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Effects of Suspension Variations on the Dynamic Wheel Loads of a Heavy Articulated Highway Vehicle
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Volume II -- Effects of Suspension Variations on the Dynamic Wheel Loads of a Heavy Articulated Highway Vehicle

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Abstract
The work reported here was supported jointly by Canroad and NRC Division of Mechanical Engineering, Vehicle Dynamics Laboratory, as part of the RTAC/CCMTA Heavy Truck Weights and Dimensions Study.

This study has investigated the dynamic wheel load behaviour of various heavy highway vehicle suspension systems under a controlled experimental program. A 45 tonne tractor-trailer was modified to measure the dynamic axle loads of all major load-carrying axles simultaneously. Suspension parameters such as suspension type, axle spread and axle load were investigated as functions of road roughness and vehicle speed.

The effects of brake torque and suspension pitch attitude on the load equalization of the suspension were also investigated. Road tests were conducted at various speeds over a variety of road roughness conditions. Findings relevant to the subject of dynamic road loading are highlighted.

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Supplementary Information
This publication is produced under the auspices of the Technical Steering Committee of the Vehicle Weights and Dimensions Study. The points of view expressed herein are exclusively those of the authors and do not necessarily reflect the opinions of the Technical Steering Committee, Canroad Transportation Research Corporation or its supporting agencies.

The test program discussed in this report was carried out using suspensions in common usage in the Canadian truck fleet. The suspensions and components used for testing were provided by the respective manufacturers, and were in brand new condition. The test results observed reflect the conditions of the equipment and test procedures used, and may be expected to vary with equipment which has been used in service, or under different test conditions.

This report has been published for the convenience of individuals or agencies with interests in the subject area. Readers are cautioned that the use and interpretation of the data, material and findings contained herein is done at their own risk. Conclusions drawn from this research, particularly as applied to regulation, should include consideration of the broader context of Vehicle Weights and Dimension issues, some of which have been examined in other elements of the research program and are reported on in other volumes in this series.

The Technical Steering Committee will be considering the findings of these research investigations in preparing its "Final Technical Report" (Volumes 1 & 2), scheduled for completion in December 1986.
PREFACE

The report which follows constitutes one volume in a series of sixteen which have been produced by contract researchers involved in the Vehicle Weights and Dimensions Study. The research procedures and findings contained herein address one or more specific technical objectives in the context of the development of a consistent knowledge base necessary to achieve the overall goal of the study; improved uniformity in interprovincial weight and dimension regulations.

The National Research Council of Canada undertook a program of testing to examine the dynamic loading characteristics of different tractor and trailer suspensions. Canroad Transportation Research Corporation gratefully acknowledges the contributions of the following companies in supplying equipment and components for testing:

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EFFECTS OF SUSPENSION VARIATIONS ON THE DYNAMIC WHEEL LOADS OF A HEAVY ARTICULATED HIGHWAY VEHICLE

Rig Development, Experimental Program and Findings

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15 July 1986
DISCLAIMER NOTICE

The experimental results presented in this paper are pertinent to specific products which have been clearly identified by the manufacturer's product number. It should be emphasized that the experimental results should in no circumstances reflect the whole product line of an individual manufacturer, nor should the results be used to generalize on the performance of a generic suspension type. Subtle design variations in suspension components could, in some cases, lead to substantial changes in the performance of a suspension group. The objective of the study was not to endorse a product but rather to investigate the range of performance possible using these diverse systems.
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The authors wish to extend their appreciation for the efforts and contributions of the following individuals and companies to this study.

We gratefully acknowledge Dr. R.E. Cagné of the Systems Laboratory, National Research Council of Canada, for his efforts in providing us with a suitable data acquisition system and related graphics software package.

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The pavement response field experiments were conducted in cooperation with the Civil Engineering Department, National Resources Division, Alberta Research Council. We wish to acknowledge in particular the efforts of Dr. T. Christison and Mr. L.M. Chase. The assistance provided by the Gouvernement du Québec, Ministère des Transports, in providing services essential to the success of the field experiments was greatly appreciated.

The road roughness index was measured using a Mays-meter device from the Ministry of Transportation and Communications of Ontario (MTC). We extend our gratitude to Mr. T. Khan, Mr. F. Marciello and Mr. W.A. Phang of MTC and to Mr. A.T. Papagianakis and Professor R.C.G. Haas of the Department of Civil Engineering, University of Waterloo, for participating in the road roughness tests and for performing the data reduction and subsequent analysis of the results.

We are indebted to Chalmers Suspension International Ltd., Hayes-Dana Inc., Hendrickson Mfg. (Canada) Ltd., Neway (A Division of Lear Siegler Ltd.) and Keyco Canada Ltd. for supplying us with heavy duty truck suspensions and related hardware. Their cooperation and interest in the study are gratefully acknowledged.
ABSTRACT

The work reported here was supported jointly by Canroad and NRC Division of Mechanical Engineering, Vehicle Dynamics Laboratory, as part of the RTAC/CCMTA Heavy Truck Weights and Dimensions Study.

This study has investigated the dynamic wheel load behaviour of various heavy highway vehicle suspension systems under a controlled experimental program. A 45 tonne tractor-trailer was modified to measure the dynamic axle loads of all major load carrying axles simultaneously. Suspension parameters such as suspension type, axle spread and axle load were investigated as functions of road roughness and vehicle speed.

The effects of brake torque and suspension pitch attitude on the load equalization of the suspension were also investigated. Road tests were conducted at various speeds over a variety of road roughness conditions. Findings relevant to the subject of dynamic road loading are highlighted.
# TABLE OF CONTENTS

ACKNOWLEDGEMENTS (ii)

ABSTRACT (iii)

1.0 INTRODUCTION 1

1.1 Report Structure 2
1.2 Principles and Assumptions Governing the Choice of Hardware 2

2.0 THE TEST VEHICLE 6

2.1 Tractor 6
2.2 Trailer 7

3.0 INSTRUMENTATION AND CALIBRATION 8

4.0 TEST VARIABLES 10

4.1 Hardware Variables 10
4.2 Axle Load Variations 16
4.3 Road Roughness and Speed Variations 18
4.4 Single Bump Tests 20
4.5 Grade Level Railway Crossing Bump 21
4.6 Dynamic Bridge Loadings 22
4.7 Static Wheel Load Measurements 22
4.8 Static Pitch Test 23
4.9 Shake Test 23

5.0 ANALYSIS 24

5.1 Resolution of Dynamic Wheel Load 24
5.2 Shake Test 28

6.0 RESULTS 32

6.1 Special Cases 32

6.1.1 Impact Wheel Loads Associated with a Grade Level Railway Crossing 32
6.1.2 Dynamic Bridge Loading 33
6.1.3 Tire Scuffing in Turns as a Function of Axle Spread 33
6.1.4 Suspension Load Equalization Due to Variations in Trailer Pitch Angle 34
6.2 Shake Tests

6.3 Road Tests

6.3.1 Dynamic Wheel Loads as a Function of Suspension Type

6.3.2 Dynamic Wheel Load as a Function of Trailer Suspension Spread

6.3.3 Dynamic Wheel Load as a Function of the Number of Axles in a Suspension Group

6.3.4 Axle to Axle Dynamics for Load Sharing Suspensions

6.3.5 Load Transfer Due to Braking

6.3.6 The Air Suspended Lift Axle

6.4 Pavement Deflection Due to Dynamic Wheel Loads

7.0 Concluding Remarks

7.1 Interpretation of Experimental Results

7.2 Relating Study Findings to Vehicle Weights and Dimensions Regulations

8.0 REFERENCES

APPENDICES

A. Mechanical Design Drawings of Hardware Fabricated for this Study

B. Vehicle Calibration and Instrumentation

C. Road Roughness Report

D. A Listing of Software Developed for the Analysis of Dynamic Data

E. Plots of Wheel Load vs Ride Comfort Rating for Various Suspension Combinations

F. Histograms of Digitized Wheel Load Data

DOCUMENTATION PAGE
EFFECTS OF SUSPENSION VARIATIONS ON THE DYNAMIC WHEEL LOADS OF A HEAVY ARTICULATED HIGHWAY VEHICLE

1.0 INTRODUCTION

The purpose of this study is to provide the road regulatory authorities with factual data on the first order effects of suspension variations in terms of dynamic wheel loading as seen by the pavement. Simply put, the objective of this suspension study is to answer the following questions.

1. How well do multi-axle truck suspensions equalize load?
2. What are the dynamic wheel forces associated with typical suspension types?
3. How do variations in suspension axle spacing affect the dynamic wheel loads and the load equalization capabilities of a given suspension?
4. What is the effect of variations in vehicle speed and road roughness on dynamic wheel loads?

In addition to these four basic questions, typical examples of dynamic axle loads associated with discontinuities in the road structure will be provided. Included in this category are the following:

1. dynamic bridge loading associated with smooth and rough approaches.
2. dynamic road loading associated with a grade level railway crossing.
3. dynamic road loading associated with various pavement conditions such as:
   - rigid pavement nearing the end of its acceptable life.
   - old flexible overlay on rigid pavement base with reflective cracking.
   - new smooth overlay on rigid pavement.
   - end of overlay transition bump.
   - rough and smooth flexible pavement.

Finally, the effect of vehicle braking and suspension equalization will be examined.

1.1 REPORT STRUCTURE

The main body of the report has been structured to be as concise as possible dealing with the first order results which are of primary interest to the overall study. Topics which serve to support the findings such as hardware development, instrumentation, theoretical analysis and calibration procedure are contained in the appendices.

1.2 PRINCIPLES AND ASSUMPTIONS GOVERNING THE CHOICE OF HARDWARE

In order to accurately measure the performance characteristics of heavy vehicle suspensions, as a function of road roughness variations and suspension parameter changes, considerable thought was required for the choice of vehicle to be used during the test program. It is well known that general vehicle characteristics such as vehicle mass, chassis compliance (both bending and torsion)
will effect the vertical dynamics and hence the dynamic axle loads of a vehicle. To accurately study the effects of suspension variations, these external influences must be held constant so that their contribution to vehicle response is not confused with those associated with a suspension parameter change.

Bearing in mind these concerns, the following points were used as guidelines in developing a vehicle suitable for these experiments.

(1) The vehicle must be stiff in bending (beamng) and torsion so that structural compliance of the vehicle does not interfere with the response of the vehicle when a suspension parameter change is made.

(2) The size and weight of the vehicle should be representative of large vehicles used in Canadian Interprovincial Trucking.

(3) The weight of the vehicle must be controlled and must remain constant over time.

(4) The modified chassis of the vehicle must permit rapid change out of suspensions and suspension components even when the vehicle is fully loaded.

(5) Suspension components to be tested must cover the most common suspension types found on Canadian roads.

(6) The suspensions must be fabricated in accordance with the manufacturers' instructions.

(7) All suspensions must use the same make and model of axle, brake components and the same tires and rims.
(8) The sensors used to measure force and torque cannot in any way effect the mechanical response of the suspension.

(9) The vehicle's instrumentation system must continuously record all axle loads, brake torques and vertical accelerations simultaneously in analog form.

(10) The sensors used must have minimum cross axis sensitivity and must be linear, with minimum hysteresis.
2.0 **THE TEST VEHICLE**

2.1 **TRACTOR**

A 1979 White Freighliner cab over tractor was refurbished to serve as the power unit for the study. Instrumentation recording systems were housed in the existing sleeper compartment which was fitted with a shock attenuating floating floor. All electronic data channels were routed through a connector junction box, a wiring harness and patch board were permanently fixed to the tractor. Electrical power was provided by an auxiliary power unit fixed to the tractor chassis.

The tractor was fitted with a new drive axle suspension. The suspension beams and drive axles were instrumented to yield vertical axle load, brake torque and vertical axle acceleration. A vertical accelerometer was also fitted to the steering axle of the tractor as a check on the relative road roughness between runs. Since the vertical response of the front axle is somewhat independent of the drive suspension, and since the static weight on the front axle is constant, the response of the front axle formed a reference from which runs of the vehicle over the same stretch of road could be compared with confidence. In short, if the front axle response characteristics were similar in energy content for two separate runs at the same speed over the same road but at different times of the year then one could be reasonably confident that the road roughness did not change significantly since the last time the test was run.

Finally, speed and distance were monitored by use of a trailing wheel.
2.2 TRAILER

A 1974 Fruehauf compartmentalized baffled liquid tanker was refurbished for the study. The frame structure and original suspension were removed, scrapped and replaced by a new frame specifically designed for the purpose of this study.

The replacement frame was designed to accept different suspensions each mounted on its own sub frame. The sub frames could be moved to various positions on the main frame thereby permitting changes in axle position and spacing. Design drawings of the trailer frame, suspension sub frames and other mechanical components required for the study are found in Appendix A.
3.0 INSTRUMENTATION AND CALIBRATION

All dual-tire axles of the tractor and trailer were instrumented to measure vertical axle load, vertical acceleration and brake torque. Axle load measurement was achieved by the use of strain gauges on the axles which were sensitive to vertical bending of the axle. Brake torque was measured with strain gauges measuring strain in the axle along the torsional shear axis of the tube (i.e., 45° to the axle's axis.) Both of these measurement techniques provided linear results with no significant hysteresis. This was due in part to the choice of axle design used in these experiments. The axle was fabricated from steel tubing, and the axle spindles were friction welded to the tube without the use of a pilot shaft. This manufacturing technique eliminates the need for a pilot shaft on the end of the spindle, which is commonly pressed into the tube before welding. It was felt that the presence of a pilot shaft in the vicinity of the strain gauge section of the tube would detract from the linearity of the calibration curve.

The vertical acceleration of the axles was measured by strain gauge type accelerometers mounted on the same vertical axis as the load sensing strain gauges. The acceleration component is necessary to account for the vertical inertial effects of the tires, wheels and brake components outboard of the load sensing strain gauges. This inertial component is added to the vertical axle load to determine the impact load at the pavement.

Accelerometers were also fixed to both ends of the tank. By combining the outputs of these accelerometers, both pitch and bounce of the trailer were resolved.
Vehicle velocity and distance travelled were measured by an optically instrumented idler wheel mounted on the side of the trailer.

Quite by accident, it was observed that tire side forces in low speed turns could be resolved using the load sensing strain gauges on the axle. The side force induces a moment on the axle which, when calibrated, can be resolved into the magnitude of the side force. The prime limitation is that the cornering must be done at quasi-static speeds so that there is no roll induced load transfer to the axles.

Further details on the instrumentation and calibration procedures are found in Appendix B.
4.0 TEST VARIABLES

4.1 HARDWARE VARIABLES

The study examined three generic types of heavy vehicle suspension, the walking beam, the air suspension and the spring suspension. These represent the majority of suspension found on Canadian trucks. An explanation of each of these suspensions and their variations follows.

A. The Walking Beam Suspension

The two walking beam suspensions tested are illustrated in Figures 2 and 3. The axles are fixed to a rigid beam pivoted at its center or balance point thus facilitating ideal static load sharing. Two spring elements are used, one for each side of the chassis located between the frame rail and the walking beam. The specifications of the two walking beam suspensions are as follows:

Tractor Drive Suspension

Manufacturer - Hendrickson Mfg. (Canada) Ltd.

Model - ATE 440 (extended leaf tandem)

Combined Axle Rated Capacity - 20 tonnes

Axle spacing - 1.52 meters

Outer tire track width - 2.44 meters

Spring elements - steel leaf spring, 2 stage, Part Number 45322

Trailer Suspension

Manufacturer - Chalmers Suspensions International Ltd.

Model - 754-44-LW
Combined Axle Rated Capacity - 20 tonnes

Axle spacing - 1.37 meters

Outer tire track width - 2.59 meters

Spring element - rubber with restrictor can

Figure 2  Tractor Drive Walking Beam Suspension

Hendrickson RTE 440

Figure 3  Trailer Walking Beam Suspension -

Chalmers 754-44-LW
E. The Air Suspension

Two tandem axle air suspensions were tested. The axles of both suspensions are mechanically independent of each other. The air supply to the air bags is regulated by two time delayed height sensing valves, one for each side of the vehicle. The air bags on each side of the vehicle are plumbed in parallel thus achieving load equalization.

Figure 4a. Tractor Drive Air Suspension Neway ARD 244-6

Figure 4b. Trailer Air Suspension Neway AR95-14
yet maintaining quasi-static vehicle roll resistance. Mechanical roll stiffness of the suspension is achieved by the use of trailing members semi rigidly fastened to the axle (trailer suspension) or a torsion tube (tractor suspension) to form an anti roll bar. This transfers roll moments of the vehicle to vertical forces at the wheels.

The lift axle was identical to the trailer tandem suspension unit except that the air pressure in the air bags was governed by a constant pressure regulator rather than a height sensing valve. It also had additional mechanical lifting members and air bags for the purpose of lifting the axle off the road surface. The lift axle therefore was fully independent of the tandem axle air suspension. The specifications of the air suspension axles tested are as follows:

Manufacturer - Neway (A Division of Lear Siegler Inc.)

Model - Tandem axle drive suspension ARD 244-6

Serial # C904157 EM

Combined Axle Rated Capacity - 20 tonnes

Spring Element - Air bag part number 905-57-031

Tandem Axle Spacing - 1.37 meters

Outer tire track width - 2.59 meters

Model - Tandem axle trailer suspension AR95-14

Serial # Lead - C9926210LK

Trailing - C9926211LK

Combined Axle Rated Capacity - 22.73 tonnes

Lift axle - AR95-14 (Lift)

Serial #C9926209LK
Rated Capacity per Axle - 11.36 tonnes
Spring Element - air bag Part Number 905-57-020
Tandem Axle Spacing Tested - 1.27 meters
- 1.83 meters
- 2.44 meters

Outer tire track width - 2.59 meters.

The suspensions are illustrated in Figures 4a, 4b and 5.

Figure 5 Trailer Lift Axle Neway AR95-14 (Lift)

C. Four Spring Suspension

A single tandem axle four spring trailer suspension was tested (see illustration Figure 6). As with the air suspension the four spring suspension was tested at three different axle spacings. The same springs and axles were used in all cases. The axle spacing variations were possible because of the design of the sub frame structure which accommodated various standard equalizing beams for this particular suspension. The specifications of the four spring suspensions tested are as follows:
Manufacturer - Reyco Canada Inc. (a subsidiary of Reyco Industries Inc.)

Model - 2113-FAB-222-WB-14-C-50-3564

Combined Axle Rated Capacity - 20 tonnes

Tandem axle spacing tested - 1.27 meters

- 1.83 meters

- 2.44 meters

Outer tire track width - 2.59 meters

Spring elements - multi leaf spring - T-3564

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Figure 6a

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Figure 6b

Figure 6a, 6b The Trailer Four Spring Suspension Showing Support Hardware Variations for 1.27, 1.83, 2.44 Meter Spreads - Reyco 2113-FAB-222-WB-14-C-50-3564.

TIRES - When suspension changes were made the same tires were used. Both the tractor and the trailer were fitted with dual tires. All
tires were inflated to 100 psi and checked before each run. The tires used are listed below.

Tractor - Steering axle - Uniroyal 14/80 R20
- Drive axles - Michelin 12R 22.5
Trailer - All axles - Michelin 11R 22.5

(The tires mentioned above are not tires which the NRC necessarily endorses.)

4.2 AXLE LOAD VARIATIONS

Vehicle mass and spring constants are the primary variables which have first order effects on vehicle vertical response. It was clear from the start of this study that control over the mass variable was considered to be of prime importance. To achieve constant mass, the fore and aft compartments of the trailer were completely filled with water and sealed for the duration of the test program. Changes in static axle load independent of vehicle mass variations were achieved through the use of an air suspended lift axle. The lift axle was located toward the longitudinal center of the trailer. The axle was controlled from the cab and could be raised clear of the road to increase the axle loads of the suspension being studied or lowered to decrease the axle loads.

This procedure allowed for constant control over the magnitude and the location of the suspended mass and its related properties such as pitch moment of inertia. Admittedly, the lift axle will have some influence on the vehicle response so care must be taken
in interpreting the data generated when the lift axle is down. (This task is helped by the fact that the air suspended lift axle has a linear response and a well defined spring stiffness and viscous damping characteristics.)

For example, when the lift axle is used in conjunction with the four spring trailer suspension, the spring constant and damping coefficient of the air suspension are much less than those of the four spring trailer suspension. Changes in the vehicle responses therefore may be attributed more to a reduction in static axle load of the four spring as opposed to the suspension effects of the air axle. This would be particularly true when considering the pitch dynamics of the trailer. By way of contrast, when the lift axle is used with the trailer air suspension, the spring constants and damping coefficients are nearly identical thereby playing a more dominant role in vehicle response variations.

The approximate static axle loads used during the test program are as follows:

<table>
<thead>
<tr>
<th>Condition</th>
<th>Tractor Drive Suspension Load Metric Tonnes Per Axle</th>
<th>Lift Axle Metric Tonnes</th>
<th>Trailer Suspension Load Metric Tonnes Per Axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td></td>
<td>10</td>
</tr>
<tr>
<td>2</td>
<td>8.4</td>
<td>7.6</td>
<td>7.8</td>
</tr>
</tbody>
</table>
4.3 **ROAD ROUGHNESS AND SPEED VARIATIONS**

A range of road roughness conditions were selected to cover simple smooth, medium and rough categories. The test roads chosen were uniquely different from one another thus serving different purposes during the test program. All suspensions were tested over the same test sections at the same speeds. The road roughness was determined by the use of a Mays Meter. The Mays Meter measurements were then correlated with Ride Comfort Rating (RCR) by the following equations:

\[
\text{RCR} = 9.63 - 0.02 \times \text{Mays Meter measurement}
\]

The RCR scale defines as Excellent, RCR Values 10-8

- as Good, " " 8-6
- as Fair, " " 6-4
- as Poor, " " 4-2
- as Very Poor, " " less than 2

The vertical profile of two of the three roads was measured with a rod and chain. Full particulars pertaining to road roughness can be found in Appendix C.

What follows is a brief description of the test sites and the vehicle speeds used for each site.

A. **Uplands Road North Bound Lane**

- High Roughness Section 1 - Mays roughness 254 IPM (RCR 4.6)
- Section 2 - Mays roughness 424 IPM (RCR 1.2)
- Test Speeds both sections 40, 60 km/hr
General Description - A two lane undivided road with badly deteriorating flexible pavement. There was excessive pavement cracking in a random pattern. Although the posted speed limit is 80 km/hr, the ride in the truck became unacceptable beyond 60 km/hr. For this particular road 60 km/hr is about the limit that most drivers would be prepared to push their equipment.

B. Woodroffe Ave. (Between CNR Tracks and Slack Rd,) North Bound
   - Rough level railway crossing
   - Smooth to medium rough roadway - Mays roughness 73 IPM
     (RCR 8.2)
   - Test speeds - 40, 60, 80 km/hr

General Description - A two lane undivided road with flexible pavement in good condition. Three speeds were chosen for this roadway 40, 60, and 80 km/hr. The test section commenced with a grade level railway crossing which was impacted at full running speed. The analysis of the smooth road section commenced once the reaction of the vehicle to the railway crossing had dampened out. The dynamic wheel loads resulting from the railway crossing were analyzed separately.

C. Highway 417 (Between Maitland Ave. Overpass and Rochester Street Exit) East Bound

Three Sections
   - smooth Mays roughness 59 IPM (RCR 8.5)
   - medium Mays roughness 165 IPM (RCR 6.3)
   - rough Mays roughness 217 IPM (RCR 5.3)
   - several bridge structures
   - test speed - 80 km/hr
General Description - A multi-lane divided highway through an urban area. The roadway is in the process of being reconstructed therefore there are three distinct surfaces present within the single test section. The smooth section (RCR 8.5) is new flexible overlay on a rigid pavement base. The medium surface (RCR 6.3) is an older overlay on the same rigid pavement. Reflective transverse cracking is evident. The rough surface (RCR 5.3) is the original rigid pavement in dire need of repair. These three surfaces were in close proximity to each other which allowed for continuous recording of all three surfaces during the same pass. The speed was held constant at 80 km/hr. This test section also contained several bridge structures with different approach roughnesses. Some approaches were undetectable by our instruments while some others gave very high reactions. These as well as the railway crossing data mentioned in the previous road description are presented under the section of special cases.

4.4 SINGLE BUMP TESTS

In addition to conducting tests on various roadways, there were a series of tests conducted with discrete bumps. These tests included both quasi-static or creeping over the bumps as well as dynamic impacts at various speeds.

The bumps were created by placing standard dimensional lumber across the road parallel to the axle axis of the vehicle.

The quasi-static or creep tests were used to measure quasi-static load equalization while the high speed runs were used to "pluck"
the suspension system so that the natural frequencies and apparent
damping coefficient could be resolved. The ability of the suspensions
to mitigate dynamic impact axle loads was also determined from the high
speed runs. A listing of the bump arrangements and test speeds
follows.

(a) Quasi-static creep tests (first gear deep reduction with
engine idling).
- Two planks side by side 4 cm × 48 cm
- Three planks side by side 4 cm × 72 cm
- Two planks side by side with a third plank centered on top
  of the bottom two 8 cm × 48 cm.

(b) Dynamic Impacts
Speeds - Top end of first gear
- 18 km/hr
- 40 km/hr

All dynamic impacts were done at the above speeds over a
single wooden plank fixed to the road surface having cross sectional
dimensions of 4 cm × 24 cm.

Speed control during all tests was achieved by selecting the
appropriate gear with the engine set against the maximum RPM governor.

4.5 GRADE LEVEL RAILWAY CROSSING BUMP

A single, grade level railway crossing was used in order to
get a 'feel' for the dynamic axle loads that can be expected from such
an input. The Mays meter roughness output for the 80 meter increment
of road containing the approaches and the crossing, was 252 IPM (RCR 4.6). Recognizing that this roughness figure is somewhat ambiguous, the general consensus was that in terms of roughness, the railway crossing could be considered to be typical.

The vehicle speeds used during the crossing were 40, 60 and 80 km/hr. The road roughness in the vicinity of the crossing was approximately 60 IPM (RCR 8.4).

4.6 **DYNAMIC BRIDGE LOADINGS**

A number of bridges were crossed during each of the tests conducted on Highway 417. The bridge approaches varied from smooth (undetectable) to very rough. One particular bridge on a recently repaved stretch of road produced quite a large vehicle response. The dynamic wheel loads associated with this bridge were included in this report, as a matter of interest.

4.7 **STATIC WHEEL LOAD MEASUREMENTS**

When a new suspension was installed on the vehicle or when the axle spacing was changed, the vehicle's static wheel load was measured on a flat level concrete floor.

The procedure used was to place jacks under the chassis of the fully loaded vehicle and then raise the vehicle until the wheels were off the ground. All load sensing strain gauge bridge circuits were balanced to zero and then the vehicle was lowered and the jacks removed. The voltage change across the bridge circuits was measured
using a digital voltmeter and the wheel load was then calculated using
the appropriate calibration constants.

4.8 STATIC PITCH TEST

The static pitch test was used to determine the static load
sharing characteristics of various suspensions as a function of trailer
pitch angle. The intent is to explore the magnitude of the suspension
equalization variations that can be expected when the tractor and
trailer riding heights are mismatched. Heavy jacks were used to raise
the fully loaded vehicle at the tractor's fifth wheel thereby inducing
a pitch angle to the trailer's suspension.

4.9 SHAKE TEST

NRC's four post shaker rig was used to demonstrate the
importance of considering inertial forces outboard of the strain gauge
when evaluating dynamic wheel loads. The experiment consisted of
lowering the air suspension lift axle on two load cells and exciting
the wheels of the axle with two hydraulic actuators. The analysis of
the spring mass system along with the experimental results are found in
sections 5.2 and 6.2 respectively.
5.0 ANALYSIS

5.1 RESOLUTION OF DYNAMIC WHEEL LOAD

The resolution of vertical wheel loads at the pavement surface requires two data sources. One being the dynamic axle load as measured by the strain gauged axles and the other being the vertical inertial component of the mass outboard of the strain gauges. This mass is comprised of tires, rims, brake hardware and a portion of the axle. The inertial force is resolved by multiplying the measured vertical acceleration of the axle by the above mentioned mass. The sum of the vertical axle force and the vertical inertial force yields the force as seen at the tire/roadway interface.

In algebraic terms

\[
\text{Total Dynamic Wheel Force} = \text{Vertical} \quad \text{Dynamic Axle Load} \quad \text{Acceleration} \times \text{End of Axle Mass}
\]

The following brief analysis proves the need to consider the vertical inertial forces.

The derivation of the equation of motion (1) makes use of the following assumptions. First, the axle is analysed as a simply supported beam, that is, the axle bending moments at points A and B in Figure 7 are zero and the member is free of axial load. The axle is considered to be a rigid body with two degrees of freedom, namely, bounce and roll, described respectively by \( x \) and \( \alpha \).
A free body diagram of the axle reveals that the reaction at point A, $R_A$, is a function of the forces transmitted by the leaf springs or air bags, $F_1$ and $F_2$, and the inertial forces of the axle;

$$R_A = F_1 + (F_2 - F_1) \frac{a}{\ell} - \frac{1}{2} \frac{m_a}{a} \ddot{x} + I_a \ddot{\alpha} \ell$$  \hspace{1cm} (1)

The term $m_a$ is the mass of the axle and $I_a$ is the axle roll moment of inertia about its center of mass. The linear dimensions $a$ and $\ell$ are defined in Figure 8.

