Validation of an Aeroengine Carcass Finite Element Model by Means of Computational Model Updating based on Static Stiffness Testing

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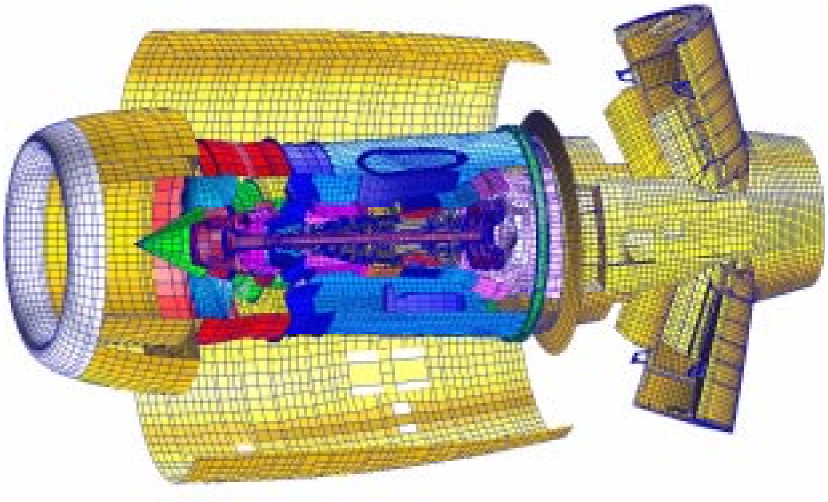

Fig. 1: BR715 Whole Engine Mechanical Model

Abstract

Whole engine mechanical finite element models (WEM) consisting of more than 400.000 degrees of freedom (DOF) perform important functions within the design and certification processes of modern aeroengines.

The application for the WEM ranges from design parameter studies during the preliminary design phase over determination of design loads up to production support during pass off tests and product support for the aircraft manufacturer during the life of an engine. In addition to the variety of tasks in the field of dynamic calculations (rotor dynamics, blade failures, bird ingestion etc.), the field of static calculations represents a wide range of applications.

During certification of the BR710 aeroengine, static stiffness tests were used (as well as other tests) to validate the BR710 WEM. Static stiffness tests have the advantage to enforce special deformations that are difficult to excite or to measure under modal testing, which is frequently used for model validation purpose. Another advantage over modal testing with regard to computational updating is that the measured deformations are independent of the mass parameter.

After completion of the testing and the correlation processes between experimental and analytical data, computational updating was developed and successfully applied. This paper presents the finite element model, the testing procedure, and the updating method as well as results of the updating processes.

The Whole Engine Model

The BR700 series aeroengines are complex technical systems which have to comply with highest requirements regarding reliability, production costs, weight, fuel consumption, noise and emissions and other important criteria.

The optimization of the aircraft installed turbomachinerie's structural behavior has a considerable influence on the performance of the whole aircraft. At BMW Rolls-Royce AeroEngines the mechanical simulation of this complex system within it's flight and landing envelopes is performed under application of the WEM, a MSC/NASTRAN finite element model.

These simulations are significant for the determination of internal load and deformation distributions under static and dynamic loading conditions. Quasi static loads are applied to simulate e.g. thrust, maneuver and landing conditions. Dynamic loads cover nonlinear transient conditions like bird impact and blade failures. Engine dynamics under windmilling, determination of imbalance induced carcass vibrations and critical speeds are part of the rotordynamic analyses required in an engine certification process.

Further applications of the WEM are represented by optimization of design parameters like e.g. tip clearances which directly influence the efficiency of the turbomachinery.

