

**GT2025-151418**

## DEVELOPMENT AND VALIDATION OF A TRANSIENT GAS TURBINE SIMULATION TOOL FOR SHAFT-FAILURE MODELLING

**Sajal Kissoon**  
Cranfield University  
Bedford, UK

**Mauro Righi**  
Rolls-Royce  
Bristol, UK

**Lucas Pawsey**  
Rolls-Royce  
Derby, UK

**Vassilios Pachidis**  
Cranfield University  
Bedford, UK

**Ioannis Roumeliotis**  
Cranfield University  
Bedford, UK

### ABSTRACT

*New civil engines must demonstrate, either by testing or analysis, that a shaft failure event will not result in hazardous engine effects. Without an accurate method of predicting levels of turbine overspeed, disk design tends to be pessimistic which leads to a weight penalty. This paper presents an alternative approach based on a highly integrated, multi-disciplinary, dynamic whole-engine simulation tool capable of predicting a gas turbine engine's behaviour following the failure of a shaft. It describes the methodology and validation of a 1D through-flow compressor model capable of post-stall simulation, coupled with a transient combustor model and a quasi-steady turbine solver for overspeed modelling. The individual components are integrated aerodynamically and mechanically to produce a transient whole-engine model which is resolved at timesteps ranging from 1 to 100  $\mu$ s. Control and aerodynamic effects impacting turbine overspeed predictions include changes in handling bleeds, fuel supply, turbine capacity reduction due to overspeed, frictional torque and variable geometry malschedule. The modelling of a gas turbine engine's response following a surge event is required for overspeed modelling which is achieved using compressor characteristics defined from normal operation to reverse flow. The whole engine simulation tool is validated against a 3-spool experimental gas turbine engine which experienced a shaft failure event on a test stand. The simulation model correctly predicted the turbine's terminal speed within a 4% error margin and blowdown rate following a surge event until recovery.*

Keywords: Gas turbine, Shaft failure, certification, safety, performance prediction, transient modelling, methods.

### NOMENCLATURE

|        |                                                                         |
|--------|-------------------------------------------------------------------------|
| ACRoSS | Axial Compressors Rotating Stall and Surge                              |
| CEA    | Chemical Equilibrium with Applications                                  |
| CFL    | Courant-Friedrichs-Lewy                                                 |
| Comb   | Combustor                                                               |
| CS-E   | Certification Specialist and Acceptable Means of Compliance for Engines |
| HP     | High Pressure                                                           |
| HPC    | High Pressure Compressor                                                |
| HPT    | High Pressure Turbine                                                   |
| IP     | Intermediate Pressure                                                   |
| IPC    | Intermediate Pressure Compressor                                        |
| IPT    | Intermediate Pressure Turbine                                           |
| LP     | Low Pressure                                                            |
| LPC    | Low Pressure Compressor                                                 |
| LPT    | Low Pressure Turbine                                                    |
| PR     | Pressure Ratio                                                          |
| PT     | Total Pressure                                                          |
| SAS    | Secondary Air System                                                    |
| TR     | Temperature Ratio                                                       |
| TT     | Total Temperature                                                       |
| NASA   | National Aeronautics and Space Administration                           |
| NGV    | Nozzle Guide Vane                                                       |
| NRT    | Non-dimensional rotational speed                                        |
| WEST   | Whole Engine Simulation Tool                                            |
| WRTF   | Non-dimensional mass flow                                               |

## 1. INTRODUCTION

Shaft failure events in gas turbine engines are potentially dangerous as the sudden decoupling of the compression side with the turbine leads to the rotor accelerating so rapidly that overspeed limits can be reached within less than a second. The sequence of events occurs so rapidly that turbine overspeed prediction is particularly hard to achieve analytically.

In 2010, an uncontained engine shaft failure happened on an A380 during a regular flight from London to Sydney via Singapore [1]. During climb, 7000 ft over Indonesia, an oil feed stub pipe feeding the left inboard engine's HP/IP turbine bearing suffered from leakage which ignited in the chamber. The fire led to the separation of the IP turbine disk from the driveshaft and to the disk going into overspeed and bursting. It released highly energetic disk fragments which punctured the leading edge of the left-wing and penetrated the left fuel tank before going through the top side of the wing. The puncture and damaged oil supply pipe led to a fire breaking out inside the tank and the engine. They were both extinguished rapidly and the pilots successfully landed without any casualties. Following this event, several steps were taken to prevent this failure mode from happening again. This included updating engine control software and revisiting the certification compliance advisory material for future engine designs.

In 2012, a certification of memorandum came out on turbine overspeed [2]. It outlines the weaknesses of prediction analysis in accurately predicting rotor overspeed following shaft failure. This memorandum affects Turbine Engine Type Certificates namely CS-E 850. It suggests testing engines by initiating shaft failure at the worst operating conditions which would maximise levels of rotor overspeed or through a validated analysis against the actual engine which encompass all relevant aspects of the failure mechanism (including surge).

The smart engine project, as presented by Greitzer, highlighted that an "interdisciplinary" (fluids, control, structures, and instrumentation) method was required to achieve improved control and operability due to the highly integrated nature of modern aero engines [3]. This limitation in understanding and predicting the complex dynamic interaction between each component of an aero engine and estimating with sufficient levels of accuracy, the system-level response impairs the ability of engine manufacturers to push the limits of the design space, operability and capability to predict with enough confidence levels of turbine overspeed.

For certification purposes, engine manufacturers rely on extensive testing campaigns or prior knowledge of engine response based on a known architecture to predict levels of turbine overspeed in case of a shaft failure. This approach is prohibitive in terms of cost and prevents step changes in engine design due to a lack of understanding of engine response following this failure mode.

This paper presents the development of a computationally inexpensive, physics-based approach which does not rely on empirical knowledge of past engine data to predict levels of turbine overspeed following a shaft failure event. It spans over

many years of research at Cranfield University (sponsored by Rolls-Royce) and this paper documents the validation of the toolset against an actual experimental engine which suffered from an IP shaft failure. The Whole Engine Simulation Tool (WEST) successfully predicted the turbine overspeed within 4%.

The WEST toolset can be used not only to assist with the certification of new engines but also to provide valuable insight into engine response during surge and stall events and assess how the whole engine responds. This could be employed at the initial stages of the design process and reduce the cost and weight penalty of engine redesign if certification requirements are not met.

### 1.1 Sequence of events

Following the onset of a shaft failure, the sudden decoupling of the compression and turbine side leads to a drop in compressor delivery pressure. This change in compressor matching coupled with the fluctuations in tip clearances and vibrations often leads to surge. Compressor stall and surge is a breakdown of normal flow conditions within compressors. Stall events can be further categorised into rotating stall or surge. Rotating stall is distinguished by one or more cells of low axial velocity rotating circumferentially around the compressor, at lower speed than the rotor. Comparatively, surge is the complete axisymmetric flow reversal. It is very important for any analytical model which aims at predicting levels of turbine overspeed to be able to accurately model surge and stall as the reduction in flow limits the air supplied to the turbine side almost instantly which has an immediate impact on turbine speed.

