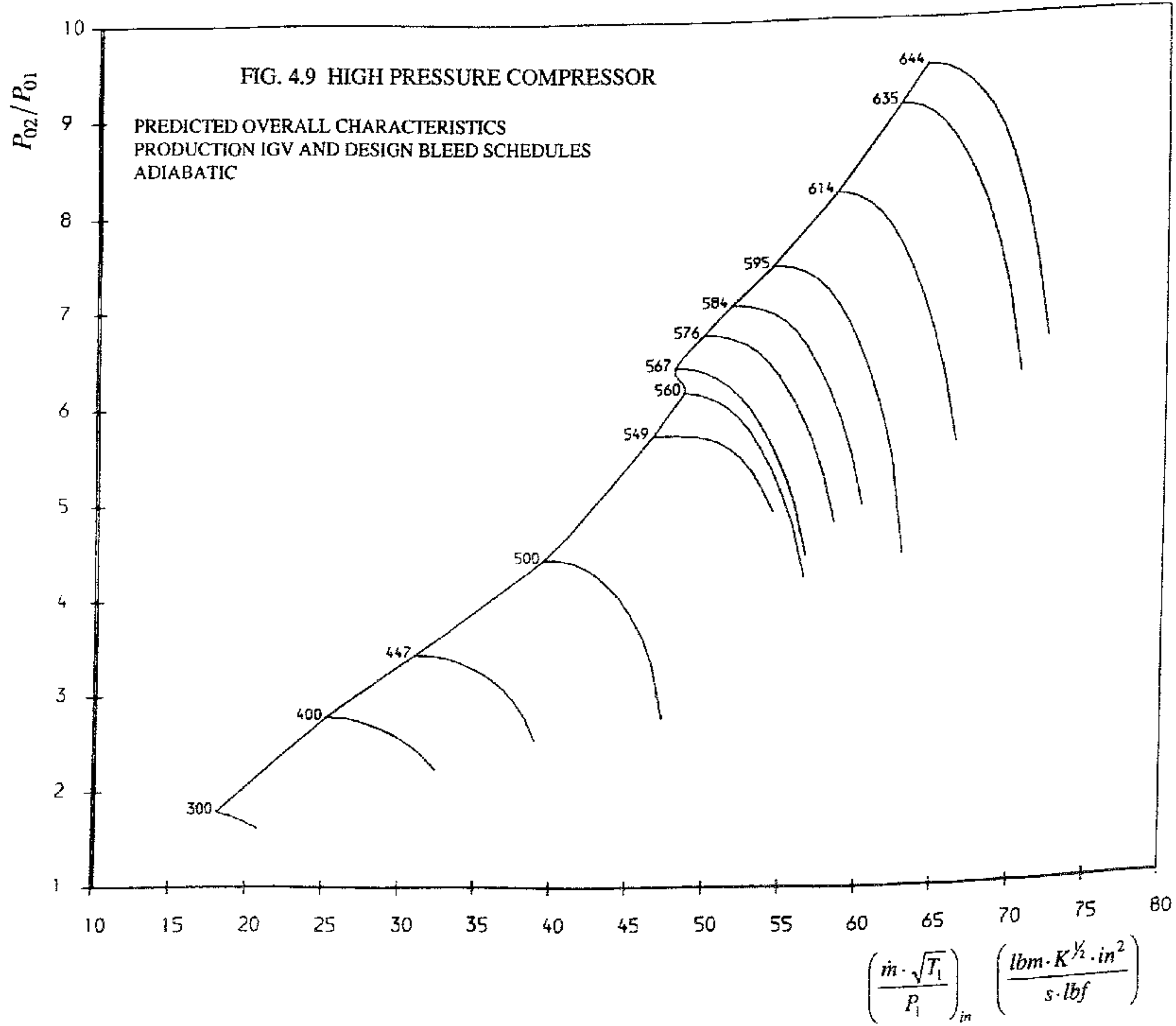


FIG. 4.9 HIGH PRESSURE COMPRESSOR

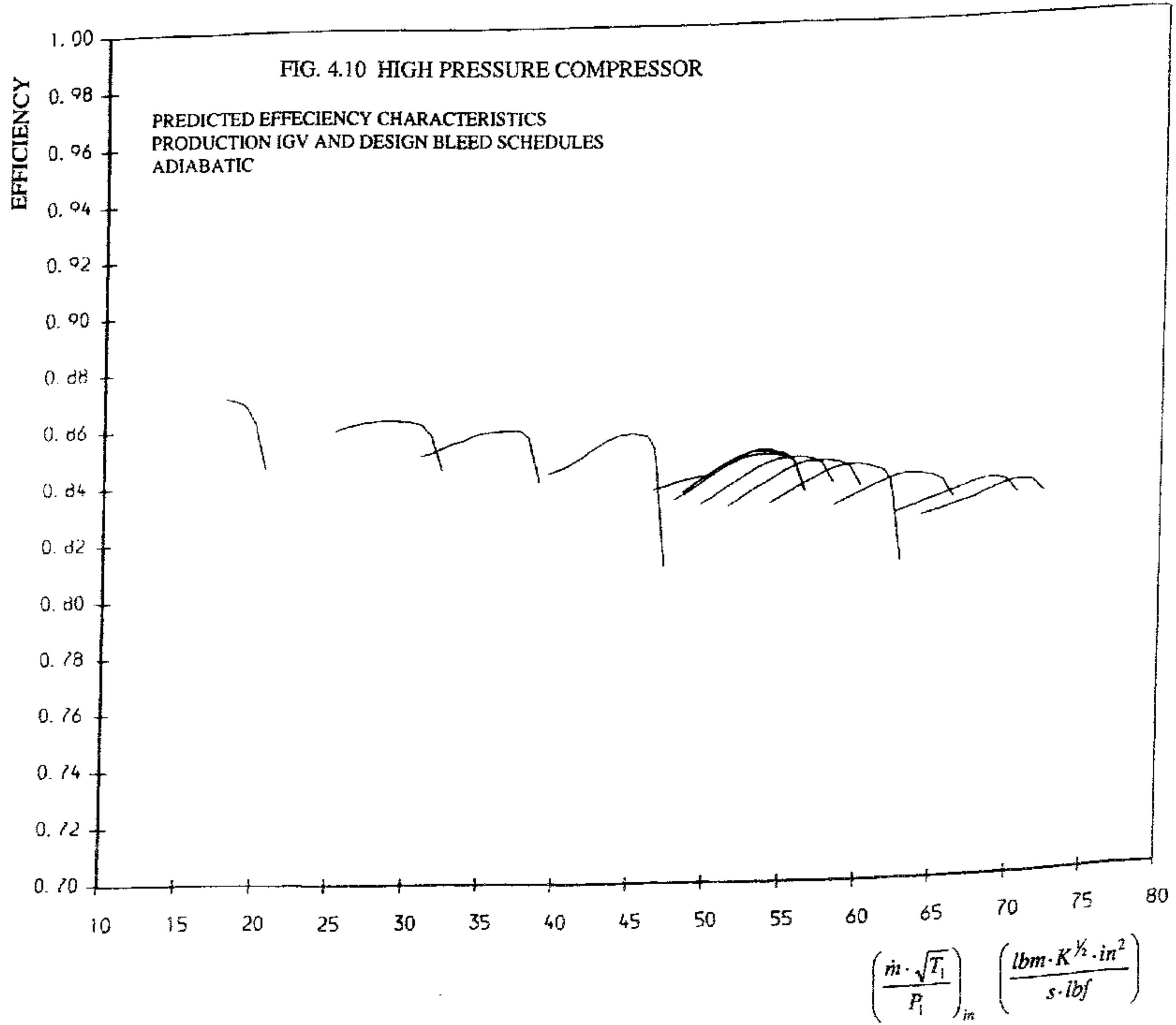


FIG. 4.10 HIGH PRESSURE COMPRESSOR

PREDICTED EFFECIENCY CHARACTERISTICS
PRODUCTION IGV AND DESIGN BLEED SCHEDULES
ADIABATIC

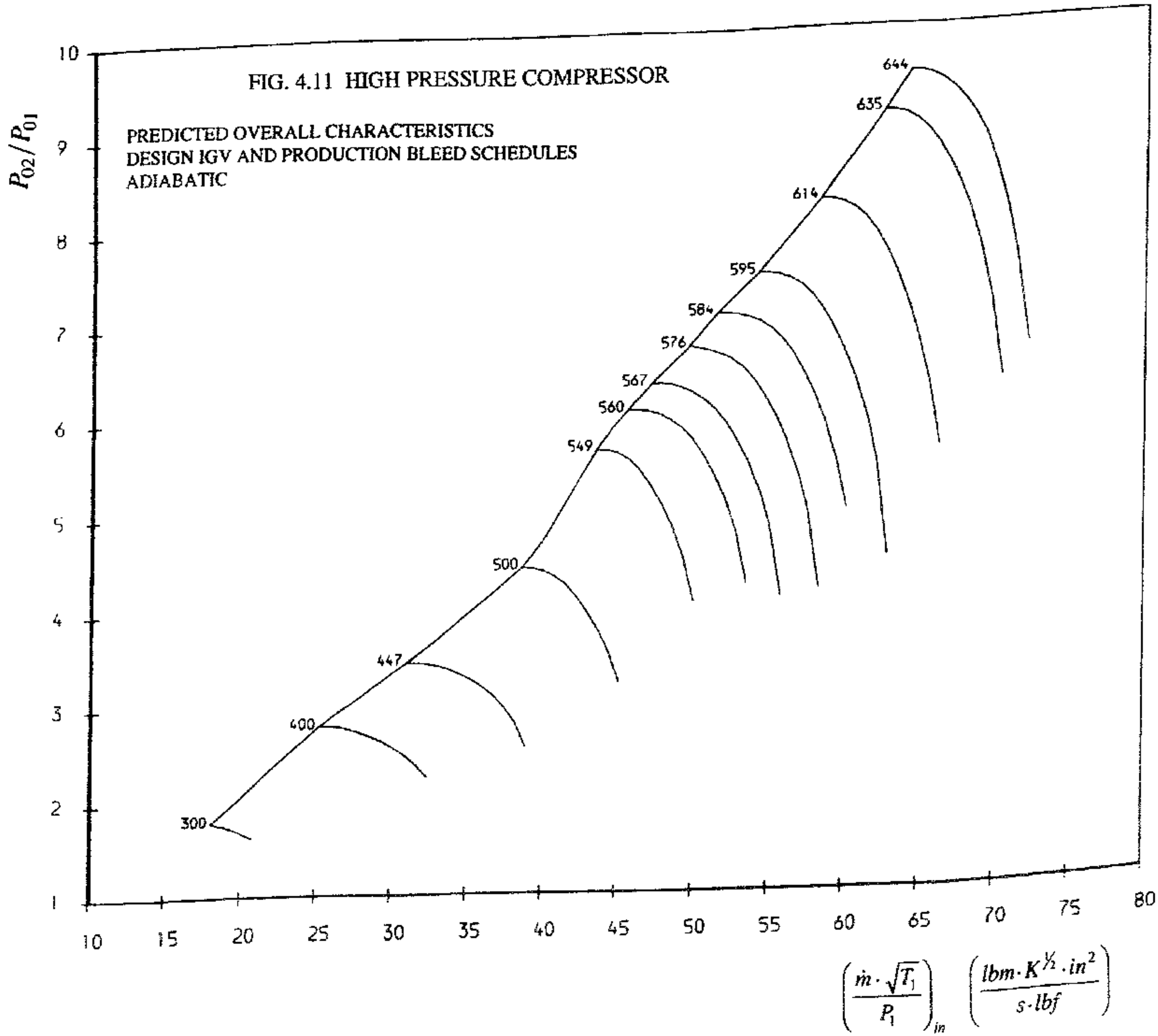


FIG. 4.11 HIGH PRESSURE COMPRESSOR

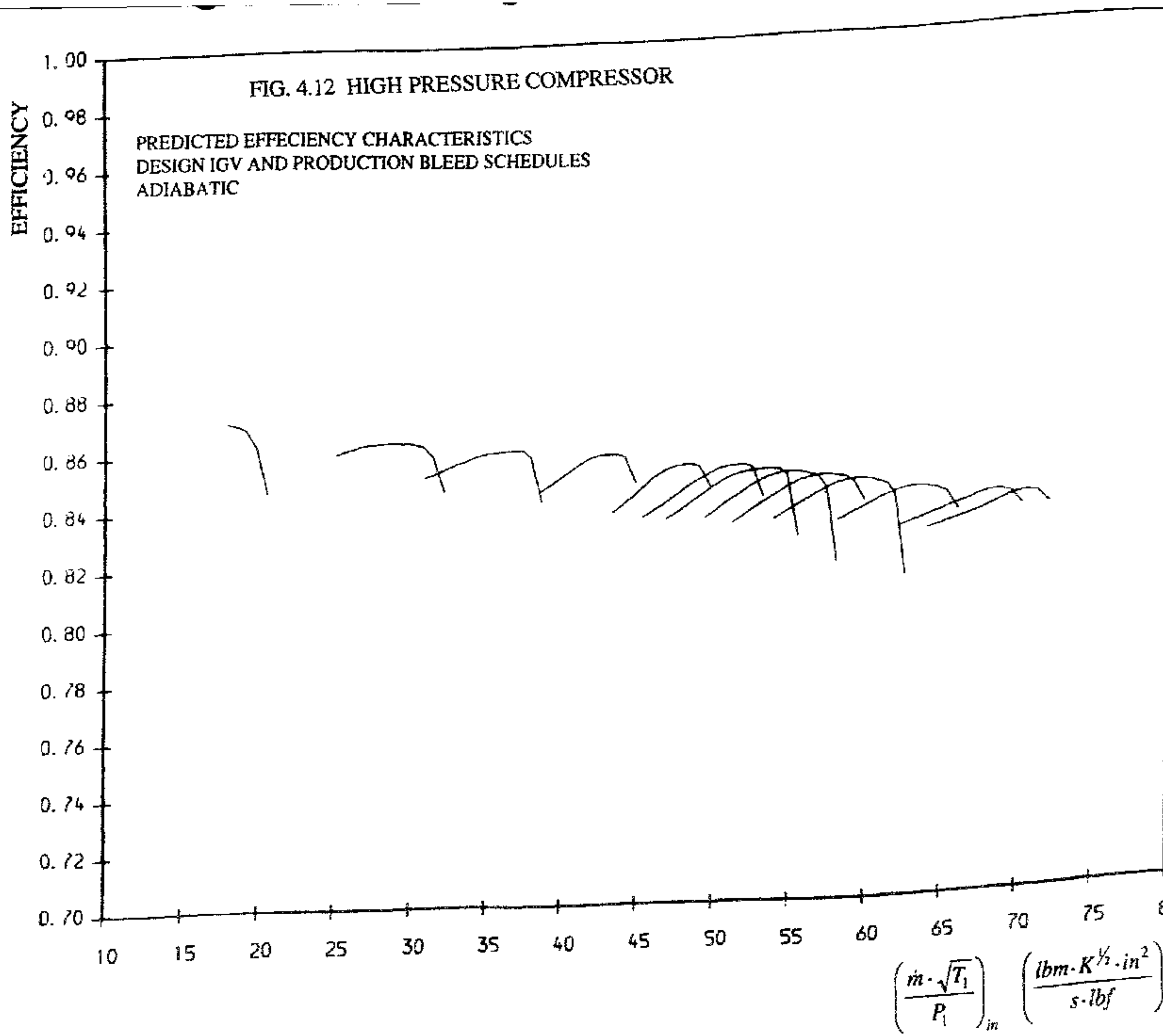


FIG. 4.12 HIGH PRESSURE COMPRESSOR

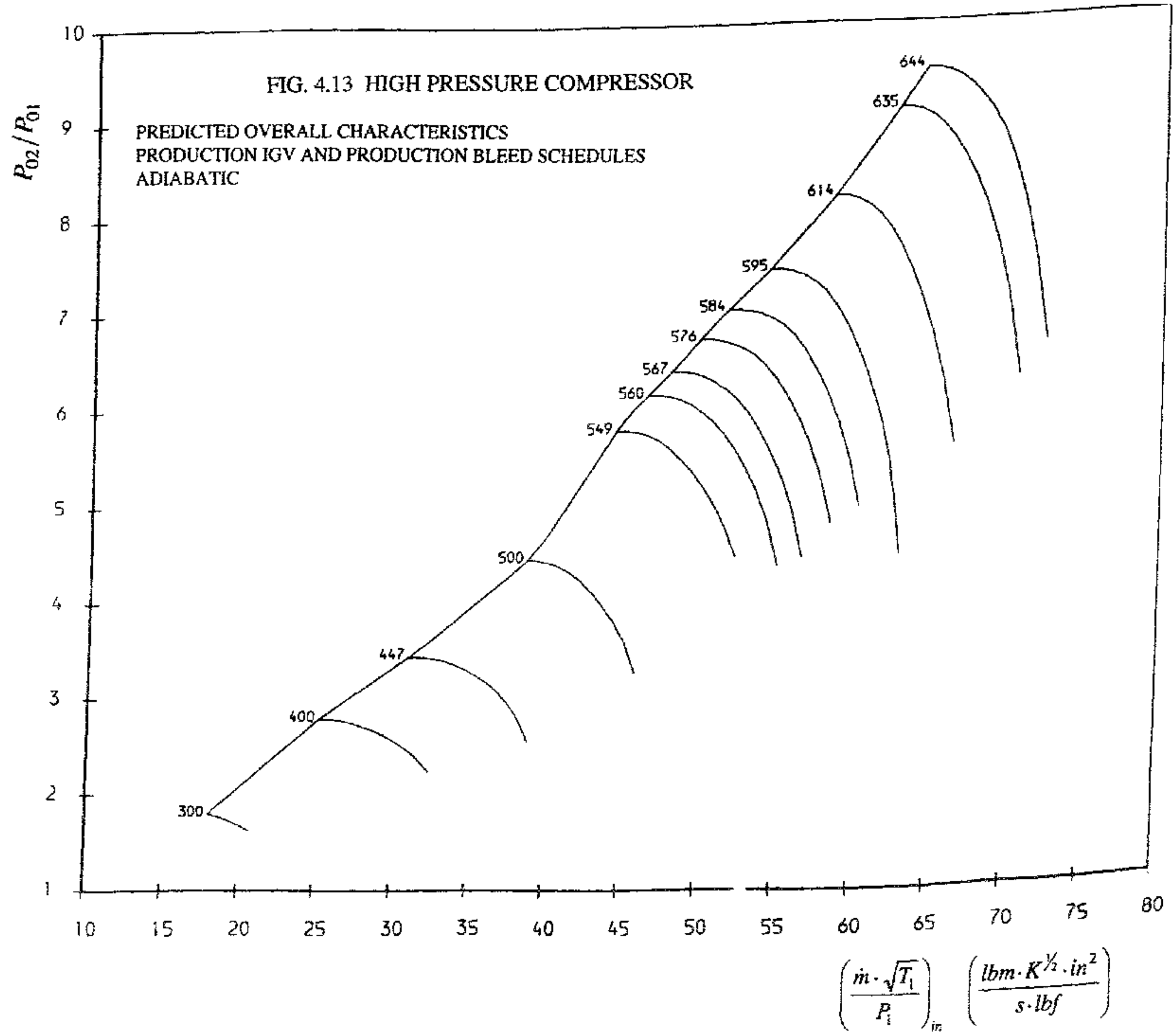
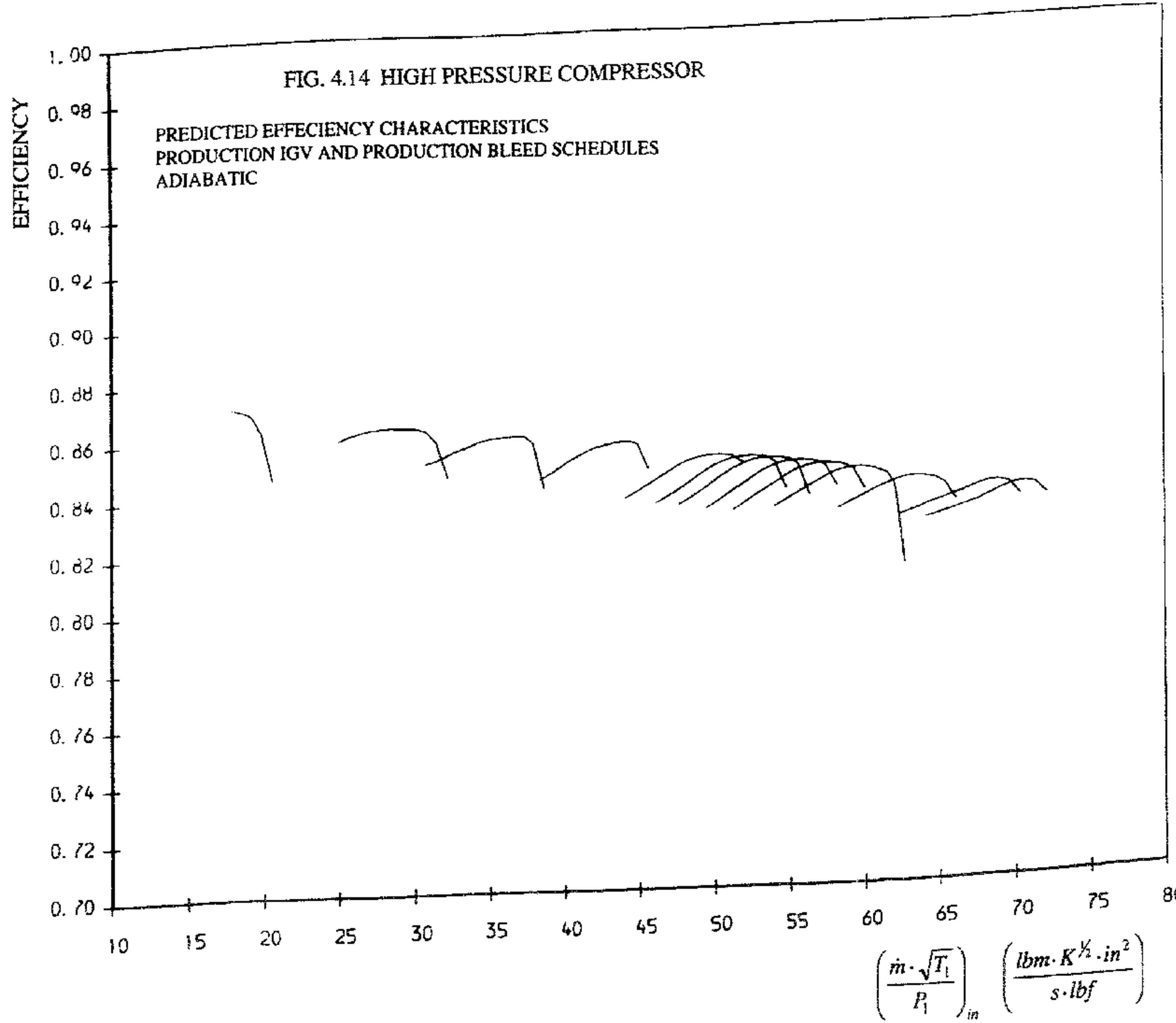


FIG. 4.13 HIGH PRESSURE COMPRESSOR

PREDICTED OVERALL CHARACTERISTICS
PRODUCTION IGV AND PRODUCTION BLEED SCHEDULES
ADIABATIC

FIG. 4.14 HIGH PRESSURE COMPRESSOR



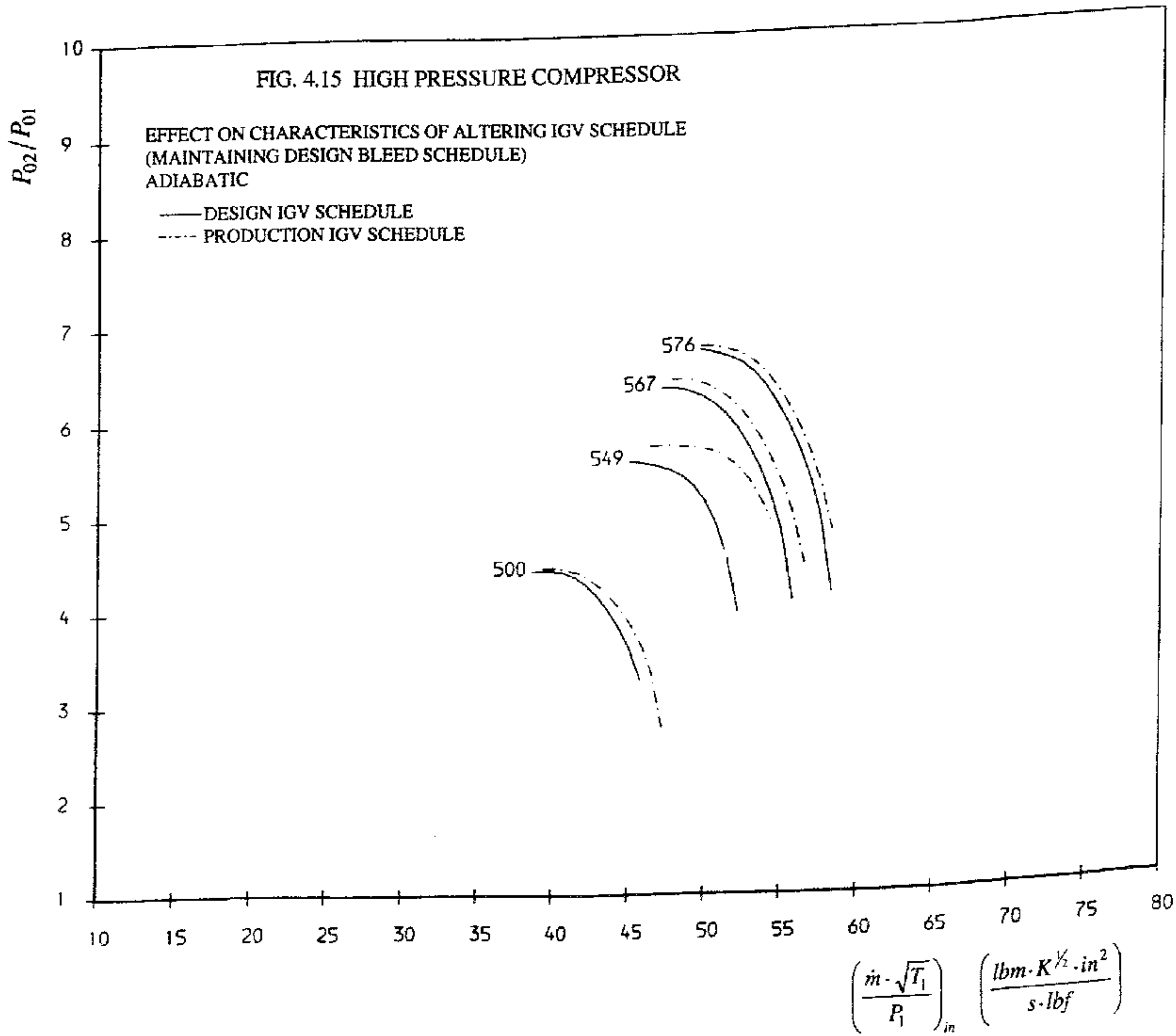


FIG. 4.15 HIGH PRESSURE COMPRESSOR

FIG. 4.16 HIGH PRESSURE COMPRESSOR

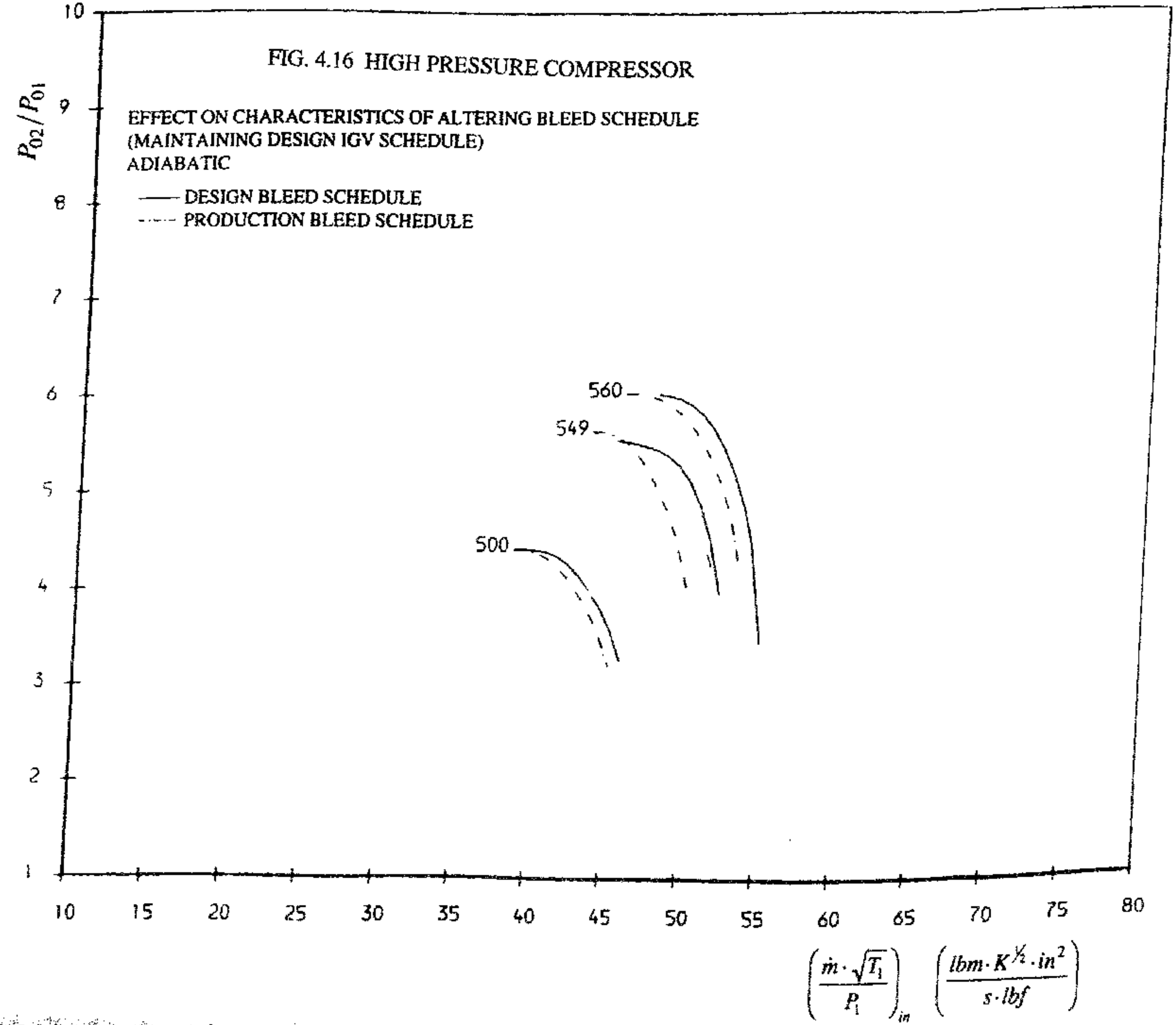
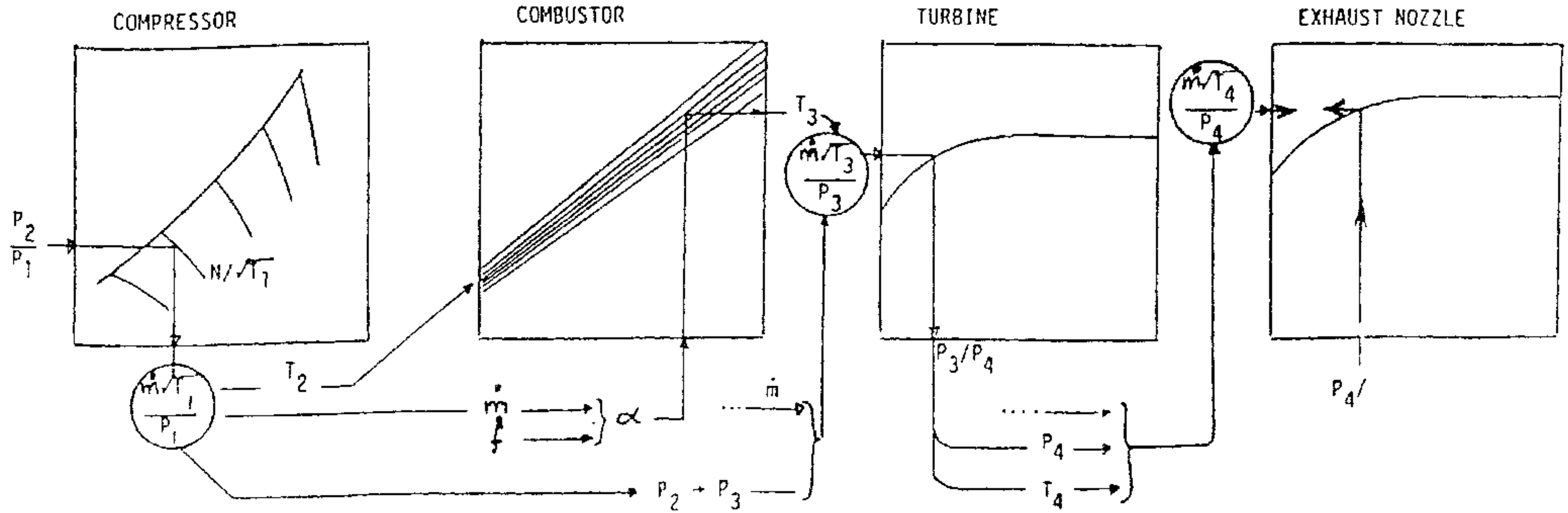


FIG. 5.1 CMF PROCEDURE FOR SINGLE-SPOOL AERO ENGINE
(Cranfield Institute of Technology)



Notes: P_1 is obtained from P_{AMB} and the flight Mach number.

$\left(\dfrac{\dot{m}\cdot\sqrt{T_4}}{P_4}\right)_{cal}$ $c.f.$ $\left(\dfrac{\dot{m}\cdot\sqrt{T_4}}{P_4}\right)_{chics}$; if not compatible alter P_2/P_1.

Once compatibility is obtained calculate accelerating power from $\dot{m}\cdot C_P\cdot(T_3-T_4)-\dot{m}\cdot C_P(T_2-T_1)$.

When the turbine is choked the iterations are carried out to obtain the choked value of $\left(\dfrac{\dot{m}\cdot\sqrt{T_3}}{P_3}\right)$.

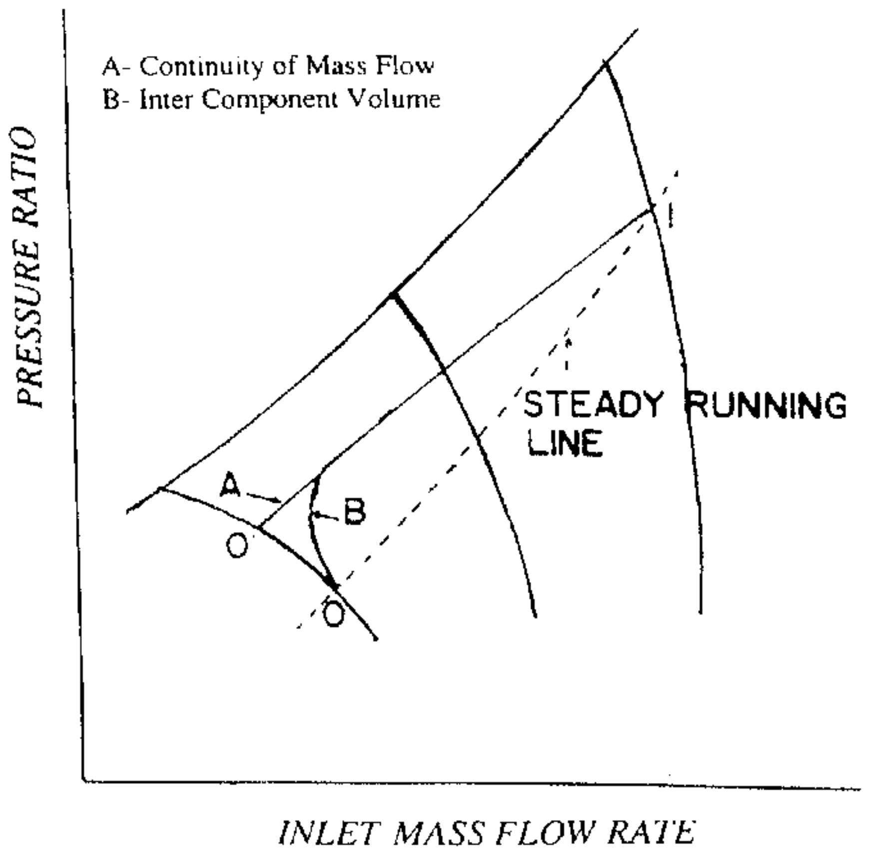


FIG. 5.2 Operating trajectories for different computer methods

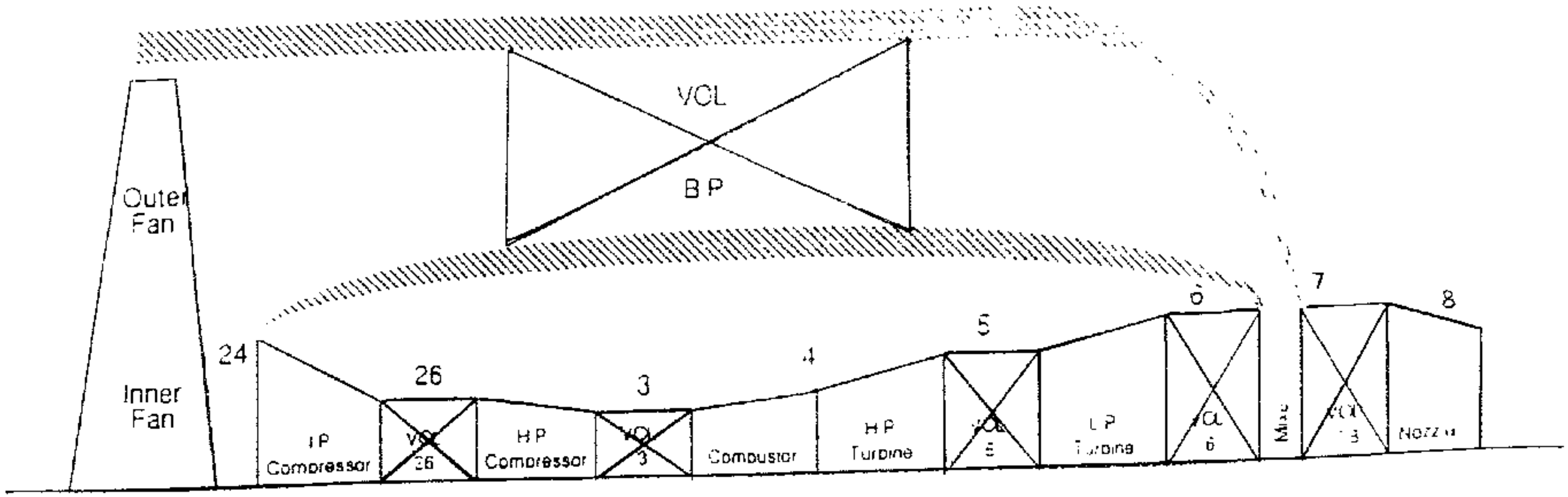
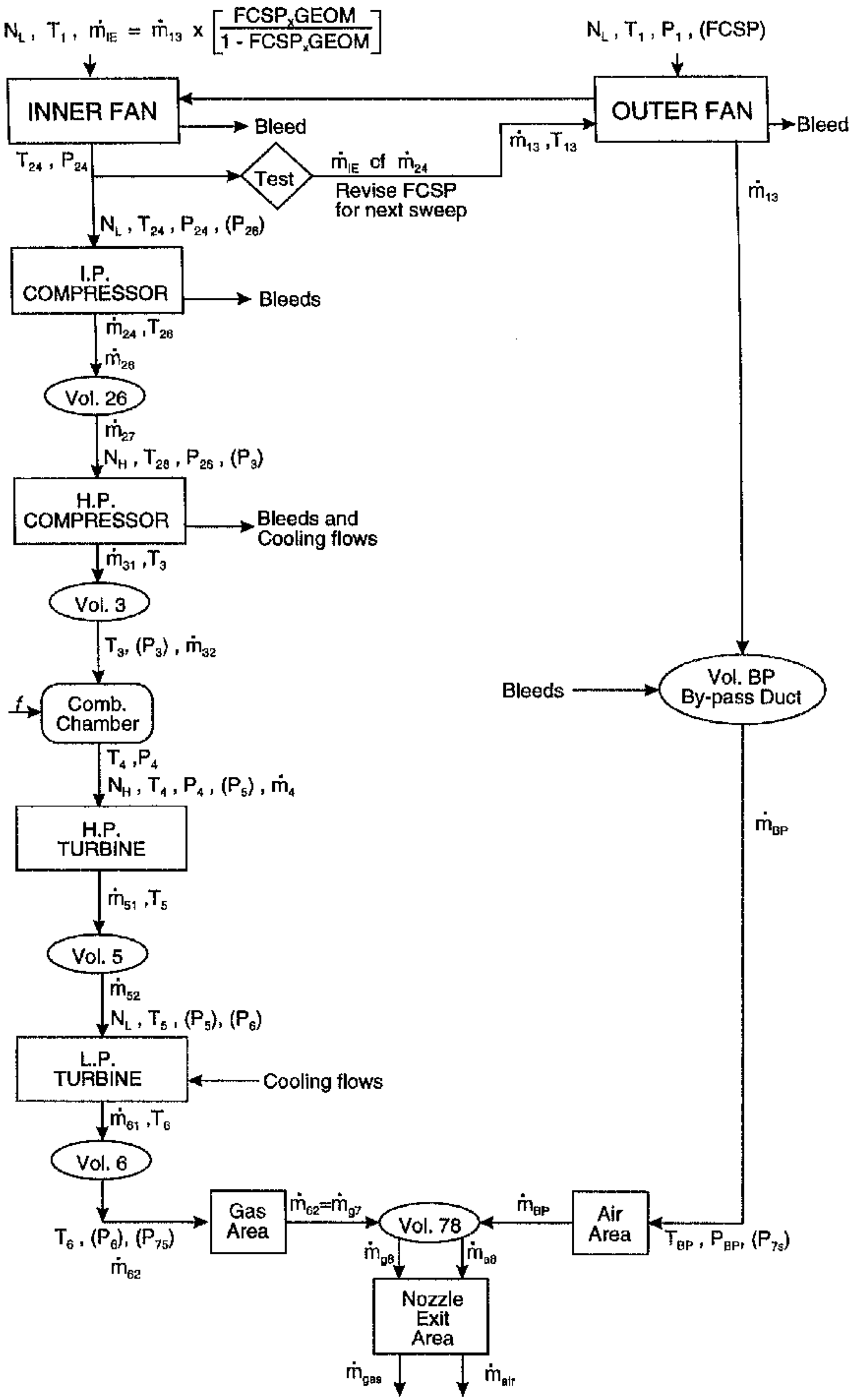


FIG. 5.3 Station numbering of the Tay turbofan engine

FIG. 5.4 ICV PROCEDURE FOR TAY TYPE ENGINE



Calculate new pressures and accelerations

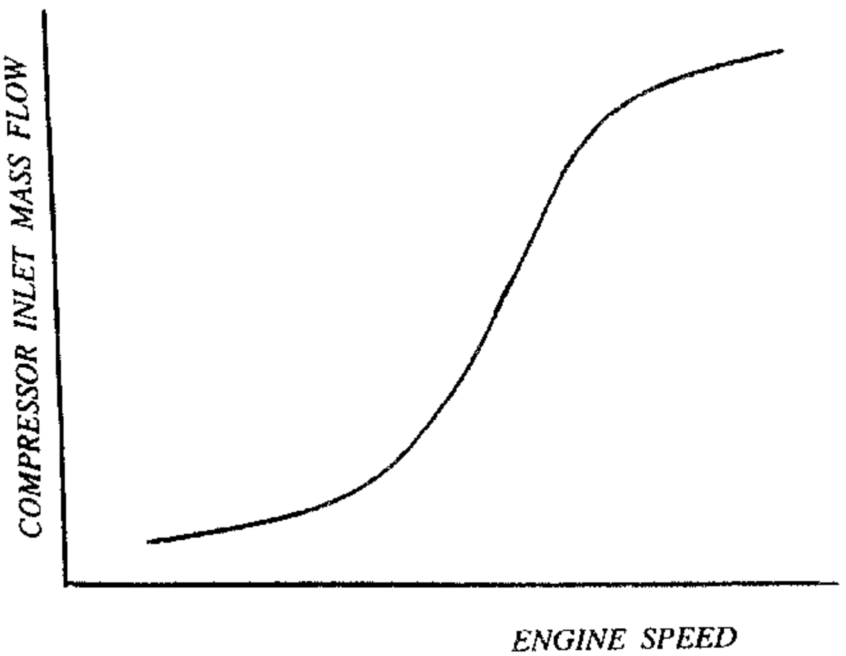


FIG. 6.1 Typical flow-speed characteristic of a compressor

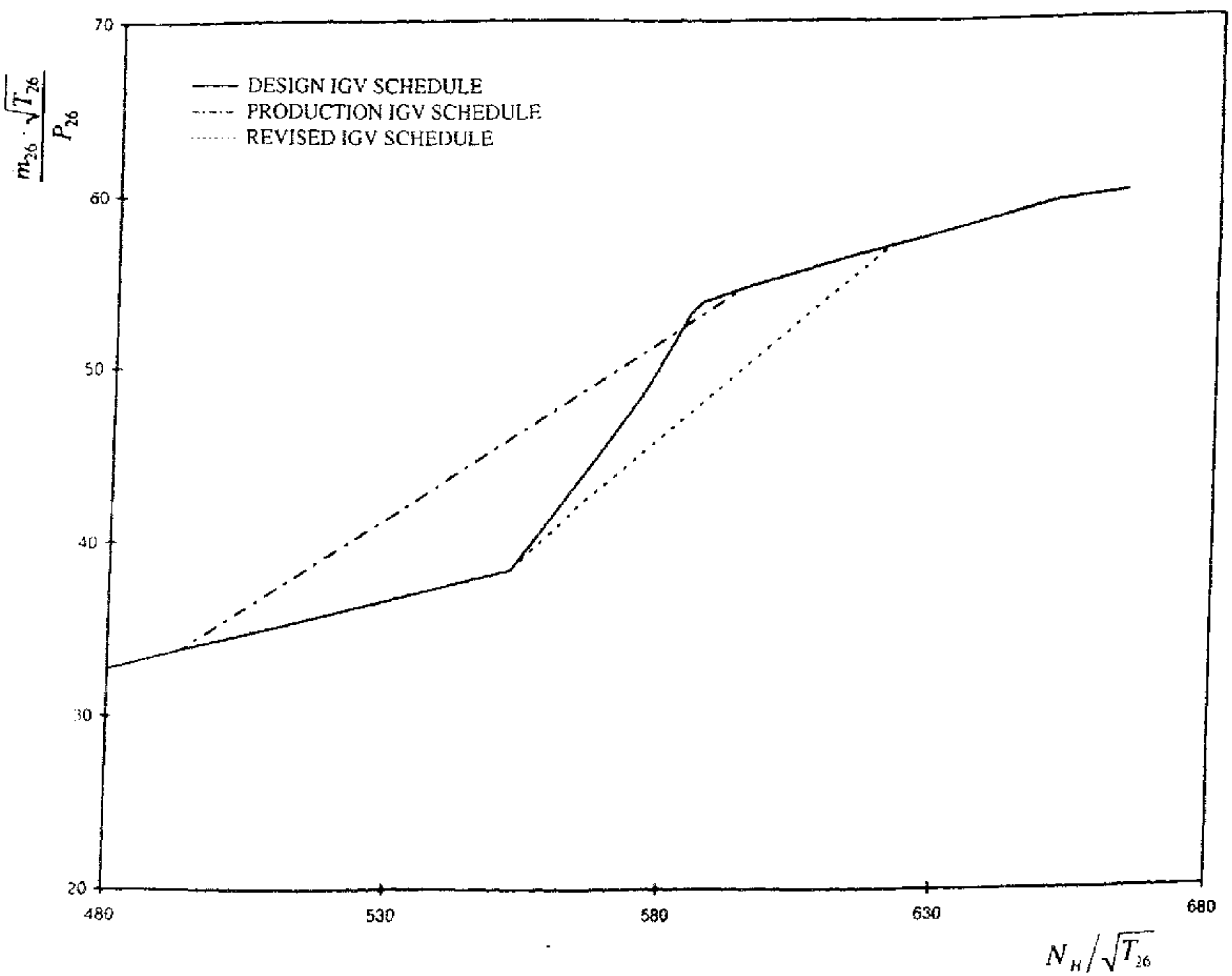


FIG. 6.2 Air flow-speed line characteristic of Tay engine

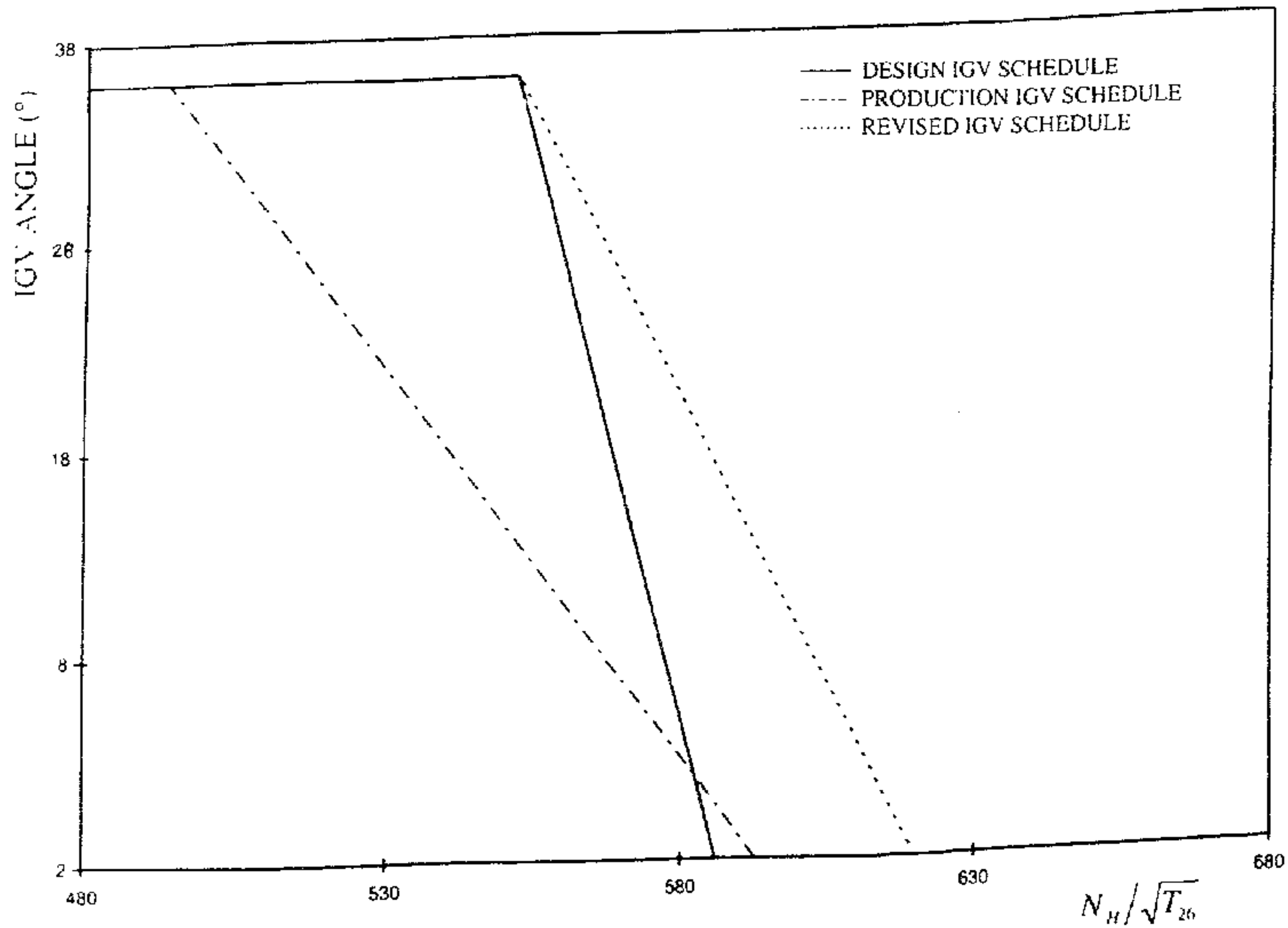


FIG. 6.3 IGV's turning angle schedules of the engine

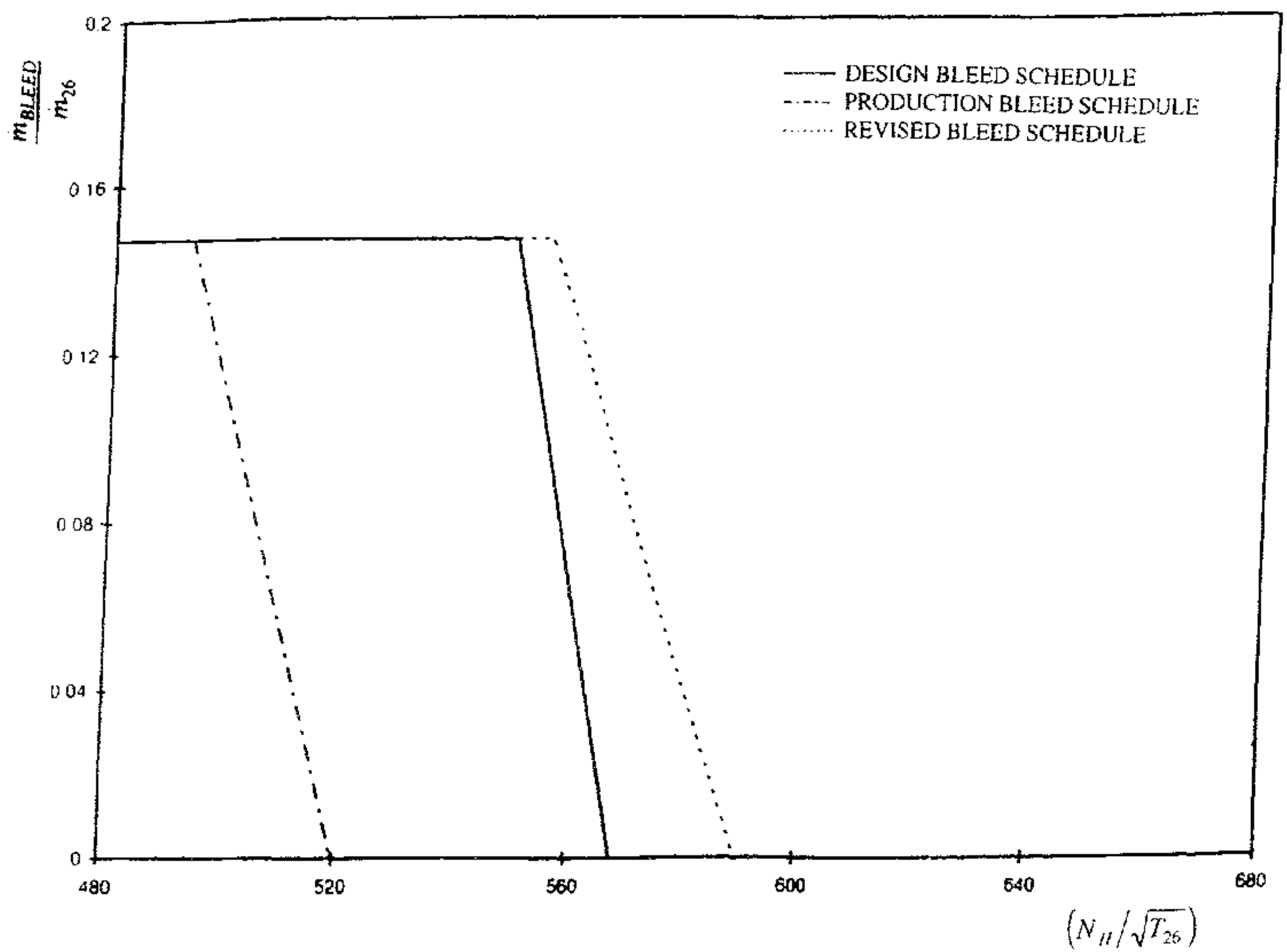


FIG. 6.4 Tay engine seventh stage bleed valve schedules

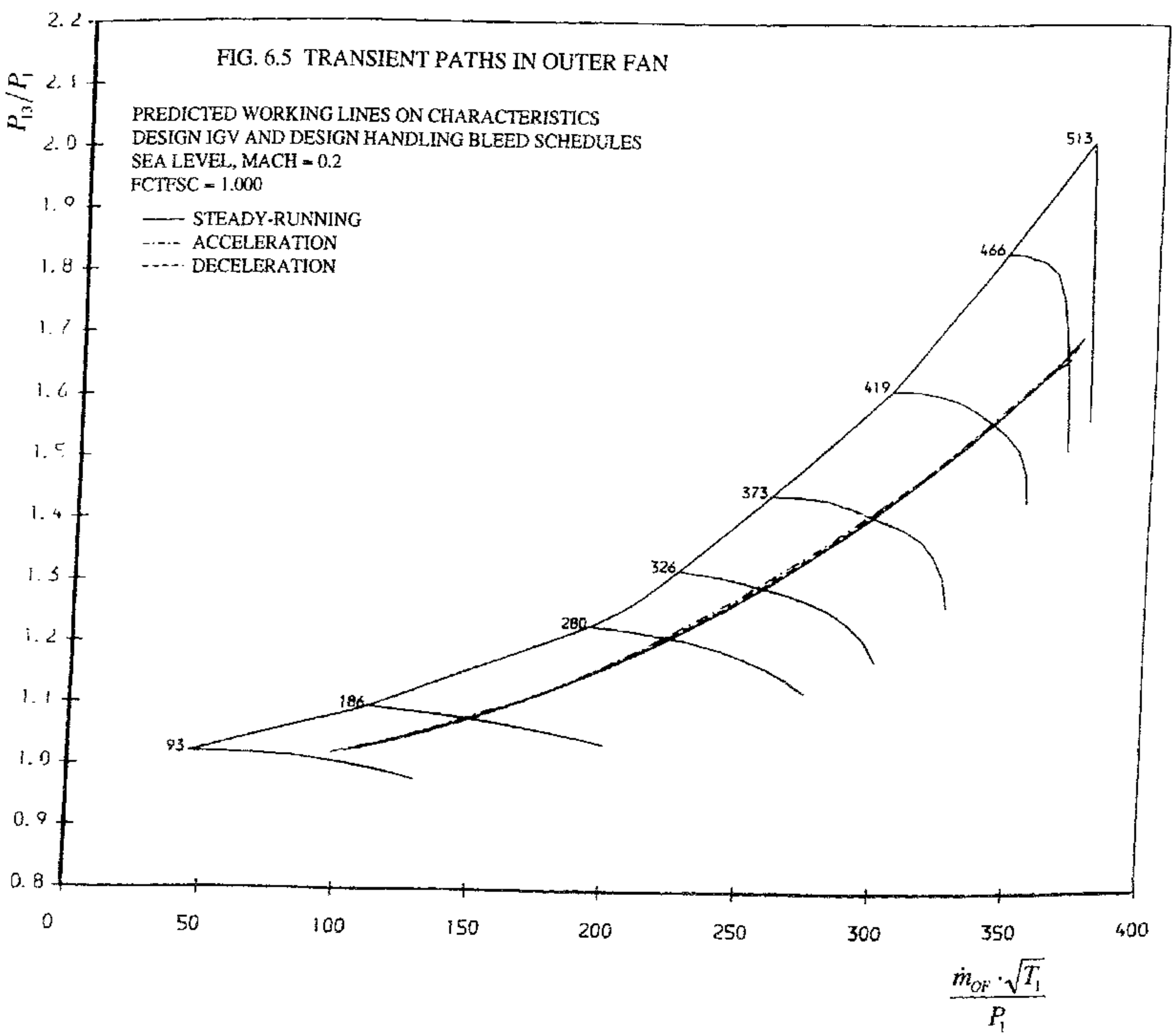


FIG. 6.5 TRANSIENT PATHS IN OUTER FAN
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{13}/P_1
$\frac{\dot{m}_{OF} \cdot \sqrt{T_1}}{P_1}$
93
186
280
326
373
419
466
513
0.8
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
2.0
2.1
2.2
0
50
100
150
200
250
300
350
400

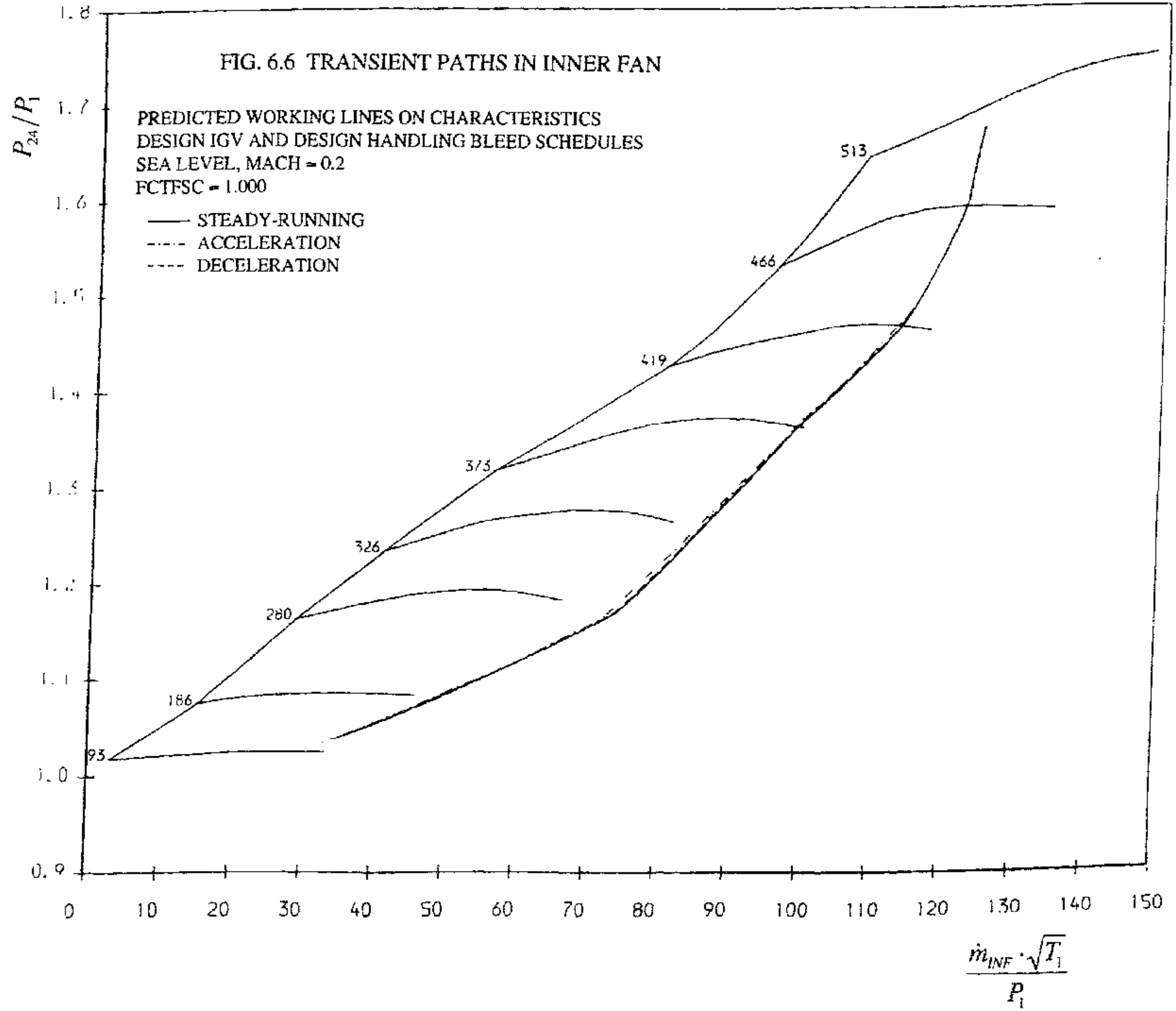
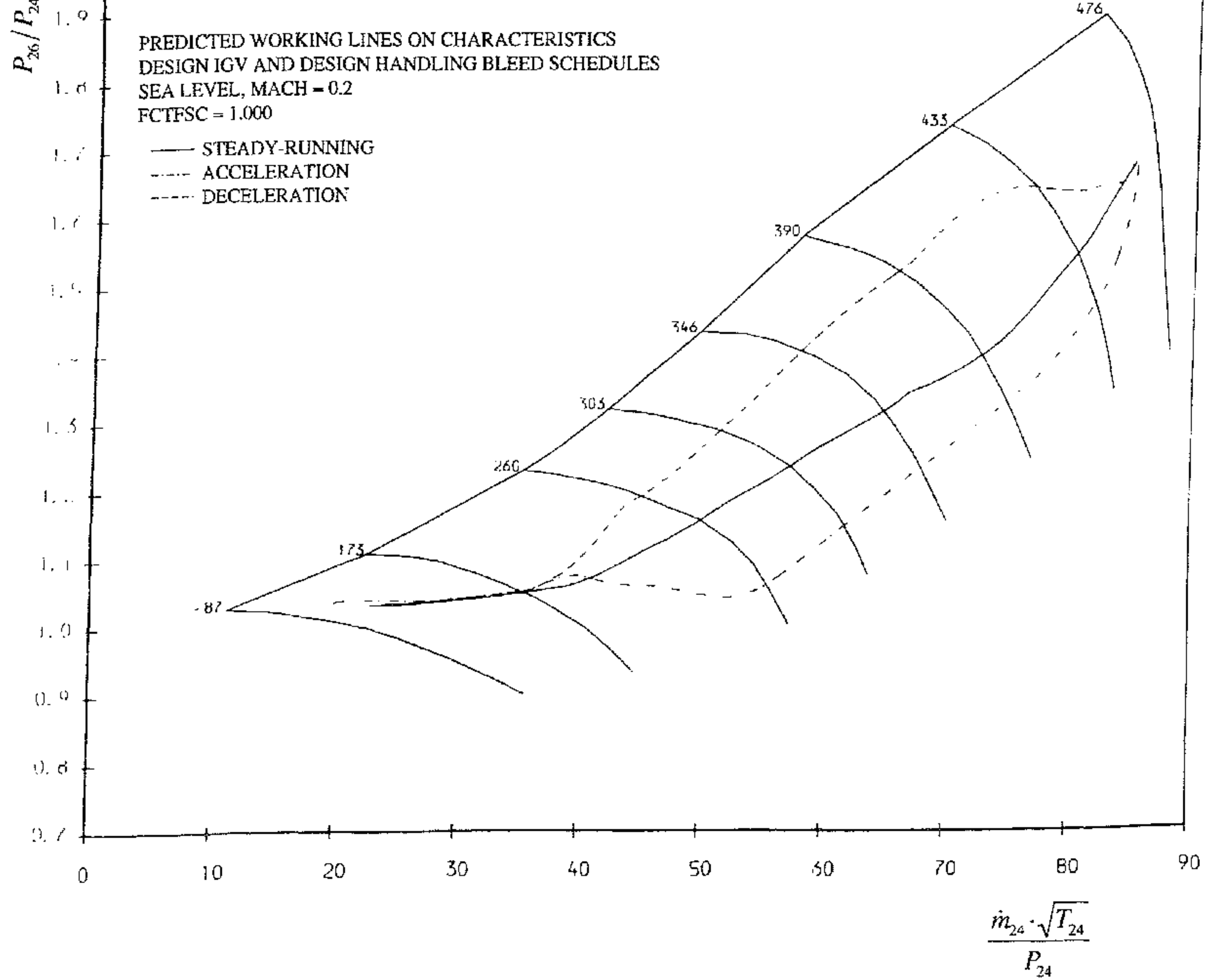


