CASE HISTORY#2

COOLING TOWER FAN UNBALANCE

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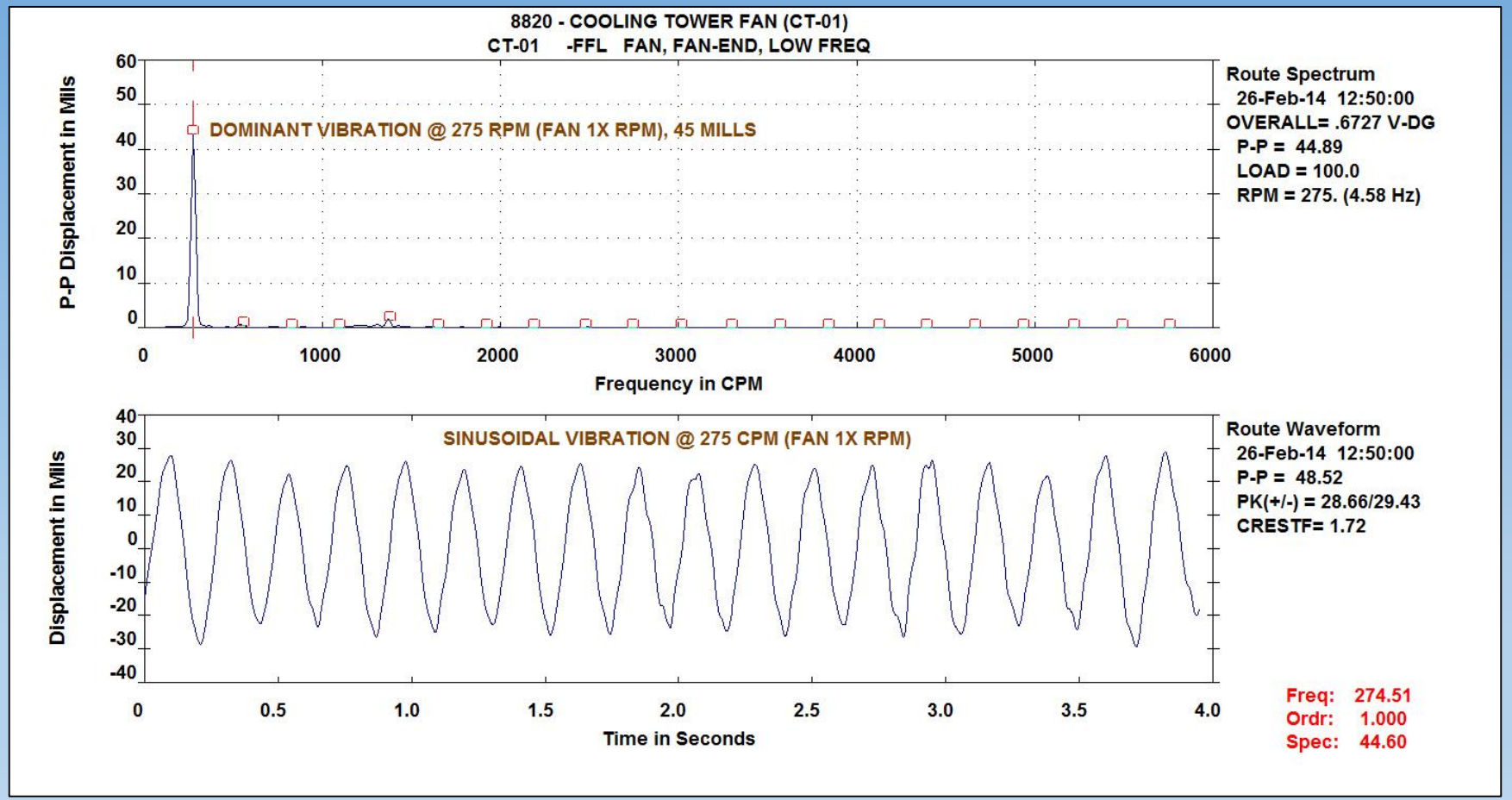
CASE HISTORY#2 – CT FAN UNBALANCE

- A baseline vibration survey was performed on a pair of identical cooling tower fans (cells 1 & 2).
- Analysis of the baseline vibration data identified high vibration levels at one of the two fans (cell 1).
- Spectral data from this fan showed dominant vibration occurring at the fan speed with relatively low levels at both the motor speed and fan blade-pass frequency.
- Both cooling tower fans were 5-blade, belt-driven by a 4-pole motor operating on a variable frequency drive. The ratio between motor & fan speeds was ~ 4.35:1.

