Control of variable frequency vibration in large span composite construction floor in a high rise building

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ABSTRACT

A new 31 storey commercial building was the subject of complaints regarding excessive floor vibration. There were some concerns regarding footfall induced floor vibration, but the majority of complaints came from the 19^{th} floor, above the 18^{th} floor plant room where a number of large, slow speed, vibration isolated, centrifugal fans were operated by variable speed drives (VSDs).

It was a simple matter to identify that the fans, when operating at 360rpm, generated significant energy that excited the floor directly above the plant room, whose first natural frequency was also identified at being around 6Hz.

Controlling and reducing the vibration was less straightforward. Due to their size, the fans could not be replaced. The greatest vibration occurred at the floor mid span and tuned mass vibration dampers (TMD's) were specified to reduce the floor response at 6Hz. Following the TMD design and installation, occupant complaints continued and it was established that the variable speed operation and the extreme occupant sensitivity at 6Hz resulted in annoyance even when the AS2670 threshold for offices was met by a factor of 10dB or more. The problem was eventually solved with a unique design of variable frequency tuned dampers conceptualized and specified by Marshall Day Acoustics (MDA) and designed and supplied by G.P. Embelton & Co. A total of 4 such dampers were successfully installed at strategic locations under the L-19 floor with very significant vibration reduction results.

INTRODUCTION

The Southern Cross Building (SX1) Melbourne is a conventional commercial office tower with a central core, and steel reinforced concrete floor slabs. The composite floor consists of 610UB101 primary beams or girders, with 9m spans between columns; and 15m long secondary beams or joists (from core to curtain wall) also 610UB101 members, with a separation of 3m.

The floor slab is 120mm thick 32MPa concrete poured on Condek formwork with a 40mm pre camber.

The Victorian Government Purchasing Guidelines (VGPG) used by the client for the SX1 project provided no vibration criteria. Accordingly three commonly used standards were considered for establishing a criterion for assessment purposes.

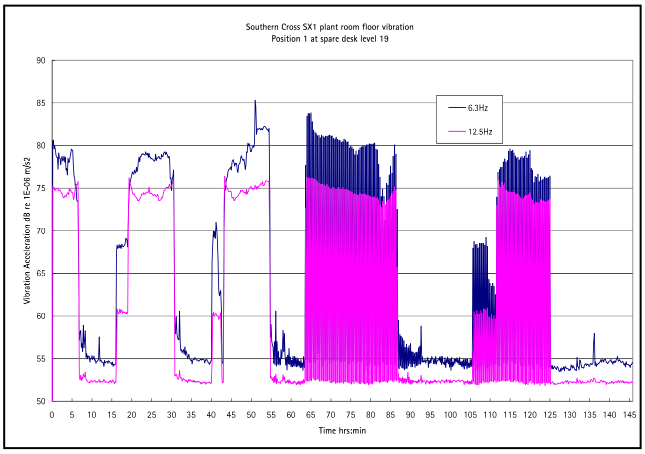

Figure 1. Overall measured vibration L-19, SX1

Figure 1 shows the floor vibration vs time with the AHU 18-1 fan operating with time of day rotational speed being significant factors for consideration. The variable speed drive (VSD) operates the fan at speeds between 0-600 rpm depending on thermal demand. MDA initially proposed limitations on the fan speed to below 80% of maximum and also not to operate within a VSD speed range of 16-25Hz (viz. 192-300rpm) in order to avoid floor resonances.

The American Institute of Steel Construction (AISC) and the National Building Code of Canada (NBCC) specify an acceptable vibration criterion for walking of 0.5%g peak acceleration, where g = $9.81m/s^2$. Hence, 0.5%g is equal to $0.05m/s^2$. Given that this criterion applies between 4-8Hz and that AISC and NBCC specify peak acceleration, this criterion is equal to $0.035m/s^2$ RMS.

Australian Standard 2670.2 – 1990 "*Evaluation of human exposure to whole-body vibration Part 2: Continuous and shock-induced vibration in buildings (1 to 80 Hz)*" recommends vibration limits for a variety of building uses. Table 1 presents vibration acceleration limits for offices, as recommended in Annex A of AS 2670.2 for continuous or intermittent vibrations.

