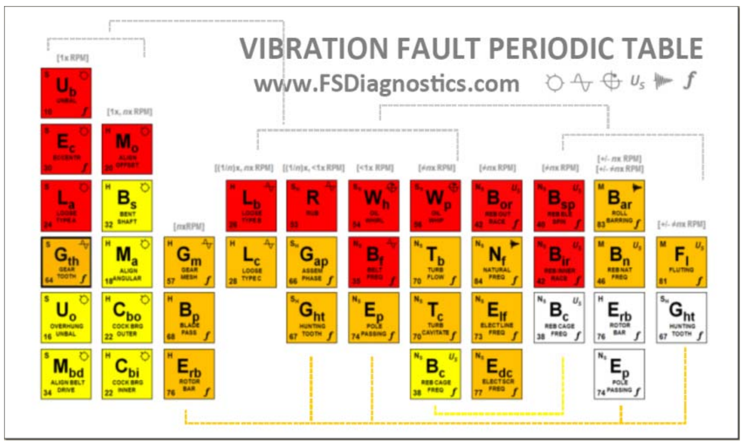
VIBRATION ANALYSIS CASE HISTORY



Dan Ambre is the founder of Full Spectrum Diagnostics, (June of 2000) providing the finest in Machinery Condition Monitoring, Specialized Analyses, and Vibration Training for Industrial, Manufacturing, Aerospace, and Automotive applications.

Dan is a graduate of ***The University of Iowa*** with a Bachelor's degree in Mechanical Engineering, and has completed additional graduate level course work in Engineering Dynamics from ***The University of Illinois at Chicago***, and ***Florida Atlantic University***.

Dan has over 30 years experience in the aviation and aerospace industries with a background in Vibration & Acoustic testing, Dynamic analysis of high speed rotor systems, Experimental Modal, and Finite Element Analysis. His consulting vibration experience base comes from positions at Sundstrand Aviation Corporation, Pratt & Whitney (United Technologies Corporation), and Technical Associates of Charlotte.

Dan is a Certified Level III / CAT IV. Vibration Training Instructor and a registered Professional Engineer in Minnesota.

The TEST ARTICLE

Machine Class: Engine (Rocket Type)
Rotating Speed: 36,000 RPM
Life Cycle: 600 seconds

PROBLEM STATEMENT:

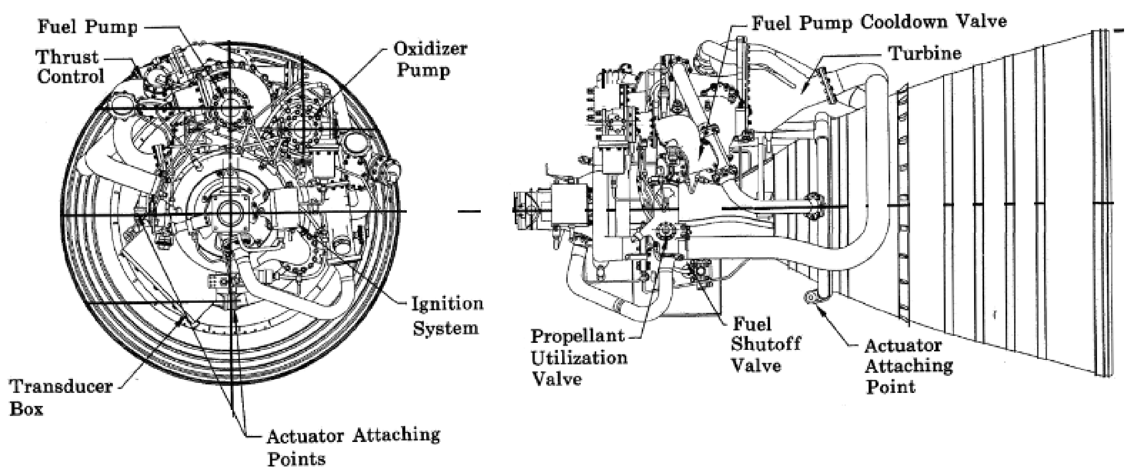
This rocket engine application was being evaluated following the modification of its extendable nozzle system. The unit was required to be resonance free in the launch vehicle “boost” flight phase.

NOZZLE :

1. Nozzle “Stowed” and locked position (Boost Phase)
2. Nozzle Free and Translating condition (non-operating)
3. Nozzle “Deployed” and locked position (non-operating and operating).

TEST REQUIREMENTS / DOCUMENTATION:

1. Natural Frequencies
2. Mode Shapes
3. Modal Damping
4. Resonance Margin



The NATURAL FREQUENCY & RESONANCE PHENOMENA

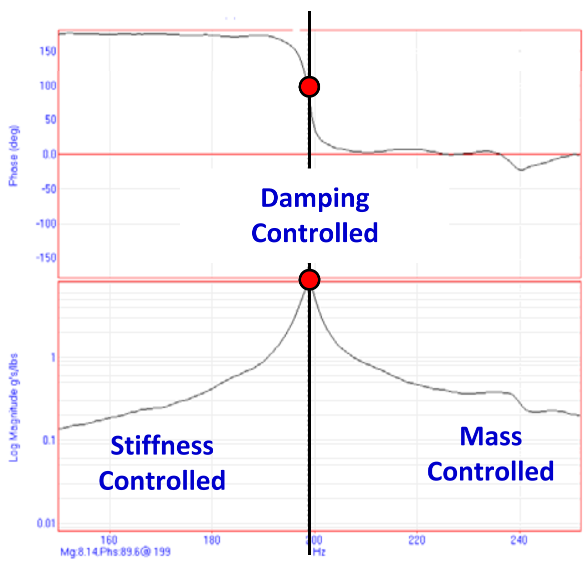
The Resonance condition creates a Phase response that is a function of the forces generated in a linear "balanced" system.

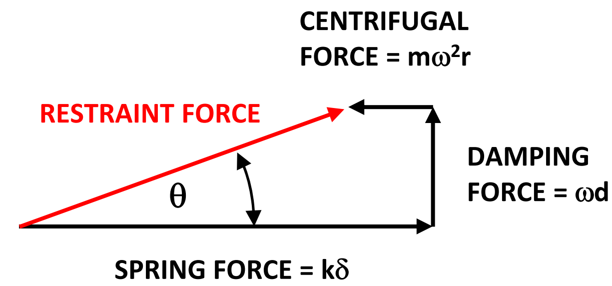
The **Physical** "structure" includes **Mass**, **Stiffness**, and **Damping**.

The **Dynamic** system includes motions in the form of **Displacement**, **Velocity** and **Acceleration**.

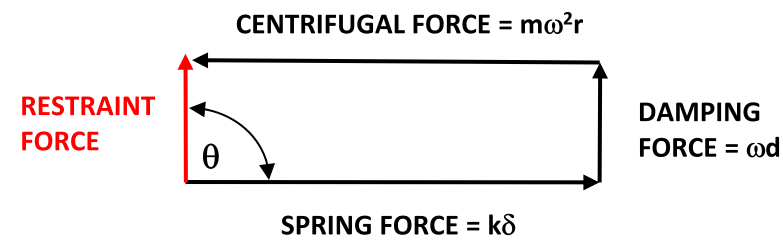
The **Forces** generated are an **Inertial** (centrifugal) force, a **Spring** (stiffness) force and a **Damping** (dissipation) force.

The diagram to the right shows the balance in forces as this system transitions into a resonance condition. Note the initially small phase angle shifts 90 degrees at resonance and includes another 90 degree shift (approaching a total 180 degrees) above resonance.