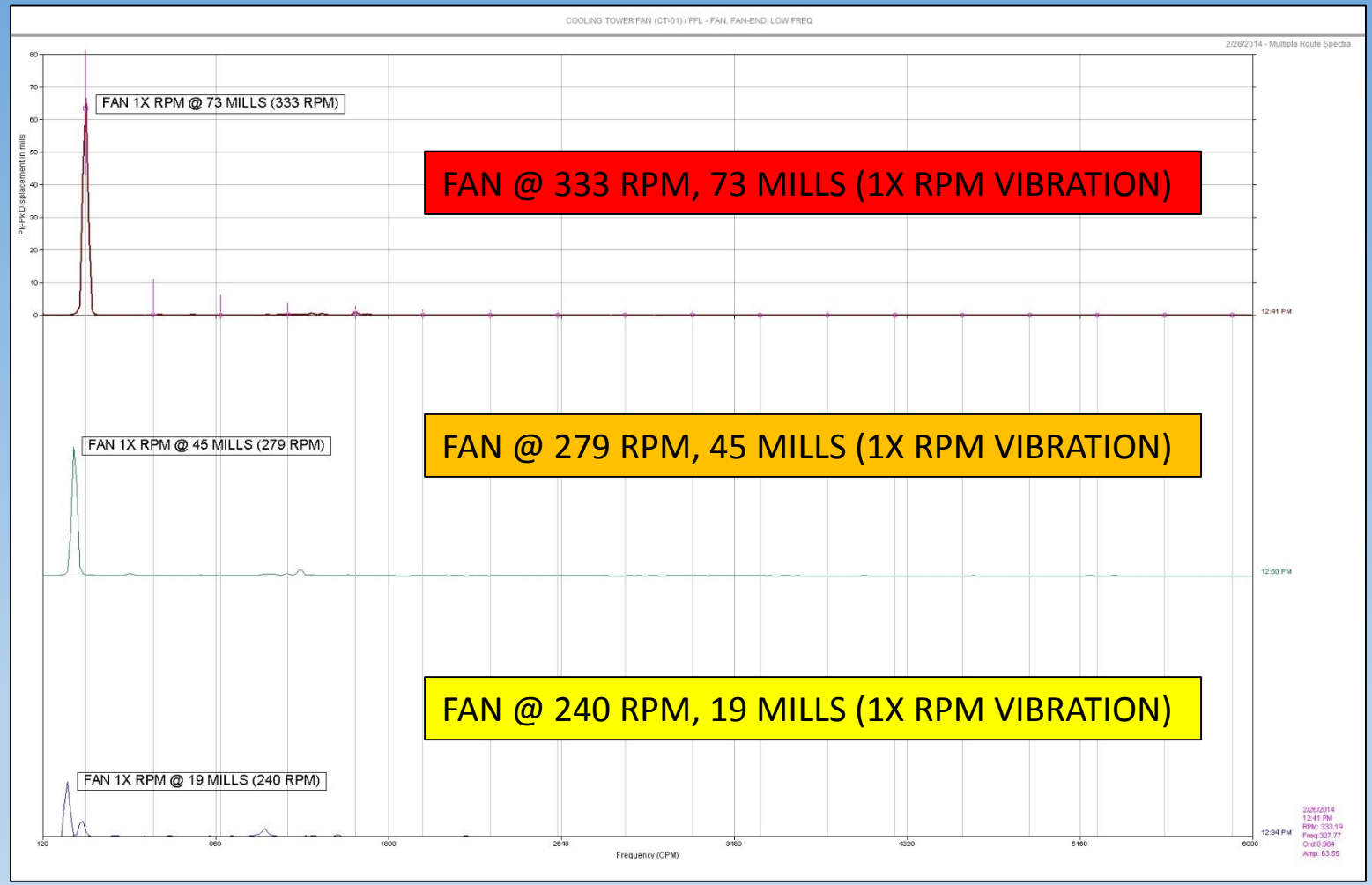
CT FAN SPEEDS & FORCING FREQUENCIES

% FULL SPEED	LINE FREQ (HZ)	MOTOR SPEED (RPM)	FAN SPEED (RPM)	BLADE-PASS (CPM)
10	6	180	41	207
20	12	360	83	414
30	18	540	124	621
40	24	720	166	828
50	30	900	207	1,034
60	36	1,080	248	1,241
70	42	1,260	290	1,448
80	48	1,440	331	1,655
90	54	1,620	372	1,862
100	60	1,800	414	2,069

