ROAD PROFILE CHARACTERISATION

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EXECUTIVE SUMMARY

Pavement managers usually characterise the condition of a road using a measure known as a roughness index. The index most commonly used in New Zealand is known as the NAASRA roughness index which was developed by the National Association of Australian State Roading Authorities. This index is measured using a standard passenger car travelling at 80 km/h with a simple mechanical device mounted in the rear which measures the accumulated motion of the rear axle relative to the vehicle chassis. Thus this measure reflects the response of a normal passenger car to the road at a reasonable speed.

A number of similar measurement systems and indices exist internationally. To enable these various measures to be compared the World Bank sponsored a project to develop a common index. The result of this was the International Roughness Index (IRI). This index is generated from a measured road profile using a computer model of a simple quarter car. By relating all roughness indices back to the IRI, results from the different measuring systems can be compared. Thus a relationship between NAASRA roughness and IRI has been established. However, these response-based indices reflect the response of a passenger car which is different to that of a heavy vehicle. This is valid because it indicates the ride experienced by passenger cars which make up the majority of road users, but it may not indicate the rate of future deterioration of the pavement which is largely determined by heavy vehicle response. This study investigates whether an alternative roughness index based on heavy vehicle characteristics could better reflect the future rate of pavement wear and thus provide additional information for prioritising pavement rehabilitation and maintenance programmes.

This study uses data obtained for an investigation into assessing heavy vehicle suspensions for road wear. During this investigation, the dynamic wheel forces of a heavy vehicle were measured on five different pavement test sections at several speeds. These dynamic wheel forces can be reduced to a simple statistic known as the dynamic load coefficient (DLC). The road profiles of these test sites were also measured. By splitting the test sites into 200m long sections, it was possible to obtain 68 data points for which the road profile, vehicle speed and DLCs were known. From the road profile information, different forms of roughness index can be calculated and together with vehicle speed fitted to the DLC values. The quality of this fit is a measure of how well this particular index represents the heavy vehicle response.

The first comparison was to use IRI as the roughness index. Using linear regression on DLC with IRI multiplied by speed produced quite a good fit. Alternatives to IRI were first generated by using spring stiffnesses and damping rates based on heavy vehicle characteristics in the IRI computer model. Four of these models were tested based on various approaches to estimating the parameters. Although the response characteristics of these models were more closely related to the response of actual heavy vehicles than that of the IRI quarter car, the quality of fit between these indices and DLC was inferior in all cases to that obtained using IRI. Following this a procedure involving band-pass filtering of the profiles was used to calculate roughness indices. This approach assumes that the dynamic response of heavy vehicles is generated by the natural modes of vibration of the vehicle which, at a particular speed, are stimulated

by particular wavelengths of road profile. Extracting these critical wavelengths with band-pass filters should then give a measure of the probable dynamic response. Although a good fit was obtained using this measure it was still inferior to that obtained using IRI. The final method tested involved replacing the simple spring in the IRI computer model with a non-linear spring whose characteristics reflect the behaviour of steel leaf springs. Again a good fit was achieved but no improvement on the fit obtained using IRI.

IRI provides a reasonably good predictor of heavy vehicle response, using dynamic wheel loads generated by heavy vehicles, and hence of the road wear caused. Although a number of indices were evaluated and tested, no other index provided a better predictor. The two reasons for this are: first, although IRI is based on passenger car characteristics, it has a relatively flat frequency response and will respond to the wavelengths which excite heavy vehicle dynamic behaviour. Second, the road profile data used show no indication of some sites having significantly different frequency distributions to others. If they had this might be expected to generate different levels of heavy vehicle response which relate better to some more frequency specific roughness index than IRI. Thus this result may be a function of the sites used rather than of the whole network.

The NAASRA roughness measurements currently used in New Zealand have several disadvantages. The measurement system is based on a passenger car which involves considerable investment and requires relatively frequent replacement. As the dynamic system involved (the vehicle's suspension) is complex, maintaining calibration is difficult and expensive. By having a measure based on suspension response rather than directly on road profile, the options for investigating other aspects of road roughness are very limited.

Basing a roughness measurement system on a trailer configuration is suggested, as it would have a longer life, be simpler to calibrate, easier to replicate and cheaper. The measurement system should be electronic and computer-based to improve reliability and storage.

Ideally the system should be based on measuring road profiles rather than vehicle response as it could then be used to extract any number of roughness indices merely by changing the analysis procedures.

ABSTRACT

Roughness indices are used to monitor road condition and prioritise maintenance and rehabilitation programs. The most commonly used measure in New Zealand is NAASRA roughness which is based on the dynamic response of a normal passenger car. It does not necessarily reflect the dynamic response of heavy vehicles which produce most of the pavement wear and hence may not reflect the rate of future deterioration of the pavement. Therefore a roughness index based on heavy vehicle response would provide useful additional information for pavement managers. The report describes attempts to develop such an index.

The International Roughness Index (IRI), determined using a computer simulation of a simplified quarter car passing over the road profile, is used to calibrate NAASRA roughness meters. Thus IRI was used as the reference roughness index for this study. Other indices were generated by: using the IRI model with heavy vehicle-based parameters, replacing the simple spring with a non-linear leaf spring model, and using band-pass filtering directly on the road profile data.

IRI provides a reasonably good predictor of heavy vehicle response, using dynamic wheel forces generated by heavy vehicles, and hence of the road wear caused, than other indices evaluated and tested.

Basing a roughness measurement system on a trailer configuration is suggested as it would have a longer life, be simpler to calibrate, easier to replicate and cheaper. The measurement system should be electronic and computer-based to improve reliability and storage.

Ideally the system should be based on measuring road profiles rather than vehicle response as it could then be used to extract any number of roughness indices merely by changing the analysis procedures.

1. INTRODUCTION

1.1 International Roughness Index

Longitudinal road profile is commonly characterised using a simple numeric value which is called the roughness index. These indices are generally obtained using either a "response-type road roughness measurement system" (RTRRMS) or by measuring the actual profile (profilometry) and calculating the index. RTRRMS is by far the most common. To date, numbers of these systems have been developed in many countries resulting in a diversity of roughness indices. Consequently, in 1982 the World Bank sponsored a research project known as the International Road Roughness Experiment (IRRE) to develop a common roughness index which is internationally acceptable, time-stable, transportable and readily measurable by all practitioners. The outcome of this project is a measure known as the International Roughness Index (IRI) which is defined in terms of the response of a mathematical model rather than a mechanical system. Thus the reference IRI measurement is obtained by applying a computation procedure to road profile data.

1.2 NAASRA Roughness Measure

In New Zealand, the most commonly used measure of road roughness is that developed for the National Association of Australian State Road Authorities (NAASRA, now AUSTROADS). The NAASRA roughness meter consists of a light chain, one end of which is attached to the centre of a live rear axle in a standard station wagon. The other end of the chain is led through the floor of the vehicle and over a sprocket mounted on the floor. The sprocket is connected via a ratchet to a simple mechanical counter. Thus the device measures the accumulated relative motion of the axle and the body of the vehicle in counts. NAASRA roughness is defined in counts/km. Originally the system was operated by manually recording the counter readings and the vehicle odometer readings as the vehicle travelled along the road at 80 km/h.

Some users have now developed computer-based recording systems to log the odometer and the NAASRA meter. The system is based on a particular vehicle (a VK Holden Commodore station wagon) and for best results care is taken to ensure correct tyre inflation, shock absorber damping rates and ballast weight (i.e. axle loading). By using a live rear axle the effects of the road profiles in the left and right wheel paths are effectively averaged.

1.3 Relationship of IRI to NAASRA Roughness

With the development of the IRI, the Australian Road Research Board (ARRB) investigated the relationship between NAASRA roughness and IRI (Prem 1989). It developed a vehicle mounted laser-based profilometer system for measuring road profiles at near highway speeds (50-80 km/h). This system is capable of measuring the profiles of both the left and right wheel paths as well as that in mid-vehicle simultaneously. The mid-vehicle profile is used to measure the degree of rutting present in the wheel paths. From this profile data Prem was able to calculate the IRI values for pavements for which the NAASRA roughness index was also available and establish the relationship between the two.

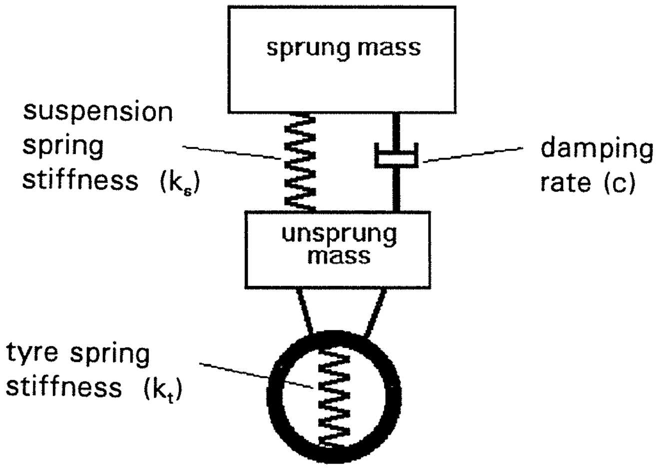


Figure 1. IRI quarter car model.

IRI is based on the response of a simple "quarter car" model, shown in Figure 1, consisting of a sprung mass, connected by means of a linear spring and a viscous damper (corresponding to the suspension), to an unsprung mass which is in turn connected by a linear spring (corresponding to the tyre) to the road profile. By normalising the system equations by the sprung mass value the model is reduced to four characteristic parameters: suspension stiffness rate (k_s); damping rate (c); tyre stiffness rate (k_t); unsprung mass ratio (mu). The IRI definition uses values which are considered representative of passenger cars. The model is excited using the pavement profile of a single wheel path at 80 km/h as the input displacement and the equations of motion are solved to obtain the suspension motion. The magnitude of the suspension motions are accumulated and divided by the distance travelled to obtain the IRI value. Where the road profile data are available at small horizontal intervals some smoothing is done prior to the model excitation to account for the enveloping effect of the tyre.

Prem (1989) compared IRI with NAASRA roughness for his test sections in a number of ways. In all cases he used least squares regression and tried to fit a second order polynomial to the data and, in nearly all cases, found that the second order coefficient was not statistically significant. Using the profile data from only one of his three tracks (inner, outer or mid) to calculate IRI he still obtained a good correlation between NAASRA roughness and IRI ($r^2 > 0.93$) with the mid track being slightly better than the other two. By combining the data from the two wheel paths an even better fit ($r^2 > 0.99$) was obtained. This was done in two ways: the first, which he calls the lane quarter car IRI, was obtained by calculating the IRI values for each of the two wheel paths and averaging them; the second, which he calls half car IRI, was obtained by averaging the two wheel path profiles and calculating the IRI from this averaged profile. Based on the results of this study, the laser profilometer is now used to calibrate NAASRA roughness meters. The measured road profiles are used to calculate a half car IRI which is converted to NAASRA roughness using the linear relationship that Prem calculated.

1.4 Use of IRI and NAASRA Roughness Values

Roughness indices are used to monitor the condition of pavements and as an indicator of the need for repair or rehabilitation. Both NAASRA roughness and IRI are based on passenger car type response and are intended to reflect the roughness as observed by the occupants of ordinary passenger cars who are generally the majority of road users.

However, it is well established that traffic-related pavement wear can be substantially attributed to heavy vehicles. Furthermore, the dynamic response characteristics of heavy vehicles are typically different to those of ordinary passenger cars.

Heavy vehicle suspensions are required to cope with much greater variations in loading (a fully laden heavy vehicle may weigh four times as much as the same vehicle unladen). Ride comfort is generally not as high a priority and considerable importance is placed on durability, robustness, low cost and weight. The net effect is that heavy vehicle suspensions are usually stiffer and less well damped than those of passenger cars.

If the usual roughness indices (NAASRA or IRI) do not accurately reflect the heavy vehicle response then, while they might reflect the current state of the pavement, they will not reflect the rate of future wear. The purpose of this project is to investigate whether a heavy vehicle based roughness index can be developed to resolve this issue. It is not envisaged that this new index would replace the existing indices but rather would provide additional information for the pavement manager to use in prioritising the maintenance programme.

1.5 A Heavy Vehicle-based Roughness Index

As part of a Transit New Zealand-funded project to investigate assessing suspension performance with respect to road damage (de Pont in press), a liquid tanker three axle trailer with steel leaf spring suspension (very typical of the New Zealand heavy vehicle fleet) was extensively instrumented to measure wheel forces and suspension behaviour and then tested at three different speeds at each of five test sites. As a full trailer rather than a semi-trailer was used, the suspension characteristics of the towing vehicle had no influence on the trailer's vertical dynamic response. The road profiles of those sites were also measured with the ARRB laser profilometer. The scope of this present study is limited to using the data from this project together with other published results. No further experimental work was undertaken.

The first stage of this project consists of analysing the data to establish the basis for developing the new index. The second stage consists of using this information to postulate the new index and then using the data to check its validity. Having established a successful index, a concept design is presented for a device to measure it.

2. HEAVY VEHICLE RESPONSE CHARACTERISTICS

RTRRMS roughness measuring systems are generally based on passenger car (or similar) suspension responses. Heavy vehicle suspensions are generally proportionally stiffer while heavy vehicle tyres, although stiffer in absolute terms, are proportionally more flexible. The overall result is vehicle response characteristics that are markedly different.

To determine the exciting frequencies of heavy vehicles that contribute most to dynamic wheel forces data collected during vehicle tests which investigated assessing heavy vehicle suspensions for road damage (de Pont in press), were analysed. Applying Fourier transforms to the wheel force signals calculated from the data collected during the vehicle road tests, shows the frequency components of those signals (Figures 2-6).

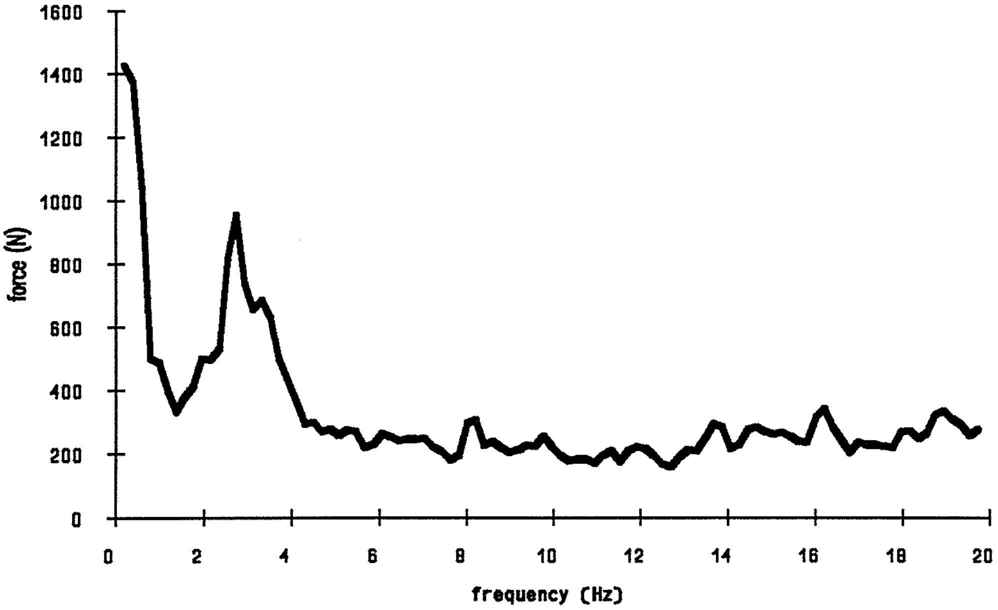


Figure 2. Wheel force spectrum for test on Mill Road, Manukau City, Auckland.