![Figure 8](image)

Assuming that the strain gauge at point A' accurately monitors the reaction $R_A$, one may proceed with the analysis of the free body diagram shown in Figure 9.

![Figure 9](image)
The equation of motion for the above figure is simply

\[ m_1 x_1 + c_1 \dot{x}_1 + k_1 x_1 = R_A \]  \hspace{1cm} (2)

The load transmitted to the pavement, \( T \), is given by

\[ T = k_1 x_1 + c_1 \dot{x}_1 \]

By making use of equation (2), \( T \) can be expressed in the form,

\[ T = R_A - m_1 \ddot{x}_1 \]  \hspace{1cm} (3)

The variable \( m_1 \) represents the inertial mass outboard of the strain gauge and \( x_1 \) is the vertical acceleration of the inertial mass.

The breakdown of the inertial mass components is as follows:

<table>
<thead>
<tr>
<th>Component</th>
<th>Mass (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tires and Rims Qty 2</td>
<td>207.5</td>
</tr>
<tr>
<td>Hub and Drum Qty 1</td>
<td>62.2</td>
</tr>
<tr>
<td>Brake Shoes Qty 2</td>
<td>16.3</td>
</tr>
<tr>
<td>Brake S Cam Shaft Qty (1)</td>
<td>3.1</td>
</tr>
<tr>
<td>Wheel Studs and Nuts Qty 10</td>
<td>1.5</td>
</tr>
<tr>
<td>Bare Axle Qty 1/10</td>
<td>10.6</td>
</tr>
<tr>
<td>Total Inertial Mass</td>
<td>301.2</td>
</tr>
</tbody>
</table>

The inertial mass was taken as 300 kg.

Summing of the axle forces and the inertial forces was done in the analogue state using operational summing amplifiers. Depending on the analysis technique required, the summed analogue signal was transferred to a strip chart recorder for direct interpretation of the data or it was digitized for numerical analysis.

The numerical analysis was performed with an IBM-AT personal computer. The computer was equipped with a four channel analogue to
digital converter, and the necessary software was developed to perform the numerical functions (see Appendix D).

The statistical functions of interest were:
- The mean
- First standard deviation
- 5th and 95th percentile and their corresponding histogram plots.

The sampling rate was 300 points/sec/channel.

The dynamic load coefficient was intended as the principle numerical quantity to be used for the analysis of continuous dynamic data. Known in statistics as the coefficient of variation, the DLC is defined as:

\[
DLC = \frac{S}{Z}
\]

where:
- \( S \) = standard deviation of the wheel forces distribution (kN)
- \( Z \) = overall mean wheel forces (kN)

The introduction of the DLC is based on the assumption that its numerical value is independent of the variation in the overall mean wheel force. Hence, the DLC allows one to compare different suspensions tested with different overall mean wheel loads. However, we find that a change in \( Z \) leads to a variation in the DLC.

Take for an example the test condition where the truck had a walking beam suspension as the drive axles and a four spring suspension as the trailer axles. A test was carried out where the drive axle wheel loads were decreased by 13%. The decrease in the standard
deviation was evaluated at 6% instead of the expected 13% decrease. By comparison, when the four spring trailer axle load was reduced by 24%
there was a reduction in the standard deviation of 19%, which is within acceptable limits.

It is evident from this exercise that the relationship between S and Z may not be linear and, moreover, it may be dependent on the suspension type, that is

\[ S = DLC \times Z \]

where

\[ DLC = DLC(Z, \text{suspension type}) \]

To eliminate possible confusion resulting from variations in the static wheel load, the DLC will not be used as the primary analysis term. It will be replaced by the standard deviation of the dynamic wheel force. This term will be examined as a function of vehicle speed, road roughness and suspension type. The DLC did prove useful however, during the final analysis with appropriate consideration.

5.2 SHAKE TESTS

To further explore the axle force and inertial force contributions of the vehicle measurement system, a vertical shake test was performed. The air lift suspension of the fully loaded vehicle was supported at the tires with two electro hydraulic vertical actuators. A low amplitude sinusoidal input, equal to the resonant frequency of the suspension, was applied. The configuration of the Vehicle Dynamics Laboratory required that the rear end of the trailer be supported by a
crane. The tractor, for its part, rested on the ground in its usual position. Figure 10 depicts a model of the spring mass system under study.

Figure 10

The span between points A and B is assumed to be small enough so that the displacement $x_1$ of the trailer mass, M, describes the motion of both points A and B, despite possible pitch of the trailer. The mass elements $m_1$, $m_2$ and $m_3$ are assumed to be rigidly attached to one another. The quantity $m_2$ represents the mass outboard of the strain gauge, while $m_1$ and $m_3$ are the mass of the radius arm and axle respectively. A free body diagram of the system is shown in Figure 11.
The sum of the moments about point 0 leads to

\[
k_2(l_2^2 + x_1 - h)l_2 + c_2(l_2^2 + x_1 - h)l_2 + k_1^1 l_1^2 + c_1 l_1^2
\]

\[+ k_3^1 l_3^2 + c_3 l_3^2 + I_0 + m_3 l_2^2 + m_1 L_1 x_1 + m_3 l_2 x_1 \]  

\[+ m_2^2 \dot{\theta} + m_2 l_2 x_1 = 0 \]  

(4)

The linear dimensions \(l_1, l_2\) and \(l_3\) are as defined on Figure 11, \(I\) is the mass moment of inertia of \(m_1\) about point 0, \(L_1\) is the distance of the mass center of \(m_1\) from point 0 and \(k_1, k_2, k_3\) and \(c_1, c_2, c_3\) are stiffness and damping coefficients respectively.

The sign convention adopted for the axle strain gauge is the following: a downward force applied to the axle inboard of the strain gauge is considered positive and results in a positive voltage output. Hence the force \(F_S\) monitored by the strain gauge (that is the inertial and spring forces inboard of the strain gauge) takes the form

\[
F_S = - (k_1 l_1^2 \dot{\theta} + c_1 l_1^2 \ddot{\theta} + k_3 l_3^2 \dot{\theta} + c_3 l_3^2 \ddot{\theta} + I_0
\]

\[+ m_3^2 \dot{\theta} + m_1 L_1 x_1 + m_3 l_2 x_1) \times l_2 \]  

(5)
Defining the actuator force $F_A$ as being positive in the upward direction leads to
\[ F_A = c_2(l_2 \ddot{\theta} + \dot{x}_1 - \ddot{h}) + k_2(l_2 \dot{\theta} + x_1 - h) \] (6)

Equation (4) can therefore be written as
\[ F_A = F_g - m_2 \ddot{x}_2 \] (7)
where $\ddot{x}_2 = l_2 \ddot{\theta} + \ddot{x}_1$ is the vertical acceleration of $m_2$ which is monitored by an accelerometer fastened to the axle right next to the wheel hub. The final result [Equation (7)] is identical to that obtained in Equation (3) on page 26.
6.0 RESULTS
6.1 SPECIAL CASES

The intent of the special cases section is to provide some typical measured values in the form of general interest material.

6.1.1 Impact Wheel Loads Associated With A Grade Level Railway Crossing

As described earlier, this single track railway crossing was considered to be typical in terms of roughness. The Mays meter roughness output for the 80 meter increment of the road containing the crossing was 252 IPM (RCR 4.6). The road roughness in the vicinity of the crossing was approximately 60 IPM (RCR 8.4).

As would be expected, the dynamic wheel loads resulting from passing over a railway crossing are velocity dependent. The peak wheel load forces associated with the impact of the crossing, and the peak level after one cycle, are listed in the table below. The forces are expressed as a ratio of the total peak load/static load.

The peak level after one cycle occurred at points 10, 7.5, and 5 meters beyond the crossing for vehicle speeds of 80, 60 and 40 km/hr, respectively. The static axle load was 10 tonnes, 5 tonnes per dual wheel. (Five tonnes is equivalent to 49 kN.)
6.1.2 Dynamic Bridge Loading

As mentioned previously, a number of bridges were crossed during each of the tests conducted on Highway 417. The bridge approach varied from smooth (undetectable) to very rough. One particular bridge on a recently repaved section of road produced quite large vehicle responses. At a speed of 80 km/hr and a static axle load of 10 tonnes, the maximum wheel force experienced while crossing the bridge was 2.1 times the static load. The peak total wheel load associated with the roadway before and after the bridge was only 1.2 times the static load.

6.1.3 Tire Scuffing in Turns as a Function of Axle Spread

A typical intersection was used as our standard for these tests. In all cases, the vehicle negotiated the turn at the same creep speed, following the same wheel path. All tests were conducted on the same suspension at three different spreads, i.e. 1.27, 1.83 and 2.44 meters. The side force generated by the tires was measured continuously through the turn.
Findings show that because tire saturation occurred in all cases, there was no appreciable difference in the magnitude of the scuffing forces as a function of spread. There is, however, a significant difference in the duration of the scuffing for a given turn. Using the 1.27 meter axle spread as a baseline, the 1.83 meter spread increases the scuffing distance by approximately 17%, and the 2.44 meter axle spread increases the scuffing distance by 30%.

Increased trailer axle spread also increases the tractive forces and side forces on the tractor drive tires. The reason for this is that the tractor drive tires must overcome the yaw moment induced by the trailer axles while in a turn. An increase in the spread of these trailer axles will see a proportional increase in the yaw moment and thus an increase in the required tractive effort from the tractor. The increase in yaw moment of the trailer also means an increase in lateral bending moment in the frame structure of the trailer.

6.1.4 Suspension Load Equalization Due to Variations in Trailer Pitch Angle

Within the industry there are variations in the 5th wheel height of tractors and the coupling heights of trailers. Mismatching tractor and trailer coupling heights will result in variations in the pitch attitude of a trailer and its suspension. To gain an appreciation for the load equalization sensitivity to pitch attitude of the various suspensions, a static pitch test was performed. Since the study is interested in wheel loads, the experiment focused on wheel load variations rather than axle load variations.
The criteria used to assess the pitch load equalization characteristics of the suspension is in the form of percentage load transfer (PLT) recorded for pitch angles varying from 0.2 to 1.2°. PLT is defined as follows:

\[
PLT = \frac{\text{change in trailing wheel load} - \text{change in lead wheel load}}{\text{Total wheel load of the group}} \times \frac{100\%}{\text{degree}}
\]

The results of the pitch test are as follows.

Both walking beam suspensions, Chalmers and Hendrickson had excellent PLT results of better than 3%.

The Neway air suspension results were the most difficult to interpret because it is an active suspension in that it utilizes height sensing control valves to maintain constant displacement from the axle to the trailer chassis. These control valves have a displacement lag feature which results in side to side differential wheel loading when the suspension is tested in a quasi-static manner. In some cases equalization was near perfect while in other cases there was a measurable difference. The highest PLT recorded with the air suspension was 6%.

The four spring suspension (Reyco) displayed the highest sensitivity to pitch variation of all suspensions examined by this study. PLT results of 14-17% were recorded depending on axle spread. 14% corresponded to the 1.27 meter spread while 17% corresponded to the 2.44 meter spread.
6.2 SHAKE TESTS

As indicated in section 5.2 the spring mass system from the shake test experiments could be modelled as a two degree of freedom system. However, to verify that the general properties of the mechanical system are properly monitored, it is less cumbersome to treat the axle-wheel combination as a one degree of freedom system with no damping. The following paragraphs justify such a simplification.

The shake tests were conducted without shock absorbers on the air suspension, thus considerably reducing the magnitude of the damping coefficients $c_1$ and $c_3$ of Equation (4) on page 30. Upon neglecting viscous forces associated with tires and air bags we obtain, for a sinusoidal forcing function, the following steady state solution for the strain gauge force;

$$F_S = [(k_{11}^2 + k_{33}^2) - \omega^2(I + m_{33}^2)]x_2/l_2^2\omega^2$$
$$+ [\omega^2(I - m_{11}^2) - (k_{11}^2 + k_{33}^2)]x_1/l_2^2\omega^2$$

The factors multiplying $x_1$ and $x_2$ are of the same order of magnitude, however, video tape of the experiment clearly showed that the displacement $x_1$ was of second order in comparison with $x_2$. Hence $F_S$ may be approximated to

$$F_S = [k_{11}^2 + k_{33}^2 - (I + m_{33}^2)]x_2/l_2^2\omega^2$$

(8)

which is precisely the result obtained from a one degree of freedom analysis. By making use of the steady state solution for a one degree of freedom system we may rewrite the expression for the actuator force, Equation (6) on page 31, as

$$F_A = [k_{11}^2 + k_{33}^2 - (m_{11}^2 + I + m_{33}^2)]x_2/l_2^2\omega^2$$

(9)
Equations (8) and (9) reveal that for small driving frequencies \( \omega \) the vertical acceleration \( x_2 \) is in phase with \( F_S \) and \( F_A \). As we increase the driving frequency we can expect a 180° phase shift between \( x_2 \) and the signals \( F_S \) and \( F_A \). If we define \( \omega_A \) and \( \omega_S \) as the frequencies at which a 180° shift occurs between \( F_A \) and \( x_2 \), and \( F_S \) and \( x_2 \), then the following results can be established from Equations (8) and (9):

\[
\omega_A < \omega_n, \quad \omega_S > \omega_A
\]

where the natural frequency for the system, \( \omega_n \), is

\[
\omega_n^2 = \frac{k_1 l_1^2 + k_2 l_2^2 + k_3 l_3^2}{(m_2 l_2^2 + I + m_3 l_2^2)}
\]

All of the above theoretical observations have been experimentally confirmed. The quantities \( \omega_A, \omega_S \) and \( \omega_n \) were measured as

\[
\omega_A \approx 4.5 \text{ Hz}
\]

\[
\omega_S \approx 6.5 \text{ Hz}
\]

\[
\omega_n \approx 11.0 \text{ Hz}
\]

The simplifications which were made in the above discussion served only to insure that the instrumentation properly monitored the general mechanical behavior of the system. It must be emphasized that these simplifications are not required in the analysis of dynamic calibration. If the model described in section 5.2 is accurate then the strain gauge and load cell will monitor precisely every term in Equations (5) and (6). Hence Equation (7), on page 31, remains an exact relation.

Shake tests were performed at several different driving frequencies and amplitudes. Using Equation (7) the inertial mass outboard of the strain gauge was evaluated at 234 kg with a standard
deviation of 9 kg. This value is 22% lower than the actual mass outboard of the strain gauge.

A portion of the discrepancy between the outboard mass measured from dynamic calibration and the actual mass of the wheel could probably be attributed to the fact that the elements of the tires are not all accelerated at $x_2$. The wheel could be modelled with say, one third of the mass of the two tires (38 kg) having a vertical acceleration of $h$ while the remaining portion of the wheel be accelerated at $x_2$. For cases where $h$ is small this would account for 57% of the difference in the recorded mass.

Although dynamic calibration is in its preliminary stages of development the experiment confirmed the presence and the importance of measuring the inertial forces associated with the mass outboard of the strain gauge. For example, when excited at the resonant frequency of 11 Hz the inertial forces accounted for 58% of the force transmitted to the shaker's load cell.

The dynamic calibration experiment suggests that a 22% lower inertial mass should be used when evaluating the inertial forces. However, before the mass obtained from dynamic calibration can be confidently used we must develop a mathematical model of the wheel which will account for its entire static mass. Also the four post shaker is a complex apparatus for which the accuracy and limitations have yet to be fully assessed.

Any error associated with the simplified model used to determine the inertial forces applies equally to all suspensions tested. Although the actual wheel force may be in error by as much as
10% depending on road roughness, the accuracy of the system for unit to unit comparison is better than 5%.

6.3 ROAD TESTS

The results presented here were derived from the road tests outlined in Section 4.3. The data from these tests has been processed and is presented in graphical form in Appendix E. Selected graphs have been included in this section to illustrate points of discussion. All graphs are plotted in the form of standard deviation of dynamic dual wheel loads verses ride comfort rating (RCR). RCR classifies road roughness on a scale of 1 to 10 where 1 is rough and 10 is smooth.

6.3.1 Dynamic Wheel Loads as a Function of Suspension Type

Figures 12, 13, 14 and 15 illustrate how the different suspensions tested compare in terms of dynamic wheel load resulting from the same road input. The following observations can be made from these graphs.

a) It is clear that as the road becomes smoother the dynamic wheel loads diminish and tend to converge to small values independent of the suspension type.

b) Dynamic wheel loads vary with vehicle speed in an exponential form. Consider for example the four spring suspension with a static dual wheel load of 50 kN (5 Tonnes) traveling over a road with a RCR of 5.2 at a speed of 80 km/hr. A marginal increase in dynamic wheel load of 20% is seen when the vehicle speed is increased from 40-60 km/hr. However, when the speed is increased from 60-80 km/hr a substantial increase in dynamic wheel load of 150% is seen.
VARIATION OF TRAILER SUSPENSION TYPE
TRACTOR SUSPENSION - HENDRICKSON
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN
NOMINAL TRAILER AXLE SPREAD OF 1.3 M

Figures 12 (upper), 13 (lower)
VARIATION OF TRACTOR SUSPENSION TYPE
TRAILER SUSPENSION - CHALMERS
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN
NOMINAL TRAILER AXLE SPREAD OF 1.3 M

Figures 14 (upper), 15 (lower)
c) Variations in dynamic wheel load are evident when comparing the different suspensions tested. The results presented in the following table are based on a vehicle speed of 80 km/hr and a RCR of 5.2, and expressed using a derivative of Sweatman's Dynamic Load coefficient

\[
DLC = \frac{\text{Standard Deviation of wheel load}}{\text{Nominal Static Wheel Load (50 kN)}}
\]

where nominal static wheel load is substituted for mean wheel load.

<table>
<thead>
<tr>
<th>SUSPENSION TYPE</th>
<th>DYNAMIC LOAD COEFFICIENT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Bag</td>
<td>16%</td>
</tr>
<tr>
<td>Four Spring</td>
<td>24%</td>
</tr>
<tr>
<td>Leaf Spring Walking Beam</td>
<td>28%</td>
</tr>
<tr>
<td>Rubber Spring Walking Beam</td>
<td>39%</td>
</tr>
</tbody>
</table>

6.3.2 Dynamic Wheel Load as a Function of Trailer Suspension Spread

Changes in dynamic wheel load associated with variations of trailer axle spread are generally small; expressed in terms of DLC they are less than 3%. See Figures 16, 17, 18 and 19. The only exception to this occurred with the four spring suspension. The largest variations occurred at 80 km/hr where changes in DLC as high as 8% were measured.

With the four spring suspension the order of preferred axle spread varies with road roughness. On rough roads at 80 km/hr, the order of suspension spread from most favorable to least favorable is 1.83, 1.27 and 2.44 meters. On smooth roads at 80 km/hr, the order is reversed, that is, 2.44, 1.27 and 1.83 meters.
Clearly, the results of Figures 16 to 19 reveal that the air suspension is only modestly sensitive to axle spread variations for all conditions of speed and roughness at which it was tested. On the other hand, at 80 km/hr the four spring suspension demonstrated acute sensitivity to changes in axle spread. The difference in axle spread sensitivity of the two suspensions is undoubtedly linked to the fact that the air suspension is mechanically independent whereas the four spring suspension is mechanically dependent. A change in axle spread on the four spring suspension does not only relocate the spring elements along the trailer but it also changes the mechanical properties of the suspension group. Hence, we may conclude that dynamic wheel load is insensitive to axle spread provided that changes in axle spread do not inherently alter the kinematic components of the tandem suspension.

6.3.3 Dynamic Wheel Load as a Function of the Number of Axles in a Suspension Group

As mentioned in section 4.2, adding an axle (air lift suspension) to a tandem axle suspension resulted in a 16% decrease of the tractor static wheel load and a 22% reduction of the trailer static wheel load. Figures 20, 21, 22 and 23 reveal that a reduction in static wheel load does not necessarily translate into a proportional decrease in dynamic wheel load. In general the percentage decrease in dynamic wheel load is less than the percentage decrease in static load.
VARIATION OF TRAILER AXLE SPREAD
TRACTOR SUSPENSION - HENDRICKSON
TRACTOR SUSPENSION - NEWAY
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

Figure 16 (upper), 17 (lower)
VARIATION OF TRAILER AXLE SPREAD
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - REYCO
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

Figure 18 (upper), 19 (lower)
EFFECT OF AIR SUSPENSION LIFT AXLE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - REYCO, AXLE SPREAD OF 1.27 M

Figure 20 (upper), 21 (lower)
EFFECT OF AIR SUSPENSION LIFT AXLE
TRACTOR SUSPENSION - MENDRICKSON
TRAILER SUSPENSION - CHALMERS, AXLE SPREAD OF 1.37 M

Figure 22 (upper), 23 (lower)
In some cases, however, the percentage reduction in dynamic wheel load can be very near the percentage decrease in static wheel load. Take for example the rubber spring walking beam suspension. With a vehicle speed of 80 km/hr and a RCR of 5.2, the triaxle group reduced the dynamic component of the wheel load by 17%. This is not surprising in light of the observation that much of the dynamic signal was generated by the low frequency whole body reaction. By adding a third suspending element the dynamic component of the sprung mass was distributed over three support members rather than two. It can be concluded therefore that the addition of a third axle will not likely result in an increase in dynamic wheel load within the bogie. The experimental results show that if the sprung mass is held constant the addition of a third axle will result in a decrease in dynamic wheel load.

6.3.4 Axle to Axle Dynamics for Load Sharing Suspensions

The series of plots on pages E20 to E29 of Appendix E show the axle dynamics for each axle of a given suspension. From these graphs it is clear that the lead drive axle has a slightly higher dynamic load (5%) than that of the trailing axle. This difference in dynamic axle load is attributed to the added mass of the interaxle differential on the lead axle.

The trailer axle dynamics do not show consistent load biasing between the lead and trailing axles. Load variations are within 5% and show similar trends. When comparing suspensions the standard deviation of the lead and trailing axles of a suspension were averaged.
6.3.5 Load Transfer Due to Braking

The four spring suspension was the only suspension tested that exhibited load transfer due to braking. For a moderate brake application of 5 KN-m (4400 lb-in) the trailing axle of the four spring suspension increased in load by 15-20%, while the lead axle decreased by a similar amount.

6.3.6 The Air Suspended Lift Axle

The air suspended lift axle was of special interest to the study. The unit tested had the same dynamic characteristics as the tandem axle air suspension. The lift axle maintains its axle loading by means of a constant pressure air regulator. The ability of this suspension to accept its share of the vehicle load over various conditions of road roughness and discontinuities was found to be excellent. Because it operates under constant air pressure settings, its load carrying ability is sensitive to vertical displacement variations. It is important, therefore, that the suspension mounting height specifications be adhered to and that axle spread be minimized so that optimum performance can be achieved. Failure to do so will result in variation of the mean axle load.

Provided that the air pressure regulator is properly set, this suspension can be classified as a load sharing suspension when used in conjunction with other suspension types in the appropriate manner. Regulators will want to consider the questions of where the control system should be mounted (i.e. cab or trailer) and when it is appropriate to lift the axle. A method of verifying that the air
regulator is properly set would be useful. Perhaps a single axle weight scale should be used at inspection stations.

6.4 **PAVEMENT DEFLECTION DUE TO DYNAMIC WHEEL LOADS**

A limited experimental study was undertaken to determine if there was a first order connection between pavement deflection and strain transducers as described in Reference 12. The intent of the experiment was to relate pavement deflection with vehicle speed and wheel loads monitored by the test vehicle.

The dynamic impulse to the axle was generated with a simple wooden plank fixed to the roadway directly over the deflection transducers and laying parallel to the axis of the vehicle's axles. The cross section of the plank measured 4 cm x 24 cm and was sufficiently long that all wheels of a given axle impacted the plank.

The axle loads for the test program are listed in Table 4. The axle number refers to the particular axle on the truck. The tractor steering axle is "1" and the last trailer axle is "6".

<table>
<thead>
<tr>
<th>Axle Number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured</td>
<td>7.5</td>
<td>7.5</td>
<td>7.0</td>
<td>6.7</td>
<td>6.7</td>
<td></td>
</tr>
</tbody>
</table>

**NOTE:** Axle loads were slightly lower than those used during the main study.

Three vehicle speeds were used for the above loading sequence, i.e. 18, 37, and 60 km/hr. Tests were run with and without a plank in place.
RESULTS

Dynamic axle loads as measured by the truck were compared with pavement deflection data recorded on site.

Upon review of the data, it became clear that selective analysis was required to eliminate confusion and convey the primary findings needed by the study. The source of confusion centered on the behaviour of the trailing axle of any of the load sharing suspensions. The fact that the lead and trailing axles of a given suspension are coupled in terms of load sharing, and the fact that the axles are in close proximity to each other, gave deflection results which were in some ways mysterious.

When, on the other hand, only the lead axles of the suspensions were studied, the results were found to be sufficiently clear and simple so that basic conclusions could be drawn.

All results are presented in graphical form.

Figure 24 shows pavement deflection as a function of vehicle speed with and without a bump. It is interesting to note that without a bump pavement deflection diminishes with increased vehicle speed. However, with a bump, the deflection increases with speed and in one case the deflection changed by a factor of 1.8 when compared with the same run without a bump.

Figure 25 shows pavement deflection versus dynamic impact factor for two separate axles at 3 test speeds.

\[
\text{dynamic impact factor} = \frac{\text{peak dynamic force}}{\text{static axle load}}
\]
PAVEMENT DEFLECTION vs. SPEED

PAVEMENT DEFLECTION vs. DYNAMIC IMPACT FACTOR
WOODEN PLANK FIXED TO ROADWAY

Figures 24 (upper), 25 (lower)
As can be expected, the experimental results show that pavement deflection increases with the dynamic impact factor. The best line fit drawn through the entire data set shows first order correlation between pavement deflection and dynamic wheel load. The scatter about the line can be attributed to secondary effects associated with the time history leading to peak dynamic forces. In turn, the time histories are related to the mechanical behaviour of individual suspensions. Also included in Figure 25 are dashed lines joining data points for each axle tested.
7.0 CONCLUDING REMARKS

7.1 INTERPRETATION OF EXPERIMENTAL RESULTS

The experimental results presented in this paper are pertinent to specific products which have been clearly indentified by the manufacturer's product number. It should be emphasized that the experimental results should in no circumstances reflect the whole product line of an individual manufacturer nor should the results be used to generalize on the performance of a generic suspension type. Subtle design variations in suspension components could in some cases lead to substantial changes in the performance of a suspension group. The objective of the study was not to endorse a product but rather to investigate the range of performance possible with these diverse systems.

7.2 RELATING STUDY FINDINGS TO VEHICLE WEIGHTS AND DIMENSION REGULATION

Findings from this study can be applied to the decision making process of heavy vehicle weights and dimensions regulations. Some of the results are straightforward and clear while others are rather complex and must be considered in conjunction with other findings.

Dealing with the straightforward findings first, we can draw two simple conclusions:

1. Axle Spread Sensitivities

   a) Dynamic wheel load was found to be insensitive to axle spread per se. For mechanically dependent suspensions where variations
of axle spread imply a different mechanical configuration of the suspension, then substantial variations in dynamic wheel load can be expected. For these suspensions an optimum axle spacing will be difficult to establish since the order of preferred axle spread varies with road roughness.

b) Closer axle spreads for a given suspension group reduced tire scuffing and improved the pitch induced axle load equalization. This implies that, in a multi axle suspension group, with or without lift axles, the spacing between the first and the last axle should be minimized.

2. The Number of Axles in a Suspension Group

The addition of a third axle to a tandem axle suspension group reduced the dynamic wheel loads. However, as seen in section 6.3.2, for a given percentage increase in static wheel load we can generally expect a smaller percentage increase in the dynamic wheel load. The difference between the two percentages varies according to the type of suspension tested. These differences range from no perceptible increase to an increase proportional to the change in static wheel load. Hence, a conservative approach in estimating the total pavement load due to an increase in static wheel load is to expect a proportionate increase in dynamic wheel load.

Comparative Considerations of Various Suspension Systems

To compare various suspension systems requires the consideration of a number of factors. What follows is a brief discussion focusing on some of these factors with some thoughts on how they might be treated in the evaluation process.
Heavy truck load sharing suspensions serve to distribute the load of a vehicle evenly between axles and to mitigate the vertical dynamics of the whole vehicle body. It must do this over a wide range of road roughness conditions, vehicle speeds and hardware variations.

This study demonstrated that all suspensions show convergences to low dynamic activity on smooth roads even at highway speeds. On moderate to rough roads however, the dynamic characteristics of the suspensions vary widely depending on the suspension type examined. There is a clear order of suspension preference in terms of dynamic wheel loading. The following list of trailer suspensions tested is arranged in order from the lowest to the highest dynamic wheel loading.

1. Air suspension
2. Four spring suspension
3. Walking beam

On the basis of these findings one may be tempted to favour or restrict the use of particular suspensions through some regulatory means such as added or restricted load allowances. It would be inappropriate to make such a decision on the basis of the dynamic wheel loading alone. Other factors such as the sensitivity of the suspension to static load equalization, load transfer due to braking and pitch should also be considered. In addition, the contribution of the suspension to vehicle stability and control must be evaluated.