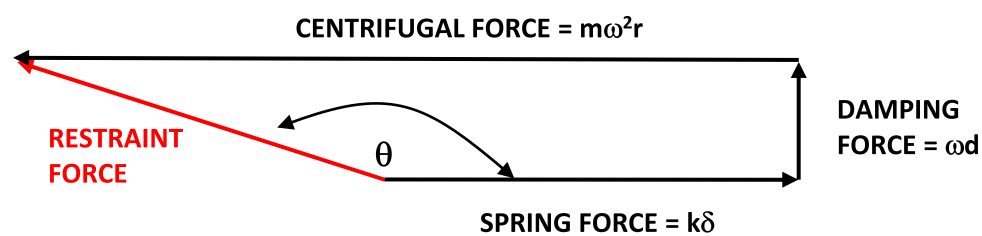




BELOW RESONANCE: STIFFNESS CONTROLLED



AT RESONANCE: DAMPING CONTROLLED



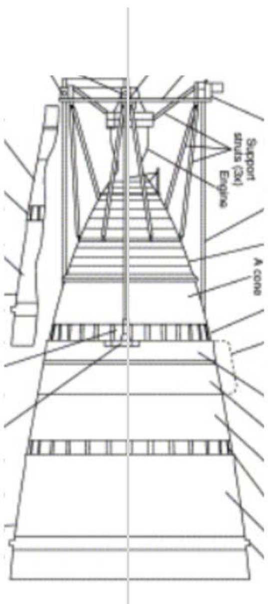
ABOVE RESONANCE: MASS CONTROLLED

Natural Frequency Testing: The Model

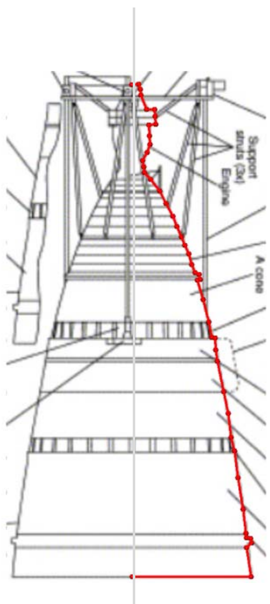
The Structural Model is constructed to approximate the physical structure. Remember that this model is **NOT** a Finite Element Analysis (FEA) model. The Experimental Modal Analysis is a purely empirical analysis. That is, the results are a result of entirely "measured" data.

Software drawing tools allow relatively quick construction of complicated structures. In this case, the symmetry of the rocket engine nozzle is used to "sweep" the centerline profile into the final three dimensional structure.

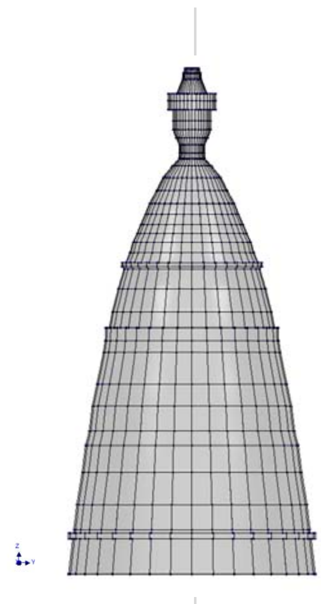
Note that the EMA measurement locations (points) are created by this method. Also, if a Finite Element Model is desired, the software drawing tools can easily convert the "drawing" surfaces to FEA surfaces. The FEA analysis is a purely analytical solution based on geometry, NOT physical measurements.



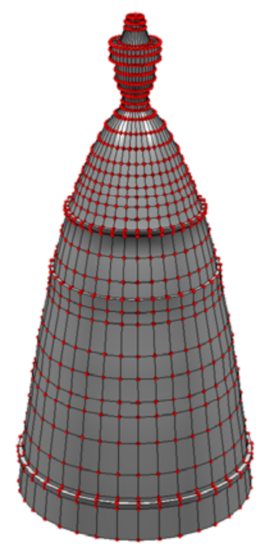
Cross-Section



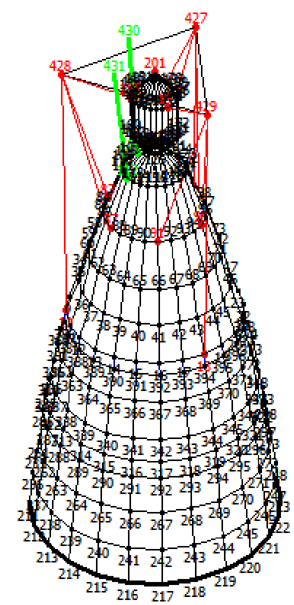
Profile



Sweep Profile (Revolve)



3-D Structure



Measurements

Natural Frequency Testing: Measurement Grid

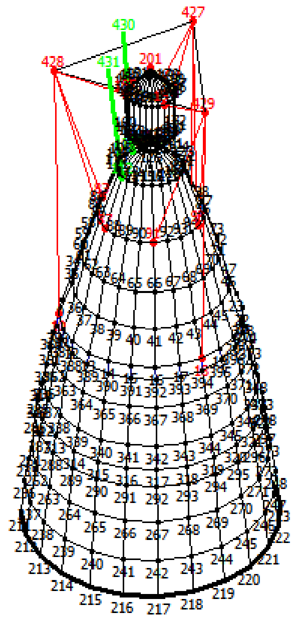
The model construction phase is a good time to consider the appropriate Measurement Grid for your analysis.

Measuring a Natural Frequency in a physical structure requires only one measurement.

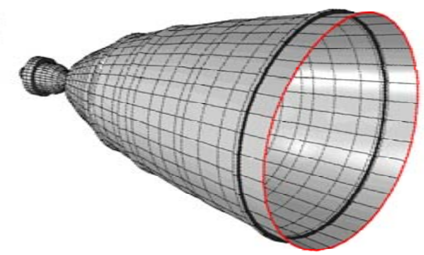
Measuring a Mode Shape in a physical structure requires a enough resolution (measurement points) to resolve the shape. In this analysis, each 360-degree nozzle "ring" included twenty-four (24) radial measurements (every 15-degrees).

Even with this measurement grid density the highest mode shape that can be resolved is the "star" pattern noted to the right.

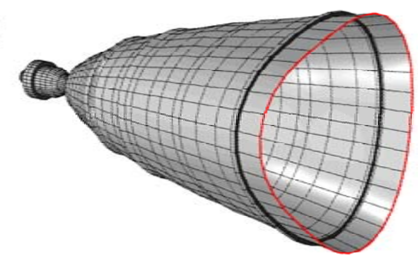
Higher natural frequencies can be measured by adjusting the frequency range, but higher frequency (complexity) Mode Shapes requires additional physical measurements.



