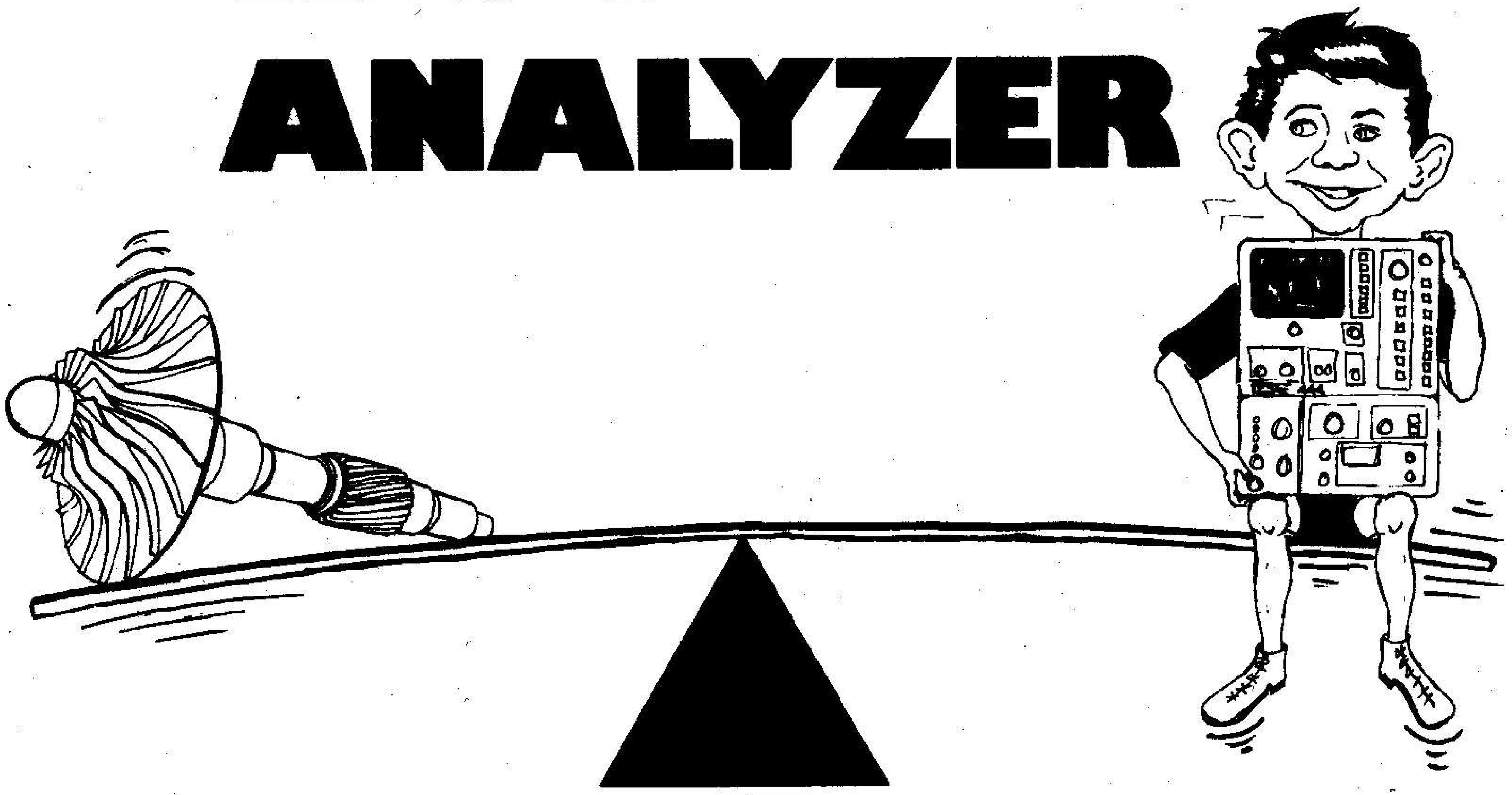
APPLICATION NOTE 11
October 1977

how to...

BALANCE with your REAL TIME ANALYZER



INCLUDING METHODS, DERIVATION OF EQUATIONS, EXPERIMENTAL EXAMPLE AND *A PROGRAM FOR YOUR POCKET CALCULATOR.*

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MODEL 444 MINI-UBIQUITOUS®
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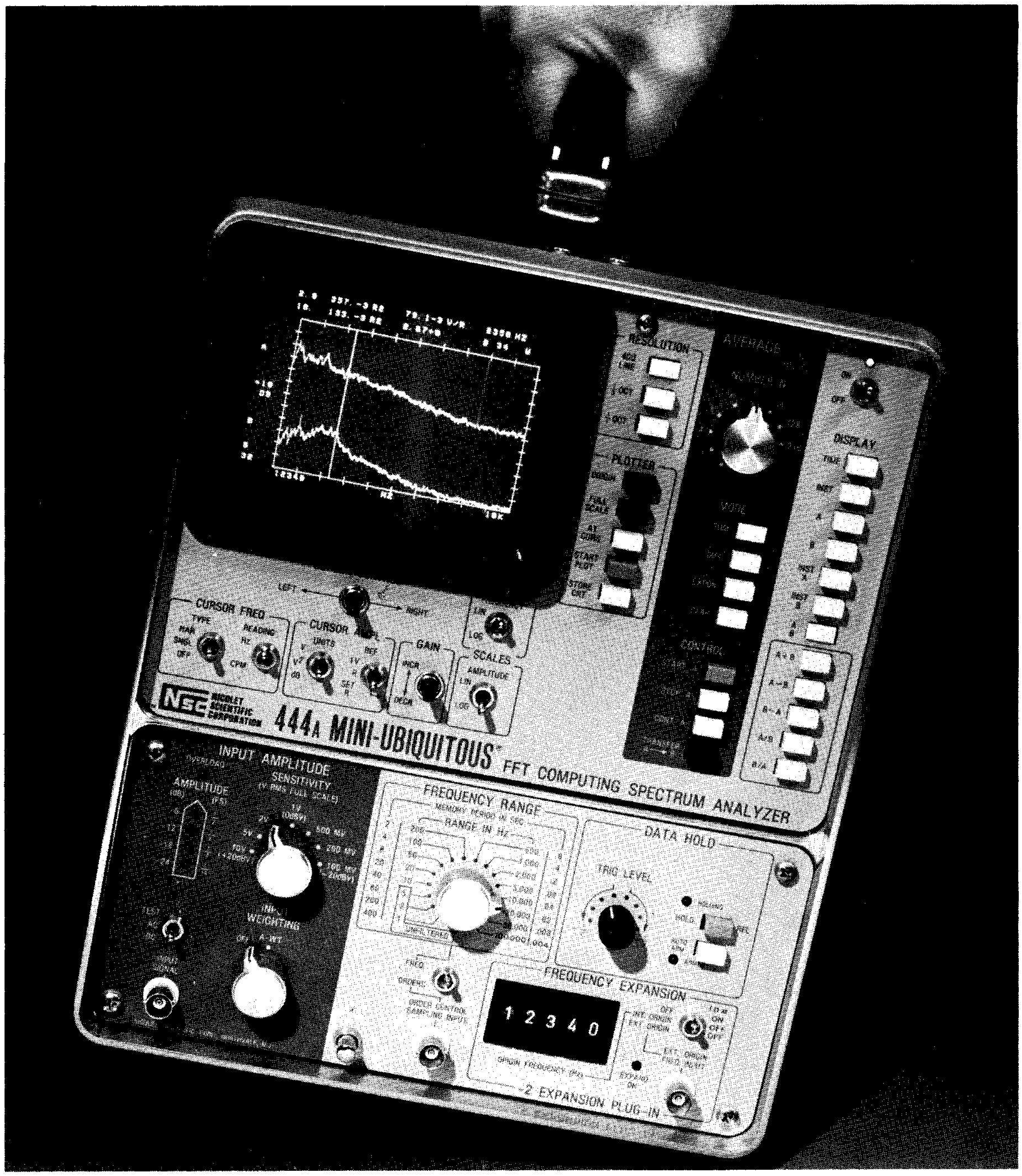


Key Features for Balancing

- **PEAK AVERAGING MODE** permits measurement of "coast down" spectra—provides balance information in minimum time when machine speed cannot be tightly controlled.
- **POWER AMPLITUDE READOUT** at cursor (volt^2 or engineering unit^2) provides data in the form required for balance computations.
- **DUAL MEMORY AVERAGER** provides immediate before/after comparison to validate balance.
- **COMPANION DIGITAL RECORDER** (Model 144) stores 1500 spectra with complete annotation—permits immediate in-field comparison of current machine status with prior condition.

WHAT IS BALANCING?

An incalculable number of rotating machines exist today, ranging from small electric motors to gigantic turbo-machines. Manufacturers of such equipment are careful to assure inertial symmetry of all rotating elements about their axes of rotation. Often, "shop balancing" techniques are used to assure this symmetry. Shop balancing is conducted on specially designed balancing machines and involves only the rotating member of the end product. The balance machine provides power to rotate the item being balanced and electronic instrumentation to measure reaction forces caused by inertial dissymmetries. Such measurements allow the manufacturer to refine the balance of the machine by adding or removing mass-offset to minimize these reaction forces.

Installed machinery requires periodic maintenance, frequently aided by electronic diagnosis. Such diagnoses often disclose unbalance, requiring *in-field* correction. *In fact, field balancing is one of the most common corrective measures applied to operating machinery to assure their continuous, trouble-free operation.* The need for field balancing can be caused by components shifting, parts wearing or corroding, or a build-up of material on rotating components.

Field balancing is a corrective measure. It minimizes the reaction forces experienced by the non-rotating parts of the machine at the frequency of rotation. This actually implies a more general concept than simply re-establishing the inertial symmetry of the rotor. Field balancing attempts to *minimize the stresses in the non-rotating components, sometimes at the expense of increasing the stresses within the rotating components.* This is a desirable trade-off because stresses experienced as operating-speed rotative loads by the fixed systems are experienced as static loads by the rotating systems. The damage producing potential of static loads is, in general, less than that of oscillatory loads.

In practice, field balancing is accomplished by measuring vibration in the *fixed* system of the machine. These measurements may be made using either *seismic* transducers (sensing velocity or acceleration) or proximity probes measuring the *relative* displacement between the rotor and the fixed system of the machine. These measurements are used in conjunction with balancing trial weights added to the rotor to introduce deliberate and known changes in the balance of the machine. A series of vibration measurements provides sufficient information to guide the installation of corrective rotor weights to assure minimum fixed-system reaction forces.

Field or trim balancing problems may be divided into three major classes. These are the single plane, two plane, and multi plane problems. The methods developed in this paper deal with the *single plane* balance problem.

Single plane balancing is the act of correcting a single rotating unbalanced force in a machine. This type of balancing is validly applied to rotor systems where the off-axis mass concentration is essentially contained within a single plane. Examples of this type of machine include large cooling fans, propellers, and single rotor steam turbines. More complex rotating equipment *can* be balanced using single plane techniques (with varying degrees of success) depending upon the geometric and dynamic characteristics of the machine being balanced.

Two plane balancing *attempts* to correct for the distributed unbalance of a rotor. The term "two plane" refers to the fact that correction weights are added to *two* specific planes of the rotor. It does *not* imply, however, that the unbalance is restricted to those two locations. The two plane balance attempts to react the radial forces generated by the rotor by introducing two additional radial forces at the balance planes. This procedure allows nulling a rotating force *and moment* occurring at the rotating speed of the machine. Frequently, such corrections will reduce the once-per-turn vibratory loading in the fixed system, while *increasing* the internal static stresses within the rotor. Two plane balancing assumes that the rotor behaves as a rigid body. That is, the assumption is made that the distributed unbalance of the rotor can be treated as an equivalent pair of discrete unbalances occurring in the balance planes.

Multi-plane balancing reduces the *distributed* unbalance within a long rotor. Corrections are applied at every plane within the rotor with significant off-axis mass distribution. This procedure is normally used in the case of long, supple rotors. Successful multi-plane balancing requires knowledge of the rotor's mode shapes (as a function of speed) and the dynamics of the fixed system components. Multi-plane balancing is not normally performed in-field.

WHY BALANCE WITH AN RTA?

Many instruments have been developed specifically to facilitate trim balancing. These instruments vary in their method of operation, precision, ease of use, and reliability. Trim balance analyzers are just a part of the instrumentation considered "standard" to the maintenance of rotating machinery. *In general, trim balance analyzers do not provide the information necessary to discriminate between the need for balancing and the need to take other corrective actions.*

Discriminating between vibration induced by unbalance, misalignment, component failure, mechanical looseness, or bearing-induced rotor instability is most readily made with a *real-time spectrum analyzer* (RTA). This instrument often gives the first indication of the need to trim balance an operating machine. (Unfortunately, it is not uncommon to recognize this need when a trim balance analyzer is not available!)

