Properties of Railway Wheels

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INTRODUCTION

The urban railway is ideally suited for high density corridors or for those areas where urban growth is considered desirable. However, urban railways are limited in their application by high capital requirements and by train noise which can disturb tne community served by the system (1).

Previous investigations into the nature of railway noise (2) have shown that wheel/rail noise is dominant, at least insofar as electric trains are concerned. It has been shown that sound radiated by the wheel is a significant part of the total noise. Many different types of railway wheels have been tested on transit systems with varying degrees of success. However, little has been published which would enable an operator to compare the different wheels on the basis of their fundamental mechanical properties. The purpose of this paper is to present some laboratory data on four common railway wheels.

BEHAVIOUR OF EXISTING WHEELS

Wheels which are presently manufactured can be grouped into two categories - solid and resilient. In Table 1, the standard wheel is representative of the former group since it is made entirely of steel.

The resilient wheels use an elastomeric element to separate the wheel tread from its hub. The elastomer reduces the unsprung mass of the wheel/rail system and adds damping to the wheel. This damping reduces the squeal noise which can occur on short radius curves in the track. On the other hand, field experience indicates that resilient wheels do not appreciably reduce the rolling noise which occurs on straight track.

For this study, four properties of the wheels were sought: frequencies; mode shapes; modal damping ratios; and the degree of coupling between in-plane and out-of-plane wheel vibrations. This so-called "radial/axial" coupling has been postulated as an important mechanism in the generation of rolling noise (3).

The vibratory properties of the wheels were determined by mounting the wheel on the apparatus shown in Figure 2, and striking the wheel with a hammer. The resulting vibration was detected by an accelerometer and decomposed by a real-time analyser into the fundamental frequencies. Damping ratios were found by passing the signal through a narrow band filter and measuring the decay rate. The radial/axial coupling was determined primarily by static loading. Further evident was obtained by the detection of the same frequency using accelerometers with axial and radial orientations.

Typical vibration patterns for the wheels are shown in Figures 3 through 6. In each case, P represents the direction and location of the impact and A represents the direction and location of the accelerometer. It is important to note that the scales are different in each figure and that no attempt was made to control the magnitude of the impact. Complete results for the wheels are available in (4) and (5).

The laboratory data was compared to trackside audio recordings taken at a curve on the Toronto Transit Commission subway system. (Refer to Figure 7.) It was apparent that there is strong correspondence between the noise spectrum and the wheel frequencies at 5.24, 11 and 17 kH_z.

The damping ratios for the wheels showed considerable scatter (Table 2). The standard wheel was, for all practical purposes, undamped while the Acoustaflex wheel produced the highest damping ratios. For all wheels except the Acoustaflex, only one sample was available for testing. Four Acoustaflex wheels were tested and these showed considerable variance in damping ratios. It is not known if such variance is typical of resilient wheels or if this behaviour is restricted to the sample tested.

To study radial/axial coupling in more detail, a Bochum wheel and a standard wheel were analysed using a static finite element model. In addition these wheels were subjected to radial and axial loads to determine stress and deflection behaviour. The slope of the web causes a load eccentricity which enhances radial/axial coupling. It was apparent from the model that the position of the load alters this eccentricity and modifies the coupling. Since the railway vehicle is free to oscillate across the rail head, the position of the actual load on the wheel is varied. This effect is one source of variability in the vibration behaviour of wheels (Figure 9).

The effect of load eccentricity can be reduced by various means. Two such methods are shown in Figure 10 and their deflections are compared to a standard wheel. It is apparent that a straight web reduces the deflections of the wheel and that they can be made very small by the use of an "A" frame.

RESULTS FROM TWO PROTOTYPE WHEELS

The investigation into existing practice showed that several design components of railway wheels could be improved. These included:

(a) damping ratios;
(b) radial/axial coupling;
(c) unsprung mass.

Two prototype wheels were designed to demonstrate these improvements (Figure 1). Several new features distinguished them from most existing wheels:

(a) a thin rim to reduce the unsprung mass and attempt to increase the wheel/rail contact area;

(b) a straight web (wheel A) or an "A" frame construction (wheel B) to reduce radial/axial coupling;

(c) aluminum centers to reduce the total wheel mass;

(d) more flexible elastomers to improve the damping ratios; and

(e) bolted construction for easier assembly.

In experiments, the prototype wheels were found to produce high damping ratios (Table 3). It is apparent that, by reducing the wheel rim mass and the elastomer stiffness, the damping ratio will increase for all modes which have a rim displacement component. Since these modes have already been identified as contributors to squeal noise then obviously an improvement is effected. In addition, the use of bolted construction increases the amount of damping available because of frictional effects.

The introduction of the elastomeric blocks near the center of wheel B improved the damping ratios marginally. These blocks reduced both the axial and radial stiffness of wheel B significantly when compared with wheel A. The radial stiffness of wheel A was 320,000 lb./in. (56 041 600 N/M) compared with 80,000 lb./in. (14 011 200 N/M) for wheel B. The axial stiffness measured at the rim of wheel A was 30,000 lb./in. (5 254 200 N/M) compared with only 2,000 lb./in. (350 260 N/M) for wheel B.

