

Case Study, 38th Turbomachinery Symposium

Title:

Torsional – Lateral Coupled Vibration of Centrifugal Compressor System at Interharmonic Frequencies Related to Control Loop Frequencies in Voltage Source PWM Inverter

Abstract:

Centrifugal compressor train in a refinery experienced high vibration problem due to torsional resonance. Sidebands in the VFD output current based on VFD control loop frequencies were identified as the root cause. In this VFD, stator current was used for torque and speed control, hence control loop frequencies had a potential to generate such sidebands. Frequencies of this type of sidebands widely vary with the rotation speed (proportional to harmonics of the fundamental frequency), hence it is difficult to avoid resonance at the train torsional natural frequency. In addition, even if a compressor system is proven to have sufficient safety margin against high cycle fatigue failure due to the torque pulsation by this mechanism, such minute torque pulsation may have a potential to excite high lateral vibration at speed adjusting gear. If unpredicted or overlooked during design stage, such high vibration may disturb plant operation. This case study therefore proposes guidelines to predict such vibration levels by a simplified torsional-lateral coupled vibration analysis.

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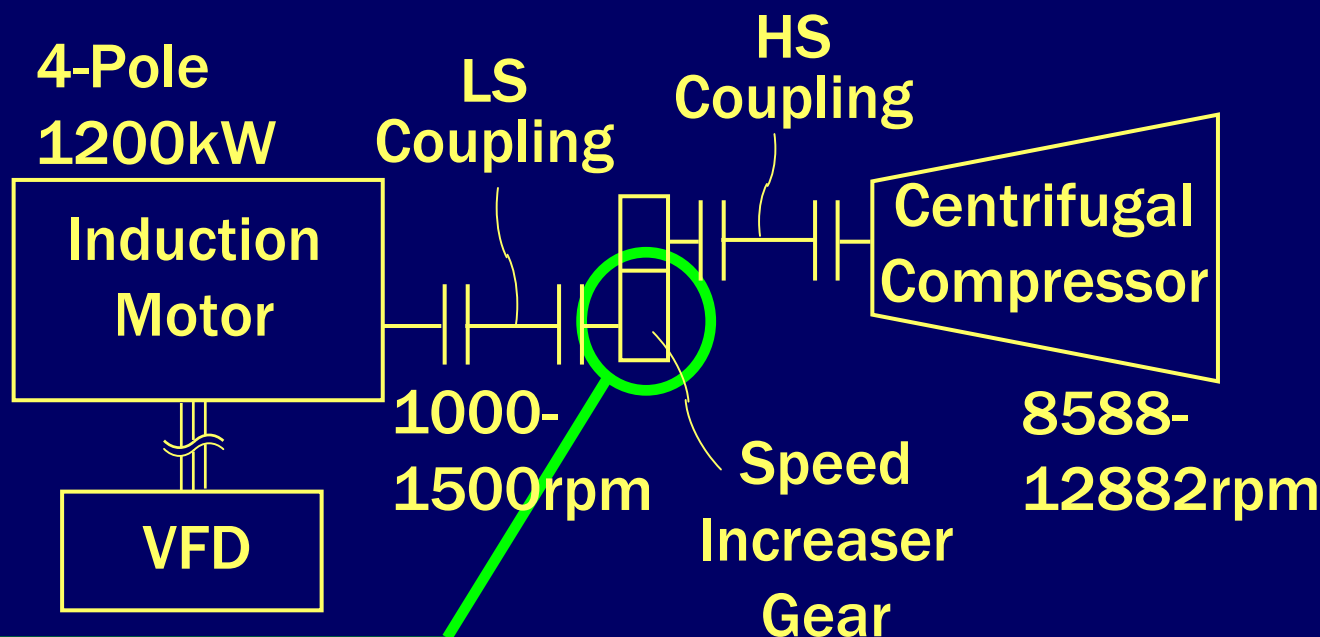
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Introduction

Train Data

VFD Induction Motor + Gear + Centrifugal Compressor



High vibration was sometimes observed at gear LS shaft

Figure 1. Compressor Train

Onset of Problem

Lateral Vibration on Low Speed Gear Shaft

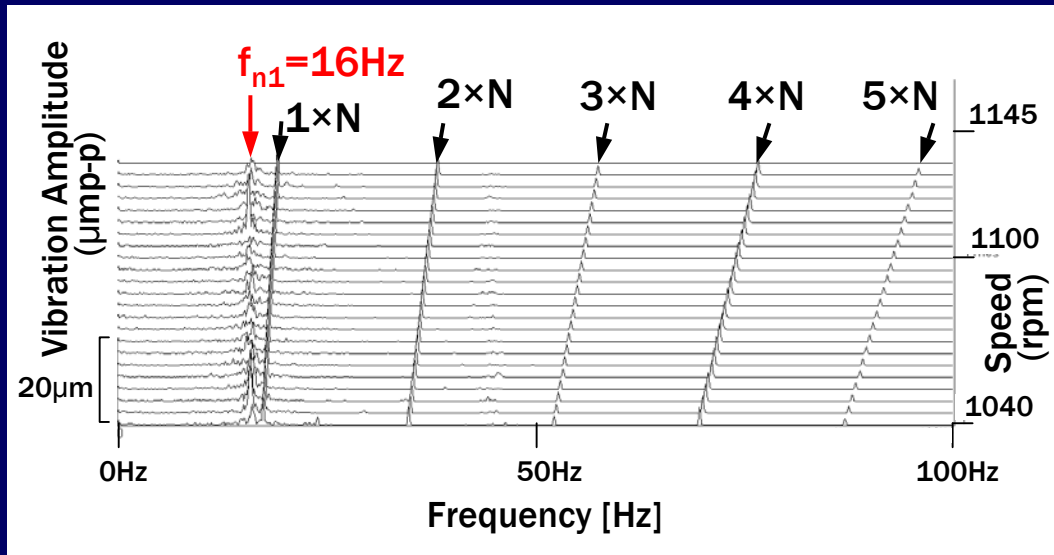


Figure 2. Cascade Plot of LS Gear Shaft Vibration

50 μm (p-p) Vibration Detected on LS Gear Shaft around 1040-1140rpm (17.3-19.0rps), 570kW