On the turbine side, the aerodynamic effects of the free-running turbine can lead to highly negative incidence angles on the nozzle guide vanes (NGV) of the following turbine. In such cases, a 'separation bubble' is formed which restricts flow capacity downstream and increases the loss coefficient. This reduction in flow capacity downstream will also affect air going through all components upstream and it was shown [4], [5] to have a non-negligible reduction in torque produced by the turbine in overspeed. In cases where the shaft failure leads to a rearward movement of the rotor, increases in tip clearances leads to work reduction which also reduces levels of overspeed.

On the combustion side, once surge occurs, the sudden drop in air supply causes a fuel-rich mixture which further reduces the amount of energy being supplied to the gas path. This reduction in combustion efficiency is a secondary effect of surge.

A shaft break event can be categorised into unlocated or located events. Based on the location of the fracture (before or after located bearings) the sequence of events can vary significantly. An unlocated shaft failure leads to the axial movement of the turbine assembly rearwards which could lead to greater damage than one which is located. The secondary air system also plays a role in cases of unlocated shaft breaks. Depending on the engine architecture rearward movement can either provide additional friction through contact with downstream components or no additional friction due to a lack of rearward load as a result of cavity sealing. A model capable of

capturing these scenarios is desirable for overspeed analysis. Gallar et al. developed a dynamic secondary air system to integrate these effects into the whole engine solver[6], [7]

From a mechanical perspective, unlocated shaft breaks are comparatively more desirable than located ones due to the large frictional loads acting between the rotating and static components. In such cases, blade tangling might occur which leads to a rapid capacity reduction on the turbine side [8], [9]. However, in cases of located shaft breaks, the frictional torque is comparatively lower which is likely to lead to higher levels of turbine overspeed.

Once a shaft break is detected, various measures can be employed. Some, such as blade tangling [10] will have a near-instantaneous impact on reducing turbine work. However, compared to the aerodynamic and mechanical effects happening right after shaft break, control mechanisms relying on detection and actuation are relatively slower.

With the world currently undergoing a technological revolution that sees innovative engine concepts being introduced, a new physics-based tool to predict the transient engine response could allow for better optimization of new engine products. This paper outlines the development and experimental validation of a unique tool, capable of predicting levels of turbine overspeed following a shaft failure event. It accounts for key aerodynamic, mechanical and control measures which can impact the terminal speed reached by the free-spinning turbine. Its prediction can help understand how various design parameters affect system-level response and improve engine development for safety, right at the conceptual design stage.

## 2. WHOLE ENGINE SIMULATION TOOL

The Whole Engine Simulation Tool (**WEST**) was developed after several years of research at Cranfield University to assess engine behaviour at both a component and system level during events of overspeed for engine certification. The unique predictive capability of this tool derives from the advanced, inter-disciplinary, analytical modelling approach which provides an “all-in-one” performance simulation tool that has the capability of simulating various shaft failure scenarios leading to overspeed. It incorporates a compressor solver capable of post-stall modelling, a quasi-steady turbine solver, a transient combustion and secondary air system model. It also includes friction modelling between stationary and rotating parts as shown in Figure 1. This tool has the capability of modelling a wide range of engine architectures and could be used right from the early stages of design, which could reduce the cost of conducting test campaigns.

Only the core gas path is modelled while the by-pass section is included by defining the fan tip power at varying rotational speed. Based on the state of the engine, it is removed from the turbine power before resolving the mechanical speed of the spools. The 1D domain starts from the fan hub to the NGV with the gas path domain geometry defined by providing the cross-

sectional area along the engine length. The flow properties at the outlet of the combustor are defined as boundary conditions to the inlet of the turbine. The state of the various systems is progressed synchronously in time with the timestep defined by the CFL condition from the 1D domain. This is usually between 1 to 100  $\mu$ s.

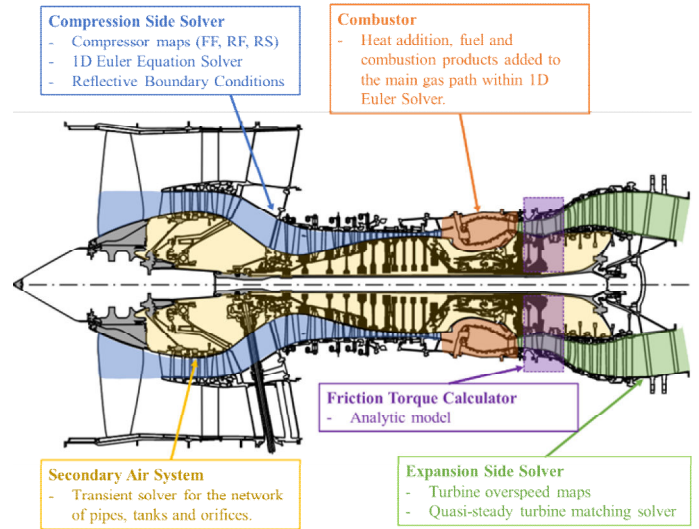


**FIGURE 1:** Subsystems used in developing a whole engine model.

### 2.1 Compression Side Solver

The compressor consists of a 1D Euler equation solver for flow properties within the main gas path from the intake up to the combustor outlet which includes the nozzle guide vane. The solver was originally based on the MacCormack scheme but transitioned to the Roe scheme following robustness issues which were resolved by a monotonicity preserving scheme. The wave strengths of the Roe solver were modified according to Mohanraj’s method [11] to account for source terms due to compressor work, bleed air, heat addition from combustion, fuel addition and pressure losses. A real gas model was implemented according to Di Mare’s method [12] and species tracking was also added to obtain accurate gas properties following combustion. The integration to the Roe scheme followed Laroutto’s semi-coupled approach [13]. The compression domain is defined as an area variation with length, with the location of the various component specified lengthwise. The work produced is read from complete compressor characteristics obtained from a 3D through-flow code, ACROSS [14], which defines the stall and reverse flow region of the map. The rotating stall and reverse flow characteristics are both obtained by operating each compressor locked-in-stall and in fully reverse flow conditions to generate enough points for each speedline.

Figure 2 provides an illustration of a compressor map used in WEST through a single speed line. It consists of 3 main regions namely: forward flow, stall and reverse flow. The forward flow map was obtained by the manufacturer while the

rotating stall and reverse flow characteristics were obtained from ACROSS. The axisymmetric characteristics is generated internally by connecting the reverse flow and forward flow characteristics through a cubic spline interpolation with the ends being at a gradient of zero. The rationale behind having 2 lines representing an NRT line within the stall region is to capture the transient behaviour of a compressor when the operating point is moving across this region. If the operating conditions across the compressor are favouring the formation of rotating stall cell/s, then the operating point will slowly transition (through a first order lag) from reading from the axisymmetric to the rotating stall line.