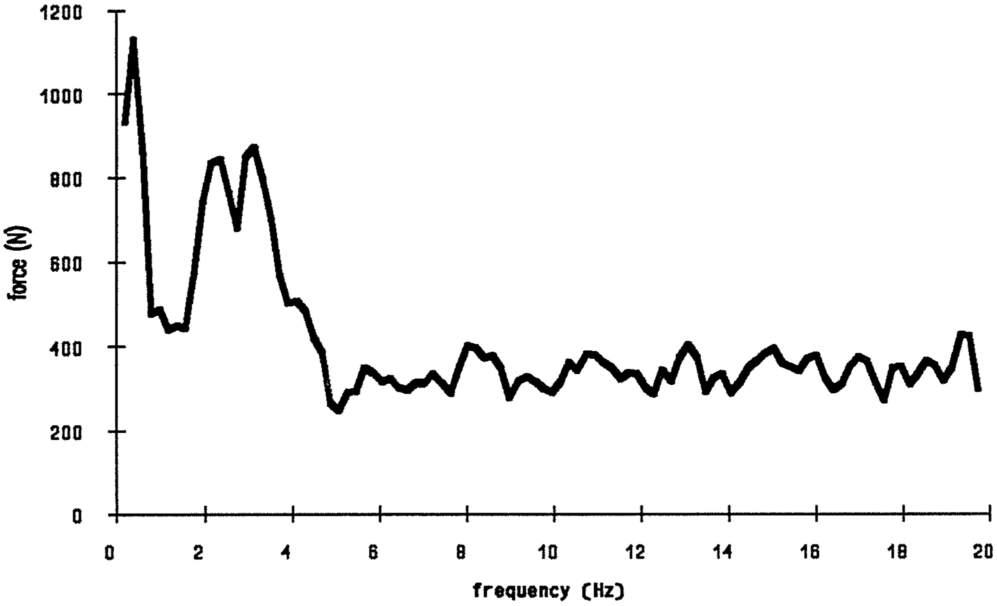


Figure 3. Wheel force spectrum for test on Takanini-Clevedon Road, Manukau City, Auckland.

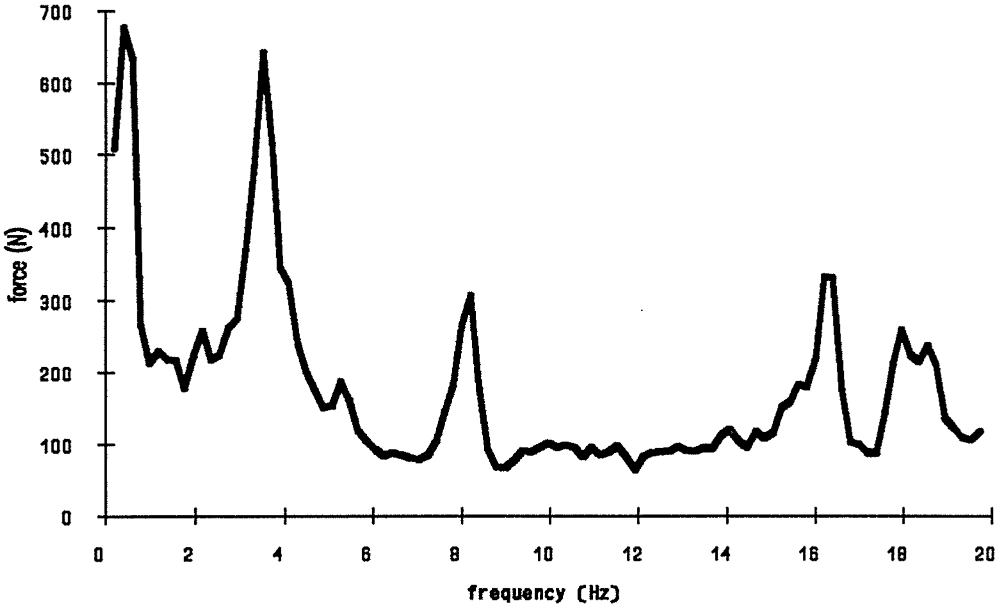


Figure 4. Wheel force spectrum for test on Southern Motorway, near Takanini, Auckland.

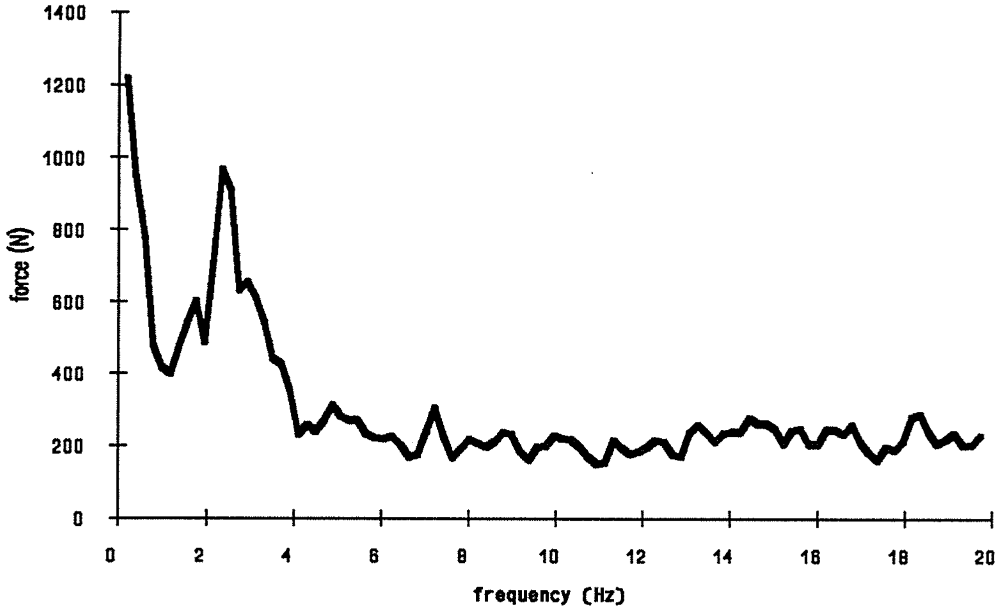


Figure 5. Wheel force spectrum for test on Airfield Road, Manukau City, Auckland.

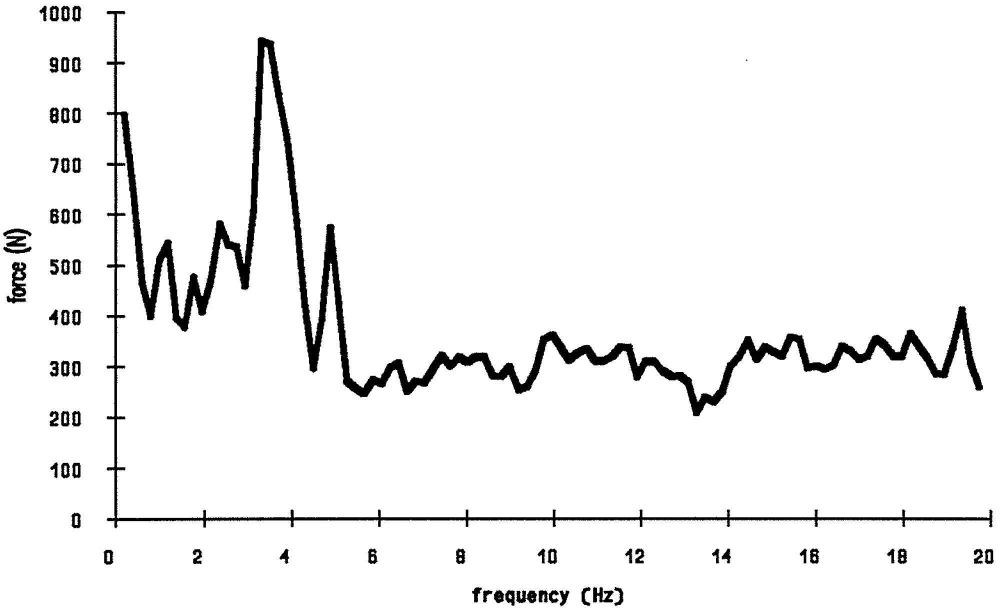


Figure 6. Wheel force spectrum for test on Church Street, Onehunga, Auckland.

The major contributions to wheel forces on the pavement occur in the range 2 - 4 Hz which are generated by the body mode resonances of the vehicle. Note that the mean frequency of this peak tends to reduce with increasing road roughness and consequently with increasing magnitude. Thus for the rougher roads around Auckland (Mill Road and Airfield Road, Manukau City), the maximum wheel force component occurs at around 2.7 Hz while for the smoother roads (Auckland Southern Motorway) it occurs at 3.5 Hz. The Takanini-Clevedon Road site has a mixture of smooth and rough sections, while Church Street, Onehunga, which is relatively rough was tested at lower speeds and thus appears to behave more like a smooth road.

This is not surprising if the physical characteristics of the suspension are considered. Steel leaf springs suffer from stiction so that at small displacements they are significantly stiffer than at larger displacements. The resonance frequencies of a dynamic system are proportional to the square root of the spring stiffnesses.

A group of resonances also occurs in the range 13 - 17 Hz, although these are of relatively small amplitude and are most obvious in the motorway site data. These correspond to resonances of the axles. The magnitude of the wheel force contributions of the resonances is small compared to the body mode contributions. For some wheels on some tests an additional peak occurs at 7.5 - 8.5 Hz which is typically of the same magnitude as the axle resonance peaks. This frequency corresponds approximately to the wheel rotation speed (for Church Street with its lower test speeds this is approximately 4.5 - 5.5 Hz) and so this is probably caused by tyre eccentricities.

Works Consultancy Services Ltd (Brown and Cenek 1989a, b) have undertaken suspension compliance tests on vehicles that did not conform to the standard regulations on heavy vehicle suspensions. In order to be permitted to operate these vehicles, the owners were required to demonstrate that the suspension performance met a number of criteria. To estimate the wheel forces generated by these vehicles, accelerometers were mounted on the axle and on the chassis at each wheel under consideration and monitored during a series of road trials. The wheel forces were calculated by summing the chassis acceleration multiplied by a sprung mass component with the axle acceleration multiplied by an unsprung mass value. No spectral analysis of the wheel force signals as such was undertaken, but the individual accelerometer signals were analysed. For the Iveco tractor unit test reported in Brown and Cenek (1989a) the resonance modes of the vehicle body occur in the 2 - 4 Hz range and those of the axle occur at 9 - 15 Hz. It is impossible to determine the magnitude of the force spectra terms from the acceleration spectra but it would appear that the lower frequency terms dominate. A second study on a mobile crane unit (Brown and Cenek 1989b) measured resonant modes around 1 - 1.5 Hz and around 8 - 10 Hz. However, this is a special purpose vehicle and its behaviour is not typical of the heavy vehicle fleet.

As Brown and Cenek's data represent a very small sample of heavy vehicles, a literature review was undertaken to expand and validate this information. Whittemore et al. (1970) measured dynamic wheel forces on three different vehicles. For the two single axle vehicles they found that the major source of dynamic wheel loads came from the vehicle body modes around 2.6 -3 Hz with a smaller high frequency component (12 - 20 Hz), particularly on rougher roads. The third vehicle tested had a tandem axle with a walking beam suspension. For this vehicle at the higher speed test, they found a high amplitude component at 11 Hz, in addition to the low frequency component. This was attributed to the poorly damped oscillation of the walking beam.

Cebon (1985) reported that dynamic forces are observed in two main frequency ranges, 0.7 - 4 Hz corresponding to the sprung mass rigid body motions and 8 - 15 Hz corresponding to the unsprung mass rigid body motions.

Woodrooffe et al. (1986) compared the dynamic wheel forces of a number of suspensions. They reported that "for suspensions showing high dynamic activity, the majority of the wheel force is from the vertical response of the whole vehicle body which is in the frequency range of 1.5 to 3.5 Hz".

Mitchell (1987) compared the performance of air and steel springs on the same tandem axle semi-trailer using a road simulator to provide the excitation. He found that with steel springs the dynamic wheel forces consisted primarily of the 4 Hz vehicle bounce (with some energy at 1 Hz due to roll and at 2.2 Hz due to pitch) and a small wheel hop component at 18 Hz. With air suspension, the bounce mode frequency was reduced to 1.8 Hz with significantly reduced amplitude, but the wheel hop component at 10 - 15 Hz was increased. The overall effect was reduced dynamic wheel forces.

Hahn (1987) calculated theoretical resonances for different suspensions based on the manufacturers' quoted technical specifications and found natural frequencies of 1.34 - 2.1 Hz for the vehicle bounce mode and 9.7 - 12.8 Hz for the axle bounce mode. However, during road tests he observed higher frequencies. For example, a tandem axle parabolic steel leaf spring system with a theoretical resonance of 2 Hz was observed to exhibit resonance at 2.5 Hz during tests on "good" and "medium-good" pavement.

Overall it appears that dynamic wheel forces occur in two frequency ranges corresponding to the rigid body vibrations of the sprung and unsprung masses respectively. These ranges are typically 1 - 4 Hz and 9 - 18 Hz. Generally, stiffer suspensions such as steel four-springs and walking beams tend to have frequencies at the higher end of these ranges and components in the lower frequency range dominate. Softer suspensions, such as air, tend to be at the lower end of the frequency ranges but the contribution of the high frequency components is more significant. These suspensions usually have a lower total dynamic load variation. The European Community "equivalent-to-air" regulations specify a natural frequency below 2 Hz which some of the modern steel parabolic leaf spring units are achieving. In the New Zealand situation, the vehicle fleet is fitted predominantly with the stiffer more robust suspensions and thus it is reasonable to assume that most of the dynamic wheel forces impacting on pavements come from sprung mass responses in the 2 - 4 Hz range.

3. PAVEMENT PROFILE CHARACTERISTICS

The dynamic loadings of vehicles on pavements are the result of the interaction of the vehicle as a dynamic system with the pavement profile and so pavement characteristics are an important factor. As part of an earlier project (de Pont in press), the ARRB laser profilometer was brought to New Zealand to measure the profile and roughness characteristics of the test sites used. Modifications were made to the profilometer so that the track width of the measurements reflected heavy vehicles rather than passenger cars. These data were analysed to determine the frequency distribution of the longitudinal profiles. The test sites were subdivided into 200m long subsections to increase the number of points for comparison. Fast Fourier Transforms (FFTs) were done on profile data for each of these subsections at each site. These are plotted as amplitude vs wavenumber (1/wavelength), that is independent of vehicle speed and shown in Appendix 1.

By multiplying the wavenumber (cycles/m) by the vehicle speed (m/sec) the ordinate axis can be converted to frequency (cycles/sec). The amplitude values are generally larger throughout for rougher roads. There are shape differences in the spectral functions with some pavement sections containing more low wavenumber components than others. However, there is no clear pattern to these differences.

