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İnsansız Hava Aracında, Ataletsel Navigasyon Sistemine ait Yapısal Yerleşim Tasarımlarının Frekans Cevap Analizi ve Modal Test Metodları ile Değerlendirilmesi

Erkin Barış Güngör[1]   **Bilgin Çelik** [2]

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Öz

Ataletsel Navigasyon Sistemleri, üzerinde bulundukları platformun konumunu ve ivmesini ölçmek için kullanılan hassas sistemlerdir. Oynadıkları kritik rol sebebiyle ölçmekte oldukları verinin doğruluğu önem arz etmektedir. Bu sistemlerin platform ile bağlantısı, yaptıkları ölçümün hassasiyetini birebir etkileyebilmektedir. Bu makalede, taktik sınıf bir İnsansız Hava Aracı için Navigasyon Sistemi yapısal yerleşimi hakkında örnek bir çalışma sunulmuştur. Alternatif yerleşim tasarımlarının mekanik davranışları, sonlu elemanlar ile modellenerek analiz edilmiş; modal test teknikleri ile incelenerek uygun tasarım çözümüne ulaşılmıştır.

Anahtar Kelimeler: Ataletsel Navigasyon Sistemi, Modal Analiz, Frekans Cevap, Modal Test, İHA, Sonlu Elemanlar.

Evaluation of Structural Behaviour of INS Device Installation Design on Unmanned Aerial Vehicle Using Finite Element Method and Modal Testing

Abstract

Inertial navigation systems are precise devices that are used to measure the orientation and accelerations of a platform they are integrated. Accuracy of the measured data is essential due to the critical role of these systems. The way that these devices are integrated to platform, directly affects the measured data accuracy. In this paper, a case study of alternative designs for Navigation device integration on a Tactical Unmanned Aerial Vehicle is presented. Mechanical behavior of the given alternative designs are evaluated with Finite Element Analyses and Modal Testing, hence the appropriate design is selected.

Key Words: Inertial Navigation System, Modal Analysis, Frequency Response, Modal Testing, UAV, FEM.

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INTRODUCTION

Inertial navigation is a self-contained navigation technique in which, measurements provided by accelerometers and gyroscopes are used to track the position and orientation of an object relative to a known starting point, orientation and velocity (Woodman, 2007). In aerospace industry this control is achieved by devices that are referred to as Inertial Navigation Systems (INS).

INS precision is essential for the aircraft, regarding the role it plays. Faulty data from this system will cause misguided aircraft navigation. The structural installation of the INS may make it vulnerable to mechanical effects such as shock and vibration but if these effects are completely eliminated, mechanical effects that platform experience which INS is meant to measure may be eliminated or faults in measurements tend to accumulate with the integrations of the acceleration data (Inertial Reference System (IRS), 2020). Hence, the mechanical integration of INS with the platform becomes essential.

In the conceptual design phase, mechanical engineer has to evaluate mechanical properties of the designed parts, assemblies or structures. This is called design analysis, and it helps engineer in resulting better or optimized designs with the appropriate behaviour to system expectations (Kurowski, 2004)

In this paper, a case study on structural installation of an INS system into a tactical UAV is studied. Two alternatives for installation design are presented with individual Finite Element Method (FEM) analysis and ground testing results. Design differences and mechanical behaviour both alternatives are evaluated.

1. METHODOLOGY

1.1. Loading & Response

Loadings cause deformations, internal stresses, motion etc. on the structure. These are called response of the structure to that loading. Most loads applied to structures are dynamic in origin. That is, their manner of application and/or removal necessarily varies with time. Likewise, the response of a structure in resisting such loads has a temporal character (Irvine, 1986). And further on, the response of a structure depends on its stiffness and mass distribution i.e.;

$$M\frac{d^2\Delta}{dt^2} + K\Delta = F$$

Where F is the external applied force, K is the structural stiffness of the system and M is its mass. Δ is displacement, hence, $d^2\Delta/dt^2$ is the acceleration (Irvine, 1986).

Any dynamic loading can cause an oscillating system response. This vibration in the system will be combination of both forced vibration and resonant vibration. Forced vibrations are caused by the ambient excitations and external loads on the system along with the unbalances and internally generated forces in the system. Whereas, resonant vibration occurs when one or more of the natural (resonant) frequencies of the system are excited (Schwarz, Brian J.; Richardson, Mark H.;, 1999). Natural frequency is the frequency of motion which system tends to move at. It depends on the stiffness and mass of the system.

Natural frequencies act as mechanical amplifiers for the system response hence, they appear as significant peaks in response on a frequency response plot. Thereby, frequency response plot is a valuable tool in evaluating system dynamic behaviour.

The deformation shape at the natural frequency is called “Mode Shape” of the structure. Mode shapes are the indicator for physical behaviour (bending) at natural frequencies. The first mode shape equals the first critical, the second mode equals the second critical, etc. The second or higher criticals are seldom a harmonic of the first or higher critical. If continuous excitation of the natural frequencies occur, the resonant vibration may amplify the response far beyond the system limits and may result in failure of the structure. Accordingly, the designer tends to avoid colliding system excitation frequency and natural frequency of the designed structures. Damping in the system may limit the resonance vibration effect to a certain level and should be accounted for.

Since damping is the measure of a system's ability to absorb energy, a relatively damped signal can be low in amplitude and relatively wide banded. Similarly, a relatively undamped vibration signal can be high in amplitude and relatively narrow banded (Taylor, 2003).

1.2. Finite Element Analysis

Finite Element Method (FEM) is a valuable tool for design analysis and helps the designer predict mechanical behaviour of the designed structure without production of the parts. FEM tool changed the traditional iterative process of designing “design, prototype, test” to a streamlined process where prototype is used for final design verification (Kurowski, 2004). The schematics of traditional design process and computer aided design processes are given in Figure 1.