Clearly, the criteria listed above do not necessarily carry the same weight in the evaluation process. Consider the following points.
Load Equalization

a) A suspension group is said to have unfavorable load equalization characteristics if the axles of a given suspension do not share load equally. Pronounced load unbalance between axles of a tandem suspension becomes critical in all cases where road and bridge structures are designed on the assumption of perfect load equalization.

b) Brake induced load transfer is of little concern on major highways except on down hill grades. Highway junctions and city intersections would suffer most from this effect.

c) Load transfer due to pitch attitude would occur when tractor and trailer coupling heights are poorly matched. This would represent a small percentage of the vehicle population and therefore may be considered to be of less significance.

Dynamic Wheel Loads

Dynamic wheel loads are a function of suspension type, vehicle speed and road roughness. The largest variation in dynamic wheel load characteristics of the various suspensions occurred at highway speeds. 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. Both axles of the suspension experience this portion of dynamic load in phase with each other. All wheel loads of a given suspension are greater than the nominal static loads for half of the period of the vehicle body oscillation. Conversely, for the other half of the period the wheel loads are less than the nominal static loads.
Consider for example a simplified vehicle with a linear suspension loaded to the legal limit traveling at 90 km/hr. We will assume that the natural frequency of the whole body is 3.5 Hz. For these conditions the vehicle will travel 7.1 meters for every complete cycle of the vehicle mass. As seen in Figure 26 this means that 3.6 meters of the highway will experience loads in excess of the desired legal limit and the following 3.6 meters of highway will experience loads below the legal limit. Consider a second and third vehicle with a natural frequency of 1.5 and 2.5 Hz, superimposed on the same stretch of roadway as shown in Figure 27. It becomes clear that, because every vehicle has different dynamic characteristics, a general band of loading can be expected. We will call this the Load Band as shown in Figure 28.

Consider the Load Band generated by the two suspensions, A and B, shown in Figure 29. Suspension A has a higher Load Band than does suspension B. Thus suspension A imposes higher maximum road loading forces than suspension B. Considering the effects of the fourth power law of pavement damage, a small reduction in Load Band should reflect a sizable gain in the expected fatigue life of the road structure.

In view of all the road loading factors discussed here, it is clear that dynamic wheel loading emerges as the single most significant suspension performance factor. When examining the dynamic wheel load performance of the various suspensions tested it is the opinion of the authors that the spread between the best and the worst performers is
Figures 26 (upper), 27 (middle), 28 (lower)
too high. This suggests the need for the establishment of an evaluation procedure which would establish acceptable suspension performance criteria. The key considerations should include vehicle stability and control performance, dynamic wheel load characteristics, and interaxle load sharing capabilities.

Further research should also consider the use of single axle enforcement weigh in motion scales. Because axle dynamics diminish with speed and roughness, precise axle load measurements should be possible with the vehicle rolling at slow speeds (≈ 10 km/hr). With the use of existing technology, this would provide information on single axle loads and total group loads. One would then be in a position to monitor axle load equalization which would likely promote longer pavement life.
REFERENCES


APPENDIX A

MECHANICAL DESIGN DRAWINGS OF HARDWARE FABRICATED FOR THIS STUDY
7 3/4
6 7/8
2
1 1/8
45° CHAMFER (both ends)

LETTER H DRILL THROUGH 2 HOLES AS SHOWN

Pivot Pin
Dynamic Suspension Study Vehicle

TOLERANCE TOLERANCE
ST ST 416

HEAT TREAT/Traitement thermique

FINISHING G3

SCALE/ÉCHELLE FULL SIZE 2

Canada CANROAD

MATERIAL/MATÉRIEL

ST 416

DRAWING/LIBRÉ
J.A. HUTCHINS 22574

CHECKED/CONFIRMÉ

APPROVED/APPUI

DATE 27-6-85

REQUEST/ETUI

ASSEMBLY NO/ENSEMBLE N° RL037B008

CANADA

NOTES

UNLESS OTHERWISE NOTED - SAUF INDICATION CONTRAIRE

UNITÉS/UNITÉS

LAB, ORDER NO/PREPAREUR N° RL037E001

ASSEMBLY NO/ENSEMBLE N° RL037B008
APPENDIX B

VEHICLE CALIBRATION

Because of the importance of this subject to the success of this study, a formal report was written so that details would be available to those interested.

The calibration report was written by Mr. Karl R. LePiane, a third year Mechanical Engineering Co-Op student from Waterloo University who spent his work term at NRC Vehicle Dynamics Laboratory. The report constituted his required work term report for which he received the Babcock and Wilcox Award for best work term report in the Mechanical Engineering Department.

Readers will note that an alternative calibration method was suggested in the report. This procedure was carried out and the results formed our final axle load calibration standard. These final results are attached to the end of this appendix.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>SUMMARY</td>
<td>1</td>
</tr>
<tr>
<td>CONCLUSION</td>
<td>11</td>
</tr>
<tr>
<td>RECOMMENDATIONS</td>
<td>111</td>
</tr>
<tr>
<td>1. INTRODUCTION</td>
<td></td>
</tr>
<tr>
<td>1.1 Background</td>
<td>1</td>
</tr>
<tr>
<td>1.2 Task</td>
<td>1</td>
</tr>
<tr>
<td>2. THE INSTRUMENTS USED AND THE MANUFACTURERS' SPECIFICATIONS</td>
<td></td>
</tr>
<tr>
<td>2.1 Axles</td>
<td>1</td>
</tr>
<tr>
<td>2.2 Strain Gauge Transducers</td>
<td>3</td>
</tr>
<tr>
<td>2.3 Strain Gauge Conditioning Amplifier</td>
<td>3</td>
</tr>
<tr>
<td>2.4 Instruments Used for Torque Calibration</td>
<td>4</td>
</tr>
<tr>
<td>2.5 Instruments Used for Vertical Load Calibration</td>
<td>4</td>
</tr>
<tr>
<td>3. THEORETICAL BACKGROUND</td>
<td></td>
</tr>
<tr>
<td>3.1 Torsional Shear Stress</td>
<td>5</td>
</tr>
<tr>
<td>3.2 Vertical Loading Stresses</td>
<td>7</td>
</tr>
<tr>
<td>3.3 Stress Strain Relationships</td>
<td>9</td>
</tr>
<tr>
<td>3.4 The Strain Gauge Transducer</td>
<td>10</td>
</tr>
<tr>
<td>3.5 Strain Gauge Orientation for Torsion</td>
<td>10</td>
</tr>
<tr>
<td>3.5.1 Transformation of Strain</td>
<td>12</td>
</tr>
<tr>
<td>3.5.2 Gauge Positioning on the Axle and the Wheatstone Bridge</td>
<td>13</td>
</tr>
<tr>
<td>3.5.3 Vertical Loading Effect on the Torsion Strain Gauges</td>
<td>17</td>
</tr>
<tr>
<td>3.5.4 Brake Force Effect on the Torsion Strain Gauges</td>
<td>19</td>
</tr>
<tr>
<td>3.6 Strain Gauge Orientation for Vertical Loading</td>
<td>20</td>
</tr>
<tr>
<td>3.6.1 Gauge Positioning on the Axle and the Wheatstone Bridge</td>
<td>22</td>
</tr>
<tr>
<td>3.7 The Strain Gauge Conditioning Amplifier</td>
<td>23</td>
</tr>
<tr>
<td>Section</td>
<td>Title</td>
</tr>
<tr>
<td>---------</td>
<td>-----------------------------------------------------------------------</td>
</tr>
<tr>
<td>3.8</td>
<td>Calibration</td>
</tr>
<tr>
<td>3.8.1</td>
<td>Least Squares Method of Best Fit</td>
</tr>
<tr>
<td>3.8.2</td>
<td>Gauge Sensitivity</td>
</tr>
<tr>
<td>3.8.3</td>
<td>Electrical Calibration</td>
</tr>
<tr>
<td>3.9</td>
<td>Error Analysis</td>
</tr>
<tr>
<td>3.9.1</td>
<td>Error in Least Squares Calculation</td>
</tr>
<tr>
<td>3.9.2</td>
<td>Method of Application</td>
</tr>
<tr>
<td>4.</td>
<td>TORQUE CALIBRATION</td>
</tr>
<tr>
<td>4.1</td>
<td>Calibration Procedure</td>
</tr>
<tr>
<td>4.2</td>
<td>Data Analysis</td>
</tr>
<tr>
<td>4.2.1</td>
<td>Initial Tests</td>
</tr>
<tr>
<td>4.2.2</td>
<td>Brake Application Torque</td>
</tr>
<tr>
<td>4.2.3</td>
<td>Beam Torque and Bending Effect</td>
</tr>
<tr>
<td>4.2.4</td>
<td>Sign Convention</td>
</tr>
<tr>
<td>4.3</td>
<td>Final Calibration Tests</td>
</tr>
<tr>
<td>4.4</td>
<td>Discussion of Results</td>
</tr>
<tr>
<td>4.5</td>
<td>Sample Calculation</td>
</tr>
<tr>
<td>5.</td>
<td>VERTICAL LOAD CALIBRATION</td>
</tr>
<tr>
<td>5.1</td>
<td>Weigh Scale Preparation</td>
</tr>
<tr>
<td>5.1.1</td>
<td>Tractor Trailer Preparation</td>
</tr>
<tr>
<td>5.2</td>
<td>Calibration of Drive Axles</td>
</tr>
<tr>
<td>5.2.1</td>
<td>Procedure</td>
</tr>
<tr>
<td>5.2.2</td>
<td>Discussion of Results</td>
</tr>
<tr>
<td>5.3</td>
<td>Trailer Axle Calibration</td>
</tr>
<tr>
<td>5.3.1</td>
<td>Procedure Followed for the Reyco Tandem Set</td>
</tr>
<tr>
<td>5.3.2</td>
<td>Procedures Followed for the Neway Axle</td>
</tr>
<tr>
<td>5.4</td>
<td>Sensitivity of the Measurement System to Bias</td>
</tr>
<tr>
<td></td>
<td>Tire Loading</td>
</tr>
<tr>
<td>5.4.1</td>
<td>Test Procedure</td>
</tr>
<tr>
<td>5.4.2</td>
<td>Results</td>
</tr>
<tr>
<td>5.4.3</td>
<td>Discussion of Results</td>
</tr>
<tr>
<td>5.5</td>
<td>An Alternate Solution</td>
</tr>
</tbody>
</table>

REFERENCES

57
Table of Contents (Cont'd)

APPENDIX A  Vishay Error
APPENDIX B  Confidence Intervals and the Student's t Distribution
APPENDIX C  Initial Torque Calibration Data Tables
APPENDIX D  Torque Calibration Results for Gauge ADS1 and BPS1
APPENDIX E  Torque Calibration Error Analysis Calculations and Results
APPENDIX F  Drive Axle Vertical Load Results
APPENDIX G  Trailer Axle Vertical Load Results
APPENDIX H  Reyco Axle Component Weights
<table>
<thead>
<tr>
<th>Fig. #</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>Axle Types</td>
<td>2</td>
</tr>
<tr>
<td>2.2</td>
<td>Strain Gauges Used</td>
<td>3</td>
</tr>
<tr>
<td>3.1</td>
<td>Torsional Shear Stress Effect</td>
<td>6</td>
</tr>
<tr>
<td>3.2</td>
<td>Vertical Load Shear Stress</td>
<td>8</td>
</tr>
<tr>
<td>3.3</td>
<td>Direct Stress Due to Bending</td>
<td>9</td>
</tr>
<tr>
<td>3.4</td>
<td>Direct Stress Due to Torsion</td>
<td>12</td>
</tr>
<tr>
<td>3.5</td>
<td>Gauge Orientation and Circuitry</td>
<td>14</td>
</tr>
<tr>
<td>3.6</td>
<td>Gauge Response to a C.W. Torque</td>
<td>15</td>
</tr>
<tr>
<td>3.7</td>
<td>Vertical Load Effect on Torque Gauges</td>
<td>18</td>
</tr>
<tr>
<td>3.8</td>
<td>Friction Force on Torque Gauges</td>
<td>21</td>
</tr>
<tr>
<td>3.9</td>
<td>Vertical Load Gauge Orientation and Circuitry</td>
<td>23</td>
</tr>
<tr>
<td>3.10</td>
<td>The Vishay and Its Functions</td>
<td>24</td>
</tr>
<tr>
<td>3.11</td>
<td>Least Squares Error</td>
<td>26</td>
</tr>
<tr>
<td>3.12</td>
<td>Least Squares Example</td>
<td>29</td>
</tr>
<tr>
<td>3.13</td>
<td>Electrical Calibration</td>
<td>31</td>
</tr>
<tr>
<td>3.14</td>
<td>Normal Distribution of Y Data Values</td>
<td>32</td>
</tr>
<tr>
<td>3.15</td>
<td>Error Analysis</td>
<td>35</td>
</tr>
<tr>
<td>4.1</td>
<td>Torque Gauge Locations</td>
<td>36</td>
</tr>
<tr>
<td>4.2</td>
<td>Calibration Configuration</td>
<td>37</td>
</tr>
<tr>
<td>4.3</td>
<td>Initial Torque Calibration</td>
<td>39</td>
</tr>
<tr>
<td>4.4</td>
<td>Brake Application Torque</td>
<td>40</td>
</tr>
<tr>
<td>4.5</td>
<td>Beam Torque</td>
<td>41</td>
</tr>
<tr>
<td>4.6</td>
<td>Sign Convention</td>
<td>43</td>
</tr>
<tr>
<td>4.7</td>
<td>Torque Calibration of ADS1</td>
<td>44</td>
</tr>
<tr>
<td>4.8</td>
<td>Torque Calibration of BPS1</td>
<td>45</td>
</tr>
</tbody>
</table>
LIST OF FIGURES (Cont'd)

<table>
<thead>
<tr>
<th>Fig. #</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.9</td>
<td>Actual Orientation of Calibration Plots</td>
<td>46</td>
</tr>
<tr>
<td>5.1</td>
<td>Drive Axle Gauge Locations</td>
<td>50</td>
</tr>
<tr>
<td>5.2</td>
<td>Datum Level</td>
<td>51</td>
</tr>
<tr>
<td>5.3</td>
<td>Trailer Gauge Locations</td>
<td>52</td>
</tr>
<tr>
<td>5.4</td>
<td>Proposed Vertical Load Circuit</td>
<td>55</td>
</tr>
<tr>
<td>Table #</td>
<td>Title</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>1</td>
<td>Material Properties and Dimensions of the Axle</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>Strain Gauge Specifications</td>
<td>2</td>
</tr>
<tr>
<td>3</td>
<td>Vishay Unit Serial Numbers</td>
<td>3</td>
</tr>
<tr>
<td>4</td>
<td>Torque Calibration Instruments</td>
<td>4</td>
</tr>
<tr>
<td>5</td>
<td>Gauge Sensitivities</td>
<td>38</td>
</tr>
<tr>
<td>6</td>
<td>Bending Effect for Gauges ADS1 and BPS1</td>
<td>42</td>
</tr>
<tr>
<td>7</td>
<td>Range For Electrical Calibration and Gauge Sensitivity of ADS1 and BPS1</td>
<td>47</td>
</tr>
</tbody>
</table>
SUMMARY

This report describes a calibration procedure performed on the strain gauged axles of a tractor-trailer. The calibration was performed to determine the loads imparted to the road during normal vehicle operation. A theoretical discussion of stresses and strains is provided as background to the calibration exercise.

The procedures followed in calibrating the vehicle are outlined. Errors inherent within the system were analysed and considered.

All calibrations produced linear results and good repeatability except those calibrations performed on the drive axles of the tractor-trailer unit. The tractor axle results were less linear but still within acceptable requirements.

The instrumentation used to monitor vertical road loading was found to be sensitive to bias tire loading. This requires that test sites be chosen with level pavement free of ruts.

Further studies are recommended for the drive axle calibration. Also consideration should be given to strain gauging sensitive to shear for vertical load instead of strain gauging sensitive to bending.
CONCLUSIONS

All calibration results showed very little hysteresis.

The trailer axle calibrations for torque and vertical load displayed excellent linearity and good sensitivity when plotted against voltage output. The drive axle gauges exhibited acceptable linearity but lower sensitivity.

It is imperative to maintain constant tire pressure in all tires to avoid bias tire loading which will lead to erroneous results while testing. Rutted roads will result in unequal tire loading also.

Three of the six torque gauges on the trailer were calibrated initially. Each displayed very similar gauge sensitivity. Therefore calibrating two torque gauges and documenting their behaviour was deemed sufficient to predict the torque experienced by all six torque gauges on the trailer.

Electrical calibration is required for all torque gauges because the load wire lengths are not constant and therefore the wire resistance will cause different output voltages.

Brake application hardware induced a torque on the axle of large magnitude. This torque was considered in the data analysis.
RECOMMENDATIONS

The test sites chosen should be free of severe longitudinal road ruts as this may induce bias tire loading.

Each vertical load gauge should be calibrated independently that is, the actual load imparted to the road by one tire set should be plotted against its respective output voltage. The vertical load calibration followed in this report ordered the above output voltage with the total road load imparted via the axle. Independent calibration will serve to better predict axle load sharing.

Experiments should be performed to determine the appropriateness of using shear gauges in assessing vertical load. The purpose would be to investigate if there is a marked decrease in the bias tire loading errors experienced by the bending instrumentation. The voltage output cross talk of bending due to vertical loading and shear stress due to braking torque should be investigated.

The calibration loading procedure followed starts at a chosen datum level and is loaded incrementally. From the maximum, load is decremented by the same amount back to the minimum for the completion of one loading cycle. At least 30 data points should be recorded for calibration. Each increasing and decreasing cycle should contain approximately five increments. This will result in a scatter of points with no identical load values except at the extremes. A best fit line will then be calculated using all the data points.
1. INTRODUCTION

1.1 BACKGROUND

A nationwide weights and dimensions study is underway to determine the loading characteristics of different heavy vehicle suspension types under dynamic conditions. Under the supervision of Mr. J. Woodroofe, at the National Research Council, an investigation of load sharing between axles, axle to road surface impact forces and axle load transfer resulting from normal vehicle braking are being considered. Once complete, the results will be presented to the Road Transportation Association of Canada (RTAC) in an attempt to equalize the heavy vehicle weights and dimensions regulations across Canada.

1.2 TASK

A heavy truck was instrumental to determine the dynamic axle loading characteristics of the vehicle. The task was to calibrate the instrumentation to accurately predict the dynamic vehicle loading.

This report covers the calibration procedures followed, a theoretical discussion describing the best locations of strain gauges to measure load and torque, and an error analysis such that the results are stated with confidence. These discussions are presented for the benefit of those not familiar with the subject.

2. THE INSTRUMENTS USED AND THE MANUFACTURER'S SPECIFICATIONS

2.1 AXLES

There are two round axles types commercially available. One type is designed with a forged hub and pilot shaft. The pilot shaft is inserted into the hollow axle and welded. The second axle type is manufactured using a hub without a pilot shaft. The hub is friction welded to the hollow axle shaft. This eliminates the pilot shaft from the axle tubing thereby providing a uniform cross section which can be strain gauged without the influence of localized stress anomalies associated with an abrupt cross sectional change.

Table 1 lists the material properties of the axle

<table>
<thead>
<tr>
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</tr>
</thead>
<tbody>
<tr>
<td>Modulus of Elasticity (E)</td>
<td>30 x 10^6 psi</td>
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<tr>
<td>Modulus of Rigidity (G)</td>
<td>11.5 x 10^6 psi</td>
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<tr>
<td>O.D.</td>
<td>5&quot;</td>
</tr>
<tr>
<td>I.D.</td>
<td>3.75&quot;</td>
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</tbody>
</table>

Table 1

MATERIAL PROPERTIES AND DIMENSIONS OF THE AXLE
2.2 STRAIN GAUGE TRANSDUCERS

Two strain gauge patterns are used. One type is used to monitor brake torque, and the other type is used to monitor vertical loading. The gauges are manufactured by Micro-Measurements. The manufacturer's specifications are given in Table 2. Figure 2.2 shows the detail of both strain gauge types.

Use: Vertical Load Gauge
Type: CEA-06-250UW-120
Gage Resistance (R_G): 120±0.3% ohms
Gage Factor (K_G): 2.045±0.5%

Use: Torsion Gauge
Type: EA-06-125TH-120
Gage Resistance (R_G): 120±0.2% ohms
Gage Factor (K_G): 2.01±0.5%

TABLE 2
STRAIN GAUGE SPECIFICATIONS
2.3 STRAIN GAUGE CONDITIONING AMPLIFIER

The conditioning amplifier (Vishay units) are used to condition the input voltage needed to activate the strain gauges. It also amplifies and filters the output signal. These units are supplied by Intertechnology Limited. Table 3 contains the serial numbers of the units used during the calibration procedure. All units are Model 2310.

<table>
<thead>
<tr>
<th>024155</th>
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</tr>
</thead>
<tbody>
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<td>045290</td>
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</tr>
</tbody>
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TABLE 3
VISHAY UNIT SERIAL NUMBERS
Before the calibration a bench test was performed on the Vishay units to check their outputs and stability. The procedure followed and results are documented in Appendix A. From the test it was concluded that each signal conditioned by the Vishay is within \( \pm 1\% \).

2.4 INSTRUMENTS USED FOR TORQUE CALIBRATION

Table 4 lists the use and specifications of all the instruments used for calibrating for torque.

<table>
<thead>
<tr>
<th>Manufacturer</th>
<th>Instrument Type</th>
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<th>Serial No.</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intertechnology</td>
<td>Load Cell</td>
<td>EP2-100K-10P3</td>
<td>30393</td>
<td></td>
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<td>Omega</td>
<td>Transducer indicator</td>
<td>DP420</td>
<td>420126</td>
<td>( \pm 5 \text{ lbs} )</td>
</tr>
<tr>
<td>Owatonna Tool</td>
<td>55 ton hydraulic jack</td>
<td>RA556</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Owatonna Tool</td>
<td>hydraulic hand pump</td>
<td>3J</td>
<td>135510</td>
<td></td>
</tr>
</tbody>
</table>

TABLE 4
TORQUE CALIBRATION INSTRUMENTS

2.5 INSTRUMENTS USED FOR VERTICAL LOAD CALIBRATION

The only instrument used to calibrate the vertical load, excluding the Vishay units, was a Fairbanks Morse Balance Scale, serial #E17049. Its rated capacity is 250000 lbs and has an accuracy of \( \pm 25 \text{ lbs} \).

3. THEORETICAL BACKGROUND

During vehicle operation the axle is subjected to forces. These forces cause the axle to deform resulting in strain. Knowing the strain, which is easily measured using strategically placed strain gauges, the forces can be calculated.

An analysis of the stress strain behaviour due to torque and vertical loading are discussed in this section.
3.1 TORSIONAL STRESS

Torque caused by braking induces a shear stress on the axle. Torsional shear stress of a uniform cross section can be calculated knowing the following relationship:

\[ \tau = \frac{Tc}{J} = \frac{\text{FORCE}}{\text{AREA}} \]  \hspace{1cm} [3.11]

where:
- \( \tau \) = magnitude and direction of the applied torque (units - ft*lbs)*
- \( c \) = distance from the center of the member (ft)
- \( J \) = polar moment of inertia (ft^4)

The polar moment of inertia for the axle, a hollow tube, is

\[ J = \frac{1}{2} \pi (c_2^4 - c_1^4) \]

where:
- \( c_2 \) = outer axle radius
- \( c_1 \) = inner axle radius

In reference to Fig. 3.1 and recalling one of the fundamental laws of nature, everything must maintain equilibrium, the behaviour of torsional shear stress on a small element will be discussed. The small element represents any point on the shaft, it is enlarged for clarity.

Logic dictates that twisting the member causes a stress in the positive y direction on the A face of the element. Multiplying this stress by the area of face A yields a force \( F_1 \)

\[ F_1 = \frac{\text{FORCE}}{\text{AREA}} \cdot \text{AREA} = \text{FORCE} \] in the direction of the shear stress (see Fig. 3.1a). By equilibrium; the sum of the forces in the y direction must equal zero. Therefore an equal but opposite force must be acting on the B face of the element. Since the member has a uniform cross section, the area of the A and B faces are equal, the result is \( \tau_1 = \tau_2 \) (see Fig. 3.1b).

The element is not yet in equilibrium. With just the forces, or stresses \( \tau_1 \) and \( \tau_2 \) the element would rotate in a counterclockwise direction. To maintain equilibrium the element must be experiencing equal but opposite forces to stop the rotational tendency (see Fig. 3.1c) Figure 3.1d shows the stresses experienced by any element in equilibrium on the shaft subjected to a clockwise torque.

Figure 3.1e shows how this torsional shear stress flows through any cross sectional portion of the axle. Figure 3.1f describes the relationship between the magnitude of shear stress and \( c \) the distance from the shaft's center.

\(^*\text{ft*lbs were chosen as the units for torque because it was felt that people have a better physical feel for 1 ft*lb versus 1 Newton meter the SI equivalent.}\)
FIGURE 3.1
TORSIONAL SHEAR STRESS EFFECT
3.2 VERTICAL LOADING STRESSES

The axle experiences vertical loading from tire contact with the road surface and the trailer weight at the points of suspension spring attachment. Vertical forces are termed shear forces.

Two types of stresses are associated with shear forces;
1) direct stress due to bending and,
2) shear stress

Shear stress is calculated in the following manner:

\[ \tau = \frac{F_0}{I} \frac{Q}{t} = \frac{\text{FORCE}}{\text{AREA}} \]  \[3.21\]

where
- \( F \) = magnitude and direction of the shear force (lbs)
- \( Q \) = first moment of area (ft³)
- \( I \) = moment of inertia (ft⁴)
- \( t \) = material thickness (ft)

Figure 3.2a shows a small element on a member experiencing a shear stress*. Figure 3.2b describes how this stress flows through any cross section of the axle. Notice along the vertical axis, through the center of the member, the shear stress is zero.

Direct stress due to bending is calculated by:

\[ \sigma_{\text{BEND}} = \frac{F(x)y}{I} \frac{\text{FORCE}}{\text{AREA}} \]  \[3.21b\]

where
- \( F \) = direction and magnitude of the shear force (lbs)
- \( x \) = distance from the force to the element in question along the axle’s horizontal axis (ft)
- \( y \) = distance along the vertical axis from the center of the axle or its neutral axis (ft)
- \( I \) = moment of inertia (ft⁴)

The moment of inertia is given by

\[ I = \frac{1}{4} \pi (c_2^4 - c_1^4) \]

where \( c_1 \) and \( c_2 \) = inner and outer axle radius

Figure 3.3a graphically describes the stress distribution induced by bending. The important characteristics of this distribution are:

*The same equilibrium argument can be used to reach the shear stress directions on the element as in the previous section.
FIGURE 3.2
VERTICAL LOAD SHEAR STRESS

i) the shaft experiences zero stress anywhere along its neutral axis

ii) any portion of the material above the neutral axis experiences tension

iii) any portion of the material below the neutral axis experiences compression

iv) if the applied moment (Fx) is in the opposite direction the top half would be in compression, the bottom half would be in tension, and zero stress is along the neutral axis.

Figure 3.3b depicts the equilibrium configuration of a small element under direct stress.
3.3 STRESS STRAIN RELATIONSHIPS

When a material is subjected to a direct stress it behaves either elastically or plastically depending upon the magnitude of the stress and the material properties. In the plastic range the material suffers permanent deformation, in contrast, the material returns to its original length when the stress inducing force is removed in the elastic range. The change in length divided by the true length of the material in the elastic region is termed strain:

\[ \varepsilon = \frac{\Delta L}{L} \]  [3.31]

where \( \varepsilon \) = denotes strain
Stress is proportional to strain when no plastic deformation occurs. The proportionality constant is called the modulus of elasticity and is a material property. It is denoted by a capital $E$.

$$\varepsilon = E \varepsilon$$ [3.31i]

A material subjected to a stress along the $x$ axis, for example, experiences strains in all three dimensions but for our purpose these relationships are unnecessary. We need only to consider the strain in the $x$ direction, therefore:

$$\sigma_x = E \varepsilon_x$$ [3.31ii]

Shear stress also induces a strain, called a shear strain. These two physical properties are also proportional in the elastic range of the material. They are related by the modulus of rigidity ($G$) also a material property.

$$\tau = G \gamma$$ [3.31iv]

3.4 THE STRAIN GAUGE TRANSUDER (From Ref. #4 p. 729)

A bonded resistance strain gauge is an electrically excited device used to measure strain. It is made of a grid of fine resistance wire bonded to a thin paper backing. When the gauge is cemented to a member under test, any deformation of the member results in a change in dimension and therefore a change in resistance of the gauge if the gauge's effective axis is oriented along an axis of either compression or tension. The relation between strain and resistance change is:

$$\frac{\Delta R}{R_G} = K_G \varepsilon = K_G \frac{\Delta L}{L}$$ [3.41]

where

- $\Delta R$ = gauge resistance change
- $R_G$ = unstrained gauge resistance
- $K_G$ = the gauge factor
- $\varepsilon$ = the strain seen by the gauge

3.5 STRAIN GAUGE ORIENTATION FOR TORSION

When a small element in equilibrium, drawn parallel to the shaft's neutral axis, is subjected to pure torsion it does not experience direct stresses. Therefore, to instrument the axle with strain gauges it is necessary to find the orientation of an element which will sustain direct stress.

