STAMFORD | AvK ™

POWERING TOMORROW, TOGETHER

Application Guidance Notes: Technical Information from STAMFORD | AvK

AGN 235 – Generating Set Assembly – Torsional Vibration Analysis

INTRODUCTION

This Applications Guidance Note is the fourth in a series of four AGNs that look at assembling an alternator to a prime mover:

- AGN232 – Generating Set Assembly – Coupling Arrangements
- AGN233 – Generating Set Assembly – Mounting Arrangements
- AGN234 – Generating Set Assembly – Alignment
- AGN235 – Generating Set Assembly – Torsional Vibration Analysis

TORSIONAL VIBRATION

Torsional vibrations are angular vibrations of an object, typically a shaft along its axis of rotation. Torsional vibration is often a concern in power transmission systems using rotating shafts or couplings where it can cause failures if not controlled.

An internal combustion engine produces power using the extremely rapid pressure pulse of a burning air fuel mixture above the piston. The engine produces power in pulses and the pulses can excite very significant resonant responses. The pulsing of the power produced by the combustion process translates to angular vibrations in the crank shaft of the engine. The magnitude of the vibration depends on the number of cylinders in the engine. The movement of an internal combustion engine is illustrated in Figure 1 on the next page.

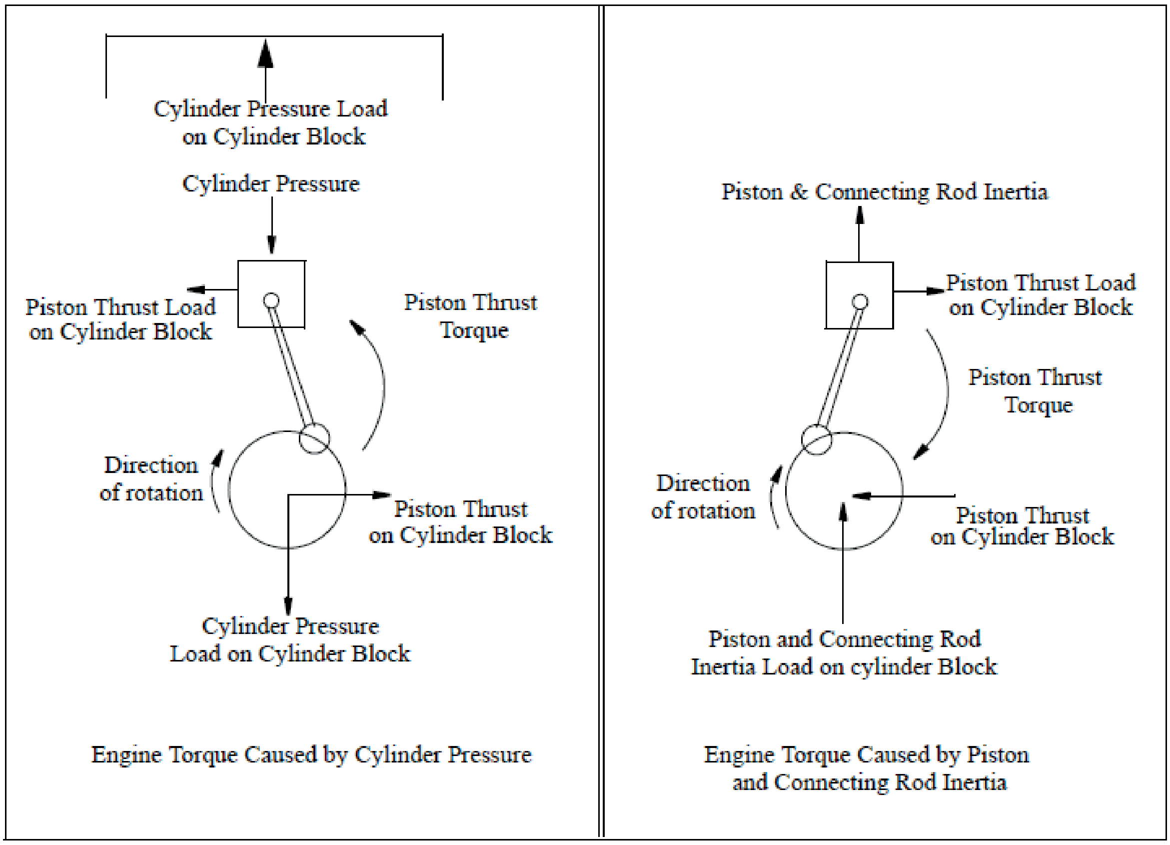

Figure 11: Movement of an internal combustion engine.

