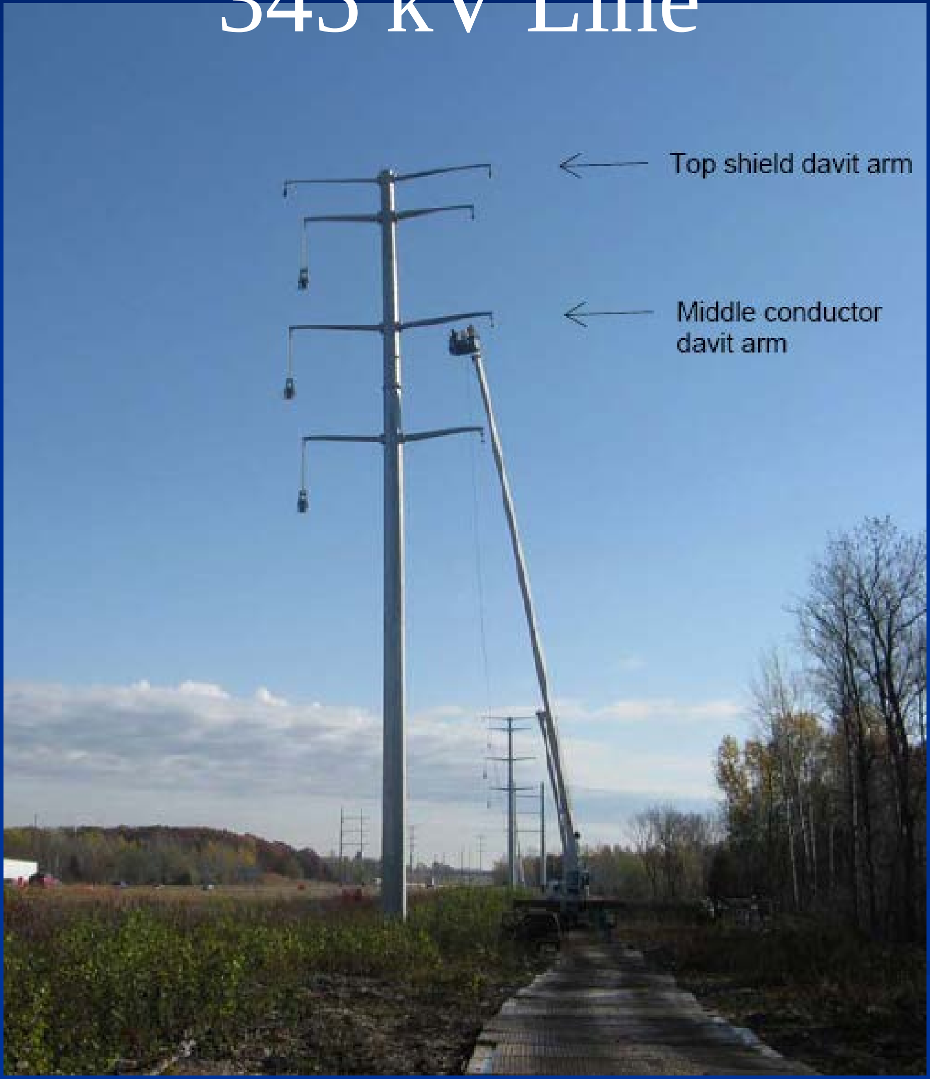
DAVIT ARM VIBRATION STUDY
Double Circuit
345 kV Line



Background for the Study

- A mid-west utility is building a new 345 kV D.C. line and is installing complete structures (both sets of arms) but only one circuit will be strung. The second circuit will be installed when electrical load demands such.
- The pole suppliers recommended either suspending 150 lbs or 10% of the arm weight at the end of the unloaded arms "as a rule of thumb".
- The utility sought a better understanding of the proposed tuned-mass damping.

Our Prior Experience: November, 2005 –Wisconsin

Our experience with this topic started in 2005 when static arms for a double circuit 345 kV line started coming down shortly after installation.

Both circuits were scheduled to be strung in but the unloaded arms stood for 39 days and failed before the static wires could be installed.

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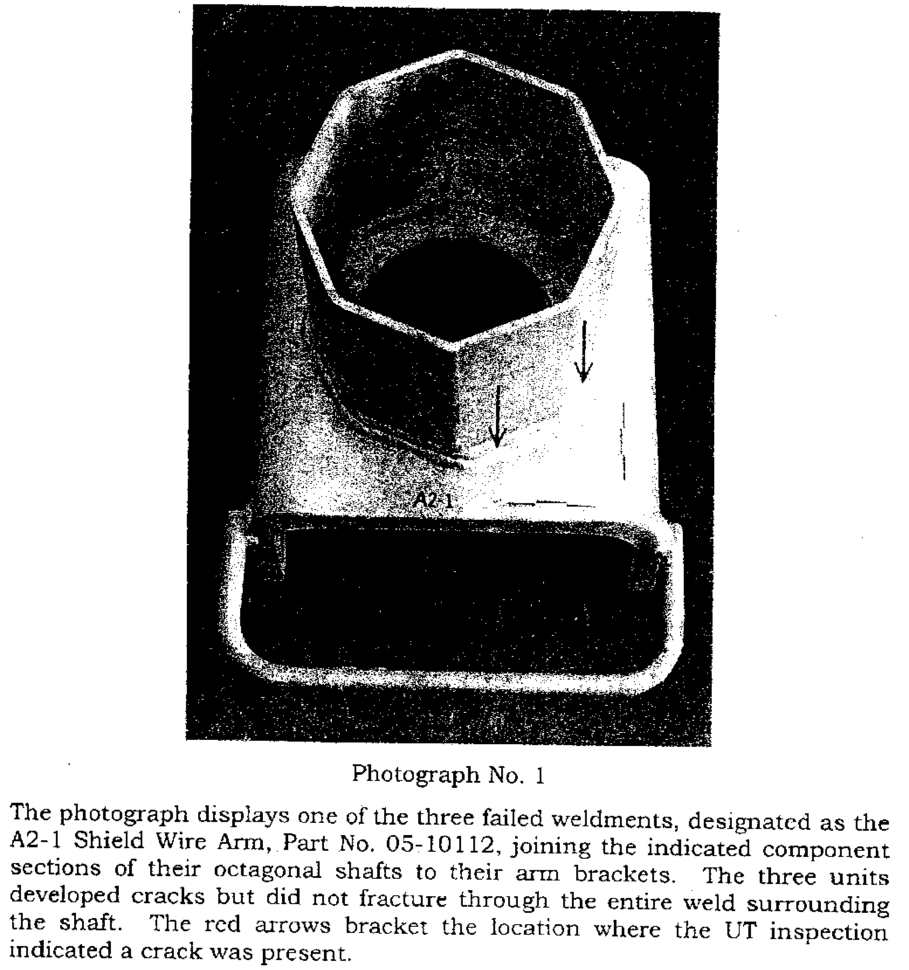
As part of the design team, we were involved in the post-failure investigation. We worked with the Owner, an independent laboratory, and the pole vendor, but not everyone shared the same theory as to the root cause of failure:



Photograph No. 1

The photograph displays one of the three failed weldments, designated as the A2-1 Shield Wire Arm, Part No. 05-10112, joining the indicated component sections of their octagonal shafts to their arm brackets. The three units developed cracks but did not fracture through the entire weld surrounding the shaft. The red arrows bracket the location where the UT inspection indicated a crack was present.

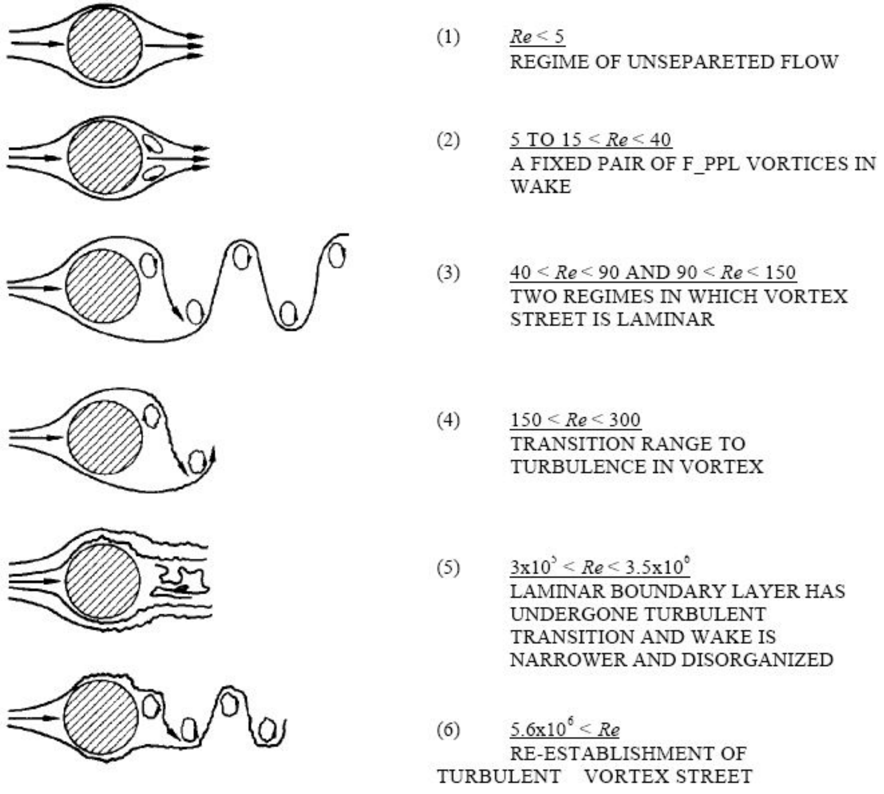
"Also the pictures of the arms that failed in the field do not support failure due to Vortex shedding. Cracking appears to have started along the sides. Vortex shedding causes movement perpendicular to the direction of the wind, therefore cracking should have propagated from the top or bottom of the arm (see attached photo)." – Vendor Engineer

- Davit arms represent a bluff structure. A bluff structure is one in which the flow separates from large sections of the structure's surface. 345 kV davit arms are very long slender structures that are prone to vortex shedding

- Re = Reynolds Number, a measure of the ratio of inertial to viscous forces. Re=Uf*D/n

  Where
  Uf = velocity of the fluid w.r.t. the object (wind speed)
  D = mean diameter
  n= kinematic viscosity

  For tubular steel arms and shafts, Re ~ 10^5



(1) $Re < 5$
REGIME OF UNSEPARETED FLOW

(2) 5 TO $15 < Re < 40$
A FIXED PAIR OF F_PPL VORTICES IN WAKE

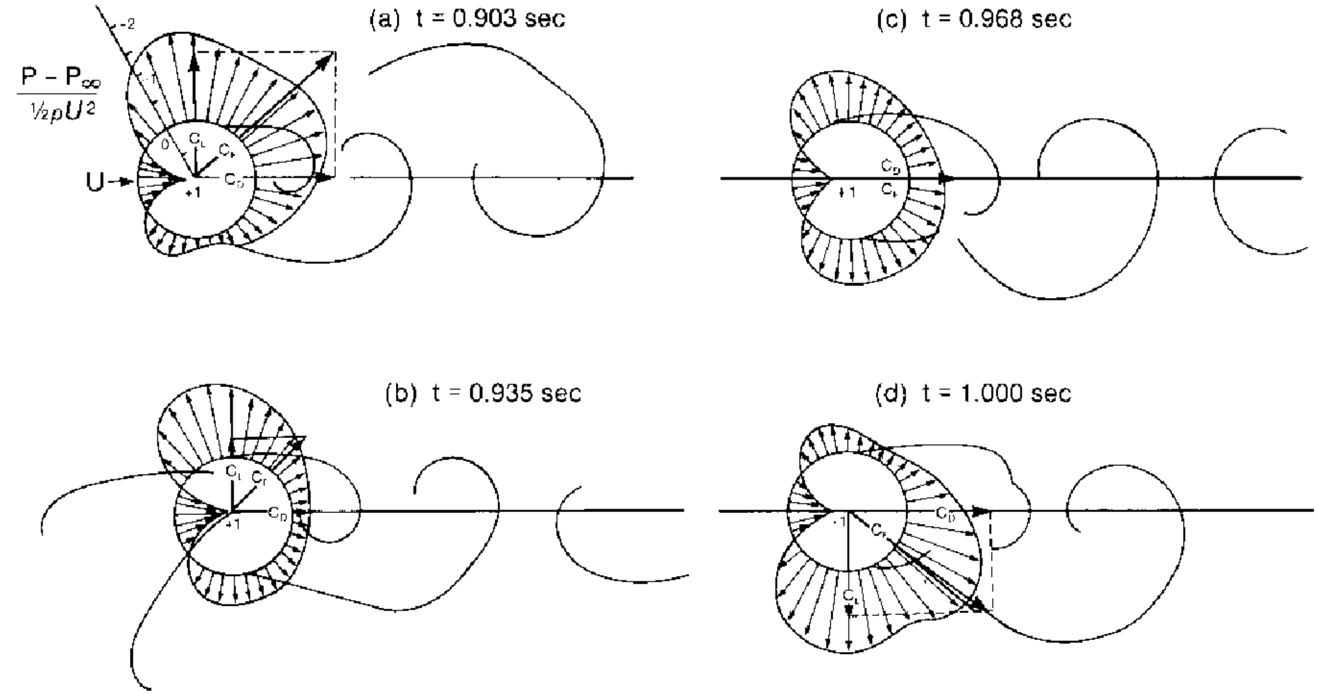
(3) $40 < Re < 90$ AND $90 < Re < 150$
TWO REGIMES IN WHICH VORTEX STREET IS LAMINAR

(4) $150 < Re < 300$
TRANSITION RANGE TO TURBULENCE IN VORTEX

(5) $3\text{x}10^5 < Re < 3.5\text{x}10^6$
LAMINAR BOUNDARY LAYER HAS UNDERGONE TURBULENT TRANSITION AND WAKE IS NARROWER AND DISORGANIZED

(6) $5.6\text{x}10^6 < Re$
RE-ESTABLISHMENT OF TURBULENT VORTEX STREET

This causes an oscillating pressure differential



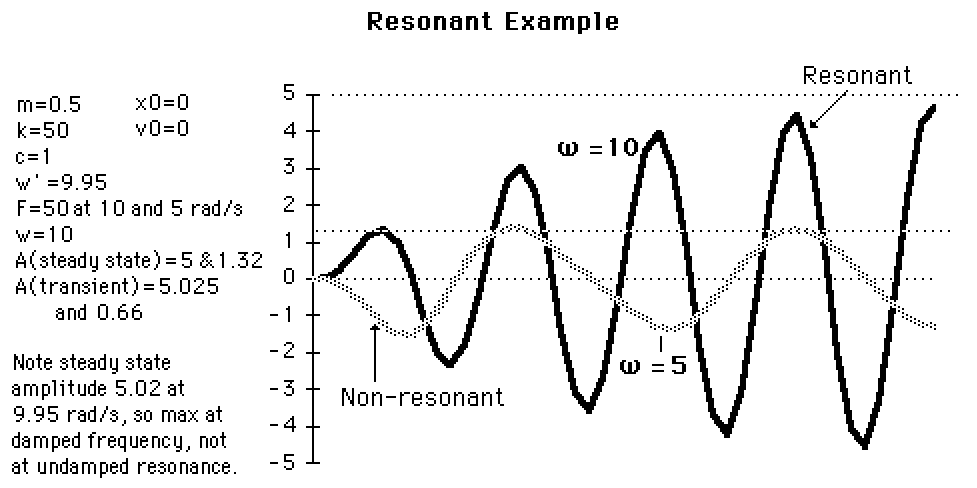
That does not act solely in the vertical plane

Things get exciting when the frequency of this oscillating pressure approach the natural frequency of the member

What is the ‘natural frequency’?