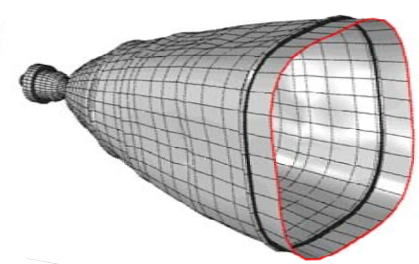
Oval



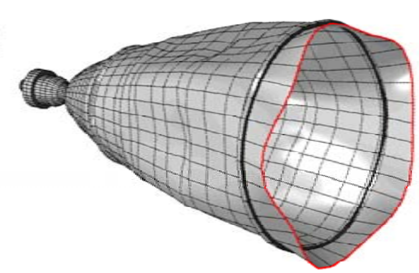
Tri-angle



Square



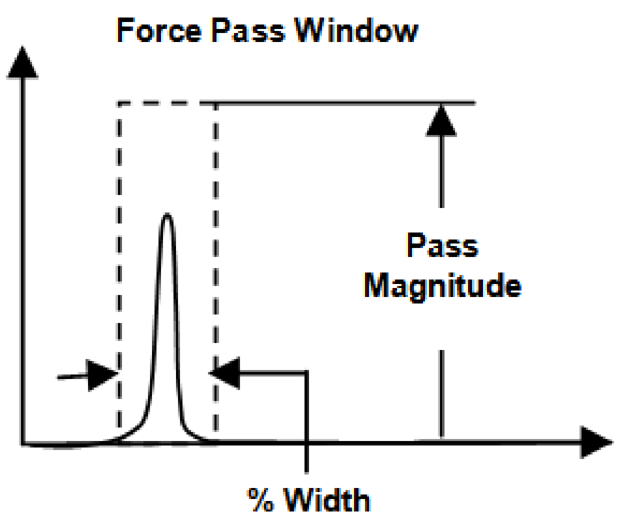
Star

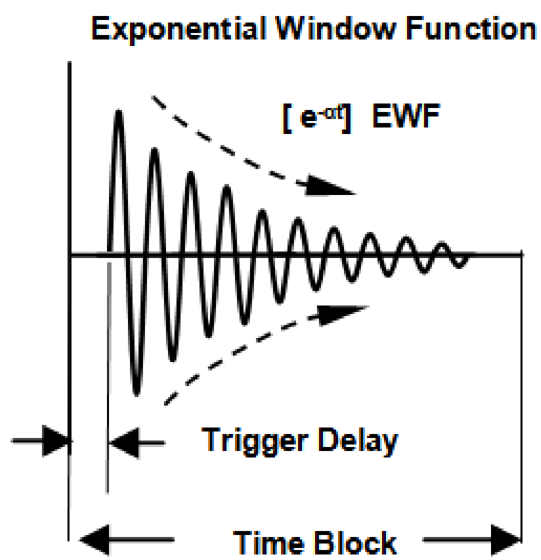


Natural Frequency Testing: IMPULSE Method

Experimental Modal Analysis accuracy relies on the application and capture of a single properly scaled impact from the test article. The size of the hammer (mass), the type of tip (hardness) used will determine the usable frequency range and the ability to detect lightly or heavily damped natural frequencies. The length of the time block will determine the frequency range in the analysis. The time block can be modified by changing the F_{MAX} and # of FFT Lines in the analysis setup.

The impact transient must be wholly contained within the time block, which will include a trigger delay and the "ring-down" of the transient response. For lightly damped systems the ring-down can be artificially forced to decay by use of an Exponential Window Function (EWF). The added damping should be considered when the final data is curve-fit and damping estimates are made. A Force Pass Window is a narrow band pass function used to reduce or eliminate noise in the Impact time waveform.





Natural Frequency Testing: Instrumentation

Massive Structures:
Lower Frequency / Heavy Hammer / Soft Tip
Floors, Walls, Building Frames, Inertial Blocks & Pedestals

Intermediate Structures:
Mid-range Frequencies / Medium Hammer / Medium-Tip
Framework, Fabricated Machine Bases, Heavy Components and Sub-Assemblies.

Light-weight Structures:
High Frequencies / Small Hammer / Hard-tipped
Turbne Blades, Gears, Piping, Light-weight Components and Sub-Assemblies

The **Rocket Engine Nozzle** was constructed of a Carbon-Carbon material (non-magnetic). The material is relatively thin (0.250 inches thick), hard and brittle. The best instrumentation choice is a small-sized hammer with a soft rubber tip.

The transducer of choice was a small tri-axial accelerometer that could be glued in-place. This testing was a "Roving Hammer Test".

PERFORMANCE

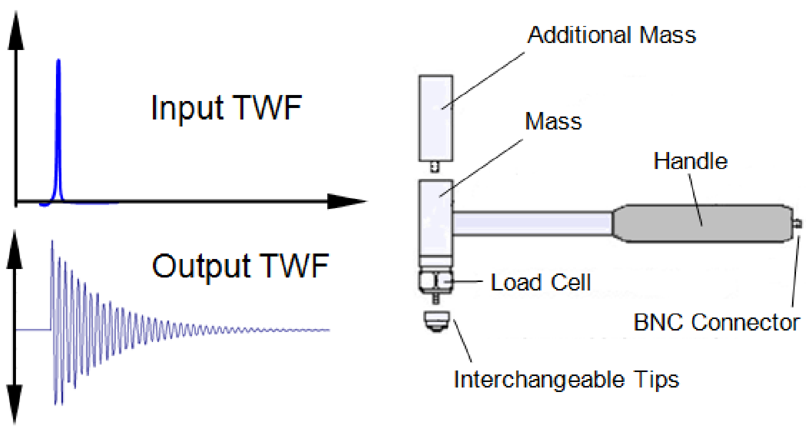
Sensitivity, ±10% [2]	10	mV/g
Acceleration Range	+/-500	Gpeak
Frequency Range, ±15%	.3-10,000	Hz
Frequency Range, ± 5%	.5 - 3000	Hz
Resonance Frequency	>24	kHz
Linearity [3]	1	%F.S.
Phase Response, ± 5°	2.3 to 3000	Hz
Transverse Sensitivity Max	5	%

SPECIFICATIONS
Sensitivity: (±15%) 1 mV/lbf (0.23 mV/N)
Measurement Range: ±5000 lbf pk (±22240 N pk)
Hammer Mass: 0.7 lb (0.32 kg)

Natural Frequency Testing: Acquisition Setup

The setup parameters are defined by the desired frequency range. It is recommended to define a time block in the 2 second range to reduce background noise.

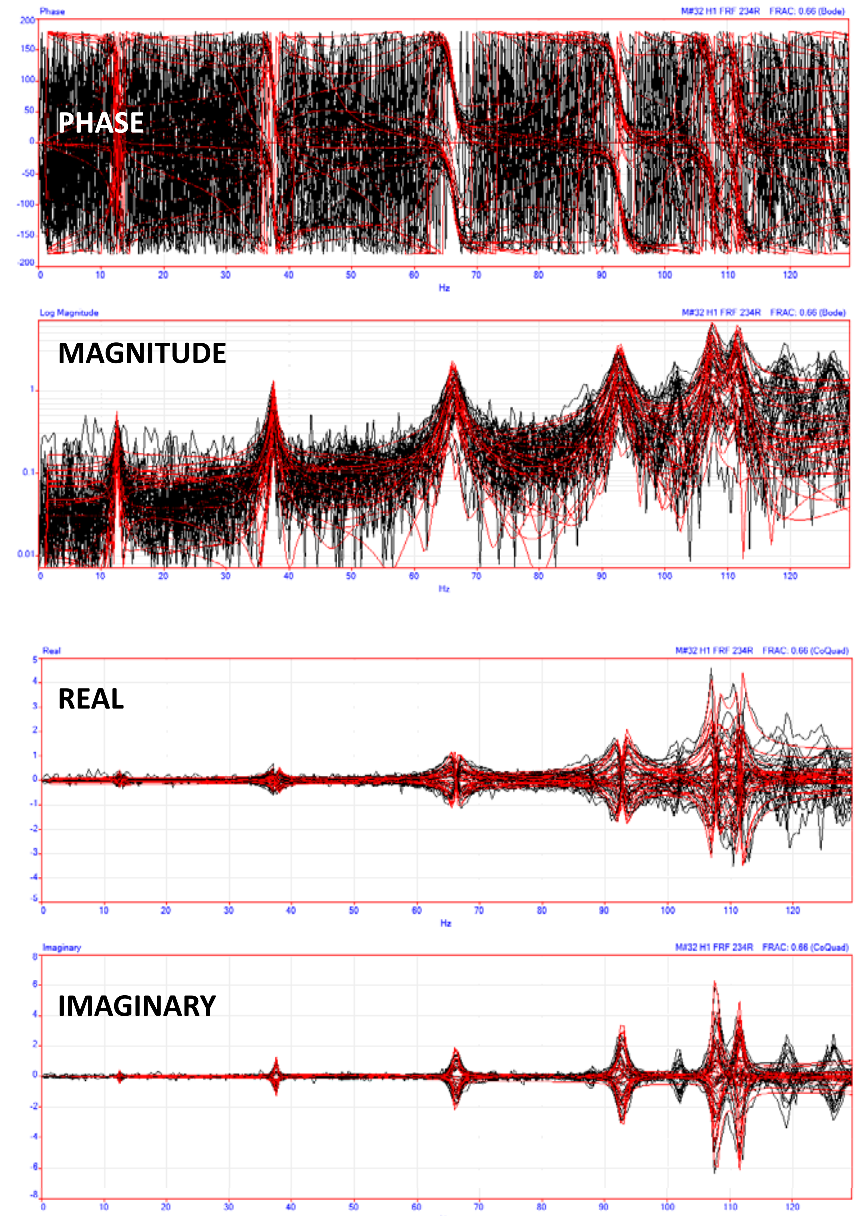
There are also options in many acquisition software for reducing noise on the input and/or output channels (H1, H2). Spend some time in the setup phase for better measurements.



