DEVELOPMENT OF VIBRATION SOURCE REQUIREMENTS FOR TMT TO ENSURE AO PERFORMANCE

Douglas G. MacMartin\textsuperscript{1a} and Hugh Thompson\textsuperscript{1b}

Thirty Meter Telescope Project, Pasadena, CA 91125

Abstract. In order for TMT to deliver the required adaptive optics (AO) image quality, vibration sources throughout the observatory need to be understood and their resulting optical response characterized. The sensitivity to vibration has been determined using a finite element model of the telescope structure and mirror segments coupled to optical models. Frequency dependent models of the AO, active optics and mount control systems are included allowing end-to-end assessment of vibration sources on AO-corrected image quality; future work will improve estimates of the propagation of vibrations from equipment in the summit support building and enclosure to the telescope pier. Modeling separately predicts effects on image jitter caused by relative rigid body motion of main optical elements, and the dynamic motion of the 492 individual primary mirror segments. These results have been used to develop allocated requirements on source amplitudes at different locations and as a function of frequency, which will lead to subsystem design requirements (e.g. for isolation systems at various locations both in the support building and enclosure and on the telescope structure). In order to meet an aggressive target for this contribution to the AO error budget, vibration forces on the telescope itself must be limited to a few Newtons in the most sensitive frequency range of 5–20 Hz; larger forces of order 100 N can be tolerated for equipment mounted off the telescope in the summit facilities building.

Keywords: Extremely Large Telescopes, Vibration

1 Introduction

The Thirty Meter Telescope (TMT), shown in Fig. 1, will be both significantly larger than existing ground-based optical telescopes, and will deliver unprecedented AO performance. Vibration due to equipment both on and off the telescope is a source of potential performance degradation for any AO system; see for example the comprehensive review by Kulcsár \textit{et al} \cite{1}. The consequences of vibration are illustrated in Fig. 2; ideally the effect of all vibration sources on the resulting AO-corrected wavefront would be at most a few tens of nm rms; our initial target is an aggressive 14 nm limit combining both image jitter and M1 segment motion.

To ensure that TMT will deliver acceptable AO image quality despite equipment vibration, we need to place requirements on source amplitudes, isolation requirements, and source locations. Our basic approach is to (i) use the telescope finite element model to evaluate the optical sensitivity to forces applied at different locations and at different frequencies, (ii) use this sensitivity analysis to place requirements on source equipment to ensure that the error budget is met, and (iii) evaluate potential vibration sources to determine what steps (e.g., isolation) would be required to meet these requirements, or alternatively assess whether the AO error budget allo-

\textsuperscript{a} macmardg@cds.caltech.edu  
\textsuperscript{b} hthompson@tmt.org
cation to vibration should be increased. We intend to complement this model-based approach with transfer-function measurements made using calibrated sources as the observatory is built.

The sensitivity analysis in Section 2 demonstrates that our performance specification requires vibration sources located on the telescope to be limited to a few N, with larger forces allowed for sources off the telescope. This stringent constraint for on-telescope sources means that equipment should be placed in the summit facilities wherever possible. Vibrations have the largest impact in the 5–20 Hz range; lower frequencies are adequately attenuated by AO, while forces at higher frequencies produce less motion due to the telescope inertia. Vibrations result in both image jitter due to rigid-body misalignments of the optical surfaces and higher-order wavefront errors due primarily to M1 segment motion. For current TMT design parameters, the image jitter contribution to wavefront error is dominant over M1 segment motion; this is partially a result of lower AO control bandwidth in tip/tilt (because natural guide stars are required) and partially a result of having chosen “soft” voice-coil actuators for control of the primary mirror segments; these provide significant attenuation of vibration relative to hard actuators.

We are in the process of a preliminary analysis of major vibration sources on the observatory; an overview is given in Section 3. Understanding the characteristics of different sources is essential for choosing a sensible allocation of requirements that does not result in an undue burden on any one source. This is also required to understand what mitigation is appropriate such as isolation, re-locating equipment, etc. Finally, this is important for assessing whether the current contribution to the AO error budget is realistic, or whether the error budget should be re-balanced to obtain the best trade-off between cost and performance. Our preliminary assessment of sources in the plant room suggests that specifying equipment to have Balance Quality Grades of G1 or better should be sufficient. Isolation mounts are also relatively simple to apply within the facilities building to reduce transmitted forces. In contrast, sources on the telescope are our dominant concern, both due to the significantly higher sensitivity to vibration and the difficulty in isolating components like cryocoolers from the instruments to which they are mounted.
Fig. 3. Smoothing applied to frequency responses; the smoothing kernel is shown at left, and an example (image jitter due to pier vibration) with and without smoothing shown at right.

