

Frequency Response Inspection of Additively Manufactured Parts for Defect Identification

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Abstract: The goal of this paper is to evaluate internal defects in AM parts using dynamic measurements. The natural frequencies of AM parts can be identified by measuring the response of the part(s) to a dynamic input. Different excitation methods such as a modal impact hammer or shakers can be used to excite the parts. Various methods exist to measure the parts' responses and find the natural frequencies. This paper will investigate the use of Doppler lasers, accelerometers and Digital Image Correlation (DIC). The parts evaluated in this work include sets of parts that are still attached to the AM build plate, this makes the identification of a faulty part much more difficult as parts on a build plate interact with each other as well as the build plate complicating the responses. Several approaches to these issues will be presented based on the above listed response measurements.

1. Introduction

Additively manufactured parts (AM) have been shown to be a good choice for low volume uses that might not justify the cost of casting molds. Along with this increase in AM parts, comes new challenges. One of which is analysis of the AM parts for structural integrity. Defects on the outer surface of AM parts can easily be found with visual inspection just as they are with traditional subtractive manufacturing methods. AM parts can possess internal defects not typically seen in subtractive manufactured parts which also must be identified. Nondestructive test methods that are currently being used to find internal defects include x-ray and CT scanning. Although very effective, these methods can be very expensive and time consuming. A low cost method, with the potential to work for either low or high volumes of AM parts, is detection based on the dynamic properties of the parts. Although not as sensitive as X-ray or CT scanning, dynamic testing is inexpensive and fast. Dynamic testing can be used as a prescreen for X-ray or CT scanning by quickly finding some defective parts which then do not need to be X-ray'd or CT'd as they are identified as "bad" parts prior to that step in the process, this has the potential to save both time and money.

If a build plate with AM parts is excited by means of a modal hammer or a shaker, the parts on the build plate will vibrate at their natural frequencies. Natural frequencies are defined by the ratio of stiffness to mass on a part as shown in Equation 1.

$$\omega = \sqrt{(k/m)} \quad (1)$$

At these frequencies, also called resonances, the parts will vibrate with a displacement/velocity/acceleration amplitude much larger than at other frequencies. These natural frequencies are indicated by peaks in the Frequency Response Function (FRF). A FRF is a function that relates the response of the structure at a particular location and direction to an input applied to the structure at a particular location and direction. The input is typically a measured force while the response is typically an acceleration or velocity. Anti-resonance frequencies are indicated by the valleys in the FRF shown in Figure 1. Assuming the input vibrational energy is evenly distributed, an anti-resonance frequency indicated little-to-no vibration at that frequency.

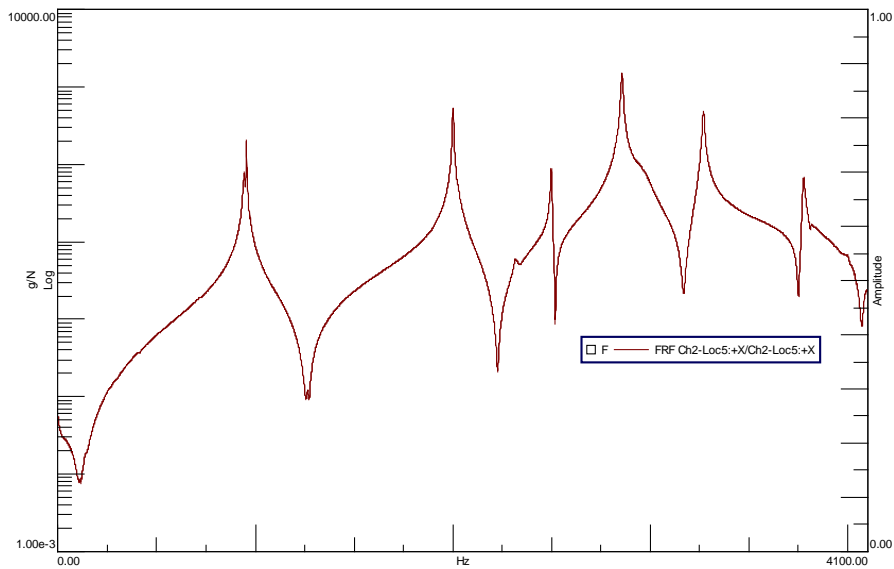


Figure 1 – Example of a FRF of an AM part on a build plate

If identical parts have the same stiffness and mass, they will have the same resonant frequencies. Respectively if identical parts have different stiffness or mass, some or all of the resonant frequencies will be different. Defective parts can be found by comparing the natural frequencies of each part with a standard deviation determined by a set of good parts.

For a natural frequency to change, a mass or stiffness variation in the part must be large enough and at a location which is significant for that natural frequency. This will be determined by the mode shape associated with the particular natural frequency, with some being sensitive to some locations of faults, and some not sensitive to the same faults. A mode shape is the physical shape at which the structure vibrates when the associated natural frequency is excited. For example, imagine strumming a guitar string. The 1st resonant frequency mode shape is the length of the string vibrating back and forth, the 2nd mode shape is the string split in two with a node not moving and the string on each side vibrating back and forth. This increasing complexity in the mode shapes continues with higher and higher resonant frequencies. Only the resonant frequencies with the mass/stiffness variation involved in the mode shape at that frequency will be sensitive to that particular fault. Due to the increasing complexity of mode shapes as frequency increases, due to more localized portions of moving mass on the part, the ability to detect small defects improves at higher frequencies.