Figure 3.4a depicts an element under the influence of shear stresses due to a clockwise torque. Recalling from section 3.1, all these shear stresses are of equal magnitude and act over face areas also of equal magnitude. Cutting this element at any angle, $\theta$, to the AD face, starting at A, results in the triangular element ADE. This
element must also maintain equilibrium, thus, the force $F$ is added acting directly on the AE face in a compressive sense as shown in Figure 3.4b. The AE face is now under the influence of a direct stress. (Recall \( \frac{\text{FORCE}}{\text{AREA}} = \text{STRESS} \)).

By equilibrium, the sum of the forces in the $x$ and $y$ directions must equal zero. All forces acting to the left are to be considered positive $x$ and all forces acting up are to be considered positive $y$ therefore:

\[
\begin{align*}
\sum F_x &= 0 \\
\tau_{ox}A_0 - F\cos\theta &= 0 \\
\tau_{ox}A_0 &= F\cos\theta \quad [3.5i] \\
\sum F_y &= 0 \\
\tau_{oy}A_0 &= F\sin\theta \quad [3.5ii]
\end{align*}
\]

Since the magnitude of the all forces are equal, dividing equation [3.5ii] by equation [3.5i] yields,

\[
\frac{\tau_{ox}A_0}{\tau_{oy}A_0} = \frac{\tau_{oy}A_0}{\tau_{ox}A_0} = \frac{F\sin\theta}{F\cos\theta}
\]

therefore,

\[
\tan\theta = 1 \\
\theta = \arctan (1) \\
\theta = 45^\circ \text{ (for this particular case)}
\]

Therefore an element oriented at $45^\circ$ to the neutral axis experiences a compressive direct stress while under the influence of a clockwise torque.

By the same argument it can be shown that a cut at $135^\circ$ to the AD face results in a tensile direct stress seen by the newly formed face (see Fig. 3.4c).
3.5.1 Transformation of Strain

Thus far, we have only discussed methods of calculating strain due to direct stress and shear stress of elements oriented parallel to the neutral axis of the member (along the x axis). It is possible to compute strains at any angle to the x axis by the following formula,

\[ \varepsilon_0 = \varepsilon_x \cos^2 \theta + \gamma \sin \theta \cos \theta \]  \[3.5111\]

where:

\[ \varepsilon_x = \frac{\sigma_x}{E} \]  [from eq'n 3.3111]

\[ \gamma = \frac{T}{G} \]  [from eq'n 3.3iv]

FIGURE 3.4
DIRECT STRESS DUE TO TORSION
Since we are concerned with strains at 45° and 135° to the x axis, equation [3.5iii] becomes:

for 45°:

\[ \varepsilon_{45°} = \frac{\sigma_x}{E} \cos^2 45° + \frac{\tau}{G} \sin 45° \cos 45° \]

\[ = \frac{\sigma_x}{E} \times \frac{1}{2} + \frac{\tau}{G} \times \frac{1}{2} \]

therefore;

\[ \varepsilon_{45°} = \frac{1}{2} \times \left( \frac{\sigma_x}{E} + \frac{\tau}{G} \right) \] [3.5iva]

for 135°

\[ \varepsilon_{135°} = \frac{1}{2} \times \left( \frac{\sigma_x}{E} - \frac{\tau}{G} \right) \] [3.5ivb]

### 3.5.2 Gauge Positioning on the Axle and the Wheatstone Bridge

Four strain gauges are used to monitor torsion, two on each side of the axle. They are placed at identical points on each side of the axle over the neutral axis plane. Figure 3.5a shows their angle of orientation and location on the axle.

The four gauges are wired into a wheatstone bridge circuit. This circuit requires an excitation voltage \( V^+ \) to activate the strain gauges. When the member is subjected to a torque the strain gauges deform resulting in a voltage output \( V_o \) due to the resistance change in the gauges. This voltage output across the nodes a and b can be calculated using the voltage divider concept (see Fig. 3.5b).

\[ V_o = V^+ \left( \frac{R_1}{R_4 + R_1} - \frac{R_2}{R_2 + R_3} \right) \times AF \] [3.5v]

where

- \( V_o \) = output voltage
- \( V^+ \) = excitation voltage
- \( R_{1,2,3,4} \) = strained resistance of the gauges
- \( AF \) = amplification factor

Equation [3.5v] can be reduced further if we consider the following implicit example.
A shaft experiences a torque of $x\text{ ft-lbs}$ applied in a clockwise direction. Describe the strain gauge behaviour.

Figure 3.6 shows the two equilibrium elements containing the strain gauges and the strain gauge orientation within these elements. Before we proceed it is important to note the following properties;

i) All shear stresses are of equal magnitude. The distance of these elements from the center of the shaft, are equal (refer to Figure 3.1f, p. 6).

ii) The two elements are identical when comparing the directions of the effective shears.

iii) The elements experience no direct stress.
Calculating the strain in gauge 1 using \([3.5ivb]\):
\[
\sigma_x = 0
\]
\[
\varepsilon_1 = -\frac{1}{2} \frac{\tau}{G}
\]
since the shear stress is constant on all elements and \(G\) (the modulus of rigidity) is also constant for all elements, let;
\[
\varepsilon_1 = -y \frac{\Delta I}{I} = -\varepsilon \text{ µ strains}^*
\]
The strain in gauge 4 is; using \([3.5iva]\):
\[
\sigma_x = 0
\]
\[
\varepsilon_4 = +\frac{1}{2} \frac{\tau}{G} = +\varepsilon \text{ µ strains}
\]

Using the appropriate formula, depending on the angle of orientation, the strains in gauges 2 and 3 can be calculated.

Summarizing,
\[
\varepsilon_1 = -\varepsilon \text{ µ strains}
\]
\[
\varepsilon_2 = +\varepsilon \text{ µ strains}
\]
\[
\varepsilon_3 = -\varepsilon \text{ µ strains}
\]
\[
\varepsilon_4 = +\varepsilon \text{ µ strains}
\]

*µ strains represents \(10^{-6} \frac{\Delta I}{I}\), the common units of strain.
from,

\[
\frac{\Delta R}{R_G} = k_G \zeta \tag{3.41}
\]

the change in gauge resistance can be calculated.

Noting that,

1) all strain gauges are identical, therefore, the gauge resistance \(R_G\) and gauge factors \(k_G\) are constant

2) the magnitudes of the strains in each of the gauges are equal

therefore for all gauges;

\[
\Delta R = +\tau \text{ (for positive strain)}
\]

\[
\Delta R = -\tau \text{ (for negative strain)}
\]

For a clockwise torque, the voltage output is, from equation [3.5v]:

\[
V_o = V^+ \left[ \frac{R_G - \tau}{(R_G + \tau) - R_G - \tau} - \frac{R_G + \tau}{(R_G + \tau) + R_G - \tau} \right] \ast AF
\]

As shown in Figure 3.5b on page 14 the lead wires connecting the instruments which provide the excitation voltage and monitor the voltage output from the Wheatstone bridge are quite long and their resistance must be considered for accurate calculation of the output voltage.

The lead wire used is hard drawn, solid copper wire, AWG size 22. Its resistance per foot is 0.016 ohms/ft. The strain gauges are connected in series with these wires (see Fig. 3.5b, p. 14). The corrected gauge resistance is:

\[
R_G^* = R_G + 2R_L \tag{3.5vi}
\]

where

\(R_G^*\) = gauge resistance

\(R_L\) = lead wire resistance (0.016 \* ft of wire)

Therefore the correct voltage output formula for a clockwise torque is:

\[
V_o = V^+ \left[ \frac{R_G^* - \tau}{R_G^* - \tau + R_G^* + \tau} - \frac{R_G^* + \tau}{R_G^* - \tau + R_G^* + \tau} \right] \ast AF
\]
Because the magnitude of the strain is in the order of $10^{-6}$, $\Delta R$ will be small. This is why an amplification factor (AF) is necessary.

To further simplify the output voltage formula the amplification factor and excitation voltage are always constant in our applications:

$$AF = 1000$$

$$V^+ = 5V$$

If the torque was applied in the opposite direction of the same magnitude the shear stress would be of the same magnitude but in the opposite sense thus reversing the sense of the strain seen by each gauge. Therefore equation [3.5v] reduces to:

$$V_o = 5000 \left[ \frac{R_G^* \tau_r}{R_G^* \tau_r + R_G \tau_r} - \frac{R_G^* \tau}{R_G^* \tau_r + R_G \tau_r} \right]$$

$$V_o = 5000 \frac{R_G^* \tau_r}{2R_G} - \frac{R_G^* \tau}{2R_G}$$

$$V_o = \frac{2500}{R_G} \left[ R_G^* \tau_r - R_G^* \tau \right]$$

$$V_o = \frac{2500}{R_G} (\tau r)$$

Therefore, for pure torsion the voltage output is,

$$V_o = \frac{5000}{R_G} (\tau r)$$

[3.5vi1]

where

- $V_o$ = output voltage
- $R_G^*$ = gauge resistance [from eq'n 3.5vi]
- $\tau r$ = calculated resistance change due to strain [from eq'n 3.4i]

The gauge resistance in equation [3.4i] is not corrected due to lead wire resistance because the change in resistance depends on the strain experienced by the gauge as a ratio to the gauge's unstrained resistance.

3.5.3 Vertical Loading Effect on the Torsion Strain Gauges

During vehicle operation the axle will experience both torsional and vertical forces. Therefore, it is necessary to determine the effect of shear stress and direct stress due to bending on the torsion strain gauges.
As discussed in the preceding section the torsion strain gauges are placed at identical points on each side of the axle straddling the plane containing the neutral axis (see Fig. 3.5a, p. 14). Referring to Figure 3.3a on page 9, the direct stress due to bending is zero at the neutral axis. The gauges are larger than a point, but, if they are placed over the center of the neutral axis at identical points on the axle the net effect of the bending stress will be zero. Thus the voltage output of the wheatstone bridge circuit is not affected by the direct stress due to bending if the gauges are accurately placed (see Fig. 3.7a).

Figure 3.7b describes the effect on the elements A and B, containing strain gauges 1, 4 and 2, 3 respectively, due to a shear stress caused by vertical loading. All stresses are of equal magnitude, but, element A is influenced by a stress opposite to that of element B. Element A will be considered as the positive sense.

\[ \text{NET BENDING STRESS} = 0 \]

**ELEMENT A**

**ELEMENT B**

**FIGURE 3.7**

**VERTICAL LOAD EFFECT ON TORQUE GAUGES**

The net effect of the bending stress is zero therefore, gauges 1 and 4 experiencing a positive shear stress of \( \tau \) see strains of:
\[ \varepsilon_1 = -y \mu \text{ strains} \quad \text{[from eq'n 3.5ivb]} \]
\[ \varepsilon_4 = +y \mu \text{ strains} \quad \text{[from eq'n 3.5iva]} \]

Gauges 2 and 3 experience negative shear stress of \(-\tau\) and see strains of:

\[ \varepsilon_2 = -y \mu \text{ strains} \quad \text{[from eq'n 3.5iva]} \]
\[ \varepsilon_3 = +y \mu \text{ strains} \quad \text{[from eq'n 3.5ivb]} \]

As discussed earlier, the resistance change in each gauge will be of the same magnitude and the sign will be determined by the direction of the strain, therefore;

\[ R_1 = R_2 = R_G - \tau \]
\[ R_3 = R_4 = R_G + \tau \]

From equation [3.5v]

\[ V_o = V^+ \left[ \frac{R_G - \tau}{R_G - \tau + R_G + \tau} - \frac{R_G - \tau}{R_G - \tau + R_G + \tau} \right] \ast AF \]
\[ V_o = V^+ \left[ \frac{R_G - \tau - R_G + \tau}{2R_G} \right] \ast AF \]

Therefore;

\[ V_o = 0 \]

Therefore, equation [3.5vii] remains unchanged. It is not sensitive to stresses induced by vertical loading.

3.5.4 *Brake Force Effect on the Torsion Strain Gauges*

The force of braking at the tire-to-ground contact point generates a torque in the axle known as brake torque.

This force induces a shear stress on the axle as shown in Figure 3.8a. The gauges are located at the point of zero shear stress, thus, they will not be strained if they are placed accurately.

Also of concern is the bending effect caused by the braking force (see Fig. 3.8b). Gauges 1 and 4 are subjected to tension under the force while gauges 2 and 3 are compressed. Using equations [3.5iva and b] and [3.4i];
\[ R_1 = R_4 = R_G^* + r \]
\[ R_2 = R_3 = R_G^* - r \]

Substituting the above values into equation [3.5v] yields:
\[ V_o = V^* \left[ \frac{R_G^* + r}{2(R_G^* + r)} - \frac{R_G^* - r}{2(R_G^* - r)} \right] * A.F. \]
\[ V_o = V^* \left[ \frac{1}{2} - \frac{1}{2} \right] * A.F. \]
\[ V_o = 0 \]

No correction is necessary for the braking force.

3.6 STRAIN GAUGE ORIENTATION FOR VERTICAL LOADING

Strain gauges respond to directional strain. The stress distribution due to bending is directed perpendicular to the plane of the neutral axis. Recalling that the bending stress distribution is a maximum at the outer surface of the material (see Figure 3.3a, page 9) the logical choice is to place the strain gauges at the top and bottom of the axles oriented as shown in Figure 3.9.

Using the transformation of strain formula [3.5iii] it will be proven that any type of shear stress does not affect a gauge oriented as in Figure 3.9a on page 23.

**Gauge orientation:** 0° to the x axis
from [3.3iv]
\[ \gamma = \tau / G \]

from [3.5iii]
\[ \varepsilon_o = \varepsilon_x \cos 20^\circ + \gamma \sin 0^\circ \cos 0^\circ \]
\[ \varepsilon_o = \varepsilon_x \]

Bending caused by the brake force will not affect the gauges if they are oriented correctly. This stress distribution is zero at the top and bottom of the axle.

Therefore the gauges experience the effect of bending only.

Recalling equations [3.2ii] and [3.3ii]
\[ \sigma_x = \frac{F}{I} x^2 \frac{y}{I} \]  \[\text{[3.2ii]}\]
\[ \varepsilon_x = \frac{\sigma_x}{E} \]  \[\text{[3.3ii]}\]
FIGURE 3.8
FRICION FORCE ON TORQUE GAUGES
3.6.1 Gauge Positioning on the Axle and the Wheatstone Bridge

Figure 3.9a depicts the gauge positioning used to monitor vertical loading. Figure 3.9b shows the wheatstone bridge circuit and the lead wire resistance.

Assuming the strain gauges are aligned without error along the axis of direct stress and are situated over each other they will see the same magnitude of strain but opposite senses. That is, one will be in compression and the other in tension depending on the direction of the applied moment, therefore from equation [3.41]

\[ \Delta R = \frac{\pi R}{r} \]

the output voltage expression for this bridge is;

\[ V_o = V^* \left[ \frac{1}{R_G R + r + 120} - \frac{120}{R_G R + r + 120} \right] \times A.P. \]

since the excitation voltage and amplification factor are constants;

\[ V_o = 5000 \left[ \frac{1}{R_G R + r + 120} - \frac{120}{R_G R + r + 120} \right] \]

\[ V_o = 6.0 \times 10^5 \left[ \frac{1}{R_G R + r + 120} - \frac{1}{R_G R + r + 120} \right] \quad [3.61] \]

where \( R_G^* \) is given by eq'n [3.5vi].

3.7 THE STRAIN GAUGE CONDITIONING AMPLIFIER

Both vertical load and torsion bridge circuits require amplification and an excitation voltage. These two functions along with three others are provided by the strain gauge conditioning amplifiers, also known as a Vishay unit. This section will summarize the functions provided by the Vishay.

1) It supplies the excitation voltage necessary for the gauges within the wheatstone bridge circuit.

2) It measures the differential voltage across the bridge.

3) Once a datum level for testing is chosen the Vishay can adjust the output voltage to zero volts. This is known as "balancing the bridge".

4) Amplification factor, or gain, is supplied by the Vishay unit. This function amplifies the bridge output voltage such that its signal is of sufficient magnitude to be recorded.
FIGURE 3.9
VERTICAL LOAD GAUGE ORIENTATION AND CIRCUITRY
FIGURE 3.10
THE VISHAY AND ITS FUNCTIONS
3.8 CALIBRATION

To calibrate the gauges a known torque or vertical load is applied to the axle. Due to the axle's elastic behaviour it is strained and an output voltage is emitted from the bridge circuit. The load* and output voltage is recorded.

A linear relationship is predicted for load and output voltage by the following reasoning: consider the torque instrumentation on one axle from [3.21];

$$\tau = \frac{F}{J}$$

the only variable in this equation is F, the applied torque all others are constant, therefore;

$$\tau = T$$

As torque increases so does the shear stress. This is a linear relationship. From [3.3.1v]:

$$\tau = \gamma$$

this is also a linear relation because the induced torque, and therefore, the induced shear stress does not exceed the elastic limit of the material. From [3.5iii] recall that $\alpha_x = 0$, therefore $\varepsilon_x = 0$

$$\varepsilon_{45^\circ}, 135^\circ = \gamma$$

and from [3.41]

$$\Delta R = \tau$$

the change in gauge resistance behaves linearly with strain. From [3.5vii]:

$$V_o = \tau (= \Delta R)$$

Therefore, the torque induced on the axle is linearly proportional to the bridge output voltage.

*the word load implies both torque and vertical load.
LOAD = V_0

A similar argument can be made of the relationship between vertical load and output voltage leading to the same conclusion.

The recorded data is plotted with the bridge output on the ordinate (y) axis and the abscissa (x axis) represents the load configuration. This convention was adopted because the output voltage depends on the induced load. A straight line is fitted to the data of the form;

\[ y = Mx + C \]

where \( M = \text{slope of the line} = \frac{\text{RISE}}{\text{RUN}} \)
\[ C = \text{y intercept}. \]

3.8.1 Least Squares Method of Best Fit

Because of errors within the system, all data points do not fall exactly on a straight line as predicted. It is necessary to calculate the best fit line using the least squares method.

Consider the following data points

\[
\begin{array}{c|c|c|c|c|c|c|c}
X & X_1 & X_2 & \cdots & X_i & \cdots & X_n \\
Y & Y_1 & Y_2 & \cdots & Y_i & \cdots & Y_n
\end{array}
\]

The data is anticipated to fall on a straight line

\[ y' = Mx + C \]

It is desired to find values of \( M \) and \( C \) which will best predict the true values of the recorded \( y \) data points. The \( i \)th value of \( y \) is given by:

\[ y'_i = Mx'_i + C \]

It is desired to minimize the total error incurred without having positive error cancelling with negative error.

\[
\text{BEST FIT LINE} \\
Y' = MX + C
\]

ERROR = \( Y_i - Y'_i \)

\[
\text{FIGURE 3.11} \]

LEAST SQUARES ERROR
To avoid the above cancellation it is required to minimize the sum of the squares of the error. Therefore,

$$\sum_{i=1}^{n} [y_i - (MX_i + C)]^2 = S$$

M and C must be determined such that S is a minimum.

Taking the partial derivative of S with respect to C and M respectively and letting the result equal zero will minimize S,

$$\frac{\partial S}{\partial C} = 0$$

therefore;

$$2 \sum_{i=1}^{n} (y_i - MX_i - C) [-1] = 0$$

$$\frac{\partial S}{\partial M} = 0$$

therefore;

$$2 \sum_{i=1}^{n} (y_i - MX_i - C) [-x_i] = 0$$

[3.81a]

Dividing equation [3.81a] by 2, therefore;

$$\sum y_i + C \sum (1) + M \sum x_i = 0$$

$$\sum y_i = Cn + M \sum x_i$$

dividing equation [3.81b] by 2,

$$\sum x_i y_i = C \sum x_i + M \sum x_i^2$$

Summarizing;

$$\sum y_i = Cn + M \sum x_i$$

[3.81ia]

$$\sum x_i y_i = C \sum x_i + M \sum x_i^2$$

[3.81ib]

where

- \( n \) = number of data points
- \( \sum x_i \) = sum of all x data points
- \( \sum x_i^2 \) = sum of all squared x data points
- \( \sum x_i y_i \) = sum of the product of corresponding pairs of data points
- \( \sum y_i \) = sum of all y data points

+ all summation signs have bounds from 1 to n data points.
The unknowns are M and C. They can be calculated from equations [3.8ia] and [3.8ib]. A demonstration of the least squares method is given here.

The following data points were recorded during a calibration:

\[
\begin{array}{c|c|c|c|c}
X & 1 & 2 & 3 & 4 \\
Y & 1.1 & 2.9 & 3.8 & 4.3 \\
\end{array}
\]

From this data it is acceptable to assume the best fit line has the form,

\[y = MX + C\]

using equations [3.8ia] and [3.8ib]:

\[n = 4\]

\[
\begin{align*}
\sum x_i &= 1 + 2 + 3 + 4 = 10 \\
\sum x_i^2 &= 1^2 + 2^2 + 3^2 + 4^2 = 30 \\
\sum x_iy_i &= 1(1.1) + 2(2.9) + 3(3.8) + 4(4.3) = 30.5 \\
\sum y_i &= 1.1 + 2.9 + 3.8 + 4.3 = 10.1
\end{align*}
\]

substituting the above values in the appropriate equations of [3.8ia] and [3.8ib]:

\[10.1 = 4C + 10M \quad [1]
\]

\[30.5 = 10C + 30M \quad [2]
\]

From [1]

\[C = \frac{10.1 - 10M}{4}\]

substituting into [2]

\[30.5 = 10 \left(\frac{10.1 - 10M}{4}\right) + 30M\]

\[M = 1.05\]

therefore from [1]

\[C = -0.1\]

\[\therefore y = 1.05 \times -0.1\]
3.8.2 Gauge Sensitivity

The gauge sensitivity describes the slope of the best fit line, $y = MX + C$ in physical units.

The general expression for the gauge sensitivity is:

$$\frac{y}{z \text{ volt excitation}} \equiv x \text{ load units} \quad [3.81a]$$

where

- $y =$ bridge voltage output in mV
- $x =$ corresponding load value
- $z =$ excitation voltage value
Gauge sensitivity is calculated in the following manner given the best fit line:

\[ Y = MX + C \]

where \( y = \) volts
\( M = \frac{\text{VOLTS}}{\text{LOAD}} \)
\( X = \text{load} \)
\( C = \text{volts} \)

let the output voltage be 1 volt; disregard the value of \( C \);

\[ l = Mx \]
\[ x(\text{Load}) = \frac{1}{M} \frac{(\text{VOLT})}{\text{LOAD}} \]

[3.8ivb]

therefore using [3.8iva] and [3.8ivb]

\[ 1 \ \text{VOLT} = \frac{1}{M} \ \text{LOAD UNITS} \]

The gain (\( G \)) and excitation voltage (\( Z V_T^* \)) affect the bridge output voltage; dividing the voltage output by these factors yields the gauge sensitivity.

\[ \frac{1}{C * ZV_T^*} = \frac{1}{M} \ \text{LOAD UNITS} \]

[3.8v]

Given the equation of a best fit line calculated for a torque calibration with a gain of 1000 and an excitation voltage of 5 volts, we will determine the gauge sensitivity

\[ y = 0.238x + 0.18 \]

let \( y = 1 \ \text{VOLT} \) and disregard the value of \( C \);

\[ l = 0.238x \]
\[ x = \frac{1}{0.238} = 4.202 \ \text{ft} * \text{lbs} \]

[from 3.8ivb]

therefore;

\[ 1 \ \text{VOLT} = 4.202 \ \text{ft} * \text{lbs} \]

dividing by the gain and excitation voltage yields the gauge sensitivity

\[ \frac{1 \ \text{VOLT}}{1000 * 5 \ V_T^*} = 4.202 \ \text{ft} * \text{lbs} \]

in its standard form the gauge sensitivity is;
3.8.3 Electrical Calibration

Electrical calibration is performed when the wheatstone bridge is balanced ($V_0 = 0$). At this point engaging a switch on the Vishay unit shunts a resistor across one arm of the bridge network (see Fig. 3.10, p. 24). This results in a voltage output. The voltage output will always be of the same value to within ±1%. The error incurred is due to the Vishay.

Once a gauge is calibrated and its best fit line is calculated, the gauge behaviour is therefore established. The electrical calibration voltage is substituted into the line equation to give a resulting load value. Consider the following implicit example:

1) gauge behaviour

$$y = MX + C$$

2) electrical calibration result;

$$y = V_0$$ (output volts)

3) resulting load

$$X_{ECAL} = \frac{V_0 - C}{M}$$

![Diagram of electrical calibration](image)

**FIGURE 3.13**

ELECTRICAL CALIBRATION
During an actual test a chart recorder is used to monitor a specific load gauge. The Wheatstone bridge is balanced and electrical calibration is performed resulting in a deflection of, for example, ten divisions on the chart paper. The gauge behaviour is known to be linear (from the line \( y = Mx + C \)) and the voltage output caused by the electrical calibration is known, therefore, the corresponding load can be calculated. On the chart recorder 10 divisions is therefore equivalent to the calculated load due to electrical calibration. The result is the magnitude of all loads can be determined from the recorded data. Electrical calibration is the most important function of calibration.

3.9 ERROR ANALYSIS

3.9.1 Error in Least Squares Calculation (From Ref. #2 p. 296)

The least squares method of curve fitting is only an estimator to the true line. This true line can be found if an infinite number of data points were collected resulting in the following equation.

\[
y_i = \alpha + \beta x_i
\]

The values of \( y_i \) recorded are independently normally distributed about the true line \( \alpha + \beta x_i \) representing the mean value of this distribution. Each \( i \)th value of \( y \) has this distribution characteristic and all of these distributions have the same shape, or, common variance (see Fig. 3.14).

![Normal Distribution of Y Data Values](image)

**FIGURE 3.14**

NORMAL DISTRIBUTION OF Y DATA VALUES
The method of least squares is used only to estimate the true line, that is, \( C \) and \( M \) are just estimates of the true values \( a \) and \( \beta \) respectively.

To evaluate the range of the true line coefficients the following expression will be used. Their development is beyond the scope of this report.

\[
S_{xx} = n \sum_{i=1}^{P} x_i^2 - (\bar{x})^2 \quad \quad \quad \text{[3.91a]}
\]
\[
S_{yy} = n \sum_{i=1}^{P} y_i^2 - (\bar{y})^2 \quad \quad \quad \text{[3.91b]}
\]
\[
S_{xy} = n \sum_{i=1}^{P} x_i y_i - (\bar{x})(\bar{y}) \quad \quad \quad \text{[3.91c]}
\]
\[
S_e = \frac{S_{xx}S_{yy} - (S_{xy})^2}{n(n-2) S_{xx}} \quad \quad \quad \text{[3.91d]}
\]

Using the Student's \( t \) distribution we can construct a confidence interval for \( a \) and \( \beta \).

For \( a \):
\[
C - \frac{t_{\alpha/2}}{n-2} \cdot S_e \left( \frac{S_{xx} + (\bar{x})^2}{n} \right)^{1/2} < a < C + \frac{t_{\alpha/2}}{n-2} \cdot S_e \left( \frac{S_{xx} + (\bar{x})^2}{n} \right)^{1/2} \quad \text{[3.91a]}
\]

For \( \beta \):
\[
M - \frac{t_{\alpha/2}}{n-2} \cdot S_e \left( \frac{n}{S_{xx}} \right)^{1/2} < \beta < M + \frac{t_{\alpha/2}}{n-2} \cdot S_e \left( \frac{n}{S_{xx}} \right)^{1/2} \quad \text{[3.91b]}
\]

The confidence level used throughout this report will be 95%. Therefore;
\[
n = 1 - 0.95 = 0.05
\]
\[
\frac{n}{2} = 0.025
\]

3.9.2 Method of Application

From the Vishay units listed in Appendix A it was determined that each output voltage recorded is in error to \( \pm 1\% \), thus, the assumption that \( y \) is normally distributed with common variance about the true line is reasonable and the \( M \) and \( C \) estimator error method is applicable. But \( x \), the load configuration, is also in error due to instrument accuracy.

*See Appendix B for discussion of confidence intervals and the Student's \( t \) distribution.
The method of incorporating these two sources of error is outlined in a step by step procedure below. The data is recorded in the following manner:

\( x_1 = \) represents the measured load  
\( y_1 = \) represents the corresponding voltage output  

1) from the measured load, add the load configuration error. For example:

\[ x_1 + 10 \text{ lbs} \]

2) calculate the best fit line through this new data set*:

\[ x_1 + 10 \text{ lbs}, y_1 \]

3) apply the least squares error calculation therefore;

\[ C_1 < \alpha < C_2 \quad [3.911a] \]

\[ M_1 < \beta < M_2 \quad [3.911b] \]

4) determine the lowest extreme line from the above. In this case;

\[ y_{\text{LOW}} = M_1x + C_1 \]

5) repeat steps 1 through 3 subtracting the load configuration error \((x_1 - 10 \text{ lbs})\). This yields;

\[ C_3 < \alpha < C_4 \quad [3.911a] \]

\[ M_3 < \beta < M_4 \quad [3.911b] \]

6) determine the highest extreme line

\[ y_{\text{HIGH}} = M_2x + C_4 \]

7) from the electrical calibration voltage add and subtract the Vishay error (±1%).

\[ V_{ECAL} + 0.01(V_{ECAL}) = V_{\text{HIGH}} \]

\[ V_{ECAL} - 0.01(V_{ECAL}) = V_{\text{LOW}} \]

8) from \( V_{\text{HIGH}} \) calculate the corresponding load using the low extreme equation

*This best fit line will be shifted to the right of the original line, having lower slope and y intercept.
9) from $V_{o,\text{LOW}}$ calculate the corresponding load using the high extreme equation:

$$x_L = \frac{V_{o,\text{LOW}} - C_4}{M_4}$$  \hspace{1cm} [3.9iiib]

Figure 3.15 graphically describes the above procedure, the end result is a range for electrical calibration.
4. TORQUE CALIBRATION

FIGURE 4.1
TORQUE GAUGE LOCATIONS

4.1 CALIBRATION PROCEDURE

To induce a torque on the axle a disk was cut from a one inch steel plate. Holes were drilled to match the trailer wheel bolt pattern. A six foot long I beam, measured from the axle center, was welded to the disk.
The axle was raised off the ground, the outside tire removed and the beam arrangement bolted to the axle. The beam was leveled and the brakes applied. A hydraulic jack was placed under the free end of the beam with a load cell situated on top of the jack (see Fig. 4.2).