Table 1. Acceleration vibration levels in building criteria, (dB re 10^{-6} m/s^2)

Criteria	Applicable	6.3Hz	12.5Hz
Curve 1	Sensitive spaces	74	78
Curve 1.4	Residential night	76	81
Curve 2	Residential day	80	84
Curve 4	Commercial Office	86	90

INVESTIGATION OF VIBRATION SOURCES

The dominant source of the observed vibration at L-19 of SX1 is AHU 18-1; when this unit was switched off, little vibration was perceptible even when standing directly above AHU 18-3. At the other end of the building on the same floor, the vibration from AHU 18-2 and 18-4 was also perceptible, but of lower magnitude and less likely to lead to such extreme complaints.

One test determined that a (horizontal) system resonance occurred at 230rpm. This is equivalent to 3.8Hz which indicated that the spring isolators under the fans were not correctly loaded. The project specification nominated 50mm deflection coil steel springs with a natural frequency of 2Hz. The 3.8Hz resonance may be a lateral (viz horizontal) mode, although since coil springs are normally less stiff horizontally than vertically, we suspected this was a measure of vertical resonance, and was too high.

Several attempts were made to dynamically balance the fans and adjust the drive belt drive tension; however there was no resulting appreciable change in vibration.

As the source of vibration was directly related to AHU's 18-1 and 8-3, the investigations were directed at identifying potential vibration transmission paths to the building structure and in particular to the L-19 floor slab above.

Following one inspection the following fundamental actions were recommended:

- Frequency banding to restrict fan operation outside the critical range of interest.
- All AHU 18-1 supply air ductwork should be suspended by combined spring and neoprene rubber hangers with a deflection of at least 50mm. Any identified bridging of isolators by incorrectly aligned hanger rods or hangers were corrected.
- The roof of the AHU enclosure had to be installed so that there was no direct connection between the AHU roof panels and the L-19 floor slab above. If connected by rigid hangers including wire or chain, then vibration isolators were required
- The walls of the AHU enclosure could not run full height from the floor of L-18 to the underside of L-19 above. Any structural elements that breached between the two that were attached to the AHU enclosure had to be decoupled using flexible isolated connections
- Where the supply air duct passed through the AHU enclosure penetration a clearance of 50mm was required on all sides to prevent transmission of duct borne vibration to the AHU walls.

Figure 2. AHU 18 -1 SA fans being lifted into position during construction

Other surveys also confirmed that the axial type return air fans (RAF), TEF and GEF fans were not major contributors to the floor vibration even though all were mounted from the L-19 slab.

UPWARDS OR DOWNWARDS PROPAGATION

Vibration level measurements were also performed on L-17 to provide a comparison with the L-19 levels. A comparison of the measured vibration on AHU 18-3, at 75% duty, showed the vibration on L-17 was significantly less than on L-19. The difference in measured levels was possibly due to the following:

- Differences in structure between L-17 and L-19
- A unique airborne or structure borne transmission path via the L-18 plantroom ceiling directly into the L-19 slab, but not into the L-17 slab

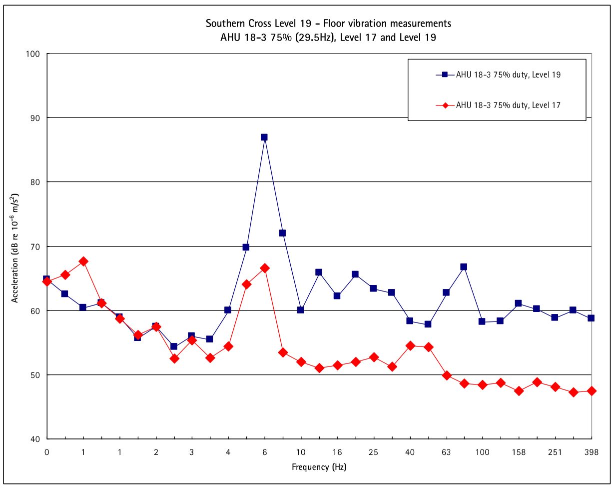

Figure 3. Relative Floor Vibration Levels 17 and 19 (with AHU 18-3 operating)

REVIEW OF OPTIONS

The rectification works and the frequency banding (fan speed range limiting) on AHU's 18-1 and 18-2 were expected to reduce the vibration levels on L-19. However, at rotational speeds outside the banding range (16-25Hz) and under certain load conditions, e.g. 100% duty, the vibration was still unacceptable to the occupants.