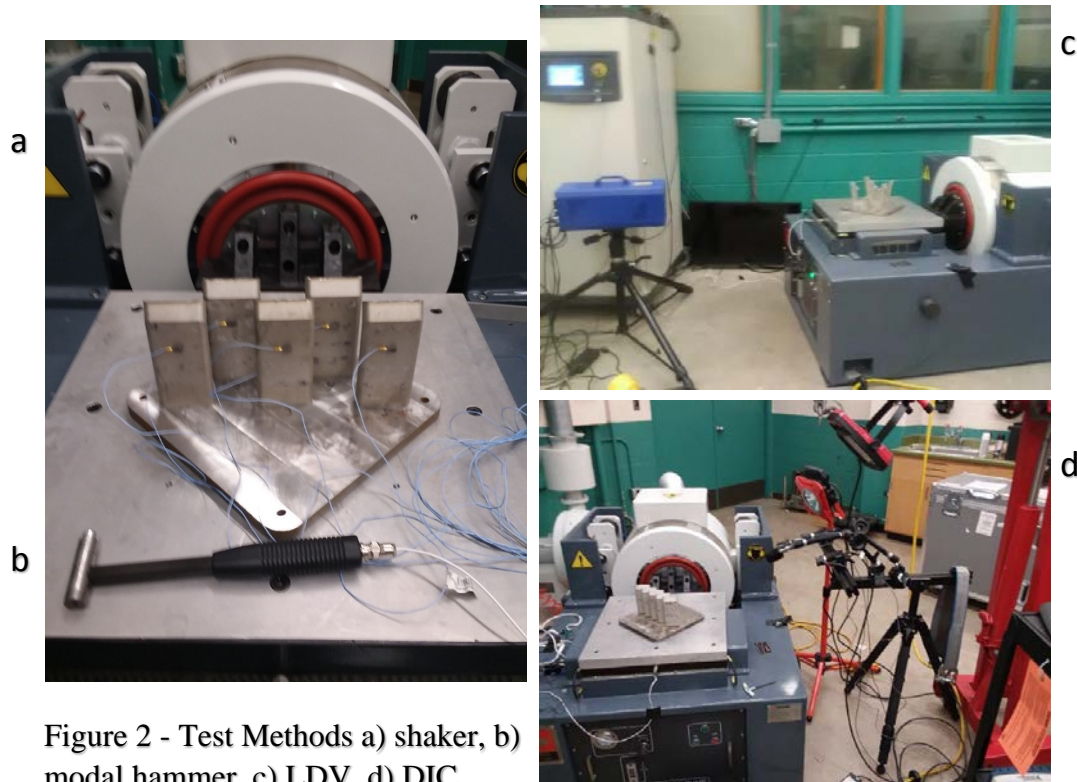
The preferred method to test a part using this modal analysis method is with free-free boundary conditions. Boundary conditions of this type would mean that the parts must first be separated from the build plate and then tested individually, this could add significant cost and time to the evaluation process. To save time and cost it is then advisable to test the parts while still attached to the build plate as multiple parts are tested simultaneously with this approach.

2. Testing Methods

The response methods used to measure the part response included several different technologies. The different technologies were chosen based on ease of use, quality of measurement, ability to discern small changes in mode shapes, and speed of testing. Accelerometers were used with both the impact hammer and the shaker input, these tend to be easy use and relatively low cost but are more time consuming in most scenarios if multiple locations are to be measured. A Laser Doppler Vibrometer (LDV) was used to measure the response velocity on the parts while shaker testing, the laser had the ability to automate the scanning of responses at many different locations making it easy to measure a large number of different response locations. However, the LDV is an expensive piece of equipment. A pair of Digital Image Correlation (DIC) cameras measuring the response displacements of a grid pattern painted on the parts was also used. The DIC is the fastest method to evaluate a large number of measurement locations as all locations are measured simultaneously. The DIC used in this case had a frame rate of only 60 Hz and thus required a sinusoidal input to the shaker which is a slow method to excite many different frequencies to find natural frequencies. A set of high frame rate cameras is being investigated to speed this testing, with an obvious equipment cost penalty.

The excitation methods used for exciting the AM builds were a modal impact hammer, which is an instrumented hammer that measures the force input to a structure or a large electrodynamic shaker as seen in Figure 2. The input signal to the shaker varied based on the technology used to measure the response of the parts. The shaker signals included periodic

frequency sweeps for the accelerometers and LDV measurements and a sinusoidal input for the DIC measurement.



3. AM Build Testing

All build plates were provided to MTU as blind specimens without MTU being informed of what or where a defect was until after testing and presentation of results to the sponsor.

3.1 Chimney 1

The first AM build test consisted of five chimneys oriented as shown in Figure 3. One chimney was printed with a defect, it was unknown prior to testing what/where the defect(s) was. The chimneys were not printed solid to the build plate but rather with a support lattice structure, as shown in Figure 3. The support structure was developed to make it easy to remove the parts from the build plate after dynamic testing.