Trim balance analyzers fall into two general categories: those which require a once-per-turn reference signal, and those that do not. Those trim balance analyzers which do not require a *once-per-turn* or "key phaser" signal *do* require an installed *degree-wheel* that can be observed with a strobe light during operation of the machine. In many situations, neither of these facilities may be available. *Balancing with a real-time analyzer does not require a key phaser signal nor does it require a degree-wheel on the machine.*

Trim balance analyzers provide *magnitude* and *phase* information. Those instruments which use a key phaser signal provide an analog (or digital) readout of the *phase angle* between the vibration and key phaser signals at the rotative speed of the machine. Those trim balance analyzers that do not require a key phaser signal provide the phase information by causing a strobe light to "freeze" the position of the machine's degree-wheel at a position indicative of this phase angle. Precise detection of phase angles is a far more difficult task than the precise determination of magnitude information. Operating dynamics of the machine (such as surge or hunting), as well as economic constraints on the design of the phase detector used in these instruments, often compromises the precision of phase data provided by trim balance analyzers. *In contrast, balance procedures utilizing an RTA depend only on magnitude information.* Today's real time analyzer is an extremely precise amplitude-at-frequency discriminator.

Trim balance analyzers require multiple runs of the machine to be made *at exactly the same operating speed.* Deviations in the operating speed, from run to run, can cause serious errors in computing the amount and placement of corrective trim weights. This is particularly important when attempting to balance machines whose speed cannot be well regulated. Here the real-time analyzer offers a unique advantage. The information required to balance a machine can be derived *either* from single speed operation or from a series of spectra acquired during the "coast down" of the machine as it is being shut off. Obtaining balance information during coast down *also* allows multiple computation of the required trim weight and its location to be made from a *single set of data.* This permits computational averaging to be used, improving the chances of getting the right balance correction made on the first try!

In summary, the advantages of using an RTA to balance are:

1. It can clearly descriminate the need to balance.
2. Neither key-phasor not degree-wheel are required.
3. Precise machine speed control is not necessary.

HOW IS IT DONE?

Single plane balancing using an RTA requires four vibration measurements. These measurements can be made with an accelerometer, velocity pickup, or shaft displacement probe. The transducer signal is spectrum analyzed using the RTA, and the amplitude at the balance speed (frequency) is read. One measurement is made from the machine in its initial (unbalanced) condition. Three subsequent measurements are made (using the same transducer and at the same frequency) with a trial balance weight attached to the machine's rotor. These three runs differ only in the angular placement of the trial weight. Computations are performed on the four vibration readings, yielding the *angular location* of the existing unbalance and the *ratio* of that unbalance to the trial weight used.

Assume that all four measurements can be made with the machine operating at a single, fixed, running speed. This operating speed shall be referred to as the balance speed. Further assume that an unbalance of an unknown amount, U, exists. Both the magnitude and angular location of the unbalance must be determined, so that corrective action may be taken.

The first measurement is made from the machine in its unbalanced state. A vibration signal (from a single, fixed-position transducer) is measured and spectrum analyzed. The amplitude of the spectrum is measured at the balance speed (frequency). Assume this amplitude is R_0.

The machine is stopped and a reference angular position is chosen and marked. A balance trial weight of mass-offset, T, is attached at the reference position. The existing unbalance to be corrected is at an unknown angle, ϕ, from the trial position. The machine is again run (at the balance speed) and the rotative-speed vibration amplitude, R_1, is measured.

The machine is stopped and the weight is removed and repositioned to a new angular position, δ, from the reference mark. *The direction in which the weight is moved in this step defines the positive angular direction.* The machine is run at the balance speed and a rotative-speed vibration level, R_2, is measured.

Again, the machine is stopped. The trial weight is moved to a position of $-\delta$ from the reference mark. The machine is run at the balancing speed and a rotative-speed vibration level, R_3, is measured.

The ratio of the unbalance to the trial unbalance (U/T) and its angular position, ϕ, (measured from the reference mark) are computed from the following equations:

$$\frac{U}{T} = \frac{1}{\sqrt{X^2 + Y^2}} \qquad \phi = \mathrm{Tan}^{-1}\frac{Y}{X}$$

where

$$X = \frac{2R_1^{\,2} - R_2^{\,2} - R_3^{\,2}}{4R_0^{\,2}\ (1 - \cos\delta)}$$

$$Y = \frac{R_2^{\,2} - R_3^{\,2}}{4R_0^{\,2}\sin\delta}$$

Derivation of the preceding equations is presented in Appendix A. A pocket calculator program (for the HP25 and 25C) implementing them is presented in Appendix B. Although a graphical solution (ref. 4) is possible, its use is discouraged because of attendant lack of precision.

Correction action is taken by applying a permanently installed balance weight of magnitude, U, at a position 180° away from the angular position, ϕ. Alternatively, a mass of U may be removed from the machine at the angular position, ϕ. The precision of computation is optimized if the three trial weight positions completely span the circumference of the balance plane. Thus, $\delta = 120°$ provides the optimum measurement situation. In practice, δ can always be selected between 90° and 120°, which will yield satisfactory precision. Selecting a smaller angle may reduce the precision of the balance weight and angular position computations.

HOW IS "COAST DOWN" DATA USED?

In many situations it is not possible to run the machine at a consistent, fixed balance speed. In this situation the PEAK AVERAGING MODE of a modern RTA, such as the 444 shown in Figure 1, can be used most effectively. The peak averaging mode captures the largest amplitude encountered at each frequency of the spectrum. If a spectrum analysis is started, using PEAK AVERAGING MODE, and the machine is then "shut off" and allowed to coast to a stop, the peak stored average will contain the amplitude vs. operating speed trace of the once-per-turn unbalanced component. (This assumes, of course, that the machine is basically "healthy" so that the rotative-speed unbalance vibrations are *dominant* in the signature during the coast down, while the average proceeds.)

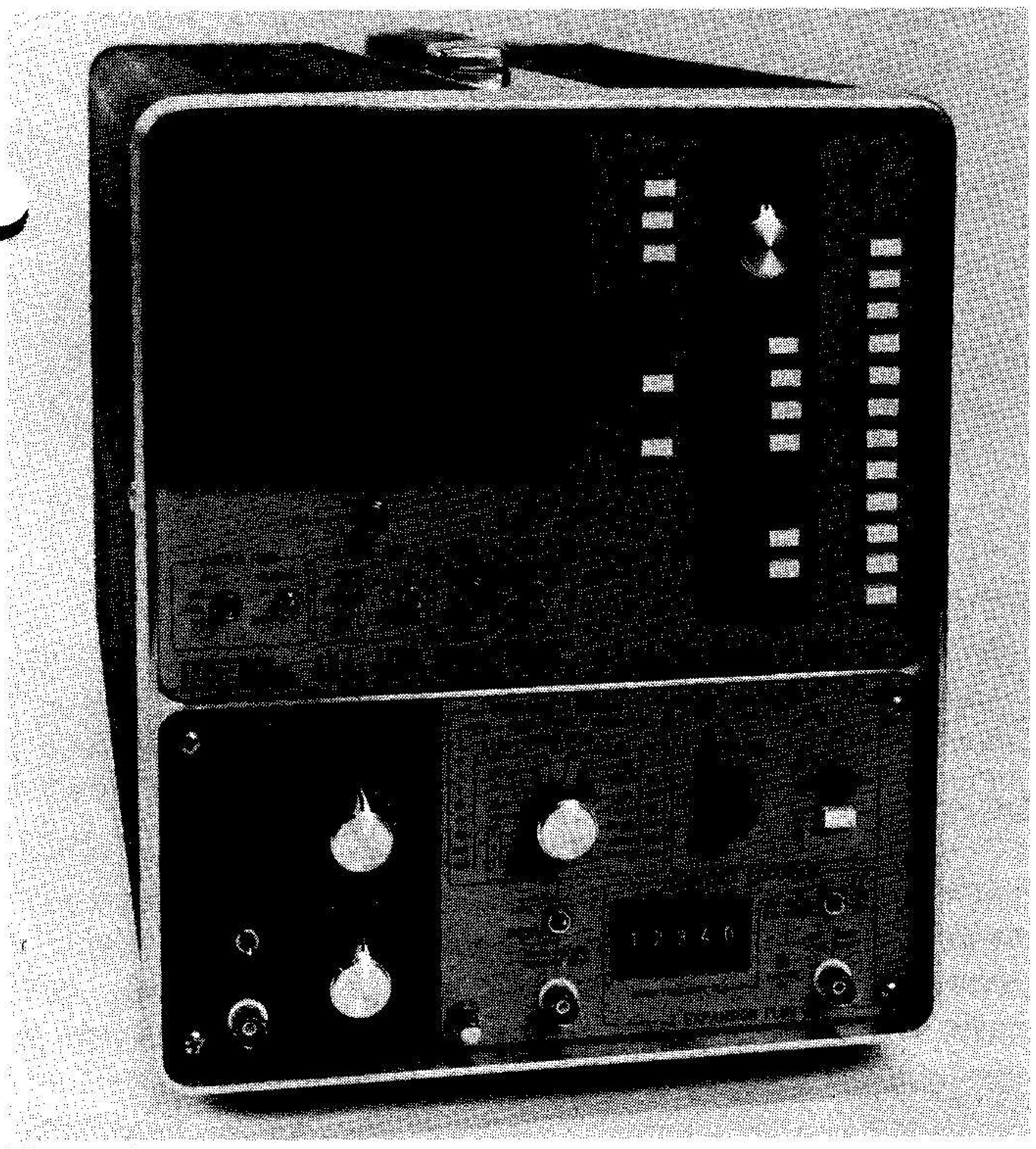


Figure 1

The peak stored average is a "smeared" version of the spectrum that would result from tracking the first order shaft speed (using an RTA and tracking adapter) during a slow speed sweep of the machine. The "smearing" will result whenever the instantaneous rate-of-change of frequency (with respect to time) exceeds the square of the RTA's resolution bandwidth. Although it affects the amplitude of the resultant spectrum, the "smearing" will only affect the precision of the balance measurement if it is *inconsistent.* If the machine can be stopped and allowed to coast down in an *unloaded or consistently loaded* manner, the "smearing" will be consistent from run-to-run and its effect will cancel out. The consistency of a coast down may be verified by measuring the same vibration signal twice during two consecutive coast downs. If consistent signatures result, the coast down data will be adequately precise to permit accurate balancing.

There are two advantages of "coast down" spectra. First, it eliminates the need to "fight" with the machine's speed control system to achieve a consistent balance speed from run-to-run. This can be a major time saver. Second, and perhaps most important, *the entire spectrum may be used for the balance computation.* That is, each frequency point in the coast-down spectrum may be thought of as a separate balancing speed. Hence, it is possible to apply the balance computation to any frequency location within the spectrum. This makes it possible to perform the computation on data above, below, and at the critical speed or to average the results of a sequence of balance computations across an entire spectrum. Since balancing is, at best, a probabilistic business, this permits improving the statistics associated with the balance computation without resorting to multiple balance speed runs.

WHERE DO I GO WHEN I CAN'T GO THERE?

Successful correction of unbalance occurs when the proper correction weight is placed at the proper angular position. Unfortunately, the "proper" angular position may not be convenient or even exist! For example, when balancing multi-bladed fans, it is not uncommon to find that the balance weight should be placed *between two blades!* Obviously, it is not possible to attach a trim weight or drill a hole in clear air! Hence, the need for "weight splitting".

"Weight splitting" is the substitution of two trim weights for one. The two trim weights are placed at angular locations that are physically convenient (i.e., on the fan blade). The masses of the two weights are selected so that the trim forces they develop have a vector resultant whose magnitude and direction corresponds to the "proper" solution.

Assume that a balance solution requires attachment of a mass, U, at an angular position ϕ. Further assume that it is not possible to attach the weight at this angular position but that attachments at angular positions, $\phi + \alpha$ and $\phi - \beta$, are possible. An effect exactly equivalent to attaching a mass, U, at angular position, ϕ, can be achieved by attaching a mass, U_1 at angular position $\phi + \alpha$ and a mass, U_2, at position, $\phi - \beta$, where:

$$U_1 = U \frac{\sin \beta}{\sin (\alpha + \beta)}$$

$$U_2 = U \frac{\sin \alpha}{\sin (\alpha + \beta)}$$

A derivation of these equations is presented in Appendix A. The calculator program presented in Appendix B contains the "weight splitting" computations.

THE PROOF OF THE PUDDING!

