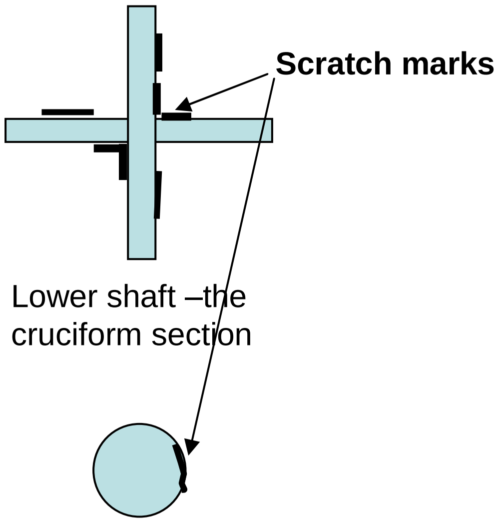
Interim report on the Target Shaft mechanical analysis and assessment -- FEA

By Stephanie Yang

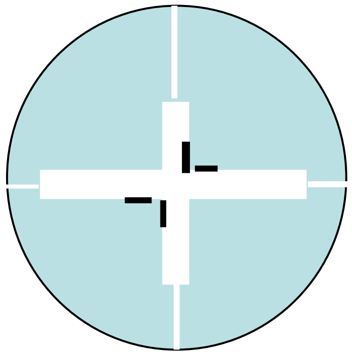
Having seen the scratch marks in the bearings and the shaft, there is evidence that the shaft was NOT travelling in a straight and vertical line. The scratch marks suggest that there is a certain degree of twisting and bending motion in the shaft. The exact nature of its source is still being investigated through field mapping and imaging using high speed camera.

There also exists a gap between the shaft and the two bearings, and an annulus gap of about 1mm between the magnet and the coil. These gaps allow the shaft to “rock”, “sway” and rotate about in between the two bearings during shuttling. These motion is further supported by the extend and locations of the scratch marks on both the shaft and the bearing as shown below.



Lower shaft –the cruciform section

The upper shaft section


Lowing bearing

Based on this observation, we believe there exists 2 primary motions of the shaft:-

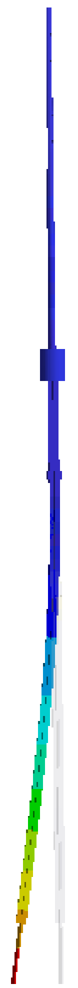
1. Twisting motion – the source of this torque is not known. Field mapping may throw some lights on this.
2. Off-set of the magnet – there is no obvious side load on the magnet drawing it off-set from its central axis. However, the combination of the inertia of the magnet, the resonance effect of the shaft and the gaps between the bearing, the shaft and the bearing could contribute to this mode;

A high speed camera may provide further understand of the exact motion path of the shaft and allow us to determine the interactive force between the shaft and the bearing.

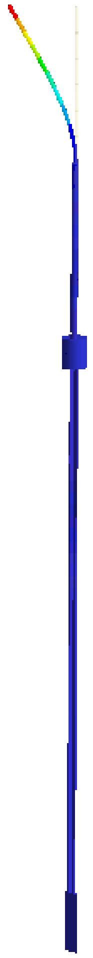
For this part of the exercise, we will concentrate on (2) of the above which requires the evaluation of the modal behaviour of the shaft unit.

The follow plots shows the first 5 modes of the shafts under 3 different support conditions:

Case A: Support condition:- fixed at the upper and lower bearings



1st and 2nd mode: 44Hz



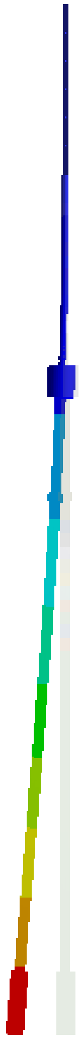
3rd mode: 145Hz

Comparing to a driving frequency of 33 Hz, this frequency is too high to be of any real impact to the bearing

Case B: Boundary condition:- fixed at the upper bearing only

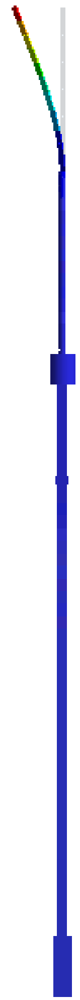
(credible if the gaps at the upper bearing is small enough to “jam” the shaft)