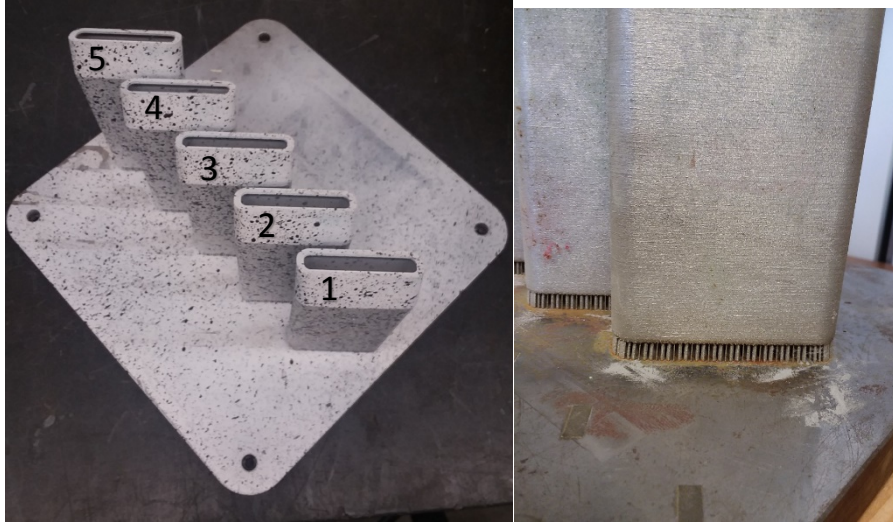


Figure 3 - 1st AM Build test – 5 chimneys

The Impact Hammer testing found the resonant frequencies of each chimney to be different by ~10 Hz and they were in order across the plate as numbered at some modes as shown in Figure 4. It was determined that the AM printed parts had a slight difference in mass or stiffness due to their position on the build plate. At other modes, chimney #4's resonant frequency shifted to a lower frequency, out of order with its position on the build plate. This indicated a more significant property difference than that caused by the location on the build plate, as shown in Figure 4.

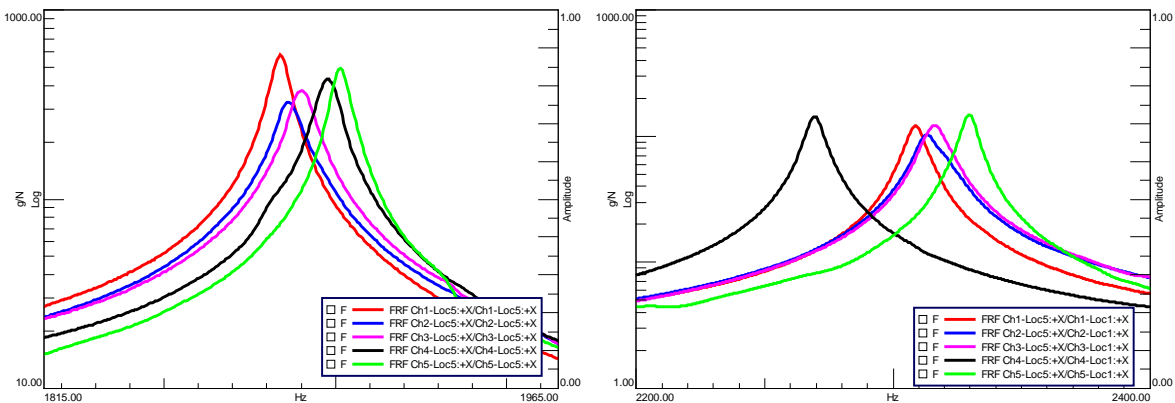


Figure 4 – FRF of Hammer Test Chimney Build 1 at 1900 Hz mode and 2300 Hz mode

The shaker testing agreed with the impact testing and also identified that the resonant frequencies of chimney 4 shifted lower at some modes as shown in Figure 5.

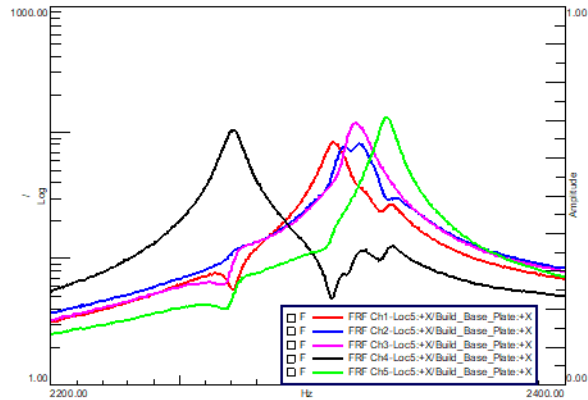


Figure 5 – FRF of Shaker Test Chimney Build 1 at 2300 Hz mode

Impact FRF's measured with a modal hammer are inherently less noisy than shaker FRF's. This can be seen in Figure 6. Therefore, it is desirable at times to use the modal hammer for finding the resonant frequencies. The difficulty with impact testing on AM parts still bolted to the base plate, is the modes of the base plate show up in the response of the parts on the build if the plate is tested in a free-free condition. This can be seen by the additional peaks (natural frequencies) in Figure 6 right hand graph that are not in the shaker data in Figure 6 left hand graph.

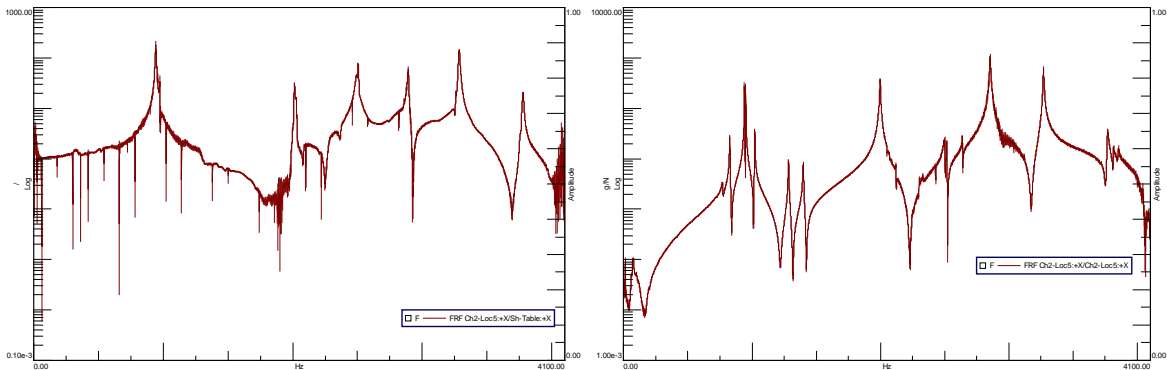


Figure 6 – FRF of Shaker LH vs Hammer free-free RH from 0-4100 Hz.

