Coarse and Fine-Grain Parallelism for Inverse Design Optimization

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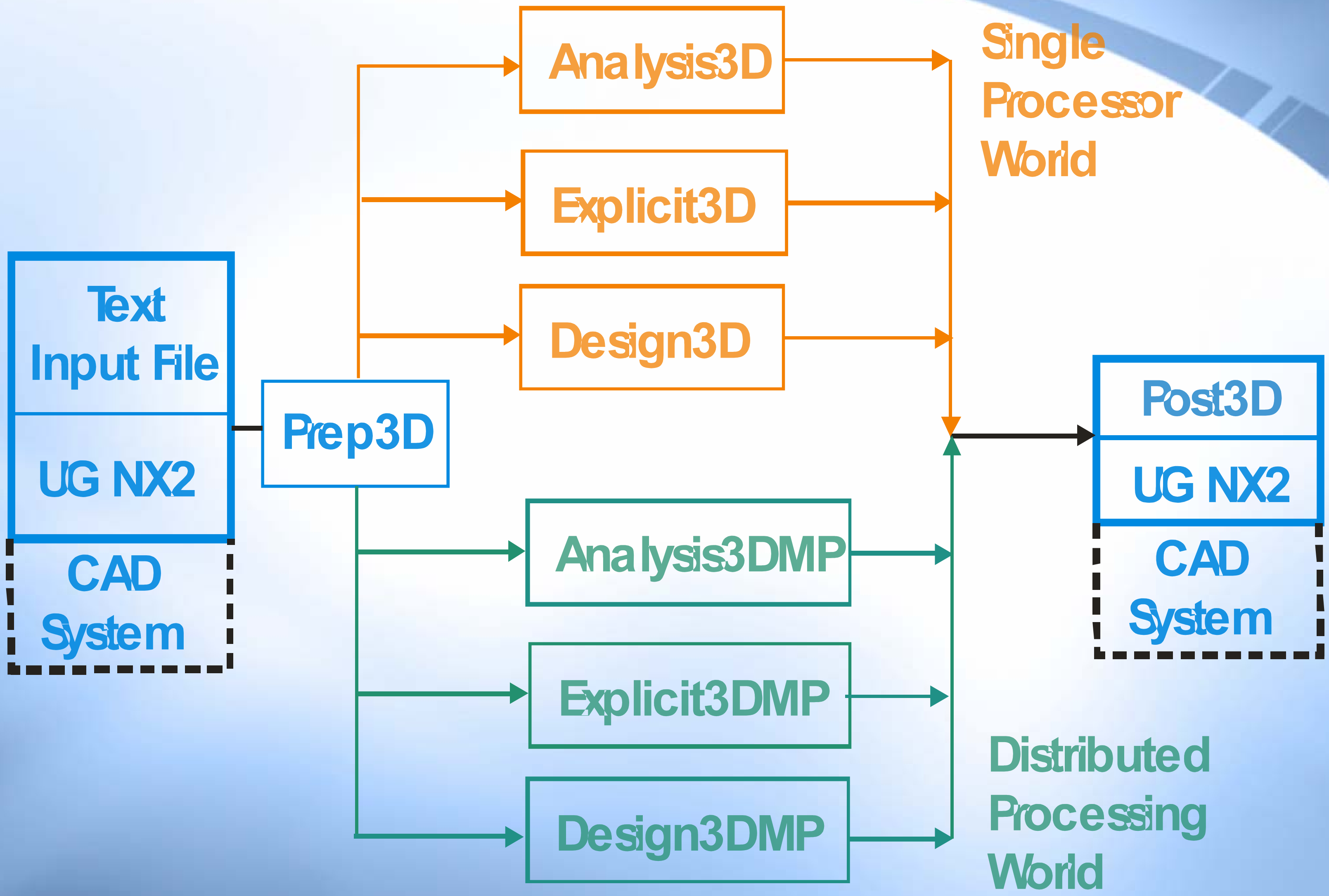
System Architecture



Computing Platforms

- Single Processor
  - Intel P3 and P4 (Windows & Linux)
  - Itanium (Red Hat Advanced Linux & Windows 2003 Server)
- Distributed Processing using MPI (TCP/IP)
  - **CML**: 25 nodes, 24 processors at ASU (P4-3.06 MHz, 2 GB RAM, Red Hat Linux)
  - **FEM**: 8 nodes, 16 processors at ASU (Dual P4-1.7 GHz, 1 GB RAM, Windows 2000)
  - **HYI**: 4 nodes, 4 processors at HYI (P3-1 GHz, 1 GB RAM, Windows 2000)

Objective Functions

- Weight, mass or volume
- Compliance
- Constrained Least-Squares
- Thermal Resistance
- Kinetic Energy, Sound Pressure

Constraints

- Constraints
  - Strength-based (failure criterion based)
  - Compliance
  - Nodal displacements
  - Frequency
  - Linearized buckling
  - Volume, mass or weight
  - Geometry

Suite of Optimizers

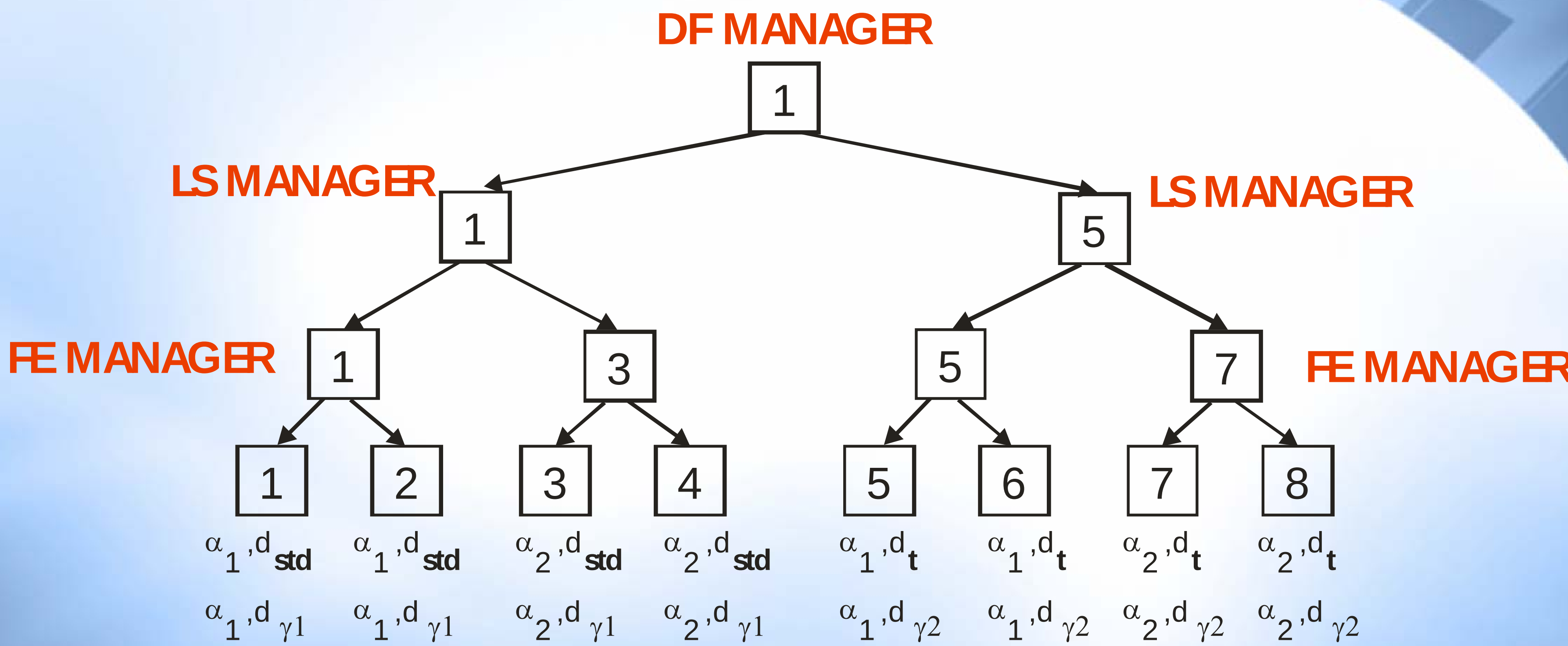
- Method of Feasible Directions
- Specialized Least-Squares Solution
- Optimality Criteria
- Genetic Algorithm