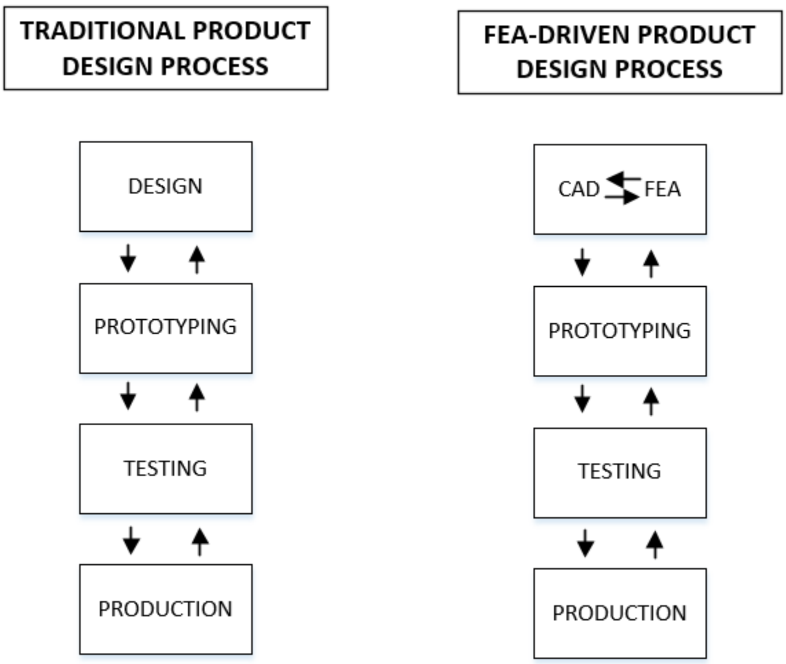


Figure 1. Design process comparison for computer aided design and traditional design

FEA calculates the structural behaviour with the following formulation;

$$[F] = [K] \cdot [d]$$

where,

[F]= Known vector of nodal loads

[K]= Known stiffness matrix

[d]= Unknown vector of nodal displacements

In this equation, displacements [d] are the primary unknown whereas [F] represents the force boundary conditions and the stiffness matrix [K] is a function of model geometry, material properties and displacement constraints.

Furthermore, Finite Element Analysis (FEA) can be used to solve a system response to frequencies at different loadings. For a frequency response solution, FEA software uses an equation of the following form:

$$[M]\{\ddot{x}(t)\} + [B]\{\dot{x}(t)\} + [K]\{x(t)\} = \{F(\omega)\}e^{i\omega t}$$

Where, M is the mass, K is the stiffness and B is the damping matrices.

The validation of the FEM can be done with testing on the structure.

1.3. Impact Testing (Modal Testing)

Impact Testing is performed with a hammer to excite a broadband of frequencies in the structure and one or more accelerometers at fixed positions to measure the response of the structure (Schwarz, Brian J.; Richardson, Mark H.;, 1999) (

Figure **2**). It is often referred to as Modal Testing, and can result in mode shapes of the structure with usage of sufficient amount of accelerometers for response measurement.

Impact testing is a fast and economical way by means of finding the modes of vibration and the dynamic characteristics of a system (Schwarz, Brian J.; Richardson, Mark H.;, 1999).

Modal testing can be done in several different methods. These are SISO (Single input single output), SIMO (single input multiple outputs), MISO (multiple inputs single output) & MIMO (Multiple input and outputs).

For the case studied in this paper, SIMO type of testing is used. This is advantageous due to the simplicity of giving single input on the system and measuring the deformation accelerations from several points helps designating more modes of the structure.

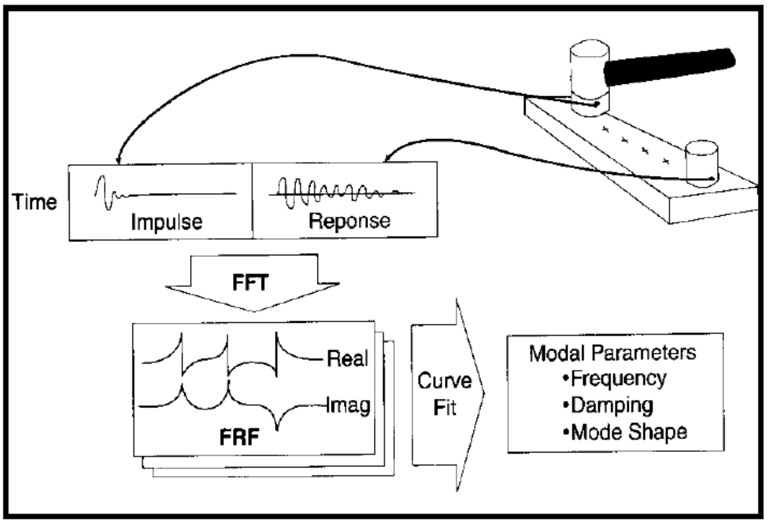


Figure 2. Impact Hammer Testing Scheme.

In vibration analysis, use of both the time domain signal and the frequency domain spectra are required for complete, accurate analysis. To move from the time domain to the frequency domain, one must perform a Fast Fourier Transform on the time domain signal. The Fourier transform of a function (signal) $f(t)$ is defined by;

$$F(\omega) = \int_{-\infty}^{\infty} f(t)e^{-j\omega t}dt = \mathcal{F}[f(t)]$$

And, is called the (amplitude) spectrum (or spectral density) of $f(t)$**.** It describes the distribution of its relative amplitude strength with respect to frequency. (Taylor, 2003).

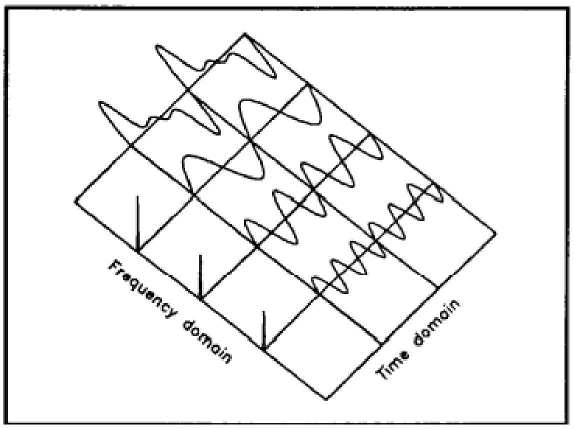


Figure 3. Relationship between Time and Frequency

FFTs are great at analyzing vibration when there are a finite number of dominant frequency components; but power density spectrum (PDS) are used to characterize random vibration signals. A PDS is computed by multiplying each frequency bin in an FFT by its complex conjugate which results in the real only spectrum of amplitude in g^2. The key aspect of a PDS which makes it more useful than a FFT for random vibration analysis is that this amplitude value is then "*normalized*" to the frequency bin width to get units of g^2/Hz. By normalizing the result we get rid of the dependency on bin width so that we can compare vibration levels in signals of different lengths.