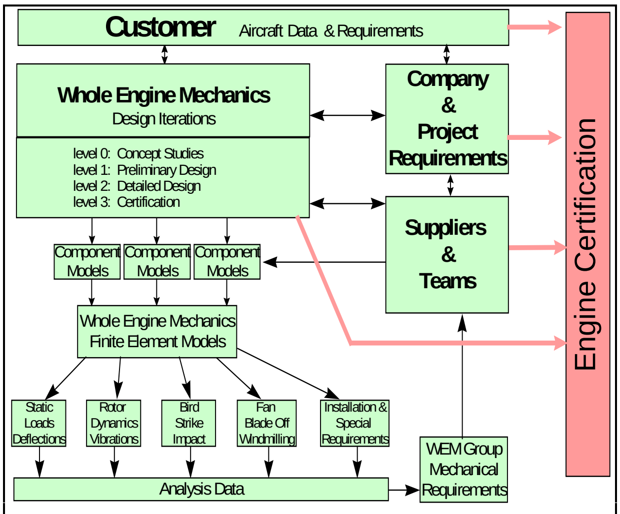

Fig. 2: Whole Engine Mechanics Design Process

The WEM is split into the following subcomponents: engine carcass, hp-rotor, lp-rotor and engine mountings. Fig. 1 shows a cutaway view of the 3-D finite element model consisting of over 0.5 million degrees of freedom.

Fig. 2 shows the complex correlations between input and output of the WEM. One example is section loads derived from WEM analyses which are used as boundary loads for detailed component models. There is hardly any area within engine development that has no relations to the WEM. Additionally the WEM is used for in service product support and for production pass off support.

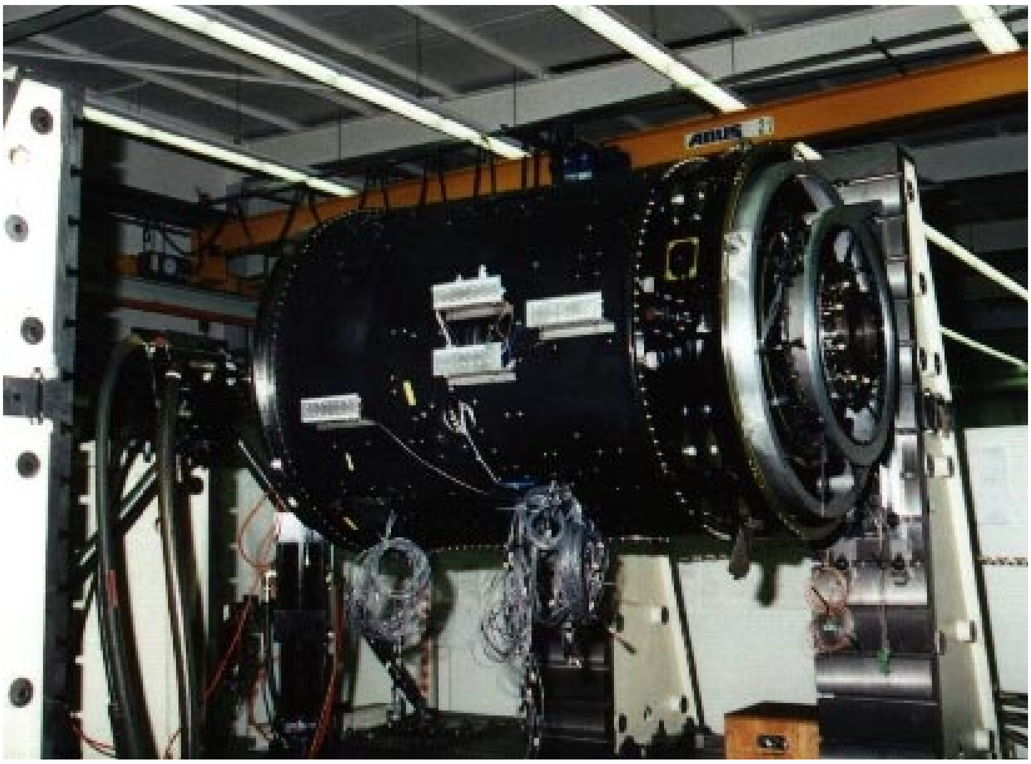

Fig. 3: Carcass Stiffness Test Rig Mounted on Base Plate

It is obvious that the accuracy of the model predictions is essential for the ambitious tasks the WEM has to fulfill in this optimization process. Additionally, in order to reduce the amount of destructive testing, modern aeroengine manufacturers intend to supersede such lead time and cost producing approach by simulations with validated numerical models.