Gradient-Based Optimization

- 4 Major Steps
  - Function Evaluation (FE)
  - Gradient Evaluation (GE)
  - Direction-Finding (DF) Step
  - Line Search (LS)

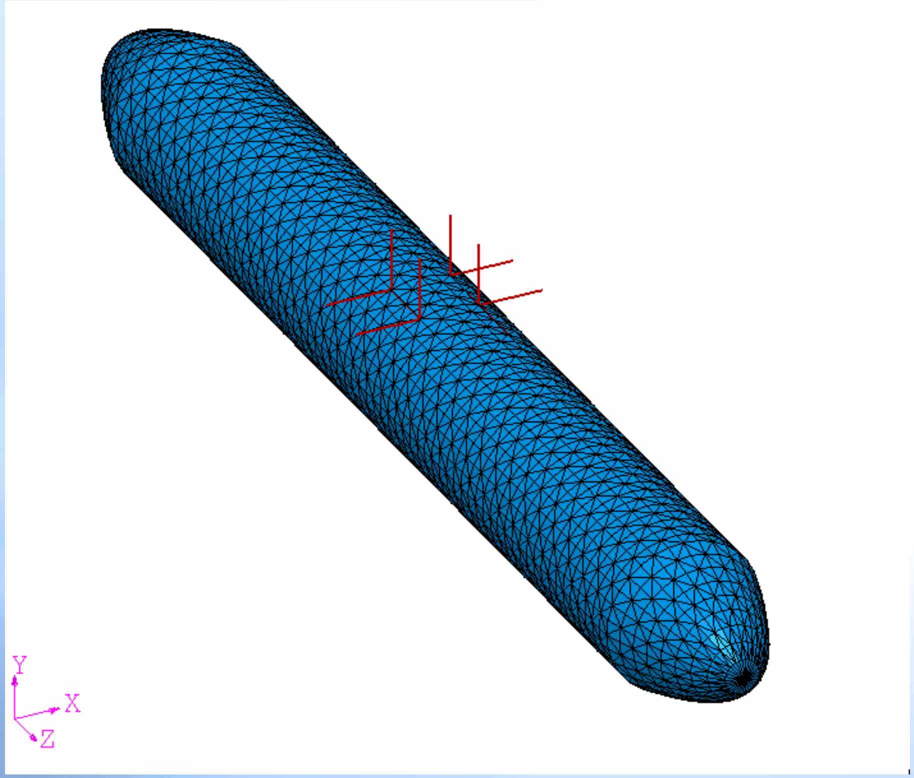
3-Level Parallelism

Scenario: NP= 8, d= 4, f= 2



Sizing Optimization: Sequential versus Distributed Processing

Fuel Tank Design



- FE Model
  - 2722 nodes
  - 5440 elements
  - Uniform internal pressure
- Design Model
  - 40 design variables
  - Von Mises FC
  - 30 iterations

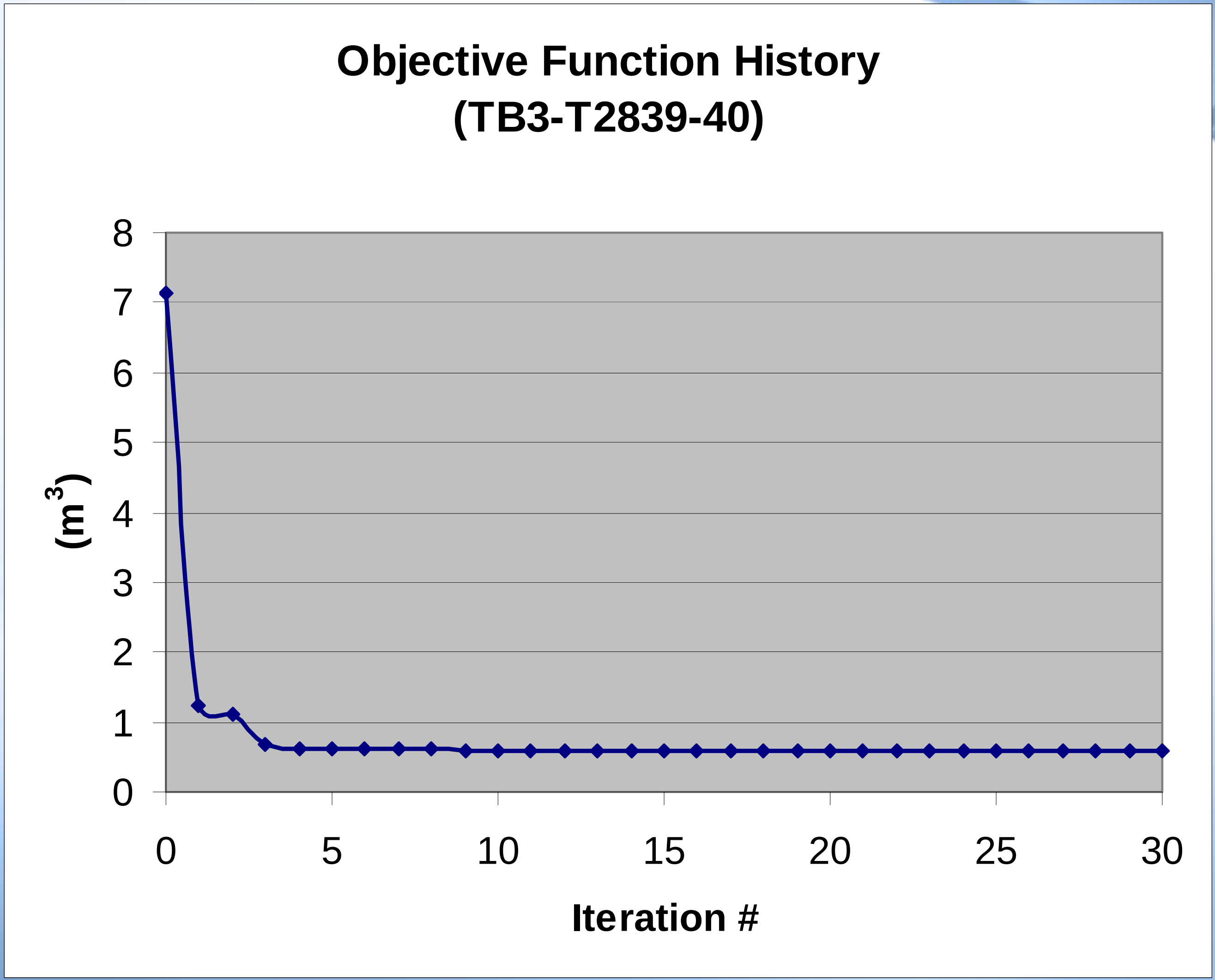
Fuel Tank Design (FEM)

# of Procs	Parallel Comps	Time (seconds)	Speedup
1	NA	8896	1.0
8 (1PN)	-ge	4792	1.86
8 (1PN)	-ge –ls	2265	3.93
8 (1PN)	-ge –df:4	2166	4.11
8 (1PN)	-ge –df:4 –ls:2	1594	5.58
16 (2PN)	-ge –df:4 –ls:4	1254	7.09