Experimental Modal Test Instrument Setup Parameters

FMAX [CPM / Hz]	#LINES	SAMP RATE	#AVE	OVER LAP	RESOLUTION [CPM/LINE / Hz/LINE]	TMAX [SEC]	DELAY [SEC]
12,000 / 200	400	1,024	4 - 6	0%	30.0 / 0.500	2.00	-0.050
30,000 / 500	800	2,048	4 - 6	0%	37.5 / 0.625	1.60	-0.050
60,000 / 1000	1,600	4,096	4 - 6	0%	37.5 / 0.625	1.60	-0.050
90,000 / 1500	3,200	8,192	4 - 6	0%	28.1 / 0.468	2.13	-0.050

Note: Use of Trigger Delay is dependent on the Analyzer Setup Parameters.

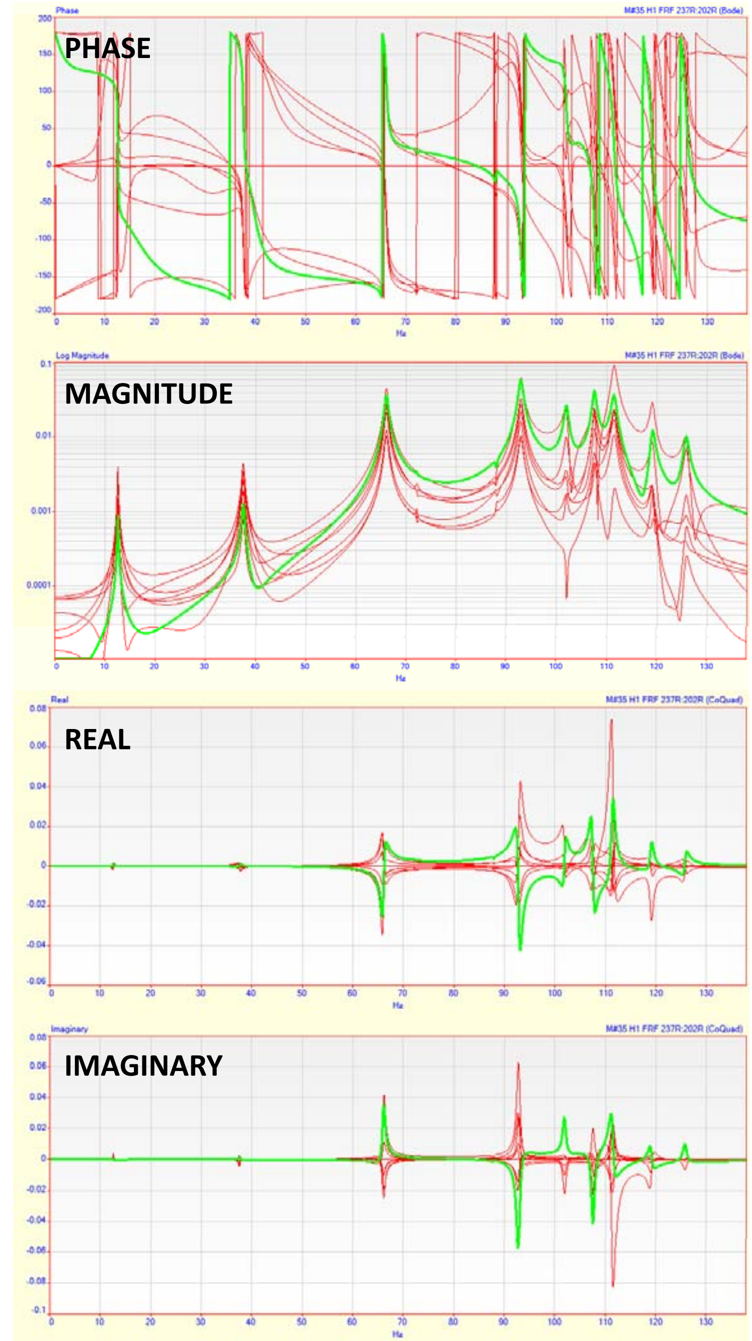


Natural Frequency Testing: Curve-Fit Measurements

Modal analysis is used to characterize resonant vibration in mechanical structures. Each resonance has a specific “natural” or modal frequency, a modal damping or decay value, and a mode shape. FRF-Based parameter estimation (or curve fitting) is used to estimate the modal parameters of a structure from a set of FRFs.

ME’scope Curve-Fitting Table

Select Shape	Frequency (or Time)	Damping	Units	Damping (%)	Label	MPC
1	12.55	0.081733	Hz	0.65123	Global-Poly	0.46277
2	37.554	0.23333	Hz	0.62131	Global-Poly	0.86549
3	66.066	0.35268	Hz	0.53383	Global-Poly	0.83472
4	72.153	0.15646	Hz	0.21684	Global-Poly	0.22931
5	87.911	0.22777	Hz	0.25909	Global-Poly	0.58547
6	92.862	0.41476	Hz	0.44664	Global-Poly	0.81477
7	101.94	0.39434	Hz	0.38683	Global-Poly	0.75748
8	107.53	0.37945	Hz	0.35288	Global-Poly	0.81848
9	111.45	0.47721	Hz	0.42818	Global-Poly	0.44497
10	119.13	0.40513	Hz	0.34008	Global-Poly	0.34223
11	125.82	0.45564	Hz	0.36214	Global-Poly	0.12838



The DAMPING FACTOR

Damping Estimation:

There are several damping estimation techniques derived from the modal (FRF) or the time plot data collected during an Experimental Modal Analysis.

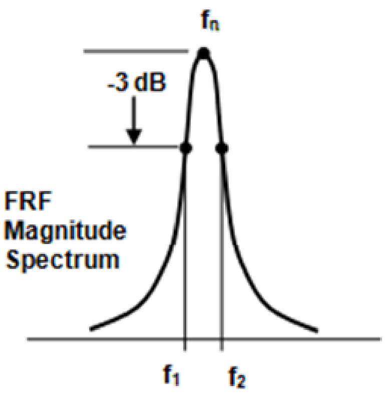
These methods allow the analyst to calculate the amplification factor (Q) and the percent critical damping of a system operating near resonance.

Half Power Point Method:

In this calculation, f_n is the natural frequency, f_1 and f_2 are frequencies above and below the resonance at the half power point or -3 dB (70.7%) amplitude down from the f_n peak. This method is only as accurate as allowed by the FFT spectrum resolution.

The ME'scope software will automatically calculate the damping as a part of the curve-fitting algorithm.