ANALYSIS

A Torsional Vibration Analysis (TVA) is required for each unique driveline. A driveline could be an engine and alternator combination in a Generating Set. A TVA is required to measure the angular vibration against acceptable limits.

A TVA should be performed:

- To prevent fatigue or wear damage to the engine, alternator or the attached drivetrain system.
- To minimize product noise and sound quality.
- To improve equipment performance.
- To avoid annoying vibrations transmitted to other parts of the machine.
- To avoid adverse effects on engine governing or other machine control functions.
- To ensure engine/alternator compatibility is achieved.

A reciprocating engine produces power in pulses and these pulses can excite very significant resonant responses. As an engine translates reciprocal motion of the piston into rotational motion of the crankshaft, the output is delivered to the flywheel and drive system. Most crankshafts are not stiff enough to be modelled as one rigid body; even if there are no significant resonant behaviors. Due to cylinder pressure induced torque alone, a crankshaft can twist enough to impact the system's forced responses. Due to this flexibility, it is common practice to simulate a crankshaft and alternator with some degree of torsional flexibility. Given current computer processing/simulation capabilities, these torsional systems are typically simulated as a series of rigid bodies (inertias) connected by torsional springs. This is called a mass-elastic model. A typical engine model will have one inertia for each crankshaft throw and/or each cylinder, with additional inertias at each end; representing such things as a pulley and a flywheel as shown in Figure 2.