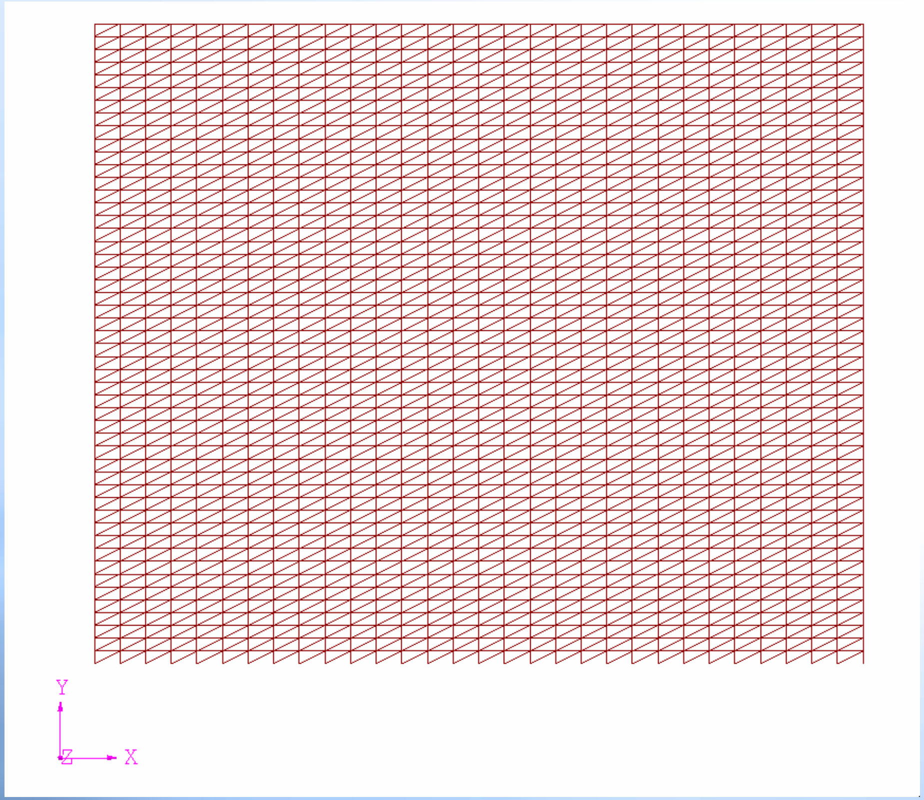
Fuel Tank Design (CML)

# of Procs	Parallel Comps	Time (seconds)	Speedup
1	NA	3602	1.0
8	-ge	2395	1.50
8	-ge –ls	1652	2.18
8	-ge –df:4 –ls:2	920	3.92
16	-ge –df:4 –ls:4	732	4.92
24	-ge –df:4 –ls:6	783	4.60

Fuel Tank Design



Truss Design

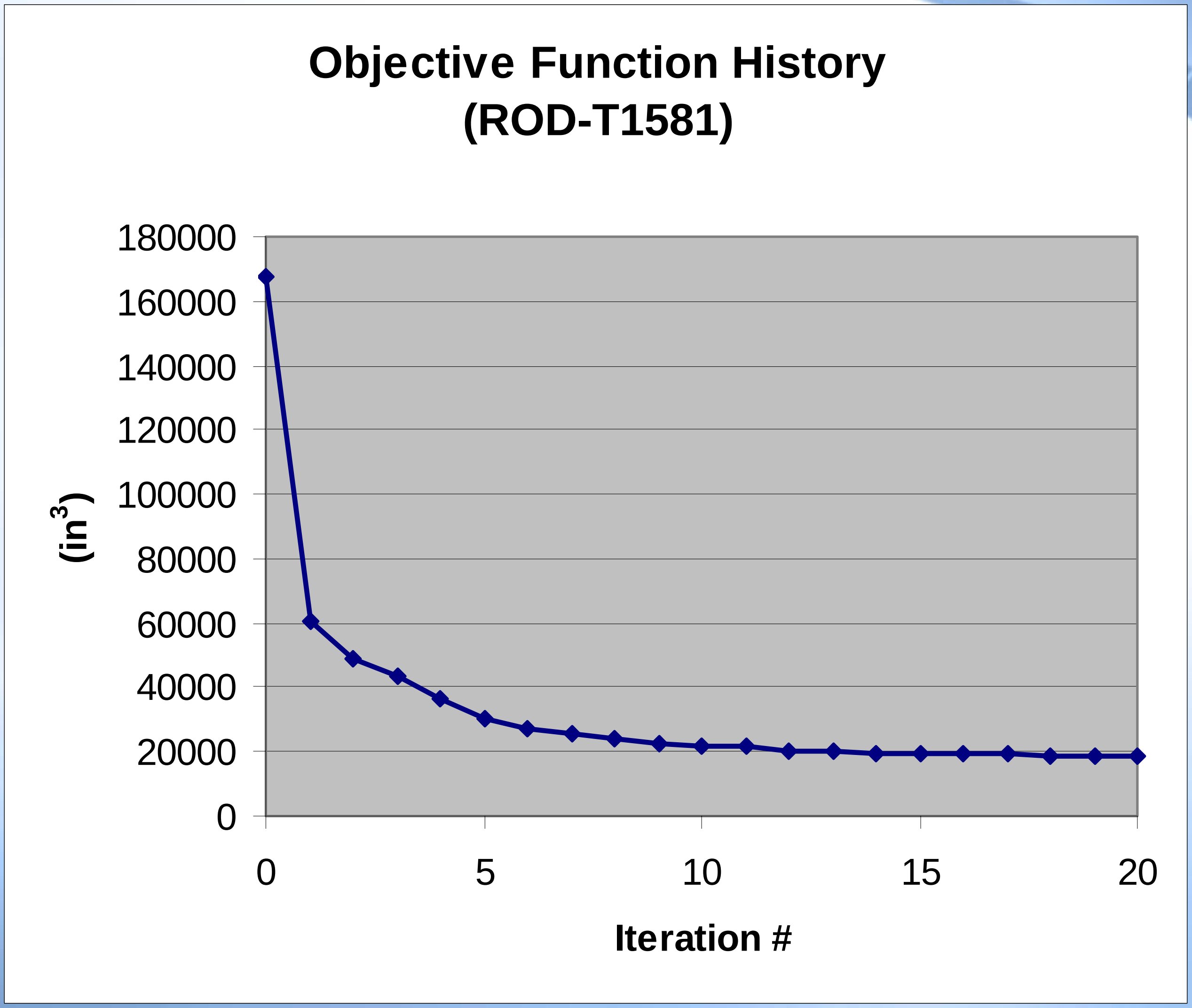


- FE Model
  - 1581 nodes
  - 4550 elements
  - Wind Loads
- Design Model
  - 150 design variables
  - Axial stress
  - 20 iterations

Truss Design (FEM)