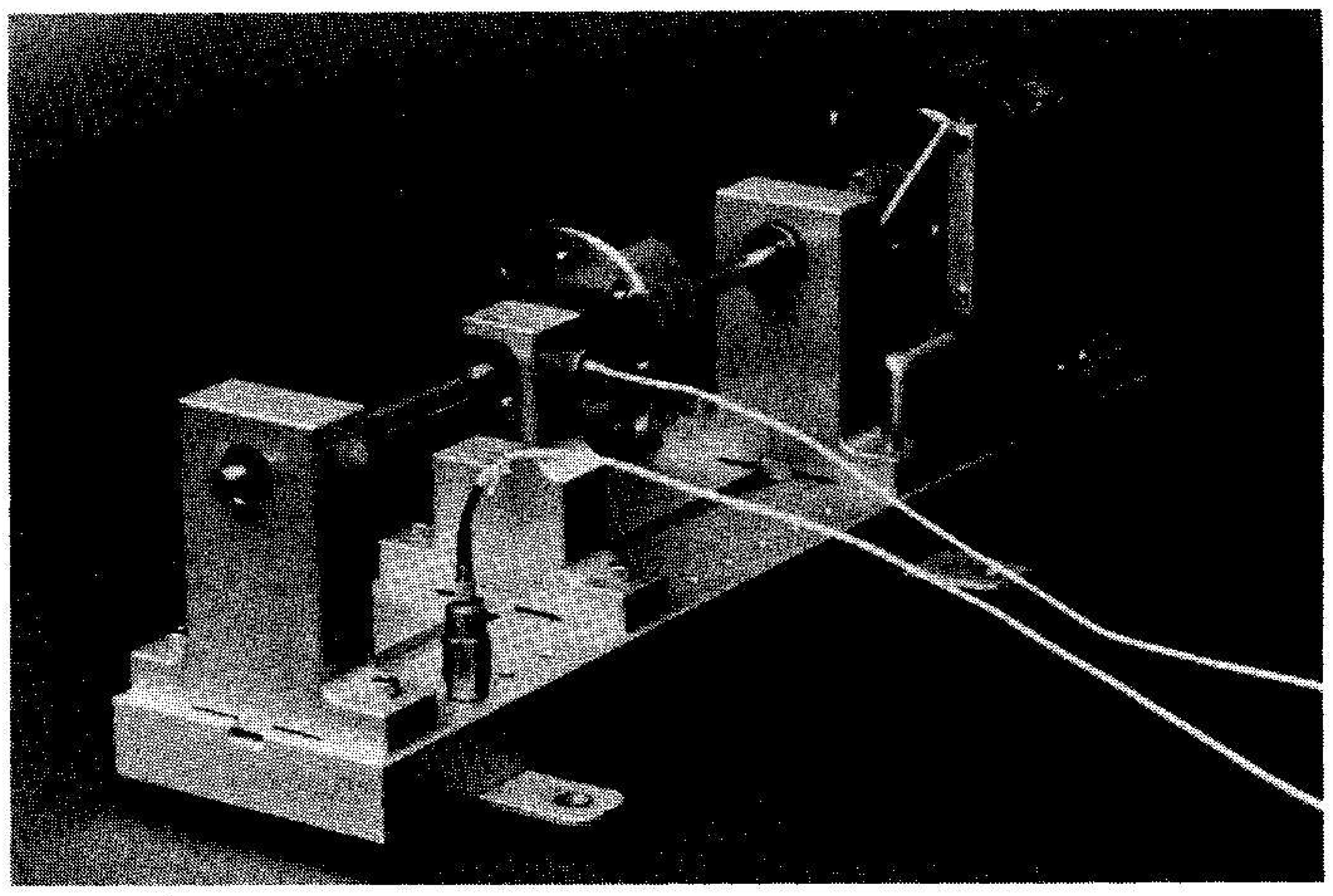


Figure 2

Figure 2 illustrates a small rotor model used to test the balancing procedure. This apparatus consisted of a flexible shaft, supported in plain bearings with a single substantial rotor mass at its center. The rotor is driven through a flexible rubber coupling by a small universal motor (with variable speed controller). A vertically oriented accelerometer on the base of the machine and a horizontally oriented displacement probe viewing the rotor shaft provide vibration signatures. The accelerometer (PCB Model 302A04) has a sensitivity of 141. mv/g while the displacement probe (Bentley Nevada Model 306) provides a sensitivity of 200 mv/mil.

Although seemingly symmetric, the rotor exhibited considerable unbalance when first assembled. It was necessary to add a trim weight of approximately 2.5 grams, 30 millimeters off the axis of rotation before proceeding with the experiment. This trim weight (2 allen head set screws) may be seen at the nine o'clock location on the rotor in Fig. 3. This initial balance was obtained based on data from the accelerometer, using a constant balance speed of 3000 RPM, the procedures previously discussed, and the program documented in Appendix B.

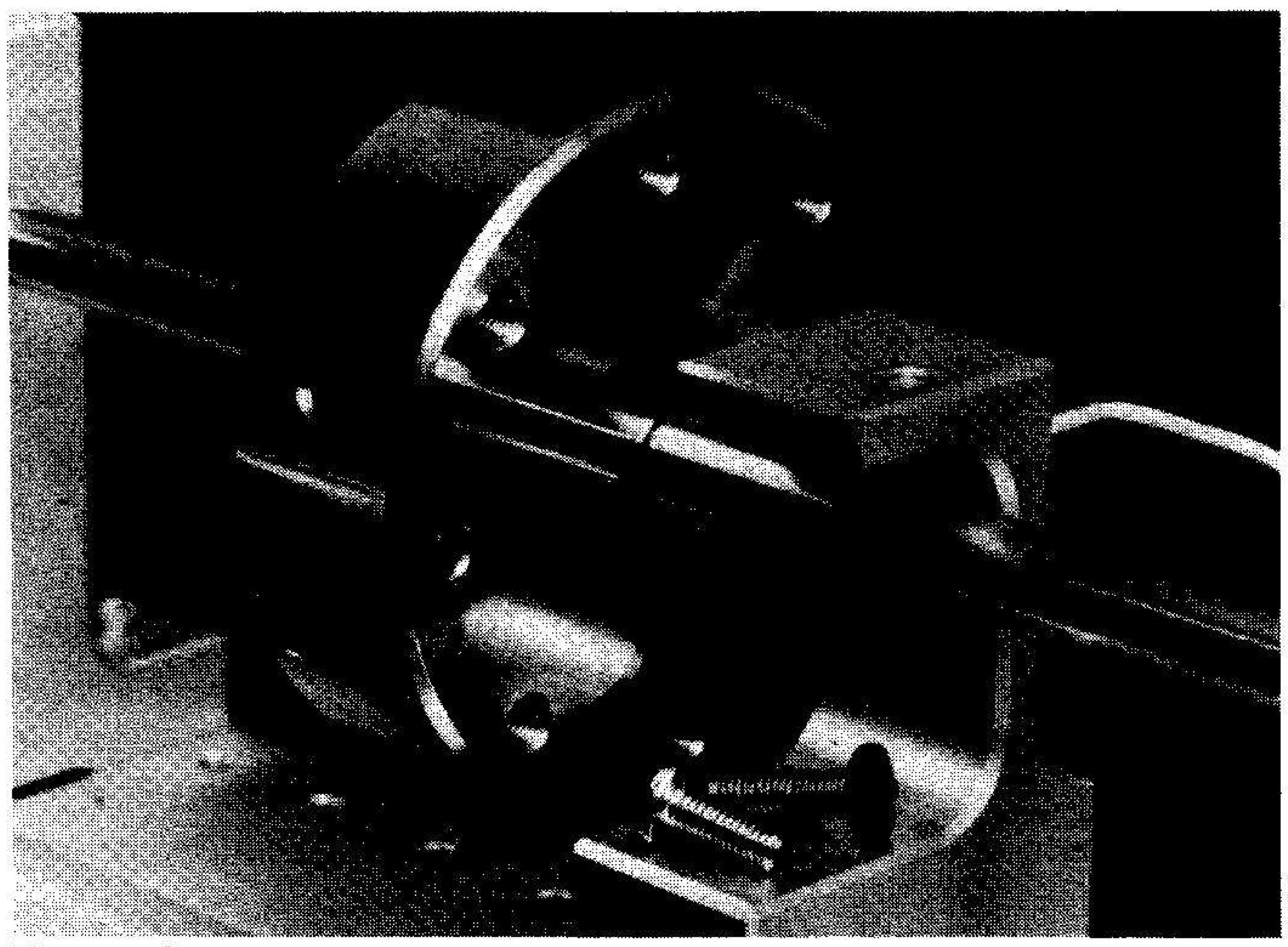


Figure 3

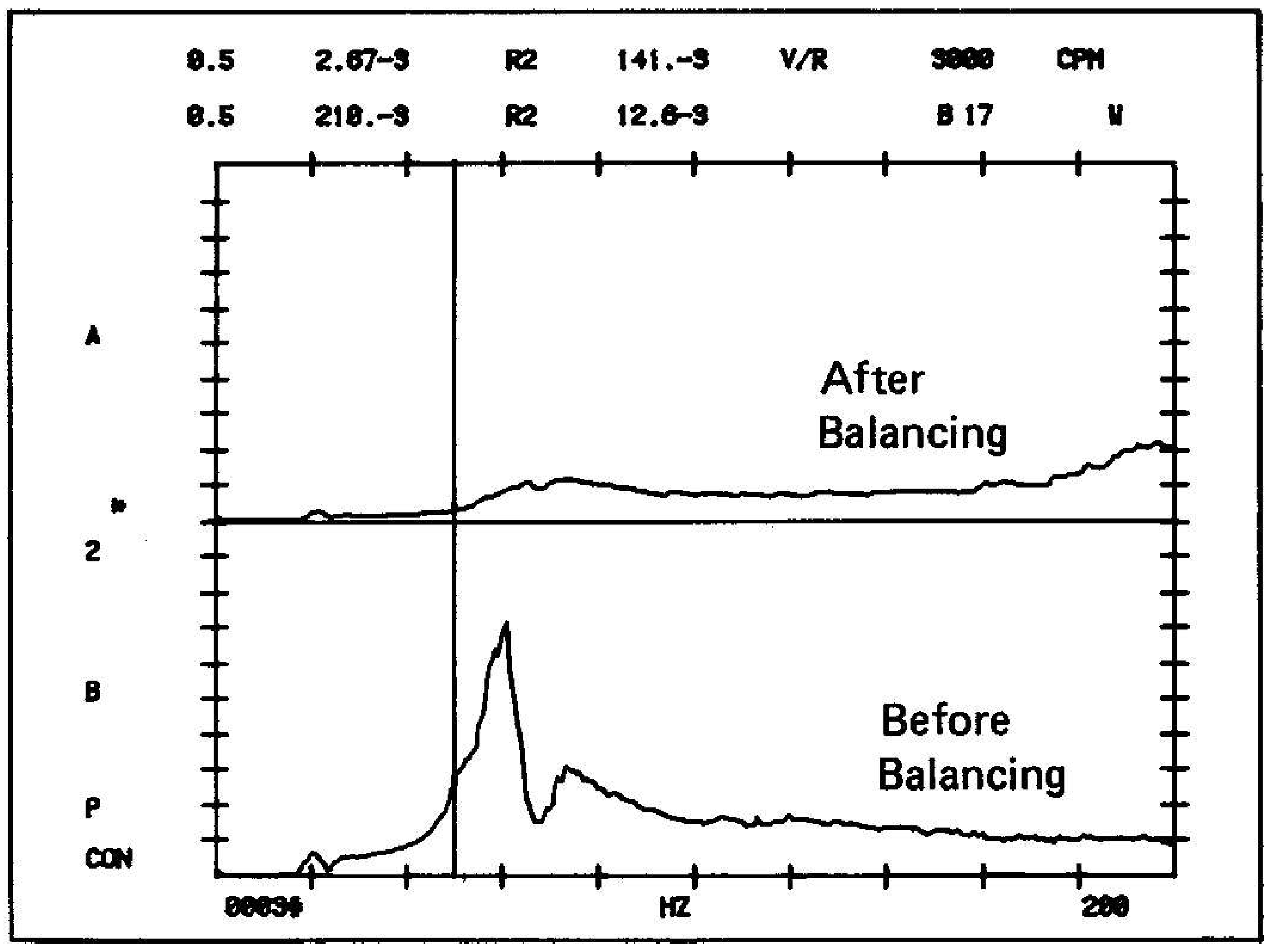


Figure 4

Figure 4 illustrates the effect of this initial balancing. The spectra presented in Figure 4 are peak stored "coast down" spectra derived by turning the machine off and allowing it to coast down from an operating speed of 24000 RPM (200 Hz). The lower spectrum (B for before) shows the acceleration prior to balancing. The upper spectrum (A for after) shows the acceleration following the balancing operation. The vertical cursor marks the 3000 RPM balance speed and its associated amplitude readout illustrates that the 3000 RPM acceleration level dropped to eleven percent of its unbalanced level. Clearly, initial balancing was required!