Figure 4 shows the progressive reduction in vibration since the rectification work was commenced. These figures show reductions of about 15dB at 6.3Hz and reductions of about 11dB at 12.5Hz. The result was an improvement in vibration to a level below that required for residential premises during the daytime period (AS2670 Curve 2).

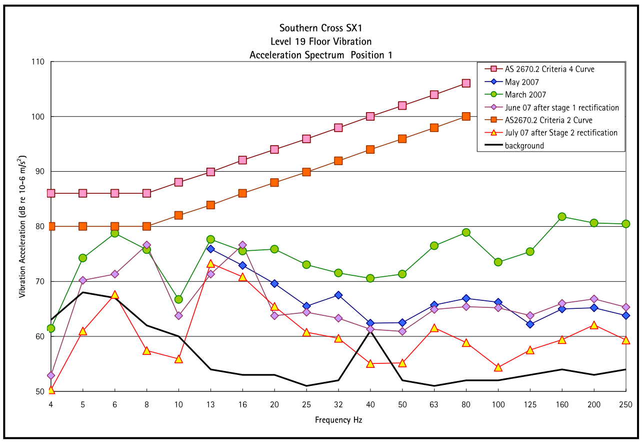

Figure 4. Vibration changes after initial rectifications

Figure 4 shows that, over a period of time, the vibration progressively decreased; however the 6Hz vibration level was below 80dB although this still represented perceptible vibration.

In discussing what was an acceptable vibration level, MDA decided that levels of between 60-75dB were acceptable, levels between 75-80dB were marginal, and levels above 80dB were in excess of what is probably fit for purpose. This target is more stringent than the AS2670.2 Curve 4 criteria for commercial buildings and is comparable to that acceptable for residential buildings (Curve 2).

A second peak at 13Hz was consistent with the fan second harmonic, which excited one of the many higher floor modes.

Shown in Figure 5 are plots of L-18 floor acceleration levels versus frequency for the four operating fans on L-17.

AIRBAG ISOLATION

The preferred option for vibration mitigation was to increase the floor slab stiffness so the vibration isolators could achieve higher performance, as a result of the increased structural rigidity. This option was evaluated extensively by Bonacci Consulting Engineers. Increasing the plant room floor stiffness, and hence raising the floor natural frequency substantially, was not practical or feasible as an increase of at least 25% was required. One alternative solution was to replace the fans with smaller, high-speed units that would operate at speeds in excess of 500rpm. Again this solution was not practical or achievable.

STRUCTURAL DAMPING OPTIONS

As neither of the above options was viable then the installation of vibration dampers was considered. There are two alternative vibration damper systems, namely: passive vibration dampers (Tuned Mass Dampers (TMD)), and active vibration control dampers which use electronic control systems to counteract the vibratory forces.

A 10dB reduction in vibration was considered to be a suitable design target for the performance of the selected vibration damping system.

INITIAL MITIGATION MEASURES

After much work MDA formed the view that the vibration on L-19 was being transmitted from the plantroom by one of two phenomena; airborne excitation or structure-borne flanking.

Surveys had indicated that vibration was not transmitting up the core or structural columns, or via the ring beam and window mullions into the L-19 floor.