Power density spectrums (PDS) are used to quantify and compare different vibration environments. Aerospace engineers shall refer PDS's used in test standards such as MIL-STD-810 and others that provide guidance on how to qualify new products & systems for various operational and transportation environments.

2. DESIGN and CALCULATIONS

Platform vibrations introduce noise in high precision equipment. In most of the applications, these vibrations are overcome by the vibration isolators introduced into the mounting of the device. These mounts are referred to as soft mounts. However, low stiffness may introduce vulnerability to disturbances with the levelling of the equipment (van der Poel, 2010). When using soft mounts, direct disturbances induce larger displacements and deformations and if the tilt modes of the structure is excited, the equipment/machinery with high center of mass may become unstable (van der Poel, 2010). These rocking motions are undesirable for the equipment operation (Rivin, 2003). To overcome these difficulties, stiffer mounts may be used and these are called hard mounts.

Inertial Navigation System (INS) equipment measure the acceleration and position of the aircraft, hence, are needed to be rigidly integrated with the aircraft structure for accurate measurements. Continuing noise in measurements will introduce accumulating errors since the acceleration is integrated to velocity and position in these systems. Hence, the displacement and acceleration response of the mounting are needed to be evaluated together.

2.1.Details of the Installation Design Alternatives

2.1.1. Alternative 1

Weight is usually a deciding factor on aircraft structure design. Due to the effective design capabilities and weight superiority of polymer-matrix fiber composites, they are widely used in aircraft structures (Baker, Dutton, & Kelly, 2004). Installation design alternative comprising of a composite and an aluminum bracket is given in Figure 4 (Alternative 1). In this design, the composite bracket is attached to fuselage structure and the aluminum bracket acts as the mount between equipment and composite bracket. Composite structure is expected to lower vibration acceleration values while retaining sufficient stiffness for the structure provided that its stiffness is lower than metal.

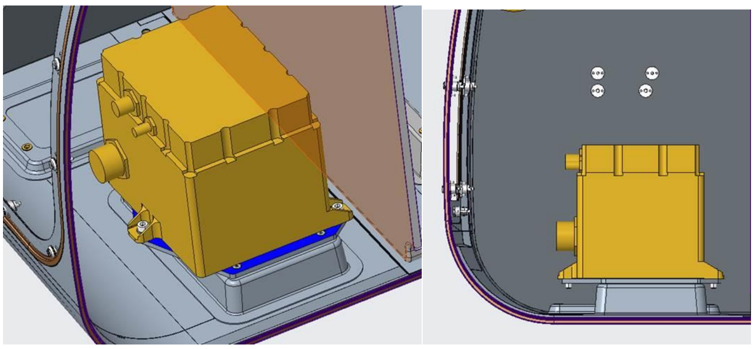
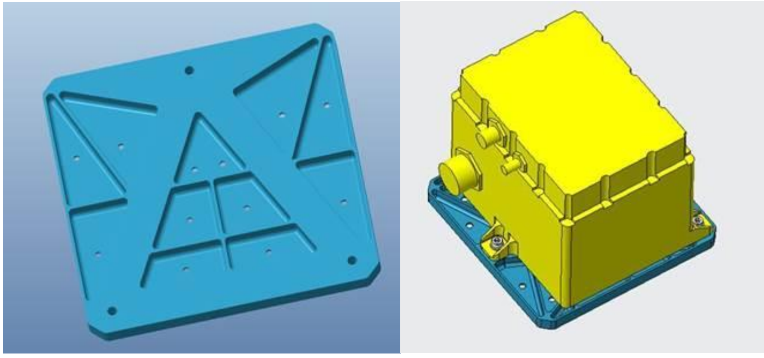


Figure 4. Alternative 1: INS device (yellow) and its mounting brackets (grey-composite, blue-metal)

2.1.2. Alternative 2

The other installation design alternative is given in Figure 5 (Alternative 2). Stiffness is the main design focus of this installation. Therefore, it is expected to experience less displacement on the INS device but the stiffness causes more of the vibration energy, hence more acceleration, to be transferred.



(a) (b)

Figure 5. Installation Design Alternative 2 (a-top view, b- INS mounted view)

2.2. FEA Results

Generated frequencies or forcing frequencies are those actually generated by a running machine and some examples are imbalance, blade passing frequency etc. (Taylor, 2003). In the studied case, generated frequencies are sourced from the engine and propeller assembly.

Subject UAV is a tactical fixed wing type. An internal combustion engine drives the propeller with 2 blades. There is no speed reduction between engine and propeller hence their rotational speeds are equal. The engine runs in a range between 1200 RPM to 3300 RPM which corresponds to 20-55Hz. Blade passing frequencies (f_b) can be calculated as follows, where, f is the rotational speed of the propeller in Hertz and n is the number of blades.

$$f_b = n * f$$

Hence the generated frequencies cover a range of 20-110 Hz and this will be the relevant frequency range for result comparison.

Forcing effects can be modelled in the FEM environment and Frequency response analysis can be run for a broadband of frequencies. This procedure is often referred to as "sine sweep" where it can model the response of the structure due to excitation in consecutive frequencies. With this type of analysis, response of the structure can be quantified whereas the Normal Modes analysis does not result in any quantification in structure response but merely the deformation shapes at natural frequencies in a relative manner.

2.2.1. Alternative 1

Although this design has advantages in weight and practicality on installation, the stiffness of the structure is low enough to result in relatively high displacement under vibration loads. Also, this assembly has 2 mode shapes in the relevant frequency range and due to the low stiffness of the structure, these modes can be excited easily by the given input.