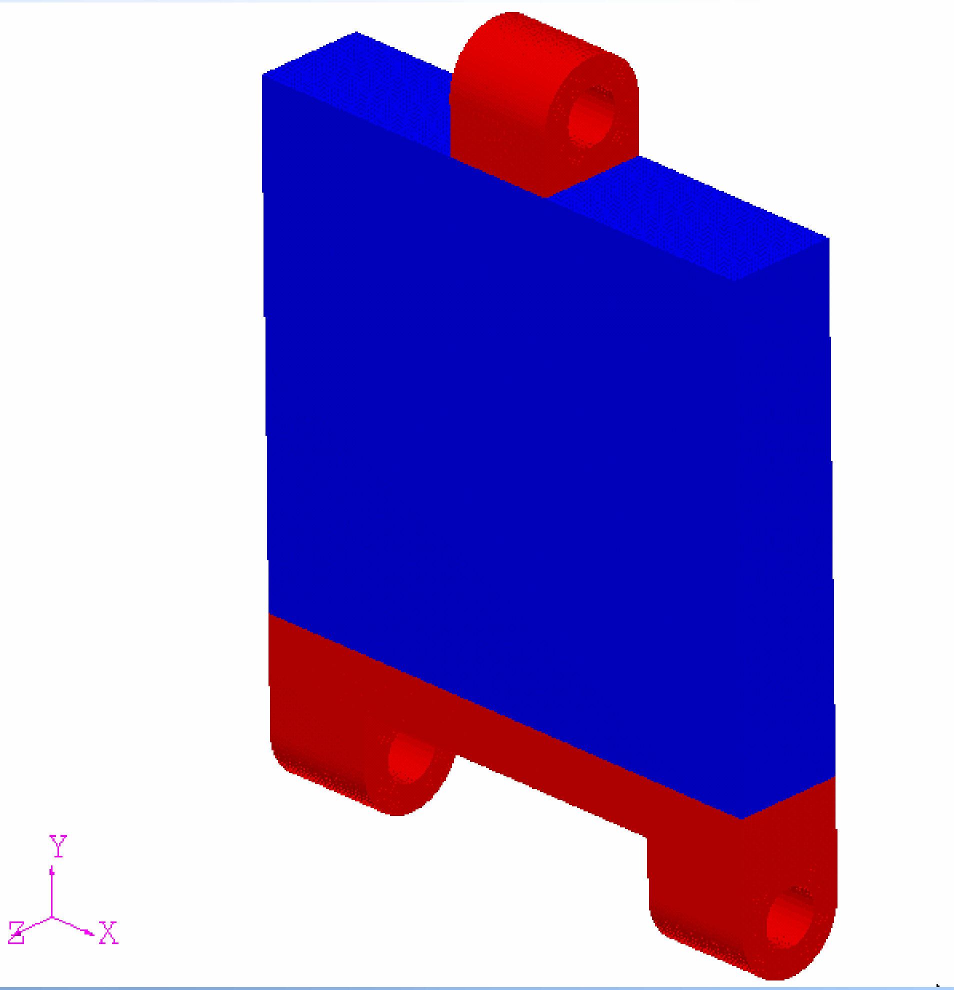
# of Procs	Parallel Comps	Time (seconds)	Speedup
1	NA	2690	1.0
8 (1PN)	-ge	1890	1.42
8 (1PN)	-ge –ls	1196	2.25
8 (1PN)	-ge –df:4	1031	2.61
8 (1PN)	-ge –df:4 –ls:2	1025	2.62
16 (2PN)	-ge –df:4 –ls:4	1117	2.41