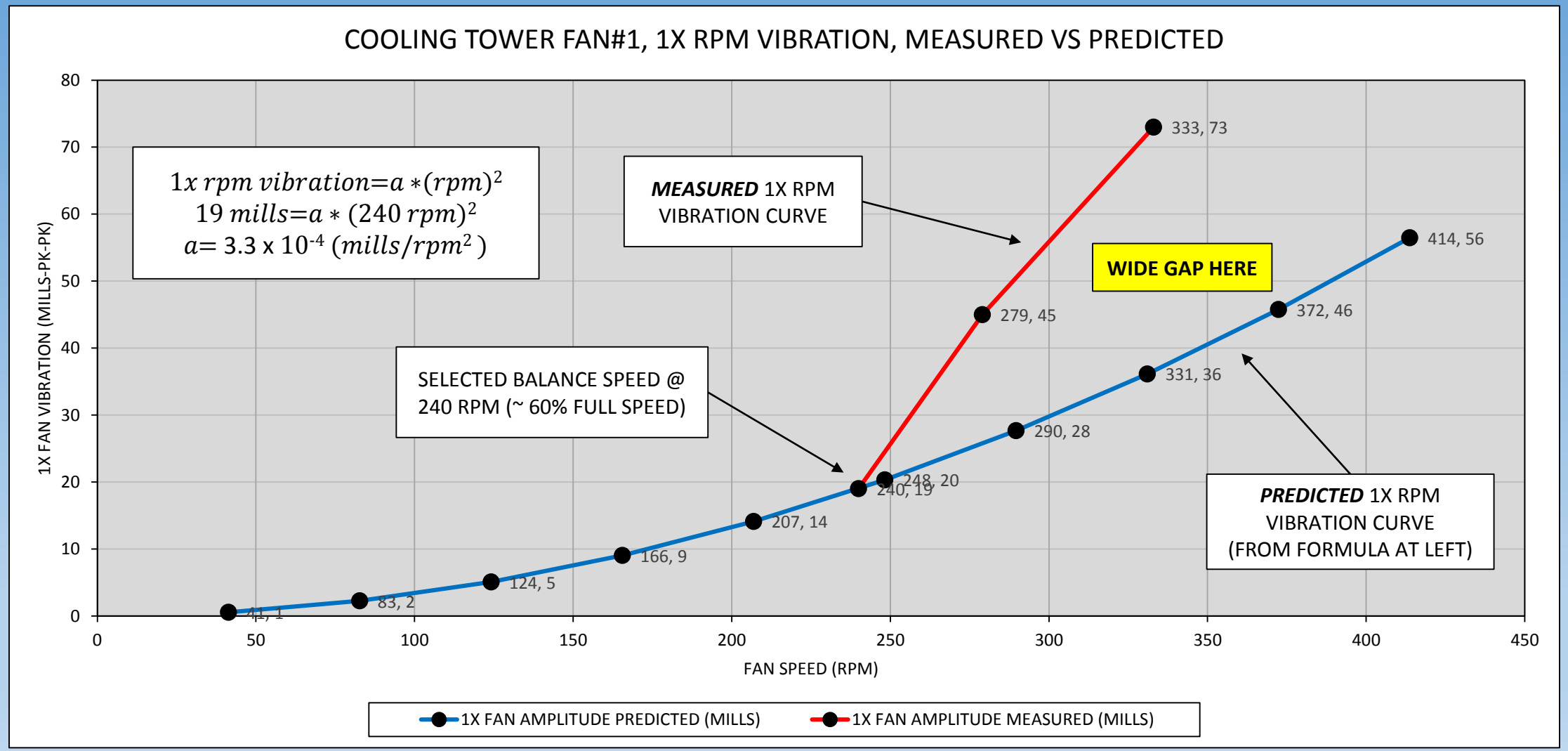
VIBRATION DATA, CT FAN 1 – BEFORE BALANCE



VIBRATION DATA AT VARIOUS SPEEDS



CT FAN 1, 1X FAN VIBRATION VS SPEED - MEASURED VS PREDICTED



FAN BALANCE – NO PHASE BALANCE (4-RUN)