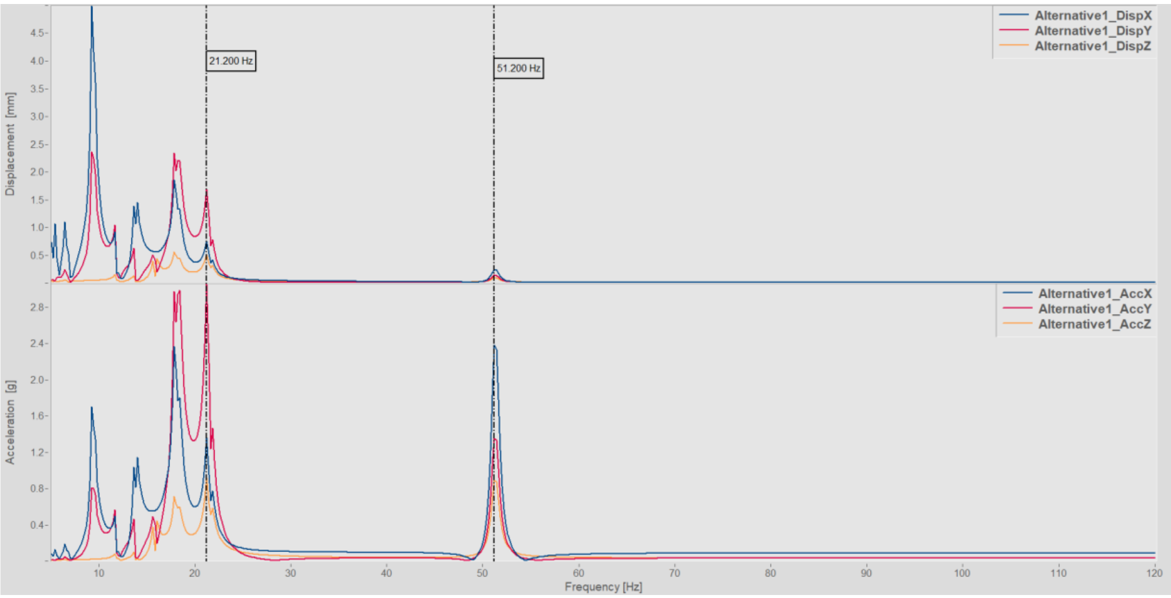
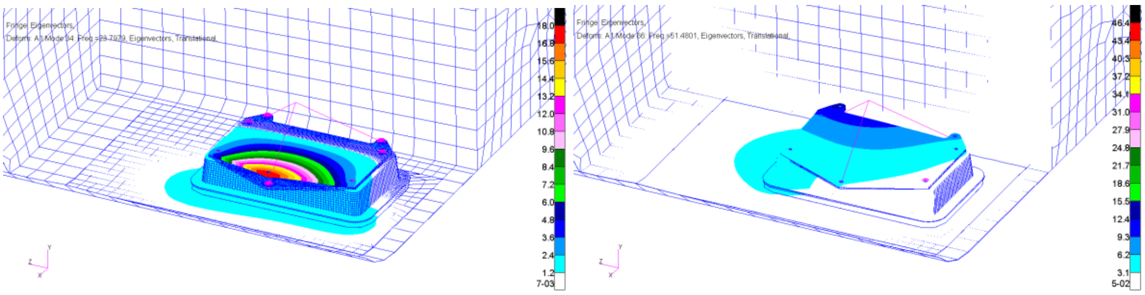


Figure 6. Frequency response results of Alternative 1 (Displacement response-top, Acceleration response-bottom)

As can be seen from the results in **Figure 6**, peak responses occur at 21.2 Hz & 51.2 Hz. These frequency values correspond to the two Mode Shapes given in **Figure 7**. Displacements at the 21.2 & 51.2 Hz excitations are 1.69mm & 0.22mm.



(a) **(b)**

Figure 7. Design Alternative 1 Mode Shapes at; **(a)**-23.8Hz & **(b)**-51.48Hz.

The results imply a tilting motion for the INS device, which is modelled as a point mass element. This motion is dangerous for INS measurements, because it affects both the gyro and accelerometer measurements. If the structure persists this motion under operational conditions, an accumulation of error due to noise in the measurements is indisputable.

2.2.2. Alternative 2

Stiffness of this installation design alternative can be interpreted through the Frequency Response results. Notice the lower displacement results in the frequency range of interest. Due to undamped model, the vibration acceleration seems excessive. Acceleration levels show high loading will occur on the INS device due to vibration. But it is expected that the damping under realistic conditions may dissipate most of this energy.

