

Increasing Machine Service Life of Large Envelope, High Acceleration AFP Machines

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ABSTRACT

Since Automated Fiber Placement (AFP) is used to manufacture twin-aisled commercial aircraft parts, extremely large envelope machines are often required and appropriate. Additionally, for very large parts, the average AFP course length may be on the order of one to two meters, and the part may have numerous contours. With courses of this length, a high acceleration machine is necessary to achieve fast laydown rates because the machine is frequently starting and stopping. Part contour also requires high acceleration machine axes to accurately maintain the AFP tow path at high feedrates.

Large machines with high accelerations result in very large loads on bearings. Large loads and the long, high speed axis travels associated with large envelope machines make achieving a long service life difficult. Designing efficient, lightweight machine structures becomes critical to provide long machine service life. This paper compares two structural configurations of large AFP machines to achieve high acceleration, high stiffness, and long service life. A single-tower and two-tower post style machine structure were compared, demonstrating the performance advantage of the two-tower structure. The two-tower machine mass was reduced 49% below an existing single-tower machine and 29% below an optimized single-tower design while maintaining required stiffness.

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INTRODUCTION

AFP machine performance can be empirically determined by laying tow with a machine of a given acceleration and stiffness on a part with sufficient contour to require high acceleration while laying tow and observing any process-related issues such as torn tows, tow feeding problems and deviation from programmed tow path. A stiffer machine structure is better for servo axis controllability but the cost of an overly stiff machine is increased weight. Increased weight causes higher drivetrain loads and higher bearing loads. Furthermore, large envelope machines often have cantilevered structure which induces large moments that bearing arrangements or drivetrains must react, [Figure 1](#).

Designing for adequate but not excess stiffness results in the most optimal design when considering both performance and machine life. For two machines of equal envelope, footprint, and stiffness, the lighter machine is considered preferable. Additionally, it is desirable that the machine be only stiff in necessary directions. A machine compliant in certain directions can allow more statically determinant

bearing loads while maintaining sufficient stiffness for the AFP process.

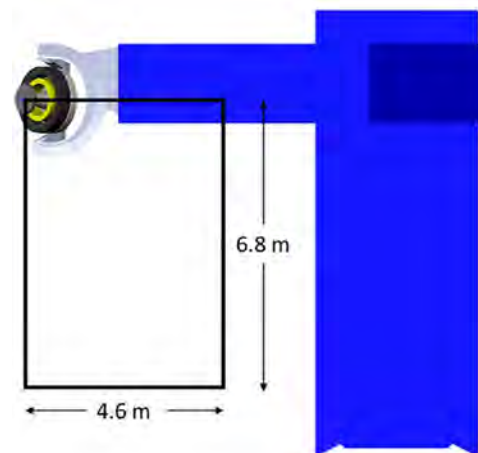


Figure 1. Large machine envelope results in large cantilever. Machine axis travel shown.

For this comparison, machine stiffness and bearing loads were designed to meet determined values and machine mass used to measure relative performance. An implicit assumption is that, if necessary, bearing loads can more easily be reduced to further increase service life on a machine of less mass.

MACHINE PERFORMANCE CONSIDERATIONS

Loads

Given a specified machine envelope and acceleration, the two main factors controlling bearing loads are machine mass and load path. The machine mass can be divided into two loads on the machine:

1. The static weight of the machine.
2. The inertial loads from machine acceleration.

To calculate the bearing life it is important to know how much time the machine spends accelerating. For short course length or highly contoured parts, the fraction of time spent accelerating may be significant. A Mean Equivalent Load is calculated which combines all acceleration and static load cases into one load which is used to calculate bearing life.

The AFP process loads due to compaction are on the order of several hundred pounds and are negligible for large machines.

Bearing Life

Linear roller bearings are typically used to achieve high axis feedrates and high load capacity. The travel life of a cylindrical rolling element bearing is governed by Equation 1, neglecting load factors.

$$L = 50 \left(\frac{C}{P} \right)^{(10/3)} \quad (1)$$

C = dynamic rated load

P = equivalent applied load

50 = km of life at load C

L = Predicted Life in km

Note that a reduction of load by 20% results in a life increase of 2.1 times. Therefore, even a modest reduction in weight can greatly increase bearing life.

For very large machines, the largest bearing cars reasonably available are typically used. Thus longer life cannot readily be achieved by using larger bearings. Additional bearings may also be used, but numerous tightly spaced bearings can result in indeterminate loads on the bearings. With typical manufacturing tolerances of the machine and bearings and a machine foundation which may shift slightly over time, predicting bearing loads becomes increasingly difficult for closely spaced bearings.

Machine Stiffness

For the AFP process to work reliably, the machine must be sufficiently stiff. AFP process loads due to compaction are on the order of several hundred pounds and negligibly contribute to deflection for a large machine. The stiffness requirement then becomes Tool Point deflection under inertial loading of the machine. This makes the stiffness requirement a function of peak acceleration. The higher the acceleration, the higher the required stiffness for acceptable displacements. The machines in this comparison are designed for 0.2G (1.96 m/s/s) acceleration for all linear axes. With 0.2G acceleration, a fundamental natural frequency of approximately 15-16 Hz for a simplified machine model has performed exceptionally well. Assessing machine stiffness as a function of natural frequency, rather than displacement, is preferred. This is because natural frequency is easily measured with an accelerometer for verification. Additionally, natural frequency is more readily calculated with a Finite Element Model (FEM) of the machine structure than all unique positive/negative combinations of machine XYZ accelerations.

An additional stiffness consideration is maintaining flexibility in directions where high stiffness is not necessary. The stiffer the machine the higher the indeterminate loads in multi-bearing car arrangements. This results in reduced bearing life with little added stiffness to the AFP process.

CONFIGURATION COMPARISON

Models Run

Four different FEMs were run comparing the two configurations. Three of a single-tower configuration and one of a two-tower configuration.

1. Production Single Tower (currently in production)
2. Simplified Single Tower (simplified model of item 1)
3. Comparison Single Tower
4. Comparison Two Tower

The Production Single Tower is included to quantify the difference with a simplified model. The Simplified Single Tower establishes the stiffness requirement and vertical bearing load magnitude for a machine model with only primary structure and no additional component mass such as gearboxes, servo motors, counterbalance, electrical cabinets and supporting equipment. The Comparison Single Tower takes advantage of structural design improvements which are also incorporated in the Comparison Two Tower configuration for a direct comparison of performance.

