IEEE Transactions on Nuclear Science, Vol. NS-32, No. 5, October 1985

ACCELERATOR VIBRATION ISSUES*

Roger A. Tennant, AT-4, MS H821
Los Alamos National Laboratory, Los Alamos, New Mexico 87545 USA

Summary

Vibrations induced in accelerator structures can cause particle-beam jitter and alignment difficulties. Sources of these vibrations may include pump oscillations, cooling-water turbulence, and vibrations transmitted through the floor to the accelerator structure. Drift tubes (DT) in a drift tube linac (DTL) are components likely to affect beam jitter and alignment because they normally have a heavy magnet structure on the end of a long and relatively small support stem. The natural vibrational frequencies of a drift-tube have been compared with theoretical predictions. In principle, by knowing natural frequencies of accelerator components and system vibrational frequencies, an accelerator can be designed that does not have these frequencies coinciding.

Introduction

To obtain a database with which to compare theoretical predictions, the natural vibrational frequencies have been measured for the drift-tube-and-girder alignment model of the accelerator test stand (ATS). Figure 1 is a photograph of this model. Figure 2 is a simplified schematic of the vibrational model used to develop the theoretical calculations. A Nicolet Model 440A frequency spectrum analyzer and an accelerometer mounted on the simulated magnet structure were used to measure the natural frequencies. The mass of the drift-tube body, Fig. 1, was varied by attaching additional brass disks 8 cm in diameter and 2.54 cm thick. Measured frequencies as a function of drift-tube body mass are shown in Table I. Note that only a few of the observed frequencies are relatively strong.

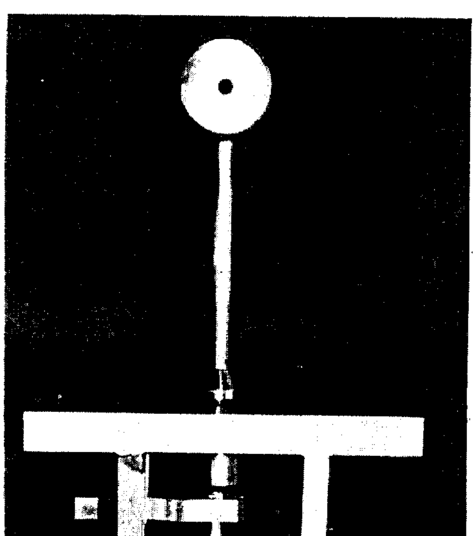
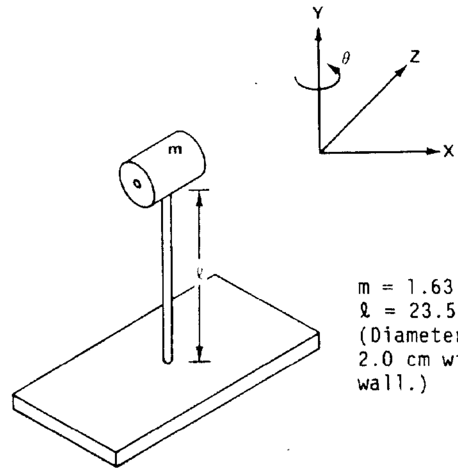


Fig. 1. ATS drift-tube and girder alignment model.



m = 1.63, 3.80, 5.97 kg
ℓ = 23.5 cm
(Diameter of ℓ is 2.0 cm with 0.365-cm wall.)

Fig. 2. Drift-tube schematic.

TABLE I

ATS DRIFT-TUBE NATURAL FREQUENCIES (MEASURED)

Drift-Tube Mass (1.63 kg)		Drift-Tube Mass (3.80 kg)		Drift-Tube Mass (5.97 kg)	
Frequency (Hz)	Strength	Frequency (Hz)	Strength	Frequency (Hz)	Strength
62.5	strong	21.3	weak	36.3	weak
180	weak	42.5	strong	180	weak
361	strong	158	weak	500	weak
500	weak	500	weak	725	strong
620	weak	775	weak	995	weak
845	weak	1160	strong	1435	weak
1190	weak	1800	weak		
		2260	weak		

Theoretical Development

Using Ref. 1 as a guide, the equations for natural frequencies in the transverse (X-Z direction), longitudinal (Y-direction), and torsional (Θ-direction) were derived. In the derivation, the differential equations were linearized by assuming the following: only small displacements were allowed, higher order terms were dropped, a point mass was assumed, and the weight of the DT stem was neglected.