To acquire a clean FRF without the extra build plate modes, the build plate was bolted to the thick base plate of the shaker for an impact test. This moved the base plate modes higher in frequency due to the added stiffness of the shaker baseplate and out of the frequency bandwidth of interest as shown in Figure 7.

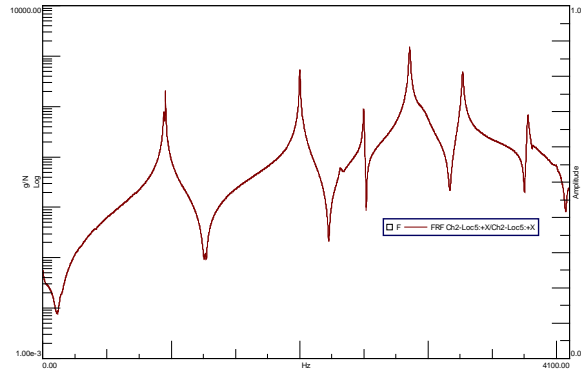


Figure 7 – FRF of Hammer bolted to shaker bed plate.

The LDV Test also showed that the natural frequencies of chimney 4 were lower at some modes. In Figure 8, the natural frequency of chimney 4 is approximately 50 Hz lower at the 2300 Hz mode, also shown is a still image of a video of build 1 shaking at chimney 4’s resonant frequency of 2289 Hz. Chimney 4 had significant amplitude in this picture while the other chimney’s exhibited much lower response amplitudes.

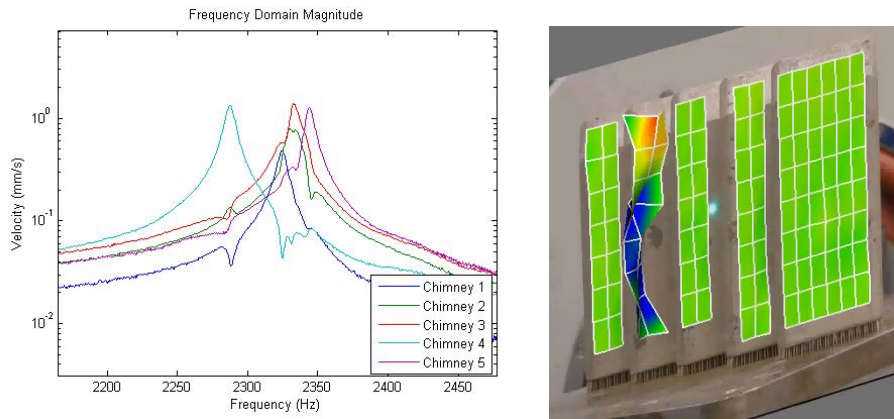


Figure 8 – LDV data at 2300 Hz mode. Image from video at 2289 Hz.

DIC was able to determine chimney 4 has resonant frequencies that are lower at some of the modes. Since the cameras used are only capable of 60 fps maximum the test was not able to be done with a frequency sweep. The resonant frequencies were previously identified through an impact test for each mode, these frequencies were then used as shaker inputs to evaluate the mode shapes and response amplitudes. By exciting the plate at the natural frequency of the “good” parts, any parts not exhibiting large amplitudes would be the “bad” parts. Likewise, if it were excited, as in this case, at the natural frequency of a “bad” part the “good” parts would not exhibit large response. The still images from videos in Figure 9 were taken at 2288 Hz which is one of the resonant frequencies of chimney 4 as shown in Figure 8.