Dominant Frequency Component: ca. 16Hz

Close to Calculated 1st Torsional Natural Freq. f_{n1} (15.71Hz)

No Significant Vibration on Compressor Shaft or Motor Shaft

Onset of Problem

Lateral Vibration on High Speed Gear (Pinion) Shaft

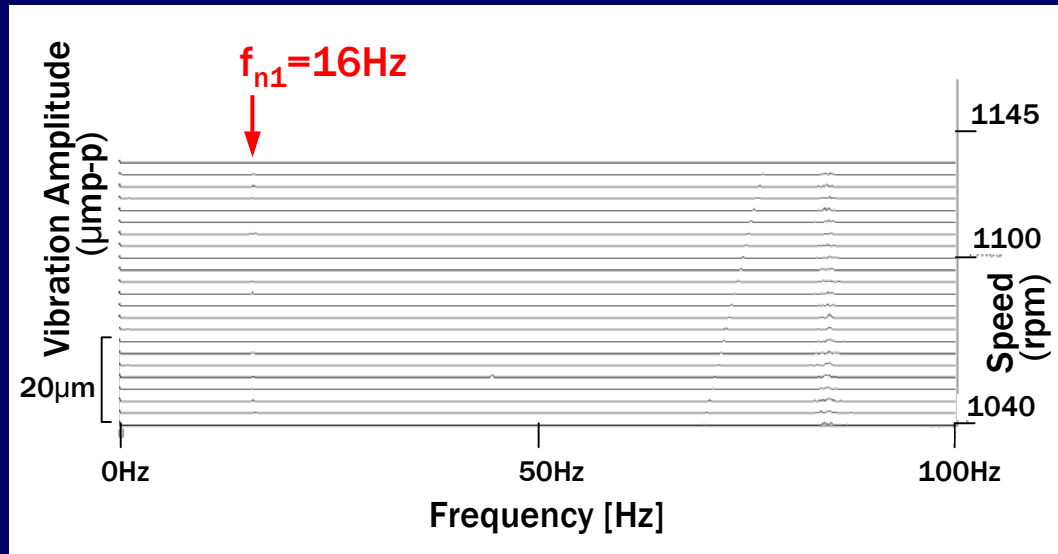


Figure 3. Cascade Plot of HS Gear Shaft Vibration

Max. $10\mu\text{m}$ (p-p) Vibration with 16Hz Detected on HS Gear Shaft

Lower Vibration on HS Gear Shaft



Stiffer Pinion Bearings Than Bull Gear Bearings at Part-Load Due to Uploading Bull Gear and Downloading Pinion

+

Pinion Lighter Than Bull Gear

Concern: What is the Excitation Source?

Torsional Natural Frequencies

Analysis for Torsional Natural Frequencies (FEM)