FIGURE 4.2
CALIBRATION CONFIGURATION

The jack and load cell were raised with a hand pump to induce a force six feet away from the axle center. The load cell indicator displayed the force, in pounds, applied by the jack.
The force was incremented to a maximum and decremented by a release valve on the hand pump, to the datum level. At predetermined levels the torque (force readout * 6 ft) and corresponding output voltage were recorded. Electrical calibration was performed at the datum level. A best fit line was plotted and the corresponding electrically calibrated torque was calculated.

4.2 DATA ANALYSIS

4.2.1 Initial Tests

The datum level is determined by the following axle configuration;

i) the brakes off

ii) the axle on the ground and

iii) the beam assembly removed

The vertical shear stress contribution is zero, as previously discussed. The effect of bending was assumed to be zero. The calibration was carried out as per the procedure outlined in the previous section.

Three gauges were calibrated: ADS1, BPS1 and CPS1. Positive torque was considered to be in the direction of the applied force induced by the jack. For these three gauges positive torque was therefore;

ADS1: counterclockwise direction
BPS1: counterclockwise direction
CPS1: clockwise direction

The bridge voltage output sign was neglected and just the magnitude of the voltage was recorded. The gauge sensitivities are listed below in Table 5 and the calibration curves generated for this test are in Figure 4.3. Refer to Appendix C for the data values plotted in Figure 4.3.

<table>
<thead>
<tr>
<th>Gauge</th>
<th>Sensitivity</th>
</tr>
</thead>
<tbody>
<tr>
<td>ADS1</td>
<td>$0.2 \text{ mV/V} = 3040 \text{ ft*lbs}$</td>
</tr>
<tr>
<td>BPS1</td>
<td>$0.2 \text{ mV/V} = 3279 \text{ ft*lbs}$</td>
</tr>
<tr>
<td>CPS1</td>
<td>$0.2 \text{ mV/V} = 3115 \text{ ft*lbs}$</td>
</tr>
</tbody>
</table>

TABLE 5
GAUGE SENSITIVITIES
INITIAL TORQUE CALIBRATION

1. Gauge ADS1: $Y = 3.29E-04 \times + 3.06E-01$
2. Gauge BPS1: $Y = 3.05E-04 \times -3.81E-01$
3. Gauge CPS1: $Y = 3.21E-04 \times + 2.28E-01$

FIGURE 4.3
Conclusions drawn from the test were all torsion gauges behave virtually the same as seen in Table 5, and not all torque contributing factors were considered as can be seen by the marked offset from the origin of the graph in Figure 4.3

4.2.2 Brake Application Torque

The mechanism which activates the brakes induces a torque on the axle.

The brake pod, which is fixed to the axle by a bracket, is supplied with air. A diaphragm expands within the pod pushing a rod which is attached to the slack adjustor. The slack adjustor pivots and turns the S arm which activates the brakes (see Fig. 4.4).

![Brake Application Torque Diagram]

**FIGURE 4.4**

**BRAKE APPLICATION TORQUE**

When charged with air the diaphragm is forced against the brake pod. This force pushes the rod and activates the slack adjustor. The moment arm of the force is the rod center to the axle center, six inches (6”) The diaphragm area is 30 square inches. Knowing the air pressure supplied to the diaphragm gives the force pushing the rod and multiplying by the moment arm results in the brake application torque.
$T_{BR} = 30 \text{ in}^2 \times \text{SUPPLIED AIR PRESSURE } \left(\frac{\text{lb}}{\text{in}^2}\right) \times \frac{1}{2} \text{ ft}$

$T_{BR} = 15(\text{in}^2 \times \text{ft}) \times \text{PSI} = \text{ft} \times \text{lbs}$

Because of the brake mechanism orientation on the axle it applies a clockwise torque to all driver's side gauges and a counterclockwise torque to all passenger side gauges when activated. The supplied air pressure does not remain constant during calibration, therefore it had to be recorded with each output voltage and applied torque.

4.2.3 Beam Torque and Bending Effect

The beam arrangement was not made from weightless material. Therefore its contribution to the results had to be considered. The beam torque always opposed the torque induced by the jack.

To ascertain the magnitude of the torque the free end of the beam arrangement was allowed to rest on a sensitive scale. The moment arm was measured and the torque obtained. The contribution of the beam arrangement used on the drivers side was 250 ft*lbs and 240 ft*lbs for the passenger side arrangement.

FIGURE 4.5
BEAM TORQUE
The effect of bending was determined during the vertical load calibration. The datum level was maintained consistent. The result being no matter what the vertical load the bending effect contaminates the voltage output on the torque gauges by a constant amount.

4.2.4 Sign Convention

A sign convention was developed for the sake of consistency.

Positive torques will be those torques which act in the direction of wheel rotation while travelling forward. Therefore torques induced in the counterclockwise (CCW) direction on the driver’s side are positive and those torques which are applied in the clockwise (CW) direction on the passengers side are positive.

These positive senses are determined by looking directly at the axle side being calibrated.

4.3 FINAL CALIBRATION TEST

For this test two gauges were monitored with consideration given to all variables. Gauge ADS1 was calibrated applying the torque via the jack, in the positive sense. Gauge BPS1 was calibrated with the applied torque in the negative direction.

Figure 4.6 shows the sense of all contributing torques with respect to the positive sign convention adopted for each gauge.

<table>
<thead>
<tr>
<th>Gauge; ADS1</th>
<th>Gauge; BPS1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical Load Range;</td>
<td>Vertical Load Range;</td>
</tr>
<tr>
<td>$V_0$ Due to Bending</td>
<td>$V_0$ Due to Bending</td>
</tr>
<tr>
<td>-0.002</td>
<td>-0.011</td>
</tr>
<tr>
<td>-0.008</td>
<td>-0.022</td>
</tr>
<tr>
<td>-0.007</td>
<td>-0.036</td>
</tr>
<tr>
<td>-0.007</td>
<td>-0.040</td>
</tr>
<tr>
<td>-0.003</td>
<td>-0.021</td>
</tr>
<tr>
<td>-0.005</td>
<td>-0.020</td>
</tr>
<tr>
<td>-0.005</td>
<td>-0.032</td>
</tr>
<tr>
<td>-0.004</td>
<td>-0.036</td>
</tr>
<tr>
<td>-0.004</td>
<td>-0.020</td>
</tr>
<tr>
<td>-0.006</td>
<td>-0.014</td>
</tr>
<tr>
<td>-0.006</td>
<td>-0.023</td>
</tr>
<tr>
<td>$\bar{X} = -0.005$</td>
<td>$\bar{X} = -0.030$</td>
</tr>
</tbody>
</table>

**TABLE 6**

BENDING EFFECT FOR GAUGES ADS1 AND BPS1
Figure 4.6
Sign Convention
TORQUE CALIBRATION OF ADS1

VISHAY SERIAL# 024155
EXCITATION VOLTAGE = 5V
FILTER SETTING = xB
GAIN = 1000

Ecal of 1.538 volts for Real of 100 kohms = 4898 ft*lbs

Best Fit for Y=MX+C where M= 3.20E-04, C=-2.81E-02

APPLIED TORQUE
+ = INCREASING
* = DECREASING

NOTE: THE ACTUAL OUTPUT VOLTAGE IS NEGATIVE.
 THEREFORE THIS PLOT IS A MIRROR IMAGE IN X

RESULTANT TORQUE (ft*lbs)

FIGURE 4.7
TORQUE CALIBRATION OF BPS1

Ecal of 1.525 volts for Rcal of 100 kohms = 4685 ft*lbs
Best Fit for Y=M*X+C where M=3.11E-04, C=6.55E-02

VISHAY SERIAL# 024155
EXCITATION VOLTAGE = 5V
FILTER SETTING = WB
GAIN = 1000

APPLIED TORQUE
+ - INCREASING
* - DECREASING

NOTE: THE ACTUAL RESULTANT TORQUE IS
NEGATIVE
THEREFORE
THIS PLOT IS A MIRROR IMAGE IN Y

RESULTANT TORQUE (ft*lbs)

FIGURE 4.8
ACTUAL ORIENTATION OF CALIBRATION PLOTS

BEST LINE FIT FOR GAUGE BPS1

BEST LINE FIT FOR GAUGE ADS1

CORRECTED VOLTAGE OUTPUT (volts)

RESULTANT TORQUE (ft*lbs)

FIGURE 4.9
Table 6 on page 42 lists the bending effect of each gauge being calibrated. The average bending cross table for gauge ADS1 is -0.005 volts therefore 0.005 volts is added to the recorded voltage readings to compensate for bending. A factor of 0.027 volts is added to gauge BPS1 by the same reasoning.

The graphical results for the calibration of gauge ADS1 are presented in Figure 4.7. Figure 4.8 exhibits the results of calibration on gauge BPS1. Appendix D gives the table of results plotted for both gauges.

4.4 DISCUSSION OF RESULTS

The two gauges ADS1 and BPS1 displayed good sensitivity, linearity, and repeatability.

For the purpose of error analysis the calibration plots (Figures 4.7 and 4.8) were transformed into the first quadrant of the Cartesian plane. The ADS1 calibration plot in Figure 4.7 is a mirror image in X and Figure 4.8 is a mirror image in Y. These transformations do not affect the magnitude of the plotted values. If our sign convention were different for torque and the voltage output recorder was connected reversing the polarity these mirror images would be the actual plots. The error analysis is documented in Appendix E the results are reported in Table 7 below;

Gauge ADS1

| ECAL Range; 1.538 ± 1% volts = 4898 ± 5.2% ft*lbs |
| Sensitivity Range; $\frac{0.2 \pm 1\% \text{ mV}}{\text{V}} = 3125 \pm 0.7\% \text{ ft*lbs}$ |

Gauge BPS1

| ECAL Range; 1.525 ± 1% volts = 4689 ± 5.4% ft*lbs |
| Sensitivity Range; $\frac{0.2 \pm 1\% \text{ mV}}{\text{V}} = 3215 \pm 1.3\% \text{ ft*lbs}$ |

TABLE 7
ERROR BOUNDS FOR GAUGES ADS1 AND BPS1

From Table 7 the gauge sensitivity ranges indicate that the estimated measurement errors listed in Appendix E are reasonable and the error analysis procedure is valid.

The best fit line for gauge BPS1 is:

$$y = 3.11 \times 10^{-4} x + 6.55 \times 10^{-2}$$
calculating the torque for the electrical calibration voltage of gauge ADS1 using the BPS1 best fit line yields 4735 ft*lb's. This value falls within the electrical calibration range calculated for ADS1.

The best fit line for gauge ADS1 is;

\[ y = 3.20 \times 10^{-4} x - 2.81 \times 10^{-2} \]

calculating the torque at 1.525 volts, BPS1 electrical calibration value, yields 4853 ft*lb's. This is within the range of electrical calibration for gauge BPS1.

Therefore all torque gauges will behave similarly when all torque contributing factors are considered.

The strain gauge misalignment errors resulting in voltage outputs due to bending may also produce a voltage output due to the shear stress caused by braking friction. All torque gauges should be tested to determine the bending and shear effects.

4.5 SAMPLE CALCULATION

A voltage output of -1.5 volts is recorded from gauge ADS1. What is magnitude of the applied torque assuming the lead wire lengths of 50 ft, and given a gain of 1000 and an excitation voltage of 5V.

From [3.5vii]

\[ V_o = \frac{5000}{R_G^*} \pi r \]
\[ \therefore -1.5 = \frac{5000}{R_G^*} \]

from [3.5vi]

\[ R_G^* = R_G + 2 R_L \]
\[ R_G^* = 120 + 2(50)(0.015) = 121.6 \Omega \]

\[ \therefore r = 3.648 \times 10^{-2} \]

from [3.41]

\[ \Delta R = \rho \frac{x}{R_G} \]
\[ \frac{\Delta R}{R_G} = K_G \epsilon \]
\[ \frac{3.648 \times 10^{-2}}{120} = 2.01 \epsilon \]

\[ \epsilon = 151.2 \mu \text{strains} = 151.2 \times 10^{-6} \frac{\Delta L}{L} \]
\[ \epsilon_{45^\circ} = \frac{1}{2} \left( \frac{\sigma_x}{E} + \frac{\tau}{\sigma_x} \right) \]

\[ \sigma_x = 0 \]
\[ C = 11.5 \times 10^6 \text{ lbs/in}^2 \]
\[ \tau = 3.48 \times 10^4 \text{ lb/in}^2 \]

from [3.11]
\[ \tau = \frac{Tc}{J} \]
\[ c = 2.5 \text{ in}^4 \]
\[ J = \frac{1}{4} \pi (2.5^4 - 1.875^4) = 41.9 \text{ in}^4 \]
\[ T = 5.83 \times 10^6 \text{ ft*in} = 4857 \text{ ft*lbs} \]

Using the best fit line of gauge ADS1 and an output voltage of 1.5 volts (due to the transformation used) results in a torque of 4775 ft*lbs. This is within 2% of the above calculated value. Therefore the assumed lead wire length of 50 ft is reasonably accurate.

5. VERTICAL LOAD CALIBRATION

5.1 WEIGH SCALE PREPARATION

The weigh scale used was designed to weigh railway cars. To serve the purpose of monitoring axle weight a \( \frac{1}{4} \)" plate was placed over the weight scales' rail section. This plate was of a sufficient width to accommodate both dual wheels of a single axle.

To monitor only one axle of a tandem set a 'C' channel was placed over the rails and raised clear of the scale by placing four one inch thick pieces of steel on both sides of each rail. The axle that is not of concern in the tandem set is driven onto this channel where it would not affect the scale reading.

5.1.1 Tractor Trailer Preparation

The front and rear compartments of the tanker trailer were filled with water. The purpose of this exercise was to induce an appropriate full scale load. The compartments were completely filled to prevent liquid slosh.
5.2 CALIBRATION OF DRIVE AXLES

FRONT OF VEHICLE

DRIVER SIDE

WALKING BEAMS

PASSENGER SIDE

FIGURE 5.1
DRIVE AXLE GAUGE LOCATIONS

5.2.1 Procedure

Initially the Front Drive Axle was calibrated with the Rear Drive Axle on the C channel. In this configuration gauges D2, D5, P2, and P5 were monitored.

The datum level was chosen as the trailer free of the fifth wheel (see Fig. 5.2), or the weight that the pavement experiences when the tractor is without the trailer. The brakes were not applied.

Load was applied in increments of 2 tonnes and the corresponding voltage output for each gauge was recorded. To apply the load in the appropriate increments the scale was preset. The landing gear was lifted off the ground letting the weight of the trailer fall on the fifth wheel until a scale deflection was detected. Raising the landing gear was stopped. An accurate scale reading was determined and the appropriate output voltages were recorded.

Once full load was reached the above process was repeated in reverse, that is, the landing gear was lowered to decrease load.
At zero, or the datum level, an electrical calibration was performed.

The data and a best fit line was plotted. The same process was followed to calibrate the Rear Drive Axle. The gauges monitored were D6, P6, D9, P9.

5.2.2 Discussion of Results

All gauges displayed a low sensitivity and non linear results. A further investigation is necessary before the results can be reported with confidence (see Appendix F for results).
5.3 TRAILER AXLE CALIBRATION

FRONT OF VEHICLE

SUSPENSION POINTS

ADS2                  APS2
NEWAY AXLE
AXLE A

DRIVER'S SIDE

BDS2                  BPS2
REYCO LEAD AXLE
AXLE B

PASSENGER'S SIDE

CDS2                  CPS2
REYCO TRAIL AXLE
AXLE C

FIGURE 5.3
TRAILER GAUGE LOCATIONS

5.3.1 Procedure Followed for the Reyco Tandem Set

Axle B, the lead tandem axle, was calibrated first. Axle C was on the C channel clear of the scale. Gauges BDS2, BPS2 and BPS1 were monitored. The reaction of gauge BPS1, the torque gauge, was monitored to establish the effect of vertical loading.
With the brakes off and the axle lifted clear of the scale by a vertical hoisting crane all gauges in question were zeroed. This configuration was chosen as the datum level because it is the point where the road experiences zero load.

The scale was preset to 2 tonnes and the axle was slowly lowered onto the plate until a deflection on the scale was detected. The hoist was stopped and an accurate scale reading was obtained. The voltage readings for all gauges were recorded.

Load was increased in approximately 2 tonne intervals following the above procedure until full axle load was achieved. The process was then reversed lifting the axle off the scale in 2 tonne intervals.

At the datum level an electrical calibration was performed. The data for each vertical load gauge was plotted on a graph of output voltage versus the total road load. The procedure was repeated for axle C.

5.3.2 Procedure Followed for the Neway Axle

Calibration of the Neway axle, axle A, followed the same basic outline, the difference being that the hoist was not needed. The Neway axle is an air lift axle and increasing and decreasing the load on the plate was accomplished by restricting the air flow to the lift mechanism.

5.4 SENSITIVITY OF THE MEASUREMENT SYSTEM TO BIAS TIRE LOADING

A test was performed to simulate the load gauge response while driving on non level ground, that is when all wheels on an axle set are not in contact with the road.

5.4.1 Test Procedure

This test was performed using axle A and monitoring gauge ADS2. The gauge was zeroed in the raised position, zero pavement load. The axle was lowered with the outside tire resting on a 1" steel plate, a reading was taken. This procedure was repeated with the plate under the inside tire and then no plate at all.
5.4.2 Results

Voltage

Axle Up
0.0

Axle Down
Plate under Outside tire +1.480
Plate under I/S tire +0.510
No Plate +0.915
Axle Up 0.0
O/S Tire +1.524
I/S Tire +0.516
No Plate +1.061

5.4.3 Discussion of Results

The strain gauges for vertical loading are affected by the direct stress due to bending.

\[ \sigma = \frac{F_{xy}}{I_{c}} \]  \[3.211\]

The same vertical load produced varying results because the bending stress depends on the induced force and the distance that force is to the strain gauge location on the axle. Therefore two variables determine the amount of stress, thus strain, experienced by the axle. Care must be taken in selecting test sites. Rutter roads should be avoided for better loading predictions. Also the tire pressure must be maintained constant in all tires to avoid bias tire loading. All calibration plots showed very little deviation from the best fit line.

Each wheel should be calibrated with respect to the load which that wheel imparts on the road to determine the load sharing activity of the axles. This will also provide a check for the calibrations performed and listed in Appendix G.

5.5 AN ALTERNATE SOLUTION

A shear stress is also produced by vertical loading:

\[ \tau = \frac{F_{y}}{I_{c}} \] \[3.21\]
The only possible variable which can change the value of this shear stress is \( F \), the vertical load or shear force. All other variables remain constant as long as the strain gauge remains fixed on the axle circumference.

Figure 3.2b on page 8 describes how the shear stress flows through the axle's cross section. The location of maximum stress is at the neutral axis.

Strain gauges will respond to shear stress if they are oriented at 45° and 135° to the neutral axis as discussed in section 3.5. A slight variation in the Wheatstone Bridge circuit used to detect strain due to torsion will result in a voltage output from strains caused by the shear stress due to vertical loading.

Figure 5.4 presents the proposed Wheatstone bridge circuit to be used for shear force. Comparing this circuit to the torsion circuit (Fig. 3.5 p. 14), the gauges will have the same orientation on the axle but gauges 2 and 3 have been interchanged.
The voltage output for the shear force bridge is given by the following equation:

\[ V_o = V_o^{tr} \left[ \frac{R_1}{R_1 + R_4} - \frac{R_3}{R_2 + R_3} \right] * \Delta F \]  \hspace{1cm} [5.41]

Figure 3.7 on page 18 describes the gauge reactions to a shear stress caused by vertical loading. Element A, on the rear face of the axle, contains gauges 1 and 4. Element B, on the front face of the axle includes gauges 2 and 3. As noted in section 3.5.3 the elements are experiencing stresses in opposite directions when compared to each other.

The shear stress in element A will be considered positive. Knowing the gauge orientation with respect to the neutral axis and using formulæ [3.5iva and b] (keeping in mind the above sign convention) strain experienced by the gauges due to the shear force \( F \) are:

\[ \varepsilon_1 = -\gamma u \text{ strains} \]
\[ \varepsilon_2 = -\gamma u \text{ strains} \]
\[ \varepsilon_3 = +\gamma u \text{ strains} \]
\[ \varepsilon_4 = +\gamma u \text{ strains} \]

From [3.41] the change in gauge resistance is:

\[ \Delta R = \varepsilon R \]

where the sign of the resistance change will be the same as the sign of strain and all gauge resistances \( R_G \) from [3.5vi] are equal. Therefore from [5.41]:

\[ V_o = V_o^{tr} \left[ \frac{R_G^{+\varepsilon}}{2R_G} - \frac{R_G^{-\varepsilon}}{2R_G} \right] * \Delta F \]

\[ V_o = V_o^{tr} \left[ -\frac{2\varepsilon}{R_G} \right] * \Delta F \]

\[ V_o = -\frac{\varepsilon R_G^{+\varepsilon} \Delta F}{R_G} \]

for a shear force applied in the upward direction.

Considering both possible loading directions where shear in the upward direction is positive:

\[ V_o = \frac{V_o^{tr} R_G^{+\varepsilon} \Delta F}{R_G} \]  \hspace{1cm} [5.4ii]
Applying a torque in either direction will have no effect on the Wheatstone bridge circuit as will be proven.

Consider a clockwise torque as positive and an oppositely applied torque as negative. Both shear stress elements on the axle containing the gauges will be oriented in the same direction. Below are listed the strains experienced by the gauges. Positively applied torque is in the top sign. From [3.3iva and b] and from [3.41]:

\[
\epsilon_1 = t_y \mu \text{ strains and } \Delta R = +t_r \\
\epsilon_2 = -t_y \mu \text{ strains and } \Delta R = -t_r \\
\epsilon_3 = t_y \mu \text{ strains and } \Delta R = +t_r \\
\epsilon_4 = -t_y \mu \text{ strains and } \Delta R = -t_r
\]

From [5.41]

\[
V_o = V^+ \left( \frac{K_{a_1} r_{t_r}}{R_{G_1} r_{t_r} + R_{G_2} r_{t_r}} - \frac{K_{a_2} r_{t_r}}{R_{G_3} r_{t_r} + R_{G_4} r_{t_r}} \right) \Delta P
\]

therefore \( V_o = 0 \) and torque has no effect.

Bending due to vertical and frictional forces will also have no effect if the gauges are placed carefully at the same spots over the neutral axis on both faces of the axle. Refer to section 3.5.3 and 3.5.4 where these effects are considered with respect to the torque instrumentation. The same arguments apply here.

This gauge arrangement should be investigated to determine if it shows good sensitivity and reduces the sensitivity to error as shown in the bending gauge arrangement due to bias tire loading.

REFERENCES


APPENDIX A

VISHAY ERROR
A bench test was performed to determine the error in each reading conditioned by a Vishay.

All Vishay units used during the calibration procedure were interfaced with an Intertechnology 1550 Strain Indicator Calibrator (S.I.C.) which simulates the different resistance changes due to varying strains.

The excitation voltage and gain were set as in the calibration procedure, 5 volts and 1000 respectively.

Test Procedure:

1. Each Vishay was balanced at zero µ strains.
2. 600 µ strains then 1600 µ strains were induced via the S.I.C. and the voltage output recorded.
3. Electrical calibration voltage output was also recorded.
4. Steps 2 and 3 were repeated 3 times for each Vishay.
5. The average output voltage of the three readings for 600 µ strains were calculated for each Vishay. The readings of 1600 µ strains and electrical calibration were also averaged for each Vishay to yield the mean Vishay reading.
6. The overall average was then calculated for each strain level.
7. The overall average was subtracted from the individual mean Vishay readings for each strain level.
8. An average of these values was calculated giving the mean change in readings.
9. The error was determined by dividing the overall reading average into the mean change in readings and multiplying by 100%. The results are documented in Table Al.
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<th>Reading at 1600 μ strains (in volts)</th>
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\[ E_{600\mu\varepsilon} = \frac{0.016}{1.498} \times 100\% = 1\% \]

\[ E_{1600\mu\varepsilon} = \frac{0.42}{4.000} \times 100\% = 1\% \]

\[ E_{ECAL} = \frac{0.016}{1.515} \times 100\% = 1\% \]

**TABLE A1**

**VISHAY ERROR TESTS AND RESULTS**
APPENDIX B

CONFIDENCE INTERVALS AND THE STUDENT'S $t$ DISTRIBUTION
Since point estimates cannot be expected to coincide with the quantities they are intended to estimate, it is sometimes preferable to replace them with interval estimates, that is, intervals for which one can assert with a reasonable degree of certainty that they contain the parameter under consideration.

The Student's t distribution is used to estimate the confidence interval because the y data points are assumed to be a normally distributed population about the mean \( \alpha \) and \( \beta \) but we do not have an infinite number of data points to use the normal distribution with certainty.

The Student's t distribution has a similar shape to that of a normal distribution, but its variance (shape) depends upon a parameter called the number of degrees of freedom (see Fig. B.1).

![NORMAL DISTRIBUTION](image)

![STUDENT'S t DISTRIBUTION](image)

**FIGURE B.1**

For our purpose we have \( n - 2 \) degrees of freedom, where \( n \) represents the number of data points. Two degrees of freedom are lost because \( \alpha \) and \( \beta \) are replaced by their least squares estimations.

A confidence interval of 95% establishes a range of acceptable numbers for \( \alpha \) and \( \beta \) doing away with the other 5%. The deleted 5% is split such the last 2.5% of the numbers lying within the distribution range are omitted from the right tail and the other 2.5% omitted from the left tail (see Fig. B.2).
### Table B.1

**Confidence Values**

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APPENDIX C

INITIAL TORQUE CALIBRATION DATA TABLES
### Initial Calibration of GUAGE ADS1

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1. APPL TORQ  
2. VOLTS OUT  
**Subfiles:** NONE

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**DATA MANIPULATION**

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2. VOLTS OUT

Subfiles: NONF

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TORQUE CALIBRATION RESULTS FOR GAUGES ADS1 AND BPS1
DATA MANIPULATION

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Number of observations: 24
Number of variables: 7

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3. BRAKE TORK
4. BEAM TORK
5. TOTAL TORK
6. BENDING Vo
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DATA MANIPULATION

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APPENDIX E

TORQUE CALIBRATION ERROR ANALYSIS CALCULATIONS AND RESULTS
The physical errors present in the system while calibrating are:

for Brake Application Torque:

\[ T_{BR} = (\text{moment arm})(\text{brake air pressure})(\text{diaphragm area}) \]

errors;

moment arm = 0.5 ft ± 0.1/12 ft
brake air pressure = recorded reading ± 3 psi
diaphragm area = 30 in\(^2\) ± 1 in\(^2\)

for Applied Torque:

\[ T_{APPL} = (\text{load cell reading})(6 \text{ ft}) \]

errors;

load cell reading = recorded reading ± 5 lbs

\[ T_{APPL} = \text{recorded } T_{APPL} ± 30 \text{ ft•lbs} \]

The beam torque contribution is error free.

It is desired to combine these errors in order to find the extreme limits of the resultant torque.

For gauge ADS1 positive is in the CCW direction. Therefore the resultant torque is given by;

\[ T_{RES} = T_{APPL} + T_{BR} - T_{BEAM} \]

\[ T_{RES} = T_{APPL} + T_{BR} - 250 \]

for the lowest extreme torque (to be used in calculating the highest extreme line).

\[ T_{RES, LOW} = (T_{APPL} - 30) + [(\text{recorded psi} - 3\text{ psi})(0.5\text{ ft} - \frac{0.1}{12} \text{ ft})] - 250 \]

\(30 \text{ in}^2 - 1 \text{ in}^2\)

for the highest extreme torque (to be used in calculating the lowest extreme line).

\[ T_{RES, HIGH} = (T_{APPL} + 30) + [(\text{recorded psi} + 3 \text{ psi})(0.5\text{ ft} + \frac{0.1}{12} \text{ ft})] - 250 \]

\(30 \text{ in}^2 + 1 \text{ in}^2\)

The error analysis is only valid if we transform the actual calibration plot into the first quadrant of the cartesian plane. This is accomplished by ignoring the negative bridge output voltage, that is, a mirror image in X.