The resulting equation for the frequency in the X-Z directions (see Fig. 2) is

$$w = \sqrt{\frac{K}{m} - \frac{g}{\ell}} \quad \text{(rad/s)} \quad . \tag{1}$$

The quantity K is a spring constant that, for a cantilevered beam, can be related to material properties as follows:

$$K = \frac{3EI}{\ell^3} \quad (g/s^2) \quad , \tag{2}$$

*Work supported by the US Dept. of Defense and the Ballistic Missile Defense Advanced Technology Center.

0018-9499/85/1000-2868$01.00© 1985 IEEE

where E = modulus of elasticity of the DT stem (g/cm s^2),
I = moment of inertia of the DT stem (cm^4),
ℓ = length of DT stem (cm),
m = mass of DT body (g), and
g = acceleration (cm/s^2).

The equation for the natural frequency in the Y-direction (see Fig. 2) is

$$w = \sqrt{\frac{K}{m}} \quad (\text{rad/s}) \quad .$$

In this instance, the spring constant is approximately

$$K = \frac{AE}{\ell} \quad (\text{g/s}^2) \quad ,$$

where A = the cross-section area of the DT stem (cm^2), and E and ℓ are as defined previously.

The equation for the natural frequency in the Θ direction (see Fig. 2) is

$$w = \sqrt{\frac{K_T}{I_m}} \quad (\text{rad/s}) \quad . \tag{3}$$

In terms of material properties, the torsional spring constant K_T is as follows:

$$K_T = \frac{GJ}{\ell} \quad (\text{g cm}^2/\text{s}^2) \quad ,$$

where G = shearing modulus of elasticity of the DT stem (g/cm s^2),
J = polar moment of inertia of the DT stem (cm^4),
ℓ = length of the DT stem (cm), and
I_m = mass moment of inertia for the DT body (g cm^2).

Comparison of Theory and Experiment

Using Eqs. (1)-(3), the drift-tube natural frequencies were calculated. Table II shows each of the three fundamental frequencies plus the first two harmonics of each. Comparing Tables I and II shows that, for the lightest DT-body case, the strongest modes are the fundamentals in the transverse and torsional directions. Some of the weaker frequencies observed may be 2nd harmonics of the transverse and torsional vibrations. The 1st harmonics do not appear to be present. The vibration at 845 Hz may be the longitudinal mode. Other frequencies are still unidentified.

For the intermediate-mass case, the transverse mode was still strong; however, the measured torsional mode was weak. Several vibrational frequencies remain unidentified, including a strong resonance at 1160 Hz.

For the heaviest mass case, the transverse mode was still observable, but weak. The torsional mode was not observed. A 500-Hz vibration was common to all three cases; perhaps this was due to a vibration in some part of the model other than the DT body-and-stem combination. Also, there was an unidentified strong vibration at 725 Hz.

TABLE II

ATS DRIFT-TUBE NATURAL FREQUENCIES (CALCULATED)

Mode	Drift-Tube Mass (1.63 kg) Frequency (Hz)	Drift-Tube Mass (3.80 kg) Frequency (Hz)	Drift-Tube Mass (5.97 kg) Frequency (Hz)
Transverse	64.7	42.4	33.8
1st Harmonic	129	84.8	67.6
2nd Harmonic	194	127	101
Longitudinal	814	533	425
1st Harmonic	1630	1070	850
2nd Harmonic	2440	1600	1275
Torsional	362	167	96.2
1st Harmonic	723	334	192
2nd Harmonic	1085	501	289

Conclusions

Measurements and theoretical predictions for natural vibrational frequencies of a drift tube have been compared for the ATS drift-tube-and-girder alignment model. The theory used to date is only partially successful at predicting the measured vibrational frequencies. In particular, as the DT body mass increases, a strong, unidentified frequency appears that is easy to excite. Clearly, additional theoretical and experimental development is needed. Further theoretical treatment should investigate such factors as the effects of the differential equation linearization, the higher order terms that were dropped, the mass of the drift-tube stem, and the finite versus the point-mass assumption of the drift-tube body-magnet structure.

Several things could be done to minimize the effects of DT vibration:

- DT stems could be stiffened, which would raise the natural frequency and reduce the actual displacement.
- DT stem and body configurations should be optimized to minimize flow-induced vibration because of factors such as large coolant flow rates, cavitation, excessive pressure loss, turbulence, flow oscillations and eddies.
- Vibration sources should be isolated and minimized to reduce excitation of the measured DT resonant frequencies.

By taking all these precautions, vibrational displacements of less than 0.001 in. could be attained.

Acknowledgment

The assistance of Joe Uher in the assembly of the experimental apparatus is gratefully acknowledged.

References

1. R. K. Vierick, Vibrational Analysis, (International Textbook Co., Scranton, Pennsylvania, 1967).

2. R. D. Blevins, Flow-Induced Vibration, (Van Nostrand Reinhold Co., New York, 1977).