Simulation of Check Valve Flapper-Housing Impact Using LS-DYNA® Fluid-Structure Interaction

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Abstract

The flapper of a check valve in the aircraft air management system may hit the housing with a very high speed due to the sudden pressure differential caused by duct rupture, which may lead to the damage of the flapper. This event can be simulated using transient dynamic finite element tool, such as LS-DYNA®. However, the flapper impact speed usually is unknown.

The current study is focused on solving for the impact speed with the developed Fluid-Structure Interaction technology in LS-DYNA, using Arbitrary Lagrangian-Eulerian (ALE) formulation. With the aid of a concept of "Source", "Moving Air", and "Sink" for the ALE model, which represent the conditions of constant cabin pressure, airflow interacting with the flapper, and non-pressurized ambient environment, respectively, the proposed method successfully simulated the check valve fluid-structure interaction behavior. The predicted impact speed has an excellent agreement with the test.

The developed methodology is accurate, easy to use, and applicable to all check valves regardless of size, material, and pressure differential.

Background

The rupture of the upstream duct leads to an instantaneous high pressure differential between the upstream and downstream of a check valve, used in an aircraft air management system to maintain the cabin pressure. As a result, the check valve flapper will hit the housing with high speed when closing, and consequently cause damage to the flapper. There are advanced transient dynamic FEA simulation tools available, such as LS-DYNA, to simulate the flapper-housing impact event. However, the impact speed of the flapper right before the impact is not known.

As such, there is a clear need to establish a methodology, which is accurate and easy to use, to simulate the interaction between the flapper and the air flow based on Fluid-Structure Interaction (FSI), from fully open to final closing of the flapper. As a result, the flapper impact speed can be obtained using this methodology.

FSI, as one of the multi-physics models, has become increasingly the focus of computational engineering community in recent years. One of the most appropriate approaches to solve the FSI coupling problems is the Arbitrary Lagrangian-Eulerian (ALE) formulation for the fluid domain and Lagrangian formulation for the structure domain. The analysis tool readily available is LS-DYNA. The current ALE method requires no topology changes in the structure, which is the case for the check valve flapper with the load of air flow.

FEA Model

The FEA model is shown in Figure 1. The half of the symmetric ALE model is displayed in order to show the Lagrangian model inside for all the structural parts. The Lagrangian model uses 8-noded solid elements for all components.

The ALE model for fluid uses 8-noded solid elements as well. In modeling of FSI, the ALE model, which takes the shape of check valve housing and upstream duct, consists of three volumes: Source, Moving Air, and Sink (see Figure 1). The Source volume has a constant high pressure of 9.7 *psi*, representing the cabin pressure, which feeds the system with air flow. The Sink has the constant low pressure or is in vacuum, representing the ambient condition, which sucks air from the system. The pressure of the Moving Air is neither constant nor uniform, depending on the interaction of the flapper with the air flow.

For ALE model, the initial condition is set such that the Source and the Moving Air elements have a cabin pressure of 9.7 *psi*. The Sink elements have a constant pressure of 0 (vacuum). There is no initial condition applied to the Lagrangian model. The displacement boundary condition for the Lagrangian elements is applied to the housing bolt holes. For the ALE elements, boundary condition is applied perpendicular to the check valve housing and upstream duct surfaces, which is in the transverse direction to the flow. The total run time for the analyses is 13 *ms*.

ALE Material Model

The air is modeled as an ideal gas, using a linear polynomial Equation of State (*EOS):

$$P = C_0 + C_1\mu + C_2\mu^2 + C_3\mu^3 + (C_4 + C_5\mu + C_6\mu^2)E$$

where P is the pressure, μ is the volumetric strain, C_i are the material coefficients, and E is internal energy per unit volume. For ideal gas,

$$C_0 = C_1 = C_2 = C_3 = C_6 = 0$$

and $C_4 = C_5 = \gamma - 1 = 0.4$ (for air)

where γ is the ratio of the specific heats. Note that initial internal energy per unit volume E_0 is determined by $P_0/(\gamma - 1)$.

Coupling

In many FSI problems, it is necessary for a Lagrangian mesh to move through an ALE mesh. Some form of interaction or "coupling" between the two meshes must then be defined. In LS-DYNA, this coupling is achieved with the penalty method. The penalty coupling algorithm searches for the fluid-structure interface at each time step and calculates the coupling forces on the nodes of the structure (slave nodes) and those of the fluid (master nodes). At the same time, flow through the Lagrangian mesh must be restricted by the algorithm to maintain the physics of the problem.