For gauge BPS1 positive torque is in the CW direction, therefore;

\[ T_{RES} = T_{BR} + T_{BEAM} - T_{APPL} \]
A transformation of the BPSL calibration plot requires all torque to change sign, a mirror image in Y.

\[-T_{RES} = -(T_{APPL} - T_{BR} - T_{BEAM})\]

To calculate the best fit line giving the lowest extreme line we require the greatest possible resultant torque where the brake torque contribution has to be the lowest possible value because of our sign convention.

\[T_{RES\text{HIGH}} = -(T_{APPL} - 30) - [(\text{recorded psi} - 3 \text{ psi})(0.5 - \frac{0.01}{12}) \times (30 \text{ in}^2 - l \text{ in}^2)] - 240\]

Using the same logic, the highest extreme line is given by the lowest resultant torque.

\[T_{RES\text{LOW}} = -(T_{APPL} + 30) - [(\text{recorded psi} + 3)(0.5 + \frac{0.01}{12})(30 \text{ in}^2 + l \text{ in}^2)] - 240\]

The procedure explained in section 3.9.2 is followed giving the error in electrical calibration and gauge sensitivity. The resultant torques and voltage outputs used for these calculations are listed in the tables which follow under columns 4 and 5 respectively.

Sample Calculation Using the Results of Gauge ADSL

For the highest extreme line, a best fit line is calculated using the ordered pairs,

\[(T_{RES\text{LOW}}, V_0)\]

to yield;

\[y = 3.20 \times 10^{-4} x + 1.22 \times 10^{-2}\]

Applying the least squares error calculation for the range of \(\alpha\): from [3.911a]

\[C - \tau_{n/2} \cdot Se\left[\frac{\sum x_i}{n} \cdot \frac{\sum x_i^2}{n} \cdot \frac{1}{2}\right] < \alpha < C + \tau_{n/2} \cdot Se\left[\frac{S_{xx} + (\sum x_i)^2}{n} \cdot \frac{1}{2}\right]\]

where using a 95% confidence interval \(\tau_{0.025}\)

for 22 degrees of freedom = 2.074

\[n = 24\]
\[Se = 0.012\]
\[S_{xx} = 229.1 \times 10^7\]
\[(\sum x_i)^2 = 1.381 \times 10^{10}\]
We want to calculate the highest extreme line therefore;

\[
C_4 = 1.22 \times 10^{-2} + 2.074(0.012) \left[ \frac{229.1 \times 10^7 + 1.381 \times 10^{10}}{24(229.1 \times 10^7)} \right]^{\frac{1}{2}}
\]

\[
C_4 = 2.485 \times 10^{-2}
\]

Calculating the highest extreme slope from [3.911b]

\[
M_4 = M + \tau \frac{n}{2} \cdot \text{Se} \left[ \frac{n}{S_{xx}} \right]^{\frac{1}{2}}
\]

\[
= 3.2 \times 10^{-4} - 2.074(0.012) \left[ \frac{24}{229.1 \times 10^7} \right]^{\frac{1}{2}}
\]

\[
M_4 = 3.221 \times 10^{-4}
\]

Therefore the highest extreme line is;

\[
y_{\text{HIGH}} = 3.221 \times 10^{-4} x + 2.485 \times 10^{-2}
\]

Using the ordered pairs;

\[(T_{\text{RES HIGH}}, V_{\text{HIGH}})\]

yields;

\[
y = 3.20 \times 10^{-4} x - 6.95 \times 10^{-2}
\]

Calculating the range of the true line to achieve the lowest system line from equations [3.911a and b]

\[
C_1 = C - \tau \frac{n}{2} \cdot \text{Se} \left[ \frac{n}{S_{xx}} \right]^{\frac{1}{2}}
\]

and

\[
M_1 = M + \tau \frac{n}{2} \cdot \text{Se} \left[ \frac{n}{S_{xx}} \right]^{\frac{1}{2}}
\]

where;

\[
\tau_{0.025} = 2.074
\]

\[
n = 24
\]

\[
\text{Se} = 0.012
\]

\[
S_{xx} = 229.1 \times 10^7
\]

\[
(\sum x_i)^2 = 1.381 \times 10^{10}
\]

\[
C = -6.95 \times 10 \times 10^{-2}
\]

\[
M = 3.2 \times 10^{-4}
\]
therefore the lowest extreme line is
\[ y_{\text{LOW}} = 3.173 \times 10^{-4} x - 8.355 \times 10^{-2} \]

summarizing;
\[ y_{\text{LOW}} = 3.173 \times 10^{-4} x - 8.355 \times 10^{-2} = M_1 x + C_1 \]
\[ y_{\text{HIGH}} = 3.221 \times 10^{-4} x + 2.485 \times 10^{-2} = M_4 x + C_4 \]

Considering the Vishay error in electrical calibration, (±1% of the voltage reading).

\[ V_{\text{OECAL}} = 1.538 \]
\[ V_{\text{OHIGH}} = (1.01)(1.538) = 1.553 \]
\[ V_{\text{OLOW}} = (0.99)(1.538) = 1.523 \]

from equation [3.9iia] the upper electrical calibration bound, using the lowest extreme line, is;
\[ x_{\text{RIGHT}} = \frac{V_{\text{OHIGH}} - C_1}{M_1} = 5158 \text{ ft}^2\text{lbs} \]

from equation [3.9iiib] the lower electrical calibration bound is;
\[ x_{\text{LEFT}} = \frac{V_{\text{OLOW}} - C_4}{M_4} = 4651 \text{ ft}^2\text{lbs} \]

The actual electrical calibration recorded is;

\[ 1.538 \text{ volts} \equiv 4898 \text{ ft}^2\text{lbs} \]

\[ \frac{x_{\text{RIGHT}} - 4898}{4898} \times 100\% = 5.3\% \]
\[ \frac{x_{\text{LEFT}} - 4898}{4898} \times 100\% = -5.0\% \]

Therefore for gauge ADS1;

\[ 1.538 \pm 1\% \text{ volts} \equiv 4898 \pm 5\% \text{ ft}^2\text{lbs} \]

Calculating the gauge sensitivity range;

\[ \text{Gain} = 1000 \]
\[ \text{Excitation Voltage} = 5V \]

calibrated gauge sensitivity from [3.8v]
\[ \frac{0.2 \text{ mV}}{V^+} \equiv 3.125 \text{ ft}^2\text{lbs} \]
from the highest extreme line;

\[
\frac{0.2 \text{ mV}}{V_r} \equiv 3105 \text{ ft*lbs}
\]

from the lowest extreme line;

\[
\frac{0.2 \text{ mV}}{V_r} \equiv 3152 \text{ ft*lbs}
\]

calculating the percentage deviation

\[
\frac{3105 - 3125}{3125} \times 100\% = -0.6\%
\]

\[
\frac{3153 - 3125}{3125} \times 100\% = +0.8\%
\]

summarizing;

electrical calibration error for ADS1;

\[
1.538 \pm 1\% \text{ volts} \equiv 4898 \pm 5.2\% \text{ ft*lbs}
\]
gauge sensitivity

\[
\frac{0.2 \pm 1\% \text{ mV}}{V_r} \equiv 3125 \pm 0.7\% \text{ ft*lbs}
\]

electrical calibration error for BPS1;

\[
n = 30
\]

\[
t_{n/2 \text{ for } 28 \text{ d.o.f.}} = 2.048
\]

\[
y_{\text{low}} = 3.08 \times 10^{-4}x + 1.69 \times 10^{-2}
\]

\[
y_{\text{high}} = 3.146 \times 10^{-4}x + 1.156 \times 10^{-1}
\]

\[
1.525 \pm 1\% \text{ volts} \equiv 4689 \pm 5.4\% \text{ ft*lbs}
\]
gauge sensitivity

\[
\frac{0.2 \pm 1\% \text{ mV}}{V_r} \equiv 3215 \pm 1.3\%
\]
**DATA MANIPULATION**

**ADJUSTED TORQUE VALUES FOR THE HIGHEST EXTREME LINE OF ABS1**

Data file name: UPERRA:F8

Number of observations: 24

Number of variables: 5

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3. BRAKE TORK
4. TOTAL TORK
5. CORRECT Vo

Subfiles: NONE

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HIGHEST EXTREME PARAMETERS FOR GAUGE ADS1

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SIGMA Y1 = 3.784E+01
SIGMA X1*Y1 = 2.158E+05
SIGMA X1*X1 = 6.709E+08
SIGMA Y1*Y1 = 6.941E+01

# UF DATA PTS. = 2.400E+01
r@0.025 = 2.074E+00
Sxx = 2.291E+09
Syy = 2.341E+02
Sxy = 7.321E+05
S_e = 1.167E-02

THE UPPER EXTREME LINE IS;

\[ y = 3.221E-04 \times 2.485E-02 \]
DATA MANIPULATION

ADJUSTED TORQUE VALUES FOR THE LOWEST EXTREME LINE OF ADI

Data file name: L\TERRA.F8
Number of observations: 24
Number of variables: 5

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3. BRAKE TORK
4. TOTAL TORK
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LOWEST FXTREME PARAMETERS FOR GAUGE ADS1

SIGMA X1 = 1.235E+05
SIGMA Y1 = 3.784E+01
SIGMA X1*Y1 = 2.251E+05
SIGMA X1*X1 = 7.303E+08
SIGMA Y1*Y1 = 6.941E+01

# OF DATA PtS. = 2.400E+01

r@0.025 = 2.074E+00
Sxx = 2.286E+09
Syy = 2.341E+02
Sxy = 7.313E+05
Se = 1.237E-02

THE LOWER EXTREME LINE IS ;

Y = 3.173E-04 X + -8.355E-02
**DATA MANIPULATION**

ADJUSTED TORQUE VALUES FOR THE HIGHEST EXTREME LINE OF BPS1

Data file name: UPERRB:F8
Number of observations: 30
Number of variables: 5

Variables names:
1. APPL TORK
2. BRAKE PSI
3. BRAKE TORK
4. TOTAL TORK
5. CORRECT Vo

Subfiles: NONE

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HIGHEST EXTREME PARAMETERS FOR GAUGF BPS1

\[ \sigma_x = 6.835E+04 \]
\[ \sigma_y = 2.443E+01 \]
\[ \sigma_{xy} = 1.036E+05 \]
\[ \sigma_{x^2} = 3.096E+08 \]
\[ \sigma_{y^2} = 3.482E+01 \]

\# OF DATA PTS. = 3.000E+01
\[ \mu_0 = 0.025 = 2.048E+00 \]
\[ \mu_x = 4.618E+09 \]
\[ \mu_y = 4.479E+02 \]
\[ \mu_{xy} = 1.438E+06 \]
\[ \mu_e = 2.016E-02 \]

THE UPPER EXTREME LINE IS;

\[ y = 3.146E-04 \times + 1.156E-01 \]
**DATA MANIPULATION**

**ADJUSTED TORQUE VALUES FOR THE LOWEST EXTREMELINE OF BPSI**

Data file name: LTERM:EQ
Number of observations: 30
Number of variables: 5

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4. TOTAL TORK
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LOWEST FEXTREME PARAMETERS FOR GAUGE BPS1

SIGMA XI = 7.599E+04
SIGMA Yi = 2.443E+01
SIGMA Xi*Yi = 1.097E+05
SIGMA Xi**Xi = 3.461E+08
SIGMA Yi*Yi = 3.482E+01

# OF DATA PTS. = 3.000E+01
r@0.025 = 2.048E+00
Sxx = 4.624E+09
Syy = 4.479E+02
Sxy = 1.439E+06
Sxy = 1.818E-02

THE LOWER EXTREME LINE IS:

Y = 3.081E-04 X + 1.690E-02
APPENDIX F

DRIVE AXLE VERTICAL LOAD RESULTS
**DATA MANIPULATION**

**NEW ZERO FOR REAR DRIVE AXLE (CALIBRATION DATA)**

Data file name: CAL2RD:F8
Number of observations: 13
Number of variables: 5

Variables names:
1. TUNNES
2. GAGE D9
3. GAGE D6
4. GAGE P9
5. GAGE P6

Subfile name  beginning observation--number of observations
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2. DEC LOAD  9  6

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VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION
GAUGE MONITERED; DS
FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 024125
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M = .100, C = -.235 & R = .993

1.5

1.0

0.5

0.0

2.0

2.5

3.0

3.5

4.0

4.5

0  2  4  6  8  10  12  LOAD (TONNES)

VOLTAGE OUTPUT (VOLTS)

* -- INCREASING LOAD
* -- DECREASING LOAD

Ecal of 1.520 VOLTS for Rea! of 100 kohms = 17.617 TONNES
VERTICAL LOAD CALIBRATION

4.5 -
HENDRICKSON SUSPENSION

GUAGE MONITORED: BG

4.0 -
FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 224279
GAIN = 1800
EXCITATION VOLTAGE = 5 V

3.5 -

3.0 -

2.5 -

Best Fit for Y = Mx + C where M = .171, C = -4.48 & R = .981

2.0 -

1.5 -

1.0 -

.5 -

.0 -

INCREASING LOAD
DECREASING LOAD

Load of 1.573 VOLTS for Real of 100 kohms = 11.815 TONES

0 2 4 6 8 10 12 12

LOAD (TONNES)
VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION
GAUGE MONITORED: H.3
FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 245345
GAIN = 1030
EXCITATION VOLTAGE = 5V

Best Fit for \( y = mx + c \) where \( m = 0.143, c = -0.219 \) & \( R = 0.555 \)

Scale of 1.503 VOLTS for Real of 100 kohms = 12.027 TONNES

\( \times \) -- INCREASING LOAD
\( * \) -- DECREASING LOAD
VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION

GAUGE MONITORED: PG

FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 024076
GAIN = 1020
EXCITATION VOLTAGE = 5V

3.0-

2.5-

2.0-

1.5-

1.0-

.5-

.0-

0.0-- 2 4 6 8 10 12

LOAD (TONNES)

Best fit for Y = Mx + C where M = .287, C = -.462 & R = .954

Ecal of 1.508 VOLTS for Rcal of 102 kohms = 6.839 TONNES

+ -- INCREASING LOAD
* -- DECREASING LOAD
Data file name: DATA
Number of observations: 13
Number of variables: 5

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2. GAGE P2
3. GAGE W5
4. GAGE P2
5. GAGE W5

Subfile name  
beginning observation—number of observations
1. INC LOAD  
   1—6
2. DEC LOAD  
   7—5
3. EXTREMES  
   12—2

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VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION
GAUGE MONITORED: D2
FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 024125
GAIN = 1000
EXCITATION VOLTAGE = 5V

Voltage Output (Volts)

Best Fit for Y=MX+C where M = .086, C = -.223 & R = .984

1.5

= 1.515 VOLTS for Real of 180 kohms = 22.177 TONNES

\[ Y = 0.086X - 0.223 \]

\[ R^2 = 0.984 \]

5

GAIN = 1000

EXCITATION VOLTAGE = 5V

Load (Tonnes)

\[ Y = 0.086X - 0.223 \]

INCREASING LOAD

DECREASING LOAD

0.0

4

6

8

10

12

Load (Tonnes)
VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION
GAUGE MONITORED; DS

FAIRBANKS MORRIS SCALE USED TO MONITOR LOAD

VISHAY SERIAL #; 024078
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M= .186, C= -.535 & R= .971

Equal of 1.525 VOLTS for read of 100 kohms = 11.060 TONNES

+ -- INCREASING LOAD
* -- DECREASING LOAD
VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION
GAUGE MONITORED: P2

FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 045349
GAIN = 1000
EXCITATION VOLTAGE = 5V

VOLTAGE OUTPUT (VOLTS)

Best Fit for Y=MX+C where M = .128, C = -.213 & R = .993

1.5

1.0

0.5

0.0

LOAD (TONNES)

Ecal of 1.323 VOLTS for Rcal of 100 kohms = 13.636 TONNES

+ -- INCREASING LOAD
* -- DECREASING LOAD
VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION
GAUGE MONITORED; P5

FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 024876
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for \( Y = MX + C \) where \( M = 0.278 \), \( C = -0.584 \) & \( R = 1.000 \)

Ecal of 1.519 VOLTS for Real of 100 kohms = 7.795 TONNES

+ -- INCREASING LOAD
* -- DECREASING LOAD
APPENDIX G

TRAILER AXLE VERTICAL LOAD RESULTS
DATA MANIPULATION

Calibration of Trailer Axle #1

Data file name: DATA
Number of observations: 10
Number of variables: 3

Variables names:
1. TONNES
2. ADS2
3. APSP

Subfiles: NONE

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VERTICAL LOAD CALIBRATION

NEWAY AIR SUSPENSION
GAUGE MONITORED; ADS2
FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISAY SERIAL #: 324125
GAIN = 1000
EXCITATION VOLTAGE = 5V

VOLTAGE OUTPUT (VOLTS)

Best Fit for \( y = mx + c \) where \( m = 0.213, c = 0.003 \) & R=1.002

2.0 - Read of 1.545 VOLTS for Read of 100 kNmas = 7.243 TONNES

+ --- INCREASING LOAD
* --- DECREASING LOAD

LOAD (TONNES)
VERTICAL LOAD CALIBRATION

NEWAY AIR SUSPENSION
GAUGE MONITORED; APS2
FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 024079
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y = MX+C where M = 0.231, C = 0.027 & R = 0.999

Ecal of 1.587 VOLTS for Resi of 100 kohms = 6.903 TONNES

+ -- INCREASING LOAD
* -- DECREASING LOAD
Data file name: DATA
Number of observations: 26
Number of variables: 3

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2. BUS2
3. DKS2

Subfiles: NONE

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VERTICAL LOAD CALIBRATION

REYCO LEAD TANDEM AXLE 4 SPRING SUSPENSION
GAUGE MONITORED: BDS2
FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 024125
GAIN = 1200
EXCITATION VOLTAGE = 5V

Best Fit for Y = MX + C where M = 0.360, C = 0.874 & R = 0.989

Scale of 2.188 VOLTS for real of 100 kohms = 5.874 TONES

+ --- INCREASING LOAD
* --- DECREASING LOAD
VERTICAL LOAD CALIBRATION

REYCO LEAD TANDEM AXLE
GUAGE MONITORED; BPS2
FAIRBANK'S MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 024075
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M = .257, C = -.039 & R=1.000

Scale of 1.578 VOLTS for Real of 100 kN/m = 6.262 TONNES

+ -- INCREASING LOAD
* -- DECREASING LOAD
CALIBRATION OF TRAILER AXLE #3

Data file name: DATA
Number of observations: 29
Number of variables: 3

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3. CPS2

Subfile: NONE

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<td>2.73400</td>
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<td>1.85800</td>
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<td>1.70650</td>
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<tr>
<td>29</td>
<td>0.00000</td>
<td>-.02020</td>
<td>.00260</td>
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</table>
VERTICAL LOAD CALIBRATION

REYCO TRAIL TANDEM AXLE 4 SPRING SUSPENSION
GAUGE MONITORED; CDS2
FAIRBANKS MORRIS SCALE USED TO MEASURE LOAD

VISHAY SERIAL #: 024125
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M=.384, C=.032 & R=1.000
Equal of 2.212 VOLTS for Load of 100 kohms = 5.673 TONNES

+ --- INCREASING LOAD
* --- DECREASING LOAD
VERTICAL LOAD CALIBRATION

REYCO TRAIL TANDEM AXLE  4 SPRING SUSPENSION
GAUGE MONITORED; CPS2
FAIRBANKS MORRIS SCALE USED TO MONITOR LOAD

VISHAY SERIAL #: 024079
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M= .238, C= -.018 & R=1.000
Ecal of 1.585 VOLTS for Real of 100 kohms = 5.735 TONNES

+ — INCREASING LOAD
* — DECREASING LOAD
APPENDIX H

HAYES DANA AXLE COMPONENT WEIGHTS
<table>
<thead>
<tr>
<th>Description</th>
<th>Weight</th>
<th>Quantity Per Axle</th>
<th>Total Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nuts</td>
<td>2 lbs*</td>
<td>2</td>
<td>4 lbs</td>
</tr>
<tr>
<td></td>
<td>(set of 10)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Studs</td>
<td>3 lbs 9.5 oz</td>
<td>2</td>
<td>7</td>
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<tr>
<td></td>
<td>(set of 10)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tire and Rim</td>
<td>228 1/3</td>
<td>4</td>
<td>913</td>
</tr>
<tr>
<td>Cam Shaft</td>
<td>14</td>
<td>2</td>
<td>28</td>
</tr>
<tr>
<td>Brake Shoes &amp; Lining</td>
<td>18</td>
<td>4</td>
<td>72</td>
</tr>
<tr>
<td>Hub &amp; Drum</td>
<td>137</td>
<td>2</td>
<td>274</td>
</tr>
<tr>
<td>Bare Axle</td>
<td>235</td>
<td>1</td>
<td>235</td>
</tr>
<tr>
<td>Spring Suspension Assembly (Reyco Leaf)</td>
<td>139</td>
<td>2</td>
<td>278</td>
</tr>
<tr>
<td>Air Chamber</td>
<td>10</td>
<td>2</td>
<td>20</td>
</tr>
<tr>
<td>Brake Arm</td>
<td>7</td>
<td>2</td>
<td>14</td>
</tr>
<tr>
<td>Springs, Cam Holder</td>
<td>24½</td>
<td>2</td>
<td>49</td>
</tr>
</tbody>
</table>

Total Weight = 1894 Lbs

Hayes Dana Axle Component Weights
FINAL CALIBRATION STANDARDS
1.0 INTRODUCTION

Nine tractor-trailer axles strain gauged to measure vertical load were calibrated using an aircraft weighing scale and a Vishay strain gauge conditioning amplifier. The gauges were located on the passenger side at midpoint between the wheel hub and the suspension point.

Of the nine axles, two are tractor drive axles while the remaining are trailer axles. The tractor front and rear drive axles are supported by a Hendrickson RTE 440 tandem suspension. The trailer axles are numbered from the front end of the trailer. Axle 01 is fixed to a Neway air suspension while axles 02 and 03 form an axle group at the rear of the trailer. The axles for three different trailer suspension types were calibrated; namely these were a Reyco 4 spring suspension, a Neway air suspension and a Chalmers walking beam rubber suspended suspension.

Equipment

- ten single element (uniaxial) strain gauges
- aircraft weighing scale
- Vishay signal conditioning amplifier,
  Ser. No. 045312
- voltmeter, Ser. No. 10343
- two 3/4" steel plates, 14" x 18"
- White Freightliner tractor
- NRC modified Pruehauf tanker-trailer
- air and hydraulic jacks
- level
Test Setup

The calibration was performed in a temperature controlled environment of 20°C.

The tractor axles were calibrated by jacking the rear end of the tractor and positioning load cells under the dual wheel of the axle being calibrated. The three load cells were sandwiched between two 3/4" steel plates. They were placed in a manner to record approximately equal loads. Lift blocks were placed under the remaining tractor drive wheels to maintain the walking beam level with the horizontal. Photographs 1 to 6 show the load cell and jacking arrangement used for calibrating the vehicle.

The same type of load cell configuration was used to record the vertical load on the trailer axles. The trailer was kept level to the horizontal (less than 1°) during all three trailer axle calibrations by jacking the unmonitored dual wheels on lift blocks.

2.0 CALIBRATION

In all cases the load cells and the Vishay unit were zeroed with the dual wheel in the jacked position (no contact between dual wheel and top steel plate).

The vertical load was generated to the tractor axles by lowering the tractor with the use of two air jacks. A level was installed on the tractor chassis to ensure zero roll angle during calibration.
3.0 RESULTS AND OBSERVATIONS

A plot of nine calibration curves is shown in Graphs 1 to 9. The nine calibrations are found to be accurate within 0.15 tonne for any given voltage output.

The voltage output for the front drive axle oscillated consistently during calibration with an amplitude of approximately 5 mV for any given vertical load. The voltage output used for the least-square fitting was the average of the maximum and minimum values recorded.

The loading system on the Neway air suspension required about five minutes for the applied load to settle to a reasonably constant load. For every observation point the voltage output was recorded followed by the three load cell readings and a final voltage output reading. Hence, the average load and average voltage output over the time of the reading was recorded. The loading increased or decreased from the average load by less than 0.02 tonne during the time of the recording.

The voltage output and the load cell force for the rear drive axle and axles #2 and #3 were recorded without any appreciable drift.

4.0 CONCLUSION

The calibration curves plotted on Graphs 1 to 9 meet the accuracy requirements for the study and were therefore used as the calibration data in the dynamic suspension response testing.
Figures 1, 2 Three load cells and two steel plates which comprised the calibration scale.
FIGURES 3, 4 LOCATION OF THE SCALE DURING CALIBRATION OF THE AXLE
FIGURES 5, 6  VIEW OF AXLES AND WEIGH SCALE DURING CALIBRATION
VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION
FRONT DRIVE AXLE
GAUGE MONITORED: P2 Calibration: B
AIRCRAFT WEIGHING KIT

VISHAY SERIAL #: 043023
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y = MX+C where M = .238, C = .007 & R = .998
Ecal of 1.521 volts for Real of 100 kohms = 6.579 tonnes

+ --- INCREASING LOAD
* --- DECREASING LOAD
VERTICAL LOAD CALIBRATION

HENDRICKSON SUSPENSION

REAR DRIVE AXLE

GAUGE MONITERED : P8     CALIBRATION : B

AIRCRAFT WEIGHING KIT

VISHAY SERIAL # 824114
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M= .259, C= -.015 & R= .999
Ecal of 1.520 volts for Rcal of 100 kohms = 5.930 tonnes

+ --- INCREASING LOAD
* --- DECREASING LOAD
VERTICAL LOAD CALIBRATION

NEWAY SUSPENSION

TRAILER AXLE # 1  GAUGE MONITERED : APS2

AIRCRAFT WEIGHING KIT  CALIBRATION : B

VISHAY SERIAL # 042954
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y = MX + C where M =  .467, C =  .810 & R=1.000
Ecal of 1.540 volts for Real of 100 kohms = 3.279 tonnes

-- INCREASING LOAD
× -- DECREASING LOAD
VERTICAL LOAD CALIBRATION

NEWAY SUSPENSION
TRAILER AXLE # 2
GAUGE MONITORED : BPS2  CALIBRATION : A
VISHAY SERIAL # 024098
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M = .475, C = -.226 & R = .998
Ecal of 1.555 volts for Read of 100 kohms = 3.330 tonnes

--- INCREASING LOAD
--- DECREASING LOAD
VERTICAL LOAD CALIBRATION

NEWAY SUSPENSION
TRAILER AXLE # 3
GAUGE MONITORED : CPS2  CALIBRATION : A
AIRCRAFT WEIGHING KIT

VISHAY SERIAL # 024155
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best fit for \( Y = MX + C \) where \( M = 0.455, \ C = 0.047 \) & \( R=1.000 \)

Ecal of 1.548 volts for Ecal of 100 kohms = 3.211 tonnes

+ --- INCREASING LOAD
* --- DECREASING LOAD
VERTICAL LOAD CALIBRATION

REYCO TANDEM LEAD AXLE  4 SPRING SUSPENSION
TRAILER AXLE #2, PS
GAUGE MONITORED: BPS2  CALIBRATION: A
AIRCRAFT WEIGHING KIT

VISHAY SERIAL #: 045312
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for \( y = mx + c \) where \( m = 0.325 \), \( c = 0.020 \) & \( R = 0.999 \)

Eps: of 1.523 Volts for Rcal of 100 k\( \Omega \) = 2.862 tonnes

+ -- INCREASING LOAD
* -- DECREASING LOAD
• -- INCREASING LOAD
○ -- DECREASING LOAD
VERTICAL LOAD CALIBRATION

REYCO TANDEM TRAIL AXLE 4 SPRING SUSPENSION
TRAILER AXLE #3, P5
GAUGE MONITORED: CPS2 CALIBRATION: B
AIRCRAFT WEIGHING KIT

VISHAY SERIAL #: 045312
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for \( y = mx + c \) where \( m = 0.500 \), \( c = 0.230 \) & \( R^2 = 0.989 \)

Ecal of 1.542 Volts for Rc of 100 kOhms = 3.025 tonnes

+ -- INCREASING LOAD
* -- DECREASING LOAD
VERTICAL LOAD CALIBRATION

CHALMERS TANDEM TRAIL AXLE
TRAILER AXLE #3, PS
GAUGE MONITORED: CPS2, CALIBRATION: A
AIRCRAFT WEIGHING KIT
VISHAY SERIAL # 824155
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M = .473, C = .056 & R = .998
Ecal of 1.570 Volts for Rcal of 100 kOhms = 3.200 tonnes

+ --- INCREASING LOAD
* --- DECREASING LOAD
VERTICAL LOAD CALIBRATION

CHALMERS TANDEM LEAD AXLE
TRAILER AXLE #2, PS
GAUGE MONITORED : BPS2   CALIBRATION : A
AIRCRAFT WEIGHING KIT

VISHAY SERIAL # 024098
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M = 0.169, C = 0.002 & R=1.000
Egal of 1.552 Volts for Real of 100 kOhms = 3.383 tonnes

* ---- INCREASING LOAD
+ ---- DECREASING LOAD

VOLTAGE OUTPUT (VOLTS)

LOAD (TONNES)
BENDING MOMENT CALIBRATION

HENDRICKSON SUSPENSION
FRONT DRIVE AXLE
GAUGE MONITORED: P2
CALIBRATION: A

VISHAY SERIAL #: 043823
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y = MX + C where M = 0.087, C = 0.289 & R = 0.995

+ -- INCREASING LOAD
* -- DECREASING LOAD

OUTPUT VOLTAGE (VOLTS)
MOMENT (kN*m)
BENDING MOMENT CALIBRATION

HENDRICKSON SUSPENSION
REAR DRIVE AXLE
GAUGE MONITORED: PS  CALIBRATION: A
AIRCRAFT WEIGHING KIT

VISHAY SERIAL #: 024114
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y=MX+C where M = .127, C = -.001 & R = .999

+ -- INCREASING LOAD
* -- DECREASING LOAD
BENDING MOMENT CALIBRATION

NEWAY SUSPENSION
TRAILER AXLE #1
GAUGE MONITORED: APS2 CALIBRATION: A
VISHAY SERIAL #: 042954
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y = MX + C where M = .137, C = .005 & R = .999

+ --- INCREASING LOAD
* --- DECREASING LOAD
BENDING MOMENT CALIBRATION

NEWAY SUSPENSION
TRAILER AXLE 2
GAUGE MONITORED: 3PS2  CALIBRATION: A

VISHAY SERIAL: 024098
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y = MX + C where M = 0.114, C = 0.026 & R = 0.952

+ -- INCREASING LOAD
* -- DECREASING LOAD
BENDING MOMENT CALIBRATION

NEWAY SUSPENSION
TRAILER AXLE #3
GAUGE MONITORED: CPS2
CALIBRATION: A

VISHAY SERIAL #: 024155
GAIN = 1000
EXCITATION VOLTAGE = 5V

Best Fit for Y = MX + C where M = .151, C = .015 & R = .997

+ --- INCREASING LOAD
* --- DECREASING LOAD
APPENDIX C

ROAD ROUGHNESS REPORT
APPENDIX

ROUGHNESS MEASUREMENTS

By

A.T. Papagianakis * R.C.G. Haas ** T. Khan *** F. Marciello ****

* Ph.D. student, Dep. of Civil Eng., U. of Waterloo, N2L 3G1
** Professor, Dep. of Civil Eng., U. of Waterloo, N2L 3G1
*** Pavement Research Technician, MTCO, Downsview, Ont., M3M 3E6
**** Highway Design Technician, MTCO, Downsview, Ont., M3M 3E6

Acknowledgment: Thanks are expressed to Mr. J.R.F. Woodroffe of the National Research Council for his cooperation. Technical support for the roughness measurements was provided by the Ministry of Transportation and Communications of Ontario. Special thanks are addressed to Mr. W.A. Phang for his assistance with the project.
TABLE OF CONTENTS

1. Content Description .......................... 1

2. Mays-Meter Roughness Measurements .... 3
   2.1 Woodroffe Avenue ...................... 3
   2.2 Uplands Road ......................... 5
   2.3 Highway 417 .......................... 7

3. Rod-and-Level Measurements ............... 10
1. Content Description

This appendix presents pavement roughness data for three street sections in the city of Ottawa. The National Research Council uses the three sections for its dynamic load measurements obtained by a specially instrumented vehicle. The instrumentation was developed as part of the Heavy Vehicles Weights and Dimensions Study carried-out under the auspices of the RTAG. The three sections tested are:

(i) Woodrofe Avenue from the CNR trucks 1.7 Km to the Slack Road approach sign
(ii) Uplands Road from the International Airport sign 1.19 Km Northly to a drainage ditch
(iii) Highway 417 from the Maitland Road overpass 5.0 Km easterly to the Preston Boulevard intersection

Two methods were used for measuring pavement roughness. First, a Mays-meter device was used to obtain a response-type statistic expressed in the form of inches/mile. This reflects the sum of the vertical displacement of the axle of a standardized trailer with respect to its frame. Measurements were obtained for the full length of all three sections.

