Prediction of Transient Loads and Perforation of Engine Casing During Blade-Off Event of Fan Rotor Assembly

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ABSTRACT
A major risk in modern military engine is the failure of aero engine Fan and Compressor blades at very high rotating speed. High energy blade fragments may penetrate the wall of the containment ring and can damage fluid lines, control cables, oil tanks and airframes, which may seriously affect the flying performance and threaten the safety of airplane and passengers. As per certification requirement engine casing should have sufficient strength to contain the blade in the event of failure. A primary problem in the optimal design of aircraft gas turbine engine internal and support structures is the accurate simulation of the Fan-blade out event. In this paper numerical simulation is carried out using non-linear finite element technique to study the impact process. It studies the effect of mesh refinement, contact modeling and high strain rate material behavior on energy absorption characteristics during blade impact with the casing. Temporal variation in kinetic energy of released blade is studied along with strain energy of casing to estimate casing perforation and residual energy of release blade. Thereafter, transient imbalance load and center of gravity of Fan rotor assembly are numerically evaluated for bearing design. Finally, deformation in engine casing is correlated with experimental test results for validating the procedure and default parameters.

INTRODUCTION
Outer casing of civil and military engine has to withstand very high-energy impact resulting from release blade of rotor disc, and yet it is desired that engine weight should be minimum. Accurate simulation of the blade out event is one of the challenging tasks for aircraft structure designer. Aero engine design philosophy demands that high energy blade profile after ejecting from the rotor should remain contained within the casing whereas containment of large and high speed rotor discs is not within the scope of aircraft design. Blade release can occur due to various reasons such as high cycle fatigue, bird impact, and foreign object damage (FOD) or domestic object damage (DOD). Due to non-contained engine fragments there are horrified incidents where there is loss of aircraft and valuable human lives as cited in paper by Norman et al. [1]. It is utmost important to accurately simulate the blade-out event to ensure structure integrity during flight. Release blade generates extreme transient loads in the aircraft structure due to high-energy blade fragments and out-of-balanced force induced by the blade release. These loads are also important inputs for the designer to design other engine components such as nacelle, struts, wings and aircraft fuselage. As per certification requirement in gas turbine engines, it is mandatory for engine manufacturers to demonstrate through certification tests on full engine that most critical blade must be contained within the casing when it is released at the maximum rotating speed at full rated thrust of the engine and there

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should not be any catastrophic failure which can result in loss of aircraft. The penetration capability of the material released is affected by its shape, orientation of impact, material properties, and kinetic energy.

Most of the blade containment concepts have been taken from the experimental and analytical studies of cylindrical, spherical and cubical shape object impact on metal target. Such type of projectile containment has always remained an interesting topic among military engineers and aircraft designers. Impact damage largely depends on the velocity of the projectile and its mass. Recht and Ipson [2] conducted several impact tests against steel plates rigidly clamped at its edges after taking slender rods with different length to diameter ratio and concluded that failure of target plate can occur due to compression and shearing of the material over the perimeter of the impacted area or due to the tensile strain exceeding the failure limit over the extended volume of the material. Most of the analytical and finite element analysis techniques developed uses energy or momentum balance approach for containment analysis. A closed form analytical solution has been derived [3] for the limiting case of bullet containment having truncated conical heads on the basis of quasi-dynamic theory. This expression can be used to evaluate the thickness of target material required to contain a cylindrical projectile of known geometry and energy. Tests carried out during impact of cylindrical projectile on flat plate show that small projectile with high velocity can cause different type of failure as compare to large projectile with small velocity having identical initial energy. Corran et al. [4] described the failure mechanism of shield for the containment of subordinate velocity missile. Such type of experimental, analytical and numerical work reviews of projectile impact with target plate is available in an article by Goldsmith [5].

Testing of engine fragment parts for containment analysis in a test rig is a very costly process and also no scientific conclusion can be drawn to predict cause of failure from a single test. Hagg and Sankey [6] derived an analytical solution to predict casing thickness in the event of disc burst by comparing strain and shear energy with the total energy of disc fragmented missile before impact. They also compared their results with experimental observations. With the advent of computer non-linear finite element codes have been developed on time marching scheme in late seventy of twentieth century using super computers facilities. Witmer and Wu [7] developed a spatial finite element method to study the response of containment rings that can undergo large elastic-plastic deformation. They assumed rotor fragment to be rigid and used conservation of linear and angular momentum to determine the post impact velocities between the fragments. Wilbeck [8] conducted test on scaled model of turbine to assess turbine missile effect in nuclear plant design. Dewhurst [9] studied failure modes of containment ring due to multiple blade shed using finite element software ANSYS using energy balance approach. Sarkar and Atluri [10] used explicit nonlinear finite element code DYNA-3D to compare T58 rotor rig test results with numerical analysis for single and multiple blade containment analysis. Transient dynamic analysis code MSC-DYTRAN has been used by Xuan and Rong-ren [11] to compare experimental results of blade deformation before and after impact after considering single blade impact with containment ring. Recently, Sinha and Dorabala [12] simulated fan blade out event of civil aircraft engine considering both LP and HP spool of aircraft engine rotating at two different speeds. Thereafter, Carney et al. [13] correlated experimental and numerical results to investigate the affect of containment shield geometry due to blade impact.