Simplifications

AFP machines are typically constructed with three linear axes (XYZ) and three rotational axes (ABC) to provide orientation, Figure 2.

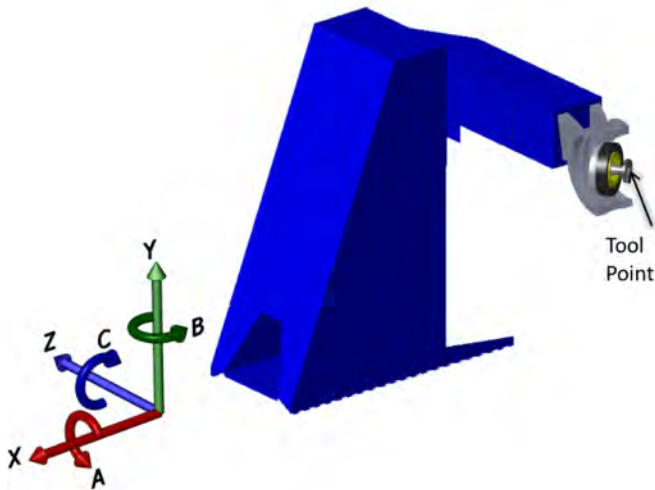


Figure 2. Axis naming convention

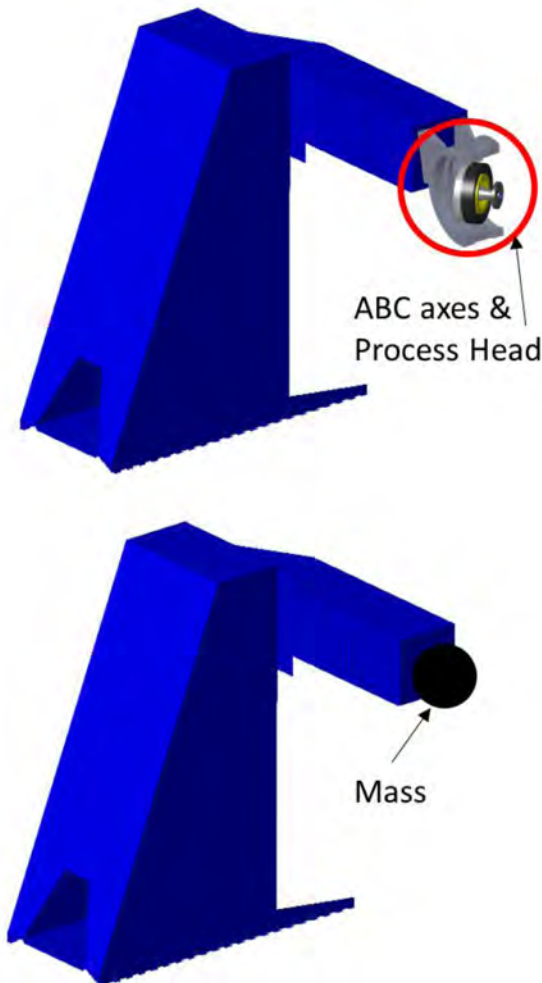


Figure 3. AFP process head and ABC axes are considered a point mass for XYZ linear axis machine structure analysis.

The worst-case deflection of the Tool Point occurs when the Y-axis is at its highest position and the Z-axis is fully extended. All analyses were conducted with the machine in this position.

The ABC axes and AFP process head are an assembly connected to the end of the three linear axis structure, [Figure 3](#). For the machines in this comparison, the ABC axis position's effect on the CG of the ABC axes and process head lumped mass is minimal relative to the size of the machine. The ABC axes and process head are replaced with a lumped mass at the CG location when $A=0$, $B=0$ and $C=0$. The stiffness of the ABC structure is not negligible, but it is excluded from all models.

Structural Similarities and Differences

The two machine configurations compared - a single post style machine and a two-tower style machine - are shown in [Figure 4](#). For the comparison, note the following:

1. The ABC Axes and process head are considered a constant lumped mass
2. Y-axis travel is the same
3. Z-axis travel is the same
4. The Z-structure is the same for both configurations (shown in grey in [Figure 4](#))
5. X footprint is within $\pm 0.3m$
6. Steel plate thickness limited to those commonly available
7. Machine structure limited to that readily fabricated with steel sheet and plate
8. Minimal mass is desired

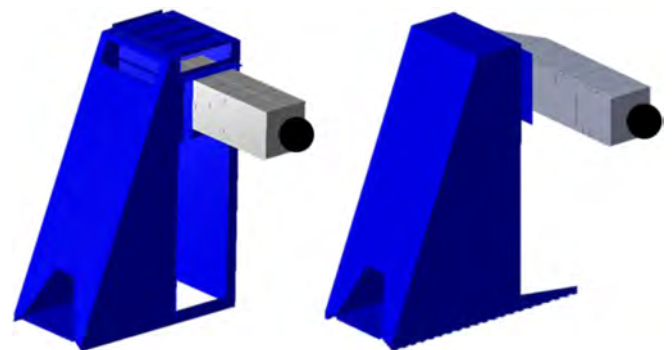


Figure 4. Machine configurations. Left is two-tower, right is single-tower. Z-structure is shown in grey and is the same for both configurations but rotated 90 degrees

X Axis Bearings

The geometry of the X-axis bearing arrangement is similar for both configurations. The distance between the X-rails is identical.

On the two tower machine, the vertical point load applied to the tower from the Y & Z moving mass is cut in half and distributed to each of the two towers, [Figure 5](#). Distributing the loads into the X-axis bearing cars is greatly improved on the two tower machine.