> “…a characteristic value of the driving frequency at which the amplitude of oscillation is a maximum.”

If a sinusoidal driving force is applied at the resonant frequency of the oscillator, then its motion will build up in amplitude to the point where it is limited only by the damping forces on the system. If the damping forces are small, a resonant system can build up to amplitudes large enough to be destructive to the system. Such was the famous case of the Tacoma Narrows Bridge, which was blown down by the wind when it responded to a component in the wind force which excited one of its resonant frequencies.



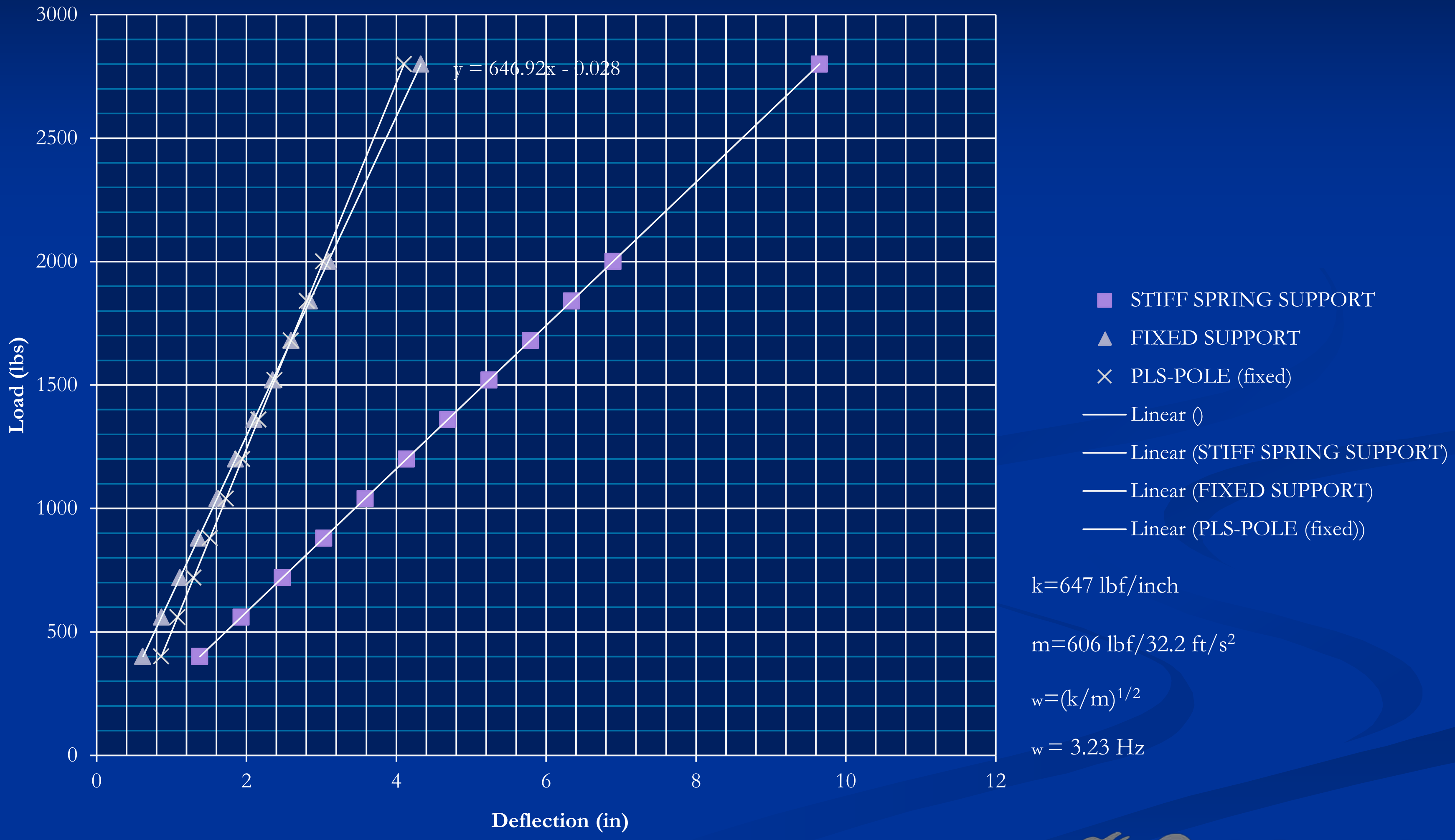
Resonant Example

m=0.5
k=50
c=1
w' =9.95
F=50 at 10 and 5 rad/s
w=10
A(steady state)=5 & 1.32
A(transient)=5.025
and 0.66

x0=0
v0=0

Note steady state amplitude 5.02 at 9.95 rad/s, so max at damped frequency, not at undamped resonance.

Hand Calculating Natural Frequency $w=(k/m)^{1/2}$



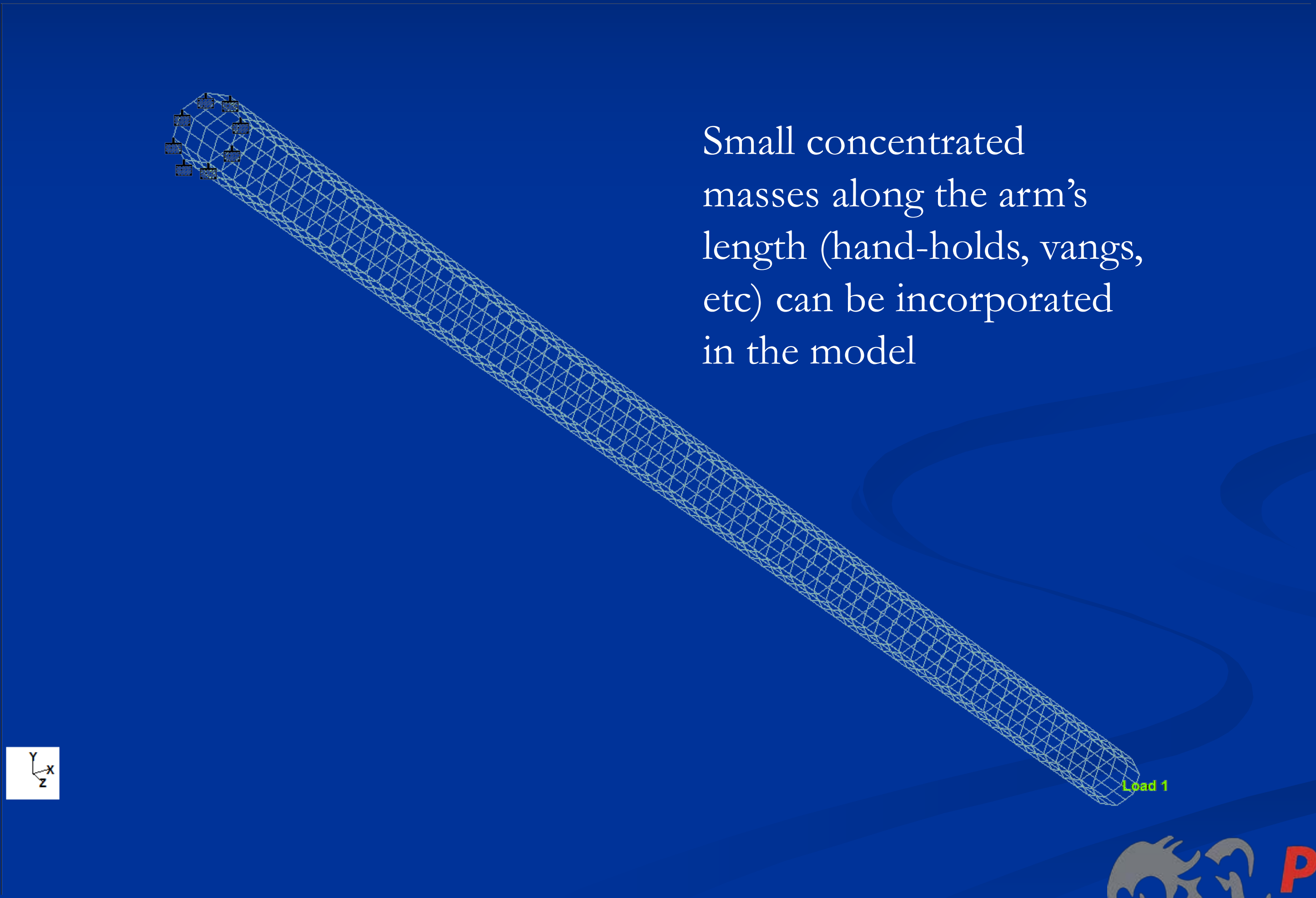
k=647 lbf/inch

$m=606\ \text{lbf}/32.2\ \text{ft/s}^2$

$w=(k/m)^{1/2}$

w = 3.23 Hz

It is more accurate to build an F.E. model of the davit arm for a more accurate calculation of the natural frequency



Small concentrated masses along the arm's length (hand-holds, vangs, etc) can be incorporated in the model

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SHIELDWIRE ARM                                        -- PAGE NO.   21