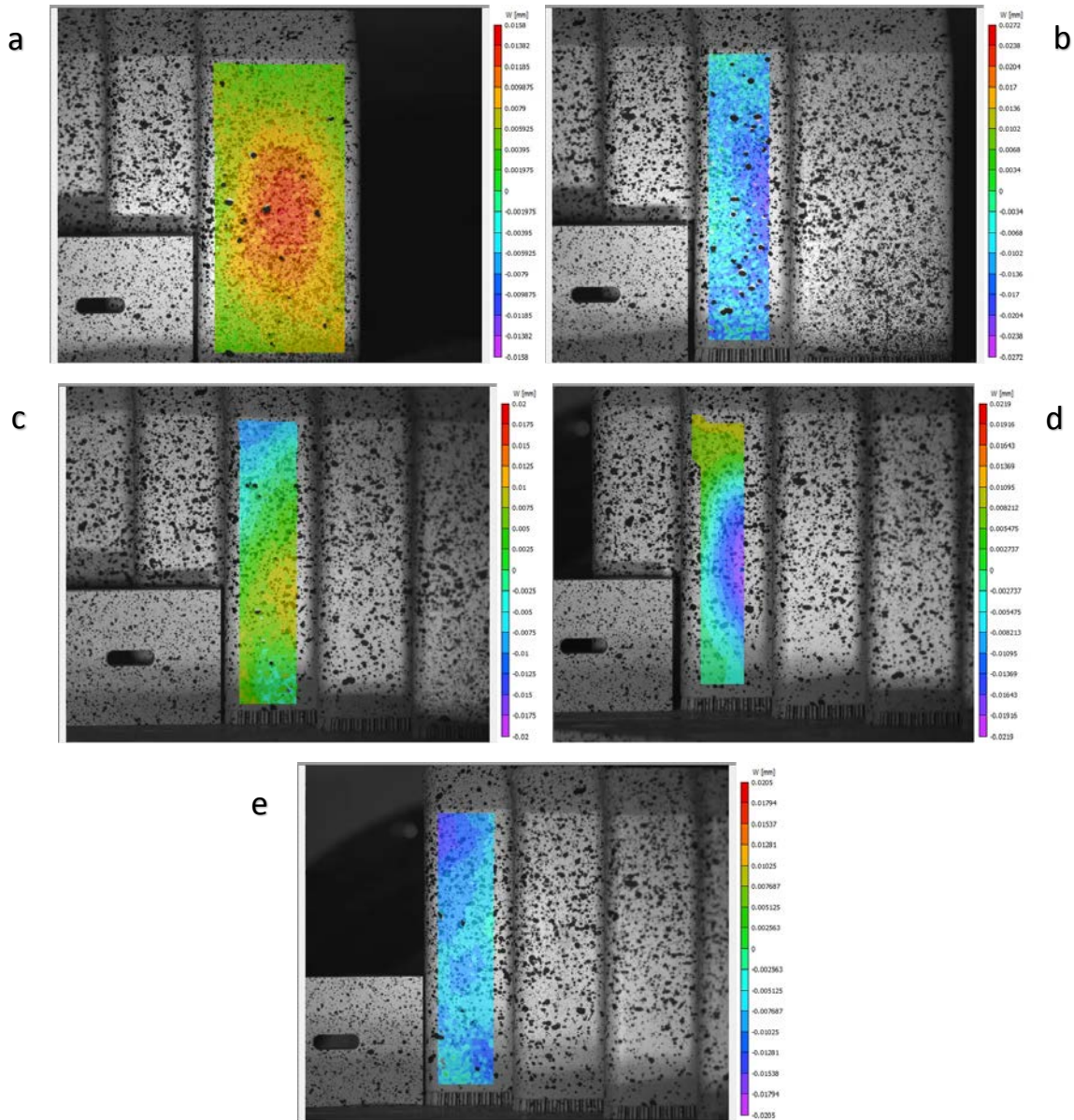


Figure 9 – DIC at 2289 Hz. a) Chimney 1, b) Chimney 2, c) Chimney 3, d) Chimney 4, e) Chimney 5

The displacement of chimney 4 is high at this frequency and the mode shape replayed from the data visually more closely matches the true mode shape than do the other chimneys. The mode shape of chimney 1 is the next closest match as it is the next closest frequency to 2288 Hz.

The chimneys on the first build were printed with support material as shown in Figure 5. During testing, the first resonant frequencies of the chimneys (the cantilever beam mode) changed between tests, this was identified to be a result of a fatigue weakening of the support material due to the vibration inputs used. This result led to the realization that modes heavily influenced by

boundary conditions may be of limited use in identifying parts with faults as the frequency could shift due to a boundary condition change and not a part fault.

Build 1 conclusions include that chimney 4 was shown to have lower natural frequencies at some modes. This was due to chimney 4 being denser. Chimney 4 was printed with 150 μ m hatch while chimneys 1, 2, 3, & 5 were printed with 85 μ m hatch. The AM build printer laser power for all chimneys was 200W.

3.2 Bracket Build

The second AM Build consisted of five brackets shown in Figure 10. It was unknown how many had defects and what the defects were. Each bracket was printed with a support structure to the base plate.

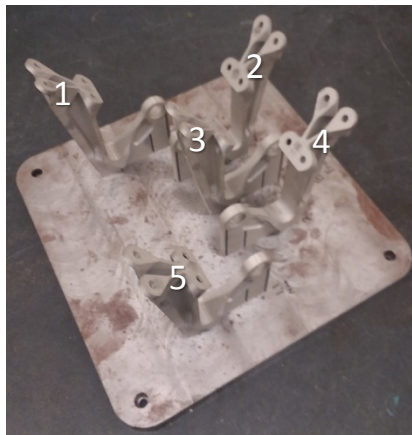


Figure 10 – Build with 5 brackets.

The impact hammer and shaker testing found the natural frequencies of brackets 1, 3, and 4 were within 1 Hz of each other while brackets 2 and 5 were higher at some modes as shown at the 600 Hz and 2250 Hz modes in Figure 11. This frequency difference increased with the higher frequency modes.

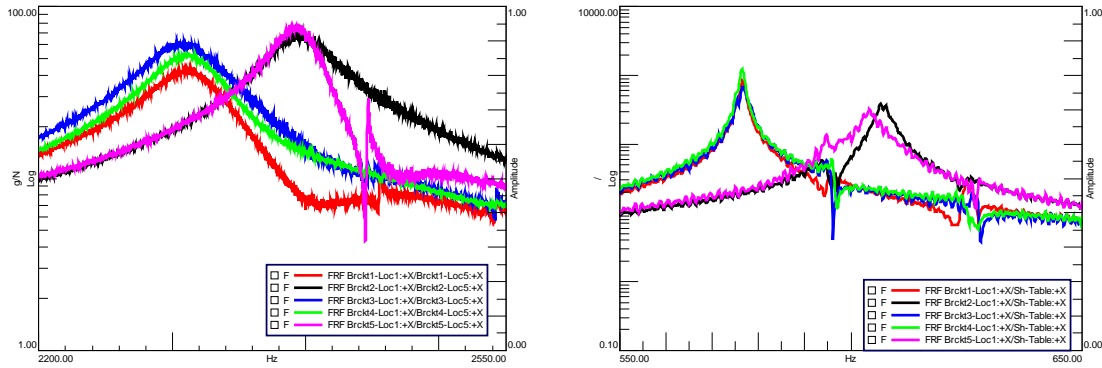


Figure 11 – Hammer FRF at 2300 Hz mode left and shaker FRF at 600 Hz mode right.