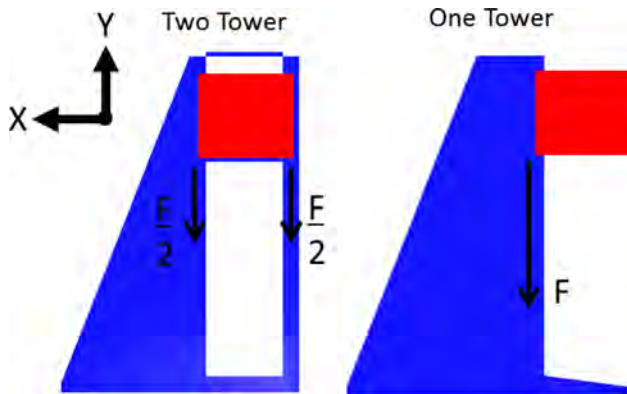


Figure 5. X-axis Loads from Y & Z mass. Y & Z mass shown in Red. Reduced vertical point load in two tower configuration.

Y Axis Bearings

For the Y-axis, the two-tower configuration has three rails, and the single-tower configuration two rails, [Figure 6](#). However, the highlighted rail on the +Z side of the two tower machine takes relatively low loads compared with the front rails. This rail completes a shear connection between the two towers. It is necessary to achieve sufficient stiffness.

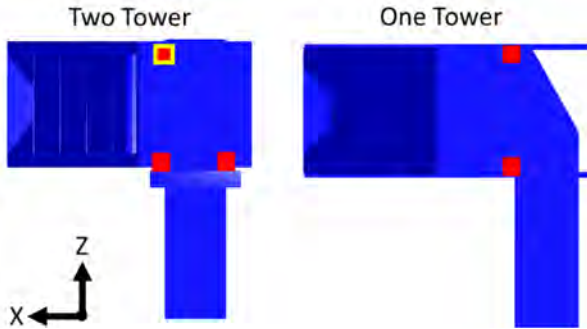


Figure 6. Y-axis rail locations shown in red. Note the two-tower configuration has 3 rails whereas the single-tower has 2 rails. The highlighted rail takes much smaller loads.

The worst case loading on the Y-axis bearing cars occurs when the Z-axis is fully extended. This load results from gravity, Y-axis acceleration, and X-axis acceleration. For these large machines, the Y & Z-moving mass CG will be

outside the Y rails when the Z-axis is extended, [Figure 7](#). The distance in the Z-direction to the midpoint of the two primary Y-axis rails is the moment arm. The two tower machine configuration has a significantly reduced moment arm, [Figure 7](#).

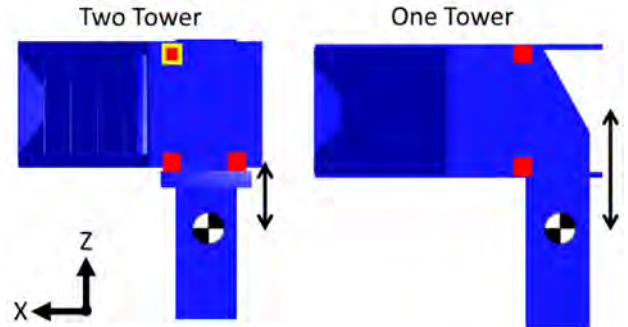


Figure 7. Y-axis Bearing Loads. Note the shorter moment arm for the two tower configuration.

Z Axis Bearings

For the Z-axis, the two tower configuration also has three rails, and the single-tower configuration two rails, [Figure 8](#). The top rail on the two tower configuration creates a shear connection between the two sides of the Y-axis and takes relatively low loads.

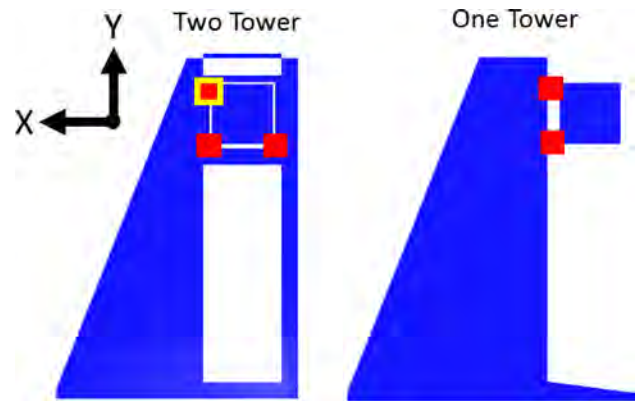


Figure 8. Z-axis rail locations shown in red. Note the two-tower configuration has three rails whereas the single-tower has two rails. The highlighted rail takes much smaller loads.

Similarly, for the Z-axis bearing cars the moment applied is proportional to the distance from the Z-moving mass CG to the centroid of the Z-axis bearing cars. For the Z-axis there is no difference in the moment load between the single-tower and two tower machine, [Figure 9](#).

Machine CG Location

The ABC and process head mass were considered a constant mass. Since this was a fixed mass, a lighter XYZ machine structure in general has a CG located closer to this

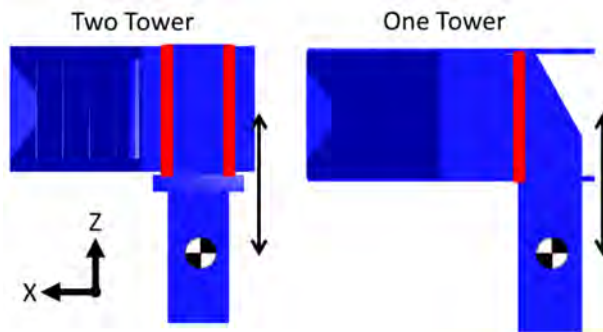


Figure 9. Z-axis bearing car span shown in red. Moment arm is no different in this case.

fixed mass. Given this, it is desirable to architect a machine such that a CG which is closer to the Tool Point can be accommodated while minimizing moments, Figure 10.