Experimental Validation

Due to the extensive application of the WEM computational requirements should be aimed to be minimized. A large degree of detailed modeling requires large computational resources. In order to minimize this, idealizations have to be made during the modeling process.

These idealizations - depending on the application - can include neglecting friction and linearized force-deflection material properties as well as linear deflection behavior of the structure, also at casing flanges, i.e. flanges deform under tension loads analog to compression loads.

Such an idealized analysis model must be calibrated by suitable validation methods in order to generate results with sufficient accuracy. This requires structural testing using original engine parts. Among these are full engine running tests, but also components tests.

The carcass stiffness test described in this paper was aimed at examining the total flexibility of the significant load carrying casing assembly in the loadpath between rotor bearings and engine-to-aircraft attachment points.

The test structure was statically determinate fixed to a very stiff mounting plate and loaded via hydraulic jacks, as represented in fig. 3. At the rotor bearings and axially midwise at the core casings, between hp-compressor and combustion chamber outer casing (core bending loadcase), radial loads were applied in several directions. Additionally the axial bearings were loaded. The magnitude of the applied forces was in the range between approx. 40 to 75 kN. In total 57 different loadcases were applied.

A stiff unloaded reference tube was attached via a slackless spherical bearing at the intermediate casing and a knife edged simple support at the rear bearing support structure. The design is depicted in fig. 4.

The maximum magnitude of the measured radial casing deflections was between 0.1 mm and 1.5 mm, depending on the axial position of the measurement plane, with a diameter around 1200 mm, which put very high requirements on the accuracy of the measurements as well as the design and build of the testrig.

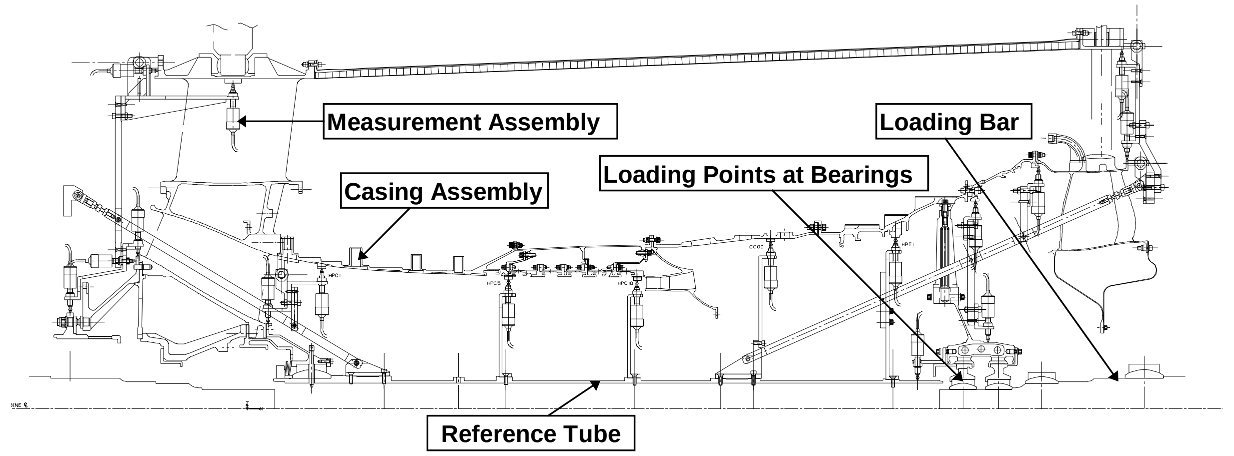

Fig. 4: Section View of Casing and Measurement Assembly for Stiffness Test

During testing it was found that the structure exhibited considerable nonlinear behavior after first time application of each new loadcase. This could be explained by settling of the structure. It was therefore decided to apply the loading 4 times for each loadcase. During the repeats the structure showed a linear behavior. The repeatability of the LVDT measurements between the second and the fourth loading was approx. 0.01 mm.