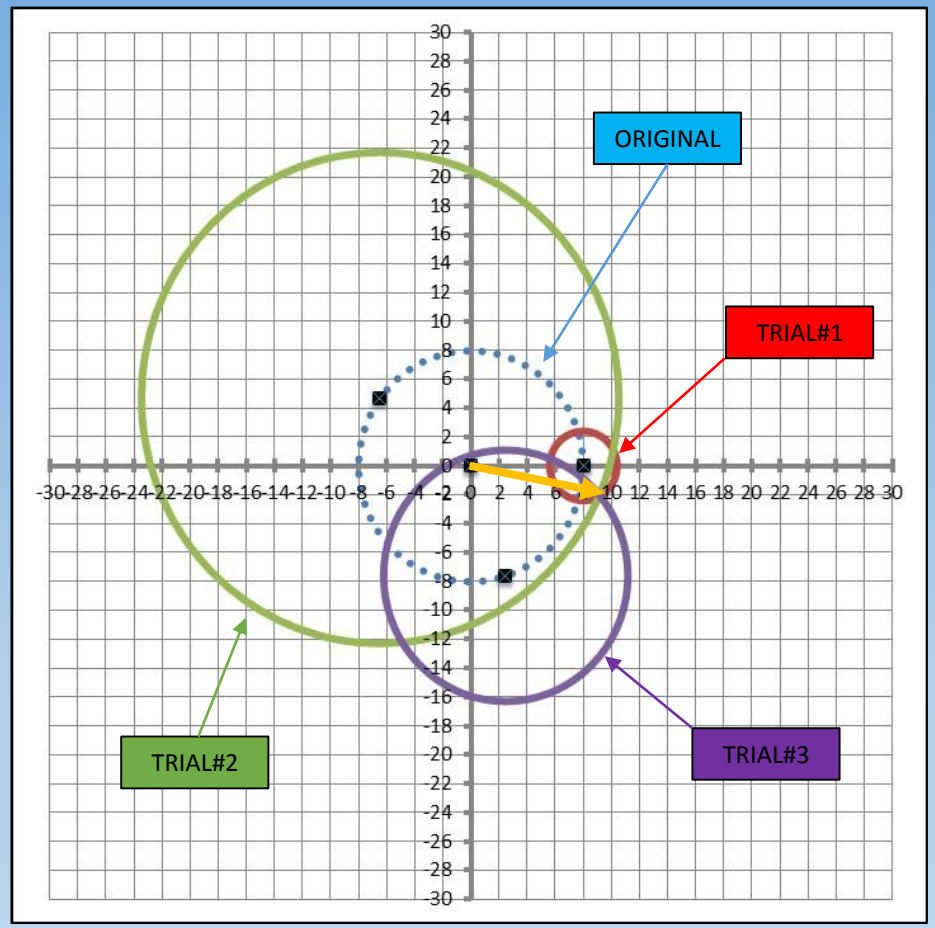
- Instead of the typical balancing process using a tach to gather phase data, a no phase or four-run balancing technique was used.
- This particular process was chosen due to the difficulty of obtaining a consistent & accurate phase angle using the optical & laser tachs available (sunlight causes strong blade-pass frequency influence in light detector of tach).
- As the name implies, the no-phase or four-run balance technique uses no tach for phase and requires a minimum of four runs to obtain a solution.
- The wide gap between measured & predicted fan 1x rpm vibration suggested the presence of a resonance at the higher speeds. For this reason, a fan balance speed of ~ 200 rpm (50% max) was selected.

NO PHASE (4-RUN) BALANCING TECHNIQUE

- During each run of the balancing technique, the fan 1x rpm vibration level is measured and recorded.
- **1st Run** – Original or reference run without any trial weights. Draw circle with *radius* equal to 1x rpm vibration level centered at origin.
- **2nd Run** – First trial run with known trial weight placed at 0 deg, reference point (center at radius of original circle at 0 deg).
- **3rd Run** – Second trial run with known trial weight placed at known location (ideally 120 deg ± 30 deg from reference point).
- **4th Run** – Third trial run with known trial weight placed at known location (ideally 240 deg ± 30 deg from reference).
- It is to your advantage when selecting trial weight positions to keep their ***average*** separation as great as possible (ideally 120 deg, but certainly > 90 deg).
- For this case, our 5-blade fan gave a separation of 72 deg per blade, so our trial runs were separated by 144 deg (2-ea blades).