A significant breakthrough in modelling surge events in 1D was the use of stage-wise characteristics in reverse flow to capture the non-uniformity of the flow during the onset of surge. The compressor map is utilised to obtain the PR and TR of the operating point which are used to calculate source terms within the Euler equation solver. More details regarding the modelling of the compressor are provided in reference 15.

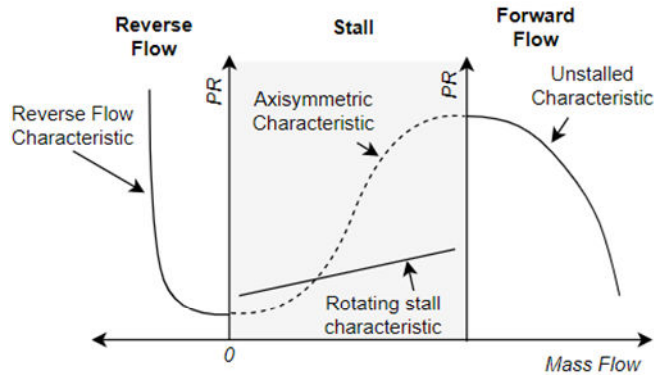


**FIGURE 2:** Complete compressor map for simulating stall and surge in one-dimension.

The compressor solver was compared and achieved very good matching against a validated 3-dimensional throughflow code for single and 2 axial compressors connected under deep surge and high-power conditions [15]. This work successfully modelled axial compressor/s under deep surge conditions using only a 1D domain connected to a plenum and using complete compressor characteristics generated by ACROSS and manufacturer's data.

## 2.2 Combustion Solver

The combustor model is based on the NPSS CEA [16] and, given the inlet conditions and fuel flow, the energy release and source terms to the Euler solver are calculated. The combustor is modelled as a plenum as shown in figure 3 where the cold section is discretised separately from the hot section. Within WEST, the hot section is where heat and combustion products are injected. A constant pressure loss is applied within this section. Current research suggests that mass flow reversal might be low between

the hot and the cold region even at high power conditions[14], reducing the inaccuracies from this assumption. For the modelling of surge, it is assumed that the NGV remains choked until recovery. This was demonstrated by Righi who analysed the evolution of pressure within the plenum and volute during surge [17]. If flow reversal is seen, extra pressure losses would need to be placed between the boundary of the cold and hot section.

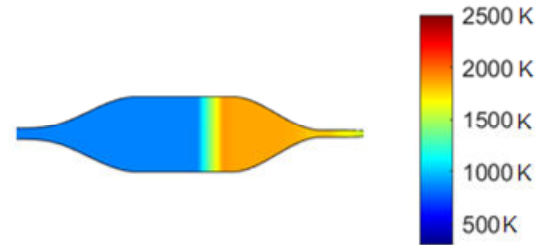


**FIGURE 3:** Representation of a 1D combustor model and temperature gradient during steady-state.

The combustor model has two ways of estimating the amount of heat released. One is through the simple use of a specified fuel heating value and the other is through a look-up table. The look-up table is generated from NASA CEA [16] which gives the combustor's energy release based on the fuel-to-air ratio and inlet conditions. [16] This enables the modelling of combustor performance ranging from fuel-rich to lean. The combustor source term can then be calculated, while considering the combustor efficiency. In normal operating conditions, the efficiency is set to be nearly unity. However, in post-stall conditions, a user-defined value, based on sponsor's recommendation, is used which tries to account for spatial variations.

## 2.3 Expansion Side Solver

The turbine solver resolves the performance of an arbitrary system of turbines in a quasi-steady manner. The extension of the compression and combustion side to the turbine was not applied due to the choked NGV assumption. This inherently assumes that any transients prior to the NGV would not propagate to the turbine side. Hence, a 0D model on the turbine side is beneficial in terms of computational cost. Information is transferred through the boundary conditions at the outlet of the NGV which is fixed at the inlet of the turbine. Having the instantaneous mechanical speed of the shaft and the outlet boundary conditions at the nozzle, the system of equations can be resolved using the turbine characteristics to determine turbine work. This is repeated at every timestep and the torque produced by every turbine is fed back to the mechanical solver routine which determines the new incremental shaft speed.

## 2.4 Secondary Air System (SAS)

A dynamic secondary air system was developed by Gallar [7] and integrated into WEST to determine the pressure build-up in blocked air cavities following an unlocated shaft failure. The SAS connects the cooling air bleeds from the compression side all the way to the turbine and cooling air system which includes turbine discs, sealing and cooling bearing chambers. The fluctuations in air properties and supply from the main gas path to the cooling chambers impact whole engine performance and most importantly the rate of pressure buildup in blocked air cavities and air supplied to the turbine side.

## 2.5 Mechanical Integration

The loading imposed from the air system is used to calculate any friction torque that results from rubbing between static and rotating parts. This is then integrated with the compressor and turbine torque to determine the net instantaneous shaft speed. In the eventuality of a shaft failure, the decoupling between the compression and turbine side is numerically achieved at a predetermined time and the resulting shaft speeds, both on the compression and turbine side are calculated separately.

The air and mechanical systems are integrated through the set-up of boundary conditions as shown in figure 4. Modelling flexibility was also retained to allow the user to transiently define external bleed requirements, end-loads and even variable geometry angles, based on the response of the control system and previous recordings to validate an engine model. Integration is achieved by progressing all the main subsystems together at each timestep. This timestep is small enough to prevent the need for reiteration.

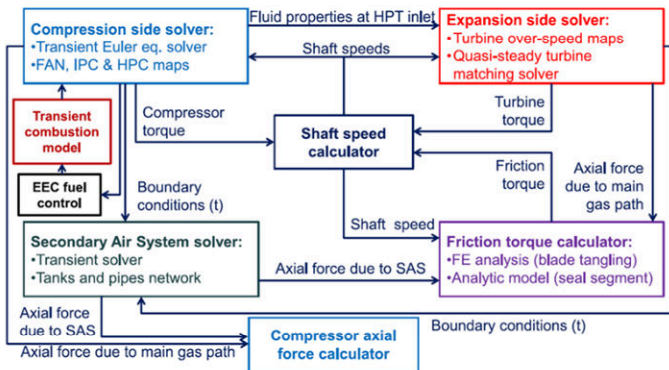


**FIGURE 4:** Flow of boundary conditions between subsystems in WEST [18].

## 3. IP SHAFT FAILURE MODELLING

The main purpose of WEST is the accurate prediction of turbine overspeed levels following a shaft failure. This segment presents the sequence of events that followed an IP shaft failure event on an experimental 3-spool engine which led to the turbine

speed reaching about twice its original value within a few seconds.