2 Sensitivity Analysis

The first step in placing requirements on allowable force levels for vibrating equipment is to understand the propagation of these forces to the delivered AO image quality (nm of rms wavefront error) as a function of source location and frequency. This includes both image jitter (tip/tilt) and M1 dynamic segment motion, combined with AO temporal correction, and AO spatial fitting error for M1 motion.

2.1 Analysis Approach and Models

Image jitter and M1 segment motion are computed separately. Image jitter is computed by connecting the finite element model to a linear optical model; this uses the identical code previously used in computing TMT wind image jitter [2]. The individual contributions to image jitter due to the motion of M1, M2, M3, or the instrument itself can be separately computed. M1 tip/tilt dominates at low frequencies and M2 motion above $\sim 10$ Hz. This is true even for vibration forces applied at the instrument itself; instrument rigid-body motion is not the dominant contributor until 50 Hz or higher. Image motion caused by internal deformation within an instrument is not captured by this analysis and may increase this term.

At the frequencies analyzed here, details of the structural response will not be accurate, however the general characteristics are assumed to be valid. Because neither specific vibration frequencies nor specific resonant frequencies are known, results are presented after smoothing in the frequency domain. This is helpful both for understanding important features that might otherwise be obscured by excessive detail, and is more representative of the expected root-mean-square response. However, this does not estimate the worst-case response, which could be significantly higher (by roughly the quality factor $Q \approx 100$). The smoothing kernel is shown in Fig. 3, with a representative example of image jitter before and after smoothing. The smoothed mean-square response at frequency $f_i$ is calculated by averaging the weighted mean-square response over frequency, with the weight at frequency $f_j$ given by $e^{-\alpha |f_j - f_i|/f_i}$, where $\alpha = 13$ and with the overall weight normalized to unity.

The dominant contribution to higher-order wavefront errors is due to segment motion of M1. (That is, the coma etc. introduced by relative motion between M1, M2, M3 and the focal plane is negligible provided that the tip/tilt from mirror relative motion is acceptable.) In order to compute segment motion, all 492 segments are coupled to the finite element model, including
the actuator dynamics and servo loop. This assembly of segment dynamics with the overall telescope model uses the identical code used in evaluating M1CS control-structure-interaction, and has been independently validated [3]. Once the segment motion has been obtained, the residual wavefront after AO correction is calculated using 2D-Fourier transforms and spatial filtering; this has been shown to approximate the performance of the TMT AO system NFIRAOS.

Input locations for vibration considered here include the following, also illustrated in Fig. 1:
- Pier motion; only the response to vertical forces is shown here. All off-telescope sources affect the optical response through motion of the pier.
- Nasmyth platforms: Input forces have been considered at all instrument mass nodes and on the Nasmyth platform itself; the response is similar and results are shown here only for a vertical force at select instruments.
- Laser Heads (mounted on the elevation journal)
- Azimuth and Elevation cable wraps are expected to result in torques about drive axes.

Additional locations where vibrations may enter the structure include mounting brackets for pipes carrying cooling fluid (the flow may be turbulent or include pressure fluctuations from compressors), and heat exchangers where cooling is used, including M2 or M3 electronics and the main drive motors. Vibration caused by actuator or bearing noise within M2 or M3 subsystems are expected to overwhelmingly result in local motion rather than vibration propagation through the structure, and there is thus no need to consider these inputs with the full structural model. For TMT, these sources are separately categorized in the error budget as “control noise”.

2.2 Soil transmission

TMT does not yet have a detailed soil/pier model to calculate the motion of the telescope pier caused by vibration sources in summit support facilities, or due to enclosure operation. All of these sources are captured here by applying forces at the telescope pier; future work will include estimating a frequency-dependent attenuation factor associated with transmission through the soil. The attenuation factor is likely to be a factor of 10 or more based on theoretical analysis of the propagation of surface Rayleigh waves [4]. Note for intuition that for the TMT site on Mauna Kea, the measured S-wave velocity is \( \sim 300 \text{ m/s} \), varying with depth (the propagation speed of Rayleigh waves is slightly less than S-waves by a factor dependent on the material Poisson ratio). Thus at 30Hz, for example, the wavelength is of order 10 m, and there are multiple wavelengths between vibration sources in the facilities buildings and the telescope pier.