Second, true pavement profile was obtained for the first 320 meters of the first two sections by a rod-and-level survey. A sampling interval of 0.5 meters was used. A self-leveling automatic level was used, generating a reference plane by a single laser beam rotating at 300 rpm. The rod contains a sensor that automatically locks into the laser beam.
The roughness measurements described in the following sections are expected to allow a direct comparison between dynamic load on the pavement and roughness profile. This is a pilot study of a project attempting to relate pavement roughness with dynamic vehicle loads. The goal of the project is to develop damage functions to be used in Pavement Management. This project is related to work carried-out by the first of the authors as part of his Ph.D. thesis.

The following sections include the roughness data organized in two parts. First, the output of the Mays-meter is presented and second, the true profile elevations are shown.
2. Mayo-meter Roughness Measurements

The following sections present the roughness statistics obtained by the Mayo-meter.

2.1 Woodroffe Avenue

<table>
<thead>
<tr>
<th>Distance (mi)</th>
<th>Landmarks</th>
<th>Roughness (I.P.M.)</th>
<th>Profile</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>POSTED 80Km/h SIGN</td>
<td>55</td>
<td>*****</td>
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<td>0.05</td>
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<td>66</td>
<td>*****</td>
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<td>0.13 STN O+000 (CNR)</td>
<td>252</td>
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<td>76</td>
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<tr>
<td>1.05</td>
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REMARKS:
### 2.2 Uplands Road

<table>
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<tr>
<th>Distance (mi)</th>
<th>Landmarks</th>
<th>Roughness (I.P.M.)</th>
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</table>

**Remarks:** Profiles for areas exceeding 350IPM not plotted.
### 2.3 Highway 417

DISTRICT: 9  
Hwy: 417  
Lane: 2  
Dir: EB  
Cont/Wp: 83-22/84-69

Location: Maitland Rd. E'ly to Preston Br.

From Lhrs/O.S. to Lhrs/O.S.  
Length Avg. IPM = 106  
Range: 32-297  
49490/0.0 49450/0.1  
5.0km Std. Dev. = 60.1  
Date: 86 04 03

<table>
<thead>
<tr>
<th>Distance (mi)</th>
<th>Landmarks</th>
<th>Roughness (I.P.M.)</th>
<th>Profile</th>
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</thead>
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<tr>
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<tr>
<td>0.10</td>
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<td>198</td>
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<td>0.15</td>
<td></td>
<td>131</td>
<td>*<em>:</em></td>
</tr>
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<td>0.25</td>
<td></td>
<td>289</td>
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<td>*<em>:</em></td>
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<td>*<em>:</em></td>
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<td>77</td>
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APPENDIX D

A LISTING OF SOFTWARE DEVELOPED FOR THE
ANALYSIS OF DYNAMIC DATA

The interactive program converts analog signals to digital values, applies calibration factors, performs the statistical analysis, stores data and plots the distribution.
Program VW

implicit integer*2 (a-z)
integer*4 size,intary(4000)
dimension nochan(3),time(3),rate(3)

common/gracom/size,intary

size=4000

50 print 1
1 format(/' main menu'/)

print 2
2 format(/' 1- convert analog signal 4- decalibrate'/)

print 3
3 format(/' 2- plot digitized signal 5- plot distribution'/)

print 4
4 format(/' 3- statistical analysis 6- store data'/)

print 7
7 format(/' 7- exit VW'/)

print 5
5 format(/' Enter code number : '/)
read 6,code

format(11)
if(code.eq.1)call convert(nochan,time,rate)
if(code.eq.2)call plotsign(nochan,time,rate,code)
if(code.eq.3)call stats(nochan,time,rate)
if(code.eq.4)call decal(nochan,time,rate)
if(code.eq.5)call plotdist(nochan,time,rate,code)
if(code.eq.6)call store(nochan,time,rate)
if(code.eq.7)goto 51

51 stop
end
subroutine convert(nochan,time,rate)

implicit integer*2 (a-z)
dimension tape(8),comp(8),vector1(30000),vector2(30000)
dimension vector3(1000),nochan(3),time(3),rate(3)
character*1 ans1,yes,ans2,ans3
integer*4 count1,count2,count3,count4,count5,r
common/comp/tape
common/conv/vector1,vector2

data yes,'y'/
do 56 i=1,3
nochan(i)=0
time(i)=0
r=0
rate(i)=0
56 call comptape
print 1
1 format('Enter first and last computer channels to be converted : ')
read 2,chanl01,chanh11
2 format(11,1x,11)
nochan(1)=chanh11-chanl01+1
print 3
3 format('Enter A to D converting time in tenths of seconds : ')
read 4,time(1)
4 format(13)
print 5
5 format('Enter A to D converting rate in counts per second : ')
read 6,rate(1)
6 format(14)
print 11
11 format('Is there a second vector to be generated? ')
read 15,ans1
15 format(a1)
if(ans1.eq.,yes)then
print 1
read 2,chanl02,chanh12
nochan(2)=chanh12-chanl02+1
print 2
read 4,time(2)
print 5
read 6,rate(2)
print 16
16 format('Is there a time lapse between the 2 vectors? ')
read 17,ans2
17 format(a1)
if(ans2.eq.,yes)then
print 18
18 format('Enter time lapse in tenths of seconds : ')

read 19, time(3)
format(13)
adapt3=0
device3=9
chanlo3=0
channh13=3
nochan(3)=4
rate(3)=10
points3=time(3)
ctrl3=0
mode3=0
stor3=0
count3=time(3)
rate43=rate(3)
stat3=0
dendif
if(ans2.ne..yes)then
print 20
format('Do you want to link vector1 with vector2?')
read 21, ans3
format(11)
if(ans3.eq..yes) time1=time(1)+time(2)
edendif
dendif
adapt1=0
adapt2=0
device1=9
device2=9
ctrl1=0
ctrl2=0
model=0
mode2=0
stor1=0
stor2=0
points1=rate(1)/10*nochan(1)*time(1)
points2=rate(2)/10*nochan(2)*time(2)
count1=rate(1)/10*time(1)
count2=rate(2)/10*time(2)
rate41=rate(1)
rate42=rate(2)
stat1=0
stat2=0
call ainsc(adapt1,device1,chanlo1,channh11,ctrl1,model,
*stor1,count1,rate41,vector1,stat1)
if(stat1.ne.0) goto 100
if(ans1.eq..yes and ans2.eq..yes) call ainsc(adapt3,device3,chanlo3,
*channh3,ctrl3,mode3,stor3,count3,rate43,vector3,stat3)
if(stat3.ne.0) goto 100
if(ans1.eq..yes) call ainsc(adapt2,device2,chanlo2,channh12,ctrl2,
*mode2,stor2,count2,rate42,vector2,stat2)
  if(stat2.ne.0) goto 100
  goto 101
100 print 7,stat1,stat2,stat3
  format(1x,'Execution errors ',3i8/)
  stop
  c
101 if(ans3.eq.1) time(1)=time1
  c
  return
  end
subroutine stats(nochan, time, rate)

implicit integer*2 (a-z)
real*4 sumv(4),sumv2(4),avg(4),sigma(4),one(4)
dimension vector1(30000),vector2(30000),tape(8)
dimension nochan(3),time(3),rate(3),prob(4,4096)
character*78 fileid(4)

common/conv/vector1,vector2
common/sta/prob,avg,sigma,code

print 1
  format(/' stats menu'
print 2
  format(/' 1- analysis of vector1   3- analysis of virtual data'
print 3
  format(/' 2- analysis of vector2   4- last menu'
print 4
  format(/' Enter code number : '
read 5,code
format(11)

if(code.eq.4)return

if(code.eq.3)then
  open(2,file='d:virtual.dat',status='old')
  read(2,14)tape
  format(8i3)
  read(2,15)nochan(1),time(1),rate(1)
  format(3i8)
  points=rate(1)/10*nochan(1)*time(1)
  if(points.gt.30000)then
    read(2,16)vector1(i),i=1,30000
    read(2,16)vector2(i),i=30001,points=30000
  endif
  if(points.le.30000)read(2,16)vector1(i),i=1,points
  format(8i8)
  read(2,17)fileid
  format(a78)
  close(2,status='keep')
  print 17,fileid
endif

calculate average and standard deviation

c
if(code.eq.1.or.code.eq.3)then
  nochan=nochan(1)
  rat=rate(1)
  tim=time(1)
endif
if(code.eq.2)then
  nochan=nochan(2)
rat = rate(2)
tim = time(2)
endif
do 50  i=1,noch
do 52  j=1,4096
52   prob(1,j) = 0
     one(i) = 0.
     sumv(1) = 0.
50   sumv2(1) = 0.
points = rat/10*noch*tim
if(code .eq. 2) goto 100
if(points .gt. 30000) then
do 59  k=1,30000-nochan(1)+1,nochan(1)
do 60  l=1,nochan(1)
ichan = k+1-1
sumv(1) = 1.0*vector1(ichan) + sumv(1)
sumv2(1) = 1.0*vector1(ichan) * vector1(ichan) + sumv2(1)
bin = vector1(ichan) + 1
prob(1,bin) = prob(1,bin) + 1
59   continue
58  p = points - 30000-nochan(1)+1
      do 58  k=1,p,nochan(1)
do 65  l=1,nochan(1)
ichan = k+1-1
sumv(1) = 1.0*vector2(ichan) + sumv(1)
sumv2(1) = 1.0*vector2(ichan) * vector2(ichan) + sumv2(1)
bin = vector1(ichan) + 1
prob(1,bin) = prob(1,bin) + 1
65   continue
endif
if(points .le. 30000) then
do 66  k=1,points-nochan(1)+1,nochan(1)
do 67  l=1,nochan(1)
ichan = k+1-1
sumv(1) = 1.0*vector1(ichan) + sumv(1)
sumv2(1) = 1.0*vector1(ichan) * vector1(ichan) + sumv2(1)
bin = vector1(ichan) + 1
prob(1,bin) = prob(1,bin) + 1
67   continue
endif
100 if(code .eq. 2) then
      do 68  k=1,points-nochan(2)+1,nochan(2)
do 69  l=1,nochan(2)
ichan = k+1-1
sumv(1) = 1.0*vector2(ichan) + sumv(1)
sumv2(1) = 1.0*vector2(ichan) * vector2(ichan) + sumv2(1)
bin = vector2(ichan) + 1
prob(1,bin) = prob(1,bin) + 1
69   continue
endif
do 53  i=1,noch
avg(i) = sumv(1) * 10./rat/tim
\[
\sigma(i) = \text{sumv2}(i) * 10. / \text{rat} / \text{tim} - \text{avg}(i) ** 2 \times \text{sumv2}(i) / \text{tim} - \text{avg}(i) ** 2 \times \text{sumv2}(i) / \text{tim}
\]

Evaluate probability for given voltage bins

\[
\text{do } 63 \ i = 1, \text{noch}
\text{do } 64 \ j = 1, 4096
\text{one}(i) = \text{one}(i) + \text{prob}(i, j) * 10. / \text{tim} / \text{rat}
\text{continue}
\]

Print statistical output

\[
\text{print } 6, (\text{avg}(i), i = 1, \text{noch})
\text{format(' average = ', 8f8.2)}
\text{print } 7, (\sigma(i), i = 1, \text{noch})
\text{format(' stand. dev. = ', 8f8.2)}
\text{print } 8, (\text{one}(i), i = 1, \text{noch})
\text{format(' integral = ', 8f8.4)}
\]

\[
\text{return}
\text{end}
\]
subsection decal(nochan,time,rate)

implicit integer*4 (a-z)
real*4 slope,constant,group,prob(4,100),eng(4,100),avgeng(4)
real*4 avg(4),sigma(4),percent,one(4),probitr(4096)
real*4 static(4),dic(4),percent5(4),percent95(4)
character*7 cal(14),dec
character*8 filename
character*12 filenamext
character*78 fileid(4)
dimension tape(8),comp(8),nochan(3),time(3),rate(3)
dimension probbit(4,4096)

common/dec/eng,prob
common/comp/tape
common/sta/probitr,avg,sigma,vect

data cal/’c01.dec’,’c02.dec’,’c03.dec’,’c04.dec’,’c05.dec’,
* ’c06.dec’,’c07.dec’,’c08.dec’,’c09.dec’,’c10.dec’,
* ’c11.dec’,’c12.dec’,’c13.dec’,’c14.dec’/

print 1
format(’decal menu’)
print 2
format(’1-decalibrate RAM data 3-last menu’)
print 3
format(’2-decalibrate raw data’)
print 4
format(’Enter code number: ’)
read 5,code
format(1)
if(code.eq.3) return
if(code.eq.1) then
if(vect.eq.1.or.vect.eq.3) then
nochan=nochan(1)
tim=time(1)
rat=rate(1)
endif
if(vect.eq.2) then
nochan=nochan(2)
tim=time(2)
rat=rate(2)
endif
endif

Read statistical data file (*.raw)

c if(code.eq.2) then
print 12
format(’Enter raw data filename: ’)
read 13,filename
format(a8)
filenamext=filename/’.aw’
print 27, filename
format(1x,312)
open(1, file=filename, status='old')
read(1,25) fileid
format(a78)
print 25, fileid
read(1,23) tape
format(b13)
read(1,24) noch, tim, rat
format(s18)
read(1,19) (avg(1), i=1, noch)
format(16x,8f8.2)
read(1,20) (sigma(i), i=1, noch)
format(16x,8f8.2)
read(1,22) (probit(i,k), k=1,4096, i=1, noch)
format(1216)
close(1, status='keep')
ochan(1)=noch
tim(1)=tim
rate(1)=rat
endif

call comptape
open(1, file='stat', status='unknown')
do 54 j=1, noch
do 50 i=1, 4096
probitr(i)=probit(j,1)*10./tim/rat

c Read calibration constants

c dec=cal(tape(j))
open(2, file=dec, status='old')
read(2,9) slope, constant, group
format(3f8.3)
close(2, status='keep')

c Decalibrate

percent=0.
percent5(j)=0.
percent95(j)=0.
one(j)=0.
do 55 i=1, 100
prob(j,i)=0.
r1=-50.*2048./slope/5.+avg(j)
probability=0.
do 53 i=1, 100
r2=(-50.+1.)*2048./slope/5.+avg(j)
if(r2.lt.1) goto 101
if(r1.lt.1 and r2.ge.1) r1=1
if(r1.gt.4096) goto 101
if(r1.lt.4096 and r2.gt.4096) r2=4096
do 51 k=r1,r2
one(j)=one(j)+probit(j,k)
prob(j,i)=prob(j,i)+probitr(k)
prob(j,i)=prob(j,i)*100.
r1=r2+1
eng(j,i)=-50.5+i
percent=percent+prob(j,i)
if(percent.gt.5.0.and.percent5(j).eq.0.0)then
  percent5(j)=eng(j,i)-(percent-5.)/prob(j,i)
endif
if(percent.gt.95.0.and.percent95(j).eq.0.0)then
  percent95(j)=eng(j,i)-(percent-95.)/prob(j,i)
endif
one(j)=one(j)*10/tim/rat*100.
avgeng(j)=0.0
dlc(j)=slope*sigma(j)*5./2048.
static(j)=constant
c File statistical analysis results
c write(1,17) dlc(j)
write(1,18) percent5(j)
write(1,26) percent95(j)
c continue
c close(1,status='keep')
c Print statistical analysis results
print 31,avgeng
format(  mean = 'f6.2, kN'
15 format('  mean = ' ,4(2x,f6.2), ' kN')
print 33,dlc
format('  standard deviation = ' ,f6.2, ' kN')
33 format('  standard deviation = ' ,4(2x,f6.2), ' kN')
print 37,percent5
format('  5th percentile = ' ,f6.2, ' kN')
37 format('  5th percentile = ' ,4(2x,f6.2), ' kN')
print 38,percent95
format('  95th percentile = ' ,f6.2, ' kN')
38 format('  95th percentile = ' ,4(2x,f6.2), ' kN')
print 36,one
format('  one = ' ,4(3x,f5.1))
c return
end
subroutine plottidm(nochan,time,rate)

implicit integer*2 (a-z)
integer*4 size,intary(4000)
real*4 echpt(2),spread
real*4 load(4,100),prob(4,100),yalign,ld(100),pb(100)
character*1 dis
character*4 lift(2)
character*10 axle(2),run
character*25 force(2),string
character*30 suspension(4)
character*37 results(10),ident(10)
dimension post(4),susp(4),nochan(3),time(3),rate(3)

c

common/gracom/size,intary
common/dec/load,prob
c
data echpt/0.0,0.0/

53 print 15
15 format(’Enter $ for screen display or H for hard copy : ’)
read 10,dis
16 format(a1)
if(dis.ne.’H’.and.dis.ne.’h’.and.dis.ne.’S’.and.dis.ne.’s’)then
print 17
17 format(’Device undefined; make other selection’)  
go to 53
endif
if(dis.eq.’S’.or.dis.eq.’s’)device=1
if(dis.eq.’H’.or.dis.eq.’h’)then
device=2
open(1,file=’runpar’,status=’old’)
read(1,43)suspension
43 format(a20)
read(1,44)axle
44 format(a10)
read(1,45)force
45 format(a25)
read(1,46)lift
46 format(a4)
close(1,status=’keep’)  
print 3
3 format(’GRAPH IDENTIFICATION’)//
print 19
19 format(’Enter run # in A10 Enter speed in I2’)  
print 20,nochan(1),nochan(1)
20 format(’i1,i1,’ Axle positions i1,i1,’ Suspension types’)  
print 21
21 format(’1 – lead axle 1 – four spring – trailer’)  
print 22
22 format(’2 – trail axle 2 – walking beam – spring,drive’)  
print 23
23 format(’3 – walking beam – rubber, trailer’)
print 24
format( ',4-- air suspension - trailer')
print 25
format( ', MAYS roughness in I3 Inertial forces')
print 26
format( ', 1-- with')
print 27
format( ', 2-- without')
print 28
format( ', Enter spread in F4.2 Air suspension')
print 29
format( ', 1-- up')
print 30
format( ', 2-- down')
print 31
format( ', Enter graph identification in sequential form : ')
read 32, run, speed, post, susp, surf, inert, spread, air
format(a10, i12, i12, B(i1, i1), i1, i1, i1, f4.2, i1, i1)
open (1, file='id', status='unknown')
do 56 i=1, nochan(1)
write(1,33) run
format( ', Run # ', a10)
write(1,34) speed
format( ', Speed = ', i2, ' km/hr')
write(1,35) surf
format( ', MAYS roughness = ', i3, ' kPM')
write(1,36)
format( ', Suspension : ')
write(1,36) suspension(susp(1))
format(3x, a30)
write(1,38) axle(post(i))
format(1x, a10)
write(1,39) force(inert)
format(1x, a25)
write(1,40) spread
format( ', Spread = ', f4.2, ' meters')
write(1,41) lift(air)
format( ', Air suspension lift axle ', a4)
continue
56 close(1, status='keep')
endif
open(1, file='stat', status='old')
open(2, file='id', status='old')
do 57 ii=1, nochan(1)
c
Open plotting system
c  status=popnps()
c
Assign plotting system output device
c  if(device.eq.1) status=ppset('display')
if(xeq(2))status=psplot('printer')

Read statistical analysis results file
read(1,18) (results(j),j=1,3)
format(a37)
if(xeq(2))then
read(2,18) (ident(j),j=1,9)
endif

Set display surface size
status=pssurf(13.7,9.1)

Define x-axis title string
status=ptxt(1,'dynamic loading (kN)')

Define y-axis title string
status=ptxt(2,'frequency of occurrence')

Set existence of chart & view area frame
status=pckfrm(1,1)
status=pavfrm(0,1)

Set axes extents
status=pxext(1,-50.,50.,10.)
status=pxext(2,0.,20.,2.)

Set axis tick label type and height
status=pxalty(1,1,2,1)
status=ptangt(1,1)
status=ptangt(2,1)

Define x/y data set
do 59 k=1,100
  ld(k)=load(k)
  pb(k)=prob(k)
  status=pdsxy(2,100,ld,pb)
59

Define data set to be a bar
status=pbar(2)

Set output primitives : data set 1 to solid bar and color
status=pdsstl(2,2)
status=pdsclr(2,3)
c Set notation height
  status=psnhgt(1)
c Define notation strings
do 54 k=1,3
   yalign=90.-2.0*(k-1.0)
  status=pnote(1,48.,yalign,results(k))
do 55 k=1,9
   yalign=90.-2.*(k-1.)
  status=pnote(1,9.5,yalign,ident(k))
c Output currently defined chart
  status=ppltit()
c View graph
  if(device.eq.1)status=ppsin('display')
  if(device.eq.2)status=ppsin('printer')
  status=prqst(1,1,echpt,26,string)
c Close I/O device
  if(device.eq.1)status=ppsd('display')
  if(device.eq.2)status=ppsd('printer')
c Reset defaults and viewing area
  status=pdeflt()
  status=pravw()
c Close plotting system
  status=pclsps()
c7 continue
  close(1,status='keep')
  close(2,status='keep')
c    return
end
subroutine store(nochan, time, rate)

implicit integer*2 (a-z)
real*4 average(4), sigma(4), eng(4,100), rprob(4,100)
character*1 ans, yes
character*8 filename
character*12 filenamext
character*40 stat(20)
character*70 fileid(4)
dimension tape(8), comp(8), prob(4,4096)
dimension nochan(3), time(3), rate(3)

common/dec/eng, rprob
common/comp/tape
common/stat/prob, average, sigma, vect

data yes/'y'/
print 9
format(/' store menu'/)
print 10
format(/' 1- store raw data 3- store raw and statistical data'/)
print 11
format(/' 2- store statistical data 4- last menu'/)
print 12
format(/' Enter code number : ')
read 13, code
format(11)

if(code.eq.4) return
if(vect.eq.0.or.vect.eq.1.or.vect.eq.3) then
nochan=nochan(1)
time=time(1)
rate=rate(1)
endif
if(vect.eq.2) then
nochan=nochan(2)
time=time(2)
rate=rate(2)
endif

sumtpe=0
do 50 i=1,8
sumtpe=sumtpe+tape(i)
do 50
if(sumtpe.eq.0) then
open(1, file='comptape.dat', status='old')
read(1,14) (comp(i), tape(i), i=1,8)
format(16i3)
close(1, status='keep')
endif

if(code.eq.1.or.code.eq.3) then
print 15
format(' Enter file name (up to 8 characters) : ')
read 16,filename
format(a8)
print 17
format(' Enter file identification (4 lines of 70 characters) : ')
read 18,fileid
format(a70)
filename=fileid//'.raw'
open(1,filenamext,status='new')
write(1,7) fileid
format(' 1.D. : ','a70,3(/8x,a70))
write(1,8) tape
format(b13)
write(1,3) noch, tim, rat
format(b18)
write(1,19) (average(i),i=1,noch)
format('  average = ','8f8.2')
write(1,20) (sigma(i),i=1,noch)
format('  stand. dev. = ','8f8.2')
write(1,21) ((prob(i,k),k=1,4096),i=1,noch)
format(d16)
close(1,status='keep')
endf
if(code.eq.2.or.code.eq.3)then
if(code.eq.2)then
print 15
read 16,filename
print 27, filename
format(' Does data file ',a8,'.raw exist ? ')
read 24, ans
format(a1)
if(ans.eq.yes)then
open(1,filenamext//'.raw',status='old')
read(1,25) fileid
format(8x,a70,3(/8x,a70))
close(1,status='keep')
endif
if(ans.ne.yes)then
print 17
read 18,fileid
endif
open(1,filenamext//'.stat',status='unknown')
read(1,26) (stat(i),i=1,3*noch)
format(a40)
close(1,status='keep')
open(1,filenamext//'.sta',status='new')
write(1,7) fileid
write(1,8) tape
write(1,3) noch, tim, rat
write(1,26) (stat(i),i=1,3*noch)
write(1,27) (eng(i,1),i=1,100)
write(1,27)((rprob(1,j),j=1,100),i=1,noch)
format(3f8.3)
close(1,status='keep')
endif
c
return
der
subroutine comptape

implicit integer*2 (a-z)
dimension comp(8), tape(8)

common/comp/tape

print 7, computer-tape recorder channel number corresponds
*denote *
open(1, file='comptape.dat', status='old')
read(1,8) (comp(i), tape(i), i=1,8)

format(16i2)
close(1, status='keep')
print 9, comp

format(' computer channel number', 8i1)
print 10, tape

format(' tape recorder channel number', 8i1)

return
end
APPENDIX E

PLOTS OF WHEEL LOAD VS RIDE COMFORT RATING FOR VARIOUS SUSPENSION COMBINATIONS
VARIATION OF TRAILER SUSPENSION TYPE
TRACTOR SUSPENSION - HENDRICKSON
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN
NOMINAL TRAILER AXLE SPREAD OF 1.3 M

TRACTOR RESPONSE

TRAILER RESPONSE
VARIATION OF TRAILER SUSPENSION TYPE
TRACTOR SUSPENSION - HENDRICKSON
AIR AXLE DOWN - NOMINAL WHEEL LOAD OF 40 KN
NOMINAL TRAILER AXLE SPREAD OF 1.3 M

TRACTOR RESPONSE

TRAILER RESPONSE
VARIATION OF TRACTOR SUSPENSION TYPE
TRAILER SUSPENSION - CHALMERS
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN
NOMINAL TRAILER AXLE SPREAD OF 1.3 M

TRACTOR RESPONSE

TRAILER RESPONSE
VARIATION OF TRACTOR SUSPENSION TYPE
TRAILER SUSPENSION - CHALMERS
AIR AXLE DOWN - NOMINAL WHEEL LOAD OF 40 KN
NOMINAL TRAILER AXLE SPREAD OF 1.3 M

TRACTOR RESPONSE

TRAILER RESPONSE
VARIATION OF TRAILER AXLE SPREAD
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

TRACTOR RESPONSE

TRAILER RESPONSE
VARIATION OF TRAILER AXLE SPREAD
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAI
AIR AXLE DOWN - NOMINAL WHEEL LOAD OF 40 KN