Fig. 5: View into Core Showing 5 LVDTs of 8 Total on Reference Ring to Measure Ovalisation of the Casing

The casing deflections were measured via 114 inductive linear variable displacements transducers (LVDT) relative to the reference tube, as can be seen on figs. 4 and 5. In addition at several locations casing stresses and link forces were detected by use of strain gauges.

Complete hysteresis loops were obtained by applying each loading cycle in 16 steps, 11 steps up to the maximum load and 5 steps of unloading. The slope between step 5 and step 10 on the force/deflection function was then taken as approximation for comparison with the linear stiffness the analytical model is based on.

Initial Correlation Results

Once settling was eliminated from the test structure it's major part showed a clear linear behavior, as depicted in fig. 6. The only exception to this was found in the loadpath between rear bearing support structure and rear mount ring. At this location two distinct stiffness lines can clearly be observed: the stiffness decreases from 30% onwards of the maximum load by 50%, as shown in fig. 7.

It was found that the differences of the maximum LVDT predictions. This is not surprising for the mounting links, because of their statical determination, but for the A-frames it confirms the accuracy of the modeled loadsplit between core casings and bypass duct.

Deviations between measured and predicted LVDT readings were generally small but locally in worst cases showed values of up to around 60%. The test-setup model (see fig. 8) consists of approx. 40.000 DOF. Measurements were picked up at 114 DOF. The proportion between the number of

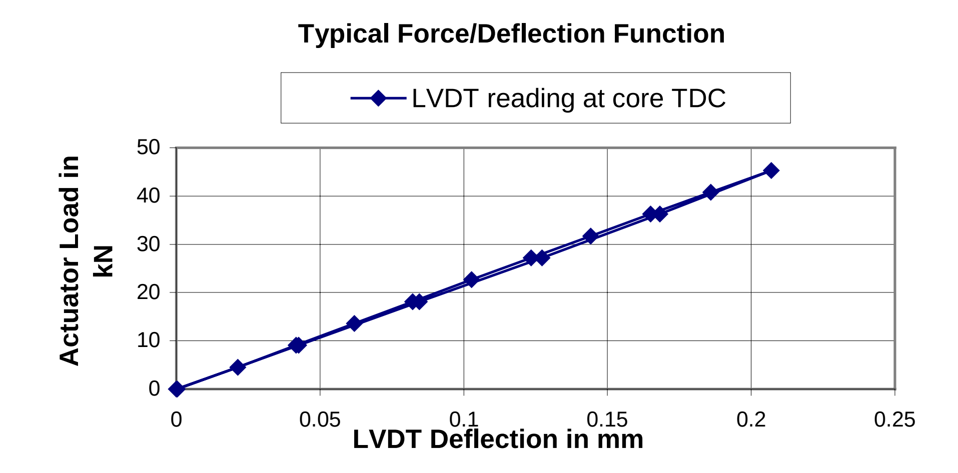

Fig. 6: Typical Force/Deflection Function

measurements between 2 loadcases in 180 degree opposite directions could vary between 1% to 10%. This was accounted for during updating by averaging the known and unknown variables and the fact that general deviations are small made it unreasonable to update the model manually. It was therefore decided to develop a