FIG. 6.6 TRANSIENT PATHS IN INNER FAN
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{24}/P_1
$\frac{\dot{m}_{INF} \cdot \sqrt{T_1}}{P_1}$
93
186
280
326
375
419
466
513
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150

FIG. 6.7 TRANSIENT PATHS IN I.P. COMPRESSOR



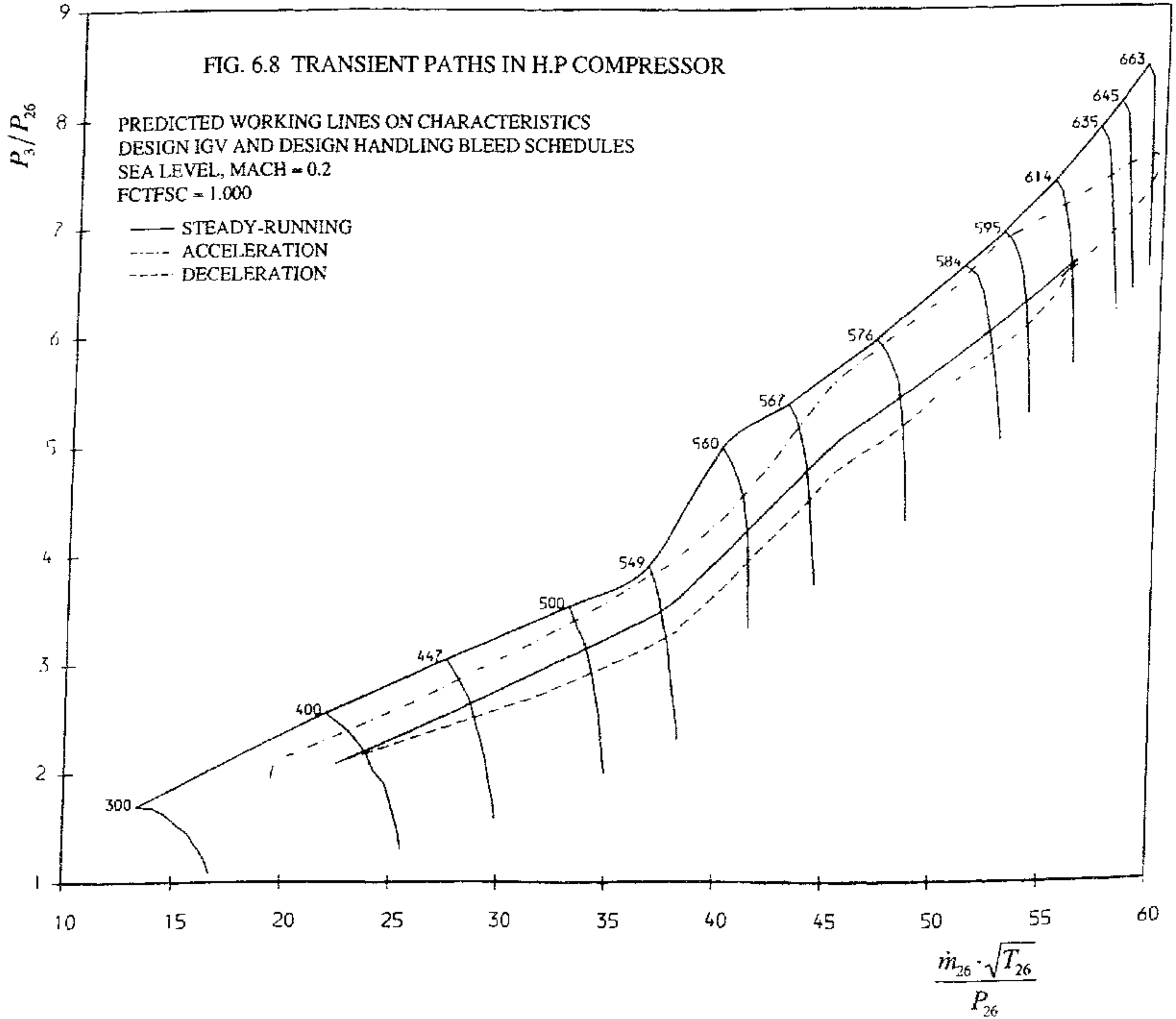


FIG. 6.8 TRANSIENT PATHS IN H.P COMPRESSOR
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_3/P_{26}
$\frac{\dot{m}_{26}\cdot\sqrt{T_{26}}}{P_{26}}$
300
400
447
500
549
560
567
576
584
595
614
635
645
663
1
2
3
4
5
6
7
8
9
10
15
20
25
30
35
40
45
50
55
60

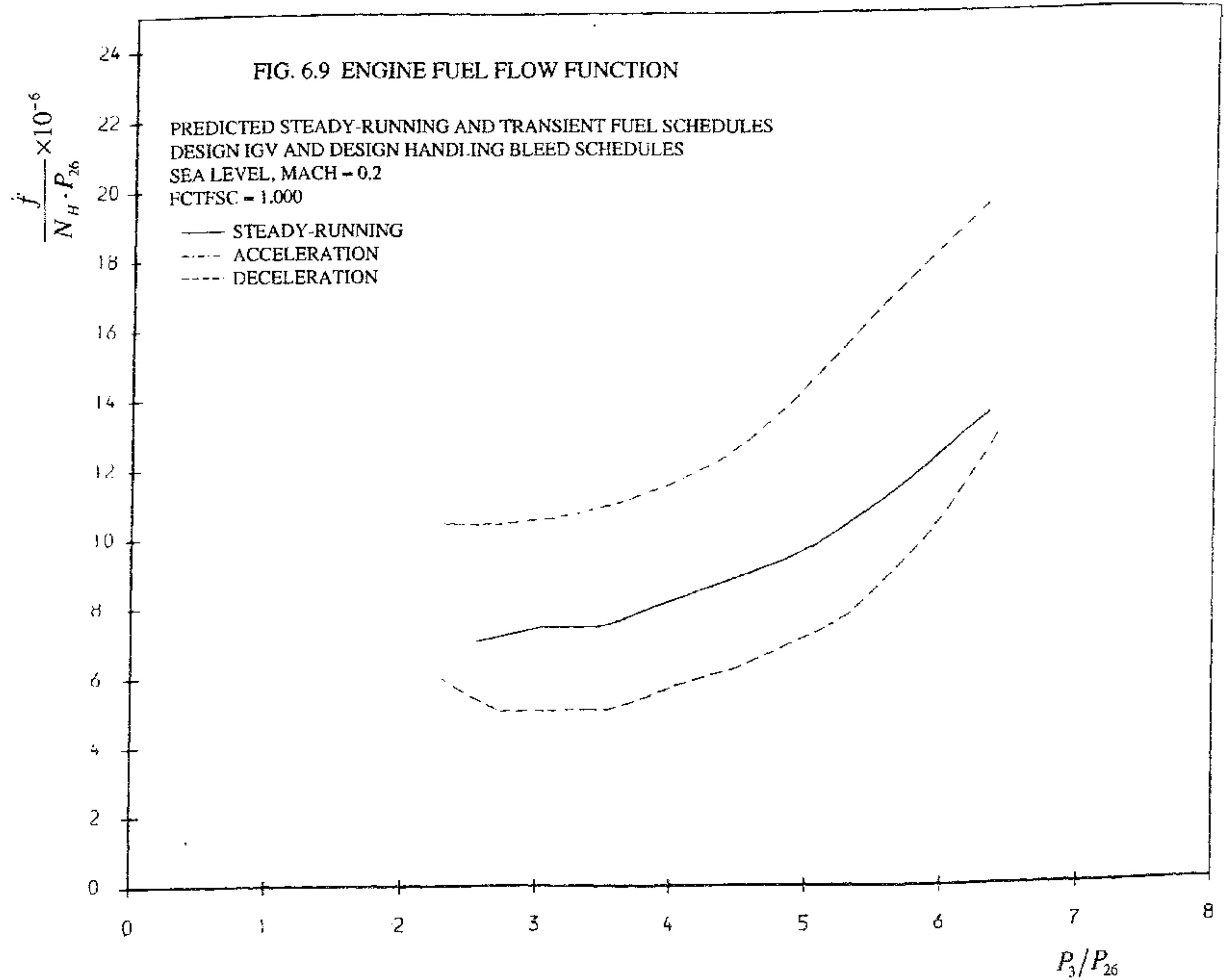


FIG. 6.9 ENGINE FUEL FLOW FUNCTION

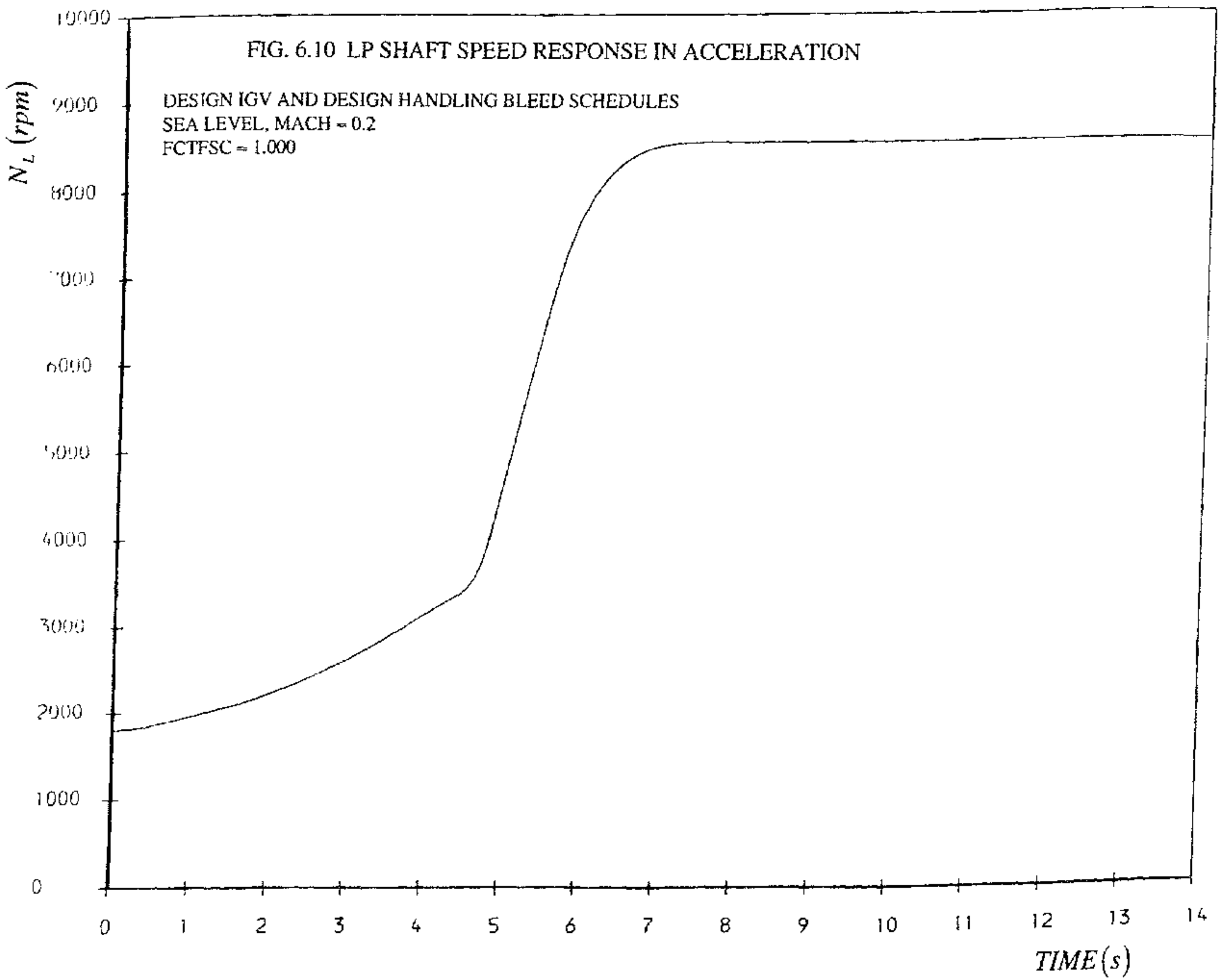


FIG. 6.10 LP SHAFT SPEED RESPONSE IN ACCELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
N_L (rpm)
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
TIME(s)

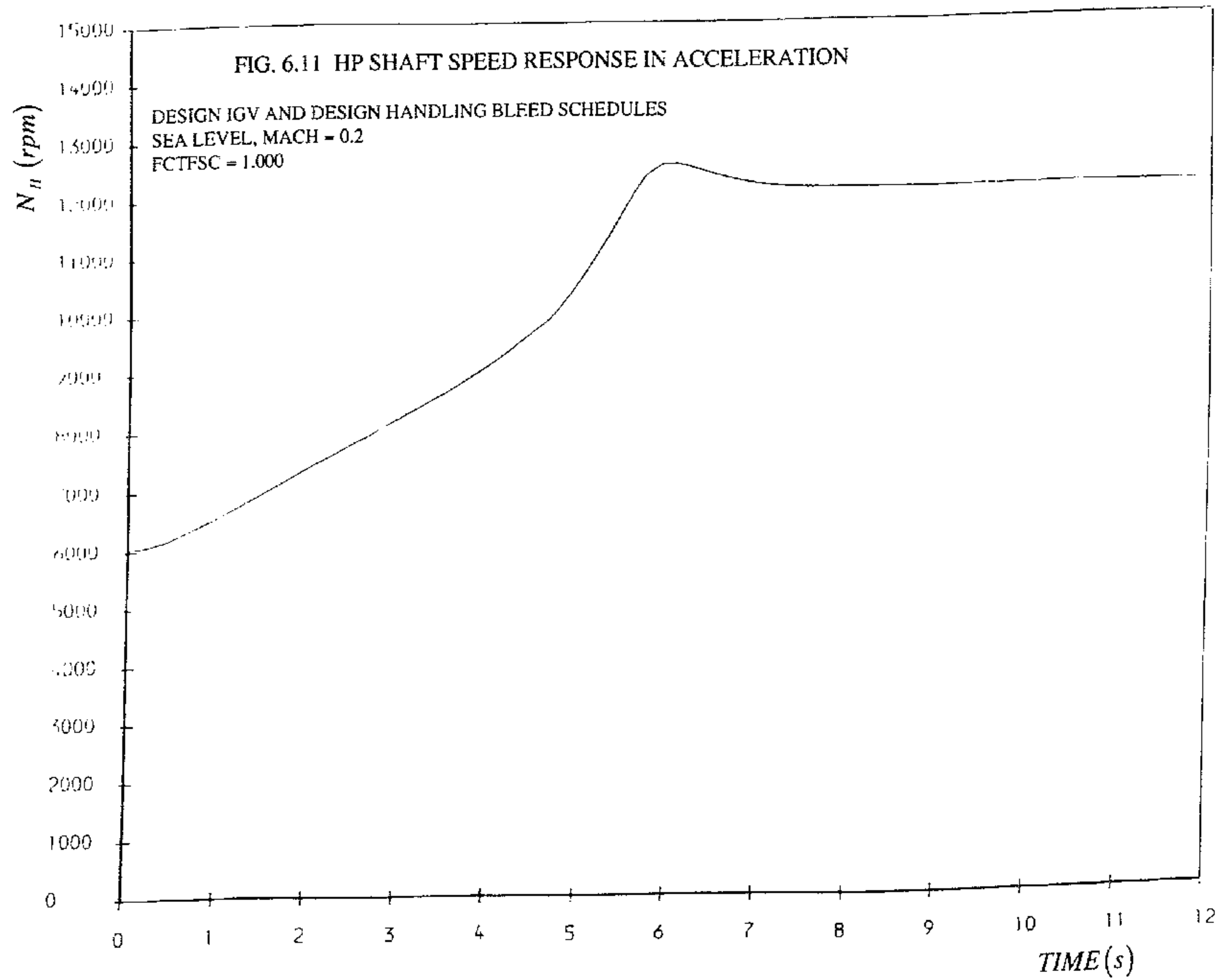


FIG. 6.11 HP SHAFT SPEED RESPONSE IN ACCELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
N_H (rpm)
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
0
1
2
3
4
5
6
7
8
9
10
11
12
TIME (s)

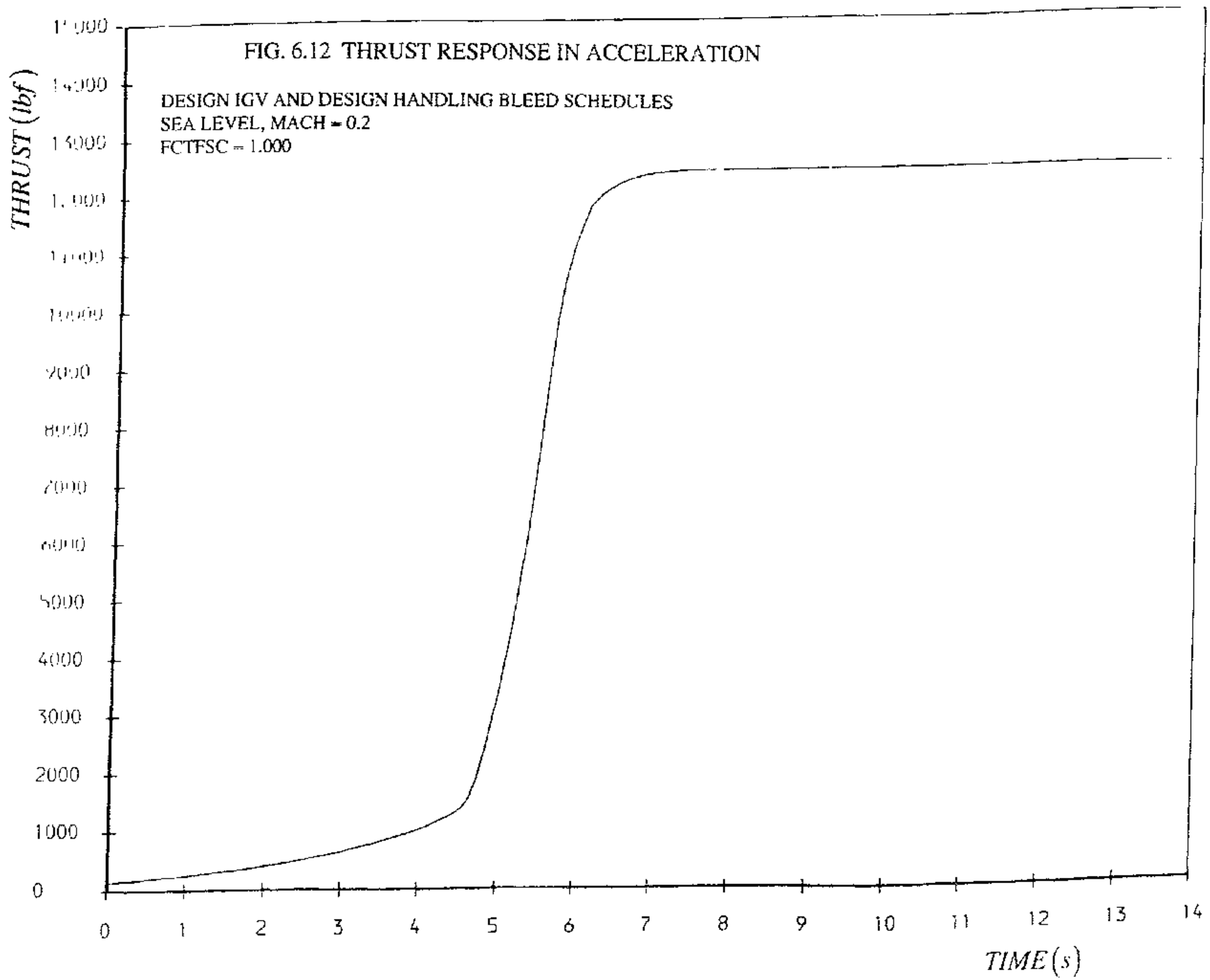


FIG. 6.12 THRUST RESPONSE IN ACCELERATION

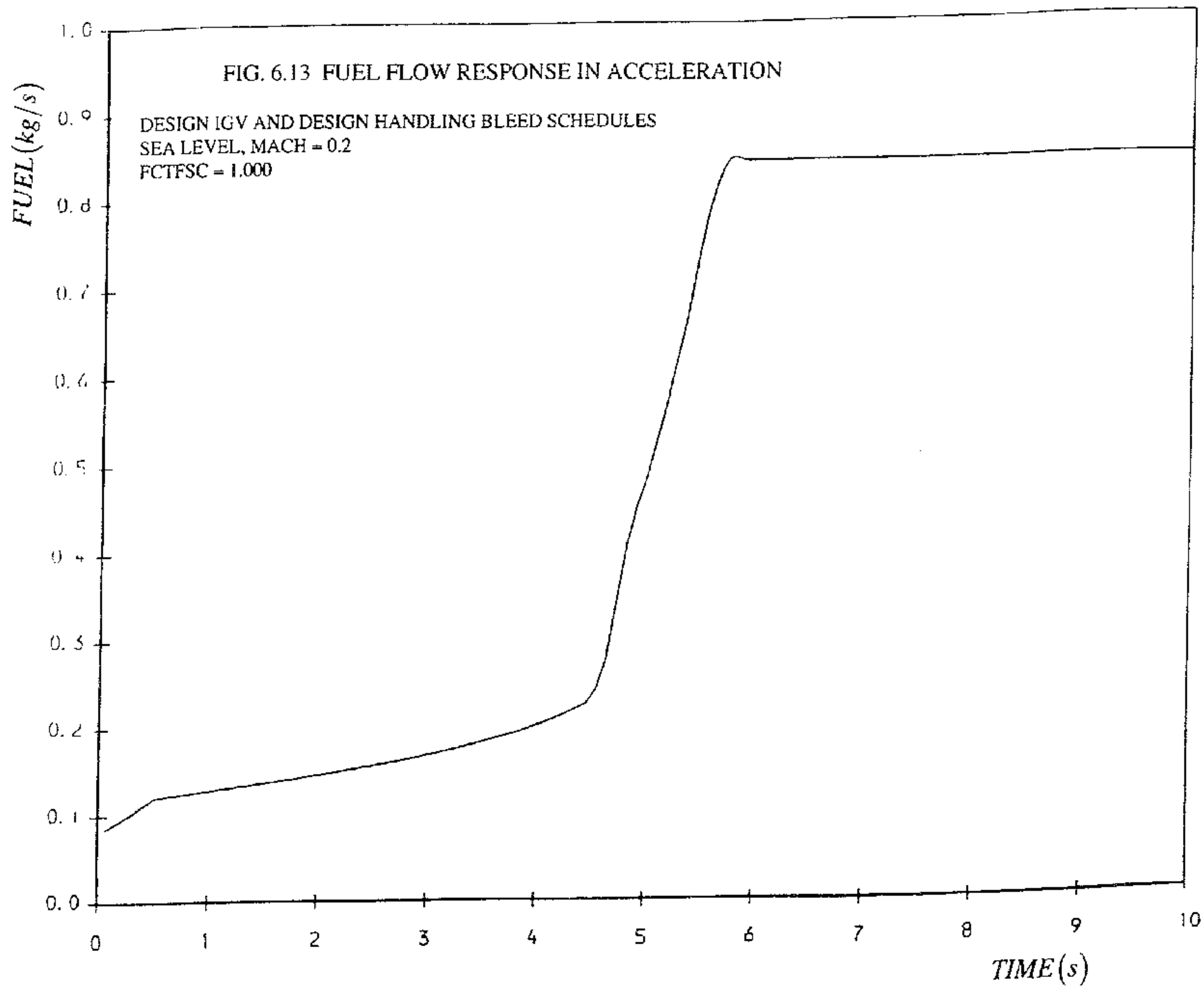


FIG. 6.13 FUEL FLOW RESPONSE IN ACCELERATION

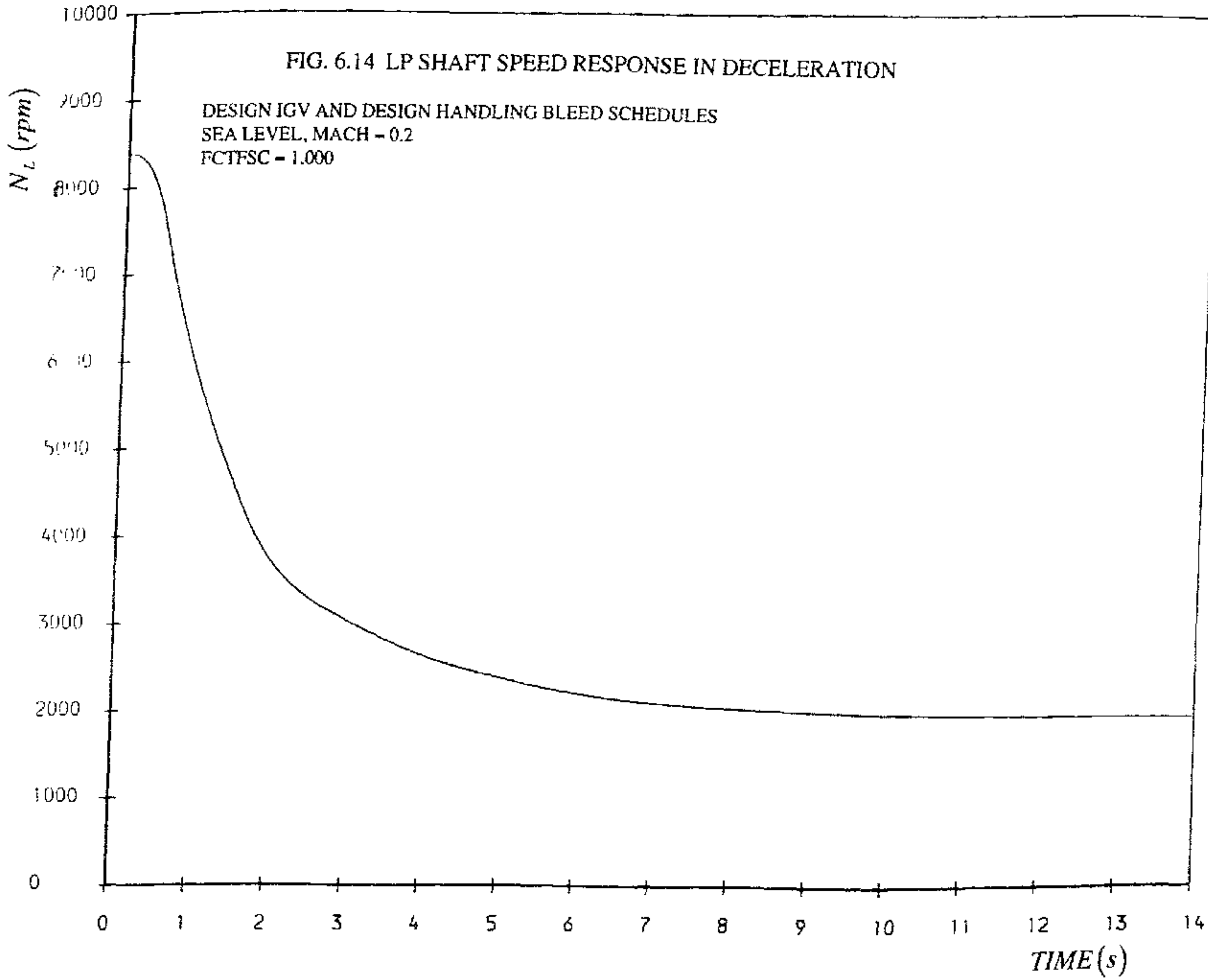


FIG. 6.14 LP SHAFT SPEED RESPONSE IN DECELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
N_L (rpm)
10000
1000
2000
3000
4000
5000
0
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
TIME (s)

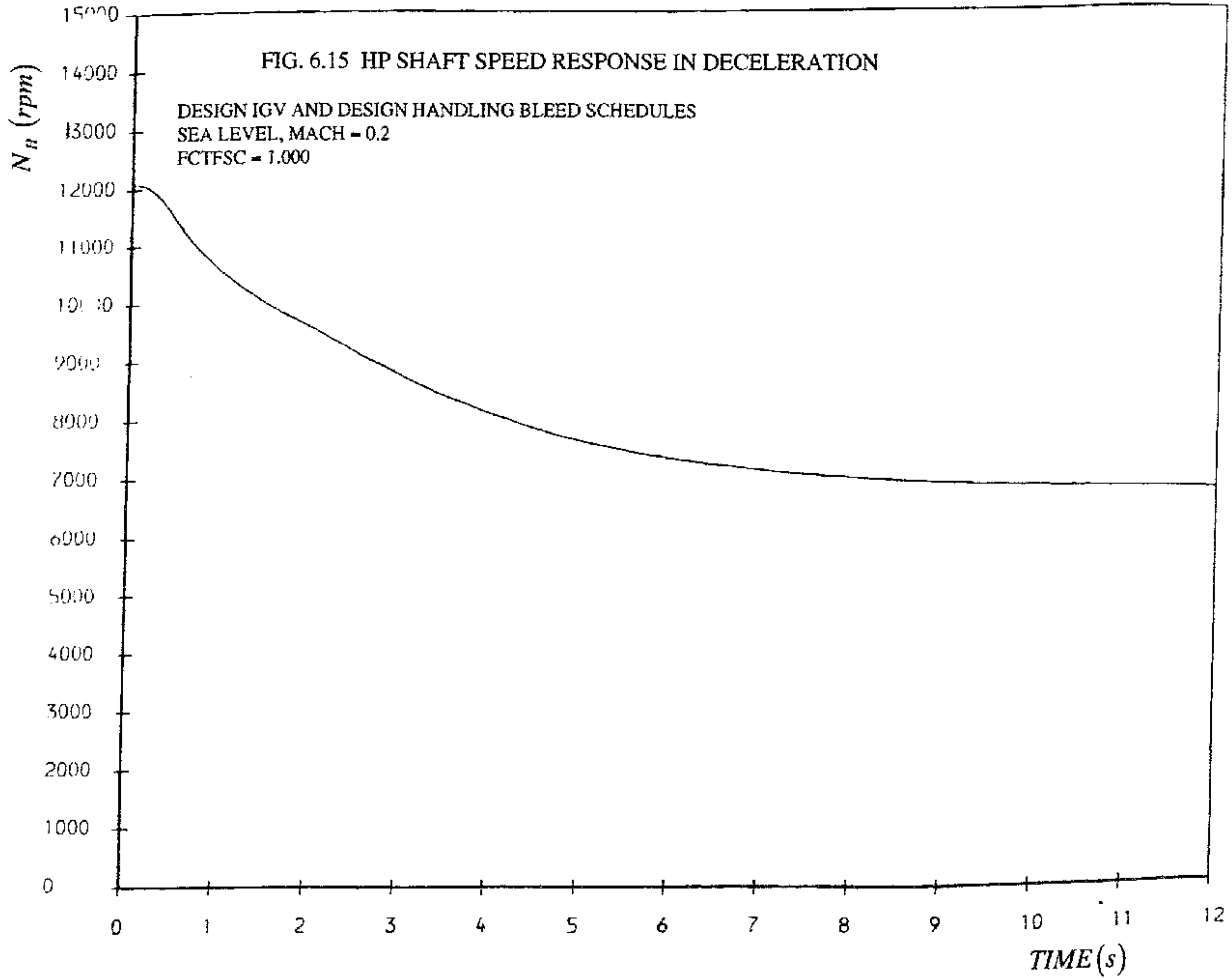


FIG. 6.15 HP SHAFT SPEED RESPONSE IN DECELERATION

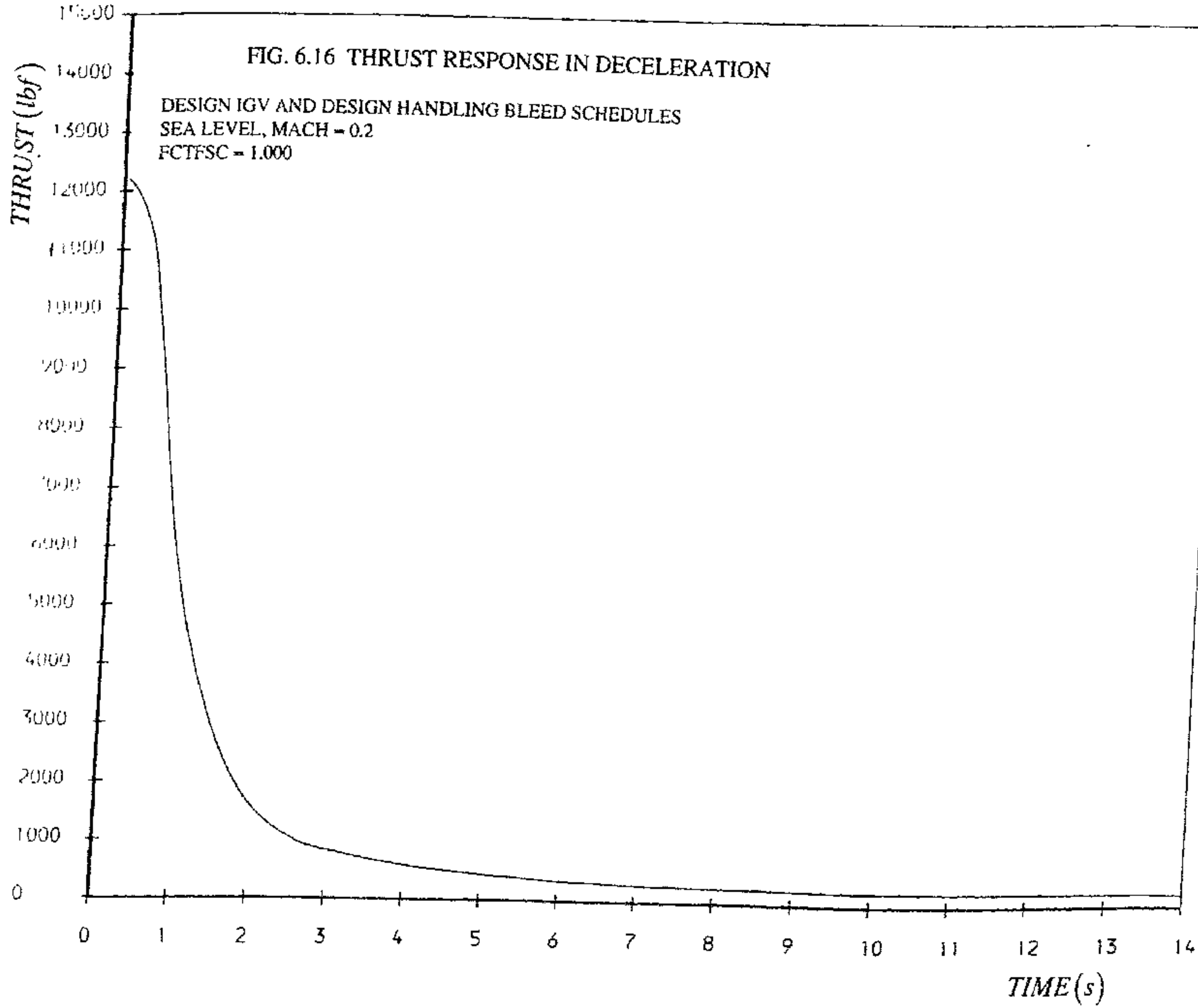


FIG. 6.16 THRUST RESPONSE IN DECELERATION

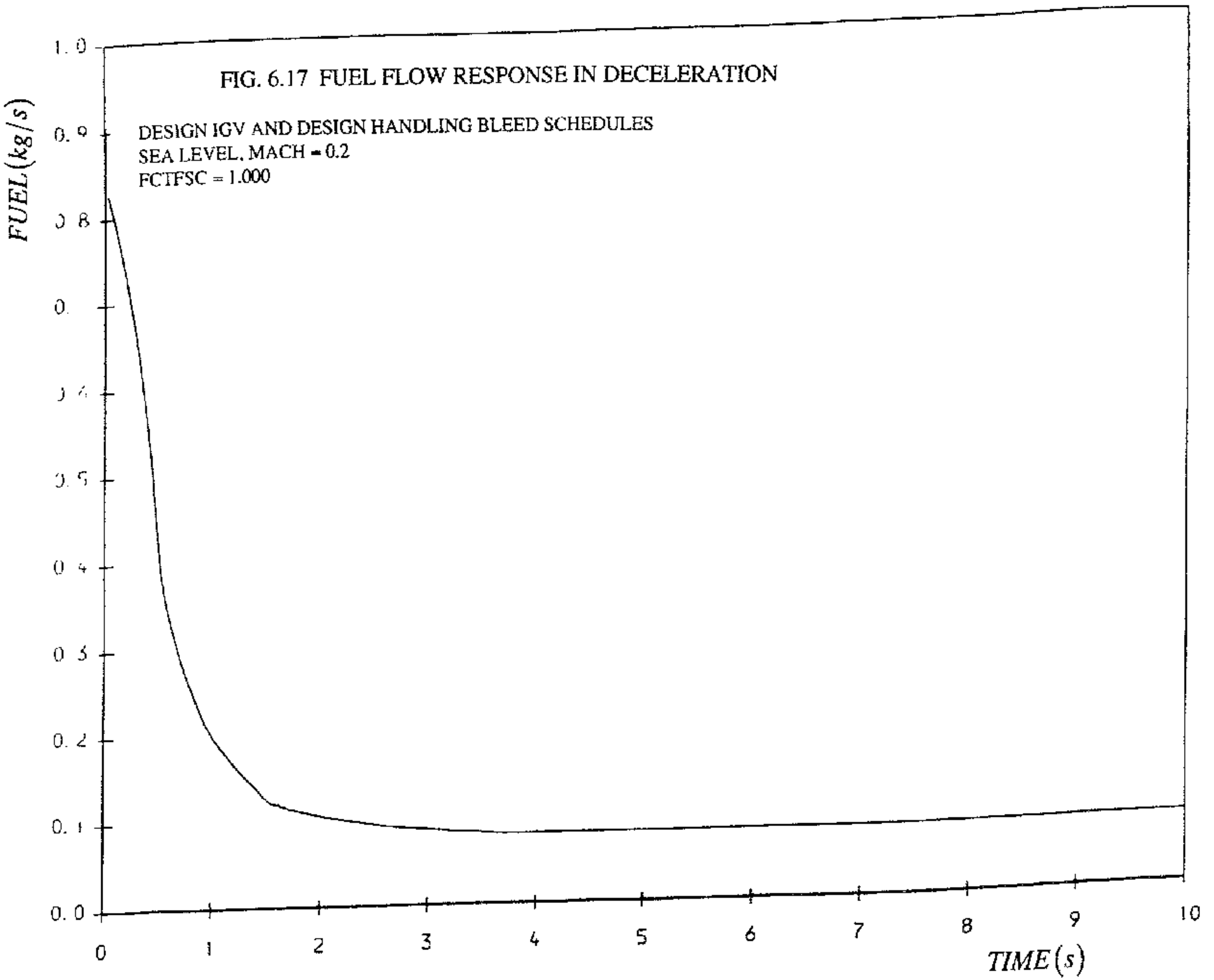


FIG. 6.17 FUEL FLOW RESPONSE IN DECELERATION

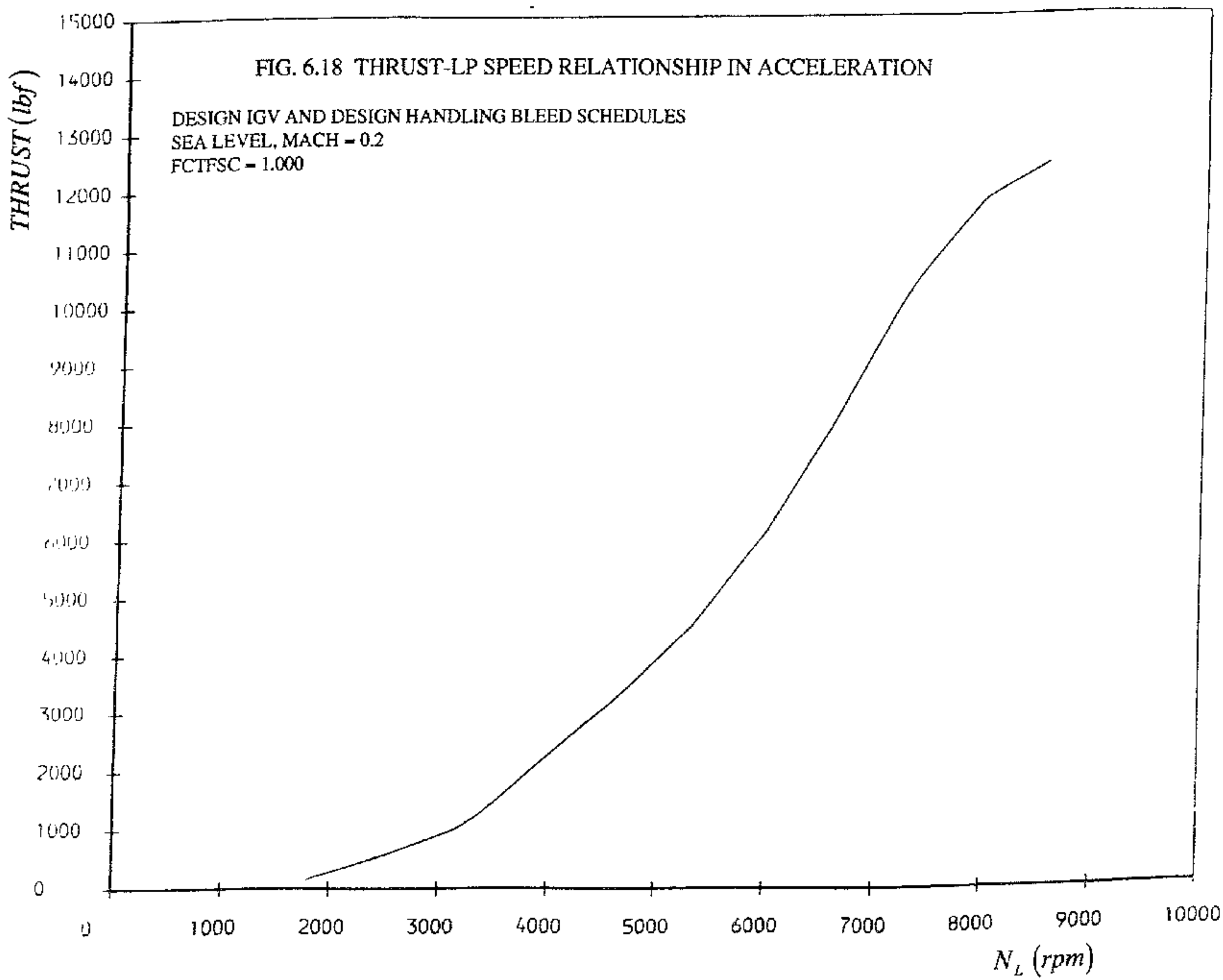


FIG. 6.18 THRUST-LP SPEED RELATIONSHIP IN ACCELERATION

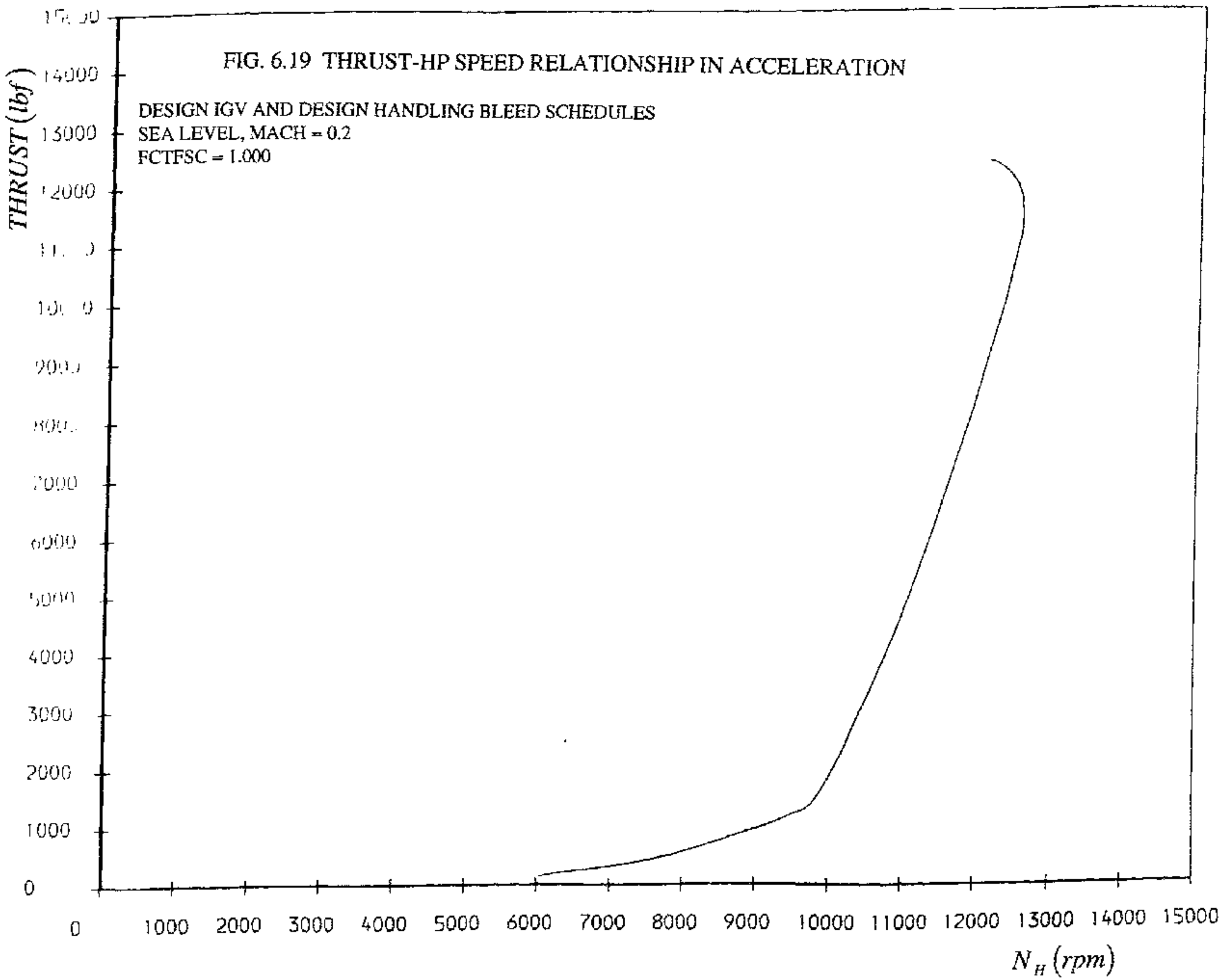


FIG. 6.19 THRUST-HP SPEED RELATIONSHIP IN ACCELERATION

FIG. 6.20 THRUST-LP SPEED RELATIONSHIP IN DECELERATION

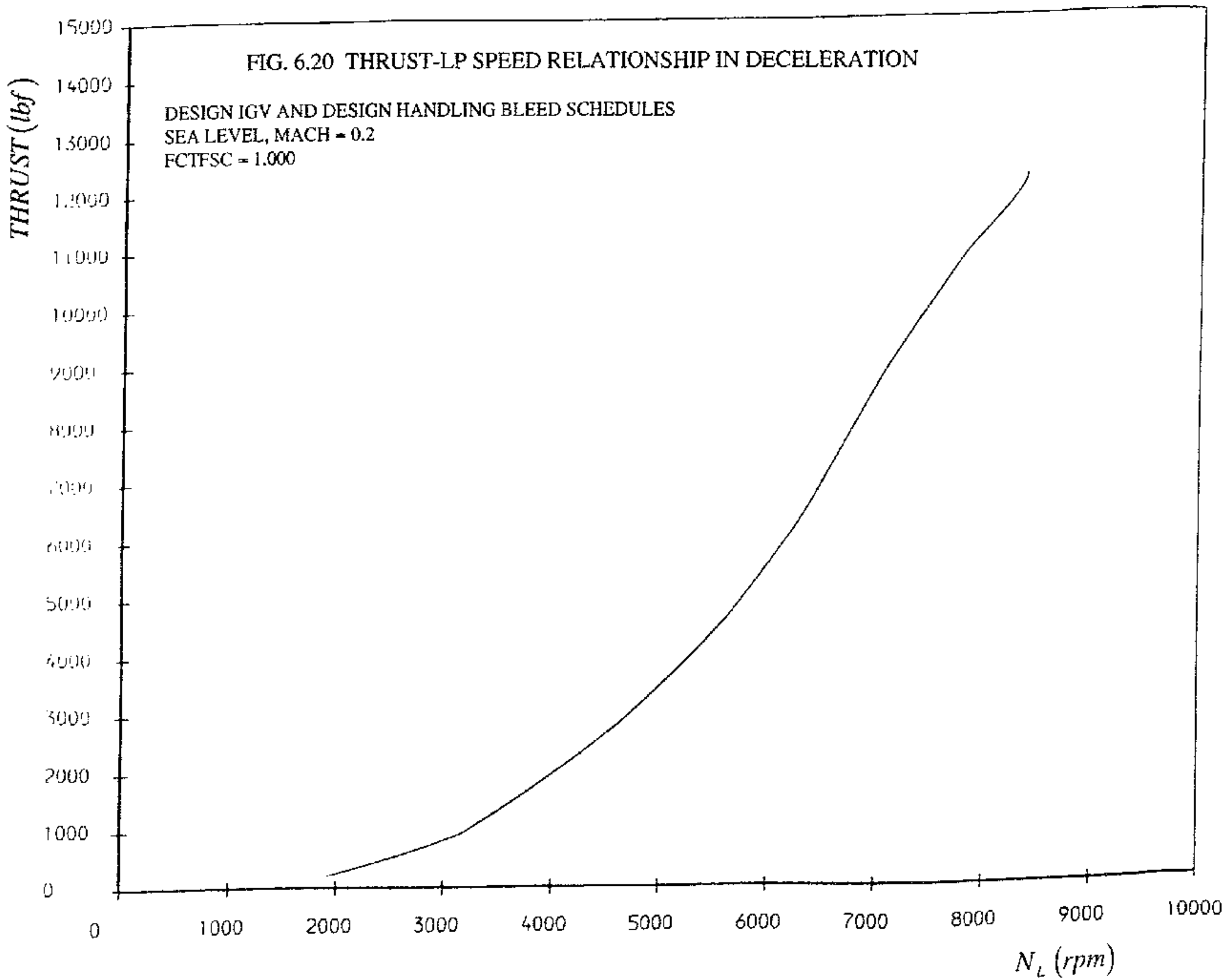
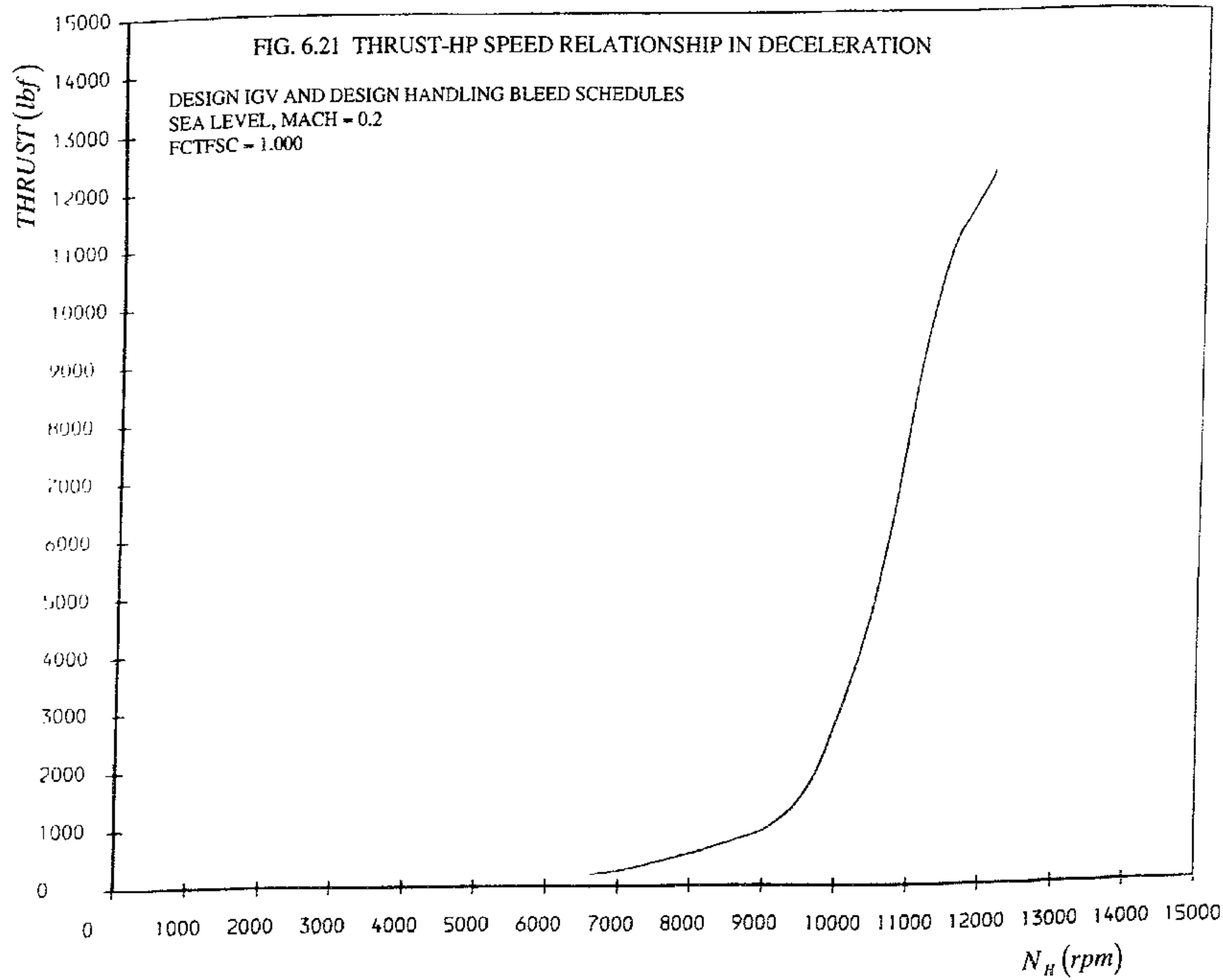


FIG. 6.21 THRUST-HP SPEED RELATIONSHIP IN DECELERATION



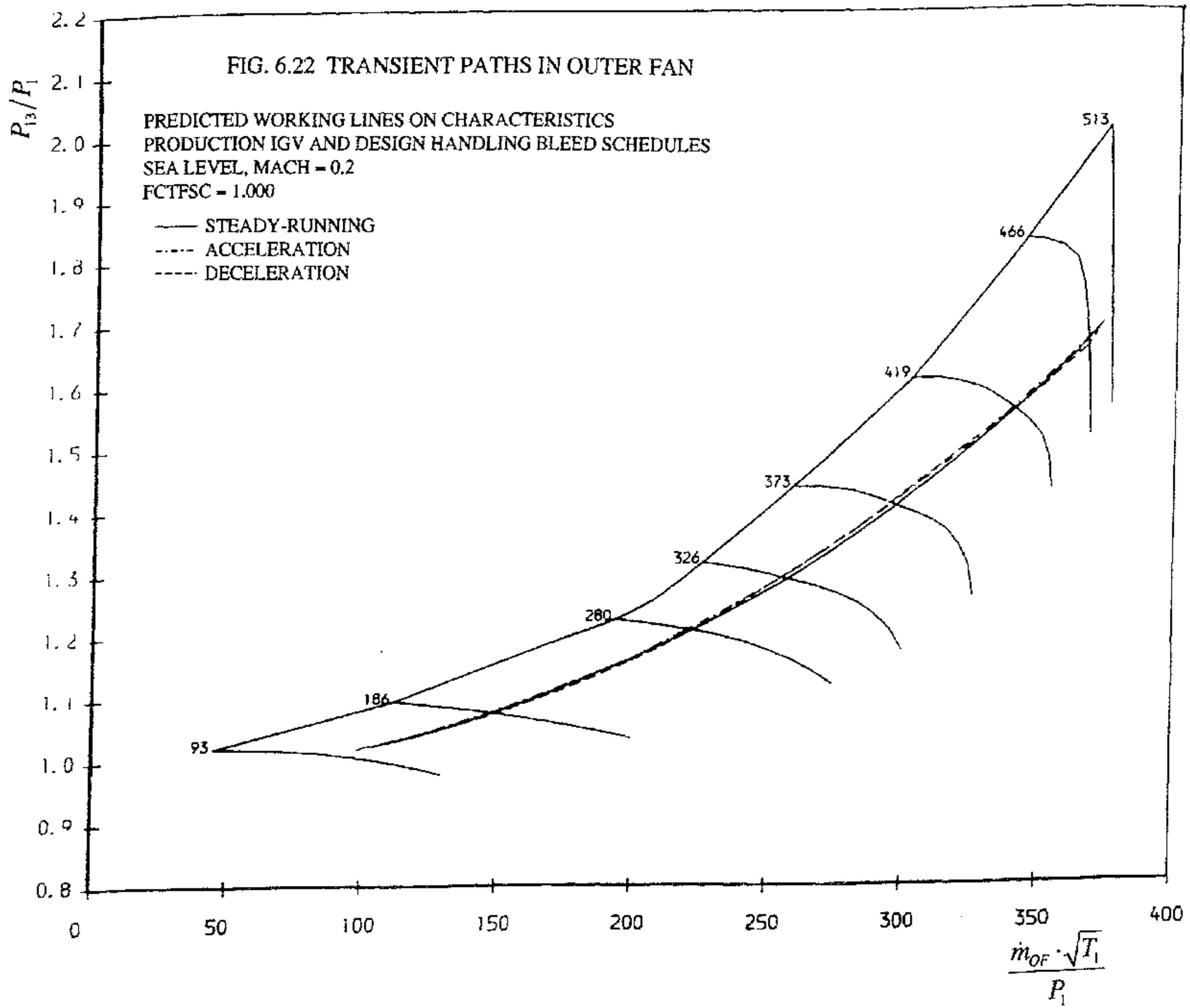


FIG. 6.22 TRANSIENT PATHS IN OUTER FAN
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{13}/P_1
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0
50
100
150
200
250
300
350
400
$\frac{\dot{m}_{OF} \cdot \sqrt{T_1}}{P_1}$
93
186
280
326
373
419
466
513

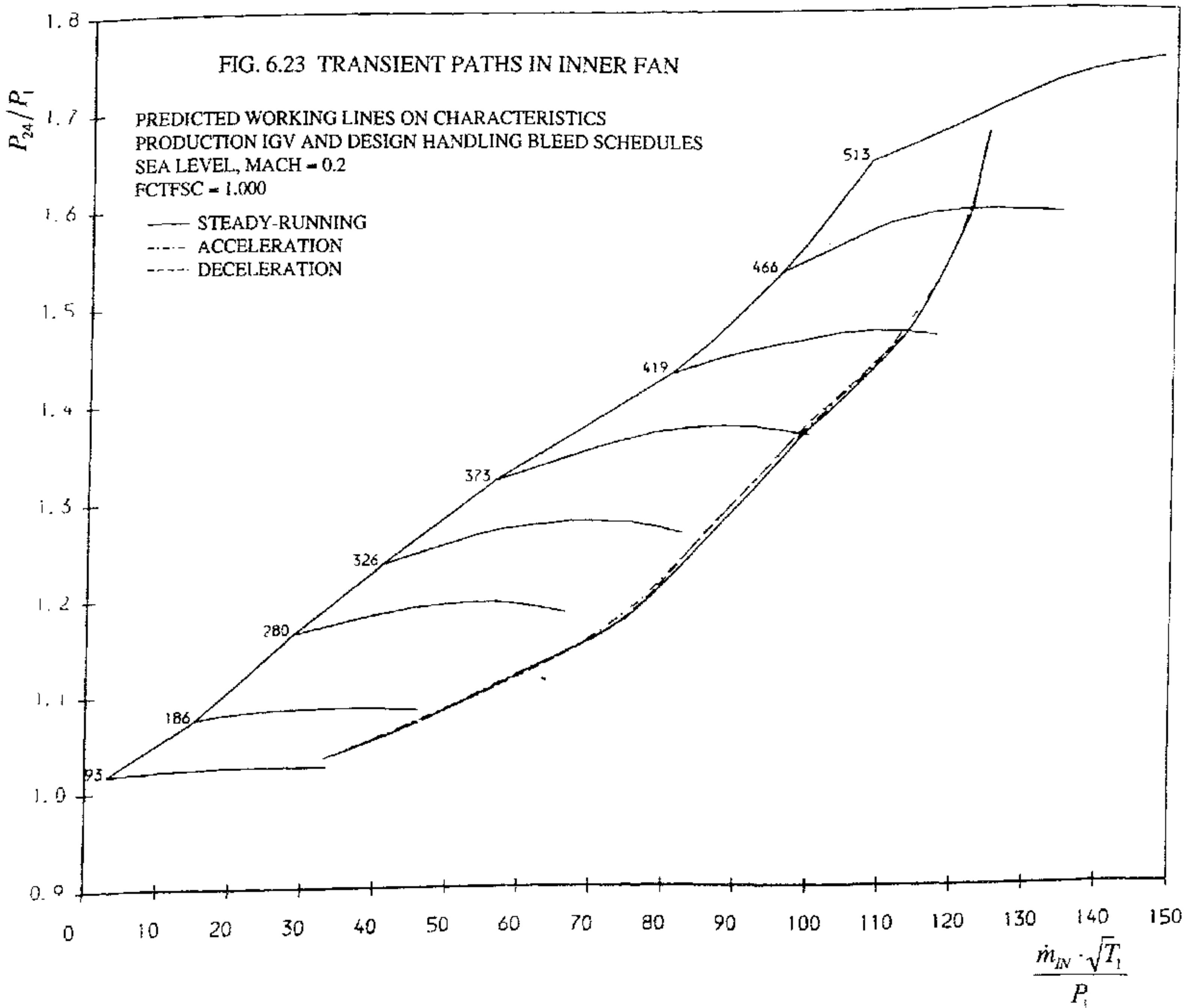


FIG. 6.23 TRANSIENT PATHS IN INNER FAN

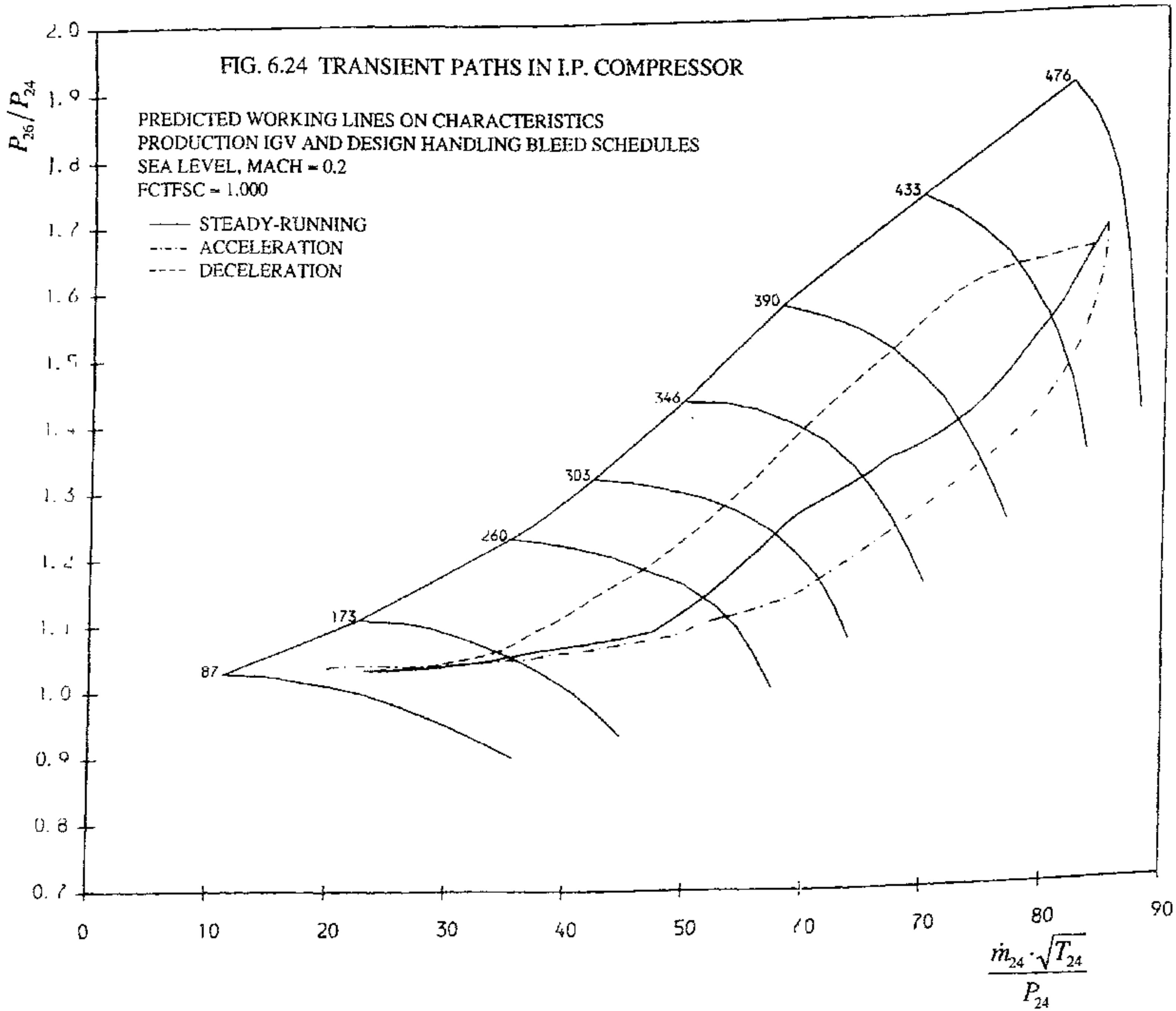


FIG. 6.24 TRANSIENT PATHS IN I.P. COMPRESSOR
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
476
433
390
346
303
260
173
87
P_{26}/P_{24}
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0
10
20
30
40
50
60
70
80
90
$\frac{\dot{m}_{24} \cdot \sqrt{T_{24}}}{P_{24}}$