TRAILER RESPONSE

TRACTOR RESPONSE
VARIATION OF TRAILER AXLE SPREAD
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - REYCO
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

TRACTOR RESPONSE

TRAILER RESPONSE
EFFECT OF AIR SUSPENSION LIFT AXLE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - REYCO, AXLE SPREAD OF 1.27 M

TRACTOR RESPONSE

TRAILER RESPONSE
EFFECT OF AIR SUSPENSION LIFT AXLE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 1.27 M

TRACTION RESPONSE

<table>
<thead>
<tr>
<th>AIR AXLE POSITION</th>
<th>Ride Comfort Rating, RCR</th>
</tr>
</thead>
<tbody>
<tr>
<td>AIR AXLE UP, 80 km/hr</td>
<td>20.0</td>
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<tr>
<td>AIR AXLE DOWN, 60 km/hr</td>
<td>16.0</td>
</tr>
<tr>
<td>AIR AXLE UP, 60 km/hr</td>
<td>12.0</td>
</tr>
<tr>
<td>AIR AXLE DOWN, 60 km/hr</td>
<td>8.0</td>
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<tr>
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<td>4.0</td>
</tr>
<tr>
<td>AIR AXLE DOWN, 40 km/hr</td>
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</table>

TRAILED RESPONSE

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<th>AIR AXLE POSITION</th>
<th>Ride Comfort Rating, RCR</th>
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</thead>
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<tr>
<td>AIR AXLE UP, 80 km/hr</td>
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<tr>
<td>AIR AXLE DOWN, 60 km/hr</td>
<td>16.0</td>
</tr>
<tr>
<td>AIR AXLE UP, 60 km/hr</td>
<td>12.0</td>
</tr>
<tr>
<td>AIR AXLE DOWN, 60 km/hr</td>
<td>8.0</td>
</tr>
<tr>
<td>AIR AXLE UP, 40 km/hr</td>
<td>4.0</td>
</tr>
<tr>
<td>AIR AXLE DOWN, 40 km/hr</td>
<td>2.0</td>
</tr>
</tbody>
</table>
E10

EFFECT OF AIR SUSPENSION LIFT AXLE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 1.83 M

TRACTOR RESPONSE

TRAILER RESPONSE
EFFECT OF AIR SUSPENSION LIFT AXLE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 2.44 M

TRACTOR RESPONSE

TRAILER RESPONSE

AIR AXLE POSITION
- AIR AXLE UP, 80 KM/HR
- AIR AXLE DOWN, 80 KM/HR
- AIR AXLE UP, 60 KM/HR
- AIR AXLE DOWN, 60 KM/HR
- AIR AXLE UP, 40 KM/HR
- AIR AXLE DOWN, 40 KM/HR
EFFECT OF AIR SUSPENSION LIFT AXLE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - CHALMERS, AXLE SPREAD OF 1.37 M

TRACTOR RESPONSE

TRAILER RESPONSE
EFFECT OF AIR SUSPENSION LIFT AXLE
TRACTOR SUSPENSION - NEWAY
TRAILER SUSPENSION - CHALMERS, AXLE SPREAD OF 1.37 M

**TRACTOR RESPONSE**

- AIR AXLE POSITION
- AIR AXLE UP, 80 KM/HR
- AIR AXLE DOWN, 80 KM/HR
- AIR AXLE UP, 60 KM/HR
- AIR AXLE DOWN, 60 KM/HR
- AIR AXLE UP, 40 KM/HR
- AIR AXLE DOWN, 40 KM/HR

**TRAILER RESPONSE**

- AIR AXLE POSITION
- AIR AXLE UP, 80 KM/HR
- AIR AXLE DOWN, 80 KM/HR
- AIR AXLE UP, 60 KM/HR
- AIR AXLE DOWN, 60 KM/HR
- AIR AXLE UP, 40 KM/HR
- AIR AXLE DOWN, 40 KM/HR
COMPARISON OF TRACTOR AND TRAILER RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - REYCO, AXLE SPREAD OF 1.27 M

AIR AXLE UP

AIR AXLE DOWN
COMPARISON OF TRACTOR AND TRAILER RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 1.27 M

AIR AXLE UP

AIR AXLE DOWN
COMPARISON OF TRACTOR AND TRAILER RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 1.83 M

AIR AXLE UP

AIR AXLE DOWN
COMPARISON OF TRACTOR AND TRAILER RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 2.44 M

AIR AXLE UP

AIR AXLE DOWN
COMPARISON OF TRACTOR AND TRAILER RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - CHALMERS, AXLE SPREAD OF 1.37 M

AIR AXLE UP

AIR AXLE DOWN
COMPARISON OF TRACTOR AND TRAILER RESPONSE
TRACTOR SUSPENSION - NEWAY
TRAILER SUSPENSION - CHALMERS, AXLE SPREAD OF 1.37 M

AIR AXLE UP

AIR AXLE DOWN

AXLE GROUP
- TRACTOR, 80 KN/HR
- TRAILER, 80 KN/HR
- TRACTOR, 60 KN/HR
- TRAILER, 60 KN/HR
- TRACTOR, 40 KN/HR
- TRAILER, 40 KN/HR

STANDARD DEVIATION, KN

RIDE COMFORT RATING, RCR
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - REYCO, AXLE SPREAD OF 1.27 M
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

TRACTOR RESPONSE

TRAILER RESPONSE
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE

TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - REYCO, AXLE SPREAD OF 1.27 M
AIR AXLE DOWN - NOMINAL WHEEL LOAD OF 40 KN

TRACTOR RESPONSE

AXLE POSITION
- LEAD AXLE, 80 KM/HR
- TRAILING AXLE, 80 KM/HR
- LEAD AXLE, 60 KM/HR
+ TRAILING AXLE, 60 KM/HR
X LEAD AXLE, 40 KM/HR
O TRAILING AXLE, 40 KM/HR

STANDARD DEVIATION, KN

RIDE COMFORT RATING, RCR

TRAILER RESPONSE

AXLE POSITION
- LEAD AXLE, 80 KM/HR
- TRAILING AXLE, 80 KM/HR
- LEAD AXLE, 60 KM/HR
+ TRAILING AXLE, 60 KM/HR
X LEAD AXLE, 40 KM/HR
O TRAILING AXLE, 40 KM/HR

STANDARD DEVIATION, KN

RIDE COMFORT RATING, RCR
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 1.27 M
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

TRACTOR RESPONSE

TRAILER RESPONSE
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 1.27 M
AIR AXLE DOWN - NOMINAL WHEEL LOAD OF 10 KN

TRACTOR RESPONSE

TRAILER RESPONSE
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 1.83 M
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

TRACTOR RESPONSE

TRAILER RESPONSE
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE

TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 1.83 M
AIR AXLE DOWN - NOMINAL WHEEL LOAD OF 40 KN

TRACTOR RESPONSE

AXLE POSITION

- LEAD AXLE, 80 KM/HR
- TRAILING AXLE, 80 KM/HR
- LEAD AXLE, 60 KM/HR
- TRAILING AXLE, 60 KM/HR
- LEAD AXLE, 40 KM/HR
- TRAILING AXLE, 40 KM/HR

TRAILER RESPONSE

AXLE POSITION

- LEAD AXLE, 80 KM/HR
- TRAILING AXLE, 80 KM/HR
- LEAD AXLE, 60 KM/HR
- TRAILING AXLE, 60 KM/HR
- LEAD AXLE, 40 KM/HR
- TRAILING AXLE, 40 KM/HR
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 2.44 M
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

TRACTOR RESPONSE

TRAILER RESPONSE
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - NEWAY, AXLE SPREAD OF 2.44 M
AIR AXLE DOWN - NOMINAL WHEEL LOAD OF 40 KN

TRACTOR RESPONSE

TRAILER RESPONSE
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE
TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - CHALMERS, AXLE SPREAD OF 1.37 M
AIR AXLE UP - NOMINAL WHEEL LOAD OF 50 KN

TRACTOR RESPONSE

---

TRAILER RESPONSE

---

AXLE POSITION
☐ - LEAD AXLE, 80 KM/HR
◊ - TRAILING AXLE, 80 KM/HR
△ - LEAD AXLE, 60 KM/HR
♦ - TRAILING AXLE, 60 KM/HR
☒ - LEAD AXLE, 40 KM/HR
♦ - TRAILING AXLE, 40 KM/HR
COMPARISON OF LEAD AND TRAILING AXLE RESPONSE

TRACTOR SUSPENSION - HENDRICKSON
TRAILER SUSPENSION - CHALMERS, AXLE SPREAD OF 1.37 M
AIR AXLE DOWN - NOMINAL WHEEL LOAD OF 40 KN

TRACTOR RESPONSE

TRAILER RESPONSE
APPENDIX F

HISTOGRAMS OF DIGITIZED WHEEL LOAD DATA

Each run represented in the appendix illustrates the dynamic wheel load distributions of the lead and trailing axles for both the tractor and trailer. The tables on pages F1 and F2 serve to correlate the probability distributions with points on the line graphs of Appendix E.
### Reyco - Hendrickson

<table>
<thead>
<tr>
<th>speed km/hr</th>
<th>roughness</th>
<th>lift up</th>
<th>lift down</th>
<th>lift up</th>
<th>lift down</th>
<th>lift up</th>
<th>lift down</th>
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Table F1  Pagination for Reyco distributions

### Neway - Hendrickson

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<th>speed km/hr</th>
<th>roughness</th>
<th>lift up</th>
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<th>lift down</th>
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</table>

Table F2  Pagination for Neway distributions
### Chalmers - Hendrickson

<table>
<thead>
<tr>
<th>Speed km/hr</th>
<th>Roughness (IPM - RCR)</th>
<th>Lift Up</th>
<th>Lift Down</th>
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<tbody>
<tr>
<td>40</td>
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<td>F87</td>
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<td>60</td>
<td>73 8.17 254 4.55</td>
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<tr>
<td>80</td>
<td>59 8.45 165 6.33 217 5.29</td>
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<td>F98</td>
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Table F3: Pagination for Chalmers distributions
Run = 122 Part 1

*No* roughness = 254 IPM

Trailer axle spread = 1.27 m

*No* suspension: spring suspended walking beam

*No* suspension: four spring

Air suspension lift axle up

**DYNAMIC WHEEL LOAD, kN**

- **Tractor lead axle**
  - Standard deviation = 6.36 kN
  - 5th percentile = 11.74 kN
  - 95th percentile = 10.23 kN

- **Trailer trailing axle**
  - Standard deviation = 5.00 kN
  - 5th percentile = 10.01 kN
  - 95th percentile = 9.24 kN
Run = 122 Part 2
Speed = 60 km/hr
Hogs roughness = 426 IPH
Tractor axle spread = 1.27 m
Tractor suspension: spring suspended walking beam
Trailer suspension: four spring
Air suspension lift axle up

**Tractor Lead axle**
- Standard deviation: 8.12 KN
- 50th percentile: 13.86 KN
- 95th percentile: 13.00 KN

**Tractor Trailing axle**
- Standard deviation: 7.76 KN
- 50th percentile: 13.69 KN
- 95th percentile: 12.00 KN

**Trailer Lead axle**
- Standard deviation: 6.63 KN
- 50th percentile: 12.06 KN
- 95th percentile: 10.70 KN

**Trailer Trailing axle**
- Standard deviation: 5.80 KN
- 50th percentile: 12.49 KN
- 95th percentile: 10.45 KN
Run = 119 Part 2
No. of roughness = 591 PM
Tractor axle spread = 1.27 m
Tractor suspension: spring suspended walking beam
Trailer suspension: four spring
Air suspension lift axle up

FREQUENCY OF OCCURRENCE, \( f \)

<table>
<thead>
<tr>
<th>DYNAMIC WHEEL LOAD, KN</th>
<th>FREQUENCY OF OCCURRENCE, ( f )</th>
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<tr>
<td>-40.0</td>
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<tr>
<td>-30.0</td>
<td>0.0</td>
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<tr>
<td>-20.0</td>
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<tr>
<td>40.0</td>
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</tbody>
</table>

Standard deviation = 3.20 kN
5th percentile = -3.32 kN
95th percentile = 6.13 kN
Run = 123 Part 1
Max. roughness = 254 lpm
Trolley axle spread = 1.27 m
Tractor suspension: spring suspended walking beam
Trailer suspension: four spring
Air suspension left axle down

- Upper left graph:
  - Tractor lead axle
  - Standard deviation = 6.76 kN
  - 5th percentile = 0 kN
  - 95th percentile = 7.25 kN

- Upper right graph:
  - Tractor trailing axle
  - Standard deviation = 7.48 kN
  - 5th percentile = -0.59 kN
  - 95th percentile = 6.47 kN

- Lower left graph:
  - Trolley lead axle
  - Standard deviation = 2.58 kN
  - 5th percentile = -7.45 kN
  - 95th percentile = 5.24 kN

- Lower right graph:
  - Trolley trailing axle
  - Standard deviation = 6.39 kN
  - 5th percentile = -7.45 kN
  - 95th percentile = 6.73 kN
Run = 44 Part 1
Maya roughness = 254 1PM
Trailer axle spread = 1.03 m
Trailer suspension: spring suspended walking beam
Trailer suspension: four spring
Four suspension lift axle up

Frequency of occurrence, %

DYNAMIC WHEEL LOAD, KN

Tractor load axle
standard deviation = 5.76 KN
5th percentile = -9.86 KN
95th percentile = 8.81 KN

Tractor trailing axle
standard deviation = 5.23 KN
5th percentile = -9.96 KN
95th percentile = 8.30 KN

DYNAMIC WHEEL LOAD, KN

Tractor load axle
standard deviation = 5.35 KN
5th percentile = -9.16 KN
95th percentile = 8.51 KN

Tractor trailing axle
standard deviation = 5.05 KN
5th percentile = -9.05 KN
95th percentile = 8.29 KN
Run = 46 Part 1
Speed = 80 km/hr
Roughness = 165 IPM
Trailer axle spread = 1.83 m
Tractor suspension = spring suspended walking beam
Trailer suspension = four spring
Air suspension lift axle up

![Histogram of Dynamic Wheel Load](image1)

**Trailer Lead axle**
- Standard deviation = 10.47 kN
- 5th percentile = 16.33 kN
- 95th percentile = 18.25 kN

![Histogram of Dynamic Wheel Load](image2)

**Trailer Trailing axle**
- Standard deviation = 9.83 kN
- 5th percentile = 16.04 kN
- 95th percentile = 15.35 kN

![Histogram of Dynamic Wheel Load](image3)

**Trailer Lead axle**
- Standard deviation = 8.74 kN
- 5th percentile = 15.83 kN
- 95th percentile = 13.88 kN

![Histogram of Dynamic Wheel Load](image4)

**Trailer Trailing axle**
- Standard deviation = 12.03 kN
- 5th percentile = 28.81 kN
- 95th percentile = 16.05 kN

Noise on channel
Run # 99 Part 1
Speed = 80 km/hr
Maze roughness = 165 JPM
Tractor axle spread = 2.44 m
Tractor suspension: spring suspended walking beam
Trailer suspension: four spring
Air suspension lift axle up

Frequency of Occurrence, %

Dynamic Wheel Load, kN

Tractor Lead axle
Standard deviation = 8.33 kN
5th percentile = 13.83 kN
95th percentile = 13.67 kN

Tractor Trailing axle
Standard deviation = 8.73 kN
5th percentile = -13.83 kN
95th percentile = 12.37 kN

Trailer Lead axle
Standard deviation = 2.45 kN
5th percentile = -11.47 kN
95th percentile = 10.36 kN

Trailer Trailing axle
Standard deviation = 3.68 kN
5th percentile = -13.88 kN
95th percentile = 10.24 kN
Run = 100 part 1, Speed = 80 km/hr
Hays roughness = 165 JPM, Tractor axle spread = 2.44 m
Tractor suspension = spring suspended walking beam
Trailer suspension = four spring air suspension lift axle down

Graph 1: Tractor Lead axle
- Standard deviation = 0.09 kN
- 5th percentile = -15.17 kN
- 95th percentile = 12.47 kN

Graph 2: Tractor trailing axle
- Standard deviation = 8.59 kN
- 5th percentile = -13.06 kN
- 95th percentile = 13.52 kN

Graph 3: Trailer leading axle
- Standard deviation = 6.53 kN
- 5th percentile = -19.12 kN
- 95th percentile = 19.03 kN

Graph 4: Trailer trailing axle
- Standard deviation = 2.29 kN
- 5th percentile = -9.47 kN
- 95th percentile = 7.9 kN
Run = 170  
Speed = 40 km/hr  
Hay roughness = 73 JPM  
Trailer axle spread = 1.27 m  
Tractor suspension = spring suspended walking beam  
Trailer suspension = air bags  
Air suspension lift axle up

Tractor lead axle  
standard deviation = 2.70 kN  
5th percentile = -5.08 kN  
95th percentile = 3.87 kN

Tractor trailing axle  
standard deviation = 2.73 kN  
5th percentile = -6.11 kN  
95th percentile = 4.65 kN

Trailer lead axle  
standard deviation = 1.83 kN  
5th percentile = -3.66 kN  
95th percentile = 2.47 kN

Trailer trailing axle  
standard deviation = 1.85 kN  
5th percentile = -3.38 kN  
95th percentile = 2.86 kN

Frequency of Occurrence, %

Dynamic Wheel Load, kN
Run = 154 Part 2

Speed = 40 km/hr

Hay roughness = 424 JPH

Tractor axle spread = 1.27 m

Tractor suspension: spring suspended walking beam

Trailer suspension: air bags

Air suspension lift axle up

---

Frequency of occurrence, %

Dynamic wheel load, kN

---

Tractor lead axle

Standard deviation = 5.42 kN

5th percentile = -10.24 kN

95th percentile = 10.24 kN

Trailer trailing axle

Standard deviation = 5.43 kN

5th percentile = -10.64 kN

95th percentile = 10.64 kN

---

Frequency of occurrence, %

Dynamic wheel load, kN

---

Tractor lead axle

Standard deviation = 6.99 kN

5th percentile = -6.36 kN

95th percentile = 6.36 kN

Trailer trailing axle

Standard deviation = 6.97 kN

5th percentile = -6.97 kN

95th percentile = 6.97 kN

---

Frequency of occurrence, %

Dynamic wheel load, kN

---

Tractor lead axle

Standard deviation = 6.31 kN

5th percentile = -6.31 kN

95th percentile = 6.31 kN

Trailer trailing axle

Standard deviation = 7.36 kN

5th percentile = -7.36 kN

95th percentile = 7.36 kN

---
Run = 157 Part 2
Speed = 60 km/hr
Nursa roughness = 424 I.P.H.
Tractor axle spread = 1.27 m
Tractor suspension: spring suspended walking beam
Air suspension: lift axle up

Tractor Lead axle:
standard deviation = 7.06 kN
5th percentile = -13.58 kN
95th percentile = 10.49 kN

Tractor Lead axle:
standard deviation = 8.00 kN
5th percentile = -12.70 kN
95th percentile = 9.70 kN

Tractor Lead axle:
standard deviation = 5.73 kN
5th percentile = -11.03 kN
95th percentile = 10.56 kN

Tractor Lead axle:
standard deviation = 5.86 kN
5th percentile = -10.61 kN
95th percentile = 8.93 kN
Run = 175  
Speed = 40 km/hr  
Mega roughness = 73 m/m  
Trailer axle spread = 1.27 m  
Trailer suspension: spring suspended walking beam  
Trailer suspension: air bags  
Air suspension lift axle down

![Graphs showing dynamic wheel load distribution.](image)
Run = 176 Part 3  
Speed = 80 km/hr  
Rays roughness = 217 RPM  
Tractor axle spread = 1.27 m  
Tractor suspension: spring suspended walking beam  
Tractor suspension: air bags  
Air suspension lift axle down  

![Dynamic Wheel Load Distribution](image1)

- **Tractor Lead axle**
  - Standard deviation = 15.65 kN
  - 5th percentile = -20.18 kN
  - 95th percentile = 20.41 kN

- **Tractor Trailing axle**
  - Standard deviation = 15.65 kN
  - 5th percentile = -20.18 kN
  - 95th percentile = 20.41 kN

![Dynamic Wheel Load Distribution](image2)

- **Tractor Lead axle**
  - Standard deviation = 15.65 kN
  - 5th percentile = -20.18 kN
  - 95th percentile = 20.41 kN

- **Tractor Trailing axle**
  - Standard deviation = 15.65 kN
  - 5th percentile = -20.18 kN
  - 95th percentile = 20.41 kN
Run # 195 Part 2
Speed = 40 km/hr
Roughness = 424 IRI
Trailer axle spread = 1.63 m
Tractor suspension: spring suspended walking beam
Trailer suspension: air bags
Air suspension lift axle up

![Graphs of dynamic wheel load distributions for tractor and trailer axles with statistical data.](image-url)
Run = 198 Part 2
Speed = 60 km/hr
Hoys roughness = 424 FH
Trailer axle spread = 1.83 m
Trailer suspension: spring suspended walking beam
Air suspension: lift axle up

Tractor leading axle
Standard deviation = 6.62 kN
5th percentile = 10.24 kN
95th percentile = 10.74 kN

Tractor trailing axle
Standard deviation = 6.62 kN
5th percentile = 10.24 kN
95th percentile = 10.74 kN

Tractor leading axle
Standard deviation = 6.02 kN
5th percentile = 11.26 kN
95th percentile = 8.70 kN

Tractor trailing axle
Standard deviation = 6.02 kN
5th percentile = 11.26 kN
95th percentile = 8.70 kN
Run = 216 Part 2  
Speed = 80 km/hr  
Ways roughness = 58 lpm  
Trailer axle spread = 1.83 m  
Tractor suspension = spring suspended walking beam  
Trailer suspension = air bags  
Air suspension lift axle up

**Dynamic Wheel Load, KN**

- **Trailer front axle**
  - Standard deviation: 2.91 KN
  - 5th percentile: -1.28 KN
  - 95th percentile: 4.36 KN

- **Trailer rear axle**
  - Standard deviation: 2.06 KN
  - 5th percentile: -1.86 KN
  - 95th percentile: 4.14 KN

- **Tractor front axle**
  - Standard deviation: 3.36 KN
  - 5th percentile: -6.12 KN
  - 95th percentile: 5.25 KN

- **Tractor rear axle**
  - Standard deviation: 3.37 KN
  - 5th percentile: -5.63 KN
  - 95th percentile: 5.87 KN
Run = 201 Part 2
Speed = 40 km/hr
Hogs roughness = 124 mm
Tractor axle spread = 1.83 m
Tractor suspension: spring suspended walking beam
Trailer suspension: air bags
Air suspension lift axle down

Tractor lead axle
standard deviation = 5.75 kN
5th percentile = 10.39 kN
95th percentile = 8.68 kN

Tractor trailing axle
standard deviation = 4.80 kN
5th percentile = -0.54 kN
95th percentile = 7.66 kN

Tractor lead axle
standard deviation = 3.28 kN
5th percentile = -6.45 kN
95th percentile = 6.91 kN

Tractor trailing axle
standard deviation = 4.59 kN
5th percentile = -8.21 kN
95th percentile = 6.40 kN
Run # 204 Part 2

Noise roughness = 424 LPA
Tractor axle spread = 1.83 m
Tractor suspension: spring suspended walking beam
Tractor suspension: air bags
Air suspension lift axle down

Run # 204 Part 2

Speed = 50 km/hr

Tractor lead axle
standard deviation = 6.55 KN
5th percentile = 10.98 KN
85th percentile = 8.36 KN

Tractor trailing axle
standard deviation = 5.69 KN
5th percentile = 10.58 KN
85th percentile = 8.79 KN

Tractor lead axle
standard deviation = 5.90 KN
5th percentile = 9.44 KN
85th percentile = 7.43 KN

Tractor trailing axle
standard deviation = 4.77 KN
5th percentile = 8.35 KN
85th percentile = 7.41 KN
Run # 214 Part 1
Speed = 80 km/hr
Hogs_roughness = 1.65 JPM
Tractor axle spread = 1.83 m
Tractor suspension = spring suspended walking beam
Air suspension = air bags
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, kN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, kN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, kN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, kN

Trailer lead axle
standard deviation = 10.31 kN
5th percentile = 7.54 kN
95th percentile = 15.06 kN

Trailer trailing axle
standard deviation = 9.05 kN
5th percentile = 13.03 kN
95th percentile = 13.81 kN

Trailer lead axle
standard deviation = 5.54 kN
5th percentile = 3.91 kN
95th percentile = 7.51 kN

Trailer trailing axle
standard deviation = 5.85 kN
5th percentile = 8.29 kN
95th percentile = 7.41 kN
Run = 263 Port 1
Mega roughness = 294 | PH
Tractor axle spread = 2.64 m
Tractor suspension = spring suspended walking beam
Trailer suspension = air bags
For suspension lift axle up

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Tractor Lead axle:
- Standard deviation = 10.62 kN
- 5th percentile = -11.62 kN
- 95th percentile = 10.62 kN

Tractor trailing axle:
- Standard deviation = 11.83 kN
- 5th percentile = -12.07 kN
- 95th percentile = 10.89 kN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Trailer Lead axle:
- Standard deviation = 11.62 kN
- 5th percentile = -12.62 kN
- 95th percentile = 10.62 kN

Trailer trailing axle:
- Standard deviation = 12.83 kN
- 5th percentile = -13.07 kN
- 95th percentile = 11.89 kN
Run #263 Part 2
Mags roughness = 424 IPM
Tractor axle spread = 2.14 m
Tractor suspension: spring suspended walking beam
Trailer suspension: air bags
Air suspension lift axle up

**Tractor Lead axle**
- Standard deviation = 7.43 kN
- 5th percentile = -12.28 kN
- 95th percentile = 11.76 kN

**Tractor Trailing axle**
- Standard deviation = 8.57 kN
- 5th percentile = -12.18 kN
- 95th percentile = 10.70 kN

**Trailer Lead axle**
- Standard deviation = 6.05 kN
- 5th percentile = -11.24 kN
- 95th percentile = 10.23 kN

**Trailer Trailing axle**
- Standard deviation = 6.08 kN
- 5th percentile = -13.29 kN
- 95th percentile = 10.43 kN
Run = 259 Part 1
Wind roughness = 165 IPH
Tractor axle spread = 2.44 m
Tractor suspension: spring suspended walking beam
Trailer suspension: air bags
Air suspension lift axle up

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, kN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, kN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, kN

Tractor lead axle
standard deviation = 6.41 kN
5th percentile = -14.22 kN
95th percentile = 15.06 kN

Tractor trailing axle
standard deviation = 8.29 kN
5th percentile = -13.15 kN
95th percentile = 13.66 kN

Trailer lead axle
standard deviation = 6.18 kN
5th percentile = -5.52 kN
95th percentile = 6.87 kN

Trailer trailing axle
standard deviation = 8.41 kN
5th percentile = -9.62 kN
95th percentile = 9.88 kN
Run = 294
Roughness = 73 IPM

Traction axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle up

- Tractor load axle
  Standard Deviation = 2.73 KN
  5th Percentile = 5.18 KN
  95th Percentile = 3.81 KN

- Trailer trailing axle
  Standard Deviation = 2.90 KN
  5th Percentile = 4.81 KN
  95th Percentile = 3.38 KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN
Run # 279 Part 1
Nays roughness - 105 lph
Tractor axle spread - 1.37 m
Tractor suspension - spring suspended walking beam
Trailer suspension - rubber suspended walking beam
Air suspension lift axle up

**Tractor Lead axle**
- Standard deviation: 17.46 kN
- 5th percentile: 16.17 kN
- 95th percentile: 17.80 kN

**Trailer Lead axle**
- Standard deviation: 16.65 kN
- 5th percentile: 15.67 kN
- 95th percentile: 17.05 kN

**Tractor Trailing axle**
- Standard deviation: 17.20 kN
- 5th percentile: 16.73 kN
- 95th percentile: 17.75 kN

**Trailer Trailing axle**
- Standard deviation: 17.61 kN
- 5th percentile: 16.91 kN
- 95th percentile: 18.41 kN

**Frequency of Occurrence**

**Dynamic Wheel Load, kN**
Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

Run = 278 Part 1
Mavg roughness = 165 IPM
Tractor axle spread = 1.37 m
Tractor suspension: spring suspended walking beam
Trailer suspension: rubber suspended walking beam
Air suspension lift axle down

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN

FREQUENCY OF OCCURRENCE, %

DYNAMIC WHEEL LOAD, KN
Run 278 Part 3

Nays roughness = 217 IPH
Tractor axles spread = 1.37 m
Tractor suspension - spring suspended walking beam
Trailer suspension - rubber suspended walking beam
Air suspension lift axle down

- Frequency of occurrence, %
- Dynamic wheel load, KN

Tractor lead axle:
- Standard deviation = 15.26 KN
- 5th percentile = -22.00 KN
- 95th percentile = 22.39 KN

Trailer trailing axle:
- Standard deviation = 15.36 KN
- 5th percentile = -22.25 KN
- 95th percentile = 22.30 KN

Tractor lead axle:
- Standard deviation = 15.57 KN
- 5th percentile = 23.12 KN
- 95th percentile = 26.03 KN

Trailer trailing axle:
- Standard deviation = 15.88 KN
- 5th percentile = -27.36 KN
- 95th percentile = 26.71 KN