CONCLUSIONS

The damping ratios of established railway wheels are low; generally below one percent. Of the wheels tested, the Bochum and Acoustaflex wheels produced the highest damping ratios. The damping ratios of resilient wheels can be improved by judicious selection of mass and stiffness properties. Two prototype wheels were tested to demonstrate this principle.

REFERENCES

(1) May, D.N., "Criteria and Limits for Wayside Noise from Trains", Journal of Sound and Vibration, 46(4), 1976, Academic Press Inc., London, England, pp. 537-550.

(2) Remington, P.J. et al, "Wheel/Rail Noise and Vibration", Report No. UMTA-MA-06-0025-75-10, U.S. Department of Transportation, Urban Mass Transportation Administration, Office of Research and Development, Rail Technology Division, Washington, D.C., 20590, 1975.

(3) Lotz, Robert, "Wheel Rail Noise: A Progress Report", Inter-Noise 74, Washington, D.C., September 30 - October 2, 1974.

(4) Perfect, N., "Vibration Testing of Damped and Undamped Steel Wheels", Report No. 76-TIL-4, Research and Development Division, Ministry of Transportation and Communications, Downsview, Ontario, Canada, 1976.

(5) Perfect, N., "Vibration Testing of Acoustaflex Wheels", Report No. 76-TIL-25, Research and Development Division, Ministry of Transportation and Communications, Downsview, Ontario, Canada, 1976.

Table 1
Four Railway Wheels

WHEEL	MANUFACTURER
Bochum	Bochumer Verein A.G. West Germany
S.A.B.	Svenska Aktiebolaget Bromsregulator Sweden
Acoustaflex	Standard Steel Company U.S.A.
Standard	Canadian Steel Wheel Division Hawker Siddeley Canada Ltd.

Table 2

Frequencies and Damping Ratios of Railway Wheels

COMPONENT	FREQUENCY kHz	DAMPING RATIO			
STANDARD WHEEL	0.62	0.00032			
	1.58	0.00014			
	2.75	0.000073			
	3.99	0.000042			
	5.28	0.000041			
	6.25	0.000041			
BOCHUM WHEEL	0.434	0.0051			
	1.24	0.0031			
	2.20	0.0083			
	4.60	0.0048			
S.A.B. WHEEL	0.433	0.0024			
	1.19	0.0011			
	2.10	0.00083			
	3.07	0.00081			
	4.09	0.00051			
	5.16	0.00039			
ACOUSTAFLEX WHEEL	0.46	0.0062	0.0027	0.0113	0.0044
	1.35	0.0048	0.0168	0.0082	0.0075
	2.45	0.003	0.005	0.0033	0.0029
	3.67	-	0.0043	-	-
	4.96	-	0.0041	-	-

Table 3
Damping Ratios of Two Prototype Wheels

COMPONENT	FREQUENCY	DAMPING RATIO
WHEEL A	330 Hz	0.008
	988 Hz	0.017
	1.83 Hz	0.026
	2.33	0.024
	2.83	0.025
	3.61	0.027
	4.54	0.027
	4.98	0.028
	5.44	0.027
	7.32	0.017
WHEEL B	360 Hz	0.032
	1.00 kHz	0.012
	1.84	0.017
	2.36	0.040
	2.84	0.034
	3.60	0.032
	4.56	0.022
	5.52	0.021
	6.44	0.016
	7.20	0.009