4. IRI CALCULATION

To complete the preliminary information gathering, software to calculate IRI values was coded and validated. Initially the algorithm to calculate IRI published in Sayers et al. (1986b) which is written in BASIC, was coded in and modified to accept profile data input from a file. However, because of unacceptably long computation times (17.5 minutes on a 16MHz 386SX for a 1 km test site from PE/25), the algorithm was recoded in Turbo C (reducing computation time to below 2 minutes). The software has been extended to calculate IRI values for both left and right wheel paths and a "half car" IRI value based on the average profile as described in Sayers et al. (1986a). The NAASRA roughness for the site is also calculated using the relationship developed at ARRB (Prem 1989).

The algorithm has been applied to the road profile data supplied by ARRB for the sites used by de Pont (in press) and compared with the IRI and NAASRA values calculated by ARRB for the same sites. The correspondence between the two sets of figures is very good though not exact. The largest discrepancies (which are less than 1%) occur at the start and finish of the data and are probably the result of the ARRB calculations using profile data from before and after the site limits in their calculations for smoothing. This information is not in the data files containing the profile values. For the other sections the correspondence is better than 0.1%.

5. A ROUGHNESS INDEX BASED ON HEAVY VEHICLE RESPONSE

5.1 Introduction

The principal aim of this project is to establish a roughness index which better reflects heavy vehicle responses than currently used measures and thus predicts the rate of future deterioration as well as the present state of the pavement. To validate whether a new index is better or not it is necessary to have measure of vehicle response to relate it to. In this work the measure used is dynamic load coefficient (DLC) which was used by Sweatman (1983), and others since, to characterise the dynamic wheel force behaviour of heavy vehicles. It is defined as:

$$DLC = \frac{standard\ deviation\ of\ wheel\ force}{static\ load}$$

From the wheel force signals used earlier to determine the vehicle response characteristics, DLC values can be calculated for each 200m of each test site at each vehicle speed tested. To assess the usefulness of the proposed indices, attempts were made to relate the DLC values calculated for a representative wheel to the roughness values as determined by the index. This was done by regression analysis and then comparing the quality of the fit.

5.2 IRI

As a reference, the first analysis was done using the IRI values as the roughness index. Sweatman (1983) found in his work that a linear regression fit of DLC to $VR^{0.5}$, where V is the vehicle speed in km/h and R is the road roughness in NAASRA counts/km, produced a good fit. NAASRA roughness meters are now calibrated using profile measurement, a "half car" IRI calculation followed by a linear transformation to obtain the NAASRA value using coefficients determined experimentally (Prem 1989). Effectively, NAASRA roughness is now defined in terms of IRI. Thus, rather than use the NAASRA roughness, IRI was used as it is a more fundamental measure.

The advantage of including the vehicle speed in the variable to fit to DLC is that all the data can be used in the fitting procedure. Speed can be eliminated as a variable if only data from tests at the same speeds are considered. For the three principal test speeds, 75, 80 and 85 km/h, a reasonable number of data points (17 or 18) was available but at the other test speeds, 45, 50, 55 and 100 km/h, only one test site was involved which generated only 4 or 5 data points.

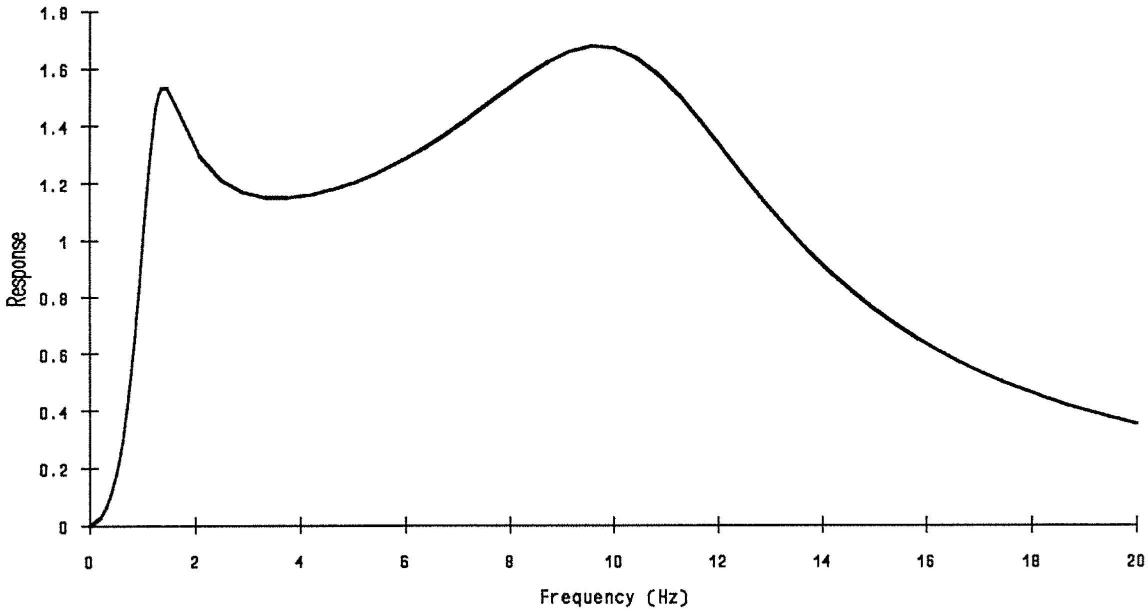
Using the data from all the test sites, VR produced a better regression fit than $VR^{0.5}$, though in this case R is the IRI value rather than the NAASRA value. The quality of fit statistic r^2 was 0.89. Using the data in groups of equal speed and thus eliminating V as part of the independent variable it was again found that using R as the variable produced a better fit than using $R^{0.5}$ in all cases. For all cases, except the low speed tests (45, 50 and 55 km/h) which were done at the same urban road site, the quality of fit as given by the r^2 statistic was good, being 0.93 - 0.98. However, for the low speed test it was poor. Removing the low speed data from the urban site and repeating the fit against VR improved the fit quality ($r^2 = 0.94$). The reason for the poor correlation in the urban road results is unclear.

The correlation between DLC and VR is quite good. From the point of view of developing a heavy vehicle-based index which is a better predictor of heavy vehicle response and hence future road damage, this raises two issues. First, it will be difficult to find an index which is better because IRI is quite good, and second, even if a better index is established, will it be worth the effort of measurement as the gain in information will be marginal?

The DLC values used for the above regression come from one wheel of the vehicle which is located on the left hand side while the IRI values used are generated using a half car model which averages the left and right pavement profiles. It might be expected that the DLC for a left hand wheel would depend more on the left hand road profile than the right and thus that a better correlation would be obtained by doing a multi-linear regression between DLC and VR for both the left and right profiles. This was not the case. The r^2 values for the multilinear fit were very similar but slightly worse than those obtained above.

Before proceeding to develop a new index it is worth investigating IRI further to determine its fundamental characteristics. IRI is calculated using a quarter car model as shown earlier in Figure 1. The variables are normalised by the sprung mass value and thus four variables remain, the normalised suspension stiffness, k_s, the normalised tyre stiffness, k_t, the normalised damping rate, c, and the unsprung mass ratio, mu. For the IRI quarter car, $k_s = 653$, $k_t = 63.3$, $c = 6$ and $mu = 0.15$. From these values the natural frequencies of the quarter car model are easily calculated. They are: 1.21 Hz which is the sprung mass bounce mode and 11 Hz which is the unsprung mass bounce mode. These values are in line with normal passenger car behaviour. However, the damping rate corresponds to approximately 0.38 of critical damping which is relatively high. Normal passenger car damping characteristics are complex with the rates dependent on the direction of motion (i.e. different bump and rebound rates) but both are normally lower than 0.38. The high damping level used for IRI is intended to reflect the behaviour of RTRRMSs. The frequency response of the suspension compression magnitude for the IRI quarter car model is shown in Figure 7.

Figure 7. Suspension deflection frequency response for IRI quarter car.



Although peaks corresponding to the two natural frequencies are clearly visible, the high level of damping means that the frequency response between these peaks is also significant. Thus, IRI is a weighted average of all the frequencies present in the road profile between approximately 0.5 and 20 Hz. It is also worth noting that IRI does not directly reflect either the wheel forces applied to the pavement or the passenger "comfort". The wheel forces applied to the pavement are indicated by the tyre compression in the IRI model. Figure 8 shows the tyre compression frequency response of the IRI model. To show the actual wheel force distribution seen by a pavement the tyre response must be multiplied by the pavement spectra. By looking at the spectra of the pavement profiles it can be seen that a general function of the form:

$$P_{spectra} = a + \frac{b}{w} + \frac{c}{w^2}$$

gives a function of approximately the correct shape.

Figure 9 shows the tyre compression response multiplied by this typical pseudo-pavement profile spectrum to give a typical pseudo wheel force frequency response as given by the IRI quarter car.

Figure 8. Tyre compression frequency response for IRI quarter car.

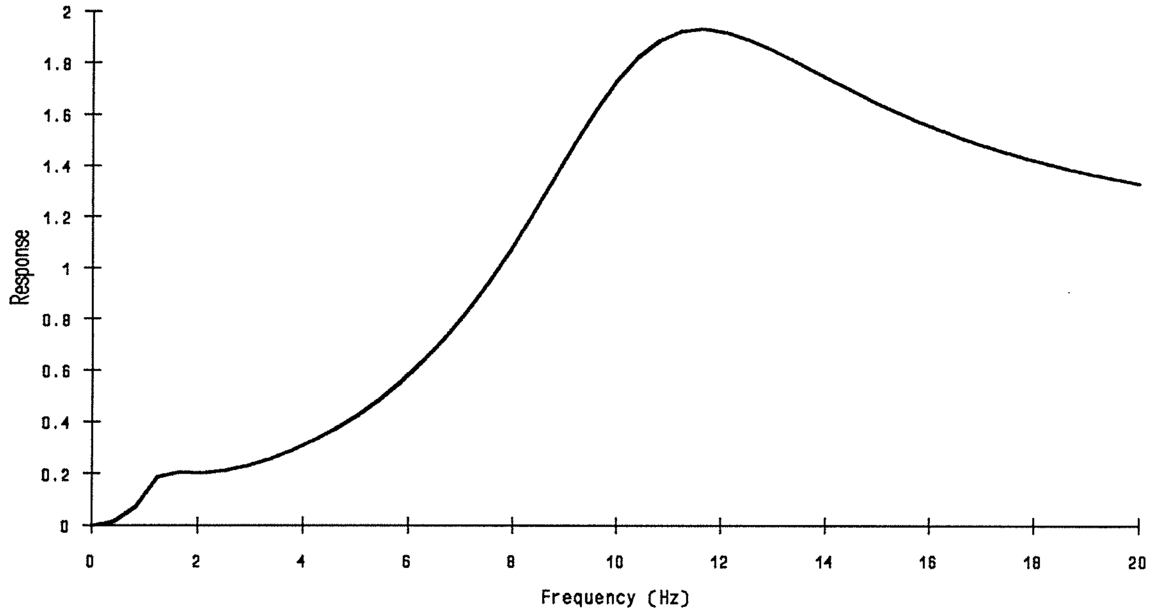
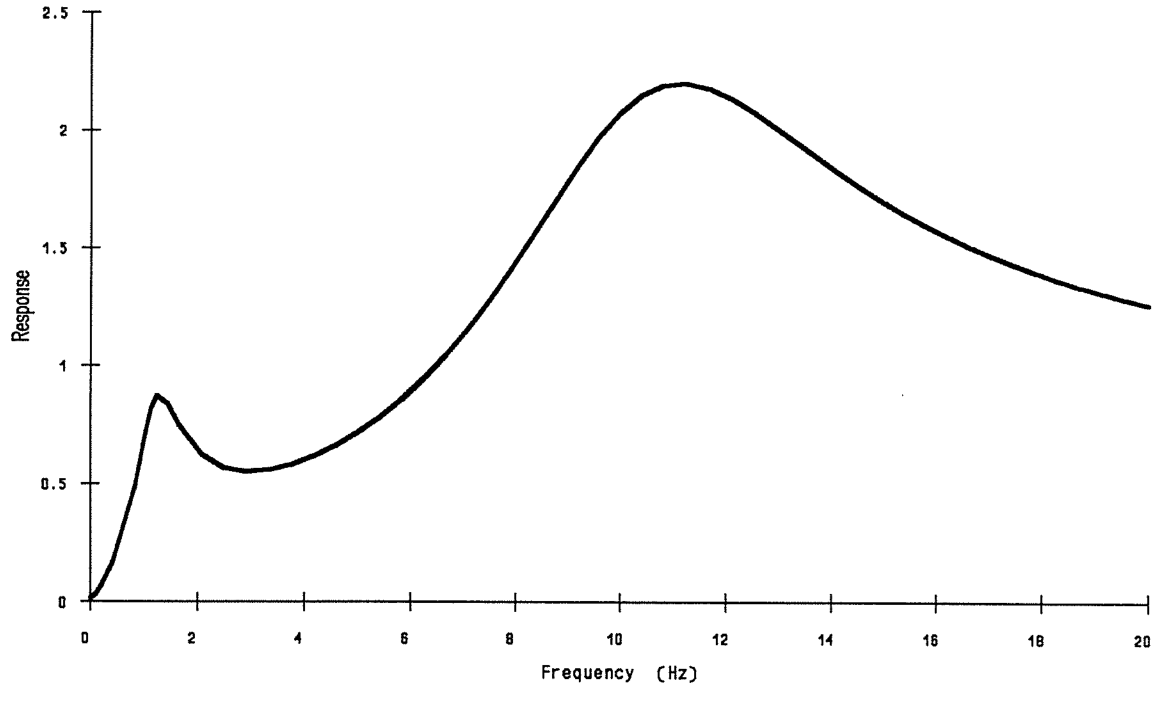
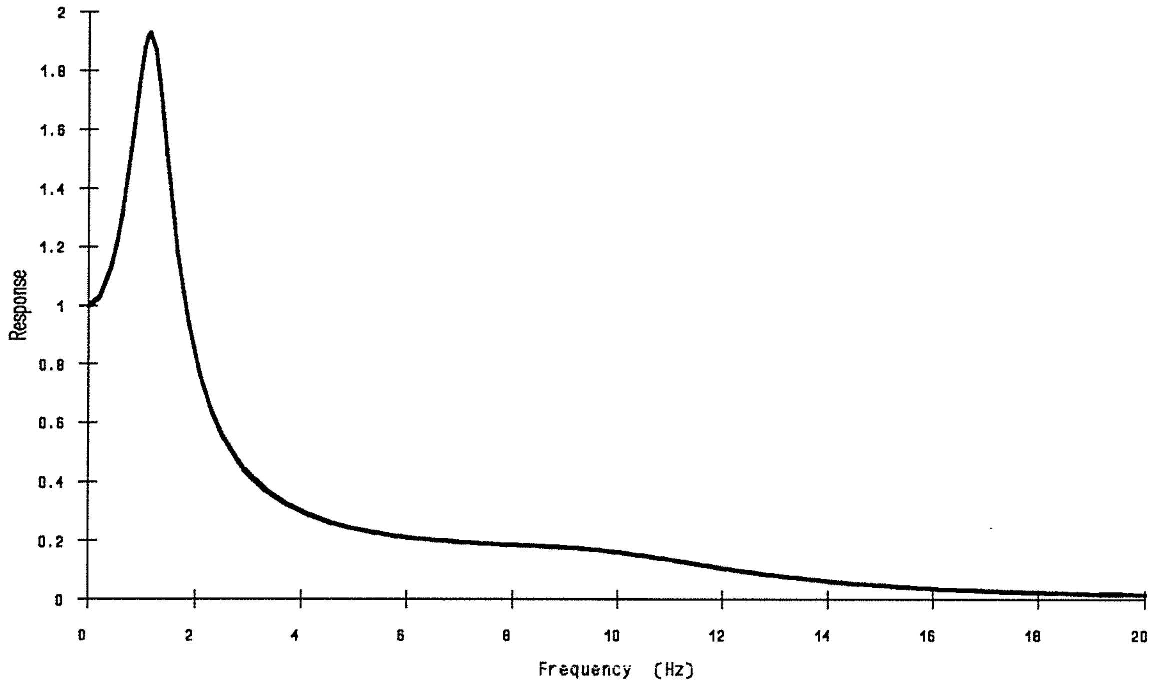


Figure 9. Pseudo wheel force frequency response for IRI quarter car.