BALANCING DATA & PLOT (200 RPM)

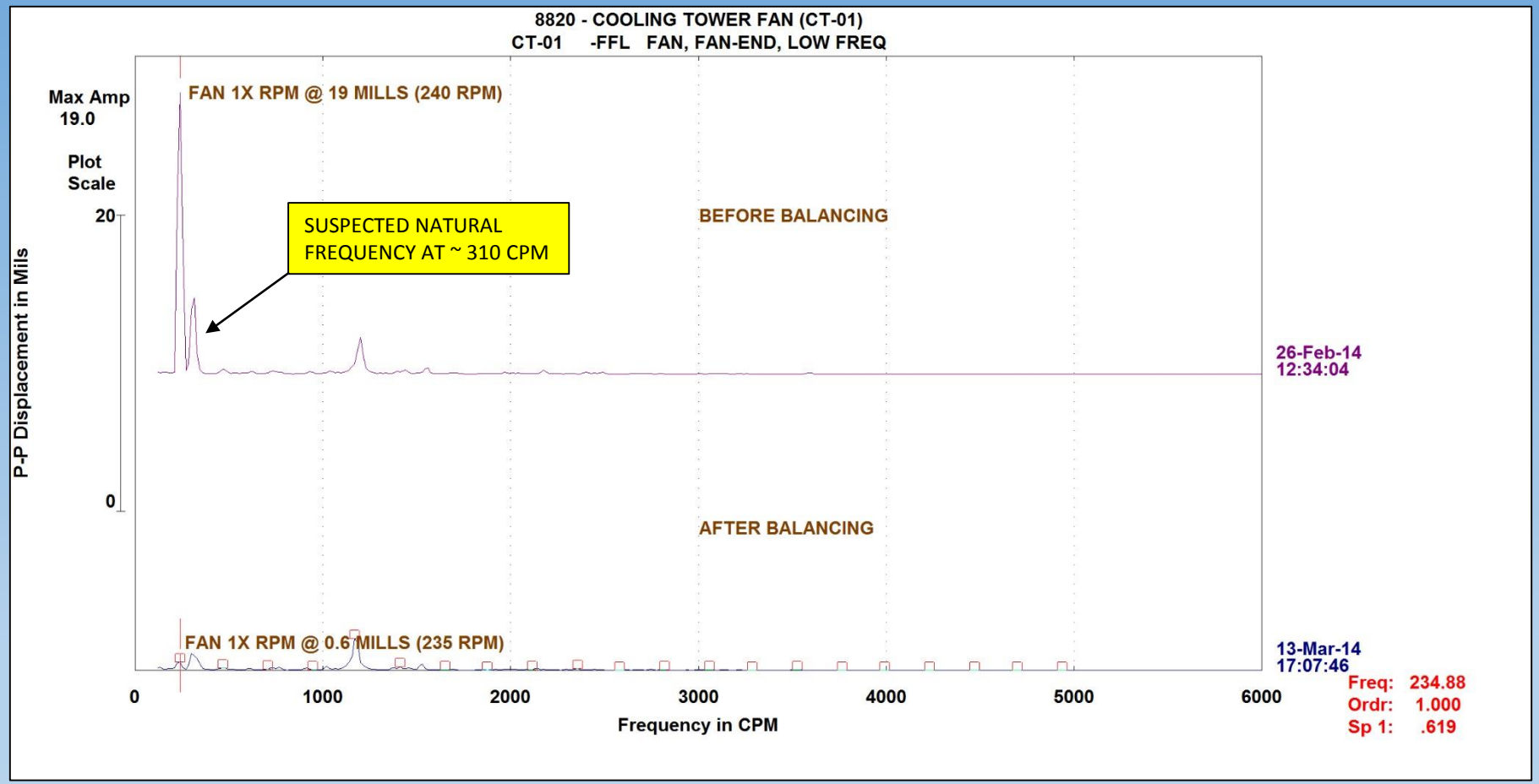
Condition	Fan 1x rpm Vibration (mills-pk-pk)	Trial Weight	
	Magnitude	Amount	Location
Original Vibration	**8**		
Trial#1	2.4	207	0
Trial#2	17	207	144
Trial#3	8.7	207	288
Trial Vector	9.3	n/a	347
Correction Weight		178	347
Applied Weights (split)		160	0
		42	288
Final Vibration	**0.5**	**94% Reduction**	
Balance Sensitivity	24 g/mill		