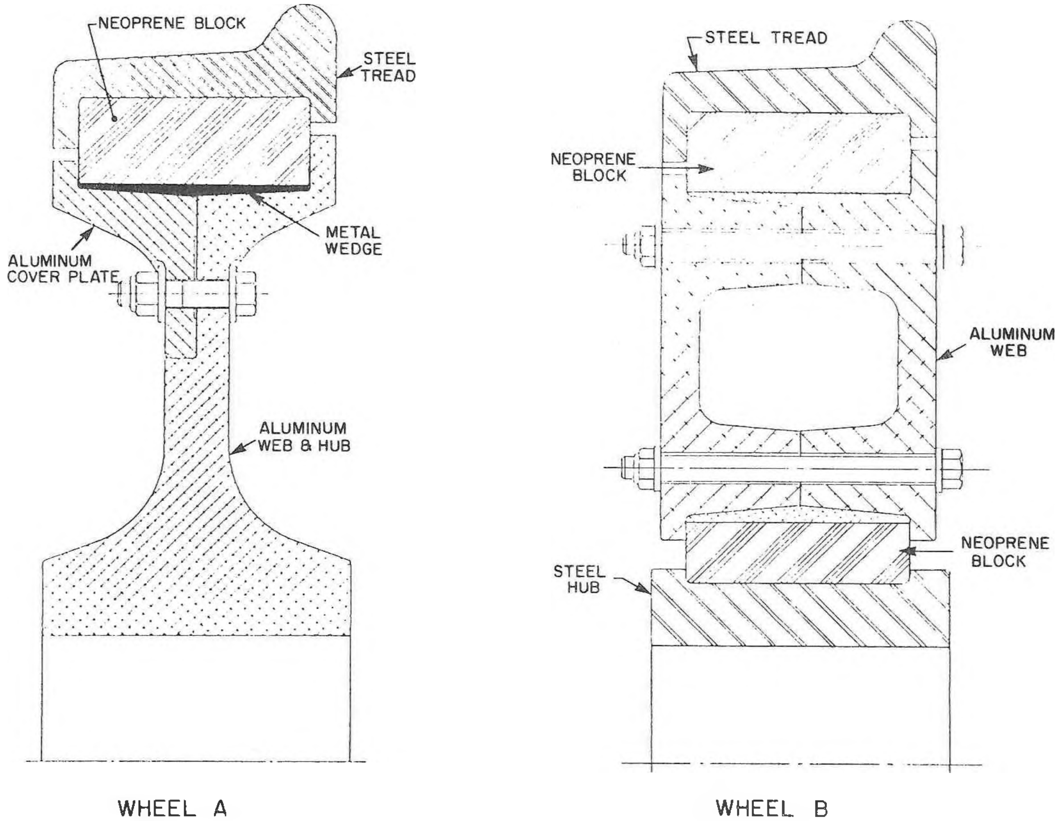


Figure 1, Cross-Section of Each Prototype Wheel

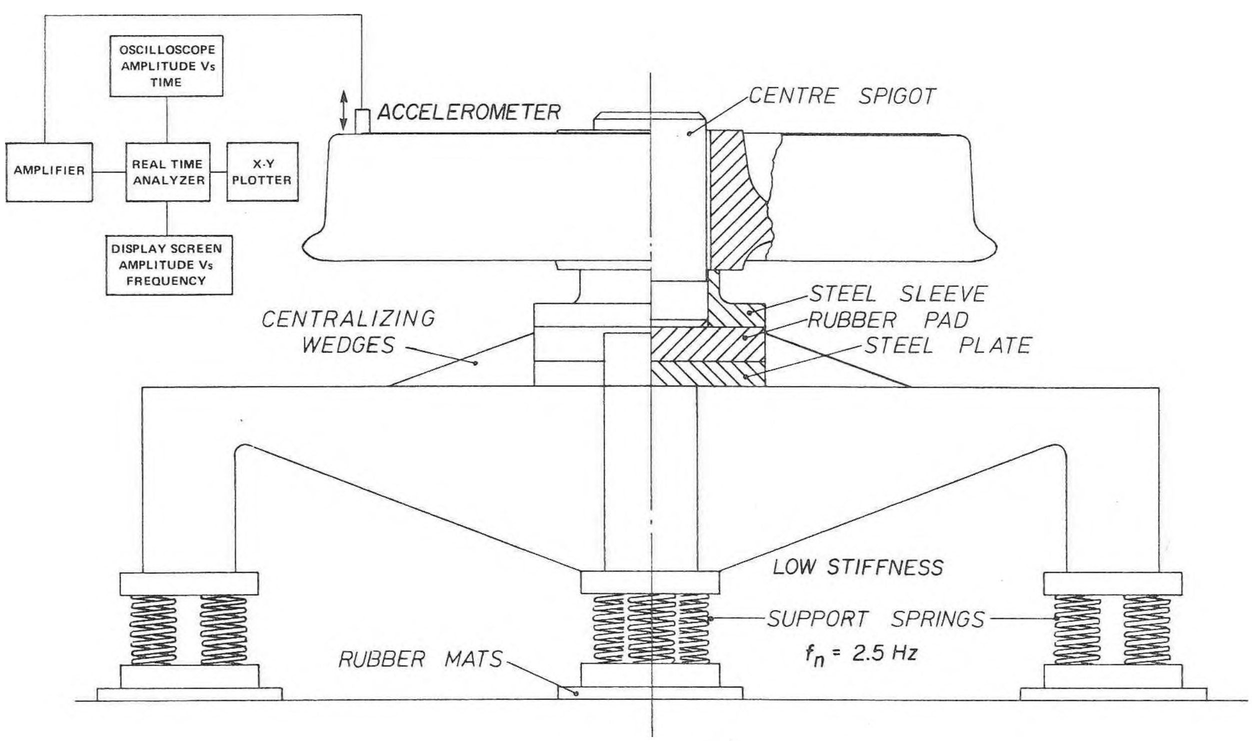


Figure 2, Test Set-Up Used to Find