The passenger "comfort", on the other hand, is a function of the sprung mass motion which, for the IRI vehicle, is shown in Figure 10.

Figure 10. Sprung mass frequency response for IRI quarter car.



5.3 The "Prem" Quarter Truck Index

The logical extension of the IRI quarter car model to an index which reflects heavy vehicle responses is to generate an analogous index using stiffness, damping and mass ratio coefficients which reflect the "typical" heavy vehicle. Prem (1987) gives a set of values for these parameters which he says are generated from Sweatman (1983) and further tests undertaken at ARRB.

The values Prem specifies are $k_s = 413$, $k_t = 381$, $c = 1.6$, and mu = 0.19 where the notation used is exactly the same as previously. With these parameters, the suspension deflection frequency response (which determines the roughness index) is that shown in Figure 11 while the expected wheel force response (tyre compression multiplied by a pseudo-profile spectra) is shown in Figure 12. This model has a high and narrow response peak at the primary sprung mass resonance which occurs at 2.2 Hz. A smaller secondary peak, corresponding to the unsprung mass resonance can be seen at 10.5 Hz. Because of the relatively low level of damping in this model, the response peaks are narrow and thus closely tuned to both the vehicle parameters and the vehicle speed.

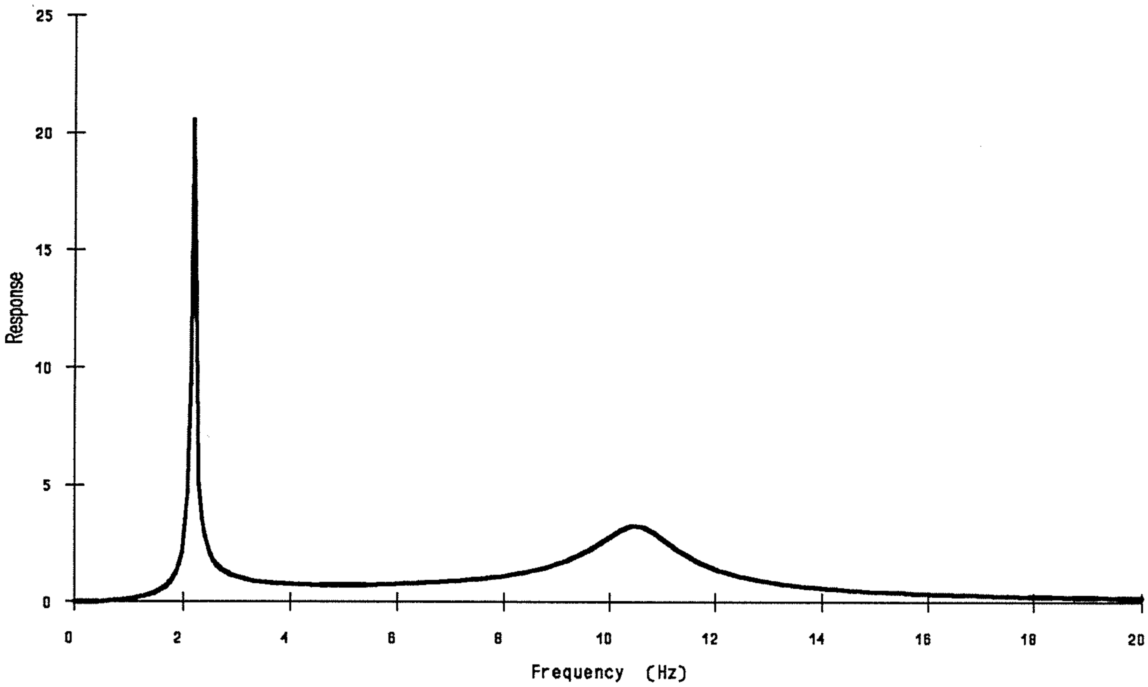


Figure 11. Suspension deflection frequency response for Prem quarter truck.

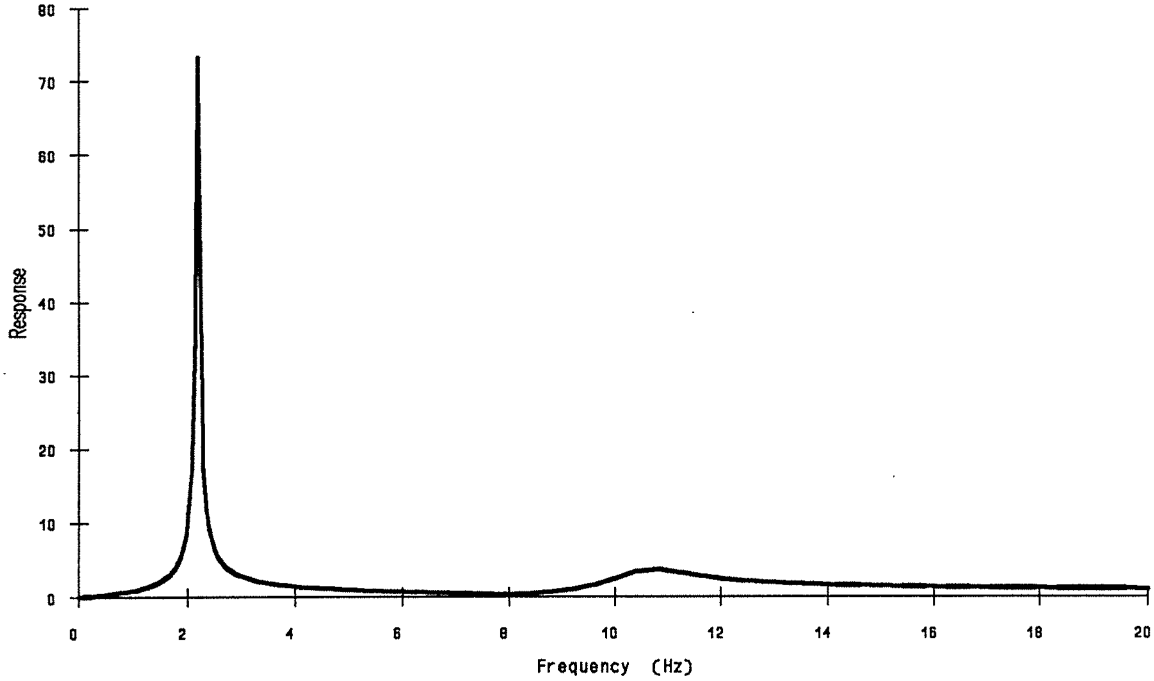


Figure 12. Pseudo wheel force frequency response for Prem quarter truck.

Using the index values generated by this model the process of fitting DLCs to roughness as undertaken for the IRI quarter car was repeated. In this case the fit was poor. As before, VR generated a better fit than $VR^{0.5}$ but the fit correlation statistic r^2 is only 0.54. Somewhat surprisingly, this roughness measure generated a better fit for the urban road which produced the poor correlations to IRI in Section 5.2. For the tests on this road, the r^2 statistic was 0.8. However, the index calculation is based on a vehicle speed of 80 km/h, while the road tests at this site were undertaken at 45, 50 and 55 km/h. The wavelengths which excite 2.2 Hz and 10.5 Hz resonances at 80 km/h would excite 3.5 Hz and 16.8 hz resonances at 50 km/h which are close to those of the test vehicle. Thus, the "Prem" quarter vehicle has coincidentally tuned in with the DSIR (now IRL) test vehicle responses at the particular speed. However, this is not a generalised effect as the fit at other speeds is poor.

As with the IRI analysis the fitting process was repeated using a multilinear regression on the roughness indices calculated independently for the left and right wheel paths. In this case the correlation improved with $r^2 = 0.62$. However, the fit is still not good and this index does not appear to be worth pursuing.

5.4 The "Heath" Quarter Truck Index

Heath (1987) reported on work on vehicle simulation models where the simplest model was a quarter vehicle with the parameters, $k_s = 270.27$, $k_t = 472.97$, $c = 2.072$ and $mu = 0.1486$. Using these in the IRI roughness program, a new set of roughness indices was calculated and the fitting procedures described above were repeated.

These vehicle parameters give resonance modes at 2.03 Hz and 11.6 Hz. The frequency response of the model in terms of roughness indices is shown in Figure 13. As can be seen, the damping is higher than for the "Prem" model and so the response peaks are not so narrow or so high. The effect of this is that the response is not so tightly tuned to specific vehicle characteristics or speeds.

The fit of the test vehicle DLC values to the roughnesses generated by this model is significantly better than for the "Prem" model. For the DLC vs VR fit for all the available data an r^2 value of 0.80 is obtained. As with the IRI fit (p. 20), the data from the low speed urban test site deviated most from the fit. Excluding these low speed data from the calculation improves the r^2 value to 0.83. Note, however, that this fit is still significantly inferior to using only the IRI roughness index.

Using both the left and right profile roughnesses and a multilinear regression procedure as for the previous two indices resulted in a considerably improved correlation with $r^2 = 0.86$ for all data and $r^2 = 0.90$ if the urban road data are eliminated.

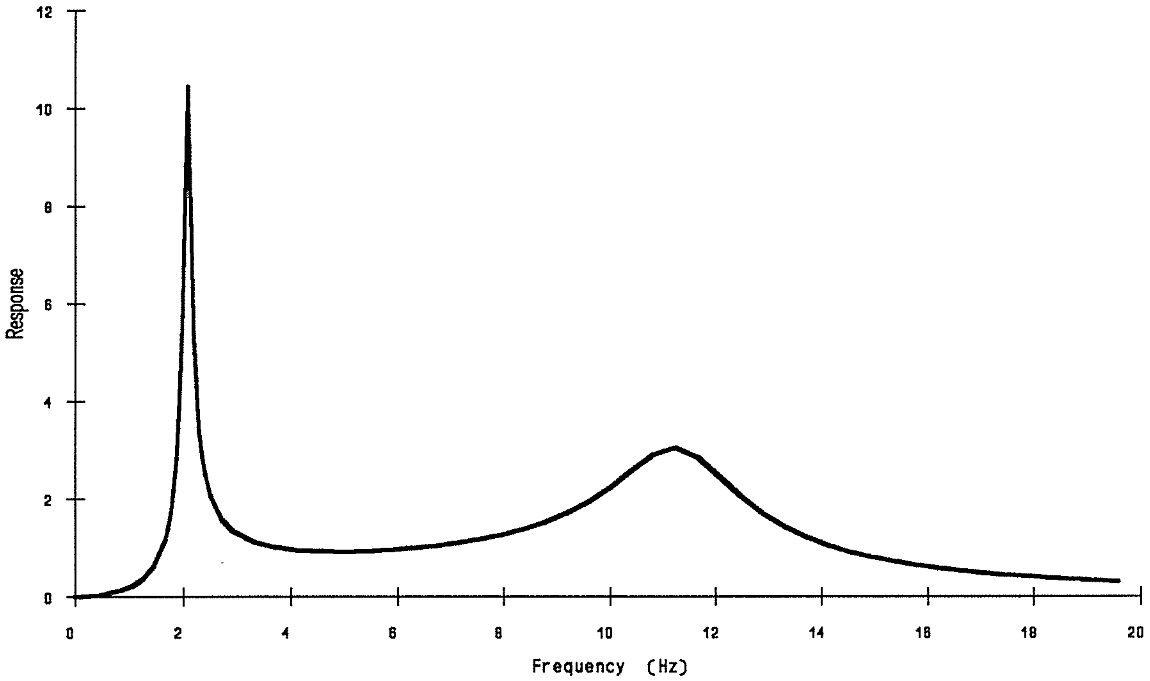


Figure 13. Suspension deflection frequency response for Heath quarter truck.

5.5 The "Factbook" Quarter Vehicle Index

The University of Michigan Transportation Research Institute (UMTRI) has prepared a "factbook" (Fancher et al. 1986) of heavy vehicle parameters for the US Department of Transport. This publication summarises the ranges of vehicle parameters observed at UMTRI over many years of vehicle testing and research. From these data, vehicle parameters based on measured values can be selected for the roughness index quarter vehicle model. The values chosen were $k_s = 237$, $k_t = 427$, and mu = 0.15 which are typical with stiffnesses at the high end of the range.

The issue of the damping ratio is more complex. The viscous damping for heavy vehicles is low, but the friction contribution is high. Friction damping can be approximated as an equivalent viscous value and thus a value of $c = 88$ was used. This damping is in fact greater than critical and shifts the location of the maximum frequency response up. The frequency response curve is shown in Figure 14. As can be seen the high damping has swamped the unsprung mass resonance, while the sprung mass resonance has a relatively broad characteristic compared to the two previous models with a peak around 2.8 Hz. The undamped sprung mass resonance of this vehicle is at 1.9 Hz.