### 3.1 Sequence of events

During testing, an early experimental engine suffered from an IP shaft failure during a slow acceleration to high power. The shaft break occurred at high power conditions and led to fast acceleration of the IPT. Due to the shaft failure, the sudden decoupling of the compressor and turbine caused a mismatch between compressor speeds leading to surge. Both IP and HP compressors surged twice before the IPC ran down stall-free and delivered a pressure ratio close to 1, while the HPC was able to recover and remain stable for a couple of seconds. The surge events had a direct impact on reducing air supply to the turbine which curbed the acceleration rate of the IPT. However, once the HPC had recovered, the acceleration resumed which led to the IPT reaching a terminal speed close to the maximum allowable speed for the disc. Other aerodynamic factors also come into play. For instance, the LP turbine flow capacity reduction was seen to have a non-negligible impact on curbing overspeed levels.

### 3.2 Model Configuration

The three-spool engine model was set up in WEST by defining the compression side domain and expansion side as shown in Figure 5. The gas path until the NGV outlet was given as an area variation with length replicating that of the actual engine with the domain of the combustor consisting of the combination of the outer and inner regions. Figure 5 starts with the IP and HP compressor. The fan core is not included in this case. Instead, the inlet boundary conditions (PT and TT) were obtained from experimental data and defined transiently as an inlet boundary condition. Due to the absence of the LPC core, the LP spool speed is specified based on recorded data of the event. The discharge of the core stream was done at sea level static conditions and the nozzle throat area is defined.

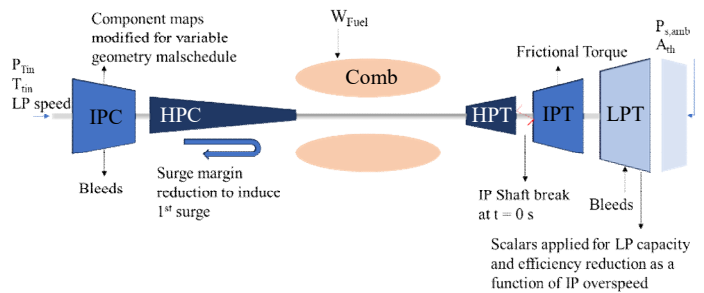


**FIGURE 5:** Engine layout for IP shaft failure modelling

Before any transient simulation is run, the first step is to initialise the conditions within the compression side. This is achieved by fixing fuel flow, bleed values and variable geometry angles to allow the engine (spool speeds) and flow properties

within the gas path to stabilise. Once achieved, this helps set the initial engine state before the transient run.

As WEST does not currently model engine control logic, the fuel-flow is specified transiently as a model input, based on recorded test data. In this case, the dynamic secondary air system was not modelled, as recorded data was available to quantify bleed from the compressor and to the turbine. As extensive work had been done in the past to quantify the frictional torque applied on the IPT following shaft break, the frictional module was not employed in WEST and the variation in torque was defined with time. Shaft break is initiated at time = 0 s by decoupling the shaft on the compressor side from the IP turbine side. Based on the inertia split which was originally defined in the input file and the instantaneous work produced by the turbine, the initial shaft acceleration can be determined. Shortly after shaft break, the compressors surge twice. As vibration-induced surge is not currently modelled, the first surge was started by reducing the surge margin.

The engine was modelled for a few seconds following shaft failure and, in this case, a CFL number of 0.6 was used. Less than 500 nodes were used to simulate the entire compression and combustion domain. Pressure loss from the S-duct was also included.

### 3.2.1 Turbine Overspeed Characteristics

Turbine overspeed within located shaft failure scenarios leads to highly distorted flow conditions downstream due to the skewed velocity triangles. This causes the formation of separation bubbles and recirculation zones which reduces the effective area through the downstream NGV. This affects the downstream turbine in terms of capacity and efficiency reduction which then throttles the overspeeding turbine. In the worst-case scenario, the turbine might reach the free running speed where it produces no work at all. However, in real engines, the terminal speed would be reached before the free running speed due to the addition of other loss mechanisms, such as friction and windage.

In the case of an unlocated shaft failure, the rearward movement of the turbine may cause tip clearances to increase which also leads to a capacity and efficiency reduction. Blade tangling can also occur in such failure scenarios which would directly reduce or stop the acceleration rate of the turbine due to the severe drop capacity.

This validation case aims at modelling a localised IP shaft failure event. To capture the drop in turbine performance, the LP turbine map was modified by applying a capacity reduction factor based on the NRT and work produced by the IPT. These correction factors were obtained from the sponsor in collaboration with Cranfield to generate tables relating incidence angles on the downstream NGV with the NRT and work done by the upstream turbine. The bigger the deviation of the incidence angle from nominal conditions, the bigger the factor on turbine capacity. These factors were obtained analytically through a validated CFD model [18]. Once they were obtained, the direct relationship between NRT and work from the overspeeding turbine was used to obtain a capacity reduction factor on the downstream turbine. Those factors were then applied on to the

LPT to generate new sets of characteristics at varying degrees of overspeed.

Within WEST, the LP turbine flow capacity was directly modified after every iteration by applying those scalar values on the non-dimensional mass flow (WRTP) calculated based on the IP turbine state. This prevents the unnecessary generation of a significant amount of turbine characteristics for the LPT. Having the capacity factor applied on the LP turbine allows the solver to reduce the capacity and efficiency on the IP more efficiently than just applying those factors on both turbine maps.

For this engine, the integration of capacity reduction due to aerodynamic swirl assists in curbing the predicted IP turbine terminal speed by approximately 4% of IP turbine terminal speed.

### 3.2.2 Variable Geometry

The compressor maps in forward flow were obtained from the sponsor. They integrate the effects of variable geometry within a singular compressor map. This means that every single speedline on the map operates at the correct VSV/VIGV schedule. By using combination compressor maps, stacked characteristics are no longer required for every VSV/VIGV angle. However, those maps were created with steady-state simulations in mind. When the engine goes through fast transients or when the control system varies the variable geometry in a way that does not follow normal schedule, the IPC map needs to be modified to match the new angle setting.

During fast transient events, the effects of compressor malschedule become important due to lags induced by the mechanical actuation of the variable geometry. This was found while trying to match the outlet total pressure of the HPC during the IP shaft failure event. Variable geometry malschedule is defined as the difference between the VSV/VIGV angle setting that the engine should be at a specific NRT and the actual geometry setting on the engine. During this shaft failure event, the rapid run down of the IPC causes the closure of the variable geometry according to the engine's schedule. During surge recovery, the operating point may diverge from the normal variable geometry setting. Based on the instantaneous specific NRT within the IPC corrections were applied to the combined IP compressor characteristics to account for the degree to which the VIGV/VSV settings diverge from its nominal schedule.

During the surge event that followed, the IP spool speed on the compression side faced a rapid deceleration. Figure 6 demonstrates the differences between the rig variable geometry settings and the actual engine settings. The degree of malschedule is also plotted on a normalised chart. At around 0.25s, the highest degree of deviation from rig angle is seen. It also coincides with the phase at which the IP compressor starts to recover from the first surge event. A high degree of mismatch between rig and actual engine angle means that the IP is more opened than expected which allows a greater mass flow through the core and delivers a higher pressure ratio.