The Laser Vibrometer testing also found brackets 2 and 5 had higher resonant frequencies at some modes. In Figure 12, brackets 2 and 5 have a higher resonant frequency than brackets 1, 3, and 4.

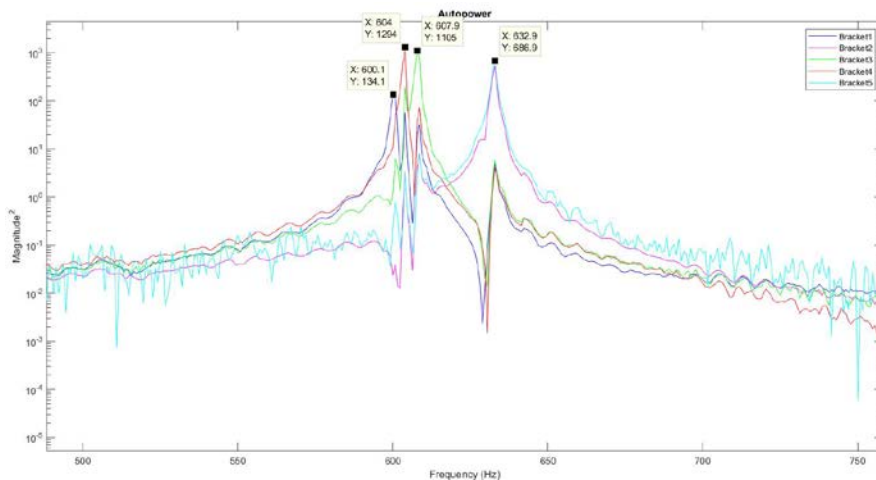


Figure 12 – LDV showing brackets 2 and 5 at higher resonant frequency at 600 Hz mode.

Bracket build conclusions include brackets 2 and 5 were shown to have higher natural frequencies. This was due to being less dense. Brackets 2 and 5 were printed with 85µm hatch where brackets 1, 3, & 4 were printed with 150µm hatch all with 200W printer laser power.

3.3 Chimney Build 2

The second chimney build consisted of five chimneys printed solid to the base plate and in the orientation shown in Figure 13. Again, it was unknown which chimneys had defects.

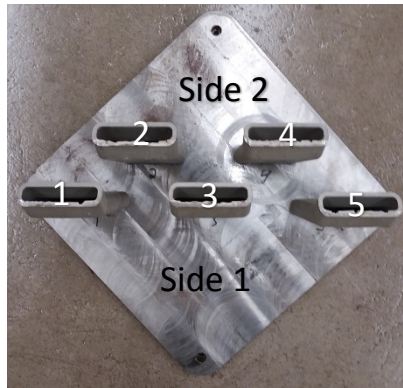


Figure 13 – 2nd chimney build.

At the first bending mode, approximately 950Hz as shown in Figure 14, chimneys 1 & 2 are coupled together similar to a tuned-mass-damper. Their resonance appear to be at the exact same frequency and they were moving 180 degrees out of phase, based on the mode shapes (not shown here). The stiffness and mass properties of chimneys 1 & 2 that affect this mode are the same. All of the chimneys also have sympathetic frequencies from the other chimneys' natural frequencies. This is shown by the fact that each chimney's FRF has a small peak at the natural frequencies of the other chimneys along with the large peak at their natural frequency. This makes the first bending mode difficult to use to find defects. In addition, chimney 3 seemed least affected by the others.

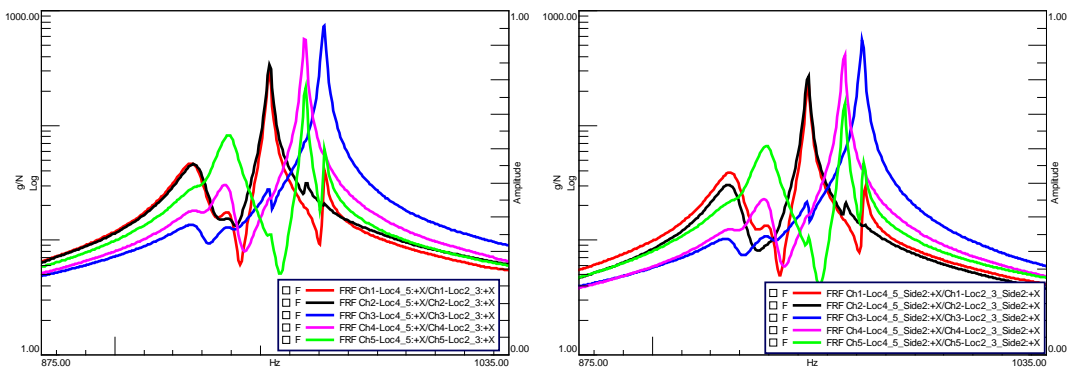


Figure 14 – 2nd chimney build side one left and side 2 right at 1st bending mode 900 Hz.