Natural Frequencies of Wheels

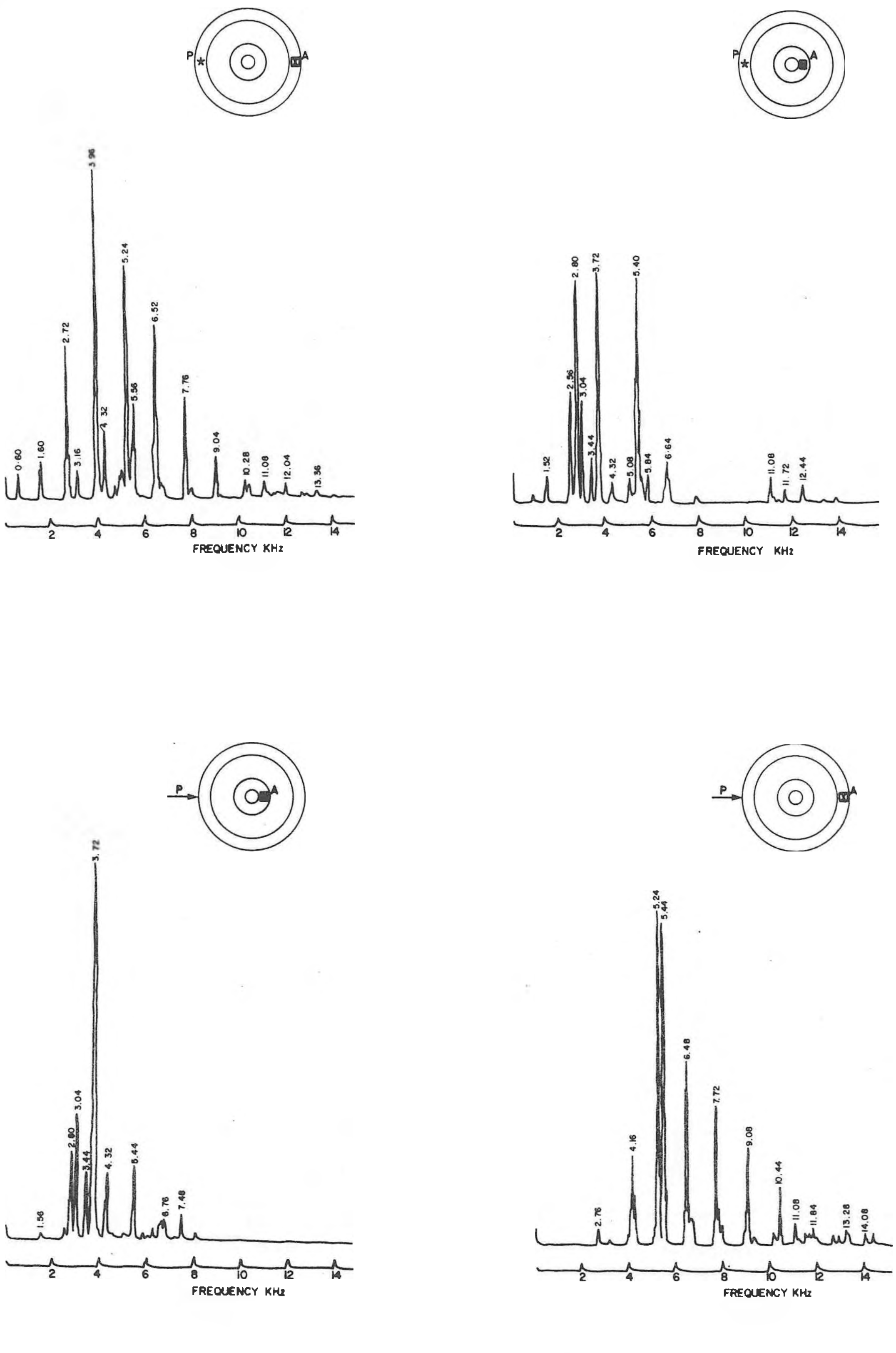


Figure 3, Natural Frequencies of a Standard Wheel

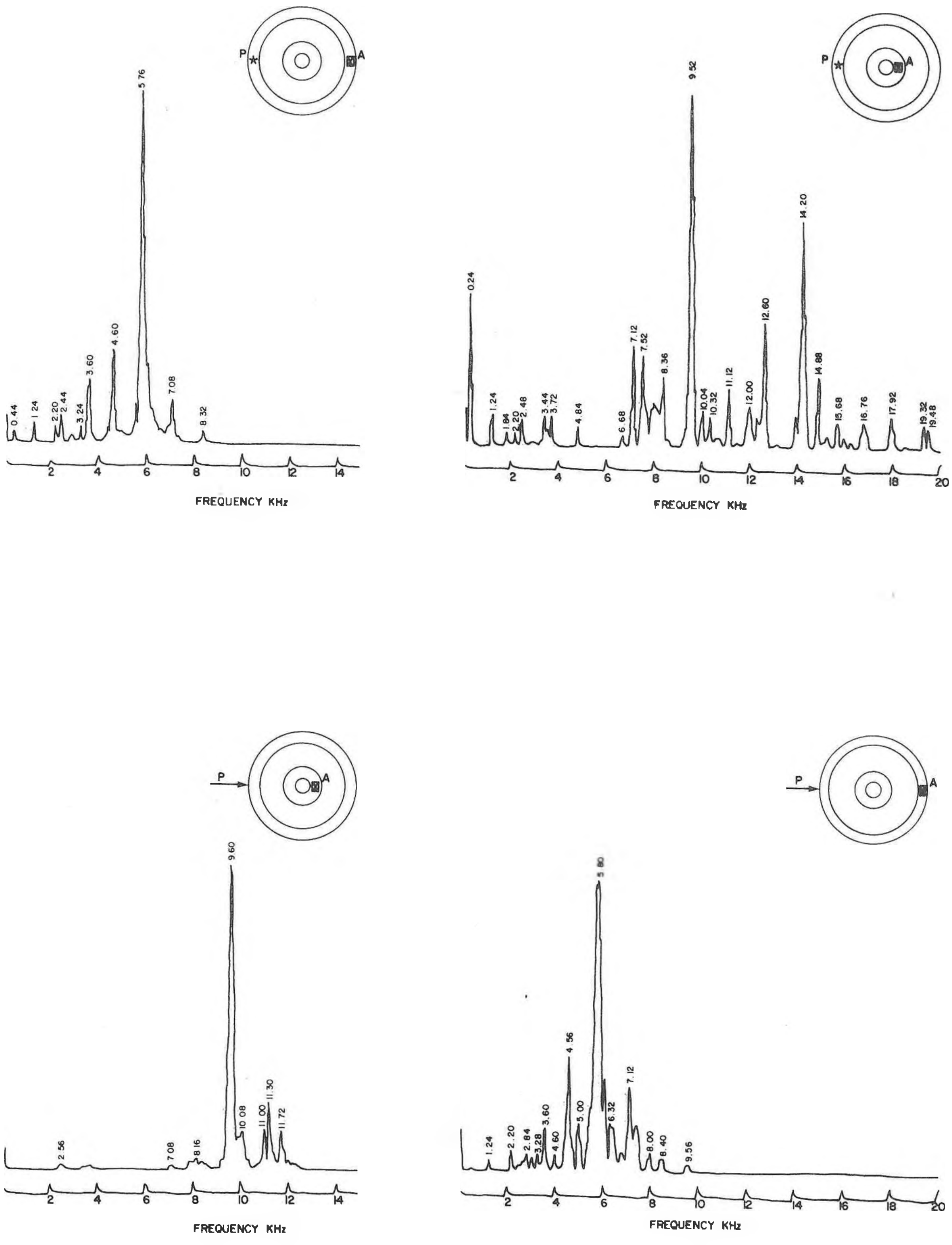