2.3 AO temporal rejection

For the rejection of image motion with the AO tip/tilt mirror, we assume a 15 Hz bandwidth and Type II controller; the sensitivity function (rejection) is \( S = \frac{s^2}{(s^2 + 2\zeta\omega_0 s + \omega_0^2)} \), for \( \omega_0 = 15 \text{ Hz} \) and \( \zeta=0.7 \). Higher-order aberrations are corrected by the deformable mirror (DM) with 62 actuators across the pupil. For the DM we assume a 63 Hz bandwidth corresponding to an integrator gain of 0.5 at an 800 Hz sample rate. Both the tip/tilt and DM bandwidth vary with guide-star brightness; these are representative values. The assumed rejection curves are shown in Fig. 4(a). Additional narrowband rejection may be possible [5], either using wavefront sensors, or feedforward of accelerometer information. This is not taken into account here but the capability will be designed into the TMT AO system to provide an additional level of risk reduction.
Fig. 4. AO temporal rejection curves (left) for tip/tilt and DM, and spatial correctability of M1 segment motion due to pier vibration (right), with the DM temporal rejection curve reproduced for comparison. The spatial characteristics of the M1 response are determined by the structure, not by the source, and thus the spatial fitting error is comparable for any source location. M1 surface motion is relatively smooth (see Fig. 5); the AO correction is dominated by temporal rather than spatial fitting errors.

Fig. 5. M1 surface response shape due to unit pier vertical force at 30 Hz; even at this frequency the characteristic length-scale of response is much longer than a single segment and the spatial pattern is relatively correctable by AO. The surface response shape due to other vibration sources is similar; the characteristic length-scale at each frequency results from the structural dynamics, not the source.

Fig. 6. AO-corrected rms WFE due to the M1 segment motion that results from pier vertical forcing if the M1CS actuators were hard (e.g., piezoelectric) or soft (voice-coil, TMT baseline); both the spatial- and temporal correctability from Fig. 4(b) are included. Above the 8-10 Hz bandwidth of the actuator servo loop the soft actuators isolate the mirror surface from the motion of the mirror cell; at 30 Hz the reduction relative to a hard actuator is roughly a factor of 10.

2.4 M1 segment motion

Predicting the response of M1 requires that the segment models, actuator models, and actuator servo loop be integrated with the telescope finite element model. This results in a computationally intensive calculation, and therefore only a few inputs are considered here. The primary mirror segments at the Keck Observatories are controlled by “hard” position actuators; in contrast, TMT has selected “soft” voice-coil actuators. Stiffness is provided by a servo loop with an 8-10 Hz bandwidth. Below this frequency the actuators are stiff and transmit forces from the mirror cell to the segments. However, at higher frequencies these actuators provide significant
attenuation of mirror cell motion, with roughly a factor of 10 less mirror motion at 30 Hz compared with a hard actuator (Fig. 6). This is a key design decision enabling TMT to meet much more stringent vibration requirements compared to existing observatories. The M1CS global edge-sensor feedback loop has a bandwidth of 1 Hz, and does not significantly affect results.

The mirror surface shape resulting from excitation at 30 Hz is shown in Figure 5. The surface motion is determined by the characteristic wavelength of structural modes excited at a given frequency, and is relatively smooth even at 30 Hz. (This is not true near the segment support resonant frequency if hard actuators are used.) As a result, the spatial correctability by the AO system is quite good, and it is the temporal AO bandwidth that limits the ability of AO to compensate for M1 segment motion due to vibration.

2.5 Combined sensitivity

The combined response due to both image jitter and M1 segment motion is shown in Figure 7. For TMT parameters this is dominated by image jitter for frequencies above roughly 10 Hz for any vibration source. If the AO tip/tilt bandwidth was higher (reducing this component), or if hard actuators were used for M1 segment control (increasing this component), then the relative importance of the two could change.

The combined image jitter and M1 segment contributions for 6 different source locations are shown in Fig. 8. The response to pier motion is significantly less than the response to on-telescope sources. Two different instruments are shown for illustration (in both cases, the image jitter response is calculated through to NFI-RAOS); the response is relatively similar.

2.6 Initial vibration budget

In order to place a simple specification on vibration sources, we first identify a shaping filter that approximates the system response to forcing. The shaping filter used here is:

\[
W(f) = \left\{ \frac{(\frac{f}{f_1})^2}{1 + \frac{f}{f_1} + (\frac{f}{f_1})^2} \right\}^2 \left\{ \frac{(\frac{f}{f_2})^2}{1 + \frac{f}{f_2} + (\frac{f}{f_2})^2} \right\}^2
\]

(1)

The comparison of this approximation with the computed (and smoothed) sensitivity for each source is shown in Fig. 8, for the AO performance including both image motion and M1 contributions. The scale factor is computed to maintain the average sensitivity in the 5–20 Hz region; this sensitivity is reproduced in the first column of Table 1. The sensitivity for the elevation cable wrap is used for both wraps for simplicity. The sensitivity evaluated from the MIRES
Fig. 8. Contribution to AO rms wavefront error due to different sources as a function of frequency, and comparison with simplified shaping filter given in Eq. (1) shown by dashed red lines. The sensitivity for each source is also given, defined as the average sensitivity between 5 and 20 Hz. Both image jitter and M1 segment contributions to the error are included.

