MODAL SPACE - IN OUR OWN LITTLE WORLD

by Pete Avitabile

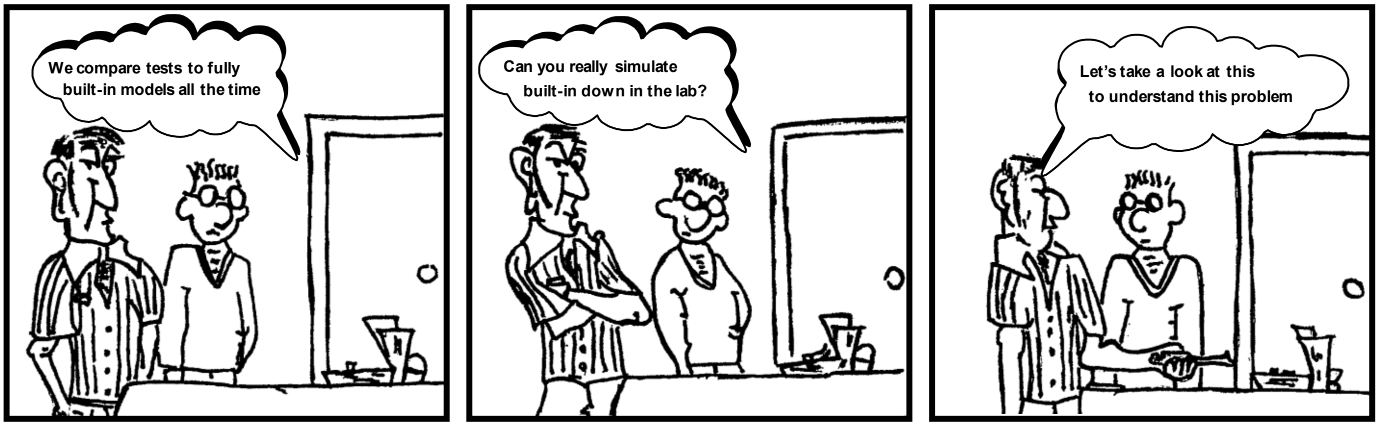

Illustration by Mike Avitabile

We compare tests to fully built-in models all the time. Can you really simulate built-in down in the lab? Let's take a look at this to understand this problem.

So this has been a constant item for discussion for as long as I can remember – and while I might forget certain things as I get older, this topic is one I remember very well. It seems to constantly pop up as an item for discussion all the time. Of course, this happens because the analytical world is quite different than the experimental world.

In analytical modeling, we can always very simply identify a boundary condition to apply to our model. We can make it totally free-free if we want. But of course the real world can be much different than the analytical world (which is filled with "assumptions" that may not be possible in the real world).

But at least with a free-free boundary condition we can often do a fairly good job of approximating that condition. In fact, we often like to test this way because then there is very little interaction with the test fixturing and related set ups for conditions that are other than free-free. (Remember in the last article, how a very seemingly insignificant change to the test set up ended up having a very significant change to the test results.)

The problem is that many times we would like to validate our analytical models with a constrained boundary condition at the attachment points to our component. From an analytical standpoint, this is very desireable. But from a practical testing standpoint, this can be very messy from a variety of different perspectives. There are all kinds of issues related to mounting surface flatness, bolting preloads, etc. that are of concern.

But one item that is always one that can be misunderstood is, eaxtly how stiff is stiff and how massive is massive in regards to creating that so called built-in condition. Many people will try to design "an infinitely stiff" support frame or test fixture. But we all know that any structure will have resonant frequencies, it is just a matter of where they occur and what effect they may have on the ability to actually create a built-in condition. The test fixture may be adequate for the first few lower order flexible modes but eventually there are fixture resonances that may interact with the test article.

Another way to simulate a built-in condition is to provide a large seismic mass. This generally tends to be a better mechanism to achieve a built-in boundary condition but often times people don't realize exactly how much mass is actually required to achieve this condition. Often times you will hear people state that the seismic mass needs to be 10 times larger than the mass of the test object. For some reason, people think that the 10:1 ratio is the answer for all problems. But what we forget is that these "rules of thumb" evolved back in the early days when all we had was a slide rule for calculations. And with that level of accuracy maybe that 10:1 rule was a good guesstimate. But now with all the sophisticated models we can evaluate today, we really should rethink that 10:1 rule.

In order to illustrate this, a simple example for a long beam like structure will be used to explain this. We have recently performed some free-free testing for flapwise modes for a 9m turbine blade and this is an excellent article to discuss in regards to a mass loaded interface to simulate a built-in condition.