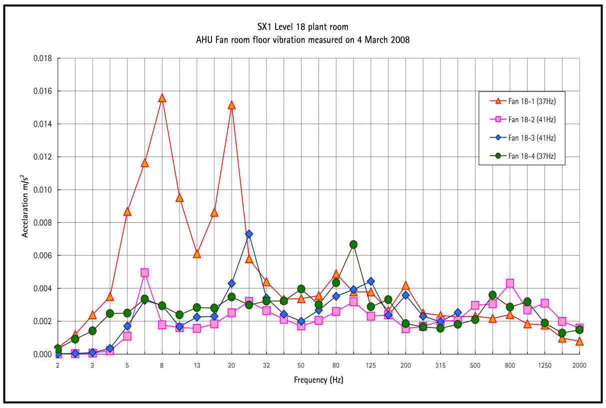

Figure 5. Acceleration levels for the four operating fans across the building

Table 2. Measured vibration acceleration levels at 6.3Hz third octave

Operating Condition	Vibration
Level 19	
AHU 18-3 alone at 75% duty	86-87 dB
AHU 18-1 alone at 75% duty	79-80 dB
Both 18-3 and 18-1 at 75%	86-87 dB
Both 18-2 and 18-4 at 75%	56-57 dB
Background	55dB

HYBRID TUNED MASS DAMPER (TMD) SOLUTION

With vibration levels at 6Hz, considered excessive; even after many modifications, a tuned mass damper was trialled to determine the reduction possible at 6Hz. The design of the TMD is based upon theory given by Rao and the prototype is shown in Figure 6 with no damping and Figure 7 with damping. The estimated effective floor mass was 20,000 kg and using a 430 kg TMD mass this gave a mass ratio of 2.15 %.

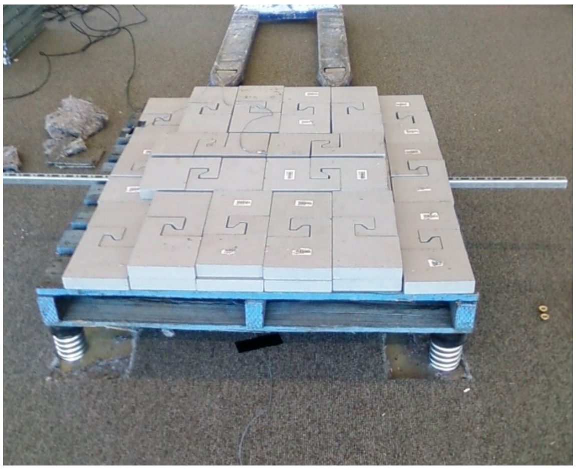

Figure 6. Prototype TMD without viscous damping

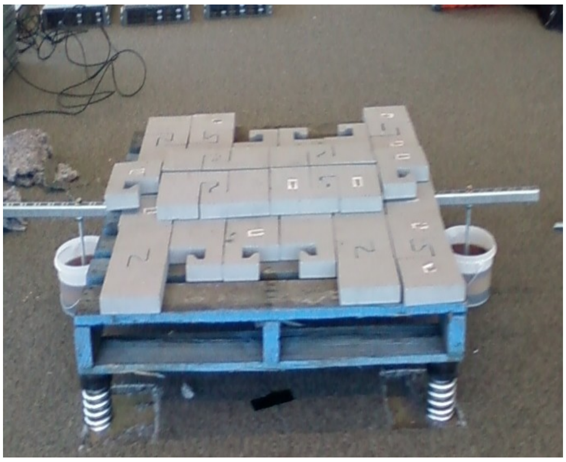

Figure 7. Prototype TMD with viscous dampers

This simple prototype TMD, was tuned to a resonant frequency of 6 Hz. Tests were then undertaken with the TMD on L-19 with the AHU 18-1 fan operating at 360 rpm. Floor acceleration spectra were obtained for the case of no TMD and the two TMD configurations (un-damped and damped). These spectra are shown in Figure 8 which shows large vibration reductions obtained at 5.9 Hz; the acceleration level results are shown in Table 3.

Table 3. Vibration reduction at 5.9Hz obtained by the prototype TMD, dB re 10^{-6} m/s^2

Scenario	Spectrum Level, dB	Vibration Reduction, dB
No TMD	70.5	-
TMD and no damping	49	21.5
TMD with damping	60	10.5
Theoretical 600 kg TMD* design	n/a	16

*With optional damping

A higher level of vibration reduction can be achieved using the un-damped TMD, but to prevent a possible secondary resonance being excited damping is required. A 3 % mass ratio TMD with optimal damping was selected with an expected vibration reduction of 16dB.