In order to ensure the coupling is properly set, the mesh density for the Lagrangian elements has to be comparable to that for the ALE elements. The ideal case would be one to one. The key to a successful coupling is the proper use of coupling stiffness. The best way to achieve is to provide a coupling stiffness with a "load curve". The guideline to use the load curve is using 10% of the average ALE element dimension for *x*-axis (penetration), and maximum pressure differential for *y*-axis (contact pressure). If the analysis with this load curve is stable, one may consider increasing the coupling stiffness. However, if the stiffness is too high, it may lead to excessive coupling force on the flapper, which results in unrealistic high stress and plastic strain.

In order to ensure an accurate FSI simulation, one must pay attention to the leakage control at fluid-structure interface. In other words, if the FSI leakage control is not properly set, the fluid can pass through the structure without activating coupling force, thus losing momentum. A parametric study suggests that using strong leakage control in the LS-DYNA coupling card lead to the most accurate results.

One of the most challenging issues for the check valve FSI problem is the lack of a clear demarcation surface between the upstream flow and downstream flow, because the flapper does not separate the Moving Air completely. As such, fluid leakage cannot be checked as the upstream material improperly penetrating into the downstream material. In other words, it is difficult to determine if the leakage control is properly set. However, leakage often occurs for Lagrangian mesh with unsmooth surfaces, which is not the case for this problem. The excellent agreement between the analysis and test, as mentioned later, further evidences that the overall leakage should be minimal.

Contact Friction

The only contact between the Lagrangian components before the flapper-housing impact is between the pin and the hinge arm bushing. It seems that the friction force should be an insignificant contributor to the final flapper impact speed. Nonetheless, an iterative study indicates that the friction is one of the most significant contributors to the final flapper impact speed. For example, if a friction of coefficient of 0.2 is used instead of 0, the final flapper speed will be reduced by approximately 20%.

The reason for that is due to the modeling of contact between the pin and the bushing. The pin and bushing do not have very fine mesh, because of the global mesh density of the Lagrangian model and the diameters of the components. As such, instead of between two concentric circular cylindrical surfaces, the actual contact is between two polygonal cylindrical surfaces. With a high contact pressure induced by high centrifugal force, it will lead to a far more resistance against relative motion than the actual hardware.

To resolve this issue, friction coefficient of 0 is used for all analyses. As a result, the minimal friction leads to an excellent correlation with the test for the flapper impact speed. Note that if the analysis is properly done, the resulting flapper speed cannot be higher than the test. It may be lower because too much energy is consumed at coupling or friction, or excessive leakage occurs. In other words, the test results can be considered as the upper bound of the analytical results. Therefore, if the correlation is good between the analysis and the test, the analysis should be right.

Results

Figures 2(a) to 2(f) show the fluid pressure distributions, and the corresponding flapper positions, at different time. The maximum pressure is capped at 9.7 *psi* for these figures in order to have a more apprehensible view of pressure distribution. The localized pressure peak could be much higher than 9.7 *psi*.

A flapper impact test was performed. A transparent housing was used so that a high-speed camera can take pictures during the impact test, with a frequency of 0.1 *ms* per frame.

The test pressure differential is 8.6 *psig*, slightly lower than the flight condition. In order to have a meaningful comparison between the test and the analytical results, a separate analysis is performed with the lower pressure.

The resultant speed of the flapper tip (node id 530644) based on the test condition is shown in Figure 3. The speed divided by the distance from the flapper tip to the rotating axis, which is 5.61 *in*, is equal to the flapper angular velocity. The resulting maximum tip speed is 2,650 *in/sec* at 9.87 *ms* as shown in Figure 3, which is equivalent to a maximum angular velocity of 472 *rad/sec*. The corresponding test result is 475 *rad/sec*., which is obtained based on the extrapolation of the high-speed camera images.

Figures 4(a) and 4(b) show the last two shots of the high-speed camera before the flapper closing. The flapper tip displacement during the two shots is 0.2503 *in*. By dividing 0.1 *ms*, which is the time between two shots, an average tip speed of 2,503 *in/sec* is obtained. The analytical results in Figure 3 show a peak flapper tip speed of 2,650 *in/sec* at 9.87 *ms*, and a speed of 2,368 *in/sec* at 9.77 *ms*, 0.1 *ms* away from the peak, which give an average speed of 2,509 *in/sec*. The correlation is excellent.

Conclusions

Based on the excellent correlation between the analytical results and the test, a methodology of FSI to simulate check valve flapper-housing impact is validated and established, and the peak flapper-housing impact speed can be analytically determined. The developed methodology is accurate, easy to use, and applicable to all check valves regardless of size, material, and pressure differential.