Figure 4, Natural Frequencies of a Bochum Wheel

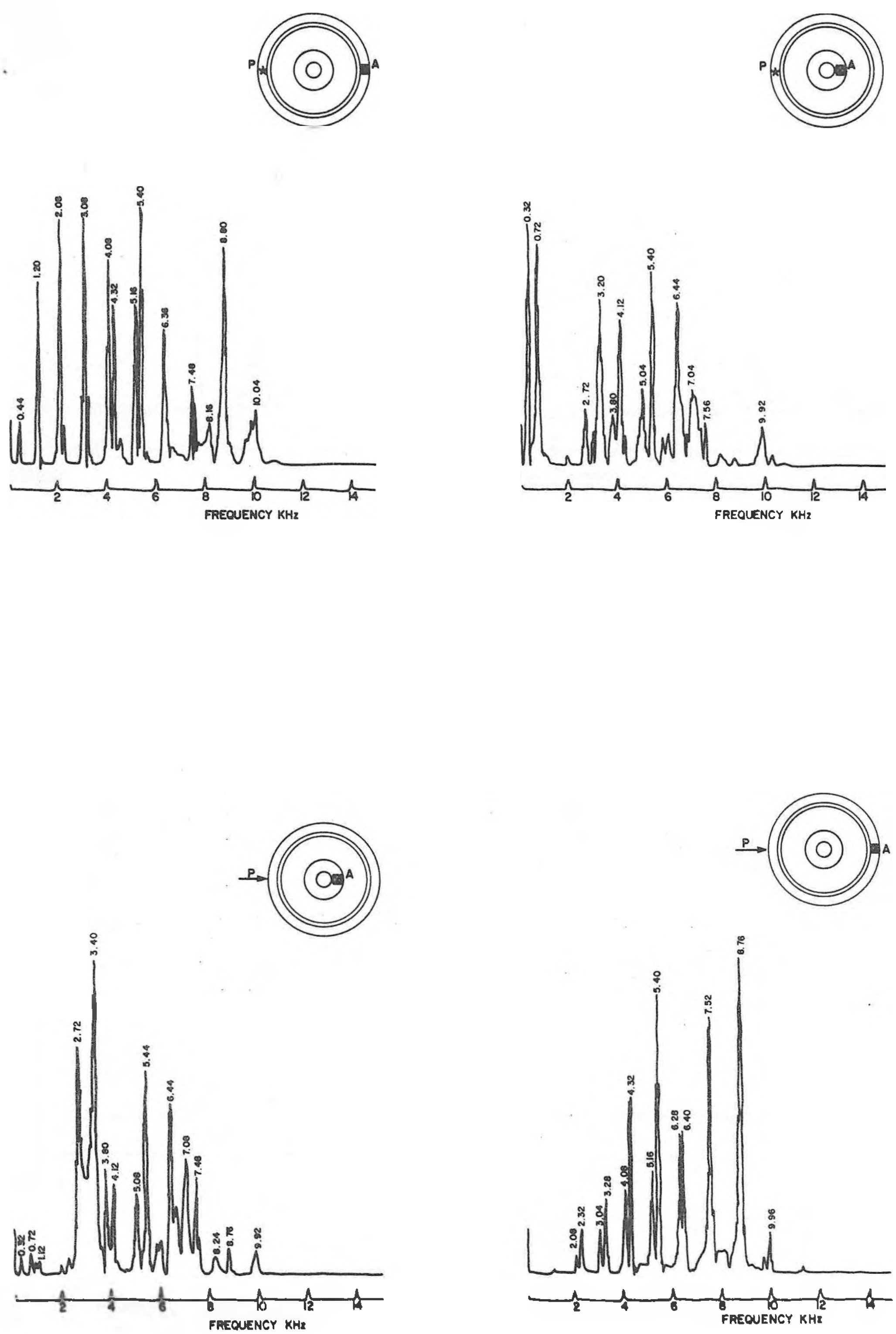


Figure 5, Natural Frequencies of an S.A.B. Wheel

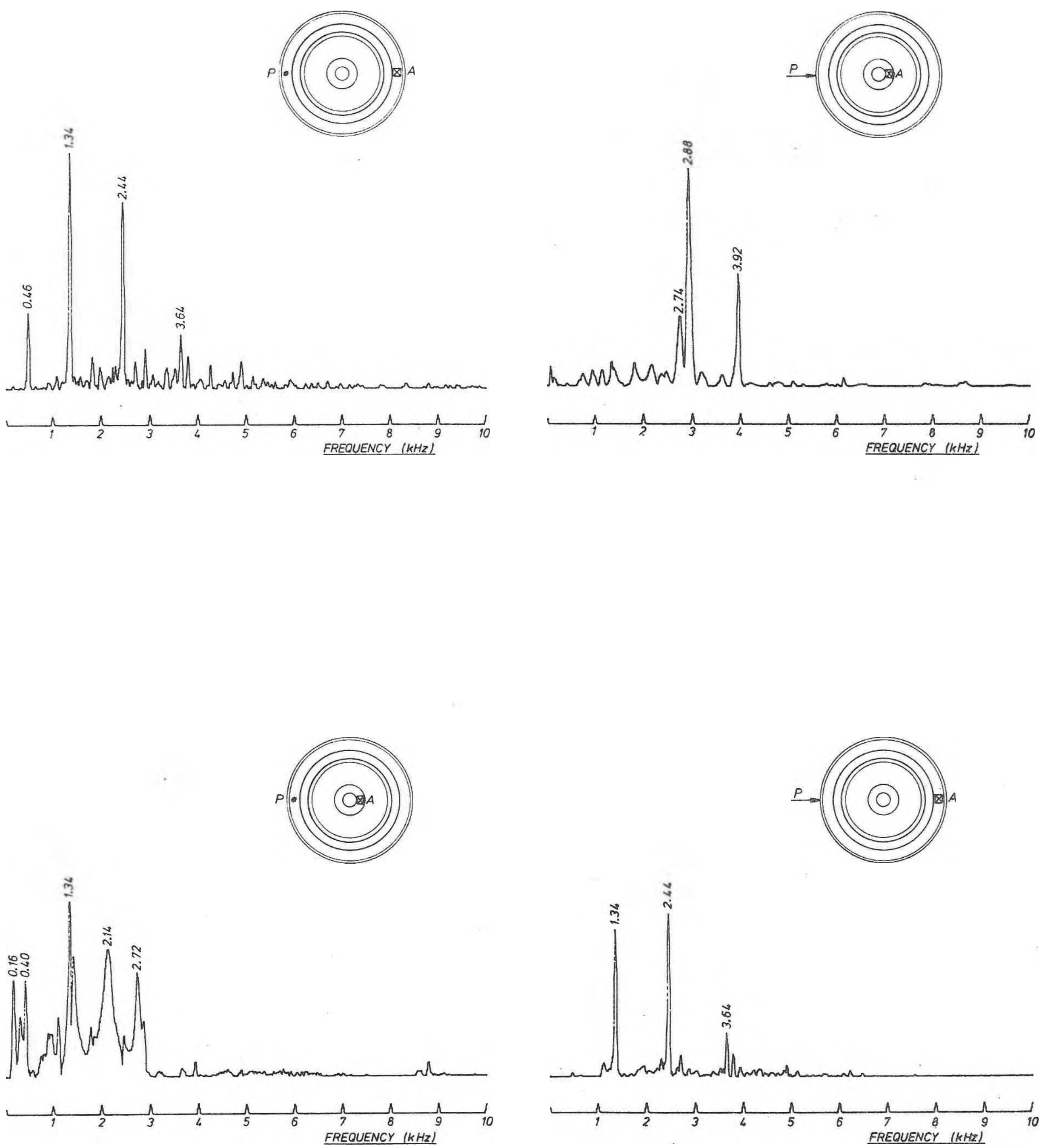