The 9m blade was tested free-free and a very coarse beam finite element model was utilized to model the turbine blade for flapwise motion alone for the first few modes. The intent was to use that free-free model and then apply a "perfect analytical built-in boundary condition" and compare it to a variety of different modes to study the effect of the amount of mass needed to actually anchor the blade to ground.

The free-free test and analytical modes are shown in Table 1 for reference. The analytical model and test performed are not described here for brevity.

Table 1 - Comparison of Model and Test Results Free-Free for a 9 m Wind Turbine Blade

Mode	Model (Hz)	Test (Hz)	MAC
1	7.84	7.76	99.85
2	18.5	21.26	98.28
3	34.52	31.34	98.85

These results are considered reasonable considering the coarseness of the rough beam finite element model.

Now the analytical model can be used to identify the "perfect" built-in boundary condition and will be used as a reference. Another analytical model will also be used to compare the effects of adding a very large "seismic mass" to the root of the blade model. This model, with different mass conditions will be compared to the "perfect" reference model. The frequencies and shapes will be compared to show the effects of the amount of mass that is needed to achieve this constrained condition. And it is very important to note that the mass is not just a lumped mass; the mass has rotational effects and it is these rotational mass effects that are the most important ones for the development of a built-in simulation for a long overhung structure such as the turbine blade. For reference the turbine blade weighs on the order of 400 lbs. One approximation of a seismic mass used a 66" x 72" and 24" thick steel plate that weighed approximately 22,000 lbs or roughly 55 times heavier than the turbine blade.

Several models were developed with various ratios of the lumped mass, designated as M, and the rotary mass, designated as MR and were compared to the "perfect" reference model, for both frequency and shape of the resulting model. Table 2 summarizes the results considering just the first mode of the turbine blade and Figure 1 shows the shape comparison for the different cases shown in Table 2.

First notice that the model with just M and MR as the seismic mass approximation does not replicate the frequency very well; realize that the anchor is over 50 times the weight of the blade. Also notice that the curvature of the mode shape does not match the "perfect" reference model well. Now doubling the lumped mass, 2*M, with the same rotary mass, MR, shows some improvement but still has differences. And if the inertia is doubled and lumped mass kept the same, the results are approximately the same. The curvatures are improved but there are still differences. Notice that the rotary mass of the seismic anchor is the most critical item to cause the frequencies and shapes to match much better. If the rotary inertia is increased by an order of magnitude, then the frequency better correlates and the curvature of the shape also starts to match very well.

So the bottom line here is that the seismic mass needs to be very large in order to approximate a built-in condition and that the rotary mass effect is much more important due to the large overhung effects of the wind turbine blade. Obviously, a different structure with a lower center of gravity will have different results but this model certainly shows the importance of identifying the proper characteristics necessary for the seismic anchor for overhung structures.

Table 2 - Comparison of Different Seismic Anchor Approximations and "Perfect" Built-in Condition for a 9 m Wind Turbine Blade

Properties of Anchor		Freq. of Mode 1 (Hz)	
Mass	Inertia	With Anchor	Truly built in
M	MR	5.29	4.36
2*M	MR	4.87	4.36
M	2*MR	4.87	4.36
M	10*MR	4.47	4.36

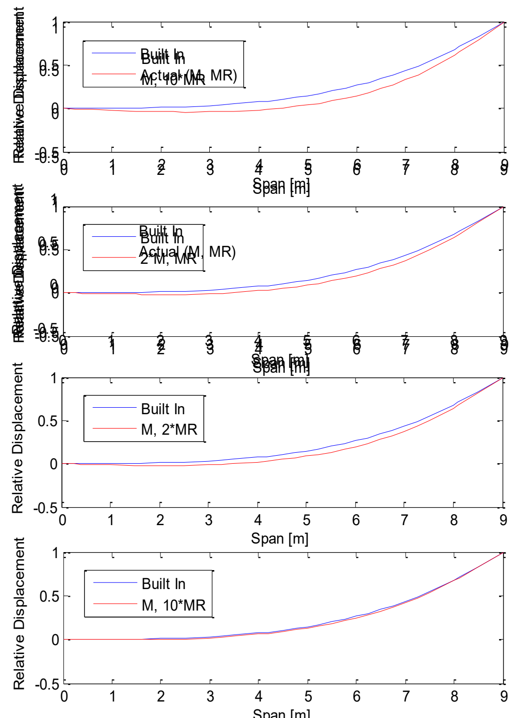

Figure 1 – Comparison of Mode Shape Curvature

But at the end of the day you have to decide how much variation from the real built-in condition is acceptable and how much deviation in the actual mode shape is acceptable for the intent of the test that is being performed. The problem is that often times people haven't considered these issues in any depth and therefore do not have clear statements as to what deviation may be acceptable. If you have any other questions about modal analysis, just ask me.