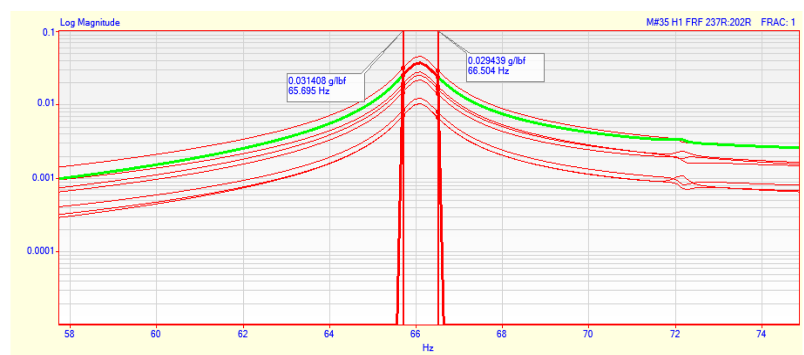
$$Q = \frac{f_N}{[f_2 - f_1]}$$



$$Q = \frac{[66.1 \text{ Hz}]}{[66.504 - 65.695]}$$

$$Q = 81.7 \text{ (Amplification Factor)}$$

$$\zeta = [1/2Q] = [1/163.4] = 0.612\%$$

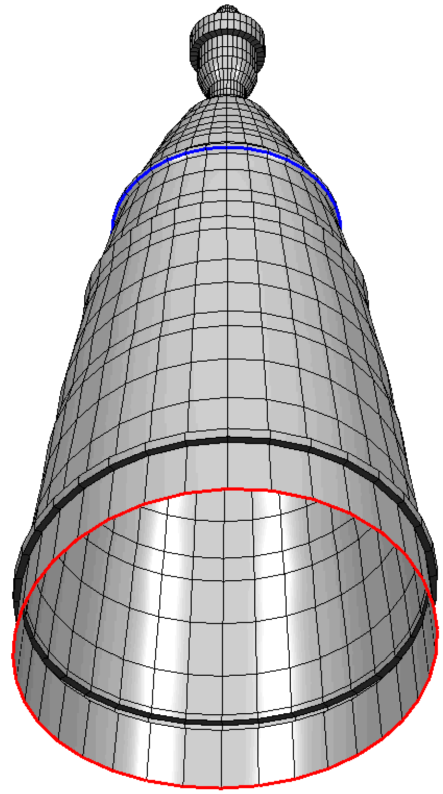


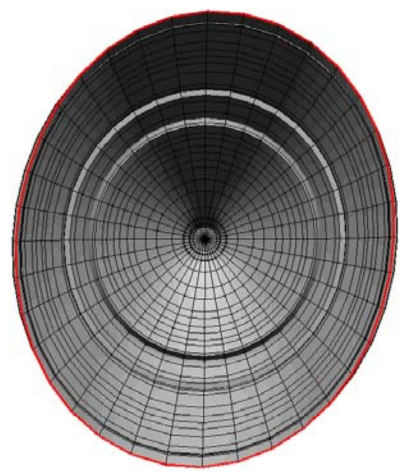
The MODE SHAPE

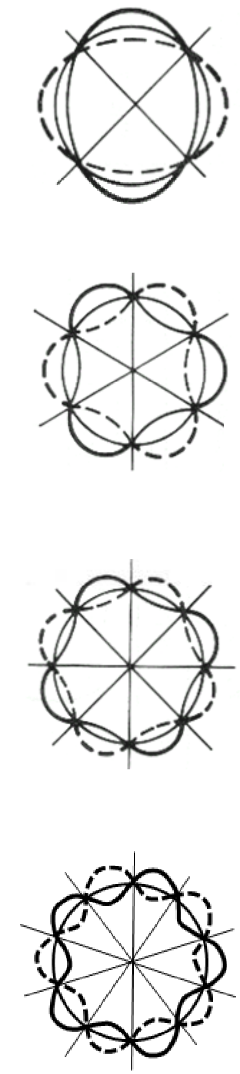
Modal Analysis is the process of defining a structure in terms of its modes of vibration. Each mode of vibration is characterized by a specific natural frequency in the structure, a modal damping (amplification) factor, and the physical mode shape itself. The mode shape is the 3-dimensional displacement of all locations on the structure due to the resonant response. Mode Shapes are divided into two classes, Rigid Body and Flexural Modes.

Rigid body modes are motions dominated by the machine or structures' mounting method. All rigid deformations are dependent on the flexibility of the supports (rocking, pitching, bouncing).

Flexural modes result from natural frequencies that tend to bend, twist, or otherwise distort the structures themselves. The examples below show basic modes of simple "ring" or "shell" structures. The mode shapes tend to be "Harmonic" multiples, however the natural frequencies associated with each shape are NOT harmonically related.