At approximately 2000 Hz there is a breathing mode and torsion mode close to each other as shown in Figure 15. These two modes are the closest and interact the most on chimney 2. The breathing mode for chimney 2 has seemed to shift to a higher frequency causing this interaction.

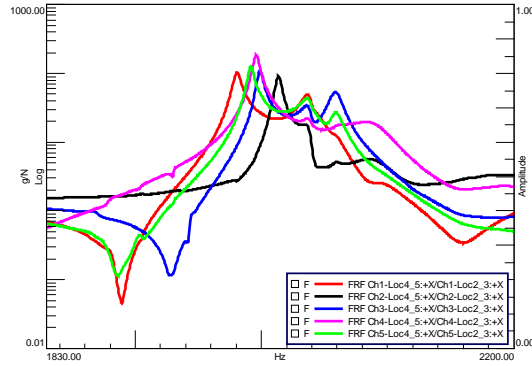


Figure 15 – 2000 Hz mode

At the 2850 Hz mode chimney 2 has shifted to an even higher frequency as shown in Figure 16.

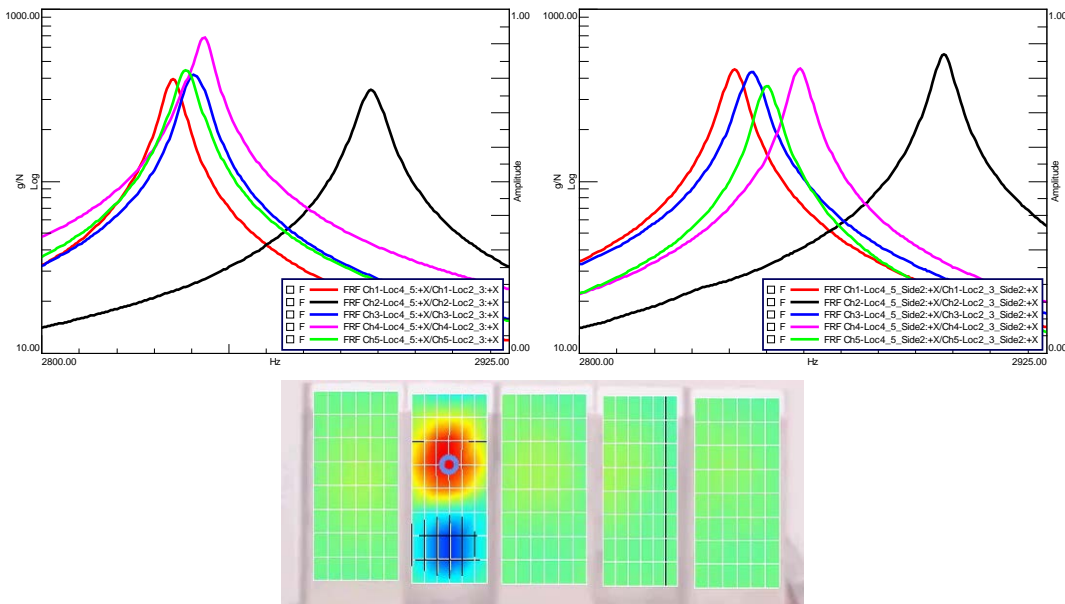


Figure 16 – 2850 Hz mode hammer data. Side 1 left and side 2 right. Image from LDV video bottom at chimney 4’s resonant frequency ~2890 Hz.

In a comparison of Side 1 and Side 2 at 2000Hz as shown in Figure 17, it can be seen in the FRF’s taken on side 1 and side 2 of the chimneys that there are differences between the chimneys. Chimneys 1 and 4 are nearly identical side to side while chimneys 2 and 5 are not as similar. Chimney 2 shows the largest differences between the two sides.