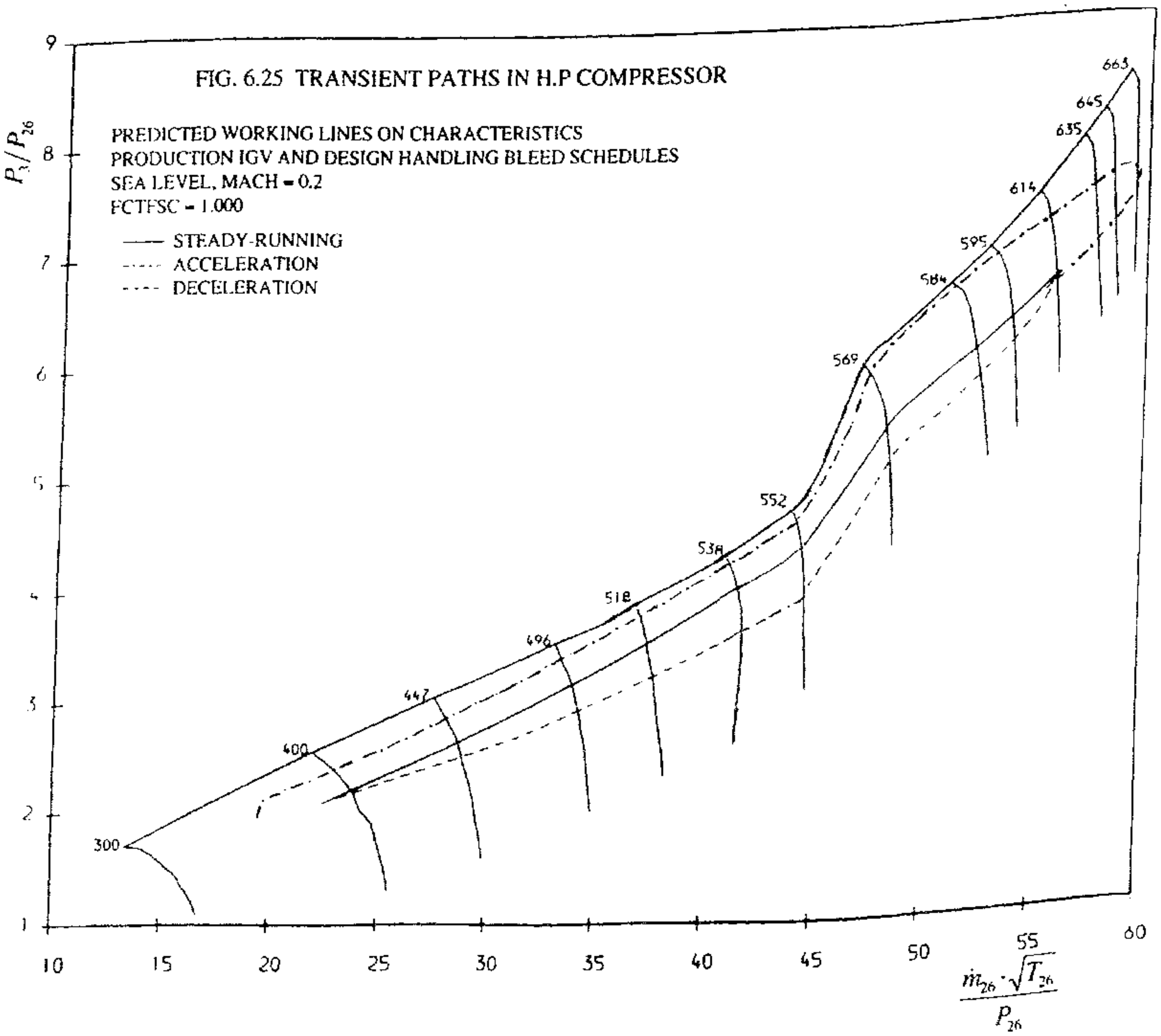


FIG. 6.25 TRANSIENT PATHS IN H.P COMPRESSOR
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_3/P_{26}
$\frac{\dot{m}_{26}\cdot\sqrt{T_{26}}}{P_{26}}$
300
400
447
496
518
538
552
569
584
595
614
635
645
663
1
2
3
4
5
6
7
8
9
10
15
20
25
30
35
40
45
50
55
60

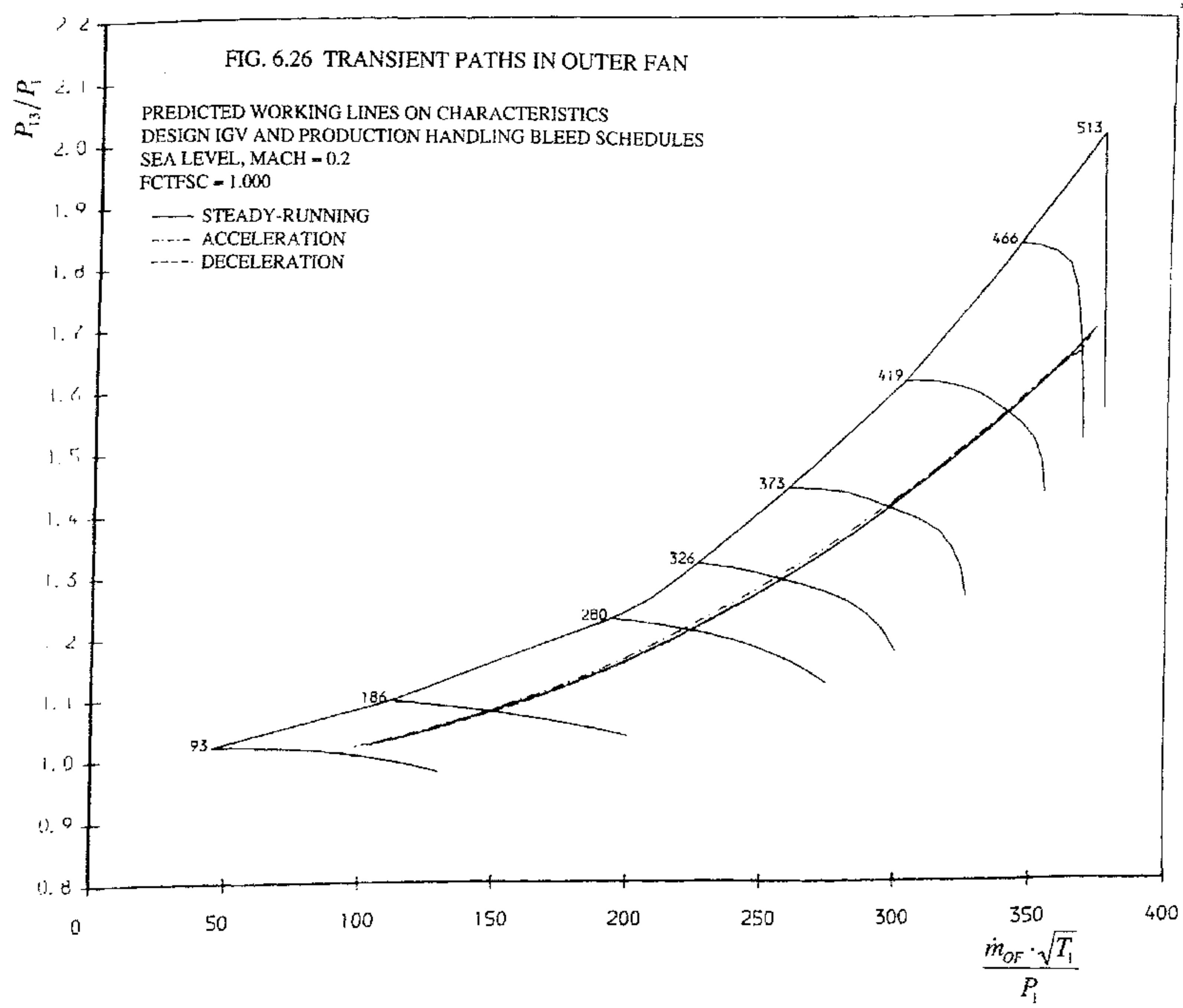


FIG. 6.26 TRANSIENT PATHS IN OUTER FAN
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
513
466
419
373
326
280
186
93
P_{13}/P_1
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0
50
100
150
200
250
300
350
400
$\frac{\dot{m}_{OF} \cdot \sqrt{T_1}}{P_1}$

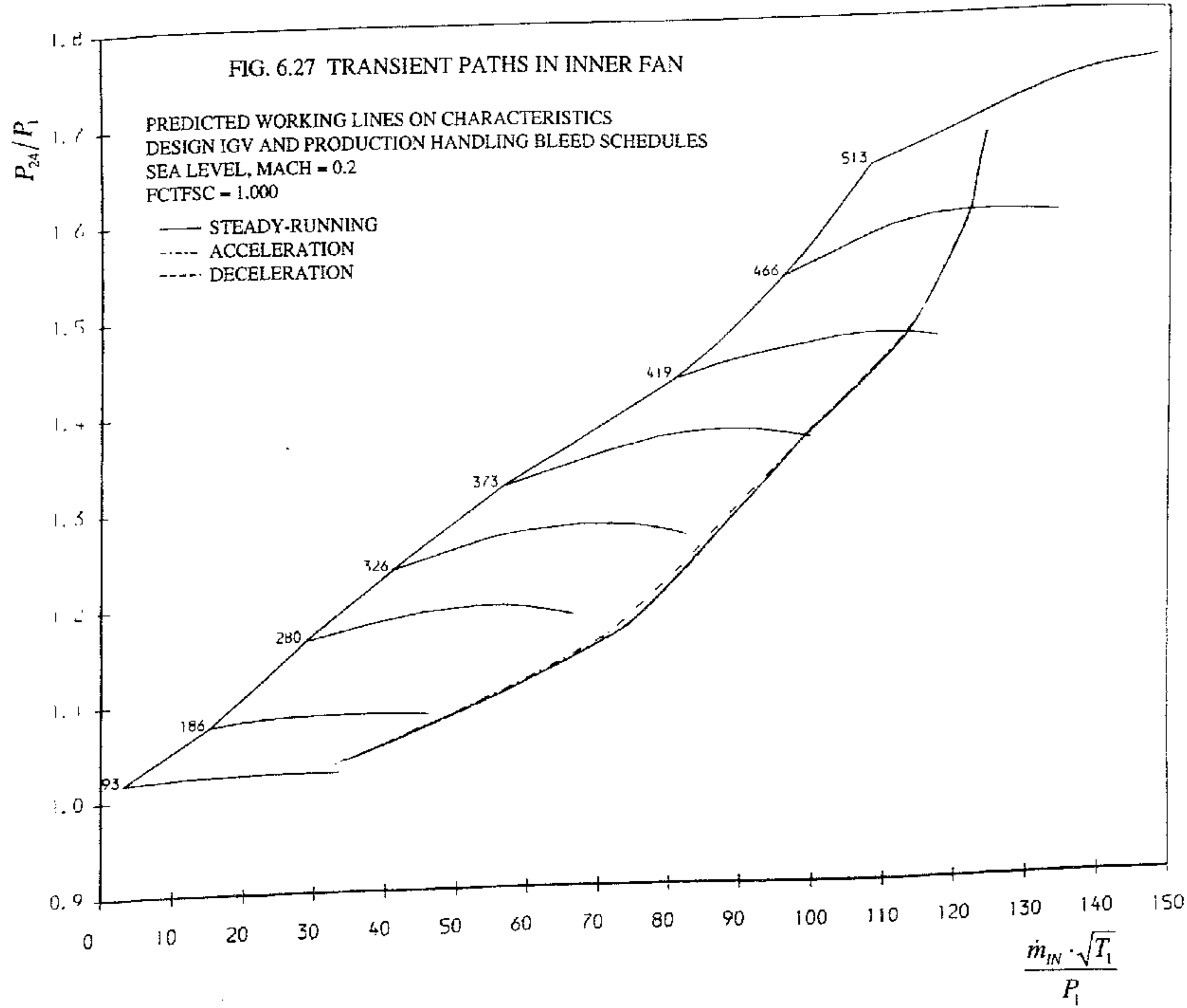


FIG. 6.27 TRANSIENT PATHS IN INNER FAN
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P24/P1
513
466
419
373
326
280
186
93
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150

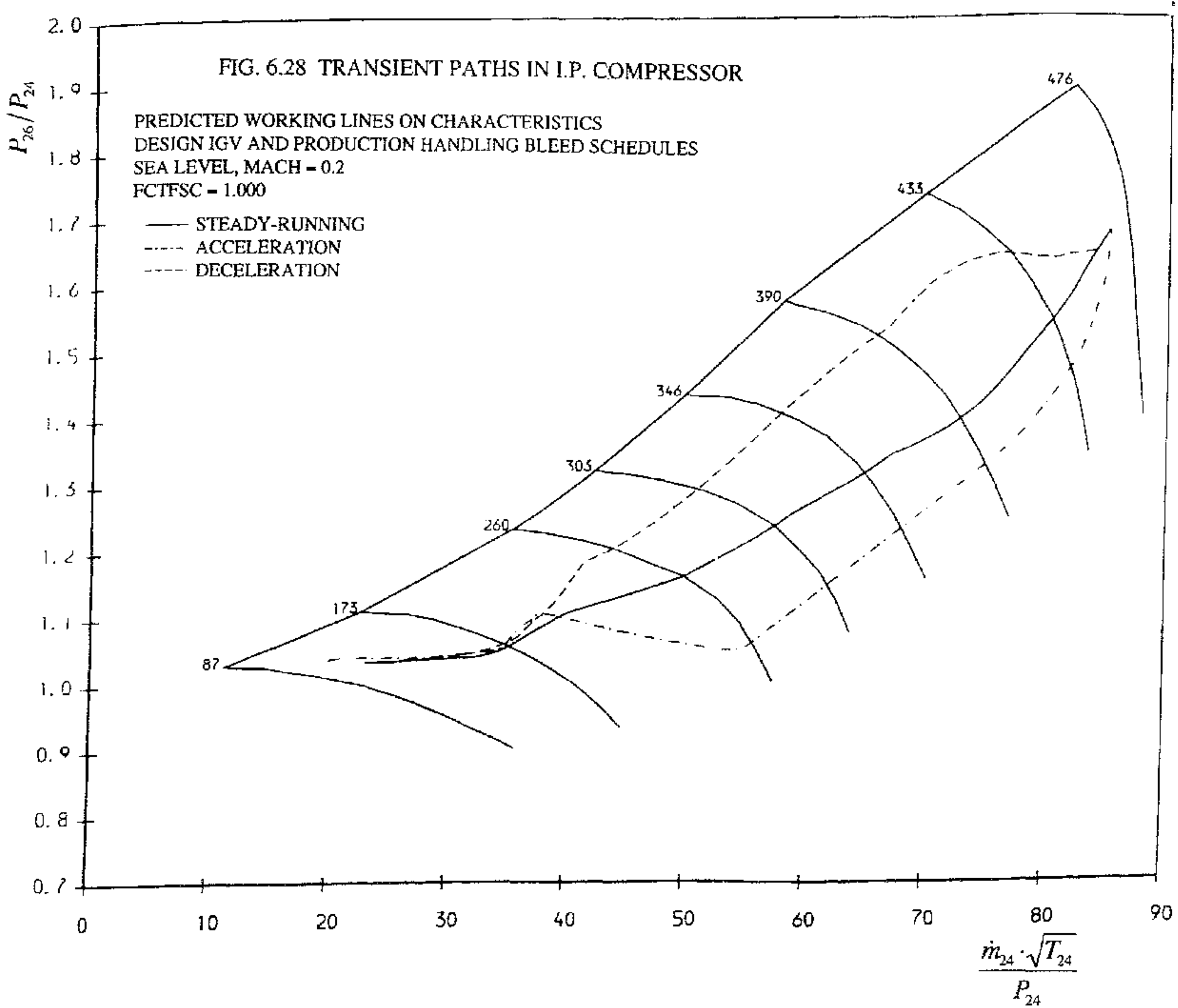


FIG. 6.28 TRANSIENT PATHS IN I.P. COMPRESSOR

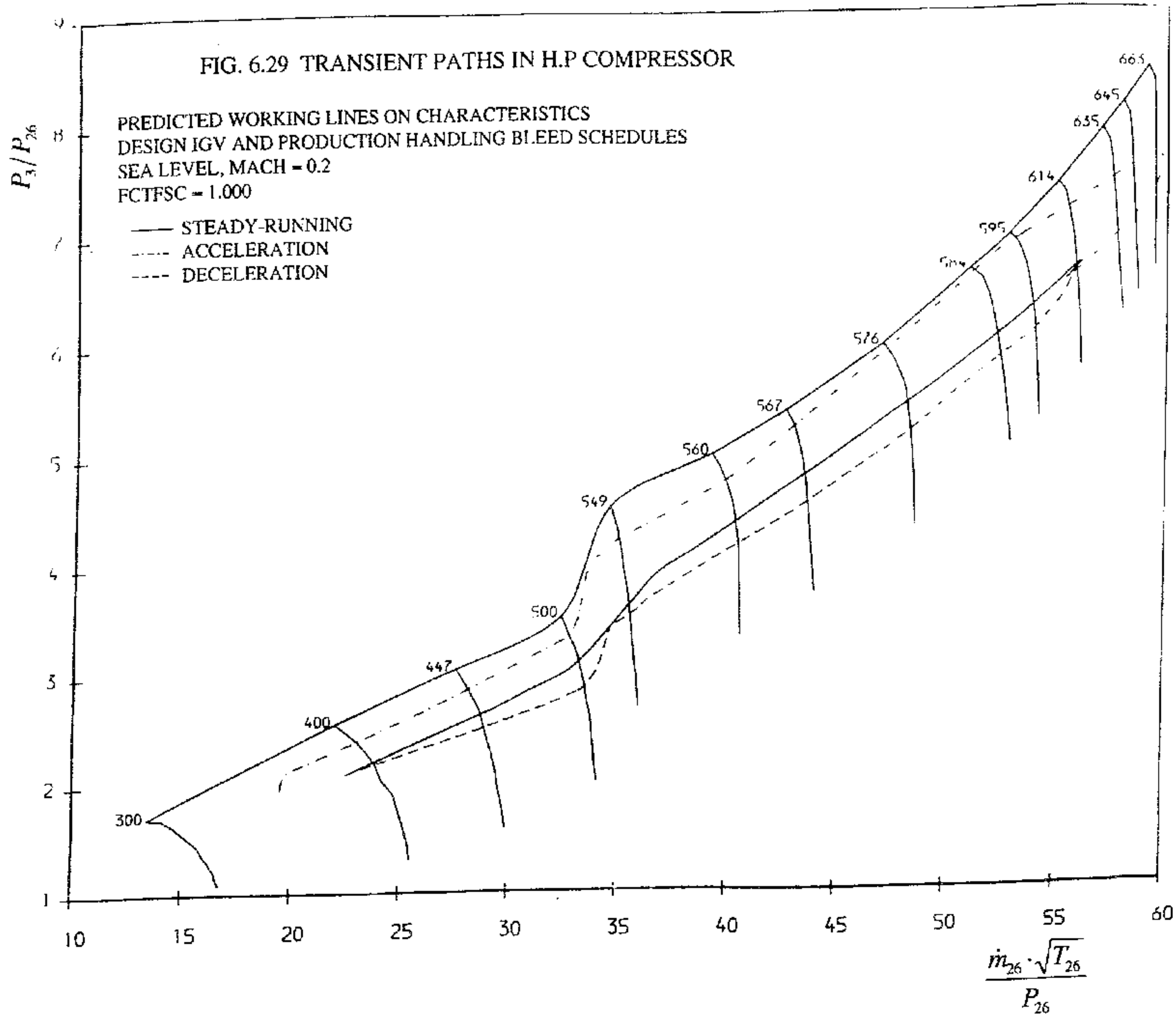


FIG. 6.29 TRANSIENT PATHS IN H.P COMPRESSOR

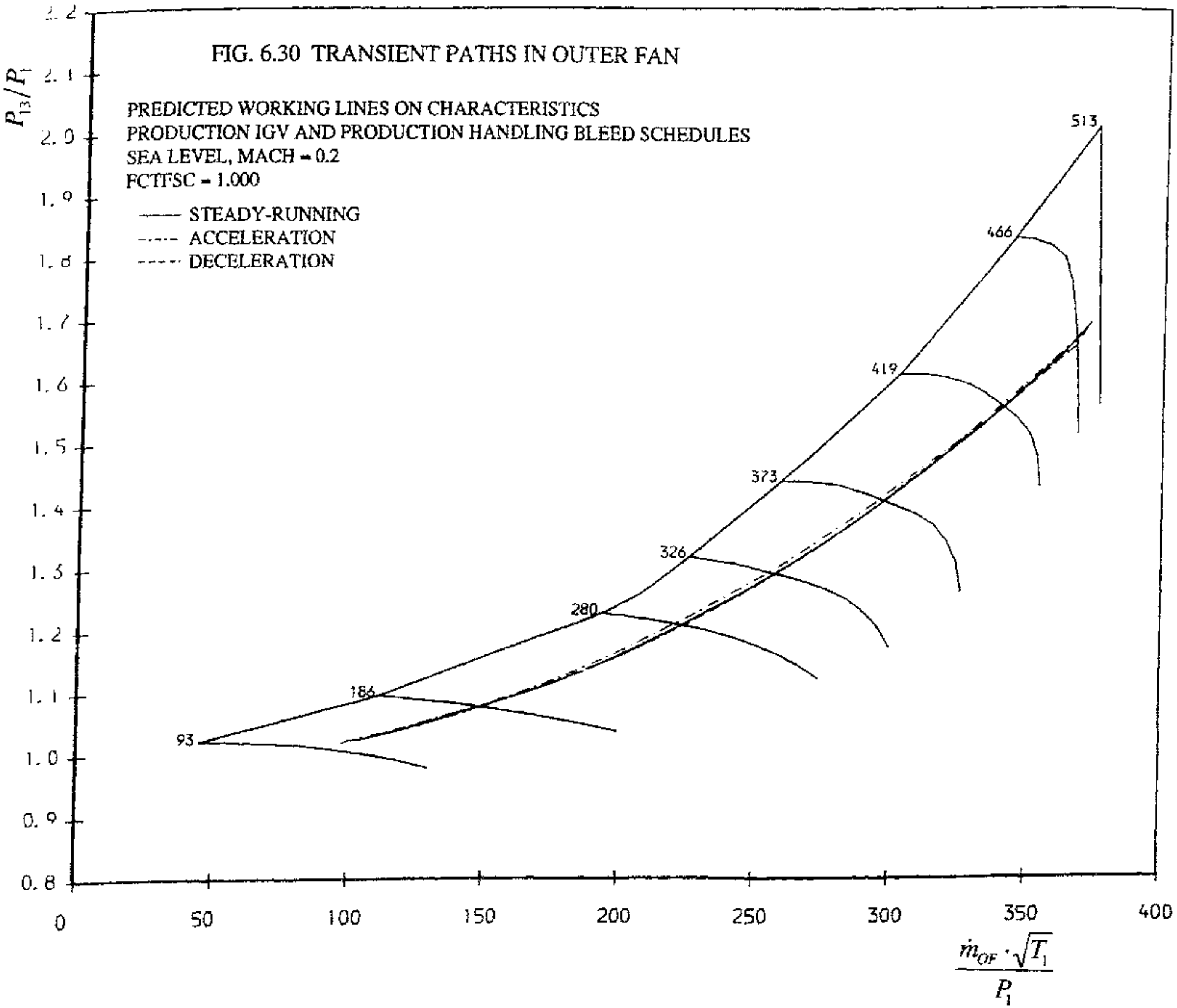


FIG. 6.30 TRANSIENT PATHS IN OUTER FAN
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{13}/P_1
$\frac{\dot{m}_{OF} \cdot \sqrt{T_1}}{P_1}$
93
186
280
326
373
419
466
513
0.8
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
2.0
2.1
2.2
0
50
100
150
200
250
300
350
400

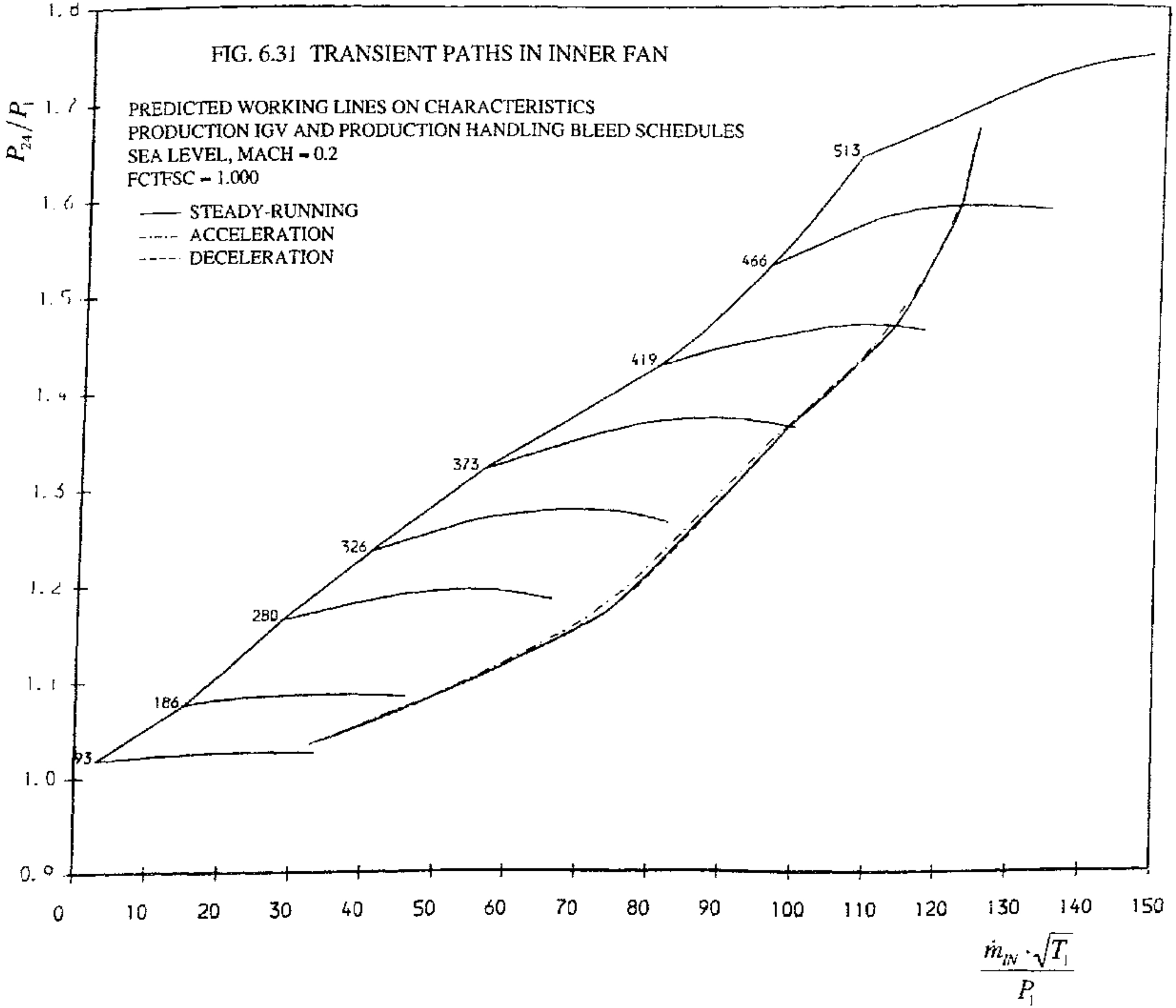
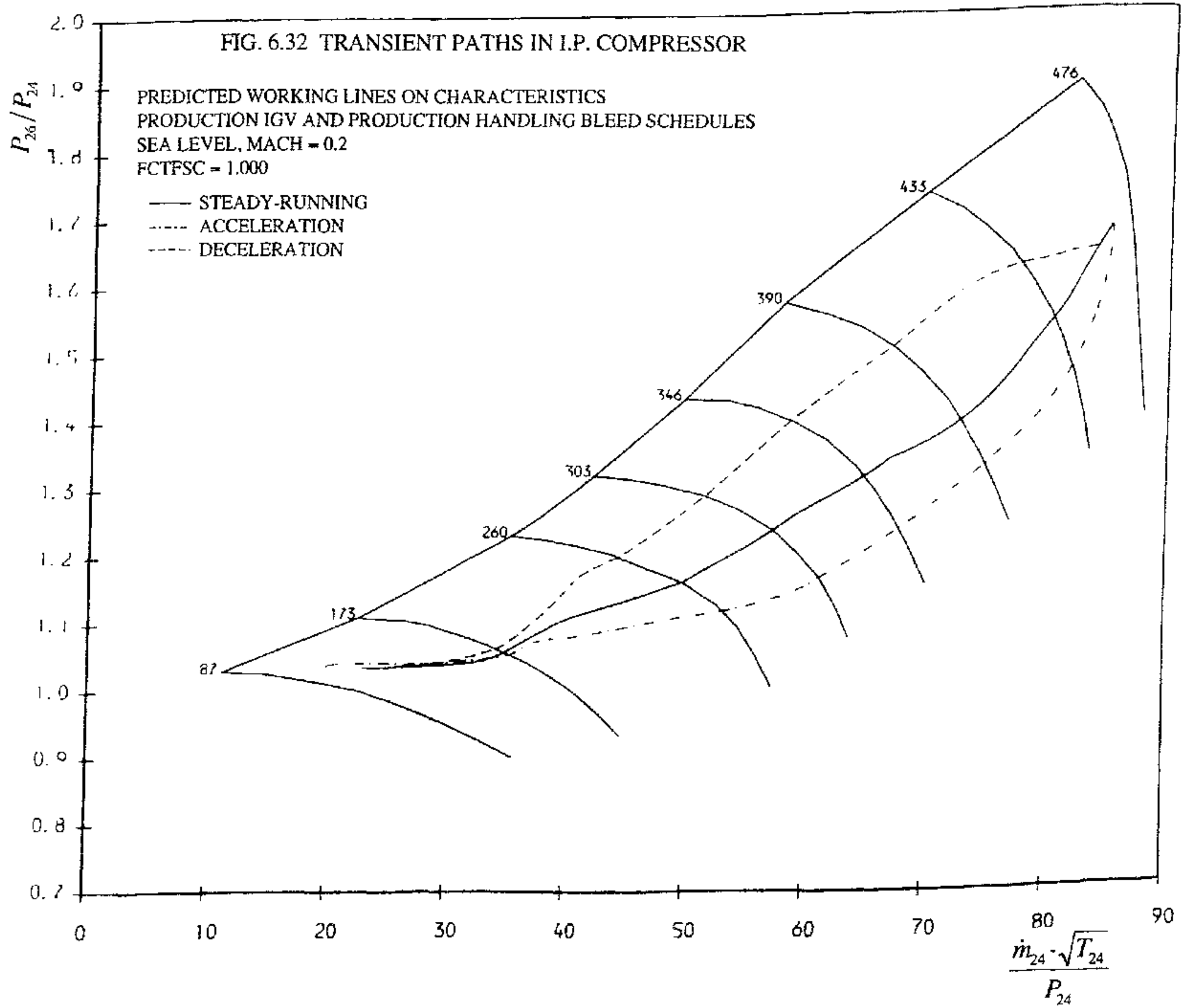


FIG. 6.31 TRANSIENT PATHS IN INNER FAN
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{24}/P_1
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.0
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
$\frac{\dot{m}_{IN} \cdot \sqrt{T_1}}{P_1}$
93
186
280
326
373
419
466
513

FIG. 6.32 TRANSIENT PATHS IN I.P. COMPRESSOR



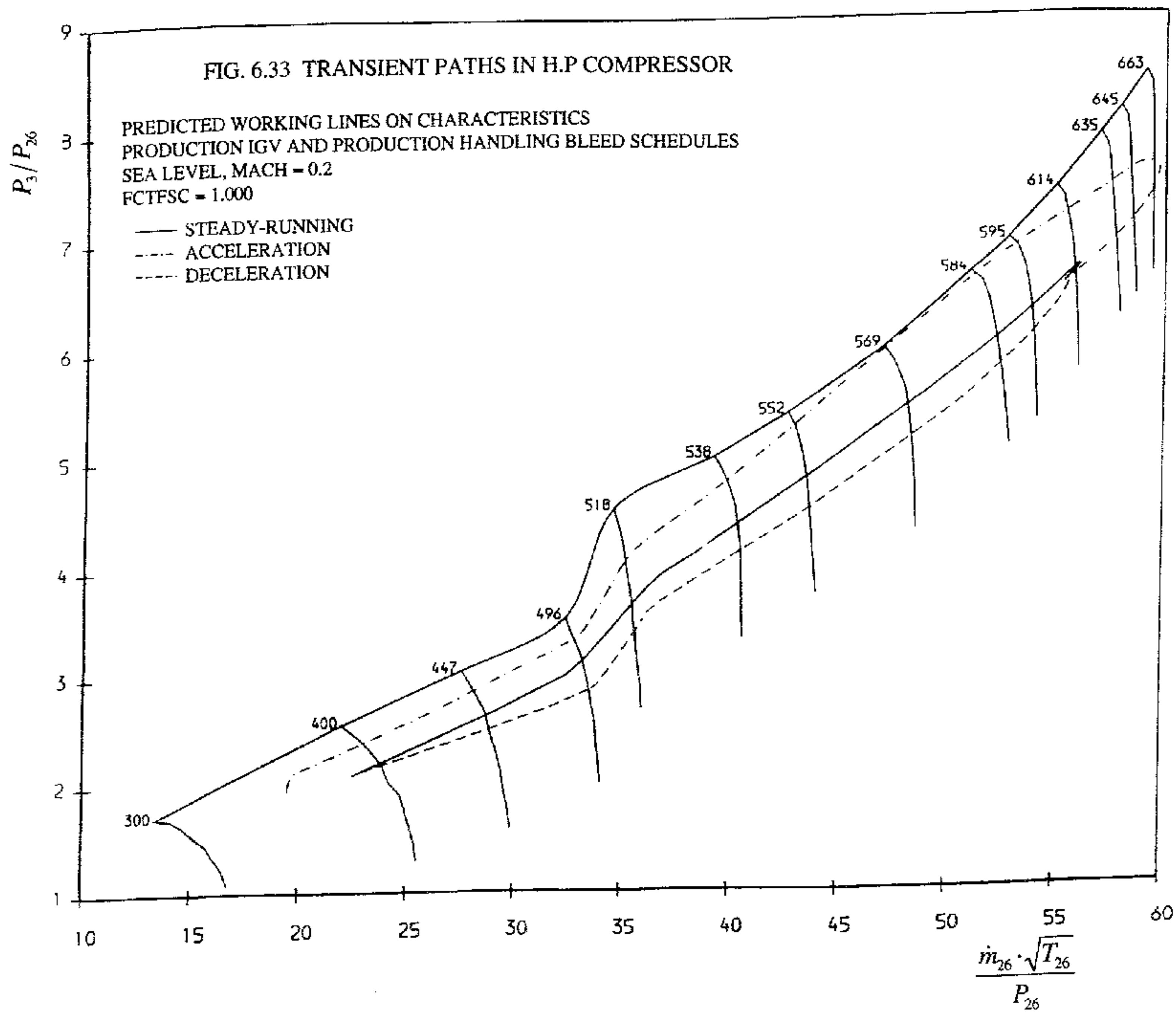
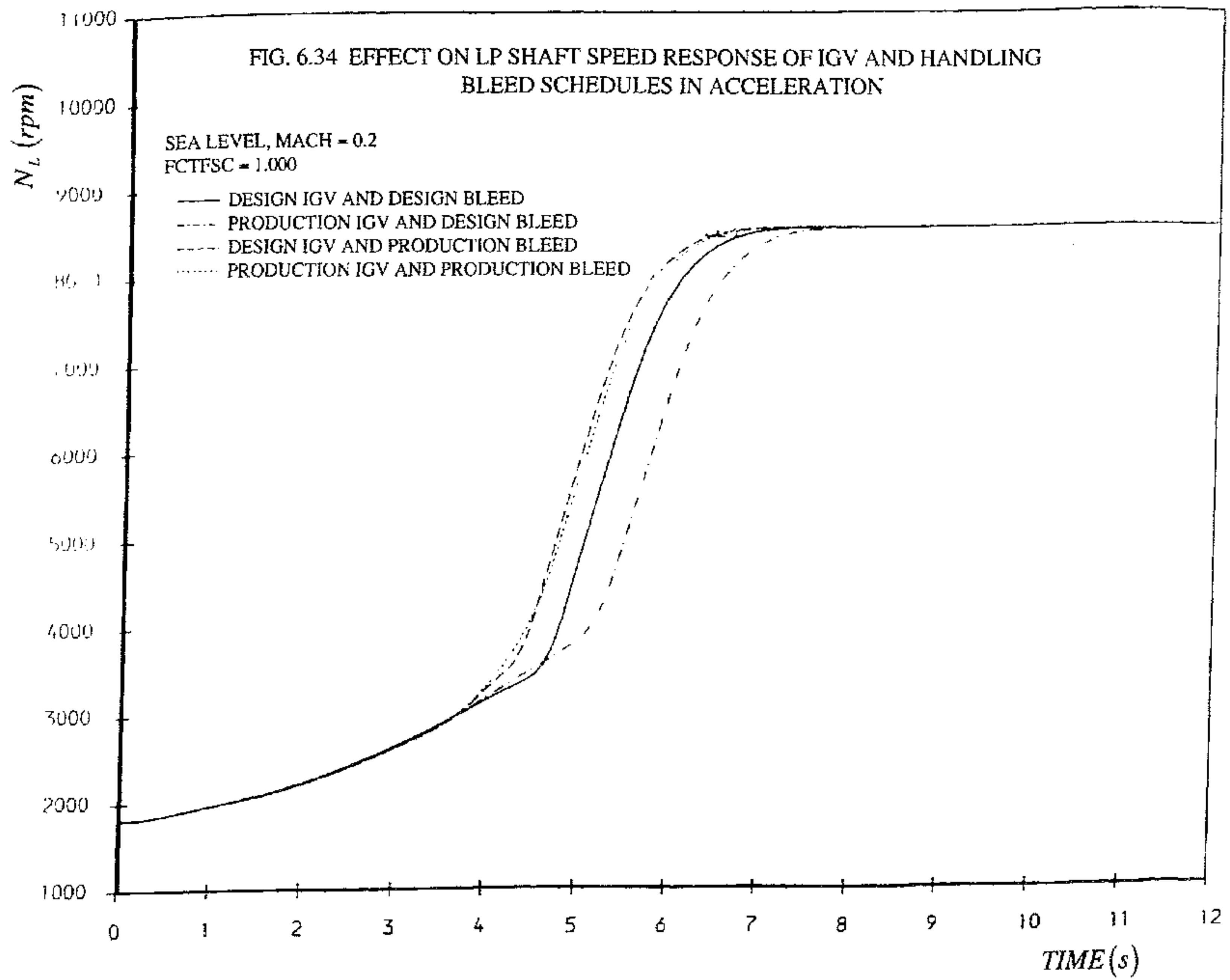


FIG. 6.33 TRANSIENT PATHS IN H.P COMPRESSOR
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_3/P_{26}
$\frac{\dot{m}_{26} \cdot \sqrt{T_{26}}}{P_{26}}$
300
400
447
496
518
538
552
569
584
595
614
635
645
663
1
2
3
4
5
6
7
8
9
10
15
20
25
30
35
40
45
50
55
60

FIG. 6.34 EFFECT ON LP SHAFT SPEED RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION



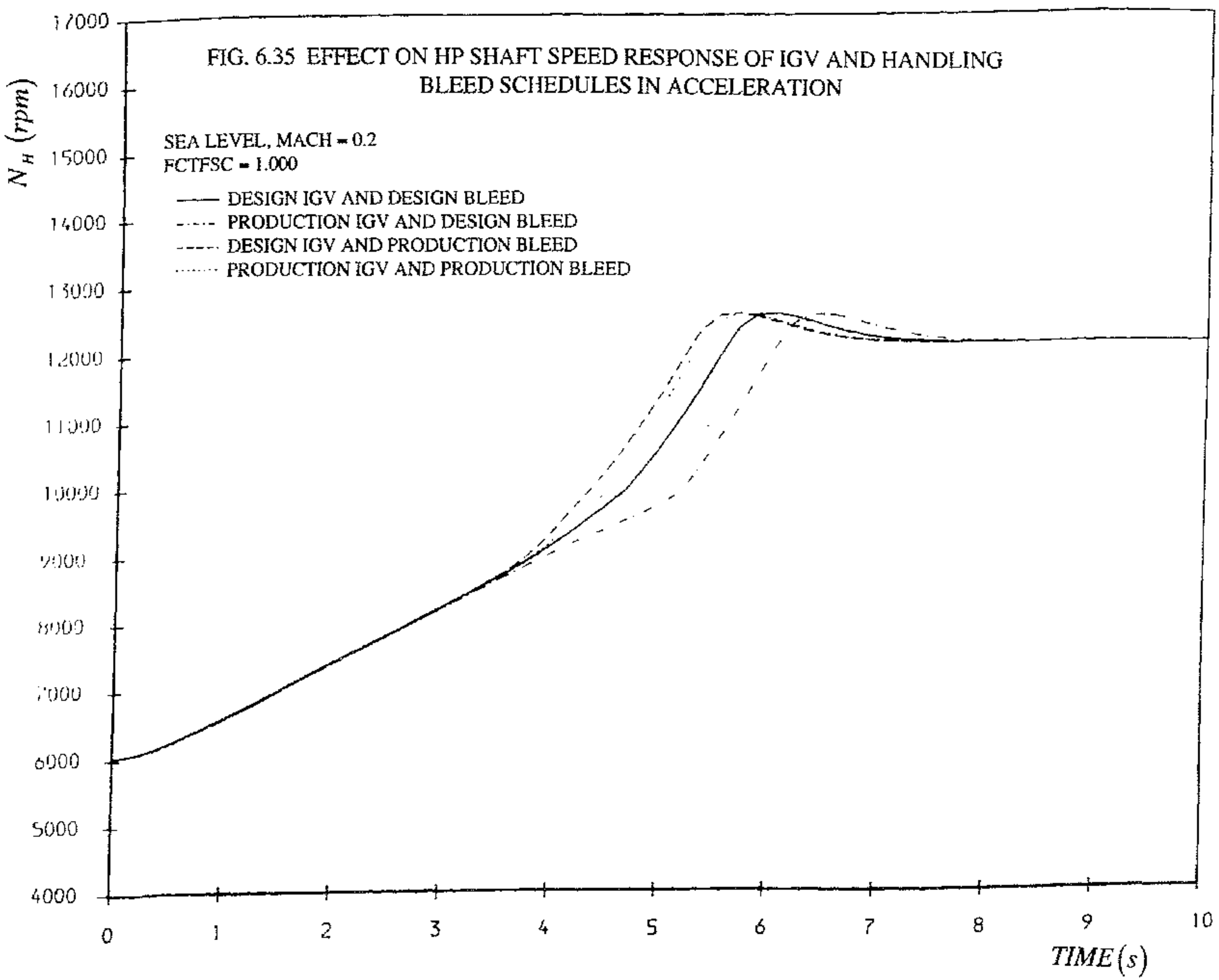


FIG. 6.35 EFFECT ON HP SHAFT SPEED RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
PRODUCTION IGV AND DESIGN BLEED
DESIGN IGV AND PRODUCTION BLEED
PRODUCTION IGV AND PRODUCTION BLEED
N_H (rpm)
17000
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
0
1
2
3
4
5
6
7
8
9
10
TIME (s)

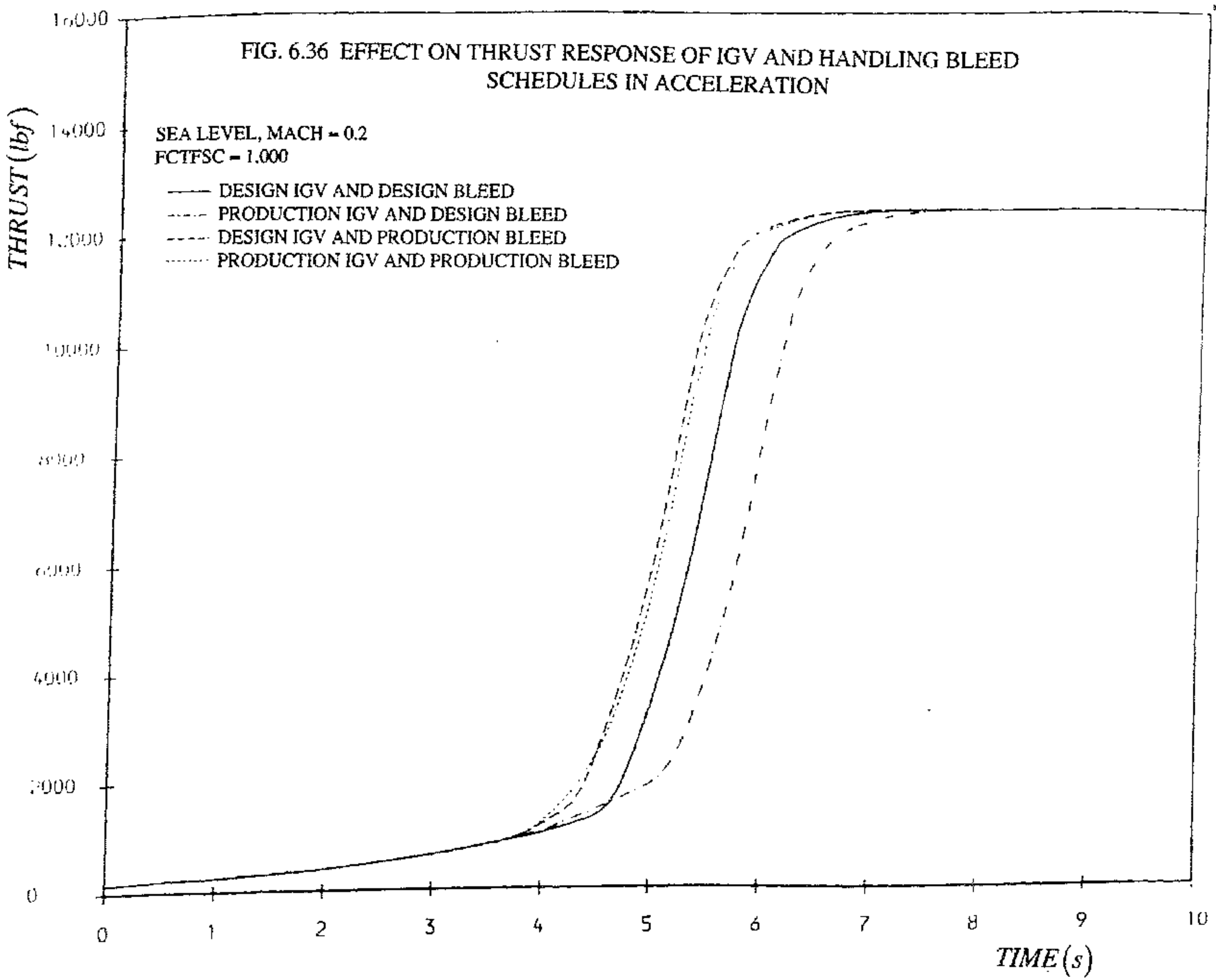


FIG. 6.36 EFFECT ON THRUST RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION

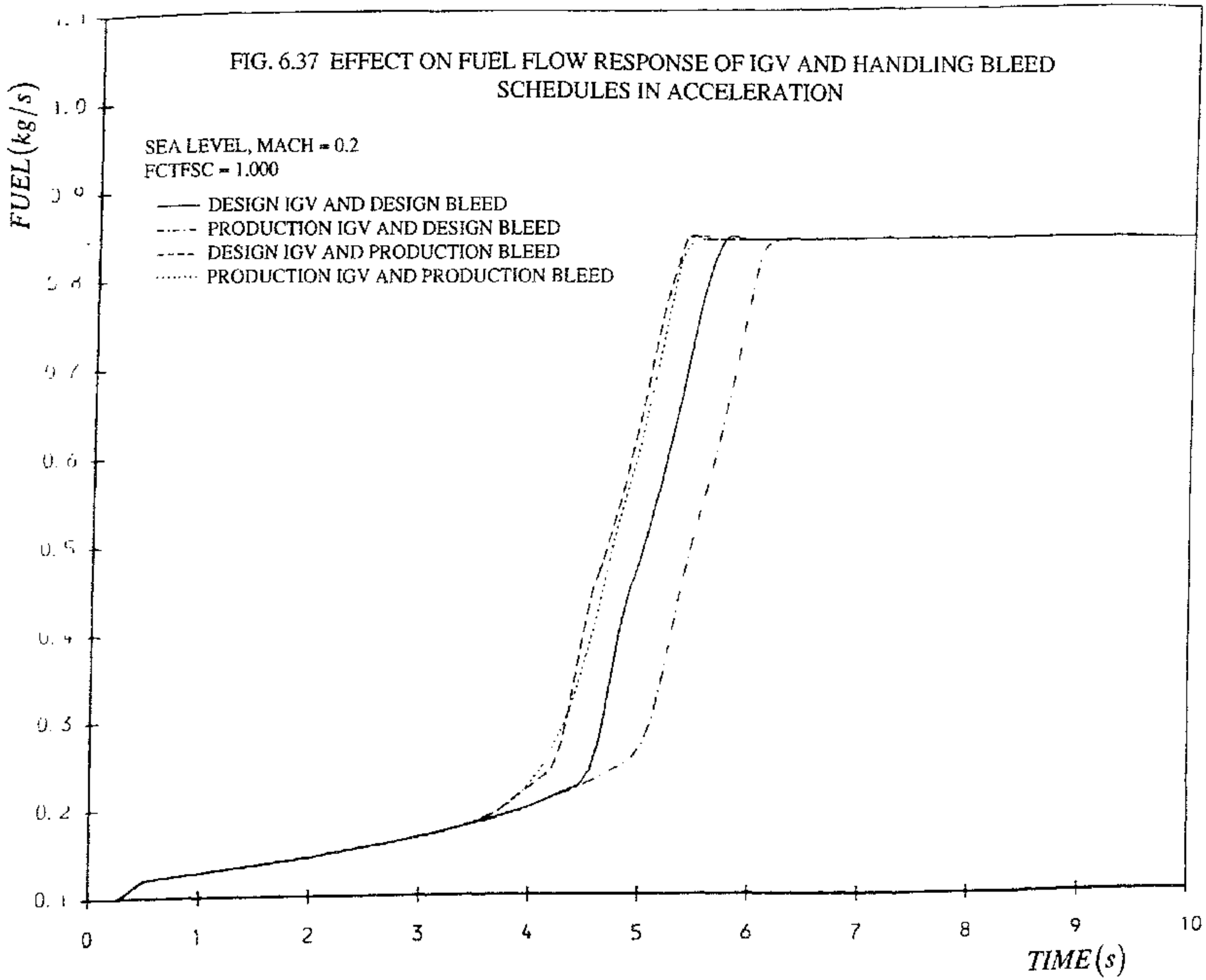


FIG. 6.37 EFFECT ON FUEL FLOW RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION

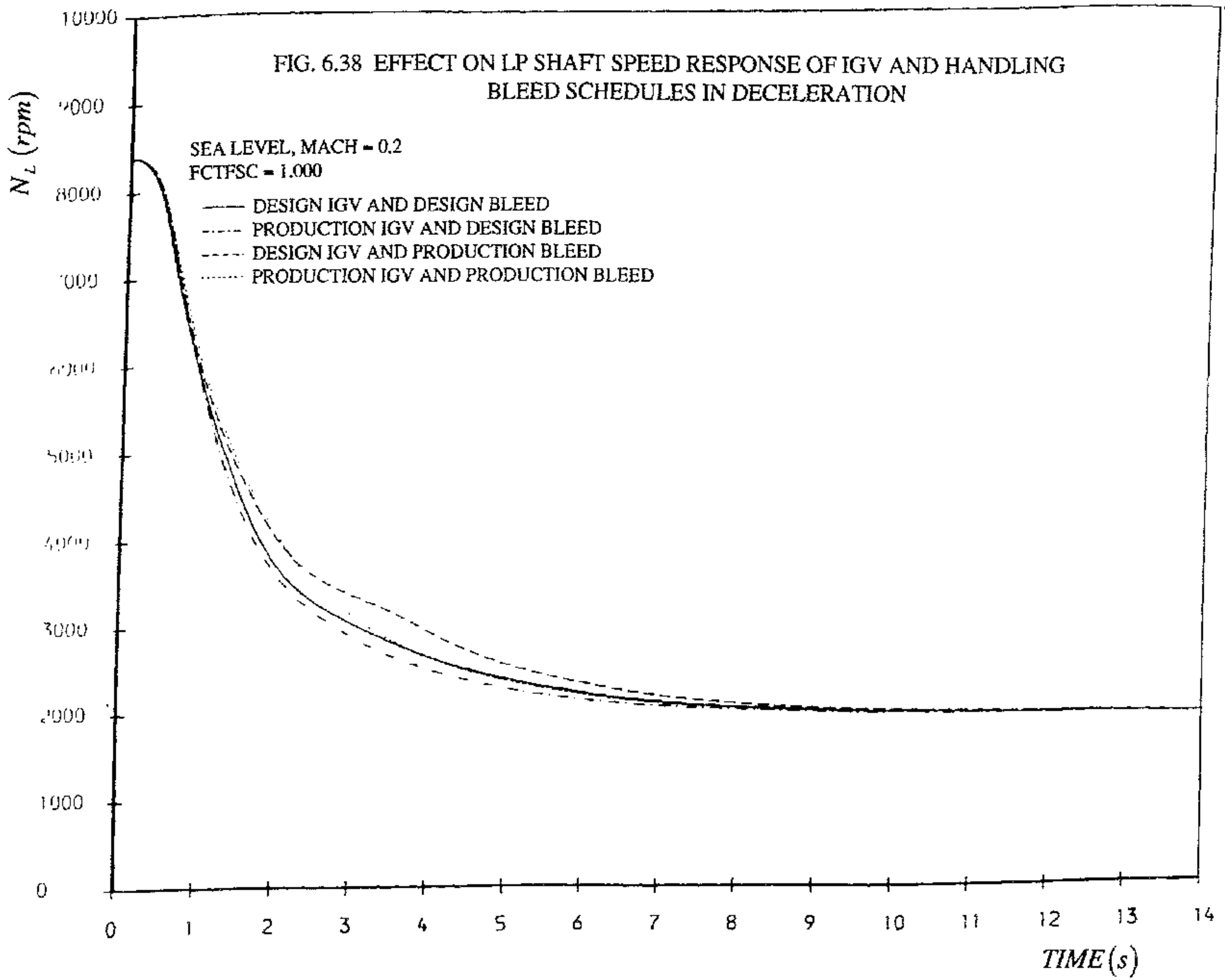


FIG. 6.38 EFFECT ON LP SHAFT SPEED RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
PRODUCTION IGV AND DESIGN BLEED
DESIGN IGV AND PRODUCTION BLEED
PRODUCTION IGV AND PRODUCTION BLEED
N_L (rpm)
10000
8000
3000
1000
0
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
TIME (s)

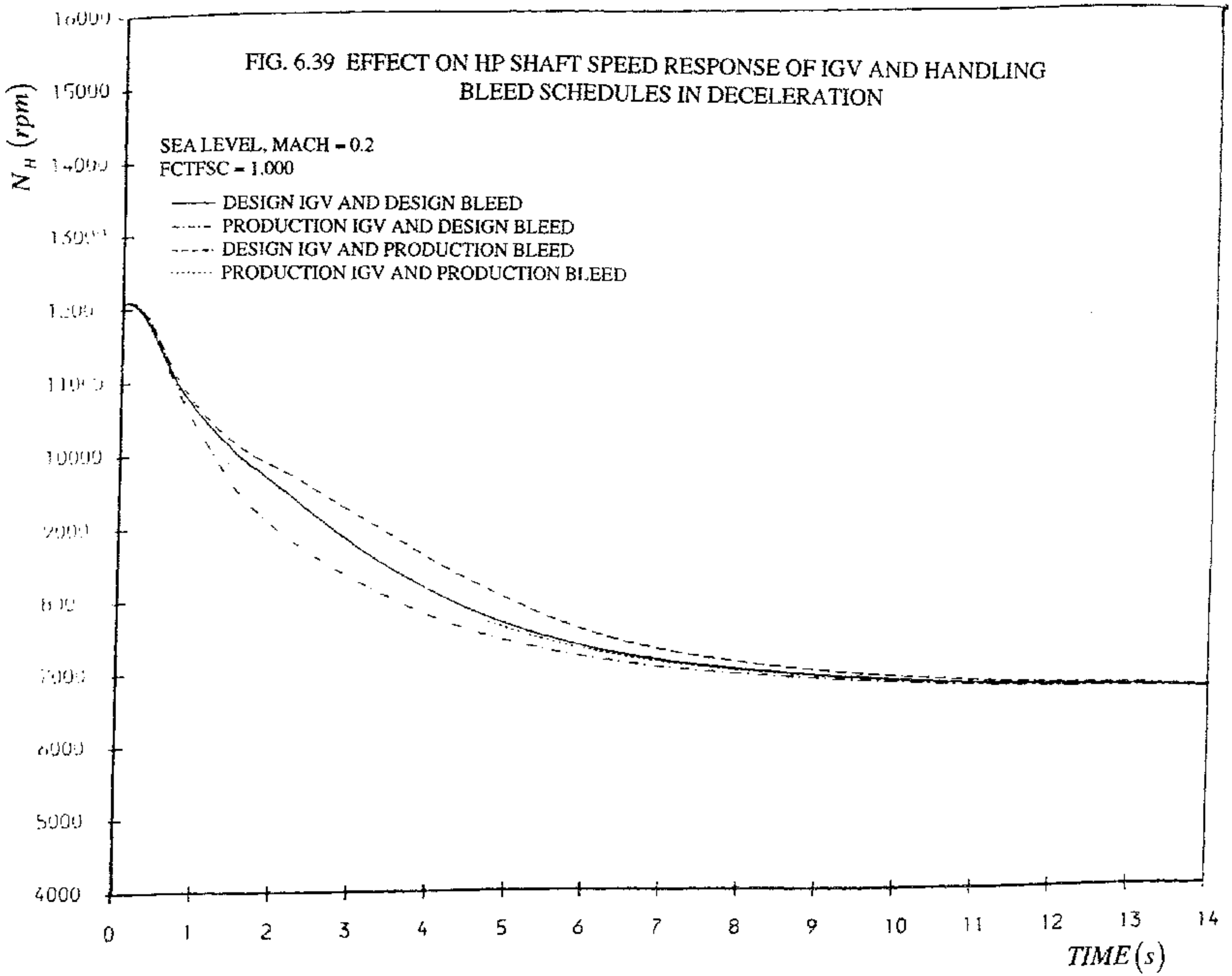


FIG. 6.39 EFFECT ON HP SHAFT SPEED RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION

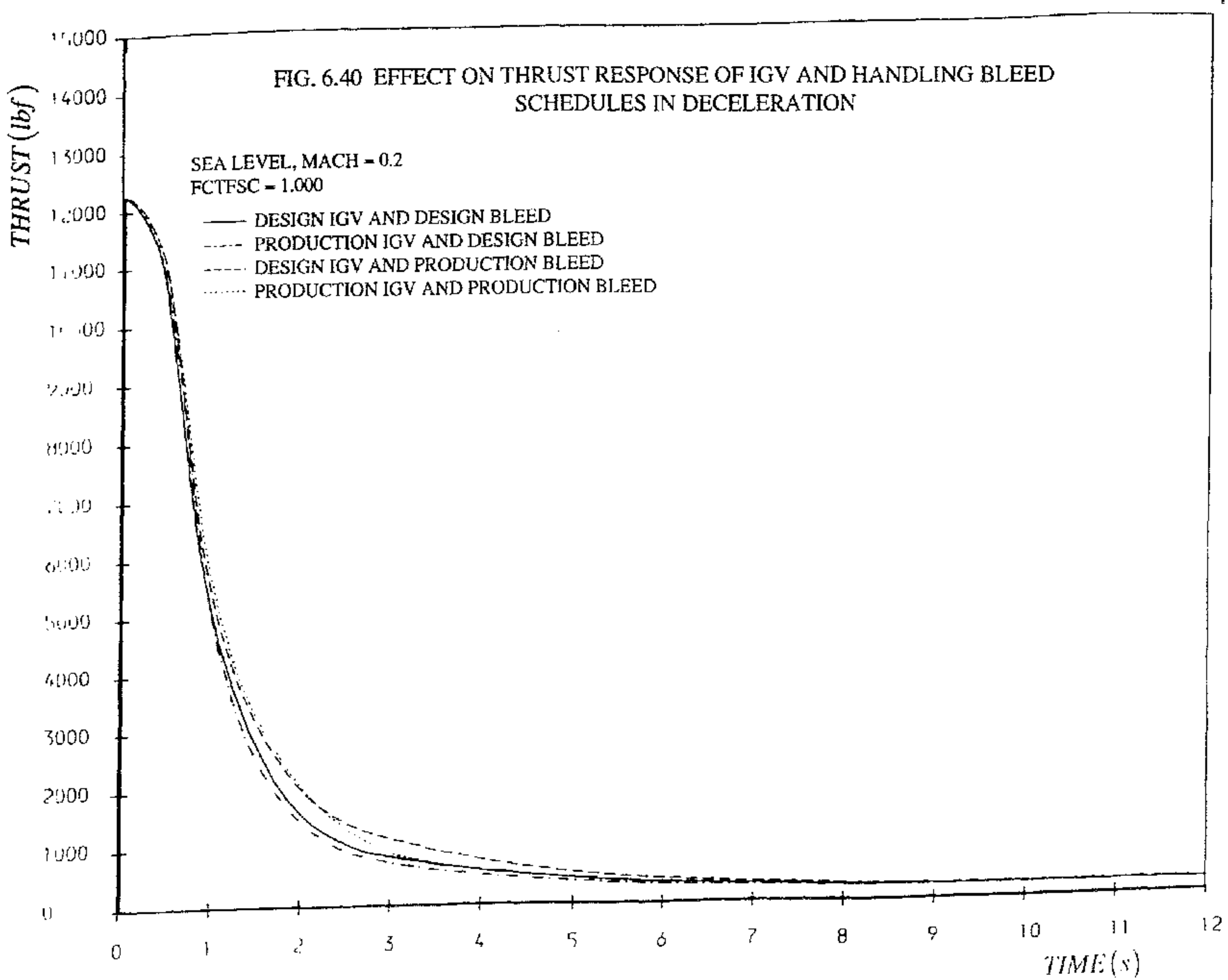


FIG. 6.40 EFFECT ON THRUST RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
PRODUCTION IGV AND DESIGN BLEED
DESIGN IGV AND PRODUCTION BLEED
PRODUCTION IGV AND PRODUCTION BLEED
THRUST (lbf)
TIME (s)
0
1
2
3
4
5
6
7
8
9
10
11
12
13000
4000
3000
2000
1000