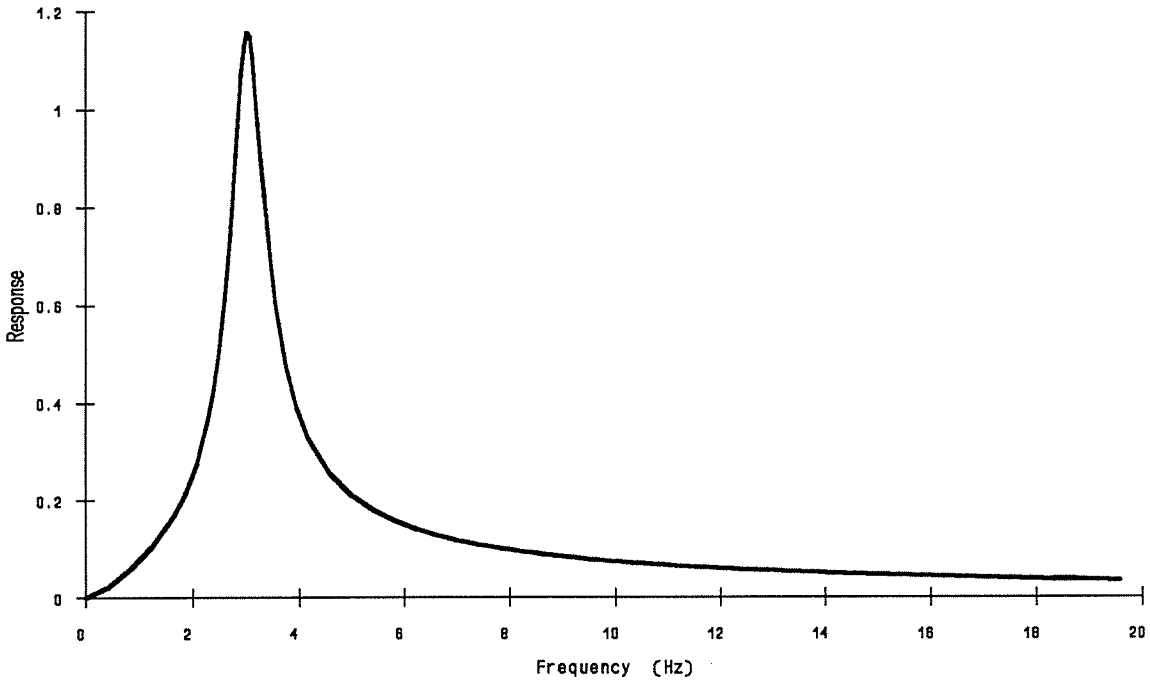


Figure 14. Suspension deflection frequency response for "Factbook" quarter truck.

Fitting the DLC data to roughness indices as before resulted in r^2 statistics slightly superior to those of the Heath model. For the fit of DLC vs VR for all data the $r^2 = 0.81$ while removing the urban site data improved it to $r^2 = 0.85$. Thus, this model provides reasonable results yet is still inferior to the original IRI.

Using independent left and right wheel path roughness indices and multilinear regression as before gives an $r^2 = 0.88$ rising to 0.90 if the urban site data are ignored. Again this is slightly better than using the "Heath" parameters.

Although the upper end of suspension stiffness values were used in this model the undamped resonances were still relatively low when compared with those observed in measured vehicle behaviour. In discussions with Winkler (pers. comm. 1992), one of the "Factbook" authors, it was established that the suspension stiffness values are incremental stiffnesses at relatively high deflections and that because of friction the small deflection stiffnesses are substantially higher. The validity of using the viscous approximation to the friction damping when the level of friction is so high is also doubtful.

5.6 The "Tuned" Quarter Truck Index

To obtain natural frequencies closer to those observed in the road tests the stiffness parameters were adjusted arbitrarily. Values of $k_s = 500$, $k_t = 600$, $c = 6$ and $mu = 0.15$ were chosen giving resonant responses of 2.57 Hz for the sprung mass and 13.9 Hz for the unsprung mass.

The frequency response of the roughness index with these parameters is shown in Figure 15. Although this has a similar appearance to the actual vehicle response, the fit obtained when fitting the observed DLC values to this roughness index were only fair. For the fit of DLC to VR for the complete data set the r^2 statistic was 0.73 while removing the results from the urban site improved it to 0.77. The probable reason for this is that the roughness index is calculated using a specific vehicle speed (80 km/h), while the road test data were obtained at a range of speeds. This range of speeds, however, are realistic and probably reflect fleet behaviour. It is also undesirable to tune the model too closely to a specific vehicle as it may then not reflect the response of other vehicles.

As with the previous three cases, applying a multilinear regression using roughness indices for both the left and right wheel paths gave better correlations with $r^2 = 0.79$ for all data improving to $r^2 = 0.82$ without the data from the urban site.

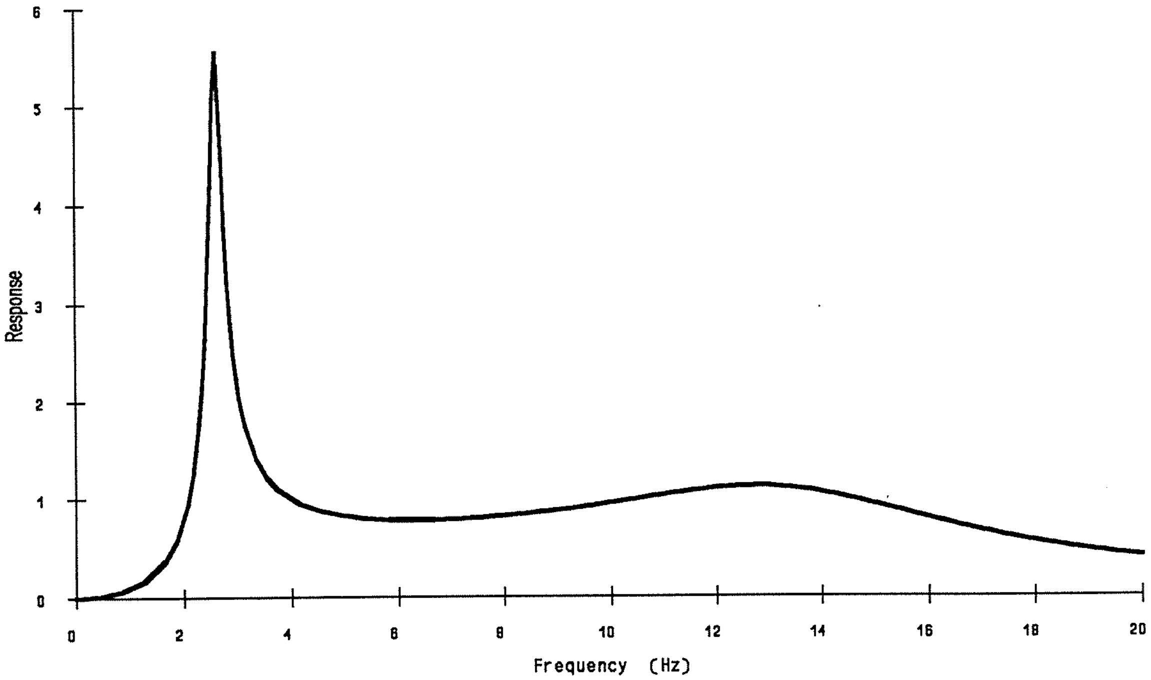


Figure 15. Suspension deflection frequency response for "tuned" quarter truck.

5.7 The UMTRI Non-linear Spring Model

The IRI quarter vehicle model used for all the previous roughness indices is based on simple linear springs. This type of model is always an approximation when representing a physical system as it assumes a linear response, i.e. constant stiffness, and no energy absorption (damping) by the spring. In the case of heavy vehicles with steel multi-leaf springs these assumptions are not valid. The interleaf friction means that the stiffness varies substantially depending on spring motion, and the spring absorbs considerable energy (so much so that additional damping is not normally required). Some of the previous models have attempted to compensate for this spring damping by increasing the damping rate of the viscous damper in the model. However, this is not correct as the damping characteristics of a friction-based system are quite different from those of a viscous system.

A model of steel multi-leaf spring behaviour has been developed at the University of Michigan Transportation Research Institute (UMTRI) (Fancher et al. 1980, Sayers and Winkler, pers. comm. 1992). This model provides an incremental method of calculating the spring force which matches the measured behaviour of steel leaf springs. By incorporating this model as a replacement for the linear spring in the quarter vehicle model it is possible to calculate a new set of roughness index values.

As the response of this system is non-linear, the frequency response is not independent of the excitation and so it is not possible to generate a generalised response function. However, Figure 16 shows the frequency response function of this model calculated as the transfer function from a particular road profile.

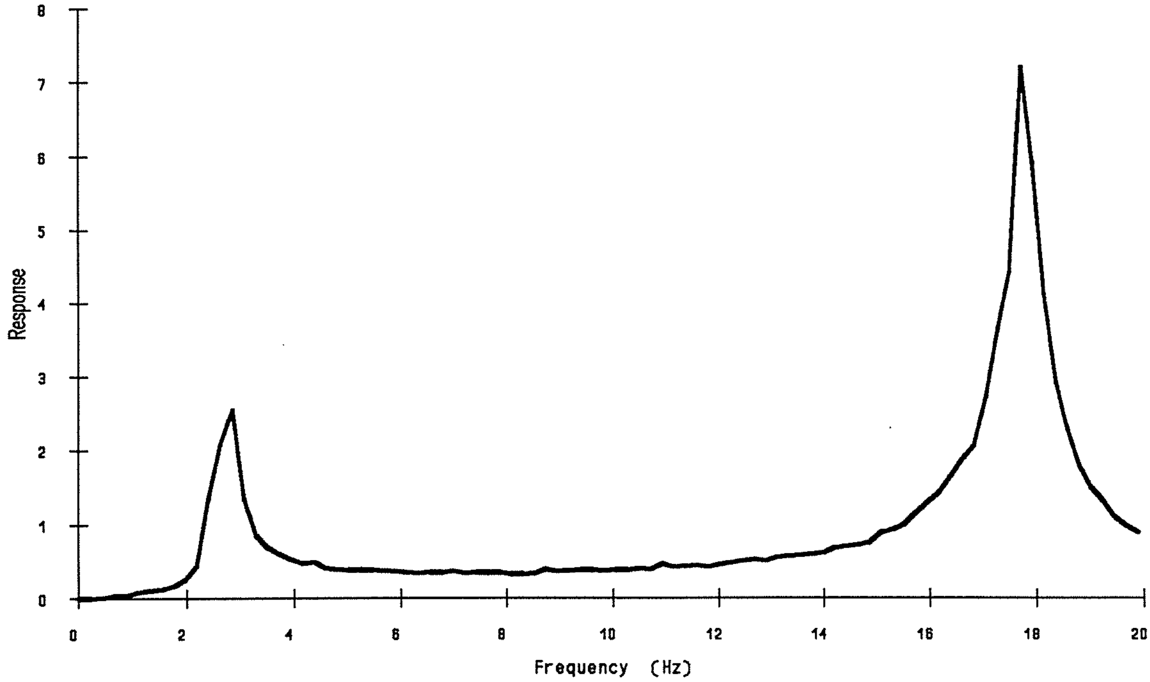


Figure 16. Frequency response function for non-linear spring IRI model.

Figure 17 shows the actual frequency response of this model to the particular road profile. As can be seen this is similar in form to the road response of the test vehicle.

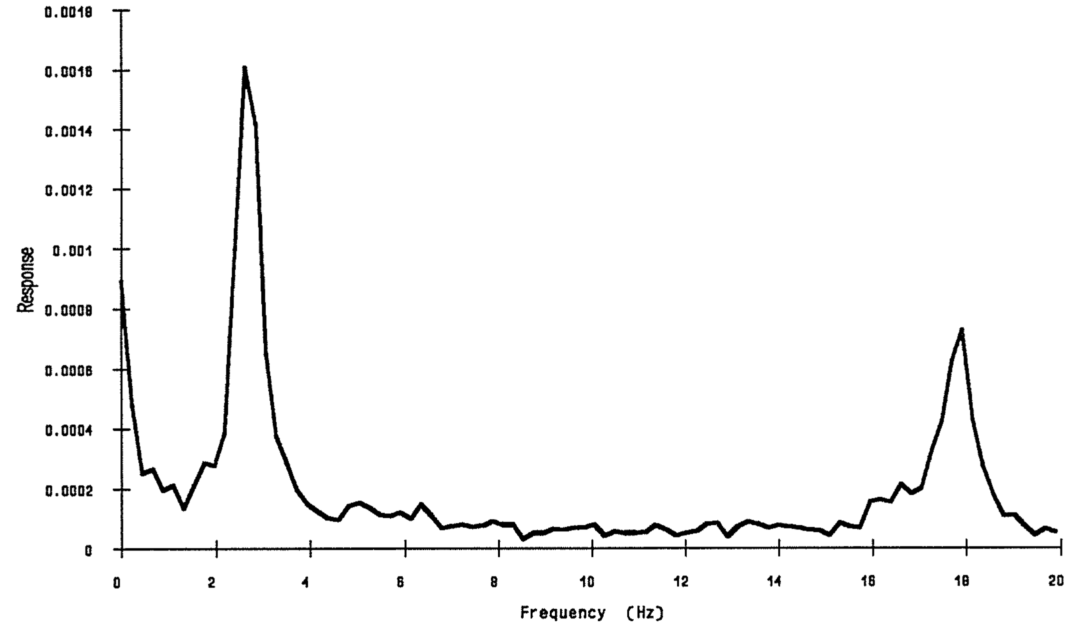


Figure 17. Actual frequency response of non-linear spring IRI model to a road profile.

In spite of this, the correlation of DLCs to the roughness indices generated by this model was only fair with $r^2 = 0.82$ for the whole dataset and $r^2 = 0.90$ when the urban site results are eliminated. With this model, applying a multilinear regression to the indices from the left and right profiles considered independently produced inferior correlation results.

5.8 The Band-pass Filter Index

All of the quarter vehicle models described previously are effectively a simulation of a mechanical filter of the road profile. The next logical step then is to eliminate the mechanical system in the filtering and apply digital filtering directly to the road profile data. This was done with four pole Butterworth band-pass filtering (Kuo 1966) over the wavelengths corresponding to 2 - 4 Hz at 80 km/h. The frequency response of this filter is shown in Figure 18.

The limits were chosen to reflect the sprung mass behaviour of the heavy vehicle fleet which are the primary contributors to dynamic wheel forces. Note that the frequency limits are not sharp cutoffs but that the frequencies outside the band also contribute but to a lesser extent. This will allow some speed variations and suspension stiffness variations to be accounted for. To convert this filtered profile to a single numeric, the amplitude of a 3 Hz sine wave which has the same energy as the filtered signal was calculated.

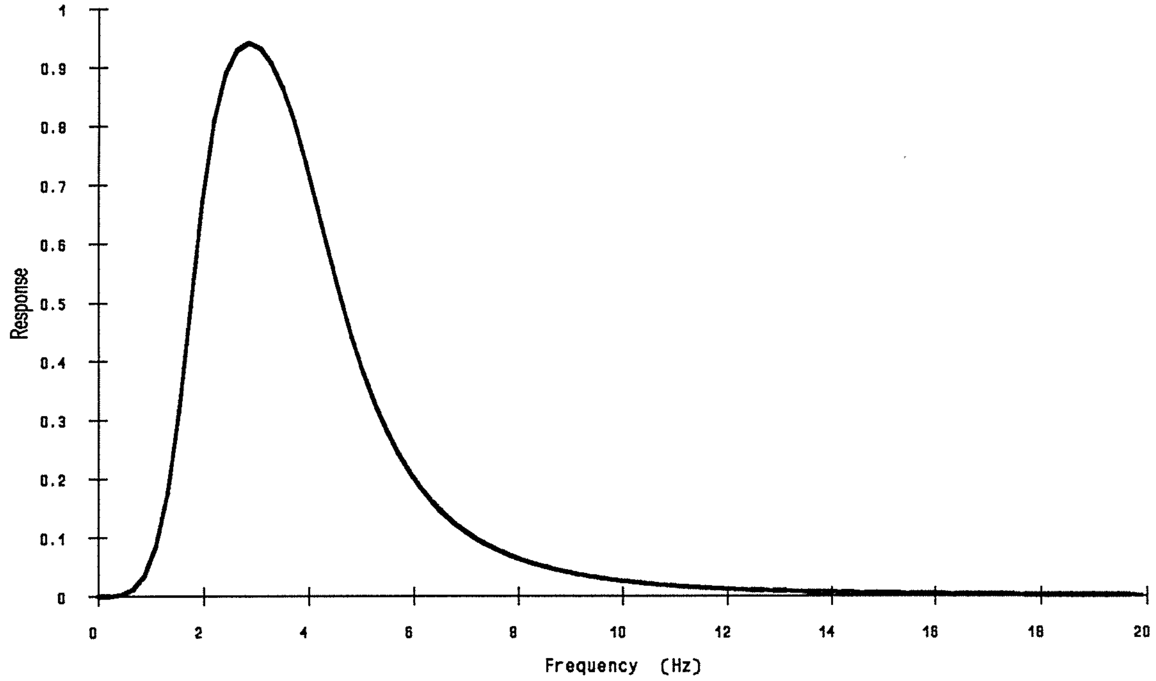


Figure 18. Frequency response for 2 - 4 Hz band-pass filter.