We do not see this scenario being credible



1st and 2nd mode: 15Hz

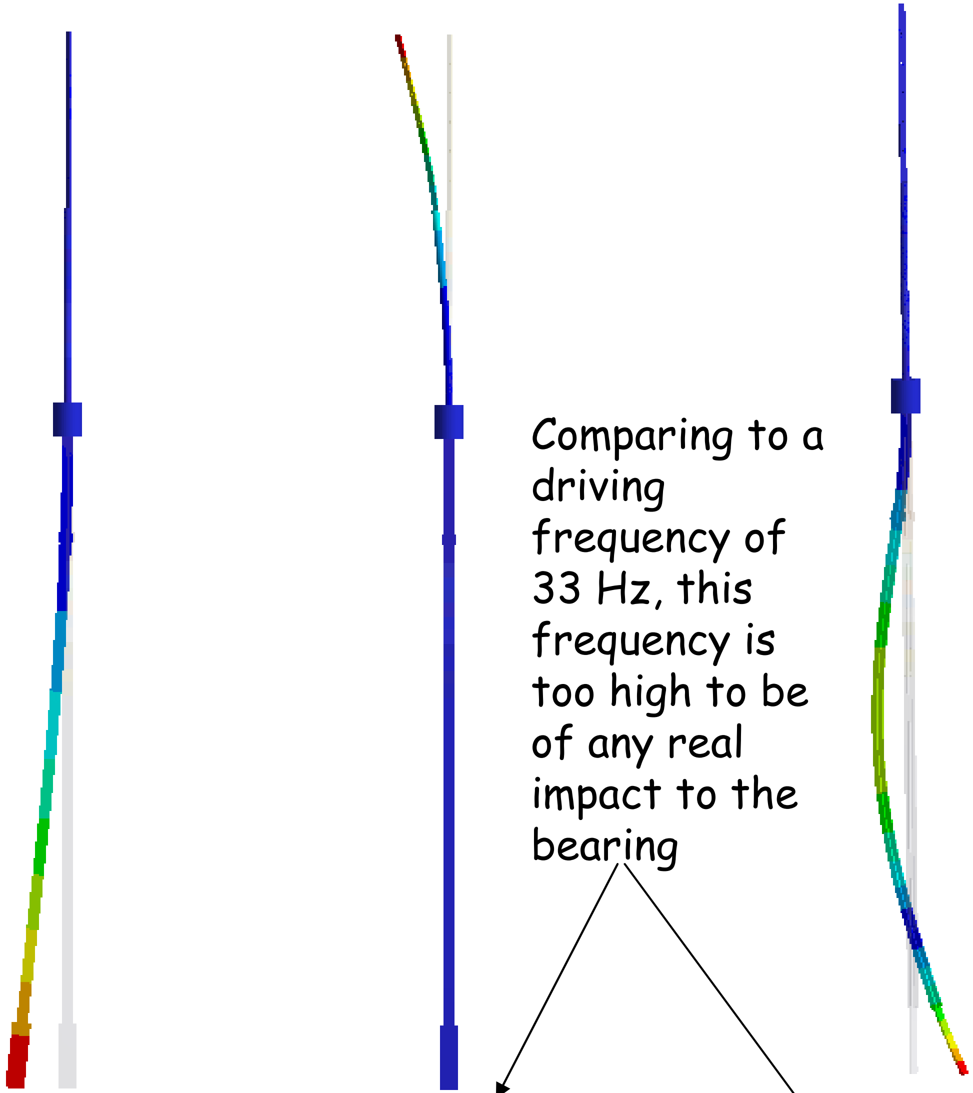
3rd and 4th mode: 79Hz



5th mode: 144Hz

Case C: fixing support on the magnet

(credible if the gaps at both the upper and lower bearings are large enough)



1st and 2nd mode: 27.6Hz

3rd and 4th mode: 80Hz

5th mode: 170Hz

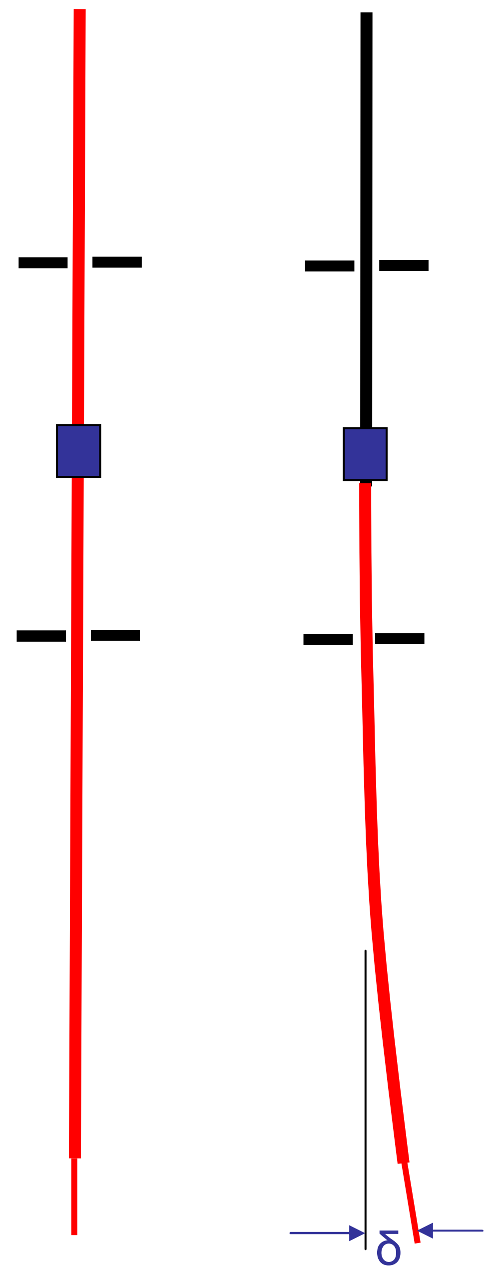
Of the 3 modal analyses studied, the first 2 modes in Case C are the mostly likely modes of vibration experienced by the shaft.