The resulting speed can be used as an initial condition for the consequent impact simulation with Lagrangian model only but a finer Lagrangian mesh for stress/strain results, which will determine if and how the flapper will damage, or lead excessive plastic strain or deformation based on the known impact speed.

Acknowledgement

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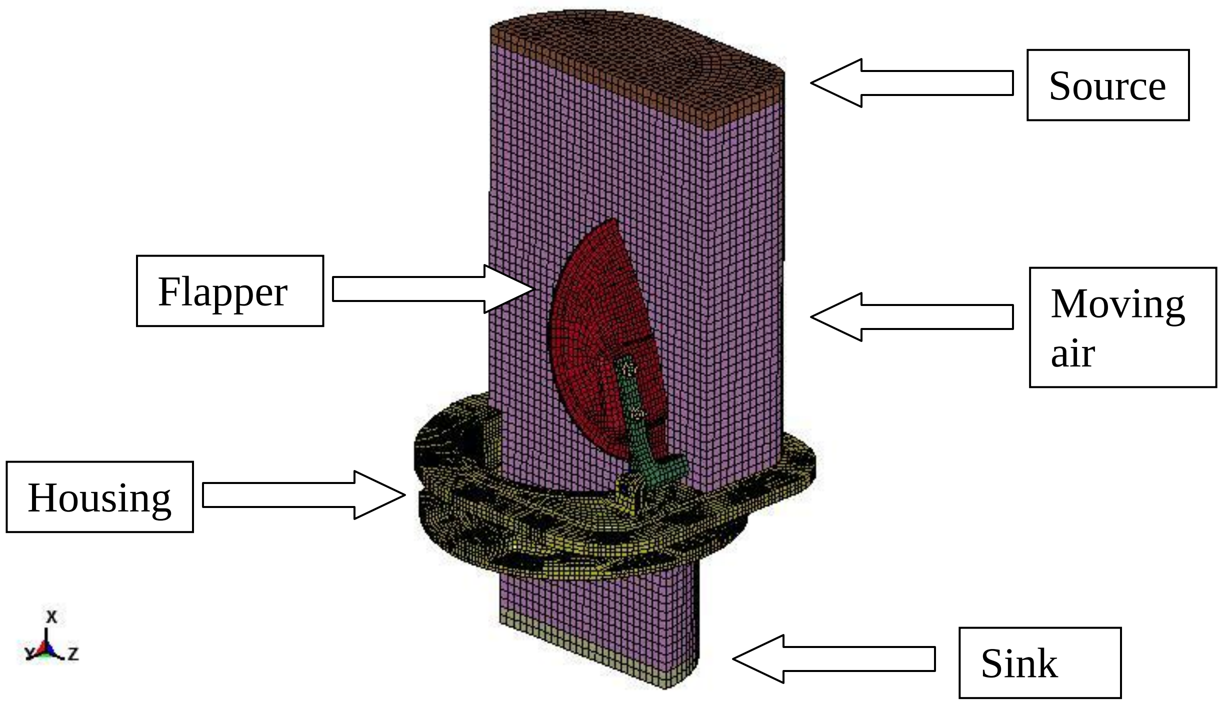
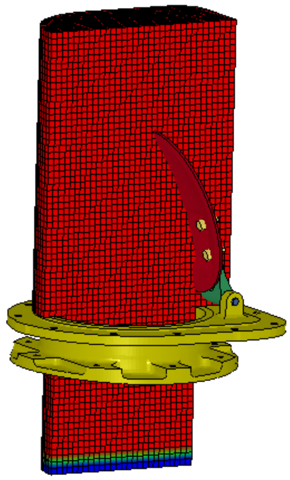
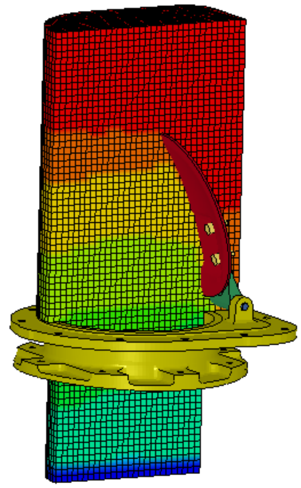


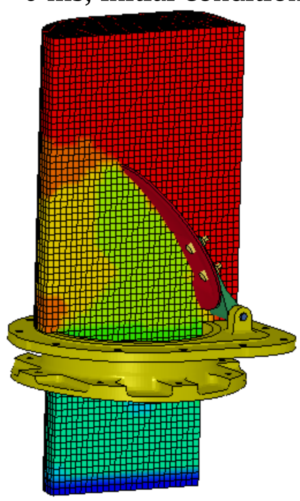
Figure 1: FEA model.