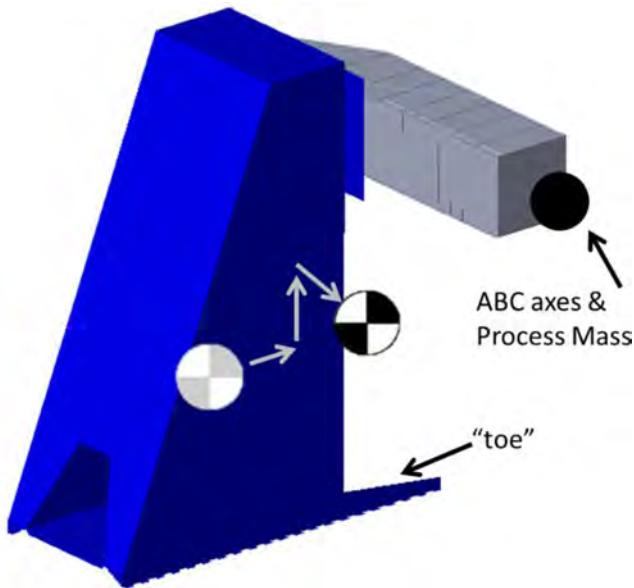


Figure 10. CG location as a function of XYZ machine structure mass. ABC and process head mass is fixed, generally a machine CG closer to this fixed point mass will be lighter.

For the X-direction CG location, it can be beneficial for a single-tower machine to translate the CG away from the process head mass to be more centered on the tower structure of the machine to reduce the loading on the “toe” of the machine. Unfortunately, for a constant ABC and process head mass and similar Z-ram, the only way to move the CG this direction is to add mass on the opposite side of the machine. Since a lighter machine is preferred, it is desirable to configure the machine such that adding mass to center the CG is unnecessary. For the two-tower configuration, the CG was naturally centered on the main X-axis bearing cars, removing the possible need to add mass.

Torsional & Bending Stiffness

Two of the substantial loads applied to the tower structure from the Y & Z mass are torsion about the Y-direction from the cantilevered Z-moving mass CG and bending about the Z-direction from the height of the Y & Z-moving mass CG. For the torsional load, consider a two tower machine with the same cross sectional area as a single-tower but part of it cut and moved to the other side of the Z-structure, Figure 11. The two towers were connected with a shear plate at the top of the machine, so when estimating the torsional stiffness it was appropriate to consider the two towers together. This resulted in a much larger torsional stiffness for the two tower machine. For bending about the Z-direction, the stiffness of the two-tower machine was difficult to assess qualitatively. This was determined with the FEM as the shear connection between the two towers is limited.

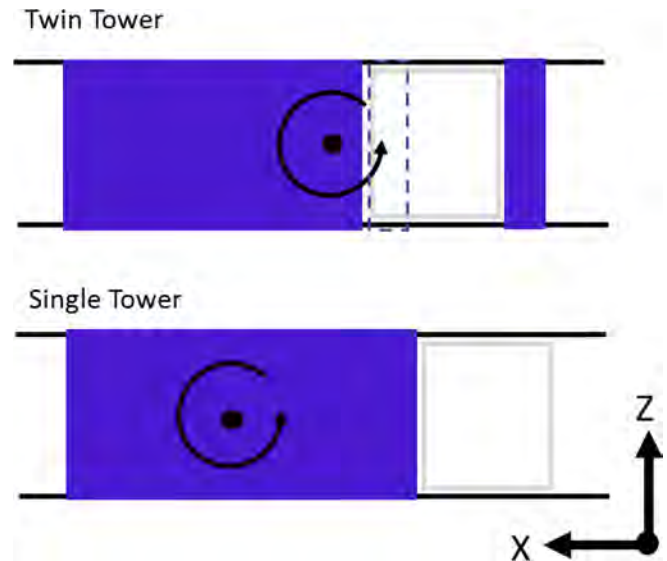


Figure 11. Tower Torsion. The two-tower machine has a torsional shear connection between the towers resulting in much great torsional stiffness. There was limited Z-bending shear connection between the two towers; the single-tower design has greater Z-bending stiffness

PERFORMANCE COMPARISON

The structural limitations applied to the single-tower and two-tower machine configurations are repeated here:

1. The ABC Axes and process head are considered a constant lumped mass
2. Y-axis travel is the same
3. Z-axis travel is the same
4. The Z-structure is the same for both configurations (shown in grey in Figure 4)
5. X footprint is within +/- 0.3m
6. Steel plate thickness limited to those commonly available
7. Machine structure limited to that readily fabricated with steel sheet and plate
8. Minimal mass is desired

The criteria to be met by each model are stiffness as measured by natural frequency and X-axis bearing loads. The baseline values will be established by the Simplified Single Tower machine model. The Production Single Tower is included to quantify the difference with a simplified model.

Production Single Tower

This is an existing Electroimpact machine in production [1]. The Production Single Tower model includes additional masses from large non-structural components. Additionally, bearing car stiffness was included for the Y and Z axes in the modal analysis. Modal results are shown in Figure 12. The machine mass was approximately 150,000 kg.

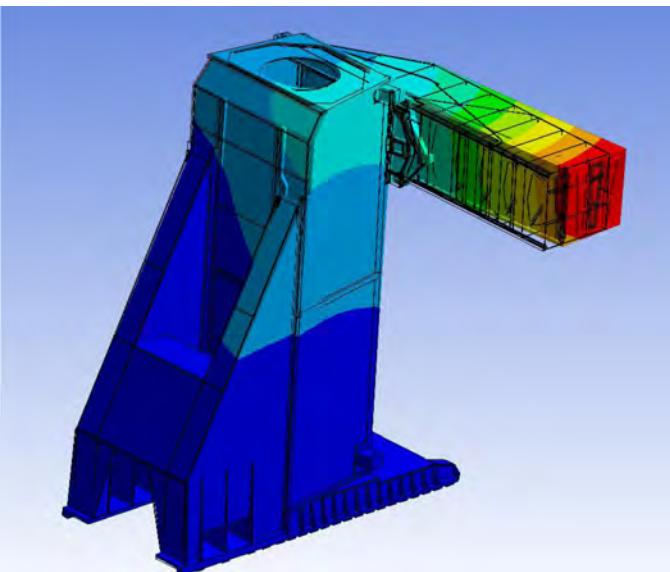


Figure 12. Production Single Tower. 1st Mode, 13.5 Hz.

The notation used for the X-axis bearing loads is shown in Figure 13. For the X-axis bearing cars loads, the model

included X-axis bearing car stiffness, the X-axis bed, and the concrete supporting the bed. X-axis bearing car loads are shown in Figure 14. In this figure, note that the contribution of Y-axis and Z-axis acceleration loads are less than the Gravity and X-axis loads on the X-axis bearing cars. This can be observed as the Y-axis and Z-axis acceleration loads closely following the plot of Gravity with +X acceleration. The Y & Z-axis acceleration combinations will not be shown for the other models run. Loads for only Gravity, Gravity with +X acceleration, and Gravity with -X acceleration are shown in Figure 15.