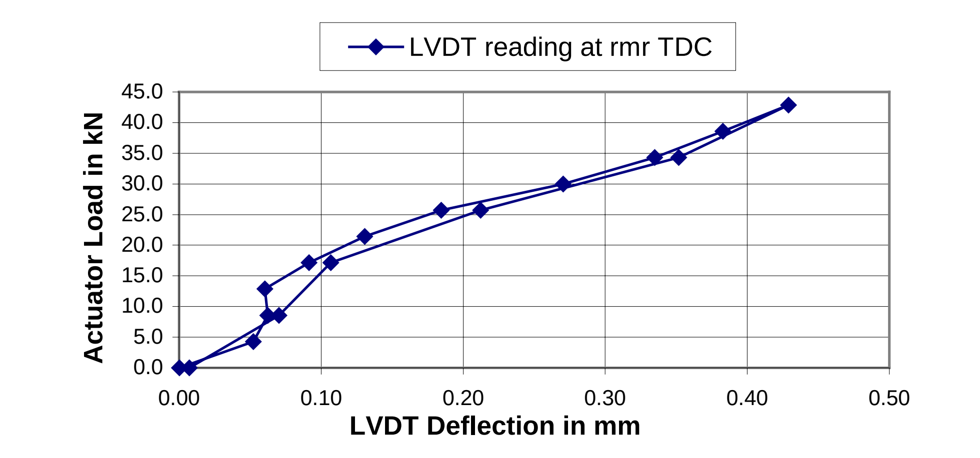

Fig. 7: Nontypical Force/Deflection Function

opposite loadcases.

All of this leaves a general small scatter that can be tolerated for the linear model. Expectations that effects like heeling of flanges would provide a considerable degree of nonlinearity, especially for the large magnitude of the applied loads, were not confirmed.

Generally the initial model predictions show an acceptable level of agreement with the test data. The measurements of mounting link forces and A-frame forces, which could precisely be determined, are within +/- 5% of the software tool assisting in the correlation and updating processes. This was carried out by DDS and implemented into the existing FEMTOOLS software.

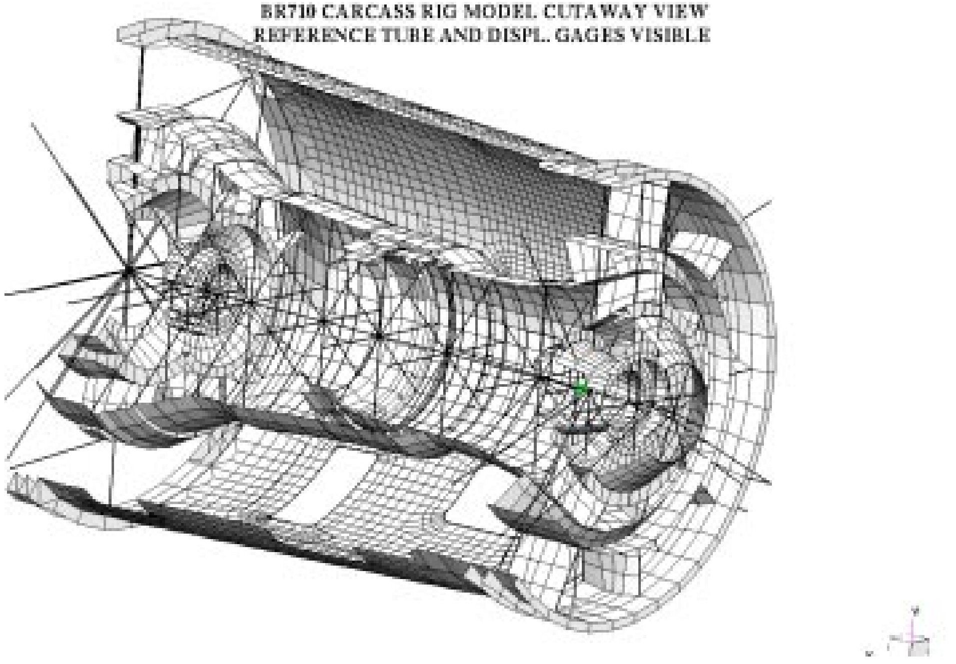

Fig. 8: Section View of the MSC/NASTRAN FE Carcass Stiffness Test Rig Model

Model Updating using Static Displacement Data

The finite element (FE) method has matured over the past three decades to a point where design, meshing, analysis and postprocessing are highly integrated and automated. However, taking into account the higher complexity of FE models like the WEM, requires the analyst to know and understand the limitations of the FE model, and to examine the results critically.