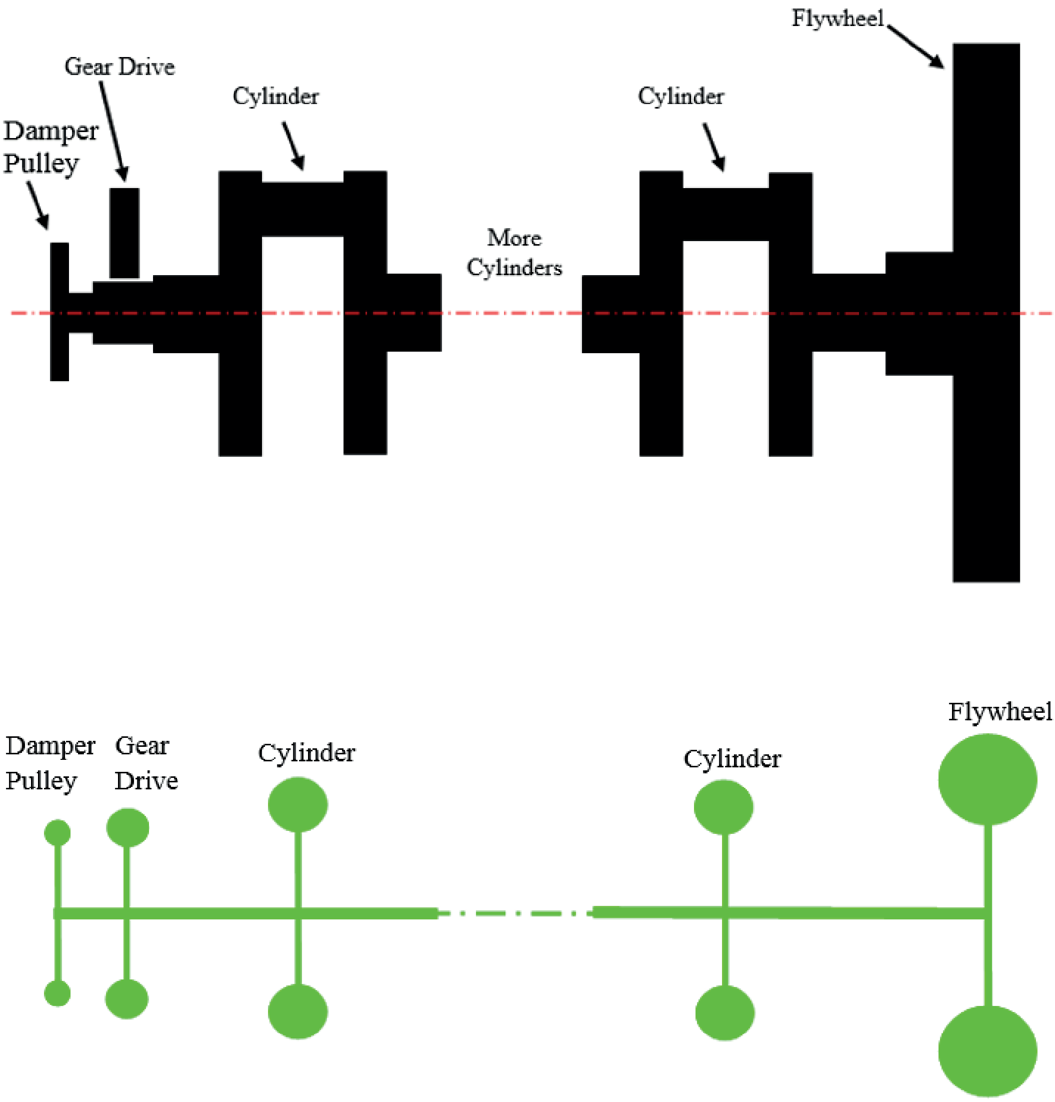

Figure 2: An equivalent linear spring/ mass system for an engine.

A typical alternator model consists of the following inertias: drive disk/hub, fan, rotor, and exciter. Dependent upon the rotor's torsional stiffness, it may take more than one inertia to represent a rotor (the hub/drive disk inertia is often just added to the flywheel inertia). This is dependent upon the required level of accuracy in the model. If the hub is simulated as a separate inertia, typically two thirds of the drive disk inertia is added to the flywheel and one third is added to the hub. Also, for long rotor cores, the core can be modelled as several separate inertias and stiffness as shown in Figure 3 on the next page.