Table 1. Initial allocation of vibration budget. Each of these categories can be further subdivided; e.g., the off-telescope sources can be divided among enclosure forces, HBS pumps, chiller, and other equipment, yielding a 10 N allocation for each of these categories. This is the allocation for the equivalent force at the telescope pier; there is a further factor of 5–10 attenuation between forces applied in the summit facilities building and the pier, resulting in an allowable force (after isolation) due to chillers of order 100 N, for example.

<table>
<thead>
<tr>
<th>Source</th>
<th>Sensitivity (nm per N)</th>
<th>Fraction of budget (%)</th>
<th>Allowable force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pier (off-telescope)</td>
<td>0.43</td>
<td>35%</td>
<td>20</td>
</tr>
<tr>
<td>Instruments/Nasymth</td>
<td>3.7</td>
<td>50%</td>
<td>3</td>
</tr>
<tr>
<td>Laser Head</td>
<td>1.9</td>
<td>5%</td>
<td>2</td>
</tr>
<tr>
<td>Cable wraps</td>
<td>1.3</td>
<td>5% each</td>
<td>2.5 each</td>
</tr>
</tbody>
</table>

The specification is then made on forces at any location after passing through this shaping filter; this allows for higher forces at low and high frequencies relative to the 5–20 Hz band. Assuming different vibration sources add in quadrature, a preliminary allocation for different locations is shown in Table 1, giving (i) the average sensitivity in the 5–20 Hz region, (ii) the % of the total budget allocated to that source, and (iii) the computed force that would consume
the designated percentage of the total current vibration error budget of 14 nm. The allocation shown for off-telescope sources does not include the additional factor of 5–10 reduction due to propagation through the ground.

3 Vibration Sources

The transmission path analysis makes it clear that vibration sources mounted on the telescope have much tighter requirements than those in the facilities buildings. This is of course expected. However, the additional understanding we want is to know the ratio between the typical source magnitudes as well as the ratio between the sensitivities at these locations. In effect how should we apportion the budget in Subsection 2.6 so as to maximize the effectiveness of vibration mitigation measures that may be undertaken to meet this budget?

We describe some of the key sources of equipment vibration below; all of these will ultimately require a detailed assessment of realistically achievable source amplitudes. Estimates of force amplitudes are given for one vibration source in particular as an example; the air handler example in Subsection 3.3 also illustrates that the vibration requirements in Table 1 for equipment in the facilities buildings seem likely to be achievable.

3.1 Instruments

There are a number of potential sources of vibration associated with the science instruments mounted on the Nasmyth platforms of TMT. Typical crycoolers require oscillating masses close to the cold tip, and are usually mounted on rigid structures tightly coupled to instrument dewars. Electronics enclosures contain circulation fans, heat exchangers and electrical transformers.

There are a number of reasons why vibration from electronics enclosures are not our primary concern. Electronics enclosures are typically simpler to isolate from the telescope structure as their position is not an integral part of the optical chain. Sources such as air circulation fans and transformers can be isolated within the enclosures themselves. Lastly, the frequency content of these sources is typically above our most sensitive 5-20 Hz band identified in Subsection 2.6. The exception to this is vibration associated with fluid distribution and fluid flow within heat exchangers. This is covered in more detail in the following Subsection 3.2.

Cryocoolers however do pose a significant challenge to the required vibration environments of telescopes using AO. In an effort to reduce this problem Jakob and Lizon [6] have performed controlled vibration testing of several commercial cryocoolers that are typical of those used on large telescope instruments. Their results show that these devices typically produce forces of ~1 N in the direction of the reciprocating head. Although these forces are usually maximum at the operating frequency of the head (1 or 2 Hz) higher harmonics can also produce significant forces. With the cooling power of the devices tested in the range of 60W at 77K, TMT would require roughly 30 such devices distributed over the full suite of instruments. TMT is actively pursuing alternate cryocooling strategies in an effort to mitigate the potentially detrimental effects of this many concurrent sources of vibration on the telescope.