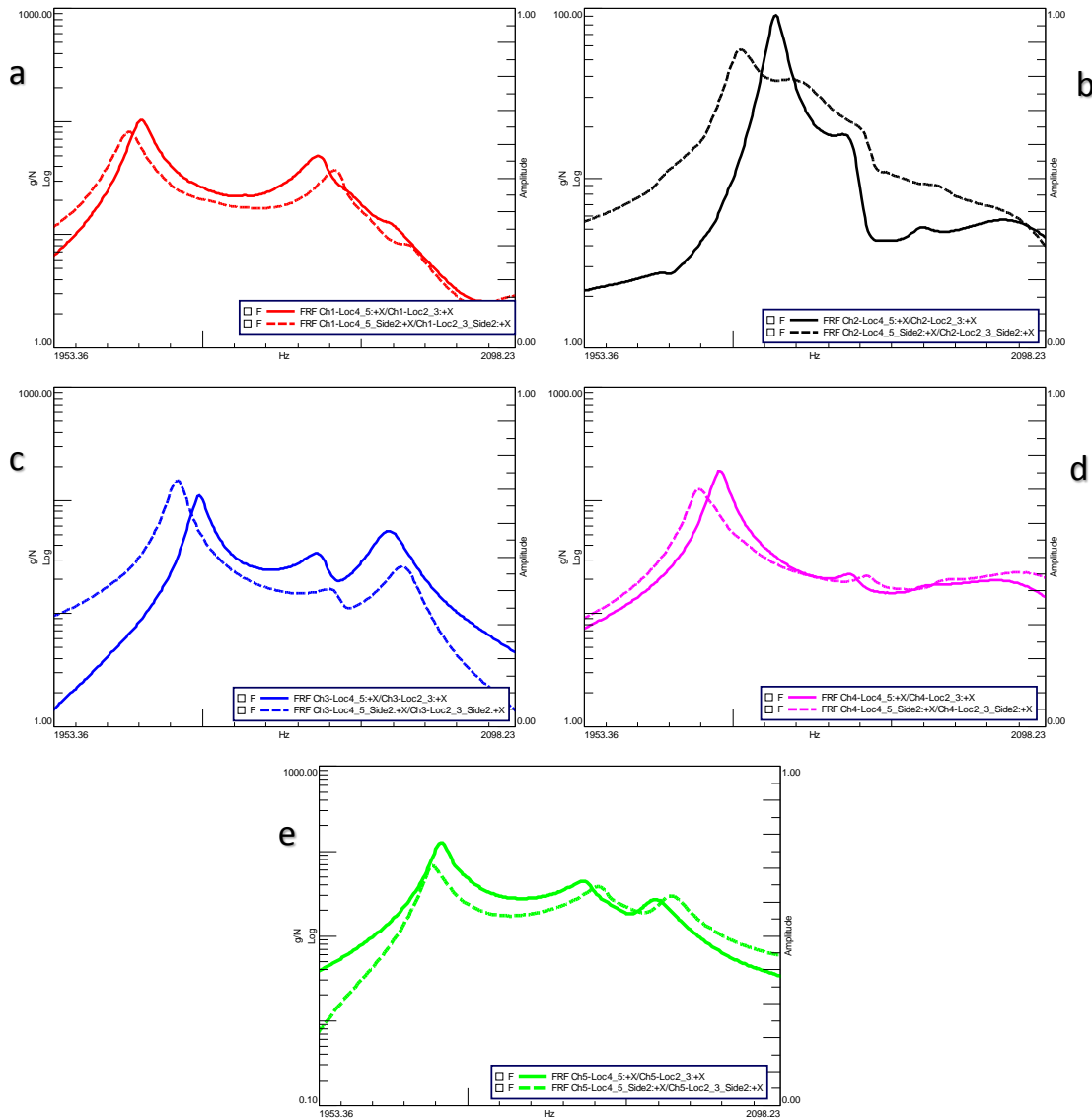


Figure 17 – 2000 Hz mode side to side comparison. a) Chimney 1, b) Chimney 2, c) Chimney 3, d) Chimney 4, e) Chimney 5,

Chimney 2 build conclusions include the observance of sympathetic frequencies showing up on all chimneys indicating the first bending mode is not an ideal mode to use for finding defects. At the 2000 Hz mode there is a difference in chimney 2 that moved the breathing mode and torsion mode closer together than in the other chimneys. At the higher frequency modes, above 2800 Hz, chimney 2 starts to show significantly higher resonant frequencies relative to the other chimneys for a given mode. This frequency difference also seems to be greater side to side on chimney 2 than the other chimneys. The higher resonance frequencies in chimney 2 are due to it being built with 85 μm hatch while the other chimneys were 150 μm hatch. All with 200W printer laser power.

4. Final Conclusions

Frequency response inspection can be used to find defective parts on an AM build. Because of the support material failures and the sympathetic frequencies seen in some build plates, the lower frequency modes may not be ideal for finding faults. The higher frequency modes are not as affected by the support material or sympathetic frequencies of the other parts on the build and appear to be a better option for detecting defects. This is due to the mass of the chimneys at the build plate not being a significant part of the mode shape at some of the higher frequency modes. Since the higher modes have smaller local movements they are more sensitive to small stiffness/mass differences and therefore more useful in finding smaller defects.

It was also shown that the boundary conditions of the base plate can have a large effect on the ability to easily identify part natural frequencies and therefore identify part defects. Before using any of the excitation and/or response methods investigated a firm understanding of the part and build plate dynamics, natural frequencies and mode shapes, are necessary to successfully be able to identify part defects.

Several different types of parts with different sizes and types of faults were evaluated in this study. The range of parts/faults evaluated shows that this dynamic evaluation method has significant potential to find “bad” parts but further development is required to bring it to a viable commercial implementation.

4. References

- [1] R. J. Allemang. *Vibrations; Experimental Modal Analysis*. Structural Dynamics Research Laboratory, Department of Mechanical Engineering, University of Cincinnati, Ohio, USA, (2008)
- [2] R. J. Allemang. *Vibrations; Analytical and Experimental Modal Analysis*. Structural Dynamics Research Laboratory, Department of Mechanical Engineering, University of Cincinnati, Ohio, USA, (2008)
- [3] J. R. Blough, K. Johnson, A. Barnard. *Survey of Dynamic Measurements to Assess Additive Manufacturing Part Integrity*. White paper submitted to Honeywell NSC-KS (2018)
- [4] I. Gibson, D. W. Rosen, B. Stucker. *Additive Manufacturing Technologies Rapid Prototyping to Direct Digital Manufacturing*. Springer, New York (2010)
- [5] Frazier, W. E., *Metal additive manufacturing: a review*. J. of Materials Engineering and Perform (2014) 23: 1917. <https://doi.org/10.1007/s11665-014-0958-z>
- [6] A. Karne, , A. Kallonen, V. Matilainen, H. Piili, A. Salminen. *Possibilities of CT scanning as analysis method in laser additive manufacturing*. Physics Procedia, ISSN: 1875-3892, Vol: 78, Page: 347-356 (2015)
- [7] M. A. Sutton, J. J. Orteu, H. Hubert. *Image Correlation for Shape, Motion and Deformation Measurements: basic concepts, theory and applications*. Springer, New York (2009)
- [8] Brown, B. Hartwig, T. Patent Application No.14/941,258, submitted Nov, 2015.