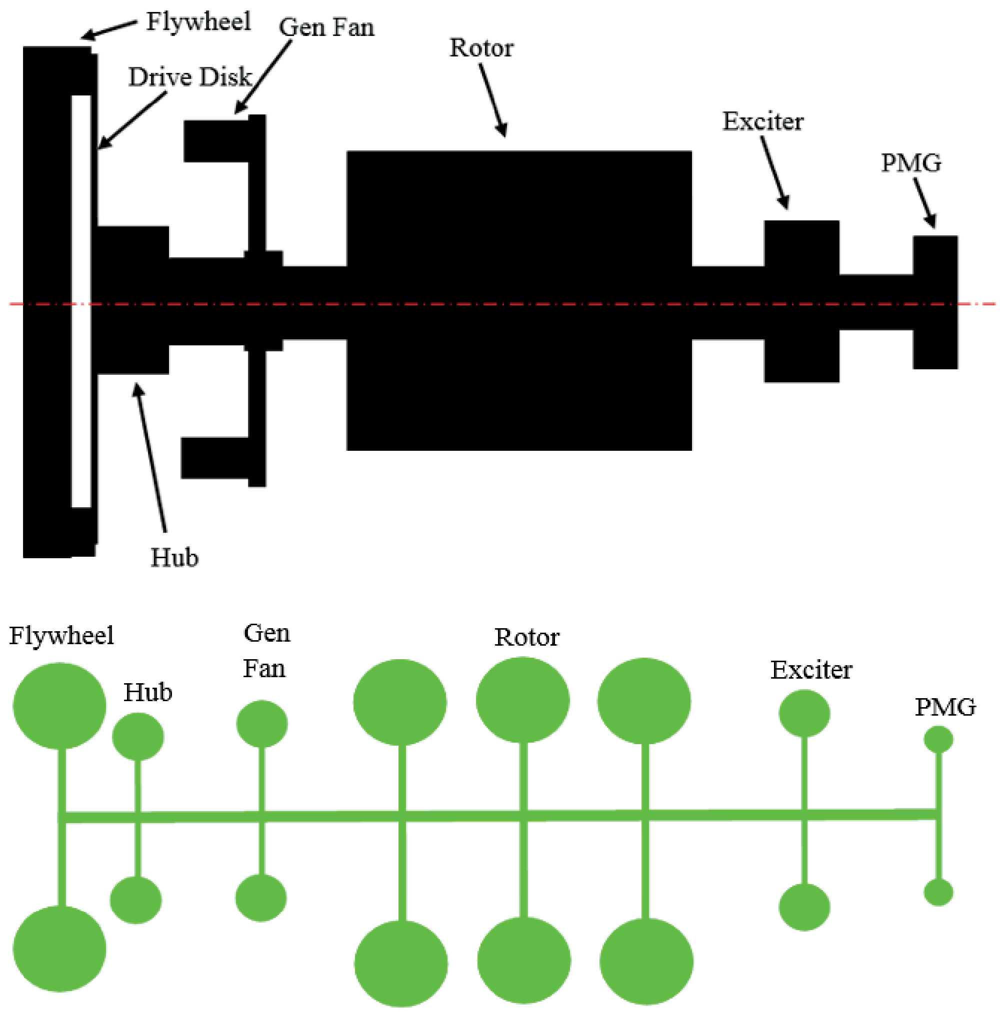

Figure 3: An equivalent linear spring/ mass system for an alternator rotor core.

Figure 4 provides an illustration of a complete Generating Set mass-elastic model.

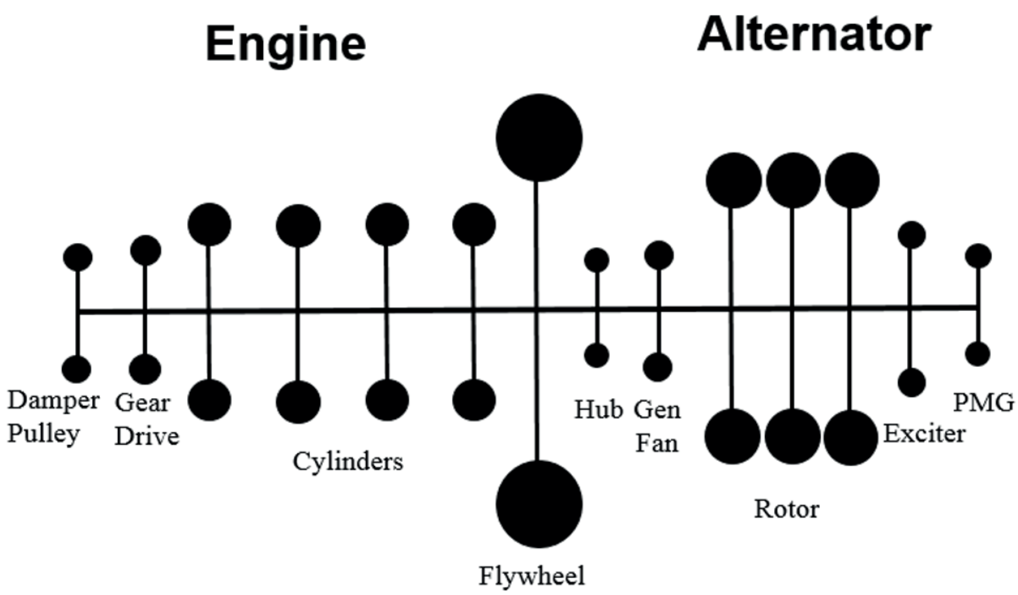

Figure 4: A complete Generating Set mass-elastic model.

TVA requirements checklist

The mass elastic data for the engine, alternator and coupling, as shown Figure 5, should be obtained from each component manufacturer prior to starting a TVA. Additionally, all relevant data such as drawings, component limits and fits and torque transmission limits, can be requested along with mass elastic data.