Truss Design



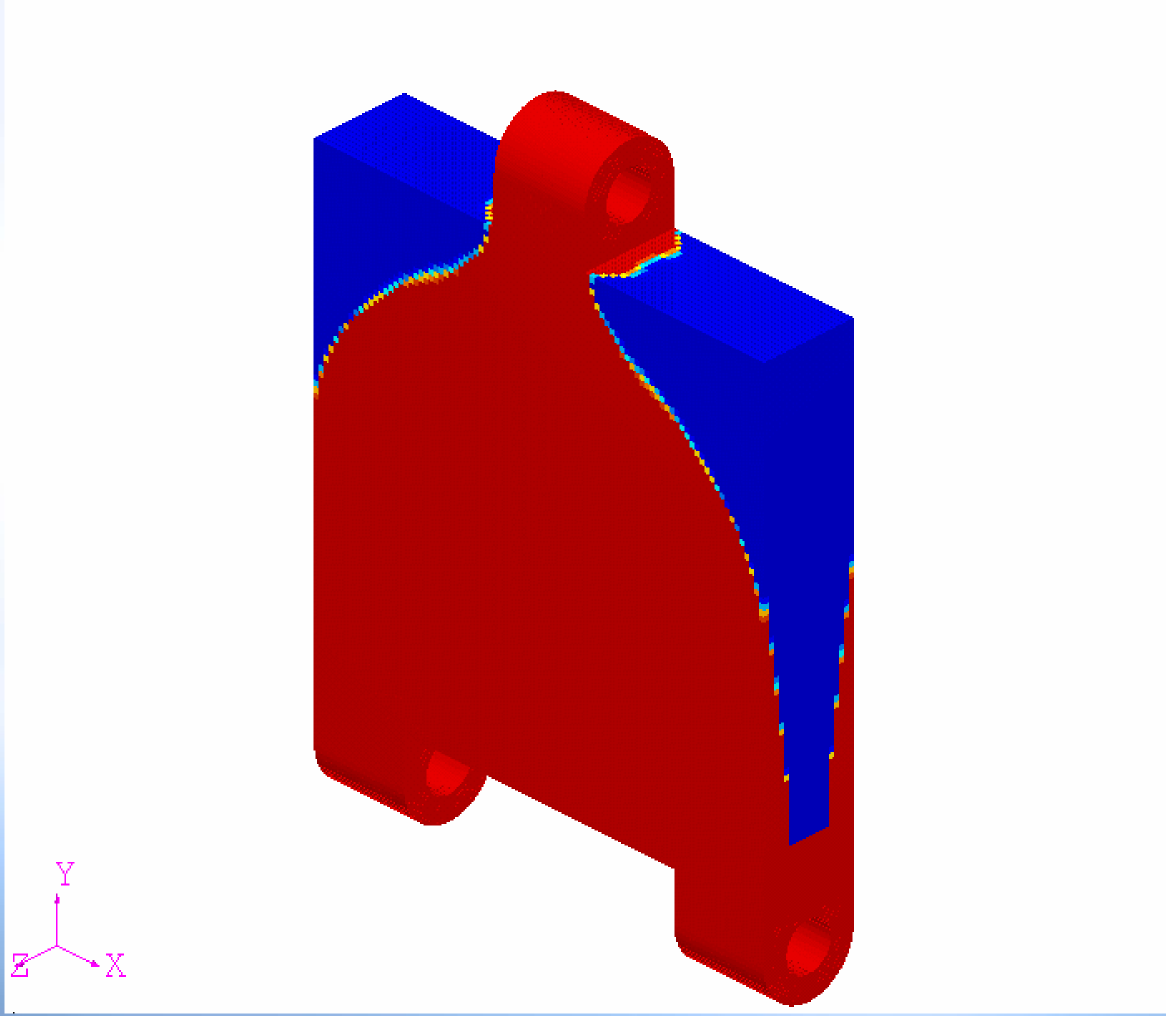
Topology Optimization

3D Topology Optimization

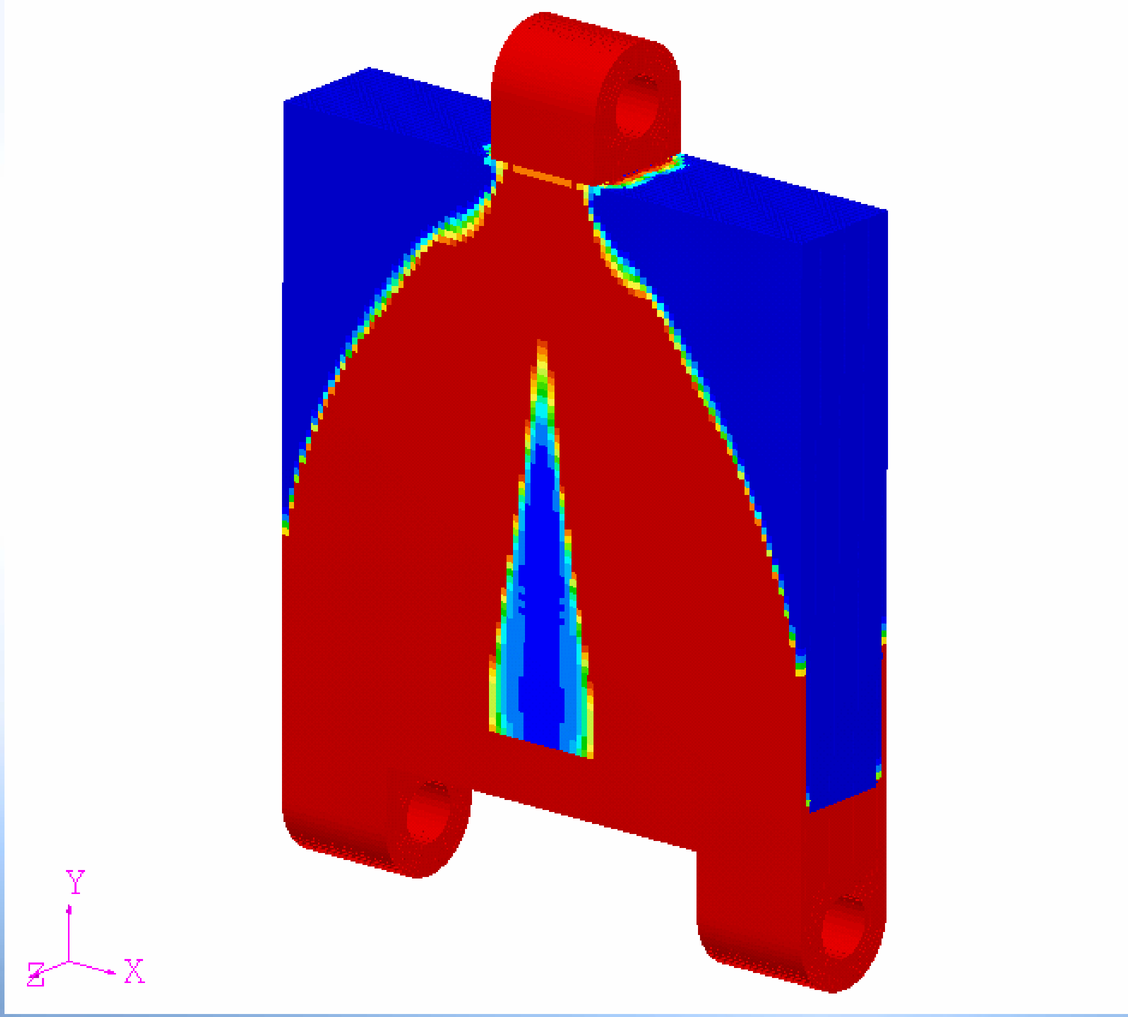


- FE Model
  - 173663 nodes
  - 159763 elements
  - Mech. Loads

3D Topology Optimization (Mass Fraction = 0.5)



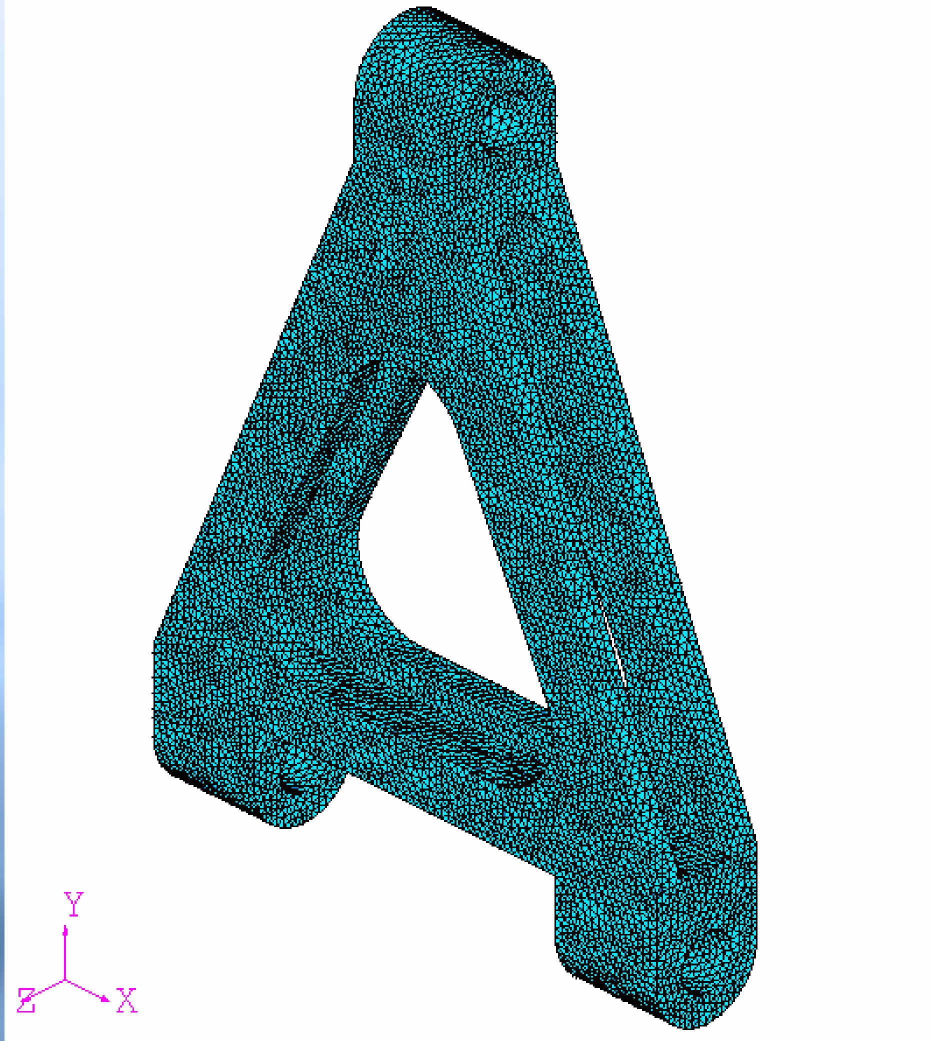
3D Topology Optimization (Mass Fraction = 0.3)

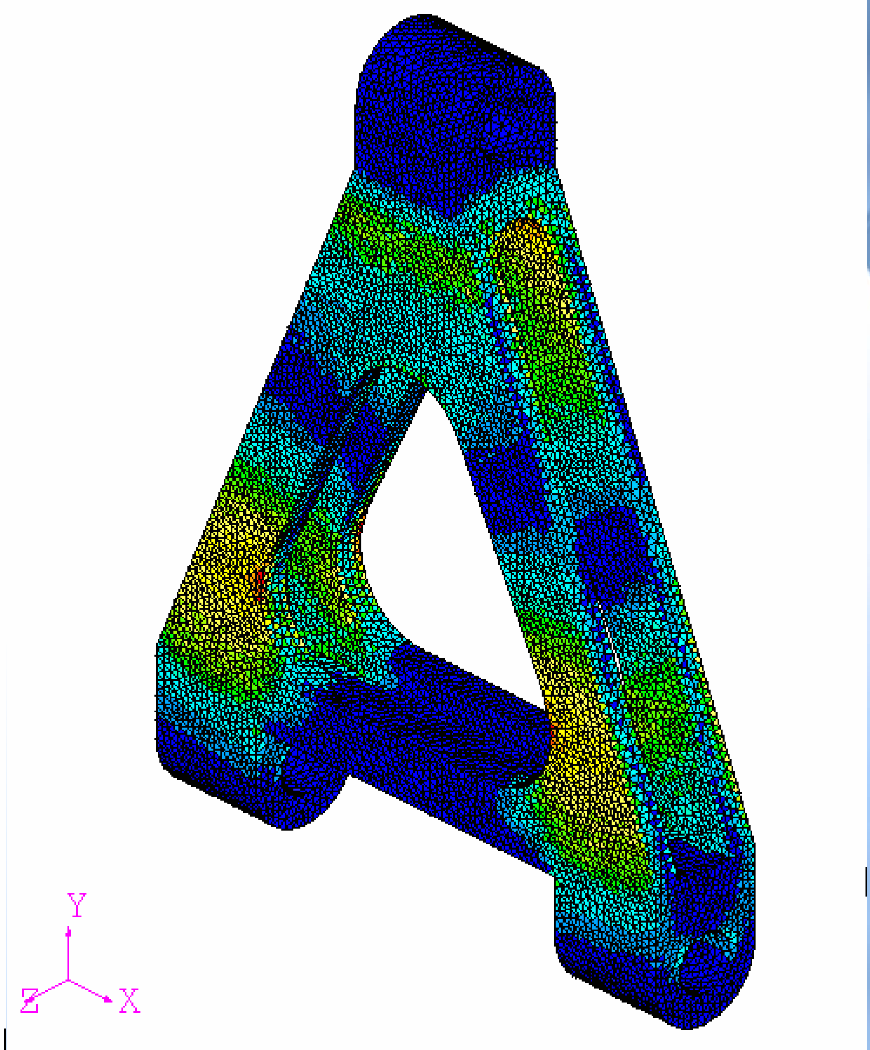


3D Topology Optimization (FEM)