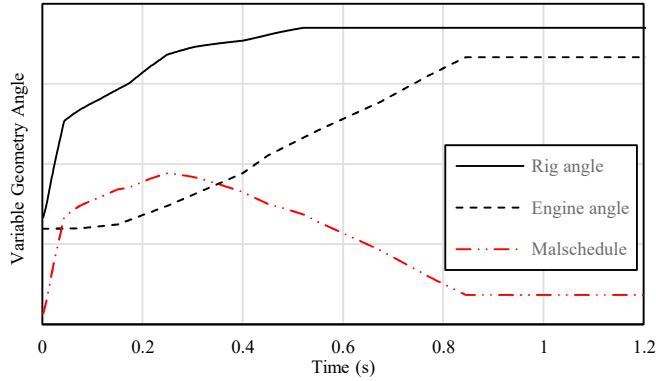


**FIGURE 6:** Normalised VSV/VIGV angle with time for calculating degree of malschedule

Originally, WEST did not account for this effect and during the re-pressurisation process following the first surge cycle, the peak pressure rise achieved at the outlet of the HPC was much lower than that recorded within experimental data. The inability to account for this led to an underestimation of the degree of turbine overspeed by a significant margin specifically due to the rapid cut-off of air being supplied to the turbine side at the most critical phase of rapid acceleration.

To account for this, the IP compressor map was modified by applying factors on WRTP, PR and TR based on the degree of malschedule. To find the instantaneous degree of malschedule, the change in actual IP compressor angle is provided with time and the rig angle or the angle under steady conditions is calculated as a function of NRT. By getting the difference between both, the malschedule angle can then be calculated and factors applied to modify the IP compressor map.

To prevent constant map manipulation for every iteration, the IP compressor map at every interval of 5 degree of malschedule was calculated beforehand. Knowing the instantaneous WRTP, NRT and engine malschedule angle, the PR and TR are obtained through a linear interpolation between the two maps. The translation of the nominal IP compressor map in forward flow operation was performed for a wide range of malschedule angles based on the factors provided by the sponsor. The surge line and recovery line were translated as well and computed for a wide range of angles. Due to the axisymmetric compressor characteristics being dependent on the point of surge, it was also translated based on the new location of the surge line.

An average VSV/VIGV angle was selected for generating the rotating stall and reverse flow characteristics in ACROSS for the IPC. This assumption was made as a means of averaging the effects of variable geometry without the added complexity of having to generate multiple characteristics within those two regions. This proved to have little impact on accurately modeling the total pressure drop at HPC exit and was carried forward for future simulations.

#### 4. VALIDATION RESULTS

The IP shaft failure was modelled numerically by disconnecting the compression side from the turbine. The first surge event was initiated by lowering the surge line. This was due to vibrations following the shaft failure. Fuel flow, handling bleed valve positions and variable geometry were provided as transient boundary conditions replicating that of the recorded test data. The whole engine model was run for a few seconds and compared against measurement data from high-speed probes.

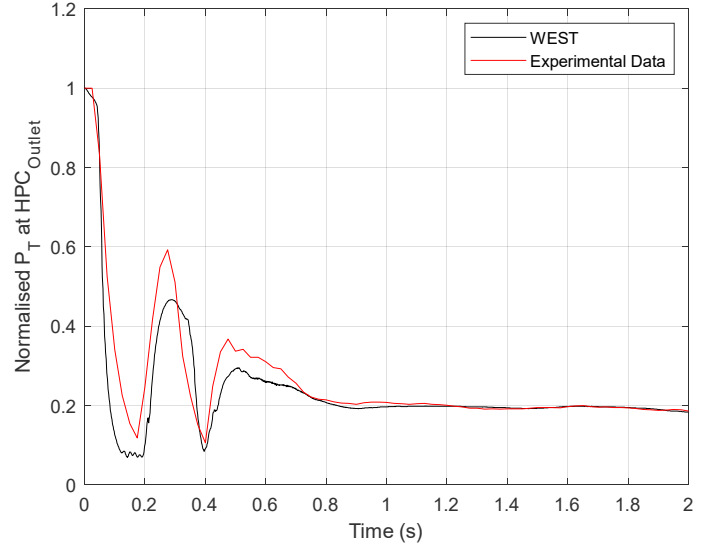


**FIGURE 7:** Total pressure comparison at outlet of HPC between WEST and experimental data

Perhaps the most critical parameter is the total pressure comparison at HPC outlet as shown in figure 7 where a good matching was achieved between WEST and experimental data. The main differences occur during re-pressurisation following the first surge. For instance, at time 0.3s, WEST underpredicts the maximum recovery pressure. There may be several reasons for this. For instance, the rotating stall characteristics were generated at a fixed angle, averaging the VSV/VIGV settings transitioning from opened to closed position. At time = 0.3s, the variable geometry settings are slightly more open than assumed when generating the stall characteristics. Alternatively, it could also be due to the assumptions made regarding the boundary conditions set which are based on recorded engine data. This data could have been shifted or lagged by a small magnitude which translate to those differences seen. Another important trend to note is the rate at which the total pressure at HPC outlet decays. When modelled in WEST the rate of decay is more rapid than that of experimental data. This is due to small measurement lags due to the cord length of the high-speed probes.

Apart from those two main differences, the frequency of the 2 surge cycles is matched and the total pressure profile after 0.8s aligns with the recorded data from the experiment. One other

aspect to note is the surge margin was reduced to induce surge at 0.04s. This was removed following the first de-pressurisation of the compressor. The second surge at time = 0.3s occurred naturally within WEST.

Between times 0.4s and 0.5s, the compressors recover once more and this time, the IP compressor loses power and the PR decays while the HP compressor recovers and operates normally. A third surge cycle is not seen as the new HP speed and conditions across the compressor allow its stable operation at a relatively high power. As the IP compressor is losing power quite rapidly after 0.5s, the total pressure at HPC outlet shows a gradual decrease.

Figure 8 provides the comparison of spool speeds on the compression side between IP and HP against experimental data. The differences this time are slightly larger. With regard to the HP spool speed, the same trend is maintained compared with experimental data. The main differences occur right after shaft failure. Instead of decelerating, the HP spool speed accelerates from time = 0s to 0.1s while the actual shaft speed decelerates slightly. This could be attributed to the differences in heat flux within the combustor. During surge, on both compressors, the operating point transitions to the stall region before going into reverse flow. For this, a fixed combustion efficiency is set which might not be constant or accurate. This could lead to the HP shaft momentarily accelerating. However, throughout the rest of the simulation, the same profile is maintained, and the error is kept relatively constant.