Figure 3 shows two weights used in the verification experiment. The larger screw (painted head) was used as a simulated *unknown* mass. The smaller screw was used as the trial weight. The mass of the large screw was 2.86 grams while that of the small screw was 2.44 grams, hence a *known* U/T ratio of 1.17 was employed.

Speed control of the universal motor was not easily achieved. Hence, the "coast down" method was deemed desirable.

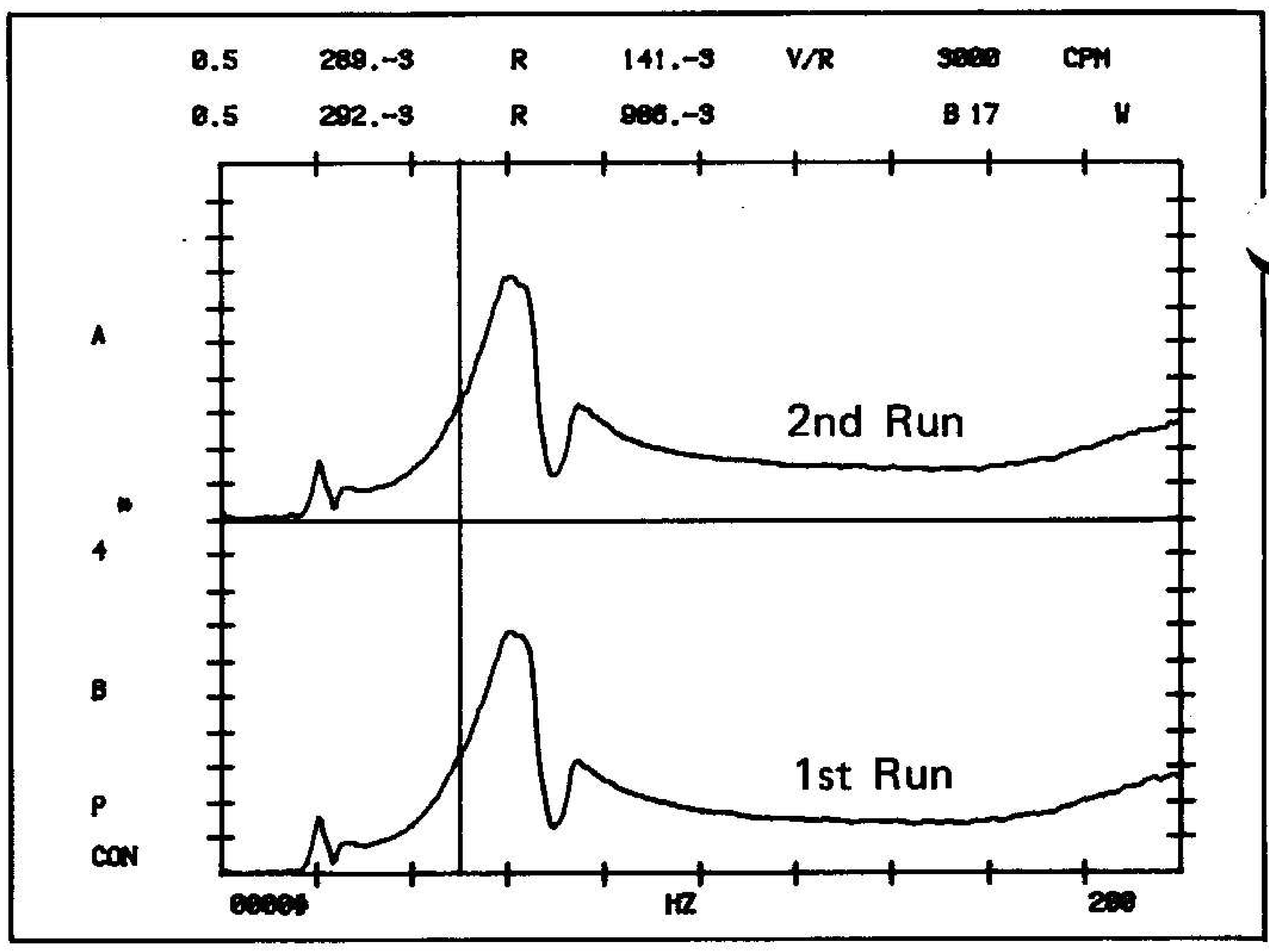


Figure 5

Figure 5 illustrates acceleration coast down spectra from two consecutive runs of the rotor with the unbalance weight installed. Clearly, these coast down measurements are consistent.

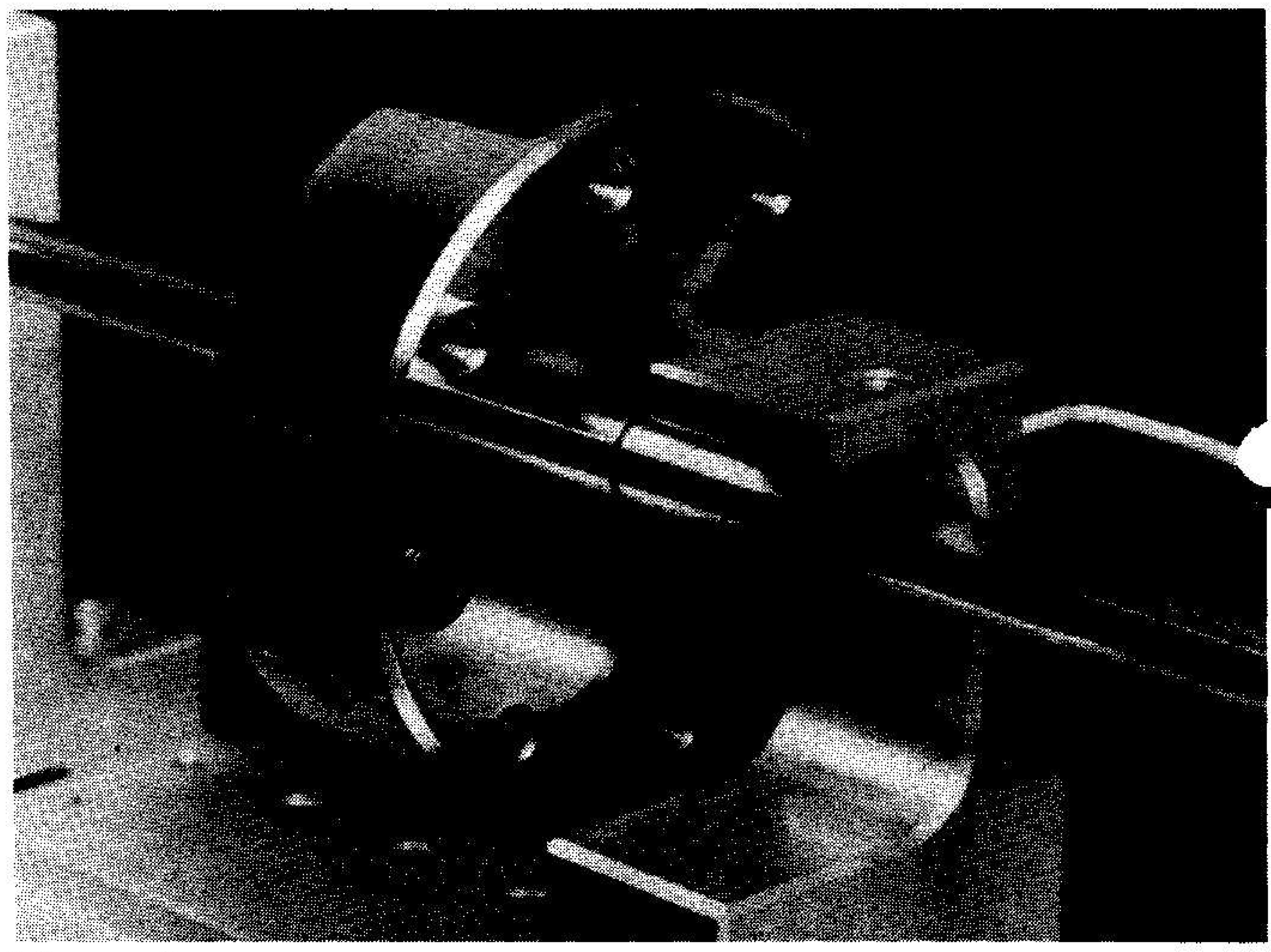


Figure 6a

Figure 6a shows the unbalance weight installed. Figure 6b presents the "coast down" spectra associated with this unbalance. The upper trace is a displacement spectrum while the lower trace is an acceleration spectrum.