# of Procs	Topology Opt. Time (s)	Normalized Time	Parallel FEA Time (s)	Normalized Time
8 (1PN)	7249	1.9	536	1.68
16 (2PN)	3813	1.0	320	1.0
8 (1PN)	8138	1.8	558	1.73
16 (2PN)	4492	1.0	322	1.0

3D Shape Optimization





3D Shape Optimization (FEM)

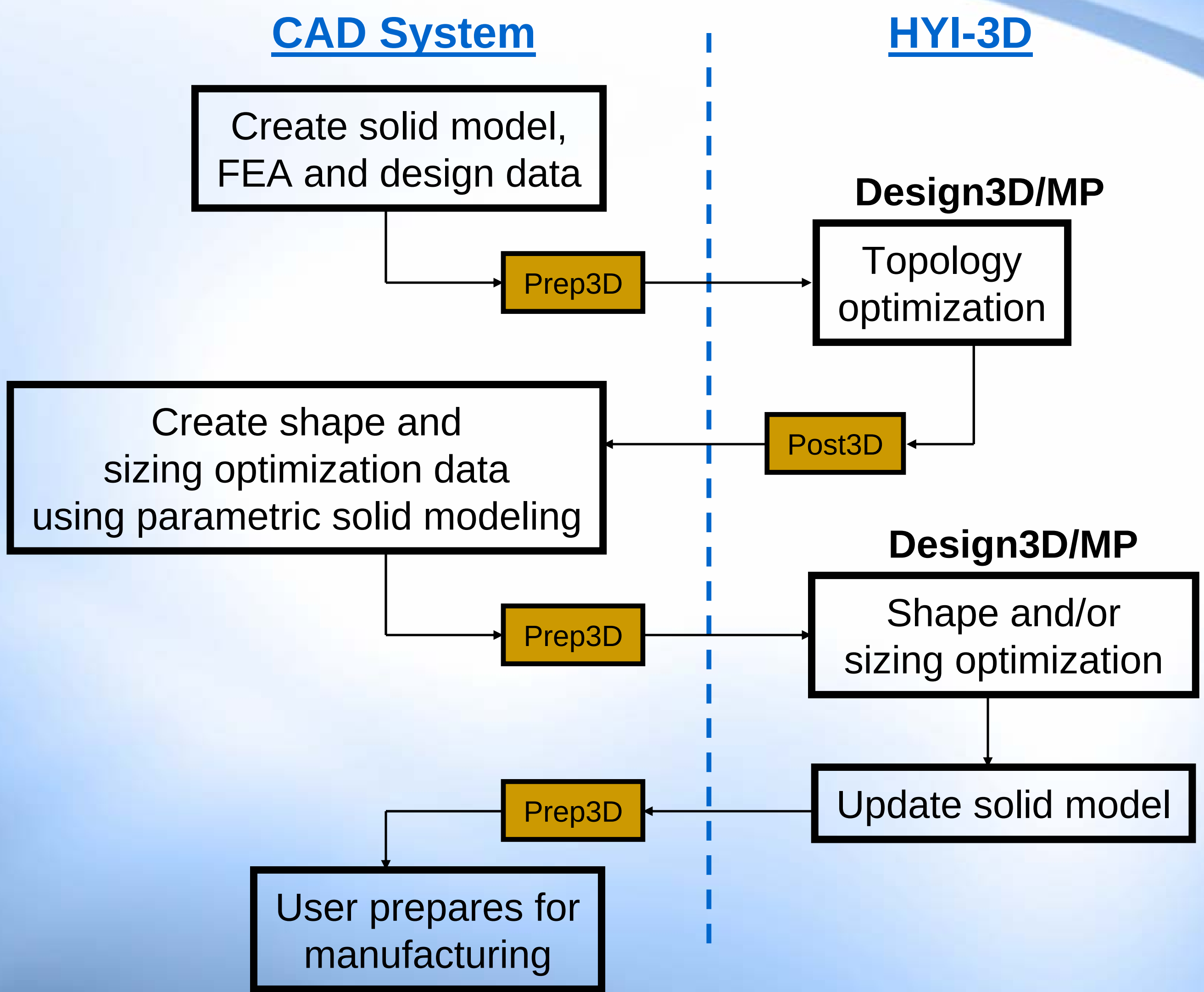
# of Procs	Parallel Comps	Time (seconds)	Speedup
1	NA	12525	1.0
8 (1PN)	-ge	9235	1.36
8 (1PN)	-ge –ls	4954	2.53
8 (1PN)	-ge –ls –fea:4	4269	2.93

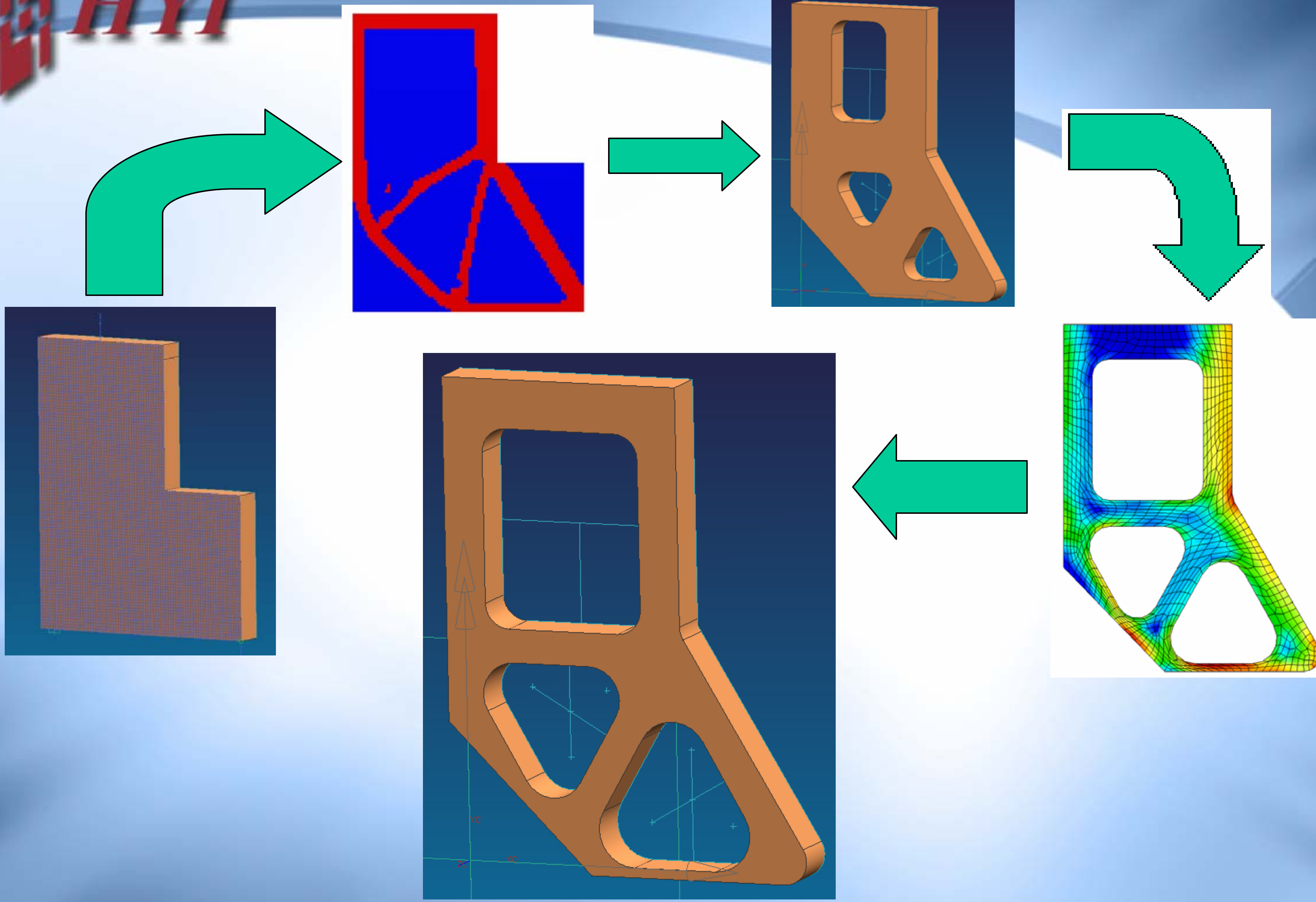
3D Shape Optimization (CML)