          CALCULATED FREQUENCIES FOR LOAD CASE      1

     MODE      FREQUENCY(CYCLES/SEC)  PERIOD(SEC)
ACCURACY

     1             7.026             0.14234     2.217E-15
     2             7.026             0.14234     8.168E-16
     3            37.624              0.02658     2.152E-13
     4            37.624              0.02658     5.859E-15
     5            99.040              0.01010     6.275E-07
```

There are no standards in our industry that address any kind of wind-induced motion.

ASME Standard STS-1 (Steel Stacks) has a section on dynamic responses
and sites Von Karman's relationship between critical wind velocities and the potential for vortex shedding:

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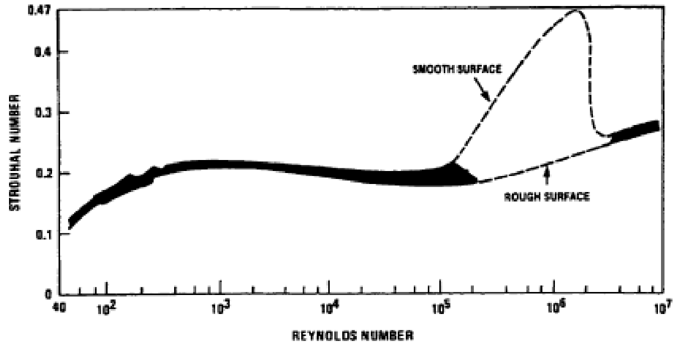
$$V_{cr} = \eta_i * D/S_t$$

Strouhal Number (~0.2)

The wind velocity at which the natural frequency of the vortex shedding equals the natural frequency of the member.

Fundamental natural frequency

D is the mean diameter in the top one third of the shaft



igure 4.2 Relationship between the Strouhal and Reynolds numbers for circular cylinders [34].

For the static arm: $f_1 = 7.02$ Hz

$V_{cr} = 17.8$ mph

For the middle phase arm: f1 = 12.3 Hz

$V_{cr} = 39.5$ mph

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- It appeared that excitation wind speeds were very probable, especially on the static arms. We planned to proceed with analyzing the dynamic loads to determine the stresses at the weld connection. Concurrently, the independent metallurgical analysis issued the following statement that strengthened our perspective:

- "…the root cause of the failures of the subject…shield wire arms was that the fatigue endurance limit of the columbium-vanadium steels, at the H.A.Z. (Heat Affected Zone) of the shaft material, was exceeded by the cyclic vibrational stresses to which the arms were subjected during the 39 days after installation."

- A statement that begged further investigation instead brought closure and shut the investigation down.
- The arms were replaced by the pole vendor, and 150 lb weights were added to the unloaded static arms.
- We (temporarily) closed the books on this topic

Segue to 2011…

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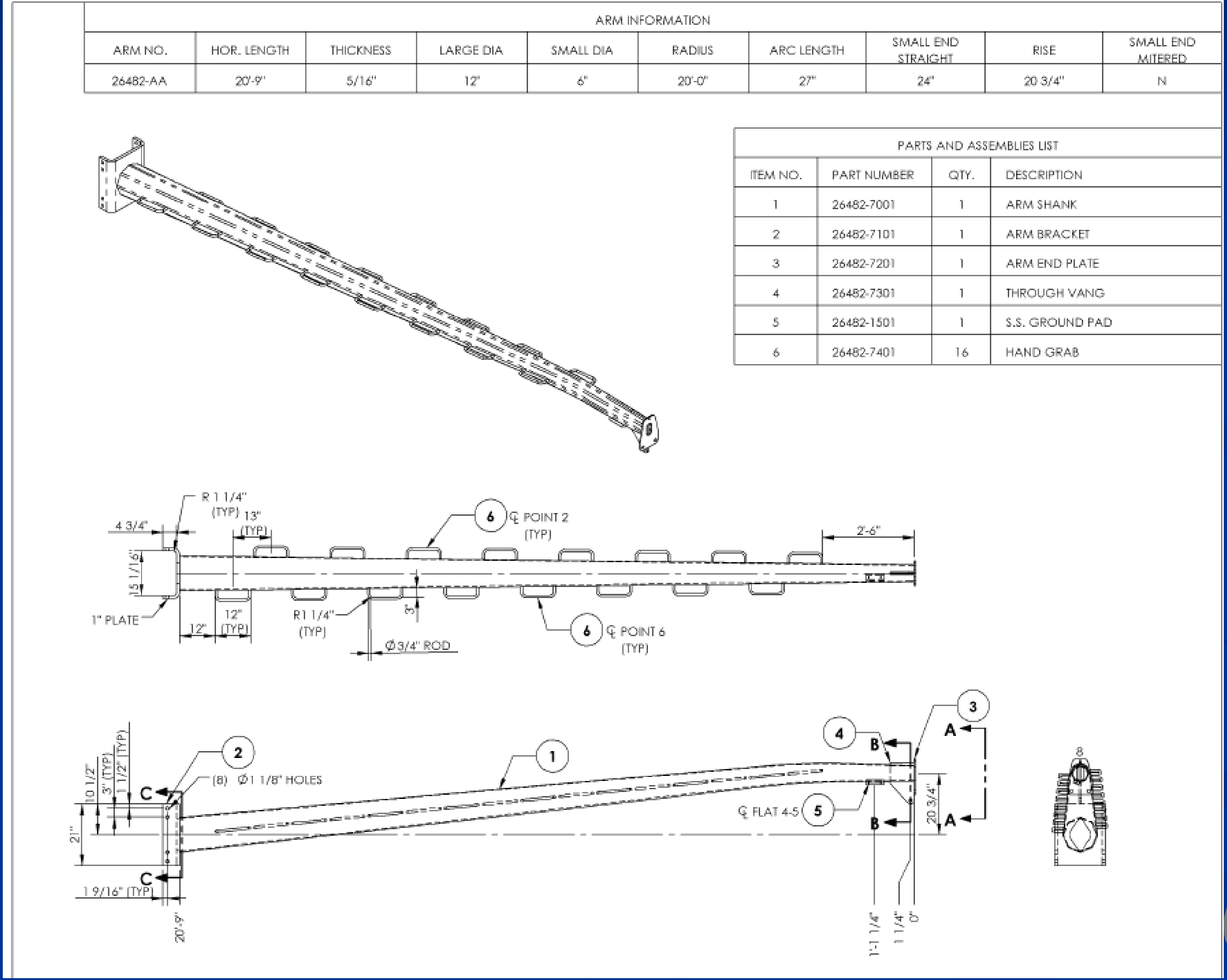
The 345 kV project in Wisconsin. Proposal to Utility:

- Two pole suppliers are providing structures for the project. POWER will build F.E. models of the shield wire arms and longer phase arms (middle phase) for both vendors that are supplying poles to the Project.
- Calculate the modal or natural frequencies associated with each unique arm – convert this to a modal excitation wind speed
- Approximate the lift and drag force along the arm due to this wind speed.
- Apply this lift force as a forcing function occurring at the resonant frequency
- Quantify the base reactions and convert those to a stress range. Compare that with recommended limits.
- Determine effective methods of damping the unloaded arms.

Part 1: Model Vendor ‘A’s tangent static arms



ARM INFORMATION

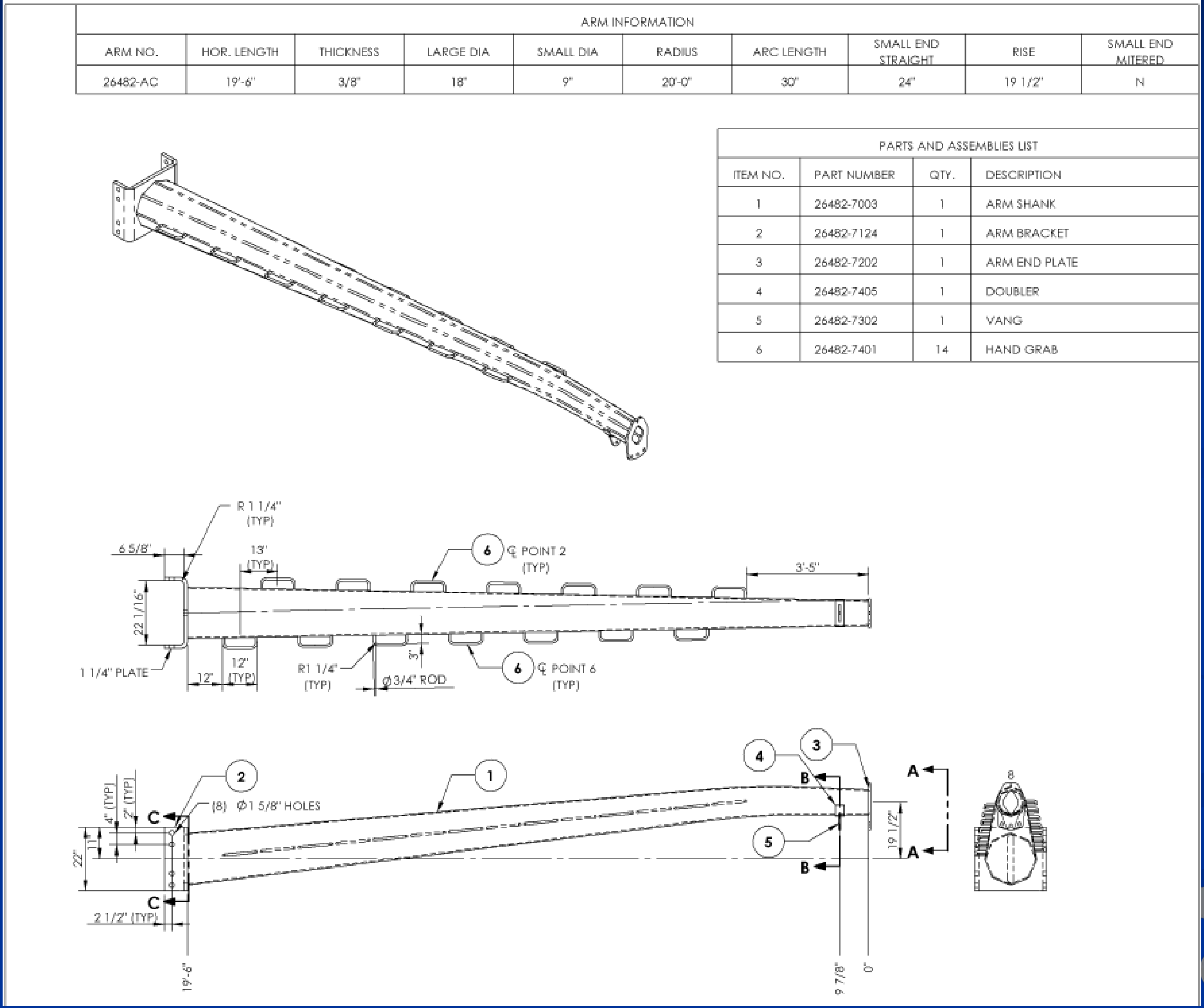
ARM NO.	HOR. LENGTH	THICKNESS	LARGE DIA	SMALL DIA	RADIUS	ARC LENGTH	SMALL END STRAIGHT	RISE	SMALL END MITERED
26482-AA	20'-9"	5/16"	12"	6"	20'-0"	27"	24"	20 3/4"	N

PARTS AND ASSEMBLIES LIST

ITEM NO.	PART NUMBER	QTY.	DESCRIPTION
1	26482-7001	1	ARM SHANK
2	26482-7101	1	ARM BRACKET
3	26482-7201	1	ARM END PLATE
4	26482-7301	1	THROUGH VANG
5	26482-1501	1	S.S. GROUND PAD
6	26482-7401	16	HAND GRAB

- 20’-9” arm
- 12” dia. base/6” dia. tip;
- 5/16” thick octagonal plate
- Wt = 800 lbs

And middle phase arms:

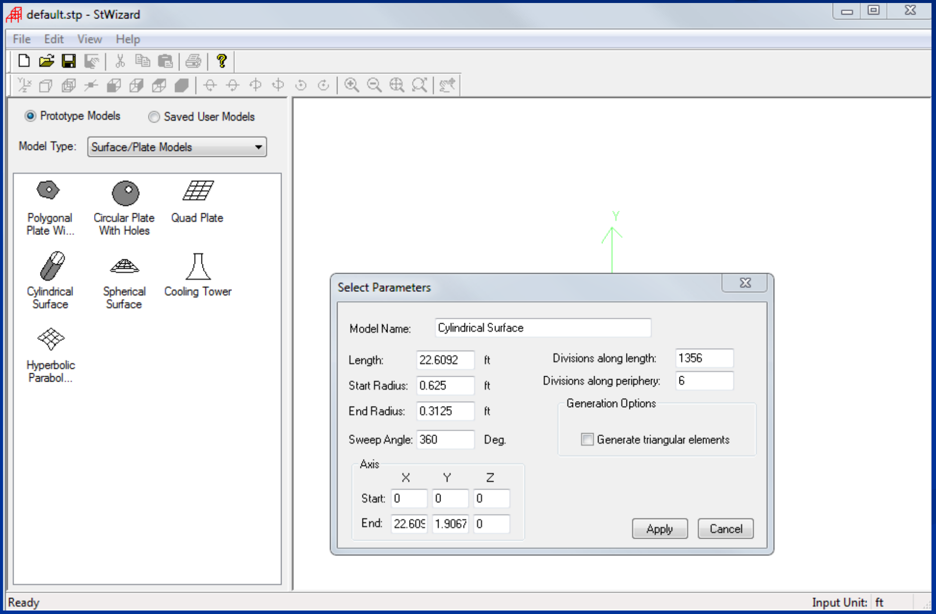


ARM INFORMATION									
ARM NO.	HOR. LENGTH	THICKNESS	LARGE DIA	SMALL DIA	RADIUS	ARC LENGTH	SMALL END STRAIGHT	RISE	SMALL END MITERED
26482-AC	19'-6"	3/8"	18"	9"	20'-0"	30"	24"	19 1/2"	N

PARTS AND ASSEMBLIES LIST			
ITEM NO.	PART NUMBER	QTY.	DESCRIPTION
1	26482-7003	1	ARM SHANK
2	26482-7124	1	ARM BRACKET
3	26482-7202	1	ARM END PLATE
4	26482-7405	1	DOUBLER
5	26482-7302	1	VANG
6	26482-7401	14	HAND GRAB

- 19'-6" arm
- 18" dia. base/9" dia. tip;
- 3/8" thick octagonal plate
- Wt = 1,400 lbs

In order to calculate the modal frequencies, the arm is modeled in STAAD-Pro using cylindrical surface prototype models



● Prototype Models

Model Type: Surface/Plate Models

Model Name: Cylindrical Surface

Length: 22.6092 ft

Start Radius: 0.625 ft

End Radius: 0.3125 ft

Sweep Angle: 360 Deg.

Divisions along length: 1356

Divisions along periphery: 6

Generation Options

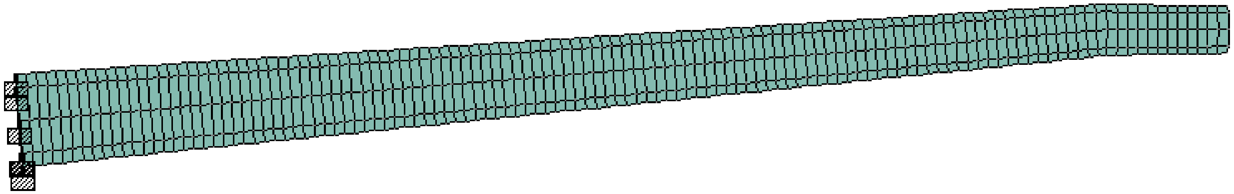
☐ Generate triangular elements

Axis

	X	Y	Z
Start:	0	0	0
End:	22.60	1.9067	0

Input Unit: ft

Middle phase arm STAAD model



- Plate elements are sub-divided to create well-behaved elements (less than 4:1 length-width ratios)
- End plates and vangs are modeled as vertical loads. These loads must be applied in all three global directions when using a dynamic analysis to calculate the natural frequency

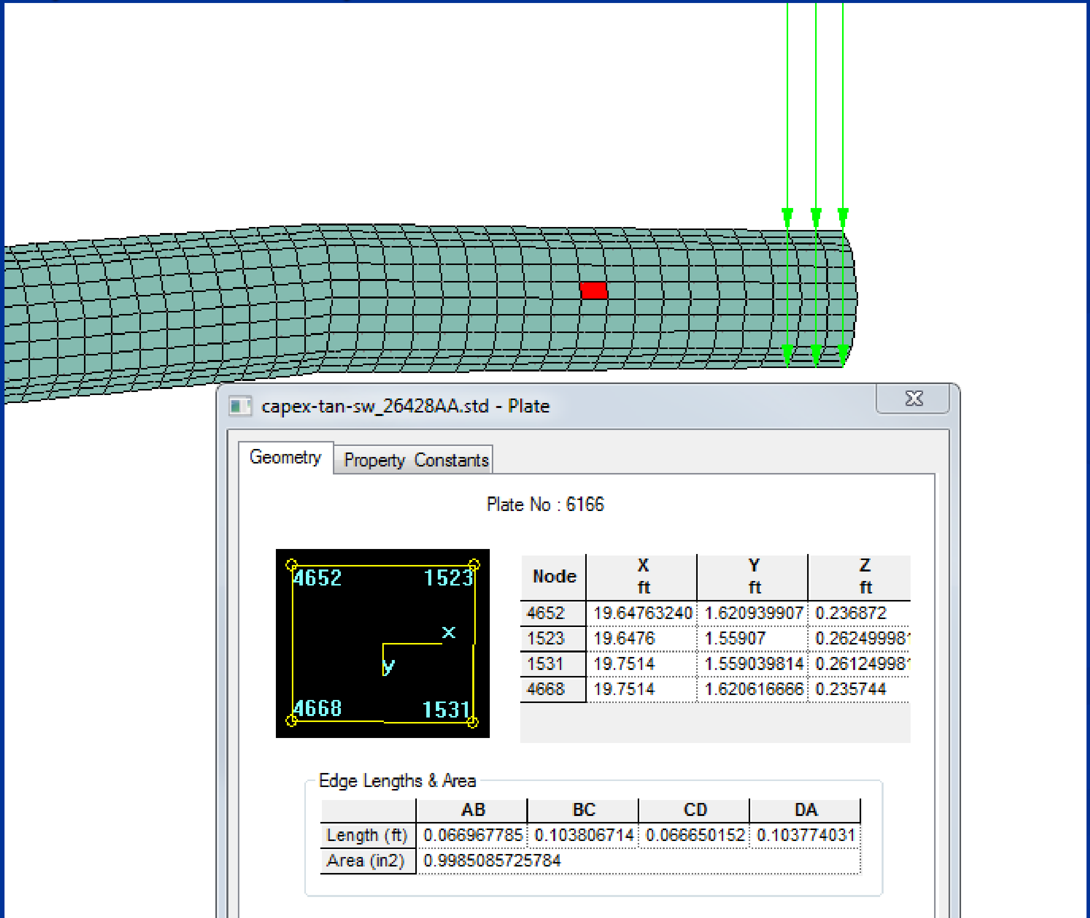


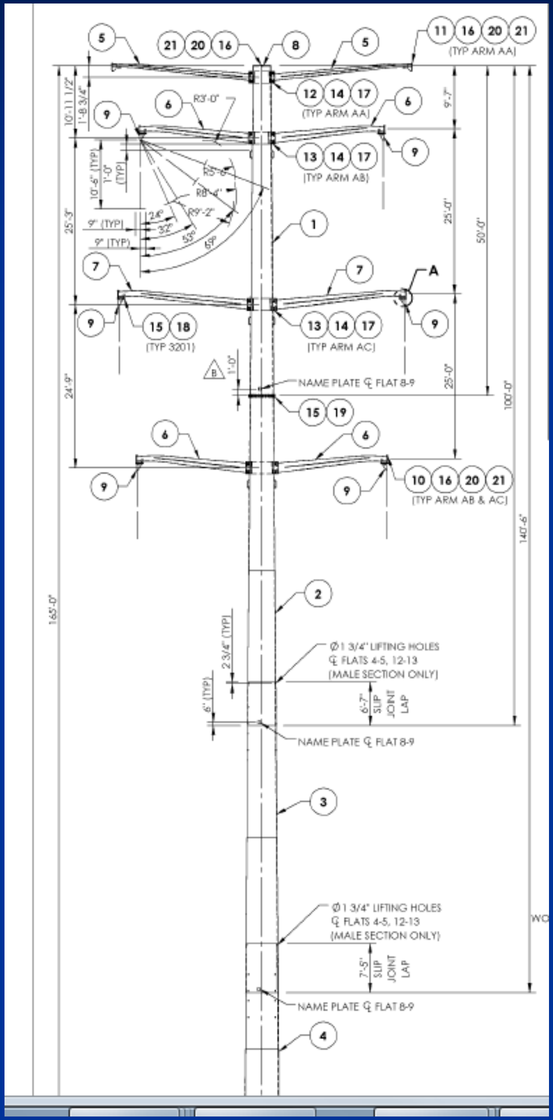
Plate No : 6166

Node	X ft	Y ft	Z ft
4652	19.64763240	1.620939907	0.236872
1523	19.6476	1.55907	0.26249998
1531	19.7514	1.559039814	0.26124998
4668	19.7514	1.620616666	0.235744

Edge Lengths & Area

	AB	BC	CD	DA
Length (ft)	0.066967785	0.103806714	0.066650152	0.103774031
Area (in2)	0.9985085725784			

ARM NATURAL FREQUENCIES



Arm Type	1st Natural Frequency, η_1	2nd Natural Frequency, η_2	Excitation Wind Speed for 1st mode	Excitation Wind Speed for 2nd mode
Tangent Static Arm	6.97 Hz	33.23 Hz	17.8 mph	85 mph
Tangent Phase Arm (Middle)	12.3 Hz	61.9Hz	39.5 mph	184.0 mph

Steady state winds that will excite the middle phase arm (~40 mph) are much less likely to occur than those speeds that will induce motion in the static arms (~18 mph). For simplicity and WOLOG, we will focus on the static wire arms.

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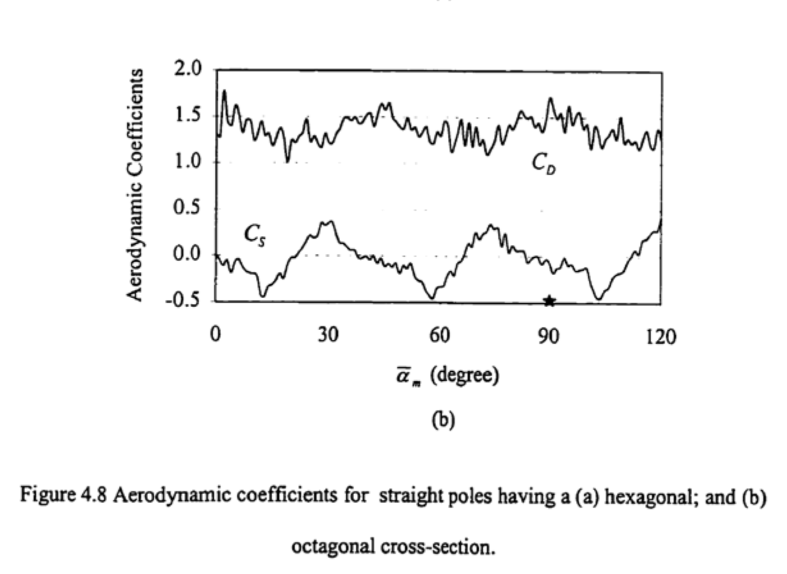
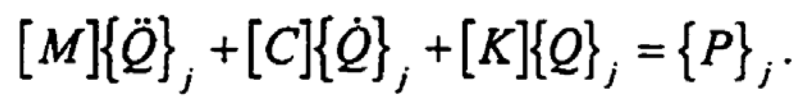
Aerodynamic Forces on an arm



(b)

Figure 4.8 Aerodynamic coefficients for straight poles having a (a) hexagonal; and (b) octagonal cross-section.



$$[M]\{\ddot{Q}\}_j + [C]\{\dot{Q}\}_j + [K]\{Q\}_j = \{P\}_j .$$

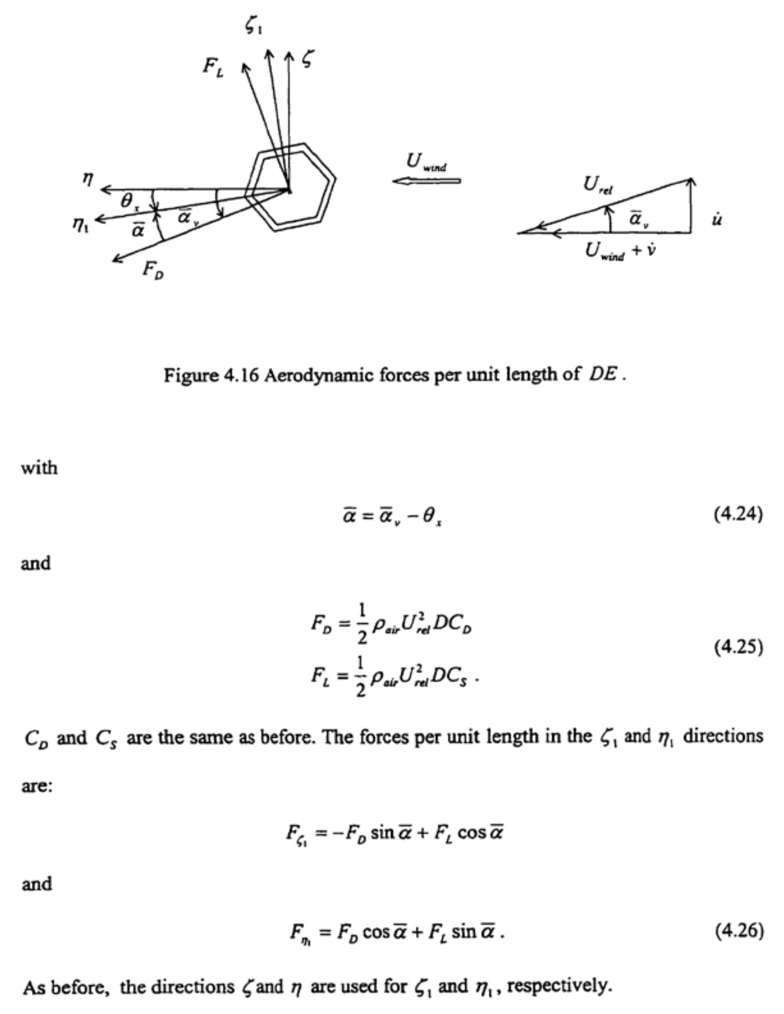


Figure 4.16 Aerodynamic forces per unit length of DE.

with

$$\bar{\alpha} = \bar{\alpha}_v - \theta_x \tag{4.24}$$

and

$$F_D = \frac{1}{2}\rho_{air} U_{rel}^2 D C_D$$
$$F_L = \frac{1}{2}\rho_{air} U_{rel}^2 D C_S . \tag{4.25}$$

C_D and C_S are the same as before. The forces per unit length in the ζ_1 and η_1 directions are:

$$F_{\zeta_1} = -F_D \sin\bar{\alpha} + F_L \cos\bar{\alpha}$$

and

$$F_{\eta_1} = F_D \cos\bar{\alpha} + F_L \sin\bar{\alpha} . \tag{4.26}$$

As before, the directions ζ and η are used for ζ_1 and η_1, respectively.

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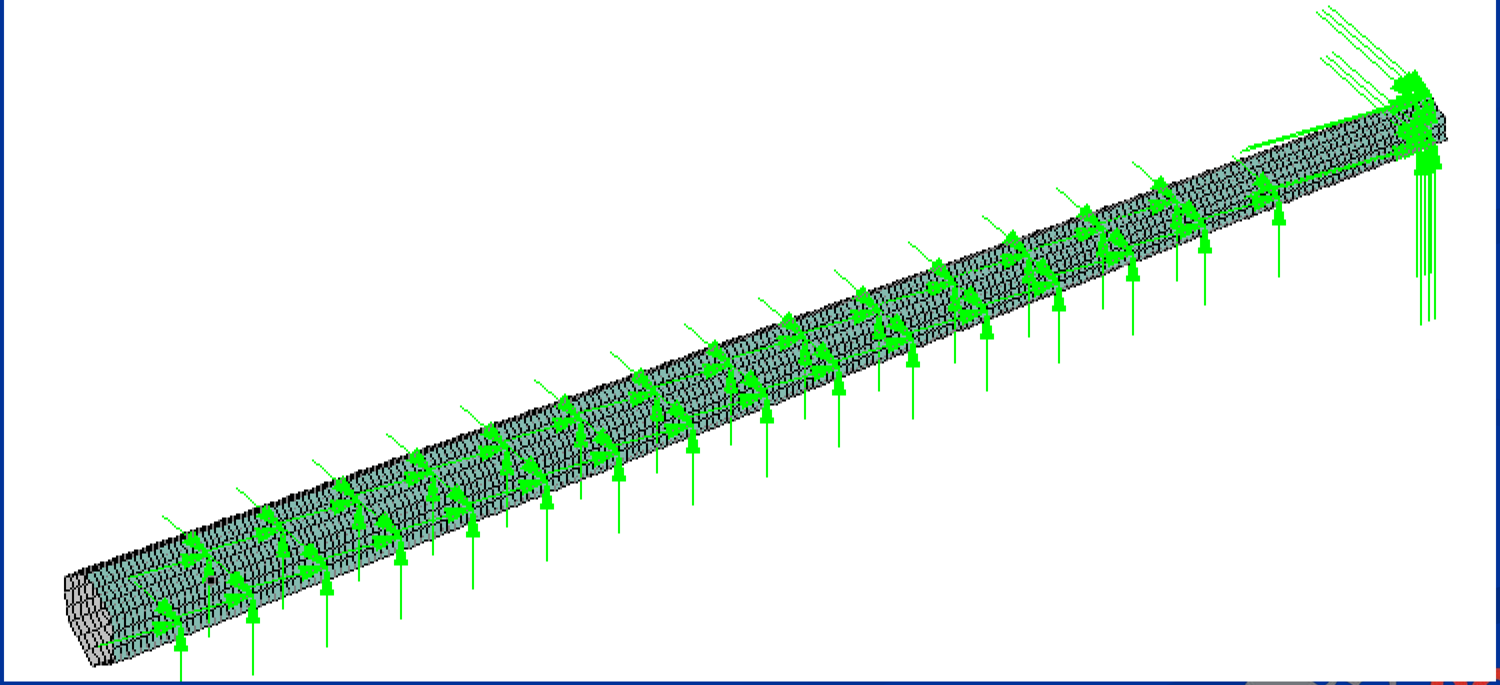
A dynamic analysis in STAAD is capable of performing a modal response based on the second order differential equation for driven harmonic oscillators:

$$\ddot{x} + 2\beta\eta\dot{x} + \eta^2 x = F(t)/m$$

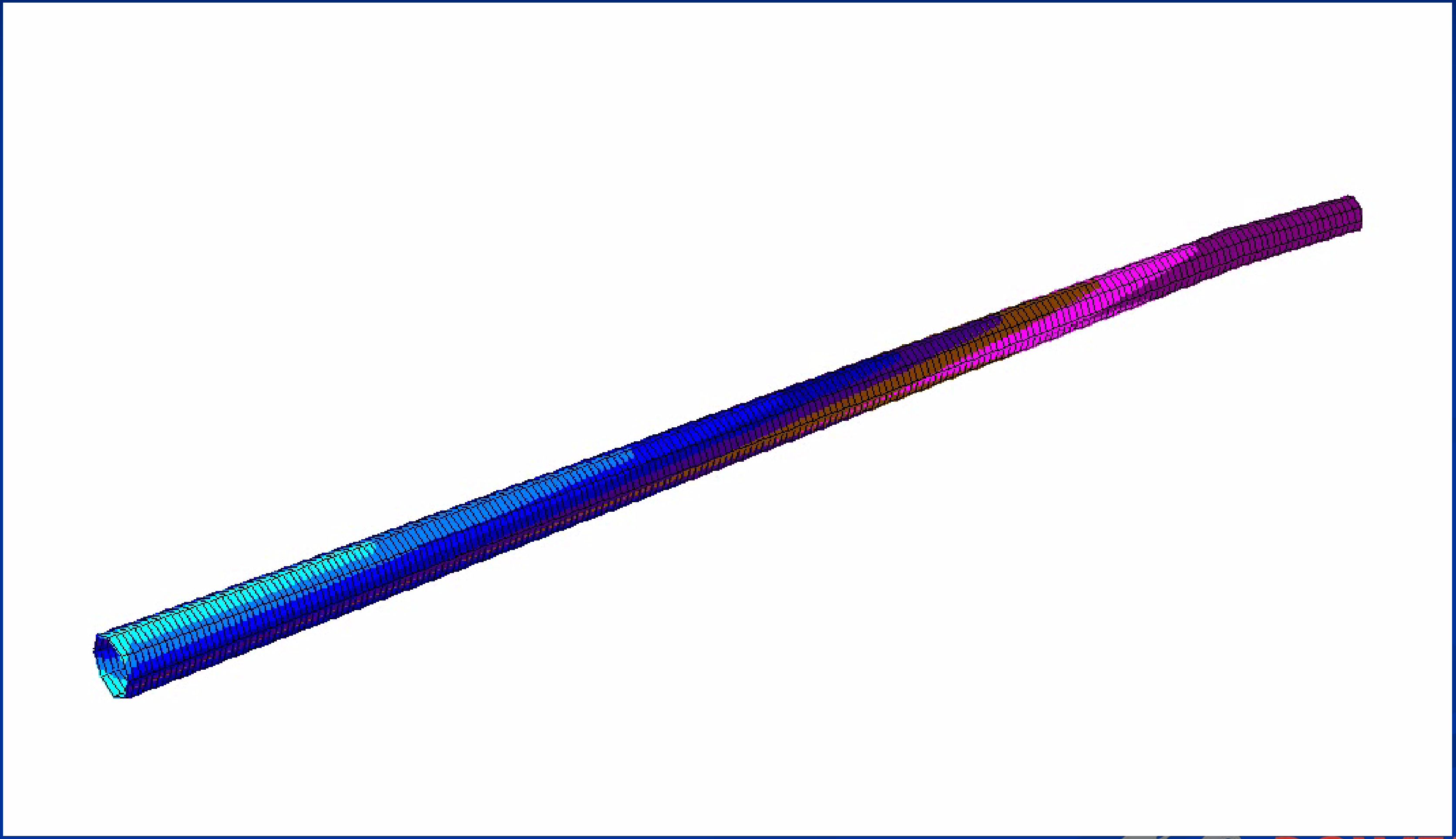