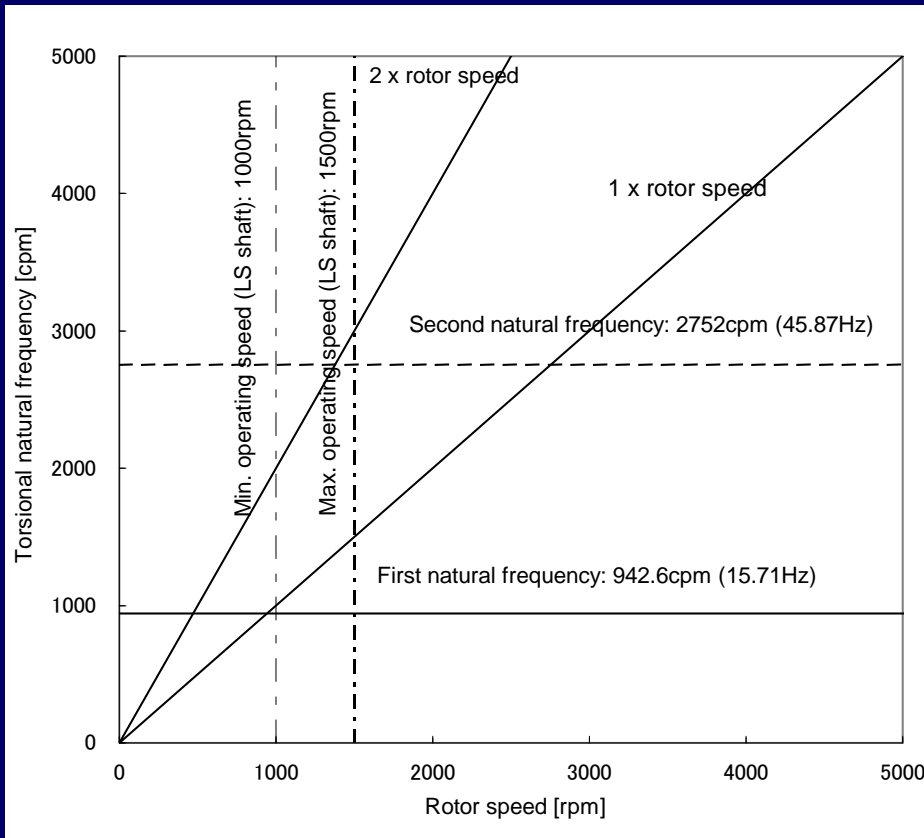


Figure 4. Campbell Diagram for Torsional Vibration

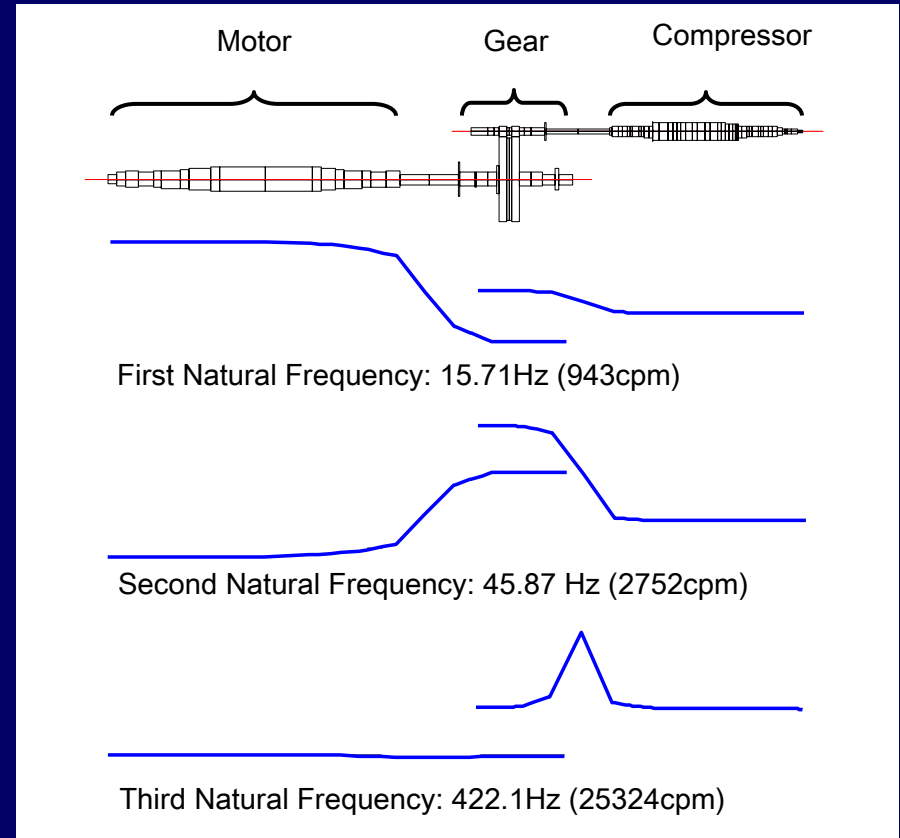


Figure 5. Torsional Vibration Mode Shapes

Critical Speeds Predicted by Rotor Analyses

Lateral & Torsional Vibration Calculation

Critical Speeds [rpm]

		First	Second	Operating Speed
Lateral	Compressor	5220	17700	8588-12882
	Pinion (HS Shaft)	21200-25600 Dependent on Load	-	8588-12882
	Bull Gear (LS Shaft)	3380-9450 Dependent on Load	-	1000-1500
	Motor	2078	-	1000-1500
Torsional		943	2752	

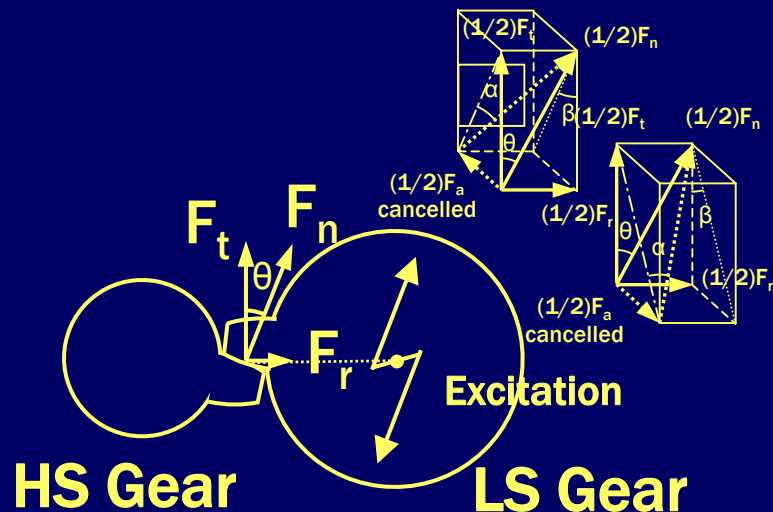
Shaft Synchronous Resonance is Excluded. Only Possibility = VFD?

Estimation of Shaft Torque and Motor Torque

Estimation of Shaft Torque Causing 50 μ m Shaft Vibration

Shaft Torque Deduced by One-Way Lateral-Torsional Analysis

Force Applied on Teeth Causing 50 μ m (p-p) Shaft Vibration



HS Gear LS Gear

Figure 6. Force Applied
on Gear Teeth
(Double Helical)

Transverse Pressure Angle θ

$$\theta = \tan^{-1} (\tan \alpha / \cos \beta)$$

α = Normal Pressure Angle

β = Helix Angle

$$\begin{aligned} T_t &= (\text{PCD}/2) \times F_t \\ &= (\text{PCD}/2) \times F_n \times \cos \theta \end{aligned}$$