Using this as a roughness index the same fitting procedures as described previously were repeated. This index produced the best results of all the alternatives tried so far. For the fit of DLC to VR for all the test data, the r^2 value was 0.83 while removing the results from the urban site improved this to 0.86. As can be seen this is still not as good as the correlation with IRI.

Applying the multilinear regression to the left and right profile roughnesses as before, again improved the quality of fit giving an $r^2 = 0.89$ for the whole dataset and $r^2 = 0.91$ without the urban test site data. The band of wavelengths included in this band-pass filter are those corresponding to 2 - 4 Hz excitations at 80 km/h. In the analysis of heavy vehicle responses given in section 2, it was established that there were also unsprung mass vibrations in the 9 -18 Hz range which contribute to dynamic wheel loads. Thus a more complete band-pass filtering approach is to use a multilinear fit of DLC to both a 2 - 4 Hz band-pass filtered index and a 9 - 18 Hz band-pass filtered index with response characteristics as shown in Figure 19. This improved the correlation giving $r^2 = 0.89$ using all the data and $r^2 = 0.91$ when the urban site data are removed.

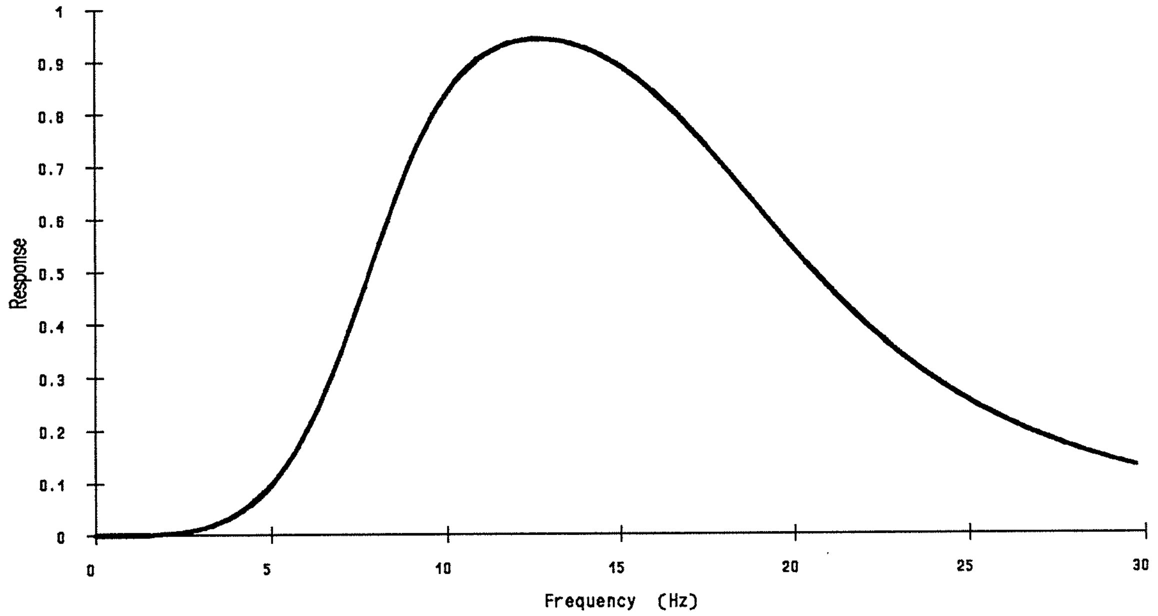


Figure 19. Frequency response for 9 - 18 Hz band-pass filter.

5.9 Summary to Section 5

A range of alternative roughness index measures have been postulated, calculated and then fitted to the test heavy vehicle response as measured by DLC. Although some of the indices provided a reasonable indicator of heavy vehicle response, none proved significantly better than IRI. It should be pointed out that, with an r^2 value of around 0.9, the IRI is quite a good indicator of heavy vehicle response. The IRI quarter car has resonances at 1.21 Hz and 11 Hz which are not typical of heavy vehicles but, because of its high damping, it has a relatively flat frequency response across the range of interest. Thus it also responds well to the frequencies which excite heavy vehicles.

6. MEASUREMENT OF ROUGHNESS INDICES

The most commonly used roughness index in New Zealand is "NAASRA roughness" which is measured with an instrument mounted in the rear of an ordinary passenger car with a live rear axle and accumulates the motion of that axle relative to the vehicle body. NAASRA roughness was originally defined in terms of the behaviour of a specific vehicle and the reference vehicle was maintained by ARRB. Because of the limited life of vehicles it was necessary to update the vehicle from time to time and at those times an extensive series of experiments was necessary to calibrate the new vehicle to the old one.

More recently, ARRB have developed a laser profilometer for measuring road profiles and established a relationship between the NAASRA roughness measure and the road profile by way of IRI. That is, the road profile is used to calculate IRI and a linear relationship has been established statistically between IRI and NAASRA roughness. The laser profilometer is now the primary means of calibrating NAASRA roughness meters and so the problems of maintaining a reference vehicle with unchanging dynamic characteristics have been eliminated.

From a New Zealand point of view there are logistic and financial difficulties in getting the laser profilometer to New Zealand and thus NAASRA roughness meters based in New Zealand are calibrated very infrequently. Thus, although the NAASRA meters are simple to use their calibration often cannot be guaranteed. They are unsuitable for use in extending this current study. Of the different indices discussed in this report only IRI can be obtained from a NAASRA meter.

Considering these issues, a number of recommendations can be made for an alternative roughness measuring device to the NAASRA meter. The first obvious one is to consider a device based on a single axle trailer. This would have a number of operational benefits:

- It would eliminate the need to have an expensive motor vehicle dedicated to roughness measurement.
- It would be a much simpler dynamic system and thus would be easier to maintain calibration on.
- It would not become obsolete and require replacement every five years or so in the way that a passenger car does.
- When replacement is necessary it will be much easier to ensure that the dynamic characteristics remain the same.
- Being so much simpler, laboratory-based calibration procedures would be possible. That is, it would not be necessary to use the laser profilometer as often. A number of technical improvements would also be possible.

If the roughness is still to be measured by accumulating suspension motion, it should be done with a displacement transducer (LVDT) and a personal computer-based data acquisition system. The current NAASRA meter's mechanical counter dates back to when computer-based data acquisition was not a practical proposition. Computer-based systems are now reliable and cheap and would ease the operator's task while increasing the accuracy.

However, if it was decided to develop a new device, it would probably be better to move away from measuring accumulated suspension motion and look instead at measuring road profile. This is done in a number of ways currently. ARRB use lasers mounted to a beam on the front of a passenger car. For the same reasons as given previously it would be better to mount the system on a trailer as is done at Transport Research Laboratory in the UK. This approach is technically quite sophisticated and expensive but produces good results.

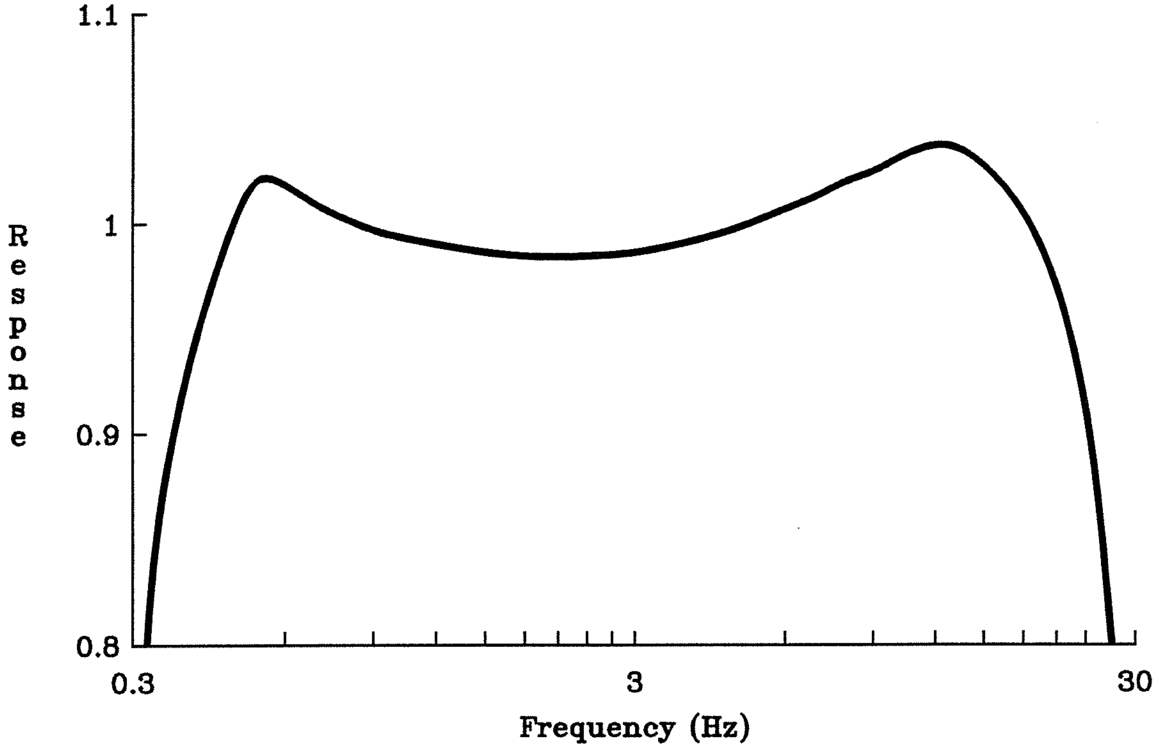


Figure 20. Frequency response of LCPC trailer to road profile.

An alternative approach is the trailer (Sayers et al. 1986a) developed by Laboratoire Central des Ponts et Chaussées (LCPC) in France which is primarily a mechanical system. This uses a horizontal pendulum to correct for the trailer's inertia and a softly sprung follower wheel which tracks the road profile. By using an LVDT to monitor the relative displacement of the follower wheel to the pendulum the actual road profile is measured. The pendulum's mass, stiffness and damping are tuned so that the system has a frequency response function very close to unity over the range 0.5 -20 Hz. This is shown in Figure 20.

This device was tested in the International Road Roughness Experiment (Sayers et al. 1986a) which developed the IRI and was found to be robust, reliable and accurate within its frequency limits. These limits include all the frequencies of interest when considering vehicle response. The wavelengths covered by this frequency range can be expanded by running the trailer at different speeds.
The two types of system just described are examples of ways in which profile measurement might be undertaken. Other techniques such as ultrasonics have also been used. The benefit of using a profile-based roughness measuring process is that it is then possible to calculate not only NAASRA roughness and IRI but also any of the indices postulated above. It would be possible to generate a whole range of possible indices from the same measurements merely by altering the analysis software.

7. CONCLUSIONS

IRI provides a reasonably good predictor of heavy vehicle response as given by DLC. Although a number of alternative roughness indices have been evaluated and tested, no other index provided a better predictor than IRI.

However, if a new system of roughness meter were to be considered, it may be better to be based on a trailer configuration as it would be simpler to calibrate, and would have a longer life, be easier to replicate and be cheaper. This however should be balanced against practical operational advantages afforded by a self-powered host vehicle.

The measurement system should be electronic and computer-based to improve reliability and storage.

Ideally the system should be based on measuring road profiles rather than vehicle response as it could then be used to extract any number of roughness indices merely by changing the analysis procedures.

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APPENDIX 1. FAST FOURIER TRANSFORMS ON PROFILE DATA FOR 4 SUBSECTIONS OF SITES 1 - 5

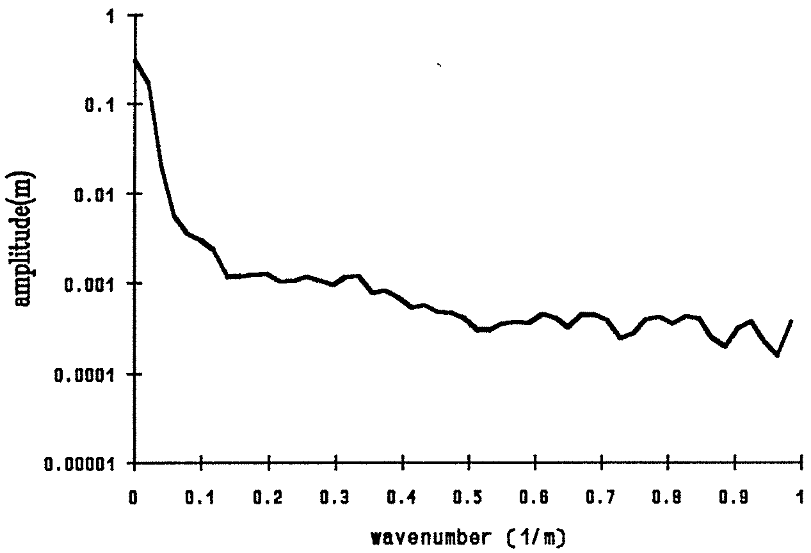


Figure 1. Profile spectrum site 1 - 1.