In this equation, β is the damping ratio and **h**is the natural frequency. The damping is the sum of the inherent structural damping (βs) and the aerodynamic damping (βa). The aerodynamic damping can be a negative value by a phenomenon known as ‘negative aerodynamic damping’ wherein the motion-induced forces are in phase with the velocity component of the structure. If the sum (βs + βa) is less than zero, this increases amplitude and the associated stress ranges on the shaft.

where $F(z,t) = 1/2C_L{*}\varrho{*}u(z)^2{*}D(z){*}\cos(\eta_i\, t + \zeta(t))$

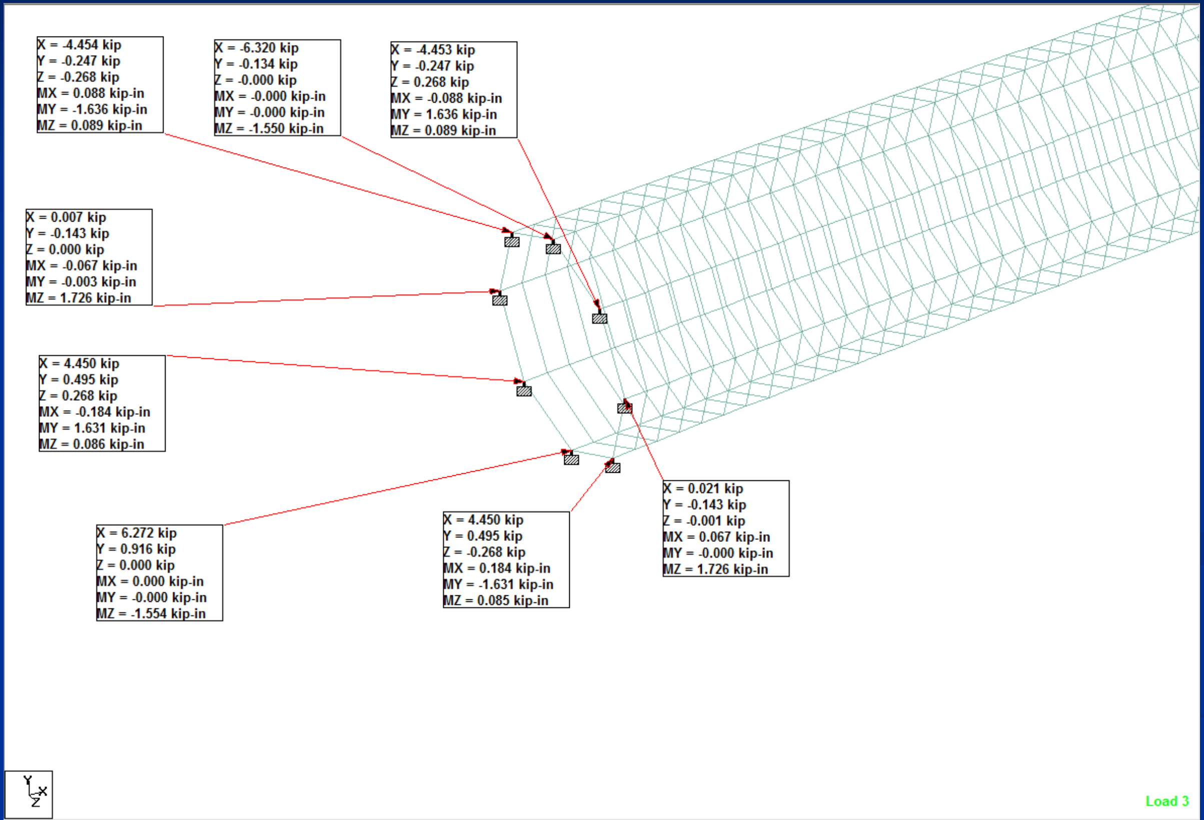
This is calculated and applied as discrete loads at nodes along the arm’s length.



Arm motion and stress contours:



Finite Element Model reactions



X = -4.454 kip
Y = -0.247 kip
Z = -0.268 kip
MX = 0.088 kip-in
MY = -1.636 kip-in
MZ = 0.089 kip-in

X = -6.320 kip
Y = -0.134 kip
Z = -0.000 kip
MX = -0.000 kip-in
MY = -0.000 kip-in
MZ = -1.550 kip-in

X = -4.453 kip
Y = -0.247 kip
Z = 0.268 kip
MX = -0.088 kip-in
MY = 1.636 kip-in
MZ = 0.089 kip-in

X = 0.007 kip
Y = -0.143 kip
Z = 0.000 kip
MX = -0.067 kip-in
MY = -0.003 kip-in
MZ = 1.726 kip-in

X = 4.450 kip
Y = 0.495 kip
Z = 0.268 kip
MX = -0.184 kip-in
MY = 1.631 kip-in
MZ = 0.086 kip-in

X = 0.021 kip
Y = -0.143 kip
Z = -0.001 kip
MX = 0.067 kip-in
MY = -0.000 kip-in
MZ = 1.726 kip-in

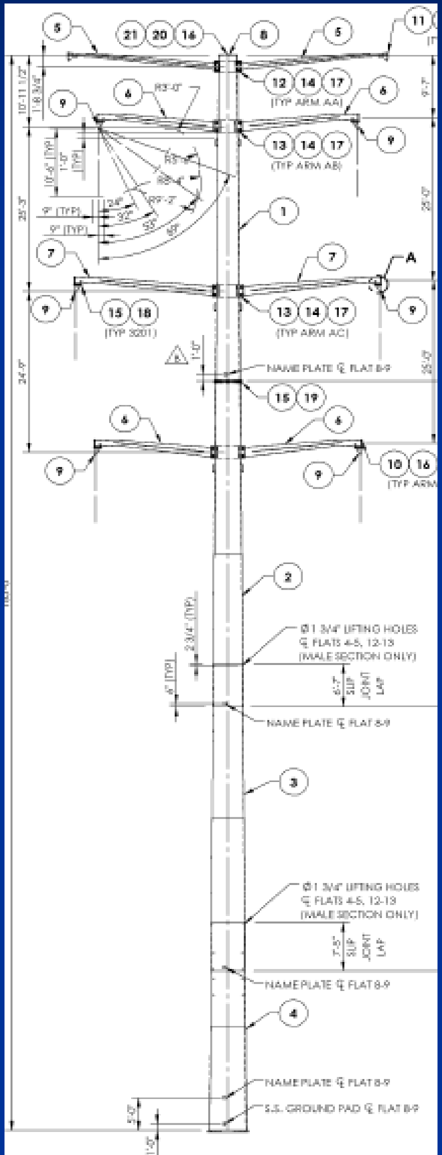
X = 6.272 kip
Y = 0.916 kip
Z = 0.000 kip
MX = 0.000 kip-in
MY = -0.000 kip-in
MZ = -1.554 kip-in

X = 4.450 kip
Y = 0.495 kip
Z = -0.268 kip
MX = 0.184 kip-in
MY = -1.631 kip-in
MZ = 0.085 kip-in

- M=Sfxi*zi
- Fb = M/S

Forcing function is applied as a time history load in the (+) and (-) y direction and combined with the gravity loads. The cyclic stress range is determined by taking the difference in the two resulting reactions.

Dynamic Stress Approximation:



Vendor A	Length	Weight	Base/Tip O.D.	Dynamic Stress Range- undamped	Dynamic Stress Range- 50 lb damper	Dynamic Stress Range- 100 lb damper
Tangent Static Arm	20’-9”	800 lbs	12”/6”	12.4 ksi	2.9 ksi	1.6 ksi

Arm Specifications and Calculated Stresses

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FIELD TESTING:

- ESI Engineering, INC performed an experimental modal analysis with the following goals:
- Determine the natural frequencies of the static and middle-phase arm
- Determine the (structural) damping in each arm

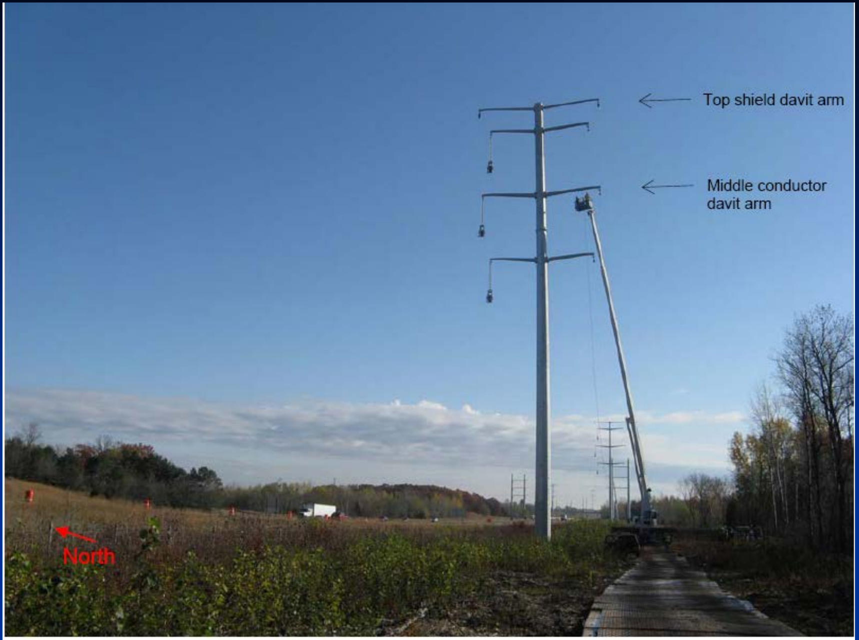
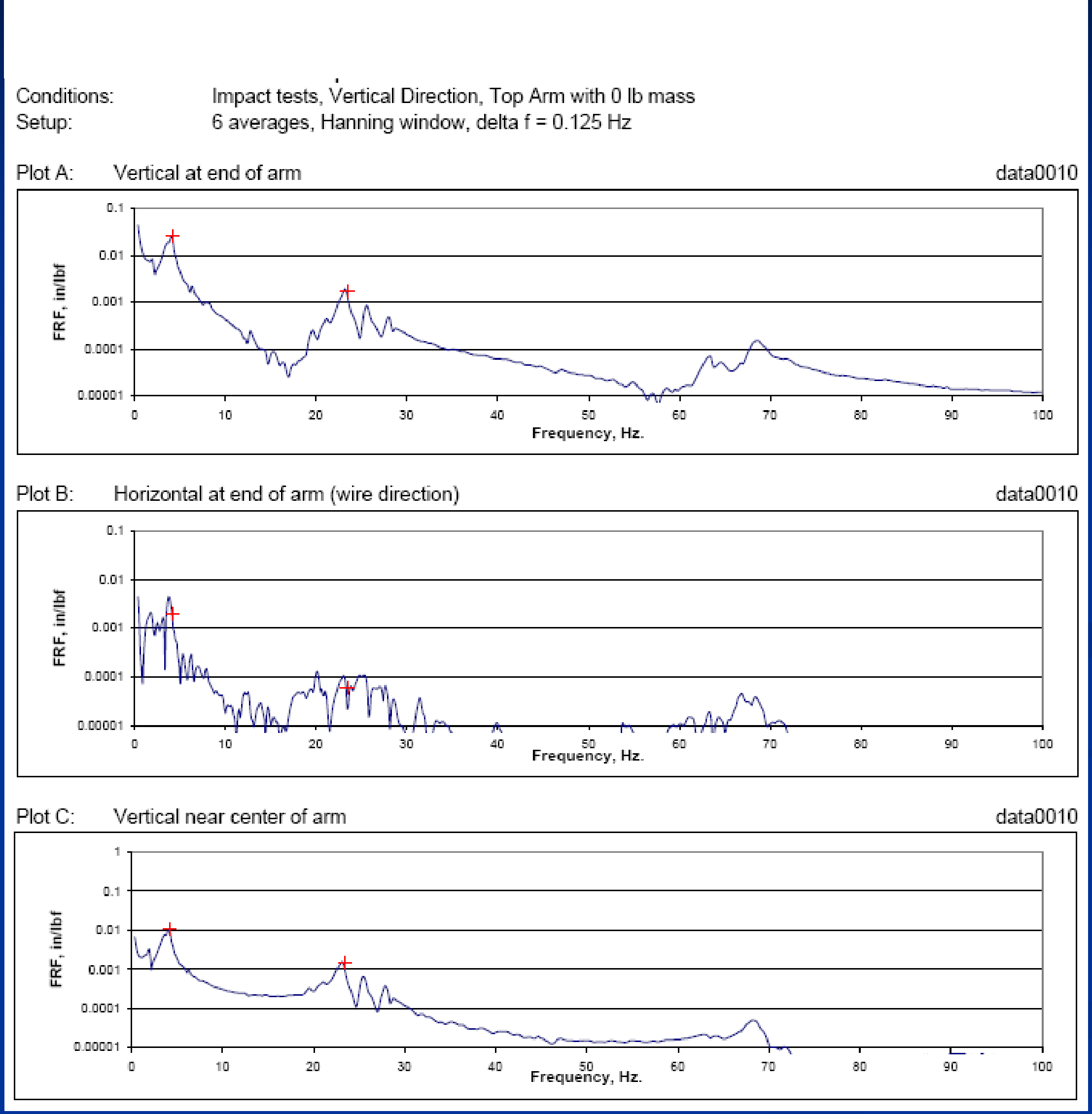


Figure 1 – Photograph of transmission pole 125 tested.

Figure 3 – Photograph showing a vertical impact of a davit arm with the modal impact hammer.

This field measurement consisted of a modal impact hammer, three accelerometers, and a FFT (Fast Fourier Transform) to get the FRF (Frequency Response Function)

Field data for static arm after the FFT. The red cross-hairs indicate the modal frequencies



Conditions: Impact tests, Vertical Direction, Top Arm with 0 lb mass

Setup: 6 averages, Hanning window, delta f = 0.125 Hz

Plot A: Vertical at end of arm data0010

Plot B: Horizontal at end of arm (wire direction) data0010

Plot C: Vertical near center of arm data0010

The field set-up and results:

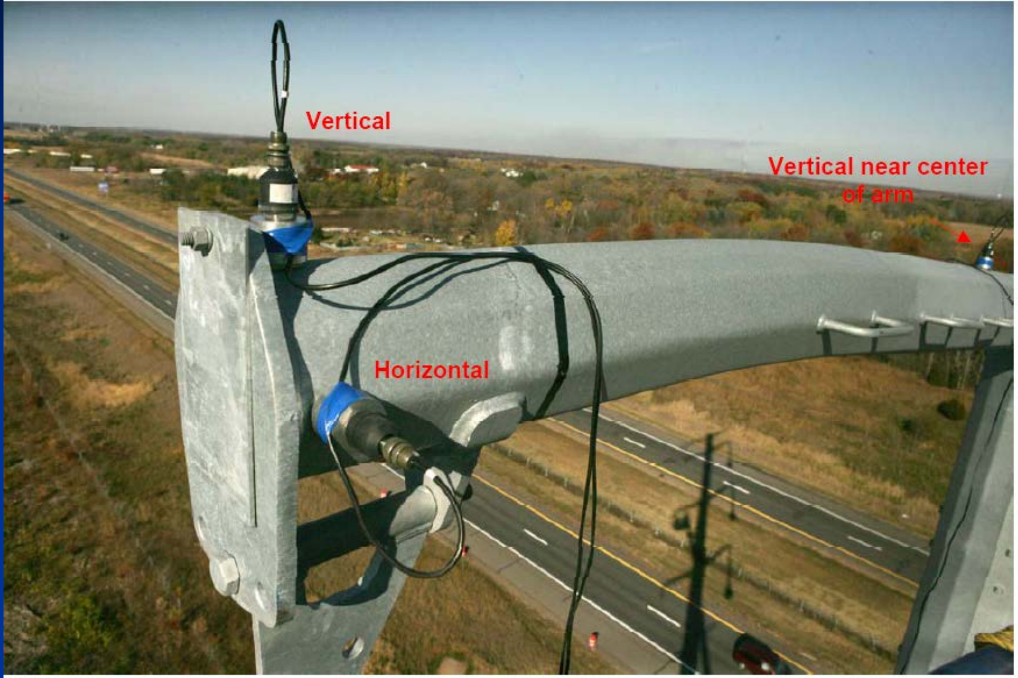


Figure 2 – Photograph of the middle conductor davit arm showing the accelerometer orientation.

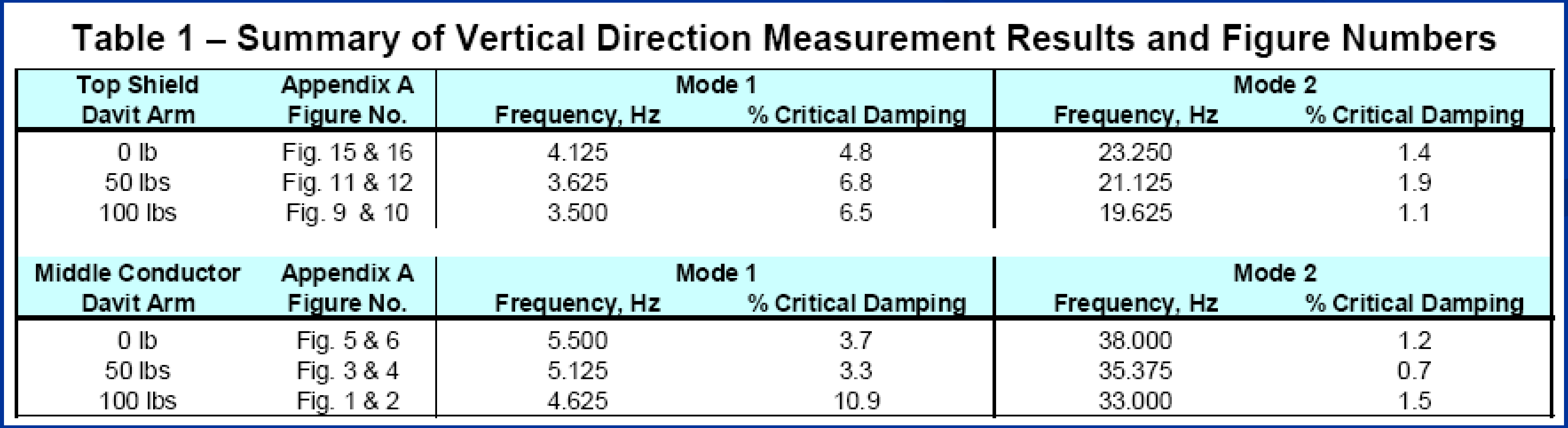


Table 1 – Summary of Vertical Direction Measurement Results and Figure Numbers

Top Shield Davit Arm	Appendix A Figure No.	Mode 1 Frequency, Hz	Mode 1 % Critical Damping	Mode 2 Frequency, Hz	Mode 2 % Critical Damping
0 lb	Fig. 15 & 16	4.125	4.8	23.250	1.4
50 lbs	Fig. 11 & 12	3.625	6.8	21.125	1.9
100 lbs	Fig. 9 & 10	3.500	6.5	19.625	1.1