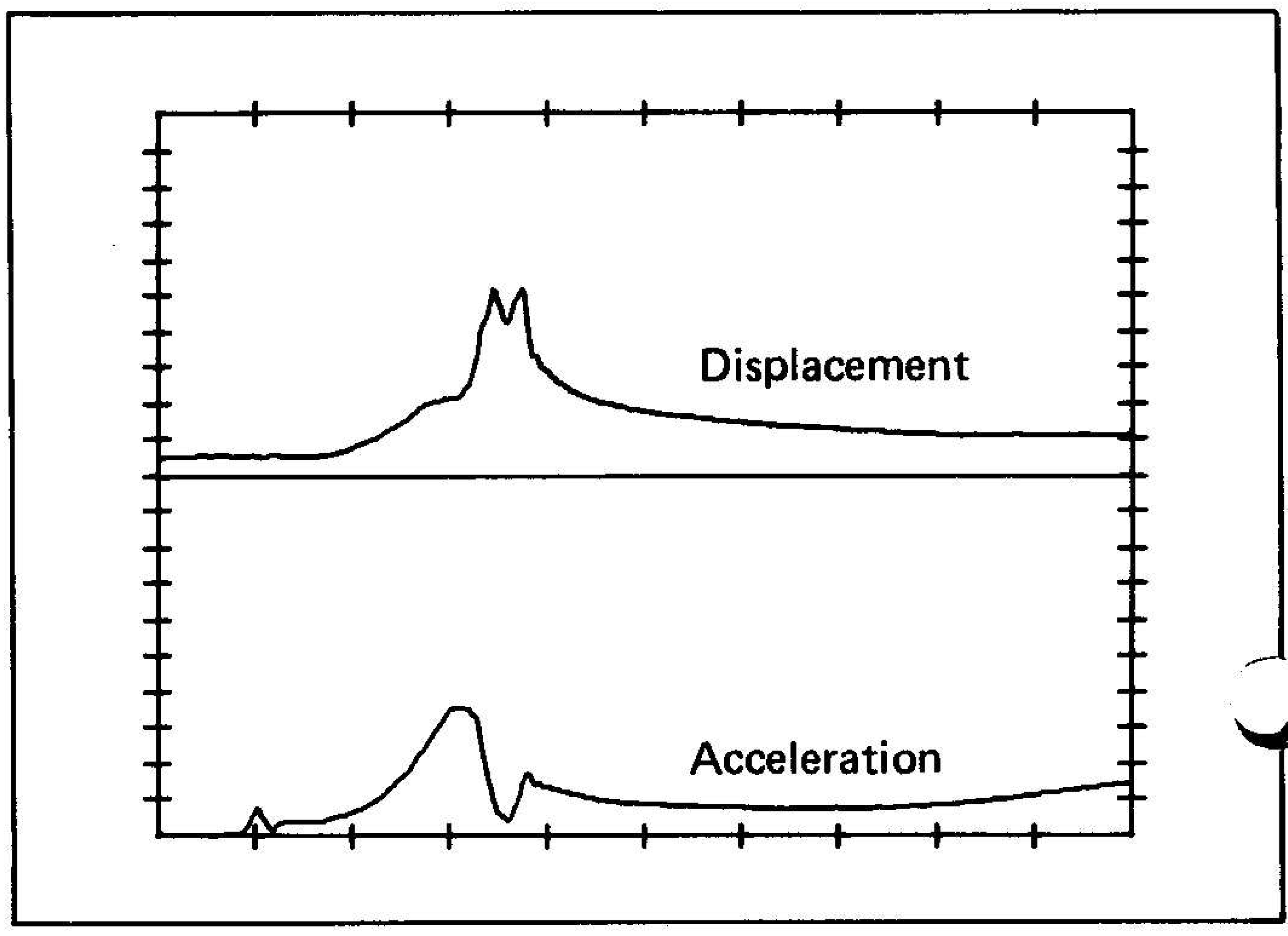


Figure 6b

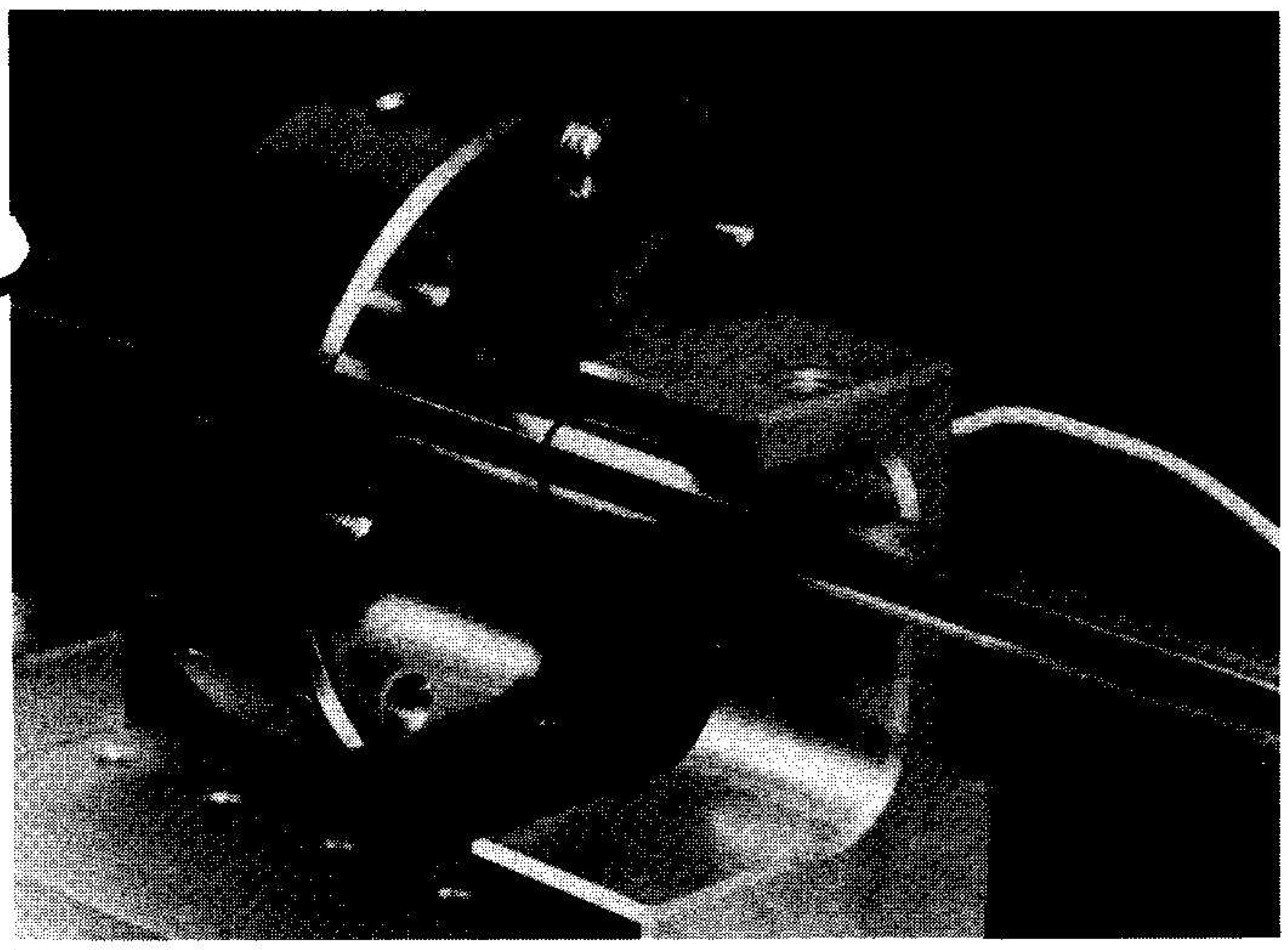


Figure 7a

Figure 7a shows a reference position marked on the rotor. The trial weight has been installed at this position. (In this example, the reference position was chosen to be 180° away from the unbalance.) Figure 7b illustrates the displacement and acceleration coast down spectra associated with this configuration.

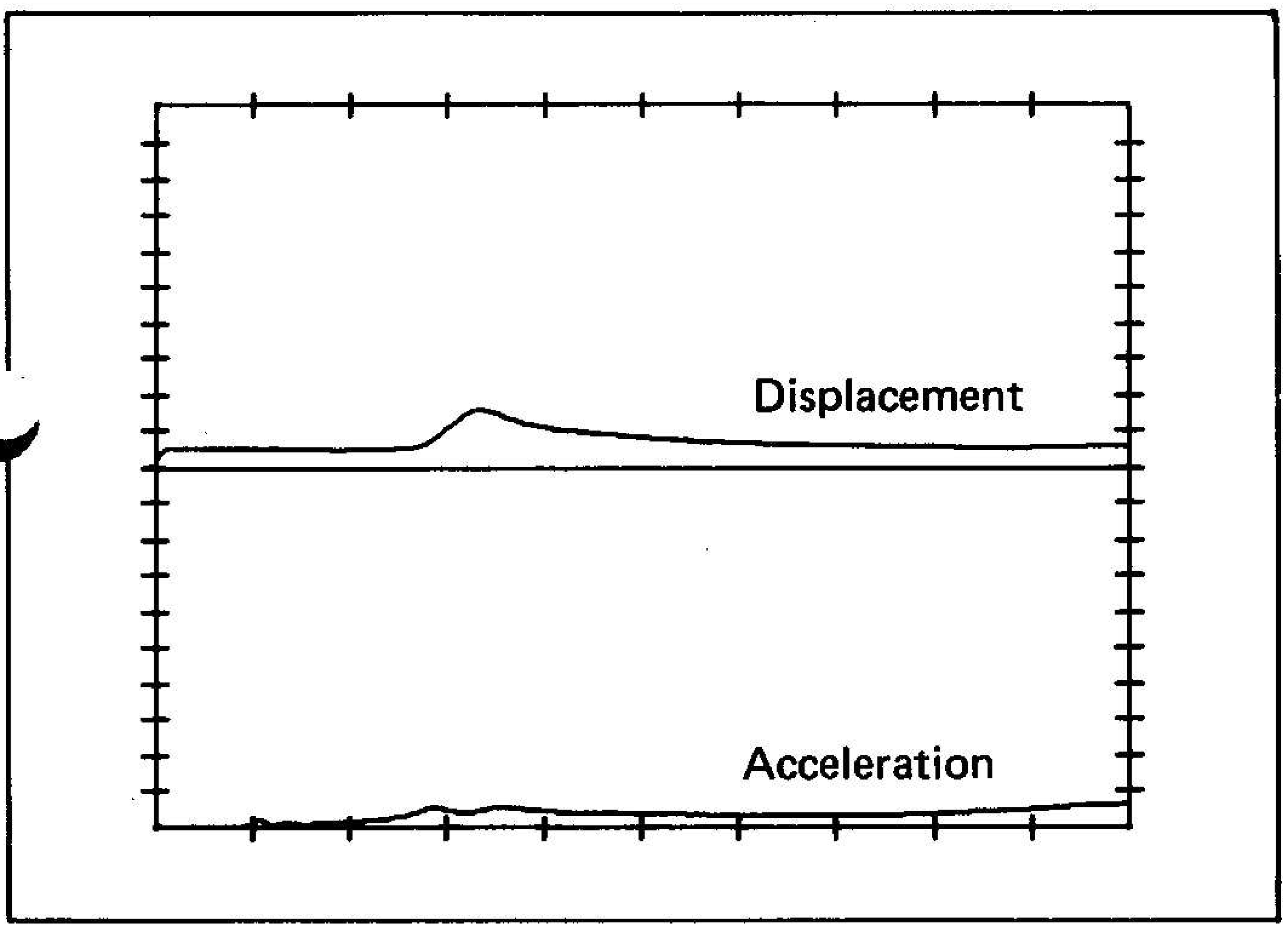


Figure 7b

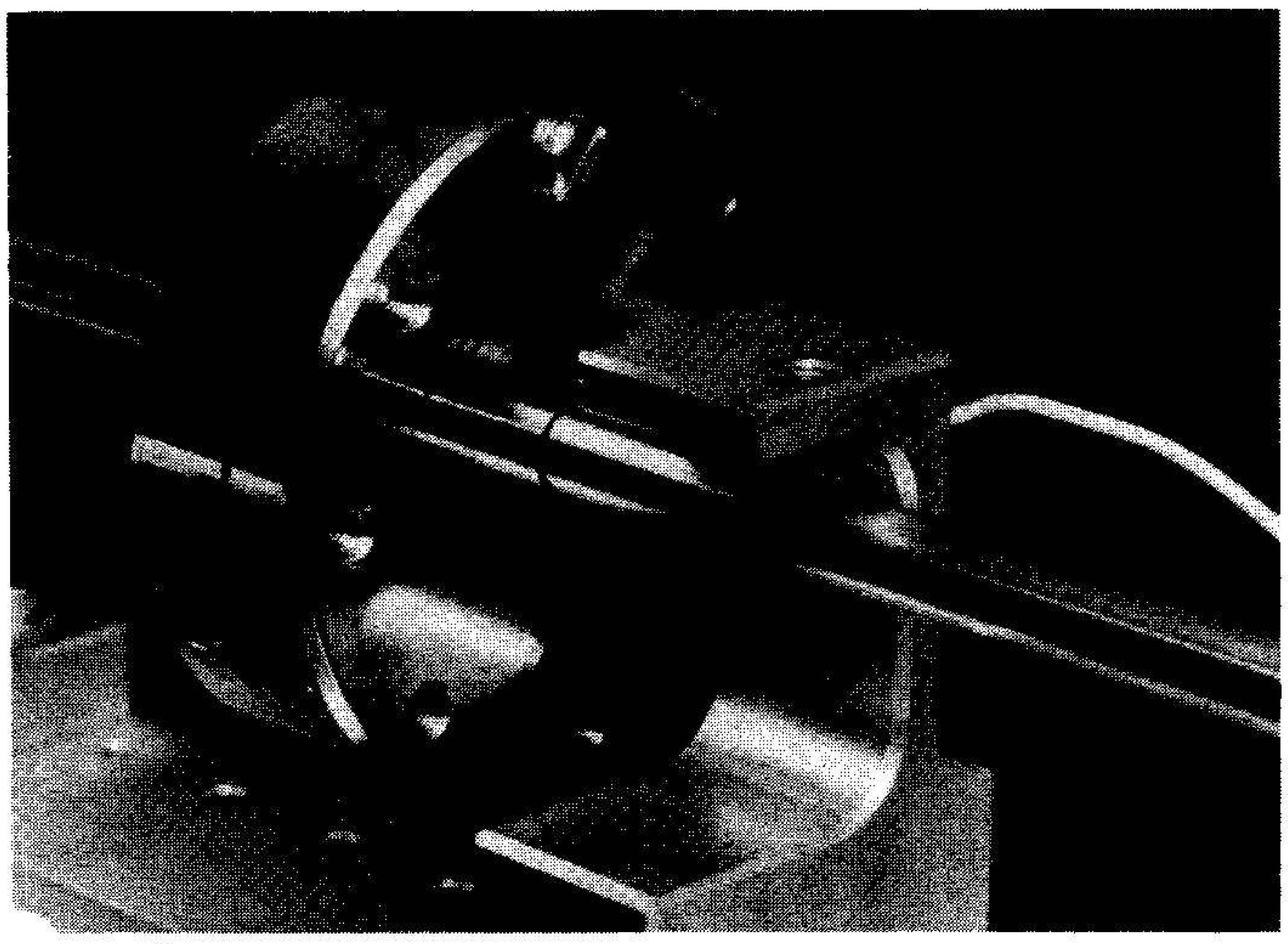


Figure 8a

In Figure 8a the trial weight has been moved through an angle, $\delta = 90^\circ$. Figure 8b shows the displacement and acceleration coast down spectra associated with this configuration.