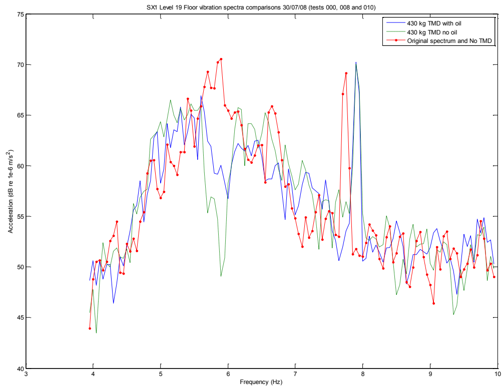

Figure 8. L-19 floor vibration spectra showing original spectrum (solid line with dot) un-damped spectrum (broken line) and the damped TMD (solid line).

Four TMD devices were installed on L-19 to control vibration generated by each AHU in the L-18 plant room. It was decided to use one TMD design for all four positions. The specifications of the TMD are as follows:

- Spring stiffness of 1,320kN/m
- Mass of 700 kg ± 200kg
- Damper variable range from 6200-7700Ns/m
- Design floor frequency of 6.1Hz for excitation frequency at 75% of fan duty
- TMD variable frequency range (dampers) between 4.9Hz and 9.1Hz.

The variation of the mass lever arm allows for tuning as it is easier to vary mass than to vary stiffness.

The resulting floor vibration above AHU 18-1 with the damped conventional TMD attached to the structure is shown in Figure 9.

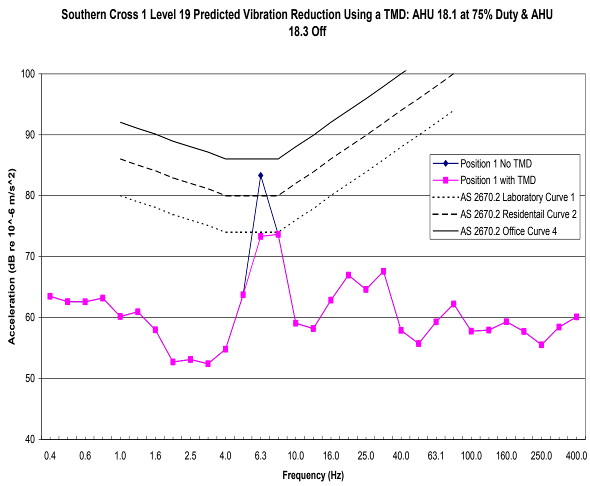

Figure 9. Measured vibration compliance. Vibration with no TMD is black line and floor vibration with TMD is red line.

The results showed a significant reduction and the achievement of Curve 1 to AS2670 (74dB at 6Hz).

VARIABLE EXCITATION DUE TO CHANGE OF AHU SPEED

The AHU fan speed is based upon load demand which is not constant and thus the excitation frequency, continuously changes. The measured acceleration above AHU 18-1 for varying fan speed is shown in Figure 10. The results in Figure 10's light indicate a non compliance at a fan operating frequency of 6.7 Hz. Thus, the fixed frequency TMD was not capable of reducing floor acceleration at all fan operating speeds.

A variable frequency TMD was designed to address this problem. The TMD mass sits on an arm at the free end; the other end is held by a pivot joint. At the free end the TMD spring and damper are located. The entire unit is rigidly attached to the L-19 floor to provide efficient force transfer with minimal displacement. On-site tuning was required to match the exact floor frequency to the TMD resonant frequency. The TMD was designed to achieve a 20 year design life with regular maintenance and servicing. Variable tuning is obtained by moving the TMD mass back and forth along the arm using a worm drive. Details of the variable frequency TMD are given in Figures 11 and 12, and schematic is shown in Figure 13.