# of Procs	Parallel Comps	Time (seconds)	Speedup
1	NA	4616	1.0
8 (1PN)	-ge –ls –fea:4	3396	1.36
16 (1PN)	-ge –ls –fea:4	2847	1.62
16 (1PN)	-ge –ls –fea:8	2122	2.18
24 (1PN)	-ge –ls –fea:4	2608	1.77
24 (1PN)	-ge –ls –fea:8	1895	2.44

Topology & Shape Optimization of an L-Bracket

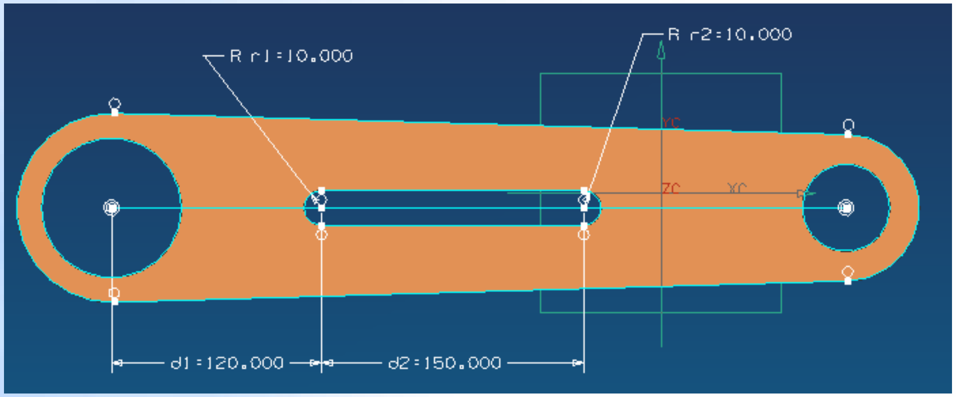
Design Flow Diagram





Shape Optimization of an Automotive Torque Arm

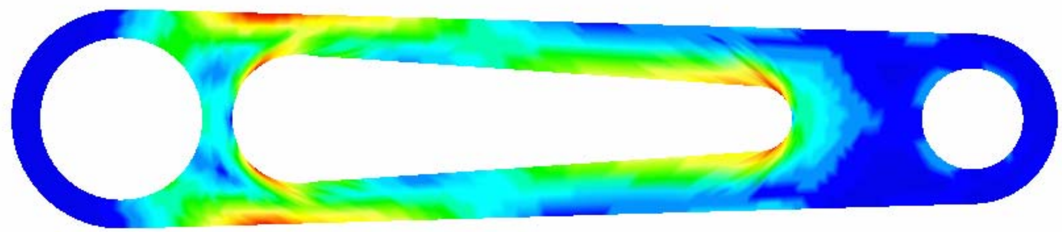
Torque Arm Design



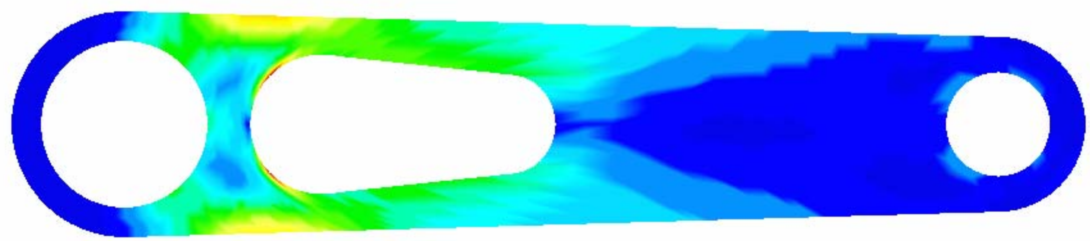
- FE Model
  - Plane stress
  - Tip loading
- Design Model
  - 4 design variables
  - Von Mises FC (800 MPa)
  - Frequency constraint (> 400 Hz)
  - Linear buckling constraint (> 750 N)

Torque Arm Problem

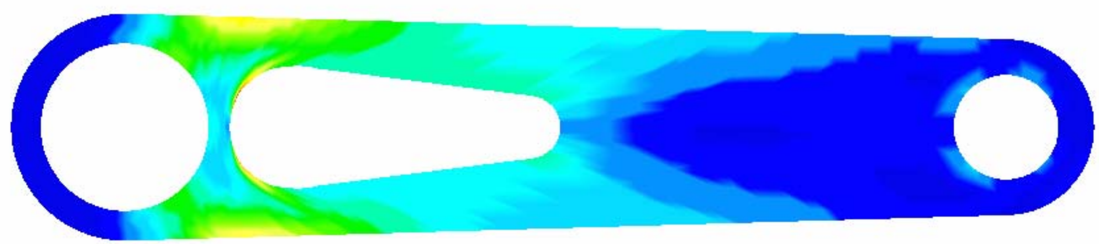
Stress Constraint



Stress +
Frequency



Stress +
Frequency +
Buckling



Torque Arm Problem

Type	Max. VMFC (MPa)	Freq. (Hz)	Buckling Load (N)	Volume (mm^3)
Stress	800	-	-	83818
Stress + Freq.	776	401	-	99036
Stress + Freq. + Buckling	751	421.5	750.1	102179

Acoustics Optimization

Acoustics Optimization

• Focus has been on experimentally verified, passive noise radiation, from vibrating plates and shells – e.g., appliance covers or side panels, oil pan, timing chain cover plate, trim panels in aircraft

• Objective is based on multiple attributes (sound power over a frequency band, weight, cost, amount of damping)

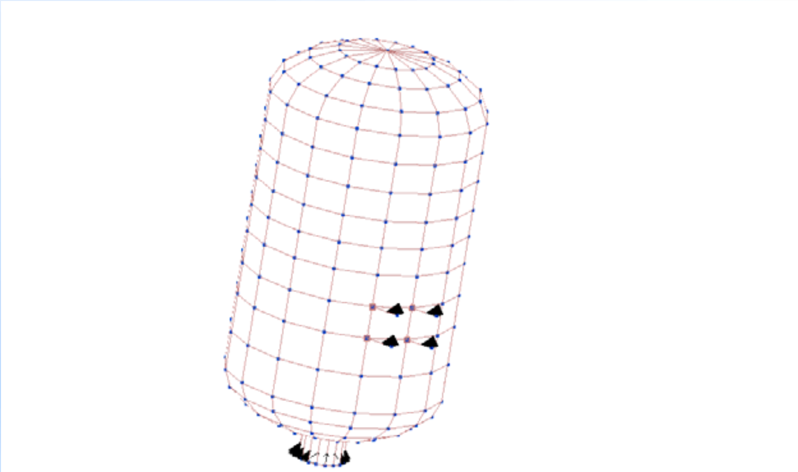
• Design Idea: *attach* masses, vibration absorbers, air-filled cavities etc *to* the existing structure