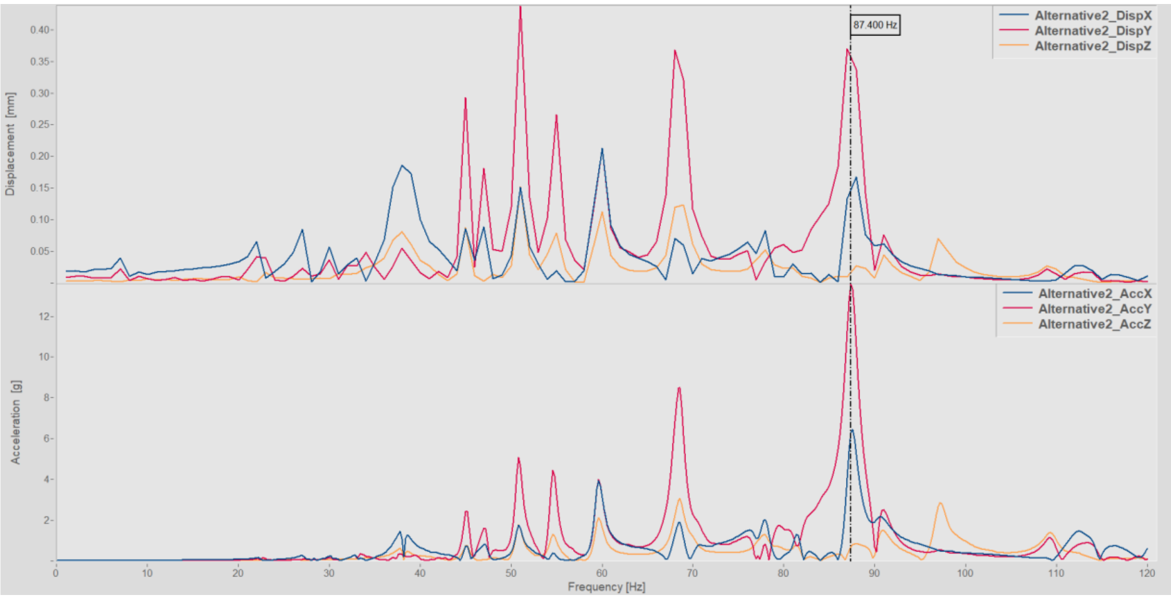


Figure 8 Frequency response results of Alternative 2 (Displacement response-top, Acceleration response-bottom)

Peak response occurs at 87.4 Hz for Alternative 2 (**Figure 8)** and corresponding Mode Shape is given in **Figure 9**. Displacement response at this point is 0.37 mm and values for the entire spectrum subceed 0.5 mm.

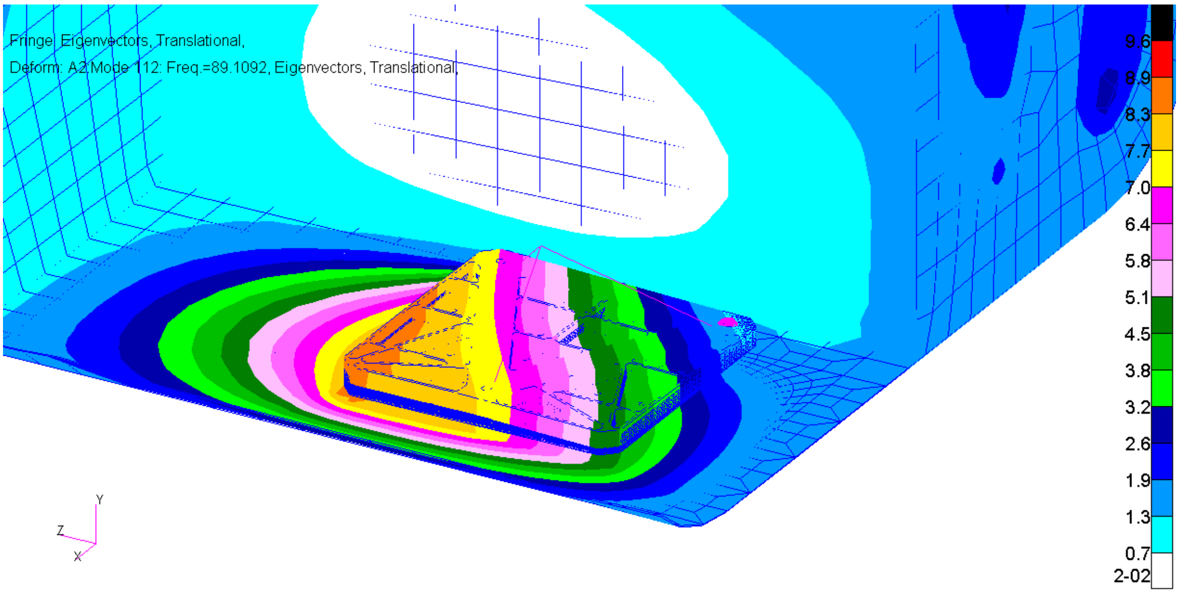


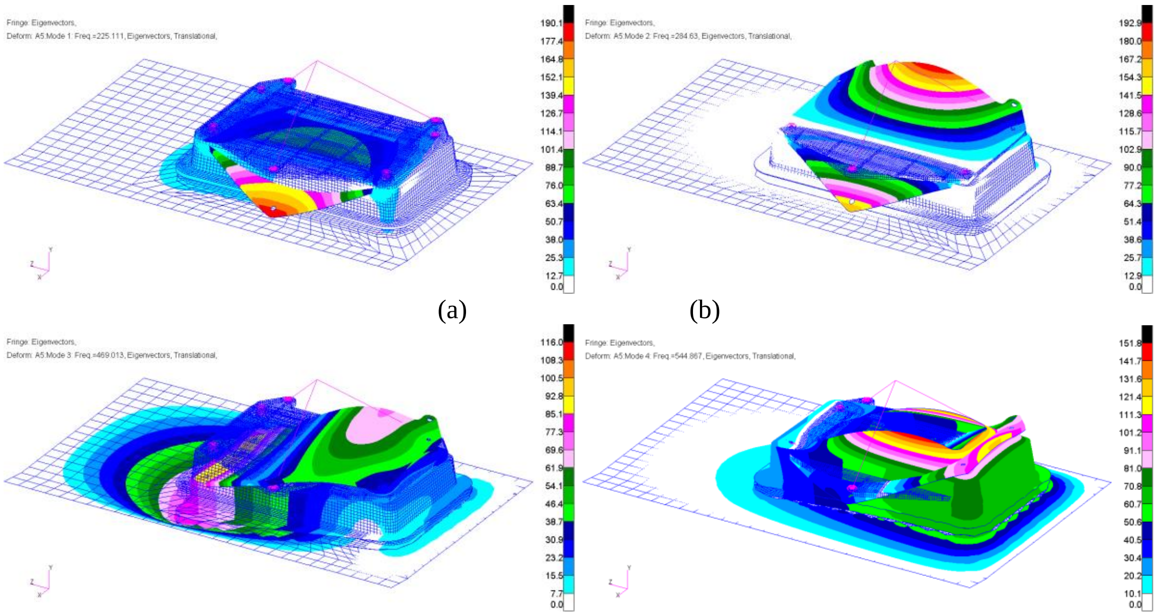
Figure 9 Design Alternative 2 Mode Shape at 89.1Hz

The resulting mode shape is expected to be excited in a relatively harder manner than that of the design alternative 1. This is due to the need of deformation of the fuselage shell in this mode shape.