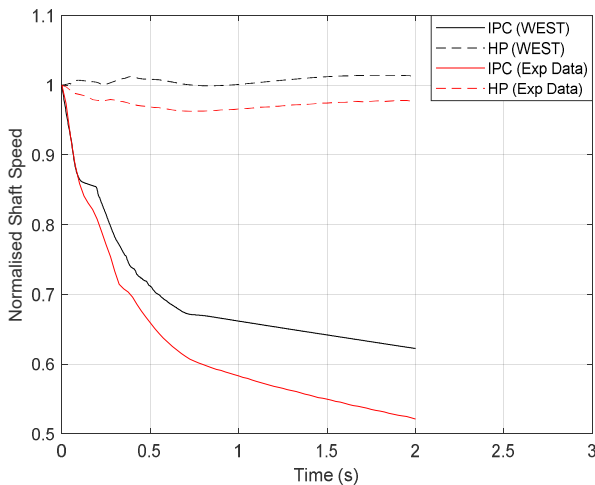


**FIGURE 8:** Normalised shaft speed comparison of the IPC (solid line) and HP (dotted line) between WEST (black) and experimental data (red).

The IPC spool shows an opposite trend. In this case, the initial deceleration rate is maintained almost perfectly until approximately 0.2s. It is where the operating point is within the stall region. Due to small differences in variable geometry assumptions within the stall region, the HP spool speed plateaus while the experimental data shows only a slight reduction in deceleration rate. It is interesting to note that the differences seen in Figure 7 increase the divergence in IPC spool speed

prediction. Once the recovery peaks are crossed, the trend in deceleration is maintained. Those differences, however, do not have a big impact on the total pressure at the HPC outlet which is the main factor for predicting IPT overspeed.

The whole purpose of this engine modelling tool is to achieve an accurate prediction of levels of turbine overspeed following a shaft break. In this case, the IP turbine spool speed is plotted against time and compared against the actual IPT spool speed derived from vibration measurements. This is shown in figure 9. There are a couple of interesting features to note on this figure. The first one is the beneficial aspect of surge events in curbing the initial acceleration rate of the IP turbine. This is seen at around time = 0.2s and 0.4s. Within time = 0.1s following shaft break, the IP turbine speed sees the sharpest rise in speed. However, the initial acceleration is immediately stopped by the first surge event which reduces air supplied to the core drastically. A surge happening post-shaft failure is highly beneficial since it can reduce the high acceleration rate before the control logic has time to respond. This is also seen when looking at recorded experimental data, especially at around time = 0.2s where the first surge occurs.

The IP turbine overspeed predicted by WEST closely follows that of experimental data up until approximately time = 1s. Following this, WEST tends to overpredict slightly. The deviation increases and reaches 4% at time = 2s which is also the location at which the IP turbine speed reached its maximum. A factor that could have contributed to this overprediction is the end-load or torque applied on the turbine end due to friction. Assumptions were made to determine this frictional torque which has some degree of uncertainty. This led to WEST overpredicting the IP turbine terminal speed by a margin of 4% as shown in Figure 9.

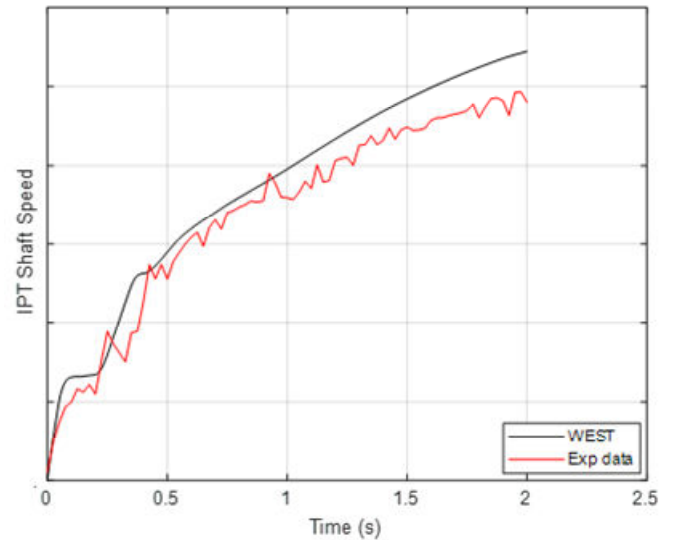


**FIGURE 9:** Shaft speed comparison of the IP Turbine between WEST (black) and experimental data (red).

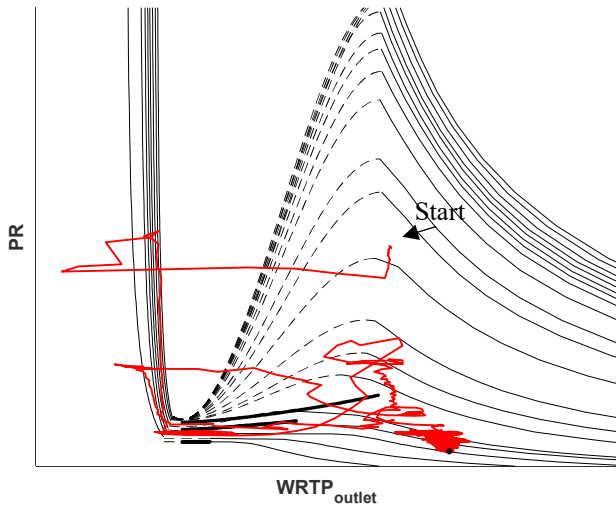


**FIGURE 10:** Trace of IP shaft failure event on IPC map

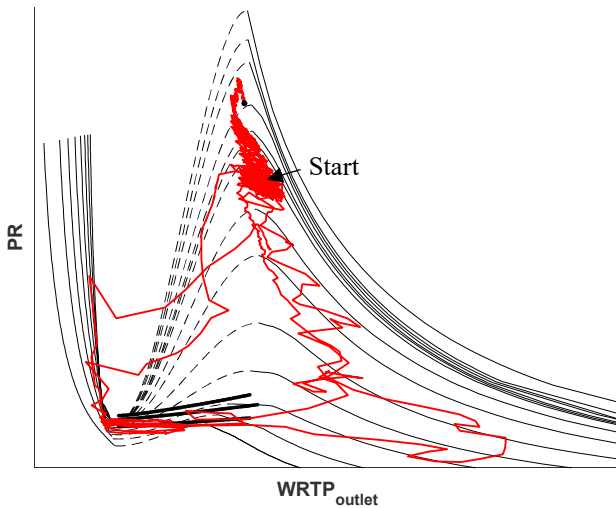


**FIGURE 11:** Trace of IP shaft failure event on HPC map

Figures 10 and 11 show the movement of the operating point on the complete characteristics for both the IP and HP compressors. On both compressor maps, this is shown by a red line with the final steady point reached being denoted by a black dot. The IP compressor operating point first goes down the NRT lines following shaft break before moving rapidly across the stall region before reaching the reverse flow region. It is seen by the upper red trace on the map. A slight deviation is seen where the operating point moves off the reverse flow lines before settling down on a specific NRT line within the reverse flow region. Then de-pressurisation happens and it is seen by the nearly vertical red line going down the reverse flow characteristics. This transition, until the operating point reaches the rotating stall line occurs very rapidly. Once the stall region is reached, it takes approximately time = 0.1s to traverse the stall region and go back into the forward flow region of the map.