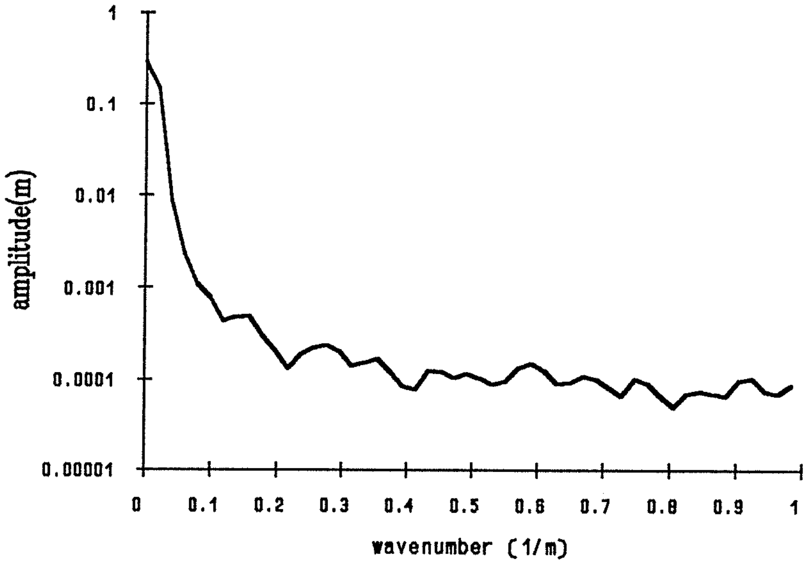
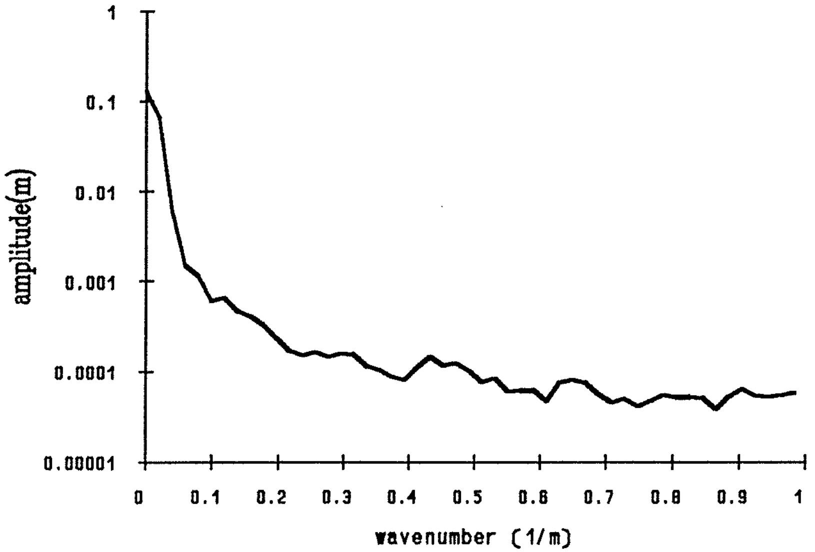


Figure 2. Profile spectrum site 1 - 2.



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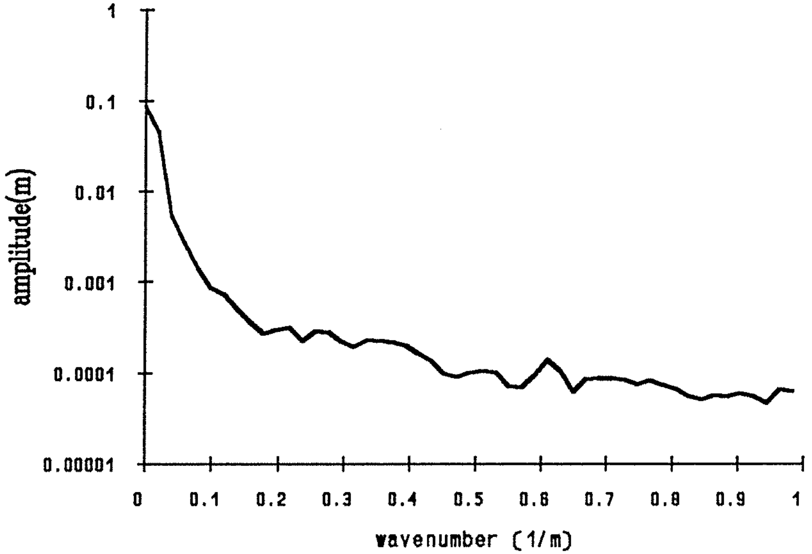

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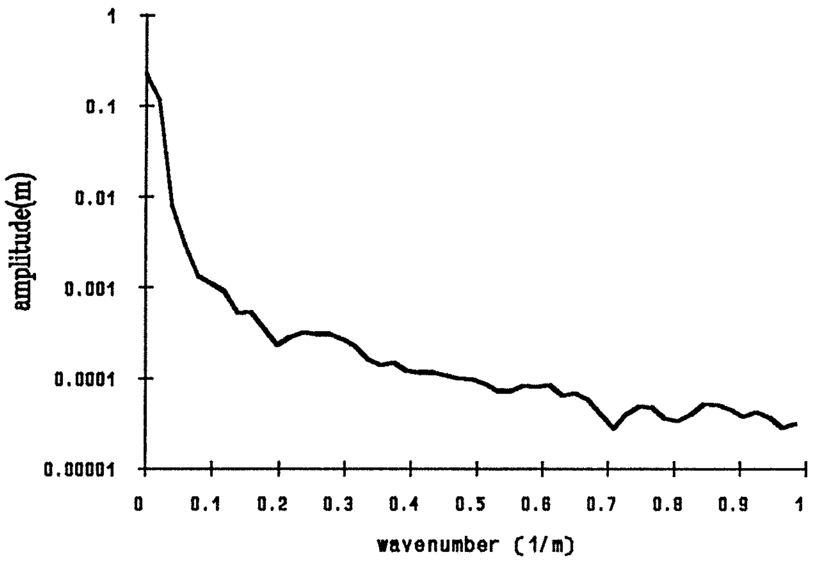

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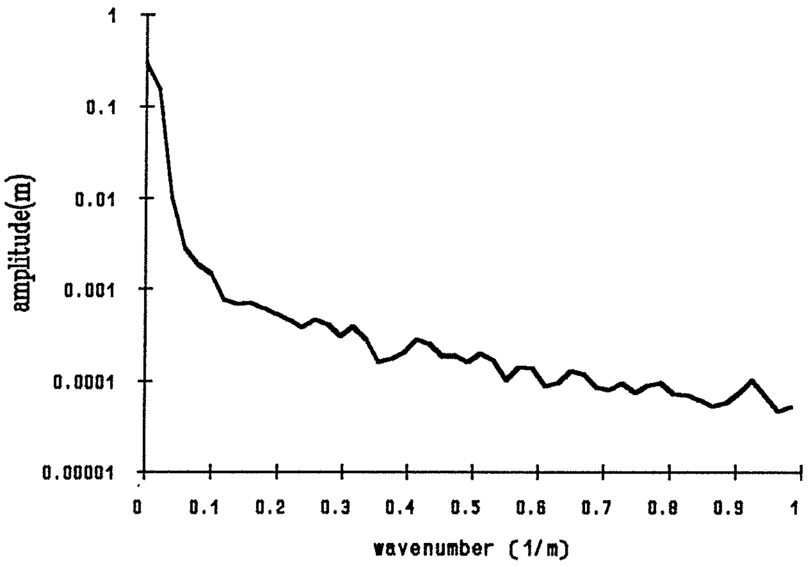

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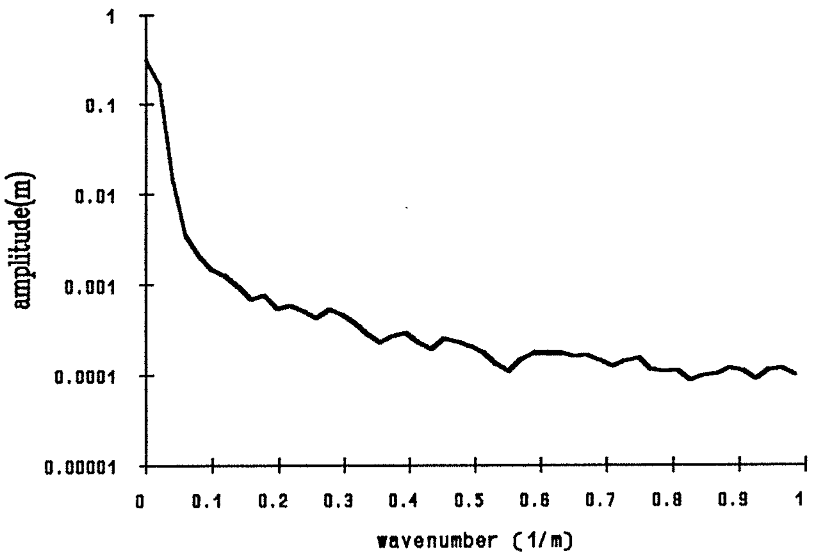

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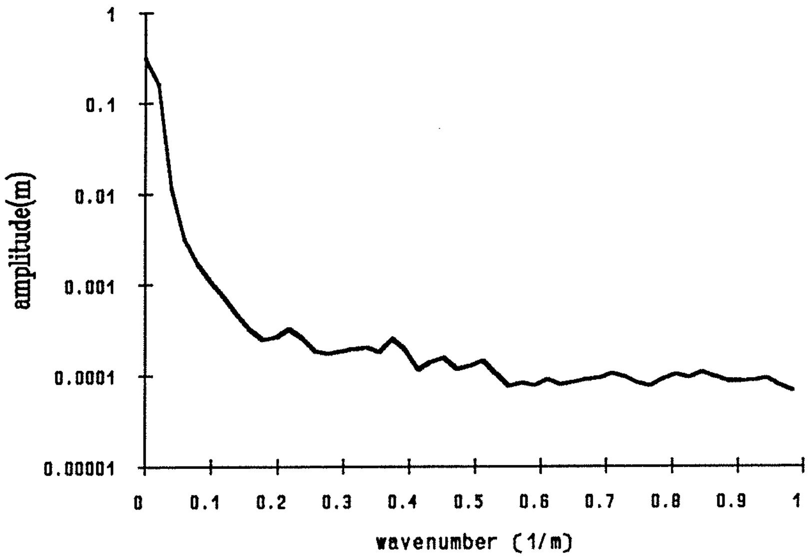

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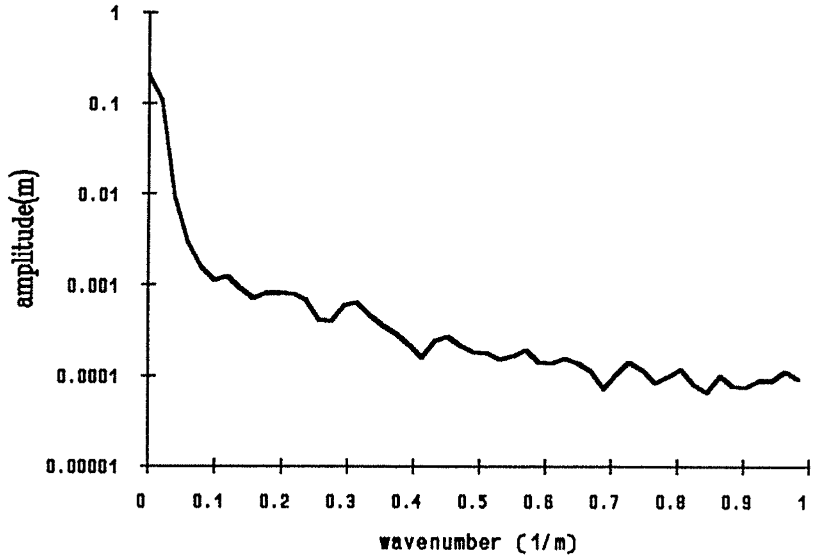

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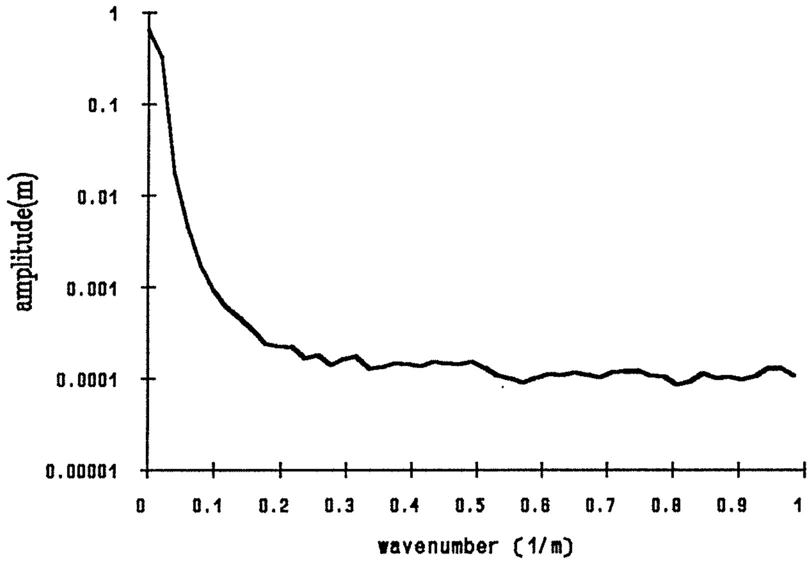

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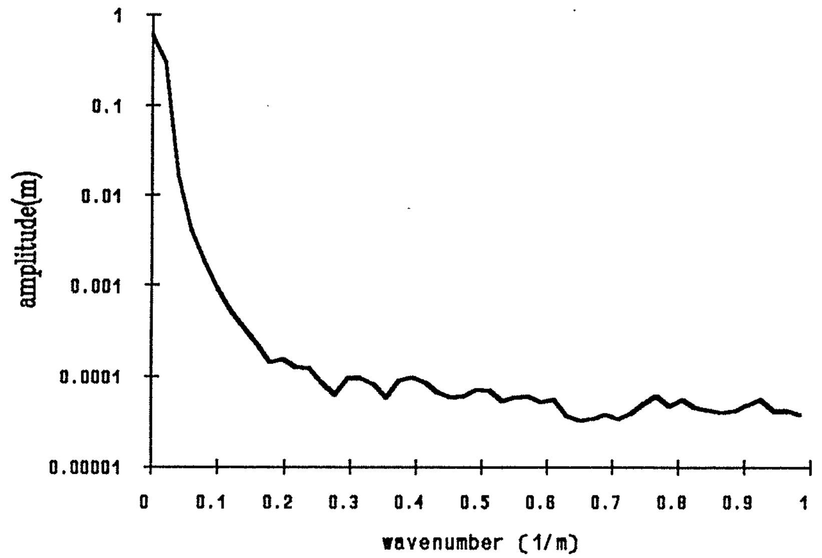

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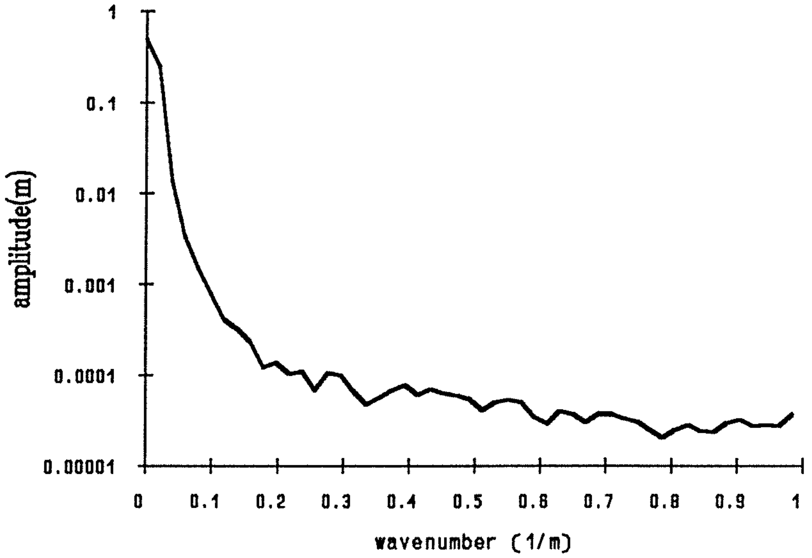