2.2.3. FEM Analyses for Validation

In an impact testing, structure response shall be measured more accurately if the output accelerations are measured from several points that is expected to deform. Hence, INS device is unmounted from the installation brackets to uncover top surface for accelerometer installation.

So far, analyses are done with the INS device included in the model. Comparison of the test results and FEM can be done if both have the same boundary conditions. Therefore, Normal Modes analyses are repeated for the condition where INS device is unmounted on the brackets. Results for Alternative 1 are given in **Figure 10** and results for Alternative 2 are given in **Figure 11**.



(a) (b)

(c) (d)

Figure 10. Design Alternative 1 Mode Shapes at, **(a)**-225Hz, **(b)**-284Hz, **(c)**-469Hz, **(d)**-544Hz <u>(INS device unmounted)</u>

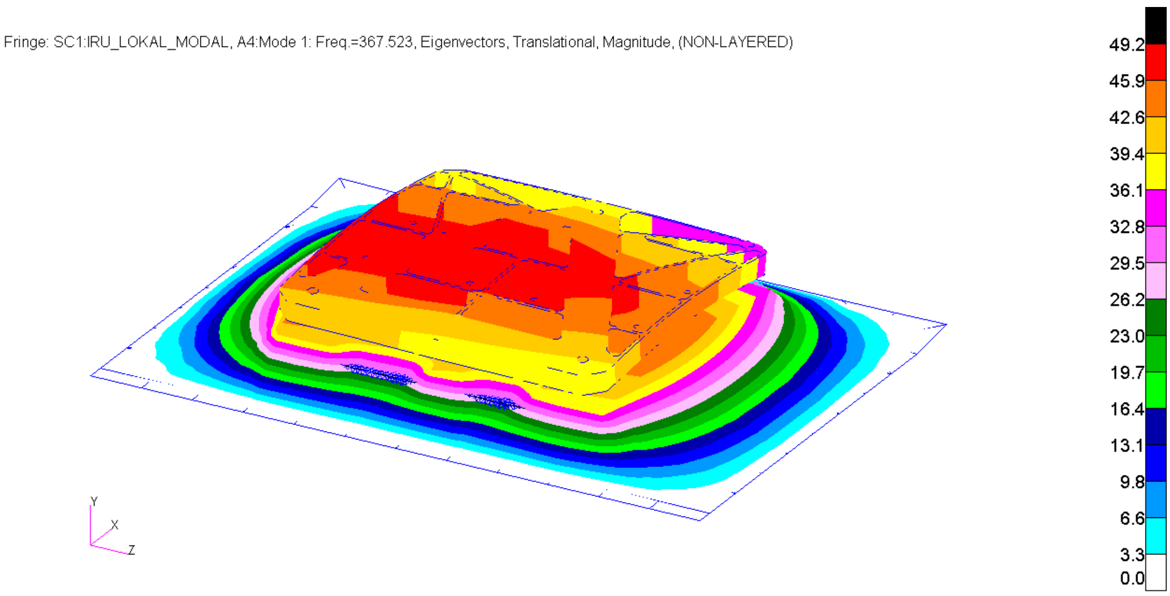


Figure 11. Design Alternative 2 Mode Shape at 367 Hz

2.3. Modal (Impact) Testing Results

Single Input Multiple Output (SIMO) type of Modal Testing is done to measure the structure response from several locations for convenience. Modal Testing of the structures took place while they are mounted accordingly on the airframe.

2.3.1. Alternative 1

Installation Alternative 1 is tested with 4 accelerometers while the INS device is not mounted (**Figure 12**).

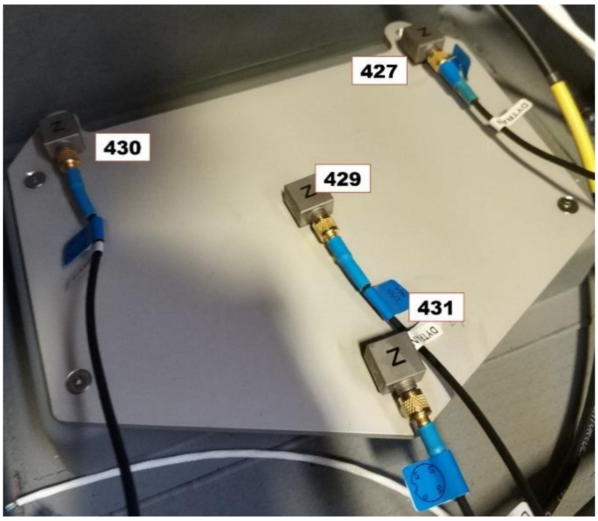


Figure 12. Modal Test setup for Alternative 1

Power density spectrum (PDS) of the measured accelerations in the normal direction to top surface is given in Figure 13. Significant peaks of response can be seen at 225 Hz, 287 Hz, 466 Hz, 494 Hz & 566 Hz in **Figure 13**.