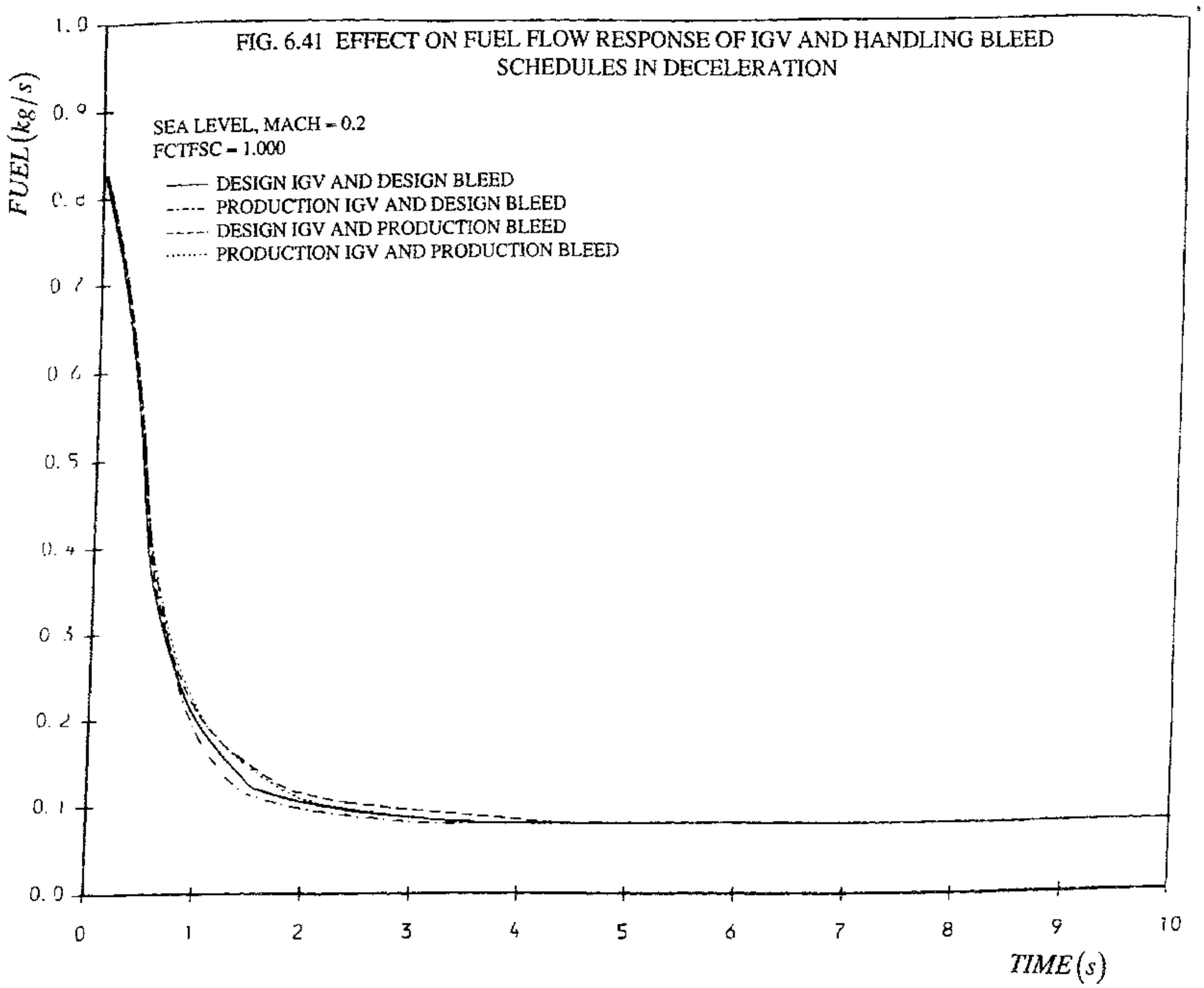


FIG. 6.41 EFFECT ON FUEL FLOW RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
PRODUCTION IGV AND DESIGN BLEED
DESIGN IGV AND PRODUCTION BLEED
PRODUCTION IGV AND PRODUCTION BLEED
FUEL (kg/s)
TIME (s)

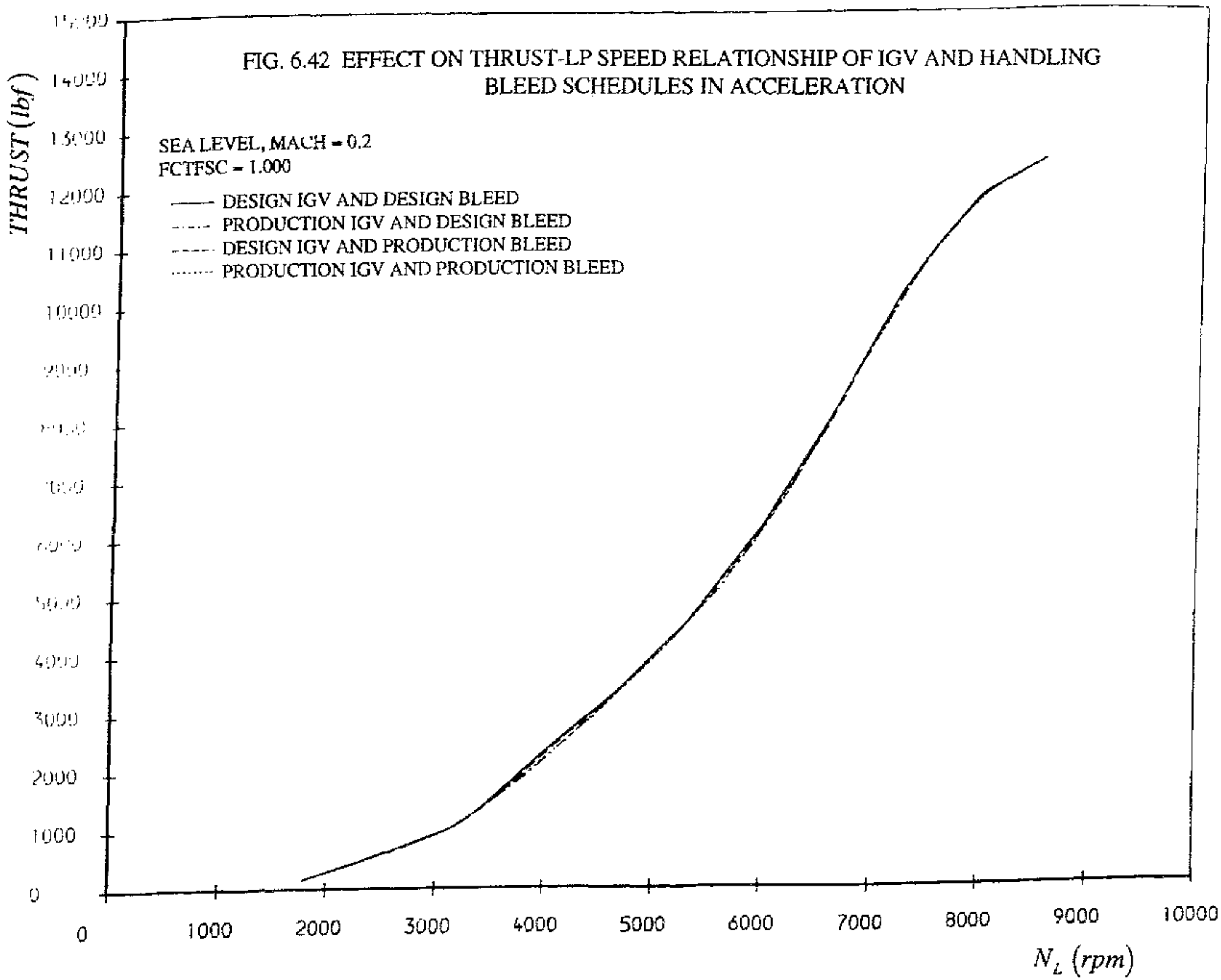


FIG. 6.42 EFFECT ON THRUST-LP SPEED RELATIONSHIP OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION

FIG. 6.43 EFFECT ON THRUST-HP SPEED RELATIONSHIP OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION

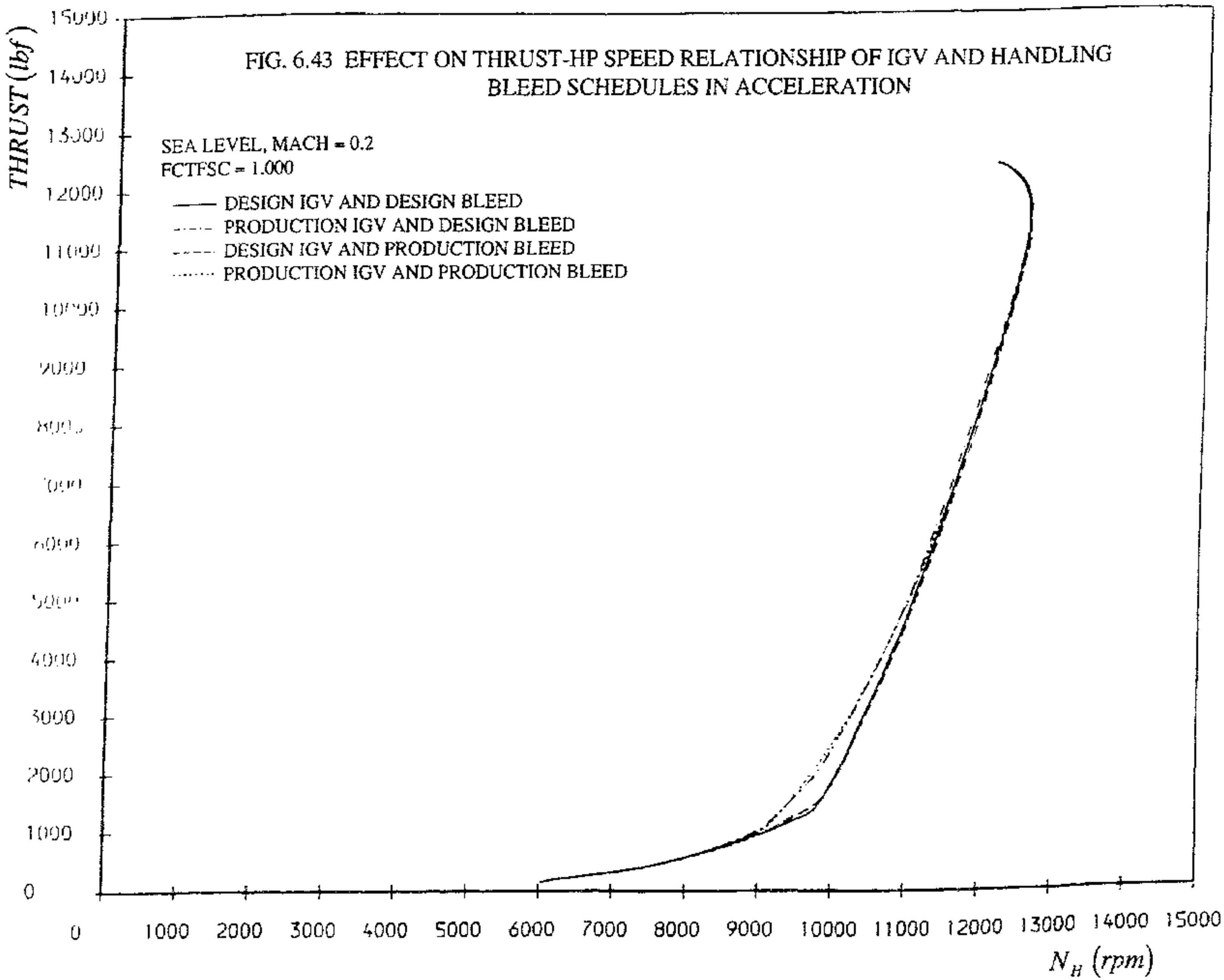
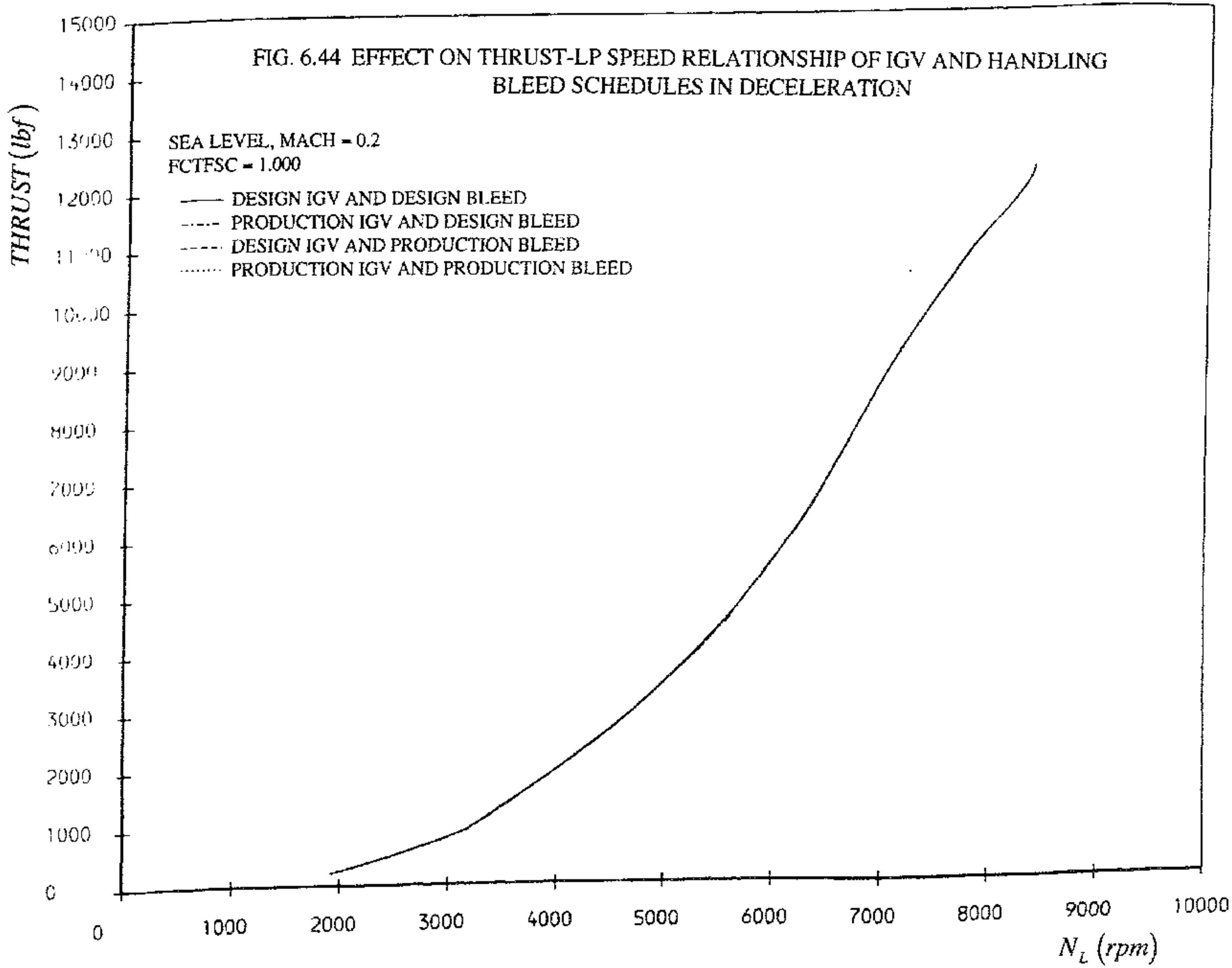


FIG. 6.44 EFFECT ON THRUST-LP SPEED RELATIONSHIP OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION



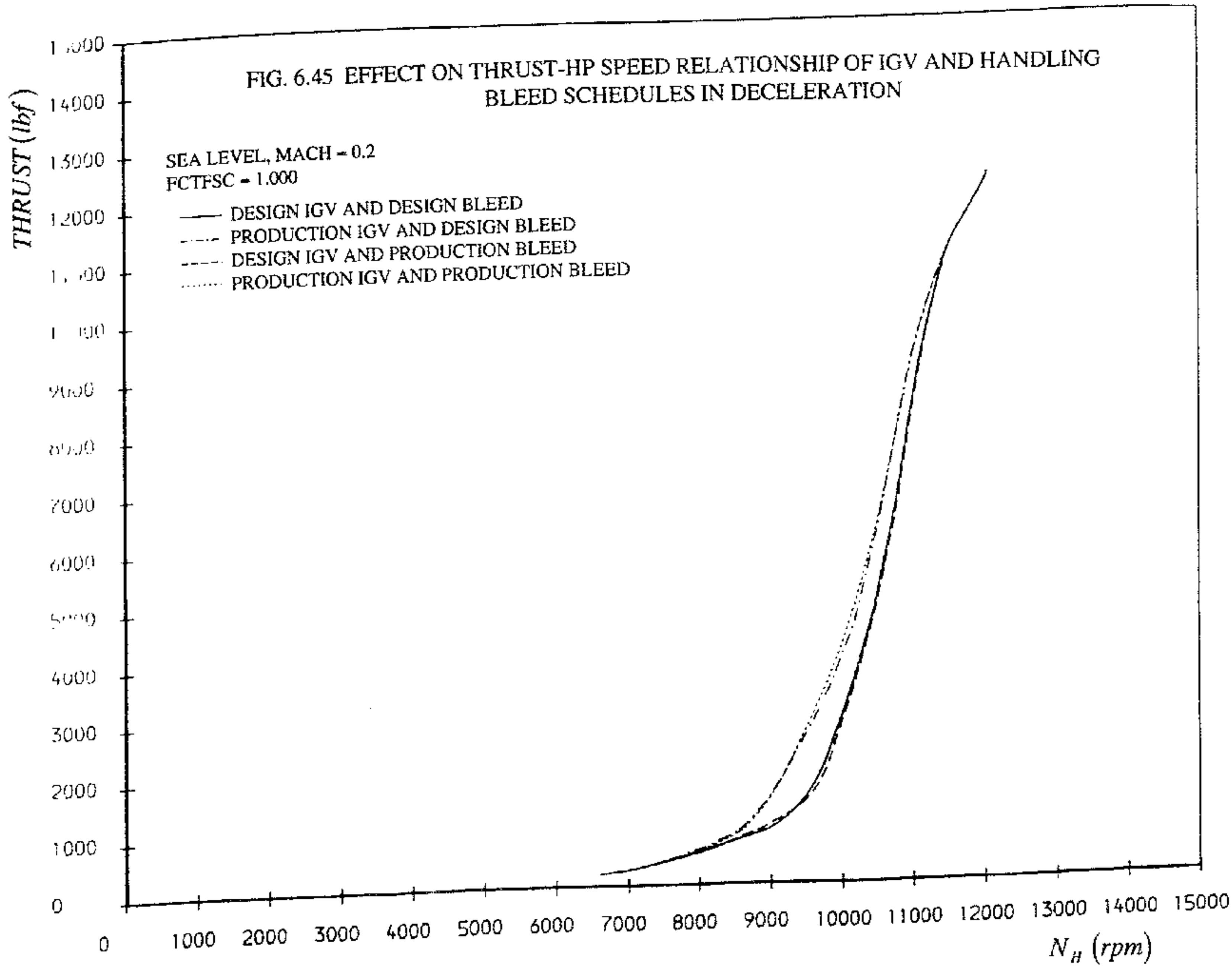


FIG. 6.45 EFFECT ON THRUST-HP SPEED RELATIONSHIP OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION

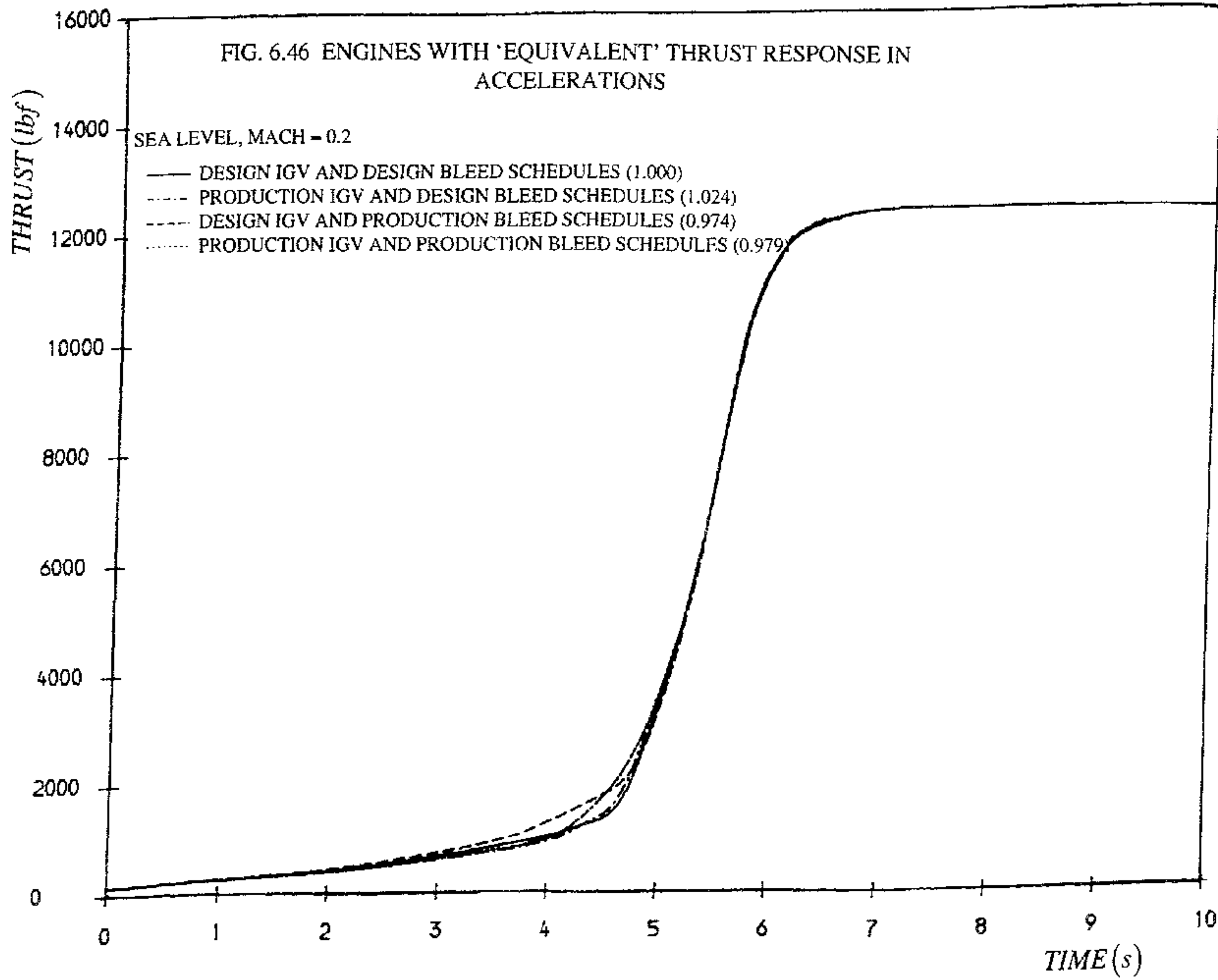


FIG. 6.46 ENGINES WITH 'EQUIVALENT' THRUST RESPONSE IN ACCELERATIONS

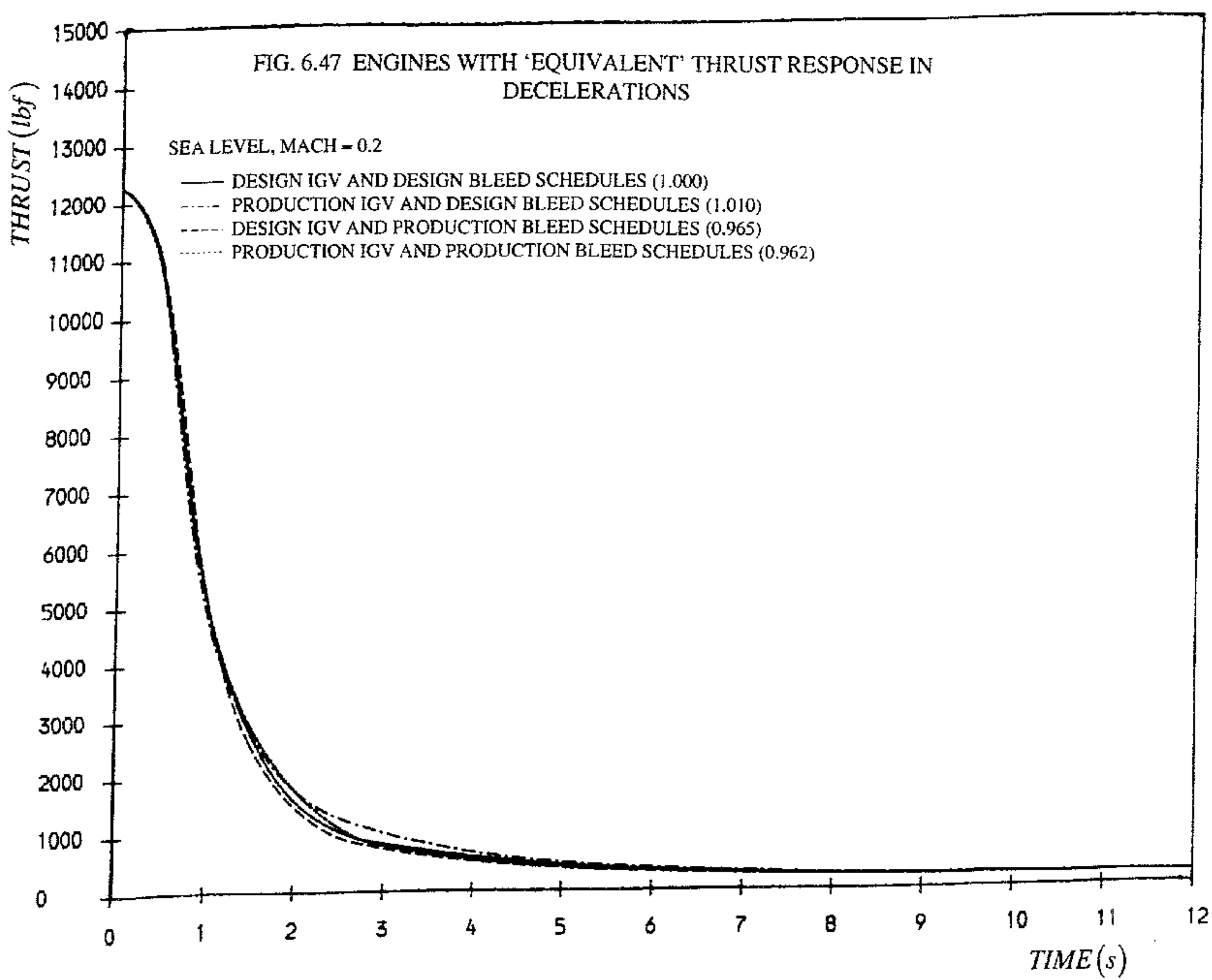
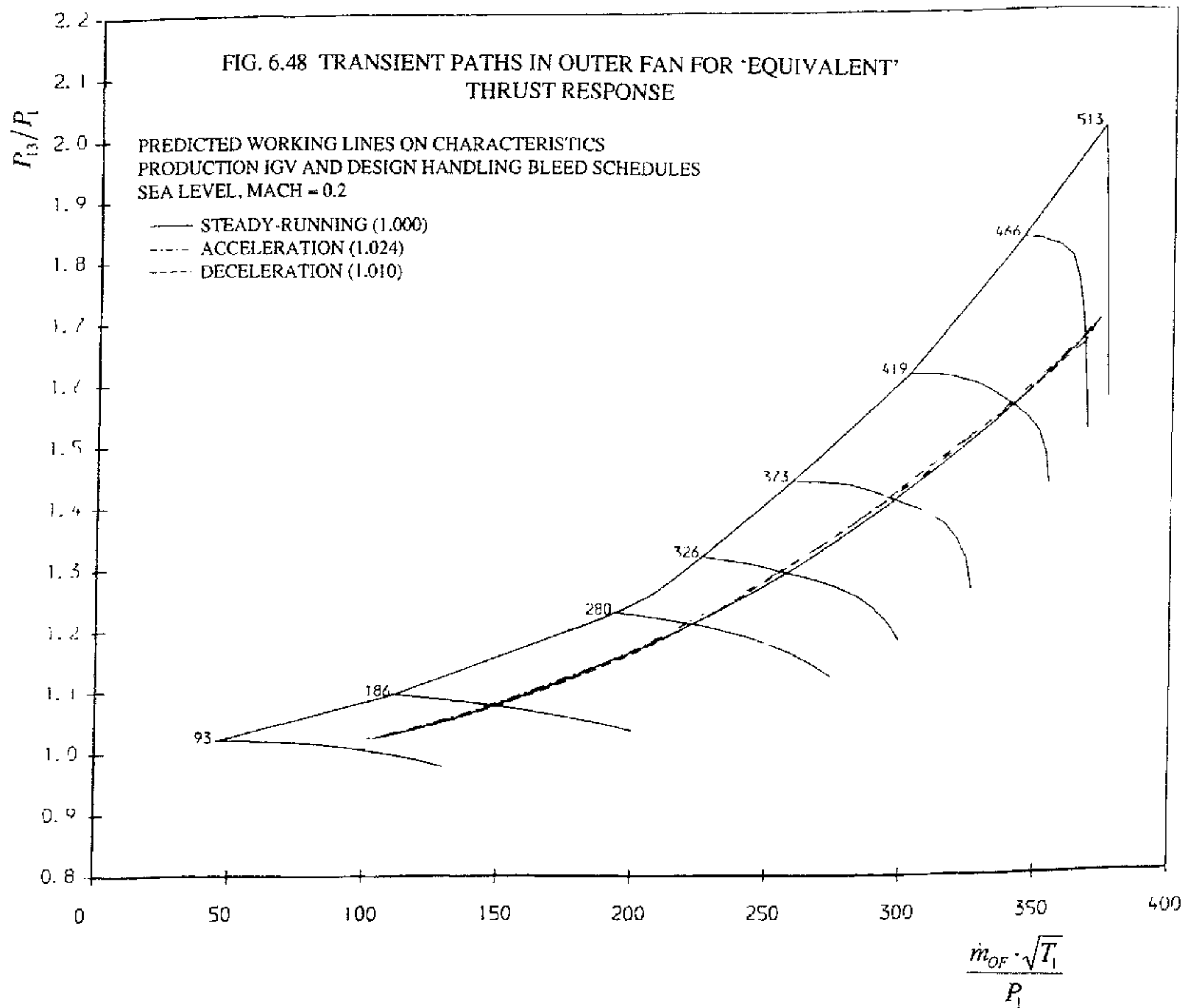


FIG. 6.47 ENGINES WITH 'EQUIVALENT' THRUST RESPONSE IN DECELERATIONS
SEA LEVEL, MACH = 0.2
DESIGN IGV AND DESIGN BLEED SCHEDULES (1.000)
PRODUCTION IGV AND DESIGN BLEED SCHEDULES (1.010)
DESIGN IGV AND PRODUCTION BLEED SCHEDULES (0.965)
PRODUCTION IGV AND PRODUCTION BLEED SCHEDULES (0.962)
THRUST (lbf)
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
0
1
2
3
4
5
6
7
8
9
10
11
12
TIME (s)

FIG. 6.48 TRANSIENT PATHS IN OUTER FAN FOR 'EQUIVALENT' THRUST RESPONSE



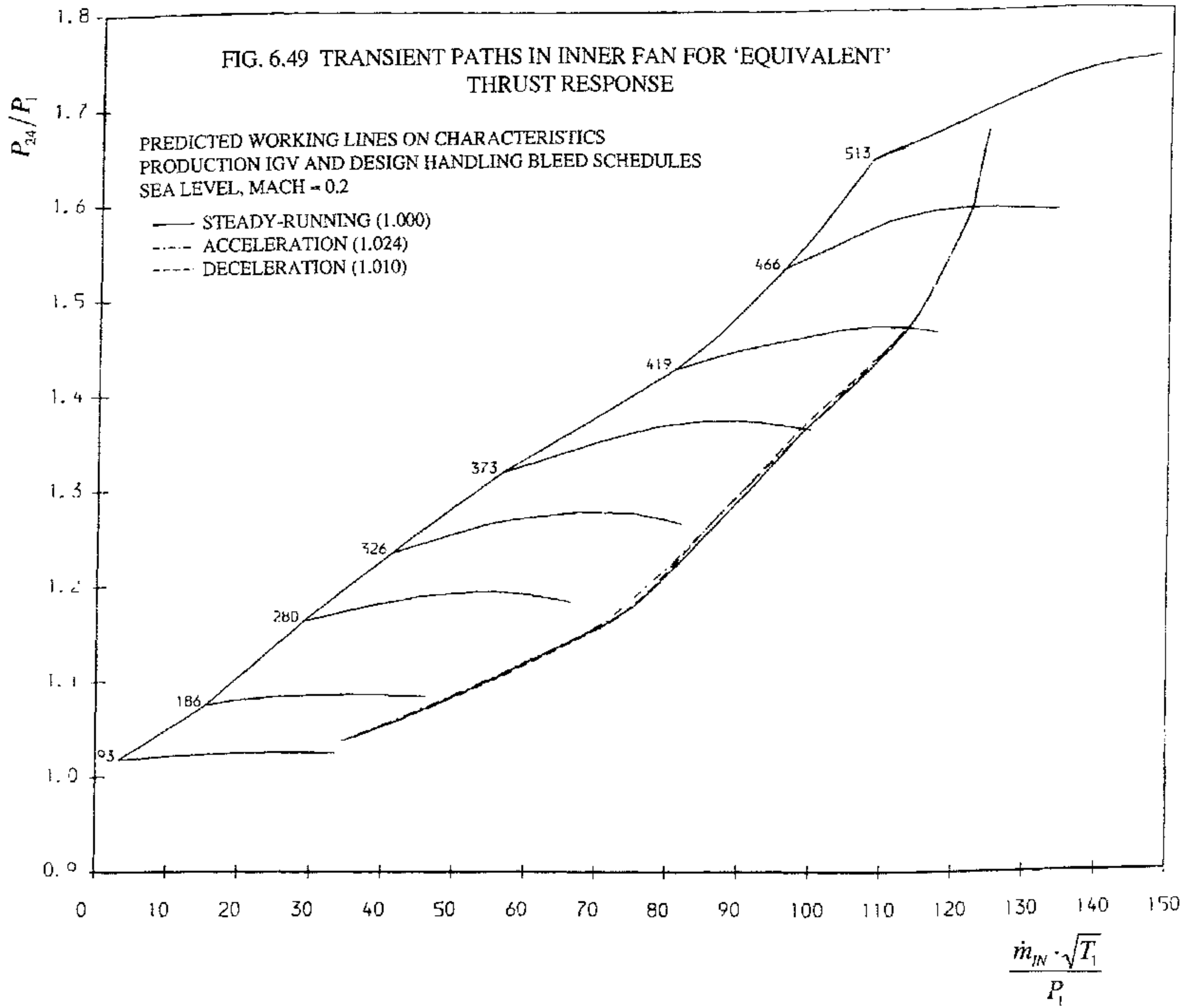


FIG. 6.49 TRANSIENT PATHS IN INNER FAN FOR 'EQUIVALENT' THRUST RESPONSE

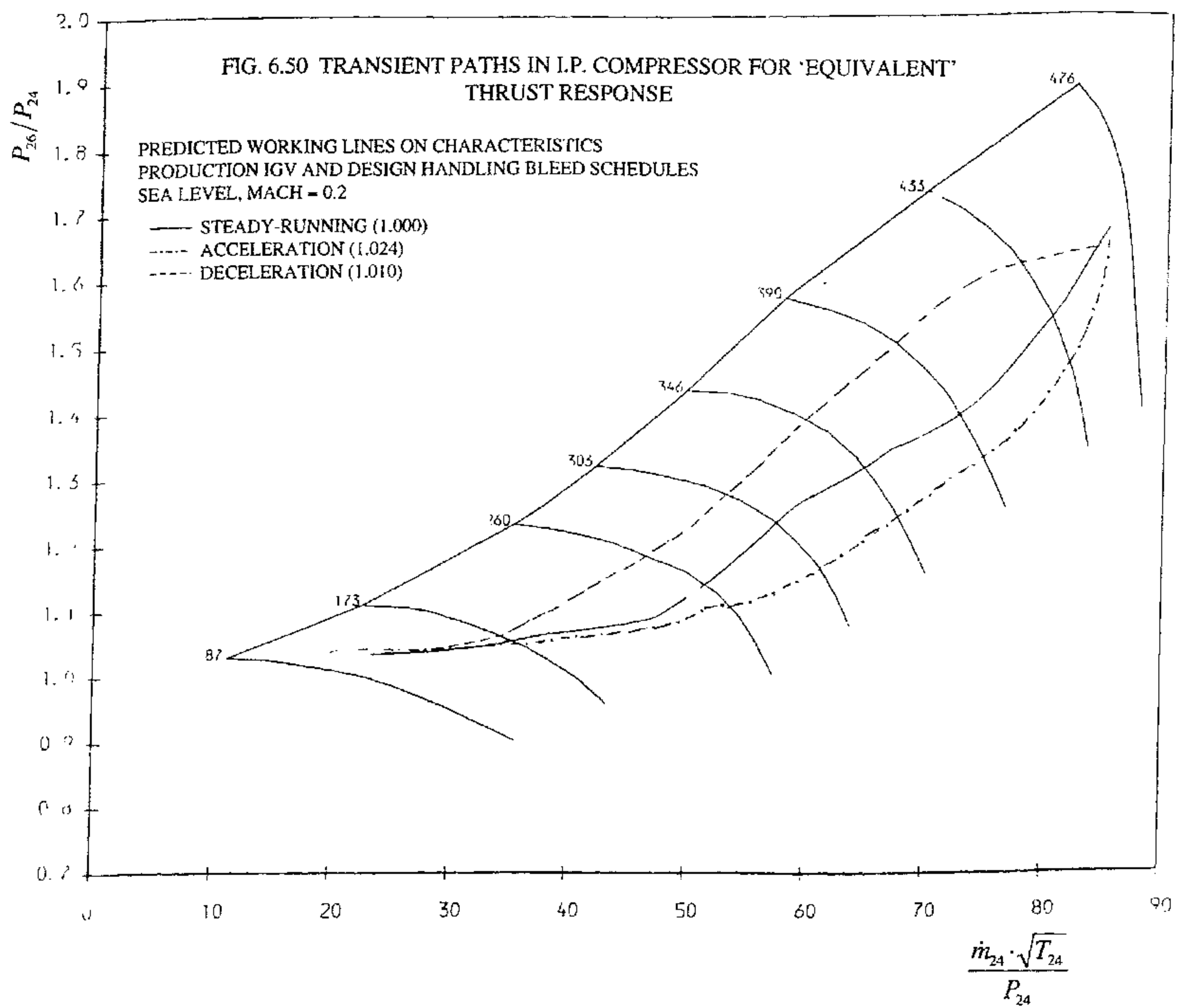


FIG. 6.50 TRANSIENT PATHS IN I.P. COMPRESSOR FOR 'EQUIVALENT' THRUST RESPONSE
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
STEADY-RUNNING (1.000)
ACCELERATION (1.024)
DECELERATION (1.010)
P_{26}/P_{24}
$\frac{\dot{m}_{24}\cdot\sqrt{T_{24}}}{P_{24}}$
87
173
260
305
346
390
433
476
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.1
0
10
20
30
40
50
60
70
80
90

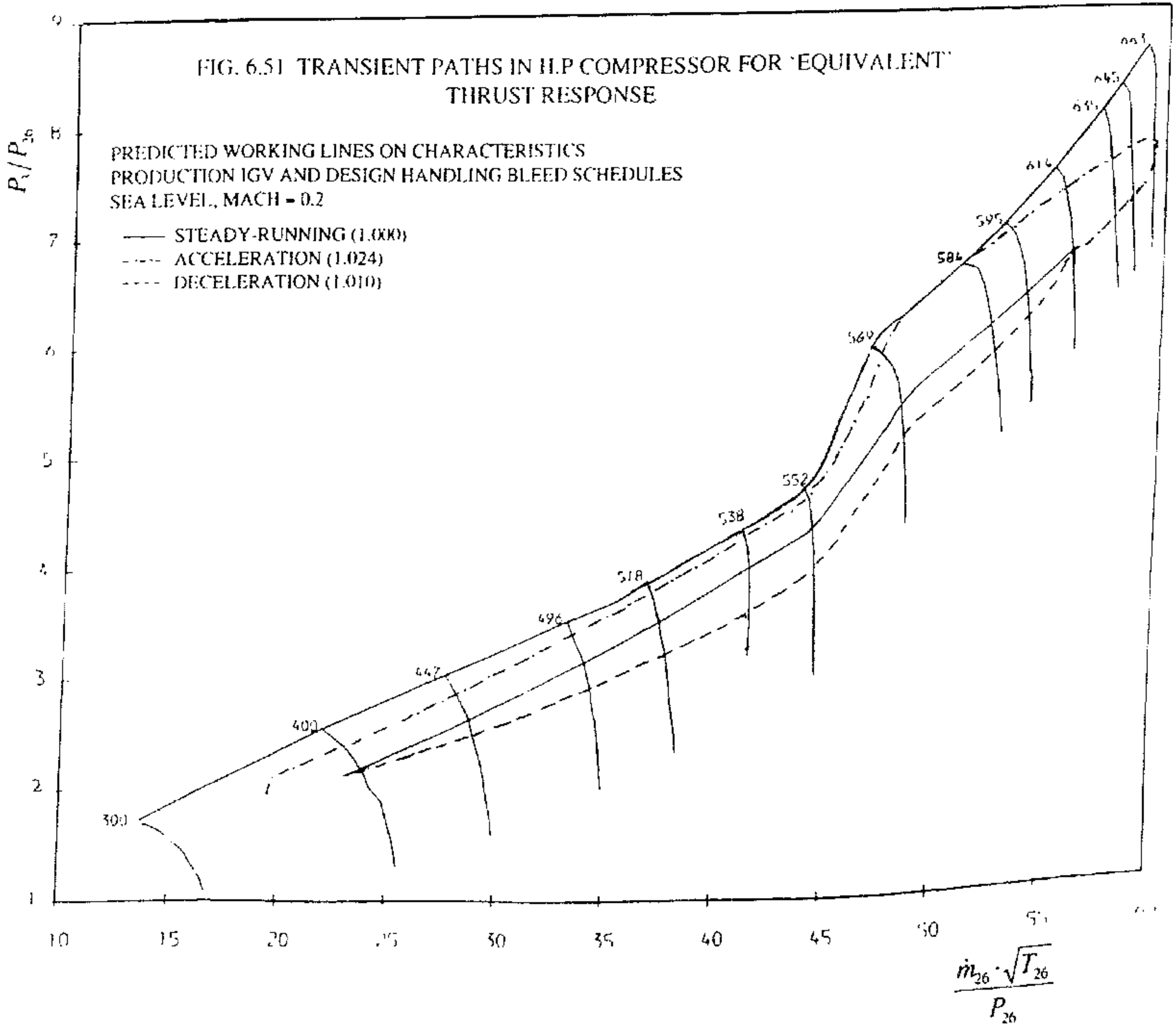


FIG. 6.51 TRANSIENT PATHS IN H.P COMPRESSOR FOR 'EQUIVALENT' THRUST RESPONSE

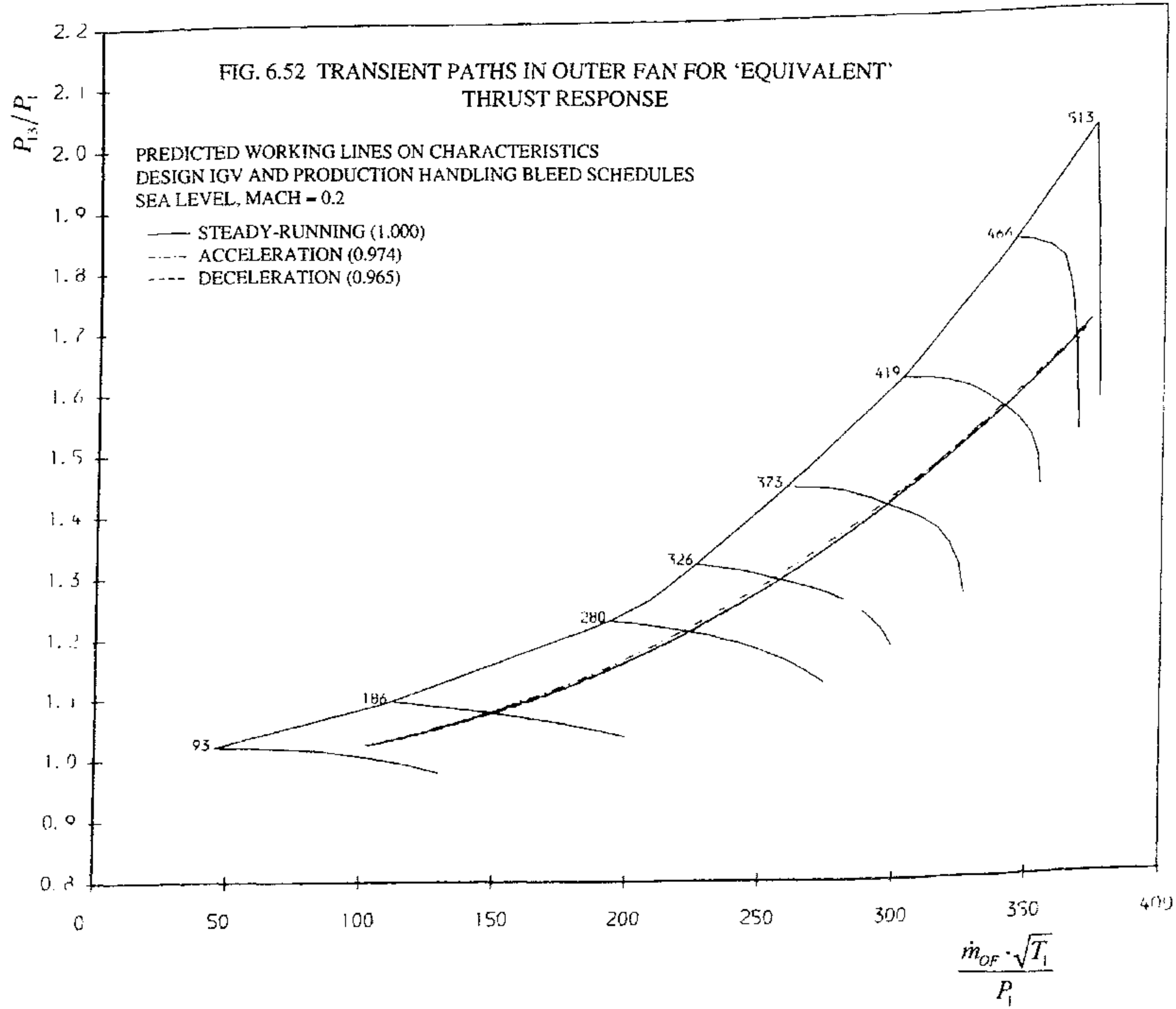
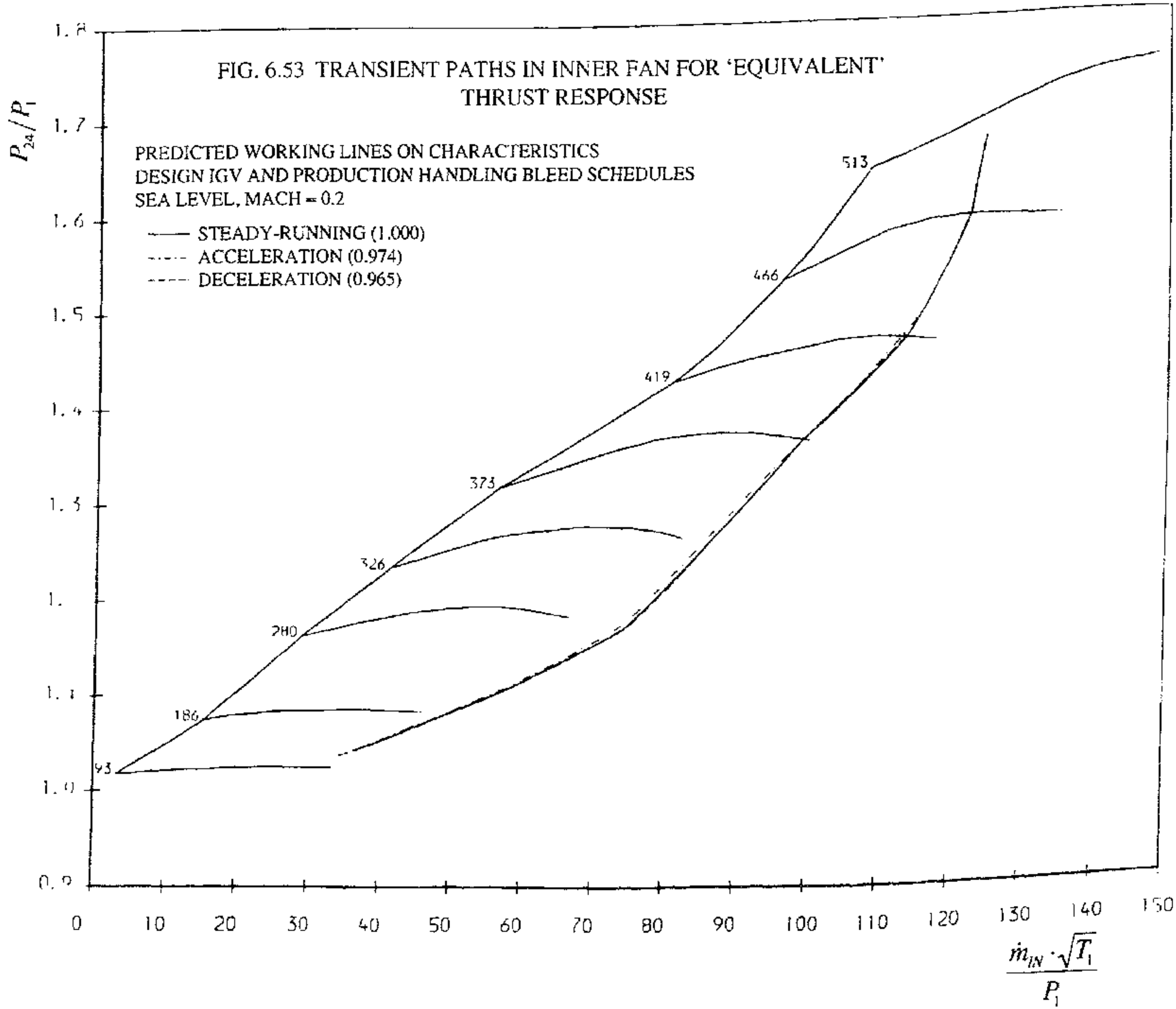


FIG. 6.52 TRANSIENT PATHS IN OUTER FAN FOR 'EQUIVALENT' THRUST RESPONSE
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
STEADY-RUNNING (1.000)
ACCELERATION (0.974)
DECELERATION (0.965)
P_{13}/P_1
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0
50
100
150
200
250
300
350
400
$\frac{\dot{m}_{OF} \cdot \sqrt{T_1}}{P_1}$
93
186
280
326
373
419
466
513

FIG. 6.53 TRANSIENT PATHS IN INNER FAN FOR 'EQUIVALENT' THRUST RESPONSE



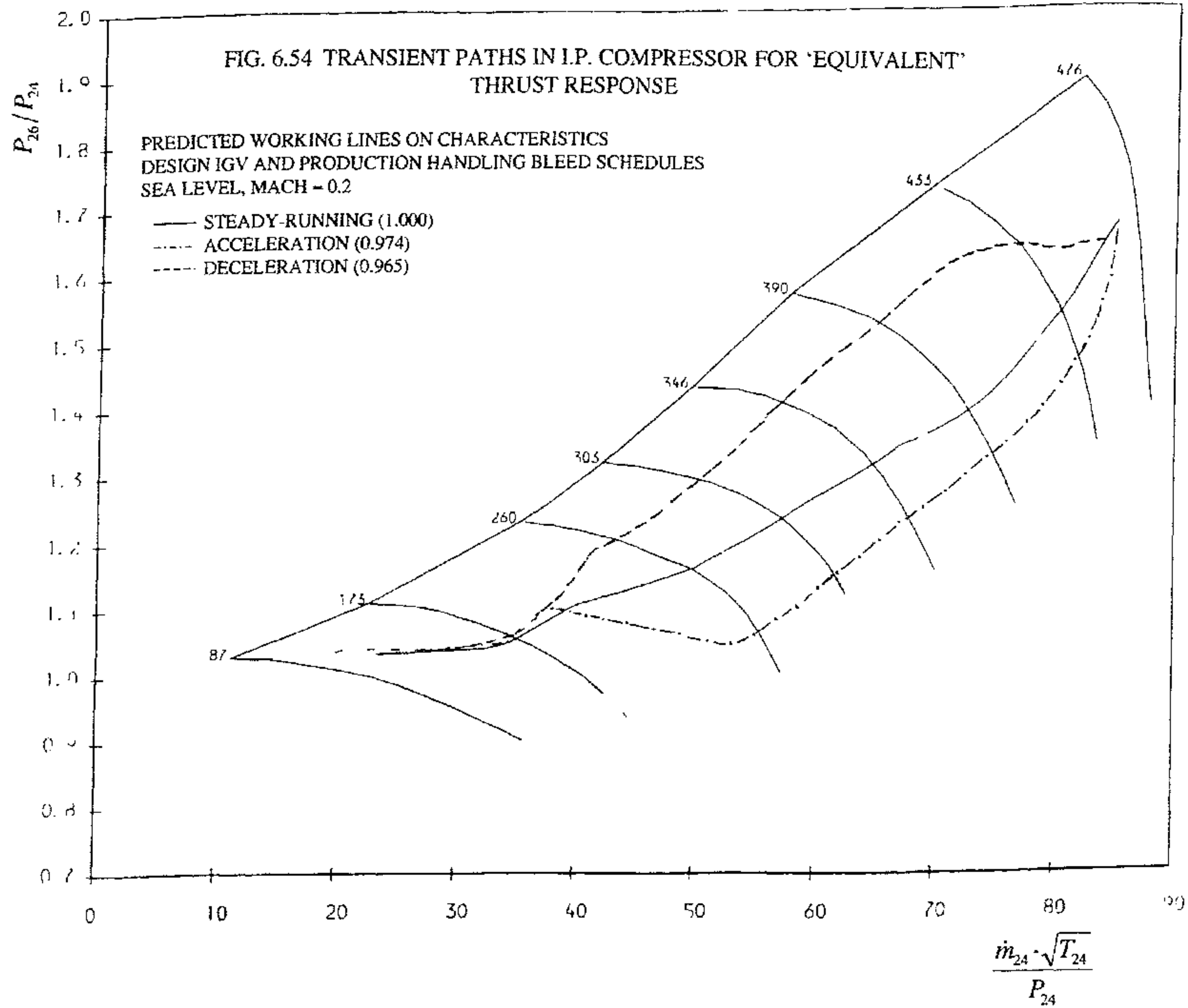
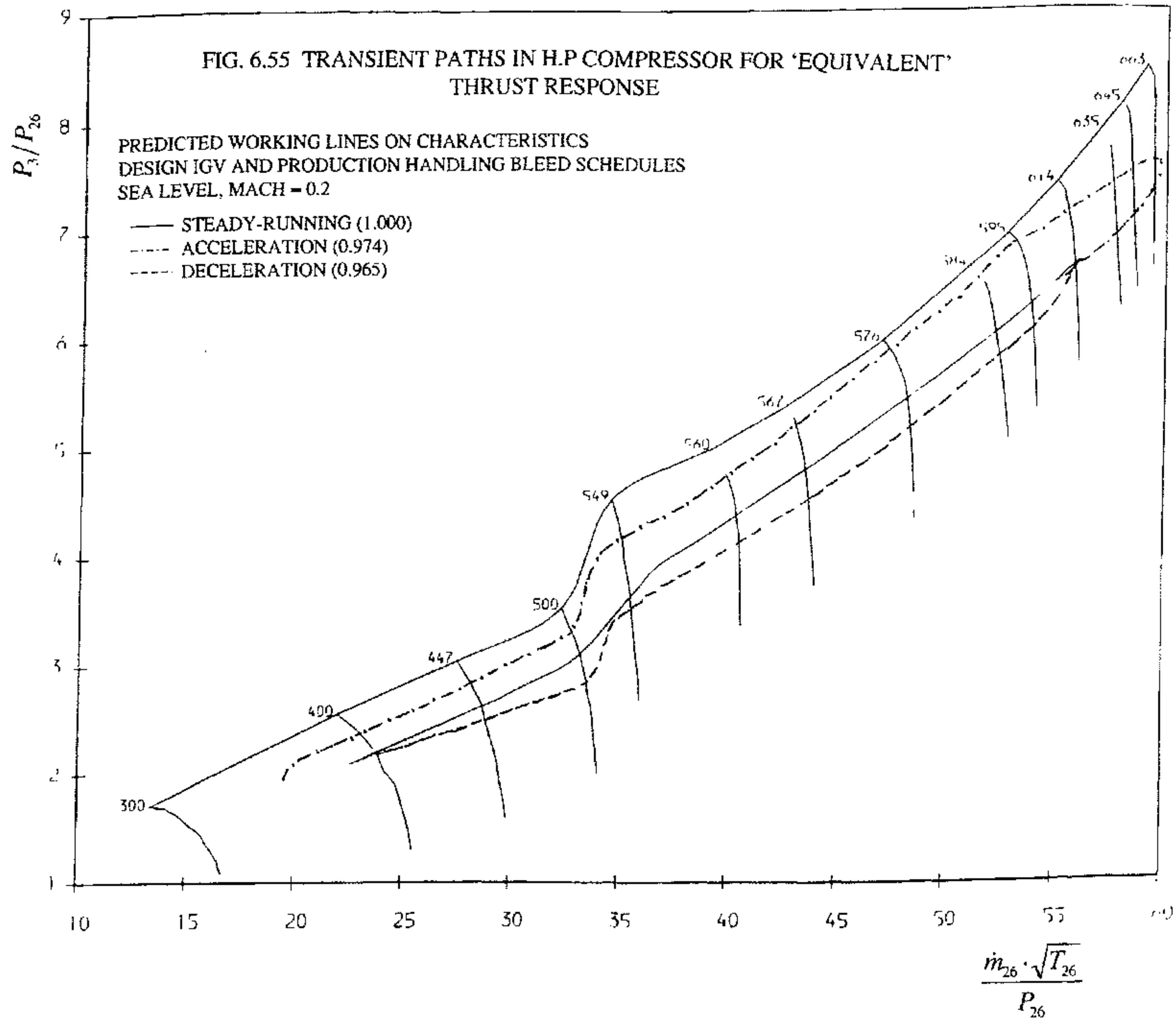


FIG. 6.54 TRANSIENT PATHS IN I.P. COMPRESSOR FOR 'EQUIVALENT' THRUST RESPONSE
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH - 0.2
STEADY-RUNNING (1.000)
ACCELERATION (0.974)
DECELERATION (0.965)
P_{26}/P_{24}
$\frac{\dot{m}_{24} \cdot \sqrt{T_{24}}}{P_{24}}$
476
453
390
346
305
260
175
87
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0
10
20
30
40
50
60
70
80

FIG. 6.55 TRANSIENT PATHS IN H.P COMPRESSOR FOR 'EQUIVALENT' THRUST RESPONSE



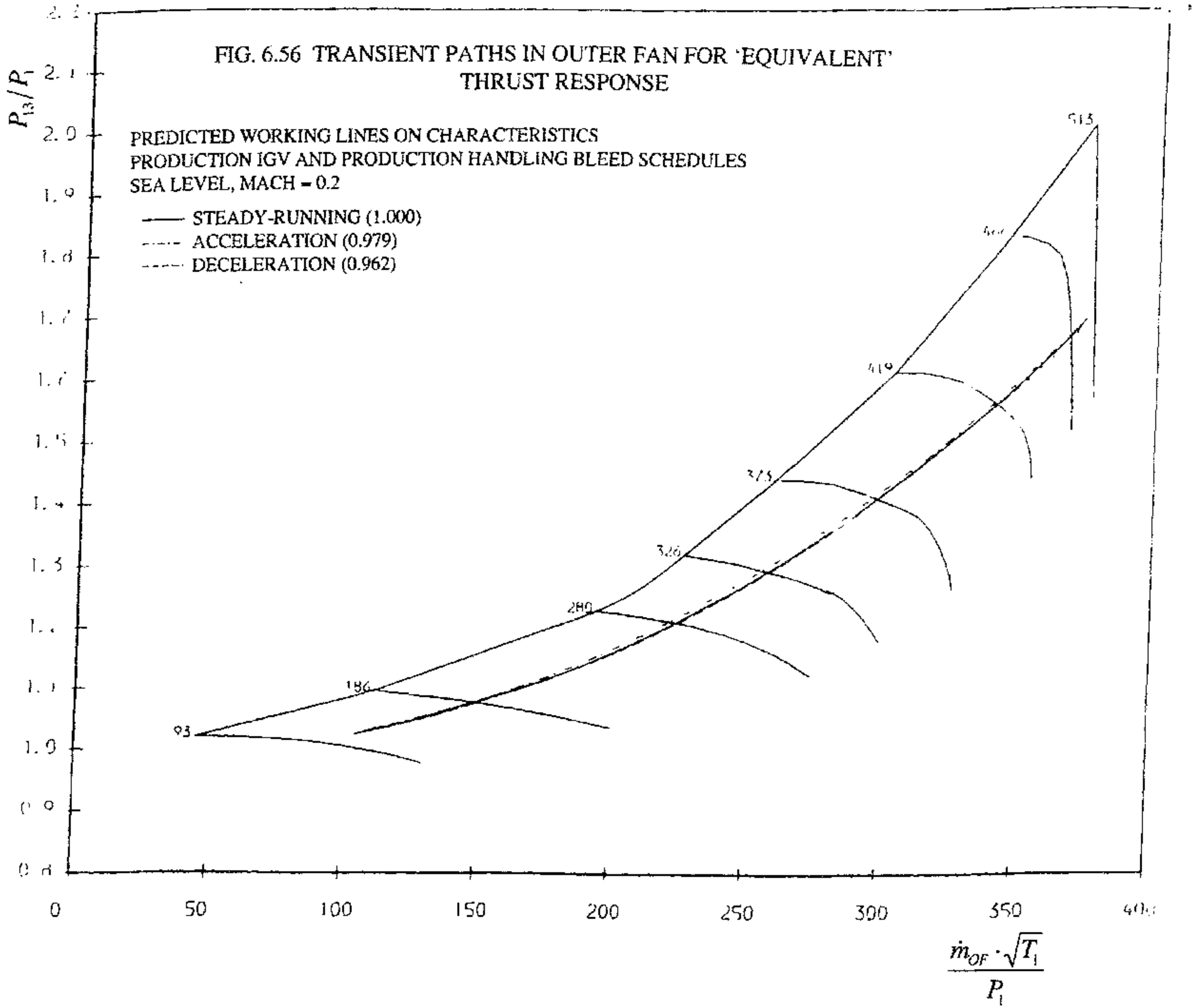


FIG. 6.56 TRANSIENT PATHS IN OUTER FAN FOR 'EQUIVALENT' THRUST RESPONSE
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
STEADY-RUNNING (1.000)
ACCELERATION (0.979)
DECELERATION (0.962)
P_{13}/P_1
$\frac{\dot{m}_{OF} \cdot \sqrt{T_1}}{P_1}$
0
50
100
150
200
250
300
350
93
186
280
326
375
419
513