Middle Conductor Davit Arm	Appendix A Figure No.	Mode 1 Frequency, Hz	Mode 1 % Critical Damping	Mode 2 Frequency, Hz	Mode 2 % Critical Damping
0 lb	Fig. 5 & 6	5.500	3.7	38.000	1.2
50 lbs	Fig. 3 & 4	5.125	3.3	35.375	0.7
100 lbs	Fig. 1 & 2	4.625	10.9	33.000	1.5

Adjustments to the STAAD models

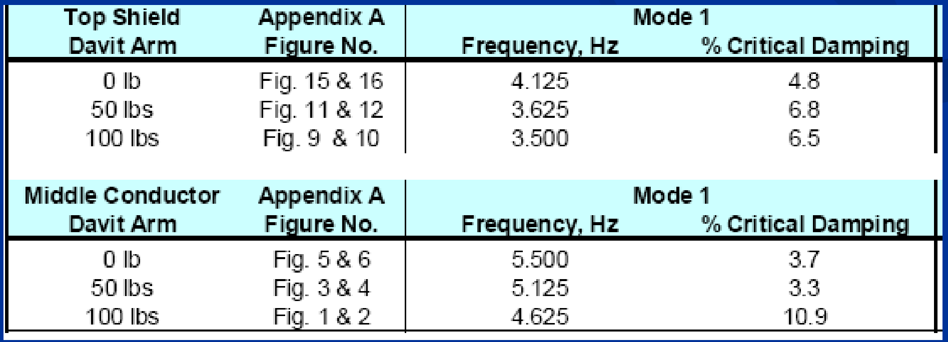
- Acknowledging that the base of the arm is not truly 'fixed', we adjusted the supports to have a spring constant of 2400 kip/ft in all three axes to match the field measured natural frequencies of the bare arm. The modal frequencies with 50lb and 100lb weights were checked against field measured values with good agreement.

Tangent Static Arm Measured	Tangent Static Arm Calculated	Tangent Static Arm STAAD w/spring supports
4.125 Hz	3.23 Hz	4.133 Hz

Damping

- Damping was a larger concern. The experimental procedure induced erroneous readings. Damping values affect cyclic stress values at the base of the arm.

- Initially assumed a value of .03, modified this to .019 based on field results and client input
- 'Slack rope effect' with suspended weights made the measured values suspect:



Top Shield Davit Arm	Appendix A Figure No.	Mode 1 Frequency, Hz	Mode 1 % Critical Damping
0 lb	Fig. 15 & 16	4.125	4.8
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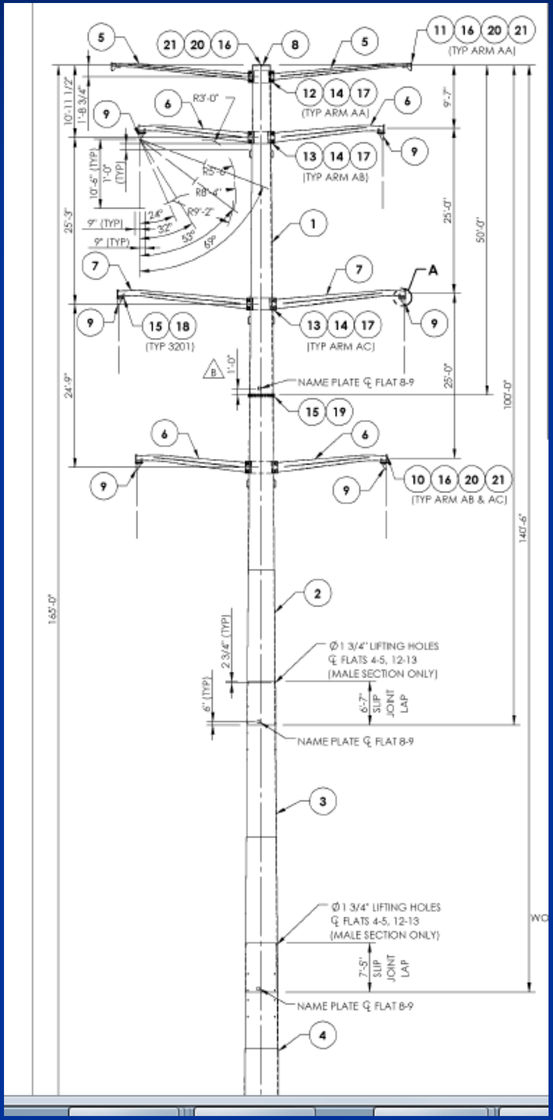
100 lbs	Fig. 9 & 10	3.500	6.5

Middle Conductor Davit Arm	Appendix A Figure No.	Mode 1 Frequency, Hz	Mode 1 % Critical Damping
0 lb	Fig. 5 & 6	5.500	3.7
50 lbs	Fig. 3 & 4	5.125	3.3
100 lbs	Fig. 1 & 2	4.625	10.9

MODIFIED RESULTS



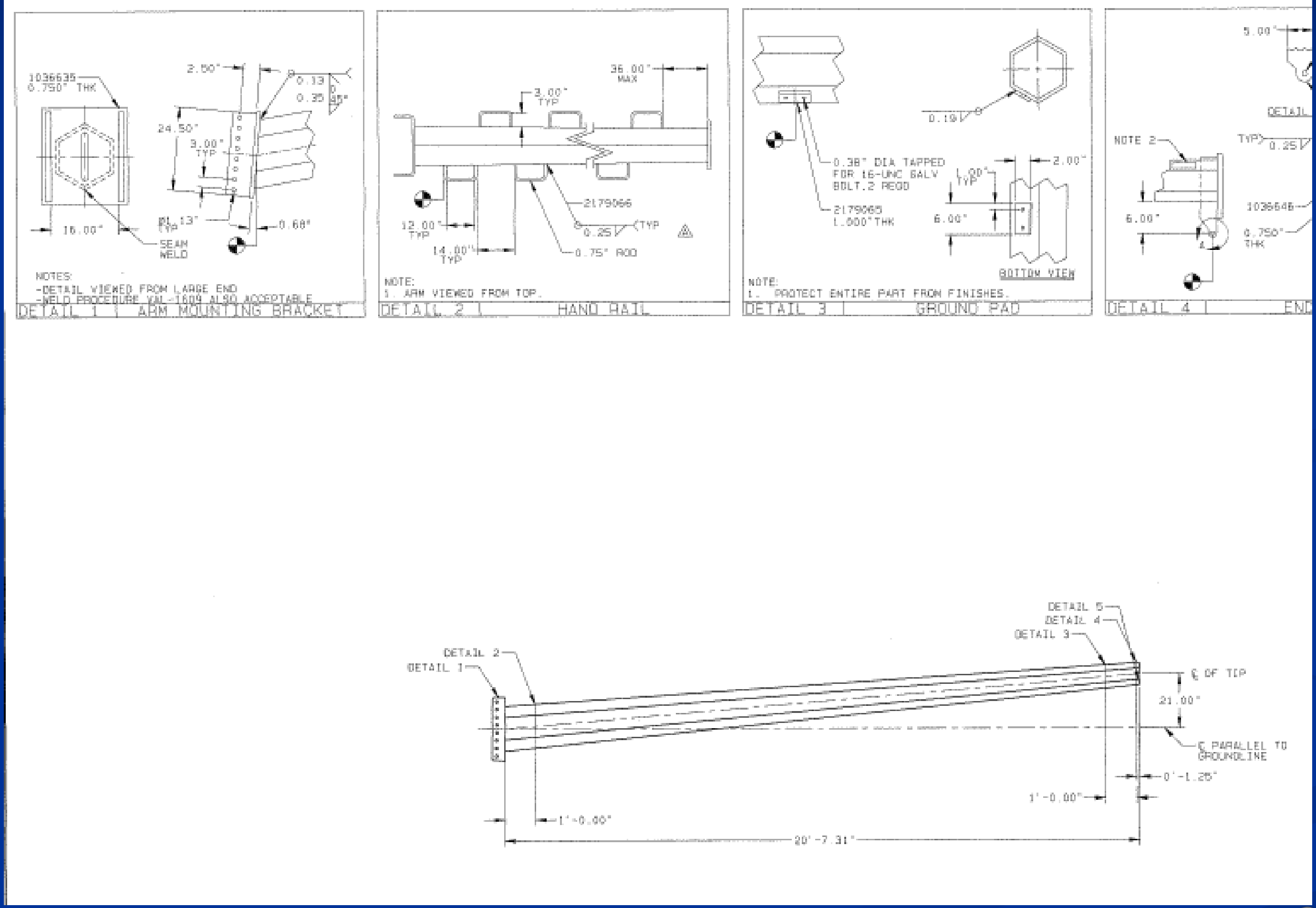
Original results:

Arm Type	Length	Weight	Base/Tip O.D.	Dynamic Stress Range- undamped	Dynamic Stress Range- 50 lb damper	Dynamic Stress Range-100 lb damper
Tangent Static Arm	20’-9”	800 lbs	12”/6”	12.4 ksi	2.9 ksi	1.6 ksi

Stresses with adjusted modal frequency and damping:

Arm Type	Length	Weight	Base/Tip O.D.	Dynamic Stress Range- undamped	Dynamic Stress Range- 50 lb damper	Dynamic Stress Range-100 lb damper
Tangent Static Arm	20’-9”	800 lbs	12”/6”	8.0 ksi	1.6 ksi	0.8 ksi

Part 2: Model Vendor B's static arm:



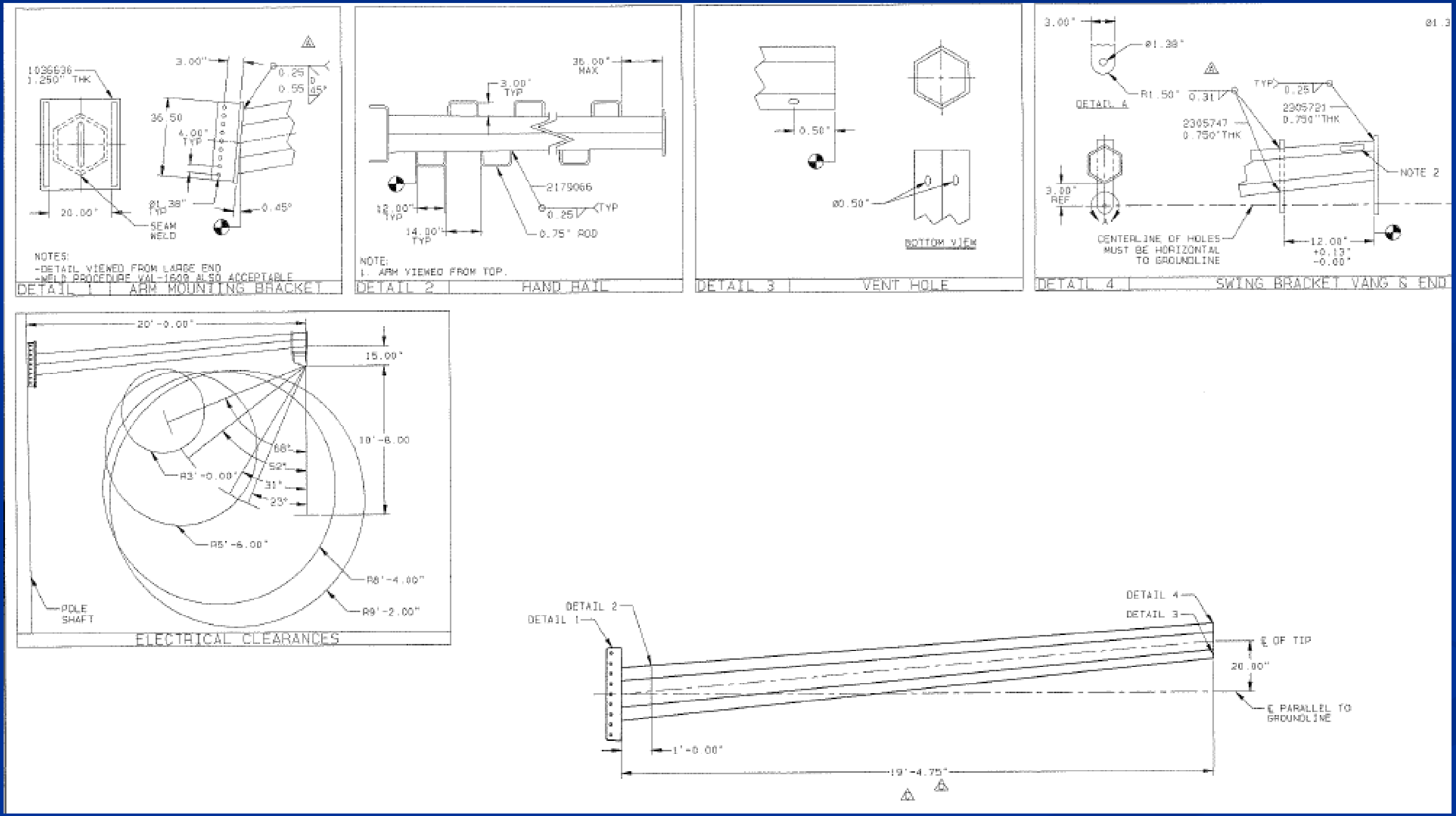
NOTES:
-DETAIL VIEWED FROM LARGE END
-WELD PROCEDURE YAL-1609 ALSO ACCEPTABLE

NOTE:
1. ARM VIEWED FROM TOP.

NOTE:
1. PROTECT ENTIRE PART FROM FINISHES.

- 21'-0" arm
- 15" dia. base/ 7.5" dia. tip
- 3/16" hexagonal plate
- Wt=756 lbs

And phase arm:



- •20’-0” arm
- •18” dia. base/ 12” dia. tip
- •5/16” hexagonal plate
- •Wt=1,627 lbs

ARM NATURAL FREQUENCIES

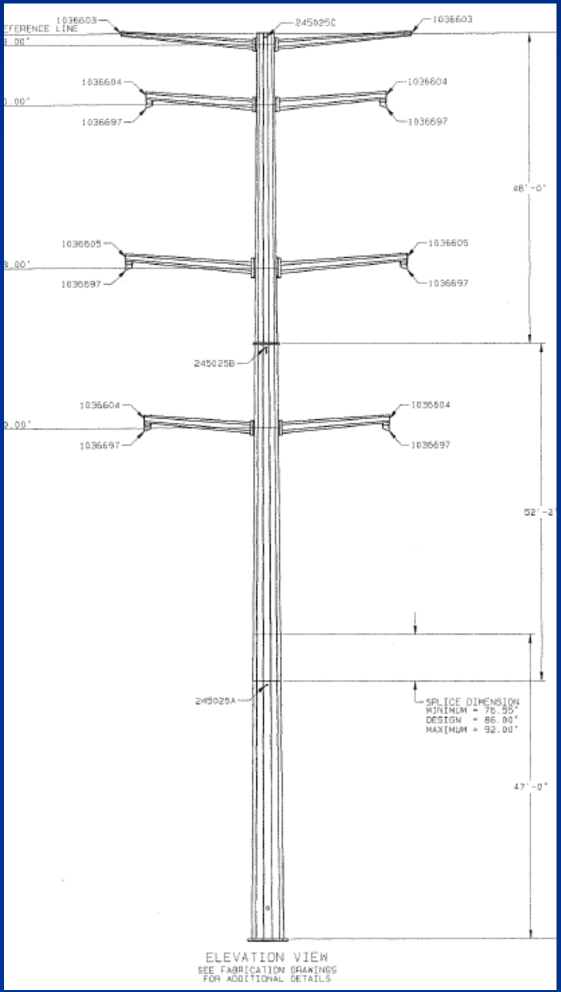
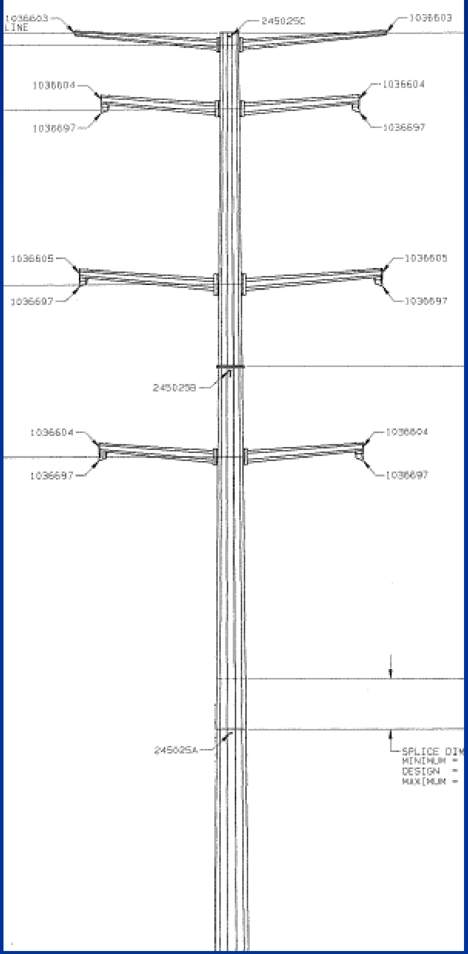


Table 1: Arm Modal Frequencies

Arm Type	1^{st} Natural Frequency, η_1	2^{nd} Natural Frequency, η_2	Excitation Wind Speed for 1^{st} mode	Excitation Wind Speed for 2^{nd} mode
Tangent Static Arm	9.6 Hz	45.4 Hz	31.0 mph	145.1 mph
Tangent Phase Arm (Middle)	12.04 Hz	49.24 Hz	38.4 mph	156.6 mph

Hexagonal arms have a higher first mode natural frequency. Thus a greater steady-state wind speed is required to induce motion.

Dynamic Stress Approximation:



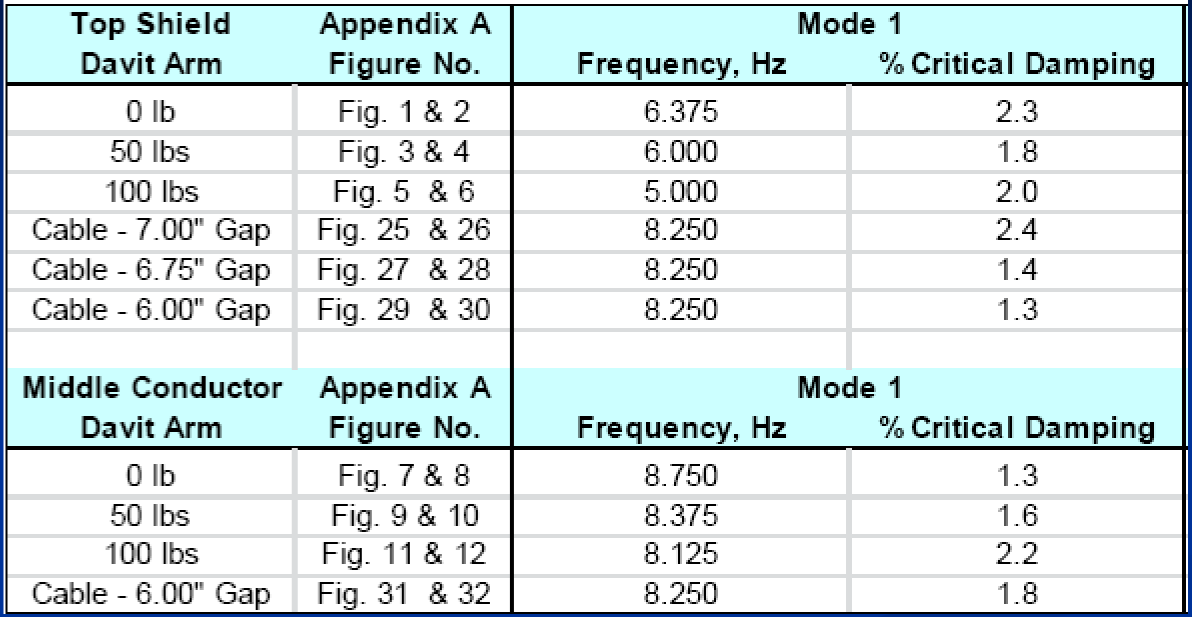
Arm Type	Length	Weight	Base/Tip O.D.	Dynamic Stress Range-undamped	Dynamic Stress- 50 lb damper	Dynamic Stress-100 lb damper
Tangent Static Arm	20’-7”	756 lbs	15”/7.5”	39 ksi	7.8 ksi	4.2 ksi

Arm Specifications and Calculated Stresses

Continued Field Testing:

- Test the shield wire arms from the Vendor B
- Improved the mass attachment to avoid ‘slack rope’ nonlinearity during the measurements.
- Investigated the effectiveness of tying the arms together during this exercise.