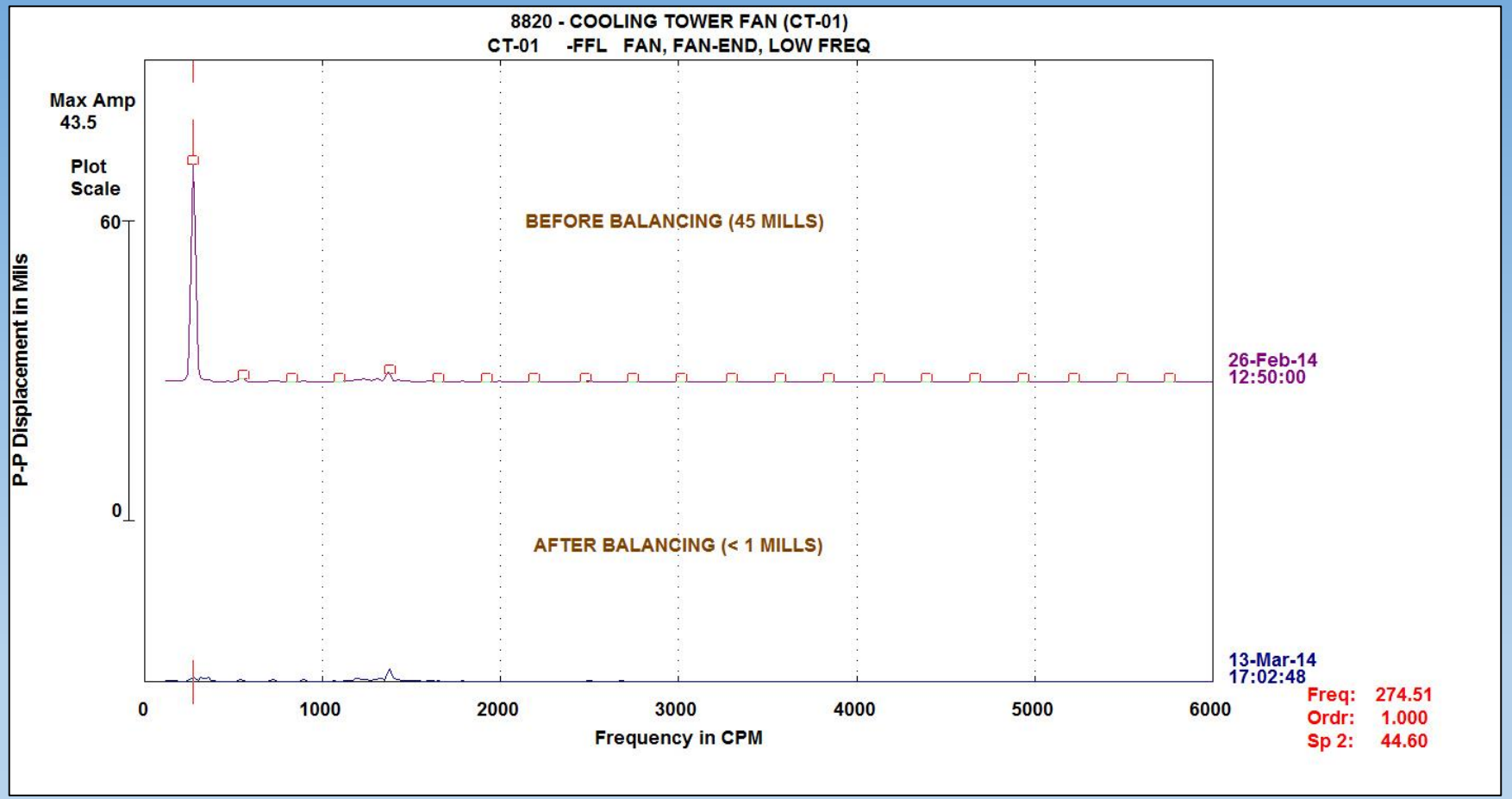
CORRECTION WEIGHT = TW * ORIGINAL / TRIAL VECTOR = 207 * 8 / 9.3 = 178