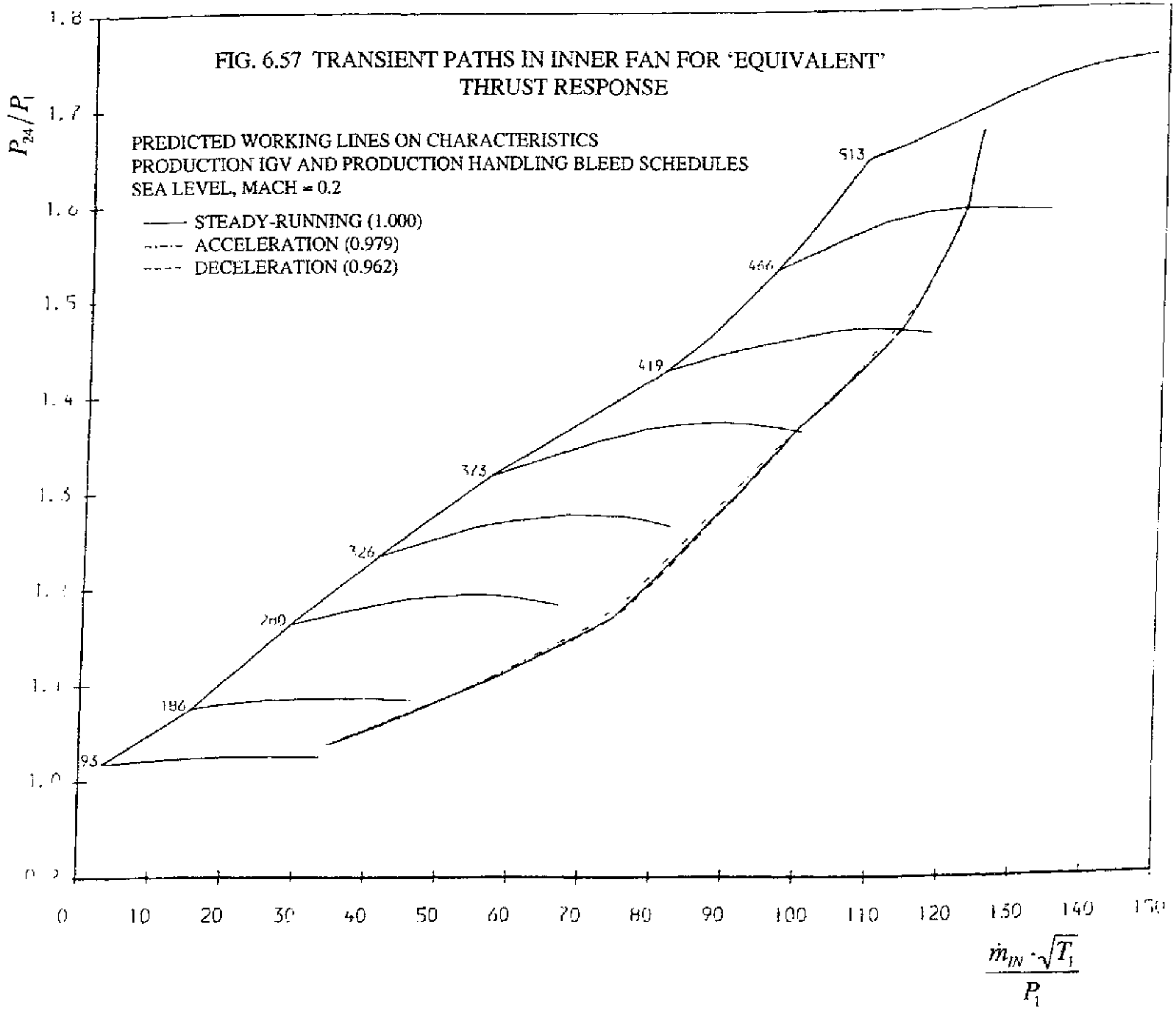


FIG. 6.57 TRANSIENT PATHS IN INNER FAN FOR 'EQUIVALENT' THRUST RESPONSE

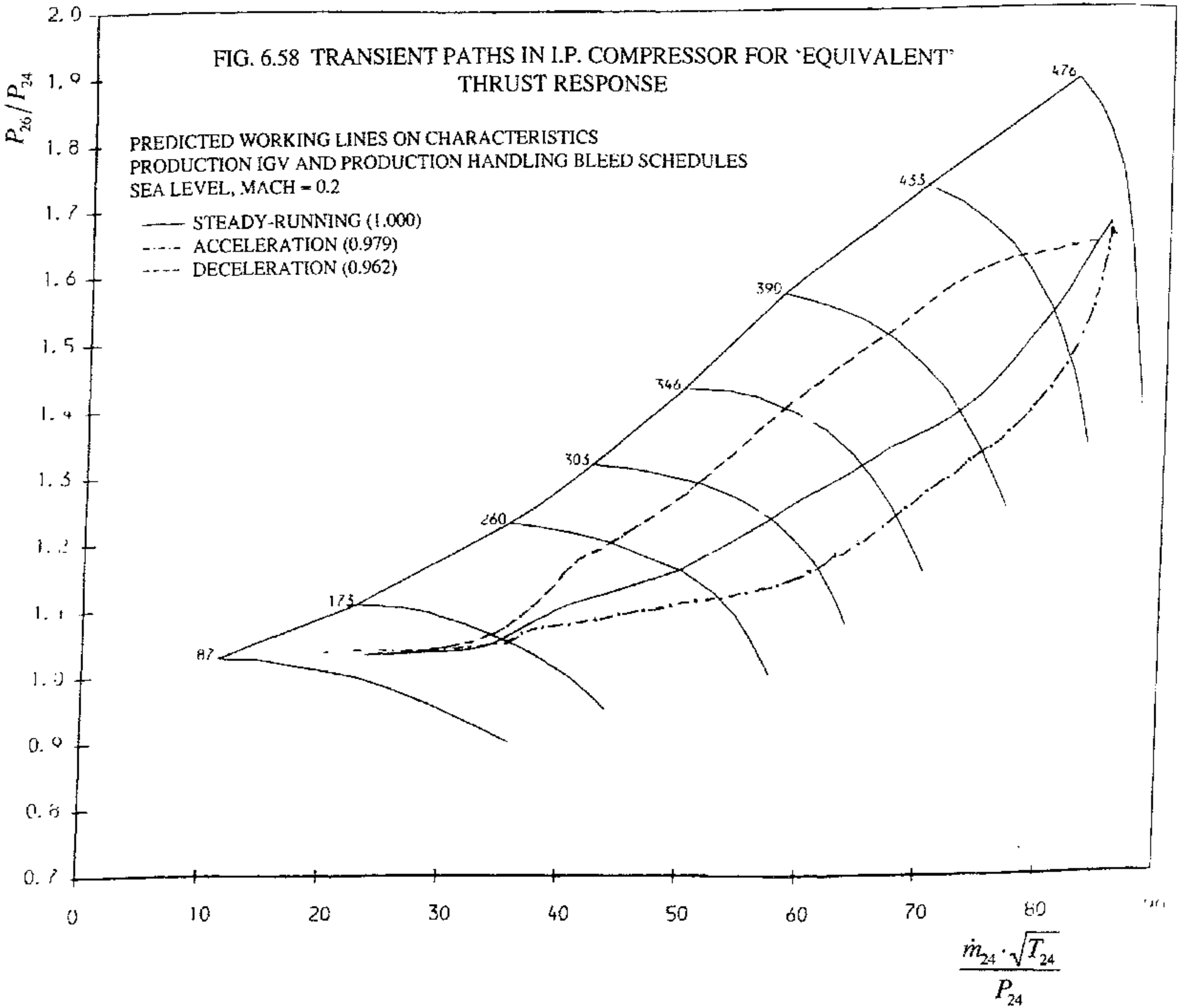


FIG. 6.58 TRANSIENT PATHS IN I.P. COMPRESSOR FOR 'EQUIVALENT' THRUST RESPONSE

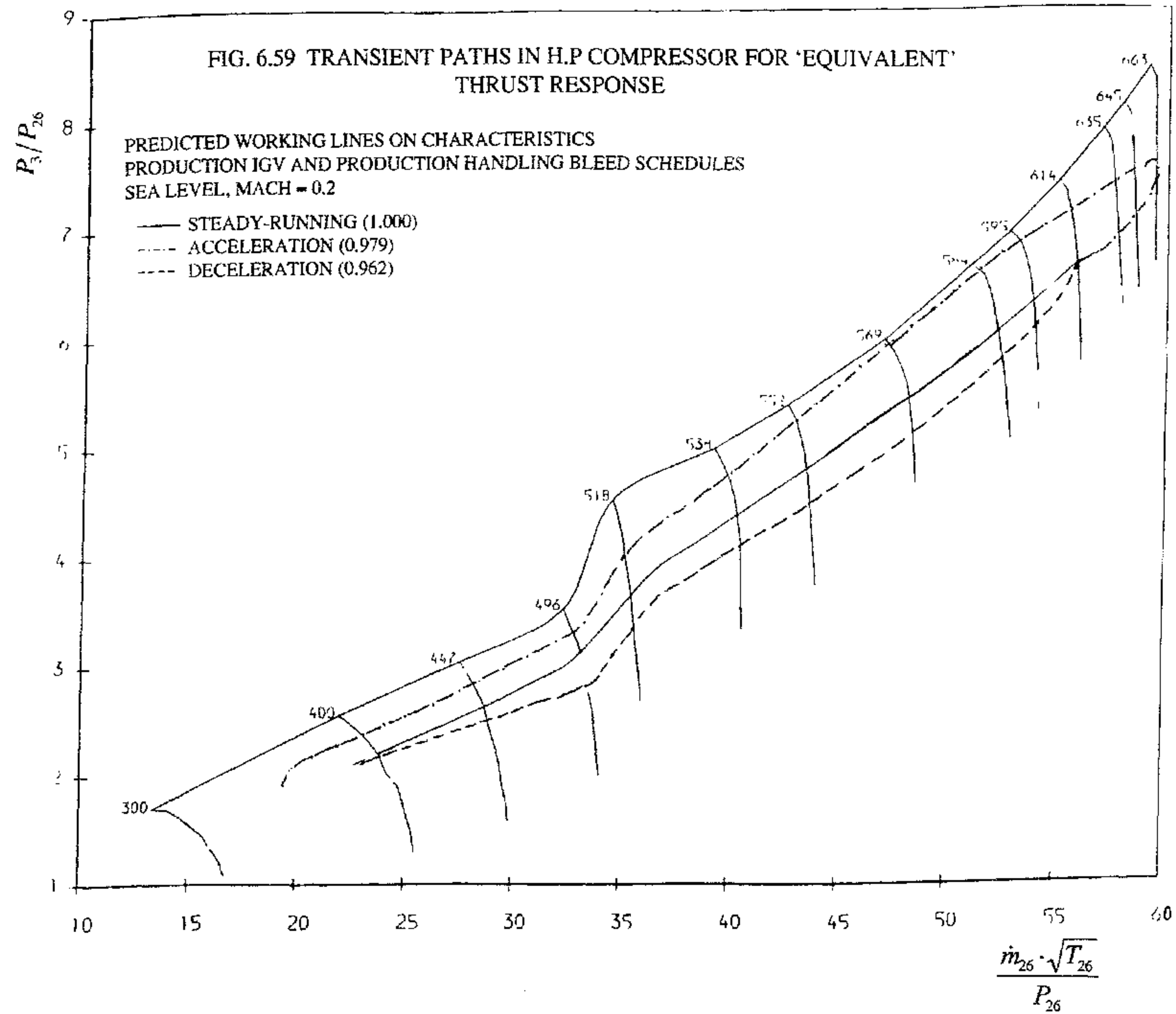


FIG. 6.59 TRANSIENT PATHS IN H.P COMPRESSOR FOR 'EQUIVALENT' THRUST RESPONSE
PREDICTED WORKING LINES ON CHARACTERISTICS
PRODUCTION IGV AND PRODUCTION HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
STEADY-RUNNING (1.000)
ACCELERATION (0.979)
DECELERATION (0.962)
P_3/P_{26}
$\frac{\dot{m}_{26}\cdot\sqrt{T_{26}}}{P_{26}}$
300
400
447
496
518
534
569
614
635
663
10
15
20
25
30
35
40
45
50
55
60
1
2
3
4
5
6
7
8
9

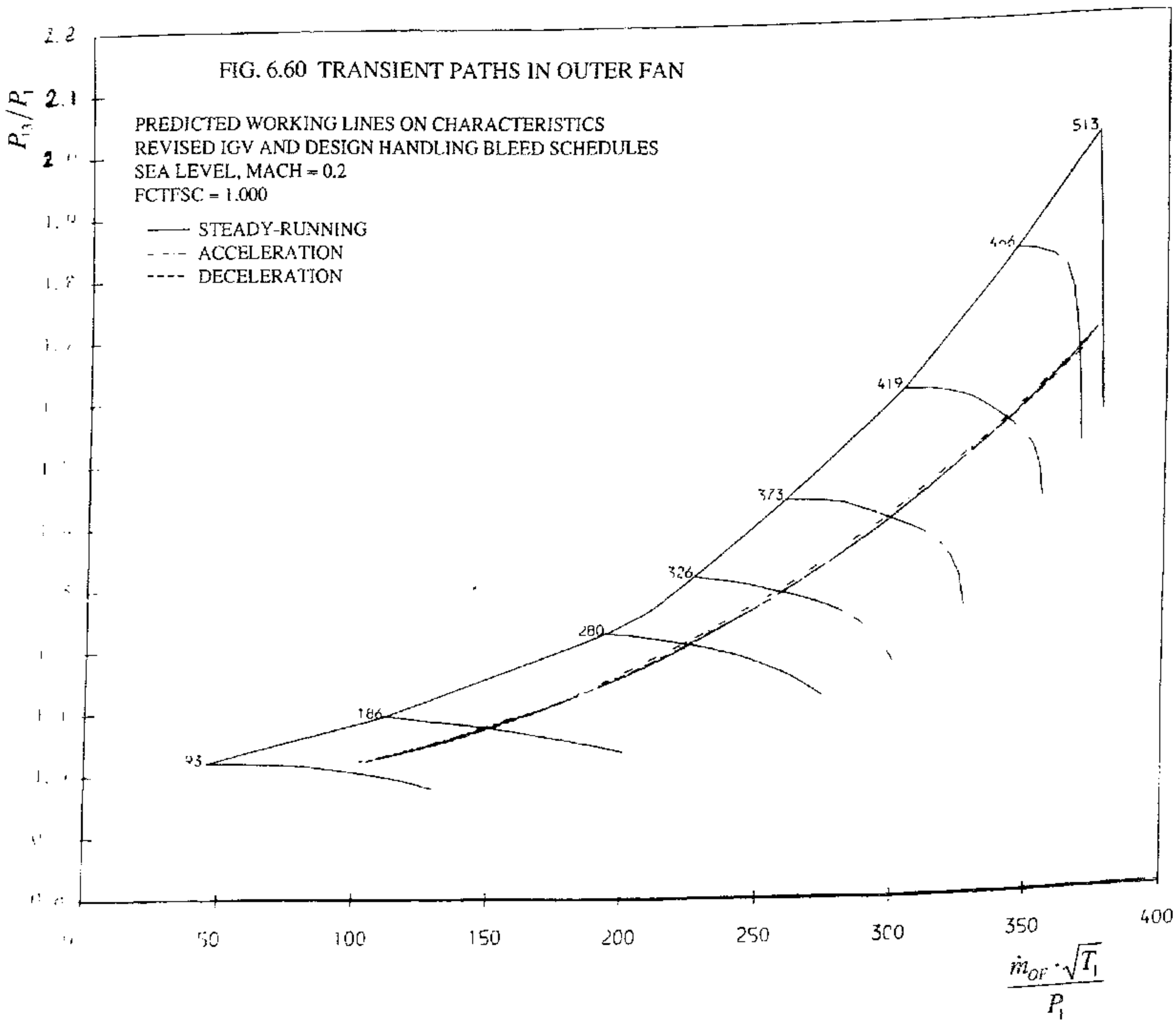


FIG. 6.60 TRANSIENT PATHS IN OUTER FAN

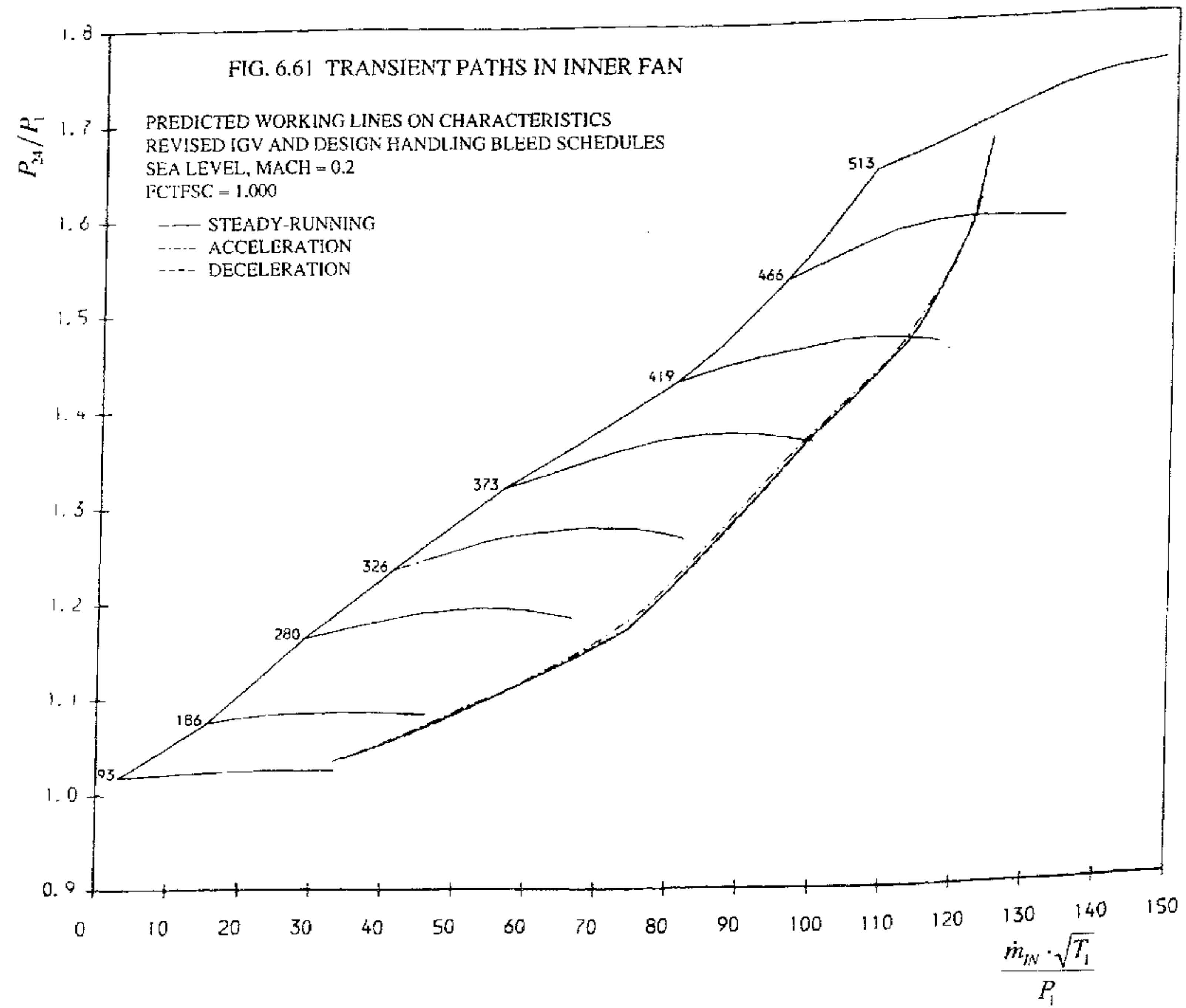


FIG. 6.61 TRANSIENT PATHS IN INNER FAN
PREDICTED WORKING LINES ON CHARACTERISTICS
REVISED IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{24}/P_1
$\frac{\dot{m}_{IN} \cdot \sqrt{T_1}}{P_1}$
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
93
186
280
326
373
419
466
513

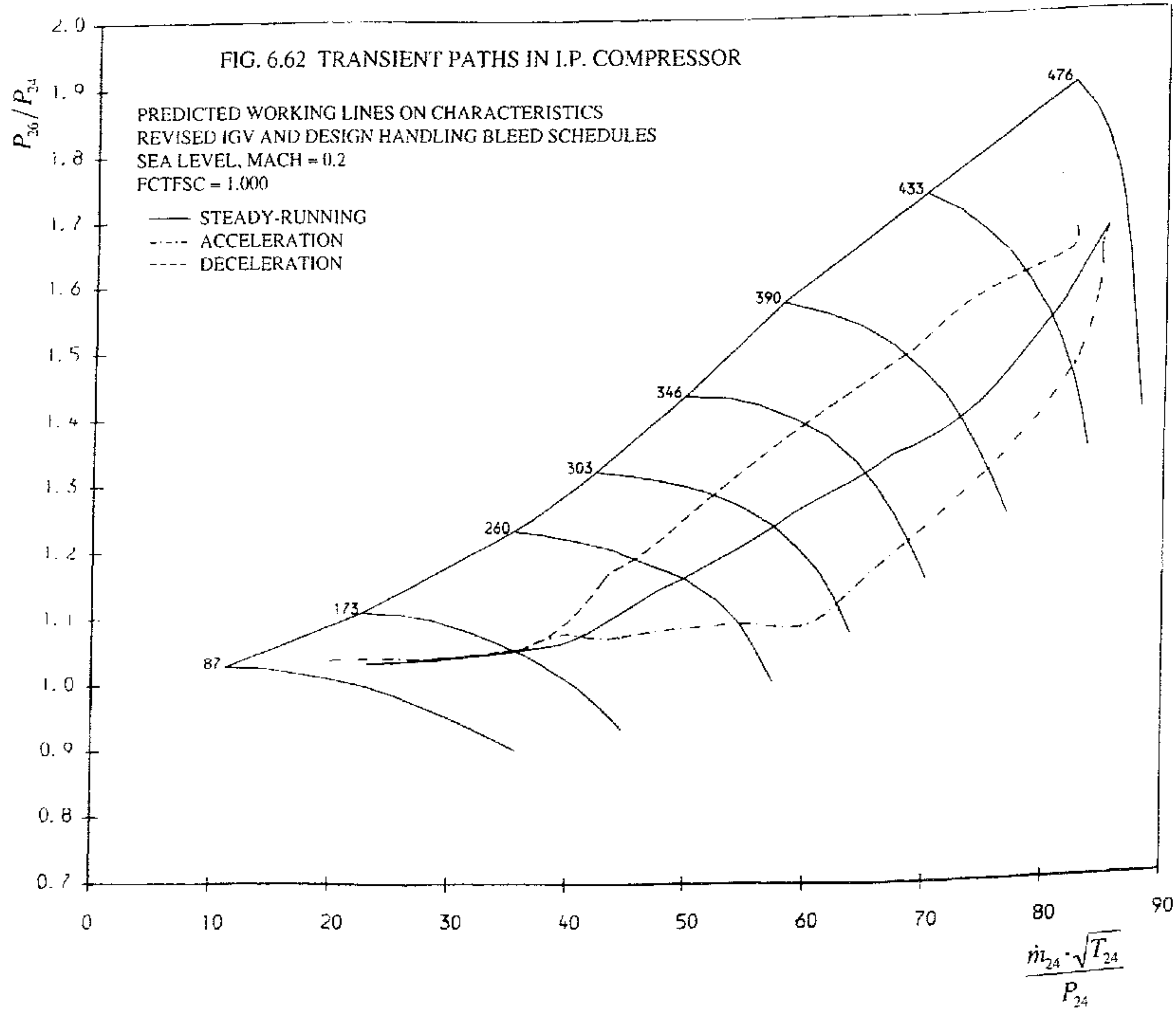


FIG. 6.62 TRANSIENT PATHS IN I.P. COMPRESSOR
PREDICTED WORKING LINES ON CHARACTERISTICS
REVISED IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{26}/P_{24}
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0
10
20
30
40
50
60
70
80
90
$\frac{\dot{m}_{24} \cdot \sqrt{T_{24}}}{P_{24}}$
87
173
260
303
346
390
433
476

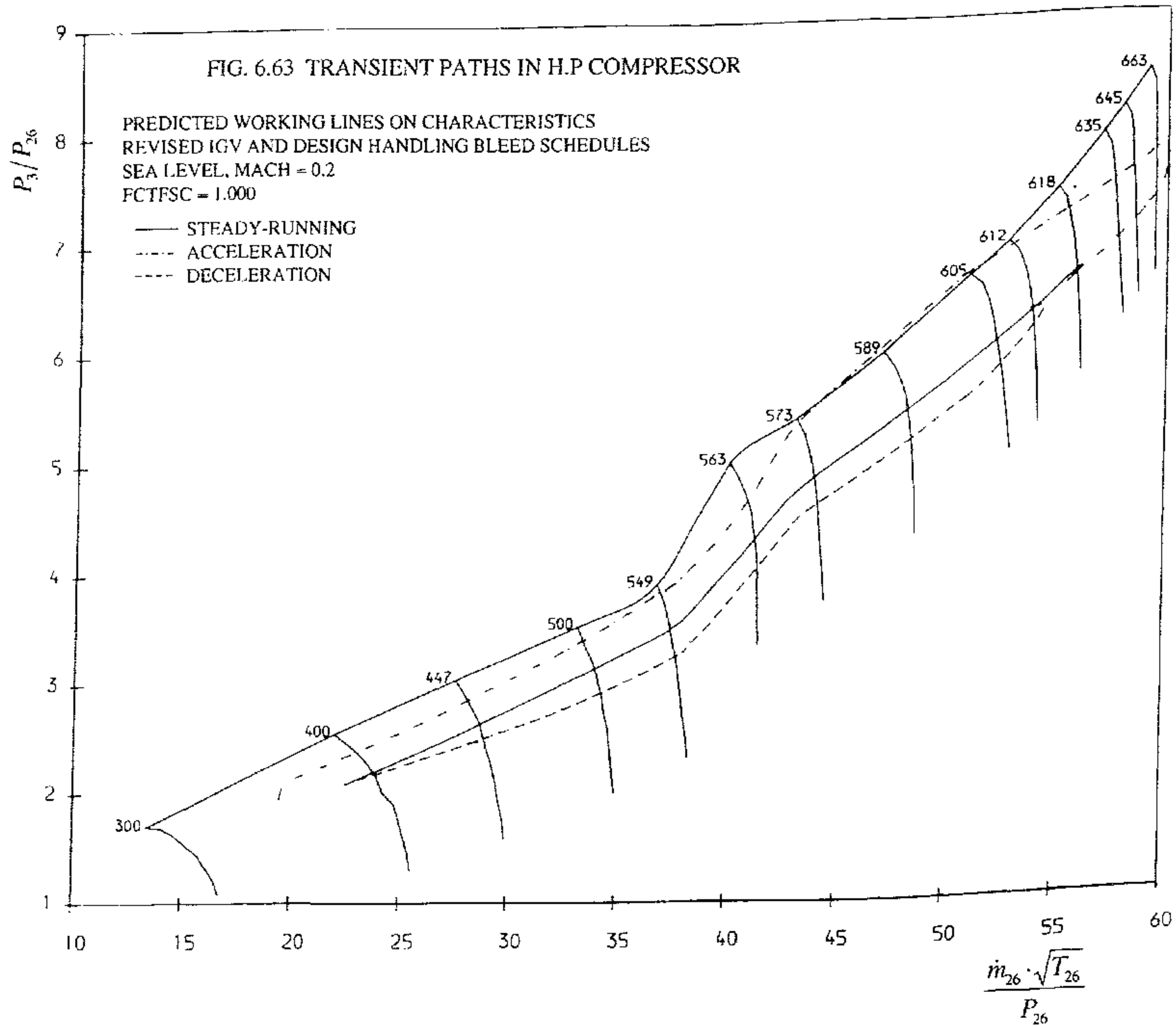


FIG. 6.63 TRANSIENT PATHS IN H.P COMPRESSOR
PREDICTED WORKING LINES ON CHARACTERISTICS
REVISED IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_3/P_{26}
$\frac{\dot{m}_{26}\cdot\sqrt{T_{26}}}{P_{26}}$
1
2
3
4
5
6
7
8
9
10
15
20
25
30
35
40
45
50
55
60
300
400
447
500
549
563
573
589
605
612
618
635
645
663

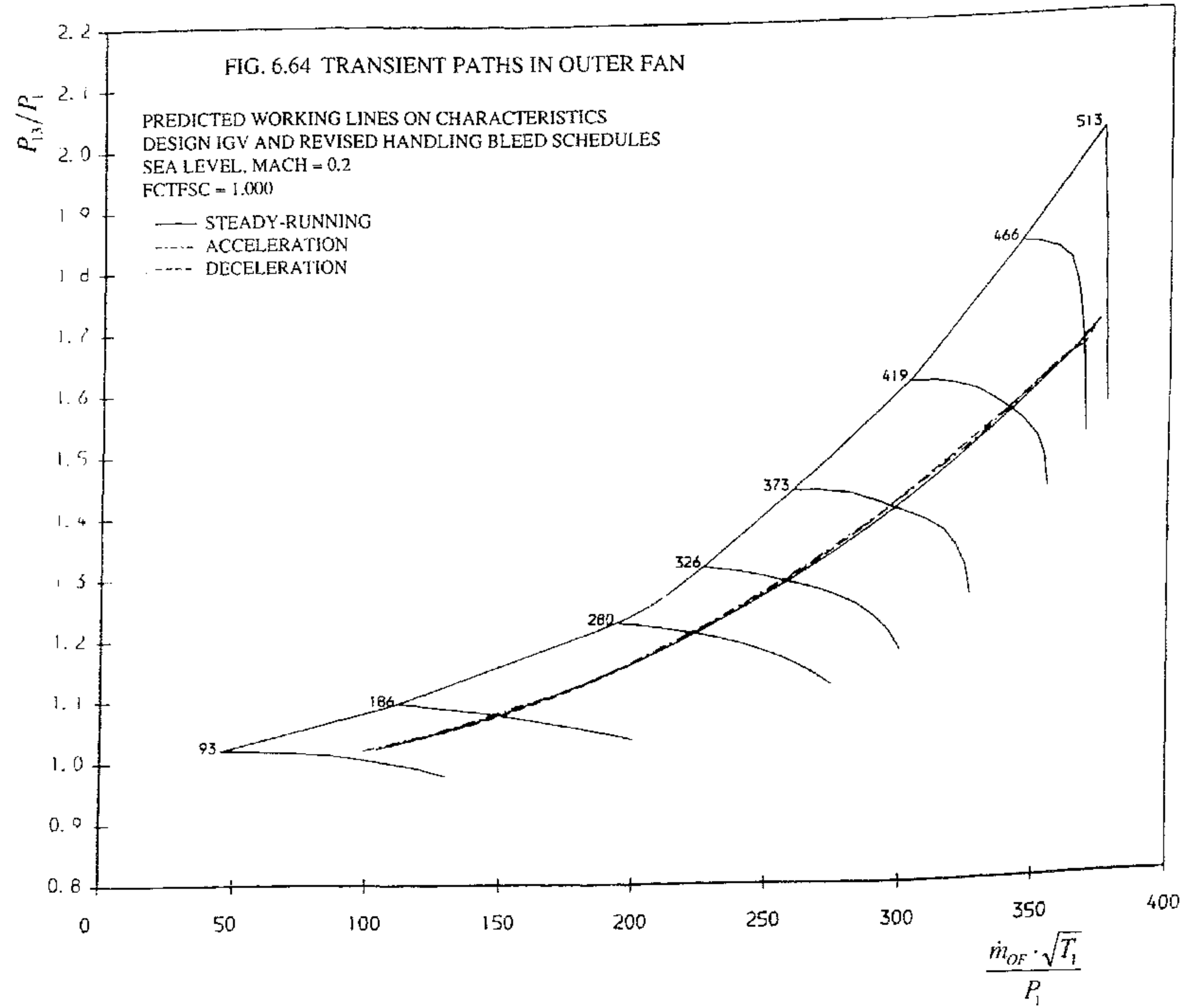
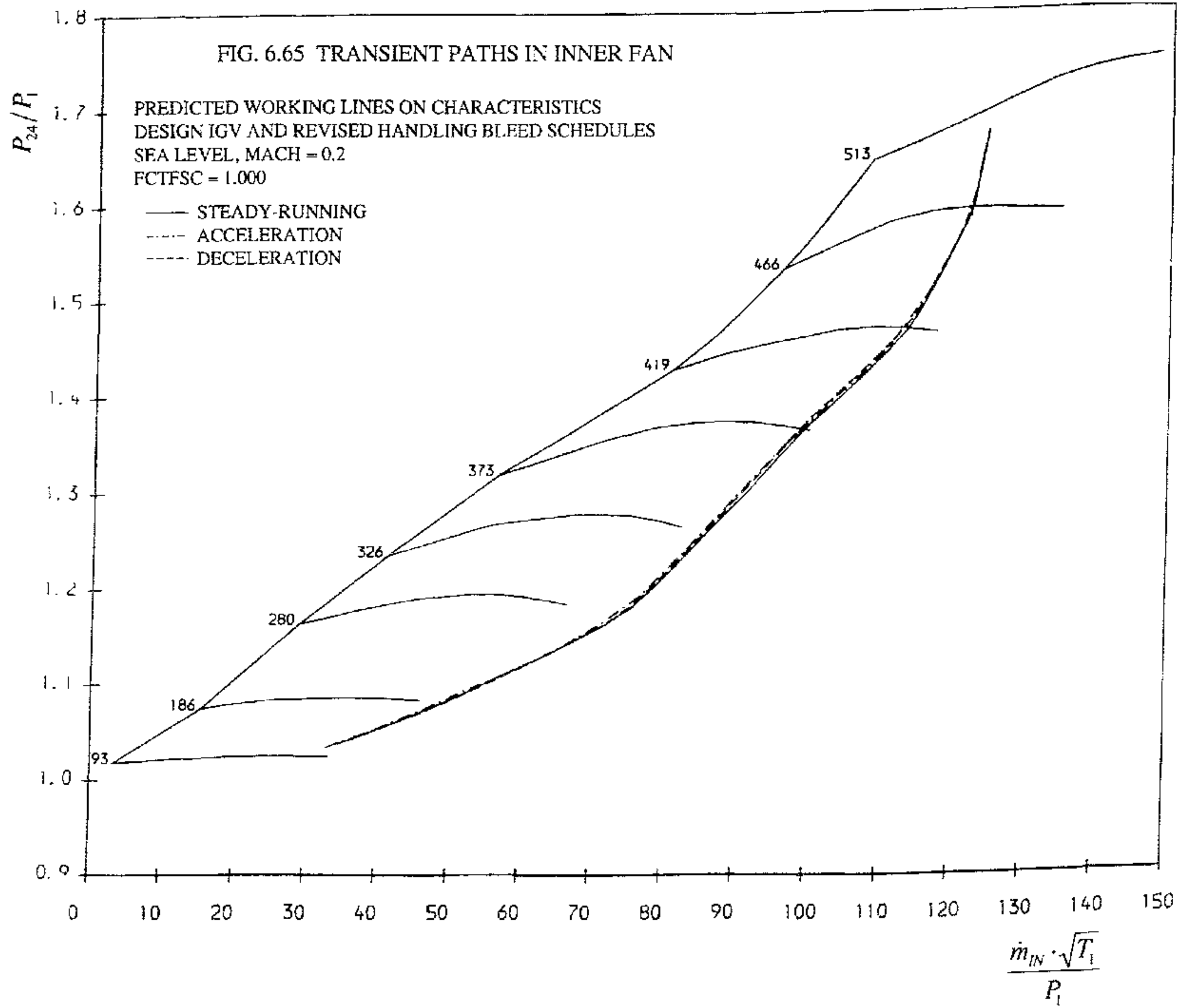


FIG. 6.64 TRANSIENT PATHS IN OUTER FAN

FIG. 6.65 TRANSIENT PATHS IN INNER FAN



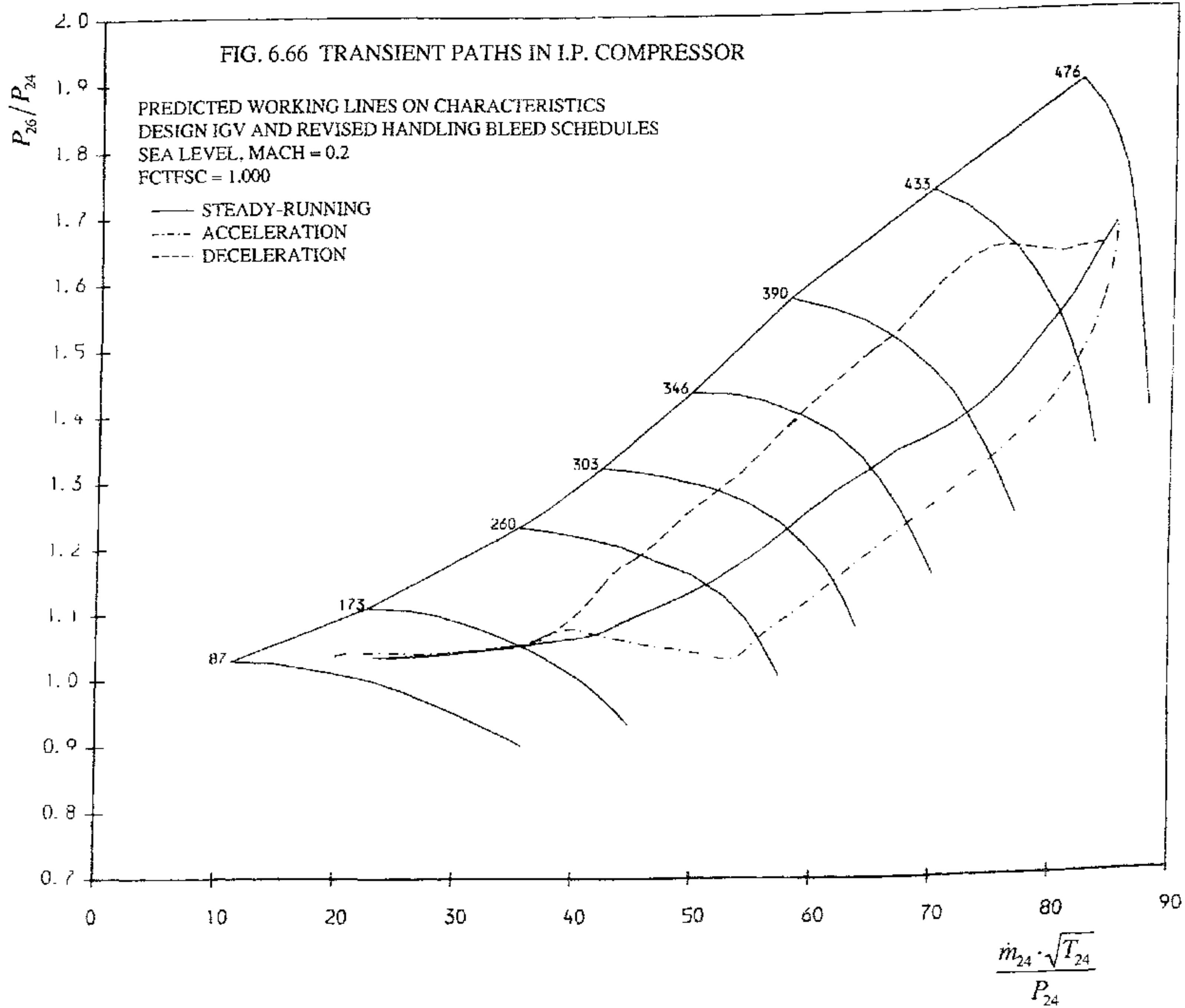


FIG. 6.66 TRANSIENT PATHS IN I.P. COMPRESSOR
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND REVISED HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{26}/P_{24}
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0
10
20
30
40
50
60
70
80
90
$\frac{\dot{m}_{24}\cdot\sqrt{T_{24}}}{P_{24}}$
87
173
260
303
346
390
433
476

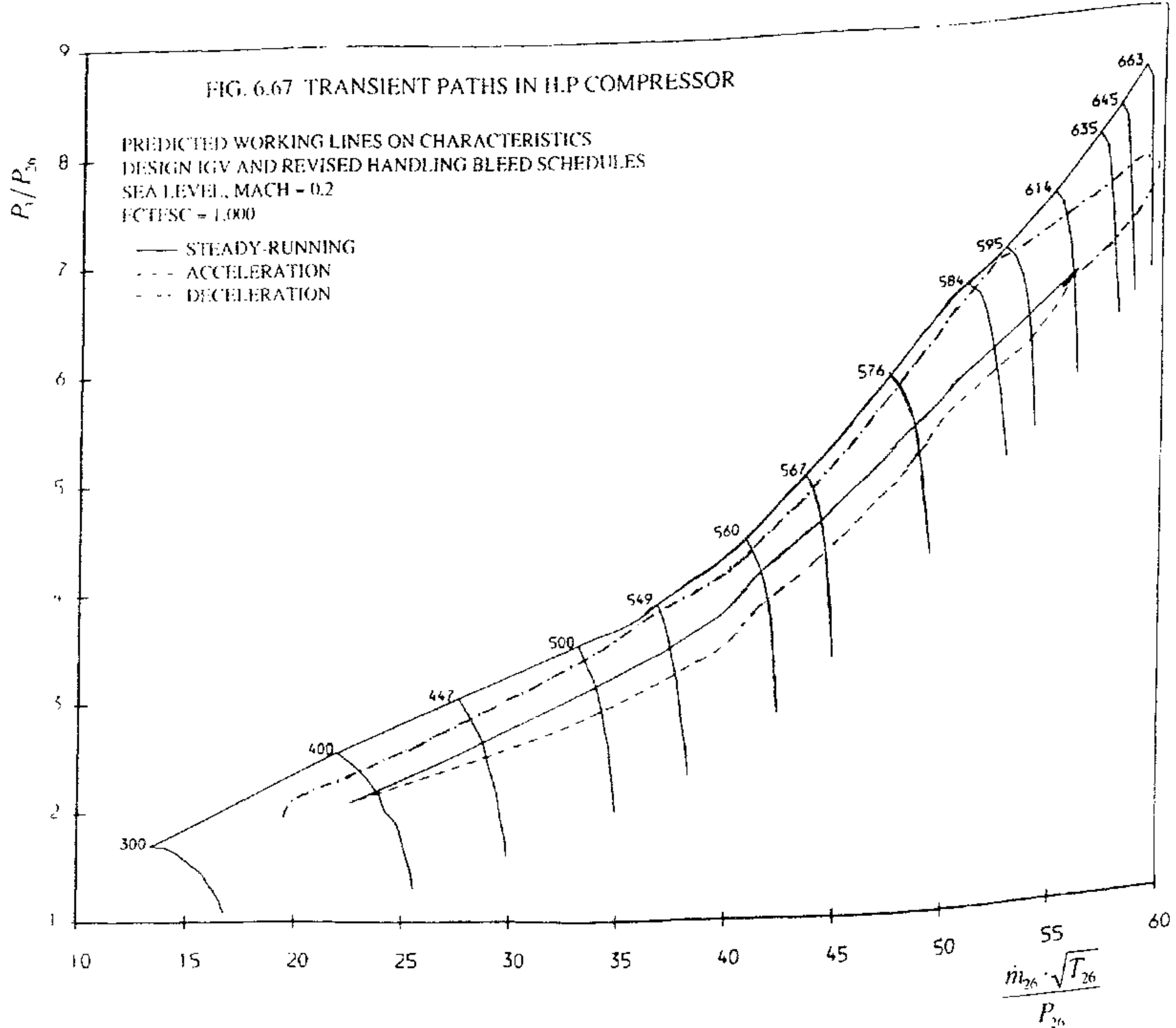
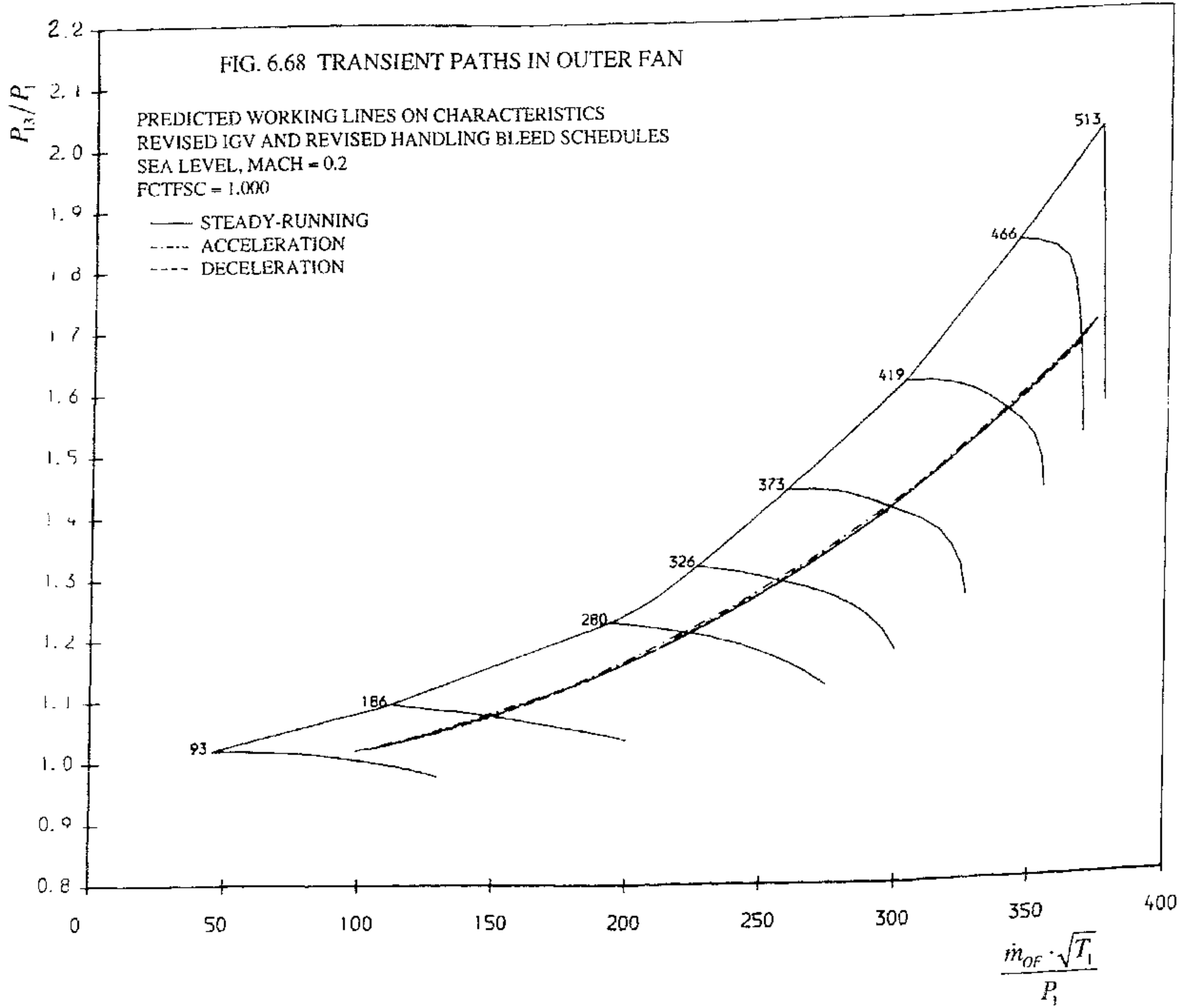


FIG. 6.67 TRANSIENT PATHS IN H.P COMPRESSOR

FIG. 6.68 TRANSIENT PATHS IN OUTER FAN



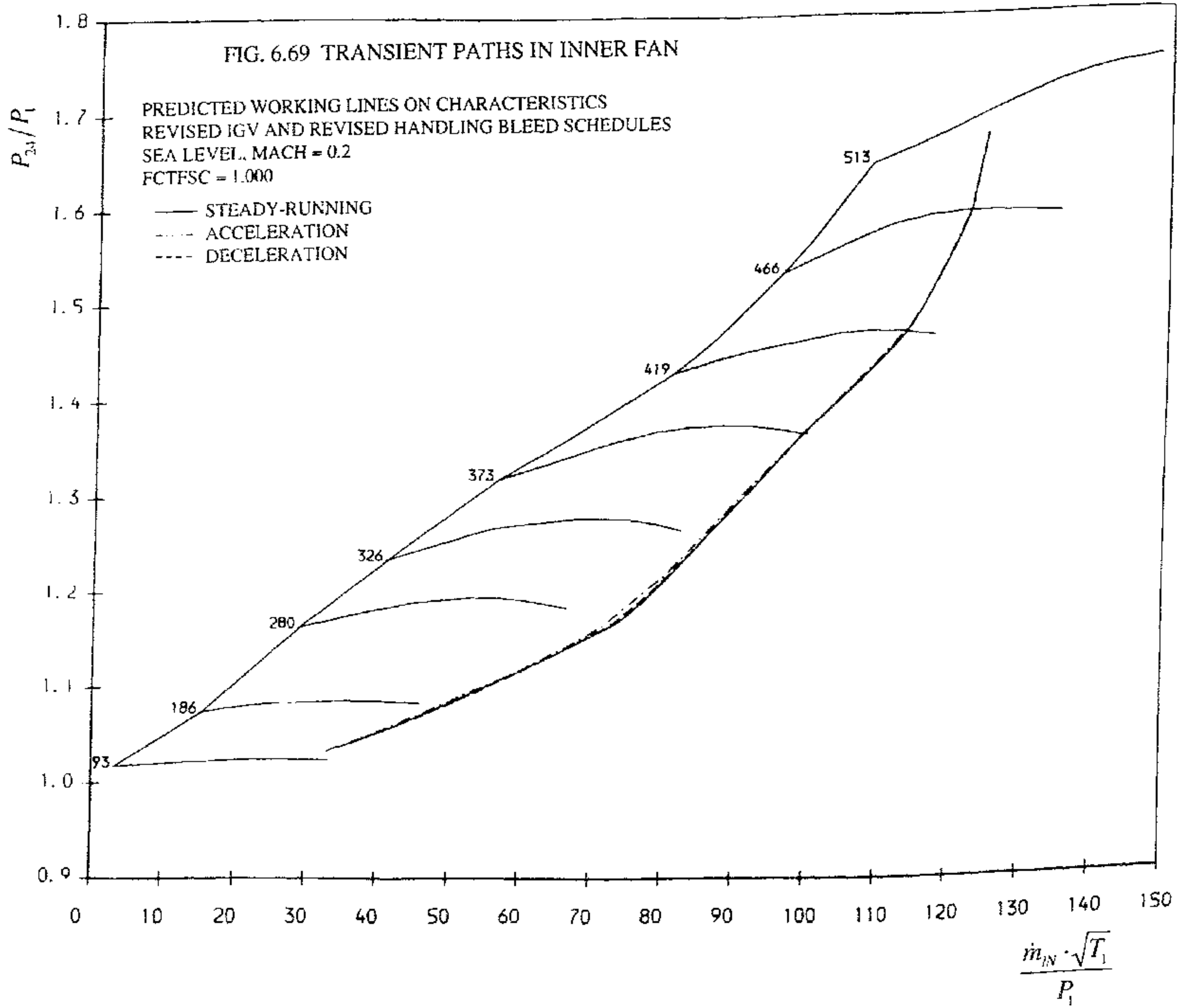
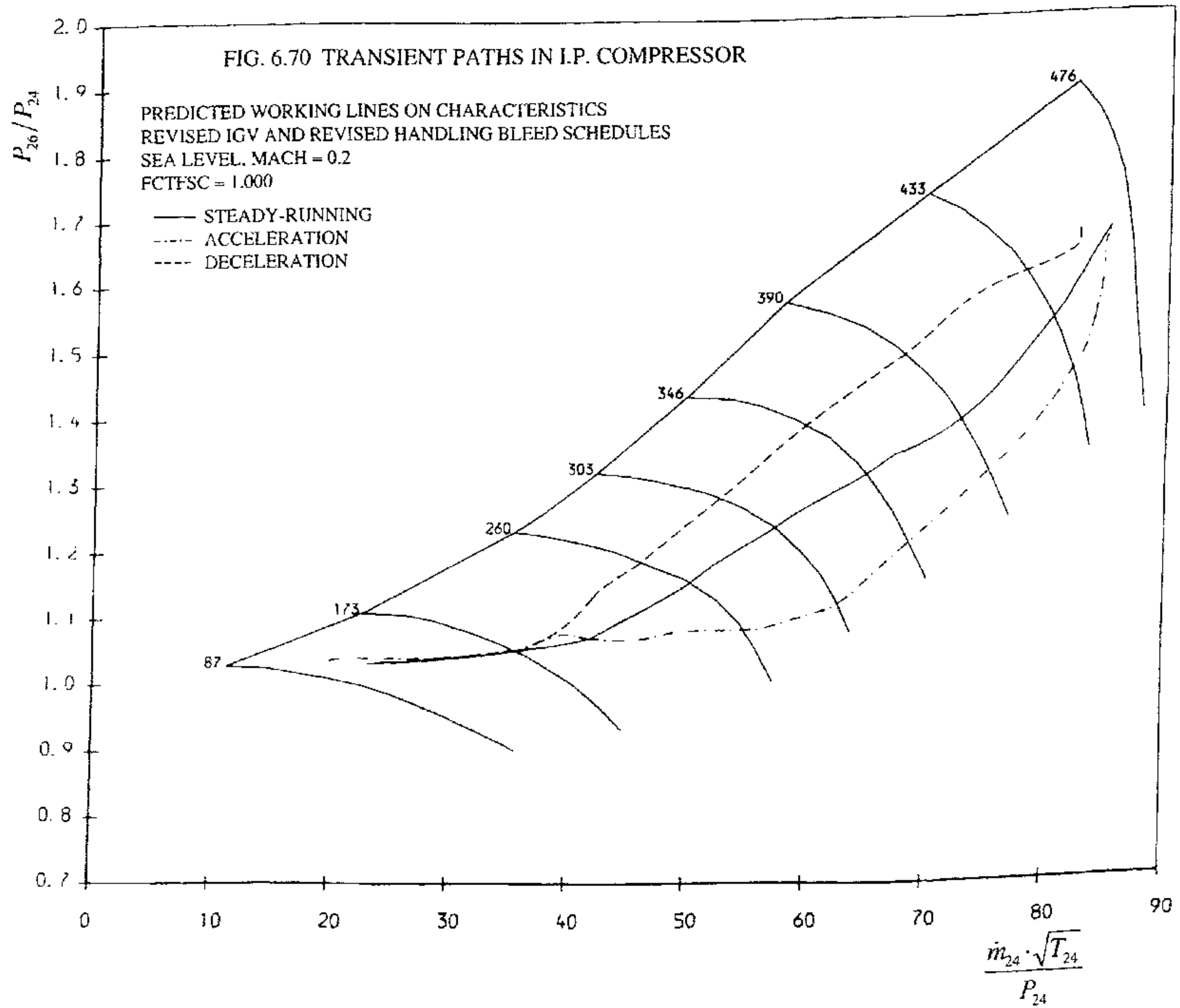


FIG. 6.69 TRANSIENT PATHS IN INNER FAN

FIG. 6.70 TRANSIENT PATHS IN I.P. COMPRESSOR



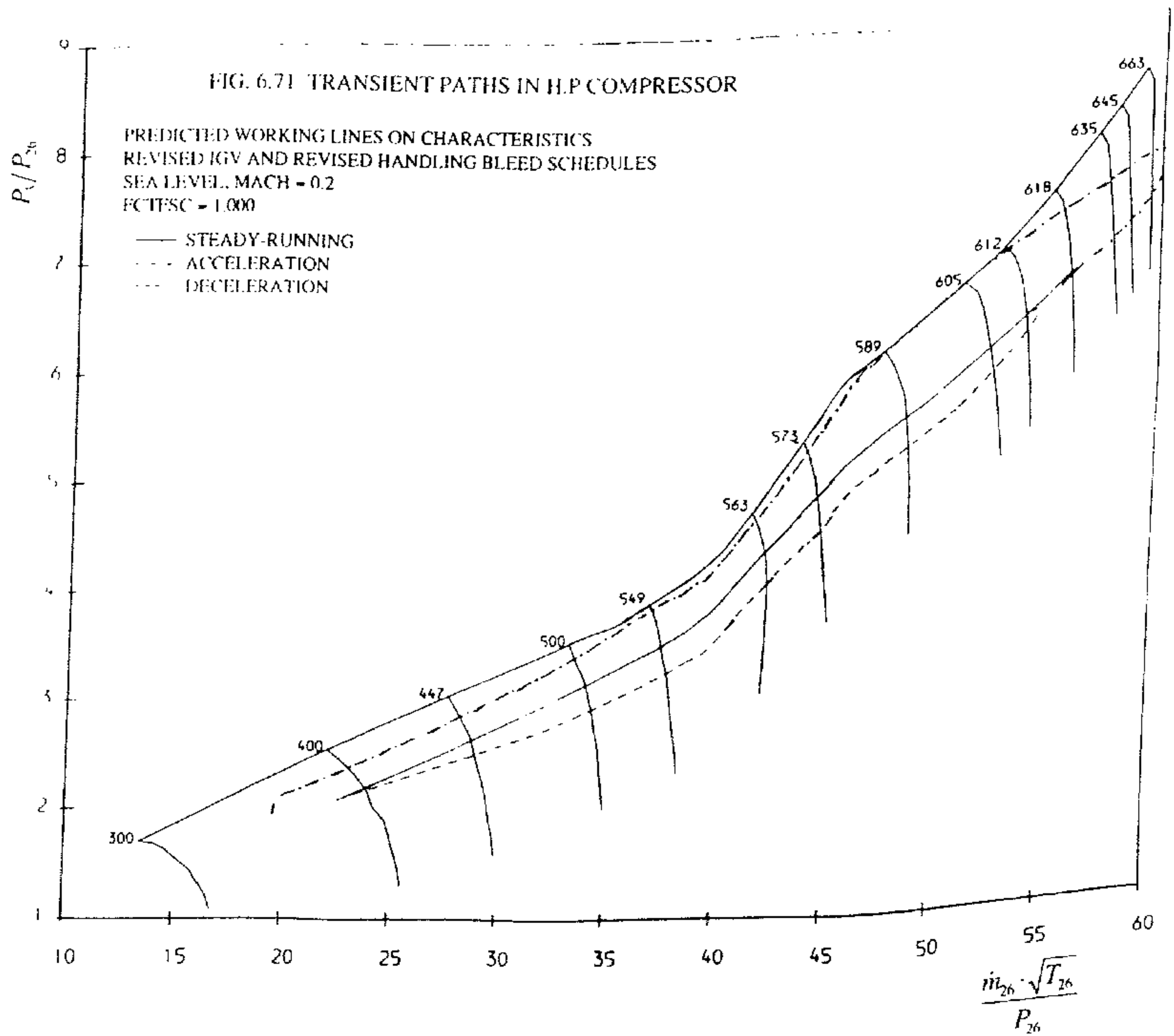


FIG. 6.71 TRANSIENT PATHS IN H.P COMPRESSOR

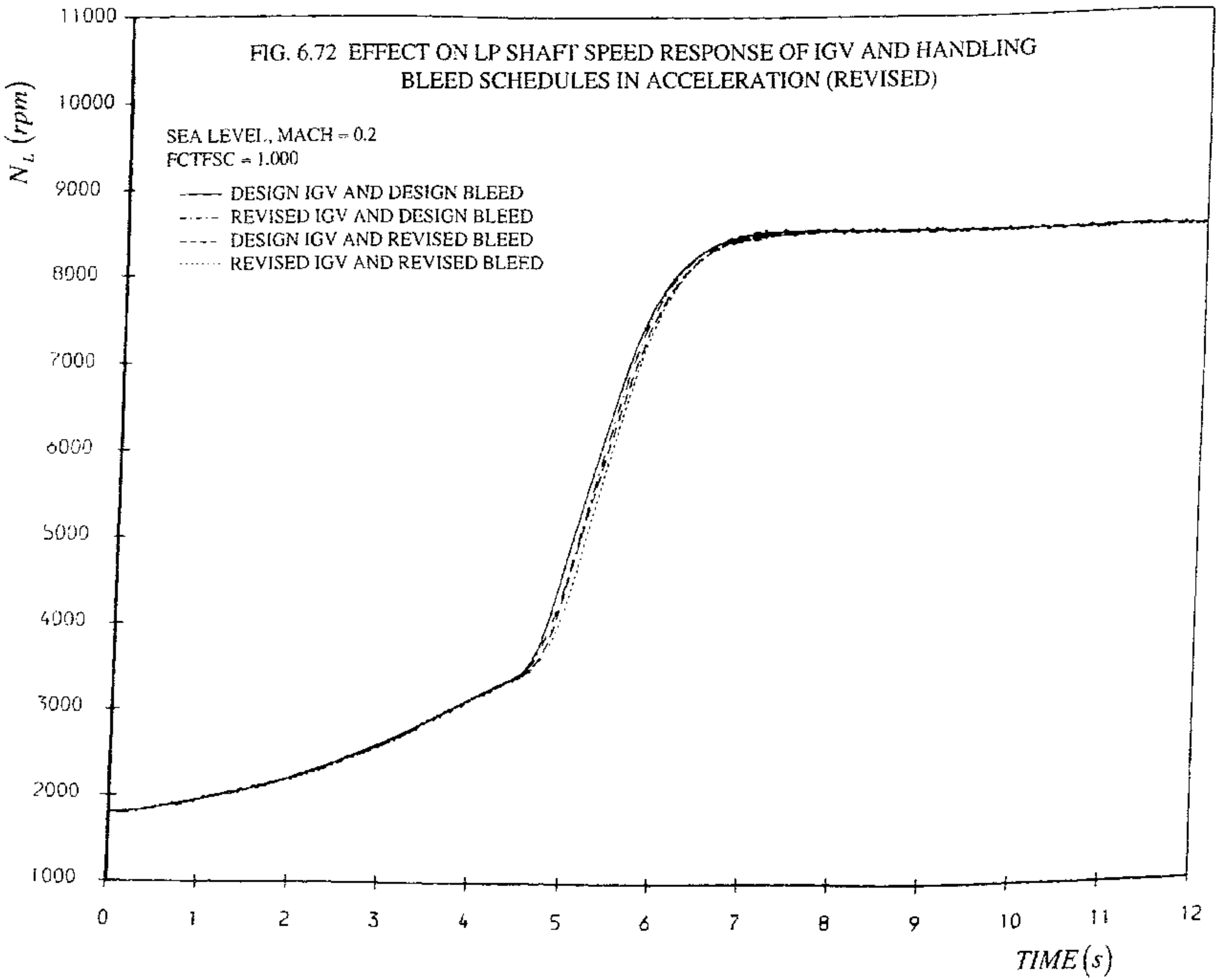


FIG. 6.72 EFFECT ON LP SHAFT SPEED RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION (REVISED)
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
REVISED IGV AND DESIGN BLEED
DESIGN IGV AND REVISED BLEED
REVISED IGV AND REVISED BLEED
N_L (rpm)
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
1
2
3
4
5
6
7
8
9
10
11
12
TIME (s)