Estimation of Shaft Torque and Motor Torque

Lateral Vibration Analysis of LS Gear Shaft

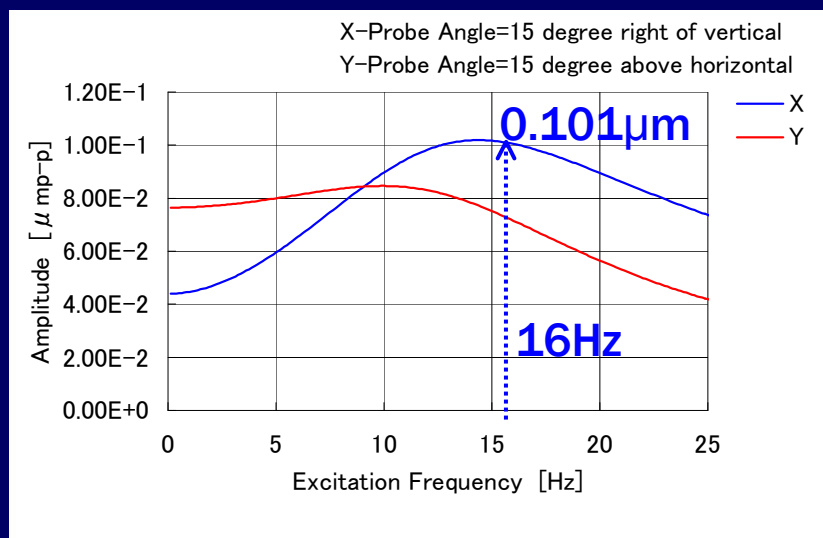


Figure 7. Frequency Response to Unit Excitation Force of Gear Shaft

(Bearing Stiffness and Damping Calculated @ 1140rpm, 570kW)

Unit Excitation on Tooth Surface

$$F_{n_p.u.} = 9.8 \cdot \sin(2\pi f_{n1} t) \text{ [N]}$$

LS Gear Shaft Vibration

Amplitude at Probe @ 16Hz

$$= 0.101 \mu\text{m (p-p) (Calculated)}$$

$$F_n = 9.8 \times (50 / 0.101) = 4851 \text{ [N]}$$

$$T_t = (\text{PCD}/2) \times F_n \times \cos \theta$$

$$= (0.806/2) \times 4851 \times \cos 21.9^\circ$$

$$= 1815 \text{ [N} \cdot \text{m]}$$

➔ 0.213 Times Rated Torque

Estimation of Shaft Torque and Motor Torque

Estimation of Excitation Torque at Motor Air Gap

Amplification Factor for Torsional System

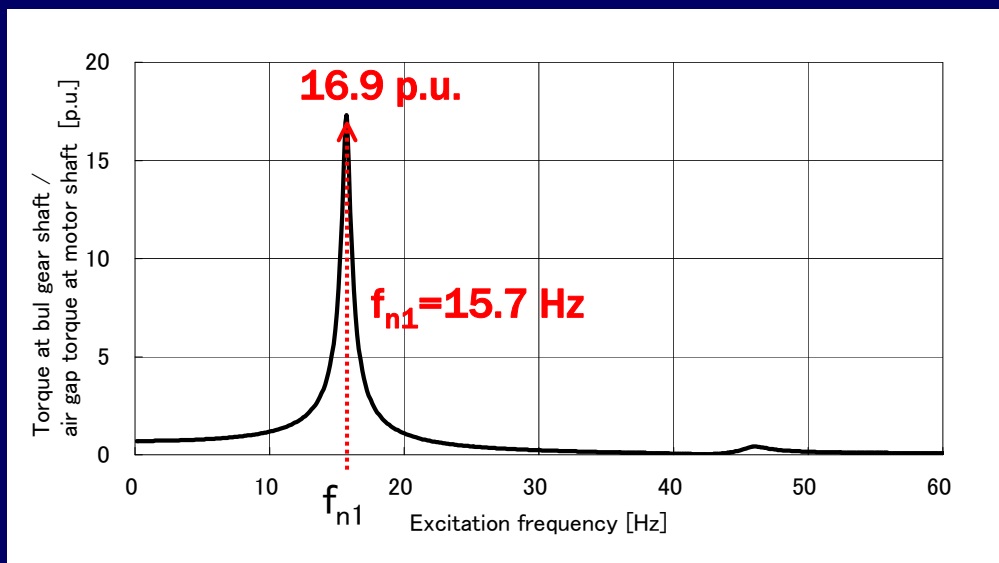


Figure 8. Frequency Response at LS Gear Shaft Assuming $\zeta = 0.02$

$$T_{AG} = 1815 / 16.9 = 107 \text{ [N} \cdot \text{m]}$$



1.3% Rated Torque at Motor Air Gap Suspected.