• Design variables are the parameters of the attachment structures

Acoustics Optimization

- Simulation Codes: In-house boundary element for sound power, FEA for vibration
- New fast re-analysis techniques have been developed for obtaining response spectrum;
  - modal calculations of original structure done once only, outside optimization loop
  - Efficient for broadband objectives since there are no peak searches
  - Absorber frequencies can be closely spaced
- Coupled structure-acoustic vibration analysis method has now been developed – can now attach a thin air-filled cavity to structure

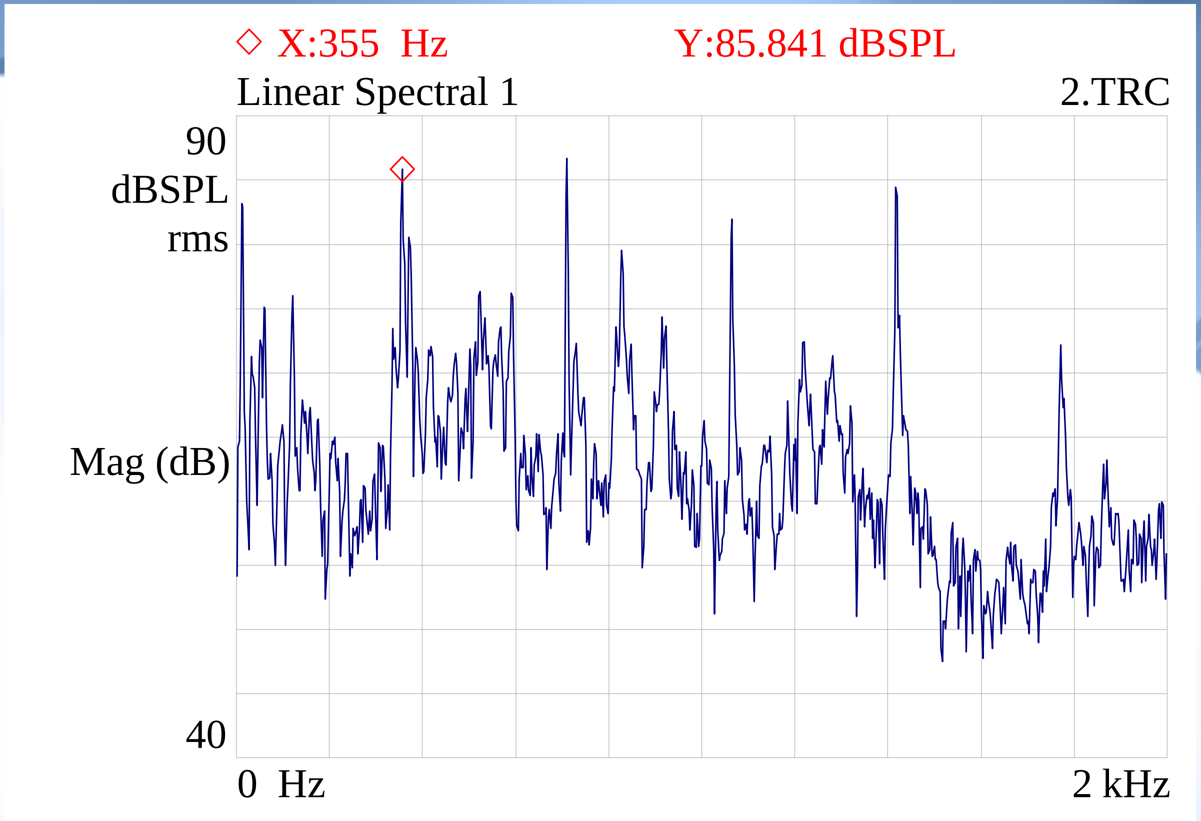
Pressure Vessel



Finite element depiction of the curved pressure vessel

A reduced eigenvalue method for broadband analysis of a structure with vibration absorbers possessing rotatory inertia, *J. of Sound and Vibration*, 281, pp. 869-886, 2005. [Grissom, Belegundu, et al]

Conjoint Analysis Based Multiattribute Optimization : Application in Acoustical Design, To Appear, *J. of Structural Optimization*, 2005. [Grissom, Belegundu, et al]



SOUND PRESSURE LEVEL 1 METER FROM THE VESSEL

The problematic noise occurs at around 360 Hz, a harmonic of the motor frequency, and at the next three harmonics, 720, 1080, and 1440

Method for Broadband Acoustic Response

Define Geometry/Materials

Modal Analysis of Base Structure

Initial Acoustic Analysis

Conjoint Analysis

Forcing Function

Calculate Frequency Response

Calculate Objective

?

Modify Many Absorber Parameters Including Rotatory Inertia Effects

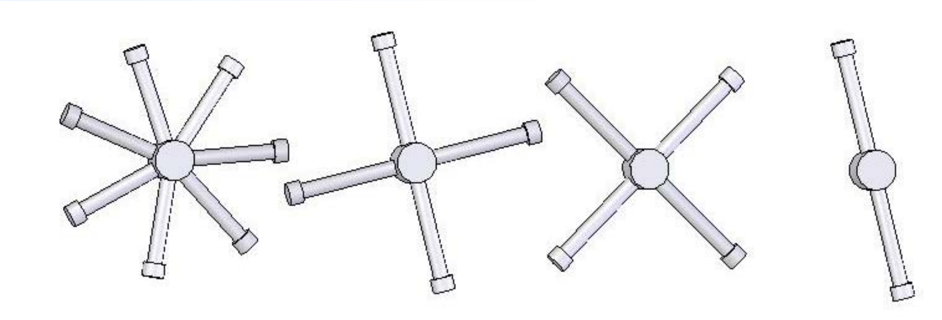
Finished

Results

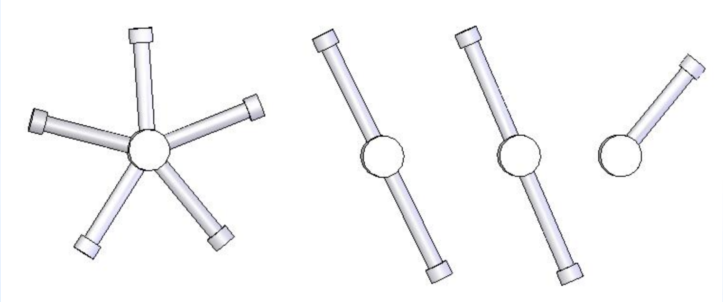
Attributes	Number of Beams	dB Reduction: low freq. range	dB Reduction: mid freq. range	dB Reduction: high freq. range	Description
Kinetic Energy Objective	17	15	10	2	Poor high freq. reduction & many beams
Sound Pressure Objective	10	14	6	7	Better than Kinetic Energy objective
Multi-attribute Objective	**6**	**13**	**6**	**7**	Almost the same reduction as Sound Pressure objective, but easier to manufacture

The "Optimized Product"

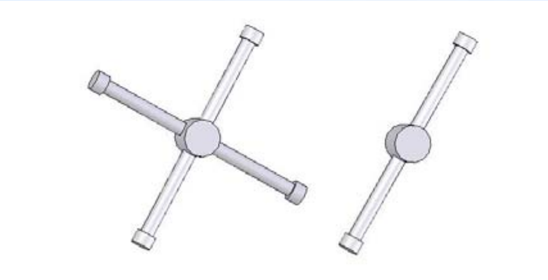
A set of optimized Broadband Vibration Absorbers with a variable number of beams, n_b



n_b = 17
KE Objective



n_b = 10
SPL Objective



n_b = 6
Multiattribute Objective

Concluding Remarks

- Future of engineering analysis and design is in some form of a combined desktop-distributed computing paradigm
- Challenges lie ahead for inverse analysis and design
- Tightly integrated multi-physics design optimization offers an attractive solution to reducing design cycle times