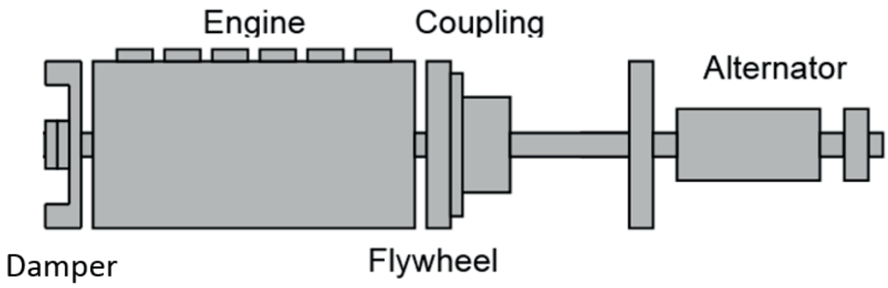

Figure 52: A complete Generating Set and areas for which data is required for TVA analysis.

To perform an effective TVA, the analyser requires adequate data on the engine (Figure 6), alternator (Figure 7) and coupling (Figure 9).

Engine data requirements

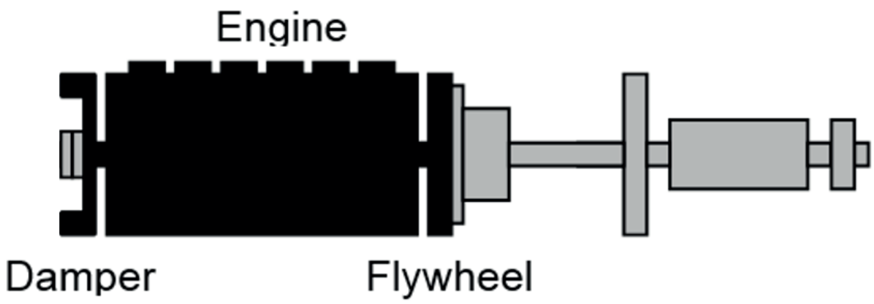

Figure 63: Engine location for TVA.

Engine Data:

- Mass Elastic data including slider crank geometry, power and speed, cylinder pressure trace and firing sequence.

Pass / Fail criteria:

- Torsional velocity at the crankshaft front end for all engine cylinder firing orders should be below the engine manufacturer's recommended maximum limit.

Parameters of interest:

- Torsional displacement at the crankshaft front end.
- Crankshaft vibratory stress.
- Crankshaft front and rear end bolted joints design margins with respect to torque capacity.
- Crankshaft damper heat load.

Alternator data requirements

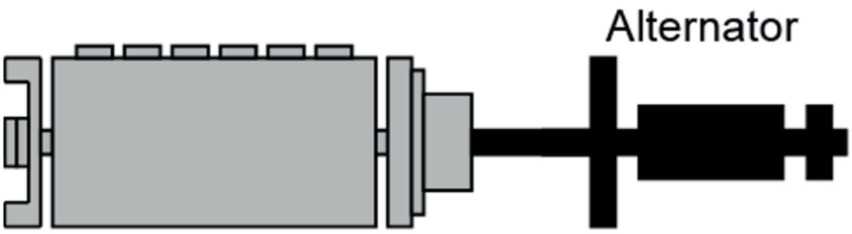

Figure 7: Alternator location for TVA analysis.

Alternator Data:

- Mass Elastic data with rotor shaft geometry and rotor inertia (discretised separately for fan, rotor and exciter, as shown in Figure 8).