Due to stringent weight criteria metallic containment shields are being replaced with composite materials. Aircraft designer are using Kevlar fibre as a possible substitute for casings material due to its high impact strength capability in comparison with all other fibers. In numerical simulation Gerstle [14] modified large deflection shell computer code to simulate fragment and fabric shield interaction. Recently Naik et al. [15, 16] carried out energy absorbing characteristics of Kevlar and Zylon fabrics after testing these fabrics in universal testing machine and later carried out numerical simulation.

This paper presents Fan blade out simulation using commercially available explicit finite element solver LS-DYNA [17]. In this study pre-stress contours obtained using dynamic relaxation technique of explicit FEM method is compared with implicit FEM solver ABAQUS. Later, the effect of high strain rate material behavior on energy absorption characteristics during blade impact with the casing has been analysed. Temporal variation in kinetic energy of released blade is studied along with strain energy of casing to predict casing perforation and residual energy of blade profile. Present study looks into the effect of mesh refinement, contact modeling during blade
casing interaction. Finally, deformation obtained numerically using explicit finite element technique in the engine casing is correlated with experimental test results.

ANALYTICAL MODEL

In this section closed form solution based on momentum balance approach derived on the lines of Hagg and Sankey [3] has been re-derived to obtain Fan casing thickness. The minimum relative perforation energy occurs with the normal force imposed by the translational motion of the impacting blade and not by the rotational energy of the release blade. Based on this assumption, energy possessed by the released blade is given as

\[ E_1 = 0.5 \, M_b \, V_b^2. \]  

(1)

In order to contain released blade within the casing, change in energy should be lesser than the energy required for straining the material in compression and shearing.

The energy \( E_2 \) required for straining the material in compression and shearing is given as,

\[ E_2 = A h \varepsilon_d + K \tau_d P h^2, \]  

(2)

where, \( A \) is contact area, \( h \) is the thickness of the casing, \( \varepsilon_d \) is per unit plastic strain and \( \tau_d \) is average dynamic plastic flow stress of the material; \( P \) is the perimeter of the sheared area and \( K \) is experimental constant with a value in the range of 0.3 to 0.5. Hence, necessary and essential condition for the containment of released blade is

\[ 0.5 \, M_b \, V_b^2 < K \tau_d P h^2 + A h \varepsilon_d. \]  

(3)

Based on this theory minimum thickness of the casing required to contain the release blade is

\[ h = \frac{1}{2k} \left( \frac{A}{P} \right) \left( \frac{\varepsilon_d}{\tau_d} \right) \left[ -1 + \sqrt{1 + 2K \left( \frac{\varepsilon_d}{\varepsilon_d} \right) \left( \frac{P}{A^2} \right) M_b V_b^2} \right] \]  

(4)

Engine house manufacturer’s uses empirical relation based on energy balance approach. According to this relation casing thickness to contain single release blade is given as

\[ h = \frac{M_b^{0.6325} V_b^{1.265}}{915 C^{0.5} R^{0.5}}, \]  

(5)

where \( C \) is true chord length in mm of the blade and \( R \) is containment strength of the casing in MPa, \( m_b \) is the mass of the blade in kg and \( V_b \) is the velocity of the blade about its CG in mm/second.

NUMERICAL SIMULATION

Numerical modeling may offer great details in a blade containment analysis. It can help in identifying the various causes which are responsible for the damage of containment shield. It is easy to segregate energy dissipated through petalling and shear processes numerically which is difficult to obtain through experimental procedures. Thereafter, limited containment tests can be used to validate the methodology of nonlinear finite element analysis. The validated numerical model may be used as a useful tool for the design of the casing. In this finite element analysis, it is assumed that there is no effect due to aerodynamic flow and temperature rise in the components due to plastic flow of the material and work done against friction between two rubbing bodies.

FINITE ELEMENT MODEL

Fan-Casing and three stages Fan-rotor assembly along with LP-Main shaft and other accessories have been modeled using solid brick elements as shown in Figure 1. This model consists of 8,81,632 elements and 12,51,636 nodes. Finite element model of the casing has been generated in such a way so that there should be at least three elements across the thickness of the casing to take into account bending effect arising during impact event in the casing. Similarly, finite element model of rotor assembly has been generated in such a way so that there should be at least three elements across the thickness of each blade.
Mesh density is one of the important considerations in the determination of model accuracy. The accuracy of the solution can increase with a higher local mesh density specifically, in the region of high stress gradient. Stress gradient increases rapidly near impact zone where blade strikes the casing. In this FEM model very high mesh density has been generated in the impact zone to capture high gradient stresses and making the FEM model more realistic.