This phase is usually slower than flow reversal because of the need to re-establish some orderly flow conditions within both compressors. Flow reversal is extremely chaotic and is characterised by the complete breakdown of flow within the compressors. This happens very rapidly. However, once the combustor reaches a low pressure, which is enough to sustain recovery, the flow needs time to reattach to the compressor blades. For multi-stage compressors and when considering more than one compressor, the air flowing within both compressors needs to be synchronised for both to recover at the same time. For instance, when the operating point crosses into the stall region from the reverse flow region, a cyclic phenomenon happens where the operating point goes up and down the rotating stall characteristics until both the HPC and IPC synchronise, and recovery occurs. This is seen more predominantly within the HPC by the loops onto the rotating stall lines.

Once both compressors have recovered, the operating point usually goes up the forward flow characteristics due to re-pressurisation. The first recovery within the HPC was particularly strong and is shown by the red trace on the right-hand corner of the map.

One interesting aspect to note is that despite the failure of the IP shaft, the trace of the operating point fluctuates less than that of the HPC. Within the IPC, the two surge cycles form relatively smooth loops on the compressor map whereas the HPC is more sensitive and will show more fluctuations. It is also seen during the transition from the stall to the forward flow region. Additionally, the HPC tends to go through more surge cycles than the IPC.

The second surge cycle is seen on both charts by the second loop where the operating point goes back to the reverse flow region. Within the IPC, it is at a lower NRT line. Hence the loop is smaller. Within the HPC, the operating point starts from a higher PR and NRT line before surging once more. This time, the operating point reaches a lower maximum PR value when in reverse flow as compared to the first cycle.

Once both compressors have recovered again, the IPC is seen to settle onto a low pressure ratio value, at the right-hand corner of the map while the operating point on the HPC map goes to a very high pressure ratio value before dropping off. Before the operating points converges, they fluctuate on the forward flow characteristics. As the mass flow probes could not accurately record flow in the negative direction, those results could not be accurately compared against experimental data. Instead, the PR across both compressors was compared and these are shown in figures 12 and 13.

In this case, as well a similar trend is achieved. The experimental data is less sensitive to transients due to measurement lags. Up until time = 0.5s, the trend predicted by WEST is similar to experimental data. The predictions start to diverge later with WEST slightly underpredicting the PR within the IP and overpredicting that of the HPC. Nevertheless, the differences cancel out and the total pressure at the outlet of the HPC matches well (refer to Figure 7). These differences are likely to stem from the IPC compressor characteristics which may not reflect sub-idle conditions accurately. Due to the new

compressor matching, the HPC compensates by producing a higher PR. Nevertheless, the same trend as experimental data is seen. Also, between approximately time = 0.5s and 1s, the results fluctuate significantly. This is mostly apparent from the HPC PR. When looking at the map, it is seen that the operating point on the HPC characteristics is very close to the surge line. This is why there is a region of high instability as the HP compressor is attempting to settle on a specific speedline. The fluctuations within the HPC also lead to instabilities of lower magnitude within the IPC. The instabilities after time = 0.5s were also captured within the recorded engine data. Despite the differences following the two surge cycles, the trend was correctly captured.

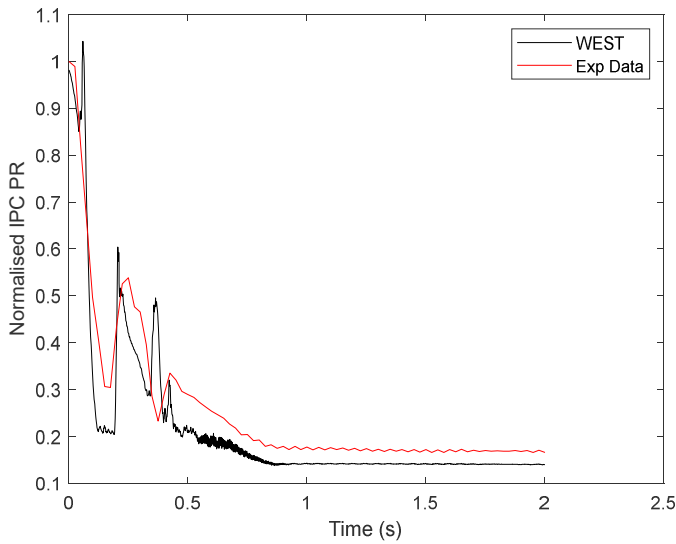


**FIGURE 12:** Comparison of normalised IPC PR with time between WEST and experimental results

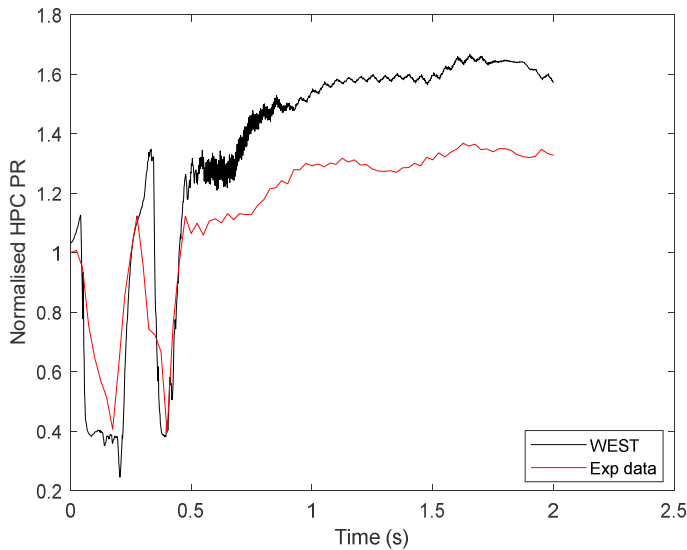


**FIGURE 13:** Comparison of normalised HPC PR with time between WEST and experimental results

## 5. CONCLUSION

This paper provides an in-depth summary of the first successfully modelled IP shaft failure event on a 3-spool engine which demonstrated good matching against real engine data. One of the main challenges was to have a thorough understanding of the aero and mechanical effects on the engine before being able to fully model those effects within a 1D engine model. The effects of variable geometry malschedule within the forward flow region allowed the model to replicate the peak in total pressure at HPC outlet of the second surge cycle. The integration of capacity reduction due to turbine overspeed allowed for better matching of the IPT terminal speed prediction by a margin of 4% against recorded test data. However, the key breakthrough was the successful development of a 1D compressor model capable of high power surge and stall modelling, even when integrated into a whole engine system. This allowed the first ever recorded and successful validation of a whole engine model following a shaft failure.