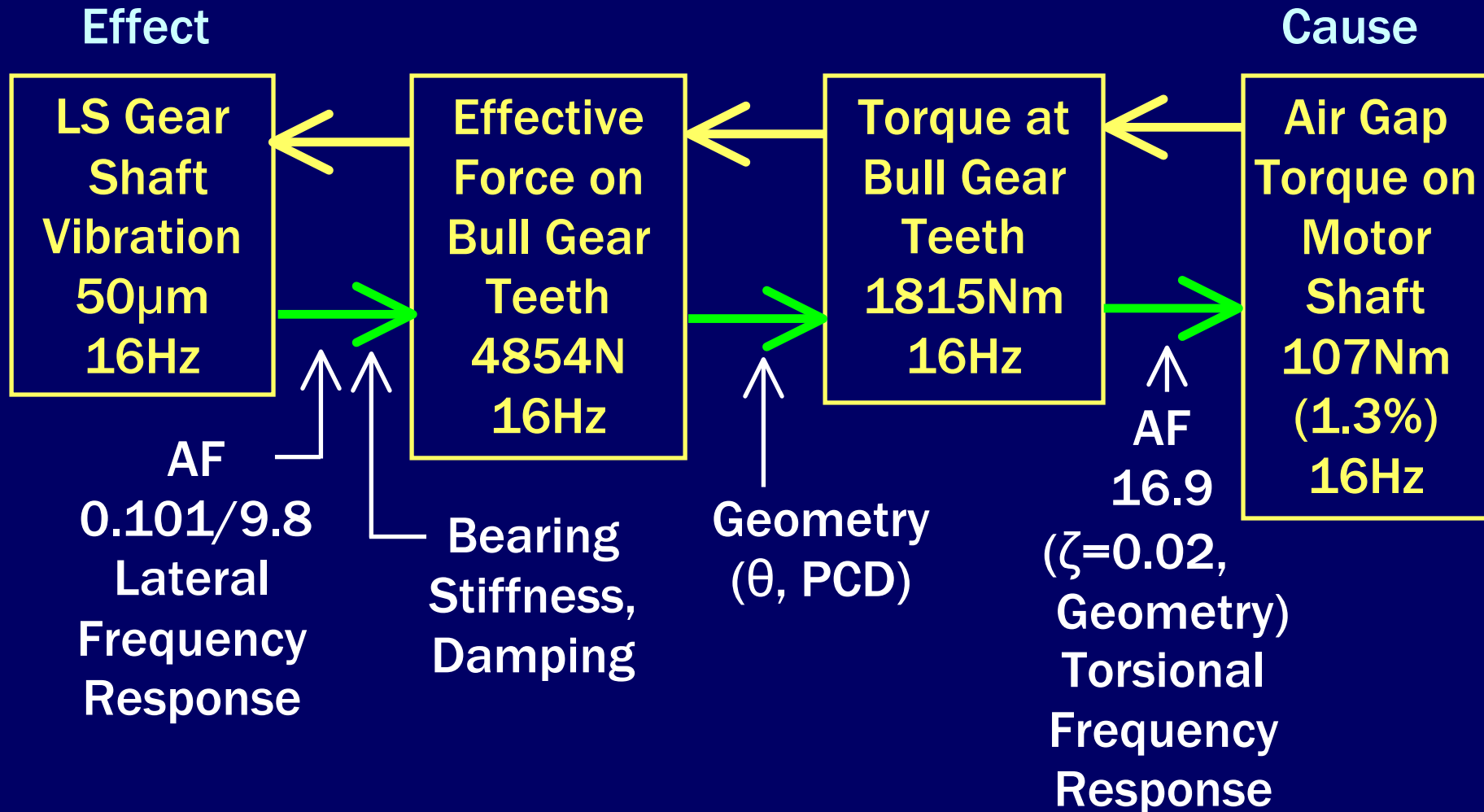
Close to Maximum Measured Data among Interharmonic Frequencies during Factory Test (1.5% Rated Torque)

Merely 1.3% of air gap torque fluctuation can cause 50 μm p-p lateral vibration!

Cause and Effect

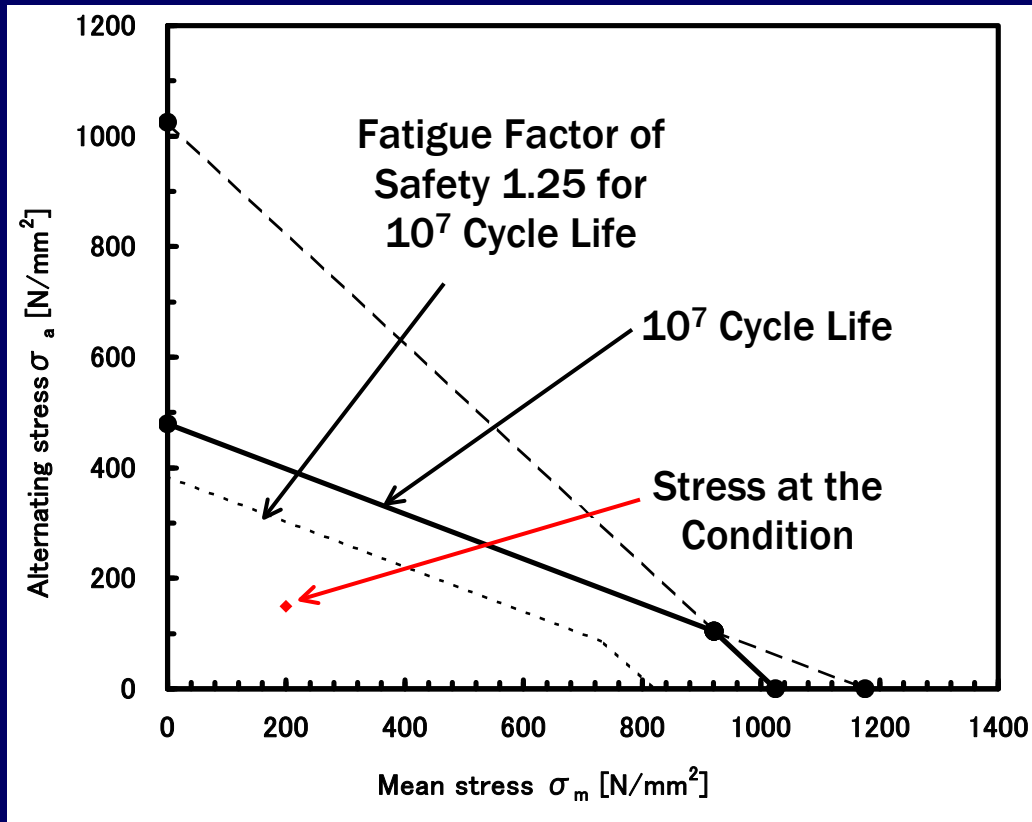
← Actual Cause and Effect

→ Sequence of Lateral-Torsional Analysis



Strength Evaluation

High Cycle Fatigue Evaluation



**Mechanical Strength
Verified**

**No Modification
Made on Machinery
or VFD**

Figure 9. Modified Goodman Diagram

Investigation of Source of Excitation Torque

Detailed Measurement (LS Gear Shaft Vibration)