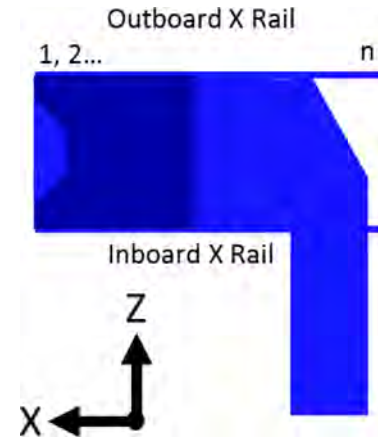


Figure 13. X-Axis Bearing Notation. Same on all Models.

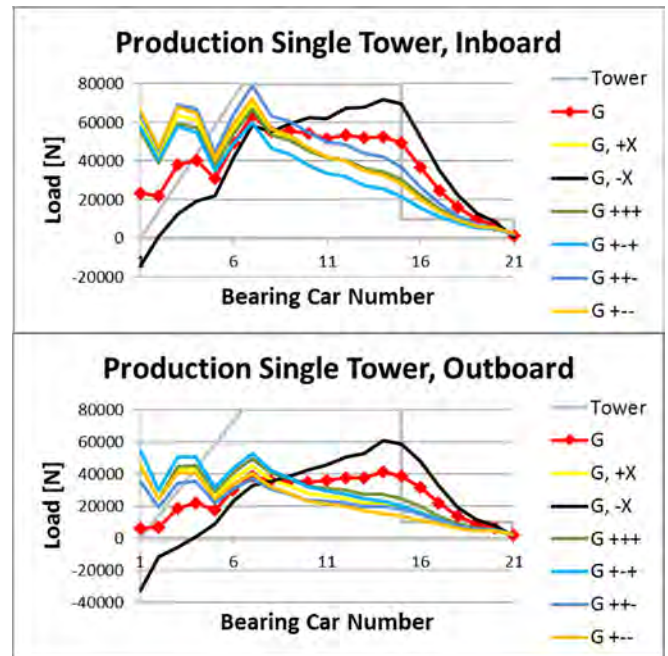


Figure 14. Production Single Tower. X-axis vertical bearing car loads. Gravity Loading, gravity with X-axis acceleration and XYZ combinations of acceleration. Outline of tower shown in grey for reference.

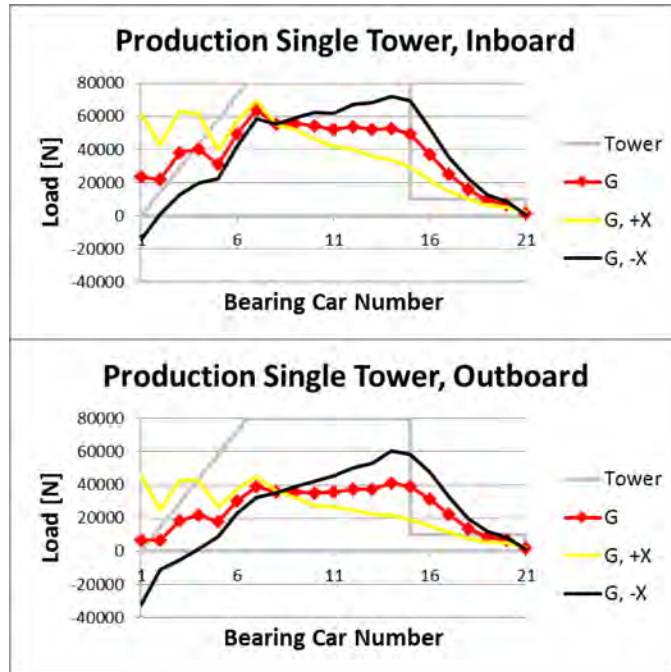


Figure 15. Production Single Tower. X-axis vertical bearing car loads. Gravity Loading, Gravity loading with +/- X-axis acceleration. Outline of tower shown in grey for reference.

Simplified Single Tower

The Simplified Single Tower is a simplified version of the Production Single Tower and sets the target values for performance for the two comparison models. Higher modal results are expected as the mass has been reduced but no primary structure has been removed, Figure 16. The machine mass is approximately 102,000 kg.

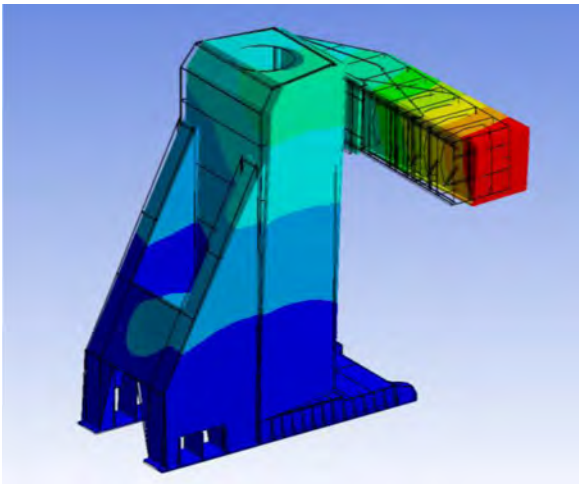


Figure 16. Simplified Single Tower. 1st Mode, 15.8 Hz.

X-axis bearing car loads were lower than the Production Single Tower Model as the mass of the machine is lower, Figure 17.