The final comparison demonstrated that the engine model in WEST replicated the very transient behaviour of a localised shaft failure event. Following the failure across the IP shaft, a surge event occurred due to vibration. This led to the drop in total pressure at the outlet of the HPC which reduced the air flow to the turbine. The drop in pressure was enough to facilitate recovery but due to the mismatch in both IPC and HPC, an other surge cycle occurred again. Unfortunately, the IPC spool speed had reduced by enough to prevent its recovery and it ran down while the HPC recovered. This trend was replicated in WEST and demonstrated within the results. The main indicator to match was the IP turbine overspeed which was matched within a 4% error margin. The terminal speed reached was just below the disk break limit set at the design stage. This would be the main parameter that could be used in the future to determine whether a preliminary engine design fails to meet certification requirements before going to testing.

The successful development of WEST has demonstrated its capability to accurately predict total pressure at the HPC outlet, a critical parameter for assessing turbine overspeed events. This achievement highlights the model's effectiveness in providing physics-based insights without the need for prior knowledge of shaft break events. This could lead to lighter disks which would be optimally designed following a better understanding of turbine terminal speed and hence improved determination of safety factors. By enabling a deeper understanding and improved prediction of terminal speeds early in the design process, WEST contributes to enhanced operability, reduced weight, and greater competitiveness in new engine designs. In the future, this study could lead to improved engine and component level understanding of surge and shaft failure events and help set up improved design methods.

## ACKNOWLEDGEMENTS

The authors would like to express their gratitude to Rolls-Royce plc. for funding this research and for permission to publish this research. Special thanks go to Mr. Richard Tunstall

and Mr Arthur Rowe for their constant support and valuable guidance.

## REFERENCES

- [1] ATSB, "In-flight uncontained engine failure A380-842, VH-OQA," 2013.
- [2] European Aviation Safety Agency, "Certification Memorandum - Turbine Over-speed Resulting from Shaft Failure," no. July, pp. 1–6, 2012.
- [3] E. M. Greitzer, "Some aerodynamic problems of aircraft engines: Fifty years after -the 2007 IGTI scholar lecture-," *J Turbomach*, vol. 131, no. 3, 2009, doi: 10.1115/1.2992515.
- [4] L. Pawsey, D. J. Rajendran, and V. Pachidis, "Characterisation of turbine behaviour for an engine overspeed prediction model," *Aerosp Sci Technol*, vol. 73, pp. 10–18, 2018, doi: 10.1016/j.ast.2017.11.037.
- [5] I. Eryilmaz, "on the Modelling of the Overspeed Performance of a Single Stage Turbine," *Proceedings of the ASME Turbo Expo*, vol. 13B, pp. 1–8, 2023, doi: 10.1115/GT2023-103487.
- [6] V. P. C. Calcagni, L. Gallar, "Development of a one dimensional dynamic gas turbine secondary air system model - Part II: Assembly and validation of a complete network," *Asme Turbo Expo 2009*, 2009.
- [7] L. Gallar, C. Calcagni, V. Pachidis, and P. Pilidis, "Model – Part I: Tool Components Development and Validation," *Asme & Gt*, 2009.
- [8] I. Eryilmaz and V. Pachidis, "Turbine thermomechanical modelling during excessive axial movement and overspeed," *Aeronautical Journal*, vol. 123, no. 1260, pp. 248–264, 2019, doi: 10.1017/aer.2018.162.
- [9] I. Eryilmaz and V. Pachidis, "A design approach for controlled blade-off in overspeeding turbines," *Eng Fail Anal*, vol. 138, no. March, p. 106323, 2022, doi: 10.1016/j.engfailanal.2022.106323.
- [10] I. Eryilmaz and V. Pachidis, "A design approach for controlled blade-off in overspeeding turbines," *Eng Fail Anal*, vol. 138, no. March, p. 106323, 2022, doi: 10.1016/j.engfailanal.2022.106323.
- [11] R. Mohanraj, Y. Neumeier, and B. T. Zinn, "Characteristic-based treatment of source terms in Euler equations for Roe scheme," *AIAA journal*, vol. 37, no. 4, pp. 417–424, 1999, doi: 10.2514/2.749.
- [12] L. Di Mare, "Modelling of variable gas properties and compositions in turbomachinery flow simulations," *Proceedings of the ASME Turbo Expo*, vol. 6, no. PART C, pp. 2523–2532, 2008, doi: 10.1115/GT2008-51085.
- [13] G. Fernandez and B. Larrouturou, "Hyperbolic Schemes for Multi-Component Euler Equations BT - Nonlinear Hyperbolic Equations — Theory, Computation Methods, and Applications: Proceedings of the Second International Conference on Nonlinear Hyperbolic Problems, Aachen, FRG, March 14 to 18,," J. Ballmann and R. Jeltsch, Eds., Wiesbaden: Vieweg+Teubner Verlag, 1989, pp. 128–138. doi: 10.1007/978-3-322-87869-4\_13.
- [14] M. Righi, "THREE-DIMENSIONAL THROUGH-FLOW MODELLING OF AXIAL COMPRESSOR ROTATING STALL AND SURGE," Cranfield University, 2018.
- [15] S. Kissoon, M. Righi, L. Pawsey, V. Pachidis, R. Tunstall, and I. Roumeliotis, "A Characteristic-Based 1D Axial Compressor Model for Stall and Surge Simulations," Jun. 26, 2023. doi: 10.1115/GT2023-100709.
- [16] NASA Glenn Research Center, "Nasa computer program for calculating of the chemical equilibrium with applications," *Cleveland Ohio*.
- [17] M. Righi, V. Pachidis, L. Könözy, T. Giersch, and S. Schrape, "Experimental validation of a three-dimensional through-flow model for high-speed compressor surge," *Aerosp Sci Technol*, vol. 128, p. 107775, 2022, doi: https://doi.org/10.1016/j.ast.2022.107775.
- [18] L. Pawsey, "Shaft failure and overspeed modelling: Systems integration and overall engine response.," Cranfield University, 2018.



**DS**SPACE

<https://dspace.org/>

## **Development and validation of a transient gas turbine simulation tool for shaft-failure modelling**

**Kissoon, Sajal; Righi, Mauro; Pawsey, Lucas; Pachidis, Vassilios; Roumeliotis, Ioannis**

**2025-06-16**

American Society of Mechanical Engineers (ASME)

<https://doi.org/10.1115/gt2025-151418>

Kissoon S, Righi M, Pawsey L, et al., (2025) Development and validation of a transient gas turbine simulation tool for shaft-failure modelling. In: ASME Turbo Expo 2025: Turbomachinery Technical Conference and Exposition, 16-20 June 2025, Memphis, USA, Volume 1: Aircraft Engine; Ceramics and Ceramic Composites, Paper number GT2025-151418