By systematically comparing the results from analytical and experimental analysis techniques, FE models can be validated and updated so that they can be used with more confidence in further analysis. FE model updating using dynamic test data has the advantage that in one analysis, information on stiffness, mass and damping is included. The drawback of this approach is that it is difficult to decide on the updating parameters to use. One way to overcome this is to separately validate mass and stiffness modeling prior to dynamic analysis. The recommended validation procedure is therefore a sequence of updates, in which mass, stiffness and force parameters are validated and updated, separately or simultaneously. This is illustrated in figure 9.

Updating stiffness modeling using static displacement tests involves minimizing the following error function [1]:

$$E = \Delta\varepsilon^t C_R \Delta\varepsilon + \Delta p^t C_p \Delta p \qquad (1)$$

with

$$\Delta\varepsilon = \frac{\partial\{U^j\}}{\partial p_k}\,\Delta p = S_{jk}\;\Delta p = [S]\Delta p \qquad (2)$$

where

$$\Delta\varepsilon = \{U^j_{ei}\} - \{U^j_{ai}\} \qquad (3)$$

is the difference between experimental (index e) and analytical (index a) static displacements, at the measured DOFs i, for a number of static load cases j, and,

$$\Delta p = p_{ku} - p_{ko} \qquad (4)$$

is the difference between updated (index u) and originally estimated (index o) parameter values.

From equations 1-4, the updated parameter values p_{ku} are obtained as

$$\Delta p = \left[[C_P] + [S]^t [C_R][S]\right]^{-1} [S]^t [C_R]\Delta\varepsilon \qquad (5)$$

C_R and C_p are respectively diagonal weighting matrices for the selected updating targets (static displacements) values and for the updating parameter values. Each weighting value is a measure of confidence in the experimental reference value, respectively in the original parameter estimation.

To compute the displacement sensitivity coefficients S_{jk}, the equation of static equilibrium is derived with respect to the updating parameters p_k:

$$[K]\{U\} = \{F\} \qquad (6)$$

$$[K]\frac{\partial\{U\}}{\partial p_k} = \frac{\partial\{F\}}{\partial p_k} - \frac{\partial[K]}{\partial p_k}\{U\} \qquad (7)$$

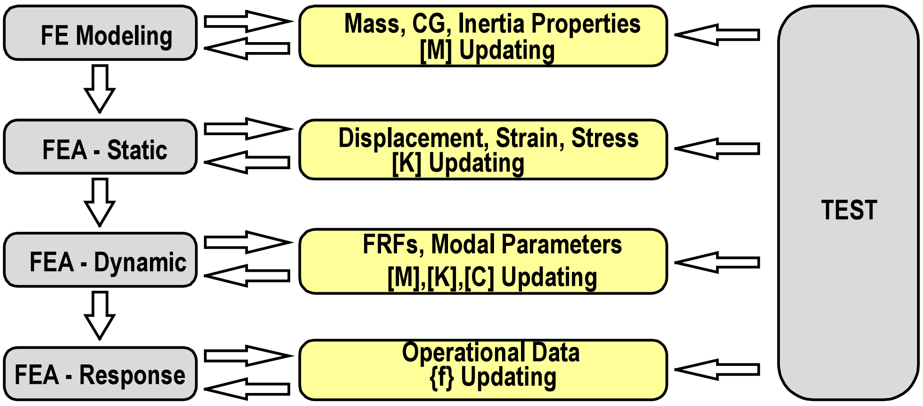

Fig. 9 Finite Element Model Analysis and Validation Phases

$\frac{\partial\{U\}}{\partial p_k}$ are the unknown displacements sensitivity coefficients.

$\frac{\partial\{F\}}{\partial p_k} = 0$ if the external loads are not dependent on structural properties.

$\frac{\partial[K]}{\partial p_k}$ is the derivative of the system stiffness matrix with respect to the parameter p_k. This derivative can be obtained via a differential or finite difference formulation with small parameter perturbation :

$$\frac{\partial[K]}{\partial p_k} \approx \frac{[K(p_k + \Delta p_k)] - [K(p_k)]}{\Delta p_k} \qquad (8)$$

To compute the displacement sensitivities, equation (7) needs to be solved. This solution is computationally identical to solving equation (6), and can be done in MSC/NASTRAN [2] by supplying $-\frac{\partial[K]}{\partial p_k}\{U\}$ as the load vector. The advantage of this formulation is that sensitivity coefficients are obtained for all DOFs of the FE model and can easily be reduced to only include the measurement DOFs.