Figure 6, Natural Frequencies of an Acoustaflex Wheel

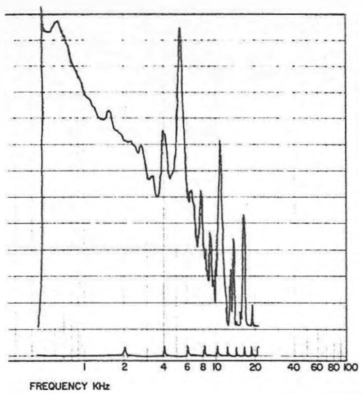


Figure 7a, TTC Train Noise Spectra from Trackside Recording on Curved Track

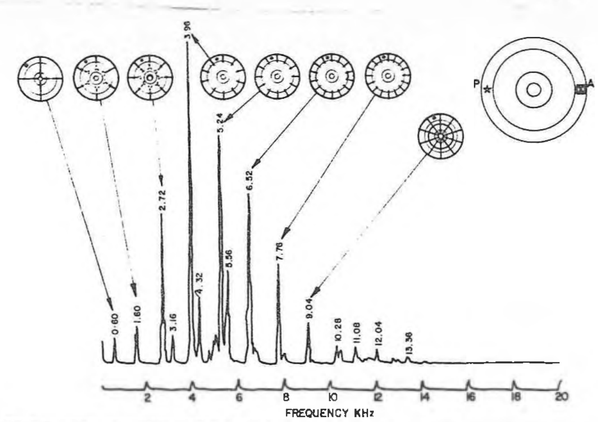


Figure 7b, Modal Patterns of Standard Wheel

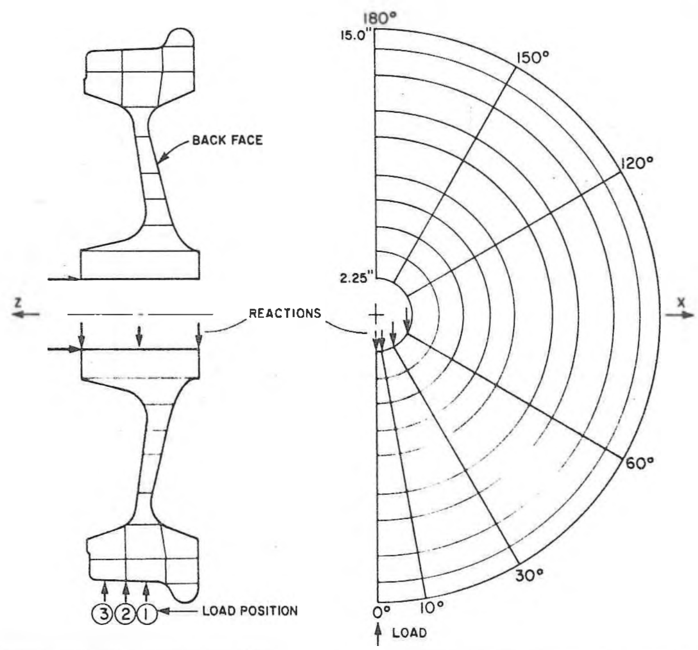
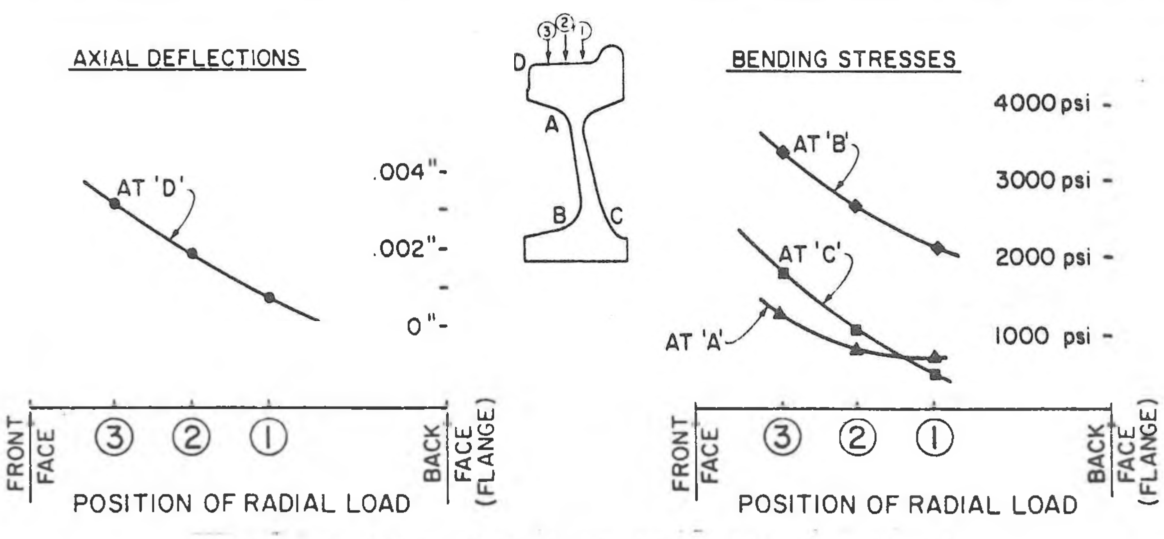