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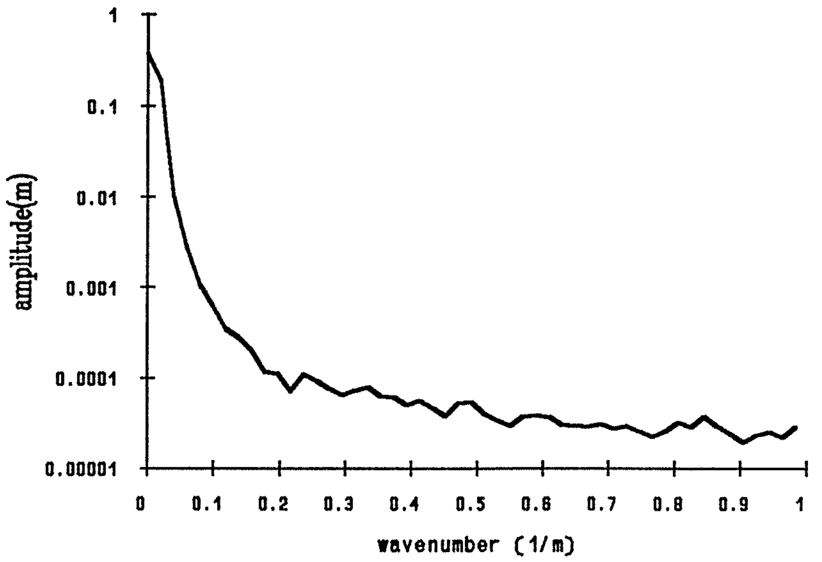

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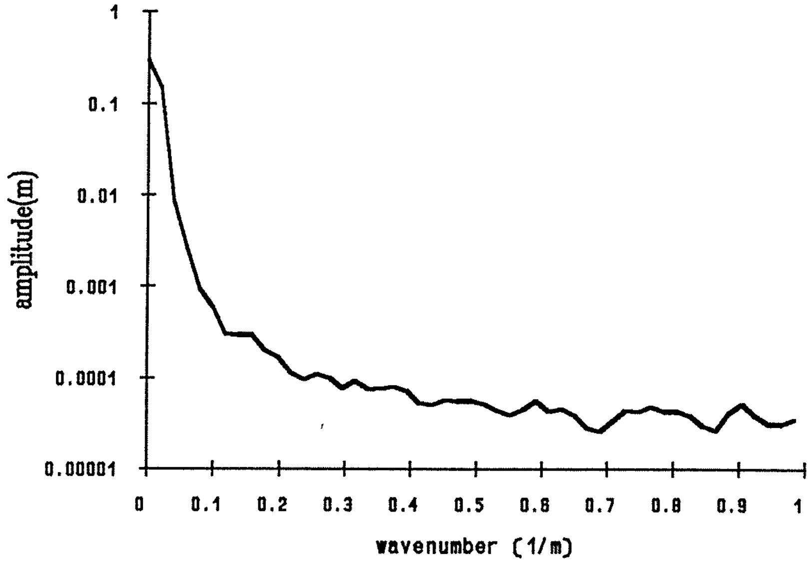

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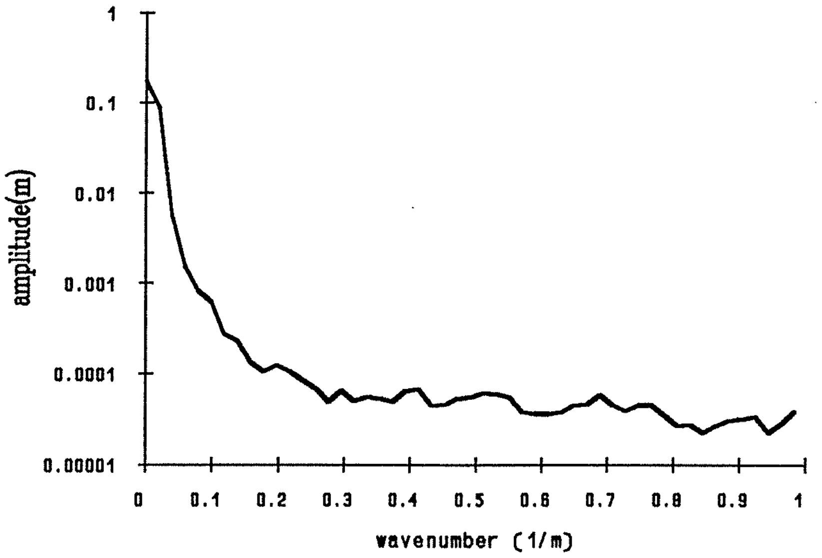

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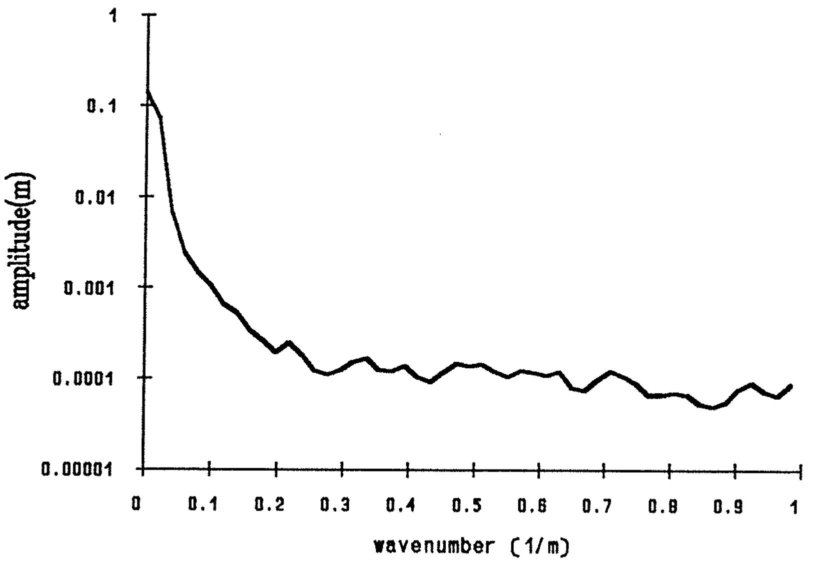

Figure 16. Profile spectrum site 4 - 1.

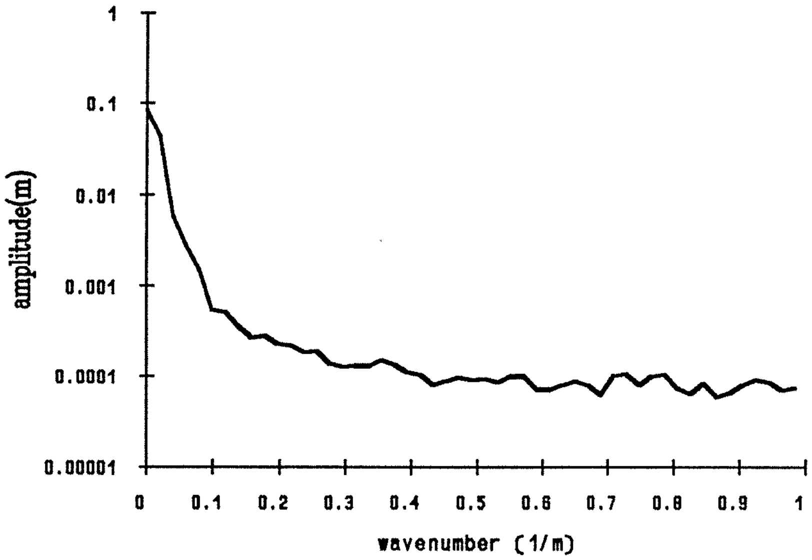

Figure 17.   Profile spectrum site 4 - 2.

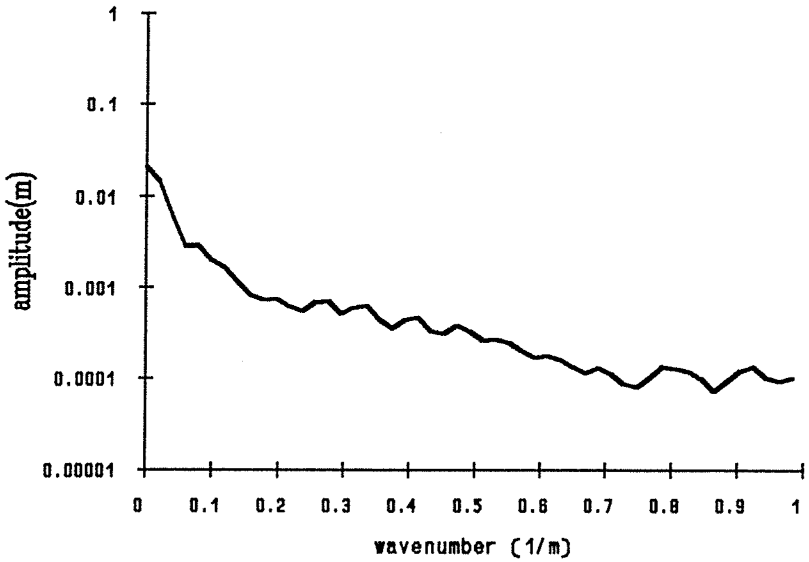

Figure 18.   Profile spectrum site 4 - 3.

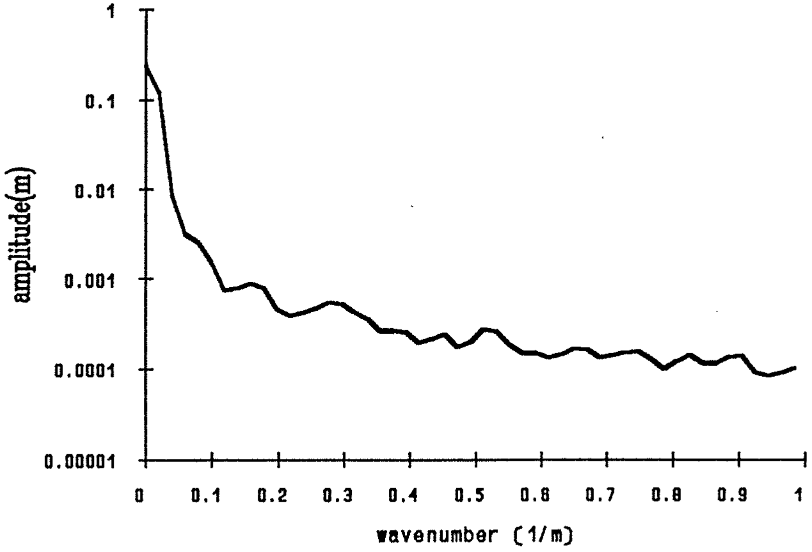

Figure 19.   Profile spectrum site 5 - 1.

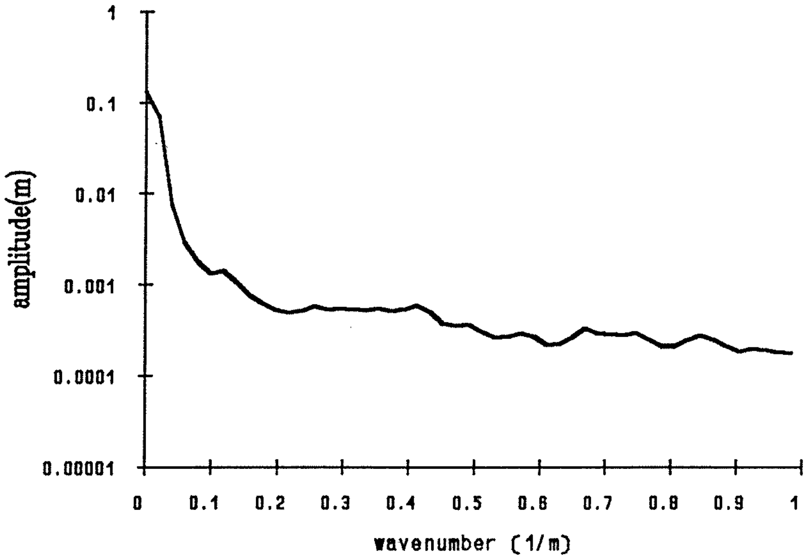

Figure 20.   Profile spectrum site 5 - 2.

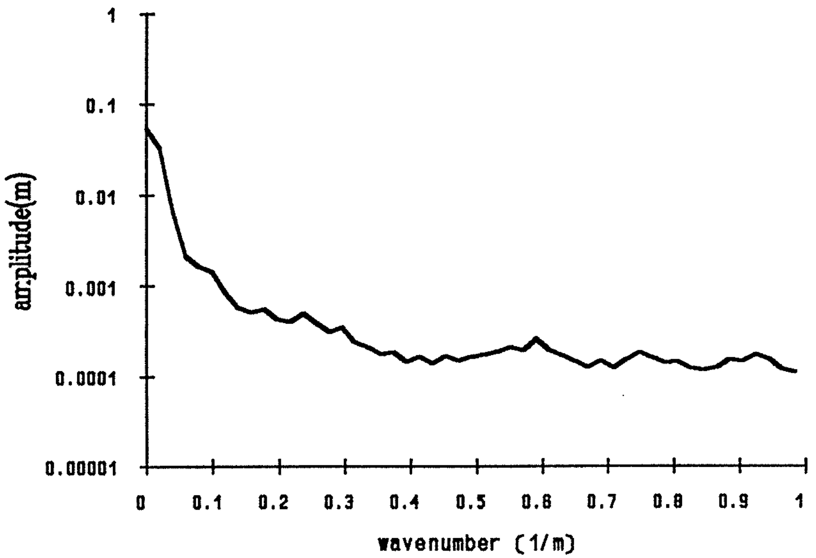

Figure 21. Profile spectrum site 5 - 3.

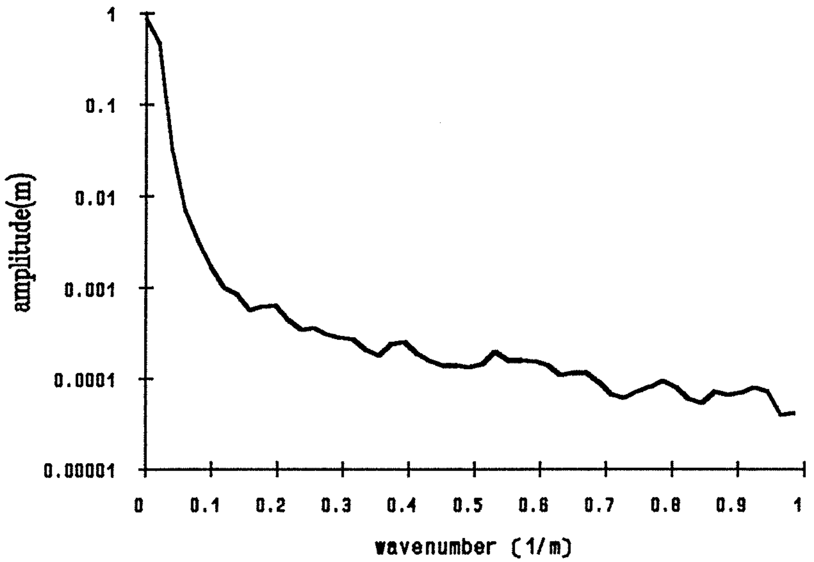

Figure 22. Profile spectrum site 5 - 4.