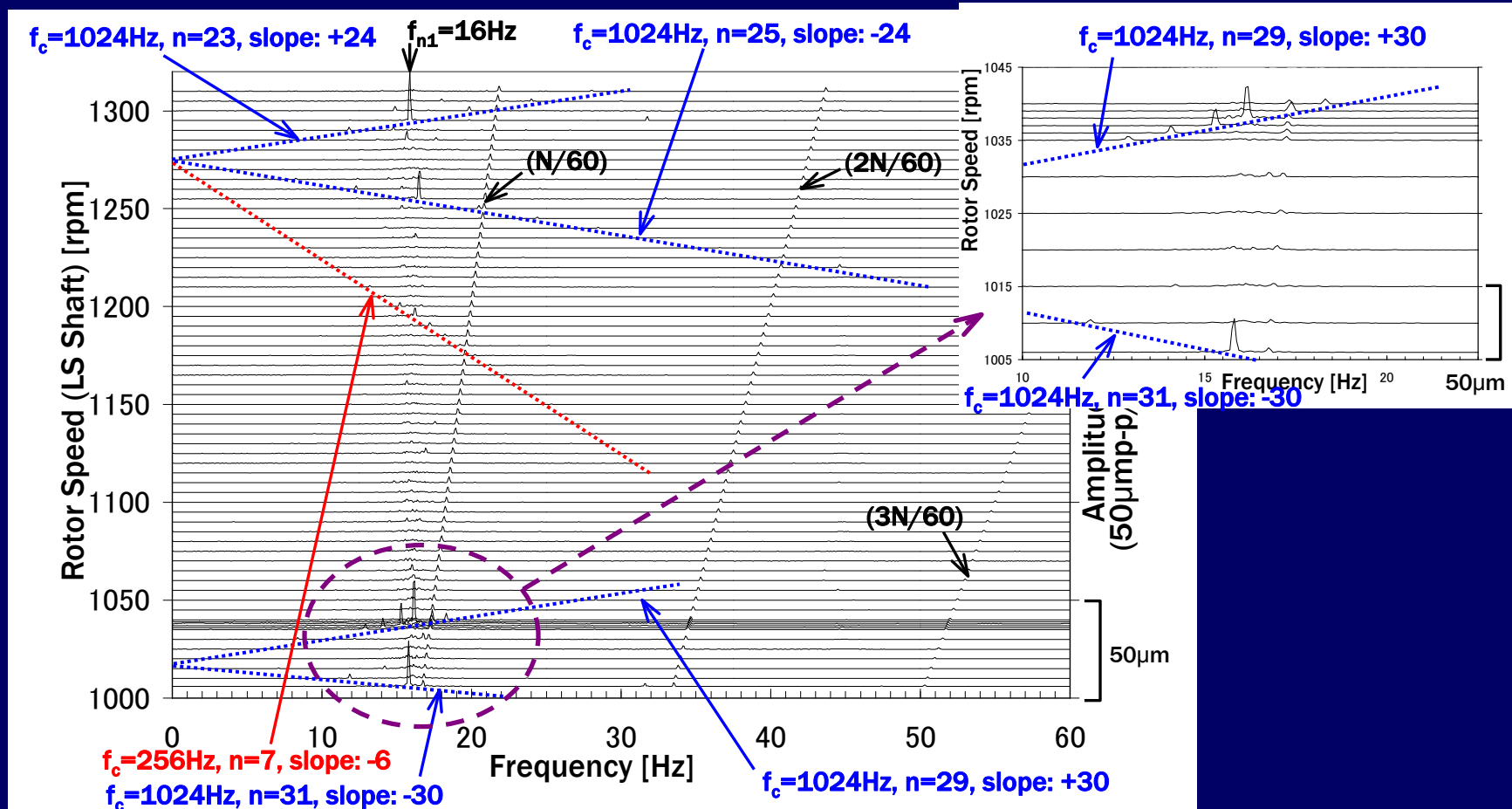


Figure 10. Cascade Plot of LS Gear Shaft Vibration

Investigation of Source of Excitation Torque

Detailed Measurement (VFD Output Current)