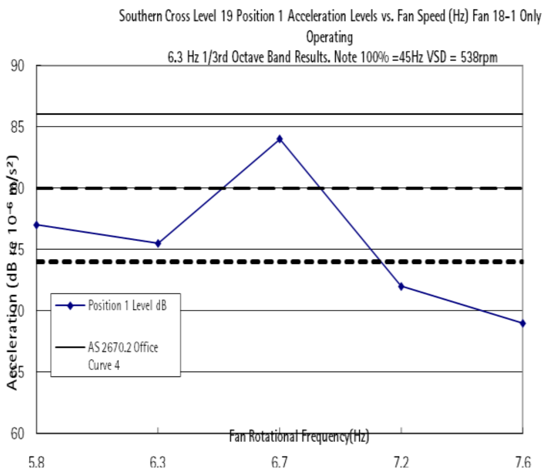

Figure 10. Measured acceleration at 6.3 Hz on L-19 with varying fan operational frequency

Figure 11. Side view of variable frequency TMD, during factory testing.

Figure 12. End view of variable frequency TMD showing dashpot and spring during commissioning.

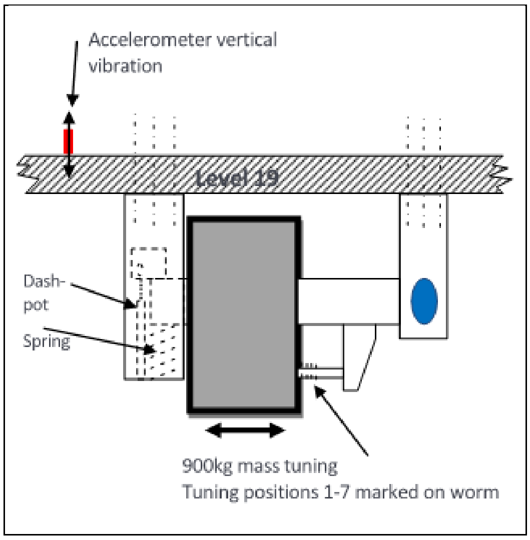

Figure 13. Details of the auto-tracking TMD attached to the L-19 floor.

The position of the mass is varied by a worm drive and motor controlled by the VSD driven rotational speed of the AHU.

The installation and commissioning was conducted jointly by MDA, Embelton and the builder, Brookfield Multiplex.

Measurements of vibration above AHU 18-1 with and without the auto-tracking TMD are given in Figure 14. The floor vibration with the auto – tracking TMD is about 20 dB less in the 6 Hz to 6.8 Hz frequency range. This figure demonstrates the usefulness of the auto- tracking TMD in achieving significant reductions in floor vibration by the use of the variable speed AHU's.

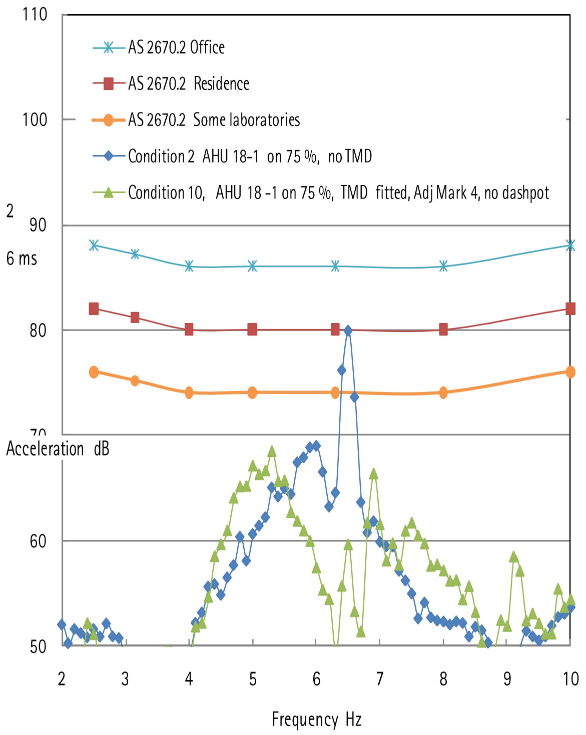

Figure 14. Comparison of floor acceleration with variable TMD fitted.

CONCLUSION

The auto tracking TMD is a new innovative and unique method of controlling vibration in structures from variable speed devices where other methods of vibration control such as replacement of equipment or increasing floor stiffness are not feasible.

REFERENCES

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