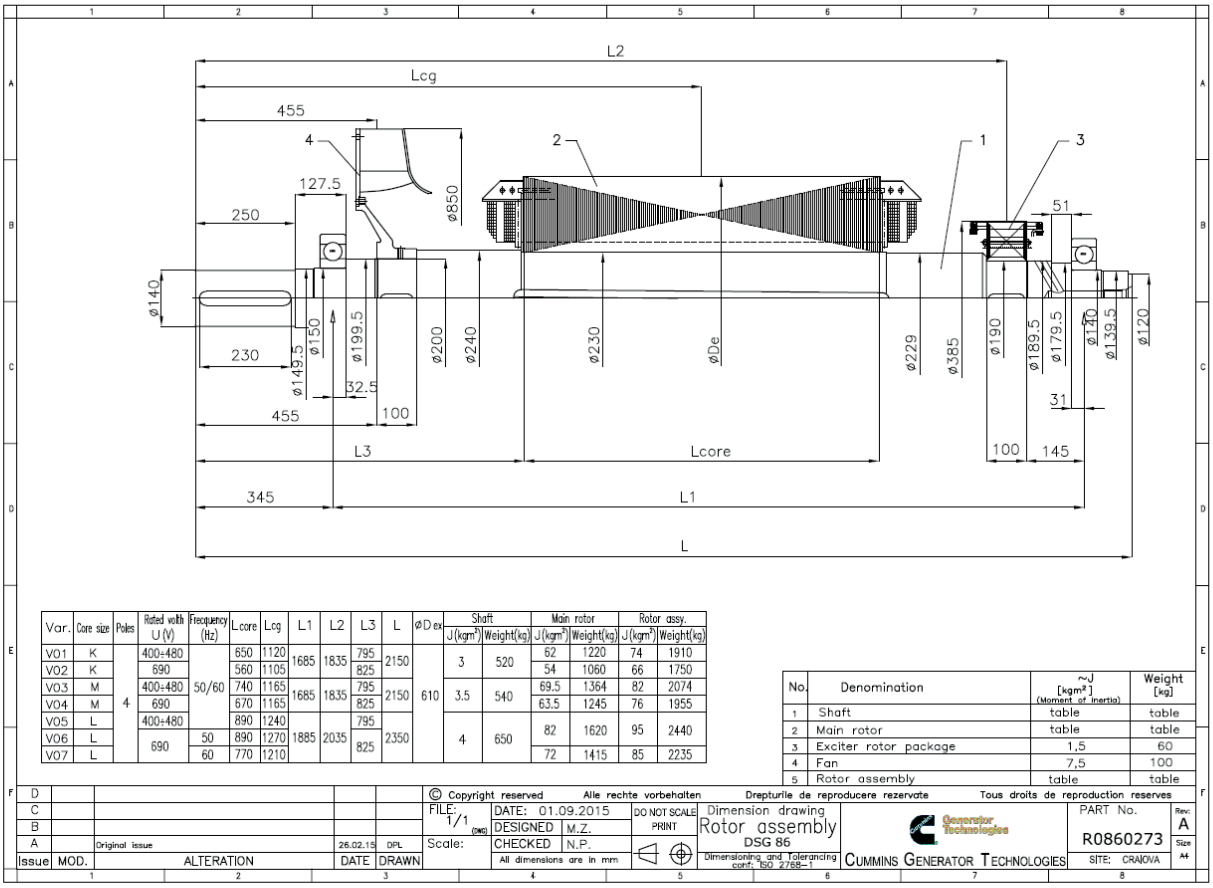

Figure 8: Typical alternator rotor drawing with shaft details.

Pass / Fail criteria:

- Maximum vibratory stress in rotor shaft should be below the alternator manufacturer’s recommended maximum limit.

Parameters of interest:

- Drive disc to flywheel and drive disc to rotor shaft bolted joints design margins with respect to torque capacity.
- Hub, rotor and exciter fit to shaft - design margins with respect to torque capacity.
- Vibratory stress at critical locations on the rotor shaft such as fillets and keyways.

Coupling data requirements

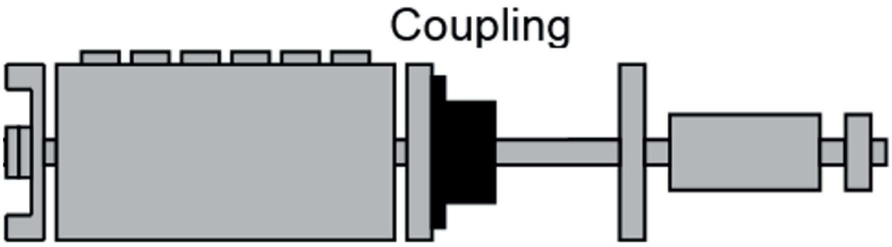

Figure 9: Coupling location for TVA analysis.