BALANCE SENSITIVITY = TW / TRIAL VECTOR = 207 / 9.3 = 24 G/MILL

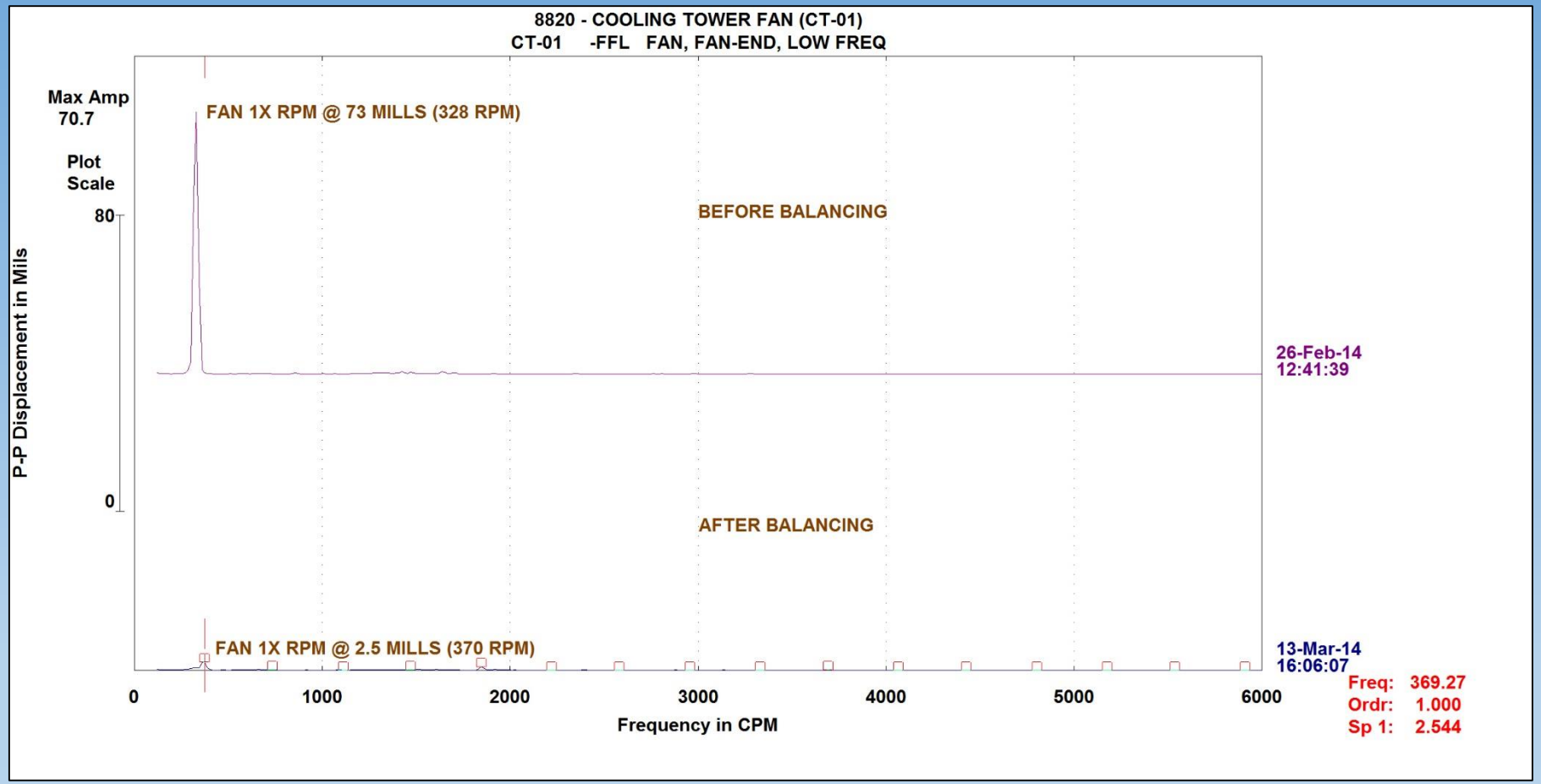
SPECTRAL DATA BEFORE & AFTER BALANCING (240 RPM)