The boundary conditions there assumed that the magnet is “held” while shuttling up and down. In reality, the magnet is “held” by the magnetic fields that drives it up and down. Provided there is clearance between the bearing and the shaft, the lower shaft would generate a natural frequency of around 27 Hz.

This is close enough to the driving frequency of the shaft which is about 33 Hz and may introduce resonance to it.

The following analyses examine the motion path of the shaft caused by this mode of vibration

> *Under the normal circumstances when the shaft is shuttling in a perfectly vertical line, there should be no side (horizontal) force acting on the tip of the shaft. However, the resonance of the lower shaft may well force the magnet to displace sideway at some stage of the shaft movement In the absence of any measured date on the shaft deflection, we assumes it takes up the full 1mm annulus gap (most onerous case).*



1. Due to the magnet being pulled sideway

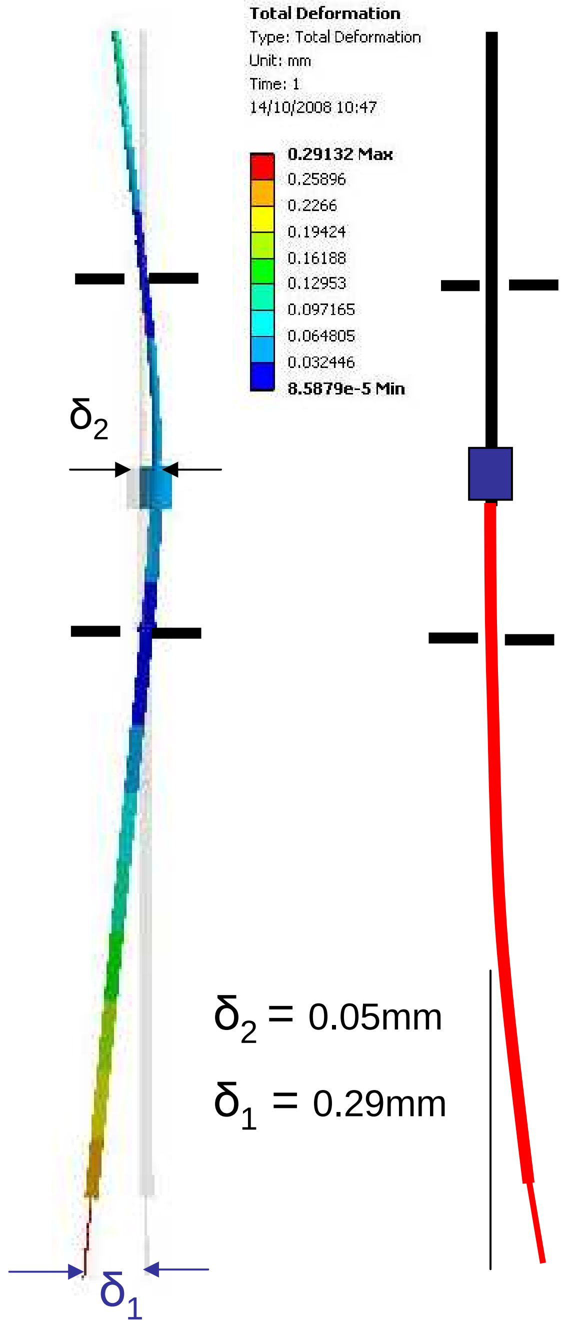
From the FEA results, when a side load of 1N is applied to the magnet, if there is sufficient gaps between the shaft and the bearings, it will create a boundary condition similar to the shaft being simply supported at the two bearing points. Such force will cause the shaft to bend towards one side by 0.05mm, the end of the shaft would bend to the opposite side by 0.29mm. The ratio of the displacements is 0.05 : 0.29.