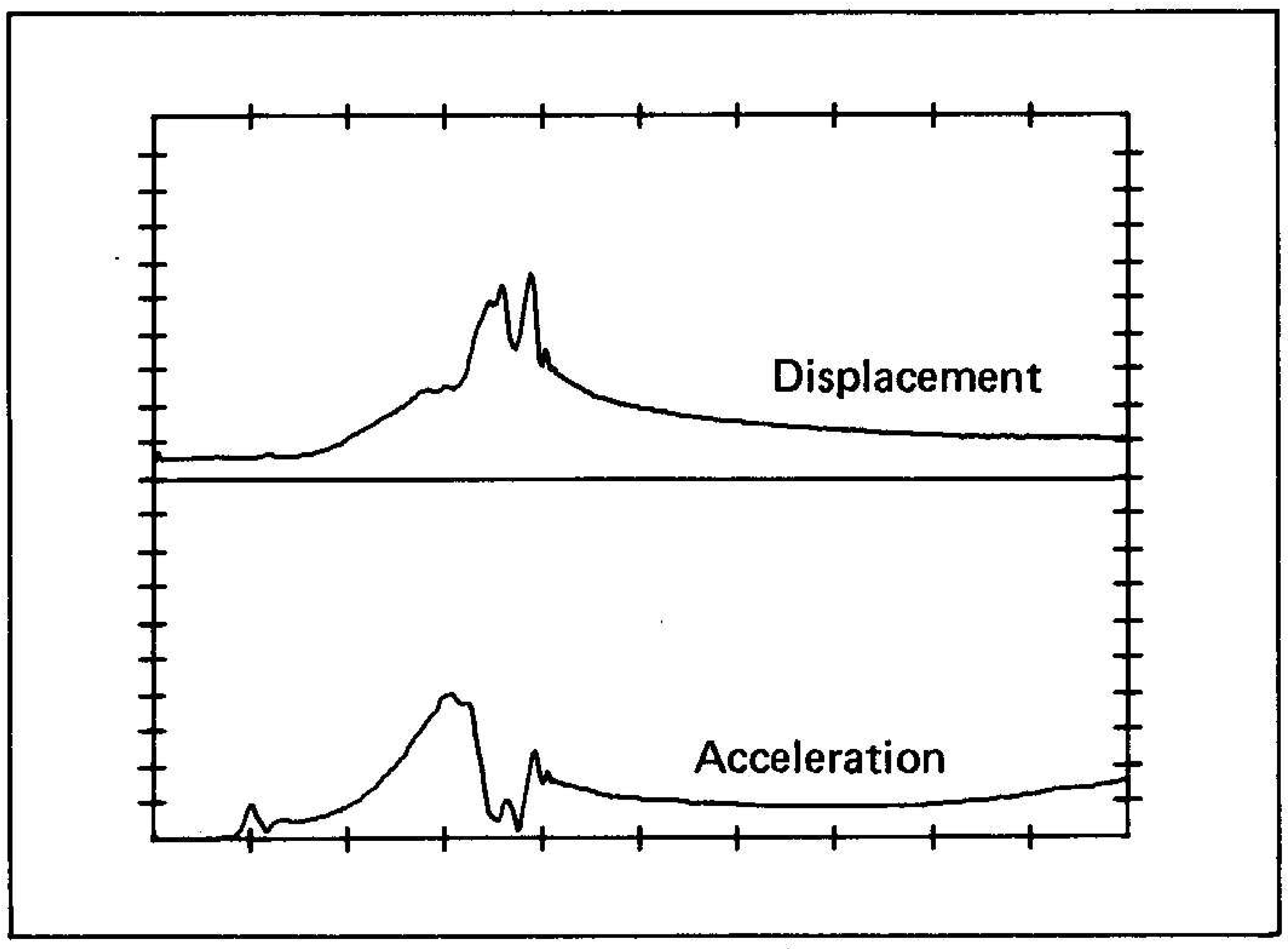


Figure 8b

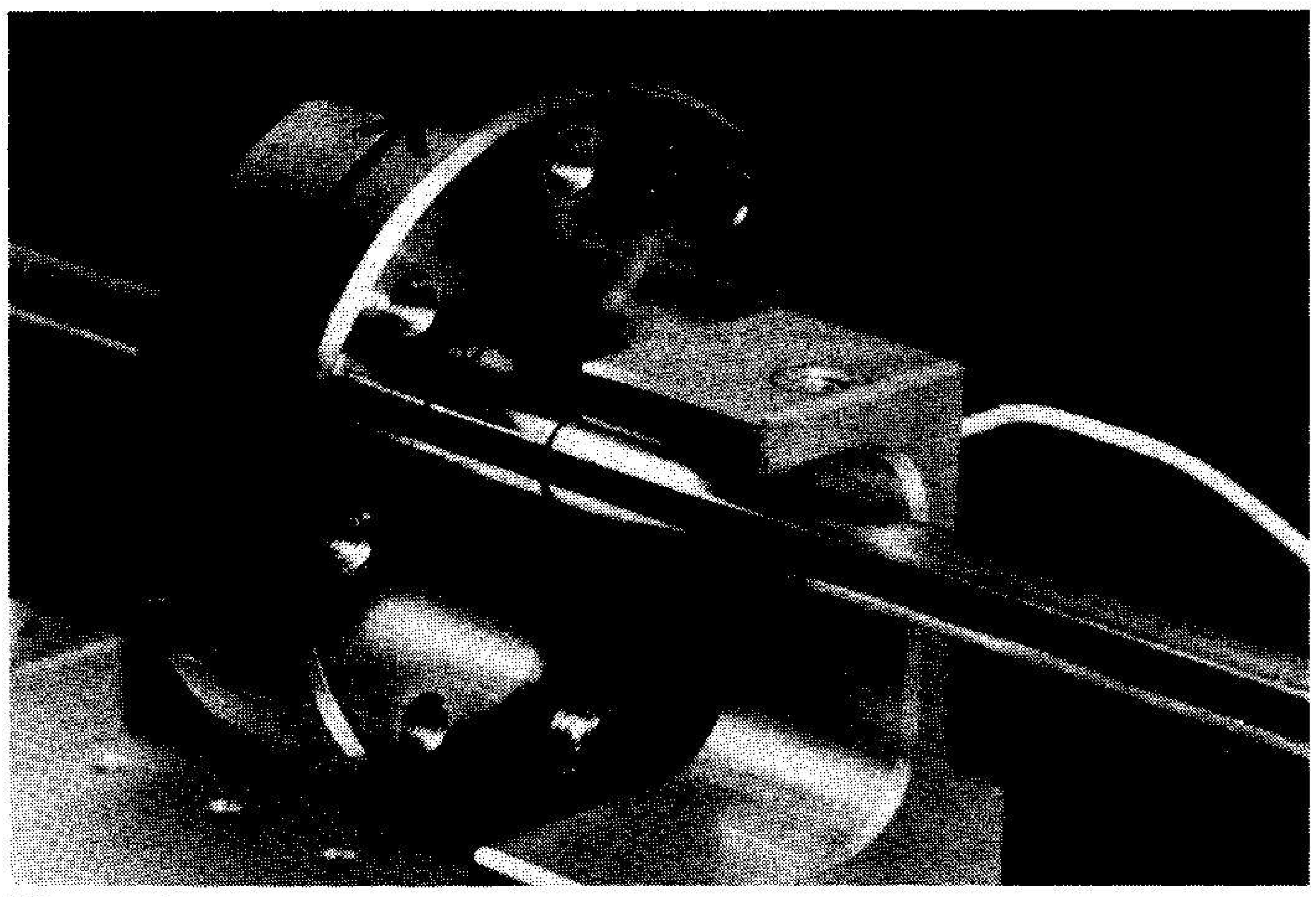


Figure 9a

Figure 9a shows the trial weight having been moved to a position, $\delta = -90^\circ$ from the reference mark. Figure 9b shows the associated displacement and acceleration coast down spectra.

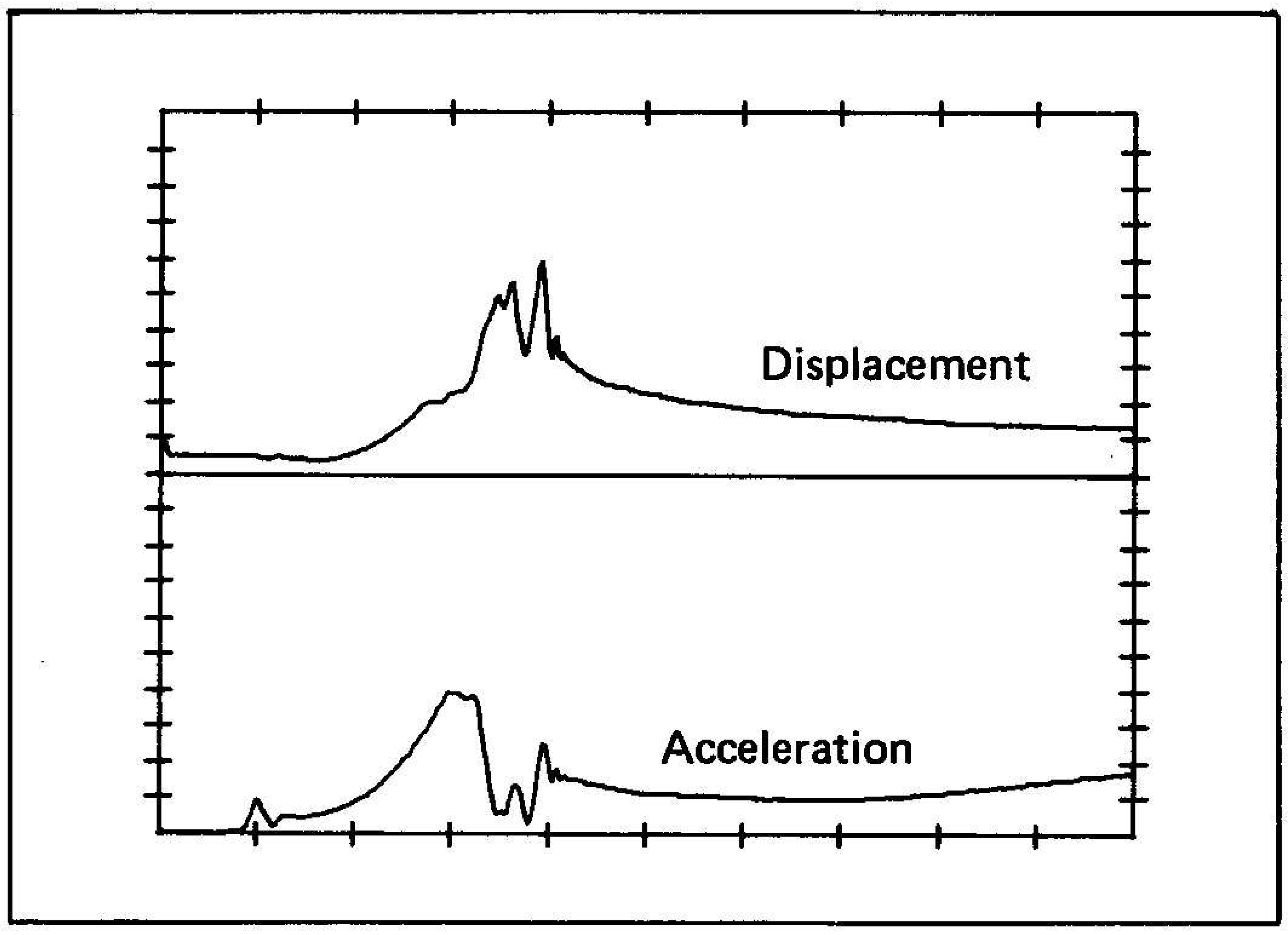


Figure 9b

Amplitudes were read from the spectra of Figures 7 - 9 using the 444's digital readout. Entering transducer scale constants, and reading out in squared units, allowed the direct readout of squared acceleration (g^2) and squared displacement (mil^2). This readout was chosen as the calculator program accepts squared amplitudes as its fundamental input.