Two parts of Fan split casing and Inlet-Casing have been joined together using pre-stress bolt connections as shown in enlarge view in Figure 2. Finite element model of Ball bearings have been simplified to journal bearing by associating equivalent stiffness as per analytical calculations. Also, finite element model of Bullet-Nose is bolted with Inlet-Strut using bolt connections. Thus this model has each and every aspect as that of real assembly model.

All the elements in this numerical model where blade will make contact with the casing have been assigned fully integrated elements formulation to avoid hour glass mode, rest of the elements have reduce integration formulation to reduce computation time. In this case, the LS-DYNA viscous-based hourglass control type 3 was used with an hourglass coefficient of 0.02. It should be noted here that each component of this finite element assembly has been modeled as per guidelines specified in the paper by Zukas and Scheffier [18] and each element of the FEM model passes through stringent quality norms.

**Figure 1 Finite element model of rotor and casing**

**Figure 2 Enlarge view of FEM mesh of Bolt connections at different locations**

**MATERIAL MODEL**

Fan casing and first stage fan blade is made up of titanium alloy. For both blade and casing, isotropic elastic-plastic material model with modified Johnson-Cook [19] parameters have been taken for defining elasto-plastic material behaviour and damage parameters. Present simulation does not considered the effect due to thermal softening of the material and titanium fire during Fan blades rub with the casing. Generally, stress-strain curves are considered as per standard practice adopted by engine house companies, but in the present analysis Johnson-Cook parameters published in standard domain [20] have been considered and listed down in Table 1 for the user’s reference.
PRE-STRESS

The stable rotation induces centrifugal forces that pre-stress the rotor and modify its original shape and dynamic behaviour. This model is subjected to rotating speed of 11000 rpm pertaining to the red line speed of the engine. Rotating Fan assembly has been pre-stress using dynamic relaxation technique [21] of explicit finite element software LS-DYNA and shown in Figure 3. In this model three stages Fan Bladed disc assembly along with shaft have been pre-stressed using commercially available LS-DYNA solver. Figure 4 shows comparison of pre-stress von-Mises stress contour obtained using explicit finite element software LS-DYNA and implicit finite element solver ABAQUS for first stage fan blade. Non-dimensionalised von-Mises stress plot obtained at the mid-section of Fan blade at different radial locations from two different solvers have also been plotted in graphical form to compare these two results. Excellent comparison has been obtained in between two results. Also, it is observed that during enforce motion in bladed disc assembly von-Mises stresses does not vary with time for about four rotations.

FAN BLADE OUT SIMULATION

Fan blade out simulation is performed on the complete assembly in a fixed frame. In this step initial velocity field is imposed at each node of the rotor including the ejected blade. The blade gets release from the disc assembly by inducing a crack in the blade which propagates at a specific speed. Identical conditions are simulated using rivet connection by disconnecting each rivet at one particular time.

CONTACT

The interaction in between released blade and Fan casing has been described using “eroding-surface-to-surface” contact with blade as a slave. All contacts are assumed to have 0.39 and 0.3 as static and dynamic coefficient of friction respectively in between the blade and the casing. The casing is clamped at the rear end with all the three degree of freedom restrained in all the three directions.
RESULTS AND DISCUSSIONS

This section presents the results for pre-stress fan bladed disc assembly in the event of blade-off analysis. First it shows the variation in kinetic and strain energies of release blade and casing to reveal whether blade can pierce through the casing or not. Thereafter, stresses and deformation in the casing at the location of impact zone has been shown to indicate severity of the impact in the casing and at last transient radial bearing loads along with transient position of radial center of gravity of bladed disc assembly has been shown for the design of engine components.

KINETIC AND STRAIN ENERGY

During interaction of blade with the casing, it is the kinetic energy of the released blade which gets converted into kinetic and strain energy of the casing and strain energy of the blade during blade impact with the casing. Also, part of the energy gets dissipated in the form of heat energy due to blade rub with the casing. As per analytical calculation carried out in the previous section, blade can be contained within the casing if kinetic energy of the blade is lesser than the sum total of shear and strain energy of the casing material.