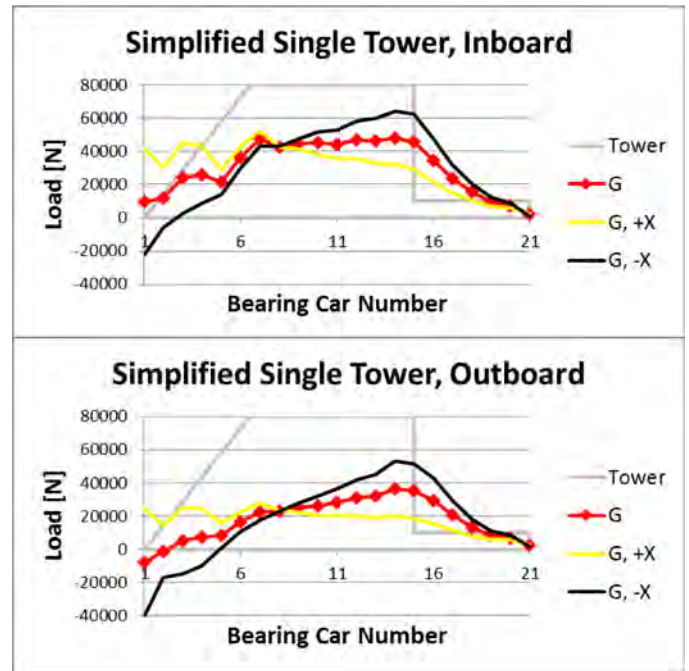


Figure 17. Simplified Single Tower. X-axis vertical bearing car loads. Gravity Loading, Gravity loading with +/- X-axis acceleration. Outline of tower shown in grey for reference.

Comparison Single Tower

For the Comparison Single Tower, the tower has been changed to a larger torsion box. The larger box results in

larger torsional rigidity. On the Production Single Tower machine, the X-drive housing is not included in the torsion box. Modal analysis results are shown in [Figure 18](#). The machine mass is 73,500 kg.

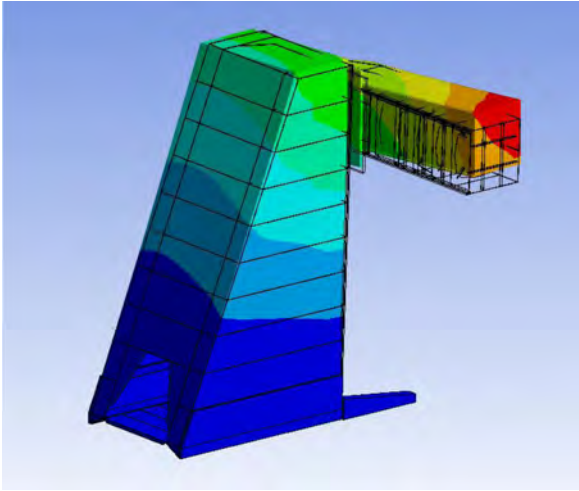


Figure 18. Comparison Single Tower. 1st Mode, 15.8 Hz.

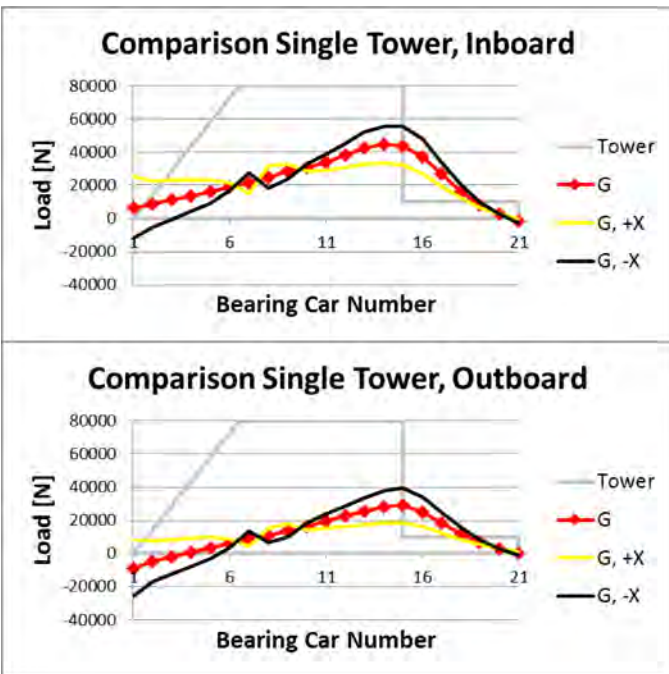


Figure 19. Comparison Single Tower. X-axis vertical bearing car loads. Gravity Loading, Gravity loading with +/- X-axis acceleration. Outline of tower shown in grey for reference.

X-axis maximum bearing car loads were targeted to be similar to the loads established by the Simplified Single Tower model, [Figure 19](#). A considerable amount of structure was added to the bottom portion of the machine to adequately distribute the load near the “toe” of the machine until the

magnitude of the bearing car loads were similar to that of the Simplified Single Tower model.

Comparison Two Tower

As the two tower structure was novel in this comparison, several structural designs were analyzed to determine the optimal design. [Figure 20](#) shows the general progression of the designs. Symmetry often has many desirable attributes so the starting point was two symmetric triangular towers. The first structure used triangles because they are a more minimalistic shape than rectangles, yet have stiffness in X, Y, & Z-directions. In the symmetric triangle case, the stiffness in bending about the Z-direction was not adequate as there is a limited shear connection between the two towers, [Figure 21](#). For a limited shear connection it is more favorable for most of the structure to be on one side of the towers, with the secondary side having high stiffness in only the Y & Z-directions. The second design was a larger triangle and triangular blade, and this resulted in greater stiffness bending about the Z-direction than the first design. The blade tapered to a small section in the Z-direction at the top, [Figure 22](#). By analyzing a series of designs ranging from triangular to rectangular for the main tower and secondary tower, a rectangular section was determined to be stiffer, leading to the third design. The principle gleaned by progressing from triangle to rectangle is that for lightweight stiffness critical designs, maintaining maximum cross sectional area is generally preferable.

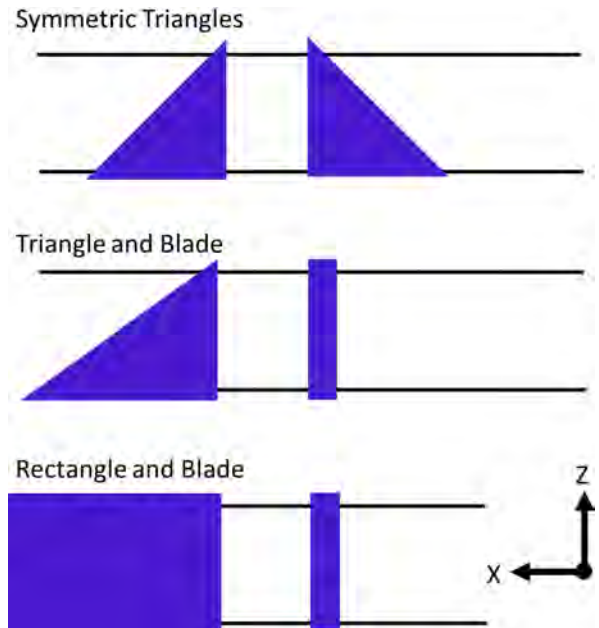


Figure 20. Comparison Two Tower. Main structural designs analyzed.