Practical Application

The FEMtools software [3] was adapted to support updating using static displacement based on the theoretical principles described in the previous section. This required development of data interfaces, correlation analysis tools, an MSC/NASTRAN driver to perform static analysis (to compute displacement sensitivities and to compute displacements after updating the FE model), and a parameter estimation algorithm (equation 5). The procedure is outlined in figure 10.

FEMtools acts like a pre- and postprocessor to MSC/NASTRAN to perform all database management and analysis except the static solution phase. The user benefits from a dedicated graphical environment to support him in the task of selecting updating targets (responses) and parameters which relies mainly on engineering judgement and insight in the assumptions and approximations during the modeling phase.

To quantify the correlation between predicted analytical results (FEA) and test, the following criterion is used:

$$DAC(U_a, U_e) = \frac{\left|\left(\{U_a\}^t\{U_e\}\right)\right|^2}{\left(\{U_a\}^t\{U_a\}\right)\left(\{U_e\}^t\{U_e\}\right)} \qquad (9)$$

The Displacement Assurance Criterion (DAC) scales and relates displacement shapes to yield values between 0 and 1. A DAC value of 1 corresponds with two displacement vectors that are completely identical.

Because first-order sensitivities are used in equation 7, solution of equation 2 is not a one-step operation but requires an iterative procedure like shown in figure 10. It is required to keep parameter changes small with each iteration in order to prevent error function from oscillating or diverging.

Although displacement sensitivity analysis is available in MSC/NASTRAN using SOL 200, using the general sensitivity analysis procedure described above with the FEMtools software, offers the following advantages:

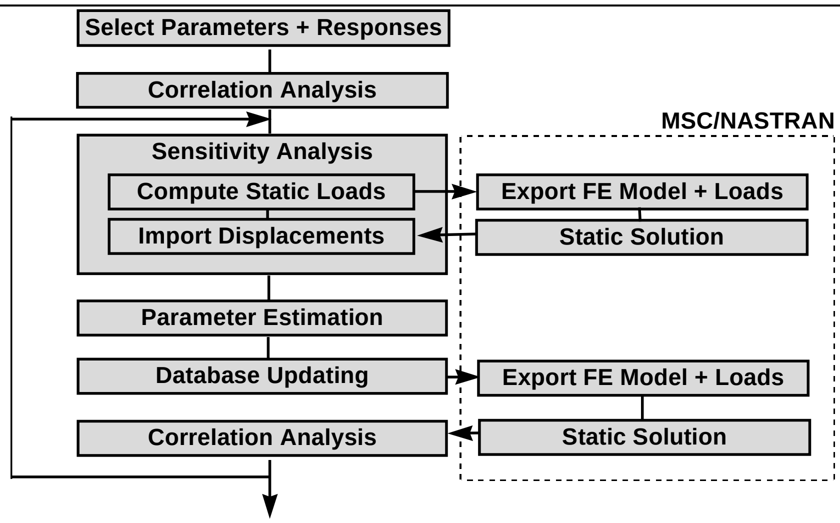

Fig. 10 Finite Element Model Analysis and Validation Phases

- Flexible selection of parameter types (material, geometry, boundary conditions, ...) and response types (absolute or relative displacements, strain or stress). Parameters can be selected at the element level or be assigned to groups of elements.
- MSC/NASTRAN or any other FEA software is used as the solver for static analysis.

Updating Results and Conclusions

Current updating techniques do not allow a completely automated updating for the size of WEM models. The number of possible mathematically correct solutions without physical meaning is enormous.

The updating software guides the user with a number of possible options where changing the model would be beneficial, but the experience and engineering judgement of the user is an indispensible condition to successful model validation [4].