*This is to say if the magnet is displaced sideway to take up the annulus gap in full, the shaft end would displace 0.29 * (1/0.05) = 5.8mm*

The exact amount of δ_1 could only be determine by measurement using, for instance, a high speed camera. If the measurement shows a value less than 5.8mm, it means the annulus gap was not fully taken up, and there would be no chance of the magnet touching the coil.

Assuming that the magnet could displace up to the full amount of the annulus gap, a force of 20N is needed. This will exert a force of roughly 2/3 of the 20N on to the lower bearing, or 13N.

Assuming a frictional coeff. of 0.06, the frictional force would be ***0.78N***



2. As regards to the inertia effect of the shaft shuttling up and down

As per the diagram on the left where the vertical inertia of the shaft is being resisted by the lower bearing through friction. The wt. of the target with magnet is ~ 35g & a = 100g

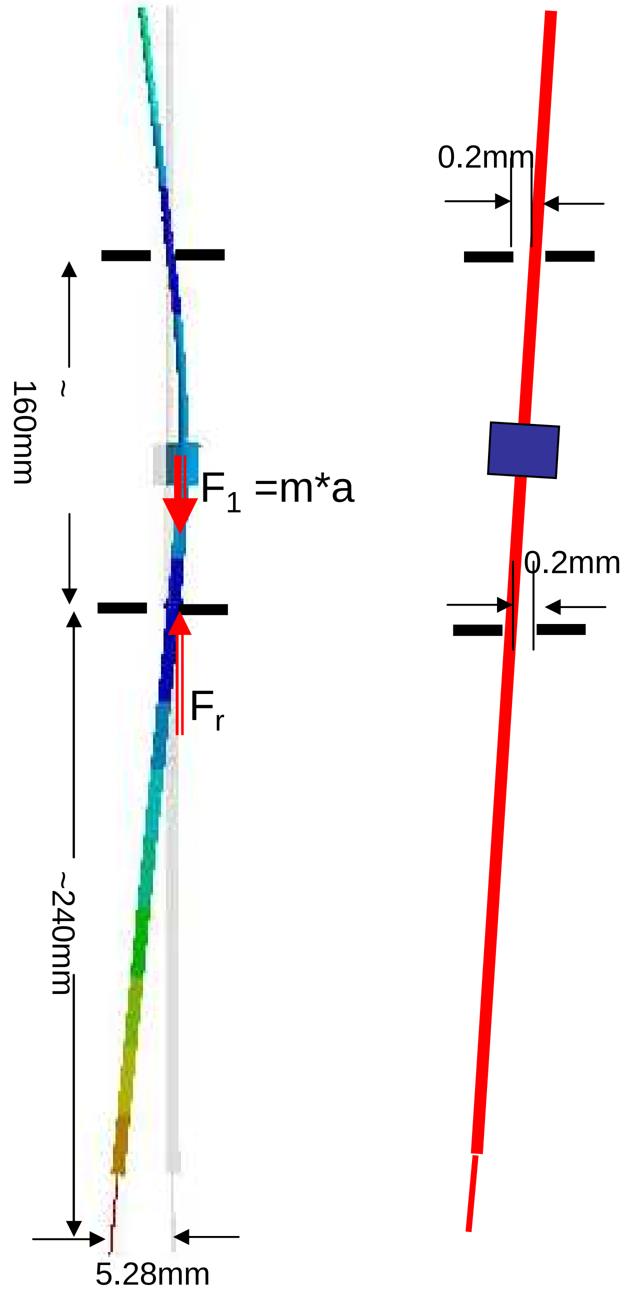
$$F_1 = 0.035 * 100 * 10 = 35N$$

F_r is a direct reaction to F_1. It is by far the most significant force that causes the shaft to rub against the bearing. All or part of this 35N could contribute to heat generation. With resonance mode showing the lower shaft swaying about the magnet, any clearance between the bearing and the shaft will create a condition allowing the shaft between the lower bearing and the magnet to “press” against the top surface of the bearing inner bore.

The inertia due to the shaft is tilted to take up the annulus gap in the bearing is negligible.

Assuming an annulus gap of 0.1mm, this allows a tilting angle of 0.2/160 ➔ 0.00125º. The horizontal component of the inertia force would be

*~ 35 * 0.001 ➔ 0.035N which is negligible*



Thermal energy equivalent:-

The speed of the shuttle is 0.088m/0.03sec or 3m/s.

The total resistance / frictional force from case 1 & 2 is 0.78 + 35 = 35.78N

Without taking into account any contribution from the Twisting effect, a total heat generation of (0.78 + 35) * 3 = ***107 W could be possible***

Since the shaft only runs once every 30th of a second, thermal energy could be as much as 3.6 W throughout.

Our next step is to examine the thermal behaviour at the local spots of the shaft and the bearing.