Results:



Top Shield Davit Arm	Appendix A Figure No.	Mode 1	
		Frequency, Hz	% Critical Damping
0 lb	Fig. 1 & 2	6.375	2.3
50 lbs	Fig. 3 & 4	6.000	1.8
100 lbs	Fig. 5 & 6	5.000	2.0
Cable - 7.00" Gap	Fig. 25 & 26	8.250	2.4
Cable - 6.75" Gap	Fig. 27 & 28	8.250	1.4
Cable - 6.00" Gap	Fig. 29 & 30	8.250	1.3
Middle Conductor Davit Arm	**Appendix A Figure No.**	**Mode 1**	
		Frequency, Hz	**% Critical Damping**
0 lb	Fig. 7 & 8	8.750	1.3
50 lbs	Fig. 9 & 10	8.375	1.6
100 lbs	Fig. 11 & 12	8.125	2.2
Cable - 6.00" Gap	Fig. 31 & 32	8.250	1.8

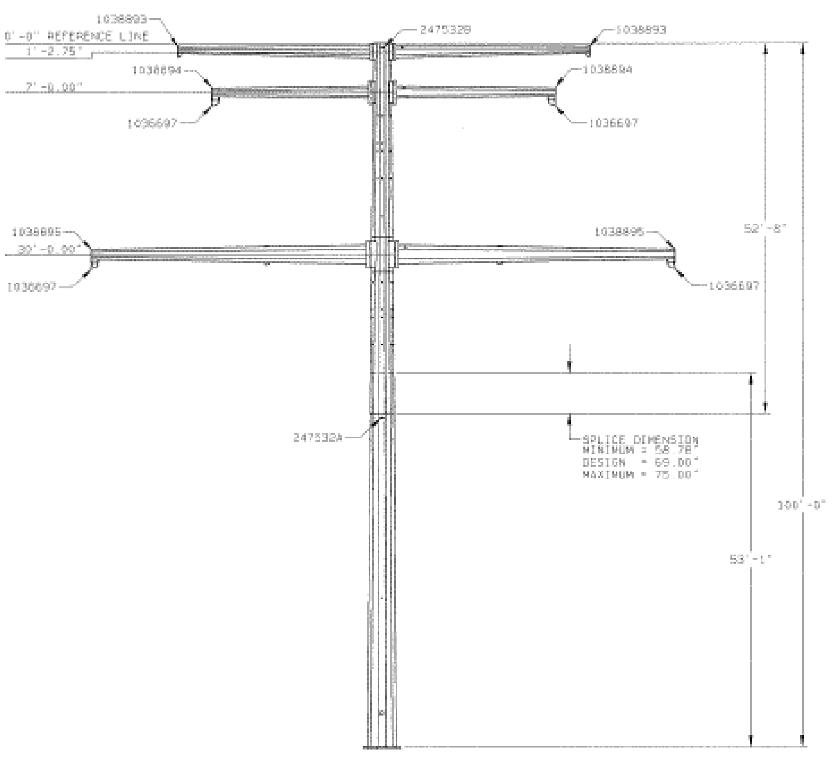
- These hexagonal arms have a higher 1st modal frequency which implies a higher steady-state wind speed is required to induce motion from vortex shedding.
- Different spring constants at the support were required to match the field measured modal frequencies. This is due to the difference in the arm connection.
- Note that changing the tensions in hold-down cables does not affect the modal frequency. The upper/lower arm system adopts a frequency close to that of the lower arm.

Dynamic stress comparison's with spring supports and adjusted % critical damping

Arm Type	Length	Weight	Base/Tip O.D.	Dynamic Stress Range-undamped	Dynamic Stress Range- 50 lb damper	Dynamic Stress-100 lb damper
Vendor A	20'-9"	800 lbs	12"/6"	8.0 ksi	1.6 ksi	0.8 ksi
Vendor B	20'-9"	756 lbs	15"/7.5"	34.7 ksi	9.6 ksi	4.1 ksi

- The fixity and critical damping were altered based on field measurements for Vendor B's arms.
- The above table compares stress ranges. NOTE: This does not imply that Vendor A's arms are superior! Recall the required steady-state wind speeds:
    - Vendor A: 9 mph
    - Vendor B: 21 mph

Part 3: Analyze a Modified Configuration

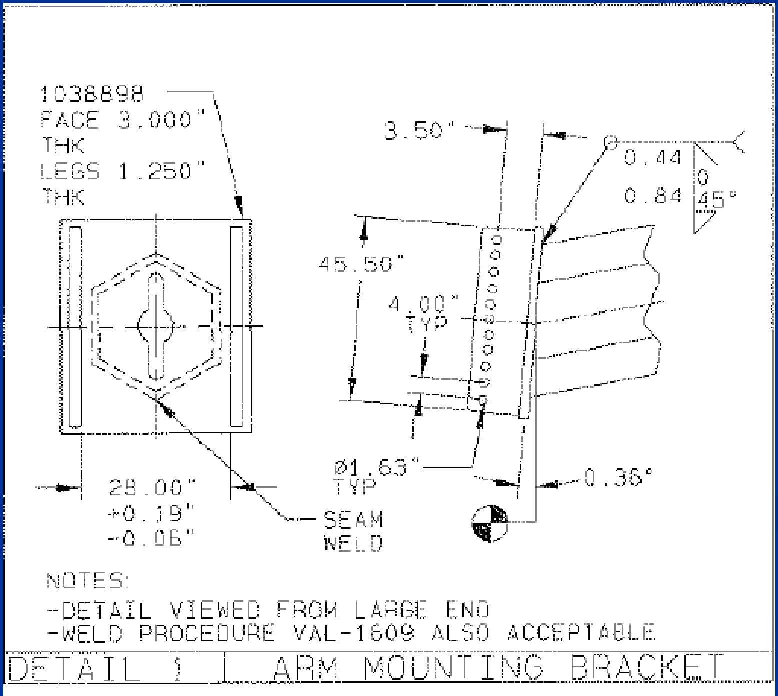


Static Arm:
- 28'-0" arm
- 18" dia. base/ 9" dia. tip
- 7/32" hexagonal plate
- Wt=1,259 lbs

Lower phase arm:
- 40'-0" arm
- 28" dia. base/ 15" dia. tip
- 1/2" hexagonal plate
- Wt=6,729 lbs

Lower phase arm is a different animal than anything studied to-date:

- 3” ‘base plate’
- Mounted at a lower elevation (stiffer section of pole)
- Can we assume the same spring constants at the supports that were used for previous models?



Static and lower phase arm results:

Arm Type	1st Natural **Frequency, η_1**	2nd Natural **Frequency, η_2**	Excitation Wind Speed for 1st mode	Excitation Wind Speed for 2nd mode
Tangent Shield Wire Arm	4.3 Hz	22 Hz	16.5 mph	86.1 mph
Tangent Lower Phase Arm	2.7 Hz	17 Hz	16.3 mph	103.8 mph

- Static arm follows same trend as previously tested static arms
- Note that the massive phase arm has a low first mode excitation wind speed.
- These arms will be field tested soon (today, in fact).

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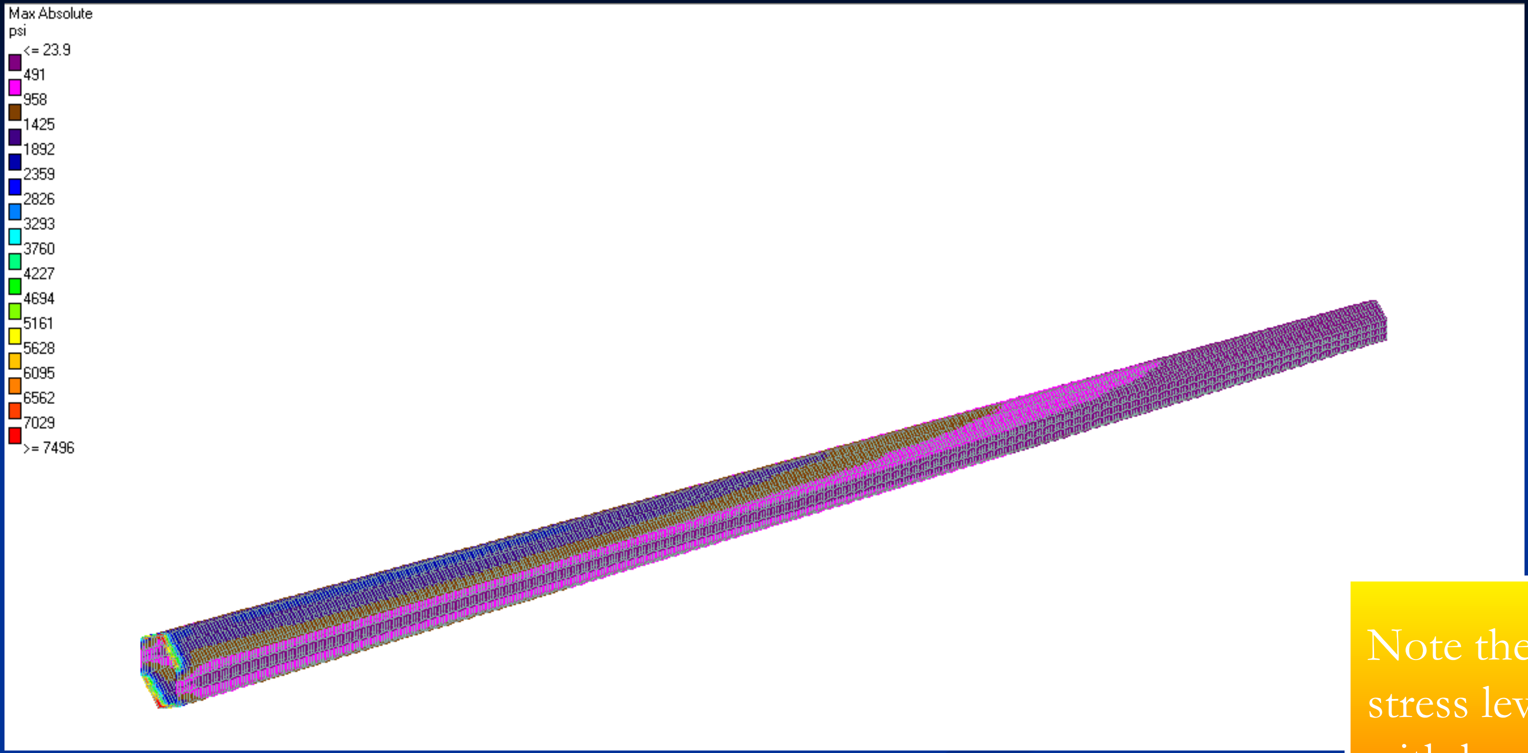

Lower Phase Arm Modeled as Fixed Supports:

Arm Type	Length	Weight	Base/ Tip O.D.	Dynamic Stress-undamped	Dynamic Stress-50 lb damper	Dynamic Stress Range-100 lb damper	Dynamic Stress Range-150 lb damper
Lower Phase Arm	39’-4”	6,729 lbs	28”/15”	24.1 ksi	17.6 ksi	13.5 ksi	12.1 ksi

Note the higher stress levels even with heavy mass damping

But what do these stress ranges mean…

The Fundamentals of Metal Fatigue Analysis

Definition: Metal fatigue is a process which causes premature failure or damage of a component subjected to repeated loading.

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Typical S-N curve for wrought Steels

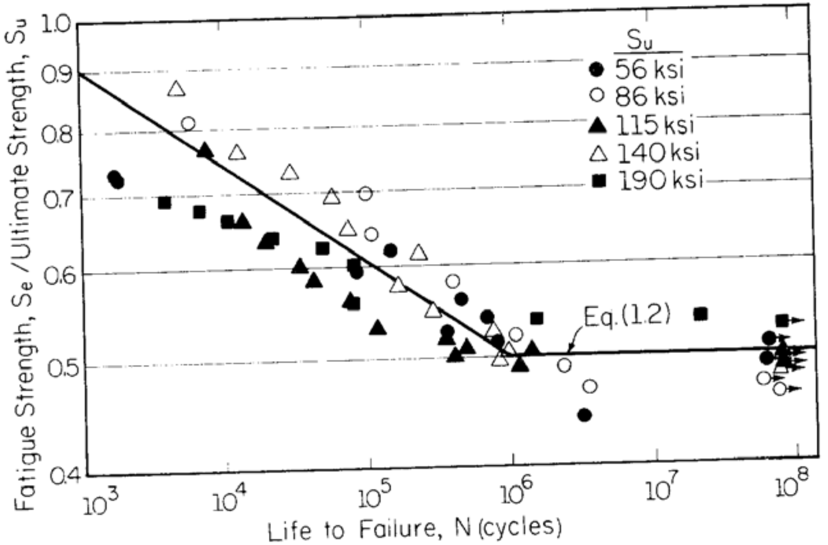

Figure 1.3 *S–N* curves for several wrought steels, plotted in ratio form (S_e/S_u).

For A572 Gr 65
Steel, Su = 80ksi

Other factors affecting the shape of an S-N curve:

- Loading Effects (variable amplitude load)
- Surface finish
- Size (Thickness adversely affects fatigue strength in welds)
- Se’ (modified endurance limit) $= S_e{*}C_{size}{*}C_{load}{*}C_{surf.\ finish\ldots}$

AVAILABLE CODES ADDRESSING FATIGUE:

1. AASHTO FATIGUE CURVES

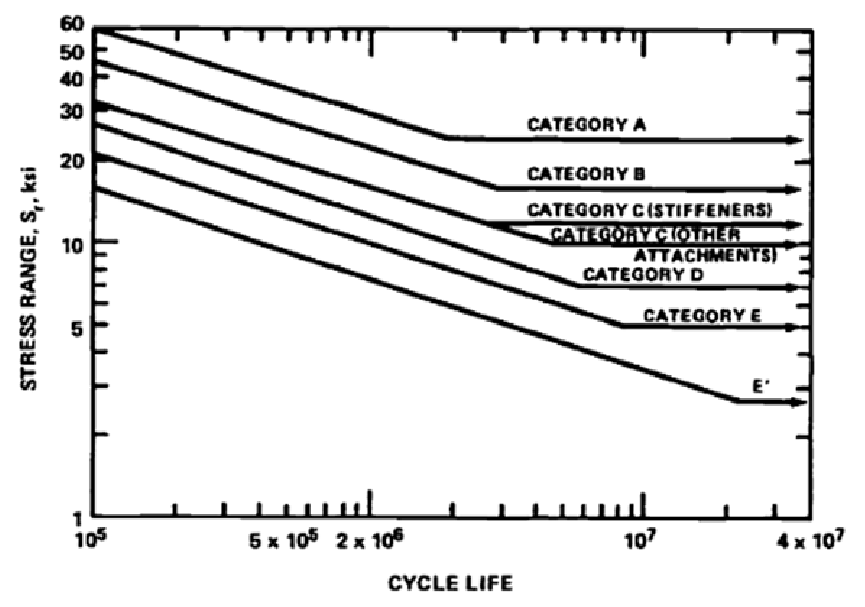

FIG. 10.23 Design stress range curves for categories A to E′.

Various curves depend on weld geometry and plate thickness. E’ is for thick plate

2. AISC Appendix K:

- Load Condition 4: 2x10^6 cycles
- Stress Category C
- F_{th} = 10 ksi (the magnitue of the change in stress due to the application or removal of the unfactored live load).

3. IEC

- F_{th}= 5 ksi

S-N curve based on laboratory testing shield wire arms at three different stress levels to initial crack

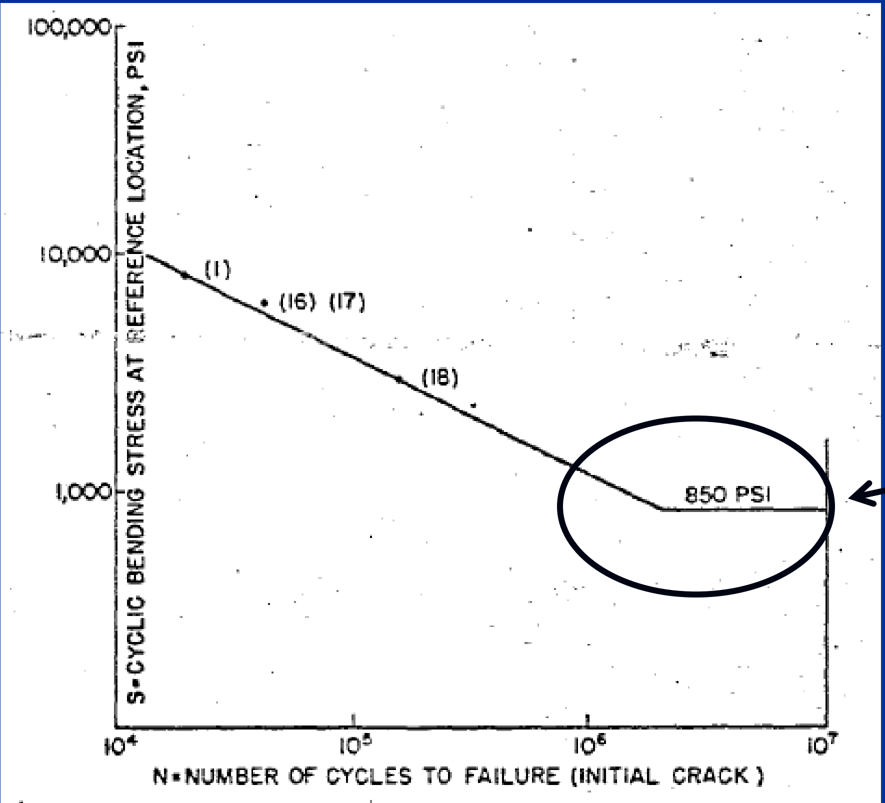

Fig. 5 Minimum Value S-N Diagram for Tests of Three Ground Wire Arms at Three Different Stress Levels.

In 1979, IEEE released a report on the effects of dynamic loading on arms. Three static arms were tested in the laboratory at different stress levels to produce the S-N curve on the left.

If arms are to be vacant for a few years, we would want to be in this area of the graph.

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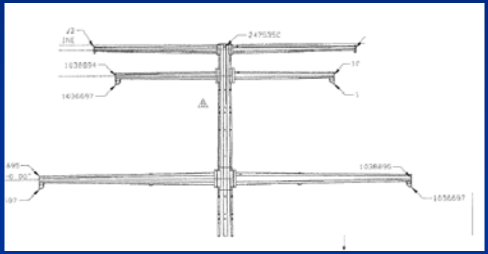
CONCLUSIONS:

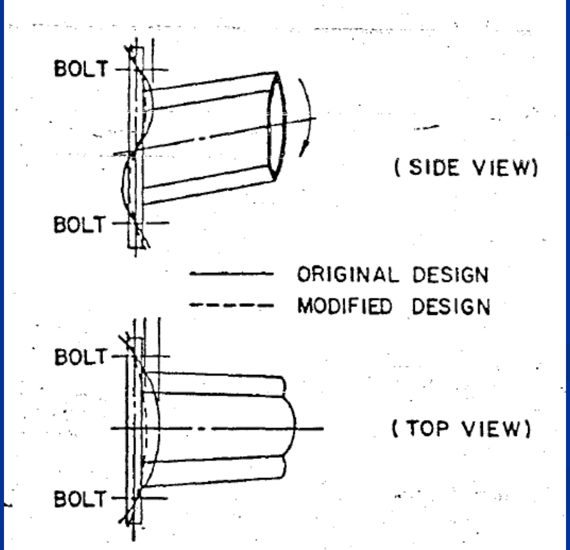
- The F.E. analysis in conjunction with parameters from field measurements shows that tuned-mass damping is effective in reducing stress levels, but many utilities are looking at other options. The weights themselves cost $3/lb. On a large scale project, this can quickly become a substantial cost.
  - Explore the use of mass-particle damping. For instance, sand or a chain inside the arm. Energy is dissipated through the friction associated with particle interaction.
  - Further explore the costs and pros/cons of tying arms together
- These F.E. models are discrete approximations at this point.
  - The models require further refinement with the assistance of additional field testing and preferably low-speed wind tunnel testing. The field testing does not incorporate the aerodynamic damping, βa, which can be negative.



Mass Particle damping may work better than tuned mass damping for a 40’ arm weighing 6,700 lbs.

Factor in Dynamic Loading Criteria

- The IEEE paper found that the best ways to minimize fatigue failure are as follows:
  1. Eliminate the drain hole that acts as a stress concentration factor
  2. Do not allow arms to be galvanized due to residual stresses
  3. Use thicker arm connection plates



The method that suppliers use to design and fabricate arms has not changed in over 40 years and therefore is unlikely to change. When a project involves unloaded arms or arms that may vibrate due to galloping conductors, we, as engineers, would be well advised to consider specifications that include dynamic loading criteria and preventative measures that can be built into the design and fabrication process.

- Collect wind data as close to the project site as possible. Use it to determine if there is a potential issue. Remember that arm vibrations can also be caused by galloping conductors. The magnitude of the driving force is not necessarily large.
- Do not force vendors (via conductor configurations or pole geometry) to design an arm that may have a short service life due to dynamic loading.
- Determine the steady-state wind speed that will induce vortex shedding. Typically, most phase arms are short and heavy with 1st mode frequencies that correlate to rare steady-state wind conditions.

THANK YOU!

QUESTIONS?