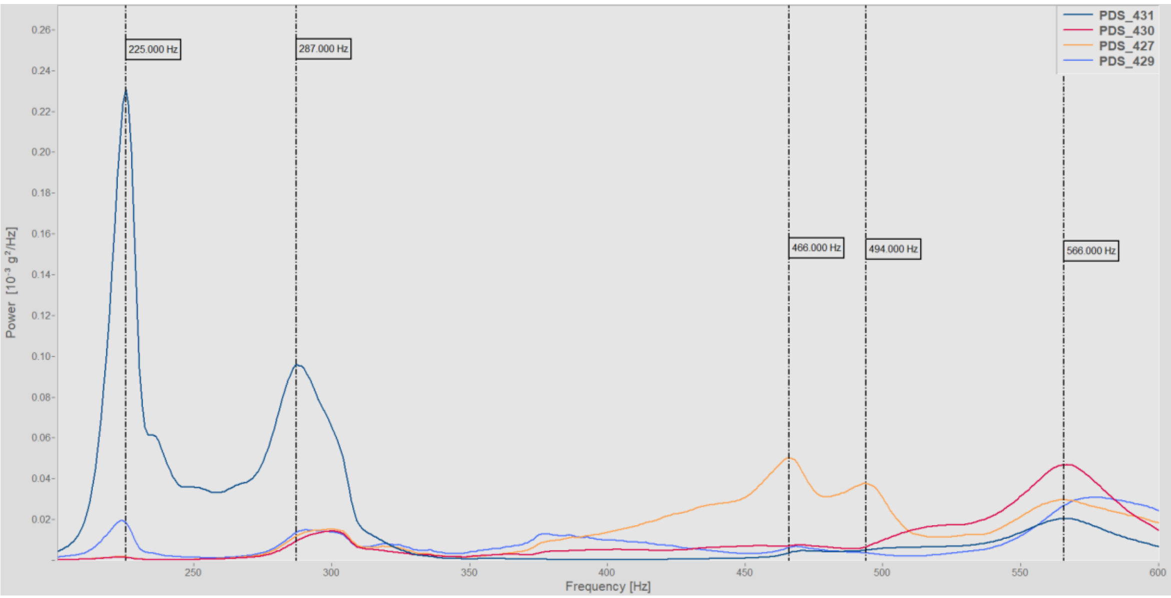


Figure 13. Modal test results for Alternative 1 (INS Unmounted), Power Density Spectrum graph.

2.3.2. Alternative 2

For FEM validation, an impact hammer test is run without the INS being mounted on the structure with the given configuration in **Figure 14**.

Figure 14. Modal Test setup for Alternative 2

High stiffness versus low mass on the structure resulted in instant decay in high frequency vibration response of the bracket structure. Such data may not give reliable results when put through a PDS algorithm. PDS graphs of the acceleration data from several runs of this test are given **Figure 15**. It is evident the test results are inconsistent whereas the vibration magnitudes are really small. There are peaks consistently seen around 7 Hz, 171Hz & 266 Hz. These results are compared with both normal modes and frequency response analysis of the total aircraft structure with similar boundary conditions. After the inspections it is evidently seen that the seen peaks are related with the total Aircraft response and not with mounting bracket. It is also convenient to mention that these peaks are related with structure response and not with significant mode shapes of the area of interest around these frequencies.

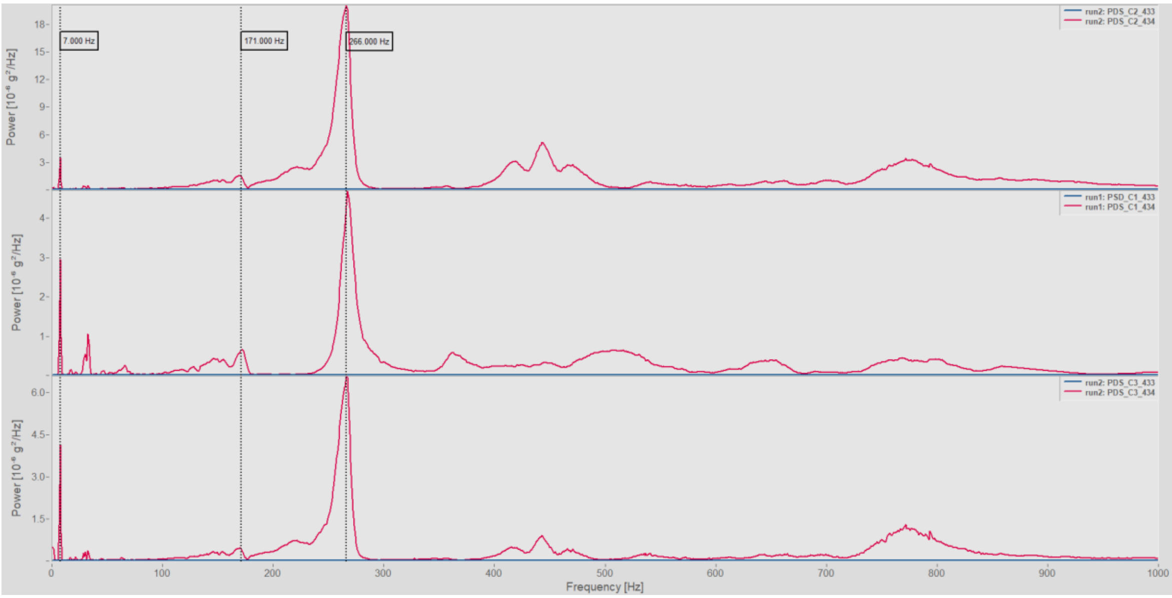


Figure 15. Modal Test results for Alternative 2 (INS Unmounted), Power Density Spectrum graph.

For better evaluation, INS device is mounted on the structure. Mass increase lowers the frequency values of the first modes and increases the response output, making the natural frequencies easy to measure. The test setup shown in **Figure 14** is updated with the INS device being mounted on the structure as can be seen in **Figure 16**.

Figure 16. INS Mounted Test Setup for Installation Design Alternative 2

Power density spectrum of the results are given in **Figure 17**. Significant peak values are seen in 88 Hz.