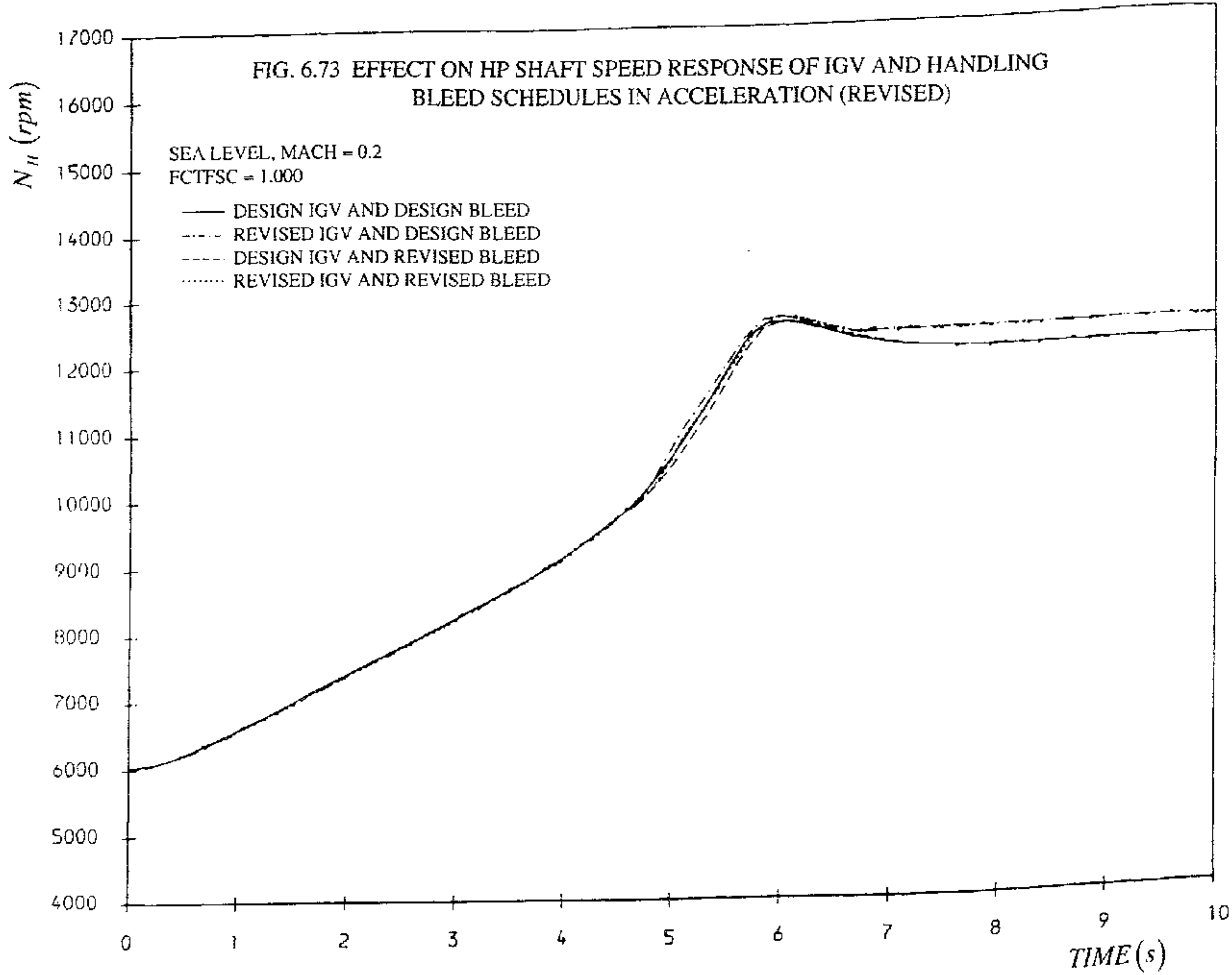


FIG. 6.73 EFFECT ON HP SHAFT SPEED RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION (REVISED)
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
REVISED IGV AND DESIGN BLEED
DESIGN IGV AND REVISED BLEED
REVISED IGV AND REVISED BLEED
N_H (rpm)
17000
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
0
1
2
3
4
5
6
7
8
9
10
TIME (s)

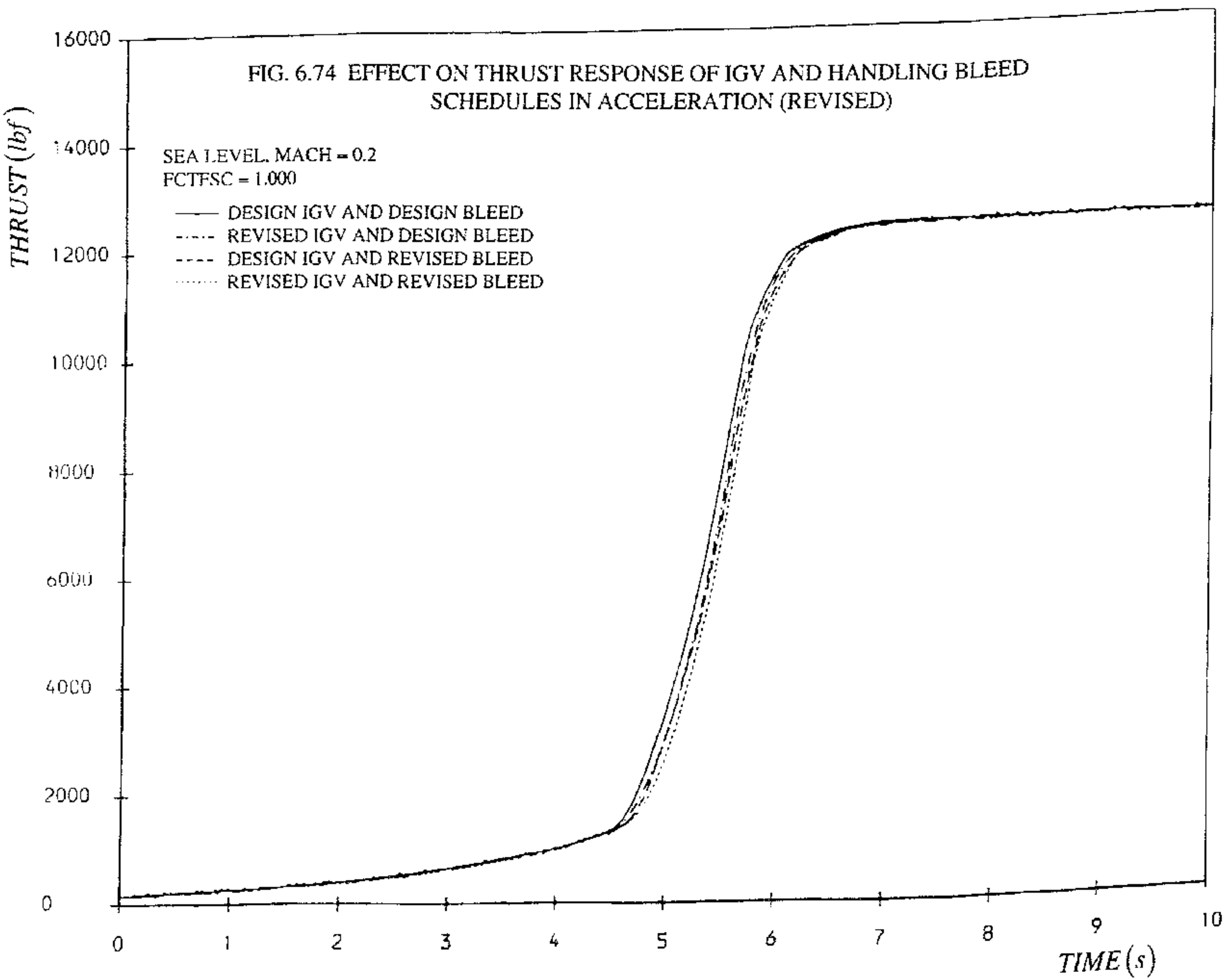


FIG. 6.74 EFFECT ON THRUST RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION (REVISED)
SEA LEVEL. MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
REVISED IGV AND DESIGN BLEED
DESIGN IGV AND REVISED BLEED
REVISED IGV AND REVISED BLEED
THRUST (lbf)
16000
14000
12000
10000
8000
6000
4000
2000
0
0
1
2
3
4
5
6
7
8
9
10
TIME (s)

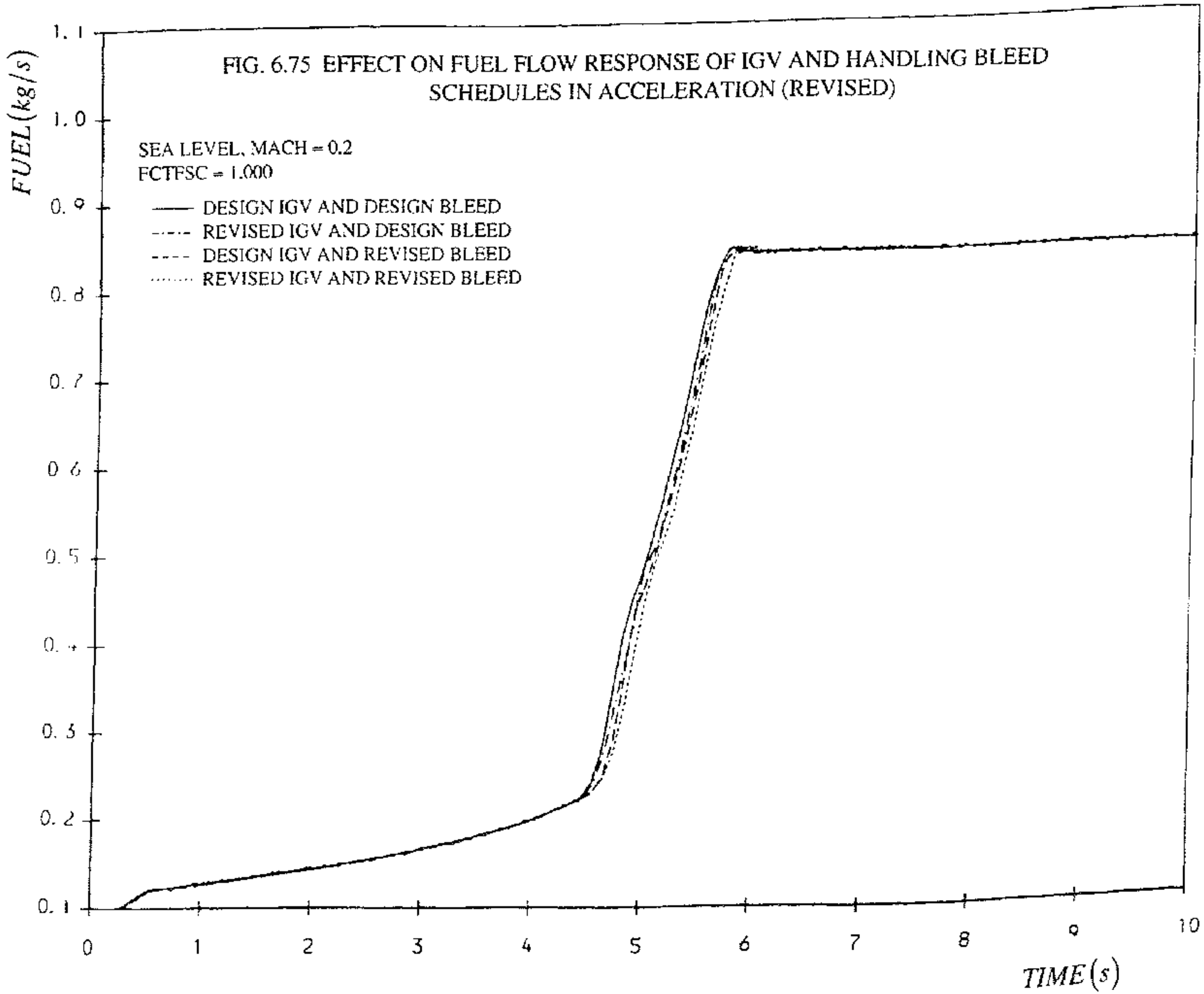


FIG. 6.75 EFFECT ON FUEL FLOW RESPONSE OF IGV AND HANDLING BLEED
SCHEDULES IN ACCELERATION (REVISED)
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
REVISED IGV AND DESIGN BLEED
DESIGN IGV AND REVISED BLEED
REVISED IGV AND REVISED BLEED
FUEL(kg/s)
TIME(s)
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1
2
3
4
5
6
7
8
9
10

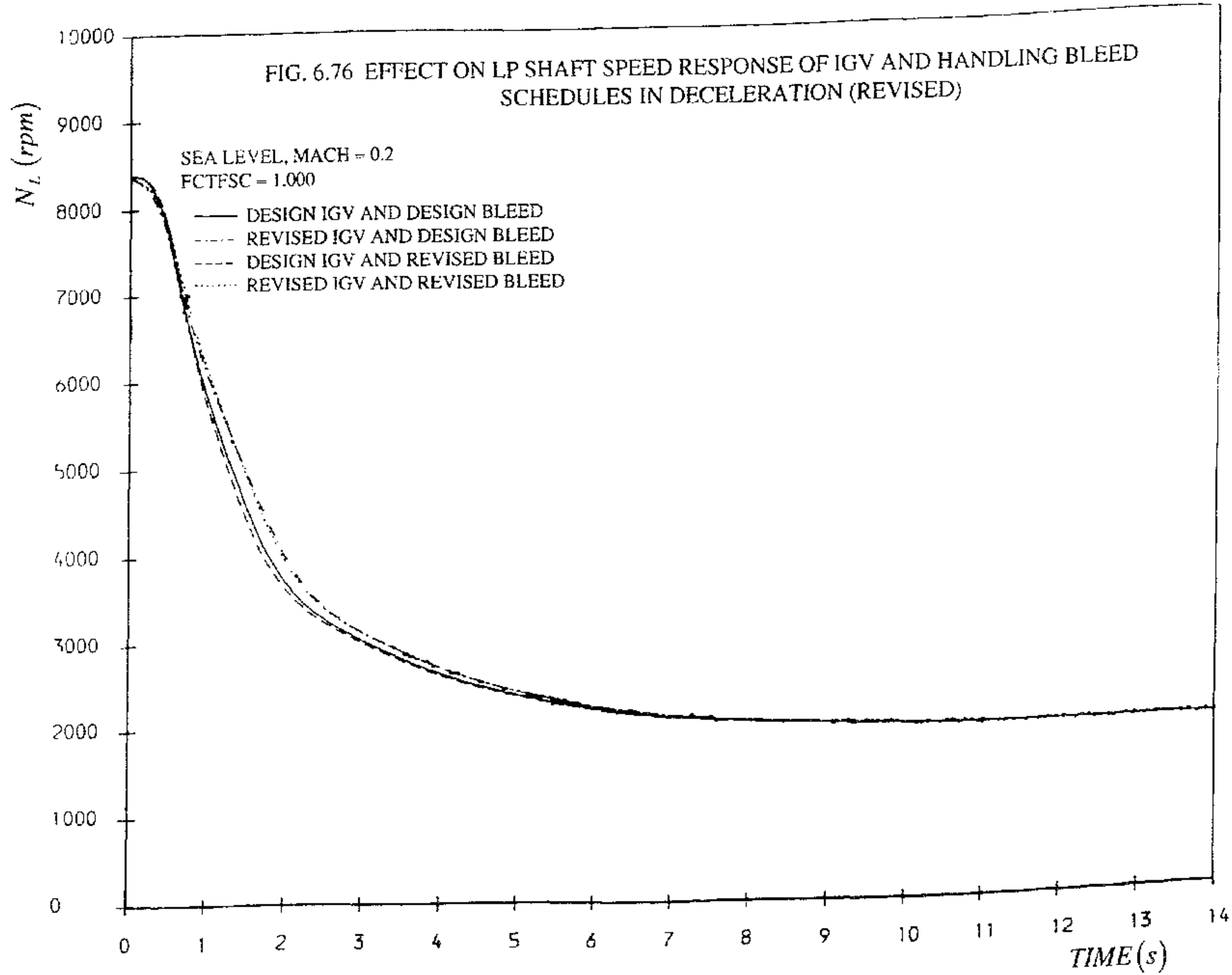


FIG. 6.76 EFFECT ON LP SHAFT SPEED RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION (REVISED)

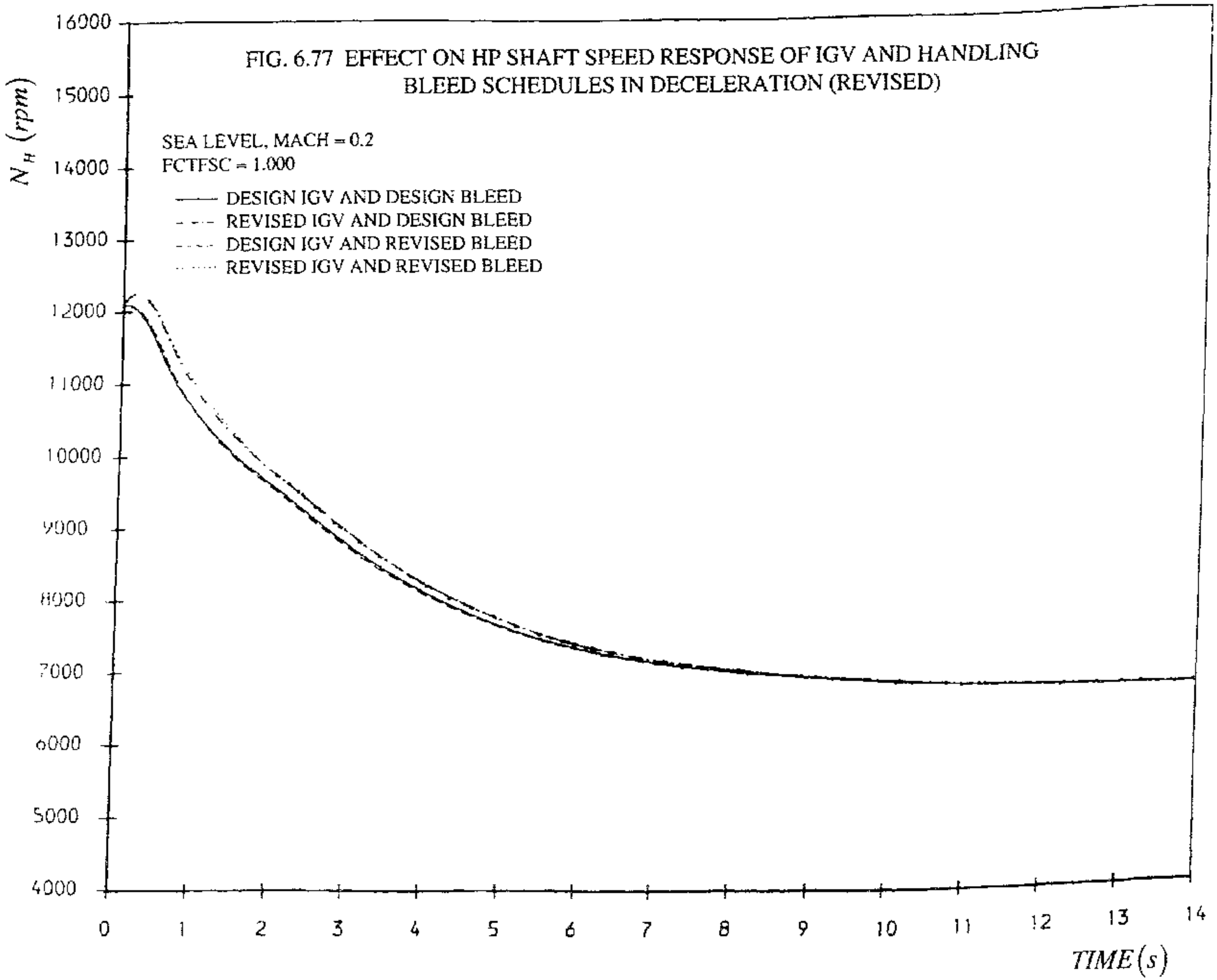


FIG. 6.77 EFFECT ON HP SHAFT SPEED RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION (REVISED)

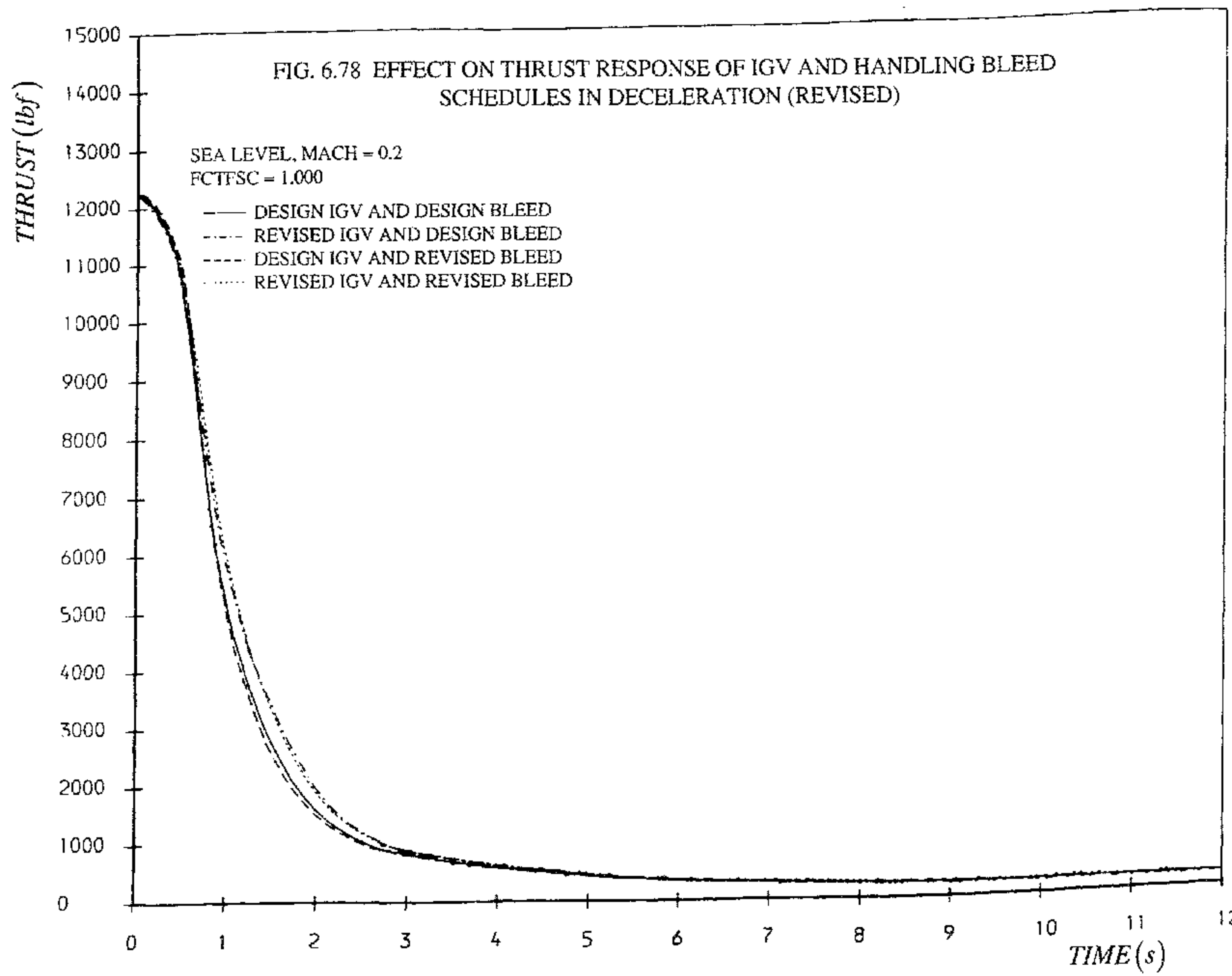


FIG. 6.78 EFFECT ON THRUST RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION (REVISED)
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
REVISED IGV AND DESIGN BLEED
DESIGN IGV AND REVISED BLEED
REVISED IGV AND REVISED BLEED
THRUST (lbf)
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
0
1
2
3
4
5
6
7
8
9
10
11
12
TIME (s)

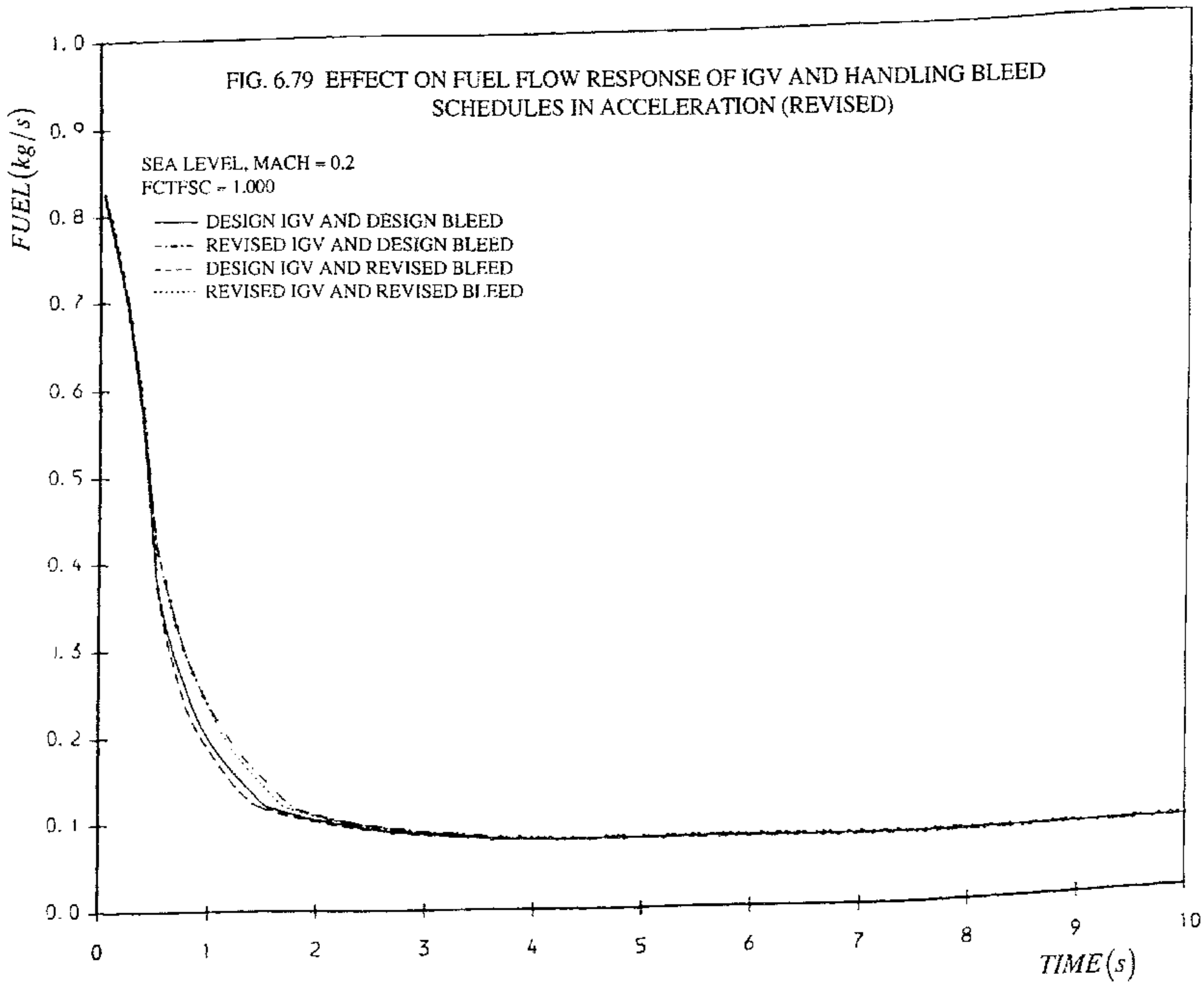


FIG. 6.79 EFFECT ON FUEL FLOW RESPONSE OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION (REVISED)
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
REVISED IGV AND DESIGN BLEED
DESIGN IGV AND REVISED BLEED
REVISED IGV AND REVISED BLEED
FUEL(kg/s)
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
0
1
2
3
4
5
6
7
8
9
10
TIME(s)

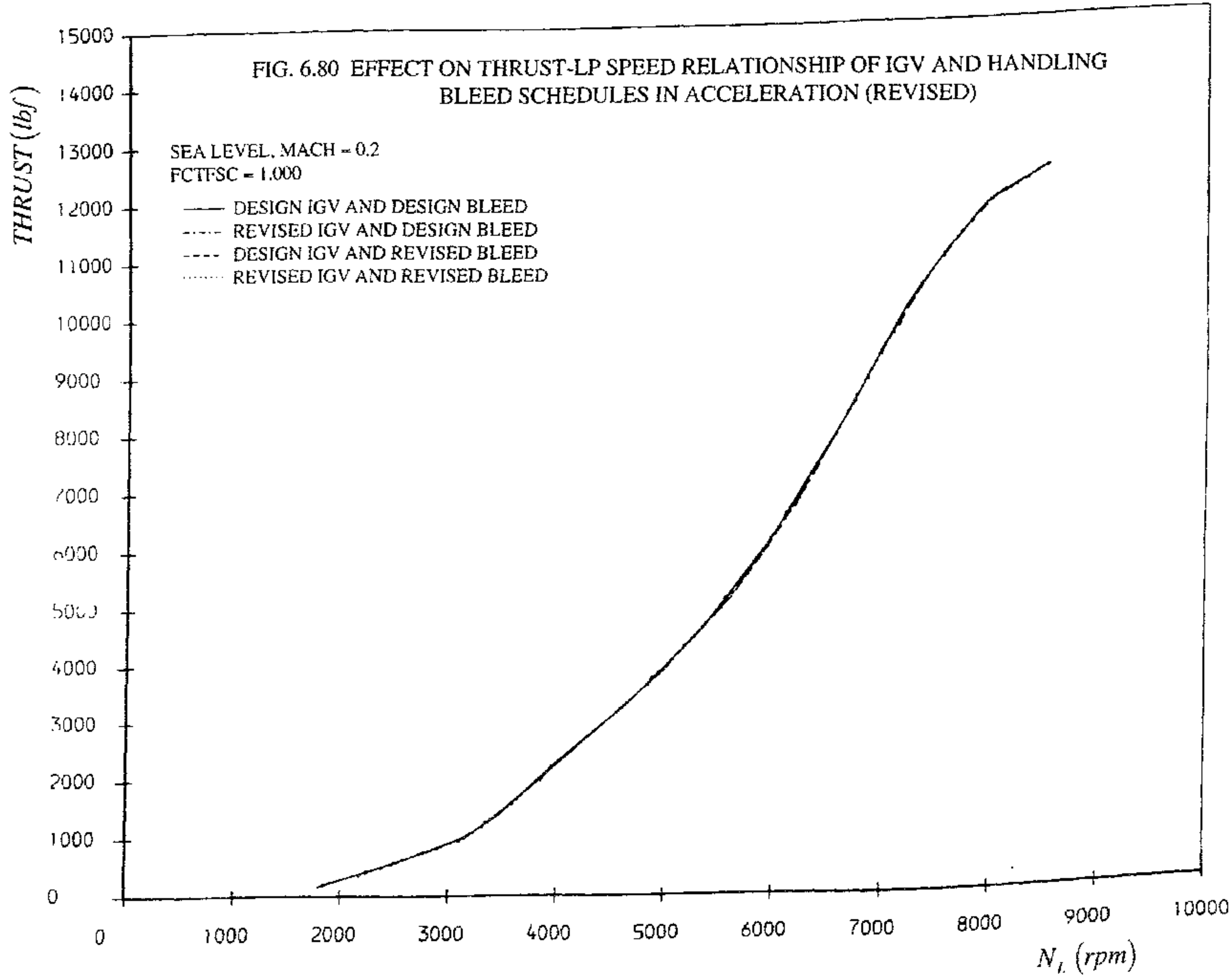


FIG. 6.80 EFFECT ON THRUST-LP SPEED RELATIONSHIP OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION (REVISED)
SEA LEVEL. MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
REVISED IGV AND DESIGN BLEED
DESIGN IGV AND REVISED BLEED
REVISED IGV AND REVISED BLEED
THRUST (lbf)
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
11000
12000
13000
14000
15000
N_L (rpm)

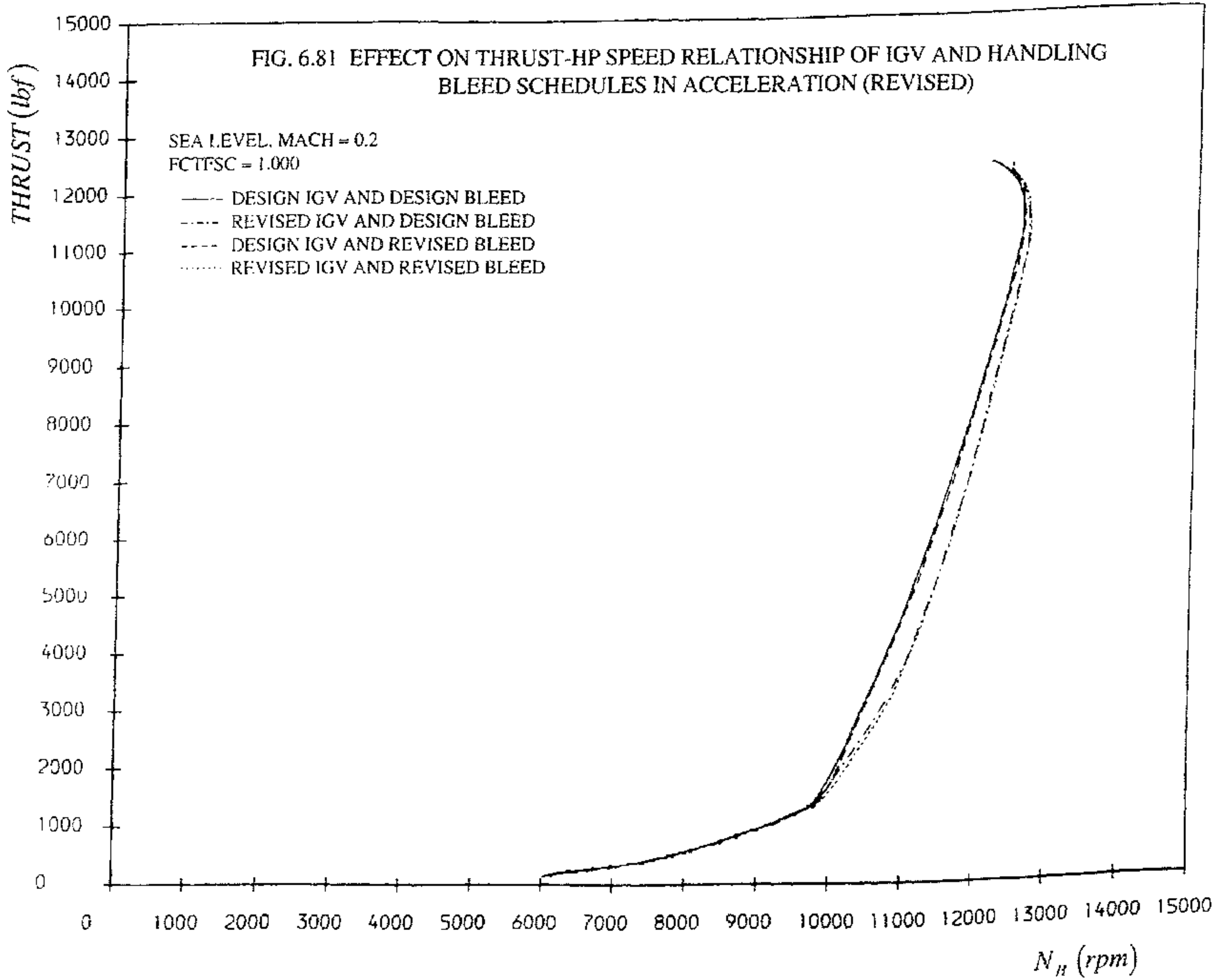


FIG. 6.81 EFFECT ON THRUST-HP SPEED RELATIONSHIP OF IGV AND HANDLING BLEED SCHEDULES IN ACCELERATION (REVISED)

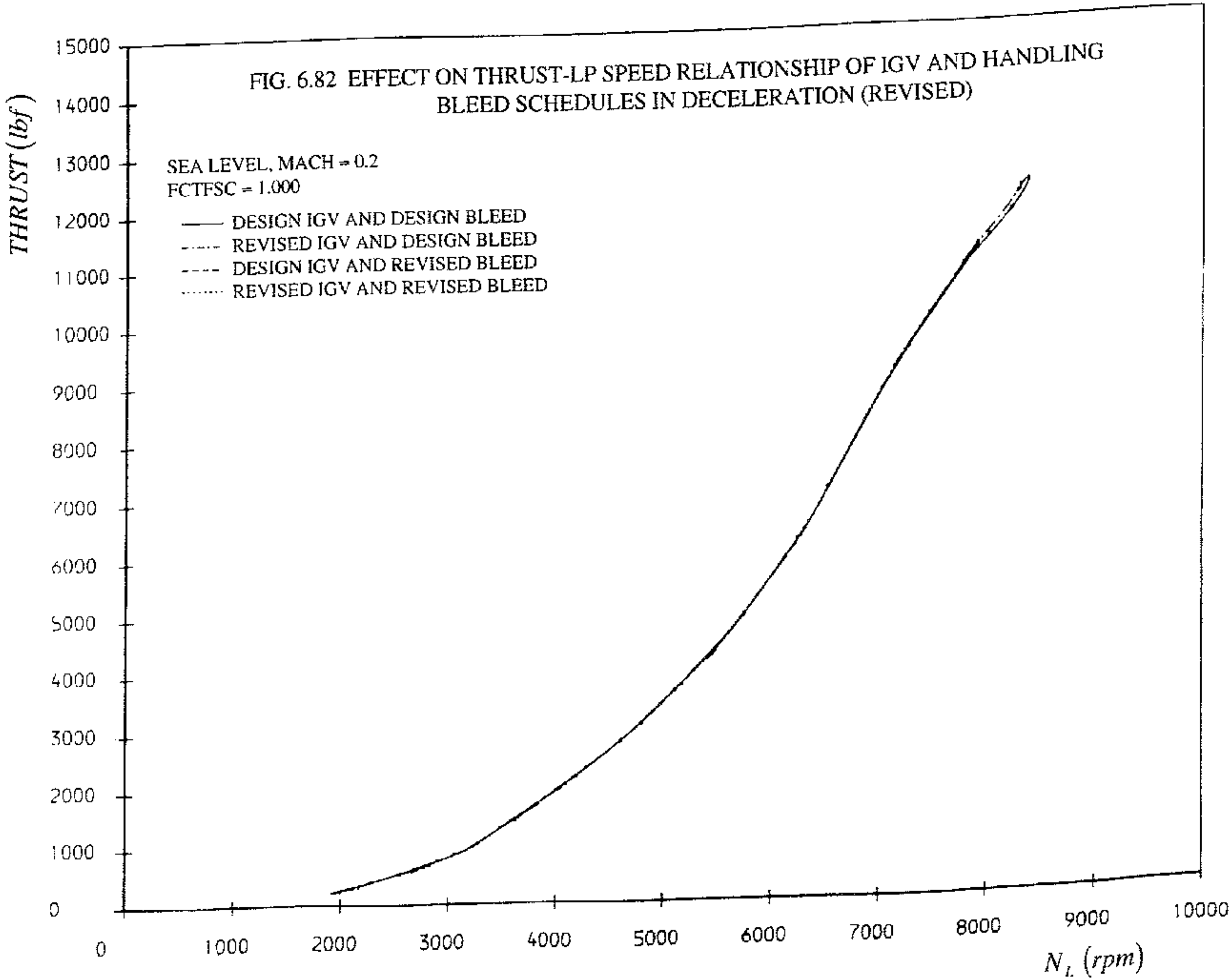


FIG. 6.82 EFFECT ON THRUST-LP SPEED RELATIONSHIP OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION (REVISED)
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN IGV AND DESIGN BLEED
REVISED IGV AND DESIGN BLEED
DESIGN IGV AND REVISED BLEED
REVISED IGV AND REVISED BLEED
THRUST (lbf)
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
N_L (rpm)

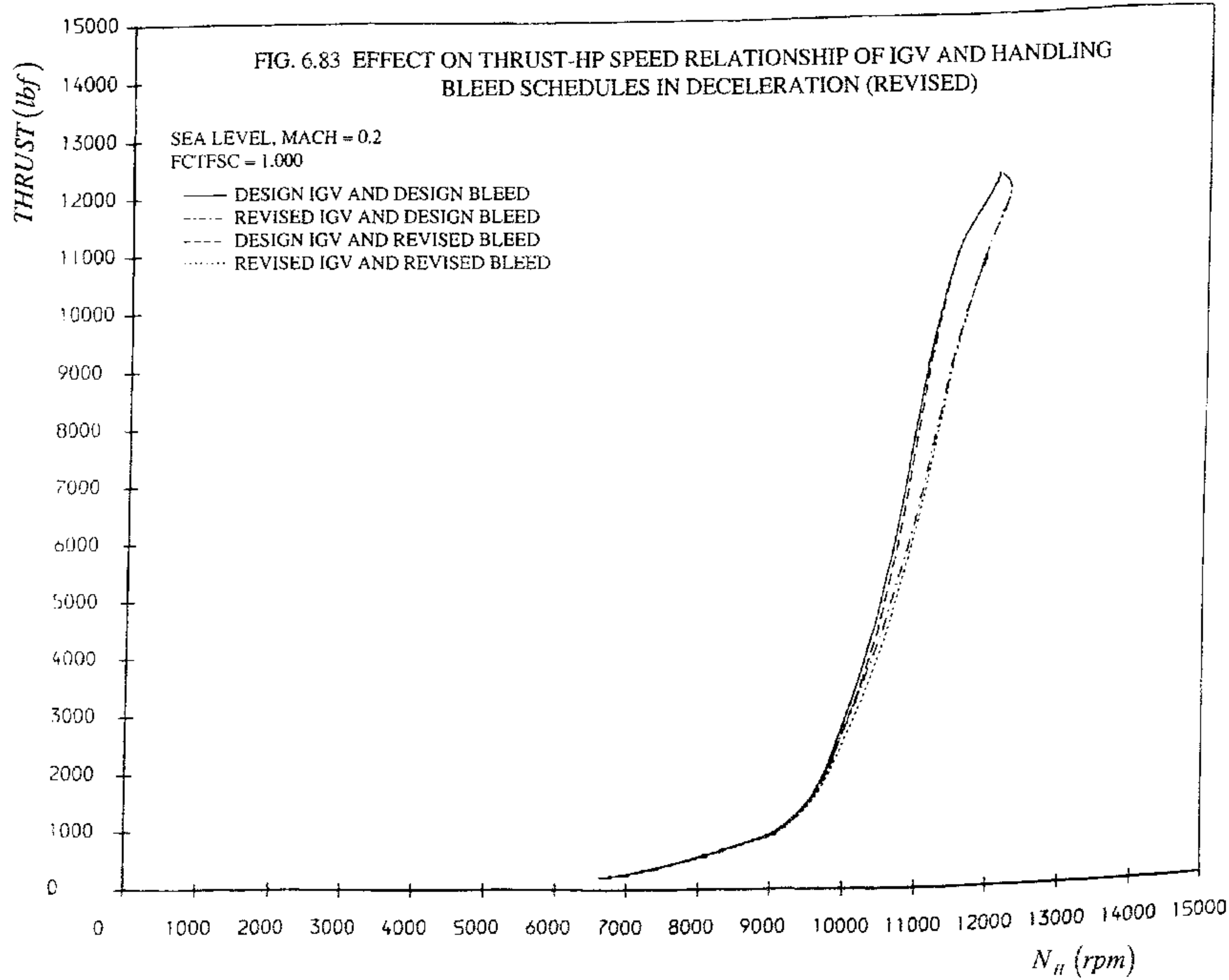


FIG. 6.83 EFFECT ON THRUST-HP SPEED RELATIONSHIP OF IGV AND HANDLING BLEED SCHEDULES IN DECELERATION (REVISED)

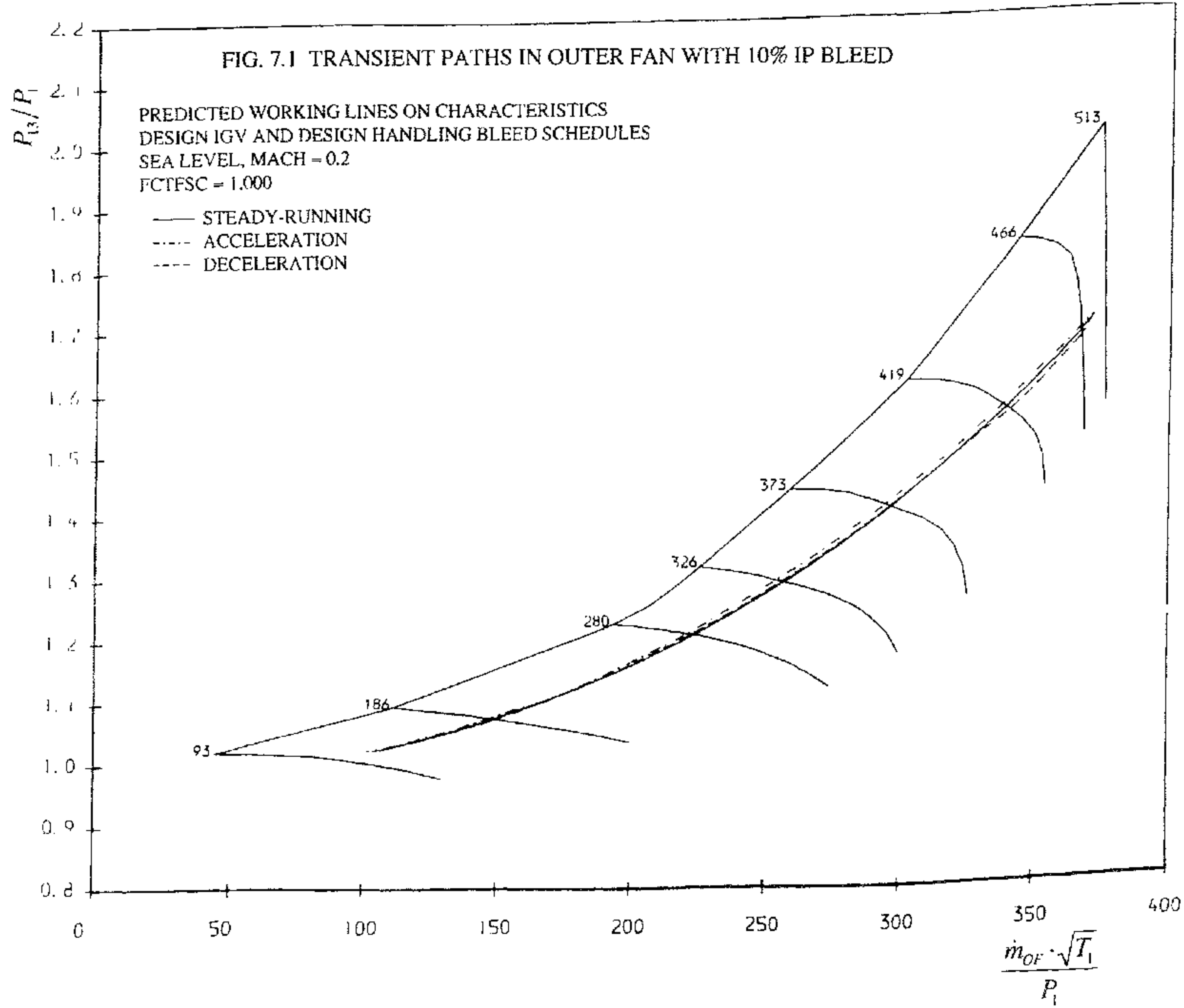
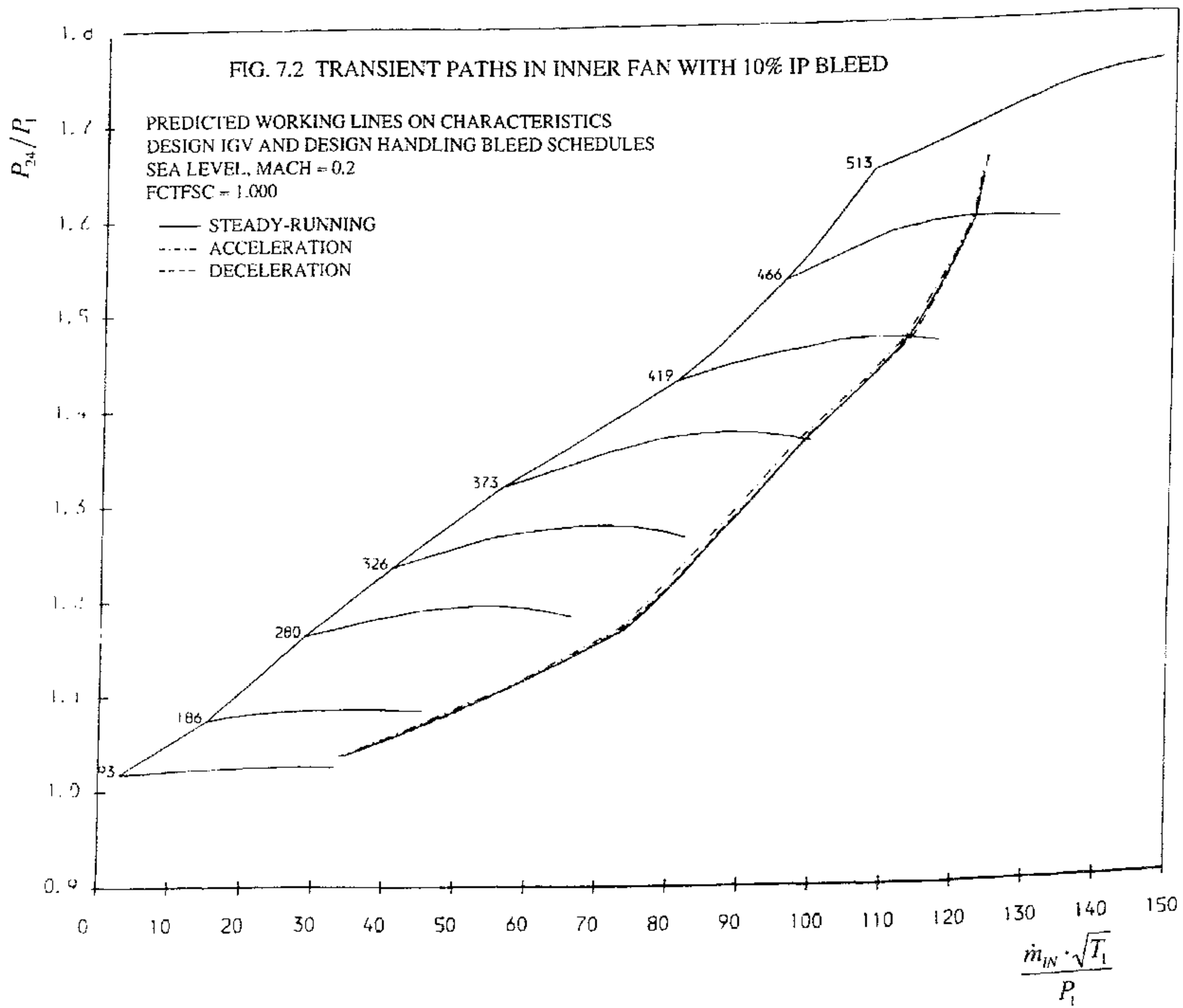


FIG. 7.1 TRANSIENT PATHS IN OUTER FAN WITH 10% IP BLEED

FIG. 7.2 TRANSIENT PATHS IN INNER FAN WITH 10% IP BLEED



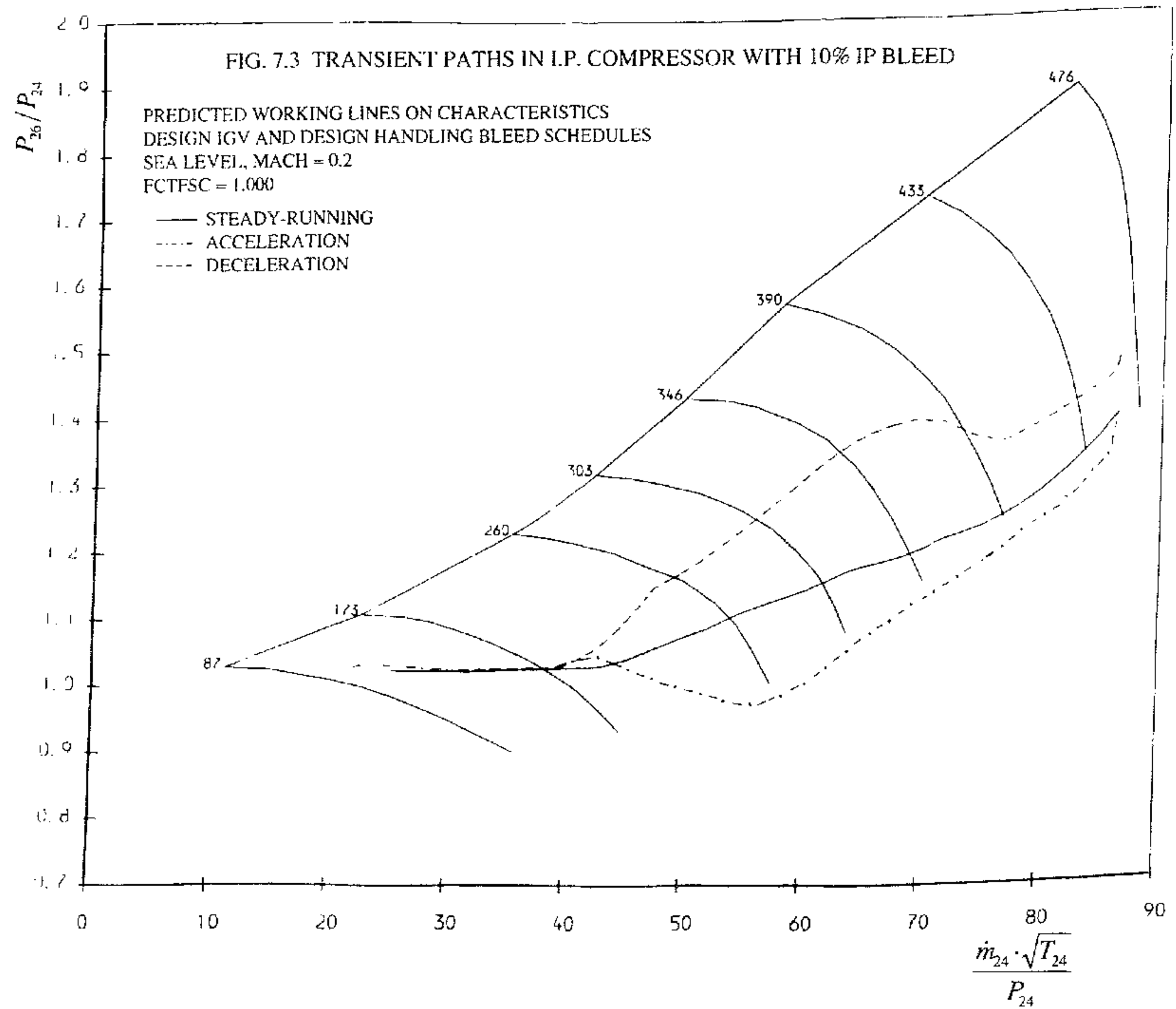


FIG. 7.3 TRANSIENT PATHS IN I.P. COMPRESSOR WITH 10% IP BLEED

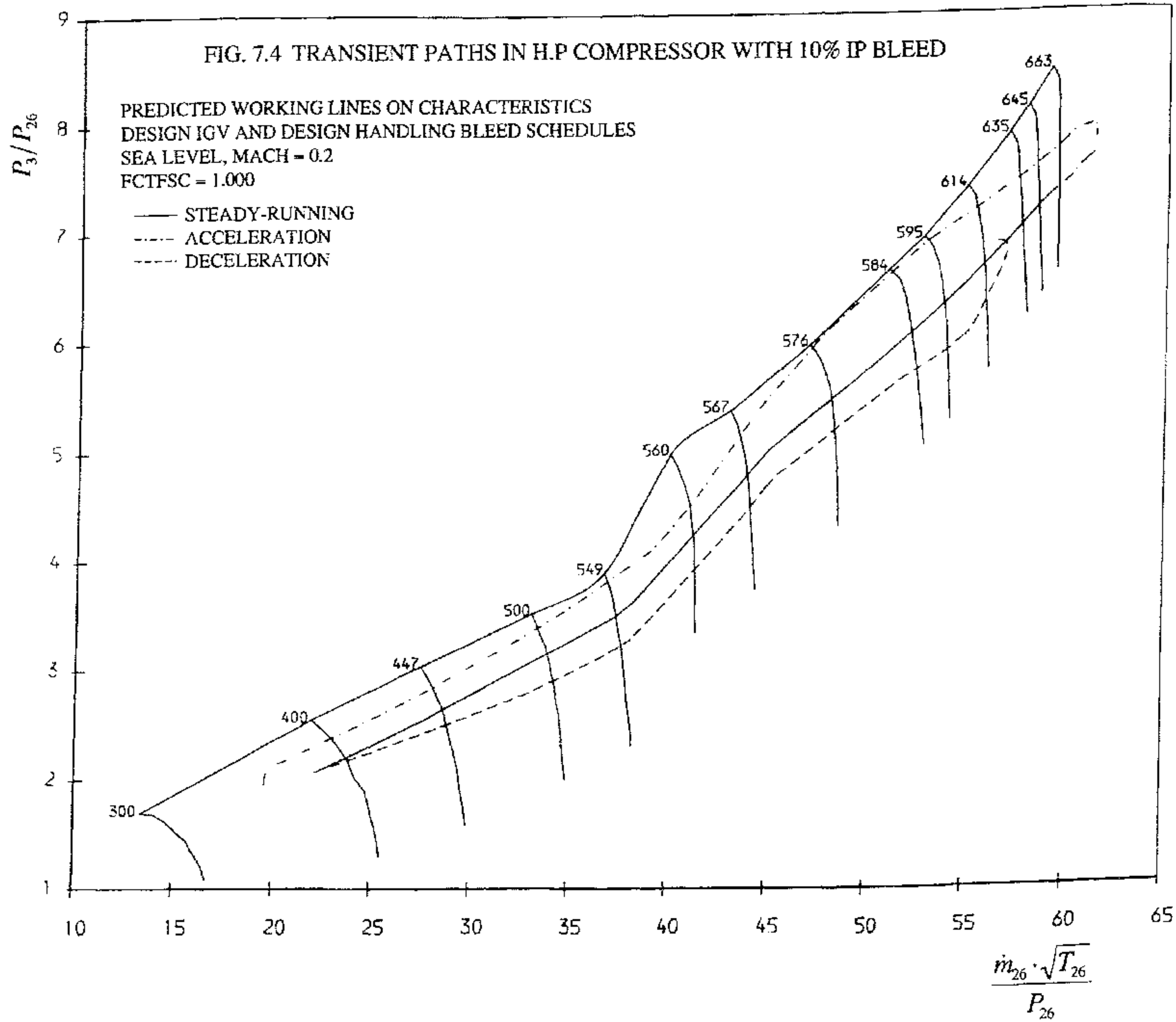


FIG. 7.4 TRANSIENT PATHS IN H.P COMPRESSOR WITH 10% IP BLEED
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_3/P_{26}
$\frac{\dot{m}_{26} \cdot \sqrt{T_{26}}}{P_{26}}$
300
400
447
500
549
560
567
576
584
595
614
635
645
663
1
2
3
4
5
6
7
8
9
10
15
20
25
30
35
40
45
50
55
60
65

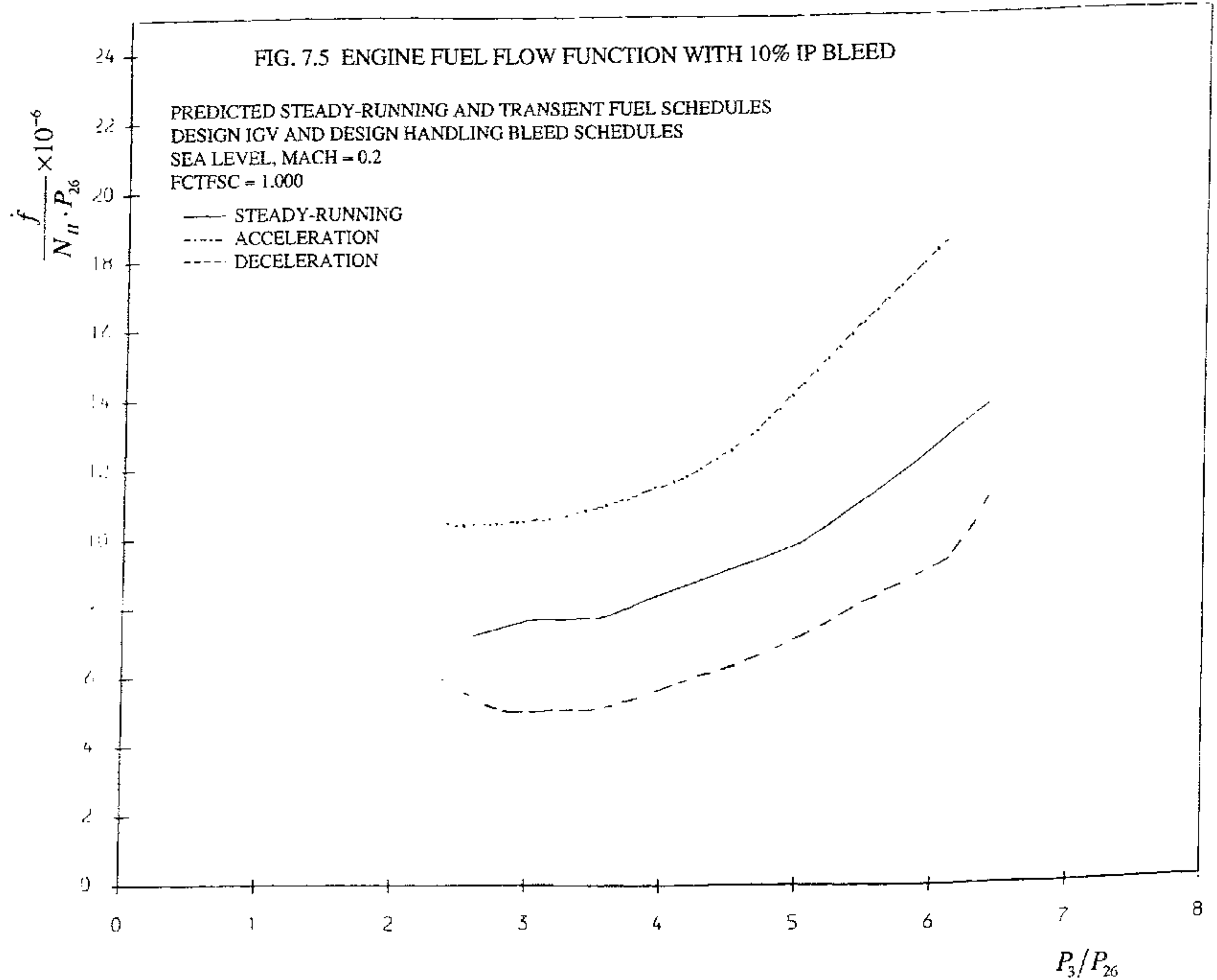


FIG. 7.5 ENGINE FUEL FLOW FUNCTION WITH 10% IP BLEED
PREDICTED STEADY-RUNNING AND TRANSIENT FUEL SCHEDULES
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
$\frac{\dot{f}}{N_{H} \cdot P_{26}} \times 10^{-6}$
24
22
20
18
10
0
0
1
2
3
4
5
6
7
8
P_3/P_{26}