Figure 5 shows temporal variation in normalised strain and kinetic energy with respect to initial energy of the released blade taken as 17840 Joule. This figure indicates that kinetic energy of the released blade drops to 40% from its initial value within 3.0 milli-second, thus 60% of energy is getting dissipated in increasing strain energy of the blade and casing. It is interesting to observe that kinetic energy of the released blade increases at 2.3 milli-seconds due to energy imparted from the following blades. Similarly, Figure 6 shows temporal variation in normalised kinetic and strain energy of the casing which goes up by 40 and 16 percent respectively during blade strike with the casing. These two figures reveal that although there is sufficient energy left with the blade after 3 milli-seconds but there is remote possibility of getting it penetrate through the casing.

Figure 5 Temporal variation in kinetic and strain energy of the released blade

Figure 6 Temporal variation in kinetic and strain energy of the casing

STRESS AND DEFORMATION

Blade after getting separated from the rotor disc travel linearly before touching with the casing as indicated in Figure 7a after 1.25 milli-second. Thereafter, as shown in Figure 7b and Figure 7c the leading edge of the blade, which is thin, deform on contact with the casing structure and impact with the following blade. The blade rotate around the point of first impact until its trailing edge hits the casing; this is indicated in Figure 7d and Figure 7e after 2.5 and 2.75 milli-second respectively. Thereafter curl blade remain entrap in between the casing and rotating blades. It is interesting to note, that curling of blade is also observed during testing and simulation by Xuan, Hai-jun and Rong-ren [11]. These figures further indicate that as soon as blade strikes the casing due to its sharp edge at the outer radius, very high stresses generated locally at the outer edge of the casing. Also, in contact zone effective-stress is very high and this location changes its position with the blade location as revealed.
by these figures. This simulation further envisages that more than three bolts of the split casing gets completely shear off and blade does not perforate the casing but remain contain within the casing.

(a) 1.25 milli-second  (b) 1.5 milli-second  (c) 1.75 milli-second
(d) 2.5 milli-second  (e) 2.75 milli-second  (f) 3.0 milli-second

Figure 7 Released blade position in Fan casing at different time interval

CASING DEFORMATION AND EXPERIMENTAL CORRELATION

Impact of the blade with the casing shows appreciable growth in the casing during interaction. Figure 8 shows normalized deform shape of casing with respect to un-deformed radius of the casing taken as approximately 370 mm above the mid section of the first stage Fan-blade. This figure shows that maximum casing deformation increases with time and it goes up to 5% from its original radius at 3 milli-second.

As per experimental test carried out on Fan Casing during blade off analysis, it is depicted that maximum radial deformation is approximately 25 mm whereas numerical simulation results indicate that this deformation is around 19 mm. Discrepancy in the result can be due to thermal softening effect due to increase in temperature of the released blade and the casing. This effect has not been included in the numerical simulation which is due to blade rub with the casing and the plastic flow of the casing and blade material. Test results further indicate that blade after getting released from the rotor remains contained within the casing as per numerical simulation conducted in this paper.

This result can also be verified using empirical equation (5). In the present investigation mass of the blade is 0.38 kg, blade velocity is 291937 mm/sec at the center of blade, true chord length of blade is 125.3 mm and containment strength R is equal to 11668 MPa. Equation (5) shows that minimum thickness of the casing require to contain 0.38 kg blade mass is 4.0 mm which matches with numerical prediction carried out in this paper.
TRANSIENT RADIAL IMBALANCE LOAD AND CENTER OF GRAVITY OF FAN ROTOR

Temporal variation in non-dimensionalised radial imbalance load in first stage Fan rotor assembly is shown in Figure 9. This figure indicates that due to sudden release of single blade, dynamic centrifugal load increases by 70 percent with respect to imbalance load arising due to single blade-off situation calculated for first stage fan rotor. Also, there is radial static shift in radius by 3.47 mm in first stage Fan rotor having assembly mass of 28 kg. The unbalanced force induced by the ejection of the fan blade makes the rotor orbiting. High loads are transmitted to the stator through the bearings and through the blades. Figure 10 shows temporal variation of centre of gravity of the first stage fan rotor. This figure further reveals that center of gravity shifted exponentially till 3 milli-second. This data is important for validating the test results.

CONCLUSIONS

Numerical simulation shows that dynamic relaxation method used in the present analysis for pre-stress analysis matches very accurately with implicit finite element solution. Decrease in kinetic energy of release blade increases strain energy of blade and casing components. Numerical and empirical calculations carried out in this paper predicts that for the casing thickness and red-line speed considered in this investigation, blade remain contain within the casing. Results further envisage that due to plastic deformation of the casing there is local bulge in the casing and its location shifted with blade position and it increases with time. Blade-off event give rise
to radial unbalance force which is 1.7 times more than the radial unbalance load in rotor assembly without single blade. Also, temporal variation in blade-off load and center of gravity of the first stage Fan blade rotor assembly increases exponentially with time.

Figure 9 Non-dimensionalised blade-off force on the bearing in radial direction

Figure 10 Temporal variation in CG of First Stage Fan Rotor

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