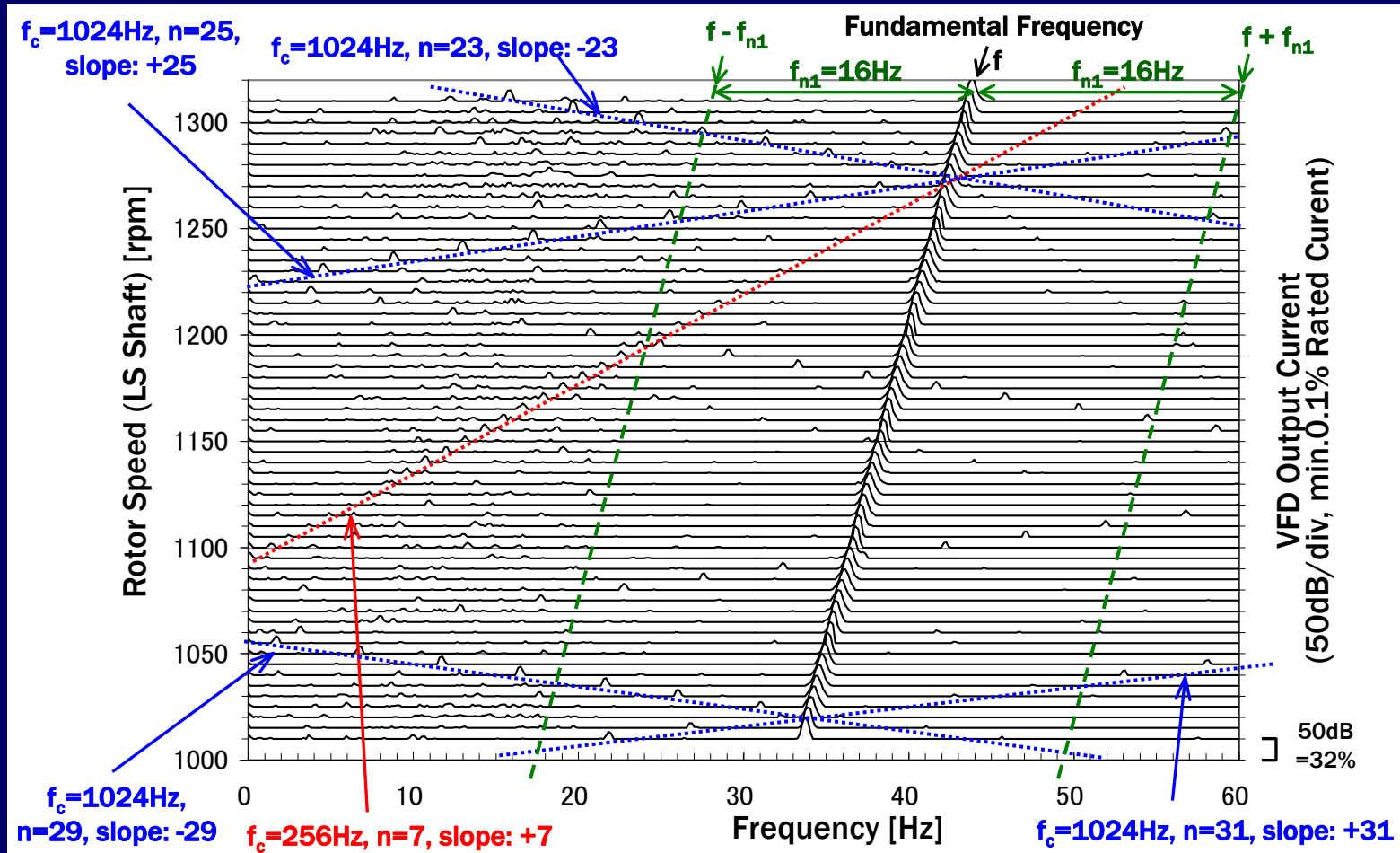


Figure 11. Cascade Plot of VFD Output Current

Pattern of Interharmonic Frequency Component

Inclined Streaks of Interharmonic Frequencies in Shaft Vibration Frequencies and VFD Output Frequencies

- **LS Shaft Vibration Frequency Content**

Difference of Harmonics of Multiples of 6 and Sampling Frequencies of VFD (1024Hz, 256Hz)

- **VFD Output Current Frequency Content**

Difference of Harmonics of Odd Numbers Other Than Multiples of 3 and Sampling Frequencies of VFD
Inclination Opposite to That of Shaft Vibration



Firm Correlation Between Shaft Vibration and VFD Output Current Suspected

Pattern of Interharmonic Frequency Component

Relation of Between Shaft Vibration Frequencies and VFD Output Frequencies

VFD Output Freq. [Hz]:

$$f_{bi} = |f_c - nf| \quad \leftarrow \text{Sideband Frequencies}$$

Shaft Vibration Freq. [Hz]:

$$f_{bt} = |f_c - (n \pm 1)f|$$

f : VFD Fundamental Freq. [Hz]

f_c : Arbitrarily Existing
Constant Freq. [Hz]


n : Positive Odd Integer
Other Than 3

Frequencies of Fluctuating Torque Generated by 3-Phase IM

$$T_e = pM' I_{s_a} I_r' \cdot (9/4) \cdot \sin(2\pi(f_a - f)t + \gamma) \quad f_a : \text{Arbitrarily Existing Current Freq. [Hz]}$$

If $f_a = f_{bi} = |f_c - nf|$

$$T_e = pM' I_{s_a} I_r' \cdot (9/4) \cdot \sin(2\pi(|f_c - (n \pm 1)f|)t + \gamma)$$