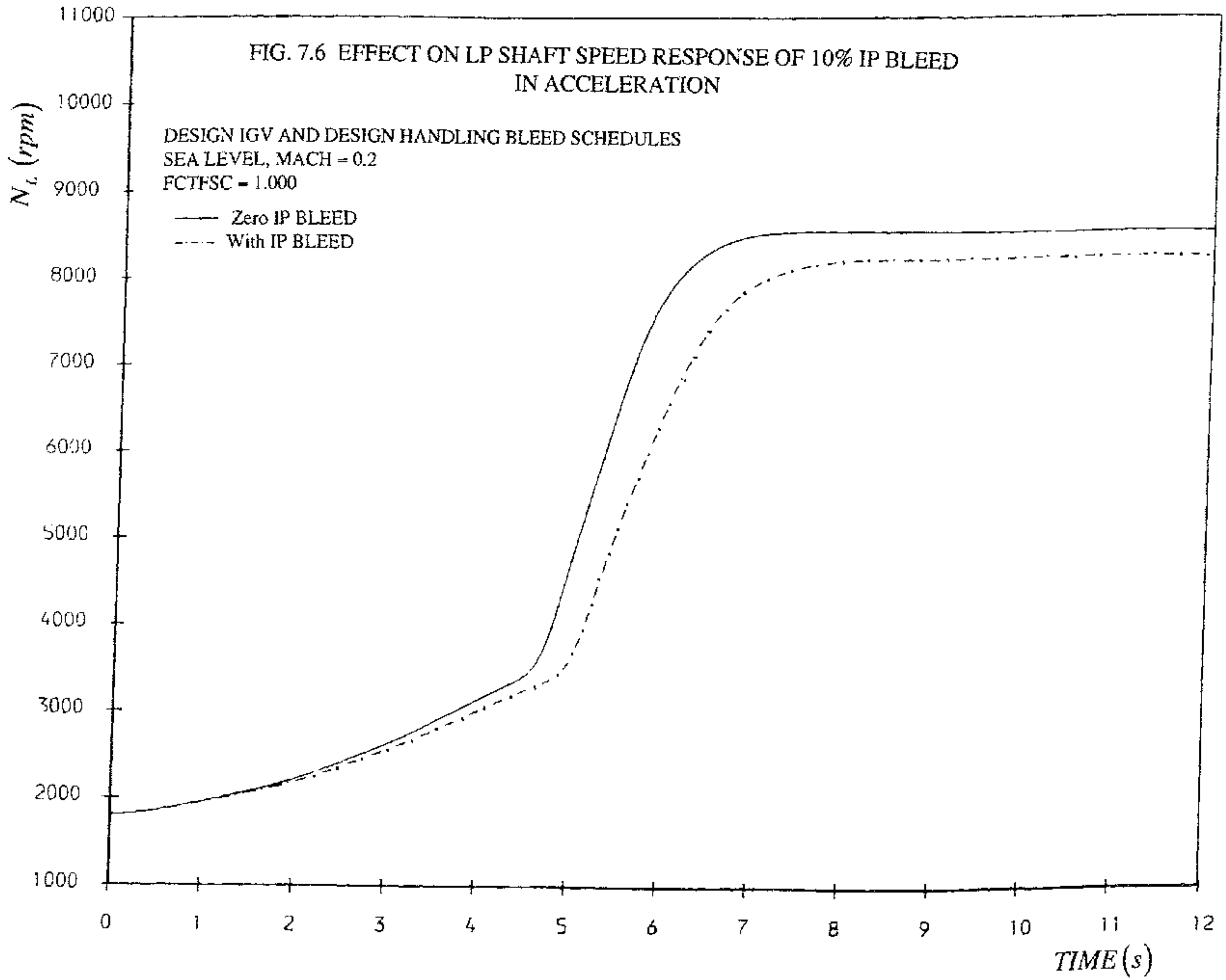


FIG. 7.6 EFFECT ON LP SHAFT SPEED RESPONSE OF 10% IP BLEED IN ACCELERATION

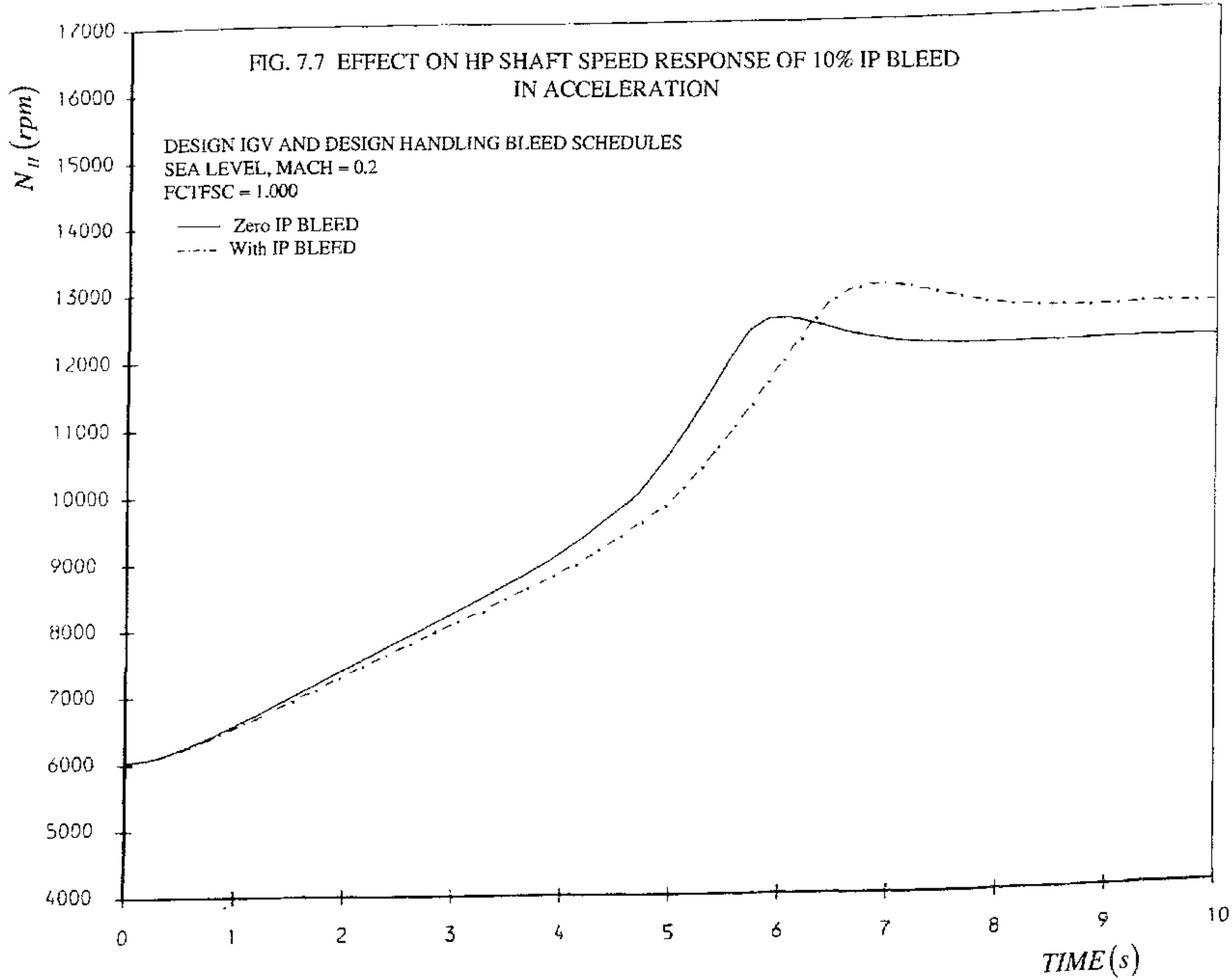


FIG. 7.7 EFFECT ON HP SHAFT SPEED RESPONSE OF 10% IP BLEED
IN ACCELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
Zero IP BLEED
With IP BLEED
$N_H\ (rpm)$
17000
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
0
1
2
3
4
5
6
7
8
9
10
$TIME\ (s)$

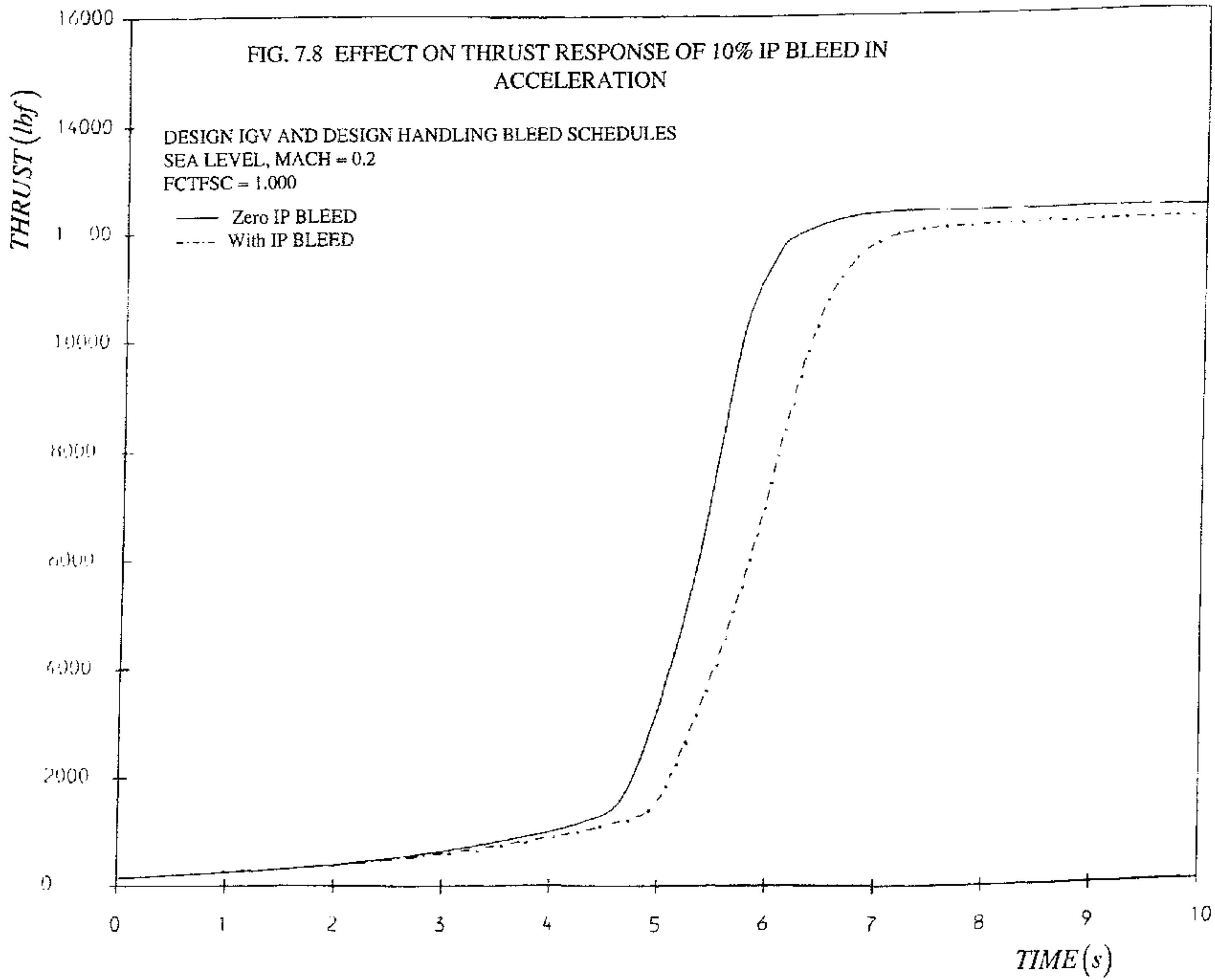


FIG. 7.8 EFFECT ON THRUST RESPONSE OF 10% IP BLEED IN ACCELERATION

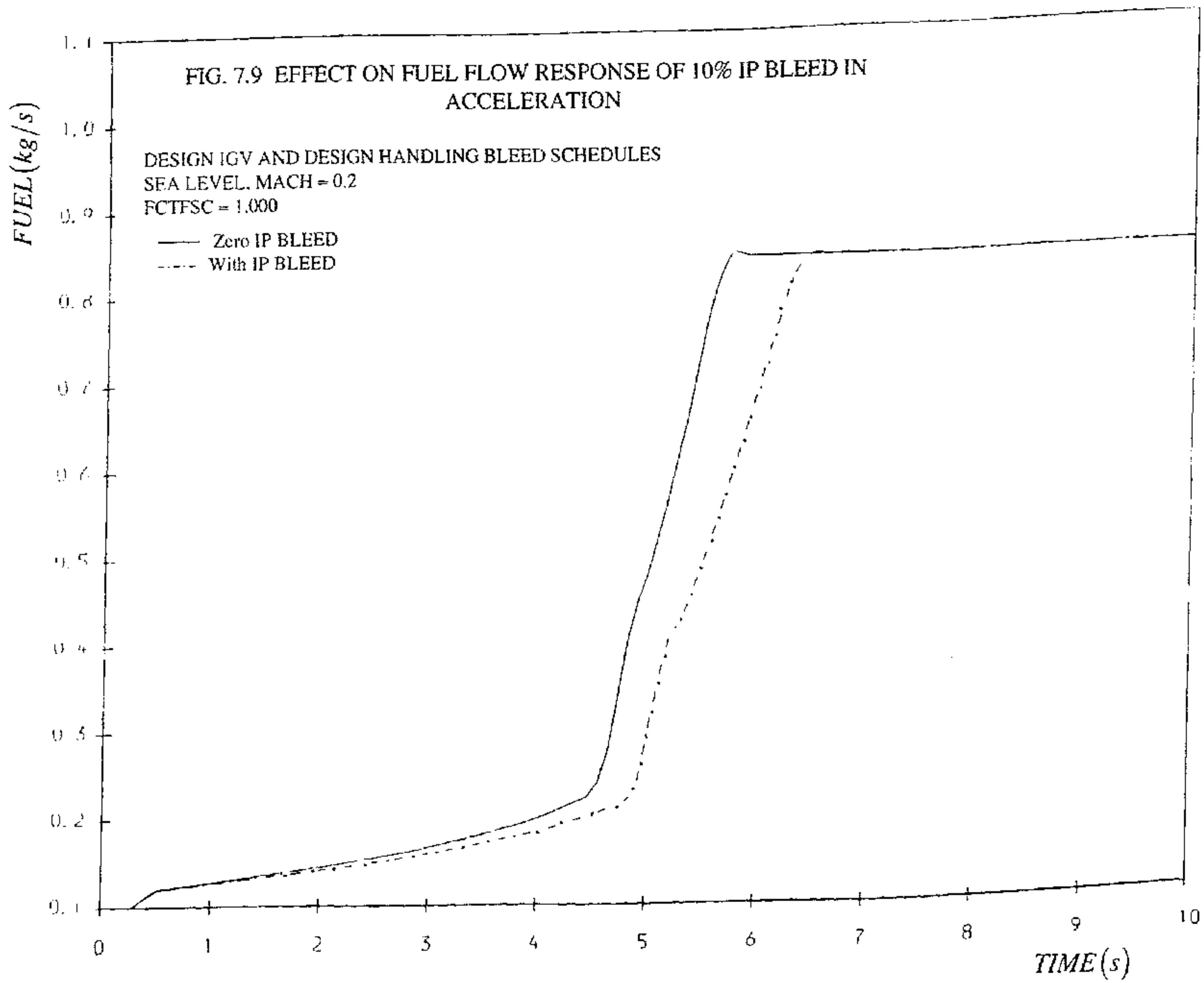


FIG. 7.9 EFFECT ON FUEL FLOW RESPONSE OF 10% IP BLEED IN ACCELERATION

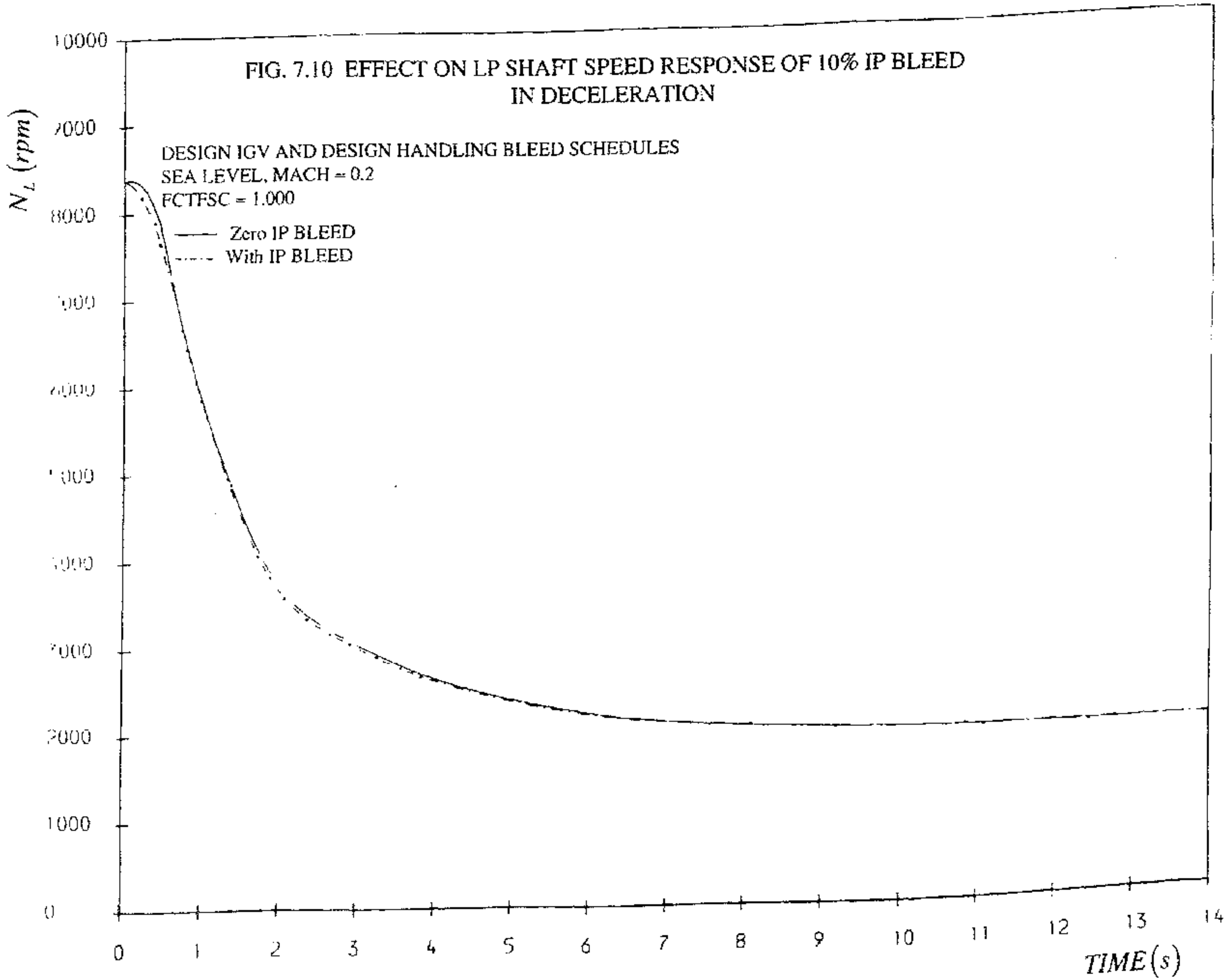


FIG. 7.10 EFFECT ON LP SHAFT SPEED RESPONSE OF 10% IP BLEED IN DECELERATION

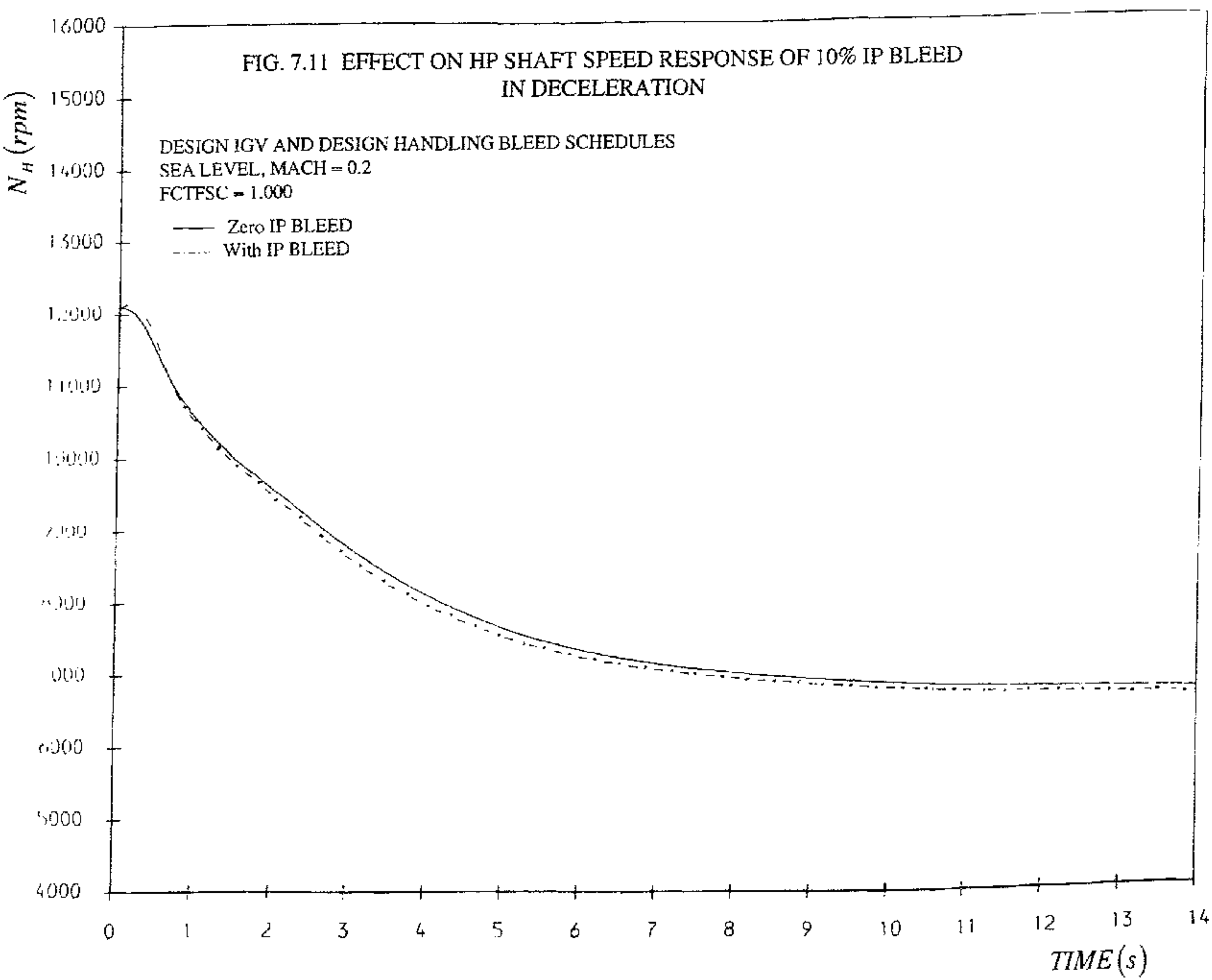


FIG. 7.11 EFFECT ON HP SHAFT SPEED RESPONSE OF 10% IP BLEED IN DECELERATION

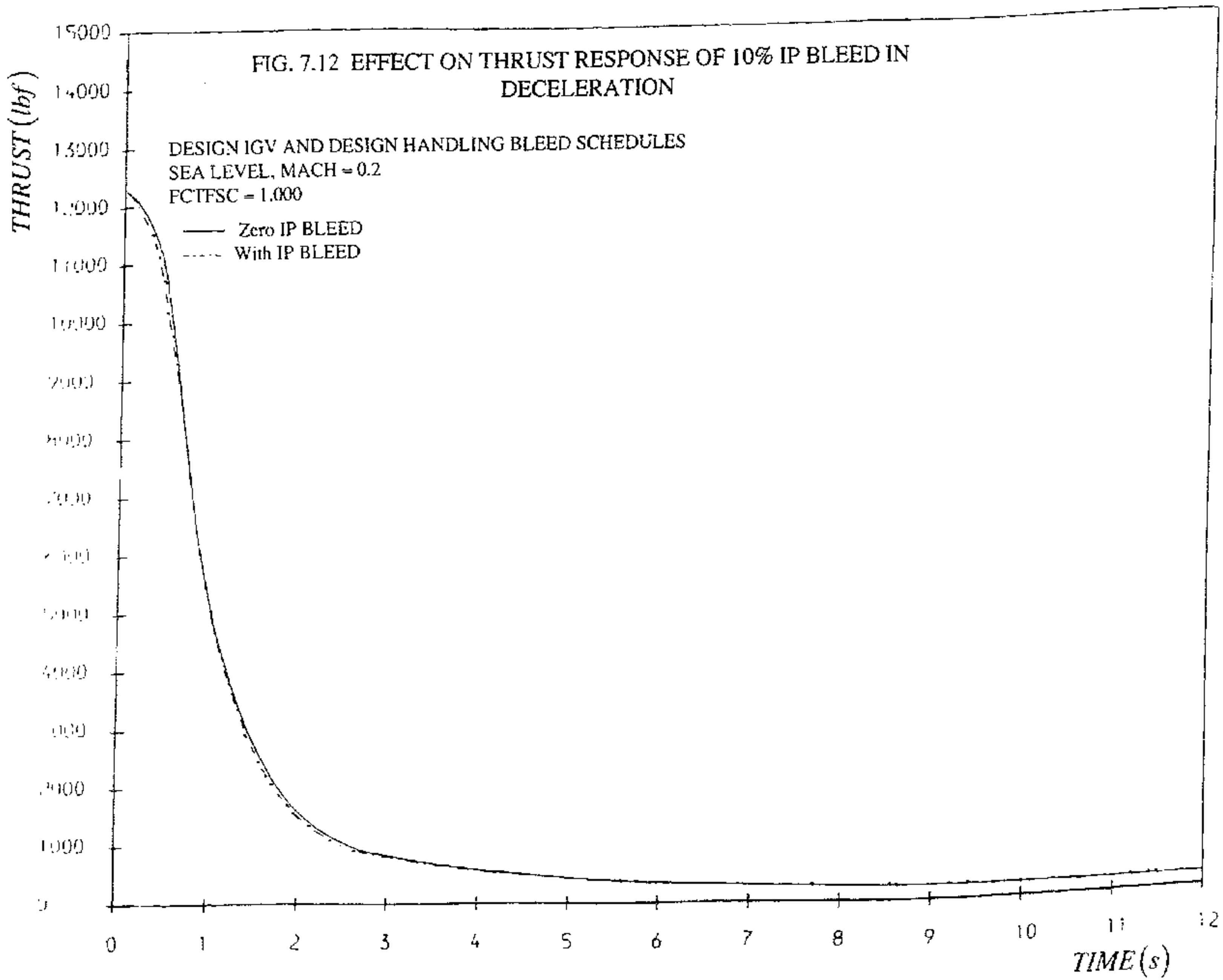


FIG. 7.12 EFFECT ON THRUST RESPONSE OF 10% IP BLEED IN DECELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
Zero IP BLEED
With IP BLEED
THRUST (lbf)
TIME (s)
15000
14000
13000
1000
0
1
2
3
4
5
6
7
8
9
10
11
12

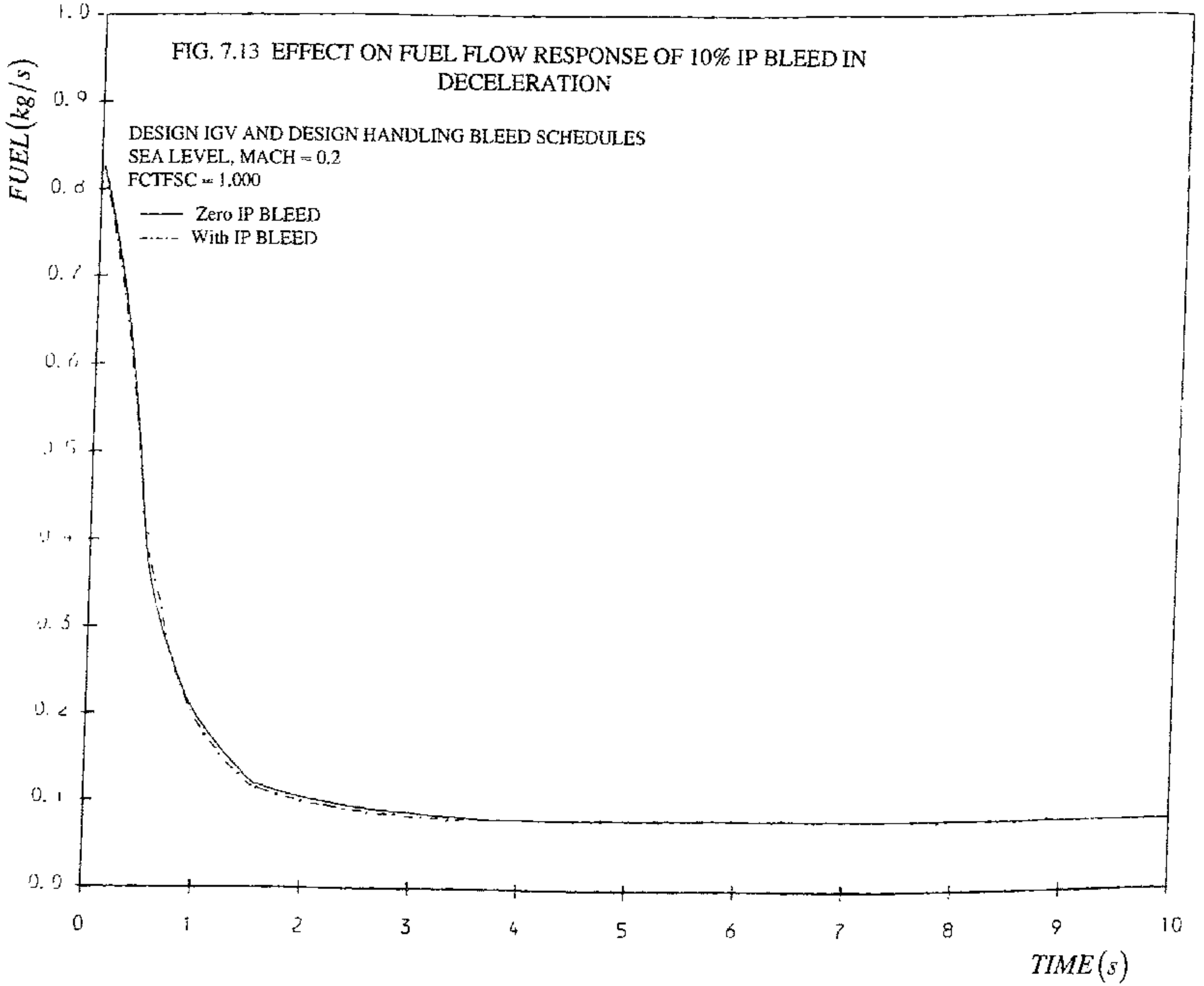


FIG. 7.13 EFFECT ON FUEL FLOW RESPONSE OF 10% IP BLEED IN DECELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
Zero IP BLEED
With IP BLEED
FUEL(kg/s)
TIME(s)
1.0
0.9
0.8
0.7
0.6
0.2
0.1
0.0
0
1
2
3
4
5
6
7
8
9
10

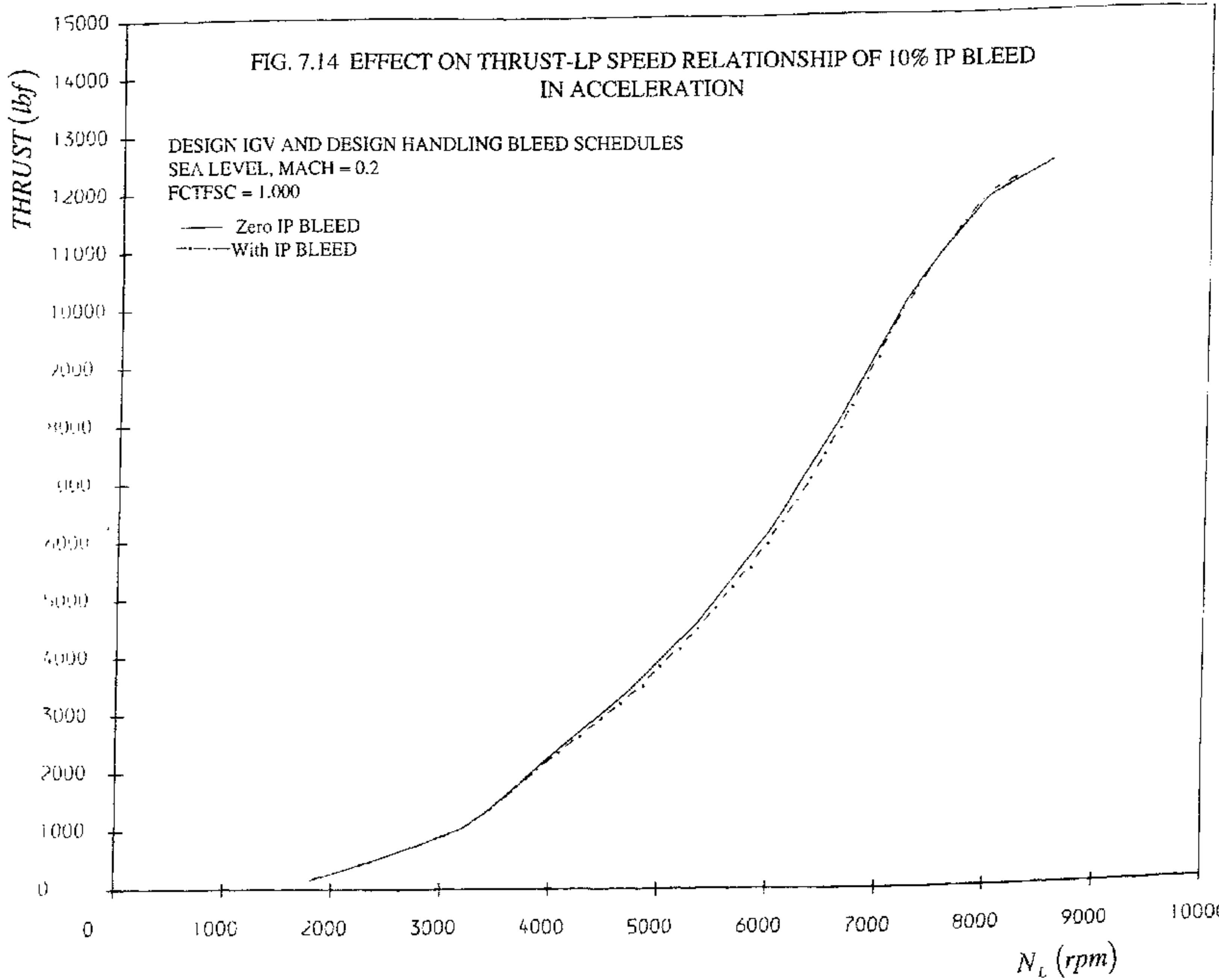


FIG. 7.14 EFFECT ON THRUST-LP SPEED RELATIONSHIP OF 10% IP BLEED IN ACCELERATION

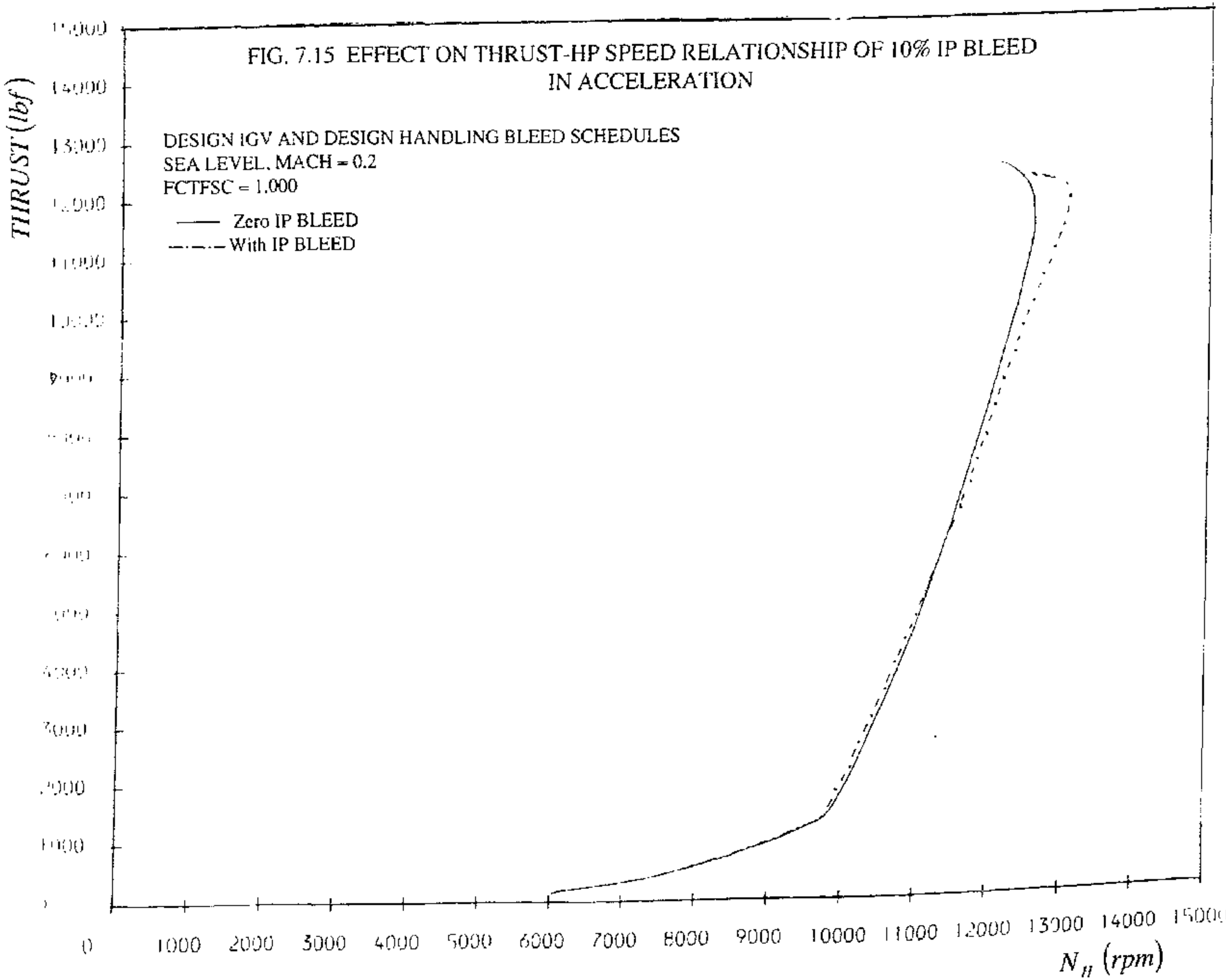


FIG. 7.15 EFFECT ON THRUST-HP SPEED RELATIONSHIP OF 10% IP BLEED IN ACCELERATION

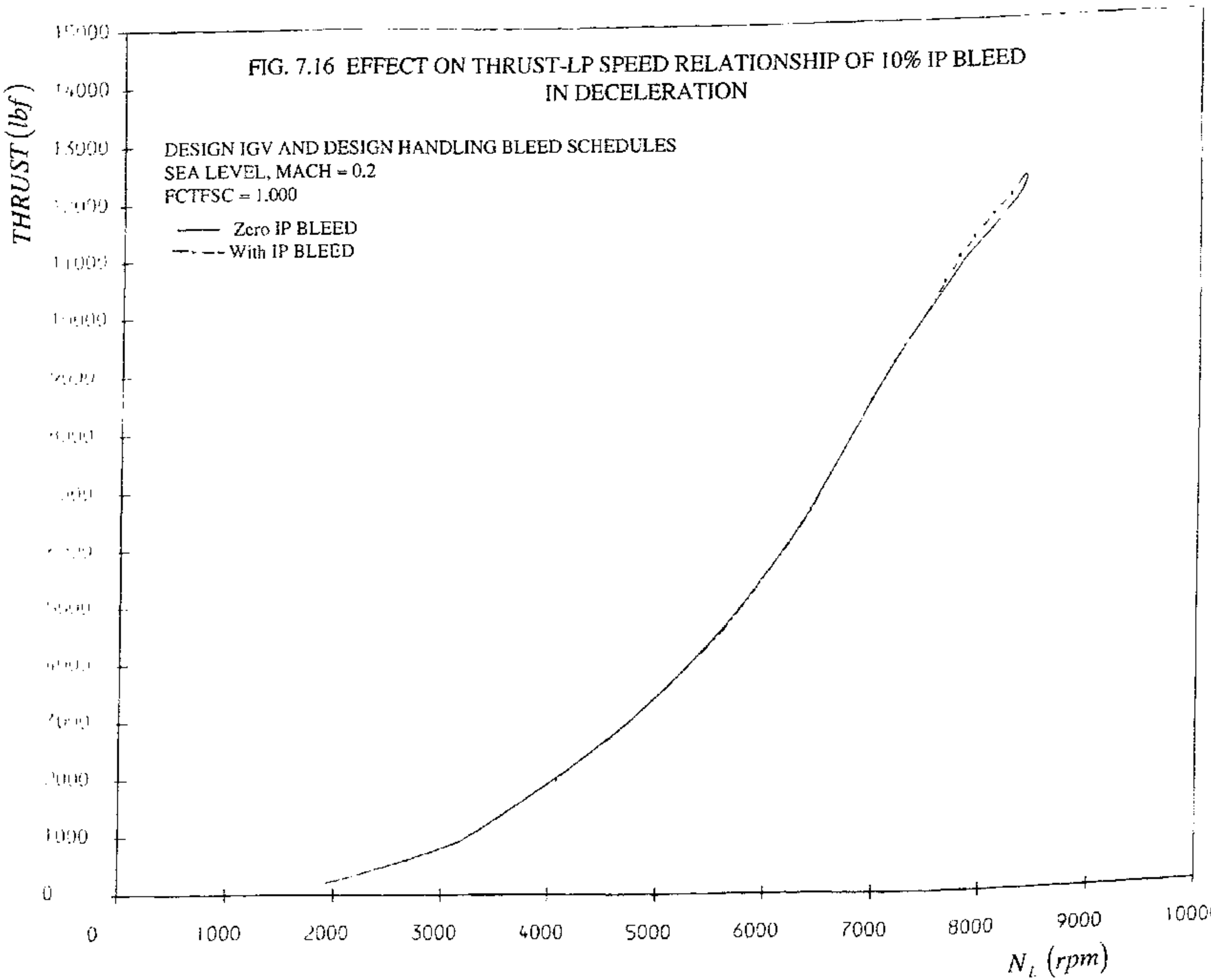


FIG. 7.16 EFFECT ON THRUST-LP SPEED RELATIONSHIP OF 10% IP BLEED
IN DECELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
Zero IP BLEED
With IP BLEED
THRUST (lbf)
N_L (rpm)
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000

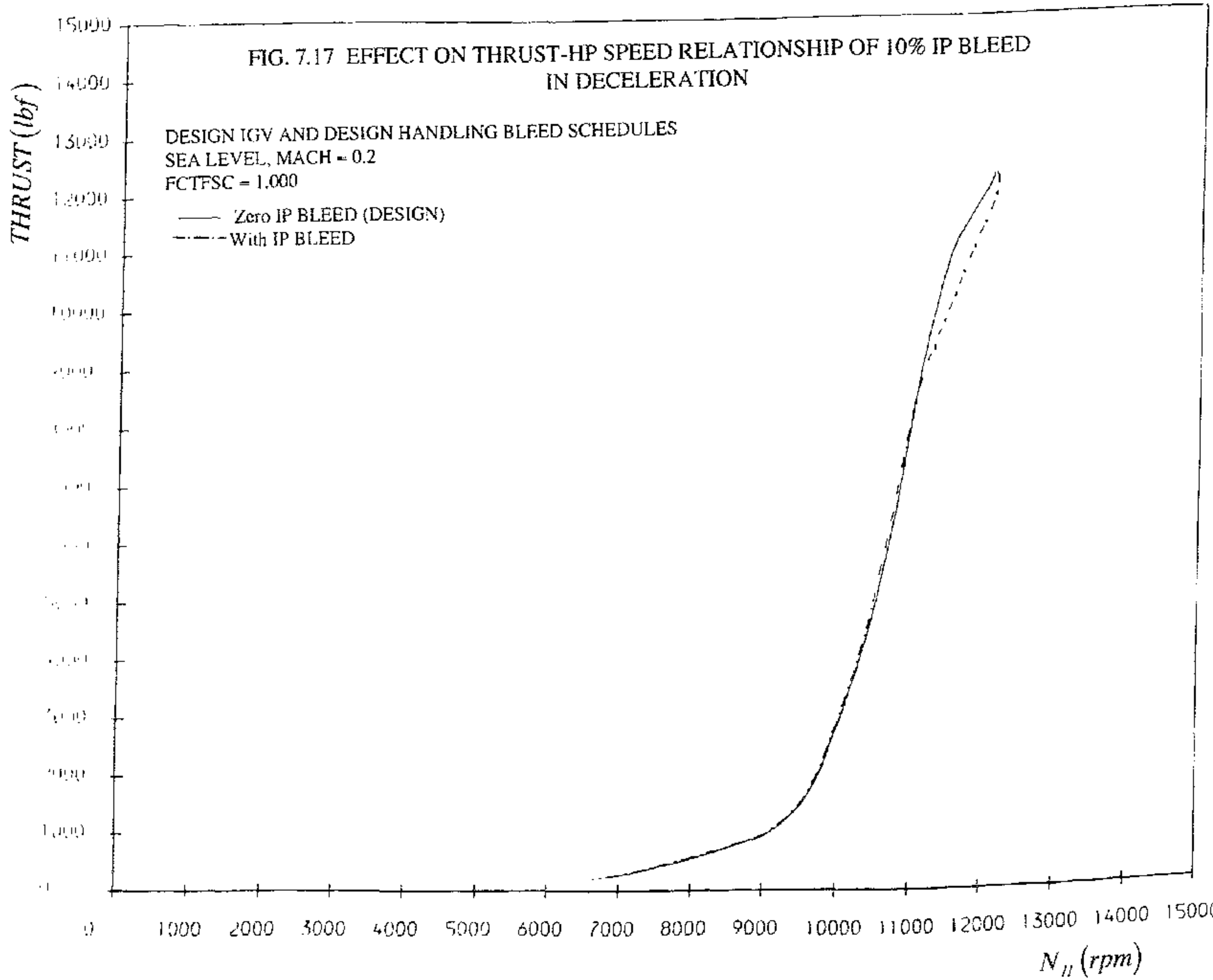


FIG. 7.17 EFFECT ON THRUST-HP SPEED RELATIONSHIP OF 10% IP BLEED IN DECELERATION

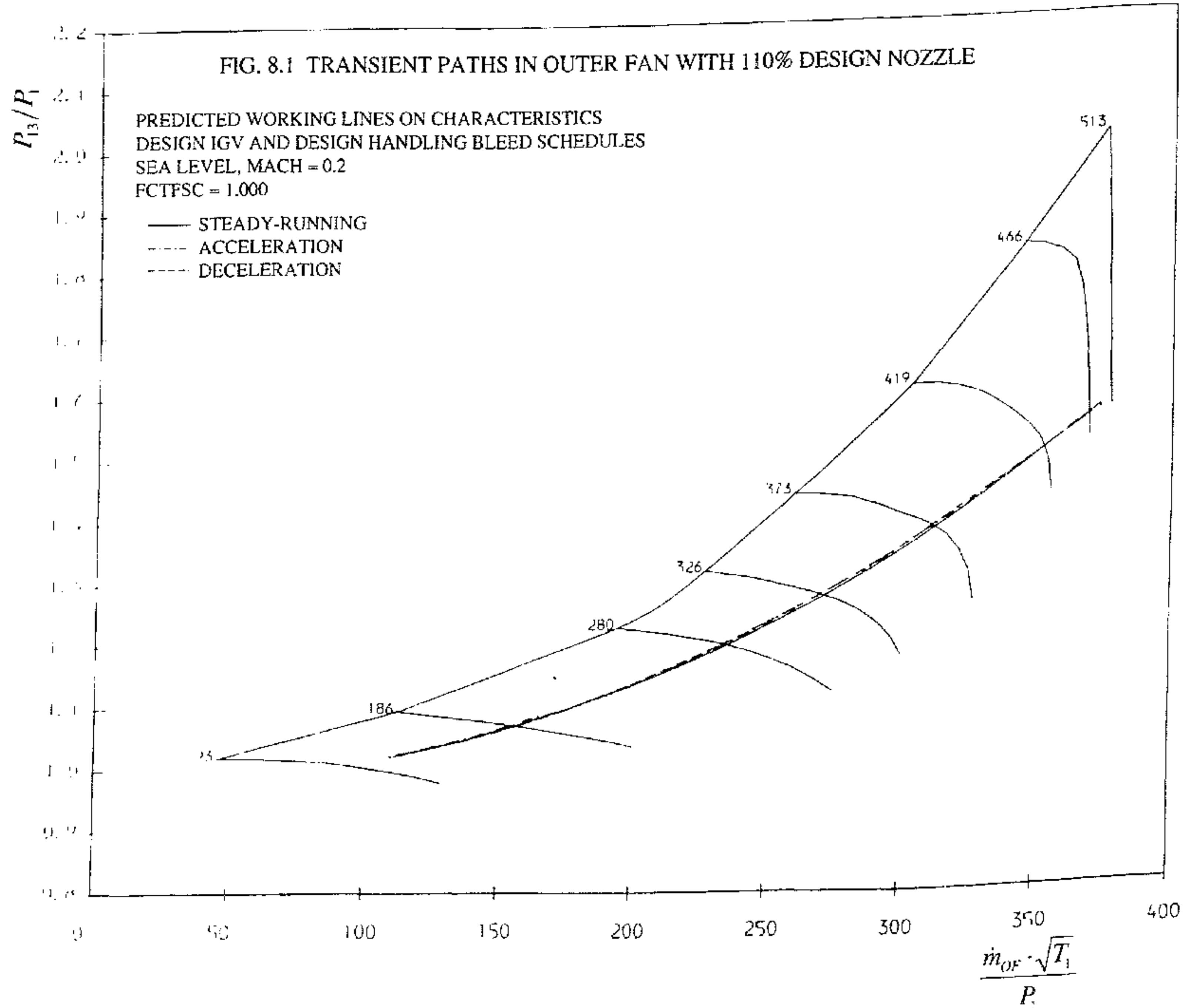


FIG. 8.1 TRANSIENT PATHS IN OUTER FAN WITH 110% DESIGN NOZZLE

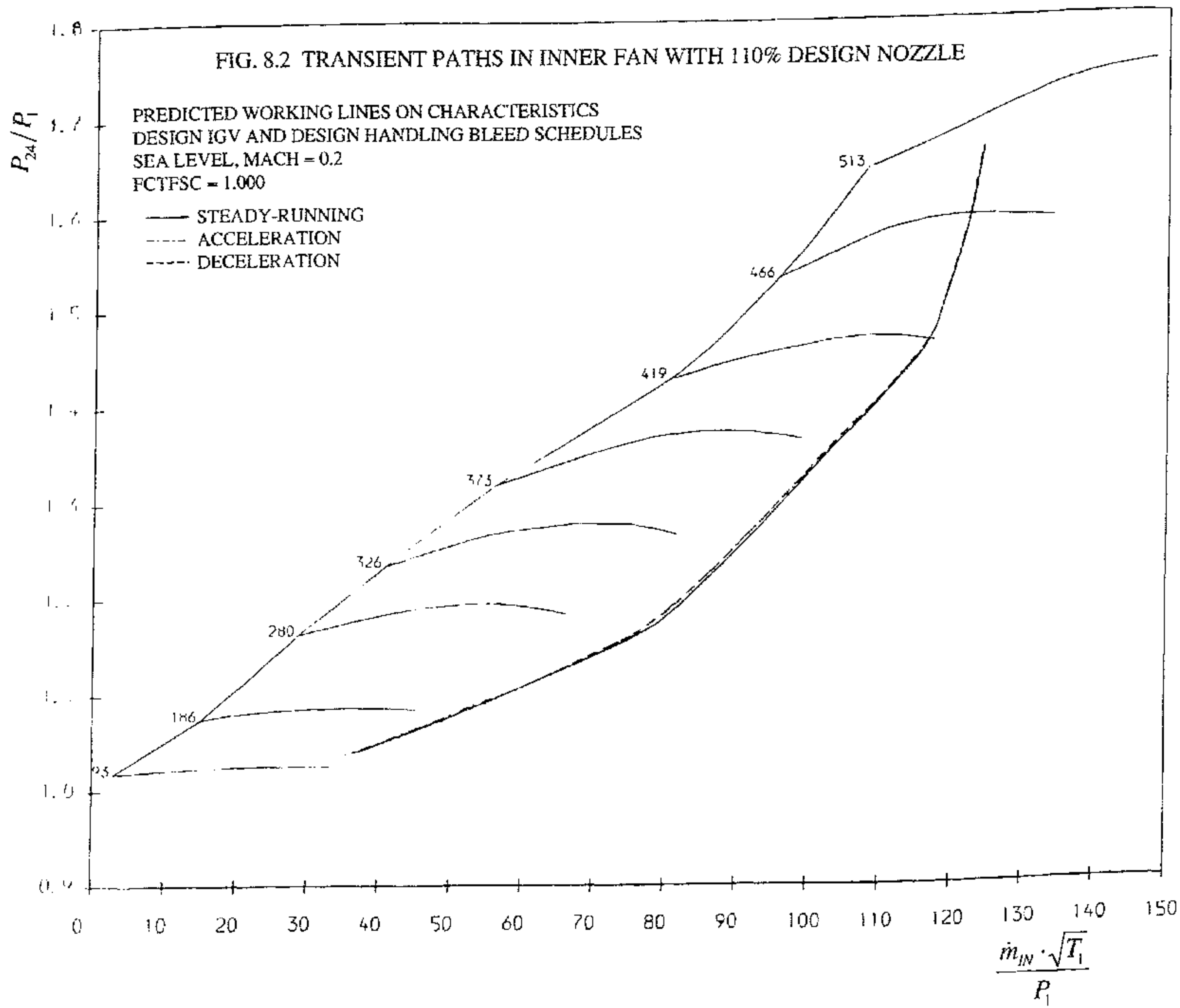


FIG. 8.2 TRANSIENT PATHS IN INNER FAN WITH 110% DESIGN NOZZLE
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
513
466
419
373
326
280
186
P_{24}/P_1
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
$\frac{\dot{m}_{IN} \cdot \sqrt{T_1}}{P_1}$

FIG. 8.3 TRANSIENT PATHS IN I.P. COMPRESSOR WITH 110% DESIGN NOZZLE

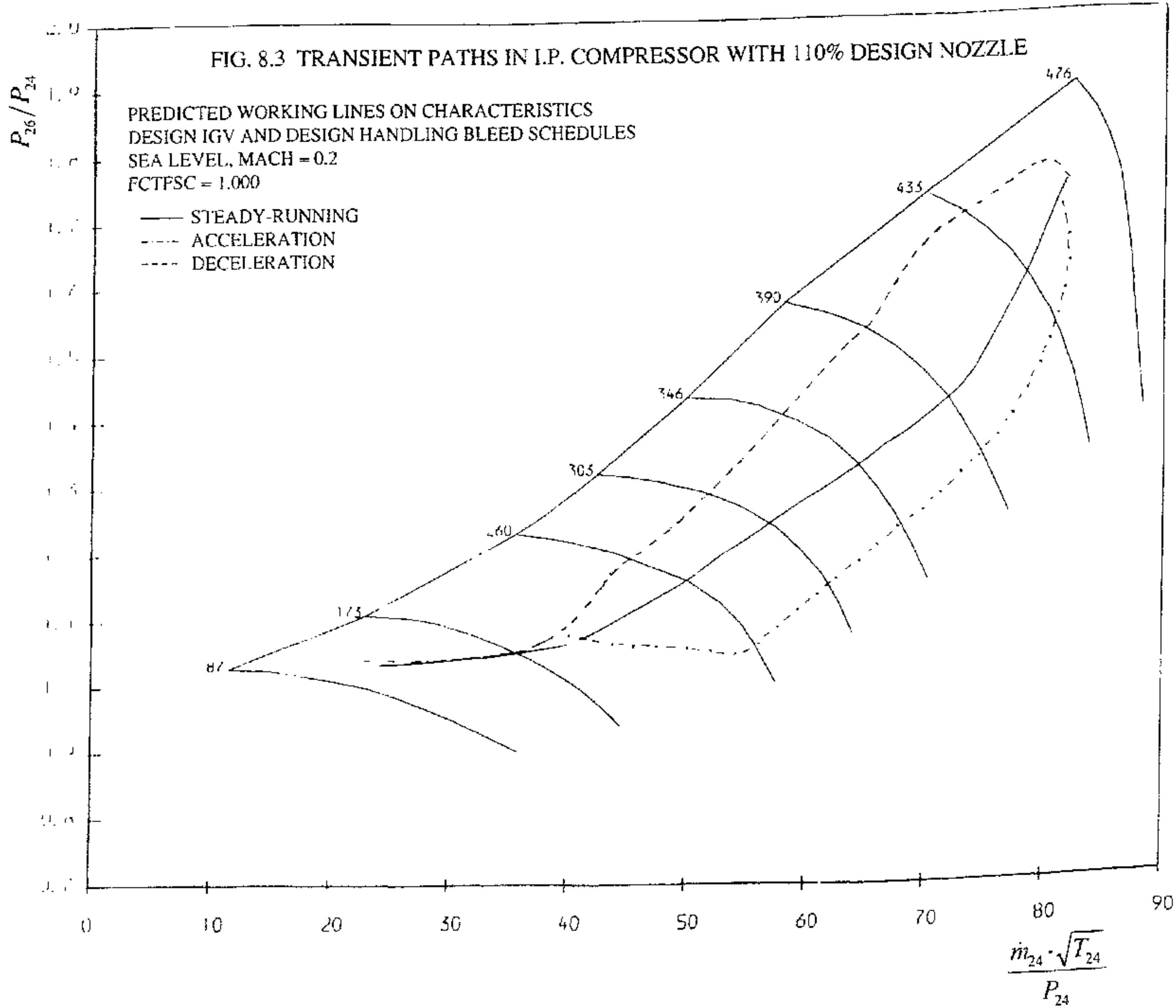
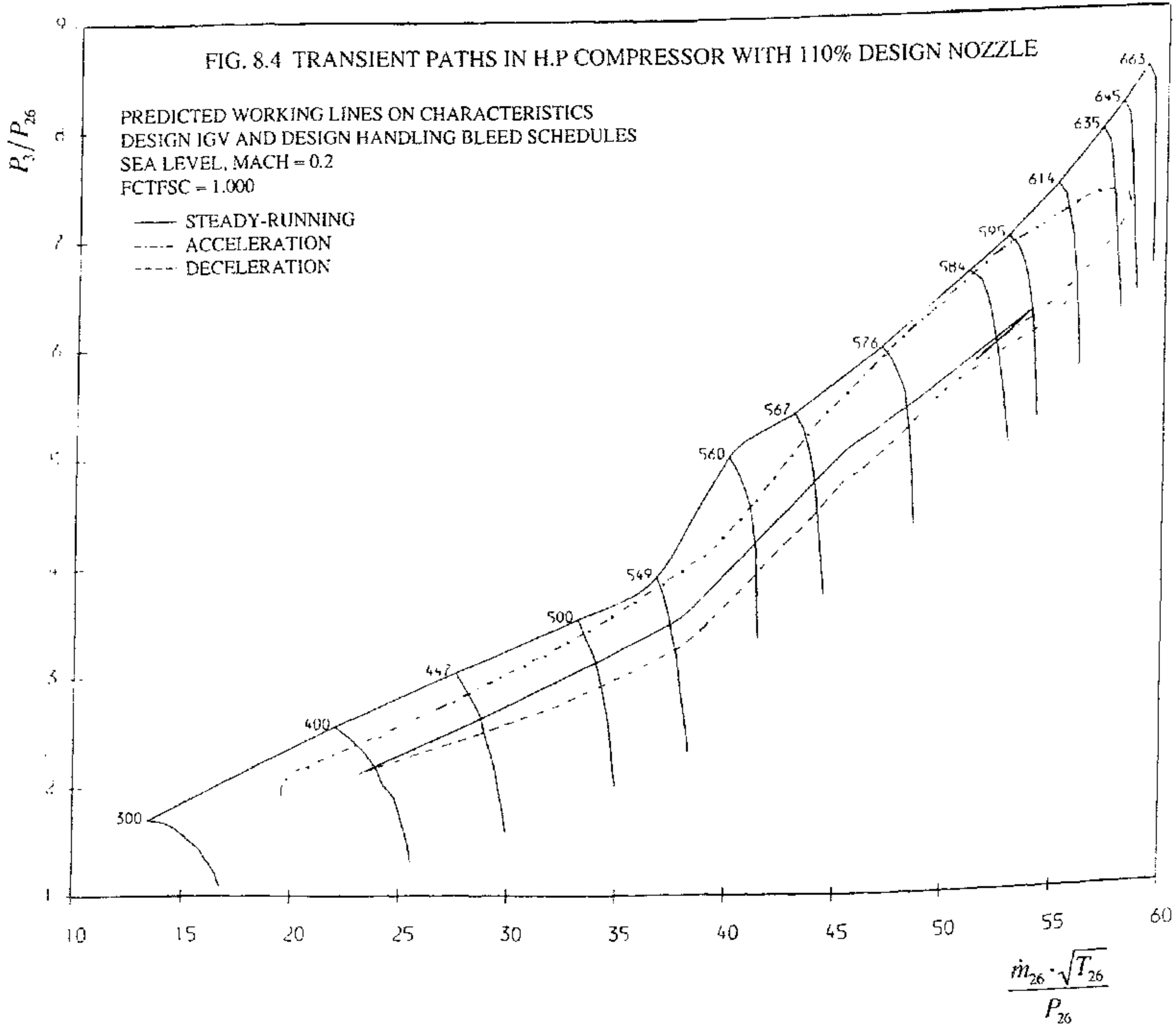


FIG. 8.4 TRANSIENT PATHS IN H.P COMPRESSOR WITH 110% DESIGN NOZZLE



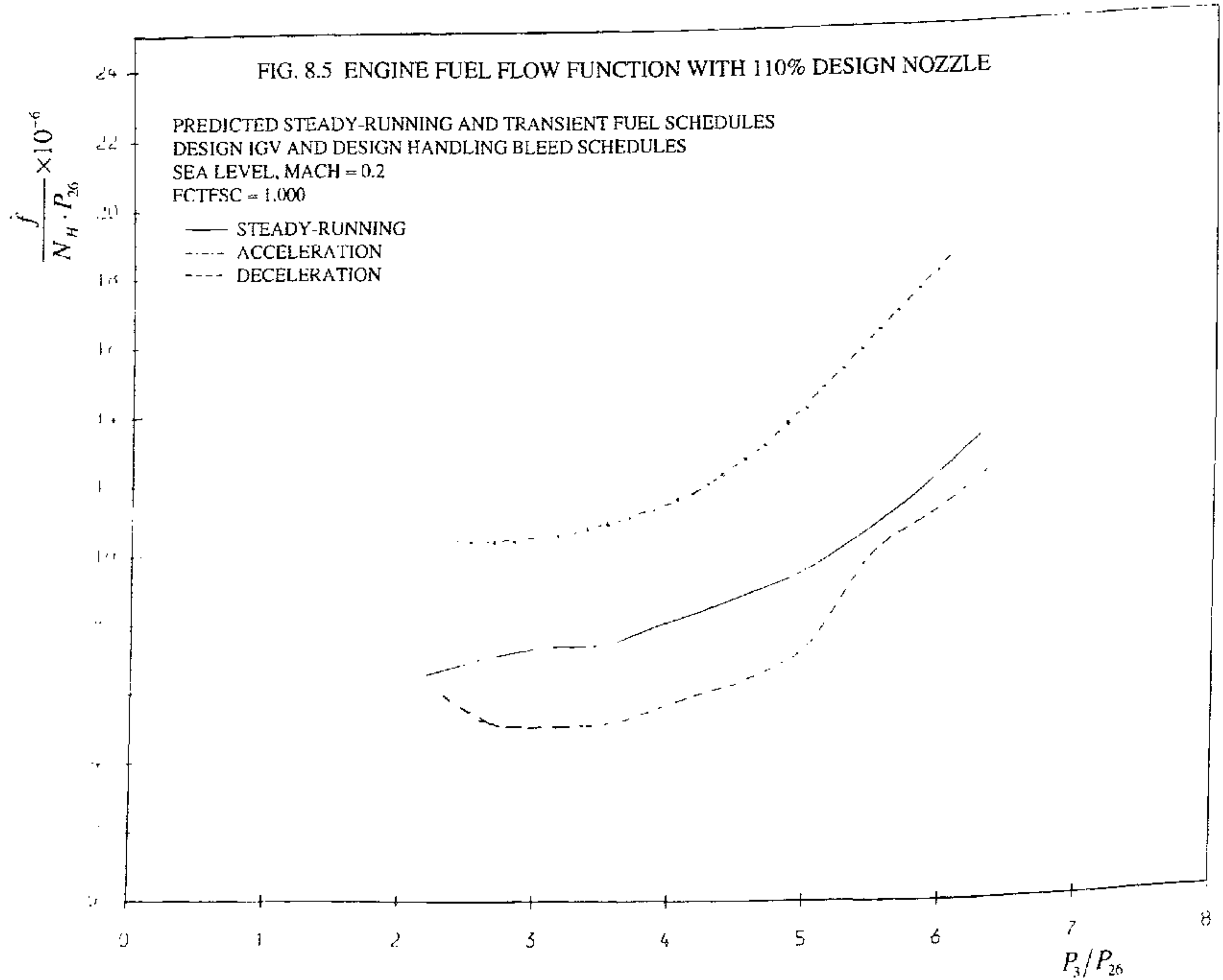


FIG. 8.5 ENGINE FUEL FLOW FUNCTION WITH 110% DESIGN NOZZLE
PREDICTED STEADY-RUNNING AND TRANSIENT FUEL SCHEDULES
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
$\frac{\dot{f}}{N_H \cdot P_{26}} \times 10^{-6}$
24
22
20
14
0
0
1
2
3
4
5
6
7
8
P_3/P_{26}