Figure 8, Finite Element Model Used for Analysis



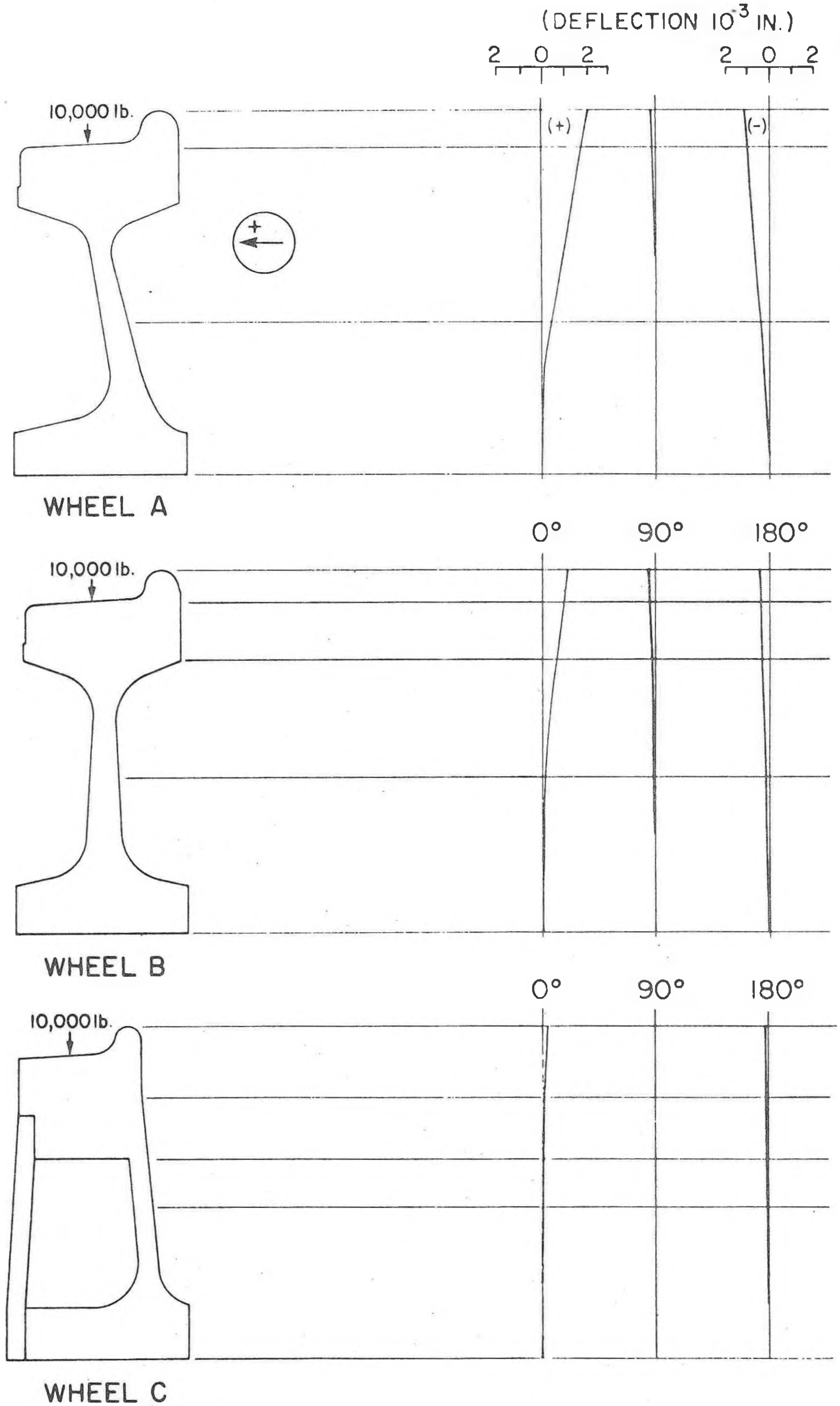


Figure 10, Axial Deflections on Back Face of Three Wheels Due to 10,000 lb. Radial Load at Position 2 of Figure 9