3.2 Sources on telescope: Cooling lines & Cable wraps

Water-glycol is typically used for cooling telescope electronics and other equipment to the ambient temperature. To maintain reasonable pipe and hose diameters (particularly through
the cable wraps) laminar flow cannot be maintained in the TMT water-glycol lines. Almost regardless of pipe diameter, the bends required for routing on the telescope structure will result in turbulent flow at least locally to the bends. Initial modelling efforts for the turbulent flow in these bends at scales typical of TMT cooling lines suggests that pipe induced vibration may be a significant contributor to our vibration budget. We intend to perform further modelling as well as laboratory experiments to assess this in more detail. For this and other reasons TMT is considering an option to dramatically reducing coolant flow rates by shifting from water-glycol to phase-change refrigerants.

Although it is likely that the lines themselves can be isolated from the telescope structure this may not be true at heat exchangers where the coupling to the equipment being cooled needs to be more rigid (such as the telescope drives). A similar issue occurs for the telescope hydrostatic bearing (HBS) lines. These lines may also transmit pressure fluctuations originating at the compressor for the high-pressure oil. Buffer volumes reduce this effect; measurements at existing facilities will quantify these effects. Gaseous Helium supply lines for instrument cryocoolers will also exhibit these pressure fluctuation effects.

It is clear that any stick-slip motion of the cable wraps or inability to drive the wraps synchronously with the telescope could easily produce forces well above our allocations in Subsection 2.6. This too is an area where additional analysis and measurement work is needed.

### 3.3 Sources in enclosure and facilities building

There are a number of sources of vibration that are not on the telescope. Rotation of the Calotte enclosure will create some vibration, with potentially large forces but likely at very low frequencies. There are many potential sources of vibration in the facilities building, however the largest amplitude sources are likely to be pumps, chillers and air handlers.

A simple analysis of the largest air handler (AHU) selected for TMT to reject the waste heat of the observatory to the surrounding air has been done. The eight fan rotors used in this AHU each have a mass of 47 kg. These rotors are balanced to “Balance Quality Grade” G 0.56 mm/s. These fans operate at 3546 rpm giving an unbalance of 0.71 kg mm. In the worst-case of all 8 rotors unbalance being in phase, and all eccentric motion constrained by the bearings, the net force perpendicular to the bearing axes would be 79 N.

Alternatively, assessing this more generally and distributing our 100 N allocation of Subsection 2.6 for the facilities among 25 sources in quadrature we can allow them 20 N each. If we assume a worst-case operating frequency of a 2-pole motor connected to 60 Hz line frequency and a rotor mass of 50 kg this results in a G value of 1.1 mm/s. This rudimentary allocation assumes no attenuation of forces between the mounting of the rotor and the floor of the facilities via isolation or any other flexibility in the bearing mounting. Lower operating frequencies would also give significant extra margin to this calculation. The conservatism inherent in these generalizations allow us to believe that specifying Balance Quality Grade of 1 or better for all rotating equipment in our facilities could achieve the foregoing budget values.

Although the specific example of the single largest AHU producing 79 N above suggests that this allocation of 20 N per source or 100 N for all sources may in fact not be sufficient, we think that these two calculations are simply indicative that we are using the right scale for setting tough but achievable limits on vibration sources in the facilities rather than any kind of explicit budget. Such detailed budgeting is future work that will require much better estimates of all sources than we currently have.
4 Conclusions and Future Work

The transmission path analysis that has been done for TMT enables a new methodology for trading vibration sources in one location against another within a modelled observatory. By understanding the relative sensitivity of AO performance to vibration disturbances of different magnitude, frequency and location, we have established a budget with which we can allocate AO wavefront error to measurable equipment vibration specifications. This enables cost-effective engineering where vibration mitigation options can be selected based on trading their cost against a quantified relative impact on AO performance. Although we have much work to do to both complete our toolset and validate our results, we believe that such techniques are necessary to achieving desired AO performance on extremely large telescopes.

In parallel with validating the sensitivity model, we must also estimate or measure the complete list of sources within the TMT observatory. A key element to completing these tasks is to make measurements at existing observatories of both vibration transmission as well as source characteristics for equipment and configurations as relevant as possible to TMT.

We recognize that several important factors are ignored in this initial analysis. Local resonances can result in significantly higher sensitivity at particular frequencies and particular locations. However, pure tone vibration problems are often both easier to find and to mitigate. Further, if the ‘integrated’ vibration environment prevents us from reaching our AO performance targets, budgeted allocations like this will be the only recourse to effectively applying solutions to reduce the problem. The framework presented here will help ensure that TMT delivers unprecedented AO performance.

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