Coupling Data (for two-bearing Generating Set):

- Mass Elastic Data including geometry, thermal capacity and torque transmission capacity.

Pass / Fail criteria:

- Vibratory torque at the coupling for engine normal and misfire scenarios.
- Heat load in the coupling for engine normal and misfire scenarios.

TVA Calculations and Results Analysis

Hand calculations and Software tools such as Matlab, FEA, Dedicated TVA software, etc., are used to assess the system, compare alternatives and visualize results across different operating conditions. Dependent upon the engine configuration and its firing sequence, cycle to cycle variation impacts different cylinder firing orders differently. All torsional vibration calculations require validation by test. Typical results from torsional vibration calculations are;

- Angular displacements.
- Angular accelerations.
- Vibratory torques and stresses at shaft sections.
- All torsional modes of vibration (natural frequencies and forced frequencies).
- Heat load in the damper.

The TVA specialist uses the design margin and approved acceptance pass / fail criteria throughout the analyses to ensure system compatibility. The first step in an analytical torsional vibration analysis is to calculate the torsional natural frequencies and mode shapes of the system, as illustrated in Figure 10.

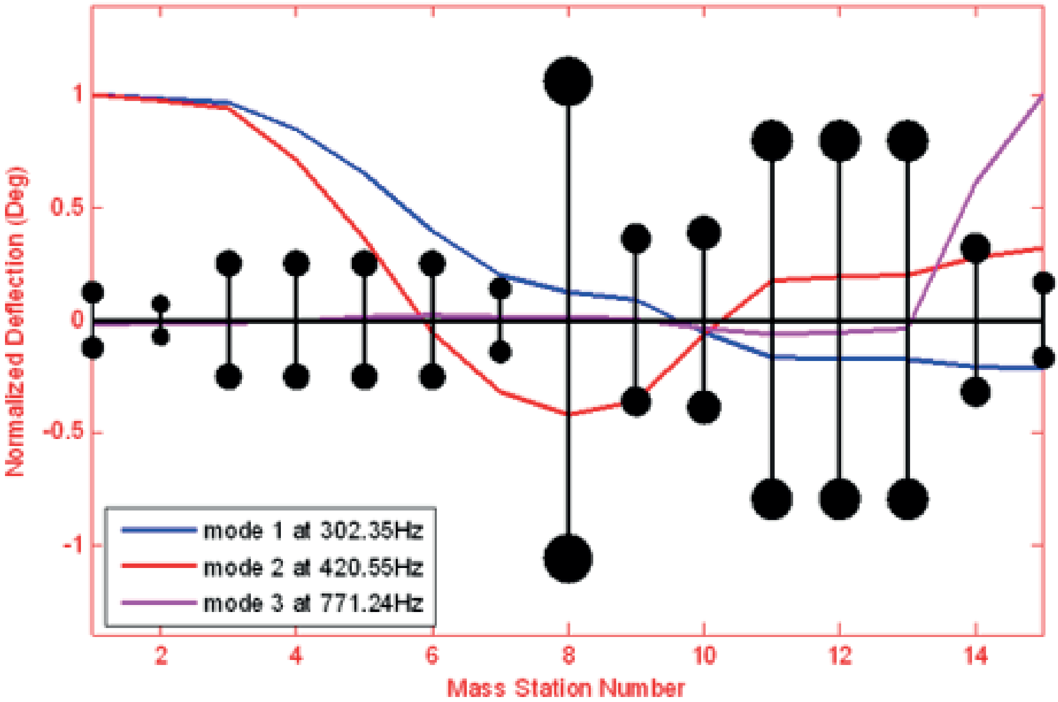

Figure 10: Typical Generating Set modal shape.

The modal shape represents how the shaft system 'wants' to move if it is driven at the appropriate frequency. The plot shows the relative angular displacement of points along the shaft system. The first mode, shown as a blue-line, is usually of most concern for the shaft system since it has the lowest frequency of all the modes.

The second step is to perform a forced response analysis. The forcing function is generated using cylinder pressures and slider crank parameters. By calculating the cylinder's excitation torque, a Fourier transform of the torque can be generated. The expansion terms are the harmonics or orders of vibration for every half order. The final step involves solving the differential equation of the system which represents the dynamic characteristics of the mechanical vibrations.