 Shaft Vibration Caused
by Excitation of Motor
Torque

Sampling in VFD

VFD Control Loop

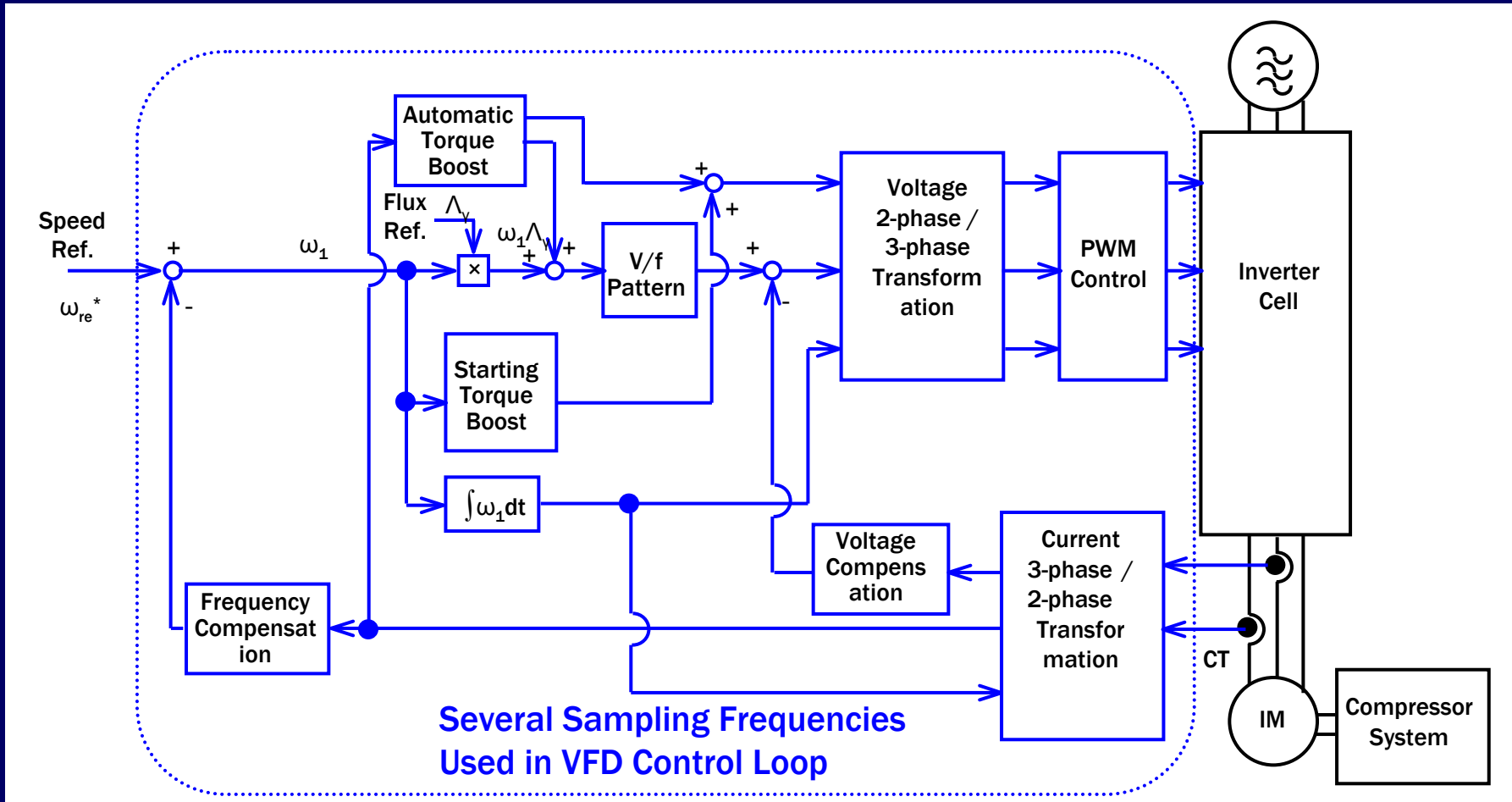


Figure 12. Block Diagram of VFD Control System

Assumed Cause of Sideband

Assumed Cause of Sideband in VFD Output Current

Coarse Pulse of Fundamental Frequency Remained in Current (e.g. Improper Dead Time Compensation)



Harmonics of Odd Numbers Produced by Pulse (Rectangular Wave) of Fundamental Frequency



Harmonics of Multiple of 3 Eliminated in Balanced Three-Wire System



Sideband Frequencies Raised by Modulation between Harmonic Frequencies and Sampling Frequencies

Occurred in VFD Control Loop → Harmonics enhance sidebands.

Other Possible Cause of Sideband

Sideband Frequencies Due to PWM

Sum and Difference of Frequencies of Harmonics of Triangular Carrier Wave (4.8kHz) and Harmonics of Signal Wave (Fundamental Frequency)

Sideband Frequencies Due to PWM Not Observed in This VFD Output Current

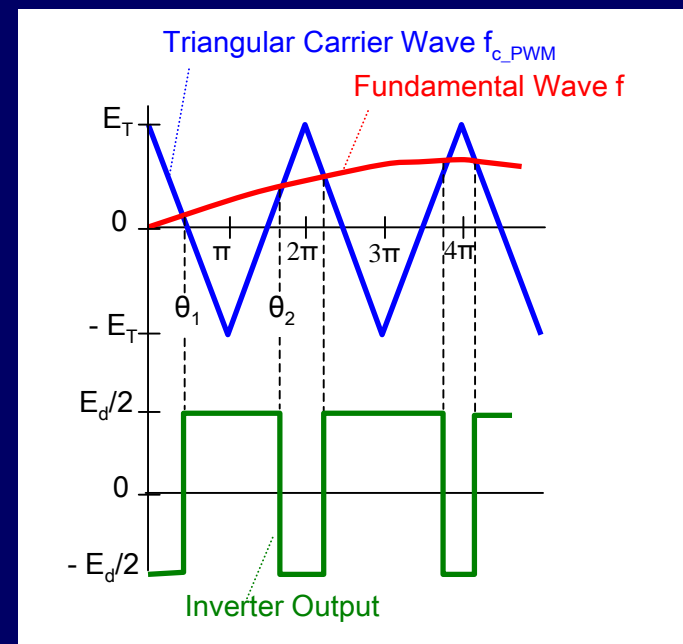


Figure 13. Mechanism of PWM

Concluding Remarks

- **Measurement of torque & current in factory test is important in case VFD characteristics are unknown.**
- **Strength evaluation by lateral-torsional analysis is essential to determine mechanical soundness.**
- **Information of any possible frequencies used in VFD control and the resulting amplitude of torque pulsation should be disclosed in advance by VFD vendor.**
- **Reduction of amplitude of fundamental frequency harmonics would decrease amplitude of sideband frequencies.**