[Figure 21](#) also illustrates that bending about the X-axis is not dependent on a shear connection between the two towers.

Bending in this direction was similar for two towers with a total cross sectional area similar to a single-tower.

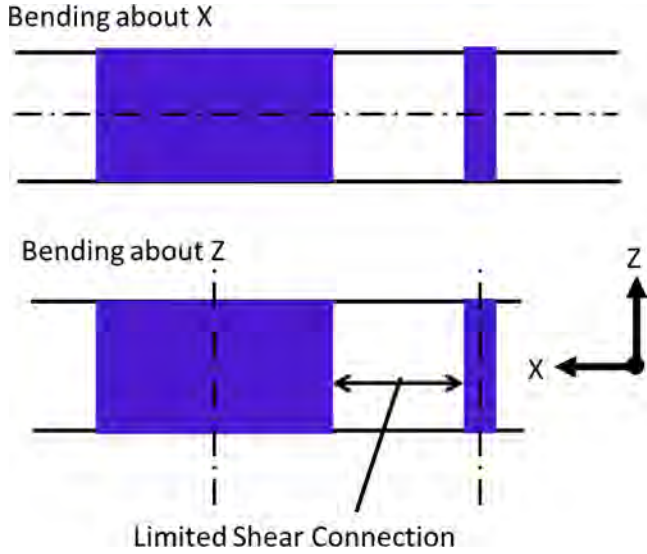


Figure 21. Comparison Two Tower. Bending stiffness differences with single tower about X and Z axes.

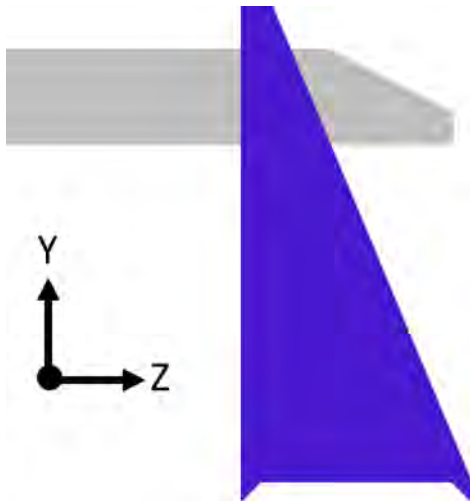


Figure 22. Taper of blade.

The resulting Comparison Two Tower modal analysis results are shown in Figure 23. The machine mass is 52,250 kg.

The X-axis bearing car loads are similar to the Simplified Single Tower Model; however, there are fewer bearing cars. The number of cars is not important, as the bearing cars are spaced closely together in the highly loaded regions, Figure 24.

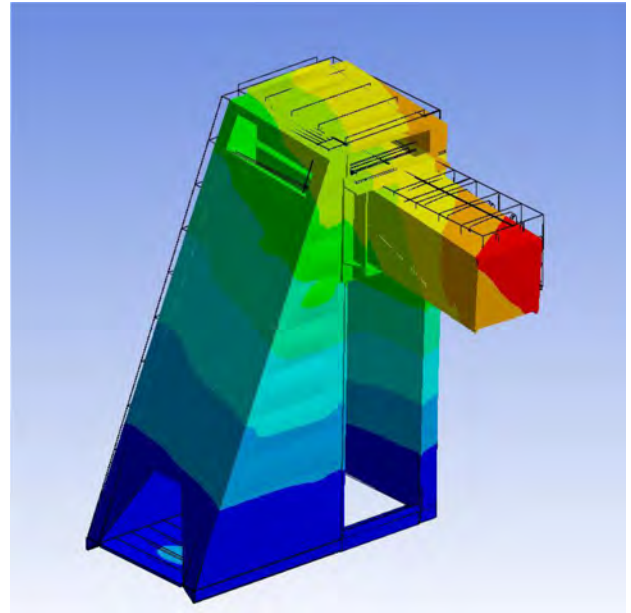


Figure 23. Comparison Two Tower. 1st Mode, 15.6 Hz.

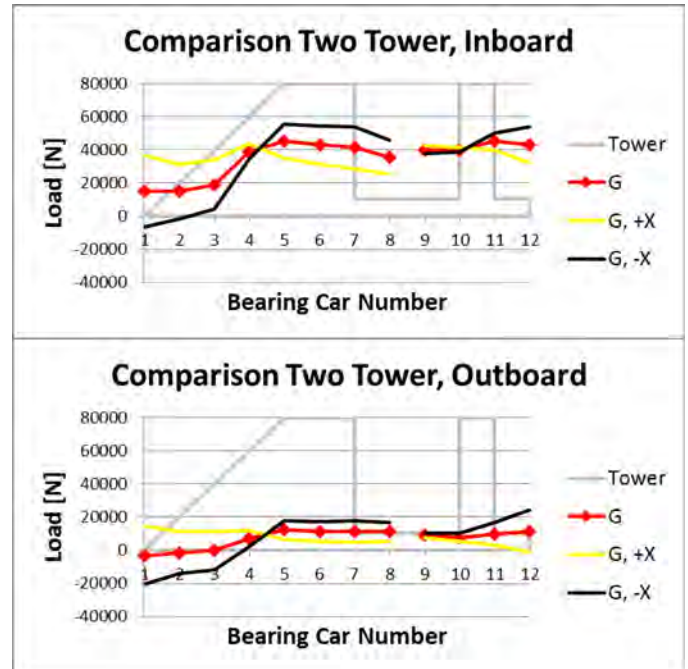


Figure 24. Comparison Two Tower. X-axis vertical bearing car loads. Gravity Loading, Gravity loading with +/- X-axis acceleration. Outline of tower shown in grey for reference.