If the TVA fails any of the pass/fail criteria, the results are shared with the component manufacturers for further consideration and review.

Hand calculation example – Stiffness of a flexible coupling

Stiffness of a flexible coupling calculation as seen below is usually the first calculation to be carried out at the start of a TVA. The example calculation has been carried out to explore the key parameters or factors affecting torsional stiffness of a flexible coupling.

For example, given the following information, find the stiffness needed for a given flexible coupling:

- The engine produces 242 kW_m (325 HP) at 1800 RPM.

- Cranking speed: 200 RPM, low idle speed: 700 RPM and high idle speed: 2100 RPM.
- Engine inertia is 2.63 kg-m^2.
- Couple drive inertia is 0.1 kg-m^2.
- Coupling driven inertia is 0.5 kg-m^2.
- Alternator inertia is 1.75 kg-m^2.

Step 1:

For the purposes of analysing the coupling, the engine-alternator drive system can be reduced to a two-mass torsional system as shown in Figure 11, where the inertia of Mass 1 includes the engine and coupling drive inertias, given by J_1, and the inertia of Mass 2 includes the coupling driven and alternator inertias, given by J_2. It should be assumed that there is a spring in between the coupling that has a torsional stiffness, given by K (Nm/radians).

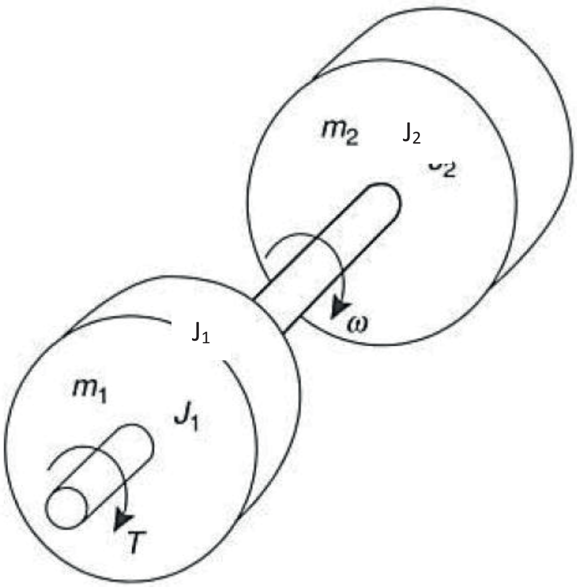

Figure 11: Two-mass torsional system.

Therefore:

Mass 1 inertia, J_1 = 2.63 kg-m^2 + 0.1 kg-m^2 = 2.73 kg-m^2.

Mass 2 inertia, J_2 = 0.05 kg-m^2 +1.75 kg-m^2 = 1.8 kg-m^2.

Step 2:

By using the torsional natural frequency equation for a two-mass system;

$$F_{ntf} = \left(\frac{1}{2\pi}\right)\sqrt{\frac{K\,(J_1+J_2)}{J_1 J_2}}$$

Where

- F_{ntf} is the torsional natural frequency (Hz).

- K is the torsional stiffness (Nm/rad).

Thus; the equation, when solved for stiffness, is;

$$K = (2\pi F_{ntf})^2 [\frac{J_1 J_2}{J_1 + J_2}]$$

For example, assuming that the coupling resonance is between 18Hz and 27.5Hz and the frequencies are substituted in the equation for F_{ntf},

$$K = (2 * 3.142 * 18)^2 [\frac{2.73 \text{ x } 1.8}{2.73 + 1.8}] = 13.87 \text{ kNm/rad.}$$

$$K = (2 * 3.142 * 27.5)^2 [\frac{2.73 \text{ x } 1.8}{2.73 + 1.8}] = 32.39 \text{ kNm/rad.}$$

Then, the stiffness range is between K = 13.87 kNm/rad and 32.39 kNm/rad respectively.

stamford-avk@cummins.com
www.stamford-avk.com

STAMFORD | AvK ™

POWERING TOMORROW, TOGETHER