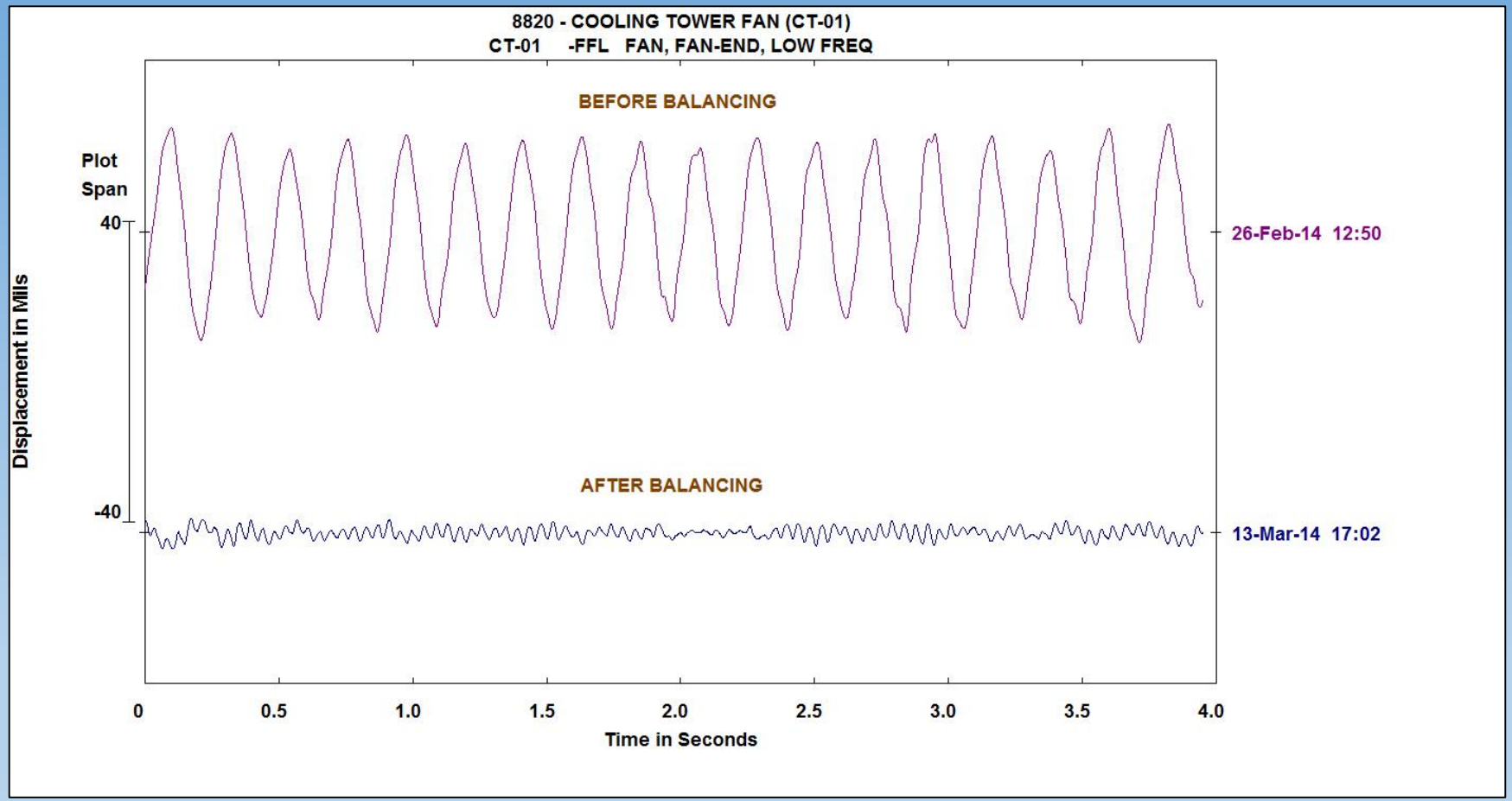
SPECTRAL DATA – BEFORE & AFTER BALANCING (275 RPM)



DATA BEFORE & AFTER BALANCING (~ 330 RPM)



WAVEFORM VIBRATION DATA – BEFORE & AFTER BALANCING (275 RPM)



CONCLUSIONS – CASE HISTORY, CT FAN UNBALANCE

- Thru balancing of the cooling tower fan we were able to reduce the 1x rpm fan vibration levels significantly (from 72 mills to 2.5 mills at ~ 330 rpm or from 19 mills to 0.6 mills at ~ 240 rpm).
- A likely structural natural frequency was identified at ~ 310 cpm.
- From our understanding of the coastdown data and due to the presence of a VFD drive, we were able to select a balancing speed (200 rpm) sufficiently away from the natural frequency (310 cpm) to obtain a good balance.
- Even though the natural frequency remained, we will now send the amplifier much lower dynamic force and thus will have much less vibration.
- Eventually, a modal analysis of the fan structure would be necessary if we wanted to pinpoint the source of the resonance and eliminate it through stiffening, etc.