RESULTS

Production/Simplified Single Tower

The results from the Production Single Tower and the Simplified Single Tower are shown in Table 1. The mass of

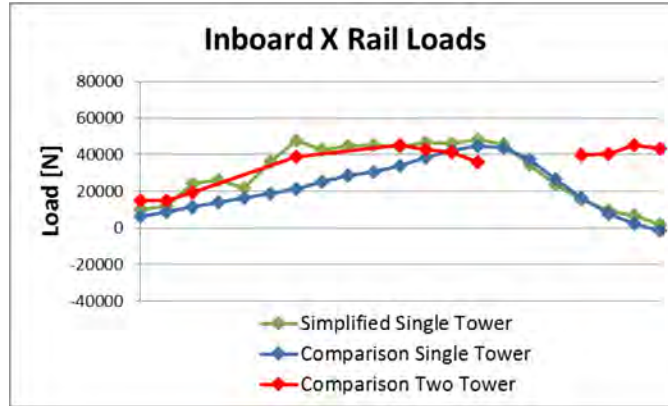


Figure 25. X-axis Loads. The baseline is the Simplified Single Tower. Max load of Comparison Single Tower and Comparison Two Tower were limited to max load of Simplified Single Tower.

the Simplified Single Tower model is reduced by a factor of 1.47. This same factor is applied to other simplified machine models to estimate the mass of the finished machine including large non-structural components. Bearing Loads for the Production Single Tower to Simplified Single Tower model are not compared as the difference is not important. The subsequent comparison model loads will be compared.

Table 1. Production Single Tower and Simplified Single Tower Mass and Stiffness Results.

	FEM Mass		1st Mode
	[kg]	[lbm]	[Hz]
Production Single Tower	150,455	331,000	13.5
Simplified Single Tower	102,273	225,000	15.8

Stiffness

The Simplified Single Tower model set the target for stiffness of the Comparison Single Tower and Comparison Two Tower models. The Comparison Single Tower and Comparison Two Tower stiffness results are summarized in Table 2.

Table 2. Simplified Single Tower, Comparison Single Tower, and Comparison Two Tower Stiffness Results.

	1st Mode	Error
	[Hz]	[-]
Simplified Single Tower	15.8	-
Comparison Single Tower	15.8	- 0%
Comparison Two Tower	15.6	- 1%

Bearing Loads

The maximum X-axis bearing car vertical loads were designed to be similar to the Simplified Single Tower model, Figure 25. This requirement ensures each model has sufficient structure to distribute large point loads into an adequate number of bearing cars. This can require substantial structure for large loads and must be included to get an accurate machine mass for comparison. Only inboard loads are shown because this was the location of absolute highest load.

Mass

The relative performance of each model is measured by mass, Table 3. The 1.47 factor was applied to ratio the FEM mass closer to that of an as-built machine.

Table 3. FEM Mass Results. Mass*1.47 is estimated mass for more detailed model.

	FEM Mass		FEM Mass * 1.47	
	[kg]	[lbm]	[kg]	[lbm]
Simplified Single Tower	102,273	225,000	150,341	330,750
Comparison Single Tower	73,636	162,000	108,245	238,140
Comparison Two Tower	52,273	115,000	76,841	169,050

Both the Comparison Single Tower and Comparison Two Tower models are significantly lighter than the Simplified Single Tower model, Table 4.

The Comparison Two Tower model is 29% lighter than the Comparison Single Tower, Table 5. In this comparison, the bearing loads were maintained at the same load as the Simplified Single Tower model. However, if loads must be reduced for longer bearing life, this will likely be more achievable by starting with a significantly lighter machine.

Table 4. Percent mass reduction from Simplified Single Tower Model

	FEM Mass * 1.47		Mass Reduction
	[kg]	[lbm]	[-]
Simplified Single Tower	150,341	330,750	-
Comparison Single Tower	108,245	238,140	- 28%
Comparison Two Tower	76,841	169,050	- 49%

Table 5. Percent mass difference of comparison models

	FEM Mass * 1.47		Mass Reduction
	[kg]	[lbm]	[-]
Comparison Single Tower	108,245	238,140	-
Comparison Two Tower	76,841	169,050	- 29%

The performance advantage of the Simplified Two Tower model has been demonstrated. Electroimpact moved forward with the design concept and developed a two tower production machine. The final two tower production machine design was not limited by all of the constraints in this comparison. The significant changes were the Z-structure and ABC-structure were lightened. With these changes, the overall mass of the simplified two tower production model remained nearly the same as the Comparison Two Tower model.

The two-tower production machine is built and running and will begin production soon. The Simplified Single Tower model is based on a machine currently in production. The mass of the simplified FEM multiplied by 1.47 and the actual machine mass are shown in Table 6, verifying the mass reduction of the Two Tower machine.

Table 6. Estimated and as built machine mass

	FEM Mass * 1.47		As Built Machine	
	[kg]	[lbm]	[kg]	[lbm]
Simplified Single Tower	150,341	330,750	160,000	352,000
Comparison Two Tower	76,841	169,050	79,500	174,900

SUMMARY/CONCLUSIONS

Large machines used in the production of twin-aisle aircraft require very large working envelopes and require long axis travel service life. Therefore efficient, lightweight machine structures must be designed to maximize bearing

service life. The comparison of a single-tower and two-tower configuration demonstrates the advantage of the two-tower design by reducing the mass 29% below that of the single-tower design.

REFERENCES

1. Rudberg, T., Flynn, R., and Nielson, J., "Production Implementation of Multiple Machine, High Speed Fiber Placement for Large Structures," *SAE Int. J. Aerosp.* 3(1):216-223, 2010, doi:10.4271/2010-01-1877.

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DEFINITIONS/ABBREVIATIONS

- AFP** - Automated Fiber Placement
- course** - A single pass of 1-16 tows
- FEM** - Finite Element Model
- mean equivalent load** - Single load calculated from all dynamic loads applied to bearing
- Tool Point** - Point of machine where position with respect to part is
- tow** - Resin impregnated carbon fiber slit to widths of 1/8", 1/4", & 1/2"