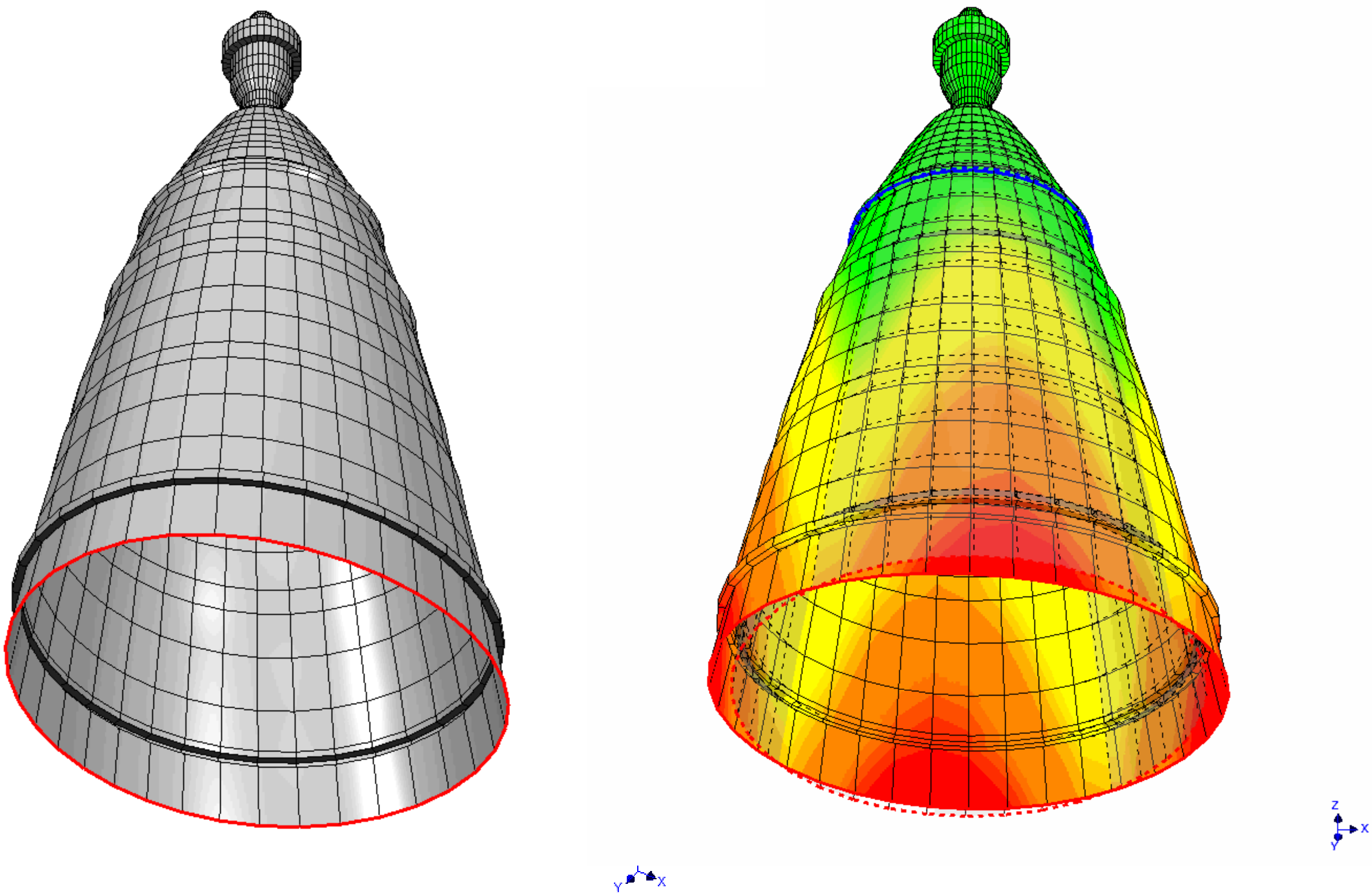
2 ND

3 ND

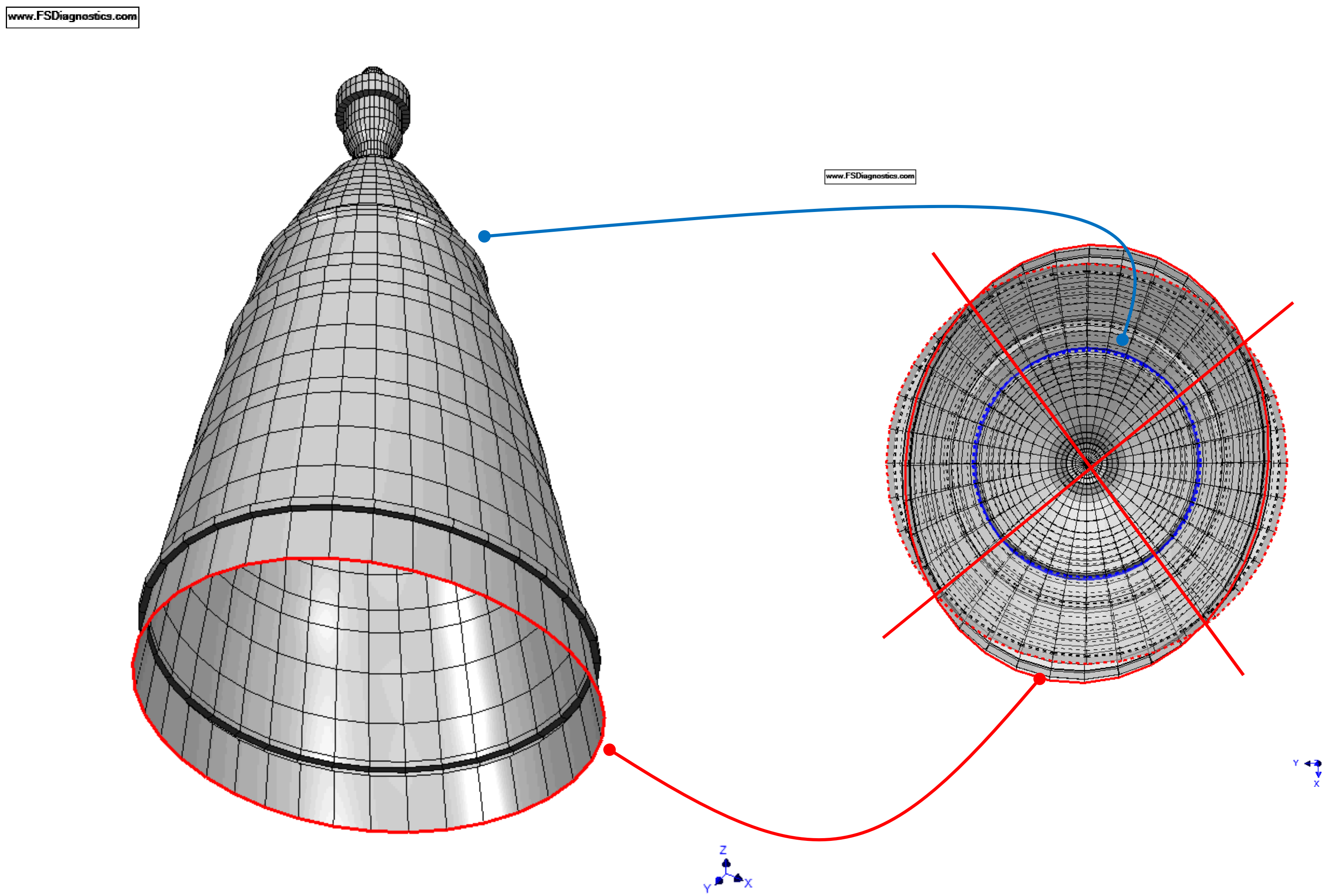
4 ND

5 ND

Natural Frequency: 12.5 Hz
Damping Factor : 0.65%
Mode Shape: Nozzle Exit 2ND (Shell Nodal Diameter)

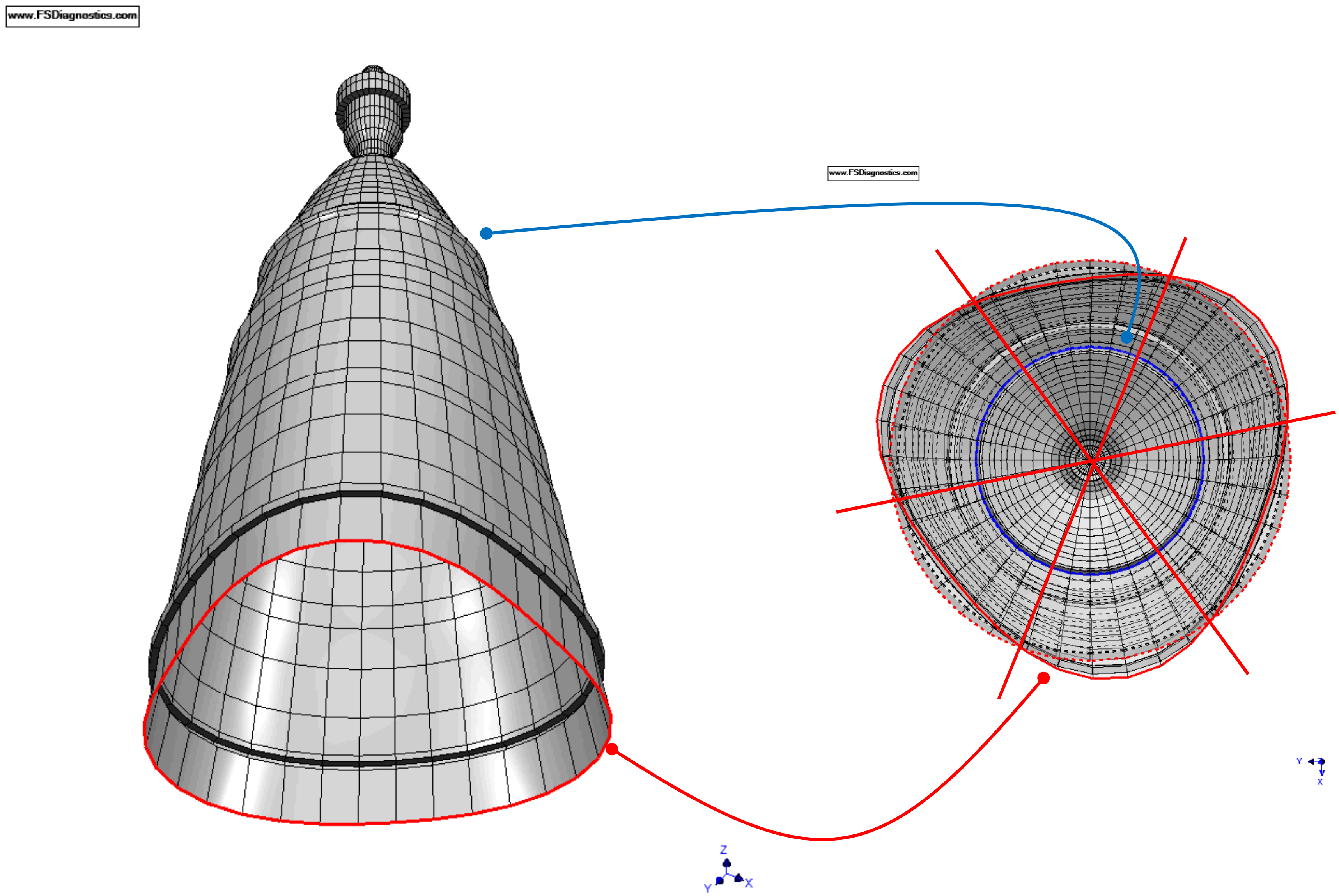


Natural Frequency: 12.5 Hz
Damping Factor : 0.65%
Mode Shape: Nozzle Exit 2ND (Shell Nodal Diameter)