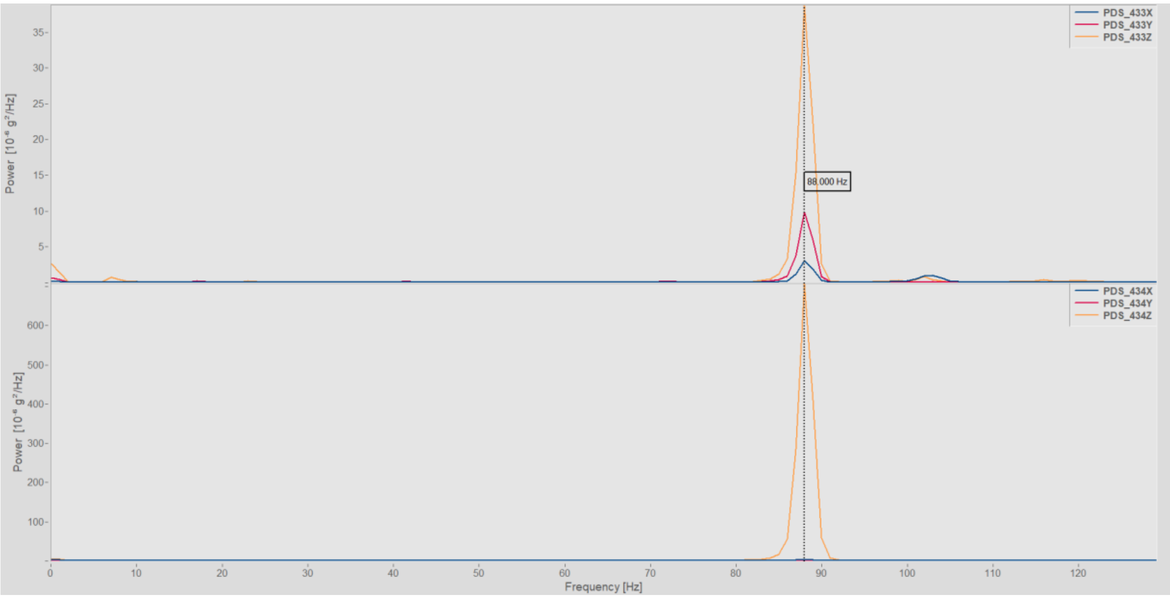


Figure 17. Modal test results for Alternative 2 (INS Mounted), Power Density Spectrum graph.

3. CONCLUSION & DISCUSSION

FEM is a powerful tool in evaluating structure behavior. Two different installation design alternatives are examined through FEM results and the advantages and disadvantages of both designs are evaluated.

Firstly, it is convenient to compare test and FEA results. This will show how accurately FEA can approximate structural dynamic behaviour.

3.1.Comparison of FEA and Test Results

Modal tests are conducted to verify credibility of the finite element model and analysis results. Test result and FEM analysis results are compared in **Table 1**. To quantify the comparison between the results, "Deviation" can be calculated as given below. However, in the frequency domain, magnitude of the Deviation does not have a physical significance and it only implies how close the results are. Therefore, given deviation values should not be taken into account in judgement of comparison between different alternatives.

$$Deviation = \frac{|Analysis\ Result - Test\ Result|}{Analysis\ Result}$$

The deviation values for the results seem acceptable considering the following;

- The Finite Element Analyses are performed without including the damping effects.
- The UAV equipment is modelled as point masses. Lower level components such as cabling, connectors are not included to the model.
- FEM allows only rigid constraints, whereas the real boundary conditions do not satisfy this condition as small deformations or friction effects occur on constraining mechanisms.
- The Power Density Spectrum distributes results in 1 Hz bins. Therefore, resulting peaks occur at integer values of freqeuncies.

Considering these facts, the test results account for the credibility of FEA.

Table 1. Modal Analysis and Test results comparison for both designs.

Design	**Mode Shape**	**Modal Analysis Result** [Hz]	**Impact Test Result** [Hz]		**Deviation**
Alternative 1	**Figure 10** – (a)	225.111	225	**Figure 13**	0.05%
	Figure 10 – (b)	284.63	287		0.83%
	Figure 10 – (c)	469.013	466		0.64%
	Figure 10 – (d)	544.867	566		3.88%
Alternative 2	**Figure 9**	89.1	88	**Figure 15**	1.23%

3.2. Comparison of Installation Design Alternatives

Alternative 1 has 2 mode shapes that lie on the relevant frequency band with lower frequency values. This indicates that this structure can be excited more easily, whereas, there exists only 1 mode shape for Alternative 2 in the frequency range of interest that is located at 89.1Hz. This high frequency mode is expected to be excited more difficultly.

When compared, Alternative 1 results in relatively high displacements and this will cause low accuracy in the INS measurements. Whereas, Alternative 2 prevents experiencing high displacement but may result in higher transmissibility of acceleration (**Figure 6**).

3.3.Discussion

The frequency response results and the mode shapes suggest that tilting may occur with usage of Alternative 1. As INS uses gyro and accelerometers for position and orientation measurements, tilting effects will introduce great noise in the measurements and digital filtering in the device may not interpret that.

The Finite Element Method requires experience based knowledge on how the modelling of structure effects the results. Best way to make sure the model represents true behaviour is by testing with the similar boundary conditions. However, testing condition and data processing may be hard for operational conditions. On the other hand, modal testing results in the mode shapes and natural frequencies with a small setup and cost effective data processing. Also, the excited mode shapes of the structure can be shown with a modal testing. Notice, design alternative 1 modal testing results in 4 natural frequencies of the structure whereas only 1 natural frequency of the design alternative 2 structure was excited and this is only when the INS device was mounted on the structure to provide enough mass distribution.

Frequency response graphs provide quantified information on system dynamic behaviour. Previous comments about the deformation and acceleration responses can be made just by using a sine-sweep method analysis. Results can be gathered for a broadband of frequencies, which provide information on structure behaviour at the whole spectrum of operational conditions. Studies in literature show the significance of Frequency Response Functions on structural behaviour and point to modification methods for the structure according to this data (Özgüven, 1990; Schmitz & Smith, 2019; Park & Park, 2000).

To sum up, using the provided method on this paper, total dynamic behaviour of the structure was evaluated with low cost testing and analysis. Information gathered so far with these analyses and testing include the excitable natural frequencies (mode shapes) of the structure and the deformation & acceleration responses due to incoming vibration effects under operational conditions. All of these information can be used to optimize the design based on system specifications and needs.

Briefly, it is concluded that the stiffer Alternative 2 is a better choice for INS application on the aircraft. The rigidity of this design seems to perevent large excursions on the assembly, preventing unwanted noise in INS measurements.

For further discussion and evaluation, one should consider performing tests in operational conditions where the structure real behaviour will be verified. Studies in literature show the excited modes can be measured in operational conditions (Ersoy, 2017). Input and output correlation can be constructed as a Frequency Response relation in operational conditions, as well. This requires input measurement, which is measured at the source of excitation and output measurement on the structure.

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