	Measured Squared Displacement (Mil^2)				Computed	
RPM	R_0^2	R_1^2	R_2^2	R_3^2	$\frac{U}{T}$	ϕ (Degree)
1500	.447	.374	.546	.347	7.26	126.08
3000	3.07	.359	5.15	2.82	1.61	162.19
6000	4.78	.980	5.59	7.68	1.66	−169.53
12000	1.78	.472	1.64	2.40	2.23	−166.21
				Mean	3.19	168.13
				Standard Deviation	2.73	31.42

Table 1
Displacement Data

	Measured Squared Acceleration (g^2)				Computed	
RPM	R_0^2	R_1^2	R_2^2	R_3^2	$\frac{U}{T}$	ϕ (Degree)
1500	.00363	.000284	.00689	.00582	1.19	174.96
3000	.0795	.00292	.130	.125	1.28	178.25
6000	.0239	.00485	.0371	.040	1.42	−177.54
12000	.0659	.0143	.0761	.0947	1.84	−172.54
				Mean	1.43	180.78
				Standard Deviation	0.29	5.41

Table 2
Acceleration Data

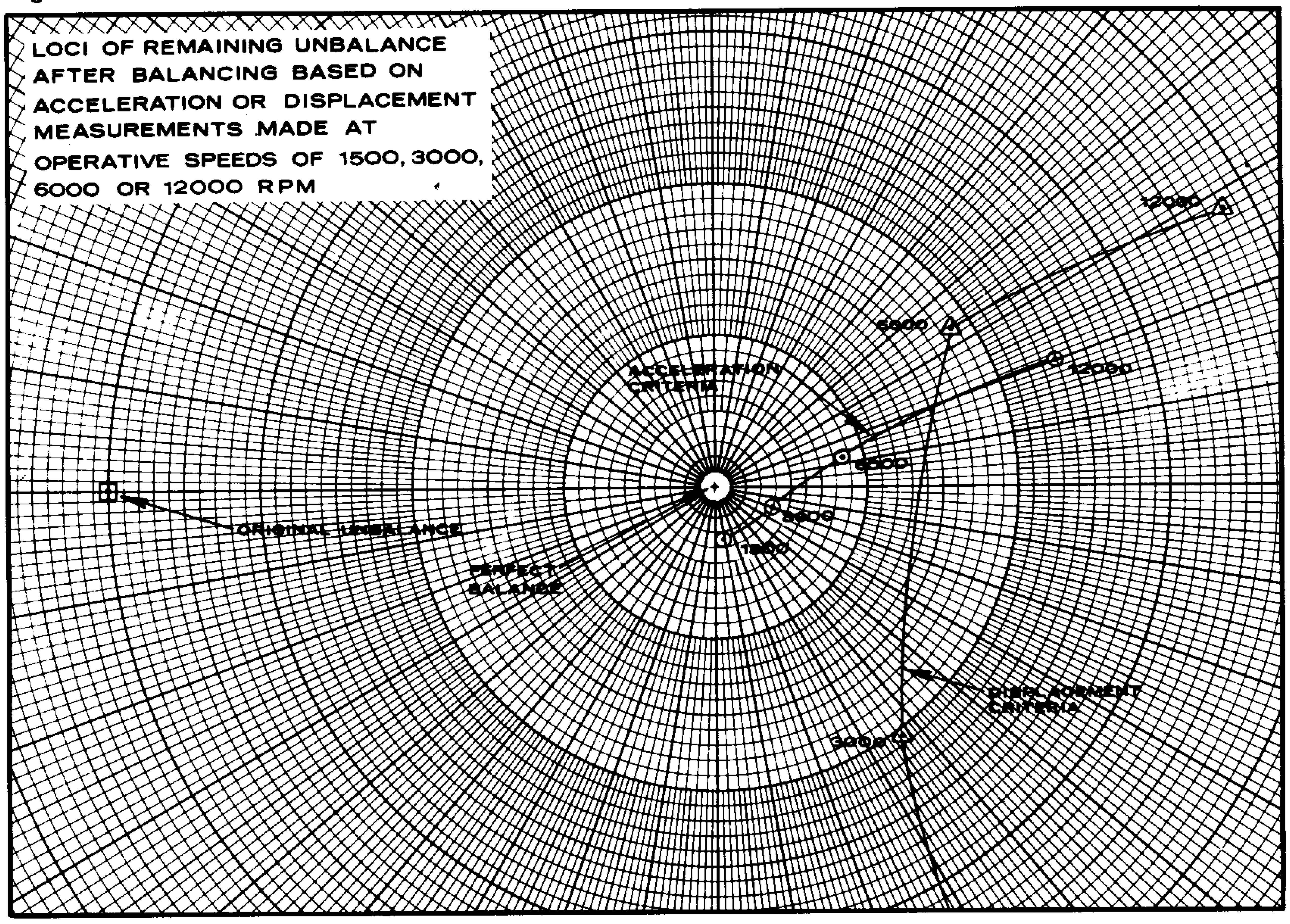
Table 1 presents data from the displacement spectra while Table 2 presents data from the acceleration spectra. Data were read at operating speeds of 1500, 3000, 6000, and 12000 RPM. This provided two balance speeds below the 3600 RPM first critical speed of the rotor, and two computations above this frequency. The data in Tables 1 and 2 were used to compute unbalance ratio and angular position using the program of Appendix B. The results of these computations are also summarized in Tables 1 and 2.

None of the eight computations gave *exactly* the "correct" (U/T = 1.17 ϕ = 180°) answer! The error (and its variability) were far less using the acceleration data than using the displacement data. This may be seen by comparing the mean and standard deviation values for mass ratio and angular position as shown in Tables 1 and 2. Even if the 1500 RPM trials are discarded, the acceleration data yields more precise and consistant answers.

The superiority of the acceleration data can be seen more clearly from Figure 10. This figure illustrates the unbalance that would result if corrective actions were taken based upon each of the previous balance computations. Using the acceleration balance at 1500, 3000, or 6000 RPM would have resulted in vibration level reductions by a factor of four or greater. Using the acceleration data at 12000 RPM would still have reduced vibration levels to 60 percent of the initial unbalance level. In contrast, the 3000 and 6000 RPM displacement data would have resulted in cutting the vibration level in half. The 12000 RPM displacement would have had virtually no change on the vibration level while using the 1500 RPM displacement data would have resulted in vibration levels more than five times as large as the original balance!

These results should not be construed as a condemnation of displacement transducers. The superior results obtained from the acceleration data *are due to the physics of the problem* and not the frailties of one transducer or another. Both the accelerometer *and the displacement probe* utilized in this experiment were top quality transducers, performing properly. The proximity probe was simply not measuring a variable linearly proportional to the unbalance force while the accelerometer was.

Figure 10



CONCLUSIONS

The real time analyzer is an ideal tool for trim balance work. Methods described in this paper and the program documented in Appendix B have been demonstrated to work successfully in reduction of unbalance-induced vibration. The time saving "coast down" method of data collection has been validated and its use is encouraged.

The experimental data previously described illustrates a clear advantage in using acceleration (or velocity), rather than relative shaft displacement, when balancing. Further, the data indicates that the most successful balance can be achieved by making measurements within a balance speed range of .5 →2 times the first critical speed of the rotor.

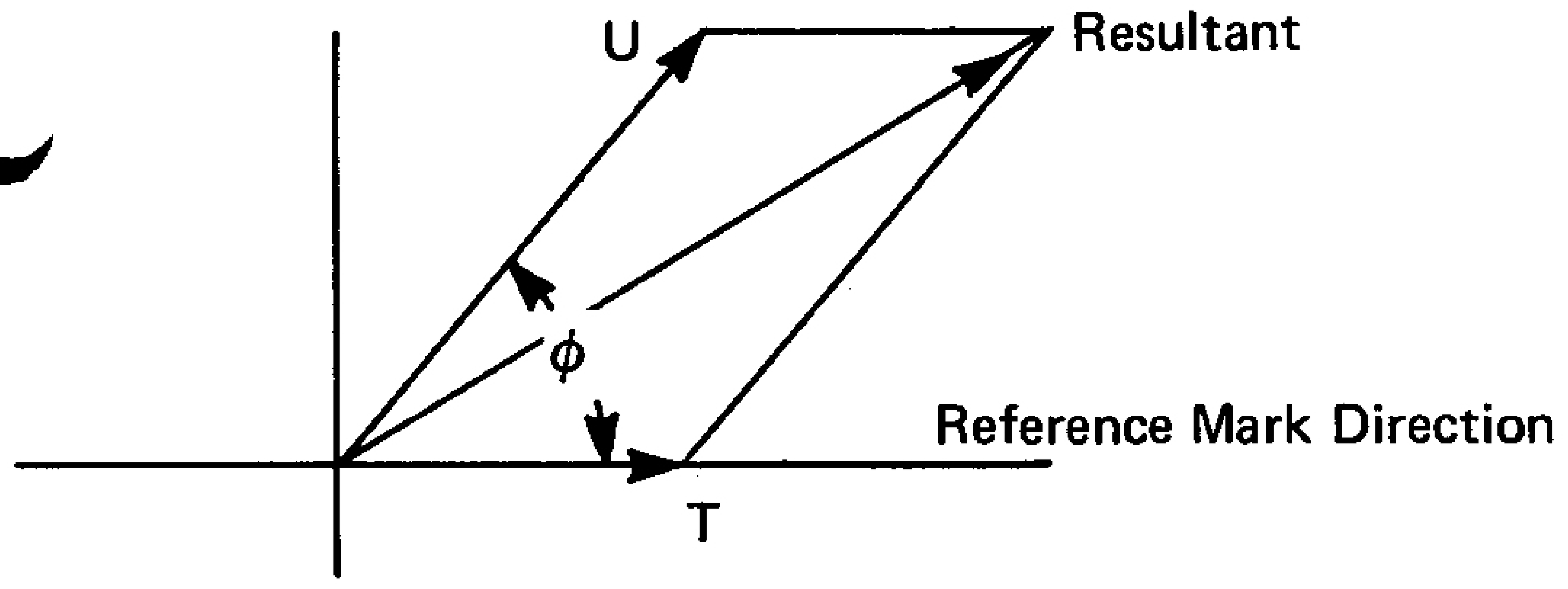
APPENDIX A: DERIVATION OF EQUATIONS

At a constant operating speed the observed rotative speed vibration level is proportional to the amount of unbalance. Hence:

(1) $R_0 = kU$

where k is a constant reflecting the amplitude of the structural transfer function between the rotor plane and the point of vibration measurement and the scale factor of the transducer employed.

The trial weight, T, is first attached at the reference position. The unbalance, U, is located at a positive angle, ϕ, from this position. The response vibration, R_1 is proportional to the vector resultant of U and T.

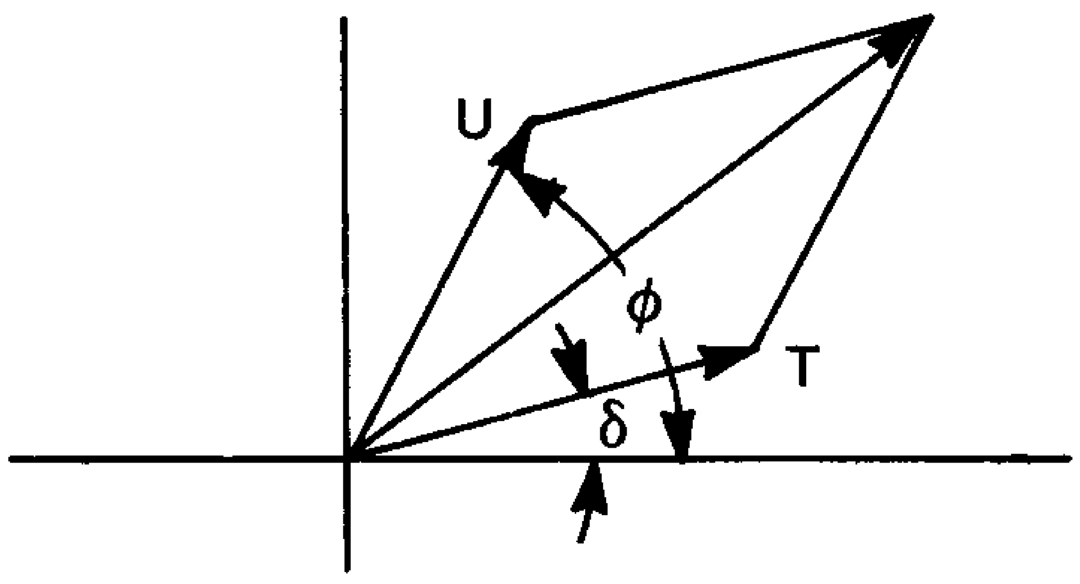


$$R_1 = k\sqrt{(U\cos\phi + T)^2 + (U\sin\phi)^2}$$

$$R_1 = k\sqrt{U^2\cos^2\phi + 2TU\cos\phi + T^2 + U^2\sin^2\phi}$$

(2) $$R_1 = k\sqrt{U^2 + T^2 + 2TU\cos\phi}$$

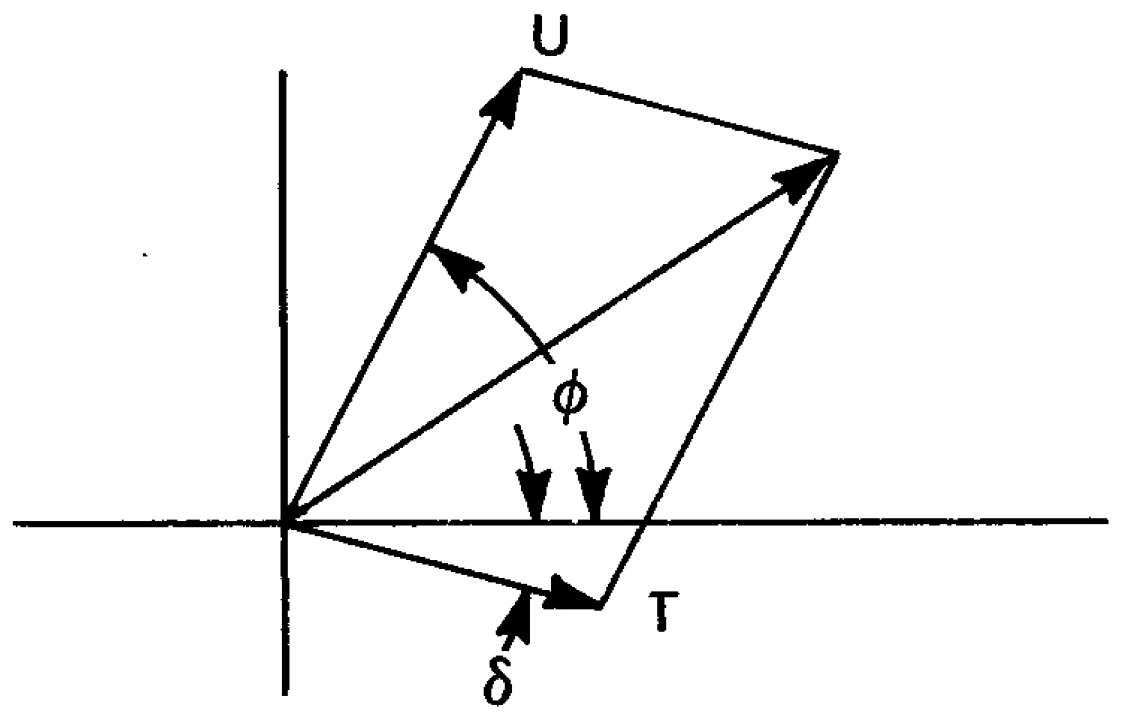
Moving the trial weight to a position ahead of the reference position:



$$R_2 = k\sqrt{U^2 + T^2 + 2TU\cos(\phi - \delta)}$$

(3) $$R_2 = k\sqrt{U^2 + T^2 + 2TU(\cos\delta\cos\phi + \sin\delta\sin\phi)}$$

Finally, moving the trial weight to the $-\delta$ position:



$$R_3 = k\sqrt{U^2 + T^2 + 2TU\cos(\phi + \delta)}$$

(4) $$R_3 = k\sqrt{U^2 + T^2 + 2TU(\cos\delta\cos\phi - \sin\delta\sin\phi)}$$

From which:

(5) $$\frac{R_2^2 - R_3^2}{4\sin\delta\, R_0^2} = \frac{T}{U}\sin\phi = Y$$

and

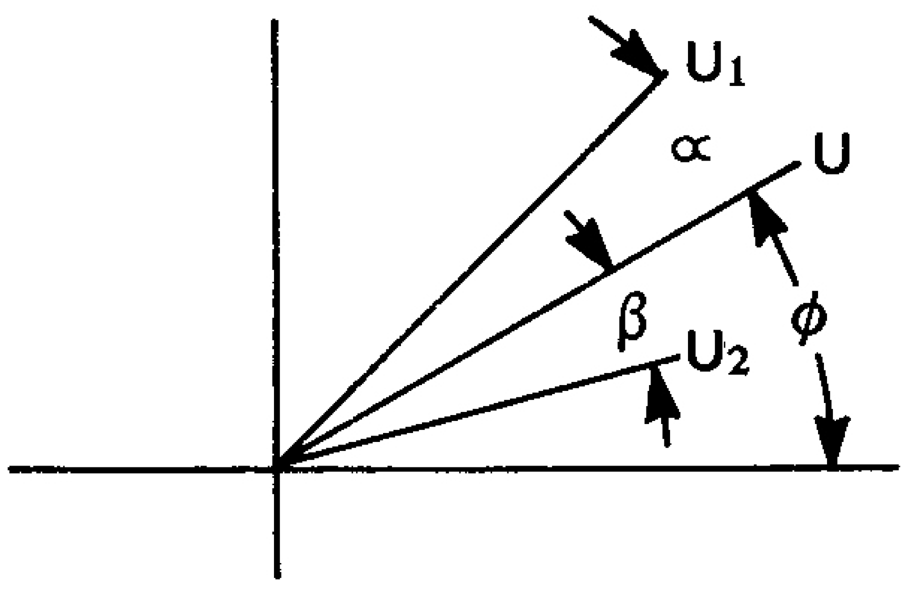
(6) $$\frac{2R_1^2 - R_2^2 - R_3^2}{4(1 - \cos\delta)\, R_0^2} = \frac{T}{U}\cos\phi = X$$

Resolving these orthogonal components to polar form yields $\frac{U}{T}$ and ϕ:

(7) $$\frac{U}{T} = \frac{1}{\sqrt{X^2 + Y^2}}$$

(8) $$\phi = \tan^{-1}\left(\frac{Y}{X}\right)$$

Weight "splitting" may be required if ϕ is inconviently located



"Splitting" a trim weight requires attachment of weights U_1 and U_2 at angular positions α and β from the ϕ direction. Their vector resultant must have magnitude, U, and direction, ϕ. Hence, it is required that:

(9) $U_1\cos\alpha + U_2\cos\beta = U$

and

(10) $U_1\sin\alpha + U_2\sin\beta = 0$

The first equation requires the ϕ direction components of U_1 and U_2 to sum to U. The second requires the components of U_1 and U_2 perpendicular to ϕ to sum to zero. From (9) and (10):

$$(11) \qquad U_1 = U \frac{\sin \beta}{\sin (\alpha + \beta)}$$

$$(12) \qquad U_2 = U \frac{\sin \alpha}{\sin (\alpha + \beta)}$$

REFERENCES

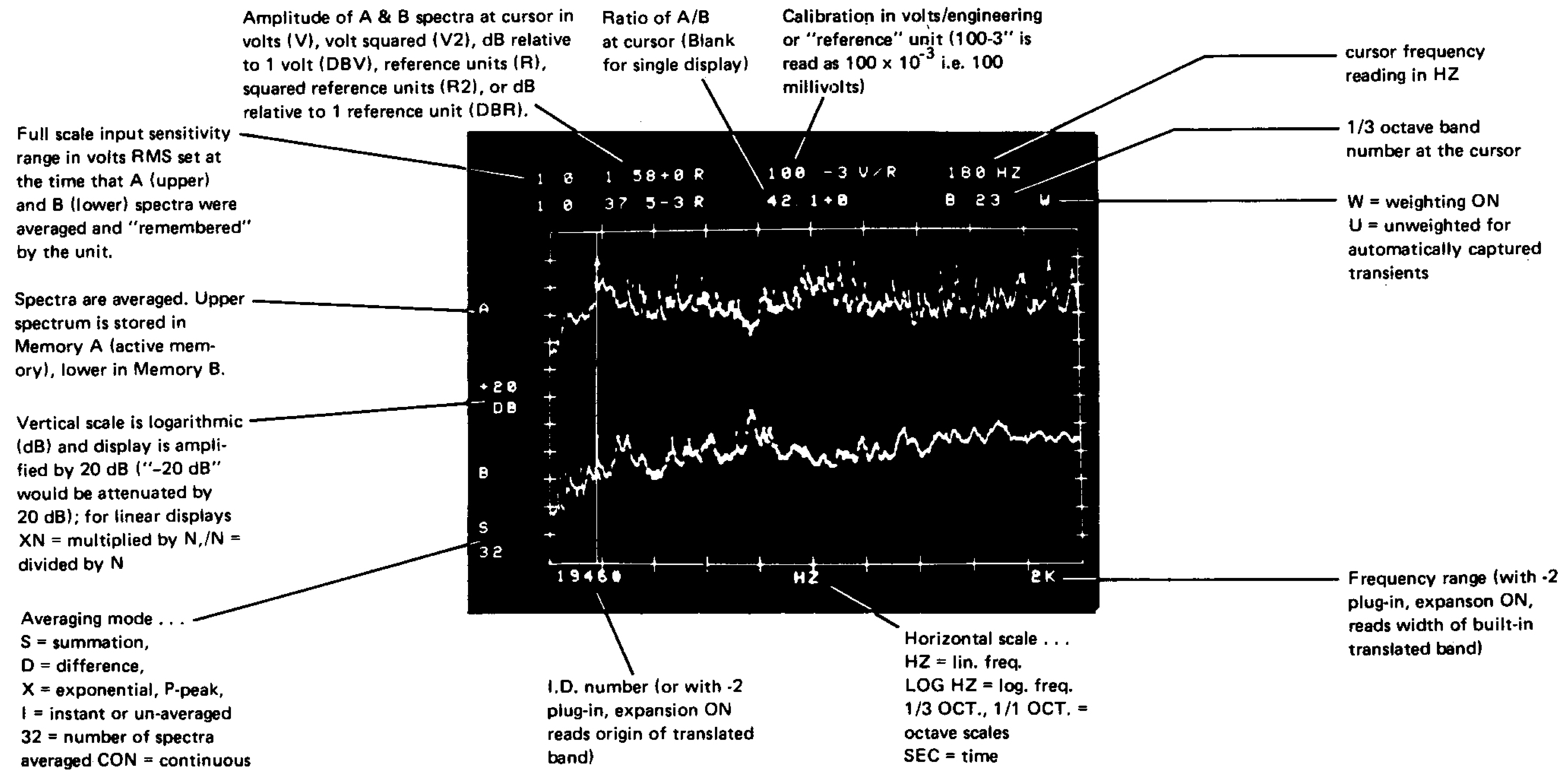
(1) "Mechanical Vibrations", J. P. Den Hartog, McGraw-Hill Book Company, 1956.

(2) "The Hidden Message in Mechanical Vibration", G.F. Lang, Machine Design, June 10, 1976.

(3) "Mini-Ubiquitous® FFT Computing Spectrum Analyzer", Nicolet Scientific Corporation, 1977.

(4) "Vibration and Acoustic Measurement Handbook", M.P. Blake and W.S. Mitchell, Sparten Books, 1972.

DUAL AVERAGE SPECTRUM DISPLAY



APPENDIX B

PROGRAM

Load the following program into your HP25 or 25C:

01	f REG	26	÷
02	f → R	27	g → P
03	CHS	28	g 1/X
04	Σ +	29	R/S
05	STO+7	30	—
06	R/S	31	g ABS
07	STO−0	32	f sin
08	STO−1	33	X⇄Y
09	R↓	34	R↓
10	STO+0	35	÷
11	STO−1	36	STO5
12	R↓	37	STO6
13	2	38	RCL1
14	X	39	RCL2
15	STO+1	40	—
16	R↓	41	g ABS
17	4	42	f sin
18	X	43	STOX6
19	STO÷0	44	RCL1
20	STO÷1	45	RCL3
21	RCL0	46	—
22	RCL4	47	g ABS
23	÷	48	f sin
24	RCL1	49	STOX5
25	RCL7		

RUNNING INSTRUCTIONS

1. Initialize program by keying: [f] [PRGM]
2. Select angle, delta, that will separate trial weight positions. (Recommended: 90° ≤ delta ≤ 120°)
3. Key in delta (degree) followed by: [ENTER ↑] [1] [R/S]
4. Measure amplitude of vibration in R2 (or V2) units from 444 cursor at the selected balance speed. Enter this number followed by: [ENTER ↑]
5. Stop machine and install trial weight (magnitude = T) at any position. Mark this location as an angular reference. Run the machine and repeat Step 4.
6. Stop the machine and move the weight to a new position delta degrees from the reference mark. The direction chosen in this step defines the direction for **positive** angles. Run the machine and repeat Step 4.
7. Stop the machine and move the weight to a new position delta degrees from the reference mark in the opposite (negative angle) direction. Run the machine, measure the balance speed vibration. Enter this number followed by: [R/S]
8. The display now shows the ratio, U/T, where U is the amount of unbalance to be corrected. Enter the mass (or weight) of the trial weight, T, followed by: [X] [STO] [4] [X⇄Y]
9. The display now shows the angular position of U. If mass is to be removed to balance, key: [STO] [1] If mass is to be added, key: 180 [+] [STO] [1]
10. If corrective action cannot be taken at the angle displayed, weight "splitting" is required. Enter the nearest convenient angle greater than the angle on display followed by [STO] [2] Then enter the nearest convenient angle less than the required angle followed by: [STO] [3] [R/S]
11. Following execution, the following may be done in any sequence:

KEY	TO DISPLAY
[RCL] [1]	Location (angle) of required total correction
[RCL] [2]	Location of 1st trim weight
[RCL] [3]	Location of 2nd trim weight
[RCL] [4]	Mass of total correction required
[RCL] [5]	Mass of 1st trim weight
[RCL] [6]	Mass of 2nd trim weight

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