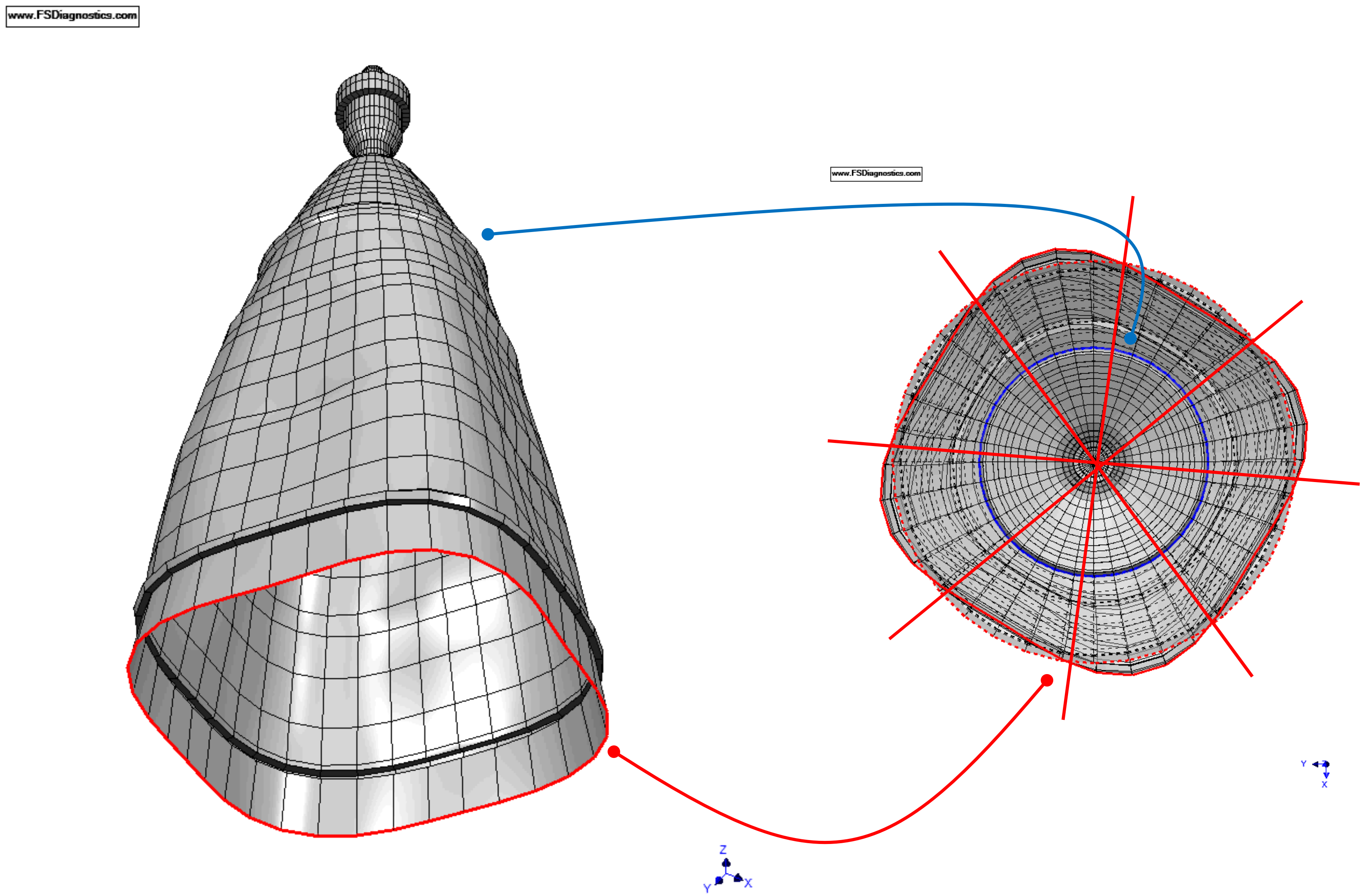
Full Spectrum Diagnostics

Natural Frequency: 37.5 Hz
Damping Factor : 0.62%
Mode Shape: Nozzle Exit 3ND (Shell Nodal Diameter)

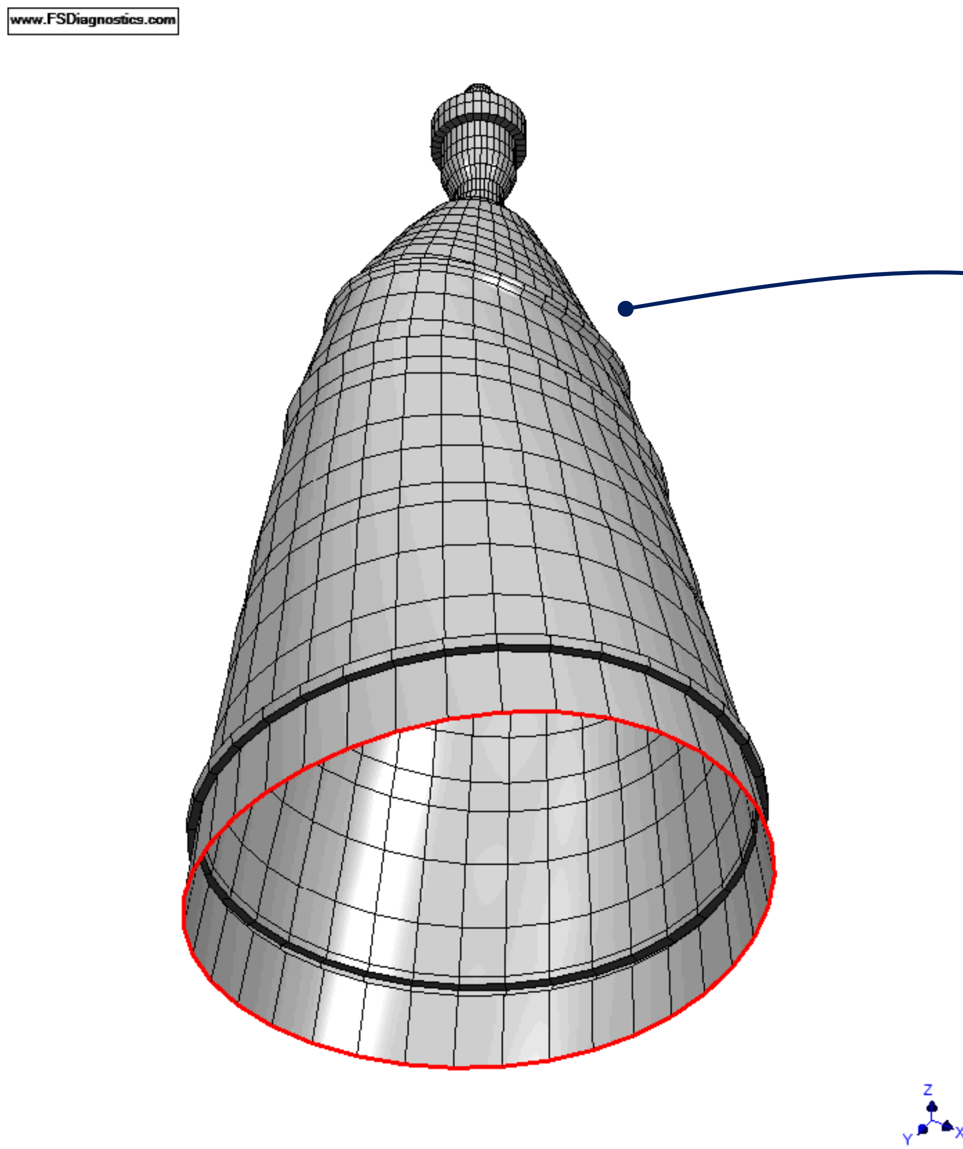


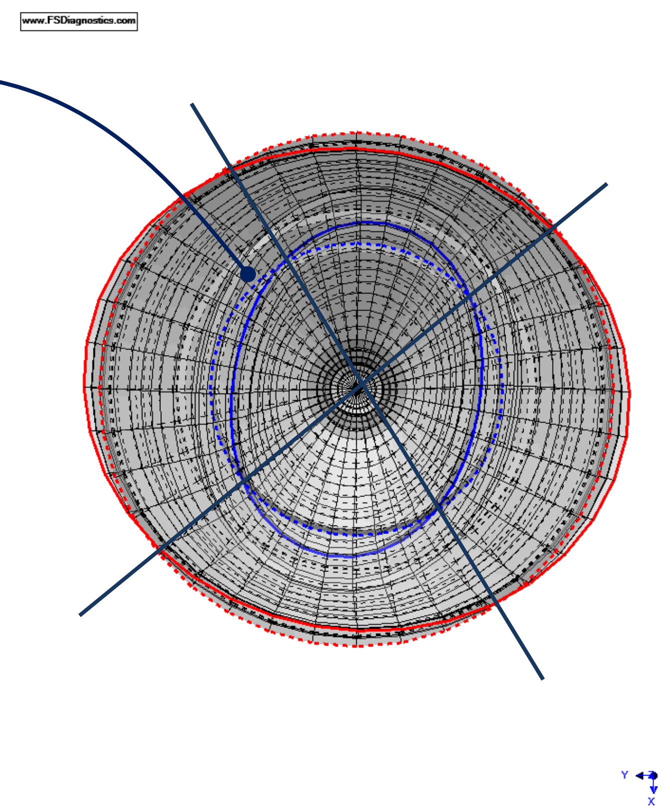
Full Spectrum Diagnostics

Natural Frequency: 66.1 Hz
Damping Factor : 0.53%
Mode Shape: Nozzle Exit 4ND (Shell Nodal Diameter)