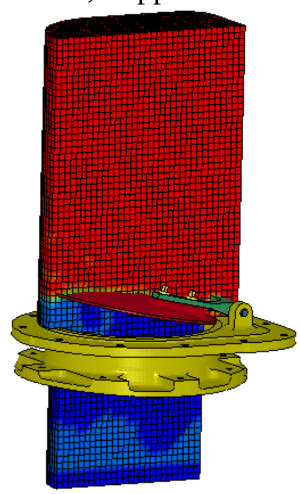
(a) t = 0 ms, initial condition



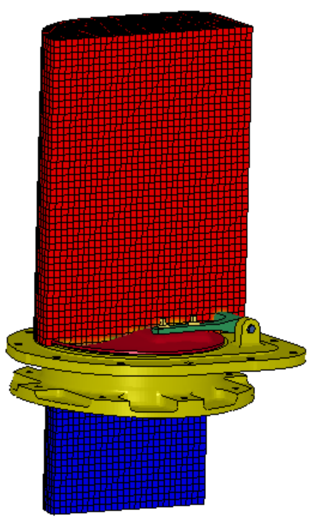
(b) t = 1.1 ms, flapper starts to move



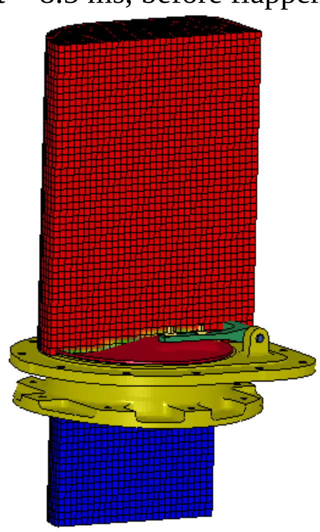
(c) t = 5 ms, flapper picks up speed



(d) t = 8.5 ms, before flapper close



(e) t = 9.2 ms, flapper has the maximum deformation



(f) t = 13 ms, the final configuration

Figure 2: Fluid pressure distributions and the flapper positions at different time.

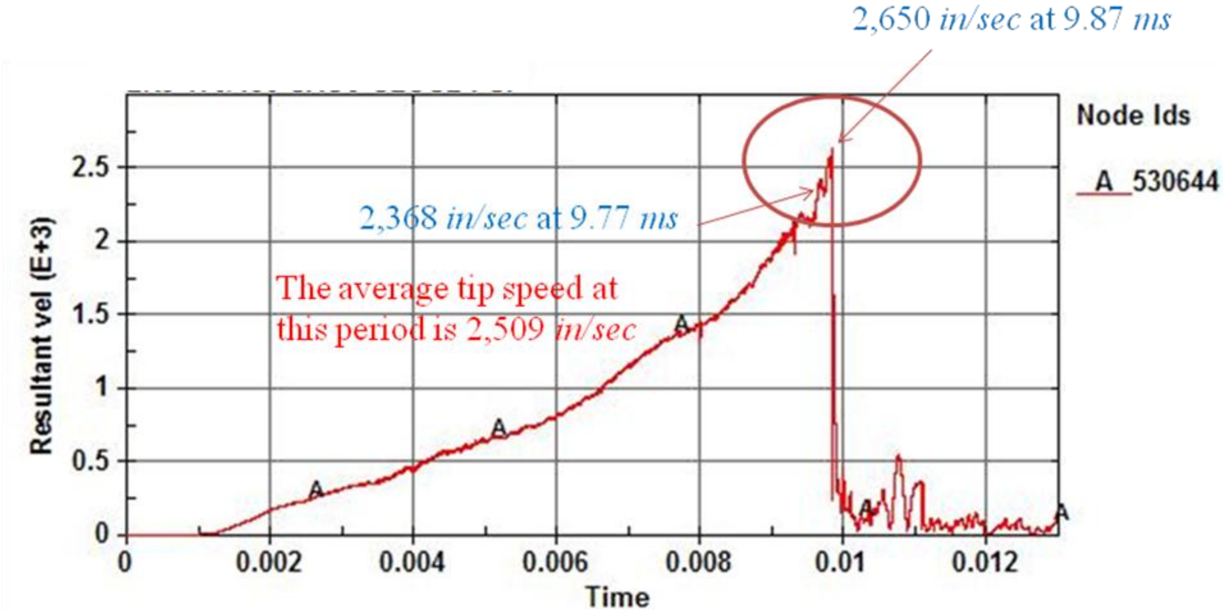


Figure 3: Time history plot for the flapper tip resultant speed (node 530644) for the test condition.

(a) (b)

Figure 4: Last two images of the test before the flapper closing, used to estimate the average tip speed.