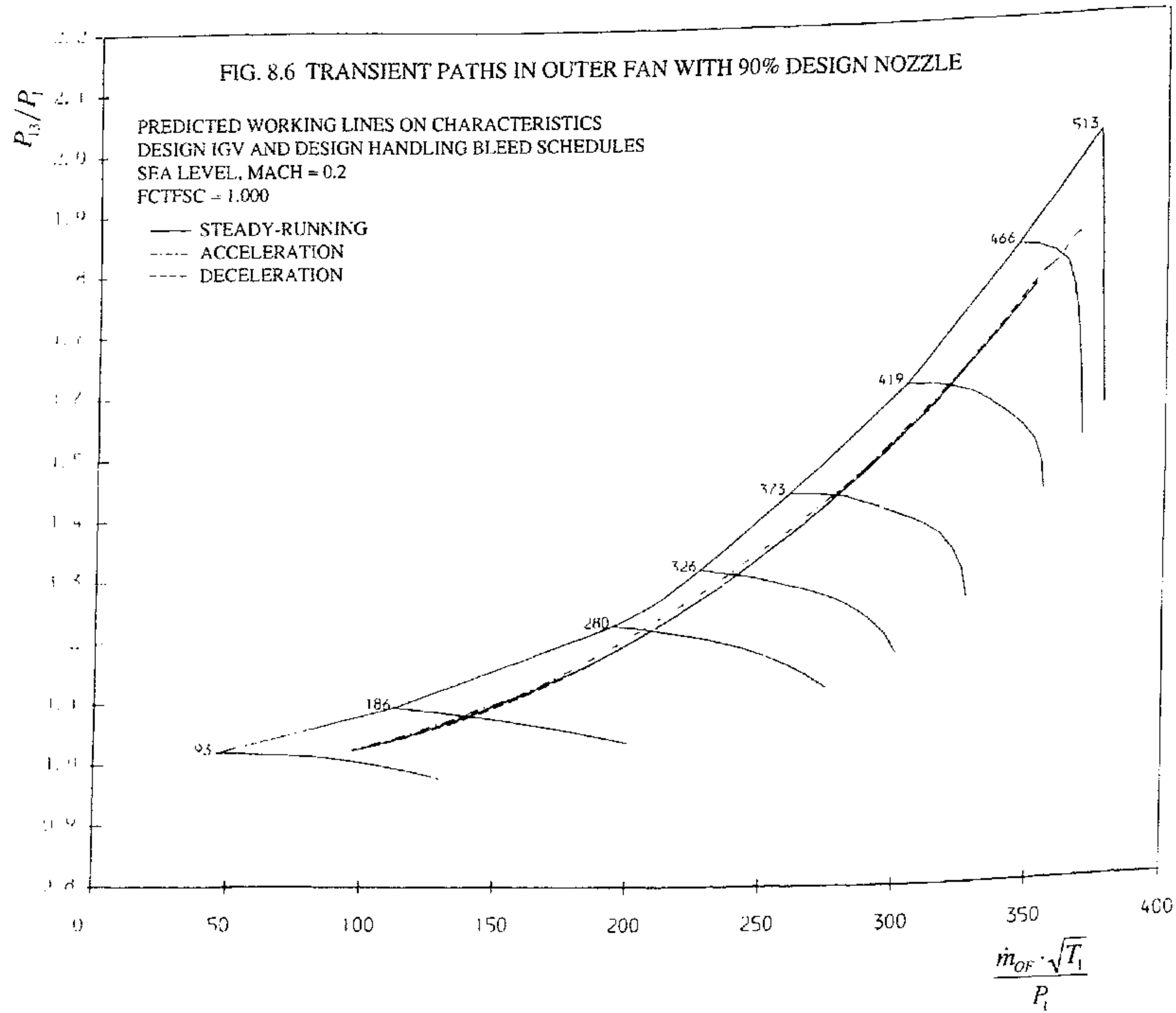


FIG. 8.6 TRANSIENT PATHS IN OUTER FAN WITH 90% DESIGN NOZZLE

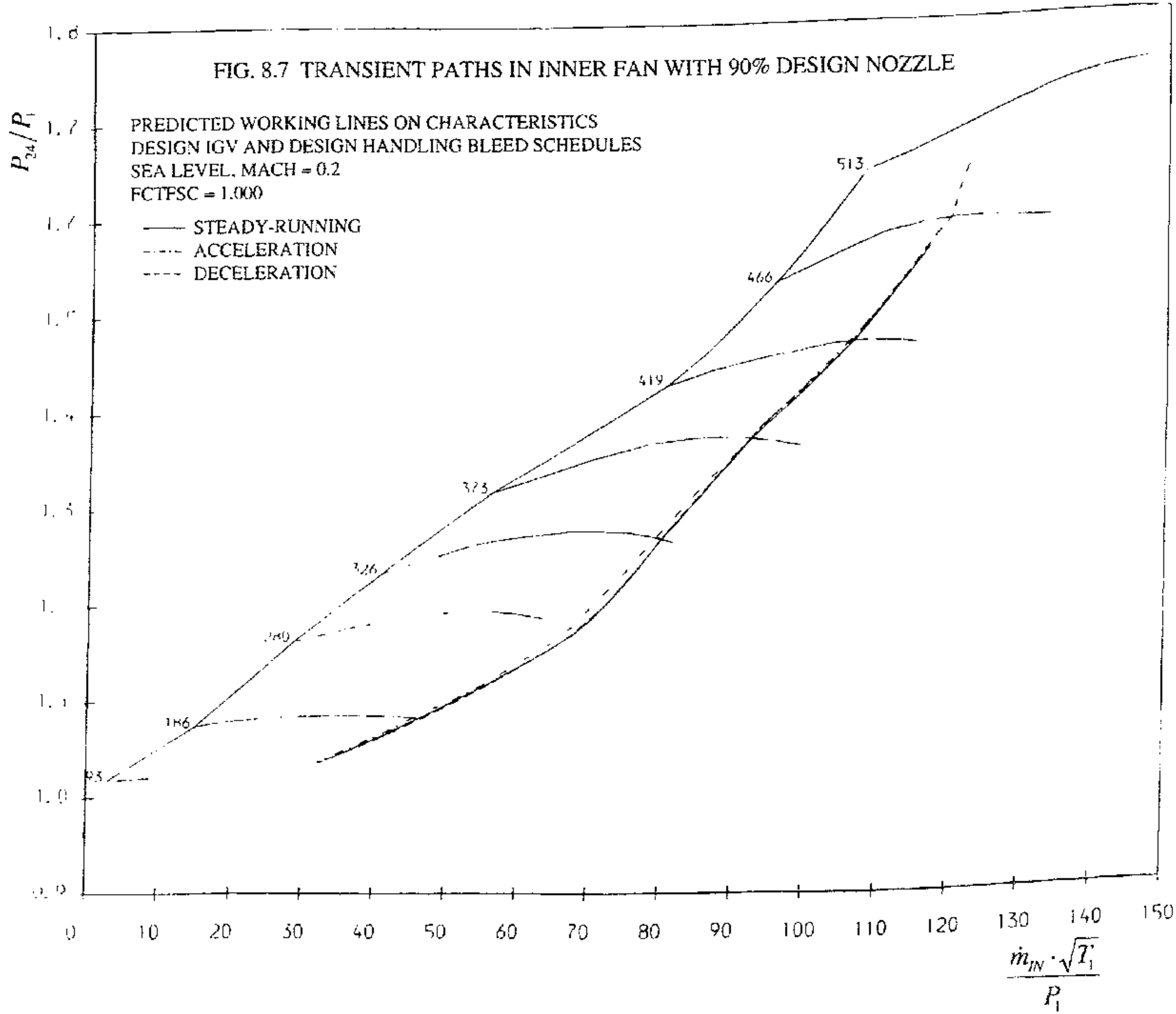


FIG. 8.7 TRANSIENT PATHS IN INNER FAN WITH 90% DESIGN NOZZLE
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
P_{24}/P_1
$\frac{\dot{m}_{IN} \cdot \sqrt{T_1}}{P_1}$
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
1.0
1.1
513
466
419
323
186

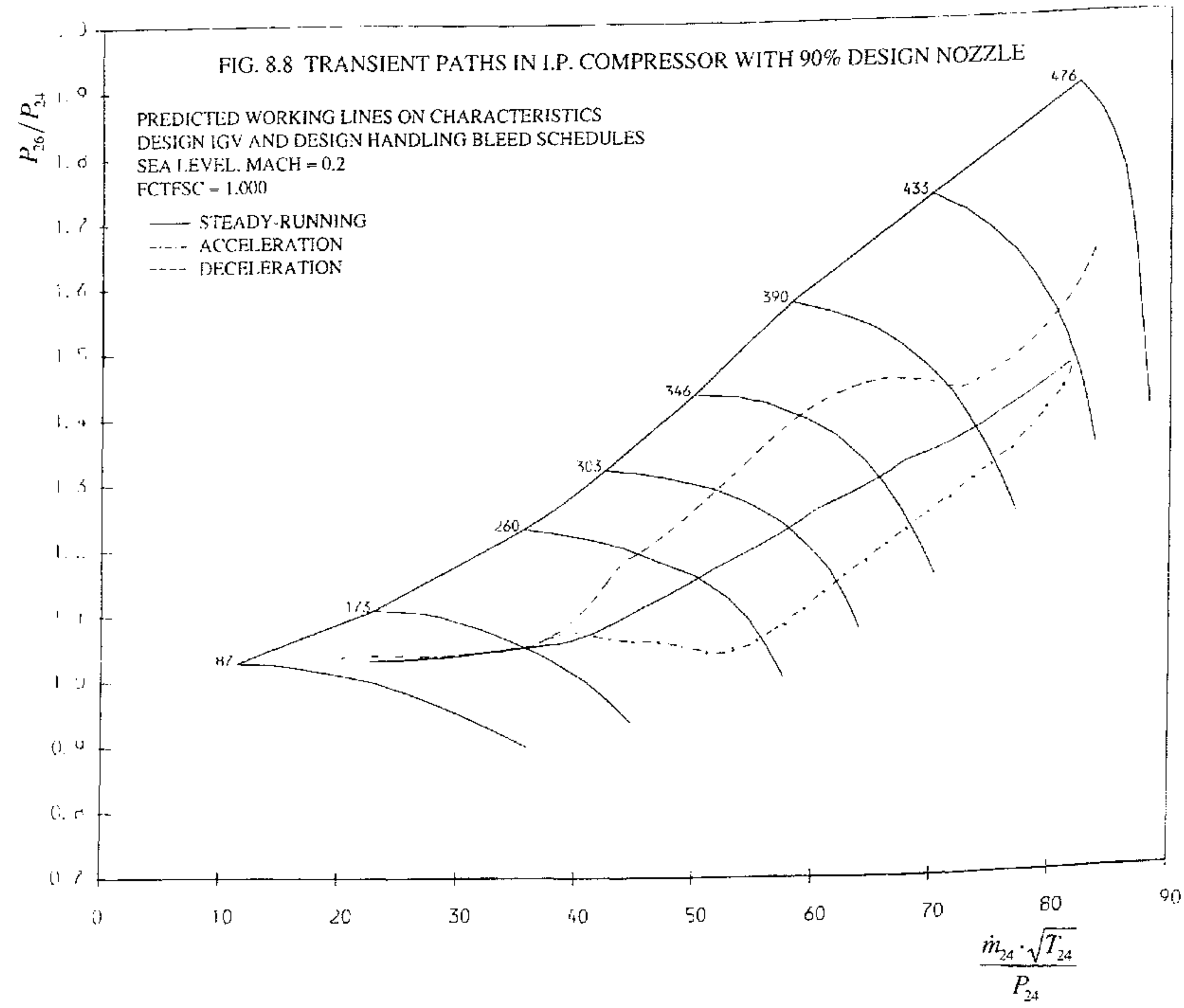


FIG. 8.8 TRANSIENT PATHS IN I.P. COMPRESSOR WITH 90% DESIGN NOZZLE
PREDICTED WORKING LINES ON CHARACTERISTICS
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL. MACH = 0.2
FCTFSC = 1.000
STEADY-RUNNING
ACCELERATION
DECELERATION
87
173
260
303
346
390
433
476
P_{26}/P_{24}
$\frac{\dot{m}_{24}\cdot\sqrt{T_{24}}}{P_{24}}$
0
10
20
30
40
50
60
70
80
90

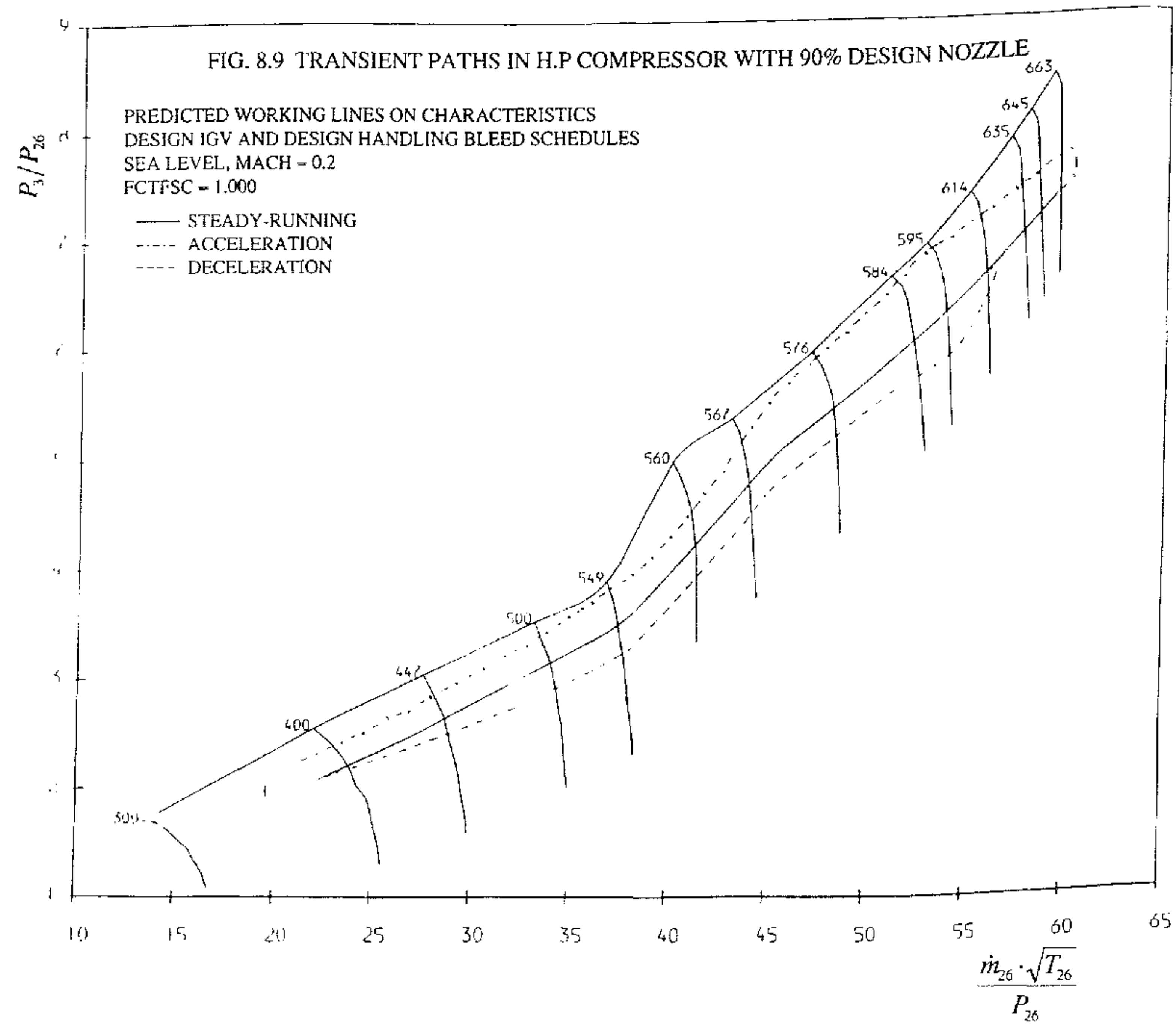
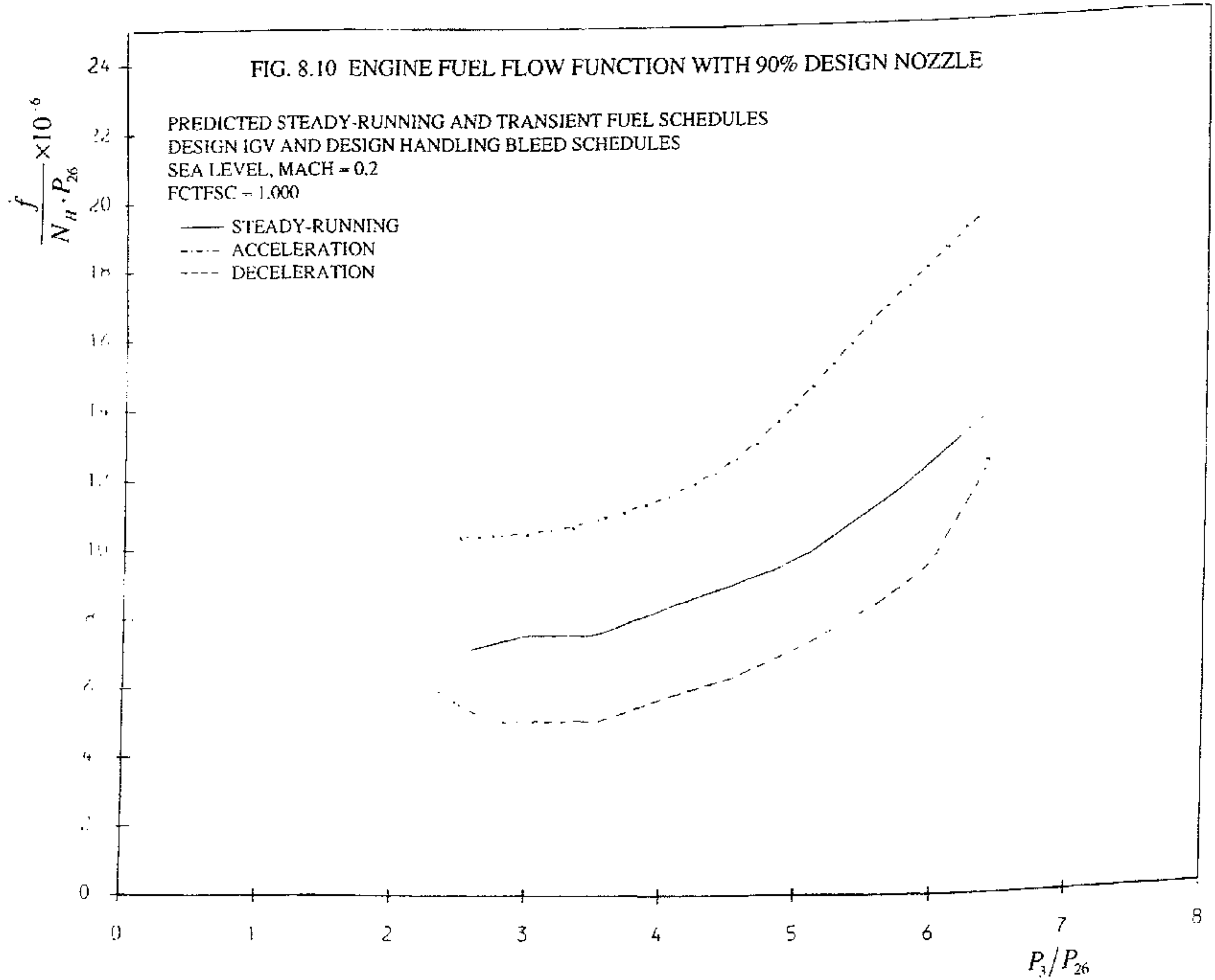


FIG. 8.9 TRANSIENT PATHS IN H.P COMPRESSOR WITH 90% DESIGN NOZZLE

FIG. 8.10 ENGINE FUEL FLOW FUNCTION WITH 90% DESIGN NOZZLE



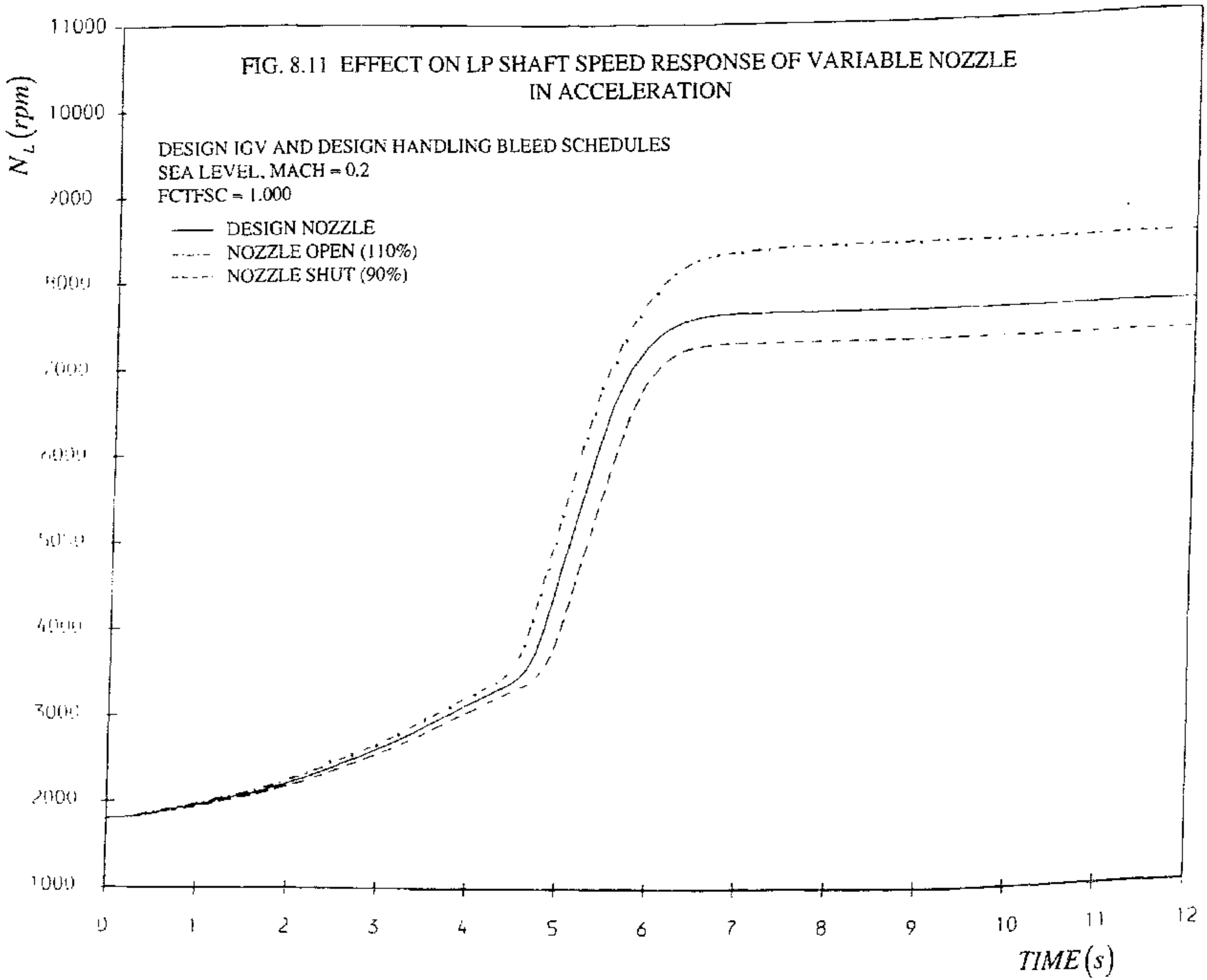


FIG. 8.11 EFFECT ON LP SHAFT SPEED RESPONSE OF VARIABLE NOZZLE IN ACCELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN NOZZLE
NOZZLE OPEN (110%)
NOZZLE SHUT (90%)
$N_L\,(rpm)$
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
1
2
3
4
5
6
7
8
9
10
11
12
$TIME\,(s)$

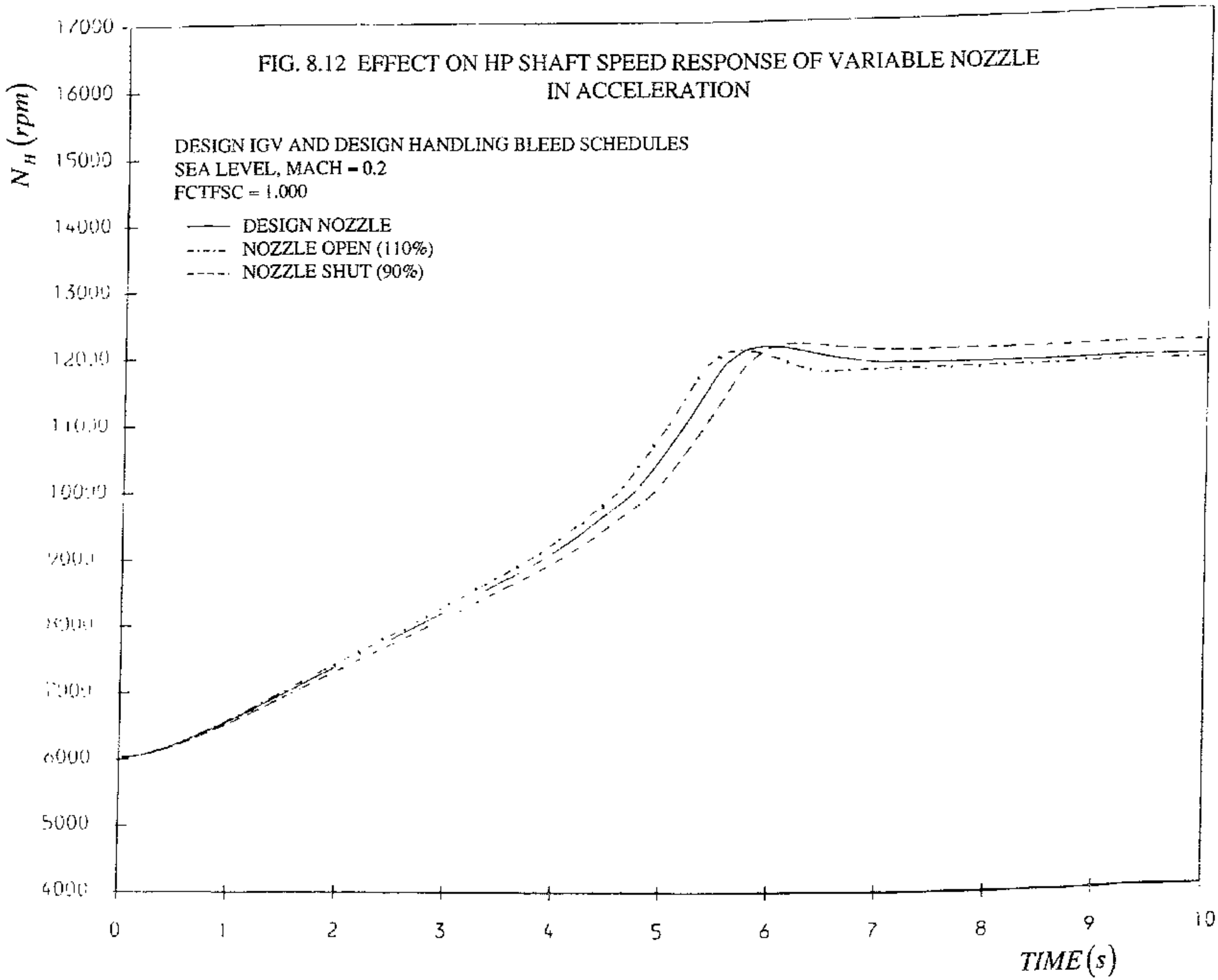


FIG. 8.12 EFFECT ON HP SHAFT SPEED RESPONSE OF VARIABLE NOZZLE IN ACCELERATION

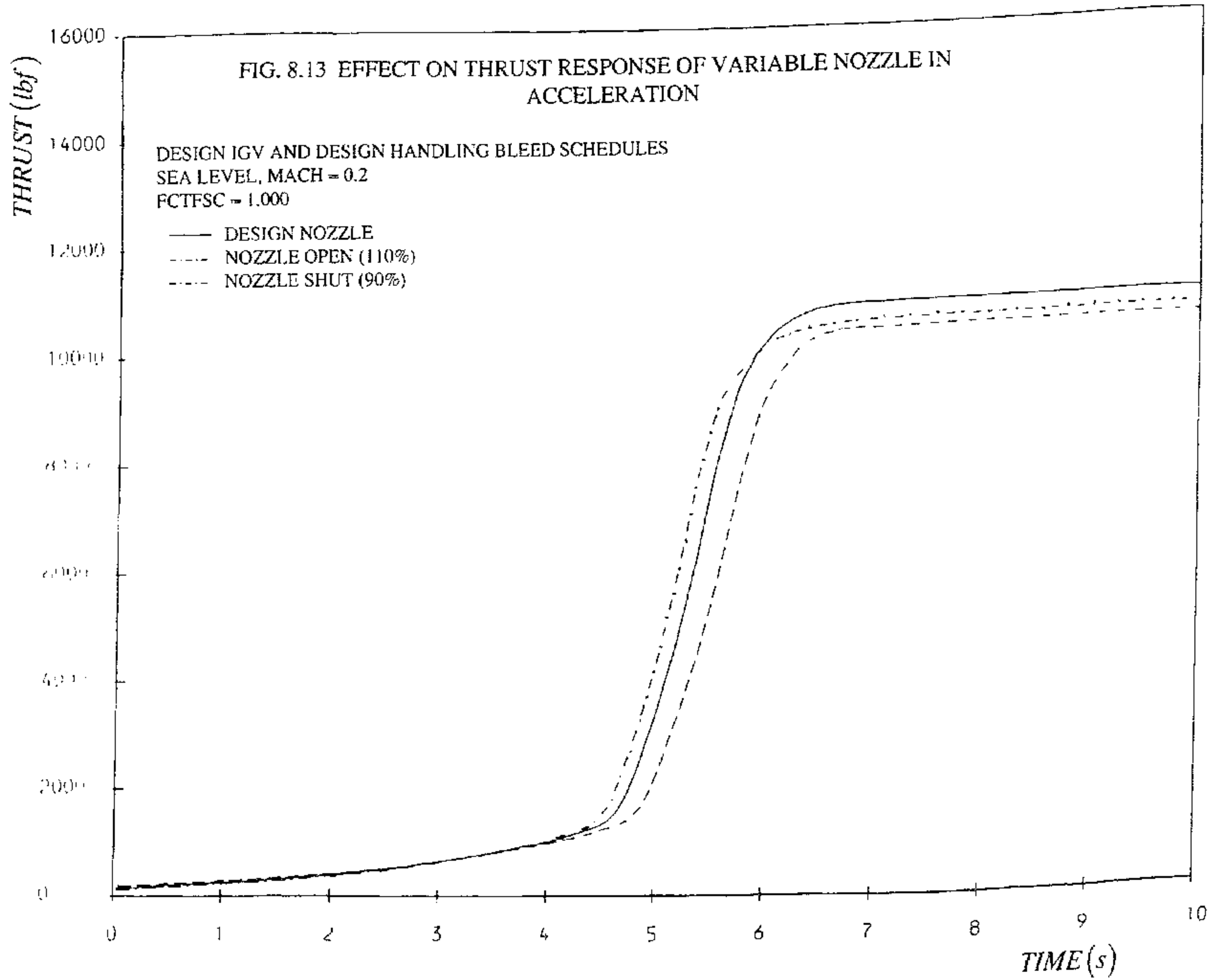
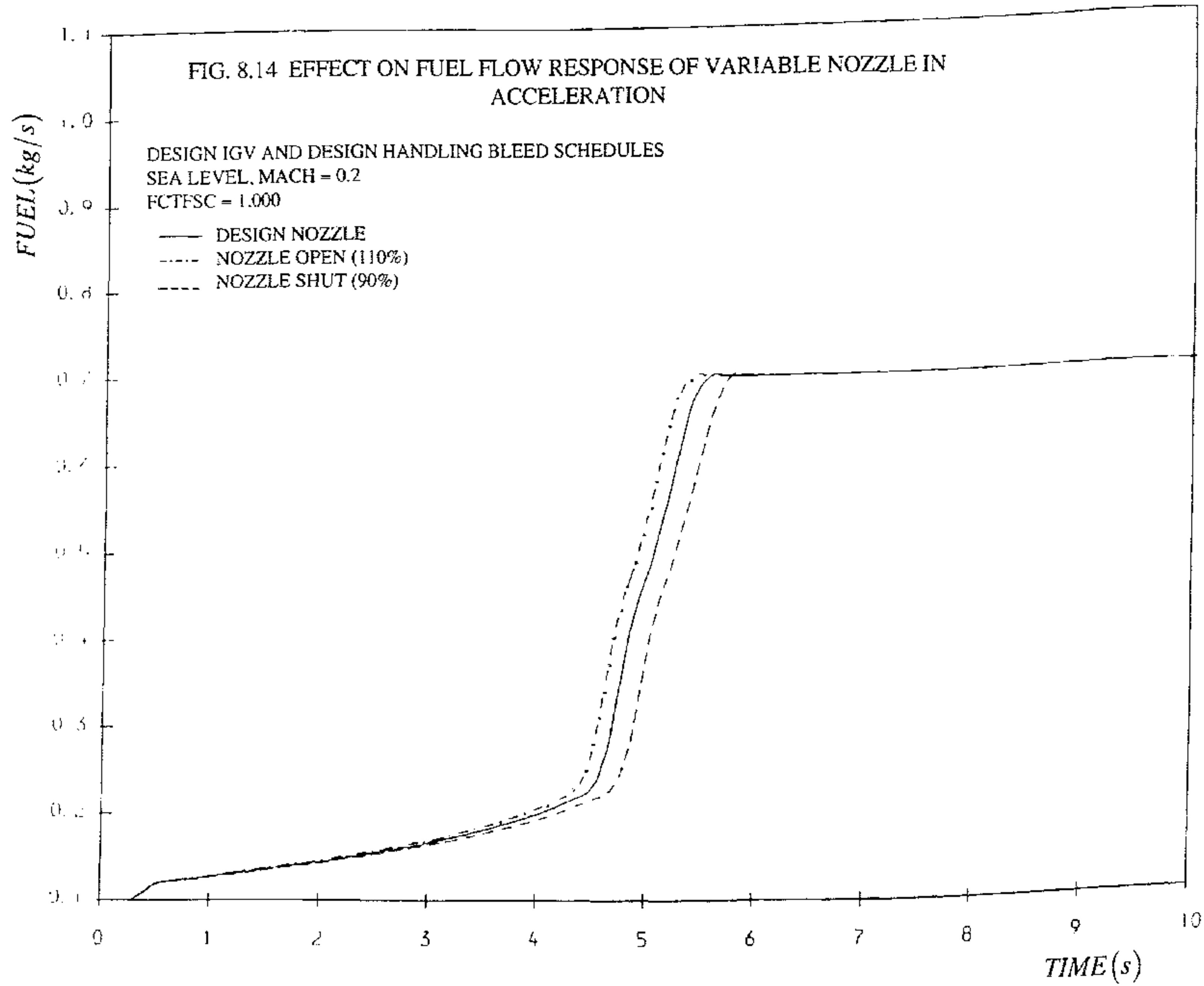


FIG. 8.13 EFFECT ON THRUST RESPONSE OF VARIABLE NOZZLE IN ACCELERATION

FIG. 8.14 EFFECT ON FUEL FLOW RESPONSE OF VARIABLE NOZZLE IN ACCELERATION



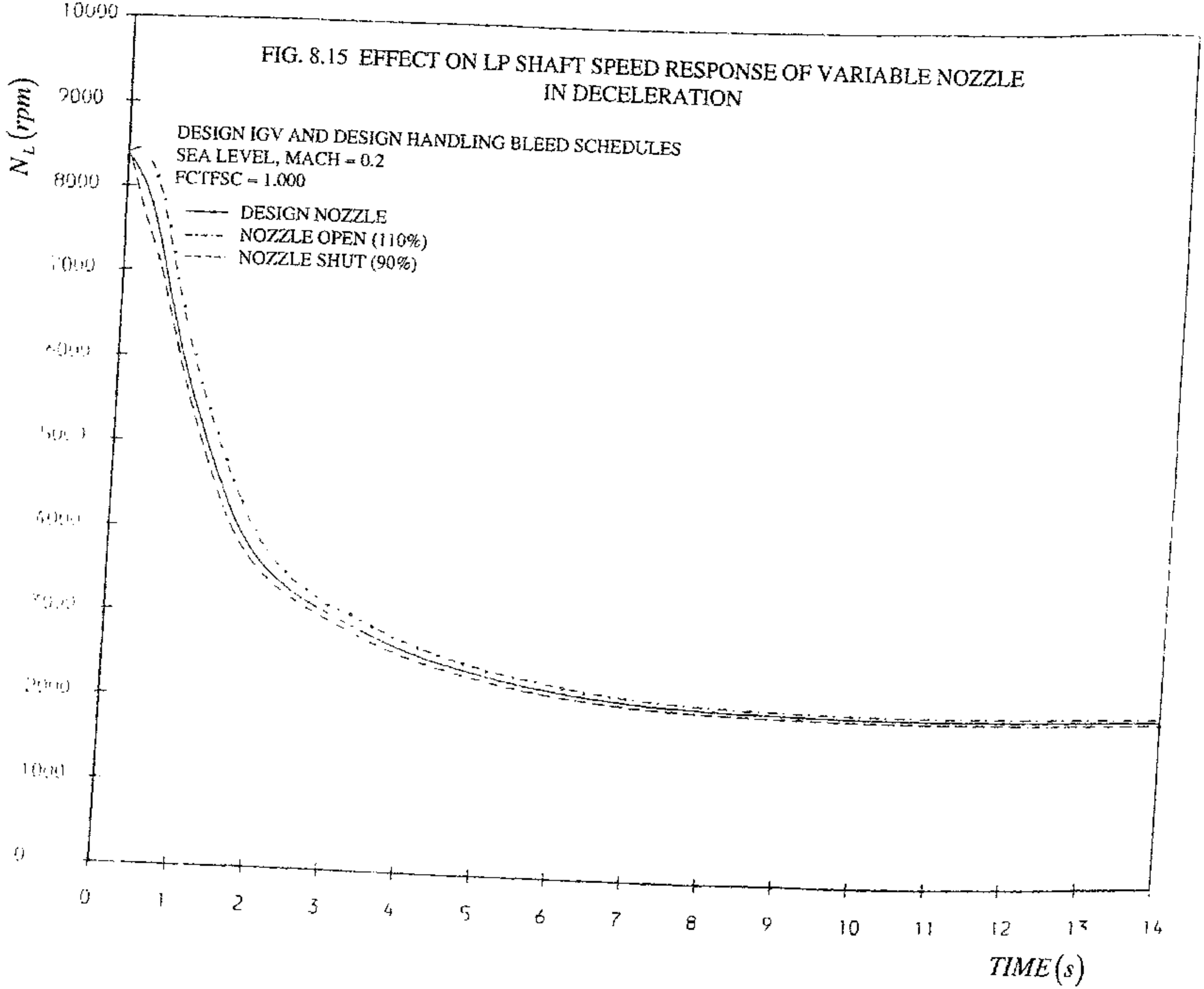


FIG. 8.15 EFFECT ON LP SHAFT SPEED RESPONSE OF VARIABLE NOZZLE IN DECELERATION

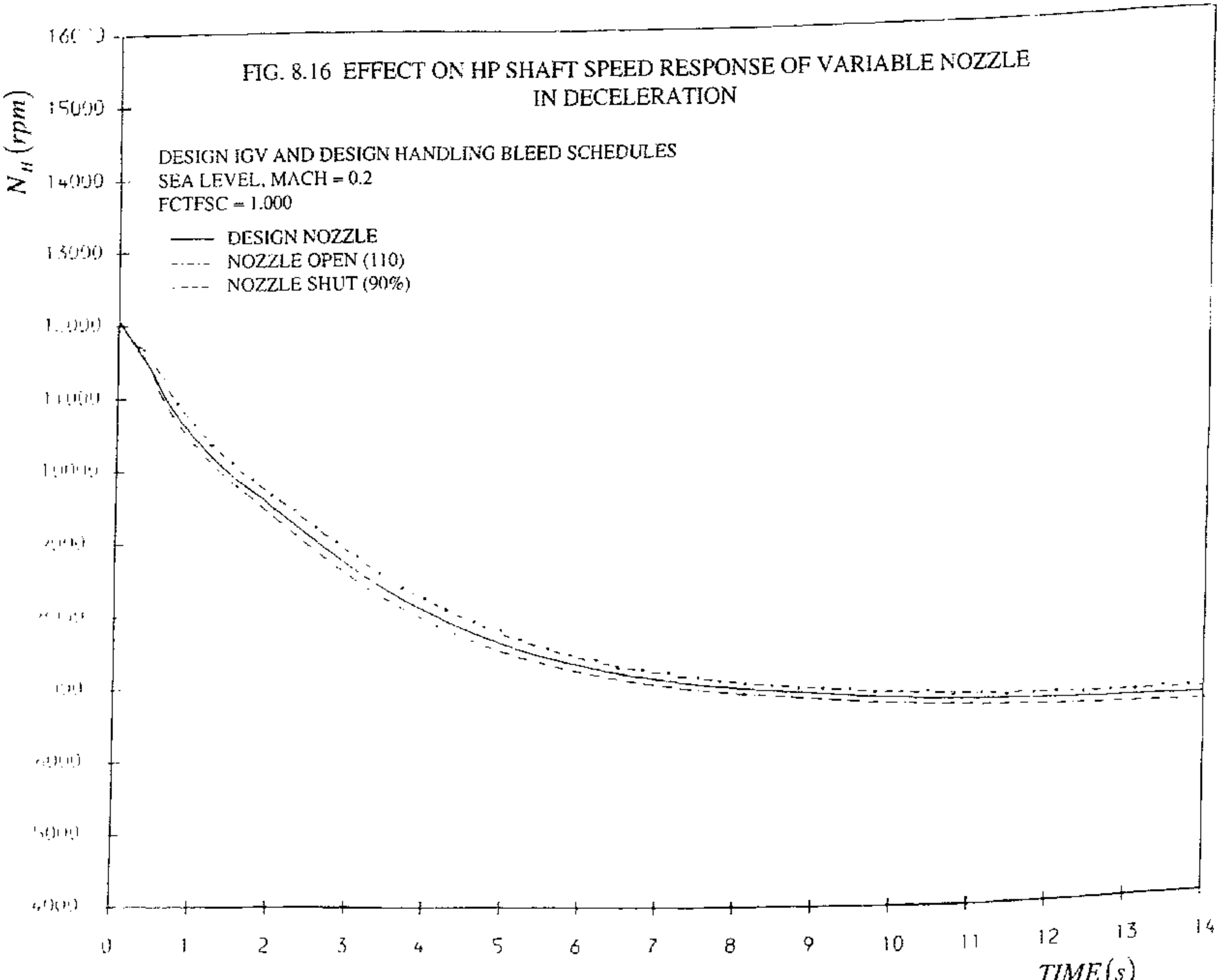


FIG. 8.16 EFFECT ON HP SHAFT SPEED RESPONSE OF VARIABLE NOZZLE IN DECELERATION

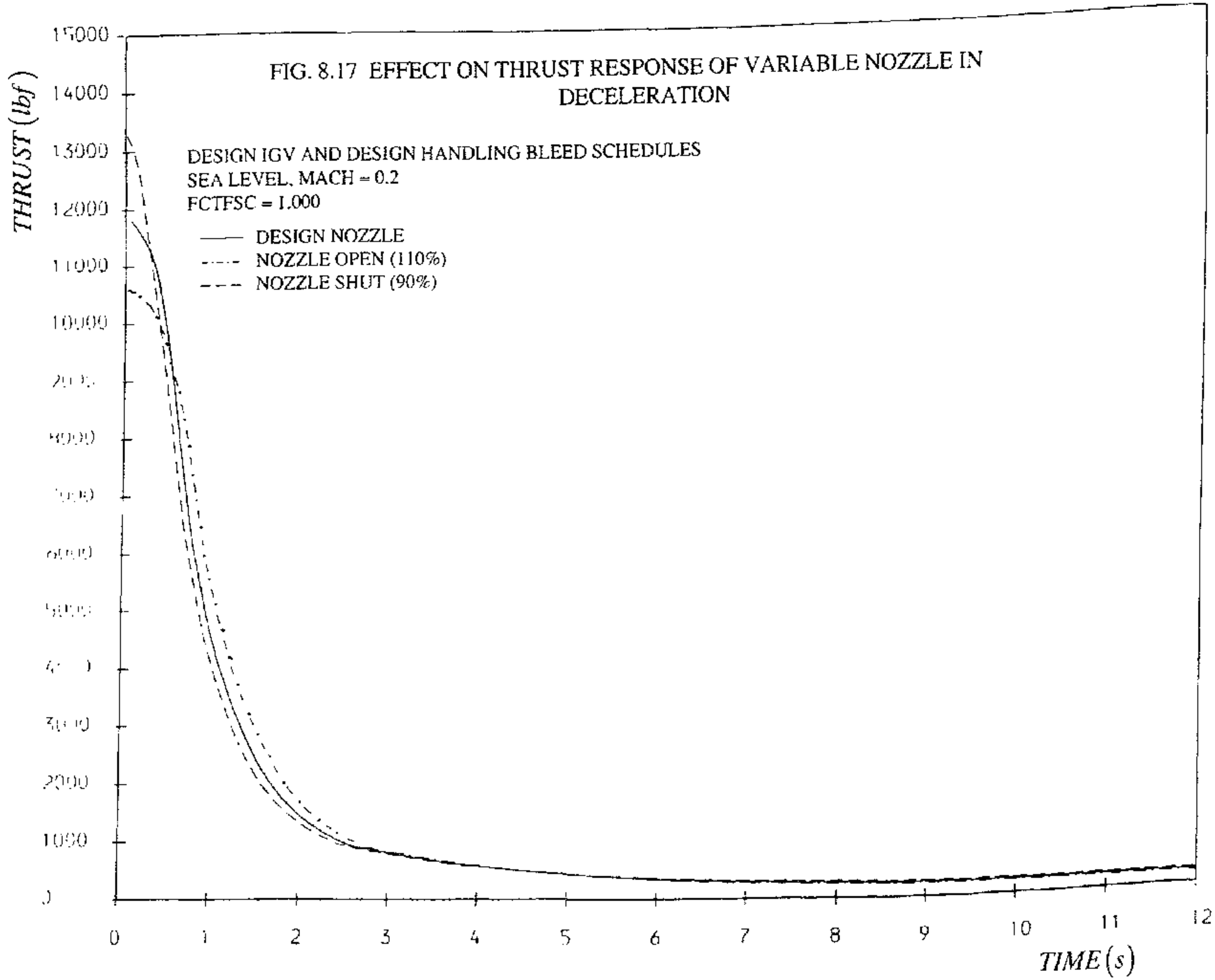


FIG. 8.17 EFFECT ON THRUST RESPONSE OF VARIABLE NOZZLE IN DECELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN NOZZLE
NOZZLE OPEN (110%)
NOZZLE SHUT (90%)
THRUST (lbf)
TIME (s)
15000
14000
13000
12000
11000
10000
2000
1000
0
1
2
3
4
5
6
7
8
9
10
11
12

FIG. 8.18 EFFECT ON FUEL FLOW RESPONSE OF VARIABLE NOZZLE IN DECELERATION

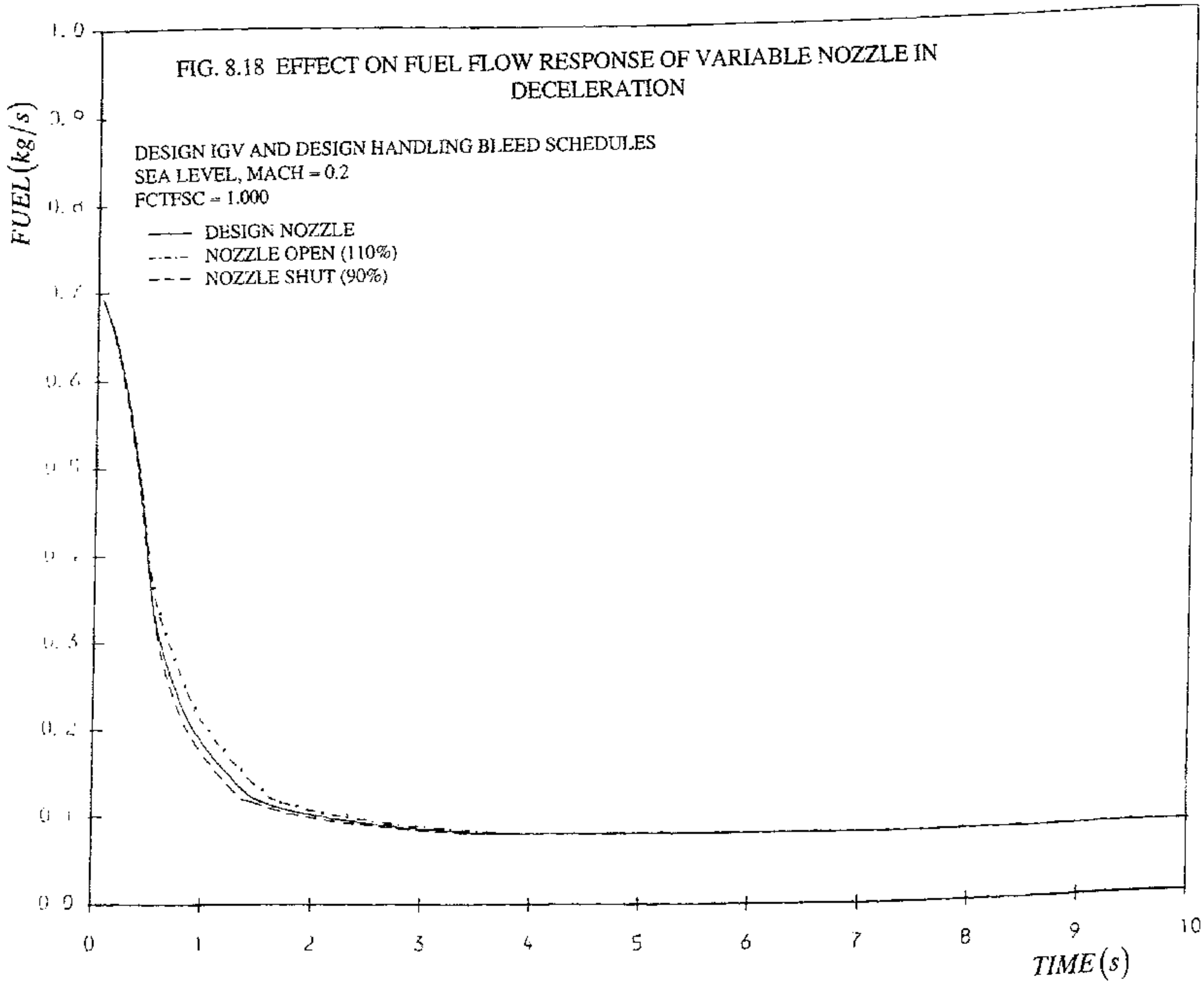
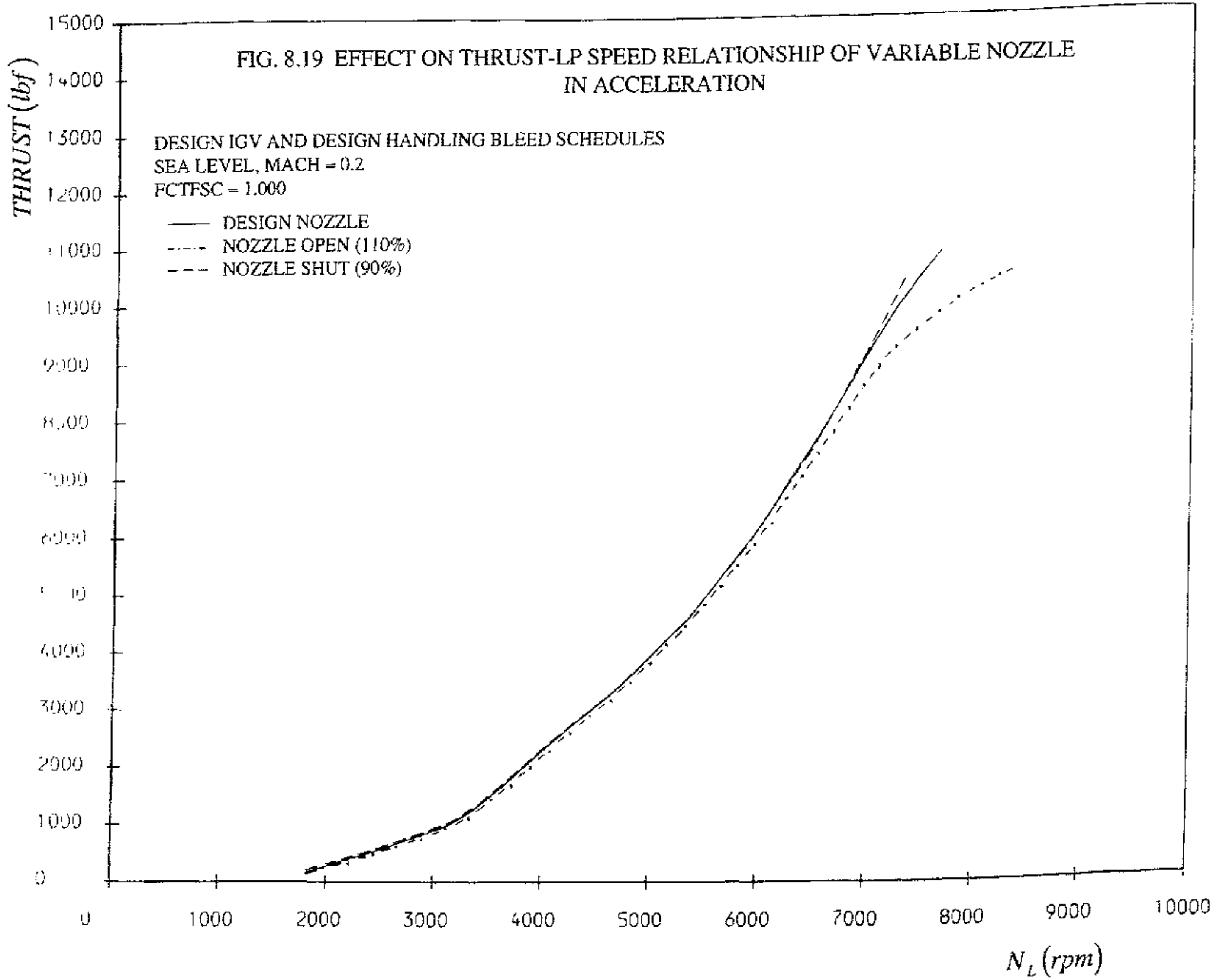


FIG. 8.19 EFFECT ON THRUST-LP SPEED RELATIONSHIP OF VARIABLE NOZZLE IN ACCELERATION



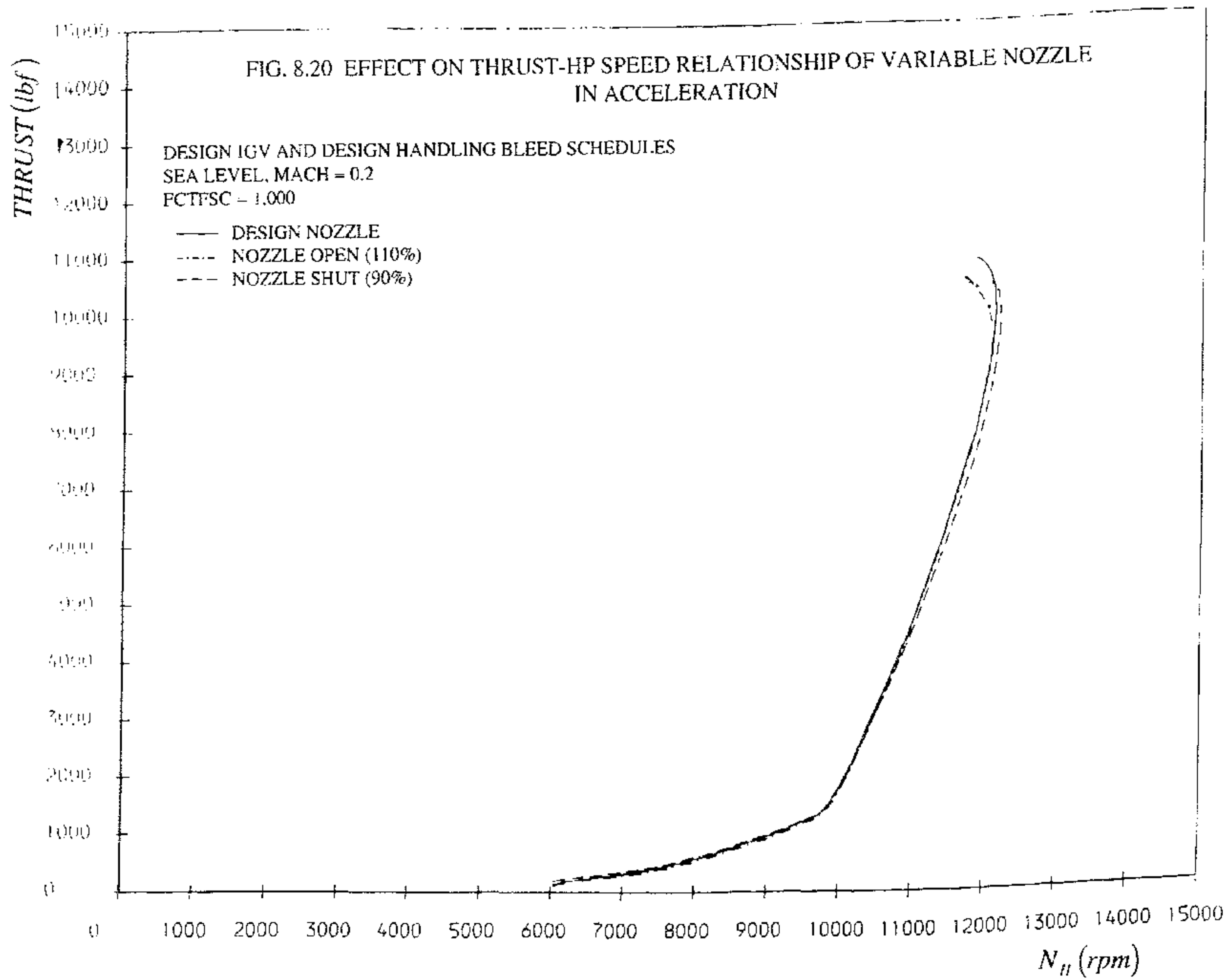


FIG. 8.20 EFFECT ON THRUST-HP SPEED RELATIONSHIP OF VARIABLE NOZZLE
IN ACCELERATION
DESIGN IGV AND DESIGN HANDLING BLEED SCHEDULES
SEA LEVEL, MACH = 0.2
FCTFSC = 1.000
DESIGN NOZZLE
NOZZLE OPEN (110%)
NOZZLE SHUT (90%)
THRUST (lbf)
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
11000
12000
13000
14000
15000
N_H (rpm)

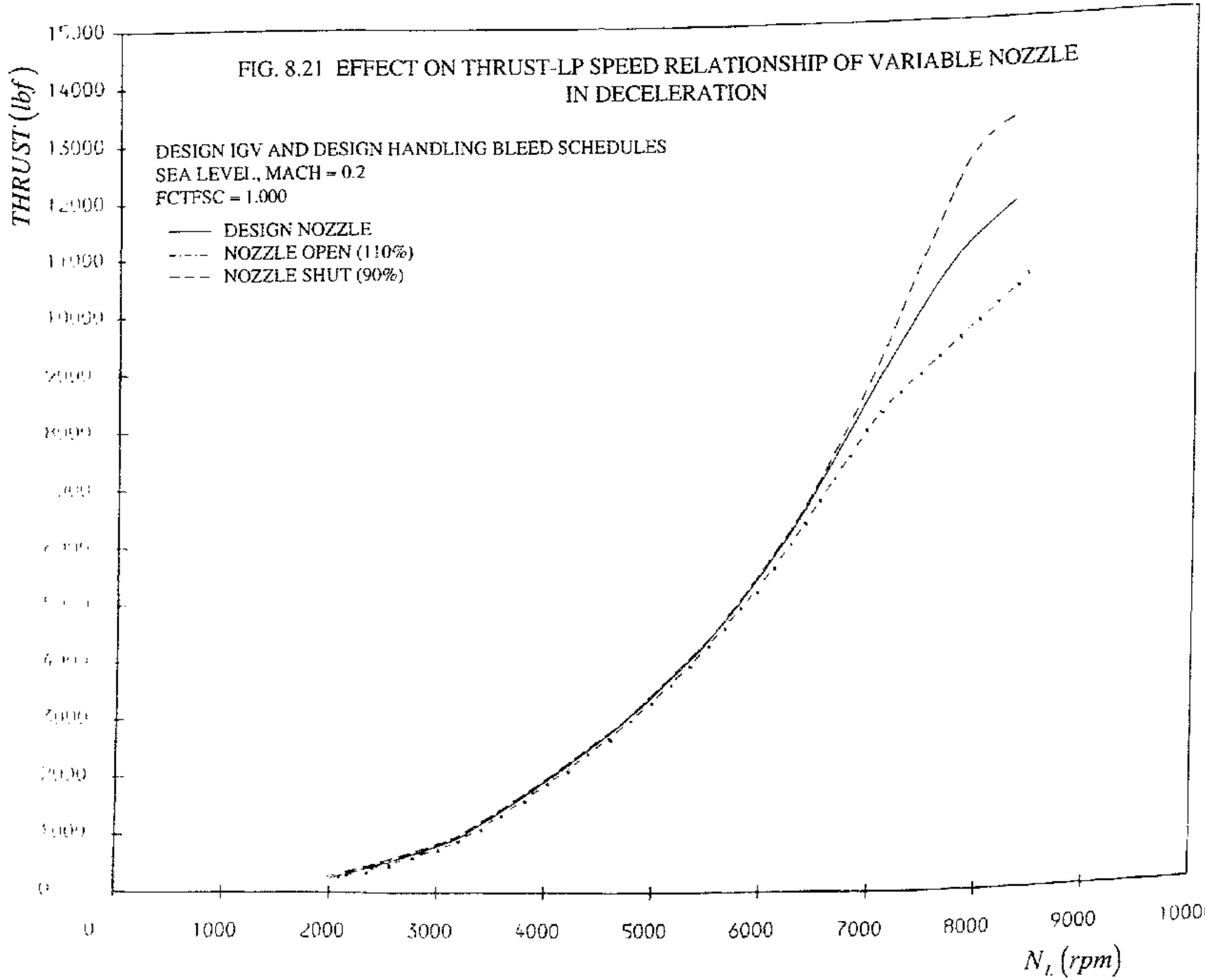


FIG. 8.21 EFFECT ON THRUST-LP SPEED RELATIONSHIP OF VARIABLE NOZZLE IN DECELERATION

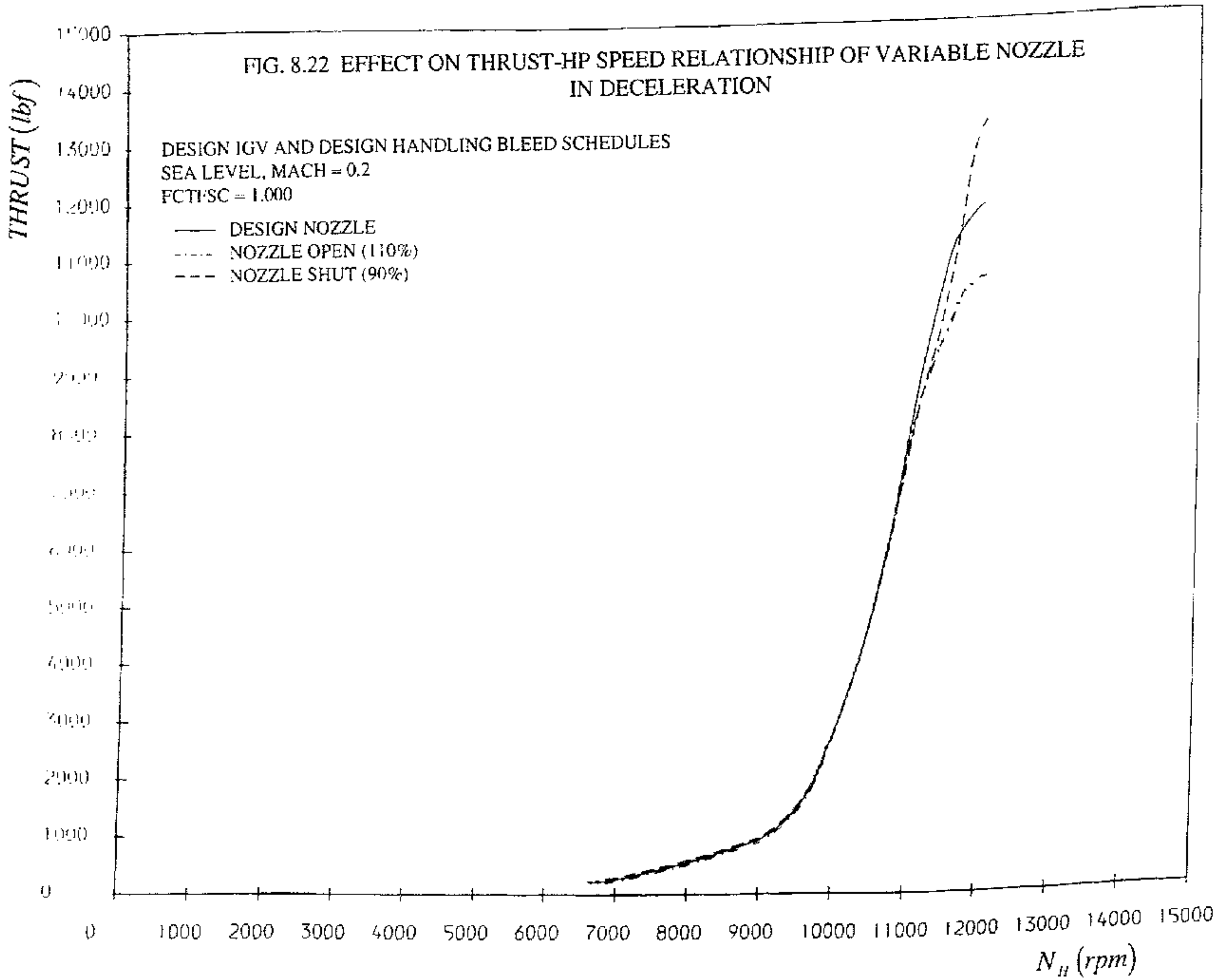


FIG. 8.22 EFFECT ON THRUST-HP SPEED RELATIONSHIP OF VARIABLE NOZZLE IN DECELERATION