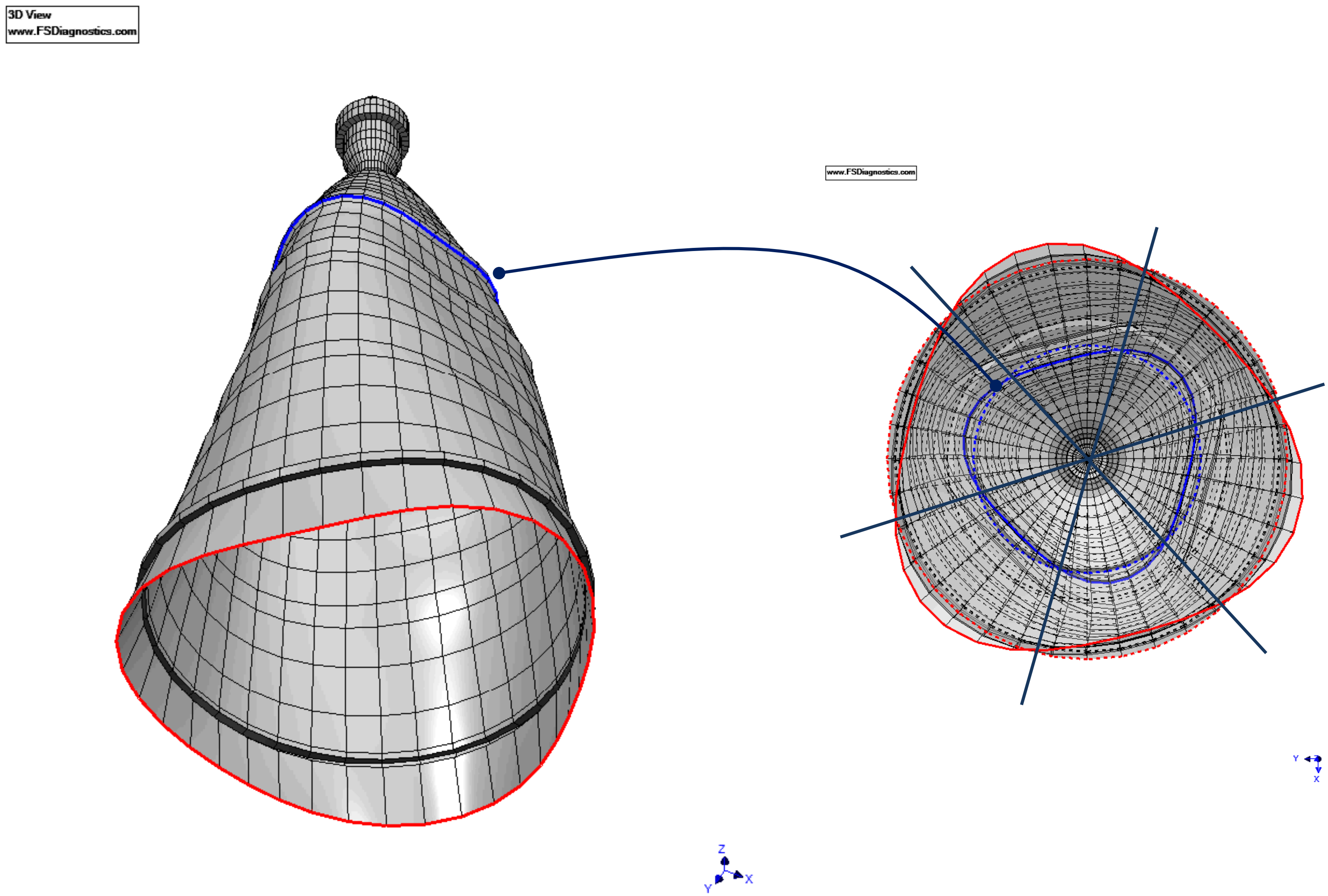


Natural Frequency: 72.0 Hz
Damping Factor : 0.21%
Mode Shape: Upper Chamber 2ND (Shell Nodal Diameter)

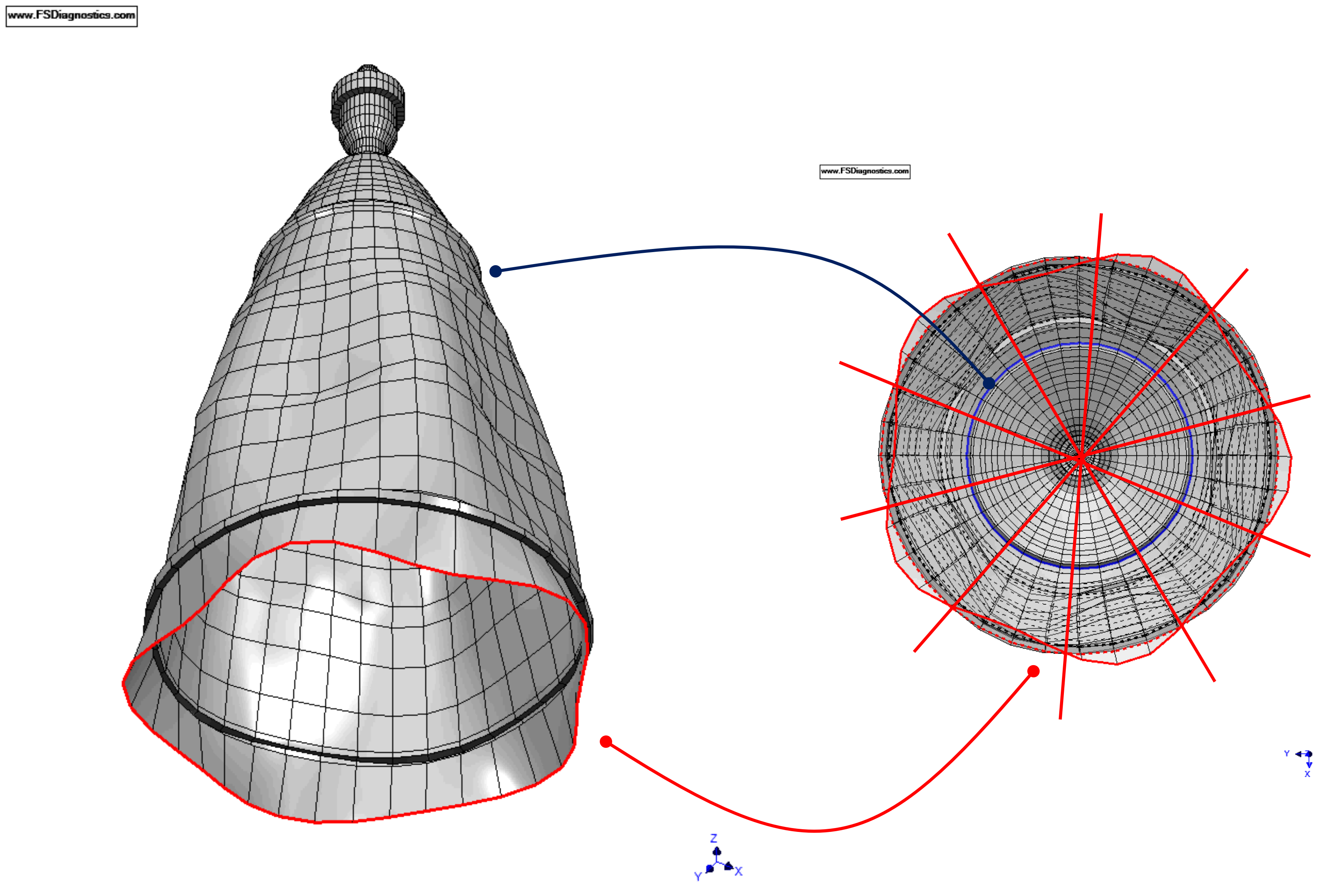




Natural Frequency: 88.0 Hz
Damping Factor : 0.22%
Mode Shape: Upper Chamber 3ND (Shell Nodal Diameter)



Natural Frequency: 92.9 Hz
Damping Factor : 0.44%
Mode Shape: Nozzle Exit 5ND (Shell Nodal Diameter)



Vibrant Technology's ME'scope