Typically the updating process requires large numbers of analyses and the results can depend on the chosen updating parameters. Large changes of a parameter can be required but also can point to it´s insensitivity. Experience shows that it may be necessary to bring a parameter near it´s required value to make it sensitive.

In this case validation of individual components prior to the updating of the assembly was applied [5]. This was done for major components by free-free modal testing. In all initial correlations from dynamic analysis the initial component models were not as stiff as required from the measurements, which is a common phenomenon for quad4 shell element structure modeling, which the model consists of to a large extent.

It was found that it was necessary to increase the value controlling the inplane rotational stiffness (variable „k6rot“ in MSC/NASTRAN) to E+06, (default value E+02) for the dynamic analyses, but for the static analysis cases a value of E+02 proved to be correct.

Updating of the model would have been impossible without prior validation of the major single components. This approach narrows the number of insecure parameters considerably and increases the confidence in variable selection and thus speeds up the validation process.

Due to the variation in the repeatability, it was decided to use only measurement values bigger than 0.1 mm for the static updating process.

The updating method developed allowed updating of physical parameters like Young's moduli and shell thicknesses. In the first step only Young's moduli were updated and in some cases led to unrealistic large changes meaning that these substitute parameters had to be exchanged for true design parameters.

Fig. 11 shows the correlation tracking for the core bending loadcases.

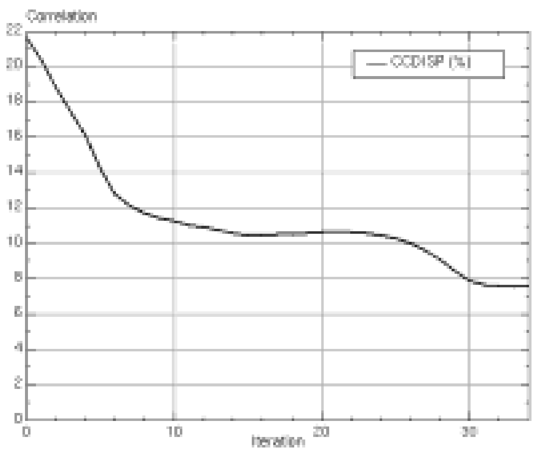

Fig. 11: Convergence of the correlation coefficient

The correlation coefficient „ccdisp“ describes the average value of weighted relative differences between predicted and reference displacements. It should be zero in case of perfect correlation and is used as the objective function that is minimised using an iterative procedure.

The starting value of 21.6% average weighted relative differences could be reduced to 7.5%. Further improvement below this value could not be achieved. It was found that the small variations in the measurement data as well as the small amount of measured nonlinearity was the reason for this. The final improvement is a best fit for matching the test data.

The design, build and performance of the test as well as the methods development and the correlation and updating of the model took large efforts in manpower. The statically determinate mounting of the BR710 engine made this approach for model validation feasible. The initial correlation between model predictions and test data was already within expected deviation tolerance but still could be improved further up to limits set by the measurement technique. Single component model improvements by experimental modal analysis approach could be verified by the carcass static stiffness test. The updating method allowed a physical interpretation of the parameter changes.

Settling of the structure under large static loads was quantified and can be used in further studies to analyse it’s structural damping capabilities. The validation process fully confirmed the confidence in the WEM model quality and predictions.

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ABBREVATIONS, SYMBOLS and INDICES

WEM	whole engine mechanical finite element model
lp-rotor	low pressure rotor
hp-rotor	high pressure rotor
LVDT	linear variable displacement transducer
DOF	degree of freedom
TDC	top dead center
$[C_P]$	Weighting matrix for the parameters
$[C_R]$	Weighting matrix for the responses
E	Error function
{F}	Static force vector
[K]	Stiffness matrix
P	Parameter
R	Response
[S]	Sensitivity matrix
{U}	Static displacement vector
Δ	Finite difference
∂	Partial derivative
ε	Convergence margin; Tolerance margin
A	Analytical value
E	Experimental value
I	DOF number
J	Load case number
K	Parameter number
O	Original value
U	Updated value
$[]^t$	Transposed matrix
$[]^{-1}$	Inverse of a matrix